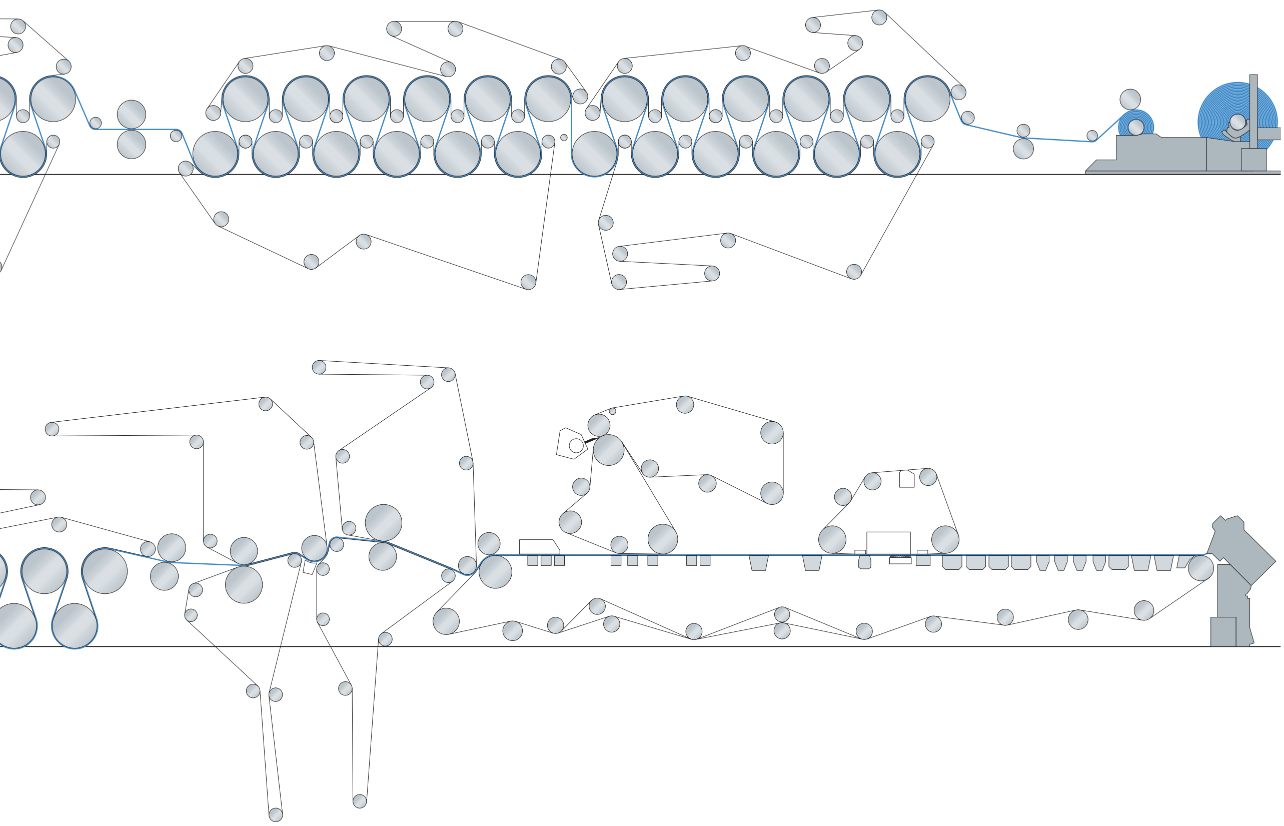
Rolling bearings in paper machines

A handbook for paper machine designers, operators and maintenance staff



5th Edition

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Rolling bearings in paper machines

The fifth edition of SKF's Rolling bearings in paper machines handbook was created as a guide for machine designers, operators and maintenance departments. It supersedes all older versions of the handbook which include obsolete recommendations due to advances in SKF bearing design and performance.

Fifth edition includes up-to-date information on SKF bearings and recommendations, but it does not cover the latest paper machine designs for confidentiality reasons.

There are some important changes from the fourth edition which was published in 2011. For example:

- There is a greater focus on issues of interest to mill staff such as root cause failure analysis and bearing remanufacturing.
- Many recommended bearing fits have changed due to the increase in bearing dynamic load ratings over the years.
- The recommendations about maximum water content are more precise.
- Information on the SensorMount method for mounting bearings with tapered bores has been included. As has detail on the bearings used in size presses, rope sheaves and spreader rolls.
- An equation for reeler bearing load calculations has been added.

We hope that you find the latest edition of SKF's Rolling bearings in paper machines useful.

Regards,
Philippe Gachet
Global application engineer master Pulp & Paper specialist

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SKF – the knowledge engineering company

From one simple but inspired solution to a misalignment problem in a textile mill in Sweden, and fifteen employees in 1907, SKF has grown to become a global industrial knowledge leader.

Over the years, we have built on our expertise in bearings, extending it to seals, mechatronics, services and lubrication systems. Our knowledge network includes 46 000 employees, 15 000 distributor partners, offices in more than 130 countries, and a growing number of SKF Solution Factory sites around the world.

Research and development

We have hands-on experience in over forty industries based on our employees' knowledge of real life conditions. In addition, our world-leading experts and university partners pioneer advanced theoretical research and development in areas including tribology, condition monitoring, asset management and bearing life theory. Our ongoing commitment to research and development helps us keep our customers at the forefront of their industries.

Meeting the toughest challenges

Our network of knowledge and experience, along with our understanding of how our core technologies can be combined, helps us create innovative solutions that meet the toughest of challenges. We work closely with our customers throughout the asset life cycle, helping them to profitably and responsibly grow their businesses.

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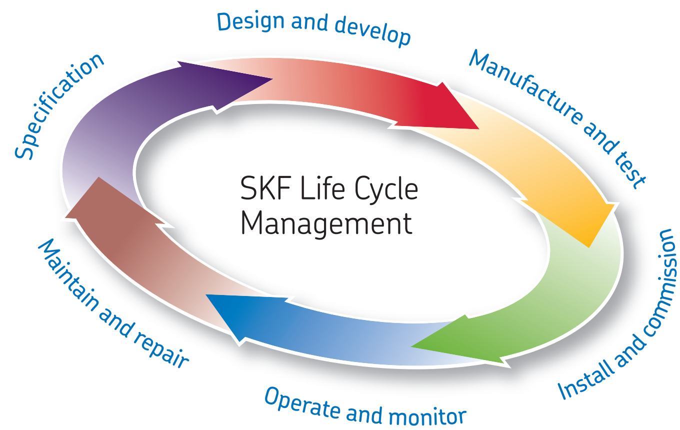
Since 2005, SKF has worked to reduce the negative environmental impact from our operations and those of our suppliers. Our continuing technology development resulted in the introduction of the SKF BeyondZero portfolio of products and services which improve efficiency and reduce energy losses, as well as enable new technologies harnessing wind, solar and ocean power. This combined approach helps reduce the environmental impact both in our operations and our customers' operations.

SKF Solution Factory makes SKF knowledge and manufacturing expertise available locally to provide unique solutions and services to our customers.

Working with SKF IT and logistics systems and application experts, SKF Authorized Distributors deliver a valuable mix of product and application knowledge to customers worldwide.



Our knowledge – your success

SKF Life Cycle Management is how we combine our technology platforms and advanced services, and apply them at each stage of the asset life cycle, to help our customers to be more successful, sustainable and profitable.

Working closely with you

Our objective is to help our customers improve productivity, minimize maintenance, achieve higher energy and resource efficiency, and optimize designs for long service life and reliability.

Innovative solutions

Whether the application is linear or rotary or a combination, SKF engineers can work with you at each stage of the asset life cycle to improve machine performance by looking at the entire application. This approach doesn't just focus on individual components like bearings or seals. It looks at the whole application to see how each component interacts with each other.

Design optimization and verification

SKF can work with you to optimize current or new designs with proprietary 3-D modelling software that can also be used as a virtual test rig to confirm the integrity of the design.

Bearings
SKF is the world leader in the design, development and manufacture of high performance rolling bearings, plain bearings, bearing units and housings.

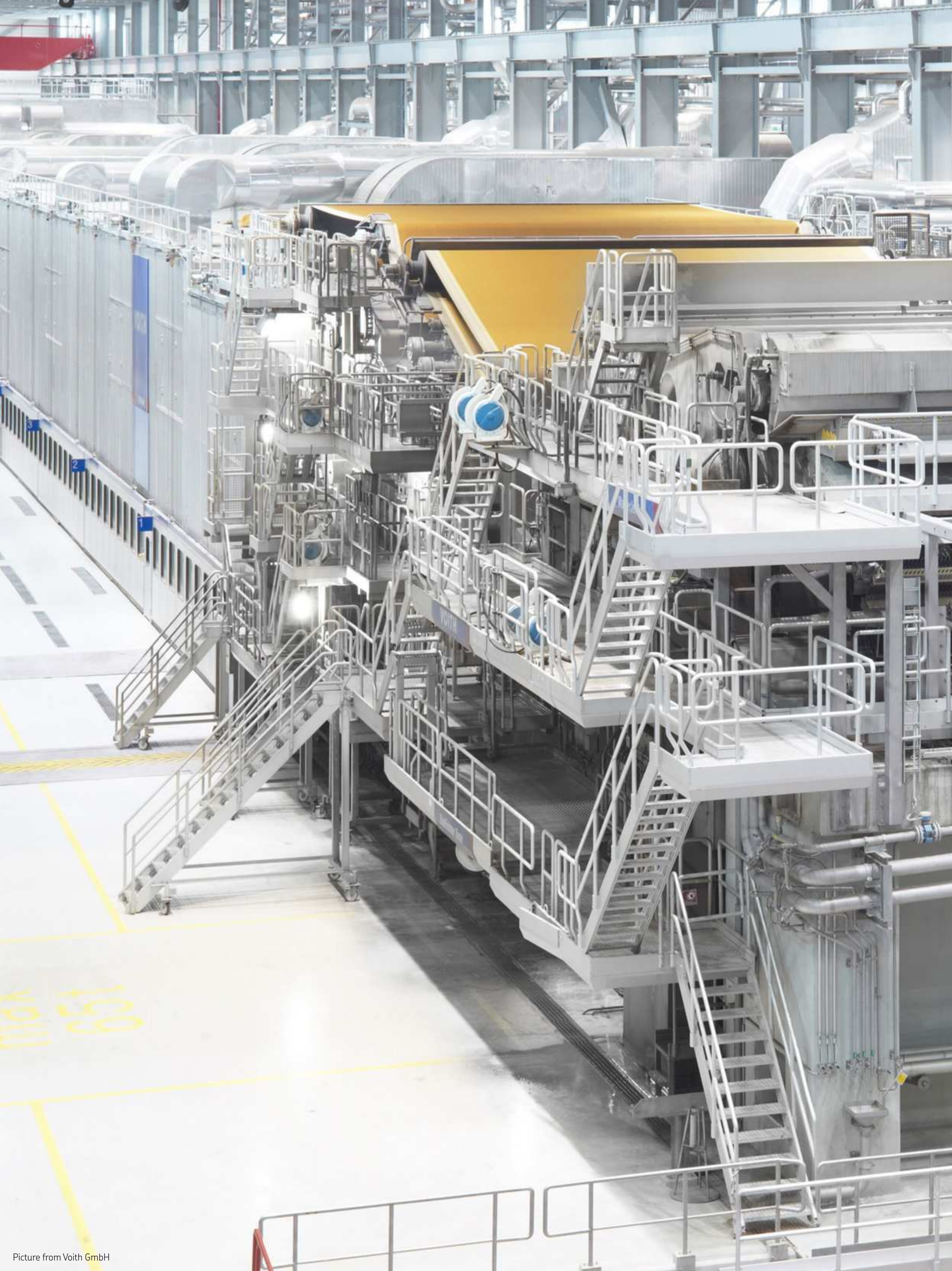
Machinery maintenance
Condition monitoring technologies and maintenance services from SKF can help minimize unplanned downtime, improve operational efficiency and reduce maintenance costs.

Sealing solutions
SKF offers standard seals and custom engineered sealing solutions to increase uptime, improve machine reliability, reduce friction and power losses, and extend lubricant life.

Mechatronics
SKF fly-by-wire systems for aircraft and drive-by-wire systems for off-road, agricultural and forklift applications replace heavy, grease or oil consuming mechanical and hydraulic systems.

Lubrication solutions
From specialized lubricants to state-of-the-art lubrication systems and lubrication management services, lubrication solutions from SKF can help to reduce lubrication related downtime and lubricant consumption.

Actuation and motion control
With a wide assortment of products – from actuators and ball screws to profile rail guides – SKF can work with you to solve your most pressing linear system challenges.



Picture from Voith GmbH

General requirements and recommendations

Modern paper machines can be more than 10 metres wide, 20 metres high and 400 metres long. Generally, they consist of a forming or wire section, a press section, a drying section, a coating section, a calender and a reeler. Large machines incorporate as many as 1 500 bearings. The operating conditions for these bearings vary greatly depending on where they are installed on the machine.

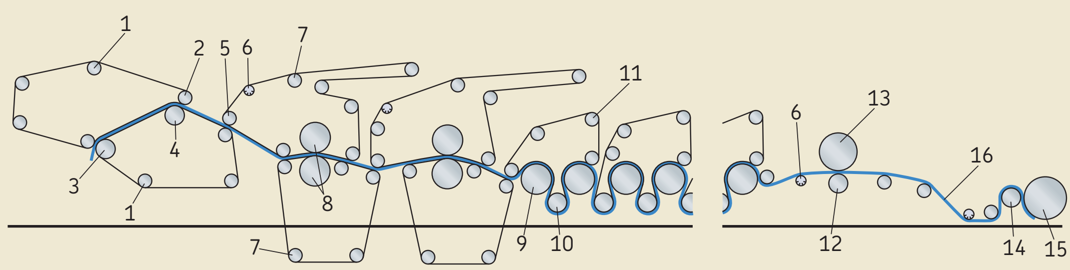
A paper machine has a large number of rolls equipped with medium and large-size rolling bearings (→ **fig. 1.1**). In a few applications, plain bearings are sometimes used.

The operating speeds of the different types of modern paper machines are typically as follows:

- Pulp drying machines 100–300 m/min
- Board machines 250–1 400 m/min
- Liner machines 300–1 800 m/min
- Fine paper machines 700–2 200 m/min
- Newsprint machines 1 000–2 200 m/min
- Tissue machines 1 100–2 200 m/min

Paper machines differ in design according to the grade of paper that is to be produced. For example, when it comes to the number of drying cylinders, board machines have the most, followed by liner and newsprint machines. A great number of drying cylinders are needed in liner and board machines on account of the product thickness, and in newsprint machines because of the very high speed. Tissue paper requires less drying and therefore the dryer section normally consists of only one large cylinder called a Yankee cylinder. The web forming part also differs from machine to machine.

Fig.1.1



Basic layout of a newsprint or fine paper machine
1 *Wire roll*
2 *Forward drive roll*
3 *Forming roll (suction roll)*
4 *Suction couch roll*
5 *Pick-up roll*
6 *Spreader roll*
7 *Felt roll*
8 *Shoe press*
9 *Drying cylinder*
10 *Vacuum roll*
11 *Guide roll (wire roll)*
12 *Deflection compensating press roll (soft calender)*
13 *Thermo roll (soft calender)*
14 *Reel drum*
15 *Reel spool*
16 *Paper web*

For modern newsprint paper machines, the steam temperature is around 140 to 150 °C. For liner and board machines, the maximum temperature is around 190 to 200 °C. Sometimes the machines operate with super-heated steam in which case temperatures up to 225 °C are common. Tissue machines normally have steam temperatures between 150 and 200 °C.

How these temperatures influence bearing temperatures is shown in the chapter *Lubrication examples*.

Selection of bearing size

Fatigue life – service life

By definition, a bearing is considered to have failed from fatigue as soon as spalling occurs. Spalling can be detected by using a vibration-sensitive instrument. If operation is continued after the first sign of spalling has been recorded, the flaked area will increase in size and the vibration level will rise.

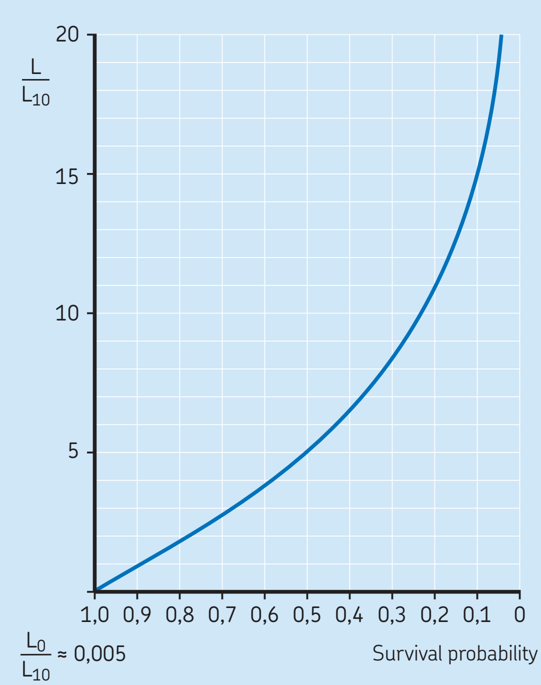
SKF has gathered a lot of statistics from endurance life testing of rolling bearings. The results of these tests are shown in **diagram 1.1**. The diagram shows that the fatigue life of one bearing can differ greatly from that of another bearing in a large population. **Diagram 1.1** can be presented in another way in **diagram 1.2**, which is more interesting for bearing users. It shows the number of bearings in a population that reach their fatigue life per unit of time

The length of time that a bearing can be left in service after reaching its defined fatigue life is generally difficult to specify.

The replacement of failed bearings as soon as possible is always recommended especially if they are in a critical position, mounted with excessive interference fits, subjected to heating through the journal, etc.

Bearing life dispersion

Diagram 1.1



L = Real bearing fatigue life
L_{10} = Basic rating life according to the ISO definition (the life that 90% of all bearings attain or exceed)
L_0 = Minimum life exceeded by all bearings

To the end-user of SKF bearings, the term "bearing life" means the time that an individual bearing works satisfactorily in the machine. SKF calls this the "service life" of the bearing. The service life of paper machine bearings is especially interesting as these bearings are usually replaced for reasons other than fatigue.

Recommended L_{10h} and L_{10mh} lives

SKF has many years of experience selecting bearing size for the paper industry using the L_{10h} and L_{10mh} bearing life equations.

When calculating the life of paper machine bearings, SKF recommends that both the basic rating life L_{10h} and the SKF rating life L_{10mh} are taken into consideration. The calculated bearing basic rating life L_{10h} should exceed 200 000 hours for dryer section bearings and 120 000 hours for bearings in other sections. The calculated SKF rating life L_{10mh} should be 100 000 hours or more for all bearing positions in the paper machine, There are some exceptions due to the fact that too low loaded bearings can fail even if the calculated life is over 100 000 hours. Rollers of a too lightly loaded bearing might slide instead of roll. For example, machine speed increase leads to increased wire and felt tension. This can add load on some bearings, but as well lift some rolls, decreasing the load on some bearings. SKF catalogue *Rolling bearings* contains formulas to calculate the minimum load in order to provide satisfactory operation.

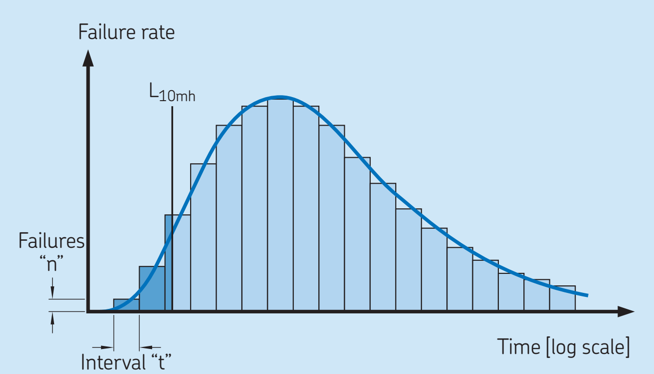
Please note that in 2015, SKF launched the concept of the "SKF Generalized Bearing Life Model" (GBLM). This new bearing life rating model, contrary to current models including ISO 281, separates the calculation of bearing rating life in surface and subsurface terms. The surface term is calculated by using the results of explicit tribological models while the subsurface term is calculated with classical Hertzian rolling contact fatigue theory.

Surface performance factors are introduced with a direct influence on surface damage risk while the basic dynamic load rating (C-value) is retained according to its original purpose in the subsurface term.

This new bearing life rating model is not yet included in the current SKF catalogue *Rolling bearings*.

The number of bearings reaching their fatigue life versus time

Diagram 1.2



The basic rating life L_{10h}

The most common way to calculate the bearing life is to use the equations in the SKF catalogue *Rolling Bearings*. The equation to be used is

$$L_{10h} = \frac{1\,000\,000}{60\ n}\left(\frac{C}{P}\right)^{p}$$

where

L_{10h} = basic rating life (at 90% reliability), operating hours
n = rotational speed, r/min
C = basic dynamic load rating, N
P = equivalent dynamic bearing load, N
p = exponent of the life equation
p = 3 for ball bearings
p = 10/3 for roller bearings

The SKF rating life L_{10mh}

The SKF rating life highlights the significant influence of cleanliness and lubricant film thickness on the fatigue life of bearings. Even though bearings in pulp and paper machinery rarely run until they are fatigued, cleanliness and lubricant film thickness have an important influence on service life.
The equation to be used is

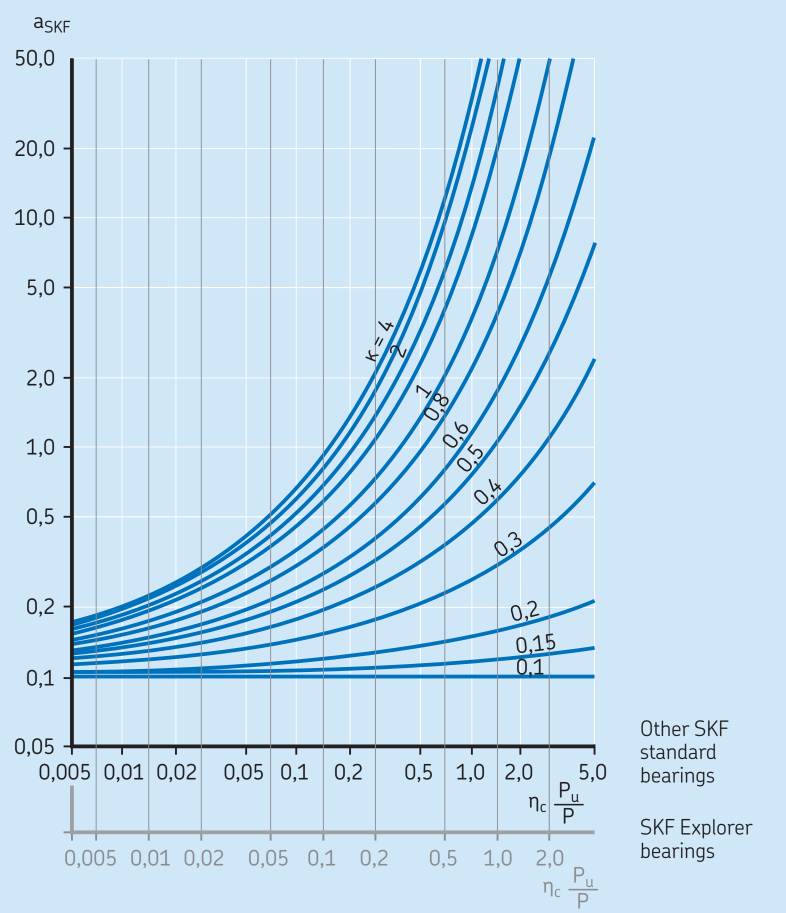
$$L_{10mh} = a_{SKF}\, L_{10h}$$

Values of a_{SKF} for roller bearings are given as a function of η_c (P_u/P) in **diagram 1.3**, where

η_c = adjustment factor for contamination
P_u = fatigue load limit, N
P = equivalent dynamic bearing load, N

Similar diagrams for radial and axial ball bearings, as well as for axial roller bearings, are given in the SKF catalogue *Rolling Bearings*.

Diagram 1.3



If κ > 4, use curve for κ = 4
As the valued of η_c (P_u/P) tends to zero, a_{SKF} tends to 0,1 for all values of κ

Factor a_{SKF} for radial roller bearings

There are advantages and disadvantages with these simplified methods. For example, it enables quick comparison with previous bearing dimensioning and field experience. This method is also easy to use because of the limited amount of input data needed. The main disadvantage is the limited ability to perform an accurate calculation where additional influential factors are taken into account. Housings and journals, for example, are assumed to be stiff and perfectly round.

Therefore, SKF has developed advanced computer programs to enable in-depth analysis that considers the influence of:

- internal design of the bearing
- clearance reduction due to heat generation in the bearing
- clearance reduction due to external heating
- clearance reduction due to housing or journal interference fit
- axial preloading
- errors of housing form
- bearing temperature due to heat generation in the bearing and external heating/cooling

Bearing types used in paper machines

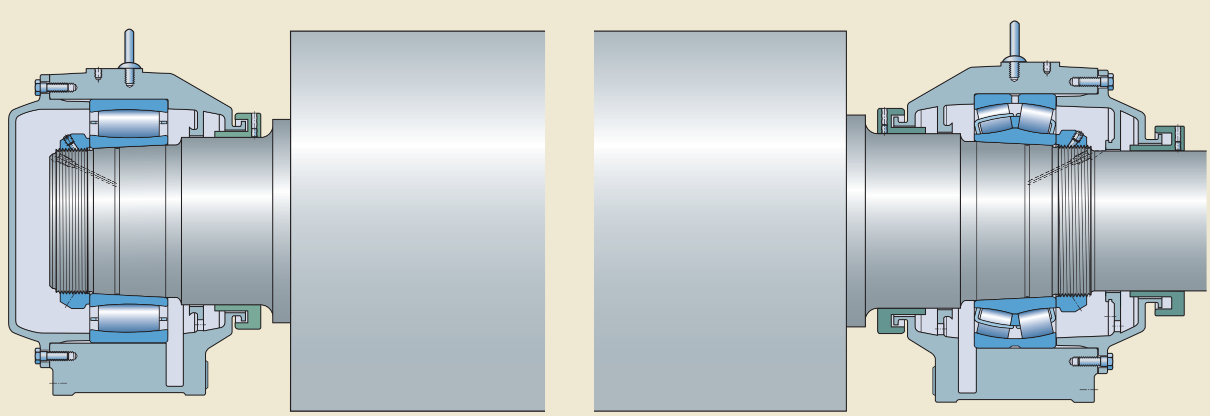
A high proportion of the bearings used in papermaking machinery are spherical roller bearings of standard design. Their ability to accommodate considerable radial loads in combination with axial loads makes these bearings very suitable for supporting the locating side (usually the drive side) of the various rolls and cylinders of paper machines. Spherical roller bearings also permit misalignment between shaft and housing, which is especially important for paper machines where bearings are mounted in separate housings spaced far apart.

In many cases, spherical roller bearings can also be used successfully at the non-locating side, usually the front side, of paper machines. However, in most cases the ideal solution is to combine a spherical roller bearing at the drive side with a CARB toroidal roller bearing at the front side (→ **fig. 1.2**). This bearing arrangement, called the SKF self-aligning bearing system, accommodates both misalignment and axial displacement internally and without frictional resistance, with no possibility of generating internal axial forces in the bearing system.

If a CARB toroidal roller bearing is used, it may be possible to downsize the bearing arrangements at both the drive and front sides. **Diagram 1.4** shows the life of different bearing systems. The coefficient of friction μ for steel against cast iron is 0,15–0,20 for new housings of good quality. For used housings, the coefficient can be considerably higher. In the example μ = 0,15 is used.

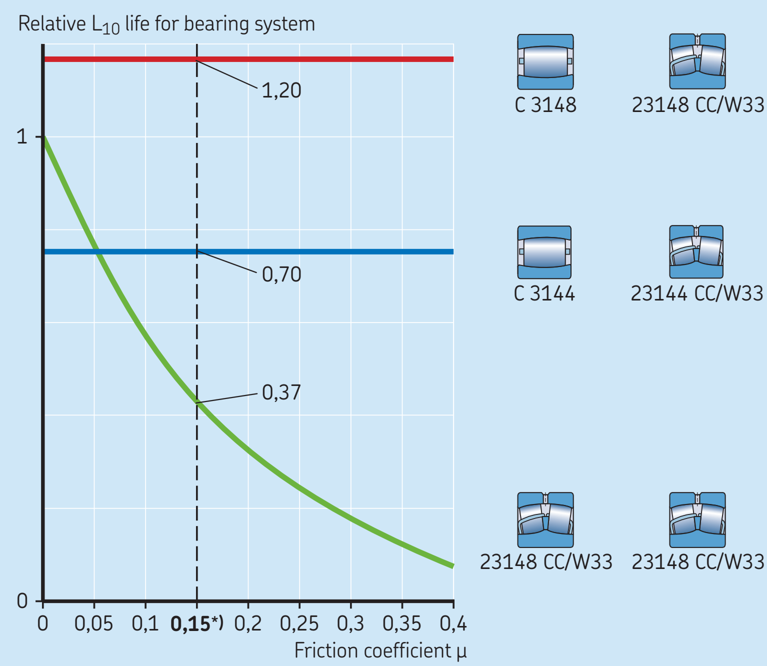
The standard solution called the SKF self-aligning bearing system – SKF spherical roller bearing on the drive side and a CARB toroidal roller bearing on the front side

Fig. 1.2



Relative life of different bearing systems

Diagram 1.4



SKF spherical roller bearings

All SKF spherical roller bearings have the C design to avoid edge stress and high friction under axial load which is typical for old bearing designs having a fixed middle flange on the inner ring. The C design, created by SKF in the early 50s, has symmetrical rollers and no fixed middle flange on the inner ring. The rollers are guided in the unloaded zone by a floating guide ring so that they enter the load zone in the optimal position (→ **fig. 1.3**).

To reduce further friction and frictional heat, rollers are self-guided by the CC principle (→ **fig. 1.4**). Today, all SKF spherical roller bearings have the CC principle even if this is not indicated in the designation. The CC principle, together with the manufacturing precision of the bearing and the adjacent elements, and also the optimized lubrication and heat removal, permits SKF bearings to run above catalogue limiting speeds in press roll applications.

C design spherical roller bearing

Fig. 1.3

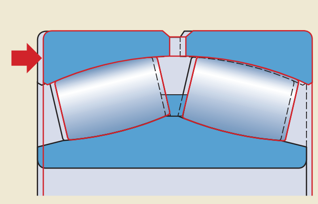
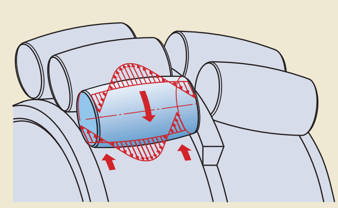


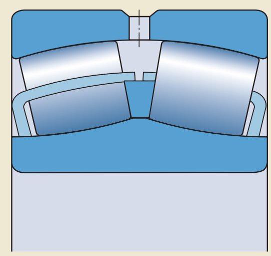
Fig. 1.4



The CC Principle: reduced friction and heat generation

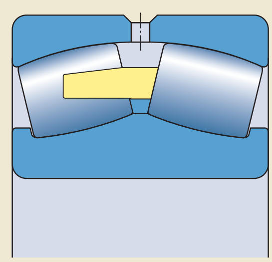
SKF spherical roller bearing of CC, C, EC and ECC designs

Fig. 1.5



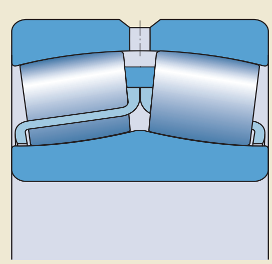
SKF spherical roller bearings of CAC, ECAC, CA and ECA designs

Fig. 1.6



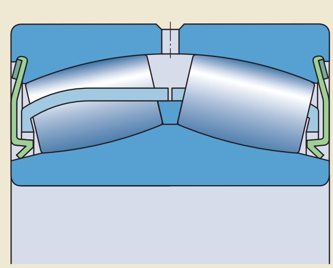
SKF spherical roller bearing of E design with bore d>65 mm

Fig. 1.7



SKF sealed spherical roller bearing

Fig. 1.8



SKF spherical roller bearings of CC, C, EC and ECC designs

These bearings have symmetrical rollers, a flangeless inner ring and a pressed steel cage for each roller row (→ **fig. 1.5**). The guide ring is centred on the inner ring. Bearings of EC and ECC designs incorporate reinforced roller sets for added load carrying capacity.

SKF spherical roller bearings of CAC, ECAC, CA and ECA designs

These designs are used for large sizes of SKF spherical roller bearings (→ **fig. 1.6**). The rollers are symmetrical and the inner ring has retaining flanges. The guide ring is centred on the inner ring between the two rows of rollers and the cage is a one-piece, double pronged machined steel or brass cage. The ECAC and ECA designs have reinforced roller sets for increased load carrying capacity.

SKF spherical roller bearings of E design

These bearings have symmetrical rollers, a flangeless inner ring, and a guide ring centered on the hardened cages (→ **fig. 1.7**). The E design bearings incorporate all the advantages of the well-proven SKF CC bearings as well as additional refinements such as higher load carrying capacity and, for the bearings with bores larger than 65 mm, better roller guidance in the unloaded raceway zone. Smaller E design bearings, with bores less than or equal to 65 mm, look like CC design spherical roller bearings.

SKF sealed spherical roller bearings

SKF has a large range of sealed spherical roller bearings from 25 mm up to 400 mm bore diameter. The integrated seals offer additional protection against contamination and permit relubrication (→ **fig. 1.8**). Such bearings have the same load carrying capacity as equivalent open spherical roller bearings. While the sealed E design is wider than the open E design, all the others have the same boundary dimensions as equivalent open bearings.

SKF Explorer and Upgraded SKF Explorer (WR) bearings

SKF introduced the SKF Explorer bearing in the late 1990s. Developments in steel production, new heat treatment procedures and manufacturing process are the factors behind the development of the SKF Explorer bearing. The main advantages are longer life for existing machines and the possibility for downsizing on new machines. To enable users to evaluate how this influences bearing rating life, SKF has introduced increased basic dynamic load ratings and shifted the η_c (P_u/P) axis in the a_{SKF} diagram when calculating L_{10mh} life using the SKF rating life.

In 2011, SKF introduced the upgraded SKF Explorer bearing. It is made with a new generation steel giving higher toughness (crack resistance) and higher hardness (better wear resistance). To estimate the influence on the SKF rating life, an additional factor has to be considered.

All Explorer bearings with a bore equal or less than 300 mm have P5 circular radial run-out, which is four times better than ISO Normal values.

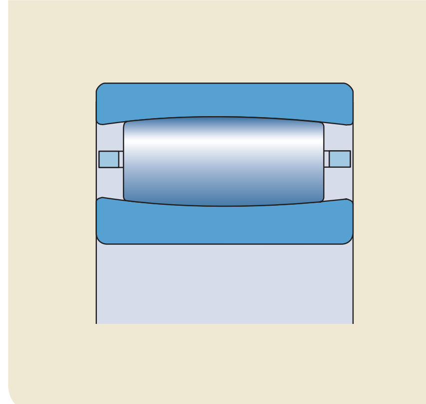
The SKF Explorer bearings retain the designations of the earlier standard bearings e.g. 22218 E or 23152 CC/W33.

However, each bearing and box is marked with the name SKF Explorer (→ **fig. 1.9**).

Upgraded SKF Explorer spherical roller bearings and packaging are marked WR in addition to the Explorer marking.

More information about SKF Explorer bearings and the present range can be obtained from your local SKF company.

Fig. 1.10



CARB toroidal roller bearing

CARB toroidal roller bearings

The best solution for most rolls and cylinders is to combine a spherical roller bearing on the drive side with a CARB toroidal roller bearing on the front side, as this is the ideal non-locating bearing (→ **fig. 1.10**).

Like spherical roller bearings, CARB toroidal roller bearings can accommodate misalignment and heavy radial loads, but are also able to accommodate axial displacement without additional frictional resistance, like a cylindrical roller bearing. This is what makes CARB toroidal roller bearings the ideal non-locating solution for most paper machine applications. It has the same friction as a spherical roller bearing and should therefore be lubricated in the same way. However, CARB toroidal roller bearings must always be lubricated from the side since they have no lubrication groove in the outer ring.

Fig. 1.9

SKF Explorer spherical roller bearing

Designations

The most common supplementary designations for spherical roller bearings and CARB toroidal roller bearings are shown in **table 1.1**. The table also shows where in the machine the bearings are used.

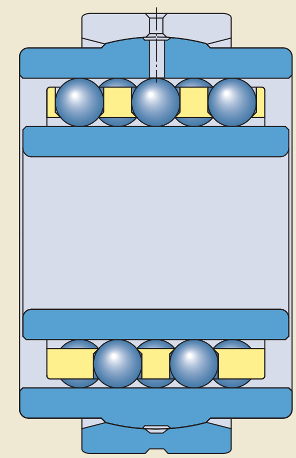
Table 1.2, gives a larger number of supplementary designations for bearings used on paper machines.

Doctor bearings

Specially designed multi-row radial ball bearings are used for doctors allowing both the axial oscillation of the doctor as well as rotating movement (→ **fig. 1.11**). Specially designed plain bearings can also be used. The rotating movement is necessary when the doctor is turned to a rest position during maintenance or when changing the blade. In addition to this degree of movement, the bearing is able to accommodate shaft misalignment. This misalignment is accommodated either by a radius on the outer diameter which adjusts the misalignment between housing and bearing or, for another bearing execution, in the sphere between the two mating parts of the bearing outer ring.

Fig. 1.11

SKF doctor bearing



Other bearing types

Other bearing types can also be seen in paper machines, e.g. cylindrical roller bearings in drying cylinder applications, tapered roller bearings in felt roll applications or reel spools, deep groove ball bearings in rope sheaves, spreader rolls and shoe press and self-aligning ball bearings in spreader rolls, and plain bearings in calender applications.

Common supplementary designations for SKF spherical roller bearings and CARB toroidal roller bearings

Table 1.1

Features	**Requirement**	**Rolls** Forming section Forming	Suction	Wire	Press section Suction	Press	Felt	Dryer section Dryers	Yankees	Felt	Calenders Thermo	Others
Clearance	std	X	X	X								
	C3				X	X	X			X		X
	C4							X	X		X	
Running accuracy	std			X			X	X		X		
	C08	X	X		X				X		X	X
	VQ424					X					X	X
Heat treatment	std	X	X	X	X	X	X	X	X	X	X	X
	HA3							(X)	(X)		(X)	
Special features	L5DA				(X)						(X)	(X)

(X) Recommended only for the specific operating conditions detailed in the relevant chapters of this handbook

The clearance requirement is a guideline that can vary depending on the effect of actual operating conditions and fit values.

Supplementary designations used for bearings in paper machines

Table 1.2

342460	Special design cylindrical roller bearing for drying cylinders. SKF recommends that such bearings are replaced with CARB toroidal roller bearings.
C2	Radial internal clearance class less than Normal
C3	Radial internal clearance class greater than Normal
C4	Radial internal clearance class greater than C3
C5	Radial internal clearance class greater than C4
C08	Extra reduced tolerance for circular radial run-out (P5) of inner ring and outer ring of assembled bearing. Standard for Explorer spherical roller bearings and CARB toroidal bearings with bores equal or less than 300 mm.
C083	C08 + C3
C084	C08 + C4
C10	Old suffix. Reduced and displaced tolerances for bore and outside diameter. For bearing with tapered bore refers to outside diameter only.
C103	C10 + C3
ECB	Spherical roller bearing with case hardened inner ring, USA production
F	Machined cage of steel or special cast iron
HA3	Case hardened inner ring
K	Tapered bore , taper 1:12 on diameter
K30	Tapered bore , taper 1:30 on diameter
M	Machined brass cage, roller guided.
S1	Bearing rings dimensionally stabilized for operating temperature up to +200 °C. Standard for SKF spherical roller bearings and CARB toroidal bearings.
S2	Bearing rings dimensionally stabilized for operating temperature up to +250 °C.
VA405	Spherical roller bearing design for vibrating applications such as vibrating screens.
VA460	Spherical roller bearing and CARB toroidal roller bearing designed for very high speeds.
VA701	Special design cylindrical roller bearing for drying cylinders. SKF recommends that such bearings are replaced with CARB toroidal roller bearings.
L4B	Black oxidized bearing, rollers and rings
L5DA	NoWear anti-smearing coating on rollers
VQ424	Circular radial run-out better than C08 and reduced width tolerance + W4 + W58 + serial number
W	Without W33 lubrication feature (when W33 standard)
W4	Eccentricity high point location marked on inner ring
W20	Three lubricating holes in outer ring of spherical roller bearing
W26	Spherical roller bearing with 6 lubricating holes drilled through the inner ring
W31	Bearing specification according to an old and obsolete Beloit quality standard
W33	Rolling bearings with 3 lubricating holes and circumferential groove in outer diameter
W58	Eccentricity high point location marked on outer ring
W77	Rolling bearings with W33 holes plugged
W503	W33 + W4
W506	W31 + W33
W507	W4 + W31 + W33
W509	W26 + W31 + W33
W513	W26 + W33
W529	W33 + W58
ZE	SKF SensorMount sensor positioned on the small bore diameter side
ZEB	SKF SensorMount sensor positioned on the large bore diameter side

Housings and journals

Due care must be given to the design of bearing housings and journals. For example, provision must be made for axial movement on the non-locating side. The space available for axial movement has to be greater than the thermal elongation of the roll.

The housings and journals should be strong enough to prevent excessive deformation under operating conditions. Additionally, the housings should fit properly into the frame of the paper machine and permit easy mounting, dismounting and inspection of the bearings. In some positions, the housings must also allow for the changing of wires and felts.

SKF has a range of special housings for felt rolls, drying cylinders and Yankee cylinders where all important functional aspects have been taken into consideration.

Housings

Historically, paper machines have been equipped with specially designed bearing housings. The manufacturer has designed a special housing for virtually every individual machine, which is costly in terms of pattern equipment and design time. Specially made housings have also been very difficult to find when a replacement has been required at the paper mill.

SKF moved into the lead in the early 1990s by introducing a standard range of bearing housings for felt rolls in the dryer section, drying cylinders and Yankee cylinders. All housings were designed for high flow circulating oil lubrication, had maintenance-free sealing arrangements and were prepared with connections for condition monitoring. In the early 2000s SKF introduced new felt and dryer bearing housings for modern paper machines with new seal designs for extra protection against liquid contamination due to high pressure cleaning.

Felt roll housings, dryer section

Felt roll housings are available in the following basic executions:

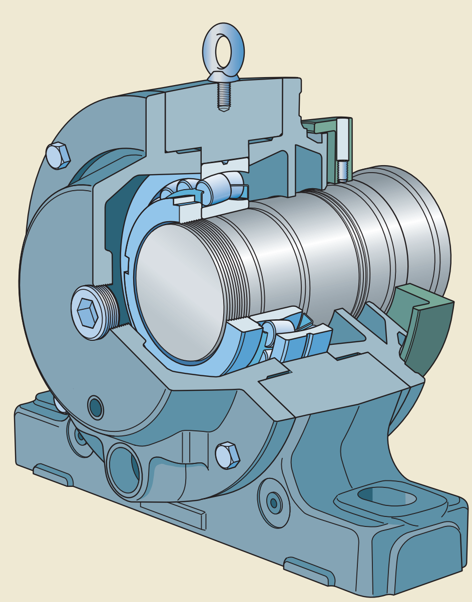
Drive side: spherical roller bearing with located outer ring (→ **fig. 1.12**)
Front side: CARB toroidal roller bearing with located outer ring.
Front side: spherical roller bearing with non-located outer ring (not applicable for the SBFN design)

SKF recommends the use of a spherical roller bearing at the drive side and a CARB toroidal roller bearing at the front side.

The SBFN design can be mounted in the wet section with some slight modifications such as additional covers.

SKF felt roll housing (SBFN design)

Fig. 1.12



Drying cylinder housings

Drying cylinder housings are available in the following basic executions:

Drive side: spherical roller bearing with located outer ring
Front side: CARB toroidal roller bearing with located outer ring (→ **fig. 1.13**).
Front side: spherical roller bearing with non-located outer ring (not applicable for the SBPN design)
Front side: spherical roller bearing with located outer ring in housing on rockers (not applicable for the SBPN design)

SKF recommends the use of a spherical roller bearing at the drive side and a CARB toroidal roller bearing at the front side.

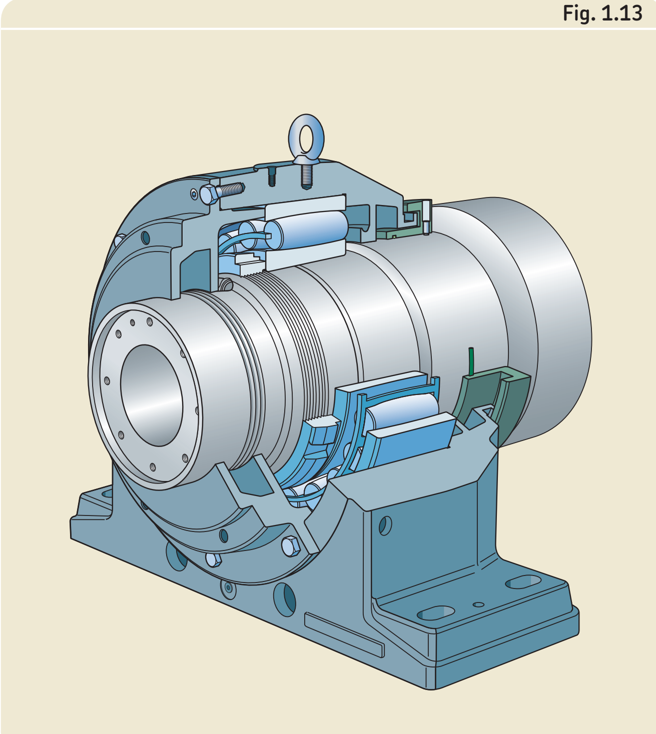
Yankee cylinder housings

Yankee cylinder housings are available in the following basic executions:

Drive side: spherical roller bearing with located outer ring
Front side: CARB toroidal roller bearing with located outer ring
Front side: spherical roller bearing with non-located outer ring
Front side: spherical roller bearing with located outer ring in housing on rockers
Front side: spherical roller bearing with located outer ring in housing on rockers with two side support rockers (→ **fig. 1.14**)

SKF drying cylinder housing
The outside machine housing side seal is of the old design and the inside machine housing side is the new design seal. The old design seal is normally not recommended unless it's combined with the bolted on steam box. The figure shows a cover designed for a steam box.

Fig. 1.13



SKF Yankee cylinder rocker housing

Fig. 1.14

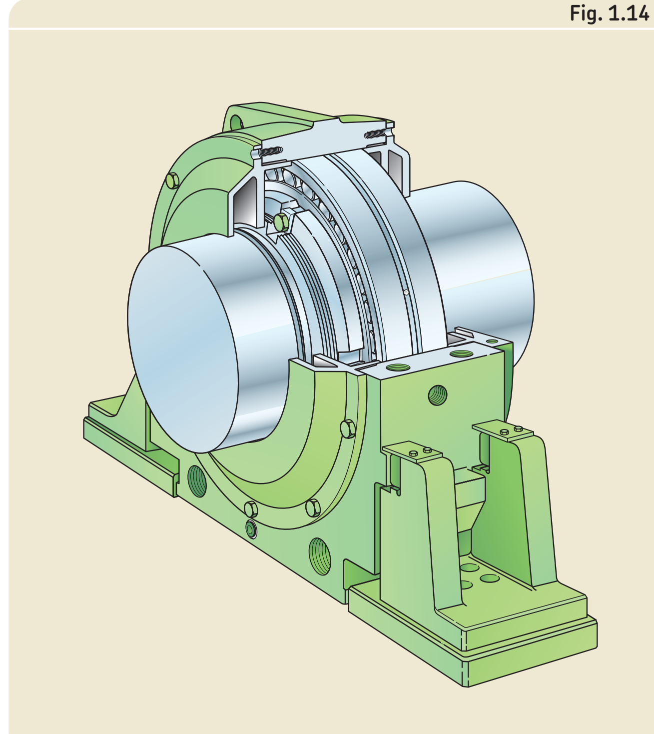
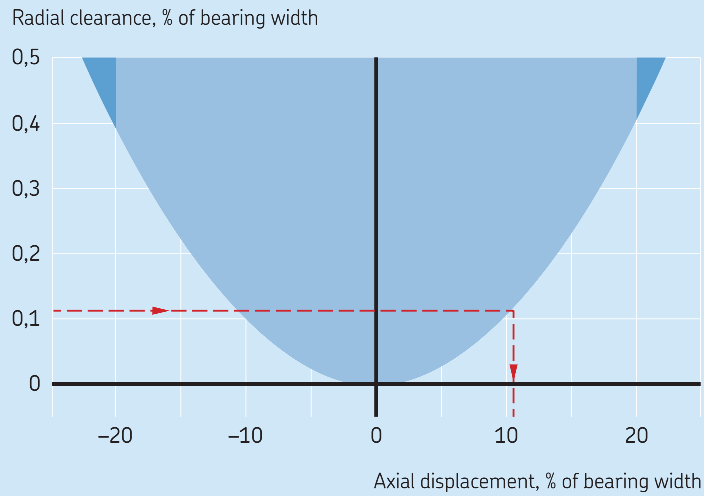


Diagram 1.5



CARB toroidal roller bearings:
Axial displacement depends on radial clearance. Rollers may protrude from the ring raceway at axial displacements above 20% of the bearing width.

Axial displacement for CARB toroidal roller bearings

The relation between radial clearance and axial displacement from a central position is shown in **diagram 1.5**. Axial displacement and radial clearance are given in relation to bearing width (B). This makes the diagram valid for all CARB toroidal roller bearings.

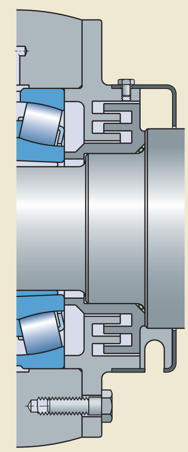
Example: Bearing C 3044/C4 with bearing width B = 90 mm.

Assuming that the operational radial clearance during start-up is 0,1 mm. That is 0,11% of the bearing width.

Diagram 1.5, then shows (dotted line) that the bearing can be axially displaced up to 11% of bearing width, which is 0,11 × 90 = 9,9 mm from the centre.

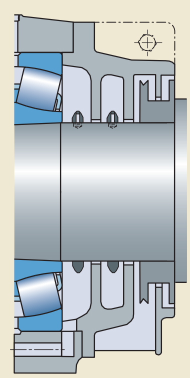
Labyrinth seal for grease lubricated bearings

Fig. 1.15



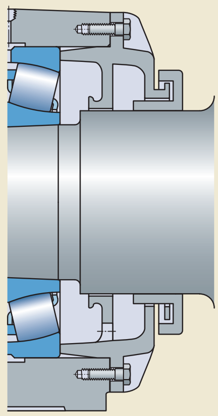
Labyrinth seal for oil lubricated bearings

Fig. 1.16



Improved seal design for oil lubricated bearings in the dryer section

Fig. 1.17



Housing seals

A very important factor for the reliable functioning of bearings in paper machines is efficient sealing of the bearing arrangements. It is important that sealing arrangements adequately protect the bearings from contamination and also prevent the lubricant from escaping and running down the machine. A rolling bearing contaminated by water and/or solid particles will become unserviceable long before its calculated life has been attained.

When designing seals, consideration must be given to the environment of each specific bearing arrangement. Depending on where they are in the machine, bearing arrangements may be subjected to flowing water, condensation, dry conditions or a high ambient temperature. The seals of the housings on the non-locating side must allow for the required axial movement due to shaft elongation. The basic design of the seal depends upon whether the bearings are lubricated by grease or oil. The application drawings in this handbook show some examples of basic designs of bearing arrangements.

Different types of sealing arrangements are shown in the lubrication examples. The need for efficient seals is greatest in the wet section where most of the bearing arrangements are subjected to very wet conditions. Experience shows that a well-greased multi-stage labyrinth seal, whether it be axial or radial, affords good protection of the bearings in the wet section especially when it is reinforced by a splash guard (→ **fig. 1.15**). If the bearings are oil lubricated, the sealing arrangements have to be of a different design. **Fig. 1.16** shows an efficient seal for oil lubricated press roll bearings. The seal must prevent water from entering the bearing housing, even during hosing down, which is often carried out with water at high pressure.

In the dryer section of the machine, bearing housings are exposed to moisture in the form of condensation or leaks from steam nozzles etc. Small soft fibres may enter the housings too. Nevertheless, simpler seals may be used for the housings in the dryer section. However, if problems occur, more efficient seals have to be applied. **Fig. 1.17** shows a design of how to improve the seals of dryer section housings.

SKF can provide a full range of radial shaft seals and V-rings. SKF can also offer customized machined seals with short delivery times (→ **fig. 1.18**). Contact SKF for more information.

SKF Seal Jet: CNC machine to manufacture customized seals

Fig. 1.18

Dimensioning of outlets in bearing housings

Many bearing positions in modern paper machines are lubricated by circulating oil. For many years, SKF has recommended larger oil flows than those normally used in older machines. One problem with older machines is draining the increased circulating oil flow through the original small oil outlets. Old machines without a heating system in the oil tank have great difficulty starting after a lengthy standstill. The problem is high oil viscosity due to low oil temperature.

It is difficult to accurately calculate the required outlet diameter because many variables influence oil drainage, e.g. oil level difference, length and diameter of the pipe, number of bends, and oil viscosity. Generally, the outlet diameters are selected from experience or by rule of thumb. **Fig. 1.19** shows a sketch of an outlet pipe with the relevant dimensions indicated.

An approximate calculation of the required minimum outlet diameter can be performed as follows

$$d = 2{,}2 \left(\frac{(2{,}5 + 0{,}2\ n)\ Q^2 \times 10^3 + 3\ \nu\ l Q}{h} \right)^{1/4}$$

where

d = minimum bore of outlet pipe, mm
n = number of 90° bends
Q = oil flow, l/min
ν = kinematic viscosity of oil at lowest operating temperature (mostly at start-up), mm^2/s
l = pipe length, mm
h = oil level difference, mm

The equation is valid when the outlet from the housing has no restrictions. Practical experience shows that if there are restrictions close to the outlet, e.g. walls in bearing housings, the resistance to the inflow increases considerably. The calculated diameter d should then be increased by 50%.

Example

A drying cylinder bearing housing has an outlet pipe with a length of 3 000 mm to the first connecting main pipe. The oil level difference to this connection is 1 000 mm and there are two bends. When starting up the machine, the oil viscosity can be 220 mm^2/s if the temperature is around 40 °C. Select minimum bore diameter for the outlet pipe to avoid flooding when the circulating oil flow is 4 l/min.

$$d = 2{,}2 \left(\frac{(2{,}5 + 0{,}2 \times 2) \times 4^2 \times 10^3 + 3 \times 220 \times 3\,000 \times 4}{1\,000} \right)^{1/4} = 20{,}8 \text{ mm}$$

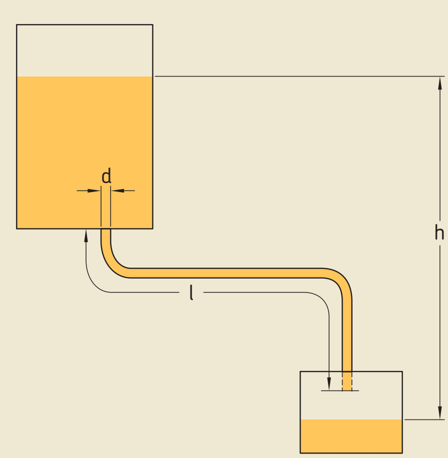
With a restricted oil inflow, the recommended minimum outlet pipe bore diameter is

$d = 1{,}5 \times 20{,}8 = 31{,}2$ mm

Select a pipe with a bore diameter larger than 32 mm.

Oil outlet definition

Fig. 1.19



Tolerances

General

From a bearing function point of view, the main aspects of the housing and journal design in paper machines are the dimensional and geometrical tolerances.

The tolerances for cylindrical bearing seats on shafts and in housings should correspond to the tolerance class of the bearings. Guideline values for the dimensional and geometrical tolerances are provided hereafter.

Dimensional tolerances

For bearings made to normal tolerances, the dimensional tolerance of the cylindrical seatings on the shaft should be at least to ISO grade IT6 and in the housing to at least grade IT7. Where adapter or withdrawal sleeves are used on cylindrical shafts, grade IT9 (h9Ⓔ) can be permitted.

Tolerances for total radial run-out

Total radial run-out tolerances as defined in ISO 1101 should be one to two IT grades tighter than the prescribed dimensional tolerance.

For example, assuming a bearing seat with shaft diameter tolerance m6Ⓔ, so corresponding to IT6, the required total radial run-out is then IT5/2 or IT4/2 because ISO 1101 defines the total radial run-out as a difference in radii of two coaxial cylinders.

Two IT grades better than the prescribed dimensional tolerance is recommended when particularly stringent run-out requirements are stipulated e.g. when bearings with extra close circular radial run-out tolerance C08, VA460 or VQ424 are used.

When bearings are to be mounted on adapter or withdrawal sleeves, the circularity and straightness of the sleeve seatings should be IT5/2.

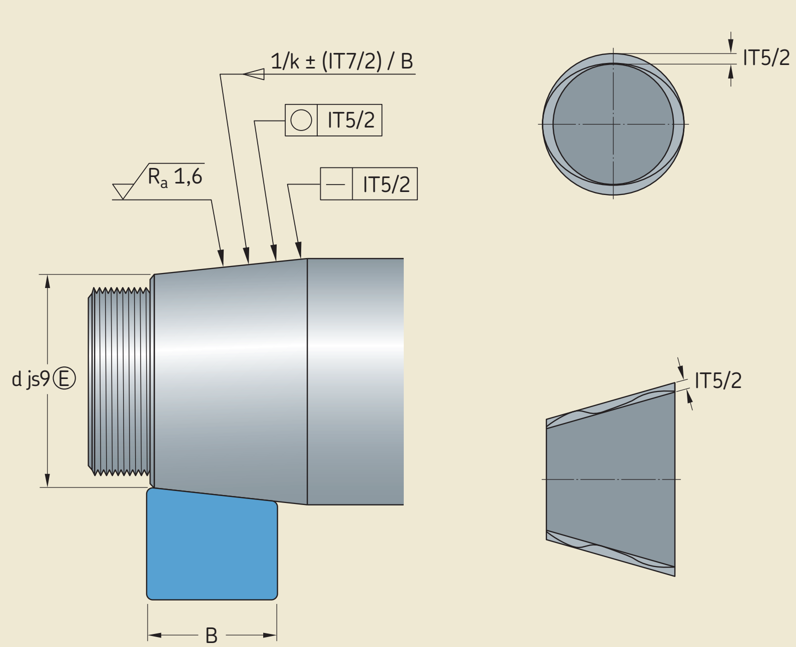
Tolerances for total axial run-out

Abutments for bearing rings should have a total axial run-out tolerance, as defined in ISO 1101, which is better by at least one IT grade than the diameter tolerance of the associated cylindrical seat. The axial run-out of the abutments corresponds to the required axial run-out of the mounted inner ring. The latter run-out requirement applies even when the bearing is mounted on a tapered journal without abutment.

Tolerances for tapered journal seatings

When the bearing is mounted directly on a tapered journal seat, the seating diameter tolerance is permitted to be wider than in the case of a cylindrical seat. **Fig. 1.20** shows a grade 9 diameter tolerance, while the form tolerance stipulations are the same as for cylindrical journal seats.

Fig. 1.20



Definition of tapered journal tolerances

When establishing tolerances for tapered journal seats for spherical roller bearings, different systems have been applied in Europe and the USA.

The European system was based on the permissible angle deviation for the journal taper being on the plus side of the nominal value i.e. following the practice applied to bearings. Moreover, the tolerance value was related to the nominal diameter of the journal.

In the USA, the corresponding permissible deviation was located on the minus side instead and the value was coupled to the nominal width of the bearing.

These divergent methods have naturally led to practical difficulties. Consequently, a common SKF recommendation was agreed upon in 1986 and this recommendation conforms well with the ISO tolerance tables. The main points of the uniform SKF recommendations for tapered journals for spherical roller bearings, which are also valid for CARB toroidal roller bearings, are as follows:

- The permissible taper deviation for machining the taper seats is a ± tolerance in accordance with IT7/2 based on the bearing width B. The IT7/2 tolerance is divided by the bearing width B, in order to convert this length tolerance into a corresponding angle tolerance. (→ **fig. 1.20**).
- The straightness tolerance is IT5/2 and is defined in accordance with **fig. 1.20**. In each axial plane through the tapered surface of the shaft, the tolerance zone is limited by two parallel lines a distance IT5/2 apart.
- The radial deviation from circularity, defined in accordance with **fig. 1.20**, is to comply with IT5/2. In each radial plane along the tapered shaft, the tolerance zone is limited by two concentric circles a distance IT5/2 apart. When particularly stringent run-out requirements are stipulated e.g. when spherical roller bearings with extra reduced circular radial run-out tolerance C08, VA460 or VQ424 are used, IT4/2 should be applied instead.

Gauging the taper deviation

The best way to check that the taper is within the recommended tolerances is to measure with dial gauges. A more practical, but less accurate, method is to use ring gauges or special taper gauges. Tapered seats up to around 150 mm in diameter are generally checked with ring gauges, and those above this size with special taper gauges. When Prussian blue is used, the area in contact should be at least 90% when shaft is new. At least 80% is accepted for an already used shaft. Using a new bearing inner ring as a ring gauge can give lower contact area than 80% even if the bearing and journal are within tolerances.

The gauge recommended by SKF in Europe can be seen in **fig. 1.21**. The taper deviation and the diameter of the seat in relation to a reference surface can be measured with this gauge. The tolerance $M_1 - M$ is calculated by means of the equation

$$M_1 - M = \pm \left(\frac{IT7}{2} \times \frac{G}{B} \right)$$

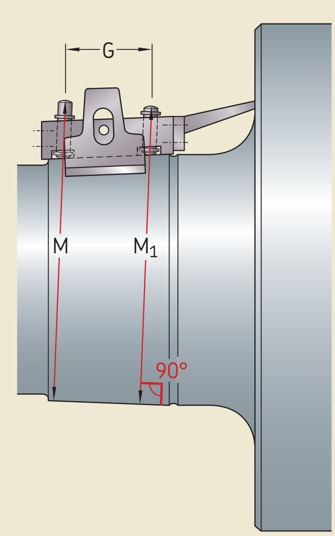
where

G = distance between the points of measurement, mm

B = bearing width, mm

Gauge for tapered journal, European design

Fig. 1.21



Example: Spherical roller bearing 23152 CCK/C4W33 (bearing width 144 mm, IT7/2 for 144 mm is 0,020 mm). The taper is to be measured using our recommended gauge in **fig. 1.21**, **page 1:19**. If a gauge with 80 mm distance between the measuring points (G) is used, the taper deviation ($M_1 - M$) is allowed to be

$$\pm 0{,}020 \times \frac{80}{144} = \pm 0{,}011 \text{ mm}$$

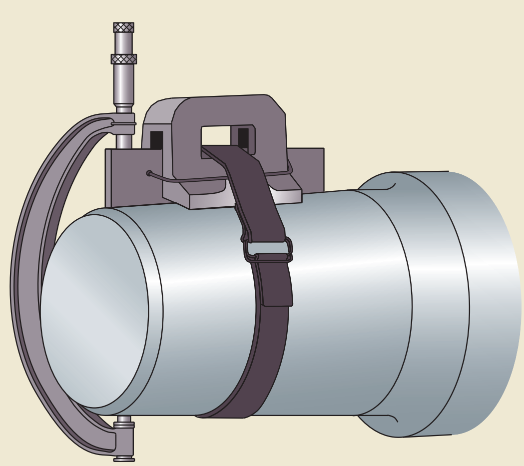
The gauge supplied by SKF in North America can be seen in **fig. 1.22**. It is called the sine bar gauge and the taper deviation can be measured with this tool.

Support surface for housing base

It is recommended that the support surface for the housing is finished to $R_a \leq 12{,}5$ µm. The flatness tolerance should be to IT7 on the length of the support surface.

SKF gauge for tapered journal, American design

Fig. 1.22



1

Oil flow resistance in a spherical roller bearing

Normally, oil flow resistance in a bearing is insignificant. Sometimes, however, the question arises as to what pressure is needed to force oil with a certain viscosity through the duct formed by the W33 lubrication groove and the housing into the interior of the bearing.

The oil pressure required to overcome the oil flow resistance of the duct can be calculated with the following equation

$$\Delta p = \frac{Q\, D\, \nu}{6\,132\, d_h^{\,4}}$$

where

Δp = required oil pressure, MPa
Q = oil flow, l/min
D = bearing outside diameter, mm
ν = kinematic viscosity of oil at lowest operating temperature (generally at start-up), mm^2/s
d_h = hydraulic diameter, mm

The hydraulic diameter d_h is a calculated value describing a virtual diameter which is equivalent to the groove cross section.

The calculation takes into consideration the resistance of the duct only. The length of the holes is very short, compared with the length of the duct, and therefore the resistance of the holes has been ignored.

The hydraulic diameter of the duct formed with different sizes of groove, as well as the groove dimensions, can be obtained from **table 1.3**. The groove numbers used for different spherical roller bearings are also listed in **table 1.3**.

When the required oil pressure is considered to be too high, and to ensure oil supply into the non-locating bearings, an extra groove can be turned in the bearing housing. The hydraulic diameter for the enlarged duct can be calculated from

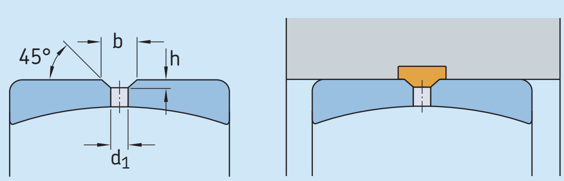
$$d_h = \frac{4\,A}{O_a}$$

where

d_h = hydraulic diameter, mm
A = enlarged duct area, mm^2
O_a = circumference of enlarged duct area, mm

Groove sizes and dimensions and hydraulic diameter for spherical roller bearings

Table 1.3



Groove No.	Groove dimension b	h	d_1	Hydraulic diameter d_h
–	mm			mm
1	5,5	1,2	3,0	1,72
2	8,3	1,8	4,6	2,59
3	11,1	2,4	6,0	3,45
4	13,9	3,0	7,5	4,32
5	16,7	3,6	9,0	5,18
6	22,3	4,8	12,0	6,92

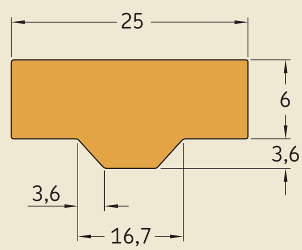
Bearing size[1)]

Series 239 over	incl.	230 over	incl.	240 over	incl.	231 over	incl.	241 over	incl.	222 over	incl.	232 over	incl.	223 over	incl.	**Groove no.**
32	38	20	24	22	30	19	20	20	26	07	18	17	18	08	11	1
38	52	24	30	30	38	20	28	26	34	18	22	18	26	11	17	2
52	72	30	34	38	56	28	30	34	48	22	28	26	30	17	20	3
72	80	34	48	56	64	30	38	48	60	28	32	30	36	20	24	4
80	96	48	64	64	76	38	60	60	76	32	44	36	44	24	34	5
96		64		76		60		76		44		44		34		6

1) The figures in the table represent the last two figures of the designation; thus, bearing 23024 has groove no.1

Example: dimensions of enlarged area

Fig. 1.23



Example

What oil pressure is needed to pump an oil flow of 5 l/min through the duct formed by the W33 groove and the housing, into the bearing 23052 CCK/C4W33? The maximum kinematic viscosity of the oil is 220 mm^2/s.

The table shows that the groove number is 5, and thus d_h = 5,18 mm.

$$\Delta p = \frac{5 \times 400 \times 220}{6\,132 \times 5{,}18^4} = 0{,}1 \text{ MPa}$$

By turning a 25 mm wide and 6 mm deep groove in the housing seating, the enlarged duct area will be (→ **fig. 1.23**)

$$A = 25 \times 6 + 3{,}6\,(16{,}7 - 2 \times 3{,}6) + 3{,}6^2 = 197{,}2 \text{ mm}^2$$

and the circumference

$$O_a = 25 + 2 \times 6 + 25 - 2 \times 3{,}6 + \frac{2 \times 3{,}6}{\cos 45} = 65 \text{ mm}$$

The hydraulic diameter thus becomes

$$d_h = \frac{4 \times 197{,}2}{65} = 12{,}14 \text{ mm}$$

and the required pump pressure

$$\Delta p = \frac{5 \times 400 \times 220}{6\,132 \times 12{,}14^4} = 0{,}0033 \text{ MPa}$$

The required pump pressure without a groove is 0,1 MPa. By turning a groove 25 × 6 mm in the housing seating, a considerable reduction, to 0,0033 MPa, is obtained.

High-speed machines

The operating speed of paper machines has increased significantly over the years.

Increased requirements on run-out tolerances

With production speeds higher than 900 m/min, there is an increased demand for bearings with extra-reduced circular radial run-out tolerances. The main reasons for these are to keep high paper quality and to avoid felt wear as well as paper tears. These aims can be achieved by using C08, VQ424 or VA460 bearings to obtain low vibration levels and a constant nip pressure. Press rolls, calender rolls and suction rolls are examples of high speed applications where these high precision bearings are often used.

C08 means that circular radial run-out of the inner and outer rings correspond to ISO tolerance class P5. Note that all SKF spherical roller bearings and CARB toroidal roller bearings up to and including 300 mm bore diameter have circular radial run-out tolerance class P5.

VQ424 means that the bearing has a circular radial run-out at least 20% better than P5 (C08) and reduced width tolerance. Additionally, the bearing is according to W4 and W58 specifications, carrying a serial number. VQ424 and C08 bearings are an advantage in all rolls that form a nip. VQ424 bearings are also an advantage when regrinding the rolls. By using such bearings, less time is needed and better total run-out of the roll is achieved.

VA460 means that the bearing has VQ424 features and is modified to withstand the effects of centrifugal forces due to very high speeds.

Reduced circular radial run-out of the bearing is not the only way to reduce vibrations in paper machines. Another way is to decrease the total run-out of the journal.

Bearings mounted on adapter or withdrawal sleeves are therefore mainly seen in old machines.

The use of a spherical roller bearing as the locating bearing, in combination with a CARB toroidal roller bearing as the non-locating bearing, has often resulted in less vibration compared to the traditional solution using two spherical roller bearings.

NoWear bearings

One way to improve the performance of a bearing is to provide a beneficial contact condition in cases where adequate lubrication is difficult to obtain. Surface engineering to obtain a low coefficient of friction is a means of achieving this. A commonly used method in industry is surface coating. For a coating to work effectively in a bearing, it has to meet a set of requirements such as hardness, ductility and fatigue resistance in order to stay on the surface during operation. The low friction ceramic coating in NoWear bearings from SKF is specially developed for rolling bearings.

NoWear bearings provide long-term low friction and low wear properties by having a surface layer with a hardness of 1 200 HV. The coefficient of friction between coated rollers and steel is roughly one third of the friction between two steel components. NoWear bearings prevent wear and smearing in bearings which operate at heavy load as well as bearings which operate at radial loads less than the recommendations in the SKF catalogue *Rolling Bearings*.

In fast machines with large and heavy bearings which may operate at radial loads less than the recommendations in the SKF catalogue *Rolling Bearings*, SKF recommends NoWear bearings with coated rollers (designation suffix L5DA). One example of such an application can be the upper thermo roll in soft calenders.

Speed rating – cooling

Sometimes the limiting speed in the SKF catalogue *Rolling Bearings* is mistakenly taken to be the maximum operating speed for the bearing. The limiting speed values in product tables are practical recommendations for general applications. They have a rather conservative safety margin.

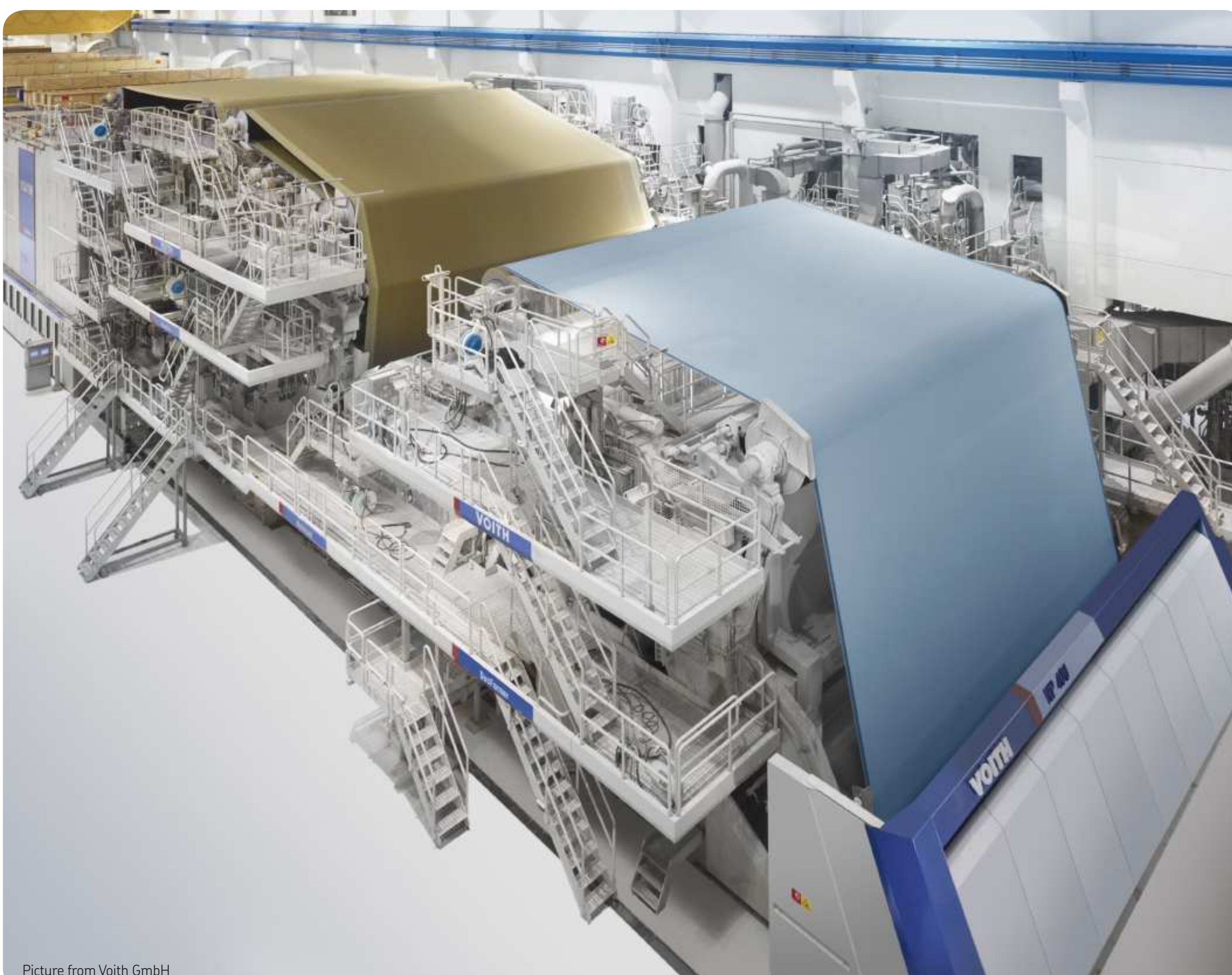
SKF has run a 230/500 spherical roller bearing with C08 running accuracy at twice the SKF catalogue *Rolling Bearings* speed limit.

So, speed limits are not the absolute maximum permissible speeds. They can be exceeded provided necessary measures have been taken regarding bearing design (e.g. running accuracy, cage), lubrication, cooling or precision of surrounding parts.

Experience has shown that for heavy spherical roller bearings and CARB toroidal roller bearings above 500 mm bore in high speed press rolls, a higher clearance class than normally recommended may be necessary to avoid uncontrolled radial preload during machine start up. Also, above a certain oil flow, oil doesn't remove more heat from the bearing assembly, but instead will create more heat due to increased drag losses.

As a guideline, SKF recommends bearings with increased running accuracy, such as C08 and VQ424, be used for speeds above the speed rating indicated in the SKF catalogue *Rolling bearings* and VA460 bearings for speeds above the limiting speed. Please contact SKF application engineering for more information.

Modern paper machine



Picture from Voith GmbH

Picture from Voith GmbH

Forming section

The forming section is the first part of the paper machine. The stock contains around 99% water at this stage. By the end of the forming section, the water content has to be about 80% to make the paper web self-supporting as it moves on to the press section. Depending on the design of the forming section, modern paper machines can be classed as either Fourdrinier machines or twin wire machines.

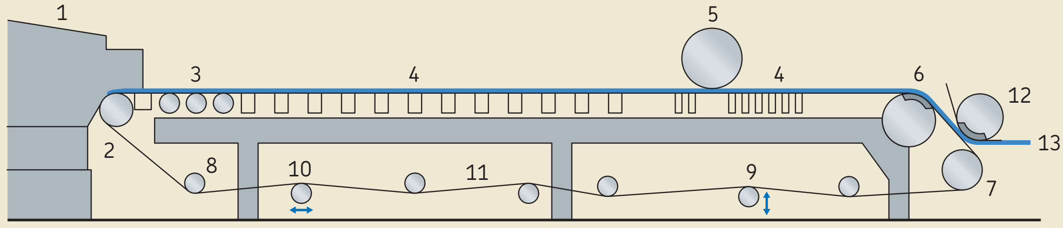
Fourdrinier machines

The Fourdrinier is the original forming unit and is still widely used on slow and medium speed machines. It can be used for virtually all types of paper and for machine speeds up to 900 m/min. **Fig. 2.1** shows the wire part of such a machine. Some wire parts are equipped with a dandy roll to flatten the top surface of the sheet, giving it a smoother finish, and to make watermarks in fine papers.

At the end of the wire part, there is a suction couch roll that reduces the water content to 80% to make the paper web self-supporting. The suction couch roll is equipped with a drive system and drives the whole wire part.

Large machines and machines operating at high speeds are usually equipped with an additional drive roll called the forward drive roll.

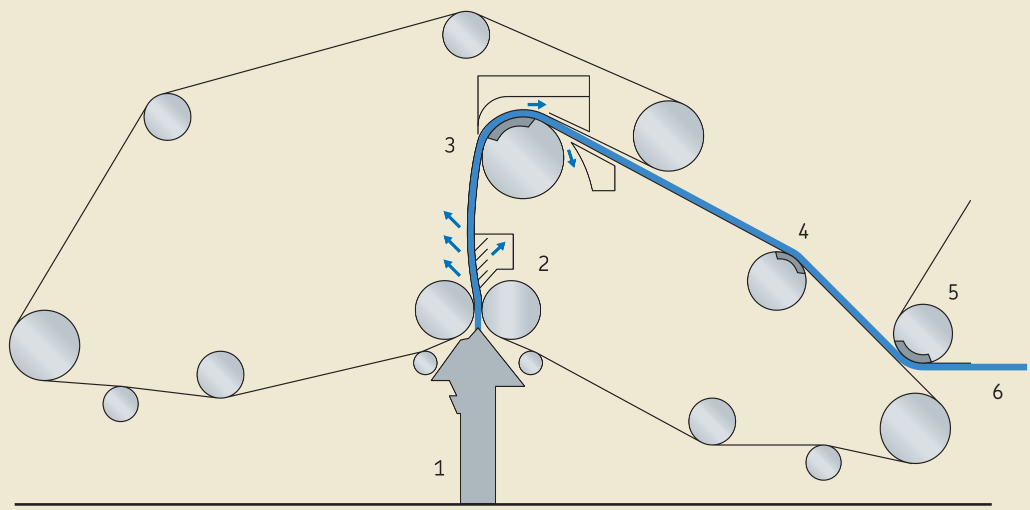
Fig. 2.1



Wire part of Fourdrinier machine
__1__ Headbox
__2__ Breast roll
__3__ Table roll
__4__ Suction box
__5__ Dandy roll
__6__ Suction couch roll
__7__ Forward drive roll
__8__ Wire roll
__9__ Wire stretch roll
__10__ Wire guide roll
__11__ Wire
__12__ Suction pickup roll
__13__ Paper web

Blade former, Beloit (Valmet) Bel Baie III
1 *Headbox*
2 *Forming shoe and blades*
3 *Primary suction roll*
4 *Secondary suction roll*
5 *Suction pickup roll*
6 *Paper web*

Fig. 2.2



Twin wire machines

It is becoming increasingly common to dewater stock between two wires. This process was developed in the early 1970s to make dewatering and forming possible at high speeds. It is possible to distinguish between three different types of twin wire machines: blade formers, roll formers and top wire formers.

Blade formers

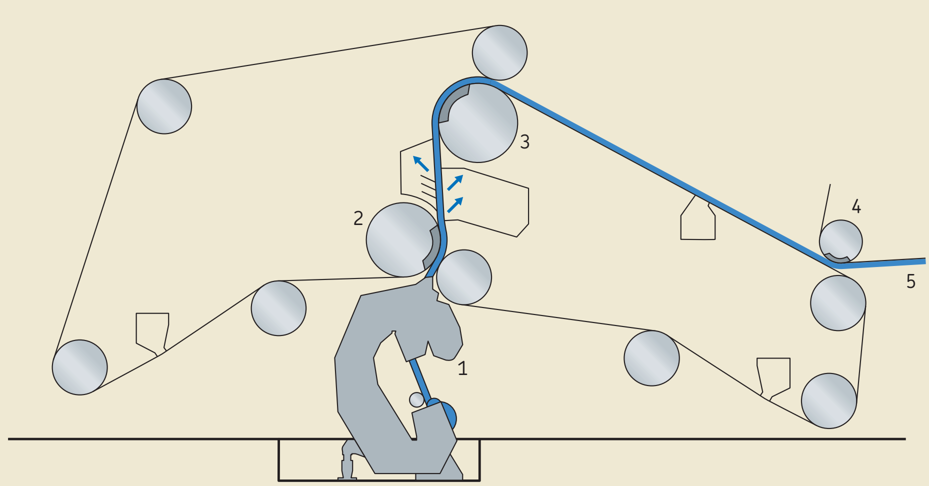
In a blade former, the dewatering zone can be curved (→ **fig. 2.2**). The curved dewatering zone is followed by suction rolls.

Roll formers

In a roll former, dewatering is accomplished between two wires which run around part of a relatively large forming roll. In most cases, a suction roll is used as the forming roll (→ **fig. 2.3**).

In the case of tissue machines with web speeds up to 2 500 m/min, centrifugal force is utilized in the dewatering process. The forming roll is plain and dewatering is outwards only (→ **fig. 2.4**).

Fig. 2.3



Roll former, Voith Paper DuoFormer TQv
1 *Headbox*
2 *Forming (suction) roll*
3 *Suction couch roll*
4 *Suction pickup roll*
5 *Paper web*

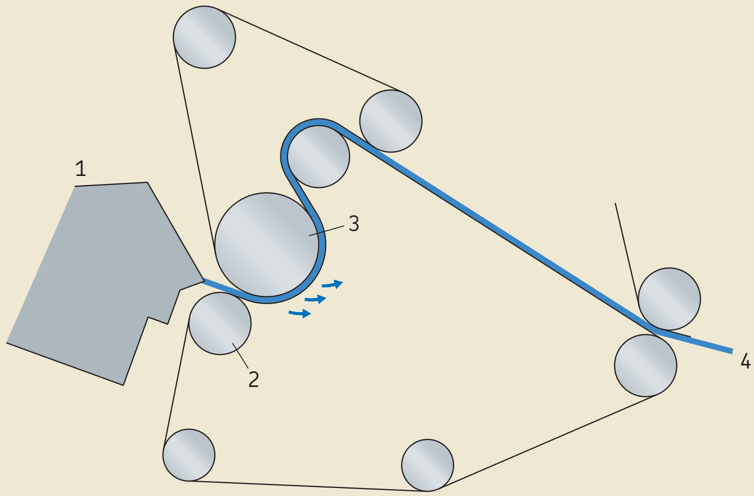


Fig. 2.4

Roll former, Valmet PeriFormer
1 *Headbox*
2 *Breast roll*
3 *Forming roll (plain)*
4 *Paper web*

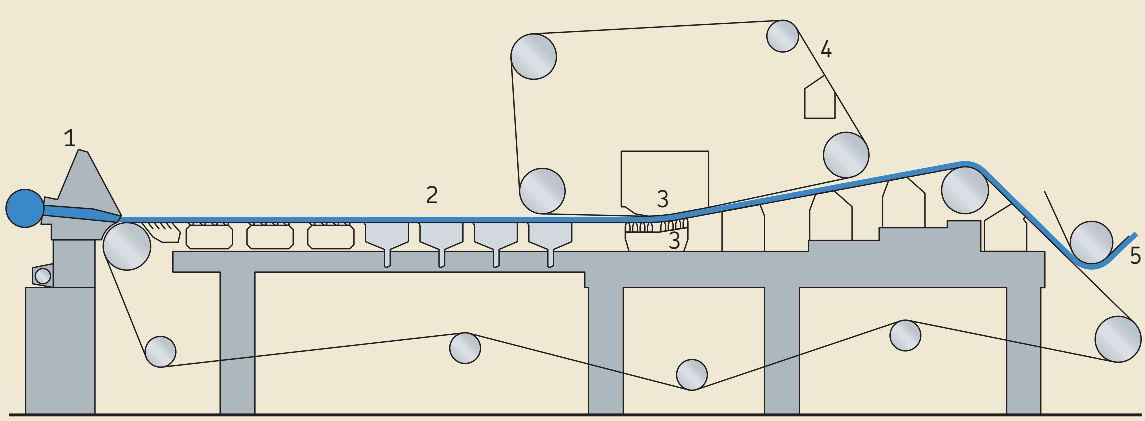
Top wire formers

In a top wire former, the dewatering and forming process starts on a Fourdrinier wire. Then the paper web passes through a nip between the Fourdrinier wire and a top wire where most of the dewatering in the wire part is carried out (→ **fig. 2.5**).

Existing Fourdrinier machines (→ **fig. 2.1**, **page 2.1**) can easily be upgraded by the addition of a top wire to increase capacity.

Top wire former, Voith DuoFormer D
1 *Headbox*
2 *Suction box*
3 *Forming blades and top suction box*
4 *Top wire*
5 *Paper web*

Fig. 2.5



Bearing arrangements

Breast and forward drive rolls

Different bearing arrangements may be used for breast and forward drive rolls. In **fig. 2.6**, the bearing housing is of the plummer (pillow) block type. The housing has its end cover designed to provide support for the extended journal during lifting.

In **fig. 2.7**, the bearing housing is sphered externally and stabilized by the use of an extra support bearing mounted on the end of the journal. The housing is carried in a sphered bracket connected to the frame. During wire changes, the roll is suspended by the bearing housing end covers in line with the support bearing. A spherical or a cylindrical roller bearing may be used as the support bearing.

In some old machines, the main bearing was positioned on the load direction line. This load was created by the contact housing/frame and, in some cases, the support bearing could become unloaded. As such, to avoid failures due to an unloaded bearing running at high speed, the main bearing is now often axially displaced a few millimetres inwards to load the support bearings. If needed, space can be provided in one of the housings to allow for axial movement of the non-locating bearing.

These bearing arrangements work under very wet conditions. The housings should therefore incorporate efficient seals irrespective of whether grease or oil lubrication is employed. For CARB toroidal roller bearings, the lubricant must be supplied from the side. For spherical roller bearings, the lubricant can be supplied either from the side or via the groove and the holes in the outer ring. An annular groove turned in the housing, so as to coincide with the holes, improves the entry of lubricant. In designs incorporating a support bearing, separate arrangements must be made for its lubrication.

Bearing types

Breast and forward drive rolls are carried by spherical roller bearings, series 232 and 223 and CARB toroidal roller bearings series C 32 and C 23, mounted on adapter or withdrawal sleeves or direct on tapered journals.

A spherical or a cylindrical roller bearing may be used as the support bearing.

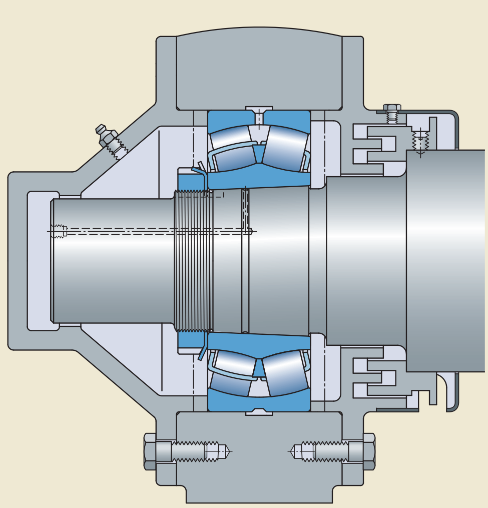
The main bearings, as well as the support bearings, are usually selected with Normal radial internal clearance. Sealed spherical roller bearings can improve service life by adding extra protection against contamination.

Selection of bearing size

Bearing size selections should be based on life calculations according to the recommendations in *chapter 1*, *General requirements and recommendations*. SKF recommends that both the basic rating life L_{10h} and the SKF rating life L_{10mh} are taken into consideration. The calculated basic rating life L_{10h} should exceed 120 000 hours, while a SKF rating life L_{10mh} of 100 000 hours is recommended. There are some exceptions due to the fact that too lightly loaded bearings can fail even if the calculated life is over 100 000 hours. SKF recommend checking that the minimum load criterion is met (for details see the product chapters in the SKF cata-

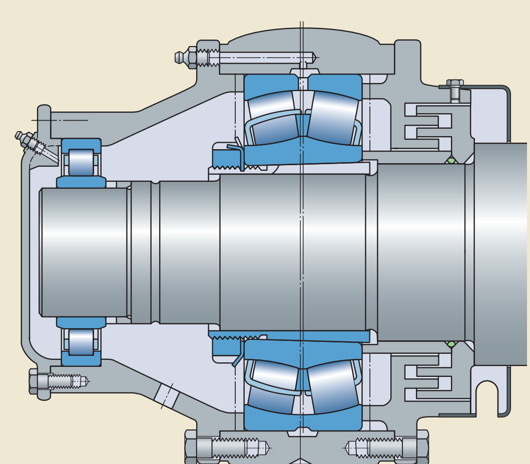
Breast and forward drive roll bearing arrangement

Fig. 2.6



Breast and forward drive roll bearing arrangement

Fig. 2.7



logue *Rolling bearings*). The wire tension of 3–7 kN/m of roll length and the mass of the roll are to be taken into consideration when calculating the bearing loads. When the housings are mounted on shake rails, the axial accelerations must be taken into account.

Lubrication

The most important factors for effective lubrication are the lubricant viscosity, achieving a satisfactory degree of surface separation between the rolling contact surfaces, and the lubricant cleanliness with respect to water and solid particles. Protection against corrosion has top priority for these bearing positions. Therefore, the lubricant must have good rust inhibiting properties and effective seals need to be provided.

Sometimes, oil rather than grease lubrication is selected for these bearings. There are several reasons for this. One is the increased operating temperatures in high-speed machines requiring excessive grease quantities when relubricating and/or too short relubrication intervals. Another reason is that there can be other oil-lubricated bearings nearby. When oil lubrication is selected, the seals must be modified to suit oil lubrication. An example of such a seal arrangement is shown in **fig. 2.8**.

For further information, see *chapter 7* and *chapter 8*.

Journal and housing tolerances for breast and drive rolls

See the indications given in *Tolerances* in *chapter 1*.

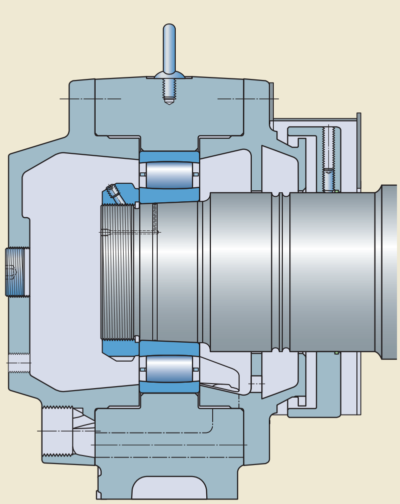
If the bearing is mounted on a tapered journal, see *Tolerances for tapered journal seats* in *chapter 1*.

If the journal is cylindrical, follow the recommendations given in **table 2.1** The recommended journal tolerances for the main bearing have changed and give tighter fits than in the 4th edition of the *Rolling bearings in paper machines* handbook. As such, the main bearings may need to have C3 radial internal clearance because of increased radial clearance reduction due to the increased tight fit.

For housing tolerances, follow the recommendations in **table 2.1**.

Wire and felt roll bearing arrangement

Fig. 2.8



Journal and housing tolerances for breast and drive rolls

Table 2.1

Cylindrical journal	Mounting on sleeve		h9Ⓔ
	Total radial run-out		IT5/2
	Direct mounting		
	(60) to 100 mm		n6Ⓔ
	(100) to 200 mm		p6Ⓔ
	Support bearing seating		
	Cylindrical roller bearing	Spherical roller bearing	
	(65) to 140 mm	(65) to 100 mm	m6Ⓔ
	(140) to 200 mm	(100) to 140 mm	n6Ⓔ
		(140) to 200 mm	p6Ⓔ
Housing			G7Ⓔ

Wire rolls

Wire roll bearings in existing machines with moderate speeds are mainly grease lubricated and mounted on sleeves but, with increasing paper speeds, the use of oil lubrication and of bearings mounted direct on the journal is preferred.

Fig. 2.8, shows an oil lubricated bearing arrangement for wire rolls. The housing is the plummer (pillow) block type. The CARB toroidal roller bearing is mounted directly on the shaft, but mounting on a withdrawal sleeve is also possible. The bearing arrangement is similar to that for press rolls.

Fig. 2.9 shows a bearing arrangement, with two bearings in the same housing, for guide and stretch rolls. The housing is sphered externally. The bearing arrangement is designed so that one of the bearings takes up the main load. The function of the support bearing is to keep the housing and journal axes in line during wire tension adjustment and wire guidance.

The design of the bearing arrangement for guide and stretch rolls depends on the design of the associated components. The support bearing may be a spherical or a cylindrical roller bearing and must be mounted at a certain distance from the load carrying bearing. The arrangement shown in **fig. 2.9** uses a spherical roller bearing as the support bearing. Note that the support bearings in both housings must be able to accommodate axial displacement.

In some old machines, the main bearing was positioned on the load direction line. This load was created by the contact housing/frame and, in some cases, the support bearing could become unloaded. As such, to avoid failures due to an unloaded bearing running at high speed, the main bearing is now often axially displaced a few millimetres inwards to load the support bearings.

If needed, space can be provided in one of the housings to allow for axial movement of the non-locating bearing.

Fig. 2.10 shows a grease lubricated CARB toroidal roller bearing arrangement for wire rolls with improved protection against water and particle entrance. The housing is provided with a protective cover.

For CARB toroidal roller bearings, the grease must be supplied from the side. For spherical roller bearings, the grease can either be supplied from the side or via the groove and the holes in the outer ring. In designs incorporating a support bearing, separate arrangements must be made for its lubrication.

Bearing types

SKF recommends the use of spherical roller bearings of series 223 and 232 and CARB toriodal roller bearings of series C 23 and C 32, but spherical roller bearings of series 222 and CARB toroidal roller bearings of series C 22 can also be used.

Both spherical roller bearings of series 222 and cylindrical roller bearings of series NUB 2, with wide inner ring, can be used as the support bearing.

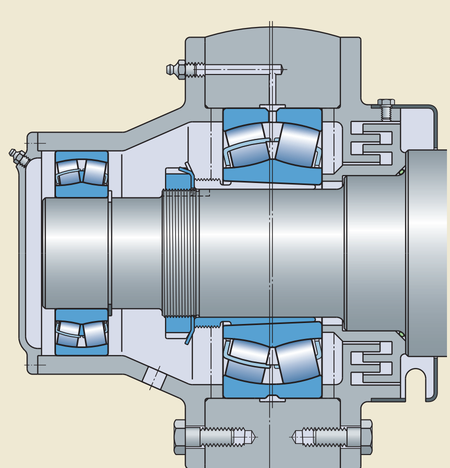
The main bearings, as well as the support bearings, are usually selected with Normal radial internal clearance.

Selection of bearing size

Bearing size selections should be based on life calculations according to the recommen-

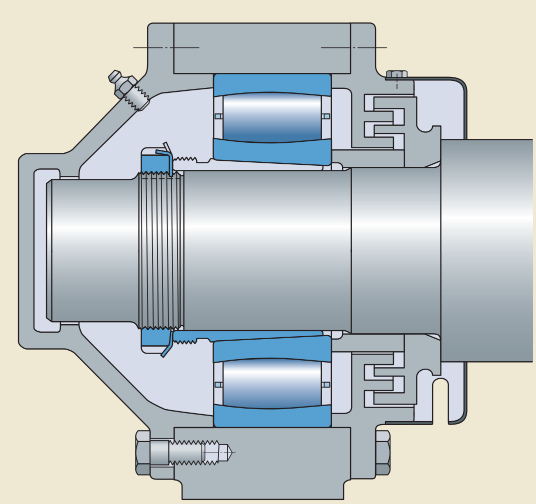
Guide and stretch roll bearing arrangement

Fig. 2.9



Wire roll bearing arrangement with grease lubrication and protective cover. As there is no O-ring between the spacer and shaft to prevent water contamination, the application of seal paste between the spacer and shaft is recommended.

Fig. 2.10



dations in *chapter 1*, *General requirements and recommendations*. SKF recommends that both the basic rating life L_{10h} and the SKF rating life L_{10mh} are taken into consideration. The calculated basic rating life L_{10h} should exceed 120 000 hours, while a SKF rating life L_{10mh} of 100 000 hours is recommended. There are some exceptions due to the fact that too lightly loaded bearings can fail even if the calculated life is over 100 000 hours. SKF recommend checking that the minimum load criterion is met (for details see the product chapters in the SKF catalogue *Rolling bearings*). If the mass of the wire roll is known, the maximum radial bearing load can be roughly estimated with the aid of the following equations:

$$G = g\ m/1\ 000$$

$$K_r = 2\ q_1\ L + G$$

$$F_r = 0,5\ Kr$$

$F_a = \mu\ F_r$ (for a spherical roller bearing as non-locating bearing)

$F_a = 0$ (for a CARB toroidal roller bearing as non-locating bearing)

where
G = roll weight, kN
g = 9,81 (acceleration of gravity), m/s^2
m = roll mass, kg
K_r = roll load, kN
q_1 = wire tension, kN/m
L = wire width, m
F_r = radial bearing load, kN
F_a = axial bearing load, kN
μ = coefficient of friction between housing and outer ring (use $\mu = 0,15$ in the calculation)

Generally, it can be assumed that the wire tension is 3–9 kN/m over the length of the roll. The axial bearing load caused by wire guidance can be ignored in the case of wire rolls. In some cases, the wire tension can lift the roll and cause the bearing to run with insufficient load, making the rollers slide instead of roll. The lowest bearing load must also be calculated. The minimum roll load K_r is obtained when the wire tension lifts the roll.

Lubrication

The most important factors for effective lubrication are the lubricant viscosity, achieving a satisfactory degree of surface separation between the rolling contact surfaces, and the lubricant cleanliness with respect to water and solid particles. Protection against corrosion has top priority for this bearing position. Therefore, the lubricant must have good rust-inhibiting properties and effective seals need to be provided.

Sometimes oil lubrication rather than grease lubrication is selected for these bearings. There are several reasons for this. One is the increased operating temperatures in high-speed machines requiring excessive grease quantities when relubricating. Another reason is that there are other oil lubricated bearings nearby.

For further information, see *chapter 7*, *Lubrication*, and *chapter 8*, *Lubrication examples*.

Journal and housing tolerances for wire rolls

See the indications given in *Tolerances* in *chapter 1*.

If the bearing is mounted on a tapered journal, see *Tolerances for tapered journal seats* in *chapter 1*.

If the journal is cylindrical please follow the recommendations in **table 2.2**. Recommended journal tolerances for the main bearing have changed and give tighter fits than in the 4th edition of the *Rolling bearing in paper machines* handbook. The main bearings may need to have C3 radial internal clearance because of increased radial clearance reduction due to the increased tight fit.

For housing tolerances, follow the recommendations in **table 2.2**.

Journal and housing tolerances for wire rolls

Table 2.2

Cylindrical journal	Mounting on sleeve		h9Ⓔ
	Total radial run-out		IT5/2
	Direct mounting		
	(60) to 100 mm		n6Ⓔ
	(100) to 200 mm		p6Ⓔ
	Support bearing seating		
	Cylindrical roller bearing	Spherical roller bearing	
	(40) to 100 mm	(40) to 65 mm	m5Ⓔ
		(65) to 100 mm	m6Ⓔ
Housing			G7Ⓔ

Suction rolls

Fig. 2.11 shows a grease lubricated bearing arrangement of older design in which the front-side bearing is mounted at the end of the roll and the drive-side bearing is mounted on a withdrawal sleeve. Note that the stationary inner ring of the front-side bearing is mounted as a non-locating ring with a clearance fit on the sleeve to allow axial freedom of movement. Grease is supplied through holes in the inner ring. SKF bearings of this design have the suffix W513 (W513 = W26 + W33) in their designation, e.g. 23060 CC/W513.

One disadvantage with the bearing being mounted at the front-side end of the roll is that it is difficult to make the internal seal efficient enough to prevent the ingress of water. The suction box support bearing can be a sealed spherical roller bearing, with the seal on one side removed, to improve protection (→ **fig. 2.12**). Suction rolls of this design are not very common today, but can be found in older low speed machines.

The design shown in **fig. 2.13** is an improved version of the design shown in **fig. 2.11**. This version has oil lubrication of the main bearings and grease lubrication of the internal bearing. In this case, the internal bearing can be a sealed spherical roller bearing with relubrication via the W33 feature in the outer ring (→ **fig. 2.14**).

Suction box support sealed spherical roller bearing with one seal removed

Fig. 2.12

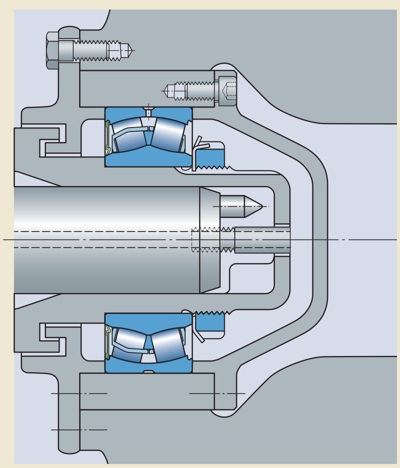
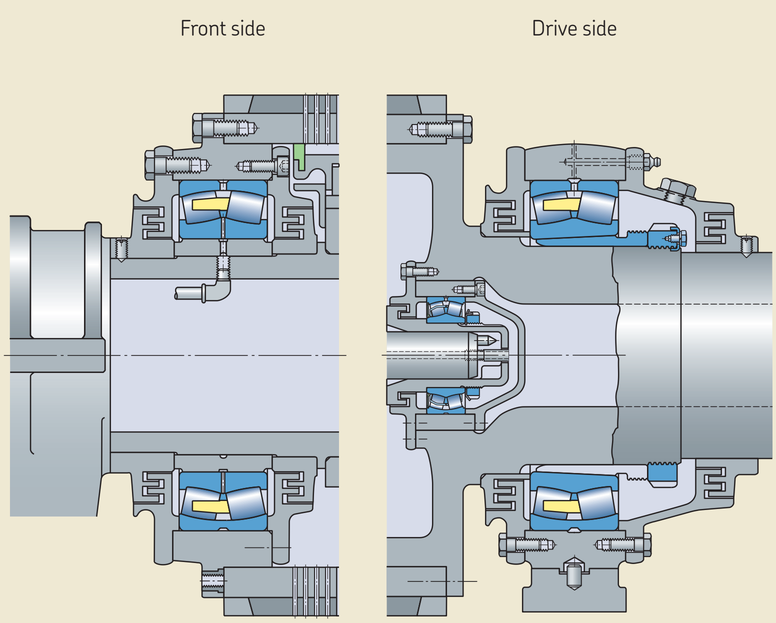


Fig. 2.11



Suction roll bearing arrangements, grease lubricated

In modern machines, the internal bearing is also oil lubricated. With this design, both main bearings are mounted directly on the roll journals and are readily accessible for inspection. When bearings are mounted directly on the journal, oil injection grooves should be provided in the journal so that the bearings can be easily dismounted when the rolls are being serviced. In this design, space is provided in the housing to allow axial movement of the non-locating bearing. The suction box support bearing is often mounted on a cylindrical sleeve and in a housing which is bolted to the back wall of the suction roll.

A development of the design shown in **fig. 2.13** replaces the solid shaft with one with a bore which makes it possible to drain the water at both sides, (→ **fig. 2.15**, **page 2:10**). With this design, the drive shaft is provided with a shaft mounted gear. With high-speed paper machines, the frictional heat in the large bearings on the suction rolls is so great that large flows of circulating oil must be passed through the bearings to dissipate the heat. Therefore, it is important that large drainage ducts are incorporated in the design to cope with these flows.

Suction box sealed spherical roller bearing

Fig. 2.14

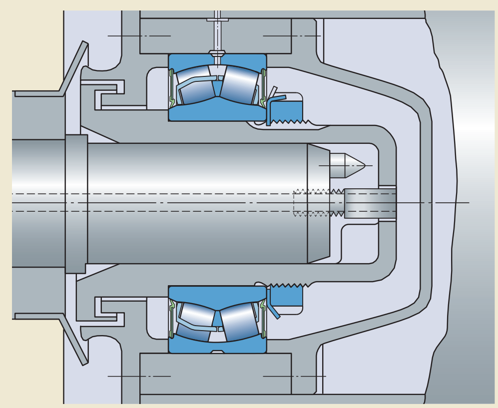
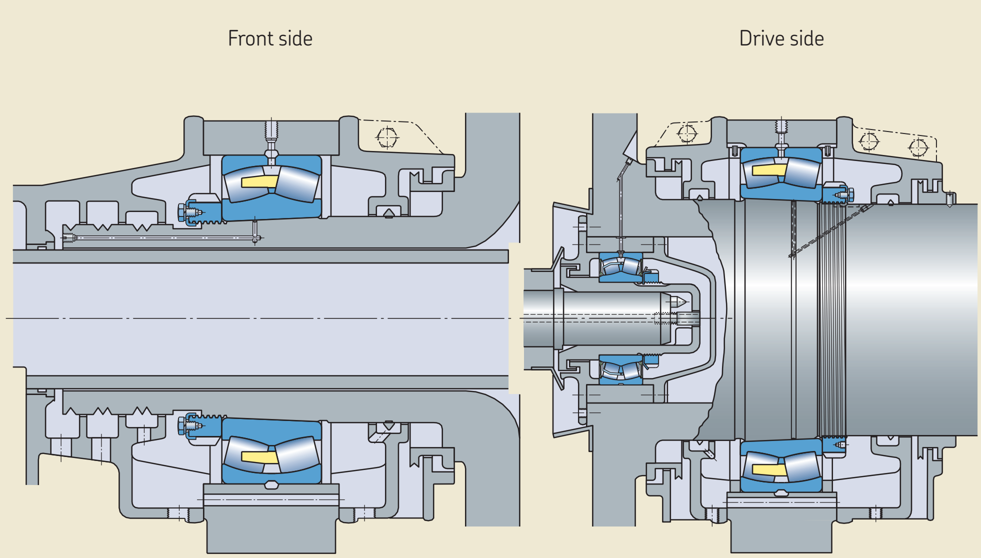


Fig. 2.13



Suction roll bearing arrangement, oil lubricated except the suction box support bearing

Important notice: The main bearings of the suction roll in **fig. 2.15** are mounted with a tight fit and a tapered journal having a large bore. Such journals will deform under load changing from a cylindrical shape into a slight oval shape. The bearing ring will follow this deformation, but if the fit isn't tight enough, there will be micro movement between bearing inner ring and journal, creating fretting corrosion and perhaps creeping and wear. For this type of suction roll design, higher bearing drive-up along the tapered seat is recommended. The internal clearance reduction due to drive-up should be at least 0,65 per thousand of the bearing bore diameter instead of the 0,50 normally recommended (see the sections about the SKF Drive-up Method and SKF SensorMount in *Chapter 9*)

Bearing types

Spherical roller bearings of series 230 and 231 are commonly used for suction rolls and bearings of series 239 are often used for very large diameter journals. Most bearings for suction rolls have C3 radial internal clearance and reduced circular radial run-out (VQ424 or C08).

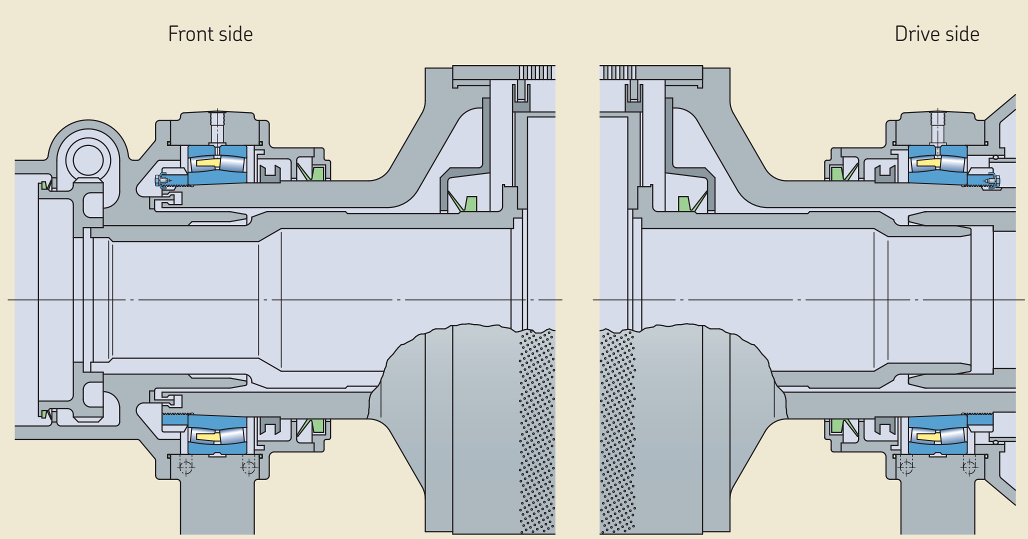
If the suction roll is provided with an internal support bearing, a bearing of series 232, 223 and 241 should be selected. Normal or C3 radial internal clearance is used.

Selection of bearing size

Bearing size selections should be based on life calculations according to the recommendations in *chapter 1*, *General requirements and recommendations*. SKF recommends that both the basic rating life L_{10h} and the SKF rating life L_{10mh} are taken into consideration. The calculated basic rating life L_{10h} should exceed 120 000 hours, while a SKF rating life L_{10mh} of 100 000 hours is recommended. There are some exceptions due to the fact that too lightly loaded bearings can fail even if the calculated life is over 100 000 hours. SKF recommend checking that the minimum load criterion is met (for details see the product chapters in the SKF catalogue *Rolling bearings*). The loads from the following sources have to be considered:

- roll mass
- wire tension
- all press nips
- vacuum

Fig. 2.15



Suction roll bearing arrangements without internal bearing, oil lubricated

Lubrication

The most important factors for effective lubrication are the lubricant viscosity, achieving a satisfactory degree of surface separation between the rolling contact surfaces, and the lubricant cleanliness with respect to water and solid particles. Protection against corrosion has top priority for this bearing position. Therefore, the lubricant must have good rust-inhibiting properties and effective seals need to be provided.

Oil lubrication is often selected for these bearings, especially in new machines. There are several reasons for this. One is the increased operating temperatures in high speed machines requiring excessive grease quantities when relubricating. Another is that the bearing speed exceeds the maximum recommended speed with grease lubrication.

The relative speeds of suction roll bearings are the highest of all bearings in paper machines. When large bearings rotate at high speeds, there is a risk of smearing i.e. sliding of unloaded rollers when they enter the loaded zone. This risk is even higher for press roll bearings because of their heavier rollers. Therefore, the lubricant requirements will be dictated by the press roll bearings.

For further information, see *chapter 7, Lubrication*, and *chapter 8, Lubrication examples*.

Journal and housing tolerances for suction roll

See the indications given in *Tolerances* in *chapter 1*.

If bearing is mounted on a tapered journal, see *Tolerances for tapered journal seats* in *chapter 1*.

If journal is cylindrical please follow the recommendations in **table 2.3**.

For housing tolerances, follow the recommendations in **table 2.3**.

Journal and housing tolerances for suction roll

Table 2.3

Journal	Mounting on sleeve Total radial run-out	h9Ⓔ IT5/2
	Front side bearing as shown in **fig 2.11** **Note:** for bearings with bore diameter above 800 mm a slightly shifted f6Ⓔ may need to be considered to assure a loose fit over the entire tolerance range.	f6Ⓔ
	Suction box support bearing	h6Ⓔ
Housing	Stationary outer ring Rotating outer ring Rotating outer ring for suction rolls with wide opening at the shell for the vacuum zone	G7Ⓔ N7Ⓔ P7Ⓔ
	Suction box support bearing	N7Ⓔ

Picture from Voith GmbH

Press section

On leaving the forming section, the paper web has a water content of around 80%. When the web leaves the press section, the water content may vary between 50 and 65%.

3

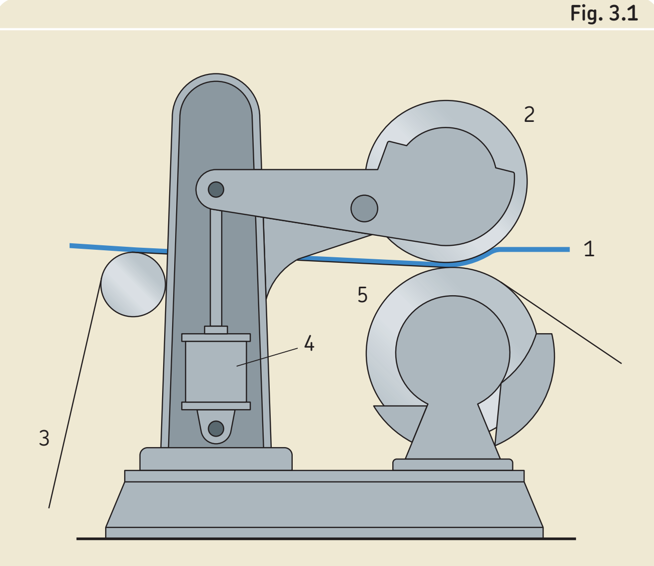
The paper web is fed through the press section on one felt or between two felts. The web passes through a number of press nips which squeeze the water out of the paper into the felt which conveys the water away. Pressing the water out of the paper web is a less expensive process than drying by using steam heating. As a general rule, it may be said that for every 1% of water that is removed in the press section, 4% of the steam utilised in the dryer section will be saved. The pressing process is accordingly being developed intensively.

The linear load, expressed in kN/m, is a benchmark for the pressing action. In ordinary presses, the linear load is between 70 and 150 kN/m, but in so-called shoe presses (wide-nip presses) it can be as much as 1 500 kN/m though 1 200 kN/m is more normal. The main advantage of a wide press nip is that there is more time for the water to be pressed out.

Presses can be basically classified as single-nip and multi-nip presses. Single-nip presses can be sub-divided into plain, suction and shoe presses.

Plain press, old design
1 *Paper web*
2 *Granite roll or steel roll with special cover*
3 *Felt*
4 *Pneumatic piston*
5 *Rubber-covered bottom roll*

Fig. 3.1



Plain presses

The plain press (→ **fig. 3.1**) with two plain rolls and one felt was the original press in paper machines. Plain presses were not particularly effective and were superseded by other types of presses as knowledge of the pressing process increased.

Plain presses have made a comeback in the form of large diameter press rolls. See the section about shoe presses later in this chapter.

Suction presses

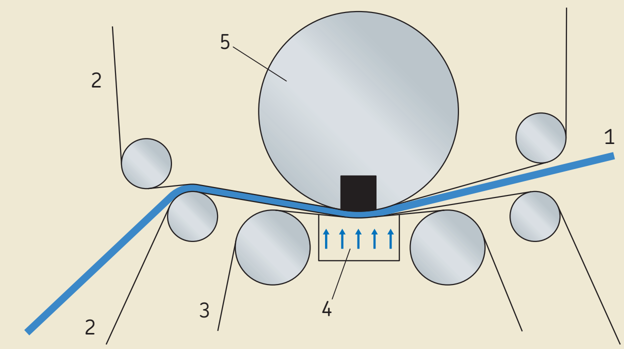
In a suction press, just like in a plain press, pressing occurs between two rolls. The difference is that in the suction press, one of the rolls, usually the bottom one, is a suction press roll. The other roll is a so-called solid press roll made of steel and covered with hard rubber. In older machines, granite rolls can also be seen. The water is pressed and sucked out of the web, via the felt, into the suction press roll.

Shoe presses

Originally, shoe presses (wide-nip presses) were primarily of interest when it came to manufacturing linerboard and fluted paper. Today, shoe presses are used in all types of machines. There are many different kinds of wide-nip presses in use such as Beloit's ENP, Valmet's SymBelt presses and Voith's Paper NipcoFlex presses.

The basic design of the Beloit ENP (Extended Nip Press) is outlined in **fig. 3.2**. The wide press nip is achieved by a concave shoe being pressed against a roll from below. The paper web is conveyed through the nip between two felts. The problem of friction in the contact with the fixed shoe has been solved by placing an endless plastic blanket between the shoe and the bottom felt and coating the blanket with oil prior to the nip so that a supporting oil film is formed between the blanket and the shoe in the nip. The top roll is a CC (crown control) roll covered with a hard plastic material. In the ENP, the nip width is 254 mm and the linear load can reach 1 050 kN/m.

Fig. 3.2



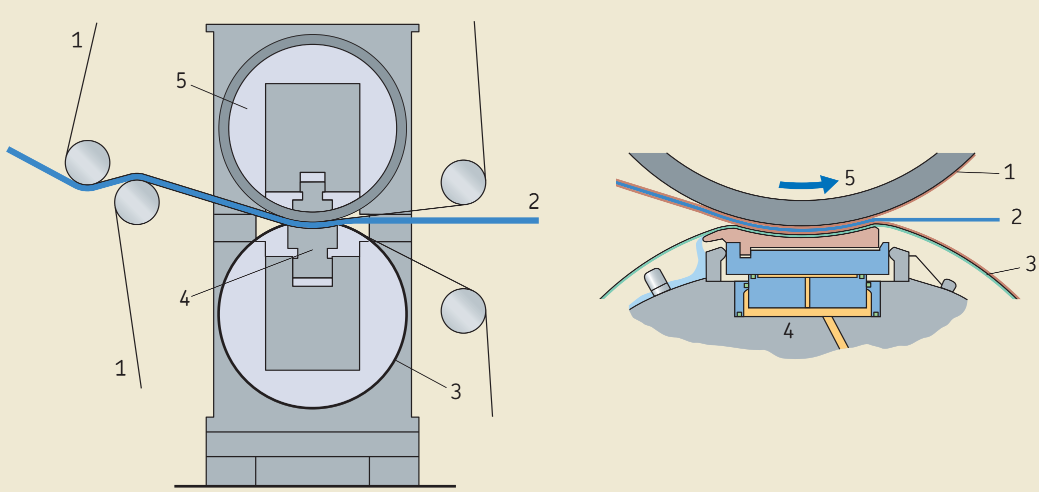
Beloit (Valmet) ENP
1 *Paper web*
2 *Felt*
3 *Plastic blanket*
4 *Concave shoe*
5 *CC roll*

The Voith Paper NipcoFlex press operates in basically the same way as the ENP. **Fig. 3.3** shows the Voith Paper NipcoFlex press.

A wide nip can also be achieved by using large diameter press rolls.

The endless plastic shell is guided by two deep groove ball bearings (→ **fig 3.4**).

Fig. 3.3



Voith Paper NipcoFlex press
1 *Felt*
2 *Paper web*
3 *Flexible plastic shell*
4 *Hydrostatic shoe*
5 *Nipco roll*

Multi-nip presses

Multi-nip presses are now a common feature on all types of paper machines. The multi-nip press is normally made up of two or more press rolls acting against a central roll. **Fig. 3.5** shows the Metso SymPress with multi-nip. In most cases, the central roll is a steel roll with a special cover, but in older machines granite rolls are sometimes used. There are also presses where a suction roll is used. The press rolls around the central roll can be plain rolls, suction rolls or deflection-compensating rolls.

Voith Nipcoflex: the endless plastic shell is guided by two deep groove ball bearings

Fig. 3.4

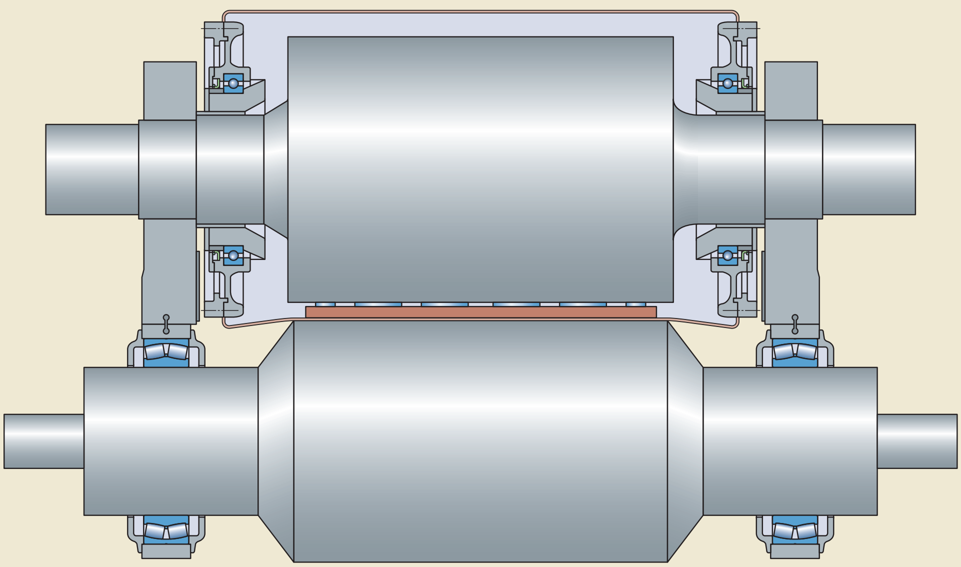
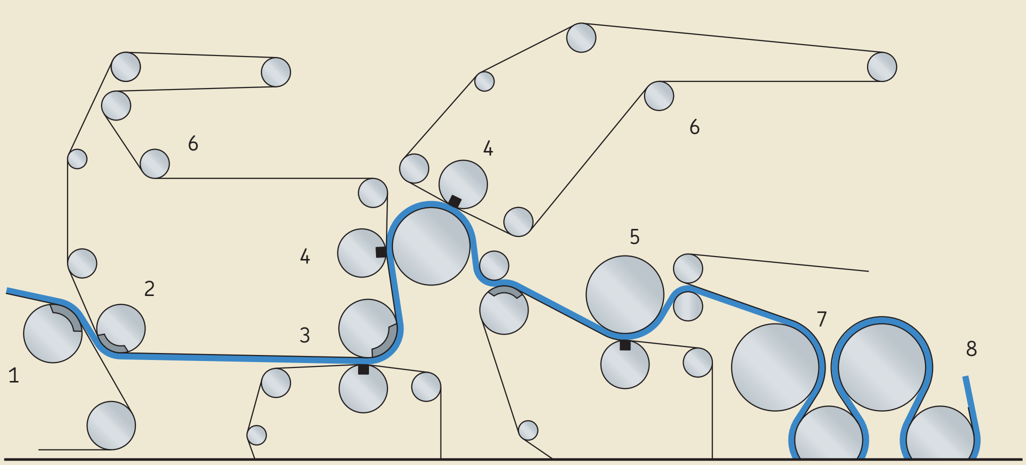


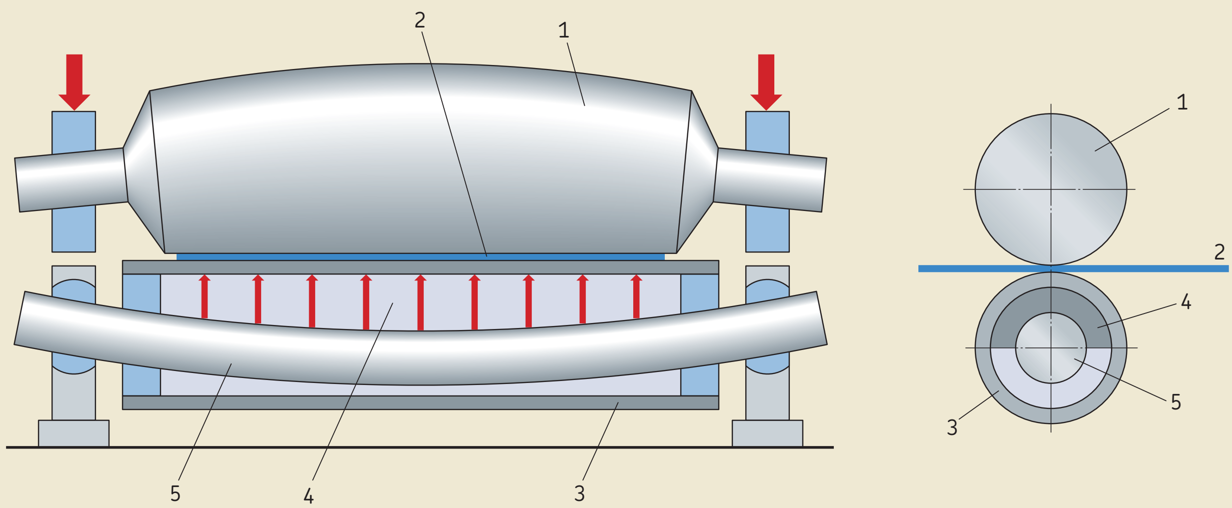
Fig. 3.5



Valmet SymPress press section with multi-nip

1 *Suction couch roll*
2 *Suction pick-up roll*
3 *First nip with a suction roll and Sym roll below*
4 *Second and third nips with two Sym rolls against a central roll*
5 *Fourth nip with a granite roll or steel roll with special cover and Sym roll below*
6 *Felt roll*
7 *Drying cylinder*
8 *Paper web*

Fig. 3.6



Deflection-compensating press roll
1 Top roll
2 Paper web
3 "Flexible" steel shell
4 Pressure chamber
5 Stationary shaft

Deflection-compensating press rolls

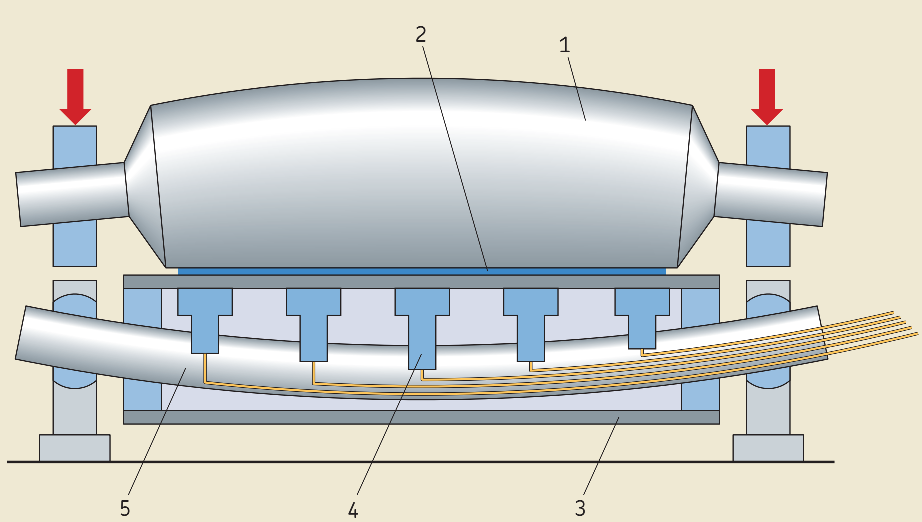
As uniform nip pressure is required to obtain an even paper quality, the ordinary plain press rolls are crowned, i.e. manufactured with a larger diameter in the middle. Deflection-compensating press rolls are also crowned, but in a different way (→ **fig. 3.6**).

Solid press rolls are crowned by grinding to give a uniform linear load over the whole length of the roll. This can only be achieved at a certain roll load. The deflection-compensating roll, on the other hand, is hydraulically crowned and, as a result, a uniform linear load can be achieved irrespective of the roll load.

A deflection-compensating roll is made up of a stationary, fixed through shaft and a shell that rotates around the shaft (→ **fig. 3.6**). Pressurized oil between the shaft and shell ensures that the latter conforms to the shape of the opposing press roll. Deflection-compensating rolls are used as top or bottom rolls and may be driven or non-driven.

In a further development of these rolls, the oil-pressurized chamber has been replaced by a number of hydrostatic shoes individually connected to a pressurized oil system (→ **fig. 3.7**). With these shoes, the shape of the roll shell can be controlled individually in each zone along the length of the roll.

Fig. 3.7



Zone-controlled deflection-compensating press roll
1 Top roll
2 Paper web
3 "Flexible" steel shell
4 Hydrostatic shoe
5 Stationary shaft

Deflection-compensating rolls can be made considerably smaller than plain rolls for the same linear load. This is a great advantage.

These rolls are probably the most important ones when it comes to the production of good quality paper. Most of the major manufacturers of paper machines include this type of roll in their product range, though the actual design used differs a little. The rolls are known by different brand names, such as:

- CC Rolls (Valmet /Beloit)
- Sym rolls (Valmet)
- Nipco rolls (Voith Paper)
- Swimming rolls (Andritz Küsters)

Bearing arrangements

Felt rolls

Spherical roller bearings and CARB toroidal roller bearings are recommended for this position. The bearings in modern machines are mainly oil lubricated and mounted directly on the journals. However, mounting on sleeves is common in old, slow-running machines.

Seals for oil lubricated bearings are normally not as efficient as the multi-stage labyrinths used for grease lubricated bearings. However, seals for oil lubricated bearings in the press section have to be equally efficient due to their exposure to water during operation and hosing down. **Fig. 3.8** shows an arrangement with improved seals for oil lubricated felt roll bearings.

A bearing arrangement for guide and stretch rolls is shown in *chapter 2, Forming section*, **fig. 2.9, page 2:6**. This shows a grease lubricated guide and stretch roll in the wire section, but this arrangement can be applied to the press section as well. However, oil lubrication can also be used for guide and stretch rolls if the seals are modified.

Bearing types

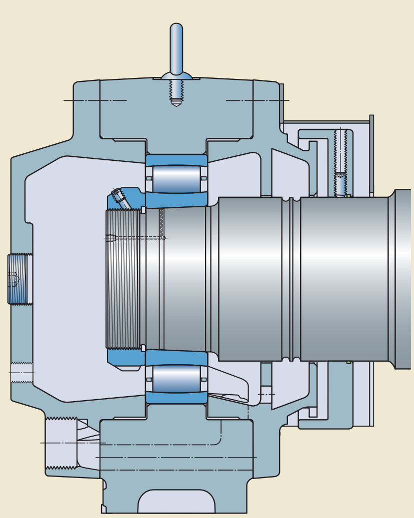
SKF recommends the use of spherical roller bearings of series 223 and 232 and CARB toroidal roller bearings of series C 23 and C 32, but spherical roller bearings of series 222 and CARB toroidal roller bearings of series C 22 can also be used.

Both spherical roller bearings of series 222 and cylindrical roller bearings of series NUB 2, with wide inner ring, can be used as the support bearing. Sealed spherical roller bearings can improve service life by protecting the bearing from contamination.

The main bearings as well as the support bearings are usually selected with C3 radial internal clearance.

Oil lubricated felt roll bearing arrangement with improved seals

Fig. 3.8



Selection of bearing size

Bearing selection should be based on life calculations according to the recommendations in *chapter 1, General requirements and recommendations*. SKF recommends that both the basic rating life L_{10h} and the SKF rating life L_{10mh} are taken into consideration. The calculated basic rating life L_{10h} should exceed 120 000 hours, while a SKF rating life L_{10mh} of 100 000 hours is recommended. There are some exceptions due to the fact that too lightly loaded bearings can fail even if the calculated life is over 100 000 hours. SKF recommend checking that the minimum load criterion is met (for details see the product chapters in the SKF catalogue *Rolling bearings*). If the mass of the felt roll is known, the maximum radial bearing load can be roughly estimated with the aid of the following equations:

$$G = g\,m/1\,000$$

$$K_r = 2\,q_1\,L + G$$

$$F_r = 0{,}5\,K_r$$

$F_a = \mu\,F_r$ (for a spherical roller bearing as non-locating bearing)

$F_a = 0$ (for a CARB toroidal roller bearing as non-locating bearing)

where
G = roll weight, kN
g = 9,81 (acceleration of gravity), m/s^2
m = roll mass, kg
K_r = roll load, kN
q_1 = felt tension, kN/m
L = felt width, m
F_r = radial bearing load, kN
F_a = axial bearing load, kN

μ = coefficient of friction between housing and outer ring (use μ = 0,15 in the calculation)

Generally, it can be assumed that the felt tension is 3–7 kN/m over the length of the roll. The axial bearing load caused by wire guidance can be ignored in the case of wire rolls. In some cases, the wire tension can lift the roll and make the bearing run with insufficient load, making rollers slide instead of roll. The lowest bearing load must also be calculated. The minimum roll load, K_r, is obtained when the wire tension lifts the roll.

Lubrication

The most important factors for effective lubrication are the lubricant viscosity, achieving a satisfactory degree of surface separation between the rolling contact surfaces, and the lubricant cleanliness with respect to water and solid particles. Protection against corrosion has top priority for this bearing position. Therefore, the lubricant must have good rust-inhibiting properties.

Oil lubrication is often selected for these bearings. There are several reasons for this. One is that if grease lubrication were to be employed for the bearings of high-speed machines, the operating temperatures would be so high that regreasing would have to be carried out far too frequently and excessive amounts of grease would contaminate the machine. Another reason is that there are other oil lubricated bearings nearby. Demands made on the oil for the felt roll bearings are not as great as for the oil for the press roll bearings. Therefore, the same oil used for the press rolls can be used, provided it has been correctly selected.

For further information, see *chapter 7*, *Lubrication*, and *chapter 8*, *Lubrication examples*.

Journal and housing tolerances for felt rolls

See the indications given in *Tolerances* in *chapter 1*.

If the bearing is mounted on a tapered journal, see *Tolerances for tapered journal seats* in *chapter 1*.

If the journal is cylindrical please follow the recommendations in **table 3.1**. The recommended journal tolerances for the main bearing have changed and give tighter fits than in the 4th edition of the Rolling bearings in paper machines handbook. As such, the main bearings may need to have C3 radial internal clearance because of increased radial clearance reduction due to the increased tight fit.

For housing tolerances, follow the recommendations in **table 3.1**.

Journal and housing tolerances for felt rolls

Table 3.1

Cylindrical Journal	Mounting on sleeve		h9Ⓔ
	Total radial run-out		IT5/2
	Direct mounting		
	(60) to 100 mm		n6Ⓔ
	(100) to 200 mm		p6Ⓔ
	Support bearing seating		
	Cylindrical roller bearing	Spherical roller bearing	
	(40) to 100 mm	(40) to 65 mm	m5Ⓔ
		(65) to 100 mm	m6Ⓔ
Housing			G7Ⓔ

Plain press rolls

Plain press rolls are sometimes called solid rolls although they are not always solid all the way through. A plain press roll features as one of the rolls in a single-nip press and as the central press roll in a multi-nip press. Plain rolls are normally steel rolls coated with different synthetic materials, but previously granite rolls were used.

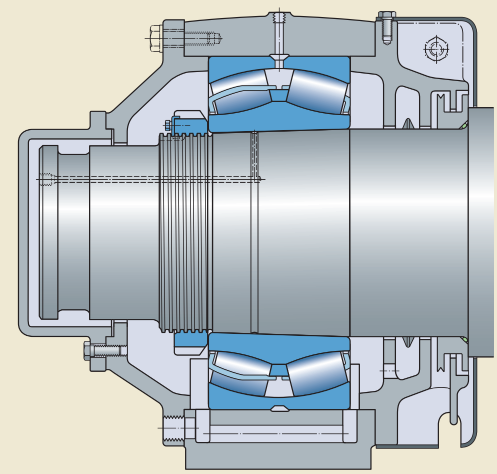
The bearing housings for press rolls are exposed to heavy loads and therefore the use of non-split housings is recommended.

The example shown in **fig. 3.9** is designed for grease lubrication and a double axial labyrinth seal is incorporated. As further protection, a protective cover is fitted to prevent the entry of water during hosing down operations.

Fig. 3.10 shows an oil lubricated press roll bearing arrangement. Where a circulating oil system is used, a flinger prevents oil escaping from the housing, while the radial labyrinths serve to keep out the water. Bearings mounted on withdrawal sleeves are sometimes used, but today machine builders generally prefer bearings mounted directly on tapered seats. If the builders have proper machining facilities, this design will be less expensive, but another important advantage is the reduced radial run-out of the roll with respect to datum, i.e. the roll axis defined by the tapered bearing seats, which among others will result in less vibrations and allow for higher speeds.

Modern press roll bearing arrangements often have one lip radial shaft seals with a garter spring and/or a V-ring (→ **fig. 3.11**)

Fig. 3.10



Oil lubricated press roll bearing arrangement

Bearing types

SKF recommends spherical roller bearings of series 231, 232 and 241 with CARB toroidal roller bearings of series C 31, C 32 and C 41. Bearings with C3 or C4 radial internal clearance are recommended if the speed, load, bearing size, lubrication and heat evacuation possibilities warrant it. For large size bearings, especially on high-speed paper machines, a higher clearance class than the steady state operating conditions would suggest is often chosen to avoid internal preload during start up. Bearings with reduced circular run-out tolerances (C08, VA460 or VQ424) minimize roll run-out. For high speeds, and if the roll is reground while supported by the bearings,

Press roll bearing arrangement, grease lubricated

Fig. 3.9

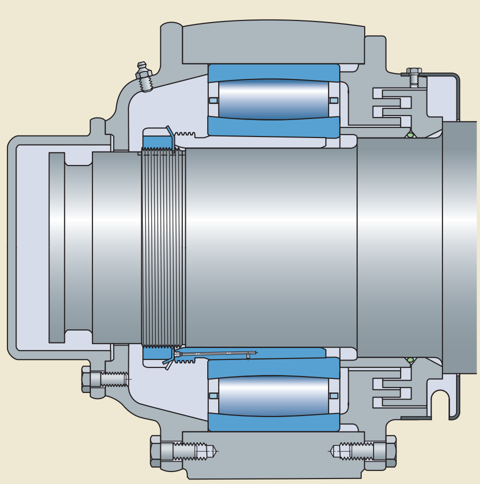
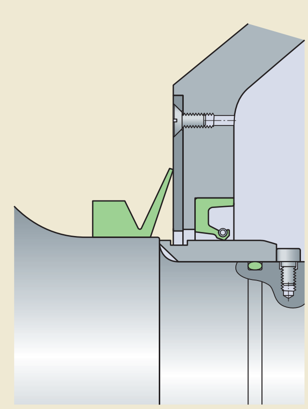


Fig. 3.11



Seal arrangement in modern press rolls

VQ424 is recommended. For very high speeds, VA460 bearing variants are recommended.

Note that the highest ring eccentricity point, when marked on the ring face, indicates the angular positioning of the ring on its seat. The ring must be positioned so that its high eccentricity point is placed diametrically opposite the journal high eccentricity point.

Due to the progress in manufacturing, ring wall thickness variation has significantly reduced over the years. On many old machines where the ring high eccentricity point was needed to position the bearing on the journal to reduce the roll run-out, the high eccentricity point is not needed anymore. Field experience, in which SKF bearings are mounted without taking the ring high eccentricity point into consideration, proves the fact.

Ring high eccentricity point indication is only recommended when VQ424 or VA460 bearings are requested.

Selection of bearing size

Bearing selection must always be made on the basis of proper calculations. This is especially important for new and modernized press parts with multi-nip presses. The life calculation for a central press roll bearing should be based on the loads from all nips involved as well as the roll gravity force.

Bearing selection should be based on life calculations according to the recommendations in *chapter 1*, General requirements and recommendations. SKF recommends that both the basic rating life L_{10h} and the SKF rating life L_{10mh} are taken into consideration. The calculated basic rating life L_{10h} should exceed 120 000 hours, while a SKF rating life L_{10mh} of 100 000 hours is recommended. The following equations should be used when calculating the magnitude of the loads:

$$G = g\, m / 1\,000$$

$$P_1 = F_{N1}\, L$$

$$P_2 = F_{N2}\, L$$

where
G = roll weight, kN
g = 9,81 (acceleration of gravity), m/s^2
m = roll mass, kg
P_1 = press load from the first press nip, kN
P_2 = press load from the second press nip, kN
F_{N1} = linear load of the first press nip, kN/m
F_{N2} = linear load of the second press nip, kN/m
L = press nip length, m

When G, P_1 and P_2 are known, the resultant roll load K_R can be established either graphically, as in **fig. 3.12**, or by trigonometric calculations.

The radial bearing load F_r is then calculated as follows:

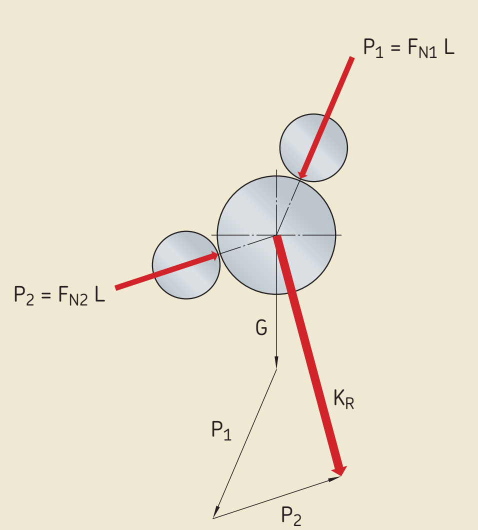
$$F_r = 0{,}5\, K_R$$

$F_a = \mu\, F_r$ (for a spherical roller bearing as non-locating bearing)
$F_a = 0$ (for a CARB toroidal roller bearing as non-locating bearing)

where
F_r = radial bearing load, kN
K_R = resultant roll load, kN
F_a = axial bearing load, kN
μ = coefficient of friction between housing and outer ring, use μ = 0,15 when calculating.

The resultant load K_R on central press roll

Fig. 3.12



Lubrication

The most important factors influencing lubrication are the lubricant viscosity, achieving a satisfactory degree of surface separation between the rolling contact surfaces, and the oil cleanliness with respect to water and solid particles.

Grease or oil lubrication may be used for press roll bearings. Grease lubrication is sometimes selected for machines with low operating speeds, e.g. pulp dryers.

With high-speed paper machines, the frictional heat in the large bearings on the press rolls is so great that large quantities of circulating oil must be passed through the bearings to dissipate heat. A low oil inlet temperature, obtained with a cooled tank in a separate system, is thus advantageous from a viscosity point of view. However, SKF recommends that oil is heated before start up and higher oil flow rates than necessary are avoided.

Press roll bearings in modern high-speed machines run at speeds above the reference speed rating in the SKF catalogue *Rolling bearings*. When large bearings rotate at high speed, there is a risk of smearing, i.e. sliding of unloaded rollers when they enter the loaded zone. In the press part, the risk is highest for press roll bearings because of their heavy rollers. Higher oil viscosity than required should be avoided.

As protection against corrosion and smearing has top priority for these bearings, oil with effective EP and rust-inhibiting additives is recommended.

For further information, see *chapter 7*, *Lubrication*, and *chapter 8*, *Lubrication examples*.

Journal and housing tolerances for plain press rolls

See the indications given in *Tolerances* in *chapter 1*.

If the bearing is mounted on a tapered journal, see *Tolerances for tapered journal seats* in *chapter 1*.

If the journal is cylindrical, please follow the recommendations in **table 3.2**.

For housing tolerances, follow the recommendations in **table 3.2**.

Suction press rolls

The design of these rolls is the same as for suction rolls in the forming section. See *chapter 2*, *Forming section*.

Journal and housing tolerances for plain press rolls

Table 3.2

Journal	Mounting on sleeve	h9Ⓔ
	Total radial run-out	IT5/2
Housing	Bore diameter up to 400 mm	G7Ⓔ
	Bore diameter above 400 mm	F7Ⓔ

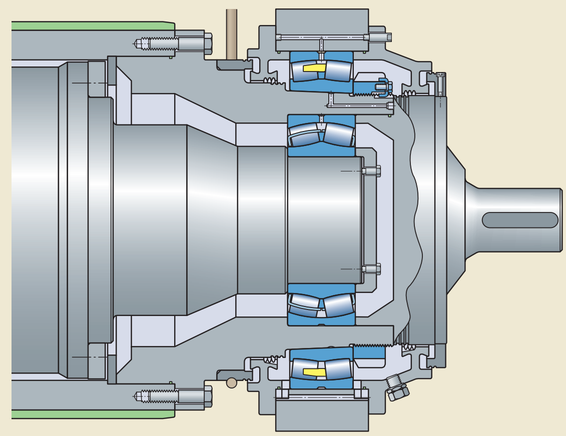
Deflection-compensating rolls

A driven deflection-compensating roll is shown in **fig. 3.13**. In this case, the stationary shaft is supported by a spherical plain bearing at the front (locating) side and by a triple ring bearing at the drive (non-locating) side. The roll shell is linked to the drive shaft via the centre ring of the triple ring bearing. Note that only the centre ring of this bearing rotates during operation. This ring must therefore have reduced run-out tolerances, e.g. C08, in order to minimize vibrations taking the form of variation of the nip pressure.

As triple ring bearings are not standard and have limited availability, designs with two spherical roller bearings are preferred as shown in **fig. 3.14**. The inner bearing has its outer ring mounted with a tight fit due to the drive-up of the inner ring of the outer bearing along its seat. To avoid the tight fit of one bearing influencing the tight fit and the radial clearance of the other, and to have the opportunity to mount two bearings with higher load capacity, the inner bearing on **fig. 3.14** can be displaced inwards towards the end of the nip.

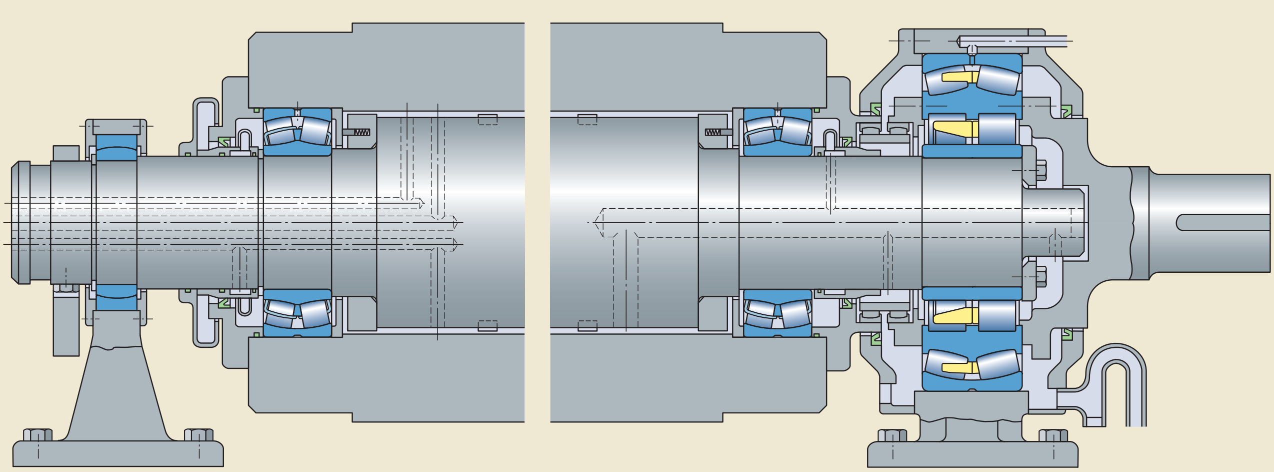
Other alternatives to triple ring bearing arrangements exist e.g. two tapered roller bearings with one spherical roller bearing, or three cylindrical bore spherical roller bearings. Such cylindrical bore bearings must be positioned so that the tight fit on one bearing doesn't influence the radial clearance and fit of the other.

Fig. 3.14



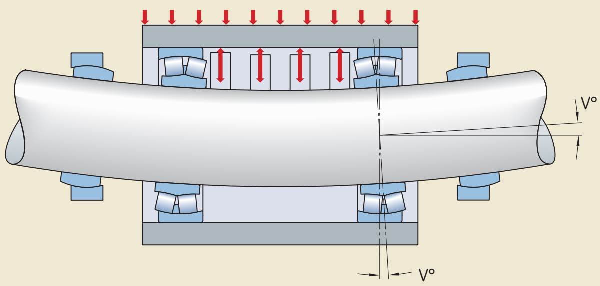
Alternative arrangement with spherical roller bearings instead of a triple ring bearing

Fig. 3.13



Old bearing arrangement for an oil lubricated, driven deflecting-compensating roll

Fig. 3.15



Rotating misalignment

Deflection-compensating rolls present the greatest challenge of all when designing reliable bearing arrangements for paper machines. This is because the main bearings that support the shell of the roll operate with rotating misalignment of the outer ring in most cases (→ **fig. 3.15**). Under such conditions, a number of phenomena detrimental to the smooth operation of bearings occur.

The greatest risk is edge loading of the rollers. The load is displaced towards one end of the roller because of the tendency of the misaligned rotating outer ring to pull the rollers with it axially as it oscillates with each revolution. The edge loading of the rollers can be so severe as to cause them to suddenly jump back to their equilibrium position. This happens if the axial component load on the roller, as a result of the edge loading, exceeds the roller-to-raceway friction. Such behaviour results in a major risk of smearing damage to rollers and raceways. Another phenomenon is that the speed of the rollers varies through each revolution of the ring. The speed of the rollers in one row is higher than that of the rollers in the other row through one half of the revolution and lower through the other half. This behaviour means that there is a risk of excessive loads being imposed on the cage bars and of peripheral sliding of the rollers, i.e. a risk of smearing.

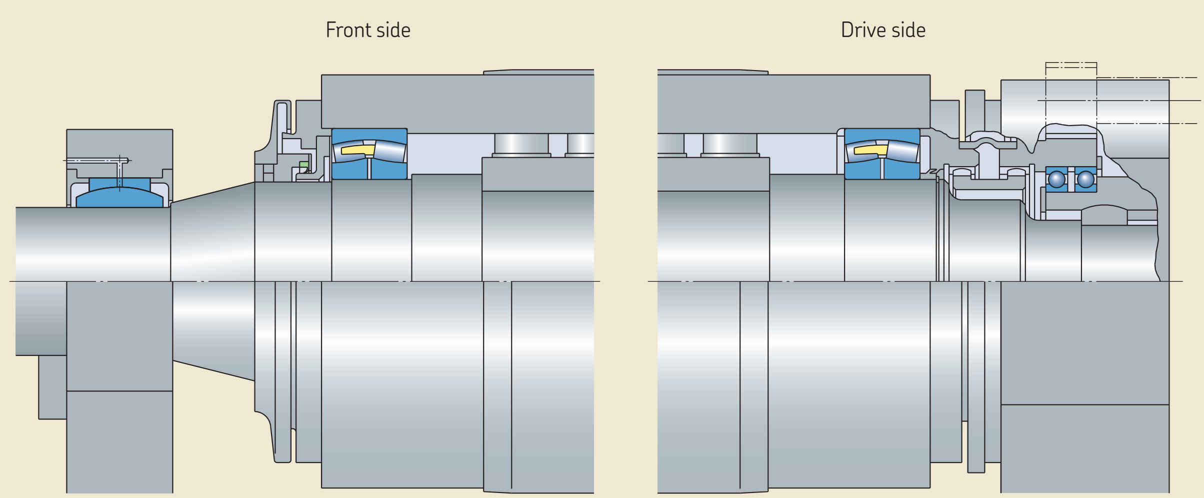
The best way to avoid problems with the bearings in this position is to minimize the angle of misalignment. The contact angle and internal clearance of the shell supporting bearing play a major role in operational performance. Various arrangements can be used to drive deflection-compensating rolls. The first such rolls were driven via a triple ring bearing but nowadays most machine manufacturers have opted for an integrated gear unit and zone-controlled nip pressure. An example of such an arrangement is shown in **fig. 3.16**.

Modern bearing arrangement for an oil lubricated, driven zone-controlled deflection-compensating roll

Bearing types

Where shell supporting bearings are subject to normal loads and misalignment close to 0,3 degrees, calculations have proved that

Fig. 3.16



bearings of series 230 and 239 give the best results. This has been verified by practical experience though each case should be considered individually, since bearings from series 238 and 248 have also been used successfully.

In high speed machines, bearings can run above the reference speed rating and even above the limiting speed indicated in the SKF catalogue *Rolling bearings*. Bearings with a VQ424 suffix are recommended or even the VA460 suffix if the speed is close or above the limiting speed.

Triple ring bearings are supplied in three different designs – spherical/spherical roller bearing, spherical/cylindrical roller bearing and cylindrical/spherical roller bearing. In all cases, the bearing centre ring connects the shell with the drive shaft and is therefore provided with tapped bolt holes in both faces. The triple ring bearings do not belong to a standard range and are identified by drawing numbers.

Selection of bearing size

The selection of the size of the shell supporting bearings is usually dictated by the available space. The calculated bearing life is often very high because the major share of the nip load is supported by the hydraulic roll crowning.

A triple ring bearing is regarded as two bearings when it comes to life calculations. The lives of triple ring bearings are very rarely calculated nowadays because these bearings are seldom used in new designs of deflection-compensating rolls.

Bearing selections should be based on life calculations according to the recommendations in *chapter 1*, *General requirements and recommendations*. SKF recommends that both the basic rating life L_{10h} and the SKF rating life L_{10mh} are taken into consideration. The calculated basic rating life L_{10h} should exceed 120 000 hours, while a SKF rating life L_{10mh} of 100 000 hours is recommended.

Lubrication

The lubricants for shell supporting bearings are required to have AW or, if the temperature allows, EP additives and ensure an adequate oil film especially if the deflection-compensating roll is heated.

Triple ring bearings must always have circulating oil lubrication. In some rolls, the outer bearing is lubricated by a separate circulating oil system. The inner bearing is sometimes lubricated with the same oil as that supplied to the hydraulic roll crowning. In the last generation of installations using triple ring bearings, the lubricating oil is pumped into the inner bearing through holes drilled for this purpose in the inner ring. In early designs of these rolls, the flow of oil used to lubricate the inner bearing was sometimes less than that used for the outer bearing. However, SKF computer simulations show that the best result, with all the three rings having almost the same temperature, will be obtained if the same flow of oil is supplied to the inner and outer bearings.

The oil flows for the shell supporting bearings and the triple ring bearing should be large enough to keep the bearing temperature below 80 °C because many of the EP additives on the market do not perform satisfactorily at higher temperatures. In heated rolls it is not always possible to keep the bearing temperature below 80 °C. In such cases, it is necessary to select an oil with AW additives.

For further information, see *chapter 7*, *Lubrication*, and *chapter 8*, *Lubrication examples*.

Journal and housing tolerances for deflection-compensating rolls

See the indications given in *Tolerances* in *Chapter 1*.

If bearing is mounted on a tapered journal, see *Tolerances for tapered journal seats* in *Chapter 1*.

If journal is cylindrical, please follow the recommendations in **table 3.3.**

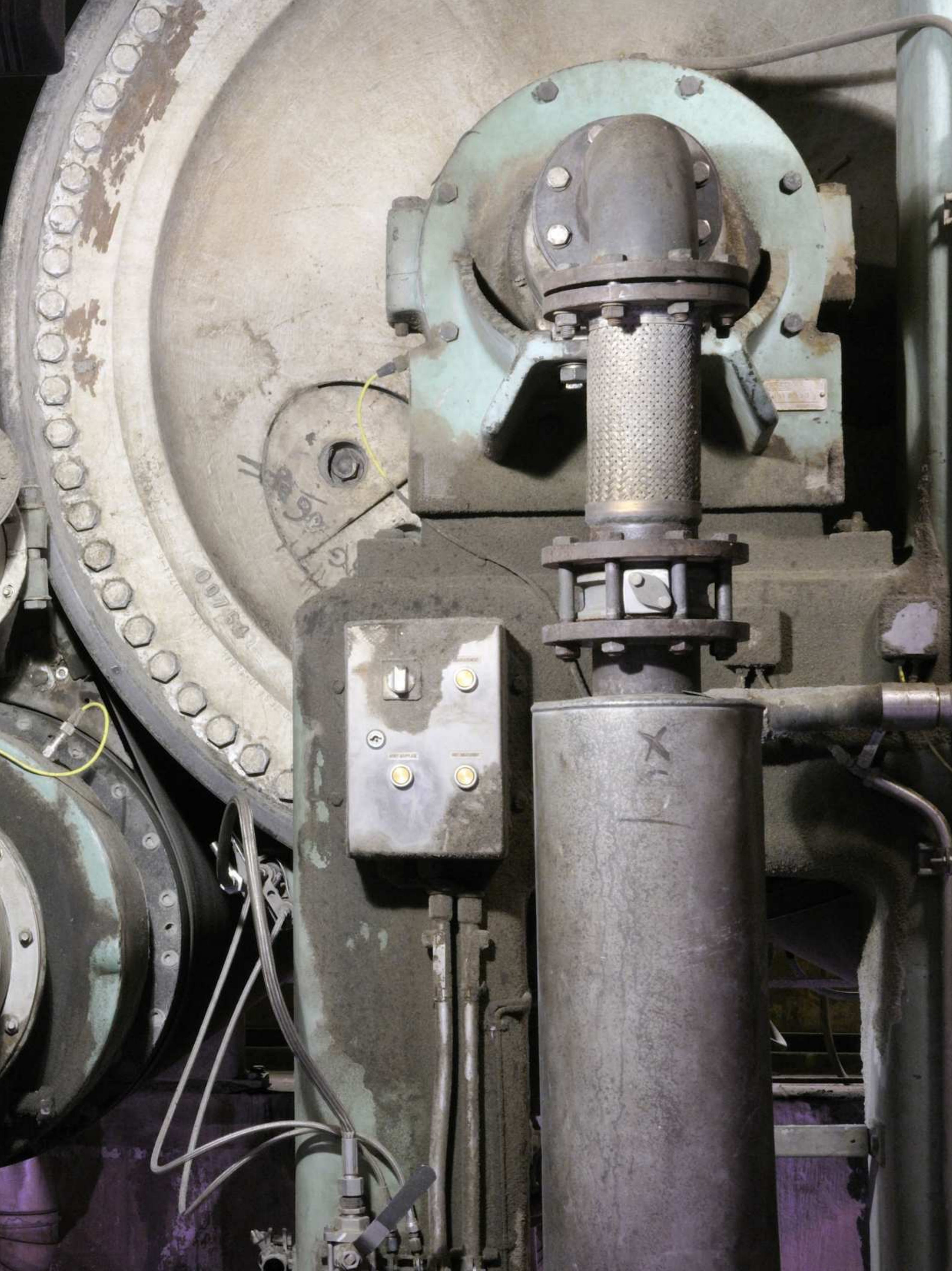
For housing tolerances, follow the recommendations in **table 3.3.**

Table 3.3

Shell (journal)	**Note:** for bearings with bore diameter above 800 mm a slightly shifted f6E may need to be considered to assure a loose fit over the entire tolerance range.	f6E
Shell (housing)	**Note:** For heated rolls where the bearing temperature may be above 140 °C, a housing tolerance P7E is recommended	N7E
Triple ring bearing	Journal Housing	f6E G7E

Journal and housing tolerances for deflection-compensating rolls

3



Dryer section

When the web has left the press section and enters the dryer section it has water content of 50–65%. Drying is completed in the dryer section so that the paper has an ambient moisture content level of 5–10%.

Drying is normally accomplished by moving the web along an S-shaped path over a double row of heated drying cylinders.

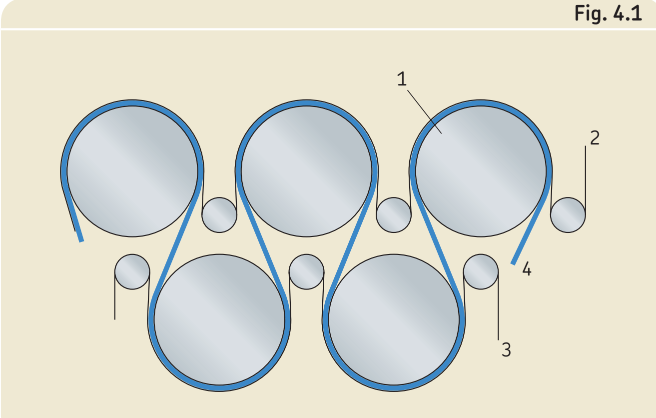
In modern paper machines, the whole dryer section is encased in an enveloping dryer hood and the ambient temperature inside the hood is 70-130 °C.

The felt run

The felt run consists of a double or a single (serpentine) felt. With the double felt, a top felt is used for the top row of cylinders and a bottom felt for the bottom row of cylinders (→ **fig. 4.1**). Double felt runs can cause web flutter at high speeds so usually a single felt run (→ **fig. 4.2**), where the web is supported the whole time by the felt, is used for high-speed machines.

Double felt run
1 *Drying cylinder*
2 *Top felt*
3 *Bottom felt*
4 *Paper web*

Fig. 4.1



Single felt run
1 *Drying cylinder*
2 *Felt*
3 *Paper web*

Fig. 4.2

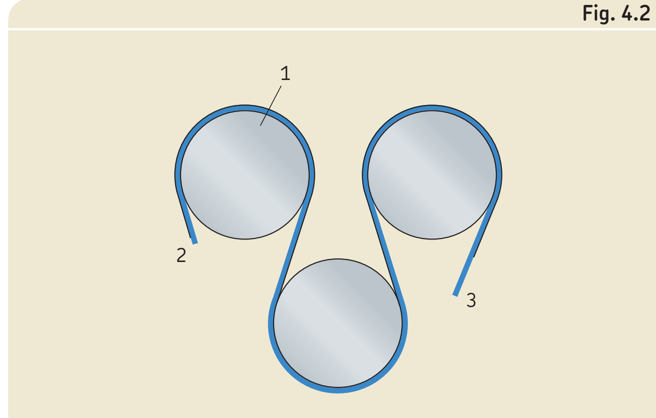
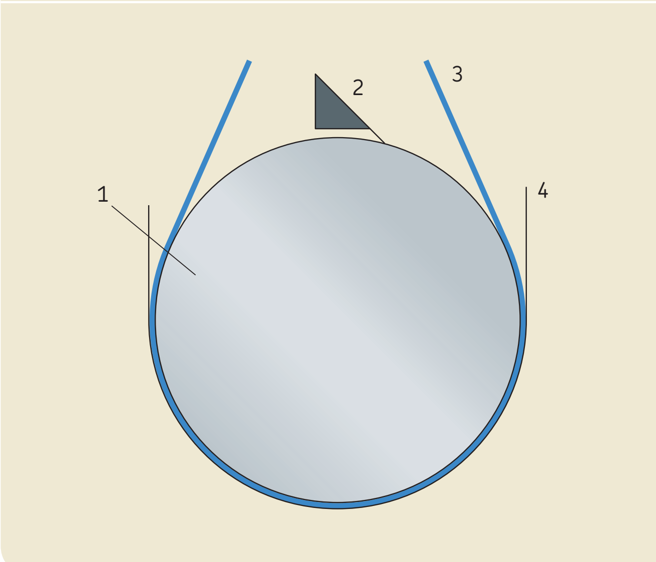


Fig. 4.3



Drying cylinder with doctor
1 Drying cylinder
2 Doctor
3 Paper web
4 Felt

Doctors

Drying cylinders are kept clean by means of doctors (→ **fig. 4.3**) that also stop the paper from getting wrapped around the cylinder in the event of web breakage. To prevent uneven wear of the cylinder surface by the doctor blade, the doctor oscillates axially by some 10–20 mm.

In the case of tissue, doctors are also used to crepe the paper.

Drying cylinders

Drying cylinders (→ **fig. 4.4**) are heated by steam. The temperature of the steam can vary between 110 and 210 °C depending on the thickness of the paper, the speed of the machine and the size of the dryer section. The steam condenses in to water when it comes into contact with the cooler surface of the cylinder and is then extracted, together with residual steam, via a siphon pipe.

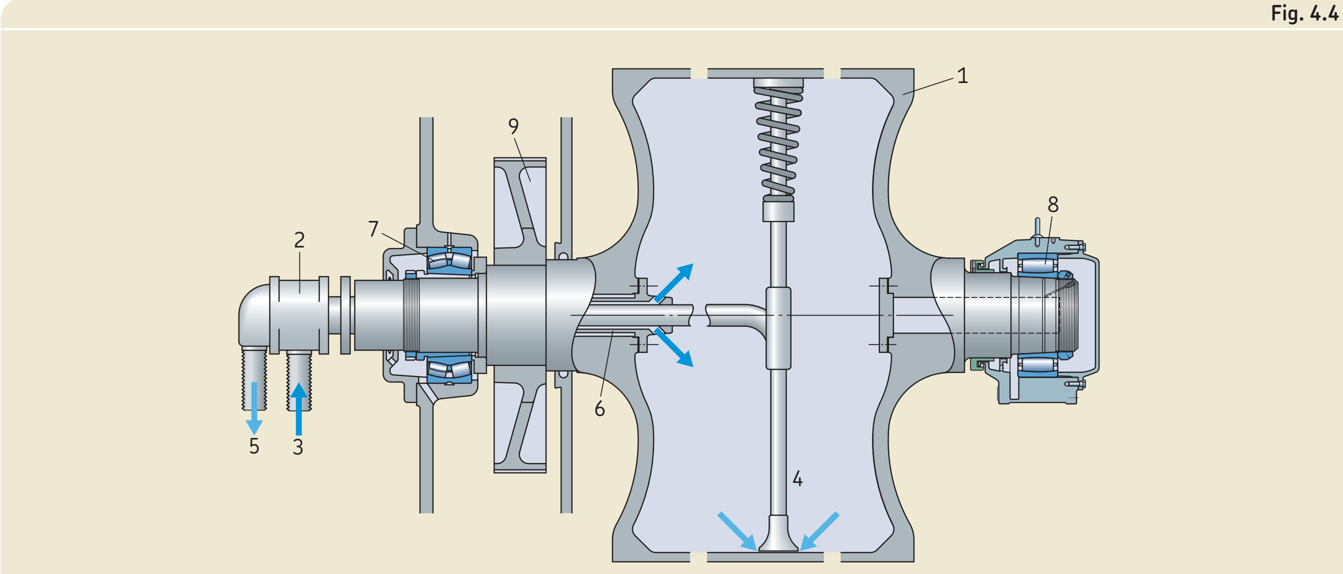
A dryer section may contain 40 to 100 drying cylinders, each 1,5 to 2,2 m in diameter. A newsprint machine usually has 40 to 50 drying cylinders and a board machine 90 to 100 such cylinders.

A traditional dryer section is split up into drive groups (→ **fig. 4.5**). The paper speed for each drive group is individually adjustable to compensate for the contraction of the web as it dries.

A feature of modern drying sections is that the gear drive of each drying cylinder is replaced by a felt drive. Normally, the drying cylinder has a gear drive built into the machine frame at the drive side. However, the strength of the felts has increased greatly over the years. This has allowed the use of felt drives where the drying cylinders are driven via the felt and felt rolls. When this type of drive system is employed, most of the drying cylinders have ordinary bearing housings at both sides.

Drying cylinder arrangement
1 Drying cylinder
2 Steam joint
3 Steam inlet
4 Siphon
5 Condensation
6 Journal insulation
7 Drive side bearing
8 Front side bearing
9 Gear drive

Fig. 4.4

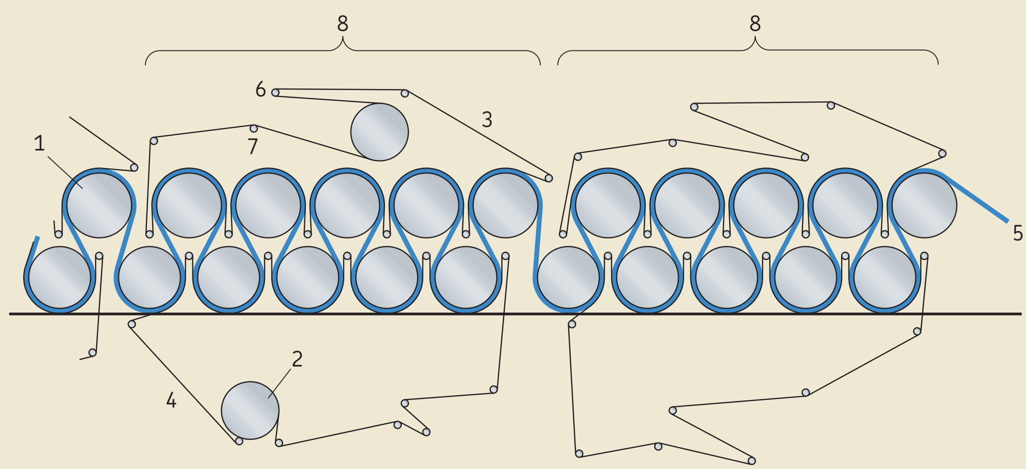


Vacuum rolls

In modern dryer sections, vented steel rolls known as vacuum or Vac rolls have replaced some of the drying cylinders. These rolls are connected to a vacuum pump and are used with single (serpentine) felt runs in positions where the web passes outside the felt. The partial vacuum in these rolls prevents the web stretching and perhaps folding under the influence of the centrifugal force. Vac rolls are placed in both bottom and top positions so that the paper web is treated equally on both sides.

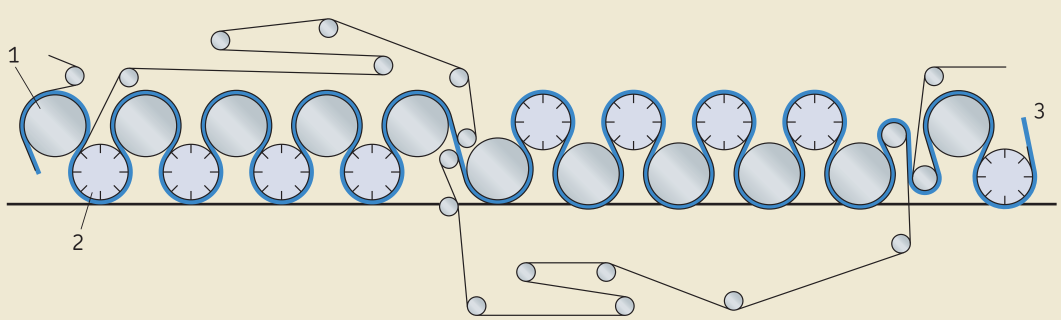
Fig. 4.6, shows a Valmet dryer section with Vac rolls and **fig. 4.7**, the Voith Paper TopDuoRun dryer section.

Fig. 4.5



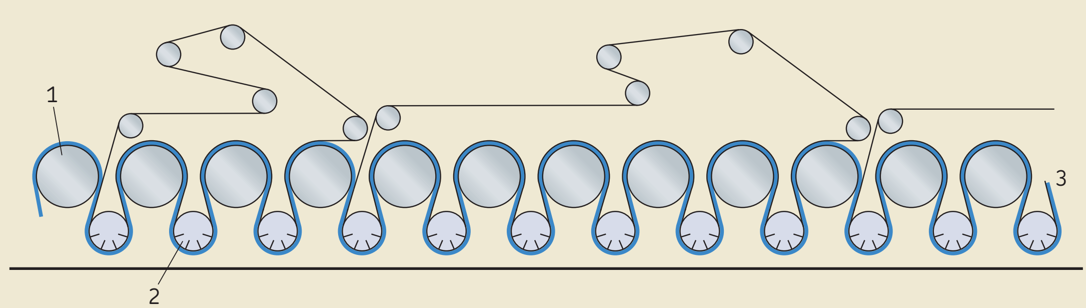
Traditional dryer section
1 *Drying cylinder*
2 *Felt drying cylinder*
3 *Top felt*
4 *Bottom felt*
5 *Paper web*
6 *Felt stretch roll*
7 *Felt guide roll*
8 *Drive group*

Fig. 4.6



Dryer section with Vac rolls
1 *Drying cylinder*
2 *Vac roll*
3 *Paper web*

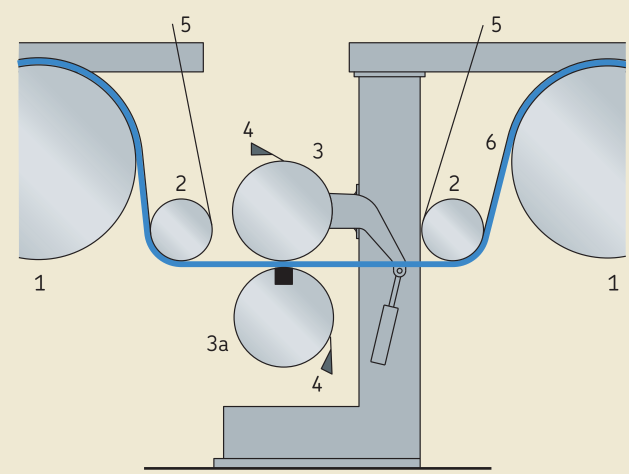
Fig. 4.7



TopDuoRun dryer section
1 *Drying cylinder*
2 *Vacuum roll*
3 *Paper web*

Breaker stack
1 *Drying cylinder*
2 *Guide roll*
3 *Solid press roll*
3a *Deflection-compensating press roll*
4 *Doctor*
5 *Felt*
6 *Paper web*

Fig. 4.8



Other drying processes for the drying section

There is an ongoing trend to shorten the drying section. This can be achieved by using:

- Some cylinders with extra high temperature. Note that the first cylinders after the press section have lower steam temperature and a special coating so that the paper web doesn't stick.
- Hot air blown with high velocity on to the paper web in addition to the traditional drying cylinders.
- Infrared heaters.

Sizing is performed both to improve the surface finish of the paper for printing and to improve its mechanical strength. The solutions used are usually starch based with a concentration between 8 and 10%. To ensure proper absorption, the dryness of the sheet must be higher than 90%. Unfortunately, the paper web needs to be dried, wetted again during sizing and dried once more in a second dryer part.

On leaving the size press, the paper web passes over a spreader roll to eliminate wrinkles.

Breaker stacks

The breaker stack (→ **fig. 4.8**) is positioned in the dryer section and consists of two driven steel rolls, one of which may be a deflection-compensating roll. The stack is used to achieve a smoother paper than can be obtained solely by the machine calendering process. It is also assumed that the breaker stack presses the water from the interior to the outer layer of the web and, by so doing, facilitates subsequent drying.

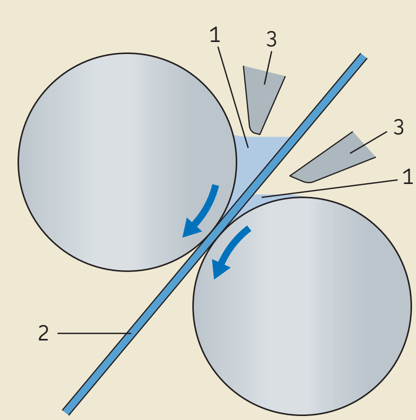
The linear load is usually 10–20 kN/m.

Size presses

The size press is made up of two pressure rolls with hard rubber coating. The size is applied at the inlet of the rolls (→ **fig. 4.9**). Many size press rolls are cooled by circulating water in the rolls. The inlet and outlet for coolant is generally on the front side.

Size press principle
1 *Size bath*
2 *Paper web*
3 *Size injectors*

Fig. 4.9



Yankee cylinders

A Yankee (MG) cylinder is used for drying the web in the manufacture of tissue and board. Yankee cylinders are normally 4–6 m in diameter. They are designed for steam pressures up to 1 100 kPa (11 bar) and the steam temperature can be more than 200 °C. One or two press rolls, positioned below the Yankee cylinder, press the web against the cylinder. The paper adheres to the heated cylinder and it is dried by the heat from the cylinder and by hot air being blown at high speed on to the outside of the web.

In the case of board, the contacting side of the paper takes on a fine-glazed shiny surface. When the board has reached a certain dryness level, the surface tension is released and the board can then easily leave the cylinder (→ **fig. 4.10**). Additional drying of board is carried out using ordinary drying cylinders.

Drying in tissue machines is performed only by a Yankee cylinder. The first press roll is usually a suction press roll in order to achieve efficient dewatering. Tissue paper does not come away from the cylinder shell automatically and has to be doctored off by means of a special crepe blade, which gives a softer paper (→ **fig. 4.11**).

Other drying processes for tissue

Developments in tissue drying include the through air drying (TAD) concept giving higher softness, strength and absorbency properties and using a shoe press directly against the Yankee cylinder.

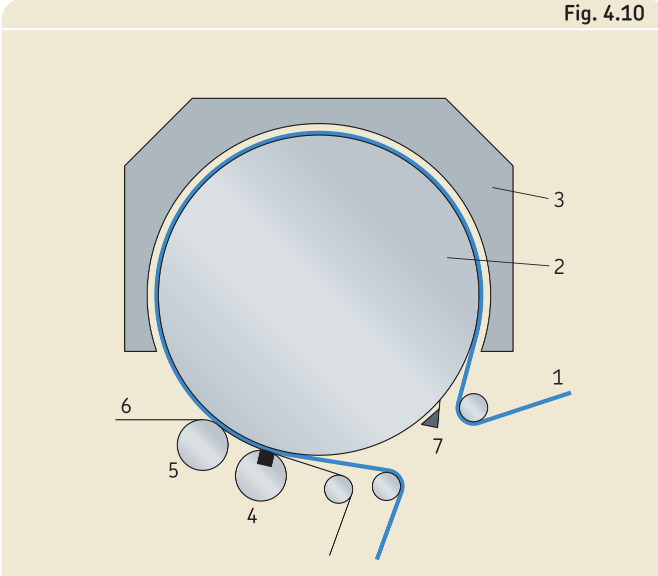
Rope sheaves

When starting up the machine or when the web breaks, a strip (tail) of paper about 200 mm wide is guided from the press section through the dryer section to the reeler either by ropes, air or vacuum. If ropes are used, they are guided by ropes sheaves through the drying section. In such cases, drying cylinders will have a groove or sheave located between the main bearing housing and the cylinder face to guide the rope.

4

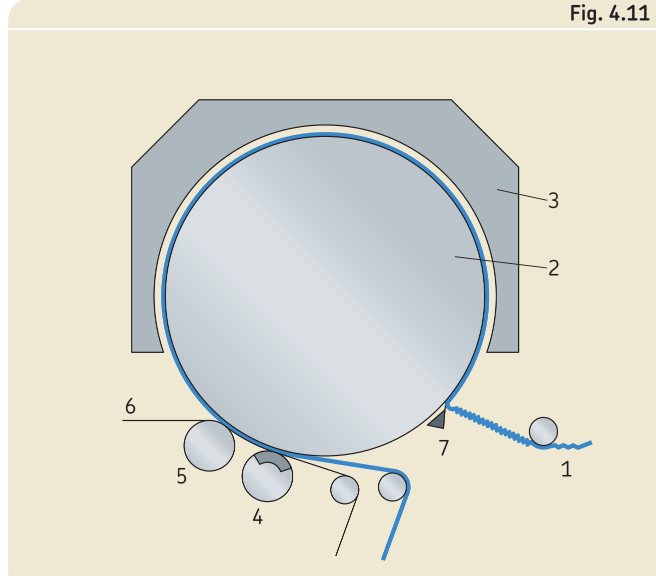
Yankee cylinder for board
1 *Paper web*
2 *Yankee cylinder*
3 *Hood*
4 *Deflection-compensating press roll*
5 *Press roll*
6 *Felt*
7 *Doctor*

Fig. 4.10



Yankee cylinder for tissue
1 *Paper web*
2 *Yankee cylinder*
3 *Hood*
4 *Suction press roll*
5 *Press roll*
6 *Felt*
7 *Doctor*

Fig. 4.11



Bearing arrangements

Warning about high temperature applications

Bearings in the dryer section are exposed to very high temperatures over long periods of time. Unfavourable lubrication conditions are quite common and short starting-up periods cause very high thermal stresses in the bearings. As such, the operating conditions for these bearings are quite severe. Over the years, the demands made on the bearings have also increased due to larger machines, higher speeds and higher steam temperatures. Production stops have become very costly, particularly unplanned stops caused by bearing failure in the dryer section. The performance and reliability of the bearings for these applications are consequently of the utmost importance.

Several factors have to be taken into consideration when choosing the bearings for high-temperature applications, like drying and Yankee cylinders and heated calendar rolls. The selection of material and heat treatment has to be based on some estimates as well as on several known parameters and serves to minimize the risk of cracked bearing inner rings. The following parameters should be taken into account:

- **The maximum total stress in the inner ring.**
  This occurs when the machine is started up as the temperature of the journal is then much higher than that of the inner ring, i.e. the inner ring is subjected to tensile stress in addition to the compressive stresses resulting from the loading of the bearing. The stress level also depends on the heat treatment method used for the inner ring.

- **The maximum difference between the temperatures of the journal and inner ring.**
  This depends mainly on the heating-up time during the starting-up period and whether or not the journal is provided with efficient insulation.

- **The drive-up of the bearing.**
  The drive-up has to be selected so that the actual operational inner ring-to-journal interference does not exceed the permissible interference for an unheated journal.

- **The initial radial internal clearance of the bearing.**
  The bearing may become preloaded during start-up if the radial internal clearance is inadequate for such conditions.

- **The ability to rapidly detect damage to the inner ring and take action to prevent cracking.**
  Since a bearing inner ring will not crack until some time after initial damage has occurred, effective condition monitoring in combination with remedial action can prevent it.

For drying and Yankee cylinders, SKF recommends the use of SKF new generation bainite steel or case carburized steel inner rings. SKF new generation bainite steel is a patented hardening method resulting in better performance than normal bainite through hardening. It has better crack, wear and fatigue resistance. All SKF spherical roller bearings and CARB toroidal roller bearings in the drying and Yankee cylinder size range have their rings made of the new generation bainite steel as standard and are marked WR. SKF bearings marked WR are the normal SKF recommendation for modern high performance paper machines. In cases where journals are not insulated and have steam temperatures above 170 °C, SKF recommends HA3 bearings which have case hardened inner rings.

Martentisitic steel bearing rings should always be avoided in such applications.

Journal insulation

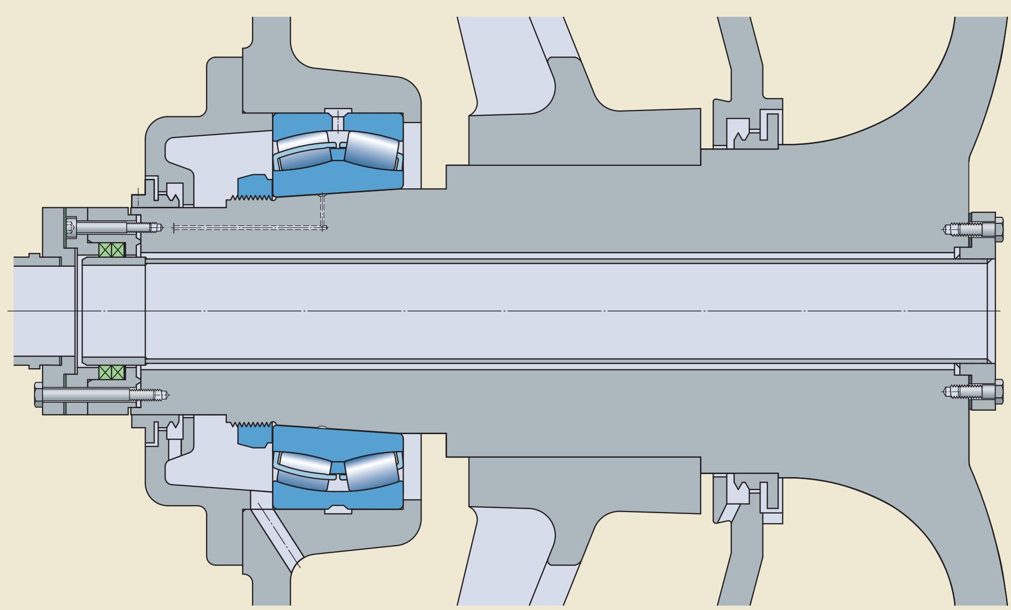
With the steam temperatures employed in modern machines, effective insulation is essential as the temperatures of the bearings would otherwise be too high. **Fig. 4.12** shows a common design of insulation in the bore of the journal of a drying cylinder bearing arrangement. The same design is, in principle, also used for Yankee cylinder applications. This type of journal insulation can reduce the bearing temperature by as much as 35 °C.

It is beneficial for the steam joint to be effectively insulated from the end face of the journal. This is often disregarded, but can easily be accomplished with an air gap as in **fig. 4.12** or with a 5 mm insulating washer with a thermal conductivity of 0,25 W/(m K) (less than 0,005 of that of steel). With this end face insulation, the bearing temperature can be reduced by another 5–6 °C which is of great value.

Special bearing housings

The ambient temperature in the dryer section is rather high. Consequently, all the cylinders will expand considerably during the heating-up period. This, in turn, puts high demands on the non-locating bearing arrangement. This is one reason why SKF developed special bearing housings for the different rolls and cylinders in the dryer section.

Fig. 4.12



Insulation of drying cylinder drive side

Felt rolls

Mounting the bearings on sleeves was standard practice in the past, but mounting them directly on the journals is more common today. **Fig. 4.13** shows SKF special housings for felt rolls in the dryer section where a spherical roller bearing and a CARB toroidal roller bearing are mounted directly on a tapered journal and are locked by an SKF KMT nut. The journal should have oil injection grooves when the bearing is mounted directly on it to facilitate dismounting.

In cases where more efficient seals are required, the housings can be provided with a reinforced sealing arrangement.

Bearing types

SKF recommends the use of spherical roller bearings of series 223 and 232 and CARB toroidal roller bearings of series C 23 and C 32, but spherical roller bearings of series 222 and CARB toroidal roller bearings of series C 22 can also be used. Both spherical roller bearings of series 222 and cylindrical roller bearings of series NUB 2, with wide inner rings, can be used as the support bearings for felt stretch and guide rolls like in the forming section of the paper machine. The main bearings and the support bearings should have C3 radial internal clearance.

Selection of bearing size

Bearings for felt rolls in the dryer section are calculated in the same way as shown for felt roll bearings in the press section. The only difference is the recommended basic rating life L_{10h} and SKF rating life L_{10mh}. These are 200 000 hours in the dryer section.

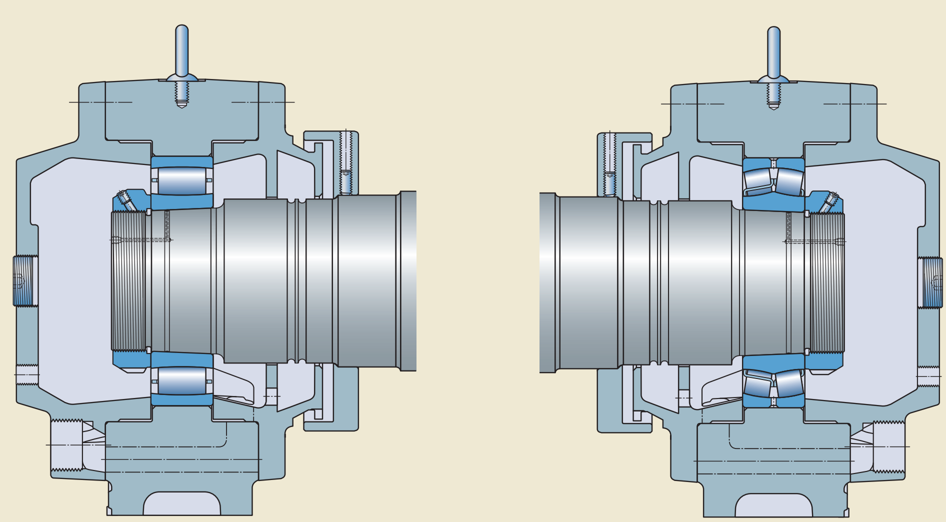
Lubrication

The felt roll bearings in the dryer section are lubricated by circulating oil from the same oil system as the drying cylinder bearings. **Fig. 4.13** shows a typical arrangement for oil circulation lubrication of spherical roller bearings and CARB toroidal roller bearings.

Requirements on the lubricating oil for the dryer section are dictated by the drying cylinders.

On older machines, felt rolls bearings in the dryer section are greased. Note that there can be an important ambient temperature difference between the bottom part and the top part of the dryer section with the top part being much hotter. As such, the desired grease characteristics can differ depending on the ambient temperature.

Fig. 4.13



SKF felt roll bearing housings designed for oil circulation lubrication

Journal and housing tolerances for felt rolls

See the indications given in *Tolerances* in *Chapter 1*

If the bearing is mounted on a tapered journal, see *Tolerances for tapered journal seats* in *Chapter 1*

If the journal is cylindrical, please follow the recommendations in **table 4.1**. The recommended journal tolerances for the main bearing have changed and give tighter fits than in the *4th edition* of the *Rolling bearings in paper machines* handbook. The main bearings may need to have a C4 radial internal clearance because of increased radial clearance reduction due to the increased tight fit.

For housing tolerances, see the recommendations in **table 4.1**.

Journal and housing tolerances for felt rolls

Table 4.1

Cylindrical journal	Mounting on sleeve		h9Ⓔ
	Total radial run-out		IT5/2
	Direct mounting		
	(60) to 100 mm		n6Ⓔ
	(100) to 200 mm		p6Ⓔ
	Support bearing seating		
	Cylindrical roller bearing	Spherical roller bearing	
	(40) to 100 mm	(40) to 65 mm	m5Ⓔ
		(65) to 100 mm	m6Ⓔ
Housings			G7Ⓔ

Drying cylinders

The bearing housing on the drive side can be an integral part of the machine frame where the circulating oil drains into the gear casing, as shown in **fig. 4.4**, **page 4:2**. These integrated gears are often supported by spherical roller bearings. **Fig. 4.12**, **page 4:7** also shows how to arrange for suitable journal insulation which is always recommended.

In machines where the housings are not an integral part of the frame, the drying cylinders have conventional bearing housings on both sides (→ **figs. 4.14**, **4.15** and **4.16**).

The bearings may be mounted directly on tapered journals or on adapter or withdrawal sleeves. If the bearings are mounted on a tapered seating, the journal should be provided with oil injection grooves to facilitate dismounting. If an adapter sleeve (with abutment spacer) will be incorporated, SKF recommends that an annular groove is machined in the journal outside the sleeve position. This groove can then be used to take a backing ring when a hydraulic nut (HMV) is employed to dismount the bearing. Where the bearings will be mounted on withdrawal sleeves, the journal would have to be threaded to take a lock nut.

Traditional solutions for the front side

Spherical roller bearing with axially free outer ring

With an arrangement as shown in **fig. 4.14**, the axial displacement is accommodated between outer ring and housing.

The friction between the outer ring and the housing may cause axial forces which are roughly 15% of the radial bearing load and this increases as soon as there is some fretting corrosion. This results in a considerable reduction of bearing life. Furthermore, at least for wide machines, the frame is mainly designed for radial loads. Therefore, the general guideline is not to use this bearing arrangement for wire widths above 4 500 mm.

Example: An induced axial load of 15% of the radial load on bearing 23052 reduces calculated bearing life by 70%.

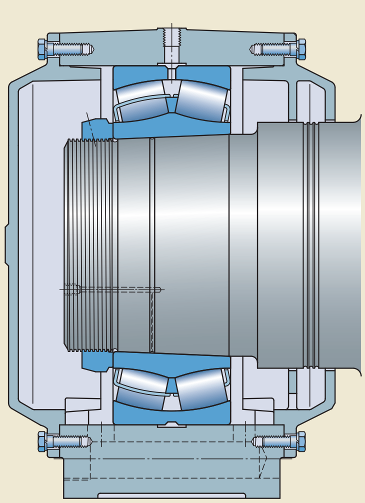
Spherical roller bearing in a housing mounted on rockers

When the housing is mounted on rockers, the axial displacement is accommodated by a slight tilting of the rockers (→ **fig. 4.15**). Due to the shape of the rockers, this gives a pure axial displacement with no displacement in the vertical direction. In the past, this was the best solution and accordingly recommended by SKF for machines with a wire width above 4 500 mm.

However, this housing arrangement is rather unstable and does not damp vibrations as well as solid housings. This may be a problem when upgrading to higher speeds. This arrangement is also sensitive to tilting

Bearing arrangement for the front side with a spherical roller bearing with axially free outer ring

Fig. 4.14



Bearing arrangement for the front side with a spherical roller bearing in a housing mounted on rockers

Fig. 4.15

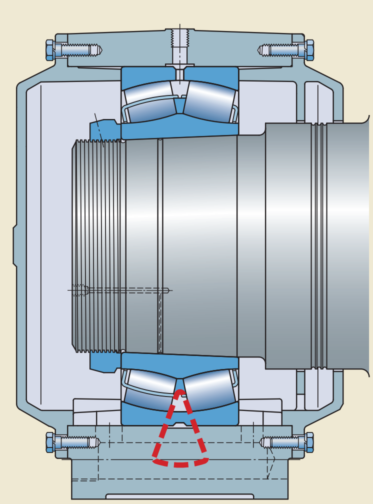
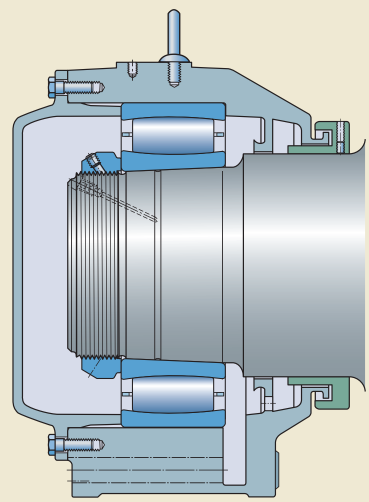


Fig. 4.16



Bearing arrangement for the front side with a CARB toroidal roller bearing in a fixed housing

forces from rope sheaves and steam joints fastened on the housing, for example.

The maintenance costs due to wear of the rocker arrangement may also be high as malfunctioning rockers produce axial loads.

To sum up, with this arrangement there is a risk of axial forces and increased maintenance costs caused by malfunctioning rockers as well as restrictions in the speed capability of the machine because of too high vibration levels.

Other solutions for the front side, such as special cylindrical roller bearings or triple ring bearings (cylindrical roller bearing with a spherical plain bearing as outer ring) have been abandoned.

Front side bearing application with CARB toroidal roller bearing

Bearing life and reliability

A typical bearing arrangement for CARB toroidal roller bearings is shown in **fig. 4.16**. The requirements for journal and housing tolerances as well as the selection of bearing size are the same as for other bearing types.

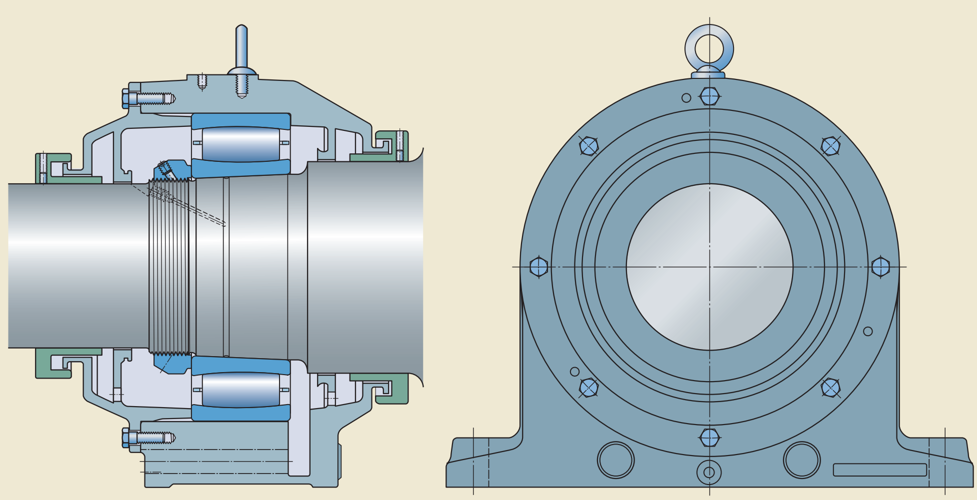
Compared to solutions with spherical roller bearings (→ **fig. 4.14** and **4.15**), the calculated bearing lives are increased due to the elimination of axial loads from the steam joint, malfunctioning housing rockers and, in the case of a fixed housing, friction between the outer ring and housing.

Housings

CARB toroidal roller bearings eliminate the need for rocker housings as the bearings will take up the thermal elongation of the cylinder. Consequently, the bearing can be mounted in a more robust and rigid fixed housing (→ **fig. 4.17**)

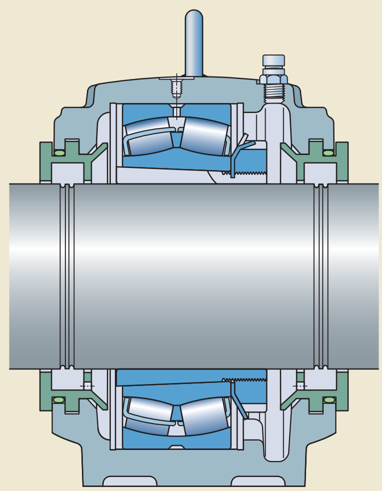
This gives a more stable arrangement and reduced vibration level which is especially important at increased speeds. Lower vibration levels also means less risk of component wear.

Fig. 4.17



The CARB toroidal roller bearing can be mounted in the SKF one-piece fixed housing which eliminates the need for rocker housings

Fig. 4.18



SNL TURP housing

For machine rebuilds, SKF can offer standard SNL housing with suitable seals (TURP suffix) (→ **fig. 4.18**). These have the advantage of a lower cost and a smaller axis height, but have oil flow rate limitations and offer comparatively low protection against water ingress when the machine is cleaned with high pressure water.

For more information about the housings, see *chapter 1*, *General requirements and recommendations*.

Steam joint on the front side

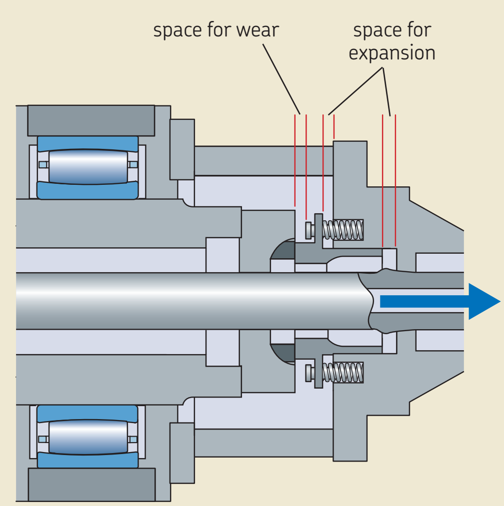
Sometimes drying cylinders are equipped with a steam joint for condensed water drainage on the front side. They are used on both rocker housings and plummer block housings (fixed).

As a CARB toroidal roller bearing is mounted in a fixed housing, the axial expansion of the cylinder will be taken up within the bearing. This means that if the steam joint is mounted directly on the cover of the housing, it has to be designed to take up all the axial cylinder expansion internally. This is no problem if the expansion sleeve can accommodate this displacement with regard to space and spring preload (→ **fig. 4.19**). Otherwise, some rework and new springs might be required. The joint is usually equipped with some sort of spherical wear washer. This wear washer is designed to prevent steam leakage and to minimize bending forces. It is normally changed at regular intervals. The total distance which the washer moves axially due to wear is roughly 5 mm. To accept both expansion and washer wear, the sleeve may have to be extended.

Different manufacturers of steam joints have their own solutions or designs. As the CARB toroidal roller bearing arrangement is more rigid and stable than a rocker housing arrangement, it is much easier to adjust the siphon to a correct position. When a CARB toroidal roller bearing arrangement is used, it is necessary to check that the siphon has sufficient space for expansion in the axial direction inside the cylinder.

Basic layout of a front side steam joint. It is important to ensure that there is room for axial cylinder expansion inside the steam joint.

Fig. 4.19



Selection of bearing size

The dryer bearings will be affected by the mass of the cylinder, the felt tension and the water content in the cylinder. The drive side bearing will also be affected by the gear forces. If the front side bearing housing rests on rockers, there is normally no axial force acting on the bearing in that position.

Depending on its location and design, the steam joint may cause one or both of the bearings to be axially loaded.

The recommended basic rating life L_{10h} and SKF rating life L_{10mh} are 200 000 hours. The bearing loads can be estimated with the aid of the following equations:

$$G = g\ m/1\ 000$$

$$G_1 = g\ m_1/1\ 000$$

$$K_R = G + G_1 + 2\ q\ L$$

where

G = cylinder weight, kN
G_1 = weight of water inside cylinder, kN
g = 9,81 (acceleration of gravity), m/s^2
m = cylinder mass, kg
m_1 = mass of water inside cylinder, kg
K_R = resultant roll load, kN
q = felt tension, kN/mm
L = felt width, mm

Drive side bearing:

$$F_r = 0{,}5\ K_R + F_2$$

$$F_a = F_3 + F_4 + F_5$$

$$F_5 = \mu\ F_r$$

Front side bearing:

$$F_r = 0{,}5\ K_R$$

$$F_a = F_4 + F_5$$

where

F_r = radial bearing load, kN
F_a = axial bearing load, kN
K_R = resultant roll load, kN
F_2 = radial gear force, kN
F_3 = axial gear force, kN
F_4 = axial force from steam box, kN
F_5 = axial bearing load, due to friction between housing and outer ring, kN (if neither CARB toroidal roller bearings nor rocker housings are used on the front side)
μ = coefficient of friction between housing and outer ring (use μ = 0,15 when calculating)

Generally, the felt tension is 3–5 kN/m. In old machines the cylinder often contains a fairly large amount of water. Modern high-speed machines are usually provided with efficient condensed water drainage and the water content is therefore reduced to a value corresponding to approximately 10–15 mm of water film around the circumference.

Lubrication

Most problems with drying cylinder bearings are related to the lubrication conditions. The best way to improve the lubrication is to provide the journals with efficient insulation. Circulating oil lubrication is used for all drying cylinder bearings except on very old paper machines. Owing to the high steam and ambient temperatures, large quantities of appropriate viscosity oil must be passed through the bearings to achieve proper lubrication. In modern machines with insulated journals it is possible to cool the bearing to temperatures below 90 °C in most cases.

As bearing housings are located under the hood, the oil inlet temperature must not be too low in order to avoid condensation and increased water content in the oil. In general, the oil inlet temperature is between 50 and 60 °C.

Many factors influence the calculation of the requisite oil flows, so these must be determined for each individual case. Consideration should be given to bearing size, speed, steam temperature, oil inlet temperature and insulation methods. The influence of the bearing load and ambient temperature on the temperature of drying cylinder bearings is small compared with that of the speed, steam temperature and insulation method. SKF can carry out a computer analysis of the lubrication and temperature conditions for drying cylinder bearings on request.

For further information see *chapter 7, Lubrication*, and the examples in *chapter 8, Lubrication examples*.

Table 4.2

Cylindrical Journal	Mounting on sleeve Total radial run-out	h9Ⓔ IT5/2
Housing	No need of bearing axial displacement in the housing: Bearing axial displacement in housing (**fig. 4.14**)	G7Ⓔ F7Ⓔ

Journal and housing tolerances for drying cylinders

Journal and housing tolerances for drying cylinders

See the indications given in *Tolerances* in *Chapter 1*.

If the bearing is mounted on a tapered journal, see *Tolerances for tapered journal seats* in *Chapter 1*.

If the journal is cylindrical, please follow the recommendations in **table 4.2**. Note that direct mounting on a cylindrical journal is not recommended.

For housing tolerances, follow the recommendations in **table 4.2**.

Yankee cylinders

Yankee cylinder bearings are exposed to very high temperatures over long periods of time. The operating conditions are similar to those of drying cylinders, but the rotational speed is usually lower as is the ambient temperature. It is always recommended that this bearing arrangement be provided with effective journal insulation similar to that shown in **fig. 4.12**, **page 4:7**. In the case of Yankee cylinder journals, the bearing and the steam joint are spaced further apart and end face insulation is not so important.

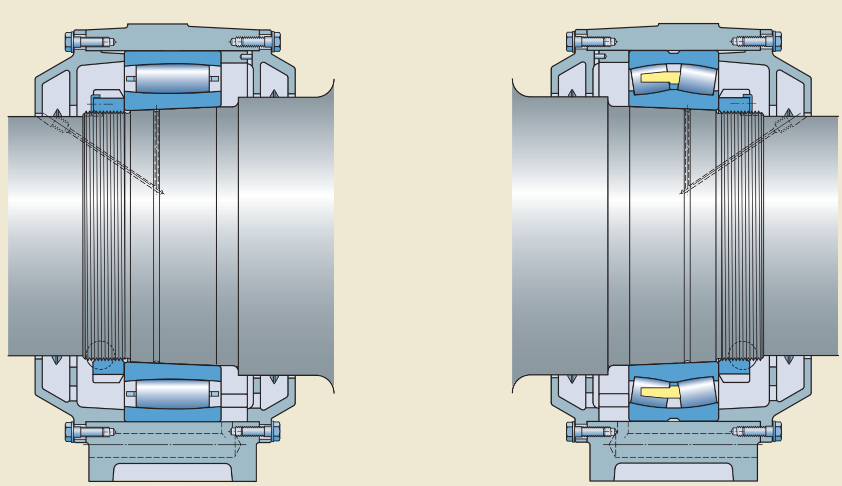
Yankee cylinders, also called MG cylinders, are carried by spherical roller and CARB toroidal roller bearings mounted directly on tapered journals or on adapter sleeves. If the bearings are to be mounted on tapered seats, the journals should be provided with oil injection grooves to facilitate dismounting of the bearings.

If an adapter sleeve (with abutment spacer) is to be incorporated, SKF recommends that an annular groove is machined in the journal outside the sleeve position. This groove can then be used to take a backing ring when a hydraulic nut (HMV) is used to dismount the bearing.

Split bearing housings with removable covers on both sides are normally used. The split cover at the inside position facilitates easy inspection, mounting and dismounting of the bearing (→ **fig. 4.20**).

Yankee bearing arrangement with fixed housings

Fig. 4.20



Before the introduction of CARB toroidal roller bearings there was a need to have special front side housings for spherical roller bearings. These housings were provided with rockers in order to reduce axial loads on the bearings and the frame caused by thermal expansion. They also had anchoring hooks in order to keep the bearing housing in position when the cylinder was being pressed upwards by the press rolls. It was necessary to equip the front side bearing housing with rockers in the horizontal plane as well if the press rolls were located in such a position that the resultant load (including gravity forces) on the cylinder diverged more than 30° from the vertical downward position (→ **fig. 4.21**).

Today, SKF recommends the use of CARB toroidal roller bearings on the front side. The CARB toroidal roller bearing eliminates the need for the rocker housings as the bearing itself takes up the thermal expansion of the cylinder. This means that the bearing can be mounted in a more robust and rigid, fixed housing (→ **fig. 4.20**). This gives a more stable arrangement and reduced vibration level which is especially important at increased speeds. Lower vibration levels also mean less risk of component wear.

For more information about the housings, see *chapter 1*, *General requirements and recommendations*.

Bearing types

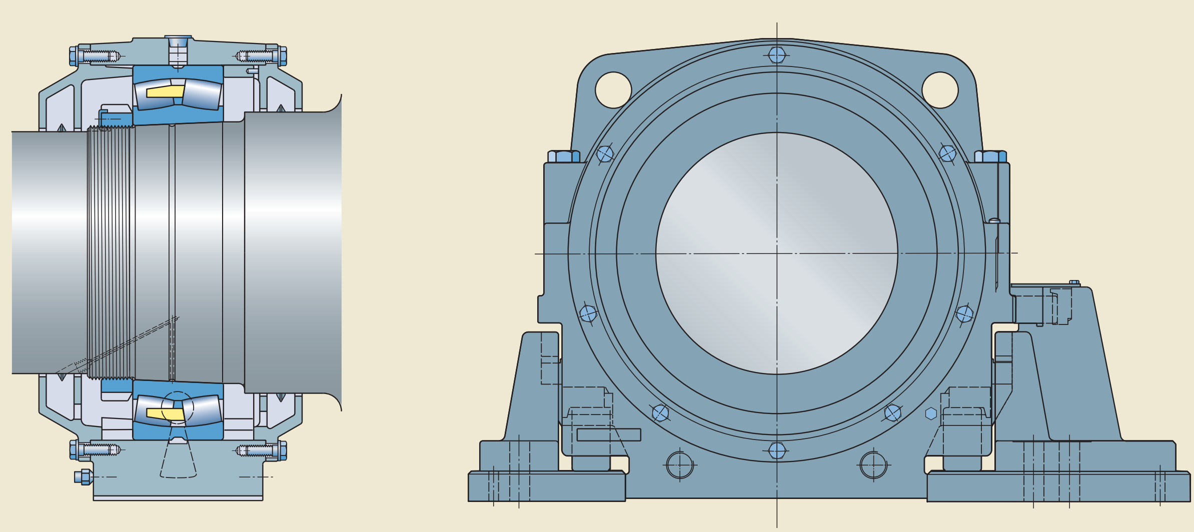
SKF recommends spherical roller bearings of series 230 and 231 for the drive side. For the front side, SKF recommends CARB toroidal roller bearing from series C 30 and C 31.

As the journals and inner rings will reach a much higher temperature than the outer rings during operation, the bearing must have a radial internal clearance greater than Normal. C4 clearance and C08 (reduced) run-out tolerances are generally recommended.

4

Yankee cylinder bearing housing with side rockers, front side

Fig. 4.21



Selection of bearing size

The radial load acting on the Yankee cylinder bearings depends on the mass of the cylinder, the water content and the position and the press load of the Yankee press rolls. The recommended basic rating life L_{10h} and SKF rating life L_{10mh} are 200 000 hours.

In modern machines, the Yankee press rolls can be placed at any suitable position around the periphery of the Yankee cylinder. When calculating the press loads, the following equations should be used:

$$G = g\ m/1\,000$$

$$G_1 = g\ m_1/1\,000$$

$$P_1 = F_{N1}\ L$$

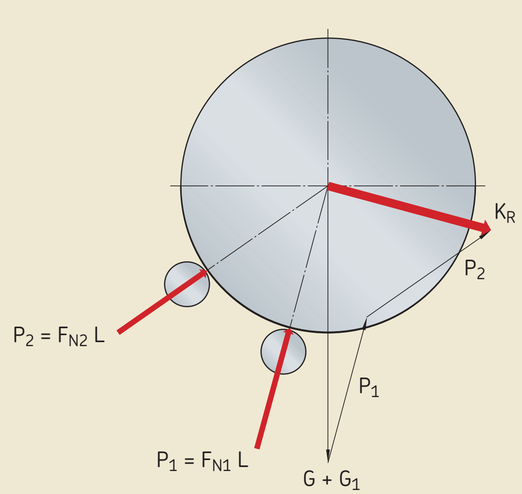
$$P_2 = F_{N2}\ L$$

where

- G = cylinder weight, kN
- G_1 = weight of water inside cylinder, kN
- g = 9,81 (acceleration of gravity), m/s^2
- m = cylinder mass, kg
- m_1 = mass of water inside cylinder, kg
- P_1 = press load from the first press nip, kN
- P_2 = press load from the second press nip, kN
- F_{N1} = linear load of the first press nip, kN/m
- F_{N2} = linear load of the second press nip, kN/m
- L = press nip length, m

The resultant load K_R on the Yankee cylinder

Fig. 4.22



When G, G_1, P_1 and P_2 are known, the resultant roll load K_R can be established either graphically (→ **fig. 4.22**) or by trigonometric calculations.

The following equations can be used in the case of shaft riding gear drive

Drive side bearing:

$$F_r = 0{,}5\ K_R + F_2$$

$$F_a = F_4 + F_5$$

$$F_5 = \mu\ F_r$$

Front side bearing:

$$F_r = 0{,}5\ K_R$$

$$F_a = F_4 + F_5$$

where

- F_r = radial bearing load, kN
- F_a = axial bearing load, kN
- K_R = resultant roll load, kN,
- F_2 = radial force from gear box, kN
- F_4 = axial force from steam box, kN
- F_5 = axial bearing load, due to friction between housing and outer ring, kN (if no CARB toroidal roller bearing or rocker housing is used at the front side)
- μ = coefficient of friction between housing and outer ring (use μ = 0,15 when calculating)

When a spherical roller bearing is used as a front side bearing, it is normally mounted in a housing with rockers and will therefore not be axially loaded by the cylinder. However, there are still some narrow machines being made where the thermal elongation of the cylinder is taken up via axial movement of the outer ring in the housing. In such cases, thrust loads coming from that movement have to be considered. The steam joint, depending on its location and design, may also cause one or both of the bearings to be axially loaded.

Lubrication

A separate circulating oil lubrication system is recommended for the Yankee cylinder bearings. Owing to the high temperature of the steam, large quantities of oil of appropriate viscosity must be passed through the bearings to achieve proper lubrication.

Many factors influence the calculation of the requisite oil flows, so these must be determined for each individual case. Consideration should be given to bearing size, speed, steam temperature, oil inlet temperature and journal insulation method. The influence of bearing load and ambient temperature on the temperature of Yankee cylinder bearings is small compared with that of the speed, steam temperature and insulation method.

SKF can carry out a computer analysis of the lubrication and temperature conditions for Yankee cylinder bearings on request.

For further information see *chapter 7*, *Lubrication*, and the examples in *chapter 8*, *Lubrication examples*.

Journal and housing tolerances for Yankee cylinders

See the indications given in *Tolerances* in *Chapter 1*.

If the bearing is mounted on a tapered journal, see *Tolerances for tapered journal seats* in *Chapter 1*.

If the journal is cylindrical please follow the recommendations in **table 4.3**. Note that direct mounting on a cylindrical journal is not recommended.

For housing tolerances, follow the recommendations in **table 4.3**.

Journal and housing tolerances for Yankee cylinders

Table 4.3

Cylindrical Journal	Mounting on sleeve Total radial run-out	h9Ⓔ IT5/2
Housing	No need of bearing axial displacement in the housing: Bearing axial displacement in housing	G7Ⓔ F7Ⓔ

Breaker stacks

The breaker stack is a single-nip press positioned in the dryer section and consists of two driven steel rolls, one of which may be a deflection-compensating roll.

The operating conditions are similar to those of ordinary press rolls except for the ambient temperature which is higher. The same sealing arrangement as for press roll bearings in the press section may be used in the dryer section even though bearings in that position are subjected to humidity only.

Bearing types

Spherical roller bearings from series 231 and 232 and CARB toroidal roller bearings from series C 31 and C 32 are recommended. Bearings with C3 radial internal clearance should be selected.

Selection of bearing size

Bearing loads are calculated as for ordinary press rolls but the recommended basic rating life L_{10h} and SKF rating life L_{10mh} life should reach 200 000 hours if possible.

Lubrication

The lubrication recommendations are basically the same as those for press rolls in the press section. The best lubrication condition is achieved by using AW or EP oil in a separate lubrication system, as in the case of deflection-compensating rolls. When the temperature allows, EP additives should be selected.

For further information, see *chapter 7, Lubrication* and the examples in *chapter 8, Lubrication examples*.

Journal and housing tolerances for breaker stack

See the indications given in *Tolerances* in *Chapter 1*.

If the bearing is mounted on a tapered journal, see *Tolerances for tapered journal seats* in *Chapter 1*.

If the journal is cylindrical please follow the recommendations in **table 4.4**. Note that direct mounting on a cylindrical journal is not recommended.

For housing tolerances, follow the recommendations in **table 4.4**.

Journal and housing tolerances for breaker stacks

Table 4.4

Cylindrical Journal	Mounting on sleeve	h9Ⓔ
	Total radial run-out	IT5/2
Housing	Bore diameter up to 400 mm	G7Ⓔ
	Bore diameter above 400 mm	F7Ⓔ

Size presses

The size press is a single-nip press positioned in the dryer section and consists of two driven steel rolls. The size press is not located under the hood.

The operating conditions are similar to those of ordinary press rolls. The same sealing arrangement as for press roll bearings in the press section is recommended.

Bearing types

Spherical roller bearings from series 230 and 231 and CARB toroidal roller bearings from series C 30 and C 31 are recommended. Bearings with normal or C3 radial internal clearance should be selected.

Selection of bearing size

Bearing loads are calculated as for ordinary press rolls but the recommended basic rating life L_{10h} and SKF rating life L_{10mh} life should reach 100 000 hours.

Lubrication

The lubrication recommendations are basically the same as those for press rolls in the press section. The best lubrication condition is achieved by using AW or EP oil in a separate lubrication system. When the temperature allows, EP additives should be selected.

For further information, see *chapter 7*, *Lubrication* and the examples in *chapter 8*, *Lubrication examples*.

Journal and housing tolerances for size press

See the indications given in *Tolerances* in *Chapter 1*.

If the bearing is mounted on a tapered journal, see *Tolerances for tapered journal seats* in *Chapter 1*.

If the journal is cylindrical please follow the recommendations in **table 4.5**. Note that direct mounting on a cylindrical journal is not recommended.

For housing tolerances, follow the recommendations in **table 4.5**.

4

Journal and housing tolerances for size presses

Table 4.5

Cylindrical Journal	Mounting on sleeve Total radial run-out	h9Ⓔ IT5/2
Housing	Bore diameter up to 400 mm Bore diameter above 400 mm	G7Ⓔ F7Ⓔ

Doctors

Specially designed multi-row ball bearings mounted direct on cylindrical journals are recommended for axially oscillating doctors. In some heavily loaded positions, like doctor blades for Yankees, plain bearings can be found (→ **fig. 4.23**).

The bearing arrangement depends largely upon how the machine manufacturer has designed and selected the associated components. A typical bearing arrangement is shown in **fig. 4.24**.

Selection of bearing size

For help with the selection of bearing size, please contact SKF.

Lubrication

Grease lubrication is normally used for the doctor bearings. Relubrication should be carried out every 48 hours to once a month depending on the position and grease. When used in the dryer section, high temperature grease should be selected. SKF recommends the grease SKF LGHB 2.

Oil bath lubrication using very high viscosity synthetic oil can also be employed. The sealing arrangement then has to be modified.

Journal and housing tolerances for doctors

See the indications given in *Tolerances* in *Chapter 1*.

For housing and journal tolerances, follow the recommendations in **table 4.6**.

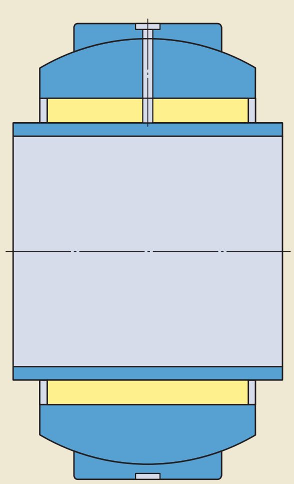
Housing and journal tolerances for doctors

Table 4.6

Journal	h6Ⓔ
Housing	J7Ⓔ

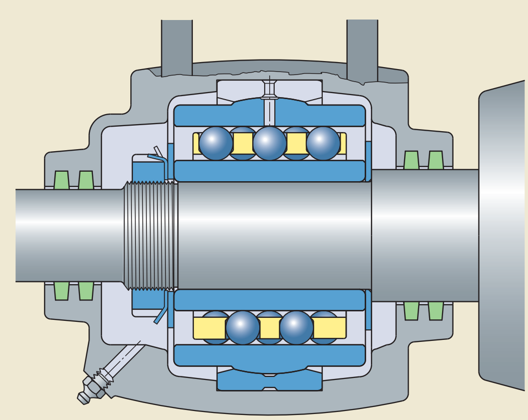
Doctor plain bearing

Fig. 4.23



Typical doctor bearing arrangement

Fig. 4.24



Rope sheaves

Rope sheaves can be found in the press, dryer and calender sections of paper machines. As such, they can be subjected to very different operating conditions. Environments vary from moist to dry with temperature ranges from ambient up to 120–130 °C. Furthermore, the operating hours of paper machines can be very different.

If drying cylinders are not equipped with rope sheaves, the rope wind around the drying cylinder and intermediate rope sheaves rotates constantly during paper machine operation. If the drying cylinders are equipped with rope sheaves at their front end, all rope sheaves can either stop or rotate slowly during normal operation once the paper has been guided through the paper machine.

Note that some felt rolls have also rope sheaves at their end.

Selection of bearing size

Rope sheaves located at the end of the drying cylinders are generally supported by one or two deep groove ball bearings of thin section 608 or 618 series with C3 clearance class (→ **fig. 4.25**). Some are equipped with angular contact ball bearings or single row thin section cylindrical roller bearings. SKF recommends thin section deep groove ball bearings of 618 series if possible.

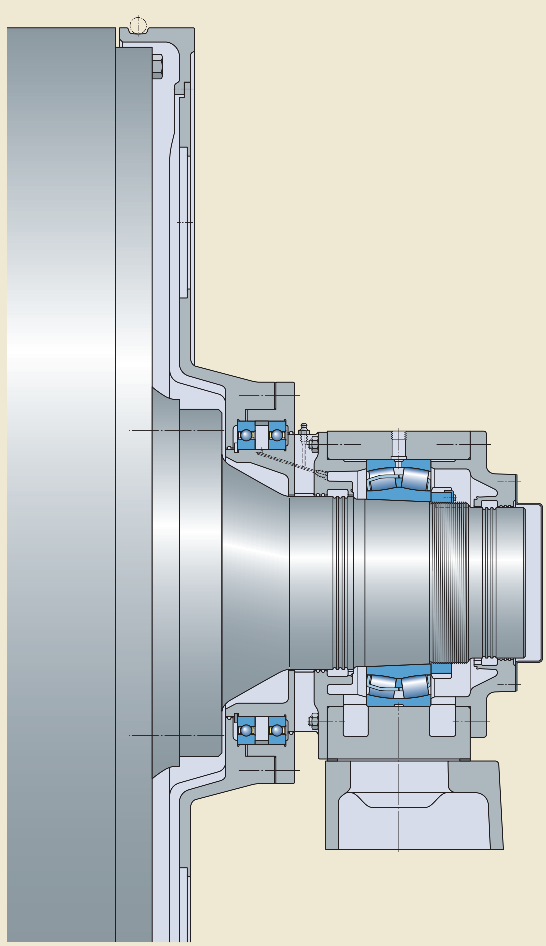
Other smaller diameter traditional intermediate rope sheaves are generally supported by two deep groove ball bearings, 60 or 62 series, C3 clearance class, with or without integrated seals (→ **fig. 4.26**).

In traditional rope sheaves, the outer ring rotates. This has two disadvantages:

- Lower grease life
- Faster rotation of the rolling elements

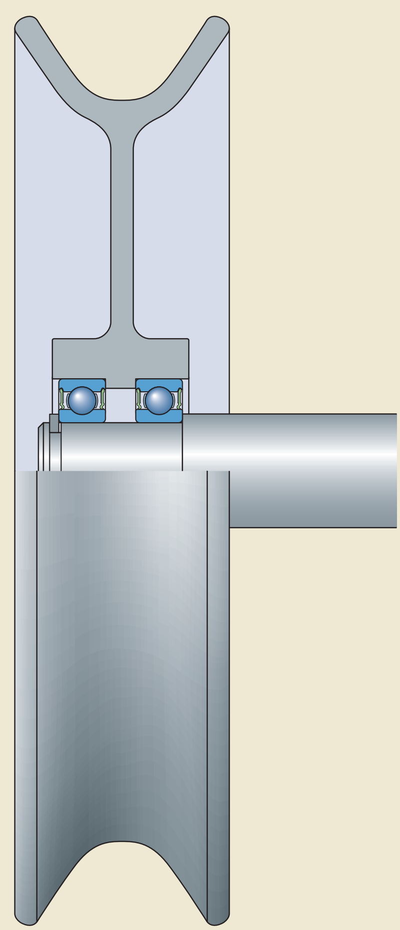
Rope sheave located at the drying cylinder front end face

Fig. 4.25



Traditional intermediate rope sheave

Fig. 4.26



They also tend to have limited protection against high-pressure cleaning as sealing is usually limited to a gap seal or integrated bearing seals (→ **fig. 4.26**, **page 4:21**). Furthermore, there is the risk that rope sheaves can fall off their shafts when the bearings are destroyed by heavy smearing.

To overcome these shortcomings, SKF lauched its first generation rope sheave unit (→ **fig. 4.27**) in 2003. These units had bearings with ceramic balls and inner ring rotation and were lubricated with long-life grease. Seal efficiency was increased using the combination of a labyrinth seal, created by the housing and rotating components, and integrated bearing seals. The outer diameter of the housing had a shoulder to prevent the sheaves falling.

A second generation SKF rope sheave unit was launched in 2012 (→ **fig. 4:28**) incorporating ceramic balls, long-life grease and additional V-ring seals for increased seal efficiency.

First generation SKF rope sheave unit

Fig. 4.27

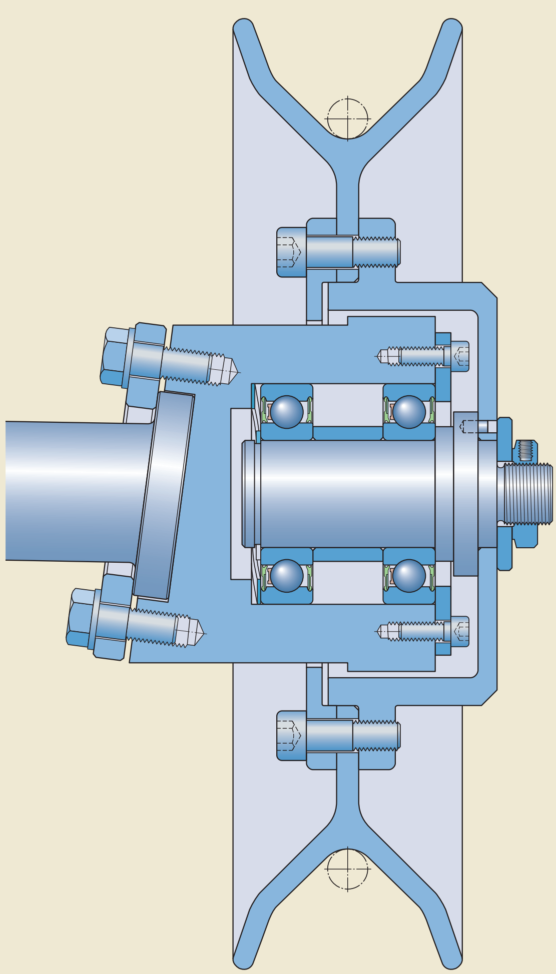


Table 4.7

Journal and housing tolerances for rope sheaves

Journal	Inner ring rotating, ball bearing and bore up to 100 mm	k5Ⓔ
	Inner ring standstill	g6Ⓔ
Housing	Outer ring rotating	N6Ⓔ
	Outer ring standstill	H7Ⓔ

Rope sheave bearings are usually oversized. The recommended basic rating life L_{10h} and SKF rating life L_{10mh} life should reach 100 000 hours.

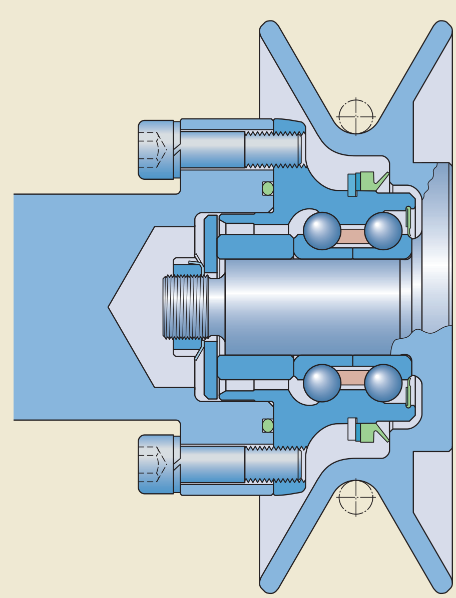
Lubrication

Rope sheaves are grease lubricated. Greased for life bearings should have, if possible, a grease life L_{10h} of at least 50 000 hours. SKF recommends high performance long-life grease with polyurea thickener type and synthetic base oil. For other bearings that are relubricated, SKF recommends SKF LGHP 2 grease.

For traditional rope sheaves located on shafts, not cylinder ends, in which the bearing outer ring rotates, SKF recommends a centralized automatic grease system if pos-

Second generation SKF rope sheave unit

Fig. 4.28



sible. Grease is supplied through ducts in the shaft.

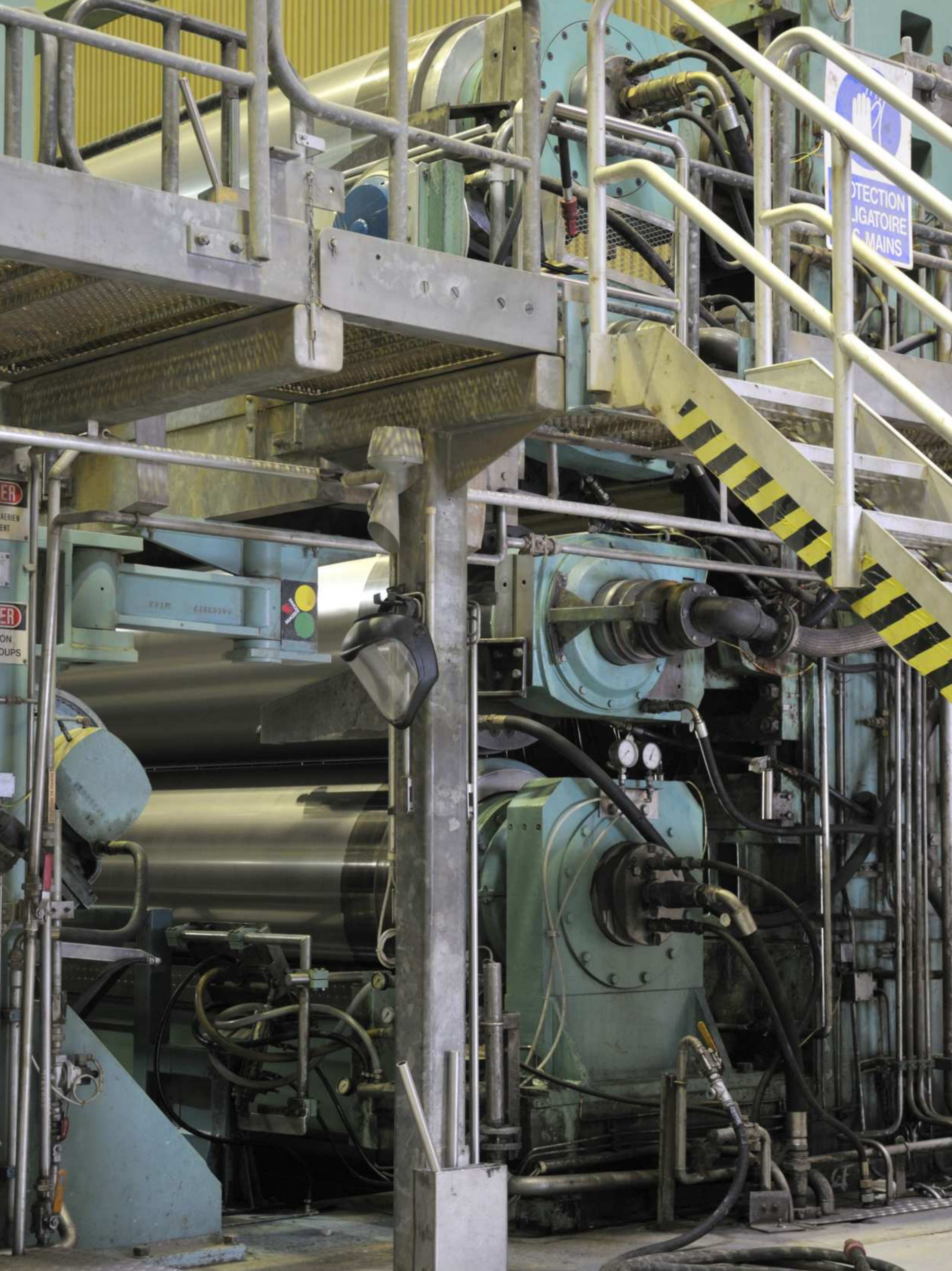
For further information, see *chapter 7*, *Lubrication*.

Journal and housing tolerances for rope sheaves.

See the indications given in *Tolerances* in *Chapter 1*.

For housing and journal tolerances, follow the recommendations in **table 4.7**. The table is based on the assumption that all drying cylinder or felt roll rope sheave bearings have outer ring rather than inner ring rotation.

Size press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Calenders

Coated paper is calendered or glazed to achieve a smoother surface and to make the paper shiny. Unglazed coated paper is quite dull and has a fairly uneven surface. Calendering can completely change the character of the paper.

5

A distinction is made between machine calendering (or machine glazing) which is performed in the paper machine itself and supercalendering which is carried out as a separate operation. The machine calender stack consists of all-steel rolls and imparts a certain smoothness to the paper. However, for most grades of coated paper this is not sufficient and a subsequent glazing operation in the supercalender is necessary.

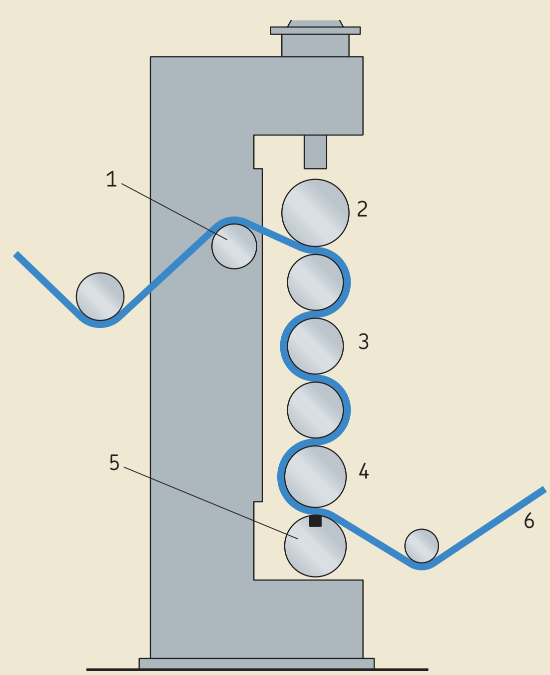
Machine calenders

The machine calender is positioned after the dryer section and consists of a stack of two to eight steel rolls resting on top of each other (→ **fig. 5.1**). On passing through the nips between the rolls, the paper is compressed and is given a smoother surface. In an eight-roll stack, the web is fed from the top downwards through seven roll nips with successively increasing nip pressure under the force of gravity. However, the nip pressures can be increased by applying pressure to the top roll. The king (bottom) roll is usually the driven roll. It is often a deflection-compensating roll which makes for a straight roll nip.

The result of the calendering operation is also temperature-dependent as the paper is more easily formable at higher temperatures. It is, therefore, customary to heat one or more of the calendar rolls using hot water, oil or steam.

Machine calender
1 *Spreader roll*
2 *Top roll*
3 *Intermediate roll*
4 *Queen roll*
5 *King roll*
6 *Paper web*

Fig. 5.1



Supercalenders

The supercalender, which is placed after the paper machine, consists of up to 14 rolls stacked one on top of the other (**fig. 5.2**).

The other rolls are carried in arrangements which allow them to move in the upright members of the machine frame and the rolls rest on top of each other. Pressure can be applied to the top roll either by means of levers in the case of old supercalenders, or hydraulically with supercalenders of more modern design.

The linear load in the bottom nip is usually 200–350 kN/m. The supercalender is usually driven by the bottom roll and the production speed may be above 2 000 m/min. The top and bottom rolls are usually of the deflection-compensating type.

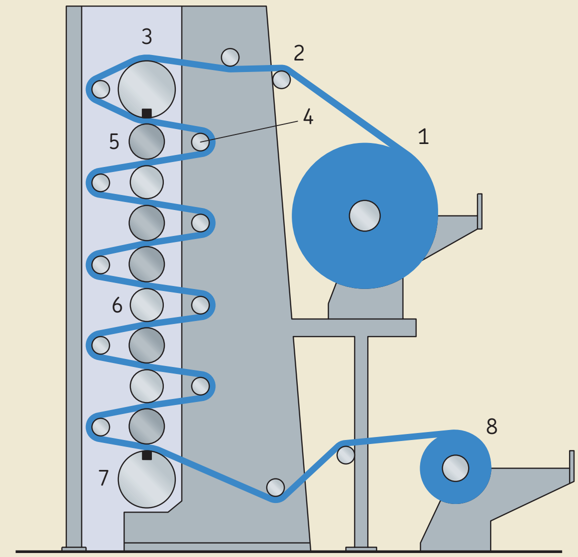
The feature that chiefly differentiates supercalenders from machine calenders is that alternate rolls in the stack are made of chilled cast iron and rolls of softer material. To treat the paper equally on both sides, the glazing process is changed by placing two fibre rolls against each other in the middle of the stack. The glazing effect is produced by the sliding that occurs in the nip between the soft and hard rolls. The surface running against the hard roll is given the highest gloss.

To increase the glazing action, some of the cast iron rolls are usually heated by hot water, steam or very hot oil.

Soft calenders

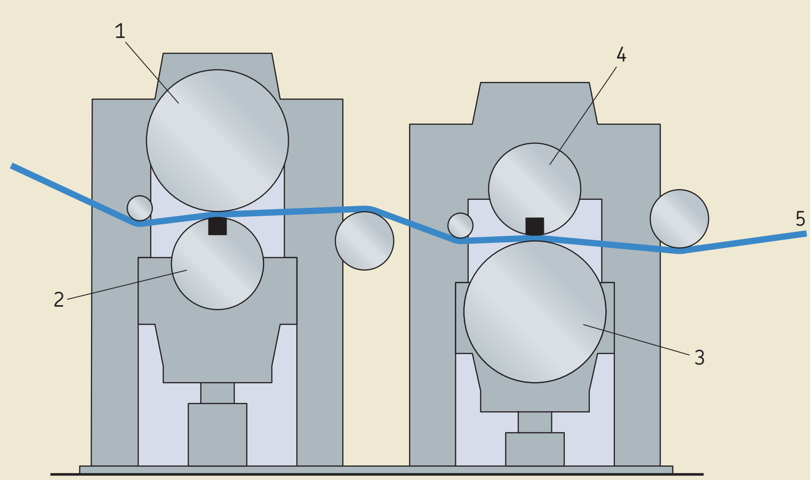
Another type of calendering process called soft-nip calendering is common today. In some cases, the soft calender (→ **fig. 5.3**) is being used to replace both the machine calender and the supercalender. The soft calender consists of one or two single-nip calenders with the linear load being roughly the same as for supercalenders. One roll is a full steel one while the other one is made of steel and coated with a layer of soft synthetic material. The main advantage with a soft press nip is that the density of the paper surface will be more even than with a hard press nip. This gives better printing properties.

Fig. 5.2

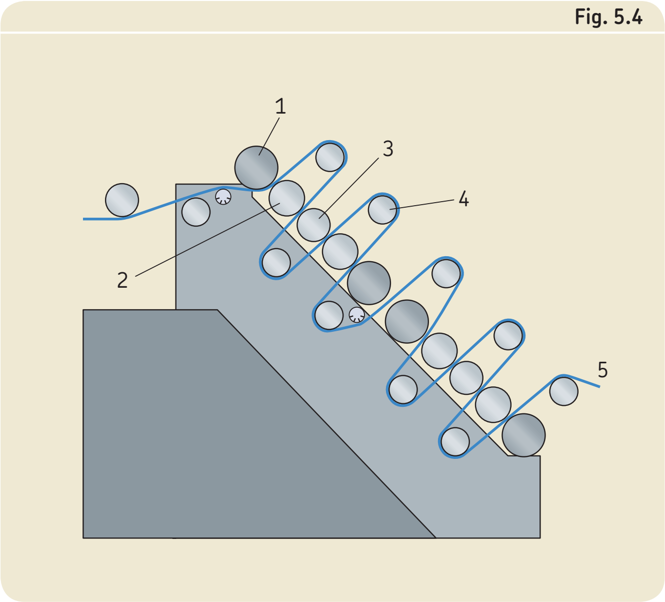


Supercalender
1 *Unwinding jumbo roll*
2 *Spreader roll*
3 *Top roll*
4 *Spreader roll*
5 *Fibre roll*
6 *Intermediate steel roll*
7 *Bottom roll*
8 *Winding jumbo roll*

Fig. 5.3



Two soft calenders in tandem for equal treatment of both sides of the paper
1, 3 *Heated plain roll*
2, 4 *Deflection-compensating press roll, soft coated*
5 *Paper web*



Voith Paper Janus calender
1 *Nipco roll*
2 *Heating roll*
3 *Elastic roll*
4 *Reversing roll*
5 *Paper web*

As a further development of the soft calendering process, one of the rolls is heated. Hot water, steam or very hot oil (at temperatures as high as 200–350 °C) is supplied to the plain roll or to the deflection-compensating roll.

Other calendering concepts

There is a trend towards a multi-nip online calendering process for printing and writing paper grades. Voith Paper has their Janus calenders (→ **fig. 5.4**) and Valmet their OptiLoad concept. In both cases, every nip can be individually loaded.

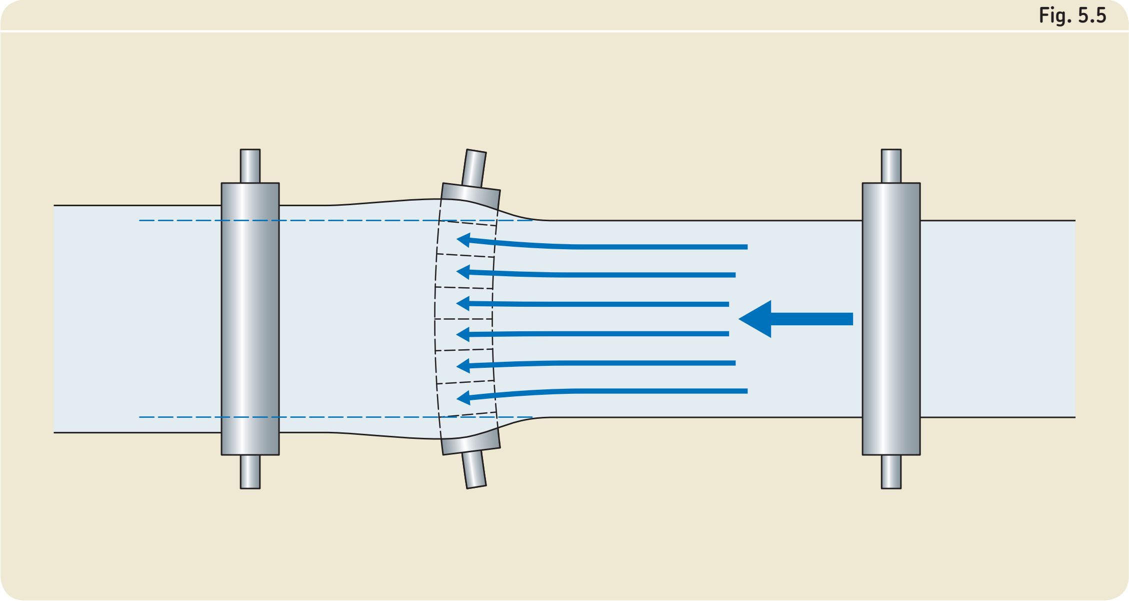
Spreader rolls

Spreader rolls eliminate wrinkles in the paper web. Spreader rolls are also used in the wet section for the forming wires and press felts and in the converting plant.

Spreader rolls are of different types:

- Crown rolls
- Straight rolls with rubber covers with two helical grooves, one being left handed, the other one being right handed, starting from the centre.
- Bowed rolls

Bowed rolls are the ones generally mounted on paper machines. Traditional bowed rolls have internal bowed stationary tubes or axles around which segmented metal rigid spools rotate (→ **fig. 5.5**). Modern spreader rolls have one rotating flexible tube around a stationary rigid tube.



Traditional spreader roll with segmented metal sleeves rotating around a bow tube to remove wrinkles

Bearing arrangements

Unheated plain calender rolls

The top, intermediate and queen rolls in a calender stack can be unheated. **Fig. 5.6** shows a basic design of the bearing housing and seals. The outer form of the housing and the fitting holes have to be made to suit the actual calender frame.

Traditionally, the bearings have been mounted on withdrawal sleeves but, owing to demands for reduced run-out, direct mounting on tapered journals is increasingly common.

Bearing types

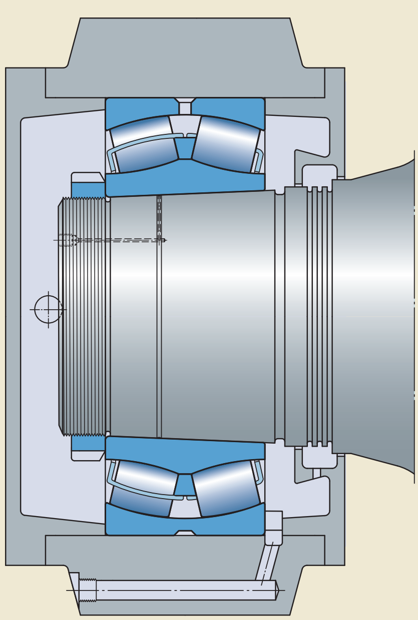
On the drive side, SKF recommends spherical roller bearings from series 230 and 231 for the top, intermediate and queen rolls while bearings from series 241 and 232 are used for the bottom rolls of plain type. On the front side, spherical roller bearings can also be used, but CARB toroidal roller bearings are more suitable. Bearings from series C 30 and C 31 are recommended for the top, intermediate and queen rolls, while bearings from series C 41 and C 32 are used for the bottom rolls of plain type.

Bearings with C3 radial internal clearance should be selected. C4 clearance class might be needed if the bearings rotate at very high speeds.

For this bearing position and all others with a nip, there is an increasing use of bearings that have reduced run-out tolerances (C08 or VQ424). The need for these bearings increases with faster paper speeds. For very high speeds, VA460 bearings are recommended.

Locating bearing for unheated plain calender roll

Fig. 5.6



Selection of bearing size

The pressure in the roll contact is brought about by the mass of the rolls and the applied press load from the top roll. The top and bottom roll bearings have to carry the main load while the bearings of the intermediate rolls are only lightly loaded. The recommended basic rating life L_{10h} and SKF rating life L_{10mh} are 120 000 hours.

Lubrication

The lubrication recommendations are the same as those for plain press rolls in *chapter 3, Press section.*

Journal and housing tolerances for unheated plain calender rolls

See the indications given in *Tolerances* in *Chapter 1*.

If the bearing is mounted on a tapered journal, see *Tolerances for tapered journal seats* in *Chapter 1*.

If the journal is cylindrical, please follow the recommendations in **table 5.1**.

For housing tolerances, follow the recommendations in **table 5.1**.

Table 5.1

Journal and housing tolerances for unheated plain calender rolls

Cylindrical Journal	Mounting on sleeve	h9Ⓔ
	Radial run-out	IT5/2
Housing	Bore diameter up to 400 mm	G7Ⓔ
	Bore diameter above 400 mm	F7Ⓔ

Heated plain calender rolls

The heating medium is normally supplied to the intermediate or the queen roll. **Fig. 5.7** shows a suitable bearing arrangement with insulation for a heated roll. The outside form of the housing and the fitting holes have to be made to suit the actual calender frame.

SKF always recommends insulation of the journals for all types of heated rolls.

Bearing types

For the intermediate and queen rolls on the drive side in a calender, SKF recommends spherical roller bearings from series 230 and 231 while bearings from series 241 and 232 are used for the bottom rolls of plain type.

Spherical roller bearings can also be used on the front side, but CARB toroidal roller bearings are more suitable. For the intermediate and queen rolls, SKF recommends bearings from series C 30 or C 31 while bearings from series C 41 and C 32 are recommended for bottom rolls of plain type.

In modern, high-speed, two-roll soft calenders, the heated plain roll is in the top position of the first nip. This may lead to very light loads on the bearing. For such applications, SKF recommends use of spherical roller bearings from series 231 as the axially locating bearing and CARB toroidal roller bearing from series C 31 as the non-locating bearing. In cases of very light loads, SKF recommends bearings with NoWear coated rollers (L5DA) (→ **fig. 5.8**).

The bearings are mounted either on withdrawal sleeves or direct on tapered journals. For this bearing position, as well as all others with a nip, there is an increasing use of bearings with reduced run-out tolerances (C08 or VQ424). The need for these bearings increases with faster paper speeds. For very high speeds, SKF recommends VA460 bearings.

If there is no journal insulation, bearings with case-hardened inner rings (HA3) may have to be selected. SKF should always be consulted where non-standard bearings are required.

The use of bearings with C3 or, when the temperature of the medium heating the roll is above 100 °C, C4 radial internal clearance is recommended. If the speed is very high, C4 and C5 radial internal clearance may be necessary.

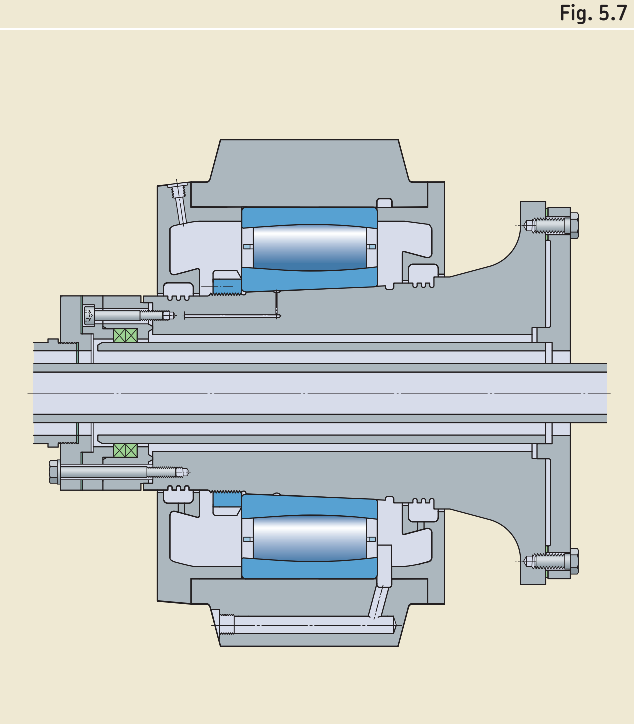
Selection of bearing size

The pressure in the roll contact is brought about by the mass of the rolls and the applied press load from the top roll. The top and bottom roll bearings have to carry the main load while the bearings of the intermediate rolls are only lightly loaded.

The recommended basic rating life L_{10h} and SKF rating life L_{10mh} are 120 000 hours.

Non-located bearing for heated plain calender roll

Fig. 5.7



Large size spherical roller bearing in NoWear execution

Fig. 5.8

Lubrication

The bearings in this particular application are lubricated by a circulating oil system. The oil is supplied via the groove and holes in the outer rings of spherical roller bearings and from the side for CARB toroidal roller bearings. Note that spherical roller bearings are also sometimes lubricated from the side.

Lubrication conditions such as oil viscosity, oil flow rate and bearing temperature for heated calender rolls are calculated principally in the same way as for drying cylinders.

Sometimes upper roll bearings in soft calenders operate with loads less than the minimum load recommendations in the SKF catalogue *Rolling bearings*. If so, bearings with NoWear coated rollers (L5DA) should be selected. Alternatively, approved polyglycol oils could be used.

For further information, see *chapter 7*, *Lubrication* and *examples 33–35* in *chapter 8*, *Lubrication examples*.

Journal and housing tolerances for heated plain calender rolls

See the indications given in *Tolerances* in *Chapter 1*.

If the bearing is mounted on a tapered journal, see *Tolerances for tapered journal seats* in *Chapter 1*.

If the journal is cylindrical, please follow the recommendations in **table 5.2**.

For housing tolerances, follow the recommendations in **table 5.2**.

Journal and housing tolerances for heated plain calender rolls

Table 5.2

Cylindrical Journal	Mounting on sleeve Total radial run out	h9Ⓔ IT5/2
Housing	No need of bearing axial displacement in the housing: Bearing axial displacement in housing required:	G7Ⓔ F7Ⓔ

Spreader rolls

On traditional spreader rolls, segmented metal spools normally rotate on deep groove ball bearings (→ **fig. 5.9**). There are exceptions like some spreader rolls for super calenders that used spherical roller bearings in the past and self-aligning ball bearings these days (→ **fig. 5.10**).

Bearing types

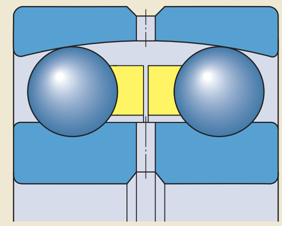
Most spreader rolls are equipped with thin section deep groove ball bearings from 160, 619 and 618 series or deep groove ball bearings in inch dimensions.

Previously, spreader rolls on supercalenders used spherical roller bearings, but excess friction led to temperature variation along the roll and problems associated with insufficient loading in some cases. Therefore, the spherical roller bearings were replaced with self-aligning ball bearings with fewer and smaller balls and very low friction (→ **fig. 5.11**). More information on these bearings can be found on *pages 558* and *559* of the SKF catalogue *Rolling bearings*.

Spreader rolls mounted on high-speed paper machines can use hybrid bearings or/and bearings with less or smaller balls. Some have reduced dimensional and radial run out tolerances such as P5 radial run-out.

C3 radial clearance class is recommended for the deep groove ball bearings to accept some tilting. The self-aligning ball bearings in the 139 and 130 series designed for spreader rolls have C3 clearance class as standard. In some cases, a higher clearance class can be needed.

Fig. 5.11



SKF self-aligning ball bearing

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Above: Spreader roll with ball bearings

Below: Supercalender spreader roll with self-aligning ball bearings replacing spherical roller bearings

Fig. 5.9

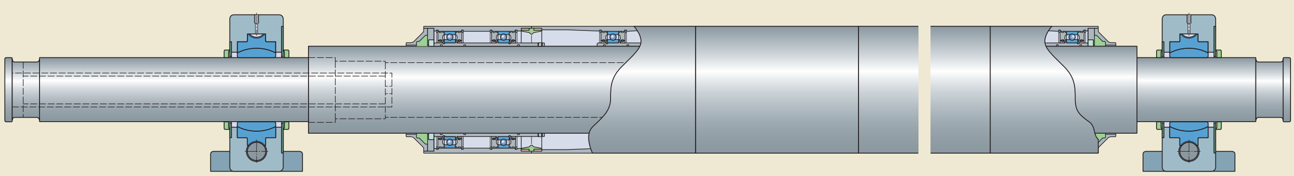
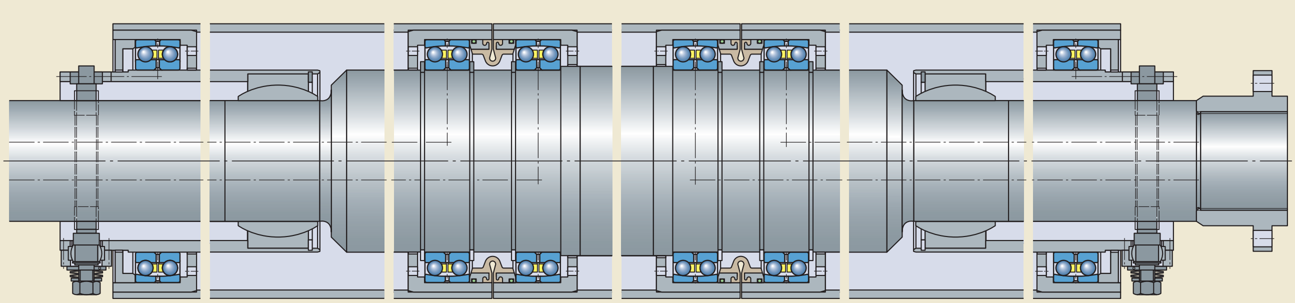


Fig. 5.10



Selection of bearing size

Bearing size depends mainly on the spreader roll size and calculated life well above the recommended basic rating life L_{10h} and SKF rating life L_{10mh} i.e. 120 000 hours. Bearing internal friction is a key parameter to avoid paper web slippage and temperature variation along the spreader roll.

The selection of the bearing size must also take in to account the minimum load needed to provide satisfactory operation. The web tension, thus the contact load on the spreader roll, is small due to the small wrap angle (20 to 35 degrees). The wrap angle is greater on supercalendars.

Service life, however, is on average between two and three years due to lubrication issues. As the majority of the bearings are greased for life, the bearing service life will depend on the grease service life.

Lubrication

The majority of the deep groove ball bearings mounted in spreader rolls are not relubricated and rely on their initial grease fill with a low viscosity base oil lubricant. As a rule of thumb, greases with an ISO VG 32-100 base oil are suitable for ambient temperatures up to 60 °C and ones with ISO VG 68-150 base oil for higher temperatures. Above 60 °C ambient temperature, synthetic base oil and polyurea thickener are recommended to increase the grease life.

Hybrid bearings are less sensitive to lubrication issues and also only rely on their initial grease fill.

Sealed deep groove ball bearings offer some advantages, but are rare in this application. Thanks to the external seal assembly in such bearings, a larger grease reservoir can be created which extends the grease service life. The external seal design must be such that it retains the grease base oil which centrifugal forces tend to force out of the bearing. Furthermore, it should have low or very low friction.

Self-aligning ball bearings are designed to be relubricated with grease. They have lubricating holes in the inner ring and a groove in the stationary axle bearing seat. Excess grease is purged.

The quantity of grease and the viscosity of the grease base oil have a direct influence on bearing temperature, friction and the necessary minimum load. SKF does not advise using higher viscosity oil or grease base oil than recommended as increased friction will generate heat.

For further information, see *chapter 7, Lubrication*.

Journal and housing tolerances for traditional spreader roll

See the indications given in *Tolerances* in *Chapter 1*.

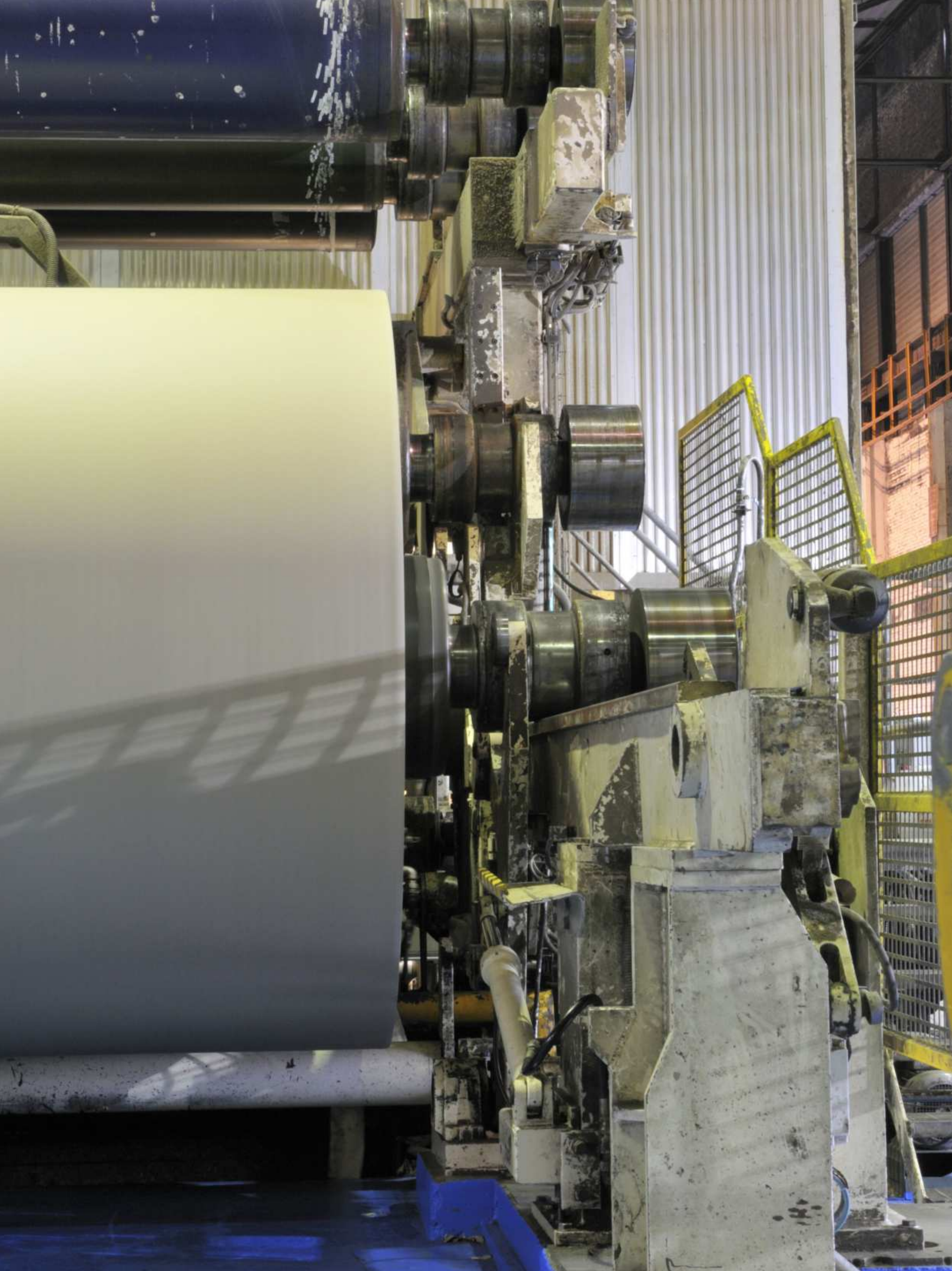
Please follow the recommendations in **table 5.3**.

Table 5.3

Journal and housing tolerances for traditional spreader rolls

Cylindrical Journal	h6Ⓔ
Housing	M6Ⓔ

Picture from Voith GmbH



Reelers

Most large paper machines produce a continuous paper web which is 8–10 metres wide. Such a wide and long web has to be cut and made into smaller rolls before distribution to the customers. The first reeling, or spool, is done in the reeler at the end of the paper machine.

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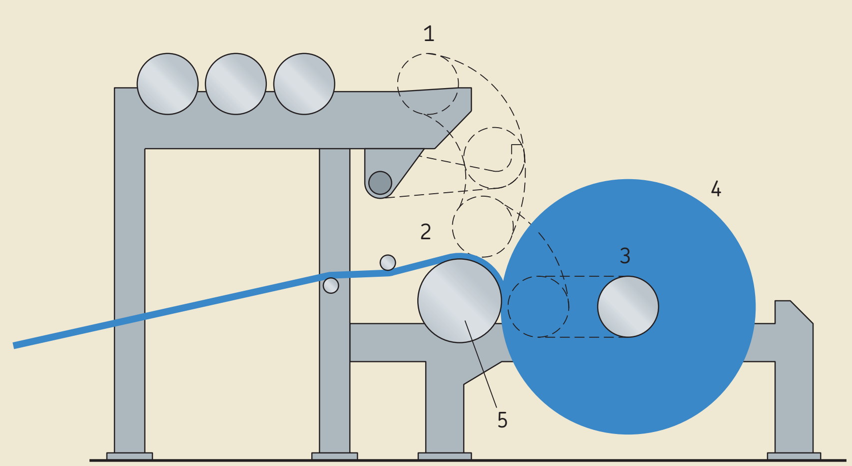
When the roll of paper on the reel reaches the desired diameter, reeling is continued on a new spool (→ **fig. 6.1**). The jumbo roll of paper is transferred to the slitter and rewinder where it is cut and rewound into rolls of the size required by the customer.

Fig. 6.1



Reeler
1 *Reel spool storage*
2 *Web turn-up to empty spool*
3 *Reeling position*
4 *Jumbo roll*
5 *Reel drum*

Bearing arrangements

The reeler bearings work in a relatively good environment. The surroundings are dry and the temperature is around 25 °C. However, it is important for the bearings to be protected from paper dust.

Reel drums

Reel drums are normally carried by spherical roller bearings on both sides or spherical roller bearings on the drive side and CARB toroidal roller bearings on the front side. Reel drum bearings are usually mounted on adapter sleeves or directly on tapered journals.

A bearing arrangement for a reel drum is shown in **fig. 6.2**. If an adapter sleeve (with abutment spacer) is incorporated in the design, SKF recommends machining an annular groove in the journal outside the sleeve position. This groove can then be used to take a backing ring when a hydraulic nut (HMV) is used to dismount the bearing.

Bearing types

SKF recommends spherical roller bearings of series 231 and 240 as well as CARB toroidal roller bearings of series C 31. Normal radial internal clearance is recommended.

Selection of bearing size

Bearing size selections should be based on life calculations according to the recommendations in *chapter 1*, *General requirements and recommendations*. SKF recommends that both the basic rating life L_{10h} and the SKF rating life L_{10mh} are taken into consideration. The calculated basic rating life L_{10h} should exceed 120 000 hours, while a SKF rating life L_{10mh} of 100 000 hours is recommended. There are some exceptions due to the fact that too lightly loaded bearings can fail even if calculated life is over 100 000 hours. SKF recommend checking that the minimum load criterion is met (for details see the product chapters in the SKF catalogue *Rolling bearings*).

Lubrication

Slow running machines can be grease-lubricated, but bearings in this particular application are mostly lubricated with a circulating oil system. For spherical roller bearings, the oil is supplied via the groove and holes in the outer ring of the bearing. An annular groove turned in the housing to coincide with the holes improves lubrication. For CARB toroidal roller bearings, the oil is supplied from the side.

For further information, see *chapter 7*, *Lubrication* and *Lubrication examples*, *chapter 8*.

Journal and housing tolerances for reel drums

Read the indications given in *Tolerances* in *Chapter 1*.

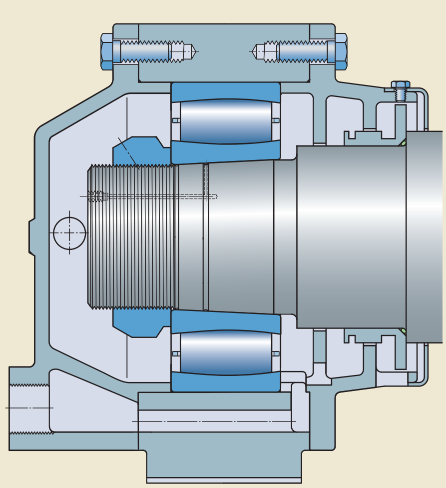
If the bearing is mounted on a tapered journal, see *Tolerances for tapered journal seats* in *Chapter 1*.

If the journal is cylindrical please follow the recommendations in **table 6.1**.

For housing tolerances, follow the recommendations in **table 6.1**.

Reel drum bearing arrangement

Fig. 6.2



Journal and housing tolerances for reel drums

Table 6.1

Cylindrical Journal	Mounting on sleeve Radial run-out	h9Ⓔ IT5/2
Housing		G7Ⓔ

Reel spools

Reel spools are generally carried by two spherical roller bearings per journal – one mounted on a withdrawal sleeve, the other mounted directly on a cylindrical journal seating.

Some reel spools are carried by two tapered roller bearings.

The bearing housings (→ **fig. 6.3**) are usually cylindrical and are designed to fit both in the reeler and the winder where the slitting takes place. A lubrication duct should be provided in each housing to facilitate lubrication of the bearings.

Bearing types

SKF recommends spherical roller bearings of series 230 and 231. While Normal radial internal clearance was recommended in the past, the recommended journal tolerances for bearings with cylindrical bores have changed and give tighter fits than in the 4th edition of the *Rolling bearings in paper machines* handbook. As such, the main bearings may need to have C3 radial internal clearance because of increased radial clearance reduction due to the increased tight fit.

Selection of bearing size

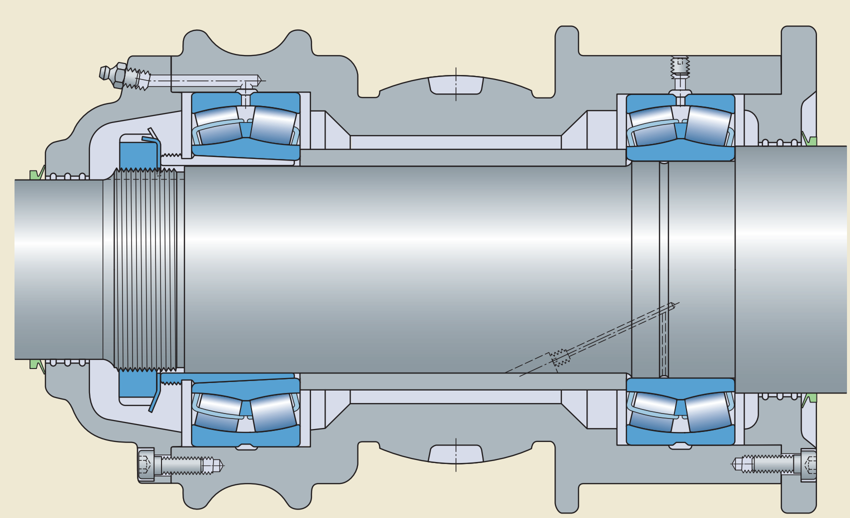
Bearing size selections should be based on life calculations according to the recommendations in *chapter 1*, *General requirements and recommendations*. SKF recommends that both the basic rating life L_{10h} and the SKF rating life L_{10mh} are taken into consideration. The calculated basic rating life L_{10h} should exceed 120 000 hours. There are some exceptions due to the fact that too lightly loaded bearings can fail even if calculated life is over 100 000 hours. SKF recommend checking that the minimum load criterion is met (for details see the product chapters in the SKF catalogue *Rolling bearings*).

During the reeling of the paper in the machine and the re-reeling from the reel spool, the paper speed is constant and the rotational speed of the reel spool varies. In order to calculate the bearing life, the equivalent speed and load have to be determined.

The constant mean rotational speed n_m which, when multiplied by the reeling time, will give the same number of revolutions as occurs in reality; can be calculated using equation 1.

Reel spool bearing arrangement

Fig. 6.3



Equation 1:

$$n_m = \frac{2V}{(D + d_0)\pi}$$

where
n_m = mean rotational speed
V = linear speed of the paper web in m/min
D = maximum external diameter of the reel in m
d_0 = diameter of the reel spool in m

A constant mean load, acting on the journal bearing arrangement and equivalent to the real load, can be calculated with the aid of equation 2:

Equation 2:

$$F_m = 9{,}81 \left[\frac{A^3}{35}\left(5\,D^3 + 20\,D^2\,d_0 + 29\,D\,{d_0}^2 + 16\,{d_0}^3\right) + \right.$$

$$+ \frac{A^2\,m_0}{5}\left(3\,D^2 + 9\,D\,d_0 + 8\,{d_0}^2\right) +$$

$$\left. + A\,{m_0}^2\left(D + 2\,d_0\right) + {m_0}^3 \right]^{1/3}$$

where

$$A = \frac{m}{D + d_0}$$

D = maximum external diameter of the reel in m
d_0 = diameter of the reel spool in m
m_0 = mass of reel spool in kg
m = mass of the reeled paper in kg
F_m = mean load in N

The equations are valid for reeling as well as re-reeling. The reel spool has two bearings per journal (→ **fig. 6.3**). In some cases, the housing rests in different ways when reeling and re-reeling. When reeling, for example, one bearing may take the full load whereas the load is shared equally between the two bearings during re-reeling.

The re-reeling speed is normally much higher, but of course the total number of revolutions is the same for the two operations. If the bearing load is different, two values, F_{m1} and F_{m2}, are calculated and then a total constant mean bearing load can be calculated using

$$F_m = \sqrt[3]{\frac{{F_{m1}}^3 + {F_{m2}}^3}{2}}$$

With $P = F_m$ the total life in number of revolutions L_{10} can be calculated with the aid of the life equation. Since half of these revolutions are for reeling and half for re-reeling, the L_{10h} life in hours should be calculated for each operation; the total L_{10h} life will then be

$$L_{10h} = \frac{(L_{10h1} + L_{10h2})}{2}$$

for reeling:

$$L_{10h1} = \frac{1\,000\,000}{60\,n_{m1}}\left(\frac{C}{P}\right)^{10/3}$$

or re-reeling:

$$L_{10h2} = \frac{1\,000\,000}{60\,n_{m2}}\left(\frac{C}{P}\right)^{10/3}$$

where
L_{10h} = basic rating life, operating hours
n_{m1} = rotational speed for reeling, r/min
n_{m2} = rotational speed for re-reeling, r/min
C = basic dynamic load rating, N
P = equivalent dynamic bearing load, N

n_{m1} and n_{m2} are calculated with the help of **equation 1**.

The required value L_{10h} is estimated in relation to the number of reel spools available. The bearings are generally oversized.

Example:

A reel spool with a bearing arrangement according to **fig. 6.3**, **page 6:3** in this chapter has a d_0 = 0,65 m and a mass of m_0 = 4 200 kg. Maximum paper mass m = 16 800 kg and then D = 2,7m. Paper speed in the machine is 600 m/min. The support in the machine is such that one bearing at each side takes the entire load when reeling. During re-reeling, the paper speed is 2 000 m/min and then all four bearings share the load equally. What bearing life can be expected for the most heavily loaded bearing?

SKF Explorer spherical roller bearing 23030 CC(K)/W33 with a basic dynamic load rating C = 510 000 N is selected.

Mean speed during reeling:

$$\frac{2 \times 600}{(2,7 + 0,65)\pi} = 114,02 \text{ rpm}$$

Mean speed during re-reeling:

$$\frac{2 \times 2\,000}{(2,7 + 0,65)\pi} = 380 \text{ rpm}$$

Mean load, which is the same during reeling and re-reeling:

$$A = \frac{16\,800}{2,7 + 0,65} = 5\,014,92 \approx 5\,015$$

$$F_m = 9,81 \Bigg[\frac{5\,015^3}{35} \left(5 \times 2,7^3 + 20 \times 2,7^2 \times 0,65 + 29 \times 2,7 \times 0,65^2 + 16 \times 0,65^3\right) + $$

$$+ \frac{5\,015^2 \times 4\,200}{5} \left(3 \times 2,7^2 + 9 \times 2,7 \times 0,65 + 8 \times 0,65^2\right) +$$

$$+ 5\,015 \times 4\,200^2 \times \left(2,7 + 2 \times 0,65\right) + 4\,200^3 \Bigg]^{1/3}$$

F_m = 126 145 N

The load when reeling is shared equally by two bearings, one on each journal. The load on one bearing is then 63 072,5 N.

The load when re-reeling is shared equally by four bearings, one on each journal. The load on one bearing is then 31 536,25 N.

The constant mean load, thus the equivalent dynamic bearing load of the most loaded bearing is then:

$$P = \sqrt[3]{\frac{63\,072,5^3 + 31\,536^3}{2}}$$

P = 52 065 N

$$L_{10} = \left(\frac{510\,000}{52\,065}\right)^{10/3}$$

L_{10} = 2 011 million revolutions

When reeling, the mean speed is equal to 114,2 rpm.

$$L_{10} = \frac{1\,000\,000}{60 \times 114,2} \times 2\,011 = 293\,500 \text{ hours}$$

When re-reeling the mean speed is equal to 380 rpm.

$$L_{10} = \frac{1\,000\,000}{60 \times 380} \times 2\,011 = 88\,200 \text{ hours}$$

The total basic rating life L_{10h} is thus 1/2 (293 500 + 88 200) = 190 850 hours.

It can be useful to check the minimum load acting on the bearings in comparison to the requisite minimum load. The lowest load and highest speed is obtained during re-reeling. The lowest load is the load of the reel spool only shared by four bearings 4 200 kg / 4 = 1 050 kg or 10 300 N

The SKF catalogue *Rolling bearings* indicates that the requisite minimum load to be applied to spherical roller bearing can be estimated using P_m = 0,01 C_0, C_0 being the basic static load rating

For a 23030 CC(K)/W33, C_0 = 710 000 N thus P_m = 7 100 N.

P_m is below 10 300 N, hence the bearings are sufficiently loaded.

For CARB bearings, as well as for spherical roller bearings under certain conditions, lower values are allowed. For more information, see the SKF catalogue *Rolling bearings*.

It has to be noted that during reeling, one bearing has, in theory, no load. In practice, the bearing will have some load and the speeds are not high enough to create problems. Nevertheless, this can become a critical issue above a certain speed. SKF does not recommend just one bearing per journal taking the main load in high-speed machines.

Lubrication

The bearing arrangements for reel spools are usually lubricated with the same grease used in the forming and press section. Relubrication should be carried out once a month.

For further information, see the *Lubrication chapter*.

Journal and housing tolerances for reel spools

See the indications given in *Tolerances* in *Chapter 1*.

If the bearing is mounted on a tapered journal, see *Tolerances for tapered journal seats* in *Chapter 1*.

If the journal is cylindrical, please follow the recommendations in **table 6.2**.

For housing tolerances, follow the recommendations in **table 6.2**.

Journal and housing tolerances for reel spools

Table 6.2

Cylindrical Journal	Mounting on sleeve	h9Ⓔ
	Radial run-out	IT5/2
	Direct mounting	
	(100) to 200 mm	p6Ⓔ
	(200) to 280 mm	r6Ⓔ
Housing		G7Ⓔ

Lubrication

This part of the handbook provides lubrication guidelines for almost all the bearing positions in the paper machine. By following these guidelines, the user should be able to reduce many common bearing failures related to lubrication.

Why lubricate?

Lubrication in earlier times

Lubricants are products that predate the industrial age. They have been used in one form or another for several thousand years to fight friction or for as long as man has needed to move bulky objects.

The first great civilisation evolved in the Middle East some six thousand years ago. Trade, construction and war necessitated the transport of heavy goods, but it was not always an easy matter.

An Egyptian illustration from some four thousand years ago demonstrates the point. What was in the jar, shown in **fig. 7.1**? It could have been water that lubricated the runners of the stone sledge as it was pulled along a roadway of clay or Nile mud.

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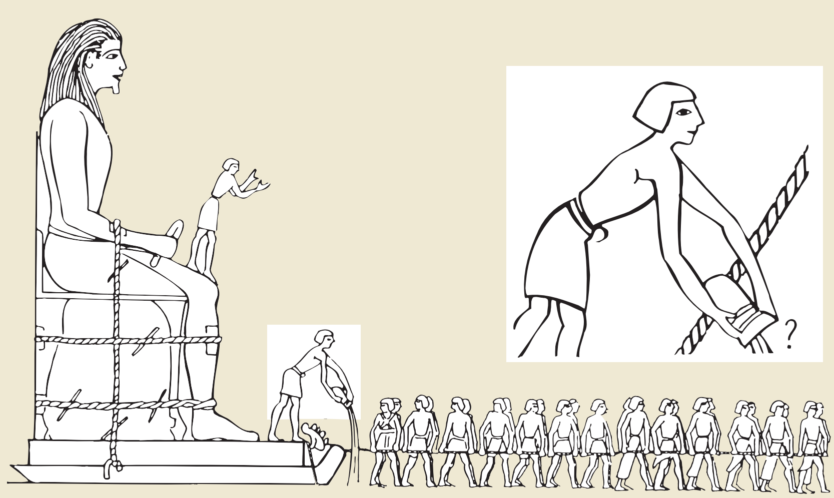
Fig. 7.1



Early use of lubricants

It was also in Egypt that an ingenious man is said to have discovered that when he stuffed animal fat between the axle and the wheel hub, the cart moved much more easily and wear was also reduced.

Olive oil was the general-purpose fat for the Mediterranean peoples. They also used it when they needed a lubricant. Analyses of remains show that the lubricating "grease" used by the ancient Egyptians consisted of olive oil mixed with lime.

Mineral oil was also known in very early times, but it was not used for lubrication. People used oil in medicines and to make their canoes watertight.

The story of lubrication began with these modest experiments using nature's various products, but it was not until the 19th century that the fight against friction really got into its stride. 1859 is the year usually stated for the birth of the oil industry. It was then that the first oil well was drilled in Pennsylvania, USA.

Oil films

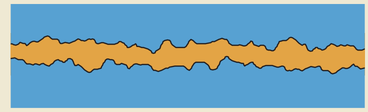
Lubrication plays a major role in rolling bearing performance. The main task of the lubricant is to build up an oil film between the rolling elements and the raceways. The oil film should be thick enough to separate these mating surfaces completely.

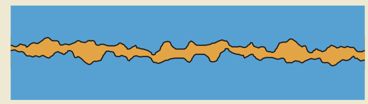
Sometimes the question is asked whether the radial internal clearance in the bearing can be too small to allow an oil film to form. This is not the case as even bearings with zero clearance can build up an oil film. An average oil film has a thickness of only 0,3 µm. By way of comparison, a sheet of printing paper has a thickness equal to about 200 oil films.

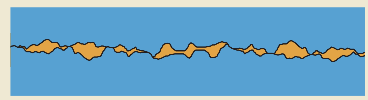
The film thickness in a rolling bearing is dictated by the bearing size, the operating speed and the viscosity at the operating temperature of the oil used. The higher the speed, the thicker the oil film. Three different terms are used to describe the lubrication conditions: full, mixed or boundary lubrication (→ **fig. 7.2**).

Fig. 7.2

Lubrication conditions


full lubrication


mixed lubrication


boundary lubrication

The bearings for high-speed suction rolls are an example of paper machine bearings with full lubrication. Unfortunately, there are also some disadvantages with increased speed. One is the risk of sliding between the rolling elements and the raceways. If the sliding speed of rolling elements is too high in relation to the rolling speed, the oil film can be broken. This normally leads to serious damage to the bearing. In such cases, NoWear bearings can solve the problem.

Felt roll bearings in the dryer section are examples of bearings with mixed lubrication. These bearings do not suffer much lubrication-related damage because the lubrication conditions are still relatively good and the speed is moderate.

The bearings for drying cylinders without insulation are examples of bearings with boundary lubrication. Whether insufficient lubrication in this case leads to serious damage or to just mild wear and running-in of the surfaces depends on the actual operating conditions.

General notes on lubrication

Basic terms

When selecting suitable lubrication for rolling bearings, there are some basic terms that need to be known.

Kinematic viscosity ν

The kinematic viscosity ν describes the resistance to flow of the oil at a certain temperature. Low values mean that the oil flows easily; high values mean that the oil flows sluggishly. The unit for viscosity is mm^2/s (previously known as Centistoke).

Table 7.1 lists the ISO classification of oils used in paper machines. Note that ISO VG 150 and ISO VG 220 are the ones commonly used in paper machine circulating oil systems.

κ value

κ is the viscosity ratio which is the ratio between the actual operating viscosity ν and the rated viscosity ν_1 of the oil, both at operating temperature, i.e.

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

The rated viscosity ν_1 which is required for adequate lubrication depends on the bearing mean diameter and rotational speed.

For greases, the viscosity of the base oil is taken.

Diagram 7.1 is used to determine ν_1. When ν_1 is known, **diagram 7.2**, **page 7:4**, is used to select an oil that gives the rated viscosity at the operating temperature.

A minimum viscosity ratio κ = 1 is required for general applications. However, a calculated viscosity ratio between 2 and 4 is recommended when possible. SKF recommends limiting κ to 4; otherwise the frictional heat decreases the operating viscosity.

Viscosity class according to ISO standards

Table 7.1

Viscosity class according to ISO	Kinematic viscosity mm^2/s at 40 °C min	mean	max
ISO VG 68	61,2	68	74,8
ISO VG 100	90,0	100	110
ISO VG 150	135	150	165
ISO VG 220	198	220	242
ISO VG 320	288	320	352
ISO VG 460	414	460	506
ISO VG 680	612	680	748
ISO VG 1000	900	1 000	1 100
ISO VG 1500	1 350	1 500	1 650

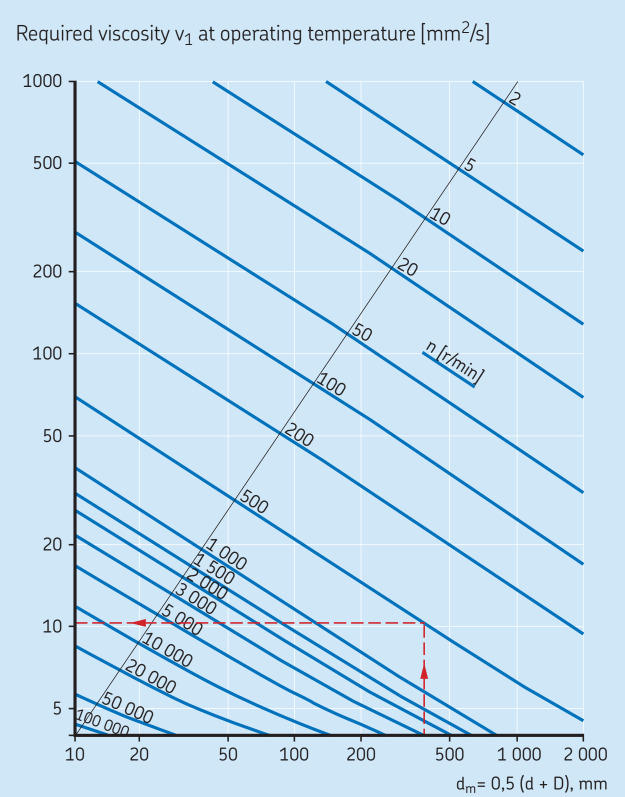
Example: A bearing having bore diameter d = 340 mm and an outside diameter D = 420 mm is required to operate at a speed n = 500 rpm. Since d_m = 0,5 (d+D), d_m = 380 mm from **diagram 7.1**, the minimum rated viscosity ν_1 required to provide adequate lubrication at the operating temperature is approximately 11 mm^2/s. If the bearing runs at 70 °C and an oil with a viscosity of 32 mm^2/s at 40 °C is selected, the obtained κ value is 1 (→ **diagram 7.2**, **page 7:4**). This means that the oil, or the base oil of the grease, to be chosen should be either an ISO VG 68 or an ISO VG 100 to obtain a κ higher than 1.

Viscosity index (VI)

The viscosity of an oil changes with the temperature. The viscosity index is a way to describe the magnitude of the change for a specific oil. Synthetic oils normally have a

ν_1 diagram with an example

Diagram 7.1



much higher viscosity index than mineral oils, i.e. their temperature dependence is less.

Diagram 7.2 is used when the viscosity at the operating temperature is converted into the reference viscosity at 40 °C. Tests at SKF show that the oil film thickness at 100 °C, i.e. at the normal bearing temperature in the dryer section, is the same for both mineral and synthetic oils if both have the same viscosity at 40 °C. Therefore, the diagram can be used for both mineral and synthetic oils. It is based on the viscosity index VI = 95. The reason is that mineral oil has a viscosity/pressure curve that increases more rapidly than the synthetic oil generally used in the dryer section. At the same viscosity at a given temperature, the oil film between roller and raceway will be thicker with a mineral oil. As mineral oils tend to have a lower viscosity than synthetic oils at higher temperature, they can generate more or less the same oil film thickness. However, at temperatures above approximately 130 °C, the advantage of the high viscosity index of synthetic oils is significant and should, therefore, be taken into consideration.

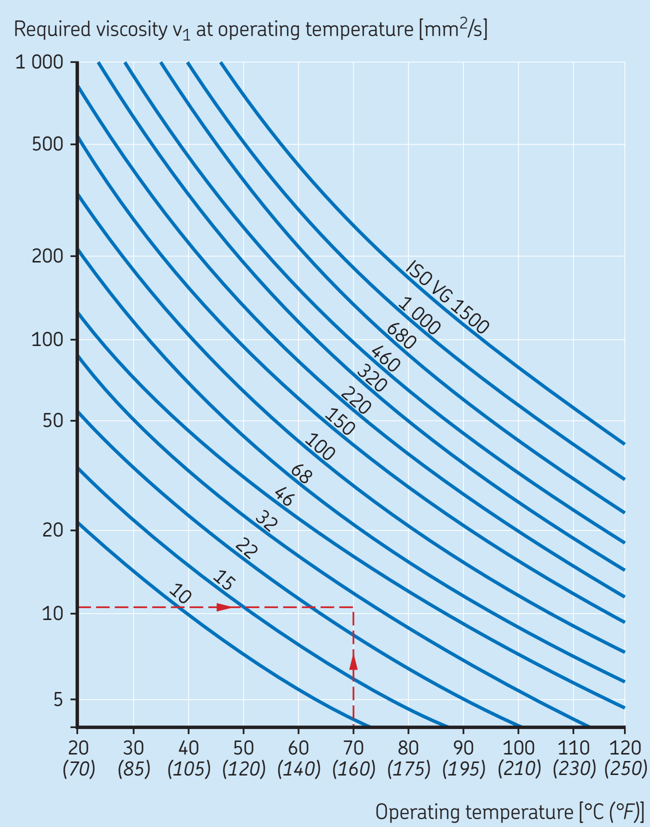
SKF life modification factor a_{SKF}

When the κ value is known, the a_{SKF} can be determined to calculate the SKF rating life of the bearing.

When dimensioning the bearing arrangements for paper machines, SKF recommends the use of the SKF rating life (L_{10mh}).

For further information, see chapter 1, *General requirements and recommendations*, *Selection of bearing size*.

Diagram 7.2



Viscosity/temperature for viscosity index of 95 diagram with an example

Different types of additives

Production of all lubricating oils begins with a straight base oil, i.e. oil without any additives. The difference between hydraulic, gear, engine and paper machine oils is the additive package. Gear oils, for example, may very well be used in paper machines if the additive package is suitable.

The function of additives is an area of very intensive development. New products are presented as replacements for products found to be toxic. For example, the use of products containing lead and chlorine is prohibited in most countries. Below are explanations of the functions of some common additives in paper machine oils.

Anti-corrosion

There are, in principle, two types of additives which offer protection. They are either water-soluble or oil-soluble chemicals, e.g. sodium nitride. Such additives normally give sufficient protection in damp operating conditions. However, they cannot give complete rust protection if the water content in the lubricant is too high. This is perhaps the biggest problem for the bearings in the forming and press section.

Anti-foam

The foam-damping action can be obtained, for example, by adding small quantities of silicone fluid. Such additives cause the bubbles to burst when they come in contact with the surface of the oil in the reservoir. This is important as air in the oil results in shorter lubricant life.

Anti-oxidant

Oil is exposed to high temperatures and air oxidizes, i.e. chemical compounds are formed. These compounds can increase the viscosity of the oil and also cause corrosion. Viscosity increase is normally used as one criterion for oil changes because these oxidation compounds have a negative influence on the lubrication effect. Anti-oxidants improve the oxidation stability of the oil by 10 to 150 times. The performance of the oil is maintained longer with obvious cost advantages. Furthermore, these additives have an anti-corrosive effect, but it is relatively limited.

EP (extreme pressure)

If the oil film in the rolling contact is not thick enough to fully separate the surfaces, there will be an interaction between the mating surfaces causing very high local temperatures. The temperature can be so high in these hot spots that asperities are welded together. This creates high friction and heavy adhesive wear like smeared surfaces in the bearing.

EP additives cause a chemical reaction in any hot spots so that the asperities shear off instead of being welded together. The result is a smoothing effect which reduces the size of surface irregularities. As the calculated κ value is based on the surface finish of the raceways before the bearing has been in operation, this smoothing effect will lead to an increase in κ value in operation when lubricants with good EP additives are used.

Many common modern EP additives are of the sulphur/phosphorus type. These additives tend to be aggressive to bearing steel at elevated temperatures and therefore reduce bearing life. As such, SKF generally recommends that lubricants with EP additives should not be used for bearings with operating temperatures above 80 °C. Of course, this can be discussed with your oil supplier.

In lightly loaded bearings where the rollers have a risk of extensive sliding during operation, EP additives normally do not work. For these applications, SKF recommends NoWear bearings with coated rollers (L5DA suffix).

AW (anti-wear)

At temperatures above 80 °C where EP additives should not be used, we recommend AW additives to reduce the risk of wear and smearing in the bearing. Additives of the AW type form a surface layer with certain beneficial properties such as a stronger adhesion to the surfaces. This surface layer causes the asperities of the mating surfaces to slide over each other instead of shearing off.

EP additives are sometimes called wear-prevention additives because they prevent adhesive wear. This is perhaps why some markets do not distinguish between AW and EP additives. In such cases, both are known as EP additives.

Detergent

These additives may be described as cleaning additives. They work in such a way that reaction products from high-temperature zones are kept floating in the oil. Without these additives, such reaction products may adhere to and discolour the surfaces in contact with the oil. These additives are normally used in engine oils for cars, but sometimes feature in paper machine oils as well.

Dispersant

One way to avoid sedimentation of contaminant particles inside the long pipes and large reservoirs of paper machine lubrication systems is to use an oil with dispersant additives. These additives can keep the particles floating in the oil until they enter the oil filter. One drawback of these additives is that they can keep small drops of water floating as well. This may cause corrosion in the bearings and clogging of the oil filters. Another drawback is that these additives can neutralize the effect of anti-wear additives.

VI-improvement

These viscosity index (and sometimes viscosity) increasing additives are often made up of large-molecule polymers. Experience with high-viscosity oils shows that these additives can be sheared to smaller molecules, i.e. the viscosity of the oil will decrease. If that happens, the thickness of the oil film in the bearings will also decrease. Therefore, additives of this type are not recommended for paper machine oils. These additives are very common in engine oils for cars, but paper machine oils must have a much longer service life without any change of basic properties like viscosity. These additives may influence the filterability of the oil.

Grease or oil lubrication?

Whether a rolling bearing is to be lubricated with grease or oil depends on a number of factors.

Grease has the advantage over oil of being easier to retain inside the bearing housing. It can also be retained in the sealing labyrinths where it protects the bearing against damp and impurities. A disadvantage when relubricating with large amounts of grease is that the used grease must come out through the labyrinths. This may cause contamination of the wire and the paper web.

Oil lubrication is used when the operating temperature with grease lubrication (due to high speed and/or heating) would be too high. A common maximum operating temperature for medium temperature greases in the forming and press section is 75 °C and for high-temperature greases in the dryer section is 120 °C.

However, SKF recommends circulating oil for all bearings in the dryer section to enable cooling and maintain high oil cleanliness. Unfortunately, circulating oil lubrication is not possible for all the bearing positions, e.g. rope sheaves, spreader rolls, doctors and steam joints. Grease is used for all these positions. In old machines, the felt rolls are grease lubricated too.

To select an appropriate lubricant, you need to know the operating temperature of the bearing. The temperature can either be calculated by SKF with the help of computer programs or by measuring in the machine.

7

Grease lubrication

Grease lubrication system

Traditionally, most of the forming and press section bearings, as well as rope sheaves and doctor bearings in the dryer section, have been grease lubricated. There are a number of reasons for this but, today, the main consideration is cost. SKF does not recommend grease lubrication in the dryer section. For the rope sheaves, doctor and spreader roll bearings, where it is very difficult to apply oil, grease lubrication is used as a compromise.

Previously, when production speeds were low, relubrication was carried out manually during maintenance stops i.e. once or twice per month. More frequent relubrication is necessary today due to increasing speeds, high-pressure cleaning of the machine and increased demand for machine reliability.

Therefore, the use of automatic lubrication is becoming increasingly popular. Automatic lubrication systems consist of a number of components (→ **fig. 7.3**) which can all be supplied by SKF.

The aim of both automatic and manual lubrication is to supply the right amount of fresh grease to the bearing arrangement.

Examples of disadvantages with grease lubrication are the difficulties in selecting grease quality, initial charge, relubrication quantity and method of grease supply into the bearing.

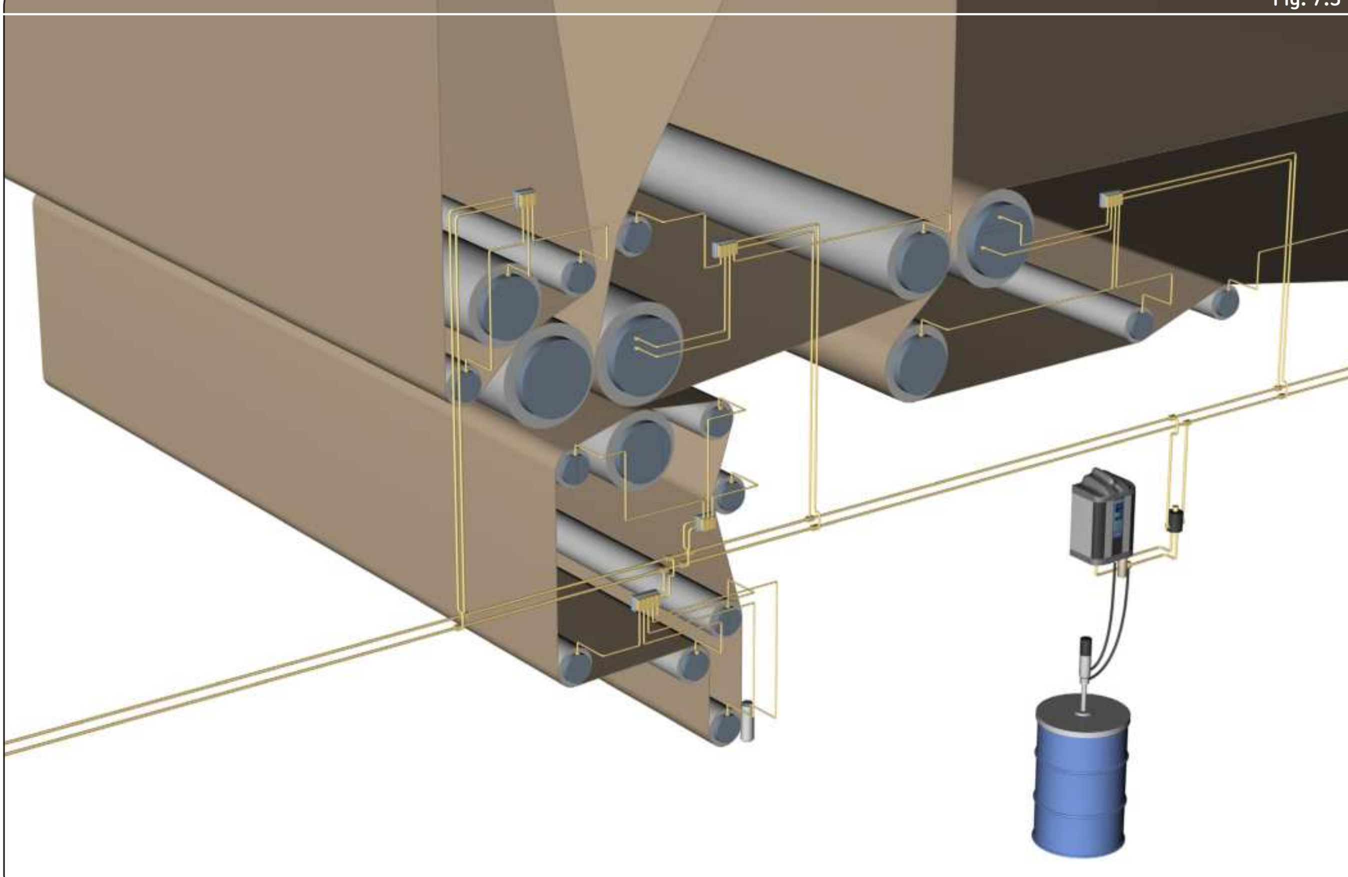
Selection of grease type

Lubricating grease is made up of a so-called base oil which is mixed with a thickener. The base oil is normally either a mineral or a synthetic oil.

The thickener in the majority of cases is a metal soap, e.g. calcium, lithium, sodium or non-soap, e.g. clay (bentonite), silica gel, polyurea. The consistency of the grease depends on the type and quantity of the thickener used. The consistency is measured as the penetration depth of a standardized cone into the grease during a certain time. **Table 7.2** shows the three consistency classes according to NLGI (National Lubricating Grease Institute) which are used for bearing applications.

Automatic grease lubrication

Fig. 7.3



Grease consistency classes

Table 7.2

NLGI index	Penetration mm/10
1	310–340 (very soft)
2	265–295 (soft)
3	222–250 (medium hard)

When selecting greases for paper machines, you need to take all operating conditions into consideration. Unfortunately, there has been a trend, mainly in response to end-user requirements, to minimize the number of different greases. That may lead to an unsatisfactory compromise. For example, it may be hard to find a grease that has all the properties required for both the forming and press section as well as the dryer section.

SKF lubricating grease range

With the improvements in bearing lubrication technology, higher operating speeds and temperatures, the demands made on lubricating greases have become more severe. SKF, in conjunction with major grease manufacturers, has developed a range of lubricating greases to satisfy these demands.

Depending on operating conditions, seal efficiency, bearing type and how the grease is fed into the bearing, SKF LGMT 2, SKF LGHB 2, SKF LGHP 2, SKF LGWM 2, SKF LGWA 2 or SKF LGLT 2 grease may be suitable. There is no grease existing on the market that is suitable for all bearing positions. As an example of this, SKF might recommend the following greases for a specific paper machine:

- SKF LGWA 2 for the roller bearings of the forming, press and reeling sections, and spreader rolls in the dry section.
- SKF LGHB 2 for the doctor bearings and roller bearings in the dryer section.
- SKF LGHP 2 for the ball bearings in the rope sheaves.
- SKF LGLT 2 for the spreader rolls except in the dryer section.

Testing the performance of fresh grease

The operating conditions of the bearings dictate the priorities given to the individual performance properties of lubricating greases. As a general guideline, this chapter gives required characteristics for a grease to meet for a paper machine bearing application. It is a guideline only and the choice of grease should be made based on the exact operating conditions, if possible. The operating conditions for papermaking machine bearings can be said to fall into three basic groups represented in the test program by positions **a**, **b** and **c** (→ **table 7.3**, **page 7:10**).

For more information, please contact SKF. SKF can undertake all of the tests to determine whether a lubricant is suitable for known bearing operating conditions. As lubricant manufacturers might change the formula of their product in the future, an infrared analysis is done on the sample tested so that it is possible to tell if the lubricant used to lubricate the bearings is still the same as the one tested.

The bearing should be completely filled with fresh clean grease during the initial mounting. Furthermore, 30 to 50% of the free space in the housing should be filled with grease.

In addition, to ensure sealing efficiency, the labyrinth seals should be completely filled with grease from the beginning.

Table 7:3

Testing the performance of fresh grease

Position **a** represents all the grease lubricated ball and roller bearing positions in the forming and press sections.
Position **b** represents rope pulleys, doctors and felt roll grease lubricated ball and roller bearing positions in the dryer section.
Be aware that some bearing positions in the lowest part of the dryer section might fail due to insufficient oil bleeding if they are lubricated with a high temperature grease.
Position **c** represents spreader rolls in the dryer section with ball bearings and an operating temperature of 100 to 140 °C.

1.1 Kinematic viscosity of the original oil including additives
Type of test: DIN 51562/1, ISO 3448 or ASTM D 2422
Required result for **a**: Min 175 mm^2/s at 40 °C
b: Min 400 mm^2/s at 40 °C
c: Min 220 mm^2/s at 40 °C

1.2 Consistency
Type of test: DIN ISO 2137 or ASTM D 217
Required result for **a**: NLGI 2
b: NLGI 2
c: normally NLGI 2 but sometimes NLGI 3

1.3 Mechanical stability 100 000 strokes
Type of test: DIN ISO 2137 or ASTM D 217
Required result for **a**: Worked penetration value should not change more than +/–40
b: Worked penetration value should not change more than +/–30
c: Worked penetration value should not change more than +/–30

1.4 Mechanical stability (shear stability in a roll stability test)
Type of test: DIN 51804/2 or ASTM D1831, 50 g grease, 50 h at 80 °C
Required result for **a**: Worked penetration value should not change more than +/–40
b: Worked penetration value should not change more than +/–30
c: Worked penetration value should not change more than +/–30

1.5 Mechanical stability (vibrations)
Type of test: SKF V2F
Required result for **a**: M
b: M
c: M

1.6 Water resistance
Type of test: DIN 51807/1
Required result for **a**: 1 max at 90 °C
b: 1 max at 90 °C
c: –

1.7 Water washout
Type of test: DIN 51807/2
Required result for **a**: Washout less than 10%
b: Washout less than 10%
c: Washout less than 20%

1.8 Rust inhibition with artificial process water or with customer's process water
Type of test: SKF EMCOR, IP 220 or DIN 51802 or ISO 11007
Required result for **a**: 0–0
b: 1–1
c: 1–1

1.9 Rust inhibition with 0,5% NaCl
Type of test: SKF EMCOR, IP 220 or DIN 51802 or ISO 11007
Required result for **a**: 0–0
b: 1–1
c: 1–1

1.10 Rust inhibition with 3% NaCl
Type of test: SKF EMCOR, IP 220 or DIN 51802 or ISO 11007
Required result for **a**: 0–0
b: –
c: –

1.11 Copper compatibility, 24 h at 90 °C
Type of test: DIN 51811, IP 112 or ASTM D4048
Required result for **a**: 1
b: –
c: –

1.12 Copper compatibility, 24 h at 120 °C
Type of test: DIN 51811, IP 112 or ASTM D4048
Required result for **a**: –
b: 1
c: 1

1.13 4 ball weld load
Type of test: DIN 51350/4
Required result for **a**: equal or higher than 2 800 N
b: equal or higher than 2 800 N but grease must pass the EP reaction test
c: equal or higher than 2 800 N but grease must pass the EP reaction test

1.14 4 ball wear test
Type of test: DIN 51350/3 (1400 N) or ASTM D2266
Required result for **a**: 0,8 mm max
b: 0,8 mm max
c: 1,0 mm max

1.15 Dropping point
Type of test: DIN ISO 2176 or ASTM 2265
Required result for **a**: 180 °C min
b: 250 °C min
c: 250 °C min

1.16 Oil bleeding
Type of test: DIN 51817 or IP121
Required result for **a**: min 1% at 40 °C and max 10% at 80 °C
b: min 3% at 80 °C and max 12% at 120 °C
c: min 3% at 100 °C and max 15% at 140 °C

Note that if the real operating conditions are known, oil bleeding for roller bearings should be at least 3% at operating temperature. For **b**, operating temperature can be very different depending on the bearing position. The bearings in the lowest position might be better lubricated with a grease for **a**.

1.17 Spherical roller bearing lubricating ability
Type of test: SKF R2F
Required result for **a**: rating 1 with procedure A (unheated)
b: rating 1 with procedure B (at 120 °C)
c: –

1.18 Pumpabilty (feedability)
Type of test: SKF test

1.19 Cleanliness
Type of test: DIN 51831
Required result for **a**: max. 5 mg/kg
b: max. 5 mg/kg
c: max. 5 mg/kg

1.20 EP additives reaction test
Type of test: SKF test 24 h at 160 °C (to be published)
Required result for **a**: –
b: rating 0
c: rating 0

1.21 Grease service life
Type of test: SKF ROF at 140 °C
Required result for **a**: –
b: –
c: min. 1 000 h

1.22 Evaporation loss
Type of test: SKF test.
Required result for **a**: –
b: To be defined
c: To be defined

The above requirements are general guidelines. Real operating conditions and seal efficiency might make a grease suitable even if it does not meet all of these requirements.

Relubrication

There are some important issues to consider when relubricating. One basic rule is to use the same grease quality for relubrication as was used when the bearing was mounted.

Another important point to consider is the method of relubrication. In most cases, there is a need for regular relubrication. Therefore, the housing must be supplied with a lubricant duct and grease nipple. For the fresh grease to penetrate effectively into the interior of the bearing, and for the old grease to be pushed away, the duct should open so that the grease is supplied immediately adjacent to the outer ring face or directly into the outer ring lubrication groove (the W33 feature in the case of spherical roller bearings). Evacuation holes should be considered for the excess or old grease that cannot be pushed out.

The bearing housings in most paper machine applications with grease lubrication are provided only with labyrinth seals. These labyrinths have to be filled with grease to obtain an efficient sealing function. Therefore, the recommended relubrication intervals and quantities are selected to secure both good lubrication and efficient sealing function.

The grease in the labyrinths can easily be washed out if the housings are hosed down under high pressure. Therefore, never direct high-pressure sprays towards the sealing gap. The housing sealing should be protected e.g. with splash covers.

Sometimes extra relubrication is carried out during a standstill in order to push the water-contaminated grease out from the labyrinth. This extra relubrication should instead be carried out under operating conditions just before standstill. More details are given in chapter 9, *Maintenance, Standstill precautions*.

Relubrication intervals and quantities

As the bearing arrangements in the forming and press section of the paper machines operate in a wet environment, there is a need for shorter relubrication intervals than those indicated in the SKF catalogue *Rolling bearings*.

There are basically three factors to consider when selecting relubrication intervals and quantities. These are the lubrication and sealing functions and the grease leakage from the housing. From a lubrication point of view, the amount of grease in the bearing should be just sufficient to supply an adequate quantity of base oil to the raceways, rollers and cages.

From a sealing point of view, it is important that the labyrinth seals are always completely filled with grease.

From a leakage point of view, the grease quantity should not be too large. Large quantities of grease generate heat in the bearing, especially at high speeds, when the bearing attempts to pump out most of the grease. When relubrication has been carried out a number of times, all the free space inside the housing can be filled with grease. From then on, the grease leakage will be equal to the relubrication quantity. Therefore, with large regreasing quantities, there is a risk that the leakage is so great that some grease may come out on the wire.

As the environment and the bearing housing design vary widely from machine to machine, the relubrication quantities should always be adjusted according to practical experience. Thus, the recommended values for relubrication intervals and quantities listed in **tables 7.4** and **7.5** should be used as guidelines only.

Use the relubrication intervals indicated in the SKF catalogue *Rolling bearings* if they give higher quantities of grease per unit of time than in **tables 7.4** or **7.5**,

Manual lubrication

Table 7.4

Forming and press section
Relubrication once a week is recommended for bearing in wet positions e.g. in the wire and press parts.

Once a month is recommended for bearings in damp positions, e.g. calenders, reelers, winders.

Quantities according to

$$G = K.D.B$$

where
G = quantity of grease, g
D = bearing outside diameter, mm
B = bearing width, mm
K = 0,002 (weekly relubrication)
0,003 (monthly relubrication)

Dryer section
Manual grease relubrication is not recommended in the dryer section because of the difficulty in reaching all the positions during operation.

Automatic lubrication

Table 7.5

Forming and press section
Automatic grease lubrication systems are becoming increasingly common. The advantage with these systems is better protection of the bearings because of shorter relubrication intervals.

Quantities according to the equation below are valid for the bearings in the forming and press sections.

$$G = 0{,}00001.H.D.B$$

where
G = grease quantity after H hours, g
D = bearing outside diameter, mm
B = bearing width, mm
H = relubricating interval, hours

Dryer section
SKF normally recommends oil lubrication for the bearings in the dryer section. However, there are some positions where circulating oil is difficult to apply. SKF recommends the same quantities as in the forming and press section.

Oil lubrication

Oil lubrication systems

Most of the bearings in old machines were grease-lubricated. Originally, only the bearings for drying cylinders had oil lubrication. The first oil lubrication systems were of the oil bath type.

Increased operating speeds and steam temperatures soon made improved lubrication necessary. The higher the operating temperature of the bearing, the more rapidly the lubricating oil would degrade.

The next step in improved lubrication was the introduction of oil drip lubrication. With this method, it was possible to extend the life of the oil. However, the continuous increase in paper speeds and steam temperatures and the introduction of hooded dryer sections very soon resulted in bearing operating conditions that required even better lubrication. This was obtained by introducing circulating oil lubrication.

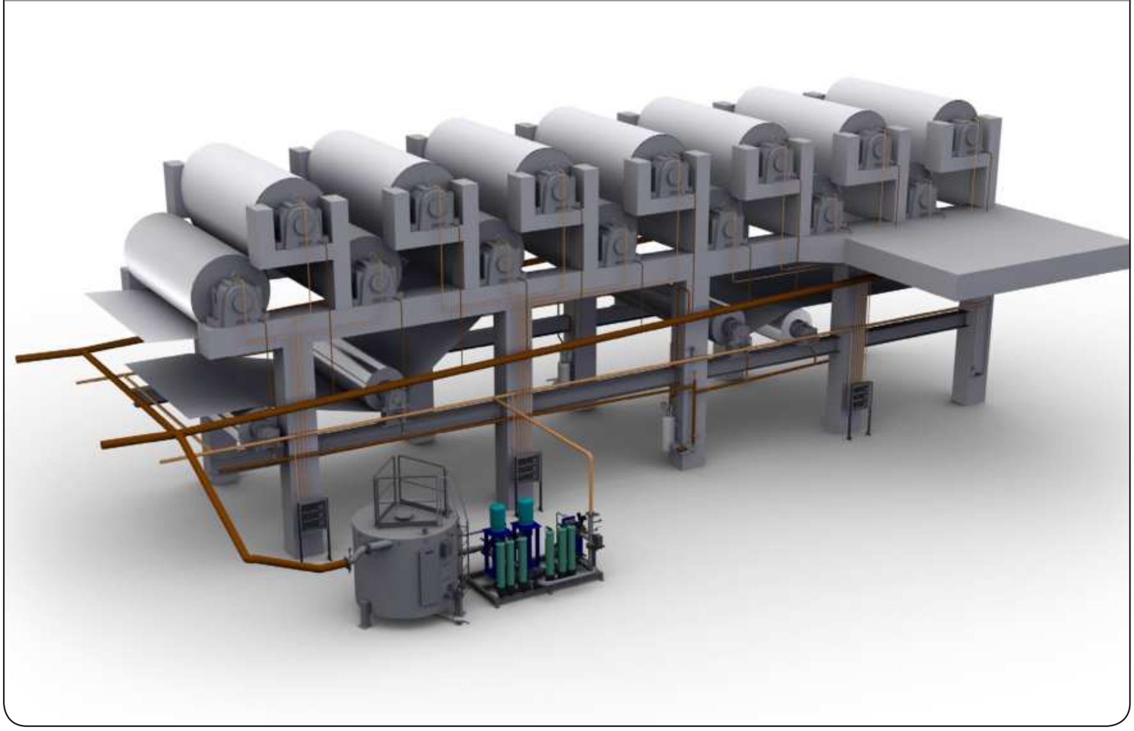
Modern circulation systems make it possible to cool and maintain high oil cleanliness (→ **fig. 7.4**). These are the main reasons why circulating oil lubrication is used for most bearings in new machines.

Over the years, the speed increase has been so great that other parts of the machine also have to be lubricated with circulating oil. Note that the speed rating in the SKF catalogue *Rolling bearings* is not the maximum permissible speed for the bearing. With oil circulation and certain additional dimensional and geometrical requirements on shafts and bearings it is possible to operate at higher speeds. Some bearings in modern high-speed machines operate above limiting speed.

Modern paper machines are often very large and the bearing operating conditions differ in the various parts of the machine. Therefore, the bearing lubrication system is split up into several smaller systems. For example, the forming and dryer sections usually have their own lubrication systems as do the Yankee cylinders and deflection compensating rolls (→ **fig. 7.5**, **page 7:14**).

Some tissue machines have the same circulating oil for the Yankee bearings, the suction and press rolls and the gearbox. This is not recommended as each application requires a suitable oil and because water ingress from the press rolls part, heat from the Yankee and the EP additives for gears will reduce bearing service life drastically.

Fig. 7.4



SKF Flowline circulating oil lubricating system for a paper machine

Selection of oil

When selecting oils for paper machines you need to take into consideration a great number of factors including:

- Lubrication method (oil bath/oil circulation).
- Operating conditions for all the bearings connected to the same lubrication system.
- Design and performance of the seals.
- Pipe dimensions.

The most common operating conditions usually result in two or three different oils being used per machine. This represents a good compromise between the different lubrication requirements and what is practical.

Unfortunately, there has been a trend, mainly in response to end-user requirements, to minimize the number of different oils. That may lead to an unsatisfactory compromise. For example, it may be hard to find an oil that has all the properties required for both the forming and dryer sections.

Mineral oils

In most cases, good-quality mineral oils are suitable as lubricants for paper machine bearings at operating temperatures up to 100 °C. However, some mineral oils developed for paper machines perform well up to 120 °C.

Synthetic oils

There are a number of different types of synthetic base oils such as synthetic hydrocarbons, esters and polyglycols.

These products have different properties with regards to their influence on rubber seals, evaporation and miscibility with polar and non-polar additives etc. Another difference is that the density of polyglycols is very close to that of water at 50 °C. This is a disadvantage since the water will not separate in the reservoirs which normally have a temperature of about 50 °C.

Poly-alpha-olefin (PAO) oils (often mixed with small amounts of other synthetic base oils) have been the most popular synthetic oil type in paper machines. This is due to the fact that these oils do not have the previously mentioned drawbacks.

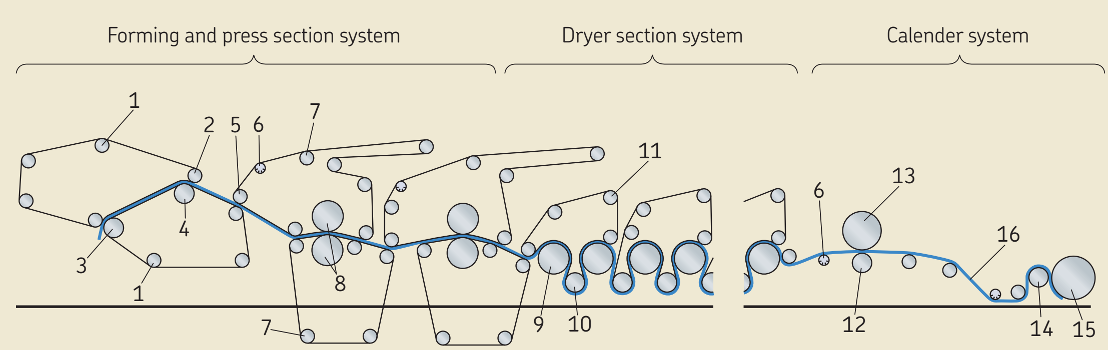
Synthetic oils have a number of advantages compared with mineral oils. The most valuable property of synthetic oils is their better performance at high operating temperatures.

Oils for the forming section

The selection of oil for the forming section should be based on operating conditions for press and suction roll bearings because these are the most demanding applications.

When large bearings with heavy rollers rotate at high speeds, the rollers may slide as they enter the loaded zone and the internal functional surfaces of such bearings may be damaged severely by smearing. That is why these bearings need a thick oil film containing EP additives. If these additives are of the sulphur-phosphorus type, the bearing operating temperature should not exceed 80 °C as such additives act aggressively at higher temperatures. Therefore, the oil flows should be large enough to keep the bearing

Fig. 7:5



Basic layout of a paper machine including a common division in different oil lubrication systems.

1 *Wire roll*
2 *Forward drive roll*
3 *Forming roll (suction roll)*
4 *Suction couch roll*
5 *Pick-up roll*
6 *Spreader roll*
7 *Felt roll*
8 *Shoe press*
9 *Drying cylinder*
10 *Vacuum roll*
11 *Guide roll (felt roll)*
12 *Deflection compensating press roll (soft calender)*
13 *Thermo roll (soft calender)*
14 *Reel drum*
15 *Reel spool*
16 *Paper web*

temperature below 80 °C. Oils with EP additives that are stable at higher temperatures or with AW additives should be used at operating temperatures above 80 °C.

Effective rust-inhibition and rapid water separation are the most important properties of oils for the forming section, but EP additives are also a top priority for high speed press and suction rolls.

Oil suitable for the forming and press section should have the following basic properties:

- viscosity class ISO VG 100 to 320, depending on the machine. Most common is the ISO VG 150. Note that many modern machines use two different oils. For example, ISO VG 150 be used for the suction roll while the oil used for the shoe press with press roll could, depending on the machine speed, be lubricated with an ISO VG 100 to ISO VG 320 oil.
- EP additives up to 80 °C
- anti-wear additives above 80 °C
- rust-inhibiting additives
- fast air release

Oils for the dryer section and machine calenders

The selection of oil for the dryer section should be based on operating conditions for drying and Yankee cylinder bearings because these are the most demanding applications.

EP or AW additives are recommended for oils in the dryer section because the high bearing operating temperatures, in combination with the viscosity of commonly used oils, leads to metal-to-metal contact between rollers and raceways. If the EP additives are of the sulphur-phosphorus type, the bearing operating temperature should not exceed 80 °C as these additives act aggressively at higher temperatures. Oils with EP additives that are stable at higher temperatures or with AW additives should be used at operating temperatures above 80 °C.

Good thermal and chemical stability are the most important properties of oils for the dryer section and machine calenders. Effective rust inhibition is desirable as well. Oil suitable for the dryer section should have the following basic properties:

- viscosity classes ISO VG 220 to 320 for drying cylinders etc. ISO VG 320 to 460 for Yankee cylinders. ISO VG 460 to 1500 for oil baths
- EP additives up to 80 °C
- AW additives above 80 °C
- rust-inhibiting additives

Oils for off-line calenders

Off-line calenders often consist of ordinary press rolls. Therefore, the requirements for calender oils are the same as for forming section oils if the bearing operating temperature is below 80 °C. However, in some off-line calenders bearings operate at the same temperatures as dryer section bearings. In these cases, the oils should fulfil the requirements for dryer section oils.

Calender development is very rapid. Heating oil in so-called hot calenders may have temperatures up to 350 °C. Special oils and lubrication systems are required if the temperatures of the press roll bearings in these hot calenders are much higher than those of the bearings in the dryer section.

Testing the performance of fresh oil

The operating conditions of the bearings dictate the priorities given to the individual performance properties of lubricating oils. The operating conditions for paper machine bearings can be said to fall into two basic groups represented in the test program by positions **a** and **b** (**→ table 7.6**)

For more information, please contact SKF. SKF can undertake all of the tests to determine whether a lubricant is adequate for known bearing operating conditions or not. As lubricant manufacturers sometimes change the formula of their product, an infrared analysis is done so that it is possible to tell if the lubricant used to lubricate the bearings is the same as the one that was originally tested.

Table 7.6

Testing the performance of fresh oil

Position **a** represents all oil-lubricated bearing positions in the forming, press and reeling section.
Position **b** represents all oil-lubricated bearings in the dryer section including machine calenders.
(N.B. : The use of so-called hot calenders is increasing, but the lubrication requirements for the bearings in these applications are not considered in this test programme).
The limits listed for the various properties are of a general nature and therefore refer to the most exacting requirements. Naturally, some of these restrictions can be eased if the operating conditions permit.

1.23 Kinematic viscosity
Type of test: DIN 51562/1 or ASTM D 445

Required result for **a**: to meet viscosity recommendation for the application
b: to meet viscosity recommendation for the application

1.24 Rust inhibition with artificial process water or with customer's process water
Type of test: SKF EMCOR, IP 220 or DIN 51802
Required result for **a**: 0–0
b: 1–1 (0–0 if oil bath)

1.25 Rust inhibition with distilled water
Type of test: SKF EMCOR, IP 220 or DIN 51802
Required result for **a**: 0–0
b: 0–0

1.26 Copper compatibility, 48 h at 80 °C
Type of test: DIN EN ISO 2160
Required result for **a**: 2
b: –

1.27 Copper compatibility, 48 h at 120 °C
Type of test: DIN EN ISO 2160
Required result for **a**: –
b: 2

1.28 4 ball weld load
Type of test: DIN 51350/4
Required result for **a**: equal or higher than 2 800 N
b: max. 2 000 N if the EP activity has been obtained by phosphorous and sulphur compounds. Other EP-additives with higher welding loads may be used if the oil passes the other required tests.

1.29 4 ball wear test
Type of test: DIN 51350/3 under 600 N
Required result for **a**: 1 mm max.
b: 1 mm max.

1.30 Water separation ability
Type of test: ISO DIN 6614
Required result for **a**: max. 20 minutes
b: max. 20 minutes

1.31 Filterability
Type of test: SKF method: at 800 mbar absolute pressure, 12 µm laboratory filter
Required result for **a**: max 15 minutes
b: max 15 minutes

1.32 Chemical reaction on bearing steel
Type of test: SKF roller test, part 1
Required result for **a**: –
b: max. 2 (mineral oils at 120 °C, synthetic oils at 120 °C and 140 °C)

1.33 Change of viscosity
Type of test: SKF roller test, part 2
Required result for **a**: –
b: +/– 20% kinematic variation from fresh oil after ageing (mineral oils at 120 °C, synthetic oils at 120 °C and 140 °C)

1.34 Sludge formation
Type of test: SKF roller test, part 3
Required result for **a**: –
b: traces, visual (mineral oils at 120 °C, synthetic oils at 120 °C and 140 °C)

1.35 Incrustation
Type of test: SKF roller test, part 4
Required result for **a**: –
b: No incrustations (mineral oils at 120 °C, synthetic oils at 120 °C and 140 °C)

1.36 Dye number
Type of test: DIN ISO 2049
Required result for **a**: –
b: max. 6,5 (mineral oils at 120 °C, synthetic oils at 120 °C and 140 °C)

1.37 Oil film ageing (first test)
Type of test: SKF film stability test
Required result for **a**: –
b: rating max. 2 (mineral oils at 120 °C, synthetic oils at 120 °C and 140 °C optional)

1.38 Oil film ageing (second test)
Type of test: SKF weight loss (evaporation test)
Required result for **a**: –
Required result for **b**: max. 20% (mineral oils at 120 °C, synthetic oils at 120 °C and 140 °C optional)

1.39 Compatibility with elastomers
Type of test: SKF method, 200 h: FPM seal at 150 °C, NBR seal at 120 °C
Required result for **a** and **b**: NBR seal: max 10% change in weight,
FPM seal: max 5% change in weight.
Less than 5 shore A hardness change for NBR and FPM.

1.40 Fast air release
Type of test : ISO 9120
Required results for **a** and **b** at 75 °C: ISO VG 100 : ≤ 10 min
ISO VG 150 : ≤ 15 min
ISO VG 220 : ≤ 20 min
ISO VG 320 : ≤ 25 min

The above requirements are general guidelines. Real operating conditions and seal efficiency might make an oil suitable even if it does not meet all of these requirements.

Cleanliness control

Lubricating oil should be continuously cleaned of impurities. It is important to remove both water and solid particles from the oil.

Cleanliness recommendation

When selecting suitable water extractors and filters, the following cleanliness guidelines should be aimed for:

- Water content should be below 200 ppm. Note that during operation with the bearing rotating and oil hot enough to dissolve all free water; 500 ppm max can be tolerated.
- Particle content should be according to ISO 4406 cleanliness class –/15/12 (using a microscope) or 18/15/12 (using an automatic particle counter) or SAE class 6B/6C, or better.

Water extractors

As mentioned earlier, SKF has found water to be one of the major reasons for the short service life of bearings. The recommendation of a water content below 200 ppm provides a good balance between the cost of water removal and increased bearing service life.

The recommended water content level can be obtained by using ordinary extractors. The most common extractors work according to two basic principles, using vacuum or centrifugal forces. The advantage of the centrifugal extractors is that they normally remove more water per minute than the vacuum extractors when there is a lot of free water. On the other hand, vacuum extractors remove dissolved water and take air out as well.

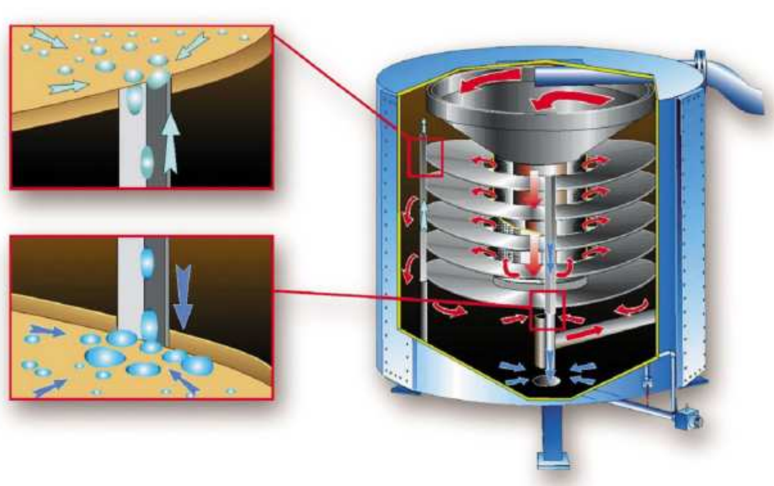
The final result of water removal depends very much on the amount of water entering the system. Therefore, an important consideration when selecting equipment is the risk of water entering the lubrication system. The most common reasons for the entry of water are inefficient housing sealing and high-pressure cleaning, but accidental leakage from oil coolers etc. has to be considered as well.

Bearings should never be exposed to oil that has a higher water content than recommended above. This is especially important during standstill. If they are, there is a higher risk that free water in the oil can start the corrosion process. Therefore, it is very important to keep water content low just before machine stoppages and to prevent entry of water during standstill.

Another cause of high water content in modern paper machine oils are so-called dispersant additives. The main task of these additives is to keep contaminant particles floating in the oil until they enter the oil filters. Unfortunately, these additives sometimes have the same effect on water molecules. This is one of the reasons for the clogging of oil filters. In such cases, the continuous use of a water extractor is required.

SKF recommends the use of the SKF Flowline tank whose design gives superior continuous water and air removal (→ **fig. 7.6**). In addition, the oil retention time is much higher than in traditional rectangular tanks, so it is possible to decrease the total amount of oil by between 50 to 65%.

Fig. 7.6



SKF Flowline tank
Water collects on the upper face of the plates while air is trapped underneath them. Air and free water are continuously extracted from the oil.

All oil lubrication systems should have equipment for continuous water removal. Without such equipment many paper machine oils may have water content higher than the recommended 200 ppm.

Oil filters

Different types of oil filters have been used for many years in the lubrication systems of paper machines. The first replaceable filters were so-called mesh elements made of woven steel wire. These filters were efficient when it came to very large particles, but research at SKF has proven that even particles smaller than 10 µm should be removed from the oil because they may have a detrimental effect on bearing functional surfaces.

Normally, there is a connection between fine filters and clean oil. However, the most important thing is to have clean lubricating oil. Therefore, SKF's recommendation is based on oil cleanliness instead of filter ratings. This recommendation is based on the optimized filter cost as well as on the bearing service life obtained.

When selecting filters, the filterability of the current oil should also be considered.

Filter ratings

Filters have developed rapidly in recent years. This means that today's standard filters are many times more efficient than the filters commonly used some years ago.

The filter rating should give an indication of the filter efficiency. Unfortunately, in the case of so-called nominal filters, there is no definition of efficiency.

The efficiency of standard filters is defined as a reduction factor β which is related to one particle size. The higher the β value, the more efficient the filter is for the specified particle size. Therefore, both the β value and the specified particle size have to be considered.

The reduction factor β is expressed as the relationship between the number of specified particles before and after the filter. This can be calculated as follows:

$$\beta_{x(c)} = \frac{n_1}{n_2}$$

where

n_1 = number of particles per volume unit (100 ml) larger than x µm upstream the filter

n_2 = number of particles per volume unit (100 ml) larger than x µm downstream the filter

Note: The β value is connected to only one particle size in µm, which is shown as the index e.g. $\beta_{3(c)}$, $\beta_{6(c)}$, $\beta_{12(c)}$, etc. For example, a complete rating $\beta_{6(c)}$ = 75 means that only 1 of 75 particles of 6 µm or larger will pass the filter.

$\beta_{40(c)}$ = 75 gives rough filtration for roller bearing lubrication oils

$\beta_{12(c)}$ = 75 is the minimum recommended

$\beta_{6(c)}$ = 200 was considered as very efficient in the late 1980s and the beginning of the 1990s. Today, it is considered as good, but not best practice.

$\beta_{3(c)}$ = 1 000 is used on modern paper machines

Particle counting

Particle counting can be based on various principles.

The simplest is to filter the oil on to a membrane and look at it through an optical microscope. The size of a particle is established by measuring the longest dimension. The particle size distribution can be estimated by comparing the sample membrane with reference membranes of different cleanliness levels. A skilled operator should do this because it is a subjective method.

Automatic Particle Counting (APC analysis) is a more commonly used method. Particles are passed in front of a light source coupled to a sensor. The amount of light passing through a window depends on the size of the particle passing that window. The particle size is derived from its cross-sectional area. Calibration of the equipment is most important. A drawback of using a sensor is the necessity of removing air bubbles, which would otherwise be counted as particles.

Yet another method for particle analysis is Spectrometric Oil Analysis Program (SOAP). This method is very good for the analysis of small particles e.g. from abrasive wear. Wear particles are in most cases smaller than 5 to 10 µm.

There are a number of additional methods and equipment on the market today and new products are launched all the time in this growing field of condition monitoring.

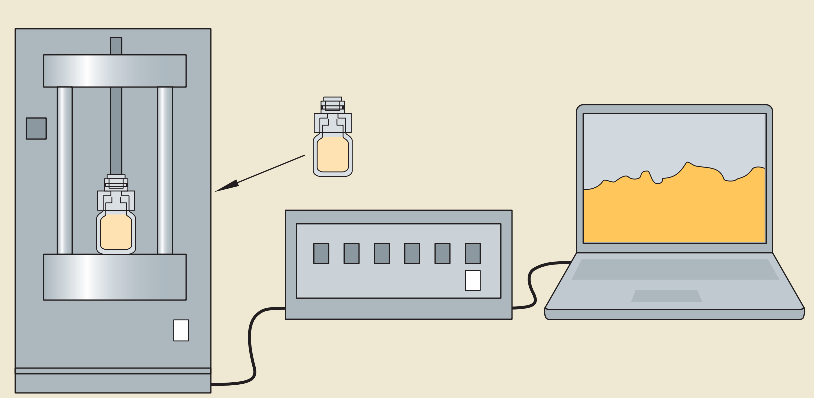
Whatever the method used, the results should be presented as a development trend. **Fig. 7.7** shows how a computer is connected to APC equipment to collect and evaluate the results over time.

Errors are to be expected if the results of an analysis show that there are very big differences between the different particle size ranges. If there are too many large particles, the filter may be damaged or bypassed. Another reason could be that the oil has not been correctly sampled. It may include sediment particles. The presence of too many particles results in a step in the distribution curve. This step shows the performance limit of the filter.

Contamination levels

Standards establish information which allows comparison and interpretation of contamination (cleanliness) levels to enable the control of particle contamination to ensure system performance and reliability.

Fig. 7.7



Equipment for particle counting

ISO classification method

The method for coding the contamination level in a lubrication system is according to ISO 4406:1999.

In order to simplify the reporting of particle count data, the quantities counted are converted to a code using scale numbers. These are allocated according to the number of particles counted per millilitre of the fluid sample (→ **table 7.7**).

The code for contamination levels using automatic particle counters (APC) comprises three scale numbers relating to the number of particles ≥ 4 µm (c), ≥ 6 µm (c) and ≥ 14 µm (c), where (c) refers to APC. The three numbers are written one after the other separated by (slashes). Example: 18/15/12 (→ **diagram 7.4**). APC calibration is according to ISO 11171.

For comparative results, the code for microscope counting comprises two scale numbers relating to the number of particles ≥ 5 µm and ≥ 15 µm. Counting is undertaken in accordance with ISO 4407. The code is stated in three part form where the first part is given as a "–". Example: –/15/12 (→ **diagram 7.4**).

There is a direct link between contamination levels and filter ratings. Any deviation should be considered as an alarm indicating that filtration efficiency has decreased (due to damaged filters or filter bypasses) or that there is contamination entering the system (from bearing damage, open oil reservoirs etc.)

Example:
$\beta_{12(c)} = 200 \rightarrow$ ISO 4406 –/15/12
$\beta_{6(c)} = 200 \rightarrow$ ISO 4406 –/13/10

ISO classification

Table 7.7

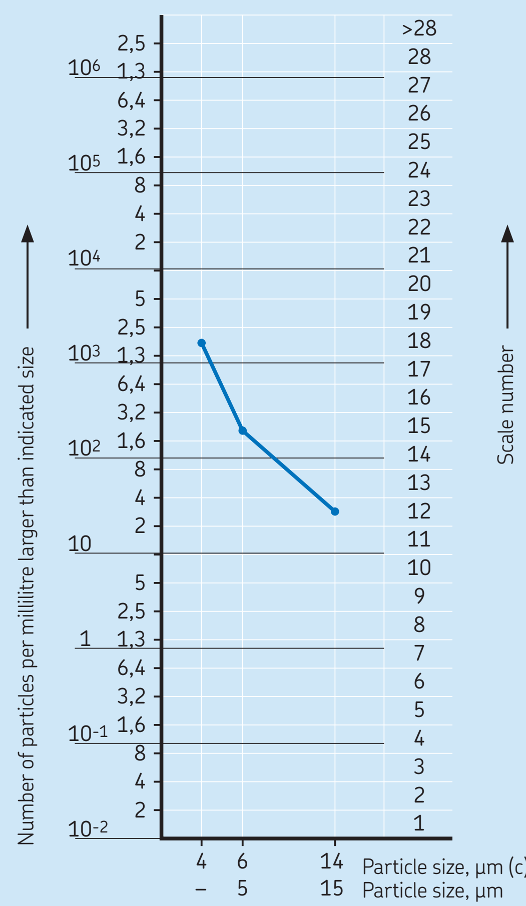
ISO classification – allocation of scale number

Number of particles per ml oil Over	Up to and including	Scale number
2 500 000		> 28
1 300 000	2 500 000	28
640 000	1 300 000	27
320 000	640 000	26
160 000	320 000	25
80 000	160 000	24
40 000	80 000	23
20 000	40 000	22
10 000	20 000	21
5 000	10 000	20
2 500	5 000	19
1 300	2 500	18
640	1 300	17
320	640	16
160	320	15
80	160	14
40	80	13
20	40	12
10	20	11
5	10	10
2,5	5	9
1,3	2,5	8
0,64	1,3	7
0,32	0,64	6
0,16	0,32	5
0,08	0,16	4
0,04	0,08	3
0,02	0,04	2
0,01	0,02	1
0,00	0,01	0

ISO 4406 method

Diagram 7.4

ISO classification and example for both microscope (–/15/12) and automatic particle counter (18/15/12)



Flushing new circulating oil systems

If there are hard contaminant particles in the system, the bearings can be damaged during the first minutes of operation. The best way to avoid this is to flush the complete lubrication system before the very first start-up. Flushing should be continued until the flushing oil reaches the recommended cleanliness level. This means that there should be a number of cleanliness tests performed during the flushing period.

The main difficulty during flushing is creating turbulence in the pipes in order to flush out all the contaminant particles attached to the walls of the pipes. SKF recommends the use of the same oil that will be used during operation. To reduce the viscosity, the oil should be heated. The oil pump should be set at its maximum flow capacity. By doing so, the system is subjected to a higher flushing effect than during subsequent normal operation. In this way, the risk of contaminants being dislodged from the pipe walls during normal operation will be minimized.

Oil sampling

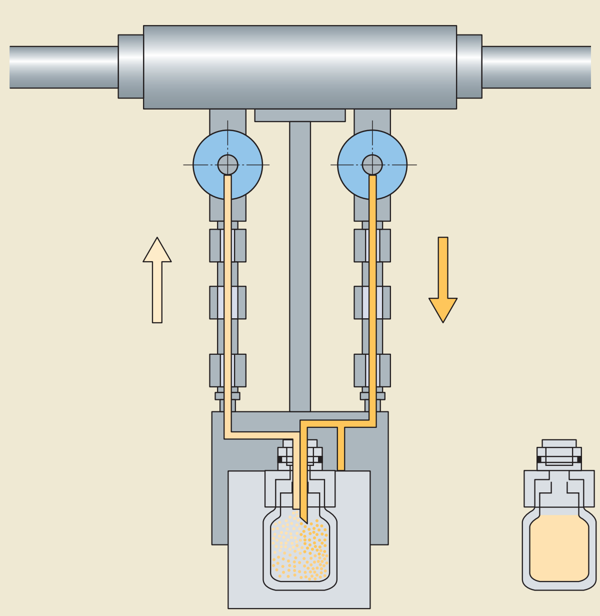
The ideal oil sample should be representative i.e. identical to the lubricant entering the rolling bearing.

Sampling from the pressurized side of the circulating oil system can be done either with a simple ball valve or with more sophisticated equipment. The main requirement is to flush the valve and the sample bottle so much that no additional contamination will enter the oil sample.

When taking oil samples from non-pressurized systems like oil baths and oil reservoirs, it is important that the sample is taken at a certain distance above the bottom sedimentation. In these cases, some kind of syringe or pump has to be used. The results are not as accurate as those obtained when sampling from pressurized pipes. On the other hand, oil samples from return pipes can be used when analyzing the source of the wear particles.

The best way to analyze the cleanliness level of the oil is to install an online automatic particle counter. However, all oil lubrication systems should be provided with good sampling points because additional testing of the oil for viscosity, water content, oxidation etc. is required. For this purpose, equipment is commercially available e.g. the sampler shown in **fig. 7.8**. This equipment uses sampling points integrated in the pressurized side of the lubrication system. When the sampler is connected to a sampling point, it is easy to flush the sample bottle with the pressurized lubricating oil. This ensures that the oil sample will be representative. There is, of course, other similar equipment on the market.

Fig. 7.8

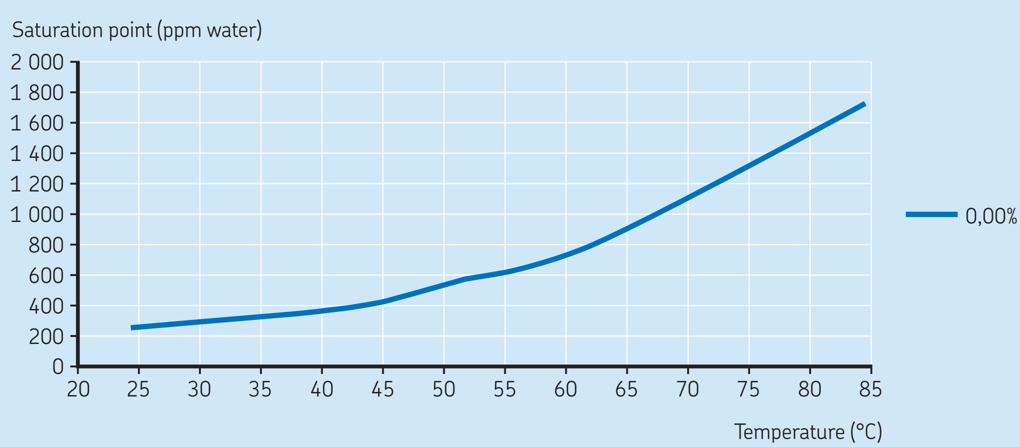


Equipment for oil sampling

Checking water content

Today, there are some products available for continuous online measurements, but the method most widely used is to take an oil sample and carry out a Karl Fischer analysis. Irrespective of the method, the results are usually presented in ppm, because the actual quantities are very small. Maximum water content should be 200 ppm. Water content up to 500 ppm can be tolerated for a short period when bearings are rotating if the oil temperature is such that all water is dissolved. 500 ppm water content can create free water in the oil inlet pipes as the oil temperature is colder than after passing through the bearing (→ **fig. 7.5**).

Diagram 7.5



Saturation curve for a typical ISO VG 220 paper machine oil

Checking oil condition

As certain properties change during operation, regular condition checks should be carried out. For example, the degradation of the oil is mainly determined by how often the oil passes through heated areas in the system like bearings, pumps etc. Contamination also influences the oil life. The greater the number of steel particles in the oil, the faster the oxidation of the oil. Suitable oil change intervals can be determined by regular checks of the oil condition.

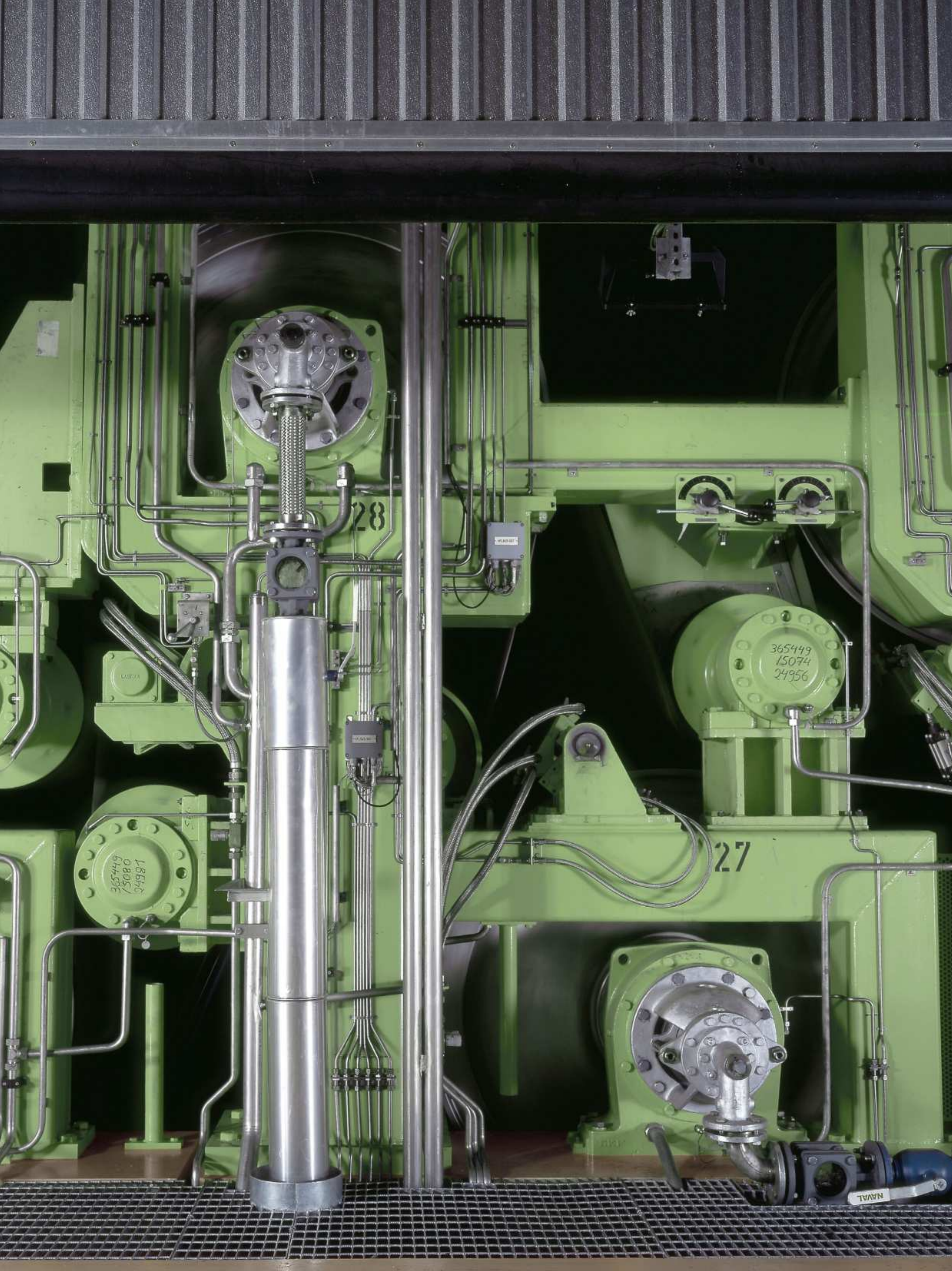
Such an analysis should include checking the following properties:

- viscosity
- oxidation
- TAN (Total acid number)
- particle distribution by size
- microscopic examination of particle type and shape
- water content
- loss of additive content

These properties dictate the life of the oil in oil baths, but oil life in circulation systems can be extended by removing particles and water from the oil.

There are no fixed rules on when oil should be changed/filtered based on the above properties. Each machine and bearing application can have different rules. However, as a general rule which should be adapted based on field experience, the following triggers can be used:

- Change of viscosity higher than 20% of the viscosity of fresh oil.
- Change of the TAN higher than 2 (mg KOH/g) compared to fresh oil.
- Water content higher than 500 ppm or water content that could be above the saturation point of the oil when the machine is cold.
- Increase of 2 maximum in the ISO 4406 code. Example: normal: –/15/12 or below, maximum : –/17/14.
- Presence of hard particles: above 200 Hv and above 6 µm size, above 100 Hv and above 15 µm size, above 50 Hv and above 30 µm size. Any hard particle that could be felt between two fingers.
- Depending on oil and additives, too great a change of the IR spectra of typical additives peak.


28
365449
15074
24956
27
NAVAL

Lubrication examples

The following examples closely represent the bearing applications described in this handbook. These examples are to be used as general guidelines only and are valid only when the bearing application and operating conditions are the same as those shown in this chapter.

Example 1

Breast, forming and forward drive rolls (grease), fig. 8.1

Lubrication guidelines

SKF's experience is that grease with a minimum base oil viscosity 175 mm^2/s provides good lubrication performance in this position. Rust protection and sealing properties have the highest priority in the wet section.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Grease lubrication*, should also be taken into account.

Comments on the diagram 8.1

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the outer ring temperature. The load at each speed corresponds to a bearing basic rating life of 120 000 hours. The highest bearing temperature shown in the diagram is 75 °C as it is customary to use oil lubrication at higher temperatures.

Table 8.1

Machine data	
Paper grades	all
Roll position	forming section
Paper speed	100–1 300 m/min
Operating conditions for the bearings	
Ambient temperature	35 °C
Lubrication	grease
Base oil viscosity	175 mm^2/s at 40 °C

Fig. 8.1

Bearing application

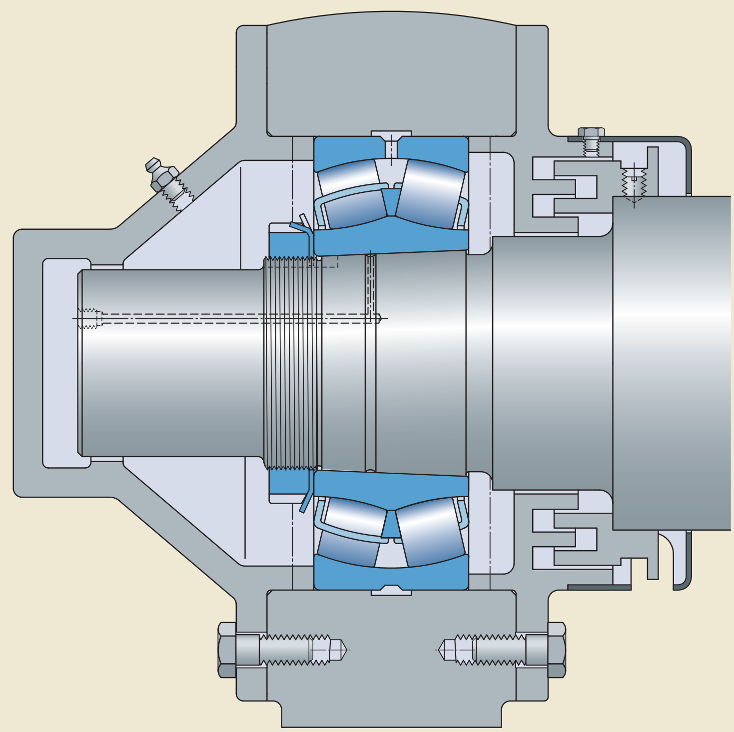
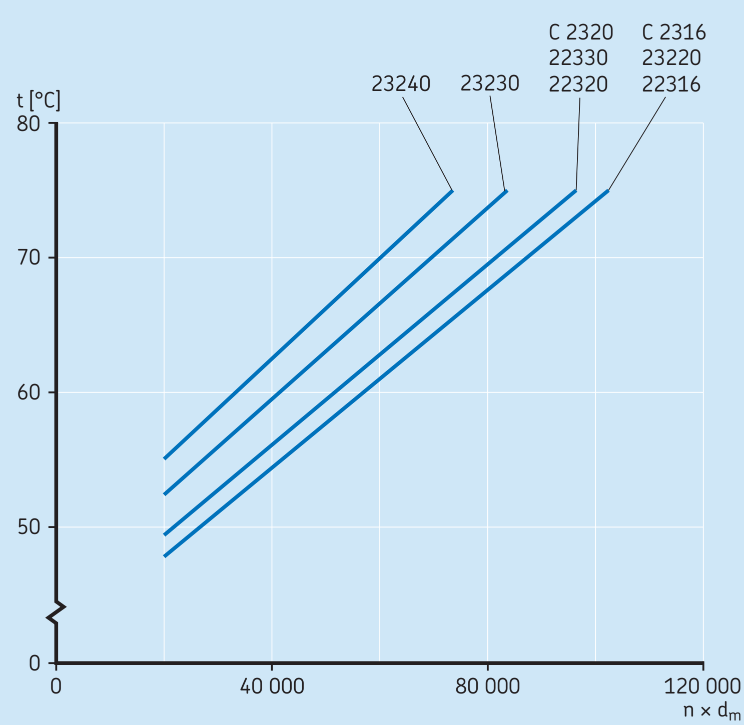


Diagram 8.1

Bearing temperature/speed factor diagram



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Example 2

Breast, forming and forward drive rolls (oil), fig. 8.2

Lubrication guidelines

Bearings for these rolls are smaller and have a lower speed factor ($n \times d_m$) than suction roll bearings which are often lubricated with the same oil. Therefore, the lubricant properties should be based on the operating conditions for the suction roll bearings (→ **example 6**). However, the bearings for the rolls in this example should have an oil flow resulting in a viscosity ratio κ according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.2

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the oil flow in the diagram is valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the outer ring temperature. The load at each speed corresponds to a bearing basic rating life of 120 000 hours. The oil flow in the diagram is based on the use of ISO VG 220 oil and selected to fulfil two criteria: the κ guidelines and a bearing operating temperature of 75 °C. If oil with lower viscosity than ISO VG 220 is used, then the oil flow must be increased.

Note that the diagram shows minimum acceptable oil flow. Slightly higher oil flow values are acceptable and beneficial in most cases.

Table 8.2

Machine data	
Paper grades	all
Roll position	forming section
Paper speed	700–2 200 m/min
Operating conditions for the bearings	
Ambient temperature	40 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	45 °C

Fig. 8.2

Bearing application

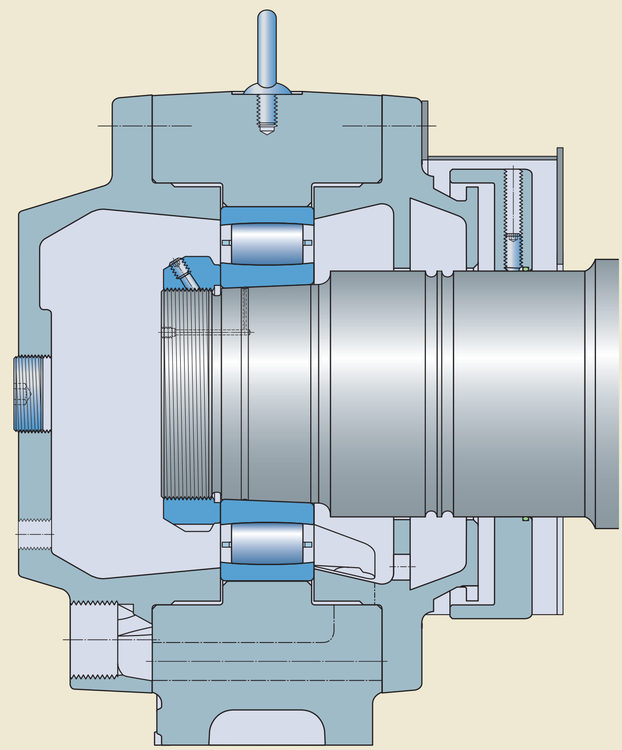
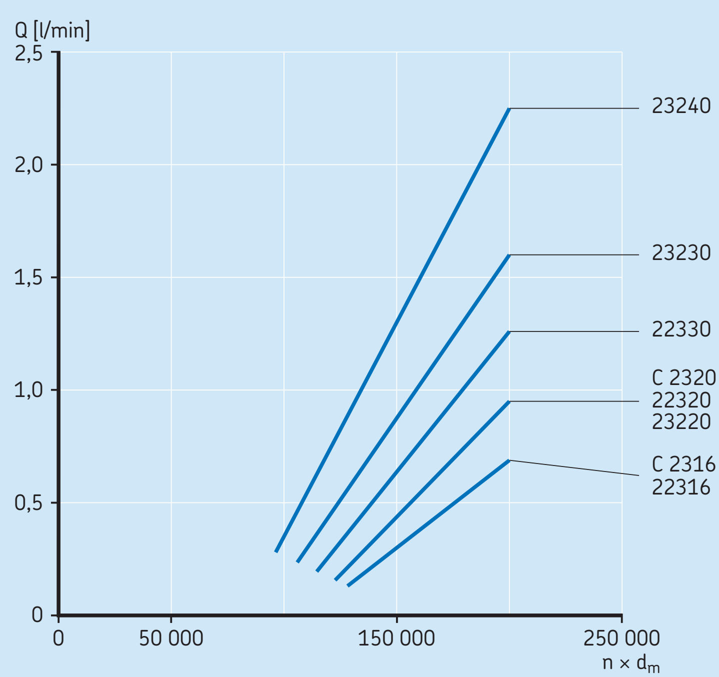


Diagram 8.2

Oil flows for bearing temperature 75 °C



Example 3

Wire roll (grease), fig. 8.3

Lubrication guidelines

SKF's experience is that grease with a minimum base oil viscosity 175 mm²/s provides good lubrication performance in this position. Rust protection and sealing properties have the highest priority in the wet section.

General guidelines, as outlined in *Chapter 7*, *Lubrication*, *Grease lubrication*, should also be taken into account.

Comments on the diagram 8.3

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the outer ring temperature. The load at each speed corresponds to a bearing basic rating life of 120 000 hours. The highest bearing temperature shown in the diagram is 75 °C as it is customary to use oil lubrication at higher temperatures.

Table 8.3

Machine data	
Paper grades	all
Roll position	forming section
Paper speed	100–1 300 m/min
Operating conditions for the bearings	
Ambient temperature	35 °C
Lubrication	grease
Base oil viscosity	175 mm^2/s at 40 °C

Fig. 8.3

Bearing application

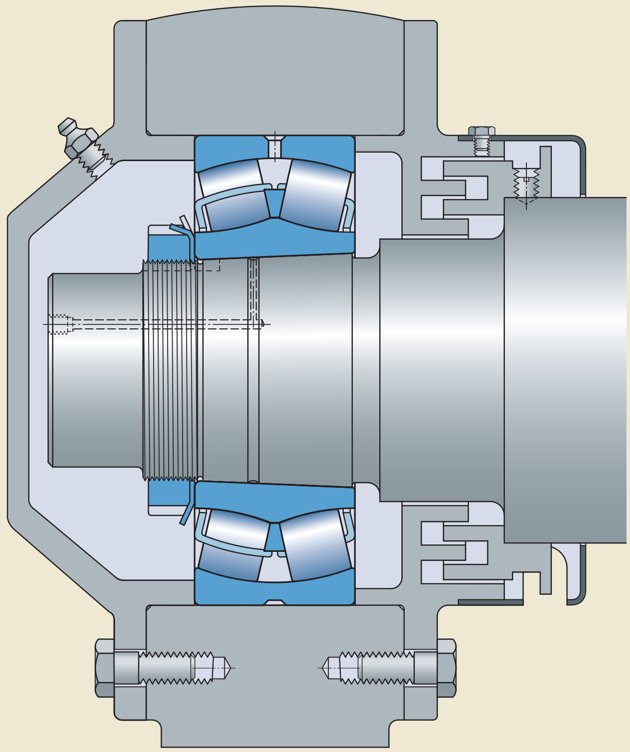
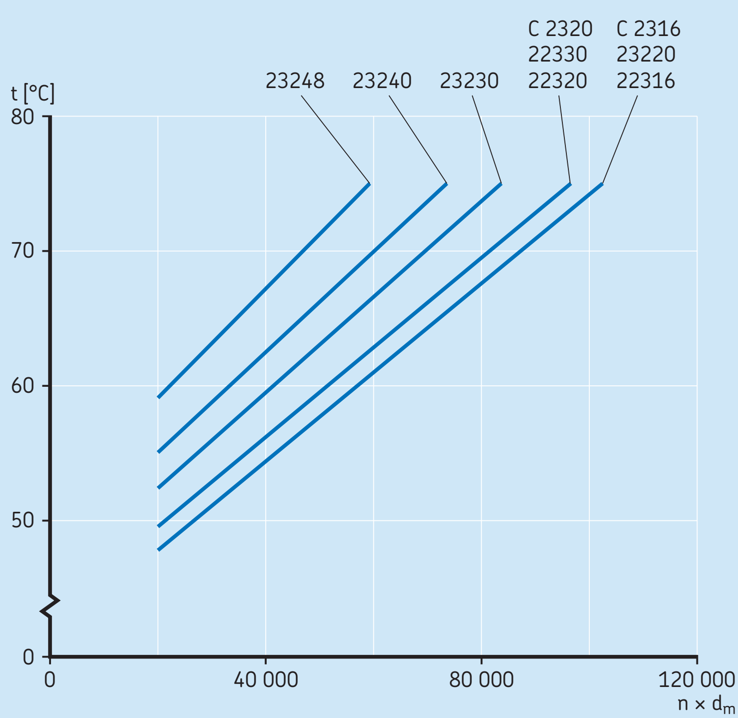


Diagram 8.3

Bearing temperature/speed factor diagram



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Example 4

Wire roll (oil), fig. 8.4

Lubrication guidelines

Wire guide roll bearings are smaller and have a lower speed factor (n × dm) than suction roll bearings which are often lubricated with the same oil. Therefore, the lubricant properties should be based on the operating conditions for the suction roll bearings (→ **example 6**). However, the wire roll bearings should have an oil flow resulting in viscosity ratio κ according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.4

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the oil flow in the diagram is valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to give the outer ring temperature. The load at each speed corresponds to a bearing basic rating life of 120 000 hours. The oil flow in the diagram is based on the use of ISO VG 220 oil and selected to fulfil two criteria: the κ guidelines and a bearing operating temperature of 75 °C. If oil with lower viscosity than ISO VG 220 is used, then the oil flow must be increased.

Note that the diagram shows minimum acceptable oil flow. Slightly higher oil flow values are acceptable and beneficial in most cases.

Table 8.4

Machine data	
Paper grades	liner, fine paper, newsprint, tissue
Roll position	forming section
Paper speed	700–2 200 m/min
Operating conditions for the bearings	
Ambient temperature	40 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	45 °C

Fig. 8.4

Bearing application

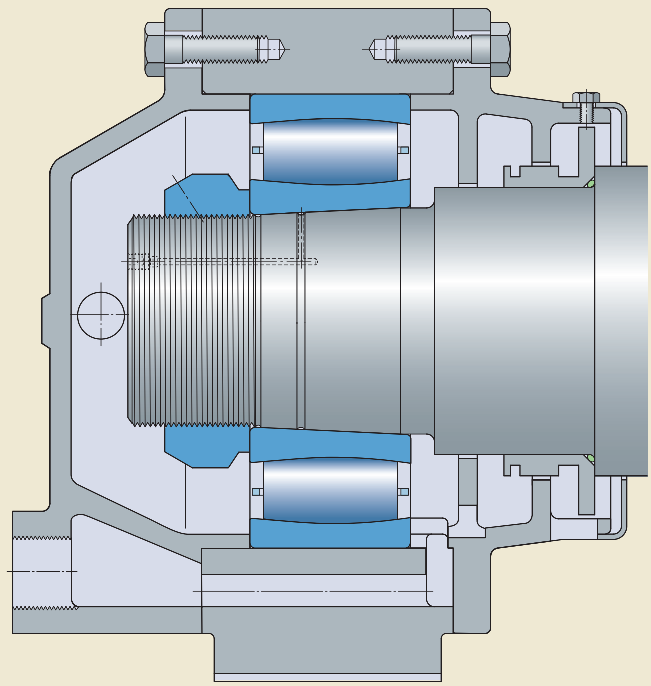
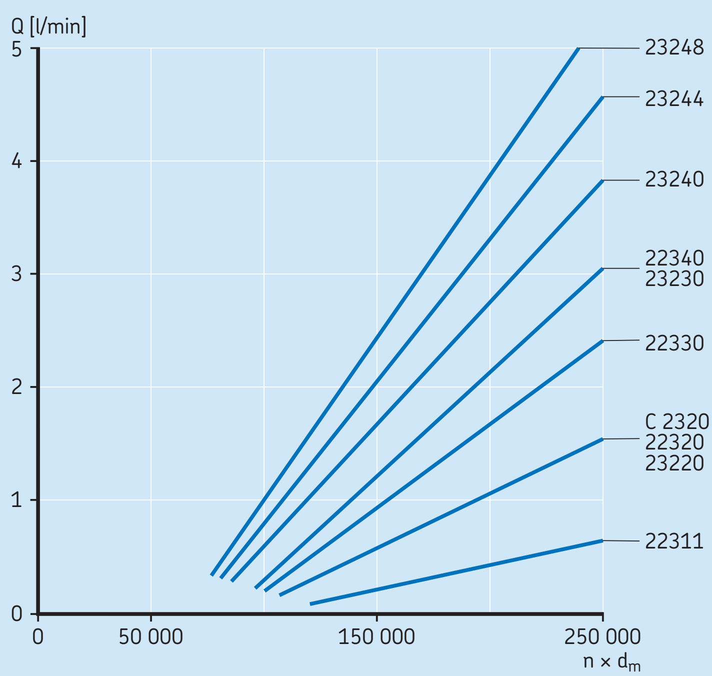


Diagram 8.4

Oil flows for bearing temperature 75 °C



Example 5

Suction roll (grease), fig. 8.5

Lubrication guidelines

SKF's experience is that grease with a minimum base oil viscosity 175 mm^2/s provides good lubrication performance in this position. Rust protection and sealing properties have the highest priority in the wet section.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Grease lubrication*, should also be taken into account.

Comments on the diagram 8.5

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to give the outer ring temperature at different speeds. The load at each speed corresponds to a bearing basic rating life of 120 000 hours.

The maximum bearing temperature shown in the diagram is 75 °C as it is customary to use oil lubrication at higher temperatures.

Table 8.5

Machine data	
Paper grades	board, liner
Roll position	forming section
Paper speed	100–500 m/min
Operating conditions for the bearings	
Ambient temperature	40 °C
Lubrication	grease
Base oil viscosity	175 mm^2/s

Fig. 8.5

Bearing application

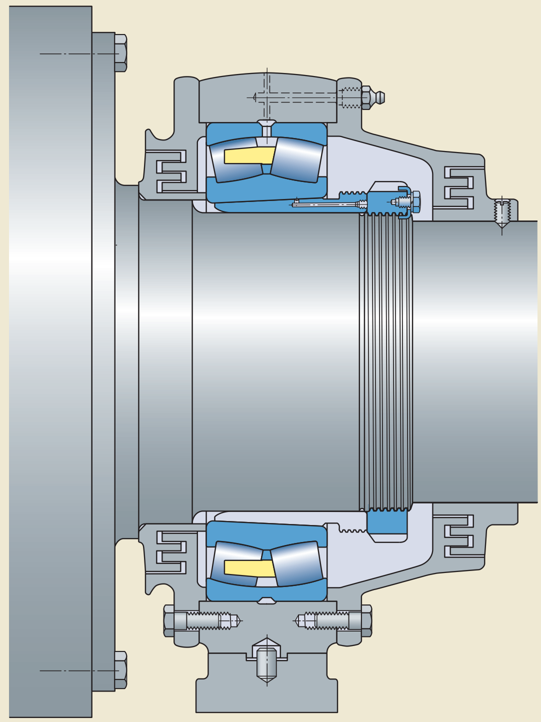
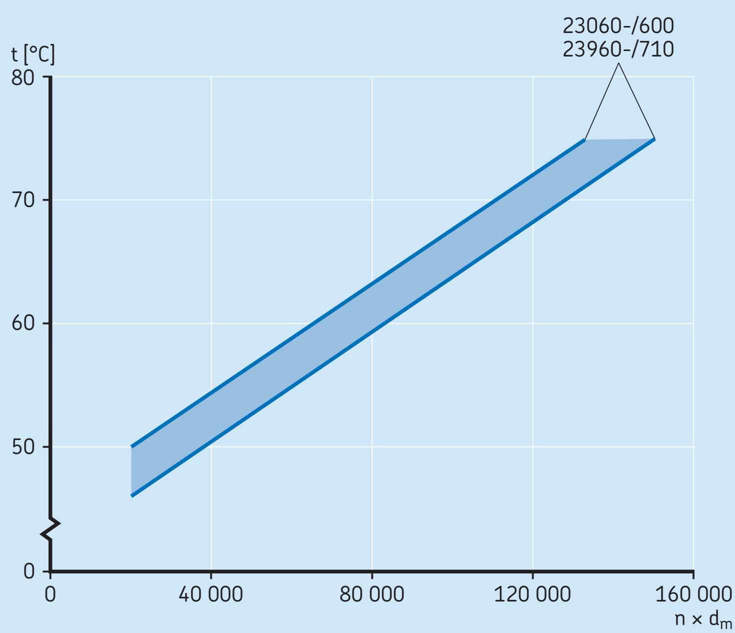


Diagram 8.5

Bearing temperature/ speed factor diagram



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Example 6

Suction roll (oil), fig. 8.6

Lubrication guidelines

As suction roll bearings are large and sometimes rotate at very high speeds, there is a risk of smearing. To avoid smearing, the oil should have EP additives and the viscosity ratio κ should be according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in Chapter 7, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.6

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the oil flow in the diagram is valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the outer ring temperature. The load at each speed corresponds to a bearing basic rating life of 120 000 hours. The oil flow in the diagram is based on the use of ISO VG 220 oil and selected to fulfil two criteria: the κ guidelines and a bearing operating temperature of 75 °C. If oil with lower viscosity than ISO VG 220 is used for these bearings, then the oil flows must be increased.

Thicker oil, e.g. ISO VG 320, is advantageous and will improve the lubrication conditions, but the bearing temperature will increase a little.

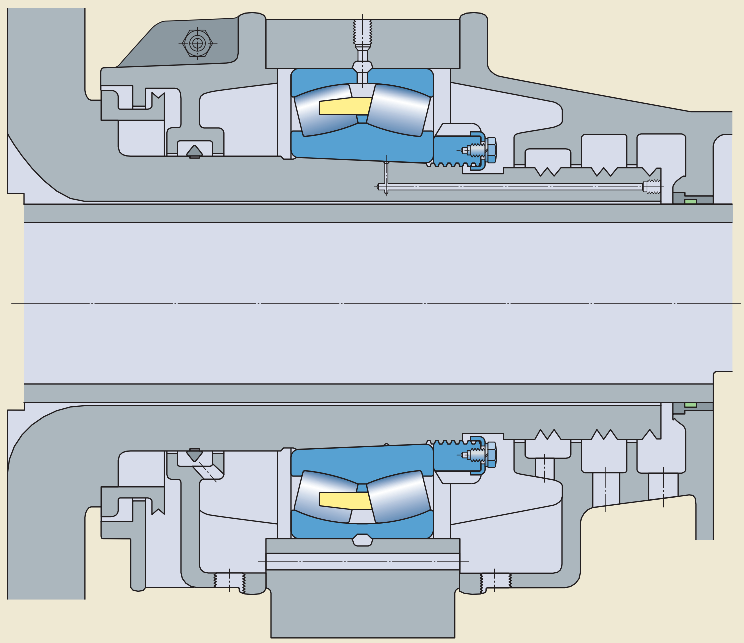
Note that the diagram shows minimum acceptable oil flow. Slightly higher oil flow values are acceptable and beneficial in most cases.

Table 8.6

Machine data	
Paper grades	board, liner, fine paper, newsprint
Roll position	forming section
Paper speed	400–2 000 m/min
Operating conditions for the bearings	
Ambient temperature	45 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	45 °C

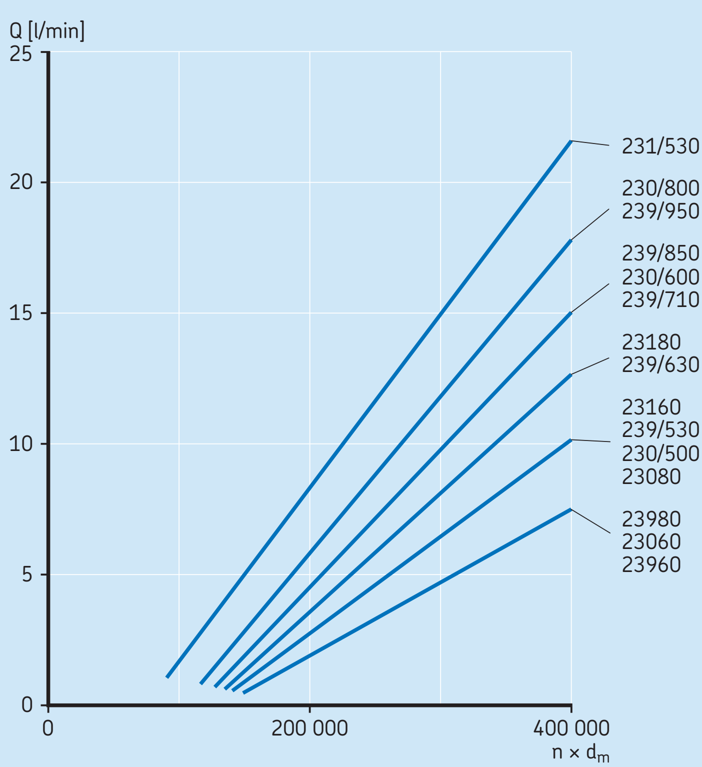
Bearing application

Fig. 8. 6



Oil flows for bearing temperature 75 °C

Diagram 8.6



Example 7

Felt roll (grease), fig. 8.7

Lubrication guidelines

SKF's experience is that grease with a minimum base oil viscosity 175 mm^2/s provides good lubrication performance in this position. Rust protection and sealing properties have the highest priority in the wet section.

The general guidelines, as outlined in Chapter 7, *Lubrication*, *Grease lubrication*, should also be taken into account.

Comments on the diagram 8.7

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to give the outer ring temperature. The load at each speed corresponds to a bearing basic rating life of 120 000 hours. The highest bearing temperature shown in the diagram is 75 °C as it is customary to use oil lubrication at higher temperatures.

Table 8.7

Machine data	
Paper grades	all
Roll position	press section
Paper speed	100–1 000 m/min
Operating conditions for the bearings	
Ambient temperature	40 °C
Lubrication	grease
Base oil viscosity	175 mm^2/s at 40 °C

Fig. 8.7

Bearing application

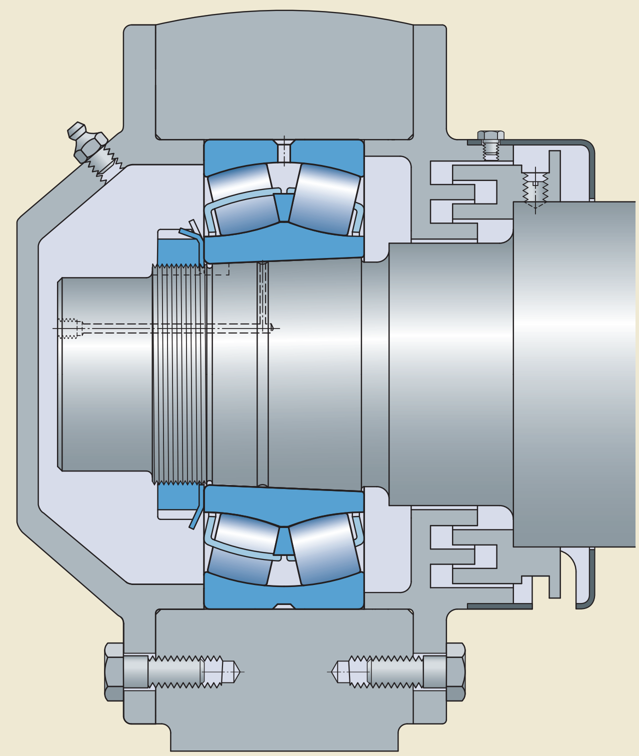
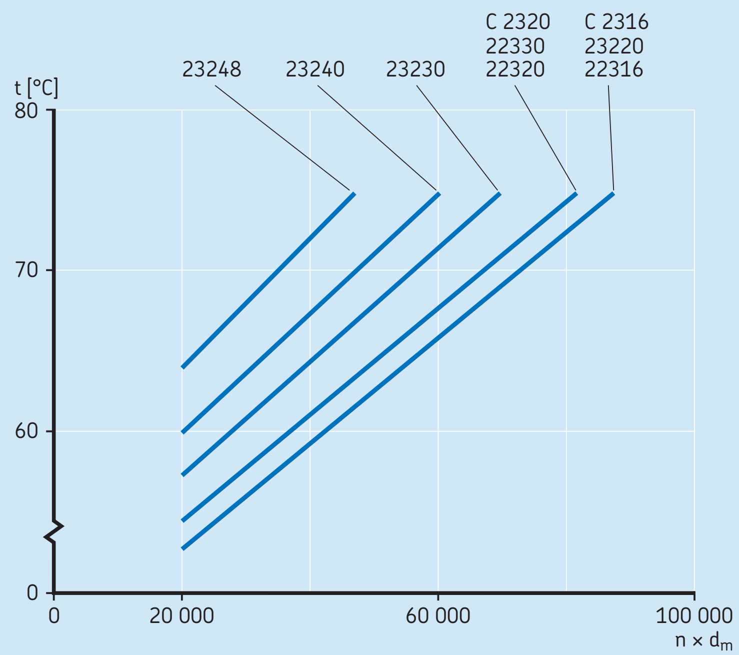


Diagram 8.7

Bearing temperature/speed factor diagram



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Example 8

Felt roll (oil), fig. 8.8

Lubrication guidelines

Felt roll bearings are smaller and have lower speed factors ($n \times d_m$) than press roll bearings which are often lubricated with the same oil. Therefore, the lubricant properties should be based on the operating conditions for the press roll bearings. However, the felt roll bearings should have an oil flow resulting in a viscosity ratio κ according to the guidelines in *Chapter 10, Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in Chapter 7, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.8

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the oil flow in the diagram is valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the outer ring temperature. The load at each speed corresponds to a bearing basic rating life of 120 000 hours. The oil flow in the diagram is based on the use of ISO VG 220 oil and selected to fulfil two criteria: the κ guidelines and a bearing operating temperature of 75 °C.

Note that the diagram shows minimum acceptable oil flow. Slightly higher oil flow values are acceptable and beneficial in most cases.

Table 8.8

Machine data	
Paper grades	all
Roll position	press section
Paper speed	400–2 200 m/min
Operating conditions for the bearings	
Ambient temperature	45 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	45 °C

Fig. 8.8

Bearing application

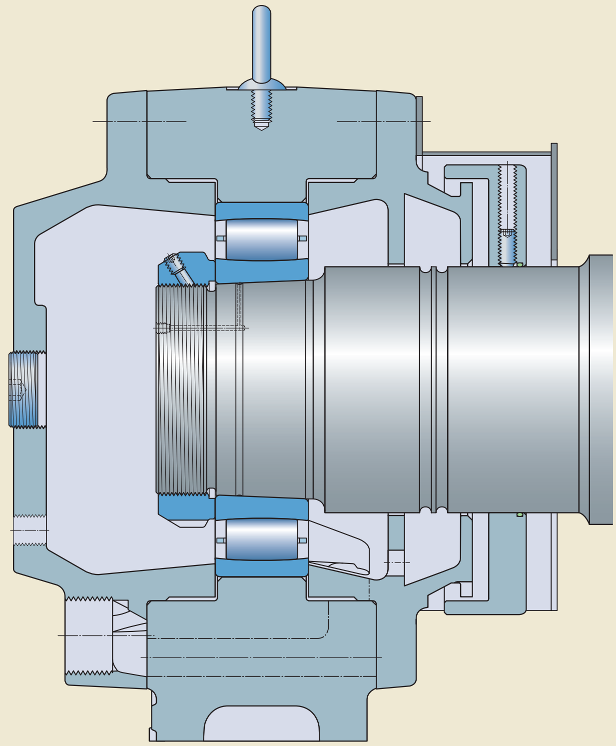
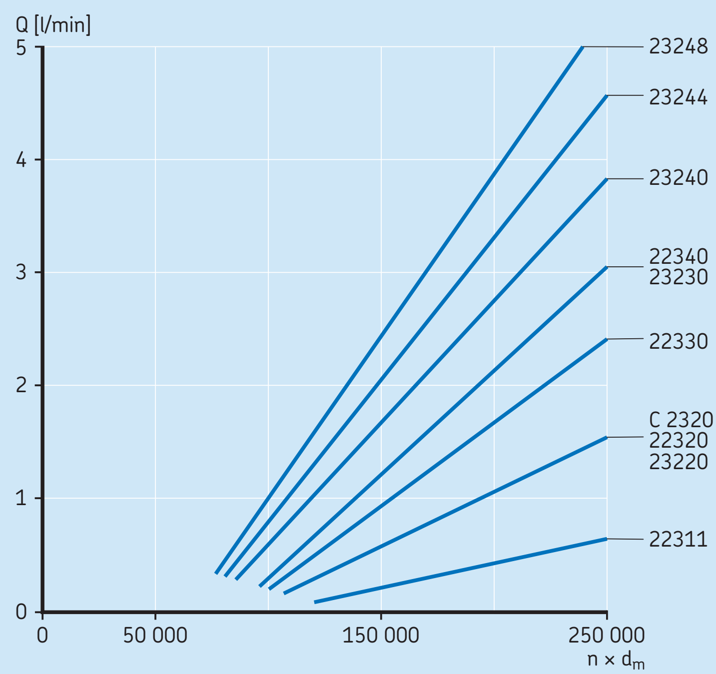


Diagram 8.8

Oil flows for bearing temperature 75 °C



Example 9

Press roll (grease), fig. 8.9

Lubrication guidelines

SKF's experience is that grease with a minimum base oil viscosity 175 mm^2/s provides good lubrication performance in this position. Rust protection and sealing properties have the highest priority in the wet section.

The general guidelines, as outlined in Chapter 7, *Lubrication*, *Grease lubrication*, should also be taken into account.

Comments on the diagram 8.9

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the calculated bearing temperatures in this example are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the outer ring temperature at different speeds. The load at each speed corresponds to a bearing basic rating life of 120 000 hours.

The maximum bearing temperature shown in the diagram is 75 °C as it is customary to use oil lubrication at higher temperatures.

Table 8.9

Machine data	
Paper grades	board, liner
Roll position	press section
Paper speed	100–500 m/min
Operating conditions for the bearings	
Ambient temperature	40 °C
Lubrication	grease
Base oil viscosity	175 mm^2/s at 40 °C

Fig. 8.9

Bearing application

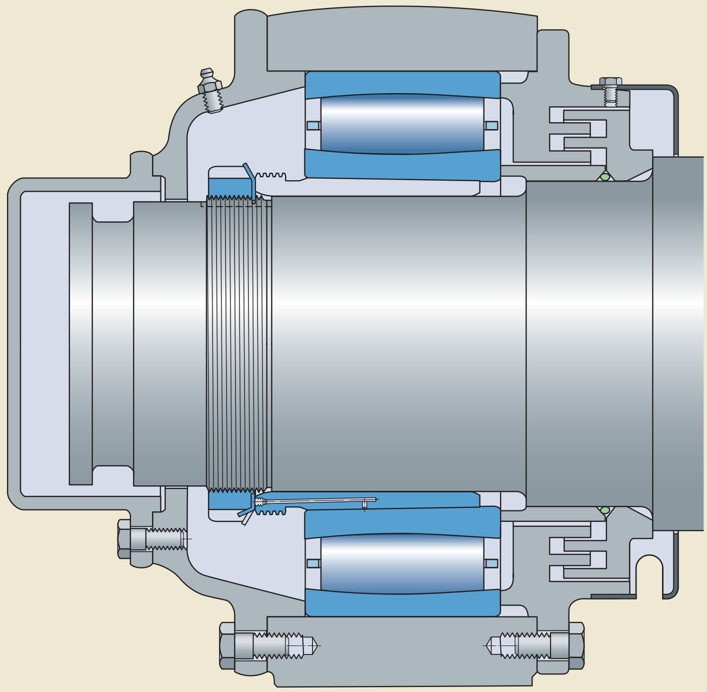
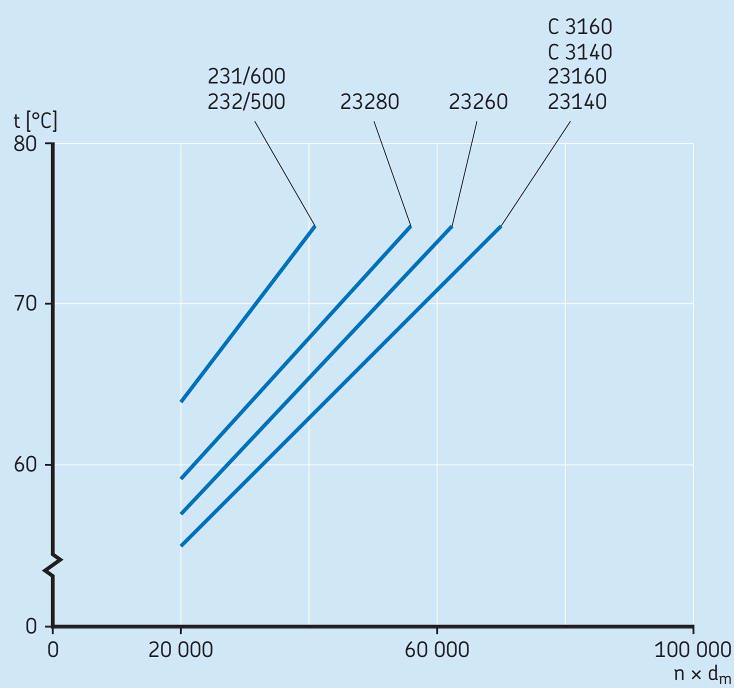


Diagram 8.9

Bearing temperature/speed factor diagram



8

Example 10

Press roll (oil), fig. 8.10

Lubrication guidelines

As press roll bearings are large and sometimes rotate at high speeds, there is a risk of smearing. To avoid smearing, the oil should have EP additives and the viscosity ratio κ should be according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.10

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the oil flow in the diagram is valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the outer ring temperature. The load at each speed corresponds to a bearing basic rating life of 120 000 hours.

The oil flow in the diagram is based on the use of ISO VG 220 oil and selected to fulfil two criteria: the κ guidelines and a bearing operating temperature of 75 °C. If oil with lower viscosity than ISO VG 220 is used for these bearings, then the oil flow must be increased. Thicker oil, e.g. ISO VG 320, is advantageous and will improve the lubrication conditions, but the bearing temperature will increase a little.

Note that the diagram shows minimum acceptable oil flow. Slightly higher oil flow values are acceptable and beneficial in most cases.

Table 8.10

Machine data	
Paper grades	board, liner, fine paper, newsprint
Roll position	press section
Paper speed	400–1 800 m/min
Operating conditions for the bearings	
Ambient temperature	45 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	45 °C

Fig. 8.10

Bearing application

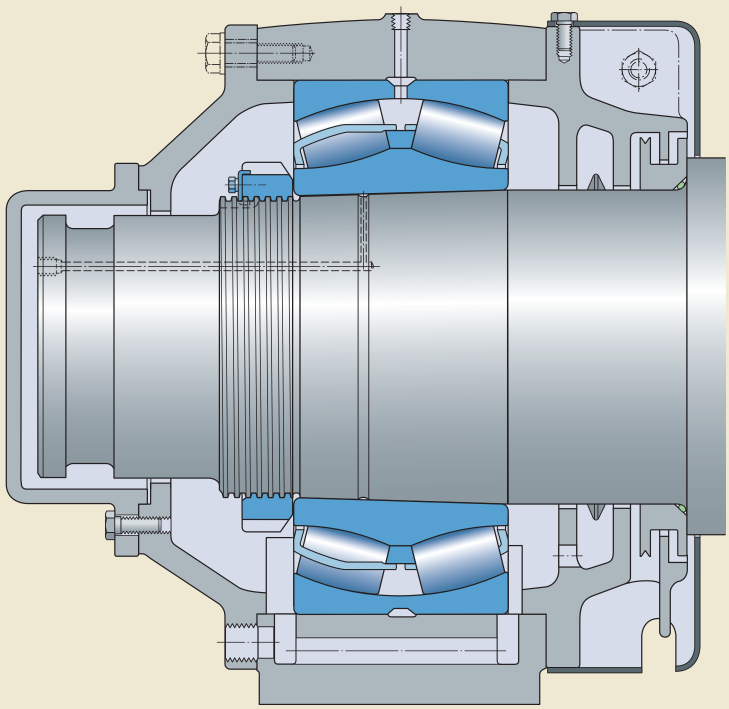
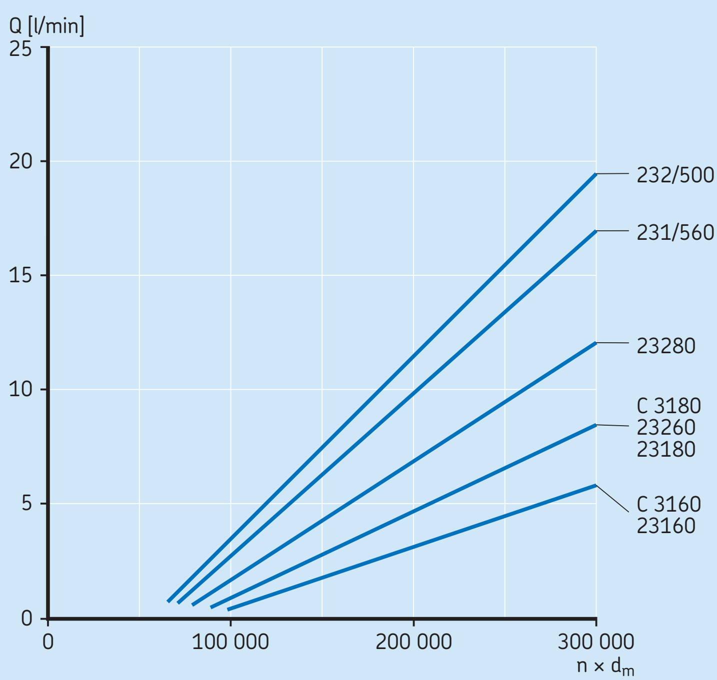


Diagram 8.10

Oil flows for bearing temperature 75 °C



Example 11

Press roll, deflection compensating (oil), fig. 8.11

Lubrication guidelines

As press roll bearings are large and sometimes rotate at very high speeds, there is a risk of smearing. To avoid smearing, the oil should have EP additives and the viscosity ratio κ should be according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.11

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the oil flow in the diagram is valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the outer ring temperature. The load at each speed corresponds to a bearing basic rating life of 120 000 hours.

The oil flow in the diagram is based on the use of ISO VG 220 oil and selected to fulfil two criteria: the κ guidelines and a bearing operating temperature of 75 °C. If oil with lower viscosity than ISO VG 220 is used for these bearings, then the oil flow must be increased. Thicker oil, e.g. ISO VG 320, is advantageous and will improve the lubrication conditions, but the bearing temperature will increase a little.

Note that the diagram shows minimum acceptable oil flow. Slightly higher oil flow values are acceptable and beneficial in most cases.

Table 8.11

Machine data	
Paper grades	all
Roll position	press section
Paper speed	400–1 800 m/min
Operating conditions for the bearings	
Ambient temperature	45 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	45 °C

Fig. 8.11

Bearing application

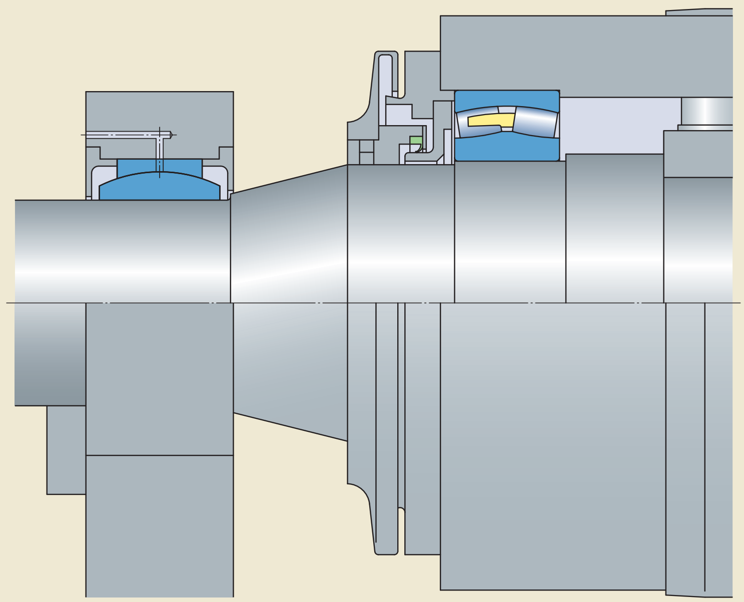
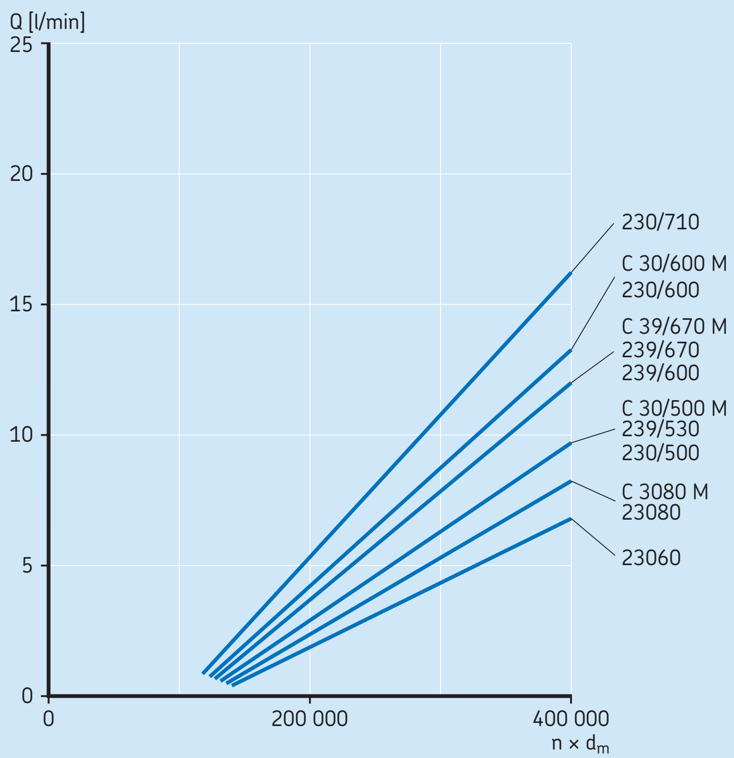


Diagram 8.11

Oil flows for bearing temperature 75 °C



Example 12

Felt roll (grease), fig. 8.12

Lubrication guidelines

This example is included in the handbook as there are still some old machines in service using grease lubrication for guide roll bearings in the dryer section.

However, SKF's experience is that it is difficult to find standard greases which will provide good lubrication properties at operating temperatures above 120 °C. Therefore, the SKF test program for high temperature greases is based on a maximum bearing operating temperature of 120 °C.

Greases which have passed the test programme should provide sufficient lubrication for the bearings in this position. Oxidation stability and oil bleeding properties have the highest priority in the dryer section.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Grease lubrication*, should also be taken into account.

Comments on the diagram 8.12

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the calculated bearing temperatures in this example are valid only when the bearing application and operating conditions are similar to those shown in this example.

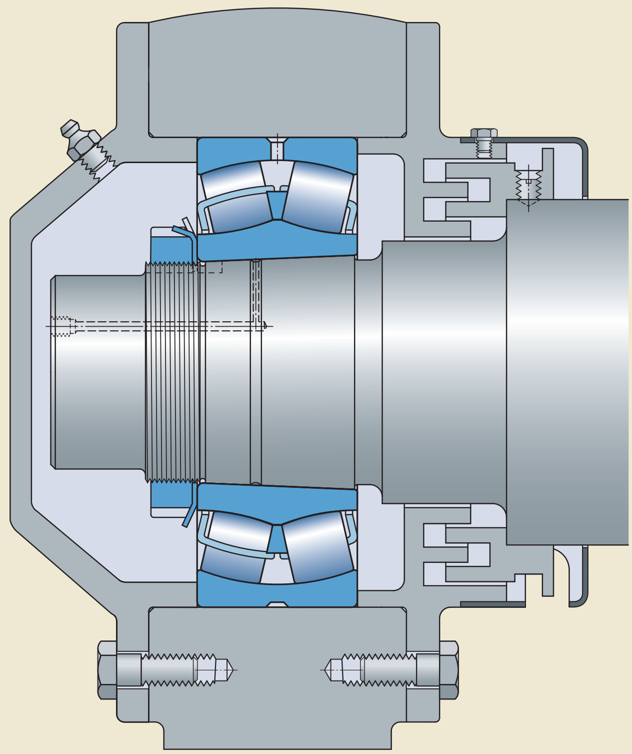
The calculations for this diagram have been made using an SKF computer program to obtain the outer ring temperature at different speeds. The load at each speed corresponds to a bearing basic rating life of 200 000 hours.

Note that the ambient temperature has a great influence on the bearing operating temperature.

Table 8.12

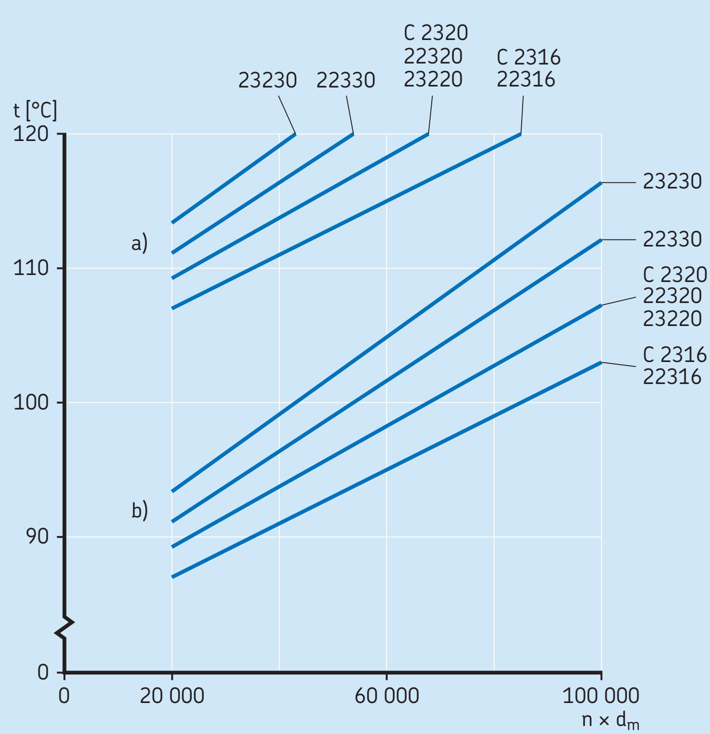
Machine data	
Paper grades	board, liner, fine paper, newsprint
Roll position	dryer section
Paper speed	100–800 m/min
Operating conditions for the bearings	
Ambient temperature	80–100 °C
Lubrication	grease
Base oil viscosity	400 mm^2/s

Fig. 8.12



Bearing application

Diagram 8.12



Bearing speed/speed factor diagram
a) ambient temperature 100 °C
b) ambient temperature 80 °C

Example 13

Felt roll (oil), fig. 8.13

Lubrication guidelines

Felt roll bearings have better operating conditions than drying cylinder bearings which are lubricated with the same circulating oil system. Therefore, the lubricant properties should be based on the operating conditions for drying cylinder bearings. However, the felt roll bearings should have an oil flow resulting in viscosity ratio κ according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter* 7, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.13

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the oil flow in the diagram is valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the outer ring temperature. The load at each speed corresponds to a bearing basic rating life of 200 000 hours.

The oil flow in the diagram is based on the use of ISO VG 220 oil and selected to fulfil two criteria: the κ guidelines and a bearing operating temperature of 90 °C. If oil with lower viscosity than ISO VG 220 is used, then the flow must be increased. Thicker oil, e.g. ISO VG 320, is advantageous and will improve the lubrication conditions, but the bearing temperature will increase a little.

Note that the diagram shows minimum acceptable oil flow. Slightly higher oil flow values are acceptable and beneficial in most cases.

Table 8.13

Machine data	
Paper grades	board, liner, fine paper, newsprint
Roll position	dryer section
Paper speed	400–1 800 m/min
Operating conditions for the bearings	
Ambient temperature	100 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	60 °C

Fig. 8.13

Bearing application

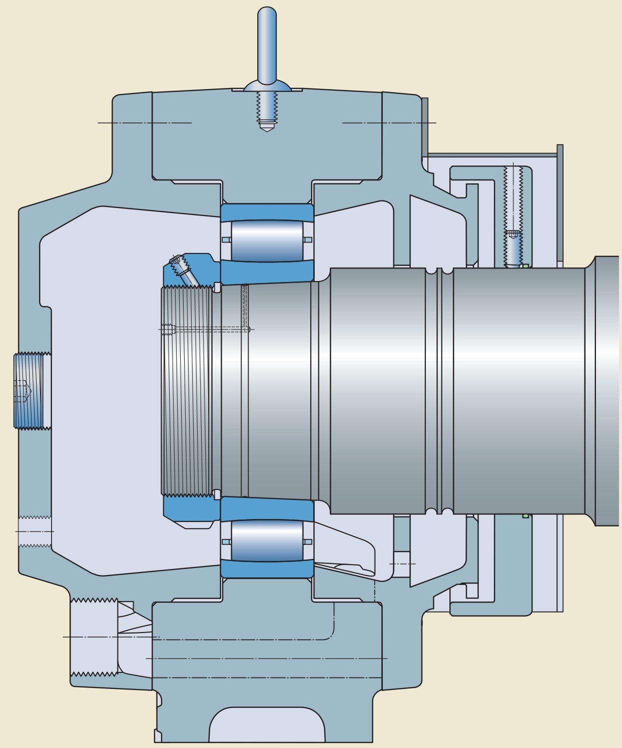
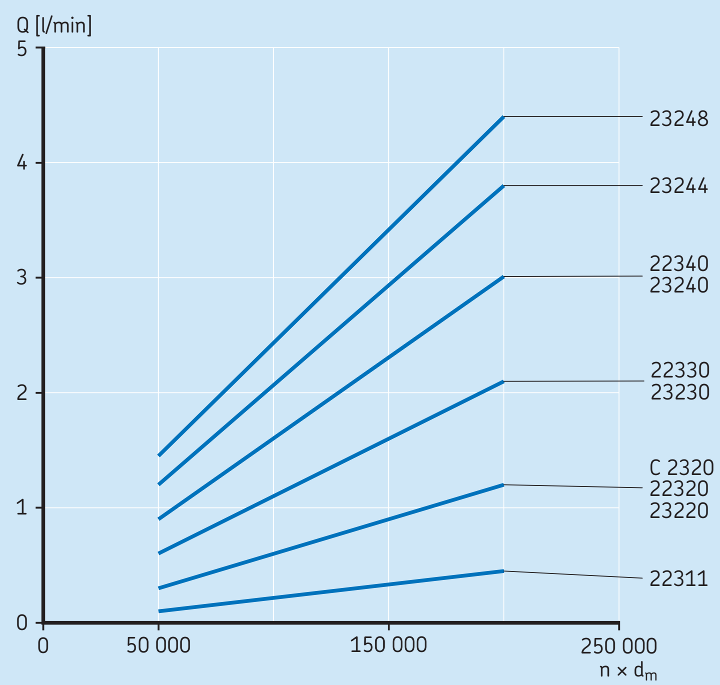


Diagram 8.13

Oil flows for bearing temperature 90 °C



8

Example 14

Drying cylinder with steam temperature 140 °C (oil bath), fig. 8.14

Lubrication guidelines

As the drying cylinder bearings in this example rotate at very low speeds and are subjected to high temperatures, there will be metallic contact between rollers and raceways. This means that there is a risk of surface distress. To avoid this risk, the lubricating oil should have efficient AW additives and a viscosity ratio κ according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

Synthetic oil is recommended because mineral oils are not suitable at temperatures above 120 °C.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.14

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the inner ring temperature. The load at each speed corresponds to a bearing basic rating life of 200 000 hours. Note that the bearing temperature is lower at high speeds than at low speeds. The reason is that the increase in cooling via the rotating journal is higher than the increase in heat generation in the bearing. At operating temperatures above 100 °C, it is not possible to fulfil the κ guidelines with commonly used ISO VG 220 oils.

The recommendation for this application is to improve the lubrication conditions as much as practically possible. The best remedy is to introduce efficient journal insulation in combination with circulating oil lubrication which will result in a bearing temperature of 85–90 °C. Changing to oil with one higher viscosity level than used for circulation systems is also beneficial.

Table 8.14

Machine data	
Paper grades	board, fine paper (old machines)
Roll position	dryer section
Paper speed	100–150 m/min
Operating conditions for the bearings	
Ambient temperature	75 °C
Lubrication	oil bath
Oil viscosity	ISO VG 220–1 500
Journal insulation	none

Fig. 8.14

Bearing application

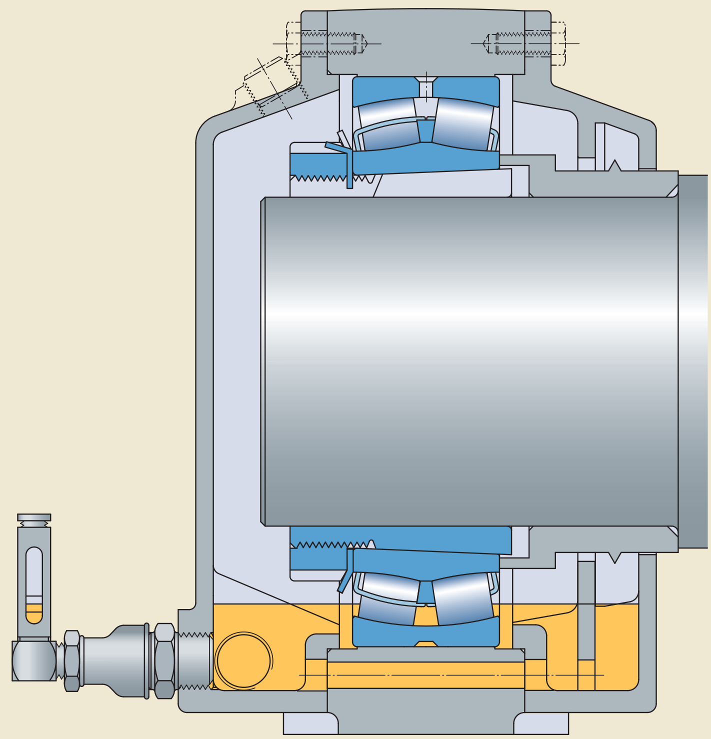
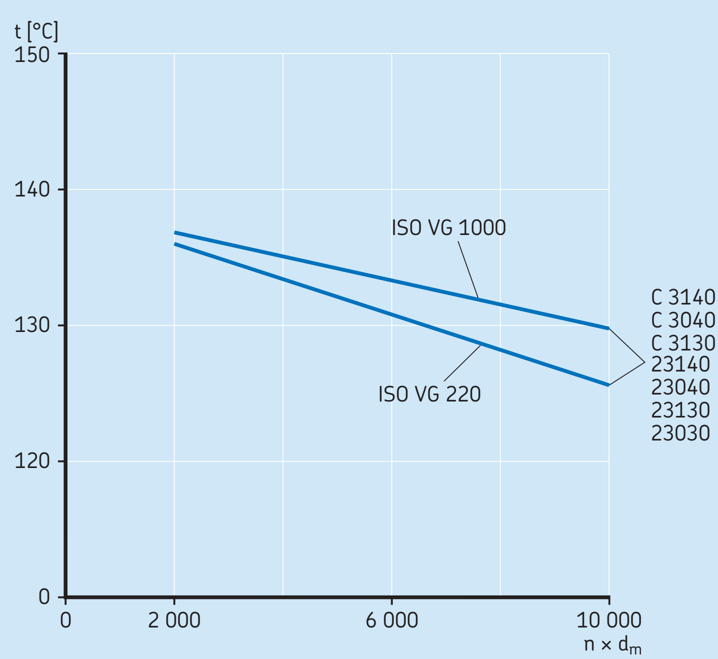


Diagram 8.14

Bearing temperature/speed factor diagram



Example 15

Drying cylinder with steam temperature 140 °C, fig. 8.15

Lubrication guidelines

As drying cylinder bearings rotate at relatively low speeds and are subjected to high temperatures, there will be metallic contact between rollers and raceways. This means that there is a risk of surface distress. To avoid this risk, the circulating oil should have efficient AW additives and a viscosity ratio κ according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.15

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the inner ring temperature. The load at each speed corresponds to a bearing basic rating life of 200 000 hours.

An oil flow of 0,5 l/min, which is common in old machines, gives a bearing temperature of around 120 °C.

At operating temperatures above 100 °C, it is not possible to fulfil the κ guidelines and some mineral oils have shown a strong tendency to carbonize. Therefore, the recommendation for such applications is to improve the lubrication conditions as much as practically possible.

The diagram shows that the influence of the oil quantity is considerable up to 2–3 l/min, but after that scarcely anything is gained by an increased oil flow except for large size bearings running at high speeds.

The best remedy is to introduce efficient journal insulation which results in a bearing temperature of about 85–90 °C. Changing to synthetic oil is also beneficial.

Table 8.15

Machine data	
Paper grades	fine paper, newsprint
Roll position	dryer section
Paper speed	700–1 800 m/min
Operating conditions for the bearings	
Ambient temperature	100 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	60 °C
Journal insulation	none

Fig. 8.15

Bearing application

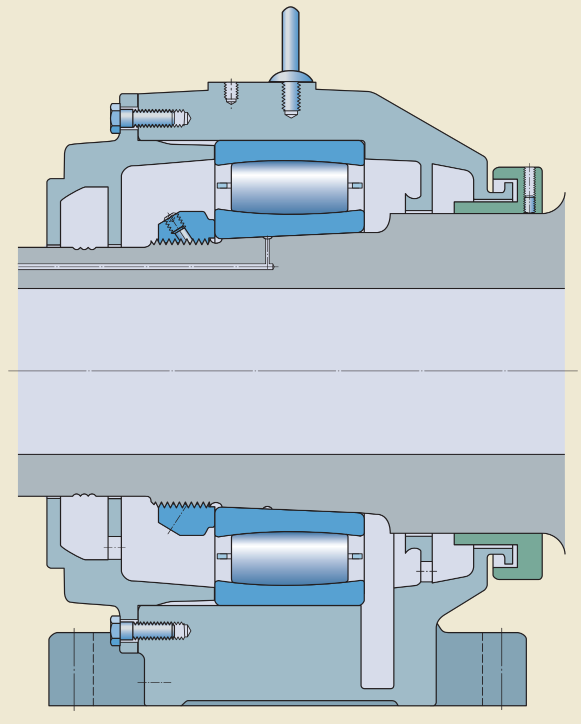
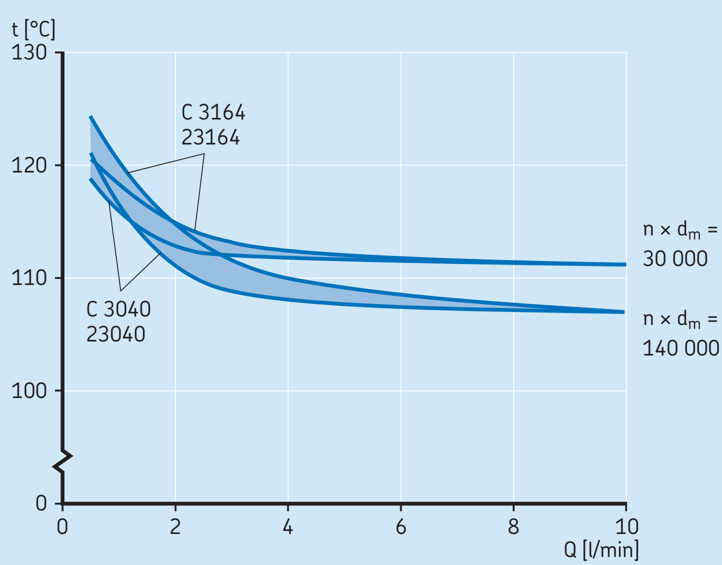


Diagram 8.15

Bearing temperature/oil flow diagram



Example 16

Drying cylinder with steam temperature 140 °C, fig. 8.16

Lubrication guidelines

As drying cylinder bearings rotate at relatively low speeds and are subjected to high temperatures, there will be metallic contact between rollers and raceways. This means that there is a risk of surface distress. To avoid this risk, the circulating oil should have efficient AW additives and a viscosity ratio κ according to the guidelines in *Chapter 10, Bearing damage and failure, How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7, Lubrication, Oil lubrication*, should also be taken into account.

Comments on the diagram 8.16

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the inner ring temperature. The load at each speed corresponds to a bearing basic rating life of 200 000 hours.

An oil flow of 0,5 l/min was common in the early machines with insulated journals. Such an oil flow gives a bearing temperature of around 90 °C at low speeds and around 100 °C at high speeds.

The aim for this application is to find the optimum oil flow which gives a bearing temperature somewhat below 90 °C.

The diagram shows that the influence of the oil quantity is considerable up to 2–3 l/min, but after that scarcely anything is gained by an increased oil flow except for large size bearings running at high speeds.

If ISO VG 320 oil is used instead of an ISO VG 220 one, the bearing temperature will increase a little. However, the use of ISO VG 320 oil would be beneficial because the viscosity ratio κ would be higher than when using an ISO VG 220 oil.

Table 8.16

Machine data	
Paper grades	fine paper, newspaper
Roll position	dryer section
Paper speed	700–1 800 m/min
Operating conditions for the bearings	
Ambient temperature	100 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	60 °C
Journal insulation	yes

Fig. 8.16

Bearing application

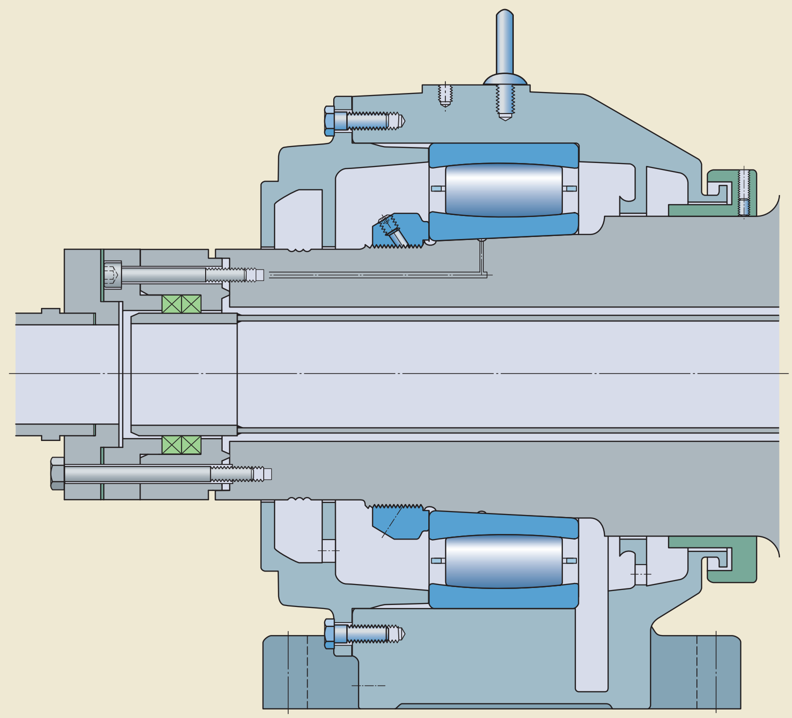
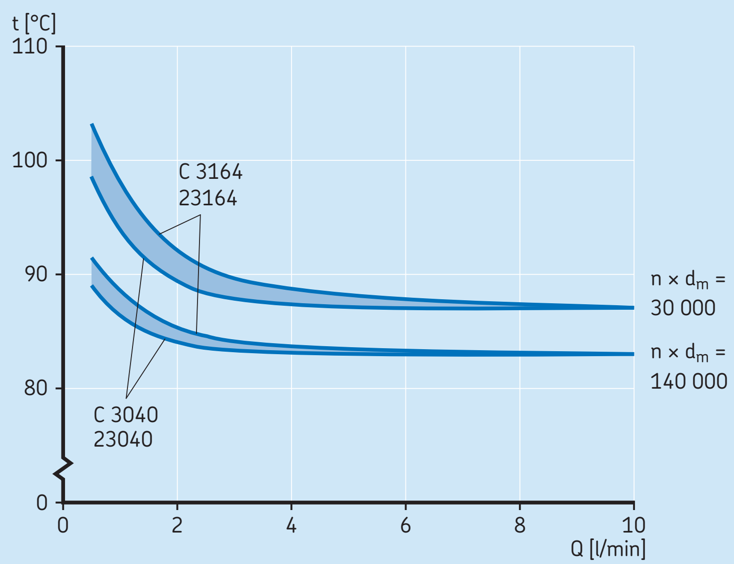


Diagram 8.16

Bearing temperature/oil flow diagram



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Example 17

Drying cylinder with steam temperature 165 °C, fig. 8.17

Lubrication guidelines

As drying cylinder bearings rotate at relatively low speeds and are subjected to high temperatures, there will be metallic contact between rollers and raceways. This means that there is a risk of surface distress. To avoid this risk, the circulating oil should have efficient AW additives and a viscosity ratio κ according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.17

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the inner ring temperature. The load at each speed corresponds to a bearing basic rating life of 200 000 hours.

An oil flow of 0,5 l/min, which is common in old machines, gives a bearing temperature of around 135 °C.

At operating temperatures above 100 °C, it is not possible to fulfil the κ guidelines and some mineral oils have shown a strong tendency to carbonize. Therefore, the recommendation for such applications is to improve the lubrication conditions as much as practically possible.

The diagram shows that the influence of the oil quantity is considerable up to 3 l/min, but after that scarcely anything is gained by an increased oil flow except for large size bearings running at high speeds.

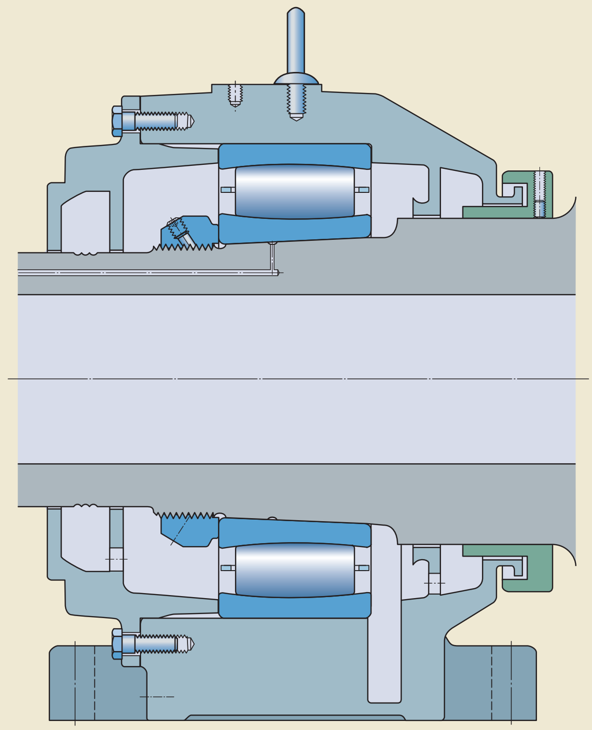
The best remedy is to introduce efficient journal insulation which gives a bearing temperature of about 90 °C. Changing to synthetic oil is also beneficial.

Table 8.17

Machine data	
Paper grades	board, liner, fine paper
Roll position	dryer section
Paper speed	400–1 200 m/min
Operating conditions for the bearings	
Ambient temperature	100 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	60 °C
Journal insulation	none

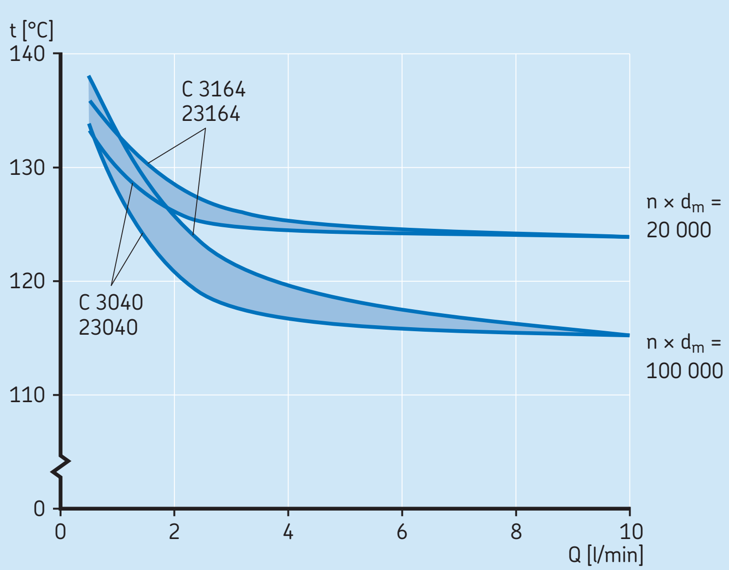
Bearing application

Fig. 8.17



Bearing temperature/oil flow diagram

Diagram 8.17



8

Example 18

Drying cylinder with steam temperature 165 °C, fig. 8.18

Lubrication guidelines

As drying cylinder bearings rotate at relatively low speeds and are subjected to high temperatures, there will be metallic contact between rollers and raceways. This means that there is a risk of surface distress. To avoid this risk, the circulating oil should have efficient AW additives and a viscosity ratio κ according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown on this page.

The calculations for this diagram have been made using an SKF computer program to obtain the inner ring temperature. The load at each speed corresponds to a bearing basic rating life of 200 000 hours.

An oil flow of 0,5 l/min was common in the early machines with insulated journals. Such an oil flow gives a bearing temperature of around 100 °C.

The aim for this application is to find the optimum oil flow which gives a bearing temperature of around 90 °C. The diagram shows that the influence of the oil quantity is considerable up to 2–3 l/min, but after that scarcely anything is gained by an increased oil flow except for large size bearings running at high speeds.

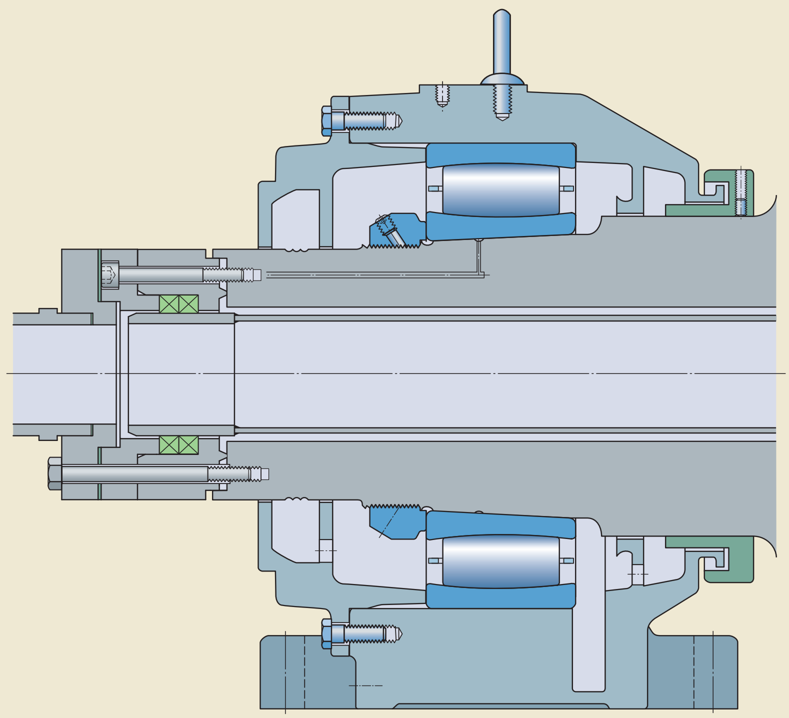
If ISO VG 320 oil is used instead, the bearing temperature will increase a little, but because the viscosity ratio κ will be higher than when using an ISO VG 220 oil, its use will be beneficial.

Table 8.18

Machine data	
Paper grades	board, liner, fine paper
Roll position	dryer section
Paper speed	400–1 200 m/min
Operating conditions for the bearings	
Ambient temperature	100 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	60 °C
Journal insulation	yes

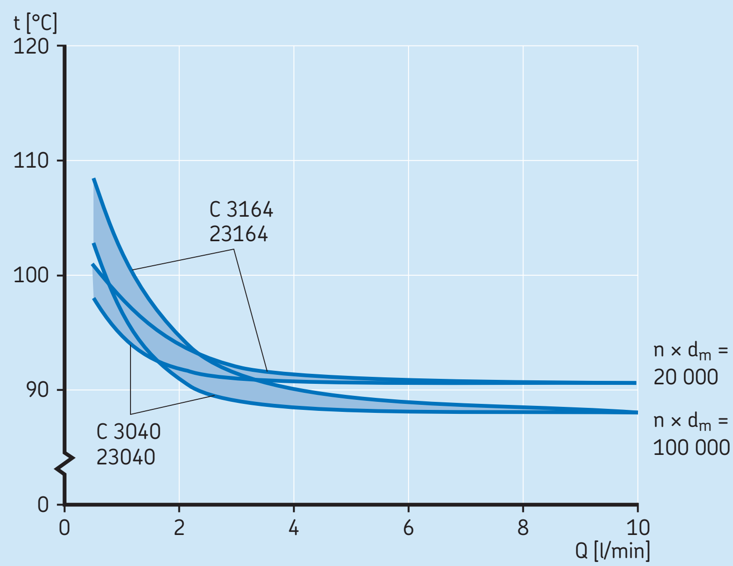
Bearing application

Fig. 8.18



Bearing temperature/oil flow diagram

Diagram 8.18



Example 19

Drying cylinder with steam temperature 190 °C, fig. 8.19

Lubrication guidelines

As drying cylinder bearings rotate at relatively low speeds and are subjected to high temperatures, there will be metallic contact between rollers and raceways. This means that there is a risk of surface distress. To avoid this risk, the circulating oil should have efficient AW additives and a viscosity ratio κ according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.19

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the inner ring temperature. The load at each speed corresponds to a bearing basic rating life of 200 000 hours.

An oil flow of 0,5 l/min, which is common in old machines, gives a bearing temperature of around 155 °C.

At operating temperatures above 100 °C, it is not possible to fulfil the κ guidelines and some mineral oils have shown a strong tendency to carbonize. Therefore, the recommendation for such applications is to improve the lubrication conditions as much as practically possible.

The diagram shows that the influence of the oil quantity is considerable up to 4 l/min, but after that scarcely anything is gained by an increased oil flow except for large size bearings running at high speeds.

The best remedy is to introduce efficient journal insulation which gives a bearing temperature of about 95–100 °C. Changing to synthetic oil is also beneficial.

Table 8.19

Machine data	
Paper grades	board, liner
Roll position	dryer section
Paper speed	400–1 200 m/min
Operating conditions for the bearings	
Ambient temperature	100 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	60 °C
Journal insulation	none

Fig. 8.19

Bearing application

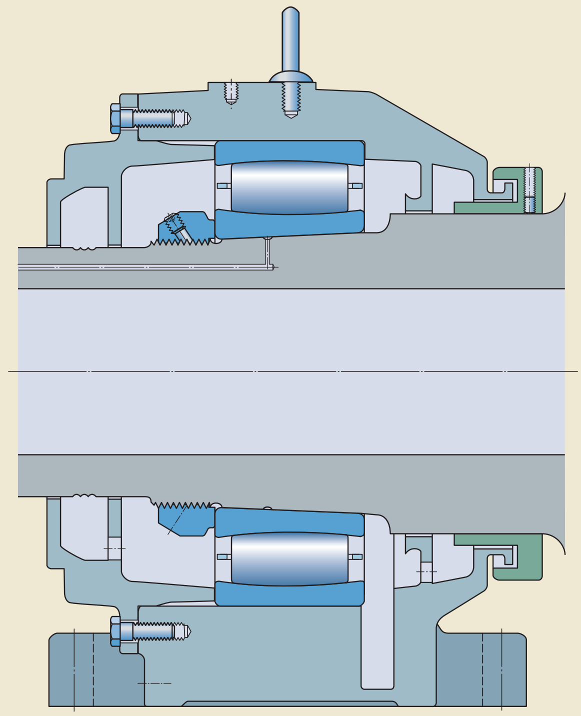
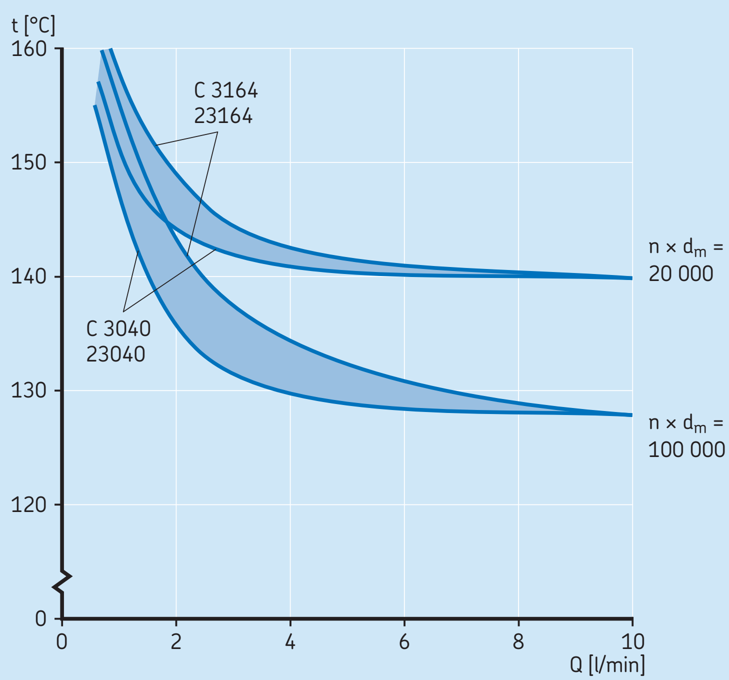


Diagram 8.19

Bearing temperature/oil flow diagram



Example 20

Drying cylinder with steam temperature 190 °C, fig. 8.20

Lubrication guidelines

As drying cylinder bearings rotate at relatively low speeds and are subjected to high temperatures, there will be metallic contact between rollers and raceways. This means that there is a risk of surface distress. To avoid this risk, the circulating oil should have efficient AW additives and a viscosity ratio κ according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.20

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example. The calculations for this diagram have been made using an SKF computer program to obtain the inner ring temperature.

The load at each speed corresponds to a bearing basic rating life of 200 000 hours.

An oil flow of 0,5 l/min was common in the early machines with insulated journals. Such an oil flow gives a bearing temperature of around 115 °C. The aim for this application is to find the optimum oil flow which gives a bearing temperature of around 95–100 °C.

The diagram shows that the influence of the oil quantity is considerable up to 3–4 l/min, but after that scarcely anything is gained by an increased oil flow except for large size bearings at high speeds.

If ISO VG 320 oil is used instead, the bearing temperature will increase a little, but because the viscosity ratio κ would be higher than when using an ISO VG 220 oil, its use will be beneficial.

Table 8.20

Machine data	
Paper grades	board, liner
Roll position	dryer section
Paper speed	400–1 200 m/min
Operating conditions for the bearings	
Ambient temperature	100 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	60 °C
Journal insulation	yes

Fig. 8.20

Bearing application

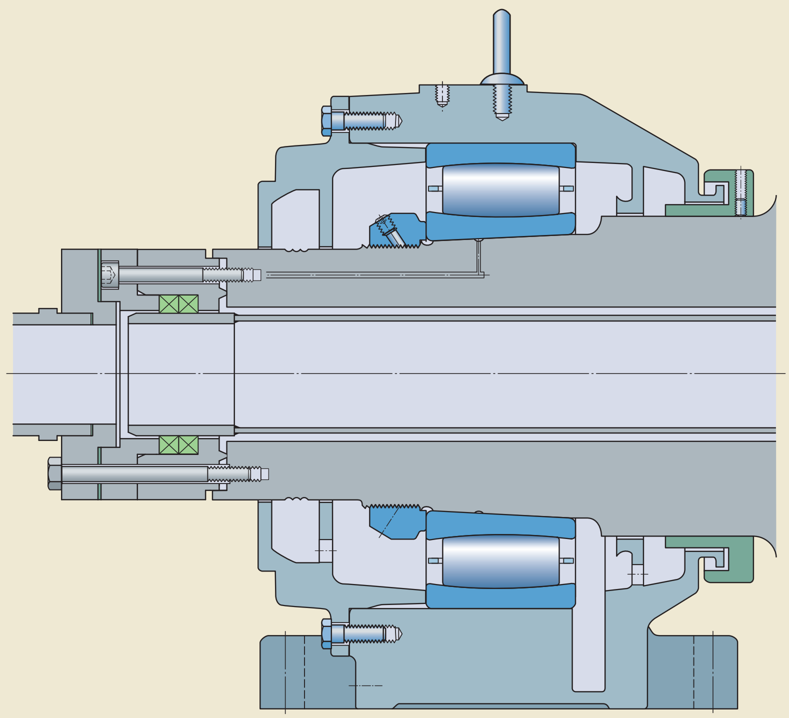
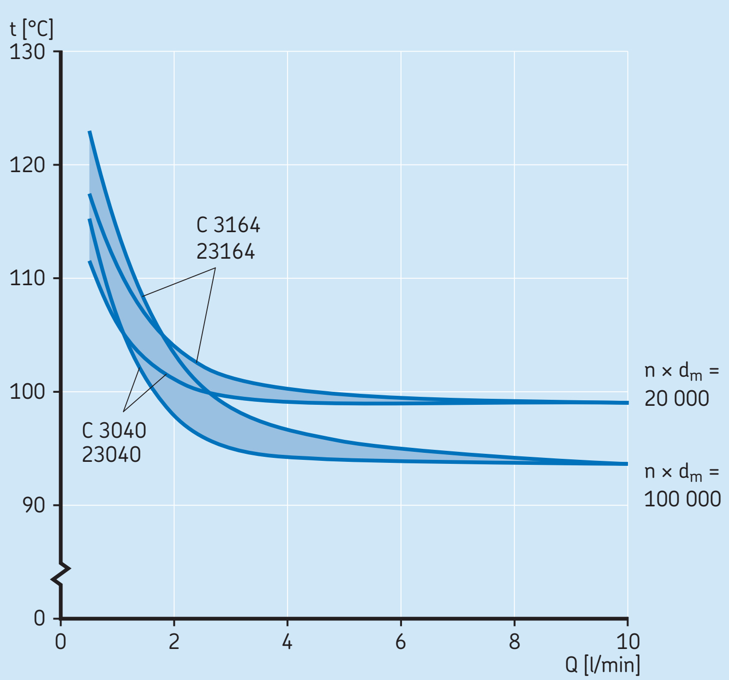


Diagram 8.20

Bearing temperature/oil flow diagram



Example 21

Yankee suction press roll (oil), fig. 8.21

Lubrication guidelines

As suction roll bearings are large and sometimes rotate at very high speeds, there is a risk of smearing. To avoid smearing, the oil should have EP additives and the viscosity ratio κ should be according to the guidelines in *Chapter 10, Bearing damage and failure, How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7, Lubrication, Oil lubrication*, should also be taken into account.

Comments on the diagram 8.21

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the oil flow in the diagram is valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the outer ring temperature. The load at each speed corresponds to a bearing basic rating life of 120 000 hours.

The oil flow in the diagram is based on the use of ISO VG 220 oil and selected to fulfil two criteria: the κ guidelines and a bearing operating temperature of 75 °C. If oil with lower viscosity than ISO VG 220 is used for these bearings, then the oil flows must be increased. Thicker oil, e.g. ISO VG 320, is advantageous and will improve the lubrication conditions, but the bearing temperature will increase a little.

Note that the diagram shows minimum acceptable oil flow. Slightly higher oil flow values are acceptable and beneficial in most cases.

Table 8.21

Machine data	
Paper grades	tissue
Roll position	dryer section
Paper speed	400–2 200 m/min
Operating conditions for the bearings	
Ambient temperature	45 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	45 °C

Fig. 8.21

Bearing application

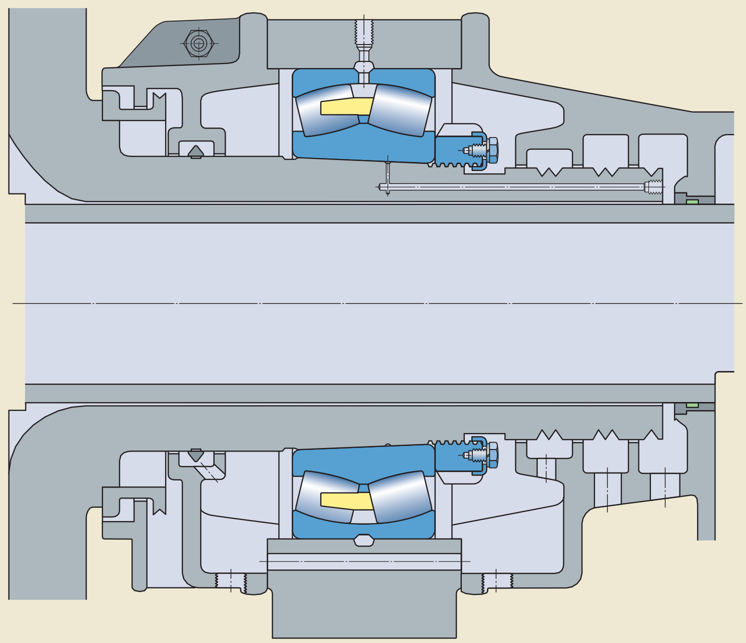
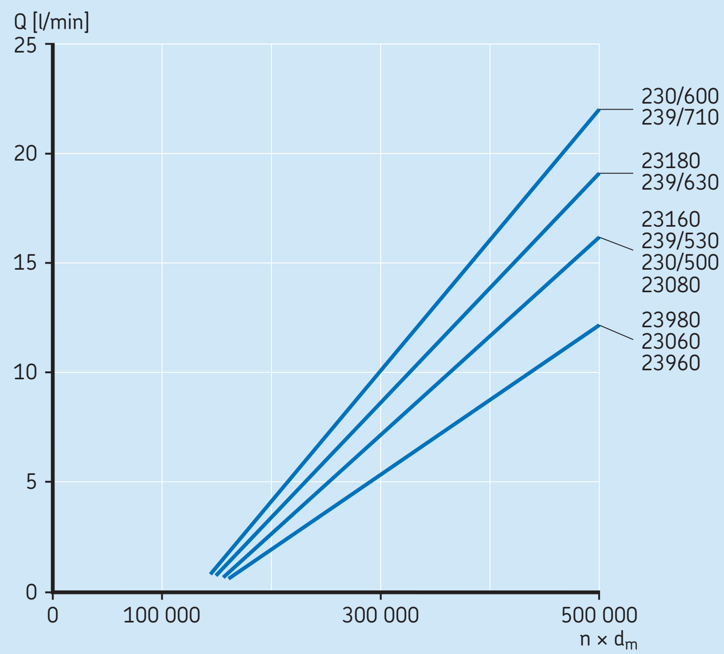


Diagram 8.21

Oil flows for bearing temperature 75°C



Example 22

Yankee cylinder with steam temperature 140 °C (oil bath), fig. 8.22

Lubrication guidelines

As Yankee cylinder bearings in this example rotate at very low speeds and are subjected to high temperatures, there will be metallic contact between rollers and raceways. This means that there is a risk of surface distress. To avoid this risk, the lubricating oil should have efficient AW additives and a viscosity ratio κ according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

A synthetic oil is recommended because mineral oils are not suitable at temperatures above 120 °C.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account

Comments on the diagram 8.22

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the inner ring temperature. The load at each speed corresponds to a bearing basic rating life of 200 000 hours. Note that the bearing temperature is lower at high speeds than at low speeds. The reason is that the increase in cooling via the rotating journal is higher than the increase in heat generation in the bearing.

At operating temperatures above 100 °C, it is not possible to fulfil the κ guidelines with commonly used ISO VG 220 oils.

The recommendation for this application is to improve the lubrication conditions as much as practically possible. The best remedy is to introduce efficient journal insulation in combination with circulating oil lubrication. This results in a bearing temperature of 85 °C. Changing to oil with higher viscosity than used for circulation systems is also beneficial.

Table 8.22

Machine data	
Paper grades	board (old machines)
Roll position	dryer section
Paper speed	100–150 m/min
Operating conditions for the bearings	
Ambient temperature	60 °C
Lubrication	oil bath
Oil viscosity	ISO VG 220–1500
Journal insulation	none

Fig. 8.22

Bearing application

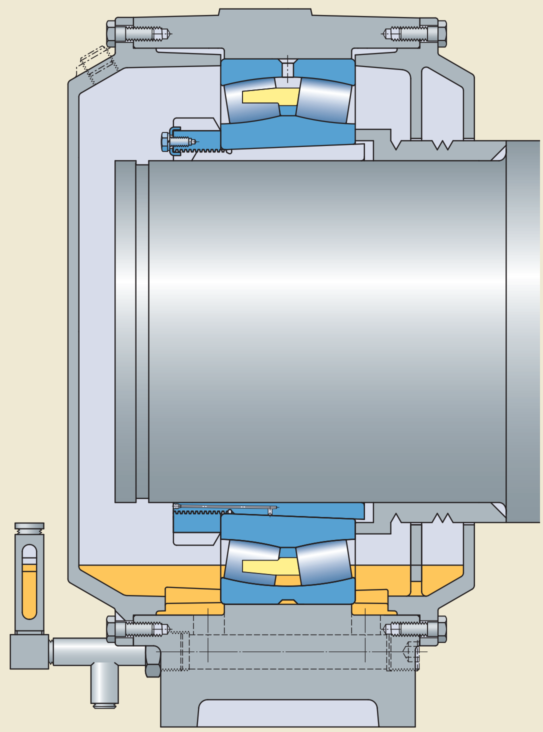
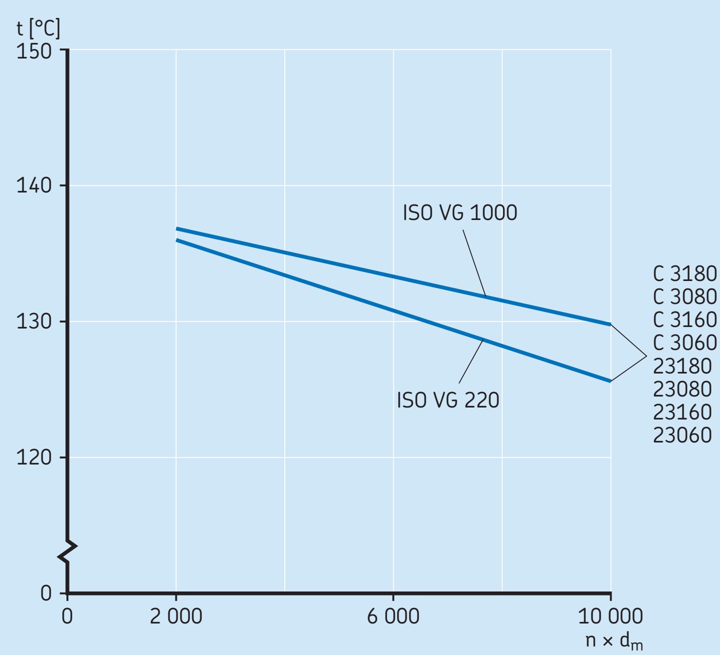


Diagram 8.22

Bearing temperature/speed factor diagram



Example 23

Yankee cylinder with steam temperature 140 °C, fig. 8.23

Lubrication guidelines

As Yankee cylinder bearings rotate at low speeds and are subjected to high temperatures there will be metallic contact between rollers and raceways. This means that there is a risk of surface distress. To avoid this risk, the circulating oil should have efficient AW additives and a viscosity ratio κ according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.23

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram are made at SKF with a computer program that indicates the inner ring temperature. The load at each speed corresponds to a bearing basic rating life of 200 000 hours.

The bearing operating temperature will be above 110 °C also at very high oil flows. At operating temperatures above 100 °C, it is not possible to fulfil the κ guidelines and some mineral oils have shown a strong tendency to carbonize. Therefore, the recommendation for such applications is to improve the lubrication conditions as much as practically possible.

The diagram shows that the influence of the oil quantity is considerable up to 4–5 l/min, but after that scarcely anything is gained by an increased oil flow except for large size bearings running at high speeds.

The best remedy is to introduce efficient journal insulation which results in a bearing temperature of about 80–85 °C. Changing to synthetic oil is also beneficial.

Table 8.23

Machine data	
Paper grades	board, tissue
Roll position	dryer section
Paper speed	100–1 000 m/min
Operating conditions for the bearings	
Ambient temperature	60 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 320 or 460
Oil inlet temperature	45 °C
Journal insulation	none

Fig. 8.23

Bearing application

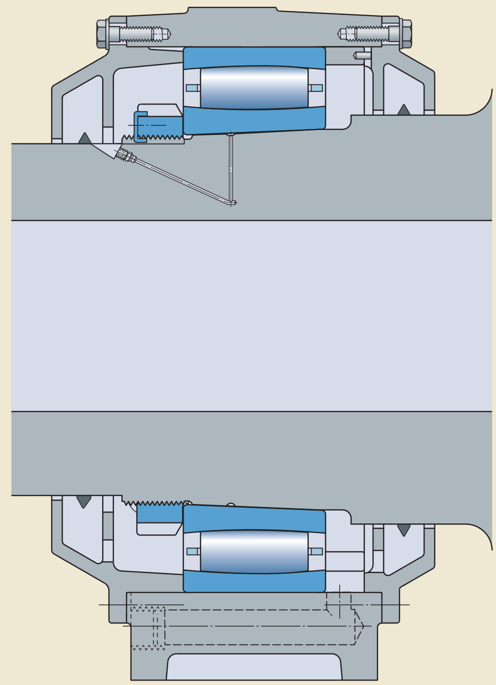
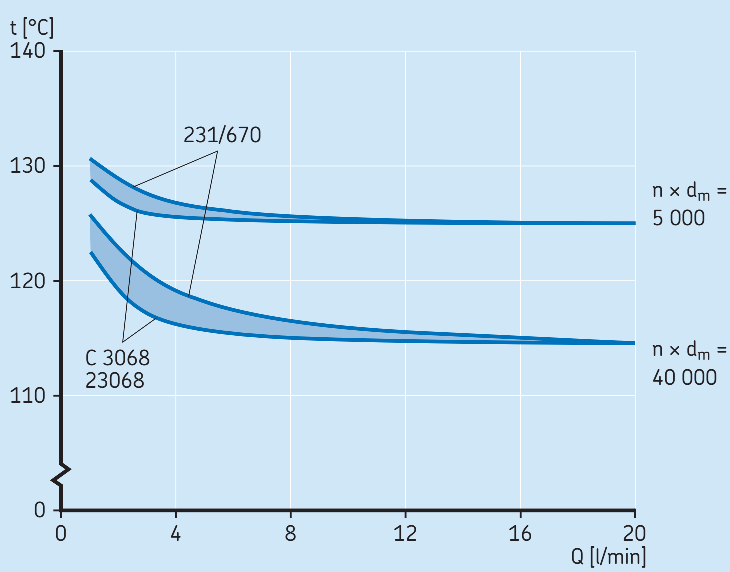


Diagram 8.23

Bearing temperature/oil flow diagram



Example 24

Yankee cylinder with steam temperature 140 °C, fig. 8.24

Lubrication guidelines

As Yankee cylinder bearings rotate at relatively low speeds and are subjected to high temperatures, there will be metallic contact between rollers and raceways. This means that there is a risk of surface distress. To avoid this the circulating oil should have efficient AW additives and a viscosity ratio κ according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.24

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the inner ring temperature. The load at each speed corresponds to a bearing basic rating life of 200 000 hours.

An oil flow of 1–2 l/min was sometimes used in the early machines with insulated journals. Such an oil flow gives a bearing temperature of around 90 °C.

The aim for this application is to find the optimum oil flow which gives a bearing temperature of around 80–85 °C.

The diagram shows that the influence of the oil quantity is considerable up to 4–6 l/min, but after that scarcely anything is gained by an increased oil flow except for large size bearings running at high speeds.

Table 8.24

Machine data	
Paper grades	board, tissue
Roll position	dryer section
Paper speed	400–1 000 m/min
Operating conditions for the bearings	
Ambient temperature	60 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 320 or 460
Oil inlet temperature	45 °C
Journal insulation	yes

Fig. 8.24

Bearing application

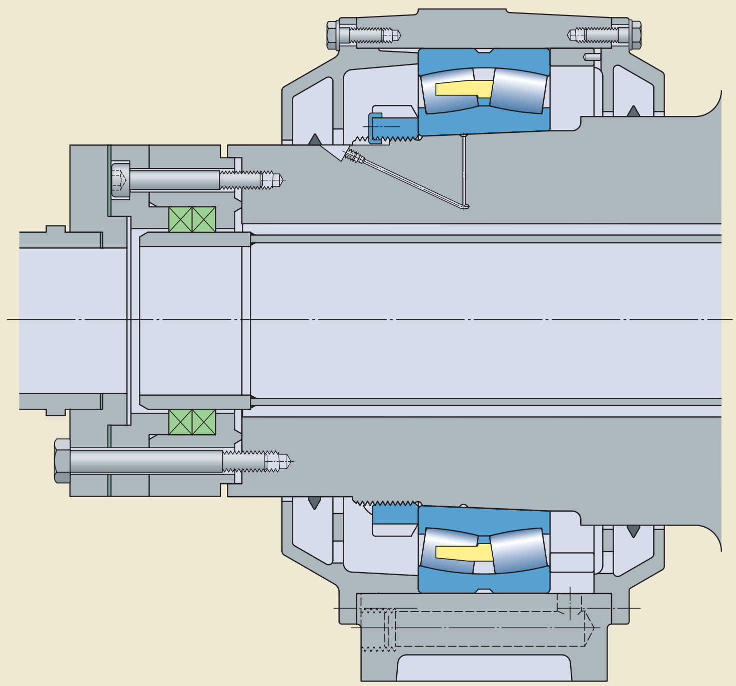
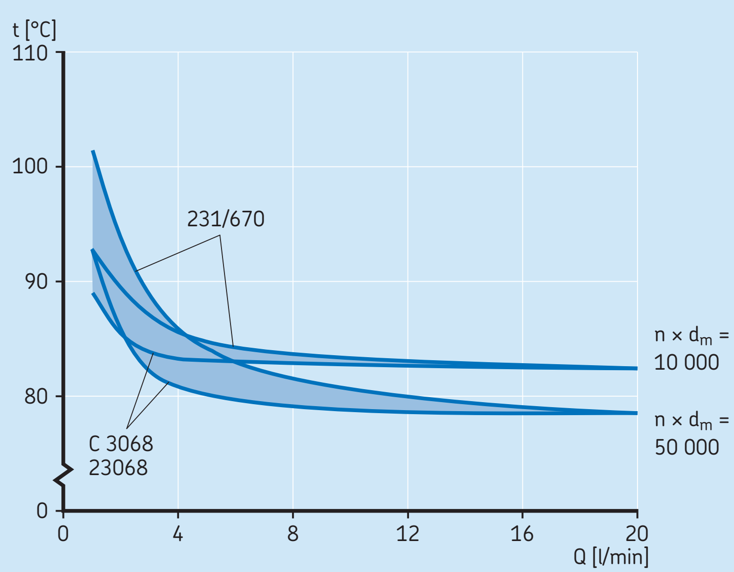


Diagram 8.24

Bearing temperature/oil flow diagram



Example 25

Yankee cylinder with steam temperature 165 °C, fig. 8.25

Lubrication guidelines

As Yankee cylinder bearings rotate at relatively low speeds and are subjected to high temperatures, there will be metallic contact between rollers and raceways. This means that there is a risk of surface distress. To avoid this risk, the circulating oil should have efficient AW additives and a viscosity ratio κ according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.25

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the inner ring temperature. The load at each speed corresponds to a bearing basic rating life of 200 000 hours.

An oil flow of 2 l/min, which is sometimes used for old machines, gives a bearing temperature of around 135 °C.

At operating temperatures above 100 °C, it is not possible to fulfil the κ guidelines and some mineral oils have shown a strong tendency to carbonize. Therefore, the recommendation for such applications is to improve the lubrication conditions as much as practically possible.

The diagram shows that the influence of the oil quantity is considerable up to 5–7 l/min, but after that scarcely anything is gained by an increased oil flow except for large size bearings running at high speeds.

The best remedy is to introduce efficient journal insulation which results in a bearing temperature of about 90 °C. Changing to synthetic oil is also beneficial.

Table 8.25

Machine data	
Paper grades	tissue, board
Roll position	dryer section
Paper speed	400–1 500 m/min
Operating conditions for the bearings	
Ambient temperature	60 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 320 or 460
Oil inlet temperature	45 °C
Journal insulation	none

Fig. 8.25

Bearing application

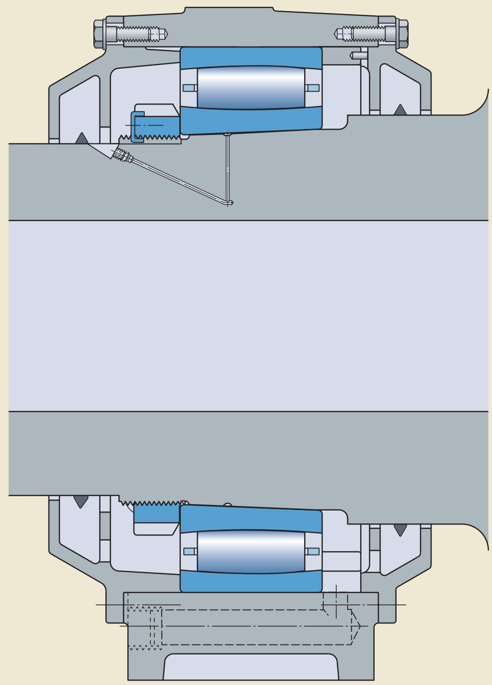
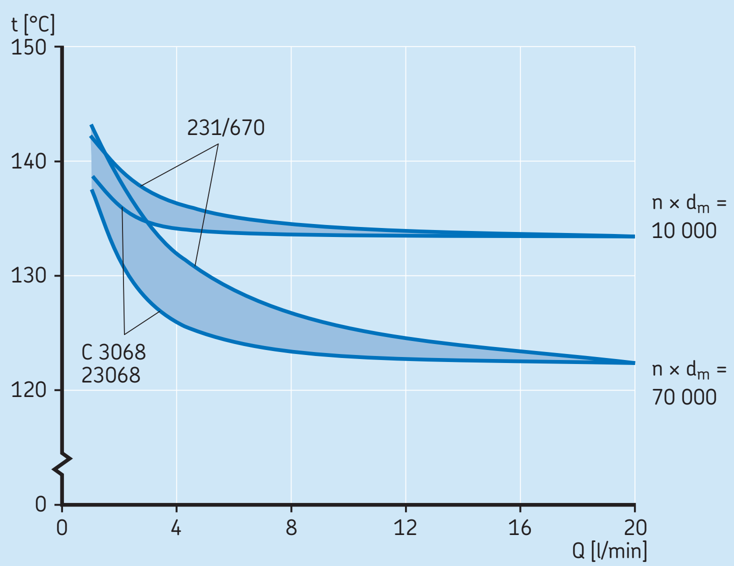


Diagram 8.25

Bearing temperature/oil flow diagram



Example 26

Yankee cylinder with steam temperature 165 °C, fig. 8.26

Lubrication guidelines

As Yankee cylinder bearings rotate at relatively low speeds and are subjected to high temperatures, there will be metallic contact between rollers and raceways. This means that there is a risk of surface distress. To avoid this risk, the circulating oil should have efficient AW additives and a viscosity ratio κ according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.26

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the inner ring temperature. The load at each speed corresponds to a bearing basic rating life of 200 000 hours.

An oil flow of 2 l/min was sometimes used in the early machines with insulated journals. Such an oil flow gives a bearing temperature of around 100 °C. If the bearing operating temperature is above 100 °C, it is impossible to fulfil the κ guidelines. The aim for this application is to find the optimum oil flow which gives a bearing temperature below 90 °C.

The diagram shows that the influence of the oil quantity is considerable up to 5–7 l/min, but after that scarcely anything is gained by an increased oil flow except for large size bearings running at high speeds.

Table 8.26

Machine data	
Paper grades	tissue, board
Roll position	dryer section
Paper speed	400–2 200 m/min
Operating conditions for the bearings	
Ambient temperature	60 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 320 or 460
Oil inlet temperature	45 °C
Journal insulation	yes

Fig. 8.26

Bearing application

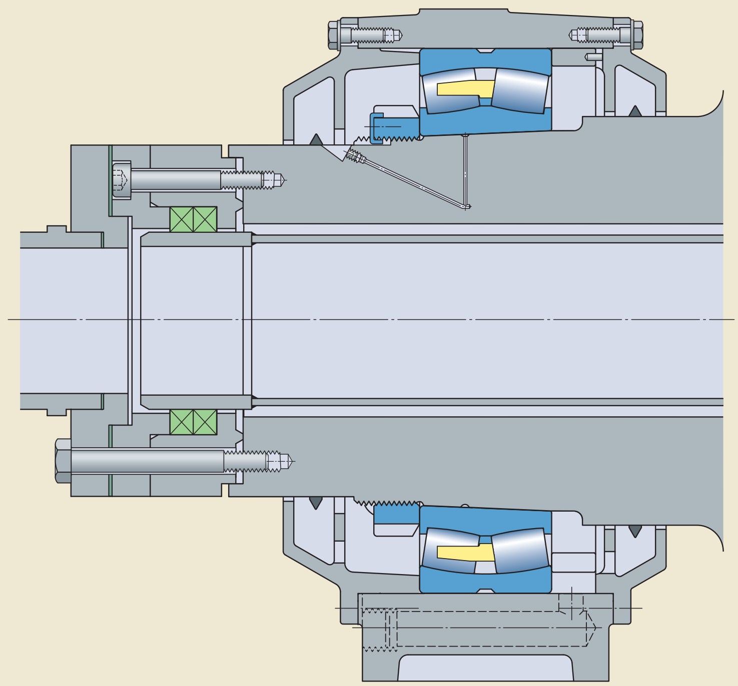
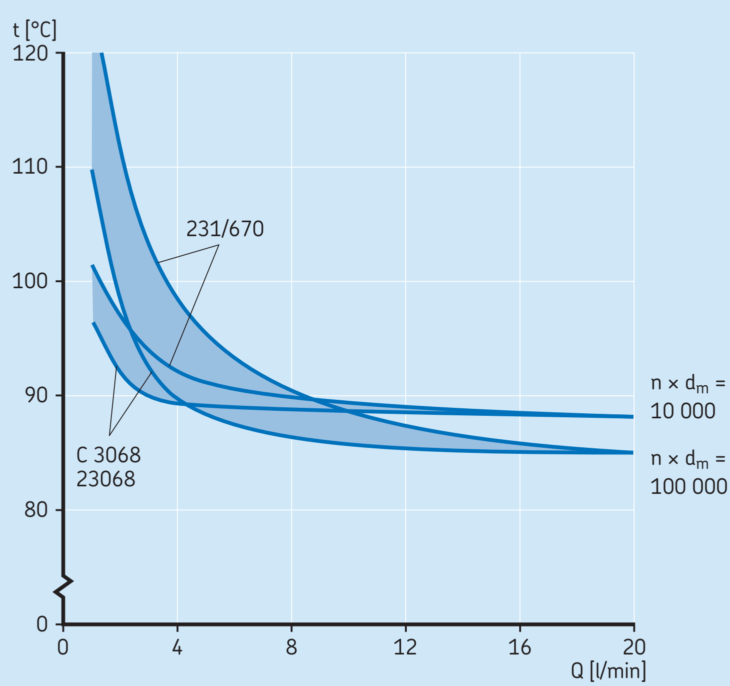


Diagram 8.26

Bearing temperature/oil flow diagram



8

Example 27

Yankee cylinder with steam temperature 190 °C, fig. 8.27

Lubrication guidelines

As Yankee cylinder bearings rotate at relatively low speeds and are subjected to high temperatures, there will be metallic contact between rollers and raceways. This means that there is a risk of surface distress. To avoid this risk, the circulating oil should have efficient AW additives and a viscosity ratio κ according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.27

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the inner ring temperature. The load at each speed corresponds to a bearing basic rating life of 200 000 hours.

An oil flow of 2–3 l/min, which is common in old machines, gives a bearing temperature of around 155 °C.

At operating temperatures above 100 °C, it is not possible to fulfil the κ guidelines and some mineral oils have shown a strong tendency to carbonize. Therefore, the recommendation for such applications is to improve the lubrication conditions as much as practically possible.

The diagram shows that the influence of the oil quantity is considerable up to 6–8 l/min, but after that scarcely anything is gained by an increased oil flow except for large size bearings running at high speeds.

The best remedy is to introduce efficient journal insulation which results in a bearing temperature of about 95 °C. Changing to synthetic oil is also beneficial.

Table 8.27

Machine data	
Paper grades	tissue, board
Roll position	dryer section
Paper speed	400–1 500 m/min
Operating conditions for the bearings	
Ambient temperature	60 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 320 or 460
Oil inlet temperature	45 °C
Journal insulation	none

Fig. 8.27

Bearing application

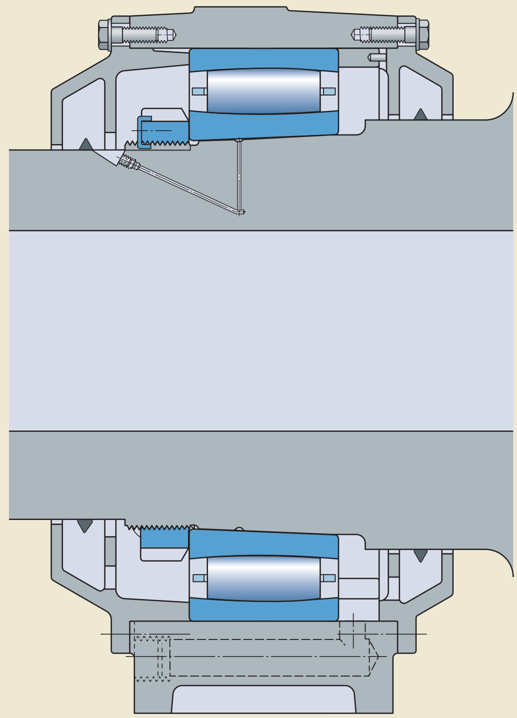
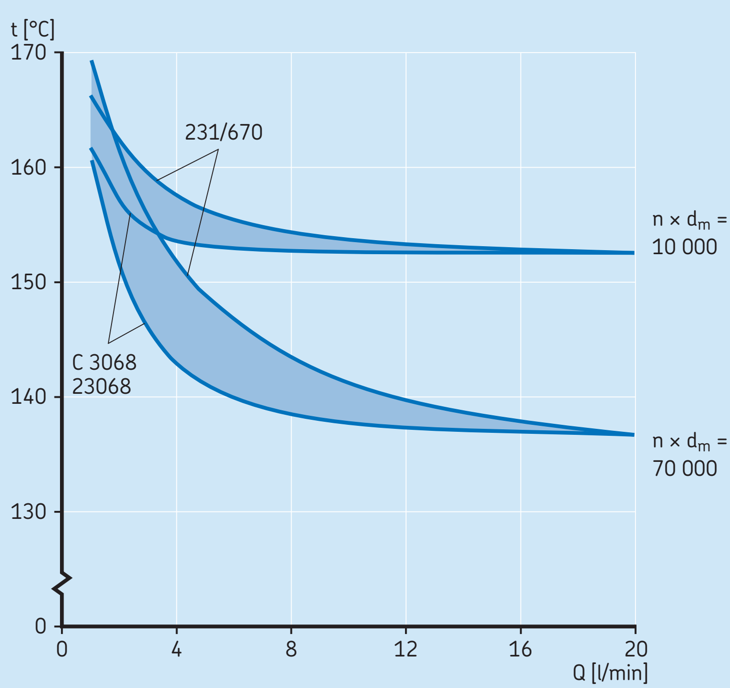


Diagram 8.27

Bearing temperature/oil flow diagram



Example 28

Yankee cylinder with steam temperature 190 °C, fig. 8.28

Lubrication guidelines

As Yankee cylinder bearings rotate at relatively low speeds and are subjected to high temperatures, there will be metallic contact between rollers and raceways. This means that there is a risk of surface distress. To avoid this risk, the circulating oil should have efficient AW additives and a viscosity ratio κ according to the guidelines in *10, Bearing damage and failure, How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7, Lubrication, Oil lubrication*, should also be taken into account.

Comments on the diagram 8.28

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the inner ring temperature. The load at each speed corresponds to a bearing basic rating life of 200 000 hours.

An oil flow of 2–3 l/min was common in the early machines with insulated journals. Such an oil flow gives a bearing temperature of around 110 °C. If the bearing operating temperature is above 100 °C, it is impossible to fulfil the κ guidelines. The aim for this application is to find the optimum oil flow which results in a bearing temperature of 90–95 °C.

The diagram shows that the influence of the oil quantity is considerable up to 6–8 l/min, but after that scarcely anything is gained by an increased oil flow except for large size bearings running at high speeds.

Table 8.28

Machine data	
Paper grades	tissue, board
Roll position	dryer section
Paper speed	400–2 200 m/min
Operating conditions for the bearings	
Ambient temperature	60 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 320 or 460
Oil inlet temperature	45 °C
Journal insulation	yes

Fig. 8.28

Bearing application

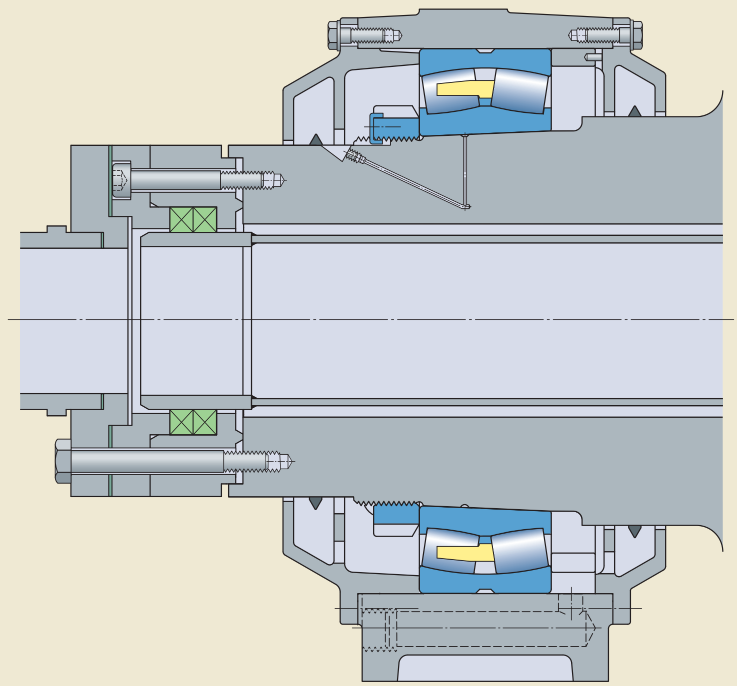
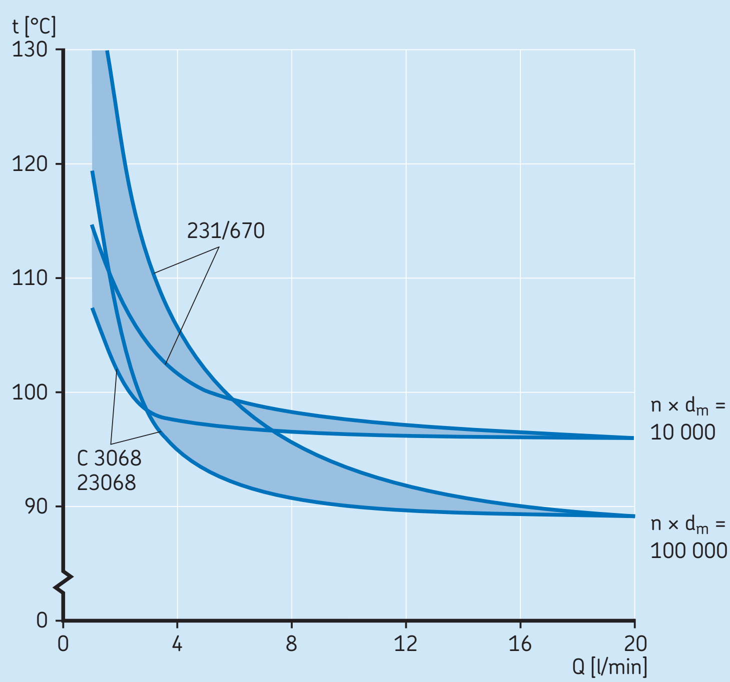


Diagram 8.28

Bearing temperature/oil flow diagram



8

Example 29

Paper guide roll (grease), fig. 8.29

Lubrication guidelines

SKF's experience is that grease with a minimum base oil viscosity 175 mm^2/s provides good lubrication performance in this position. Operating conditions for grease lubricated bearings in the forming section are more severe than in this example. Therefore, the grease selected for the forming section can also be used here.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Grease lubrication*, should also be taken into account.

Comments on the diagram 8.29

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the calculated bearing temperatures in this example are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the outer ring temperature at different speeds. The load at each speed corresponds to a bearing basic rating life of 120 000 hours.

The highest bearing temperature shown in the diagram is 75 °C as it is customary to use oil lubrication at higher temperatures.

Table 8.29

Machine data	
Paper grades	board, liner, fine paper, newsprint
Roll position	after dryer section
Paper speed	100–1 000 m/min
Operating conditions for the bearings	
Ambient temperature	45 °C
Lubrication	grease
Base oil viscosity	175 mm^2/s at 40 °C

Fig. 8.29

Bearing application

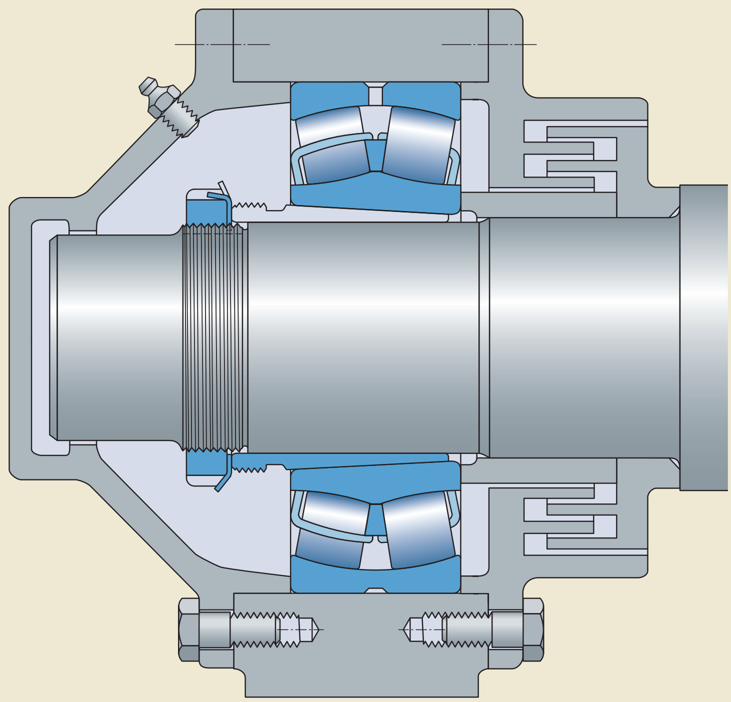
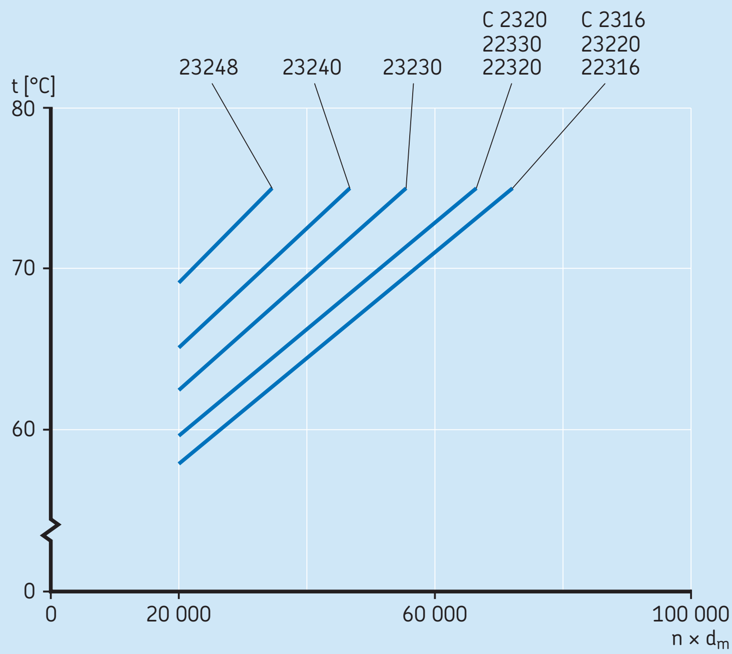


Diagram 8.29

Bearing temperature/speed factor diagram



Example 30

Paper guide roll (oil), fig. 8.30

Lubrication guidelines

Paper guide roll bearings have better operating conditions than drying cylinder bearings which are lubricated with the same circulating oil system. Therefore, the lubricant properties should be based on the operating conditions for drying cylinder bearings. However, the paper guide roll bearings should have an oil flow resulting in a viscosity ratio κ according to the guidelines in *Chapter 10, Bearing damage and failure, How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7, Lubrication, Oil lubrication*, should also be taken into account.

Comments on the diagram 8.30

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the outer ring temperature. The load at each speed corresponds to a bearing basic rating life of 200 000 hours. The oil flow in the diagram is based on the use of ISO VG 220 oil and selected to fulfil two criteria: the κ guidelines and a bearing operating temperature of 75 °C.

If oil with lower viscosity than ISO VG 220 is used, the flow must be increased in order to reach the same κ value. Thicker oil, e.g. ISO VG 320, is advantageous and will improve the lubrication conditions, but the bearing temperature will increase a little.

Note that the diagram shows minimum acceptable oil flow. Slightly higher oil flow values are acceptable and beneficial in most cases.

Table 8.30

Machine data	
Paper grades	all
Roll position	after dryer section
Paper speed	400–2 200 m/min
Operating conditions for the bearings	
Ambient temperature	45 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	50 °C

Fig. 8.30

Bearing application

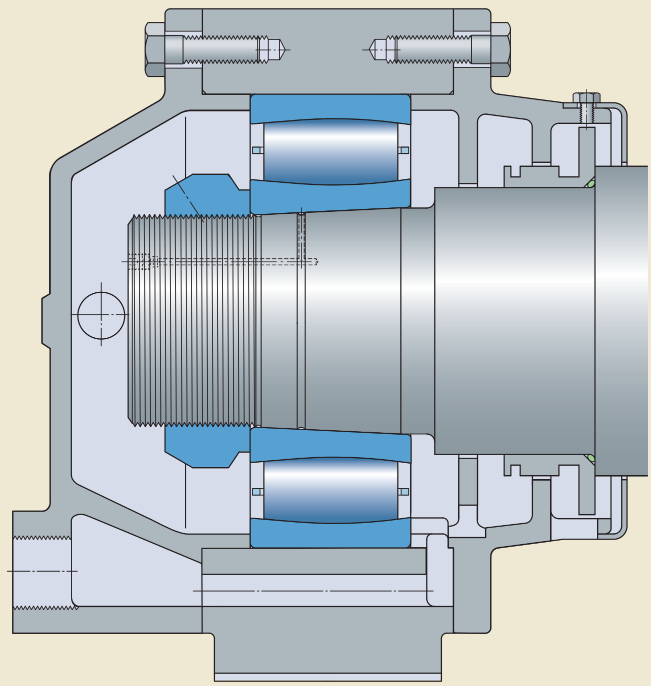
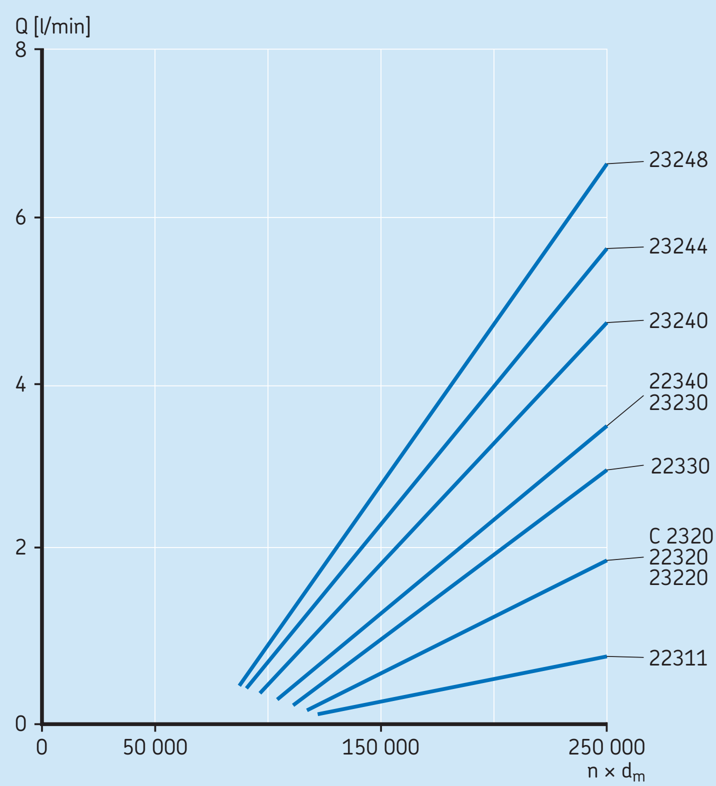


Diagram 8.30

Oil flows for bearing temperature 75 °C



Example 31

Calender roll, plain, unheated (oil bath), fig. 8.31

Lubrication guidelines

The bearings in this example have relatively good lubrication conditions. However, bearings running at low speeds will have metal-to-metal contact between rollers and raceways. This means that there is a risk of surface distress. To avoid this risk, the lubricating oil should have efficient AW additives and a viscosity ratio κ according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.31

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the outer ring temperature. The load at each speed corresponds to a bearing basic rating life of 120 000 hours.

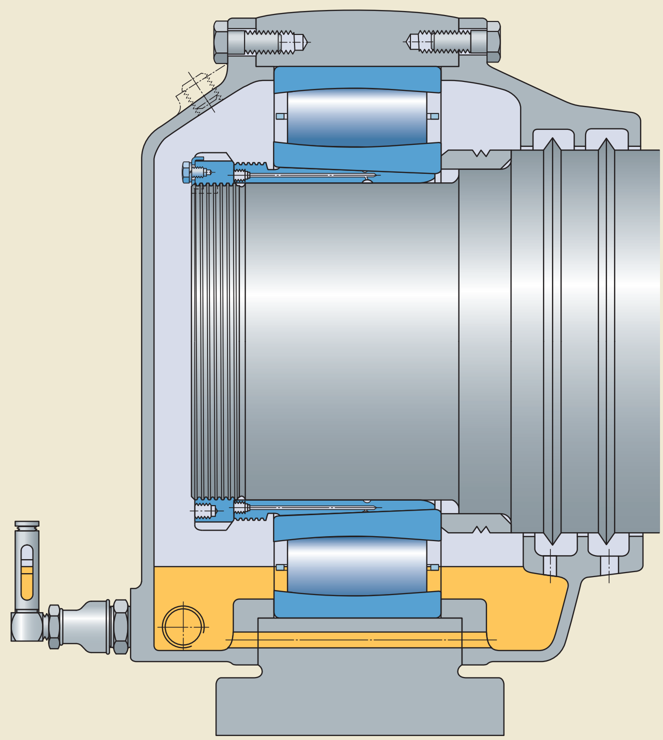
In this example, oil bath lubrication fulfils the SKF lubrication guidelines regarding the viscosity ratio κ but not the guidelines for lubricant cleanliness. Therefore, the recommendation for this application is to improve the lubricant cleanliness.

Oil cleanliness can be improved by various means e.g. by improved seals for the bearing housing, decreased intervals between oil changes or the introduction of an oil circulation system with filters and water extractors. However, the cost for these can sometimes be higher than the gains from increased machine availability and decreased bearing consumption.

Table 8.31

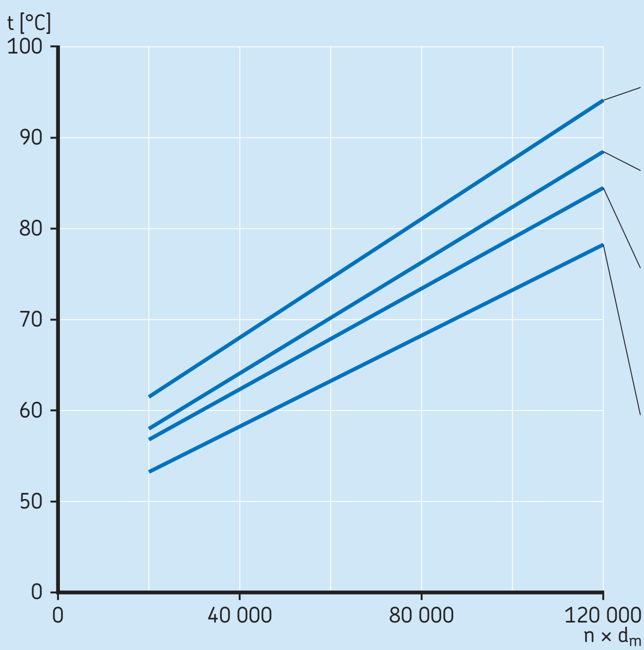
Machine data	
Paper grades	board, fine paper
Roll position	machine calender (old machines)
Paper speed	100–700 m/min
Operating conditions for the bearings	
Ambient temperature	45 °C
Lubrication	oil bath
Oil viscosity	ISO VG 220

Fig. 8.31



Bearing application

Diagram 8.31



Bearing temperature/speed factor diagram

Example 32

Calender roll, plain, unheated (oil), fig. 8.32

Lubrication guidelines

The guidelines below are based on a large size bearing rotating at high speed because these bearings have the most demanding operating conditions.

Large bearings rotating at high speed are subjected to a risk of smearing. To avoid smearing, the oil should have EP additives and the viscosity ratio κ should be according to the guidelines in *Chapter 10, Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.32

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the oil flow in the diagram is valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the outer ring temperature. The load at each speed corresponds to a bearing basic rating life of 120 000 hours.

The oil flow in the diagram is based on the use of ISO VG 220 oil and selected to fulfil two criteria: the κ guidelines and a bearing operating temperature of 75 °C. If oil with lower viscosity than ISO VG 220 is used for these bearings, the oil flows must be increased.

Oil with higher viscosity is advantageous and will improve the lubrication conditions, but the bearing temperature will increase a little. Note that the diagram shows minimum acceptable oil flow. Slightly higher oil flow values are acceptable and beneficial in most cases.

Table 8.32

Machine data	
Paper grades	board, liner, fine paper, newsprint
Roll position	machine calender
Paper speed	400–1 800 m/min
Operating conditions for the bearings	
Ambient temperature	45 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	50 °C

Fig. 8.32

Bearing application

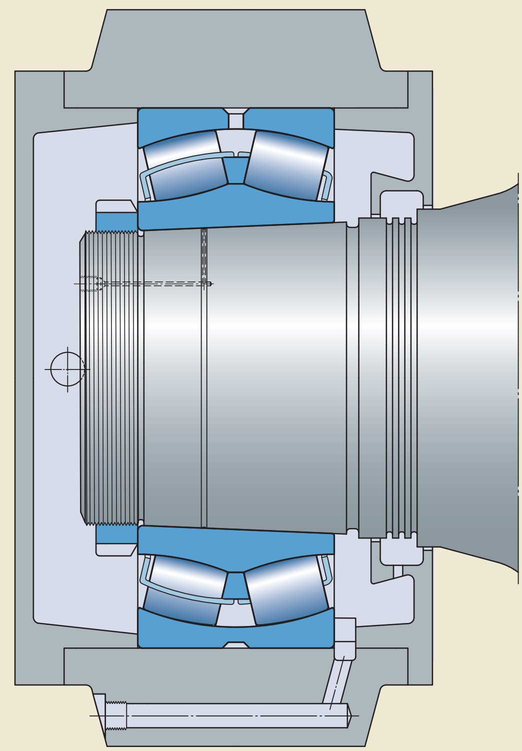
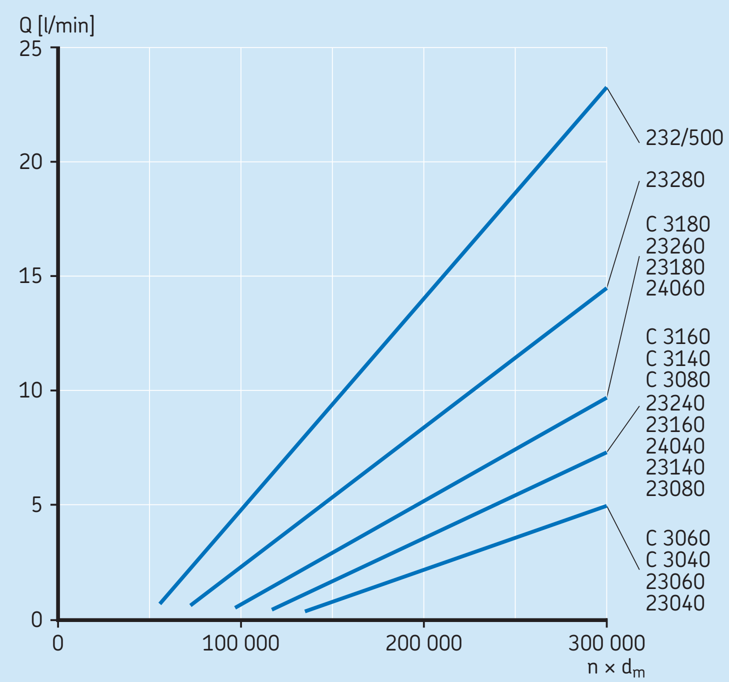


Diagram 8.32

Oil flows for bearing temperature 75 °C



Example 33

Calender roll with steam temperature 140 °C, fig. 8.33

Lubrication guidelines

As large calender roll bearings sometimes rotate at high speeds there is a risk of smearing. To avoid smearing, SKF usually recommends the use of an oil with efficient EP additives and oil flows resulting in a viscosity ratio κ according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*. However, with the above operating conditions, the bearing temperature will be too high for EP additives and accordingly, oils with efficient AW additives have to be used.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.33

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Therefore, the calculated bearing temperatures in this example are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the inner ring temperature. The load at each speed corresponds to a bearing basic rating life of 120 000 hours.

At high speeds, it is not possible to fulfil the κ guidelines and, accordingly, the best possible lubrication conditions should be aimed for, especially since EP additives cannot be used.

At low speeds, the diagram shows that the influence of the oil quantity on bearing temperature is considerable up to 2–4 l/min, but after that scarcely anything is gained by an increased oil flow. For large size bearings rotating at high speeds, the diagram shows that the influence of the oil quantity is considerable up to about 8 l/min.

Oil with higher viscosity is beneficial and will improve the lubrication conditions, but the bearing temperature will increase a little.

Table 8.33

Machine data	
Paper grades	board, liner, fine paper, newsprint
Roll position	machine calender
Paper speed	400–1 800 m/min
Operating conditions for the bearings	
Ambient temperature	45 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	50 °C
Journal insulation	bore and end face

Fig. 8.33

Bearing application

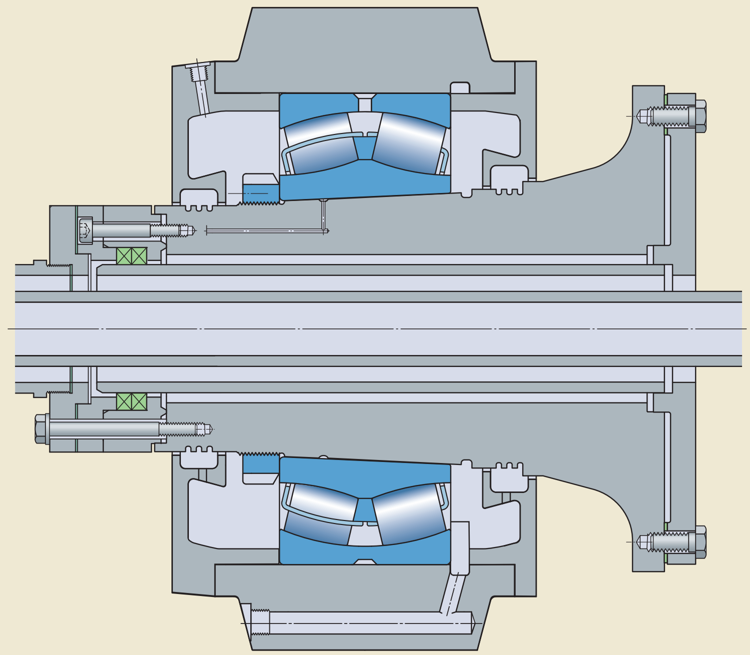
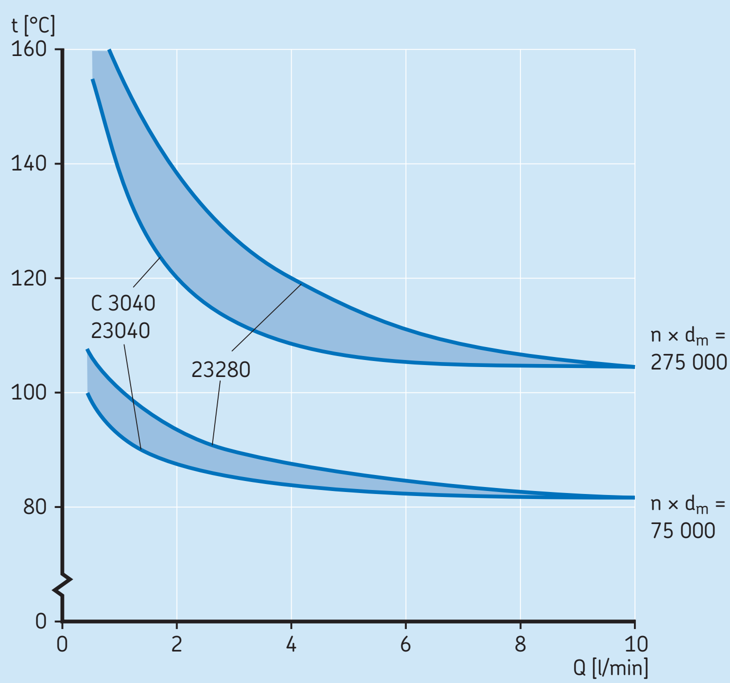


Diagram 8.33

Bearing temperature/oil flow diagram



8

Example 34

Calender roll with steam/oil temperature 175 °C, fig. 8.34

Lubrication guidelines

As large calender roll bearings sometimes rotate at high speeds, there is a risk of smearing. To avoid smearing, SKF usually recommends the use of an oil with efficient EP additives and oil flows giving a viscosity ratio κ according to the guidelines in *Chapter 10, Bearing damage and failure, How to avoid raceway and roller surface damage*. However, with the above operating conditions, the bearing temperature will be too high for EP additives and, accordingly, oils with efficient AW additives have to be used.

The general guidelines, as outlined in *Chapter 7, Lubrication, Oil lubrication*, should also be taken into account.

Comments on the diagram 8.34

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Therefore, the calculated bearing temperatures in this example are valid only when the bearing application and operating conditions are similar to those indicated here.

The calculations for this diagram have been made using an SKF computer program to obtain the inner ring temperature. The load at each speed corresponds to a bearing basic rating life of 120 000 hours.

At high speeds, it is not possible to fulfil the κ guidelines in *Chapter 10, Bearing damage and failure, How to avoid raceway and roller surface damage* and, accordingly, the best possible lubrication conditions should be aimed for, especially since EP additives cannot be used.

At low speeds, the diagram shows that the influence of the oil quantity on bearing temperature is considerable up to 3–6 l/min, but after that scarcely anything is gained by an increased oil flow. For large size bearings rotating at high speeds, the diagram shows that the influence of the oil quantity is considerable up to about 12 l/min.

Oil with higher viscosity is beneficial and will improve the lubrication conditions, but the bearing temperature will increase a little.

Table 8.34

Machine data	
Paper grades	board, liner, fine paper, newsprint
Roll position	machine/soft calender
Paper speed	400–1 500 m/min
Operating conditions for the bearings	
Ambient temperature	50 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	40 °C
Journal insulation	bore and end face

Fig. 8.34

Bearing application

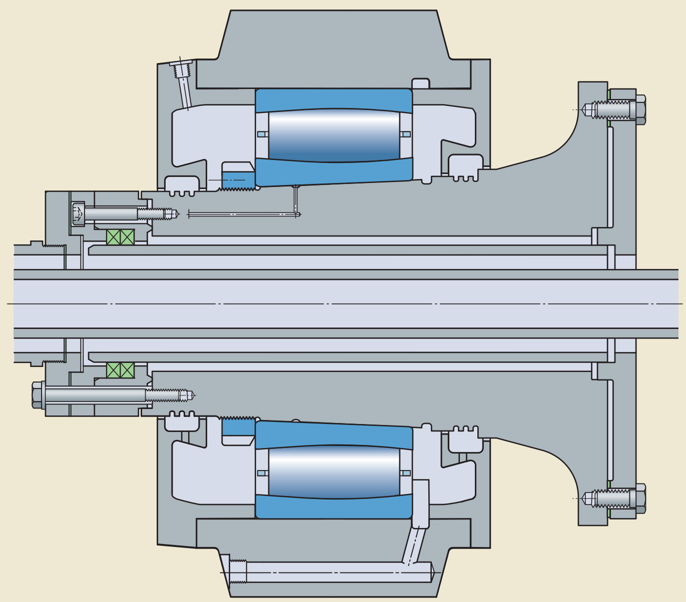
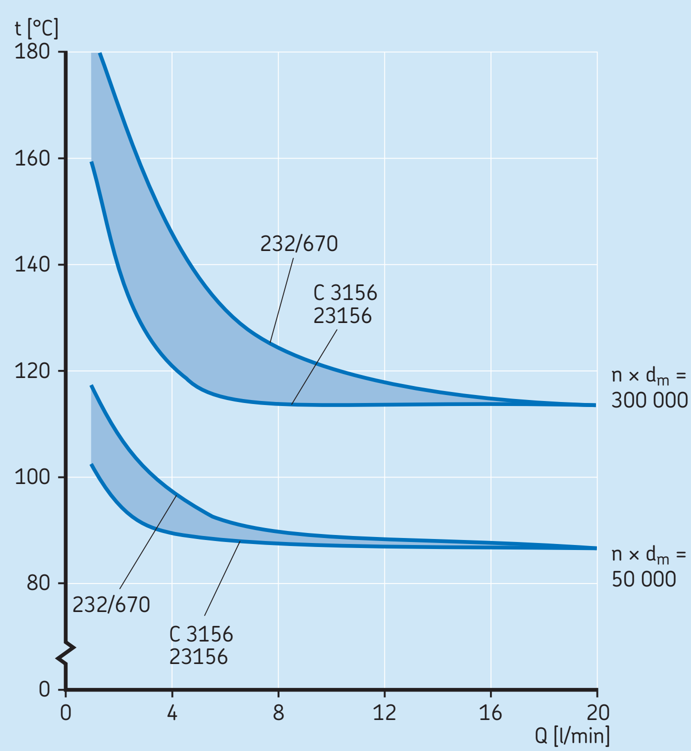


Diagram 8.34

Bearing temperature/oil flow diagram



Example 35

Calender roll with oil temperature 250 °C, fig. 8.35

Lubrication guidelines

As large calender roll bearings sometimes rotate at high speeds, there is a risk of smearing. To avoid smearing, SKF usually recommends the use of an oil with efficient EP additives and oil flows giving a viscosity ratio κ conditions, to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*. However, with the above operating conditions the bearing temperature will be too high for EP additives and, accordingly, oils with efficient AW additives have to be used.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Therefore, the calculated bearing temperatures in this example are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the inner ring temperature. The load at each speed corresponds to a bearing basic rating life of 120 000 hours.

Except for small bearings rotating at low speeds, it is not possible to fulfil the κ guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*. Accordingly, the best possible lubrication conditions should be aimed for.

At low speeds, the diagram shows that the influence of the oil quantity on bearing temperature is considerable up to 4–6 l/min, but after that scarcely anything is gained by an increased oil flow. For large size bearings rotating at high speeds, the diagram shows that the influence of the oil quantity is considerable up to about 16 l/min.

Oil with higher viscosity is beneficial and will improve the lubrication conditions, but the bearing temperature will increase a little.

Table 8.35

Machine data	
Paper grades	board, liner, fine paper, newsprint
Roll position	machine/soft calender
Paper speed	400–1 500 m/min
Operating conditions for the bearings	
Ambient temperature	50 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 320
Oil inlet temperature	40 °C
Journal insulation	bore and end face

Fig. 8.35

Bearing application

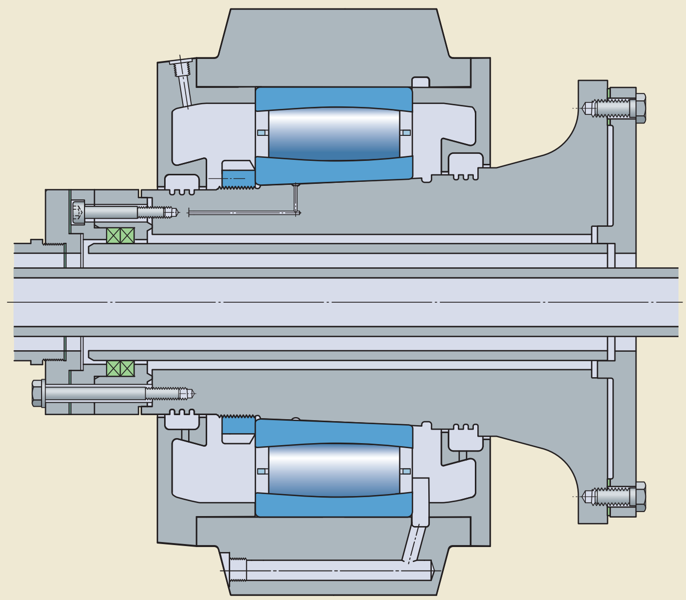
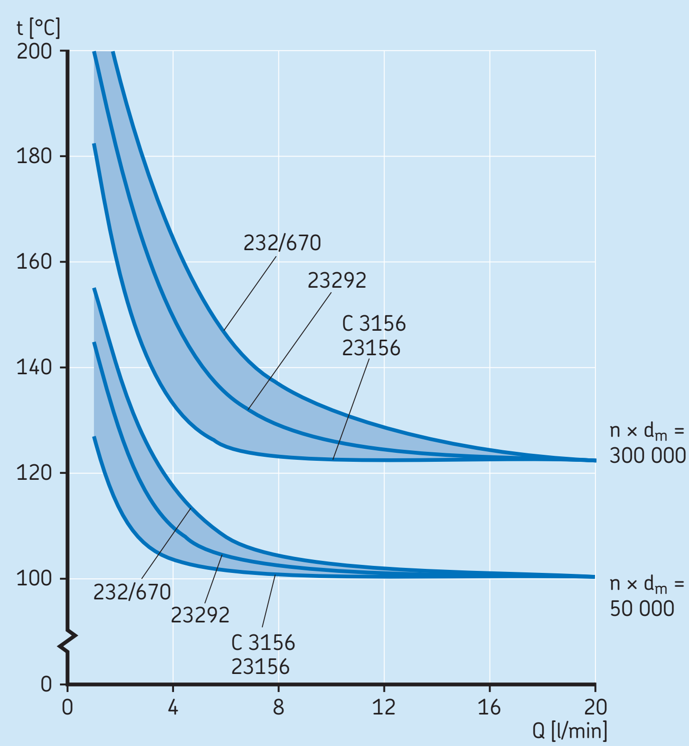


Diagram 8.35

Bearing temperature/oil flow diagram



Example 36

Reel drum (grease), fig. 8.36

Lubrication guidelines

SKF's experience is that grease with a minimum base oil viscosity 175 mm^2/s provides good lubrication performance in this position.

The same grease used in the forming section can be used in this application because the basic demands are the same and the special demands such as those for rust protection are higher in the forming section.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Grease lubrication*, should also be taken into account.

Comments on the diagram

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the outer ring temperature. The load at each speed corresponds to a bearing basic rating life of 120 000 hours. The highest bearing temperature shown in the diagram is 75 °C as it is customary to use oil lubrication at higher temperatures.

Table 8.36

Machine data	
Paper grades	all
Roll position	reeler
Paper speed	100–600 m/min
Operating conditions for the bearings	
Ambient temperature	35 °C
Lubrication	grease
Base oil viscosity	175 mm^2/s

Fig. 8.36

Bearing application

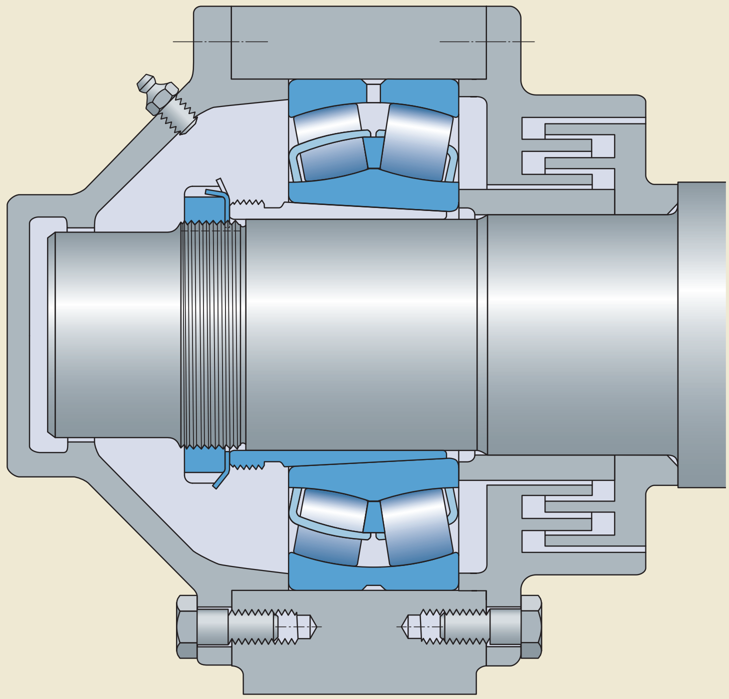
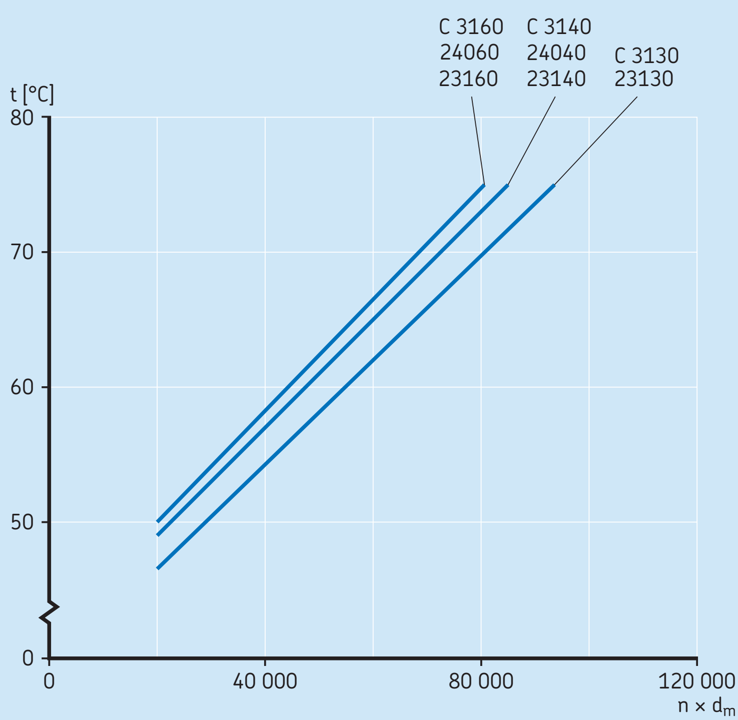


Diagram 8.36

Bearing temperature/speed factor diagram



Example 37

Reel drum (oil), fig. 8.37

Lubrication guidelines

Reel drum bearings have better operating conditions than drying cylinder bearings which often are lubricated with the same circulating oil system. Therefore, the lubricant properties should be based on the operating conditions for drying cylinder bearings. However, the reel drum bearings should have an oil flow giving a viscosity ratio κ according to the guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*.

The general guidelines, as outlined in *Chapter 7*, *Lubrication*, *Oil lubrication*, should also be taken into account.

Comments on the diagram 8.37

Bearing temperature depends on the bearing speed, applied load and external cooling or heating of the bearing arrangement. Heat transfer from the housing, shaft and the foundation is largely dependent on the bearing application and operating conditions. Therefore, the calculated bearing temperatures in the diagram are valid only when the bearing application and operating conditions are similar to those shown in this example.

The calculations for this diagram have been made using an SKF computer program to obtain the outer ring temperature. The load at each speed corresponds to a bearing basic rating life of 120 000 hours. The oil flow in the diagram is based on the use of ISO VG 220 oil and selected to fulfil two criteria: the κ guidelines in *Chapter 10*, *Bearing damage and failure*, *How to avoid raceway and roller surface damage*, and a bearing operating temperature of 75 °C. If oil with lower viscosity than ISO VG 220 is used, the flow must be increased in order to reach the same κ value.

Oil with higher viscosity is beneficial and will improve the lubrication conditions, but the bearing temperature will increase a little. Note that the diagram shows minimum acceptable oil flow. Slightly higher oil flow values are acceptable and beneficial in most cases.

Table 8.37

Machine data	
Paper grades	all
Roll position	reeler
Paper speed	400–2 200 m/min
Operating conditions for the bearings	
Ambient temperature	35 °C
Lubrication	circulating oil
Oil viscosity	ISO VG 220
Oil inlet temperature	45 °C

Fig. 8.37

Bearing application

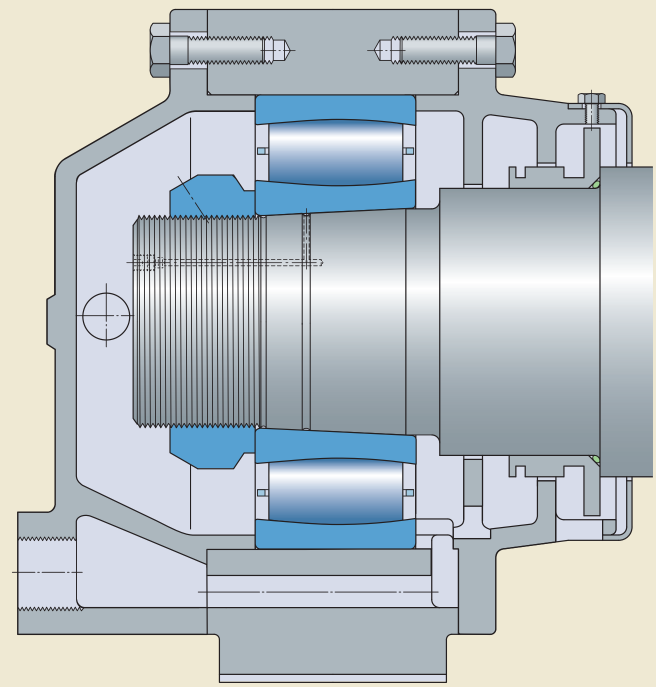
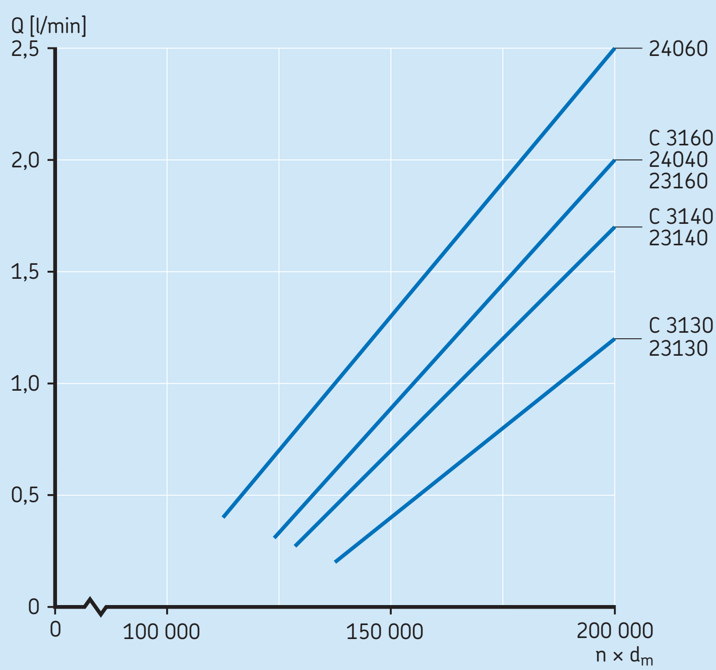
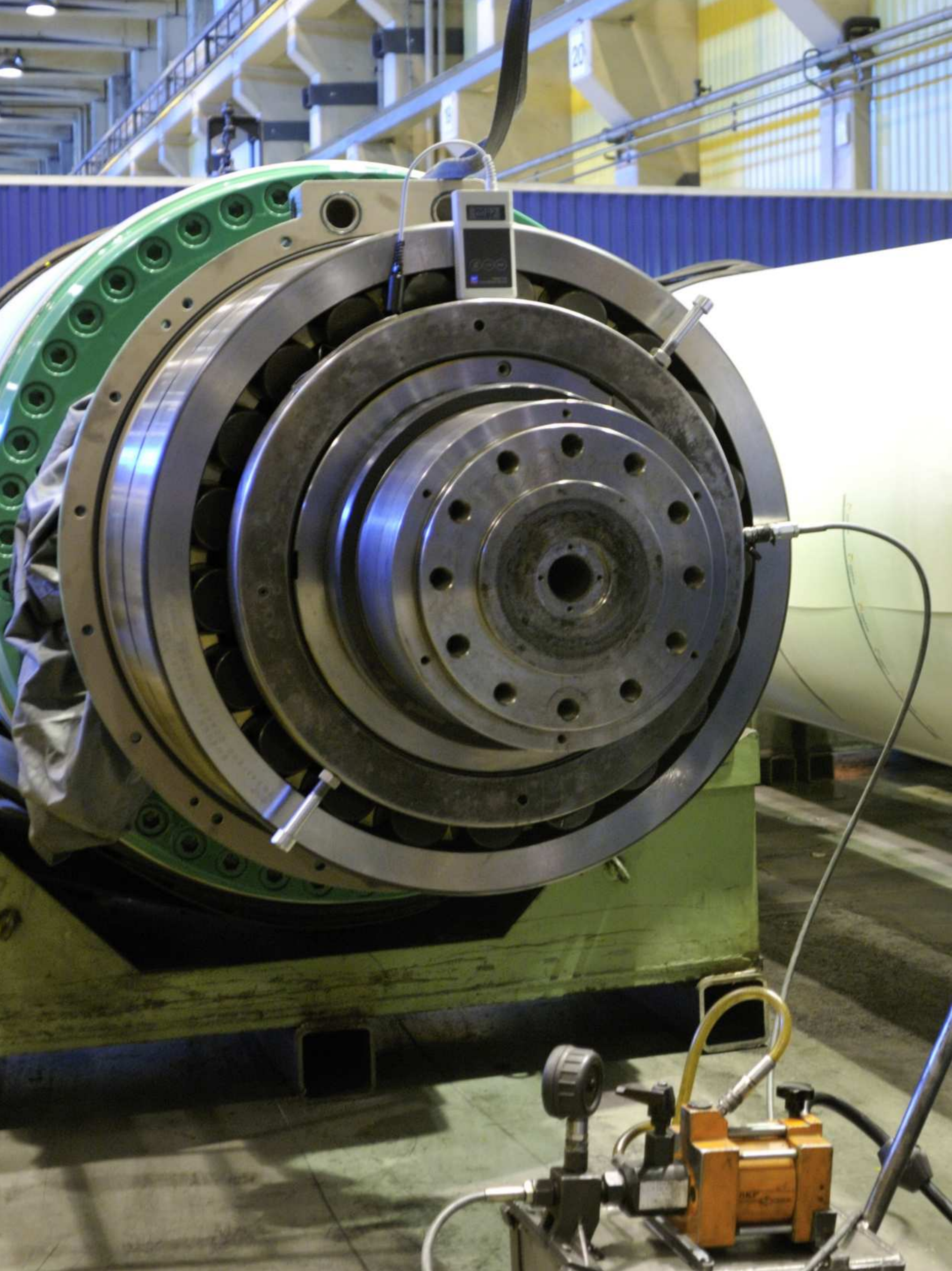


Diagram 8.37

Oil flows for bearing temperature 75 °C





Maintenance

Machine efficiency

The most common way to express machine efficiency is the overall machine efficiency (OME) percentage (→ **fig. 9.1**). OME is normally 65–95%.

Paper mills and machine builders are paying a lot of attention to the OME percentage, but also to paper speed as total production is the result of OME, trim width and paper speed.

Aims of maintenance

Maintenance certainly has some influence on paper quality and web breaks, but the major aim of maintenance is to keep the machine running without any disturbances. An unplanned stop does not reduce the need for planned stops. During a planned stop, many other problems can be dealt with at the same time. Unplanned stops are simply extra hours taken off the production capacity.

Fig. 9.1

$$\text{\% OME} = (\text{uptime}) \times (\text{saleable product}) \times 100\%$$

Where

$$\text{Uptime} = \frac{\text{annual hours the machine produces paper}}{\text{annual available hours*}}$$

$$\text{Saleable product} = \frac{\text{saleable tonnage}}{\text{produced tonnage}}$$

Source: TAPPI 0404-47 (1997)

* Excludes scheduled stops over 24 hours as well as holidays.

Measure of overall machine efficiency (OME)

9

Maintenance philosophies

Reactive maintenance

In the not so distant past, the paper industry's approach to maintenance was "fix it when it breaks". Even today, some mills still use this approach. With this type of maintenance, known as reactive or "run to failure", action is not taken until a problem results in machine failure. The failure can cause costly secondary damage along with unplanned downtime and excessive maintenance costs.

Very often the failures continue to occur since often no real analysis of the reason for the failure is performed and, therefore, no corrective actions are taken.

Preventive maintenance

Preventive maintenance implies that a machine is overhauled on a regular basis regardless of the condition of the parts. The schedule is based on experience which means that some problem areas get more service than others. While preferable to reactive maintenance, preventive maintenance is costly because of excessive planned downtime from unnecessary overhauls and the cost of replacing efficiently operating parts along with worn parts. Preventive maintenance leads to fewer unplanned machine stops than reactive maintenance (→ **fig. 9.2**).

As with reactive maintenance, the failures continue to occur since often no real analysis is done of the reason for the failure and, therefore, no corrective actions are taken.

Predictive maintenance

Predictive maintenance is the process of determining the condition of machinery while in operation. This is possible by using vibration and lubricant analysis, for example. This allows correction of problem components prior to failure. Predictive maintenance not only helps plant personnel reduce the risk of catastrophic failure, but also enables them to order parts in advance, schedule manpower and plan other repairs during scheduled downtime.

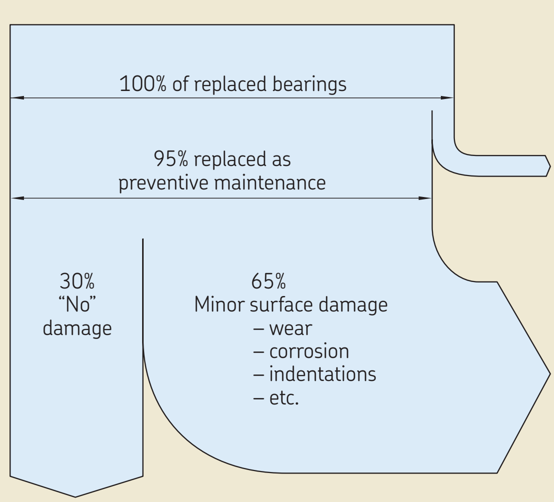
Many paper mills with efficient predictive maintenance programmes have no unplanned stops caused by bearing problems. However, as is the case with the reactive and preventive maintenance approaches, root cause analysis is often not performed.

Proactive maintenance

Proactive maintenance is a further development of the predictive method. When a failure is detected, the reason for the failure is always analyzed and corrective actions are taken. This continuous improvement of the process, focusing on the weakest link in the chain, makes it possible to eliminate unplanned stops and to have longer intervals between planned stops and shorter stops than before. The small improvements eventually make it possible to increase the production speed or load when needed.

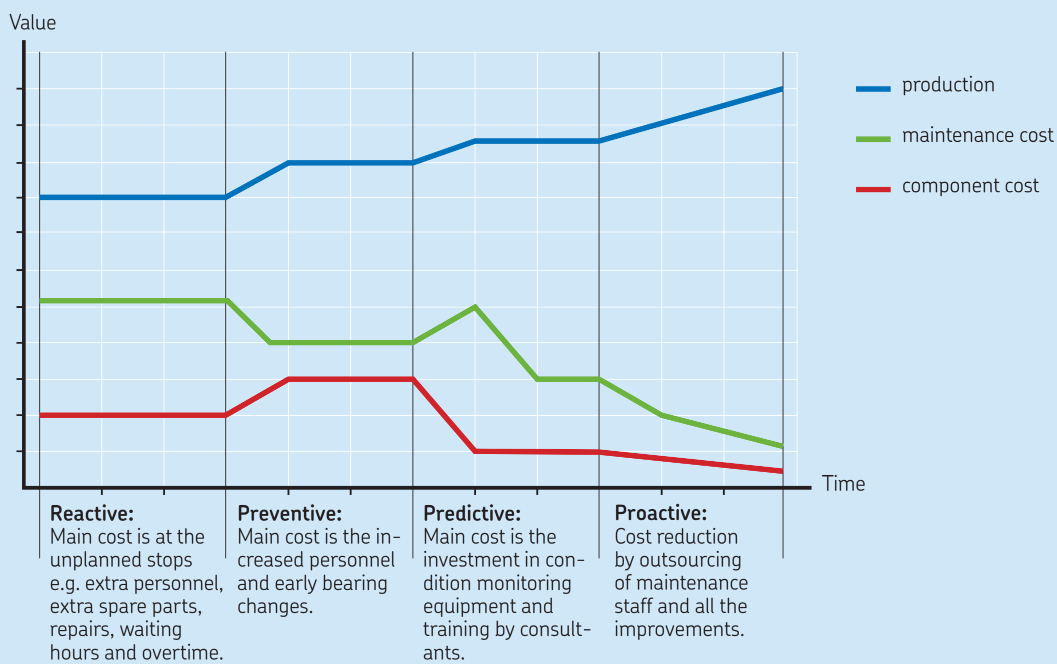
The main difference between the maintenance philosophies can be seen in **diagram 9.1** and **diagram 9.2**.

Fig. 9.2



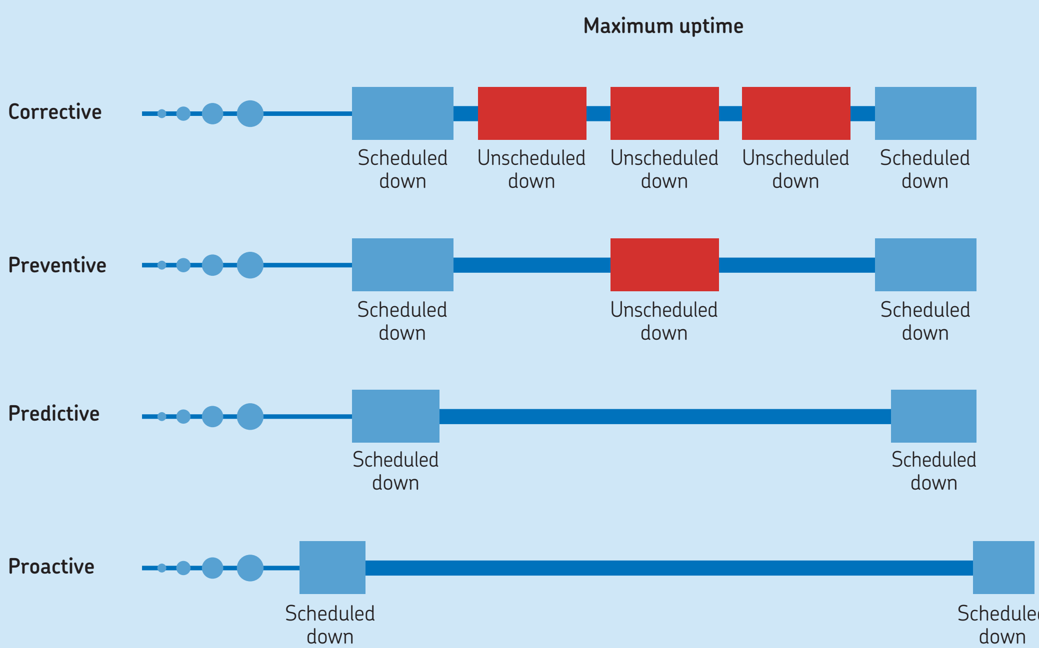
Estimate of bearing condition under preventive maintenance regime

Diagram 9.1



Development of production and cost relative to maintenance practices

Diagram 9.2



Maximum uptime versus different maintenance philosophies

9

Services and products supplied by SKF

Over the years, SKF's Trouble-Free Operation (TFO) programme has extended bearing service life and increased productivity for bearing users by providing a full menu of bearing-related products and services. The original TFO program focused on education, with most of the work performed by the mill's personnel. It also included services such as on-site troubleshooting, application engineering support, bearing failure analysis and bearing rework services.

In recent years, it has been a clear trend in many industries to outsource non-core maintenance to suppliers to reduce total maintenance costs. This is also the case for paper mills. An increasing number of mills want the suppliers of things like bearings, electrical motors, fans, pumps etc. to take over the responsibility for these products.

In response to this trend, SKF has combined its long experience with bearings in the paper industry with advanced capabilities in manufacturing and research to help customers increase productivity.

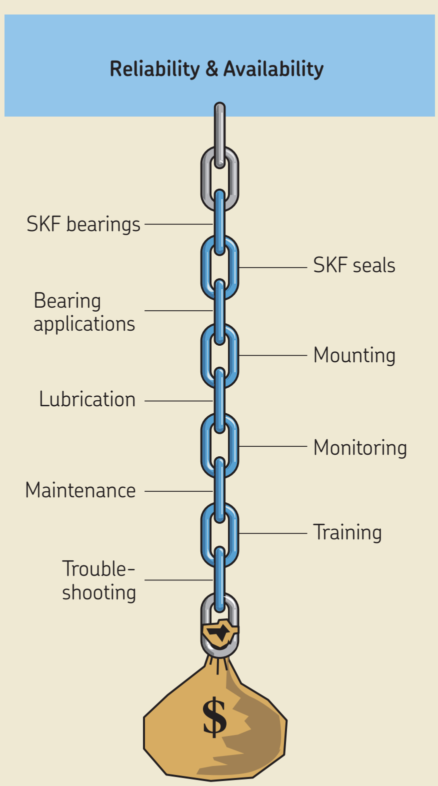
Increased profitability through products and services supplied by SKF

For example, SKF now offers performance-based contracts with a guarantee of downtime reduction or uptime increase. The foundation for the contract is SKF's proactive reliability maintenance process, a systematic method to benchmark machine efficiency and implement corrective actions that decrease total life cycle cost. It involves predictive maintenance, root cause failure analysis, corrective actions and ongoing operational service measured according to key performance indicators.

Furthermore, SKF can offer services like shaft alignment, lubricant analysis, condition monitoring, bearing remanufacturing, etc.

At SKF, we have also put together the industry's most comprehensive range of products for minimizing bearing failure. Among the products are mounting and dismounting tools, condition monitoring equipment and grease. Some of these products and services are shown in the chain (→ **fig 9.3**). This chain illustrates the connection between machine reliability/availability and profitability. Remember that no chain is stronger than its weakest link!

Fig. 9.3



Rebuild the front side of drying and Yankee cylinders to a CARB toroidal roller bearing arrangement

To obtain all the benefits of a CARB toroidal roller bearing arrangement, SKF normally recommends the use of SKF housings for CARB toroidal roller bearings when rebuilding existing machines. The reason is that reworking existing housings can be as expensive as buying new housings.

After many years in operation, the bearing seats of drying cylinder housings may have extensive fretting corrosion and be worn or oval. Therefore, in some cases the housing bore has to be reworked.

As CARB toroidal roller bearings follow the ISO standard for dimensions, a C 3152 bearing has the same boundary dimensions as a 23152 spherical roller bearing or a N 3152 cylindrical roller bearing. If the existing housings are inspected and found to be in good condition, they can in many cases be used for CARB toroidal roller bearings. Sometimes they need to be modified, but if extensive modification is required, it can be as cost-effective to invest in new housings.

When a rocker housing is modified to a fixed housing, it must be locked in all directions. This can generally be achieved by modifying the housing according to **fig 9.4**.

When rebuilding from spherical roller bearings with an axially free outer ring, distance rings have to be used in order to axially locate the outer ring of the CARB toroidal roller bearing (→ **fig. 9.5**).

When the existing bearing is lubricated from the side, it can be replaced by a CARB bearing without any changes to the lubrication.

However, if the existing bearing is lubricated through the outer ring, the housing lubrication design must be changed. One way is to displace the oil inlet to the outer side of the bearing for SKF standard housings incorporating a CARB toroidal roller bearing. However, this requires modifications in order to ensure that no oil drains out without passing through the bearing (→ **fig. 9.6**). In some existing housings, the diameter of the oil channels connecting the two sides of the bearing is small and should be enlarged to allow a suitably high oil flow.

The other alternative is to displace the oil inlet to the inner side of the bearing and to plug the oil channels connecting the two sides. Please note that this modification may

Fig. 9.5

Axially located CARB toroidal roller bearing outer ring

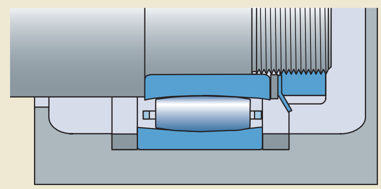


Fig. 9.4

Modified rocker housing

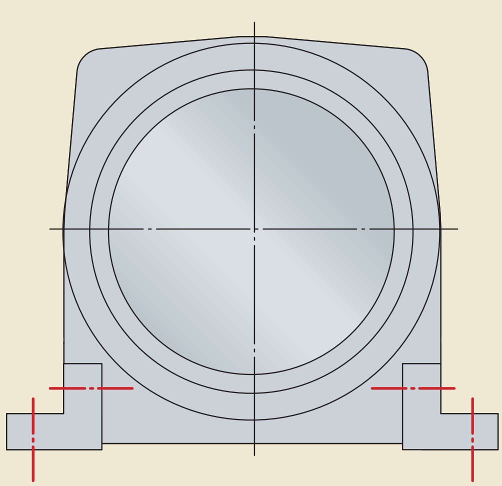
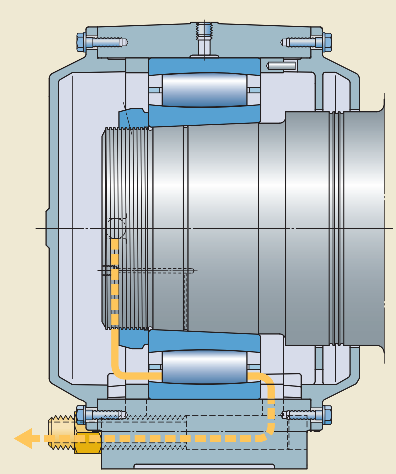


Fig. 9.6

A CARB toroidal roller bearing lubricated from the side



influence the drainage capacity of the housing.

When rebuilding to a CARB toroidal roller bearing arrangement, please contact SKF in order to optimize the arrangement.

SKF can offer a complete package for rebuilding existing housings to fixed housings incorporating CARB toroidal roller bearings. This includes:

- modification of existing housings
- supplying CARB toroidal roller bearings
- mounting service

Mounting and dismounting

The paper industry has always led the way in adopting the innovative mounting and dismounting methods developed by SKF. For modern bearing arrangements, the SKF Oil Injection Method has become the industry standard.

SKF offers a complete range of tools for efficient bearing installation and removal including:

- oil pumps and oil injectors
- hydraulic nuts HMV .. E
- pullers
- heaters

In order to facilitate mounting and dismounting, most bearings in paper machines are fitted on tapered seatings. The use of adapter sleeves and withdrawal sleeves used to be widespread, but the practice of mounting bearings directly on tapered journal seatings is becoming more and more common. Running accuracy is improved and the cost of the sleeves is saved.

Fig. 9.7

Fig. 9.8

Fig. 9.7:
Example of an SKF hydraulic pump

Fig. 9.8:
Example of an SKF puller: SKF EasyPull hydraulic puller kit

Fig. 9.9

Fig. 9.10

Fig. 9.9:
Example of an SKF heater: SKF induction heater TIH 220m

Fig. 9.10:
Tools for the SKF Drive-up Method

Fig. 9.11

Fig. 9.12

Fig. 9.11:
SKF Hydraulic nuts HMV.. E series

Fig. 9.12:
SKF Pressure gauges and dial indicators

9

SKF Drive-up Method

The SKF Drive-up Method is a more precise and less subjective method for mounting CARB toroidal roller bearings (→ **fig. 9.13**) and spherical roller bearings with tapered bores than measuring clearance reduction with feeler gauges. Furthermore, it is significantly quicker.

Therefore, SKF strongly recommends the use of this method for mounting and dismounting large bearings on drying cylinders, suction rolls, press rolls etc. The method is preferable for small bearings as well. The feeler gauge method is particularly difficult to apply for CARB toroidal roller bearings.

SKF can supply suitable tools (→ **fig. 9.14**) and mounting instructions for general applications as well as for specific ones like drying and Yankee cylinders. It is easy to make the calculations required using SKF Drive-up Method software. Simply choose the bearing designation, seating type, method and required internal clearance reduction on skf.com/mount and the calculations are made automatically.

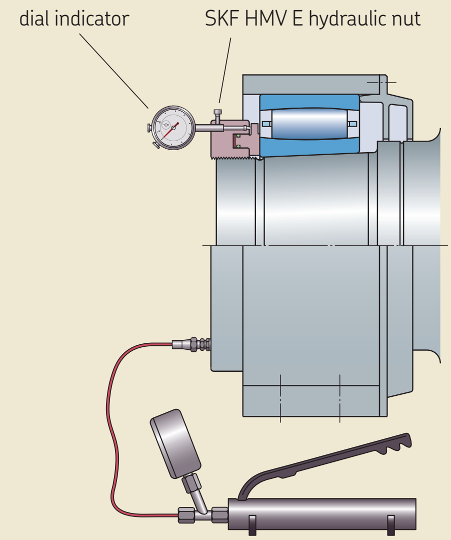
SKF Drive-up Method software can be downloaded from skf.com making it possible to do your own calculations. Note that the drive-up values obtained are only valid for SKF bearings. For more information, please contact your local SKF sales unit.

A typical printout from the program is shown on **fig. 9.21**, **pages 9:12–9:13**.

Sufficient clearance reduction

A radial clearance reduction – Δ_r – of around 0,0005 × d, where d is the bearing bore diameter, in paper machine applications is sufficient to prevent the inner ring from working loose. 0,0005 × d is also indicated as 0,50 ‰ of d.

Fig. 9.14



SKF hydraulic pump 729124 DU (for nuts ≤ HMV 54 E)
SKF hydraulic pump TMJL 100DU (for nuts ≤ HMV 92 E)
SKF hydraulic pump TMJL 50DU (for nuts ≤ HMV 200 E)

The equipment for accurate drive-up

Special situations

Sometimes the bearings are mounted with greater clearance reduction than is generally recommended. Common reasons for this are:

- hollow shafts with very large bores, e.g. suction rolls
- heavy loads (i.e. press rolls)
- experience from the same or similar applications

Fig. 9.13

Mounting a CARB toroidal roller bearing on a Yankee cylinder using the SKF Drive-up Method

One disadvantage with increased clearance reduction is an increased risk of a fractured inner ring. Therefore, the following guidelines should be applied for most unheated applications:

- The clearance reduction Δ_r for SKF standard spherical roller bearings should not exceed 0,0007 × d.
- The clearance reduction Δ_r for SKF spherical roller bearings with case hardened inner rings (suffix HA3 or prefix ECB) should not exceed 0,0009 × d

Higher values are possible in both cases, but SKF should be consulted first.

If a greater clearance reduction than normal is chosen, do not forget to check if a bearing with greater radial clearance class is needed in order to avoid preload during operation.

Starting position

In order to obtain reliable drive-up measurements, the influence of form errors must be reduced to negligible proportions. This can be done by driving the bearing up, passing the indeterminate zero position to a starting position that corresponds to a certain small initial interference (→ **fig. 9.15**). Above this initial interference, reduction in the radial internal clearance may be regarded as being directly proportional to the axial drive-up.

Note: Mating surfaces should be oiled, but note that pressurized oil must not be injected between the mating surfaces before the starting position is reached!

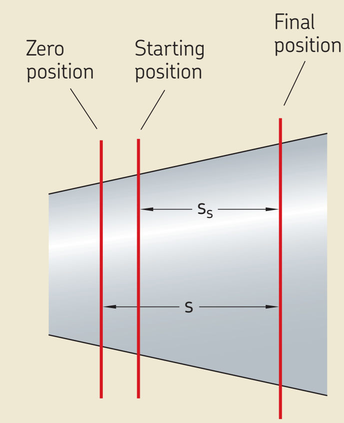
Axial drive-up from the starting position

The axial drive-up is best monitored by a dial indicator connected to an HMV .. E nut.

Pressurized oil can be injected between the mating surfaces after the starting position is reached to reduce drive-up forces. Wait a few minutes, from 5 to 30 minutes depending on the bearing size, before releasing the hydraulic nut pressure so that the oil can drain from the surfaces. The typical time is about 10 minutes for felt roll bearings, 20 minutes for drying cylinder bearings and 30 minutes for large press roll bearings. Note that a longer time may be needed if the journal temperature is low.

Fig. 9.15

Zero, starting and final positions



Interference reduction due to smoothing

Smoothing is a particularly important consideration for smaller bearings.

The drive-up values given in tables supplied by SKF include an interference compensation for smoothing and are valid for one sliding surface, new SKF components and shaft surface roughness R_a = 1,6 µm.

If the mating surfaces are worn, e.g. the bearing has been mounted several times before and/or there are two sliding surfaces, the interference reduction due to smoothing is different. A calculation is then recommended.

For large size bearings, the influence of smoothing is negligible in most cases. Only if the shaft surface is very rough will compensation have to be considered.

9

Mounting on solid shafts

1 Ensure that the bearing size is equal to the HMV .. E nut size. Otherwise, the pressure specified by SKF must be adjusted.
2 Determine whether one or two surfaces slide during drive-up
(→ **fig. 9.16**), one sliding surface
(→ **fig. 9.17**), one sliding surface
(→ **fig. 9.18**), two sliding surfaces
(→ **fig. 9.19**), two sliding surfaces
3 Lightly oil all mating surfaces with thin oil, e.g. SKF LHMF 300, and carefully put the bearing on the shaft.
4 Drive the bearing up to the starting position (→ **fig. 9.15**, **page 9:9**) by applying the specified HMV .. E nut pressure and monitor the pressure on the gauge attached to the pump.
SKF hydraulic pump 729124 DU is suitable for hydraulic nuts ≤ HMV 54E. SKF TMJL 100 DU is suitable for hydraulic nuts ≤ HMV 92E, while TMJL 50 DU is suitable for nuts ≤ HMV 200E. As an alternative, the SKF pressure gauge THGD 100 can be screwed directly into the hydraulic nut.
5 Drive the bearing up the taper the required distance S_s (→ **fig. 9.15**). The axial drive-up is best monitored by a dial indicator. The SKF hydraulic nut HMV .. E is prepared for dial indicators.

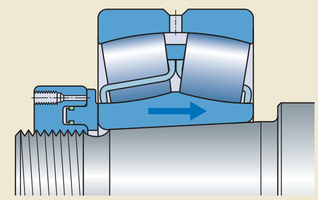
Mounting on drying cylinders

When mounting spherical roller bearings and CARB toroidal roller bearing on drying cylinder journals, the use of the SKF Drive-up Method is strongly recommended. This is a more precise and less subjective method than measuring clearance reduction with feeler gauges. Suitable tools are shown in **fig. 9.14**, **page 9:8**.

SKF spherical roller bearings

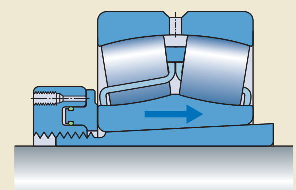
1 Lightly oil the mating surfaces with thin oil (for example SKF LHMF 300).
2 Carefully position the bearing in the housing and on the shaft.
3a For housings bolted to the frame (or frame housings): check that all rollers are unloaded, i.e. the shaft must be centred in the housing during the drive-up.
3b For housings disconnected from the frame: lift the shaft with bearing and housing about 5 mm and check that the housing is free to move axially.

Fig. 9.16



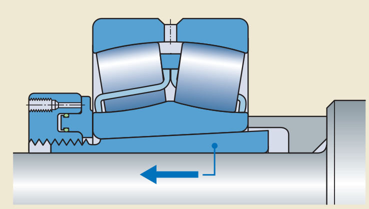
Bearing mounted directly on a tapered shaft with one sliding surface

Fig. 9.17



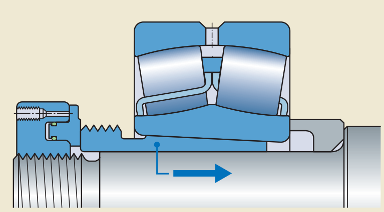
Bearing mounted on an adapter sleeve on a smooth shaft with one sliding surface

Fig. 9.18



Bearing mounted on an adapter sleeve on a stepped shaft with two sliding surfaces

Fig. 9.19



Bearing mounted on a withdrawal sleeve on a stepped shaft with two sliding surfaces

4 Drive the inner ring up to the starting position (→ **fig. 9.15**, **page 9:9**). Do not inject oil between bearing inner ring and its seat.

5 Drive the inner ring up the tapered shaft the required distance S_s (→ **fig. 9.15**) and wait a few minutes before releasing the hydraulic nut. Oil can be injected between inner ring and its seat while driven up the distance S_s.

6 If the housing has been disconnected, bolt it to the frame again.

7 Check the housing alignment, e.g. shaft centred relative to the inner cover.

8a For front side housings bolted to frame: check the axial position of the housing.

8b For front side housings on rockers: check the position of rockers and base plate.

9 If necessary, adjust the position of the housing/rockers/base plate.

CARB toroidal roller bearings

1 Lightly oil the mating surfaces with thin oil (for example SKF LHMF 300).

2 Carefully position the bearing in the housing and on the shaft.

3a For housings bolted to a frame: check that all rollers are free to move throughout the drive-up process.

3b For housings disconnected from the frame: lift the shaft with bearing and housing about 5 mm and check that the housing is free to move axially.

4 Drive the inner ring up to the starting position (→ **fig. 9.15**). Do not inject oil between bearing inner ring and its seat.

5 Drive the inner ring up the tapered shaft the required distance S_s (→ **fig. 9.15**) and wait a few minutes before releasing the hydraulic nut. Oil can be injected between inner ring and its seat while driven up the distance S_s.

6 If the housing has been disconnected, bolt it to the frame again.

7 Check the housing alignment, e.g. shaft centred relative to the inner cover.

8 Check that the axial displacement of the inner ring relative to the outer ring is within specified values.

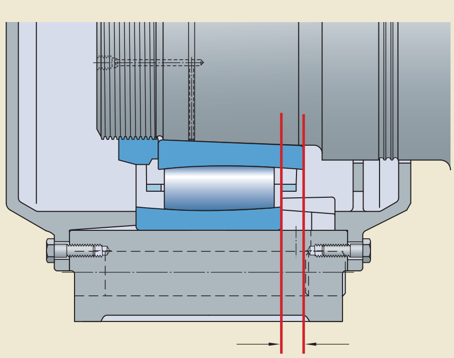
9 If necessary, adjust the position of the housing.

Initial axial displacement of housings incorporating CARB toroidal roller bearing for heated cylinders

SKF experience shows that the thermal elongation of drying cylinders is about one millimetre per metre cylinder length at a steam temperature of 150 °C. To compensate for this elongation, it is possible to displace the housing outwards from the cylinder (→ **fig. 9.20**). To achieve an equal or higher safety margin against preload for spherical roller bearings with C4 clearance, the axial mounting positions shown in **table 9.1** are recommended (valid for a cold machine).

Initial displacement can be used to increase the available axial clearance for cylinder expansion

Fig. 9.20



Housing displacement for heated drying cylinders

Table 9.1

Cylinder length over	Cylinder length incl.	Steam temperature heading	Initial axial displacement
m		°C	mm
0	4	<160	0–1
0	4	160–200	2–4
4	7	<160	2–4
4	7	160–200	4–6
7	11	<160	4–6
7	11	160–200	6–8

9

Example of SKF Drive-up calculation: bearing 23152 CCK/W33 mounted directly on a tapered shaft.

SKF Drive-up method

6/2/2015

Every care has been taken to ensure the accuracy of the information in this system but no liability can be accepted by SKF for any errors or omissions.

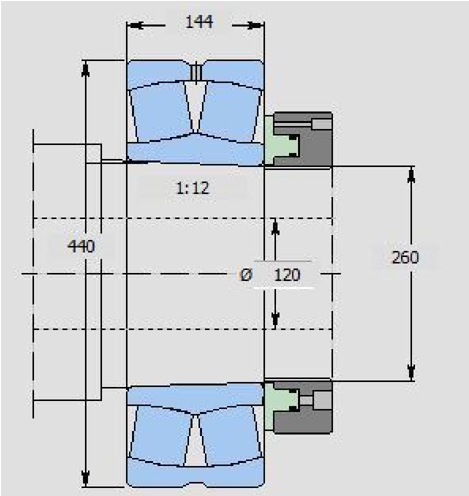
Input	
Mounting	Bearing directly on shaft, one sliding surface
Number of previous mountings	3 (of the same bearing on the same shaft and sleeve)
Required clearance reduction, mm	0.13
Shaft material	Cast iron (E=120000 N/mm2 v=0.25)

Results	
Force to starting position	54051 N = 12152 lbf
Pump pressure to starting position	2.88 MPa = 417 psi using HMV 52 E/HMVC 52 E
Drive-up distance from starting position	1.847 mm = 0.073 inch

Notes

1. Calculation OK

Bearing: 23152 CCK/W33



SKF Drive-Up Method
v04.00 (10-03-2014)

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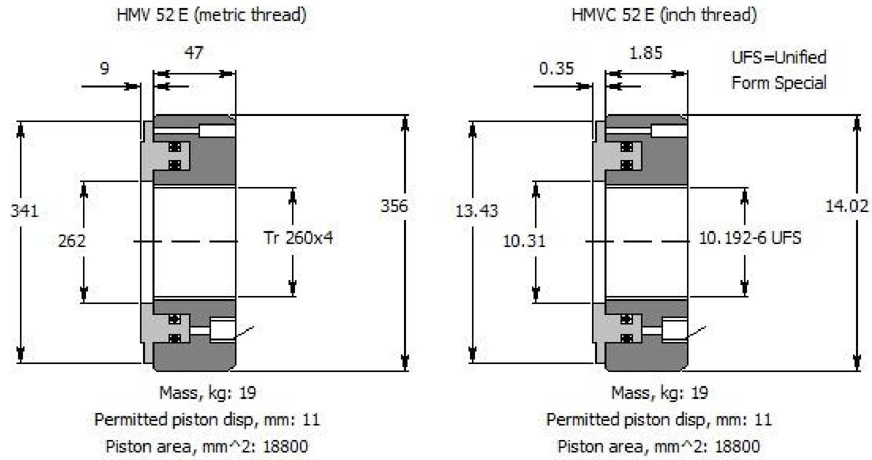
Fig. 9.21

SKF Drive-up method

6/2/2015

HMV



Suitable tools	
Hydraulic nut	HMV 52 E/HMVC 52 E
Hydraulic pump with special pressure gauge	SKF 729124 DU for nuts <= HMV 54E SKF TMJL 100DU for nuts <= HMV 92E SKF TMJL 50DU for nuts <= HMV 200E
Dial indicator	SKF TMCD 5P or SKF TMCD 10R

Mounting procedure

1. Lightly oil all mating surfaces with a thin oil, e.g. SKF LHMF 300 at 20C
2. Drive the bearing up to the starting position by applying correct pump pressure or force.
3. Drive the bearing up on the taper the required distance while measuring the axial movement of the HMV piston.

Complete bearing designation	
Machine no	
Position	
Mounted by	
Date	

SKF SensorMount

The patented SensorMount system allows large size SKF spherical roller bearings of CAK design and CARB toroidal roller bearings with a tapered bore larger than or equal to 340 mm to be mounted in an easy, fast and reliable way. The system comprises a bearing with an integrated sensor and a dedicated hand-held indicator. SKF SensorMount indicates exactly how much the inner ring expands thus ensuring that the desired interference fit between the bearing and the shaft is accurately achieved. SensorMount offers a number of benefits:

- No calculations are needed.
- Independent of bearing size, shaft material, and whether the shaft is hollow or not.
- Independent of seat surface smoothing, surface finish, and lubrication of the mating surfaces.
- Simplifies the mounting process (→ **fig. 9.22** and → **fig. 9.23**)

The displayed hand held indicator value is the internal radial clearance reduction in ‰ of the bearing bore d.

The displayed value 0,50 means that the clearance has been reduced of 0,0005 × d.

Fig. 9.22

CARB toroidal bearing being mounted on a solid roll using SKF SensorMount while oil is injected in the SKF HMV 120 E nut with an SKF air-driven oil pump

Fig. 9.23

The SKF air-driven oil pump is stopped when the value reaches 0,50

Sufficient clearance reduction

A radial clearance reduction – Δ_r – of around 0,0005 × d, where d is the bearing bore diameter, in paper machine applications is sufficient to prevent the inner ring from working loose.

Special situations

Sometimes the bearings are mounted with even greater clearance reduction than generally recommended. Common reasons for this are:

- hollow shafts with very large bores, e.g. suction rolls
- heavy loads (i.e. press rolls)
- experience from the same or similar applications

One disadvantage with increased clearance reduction is an increased risk of a fractured inner ring. Therefore, the following guidelines should be applied for most unheated applications:

- The clearance reduction Δ_r for SKF standard spherical roller bearings should not exceed 0,0007 × d.
- The clearance reduction Δ_r for SKF spherical roller bearings with case hardened inner rings (suffix HA3 or prefix ECB) should not exceed 0,0009 × d.

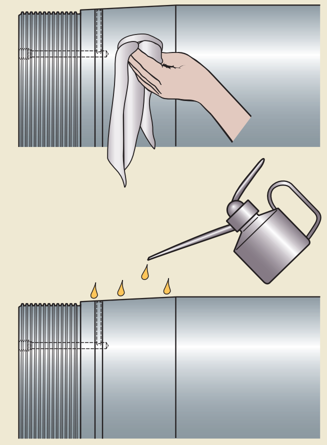
Higher values are possible in both cases, but SKF should be consulted first.

If a greater clearance reduction than normal is chosen, do not forget to check if a bearing with greater radial clearance class is needed in order to avoid preload during operation.

Mounting procedure

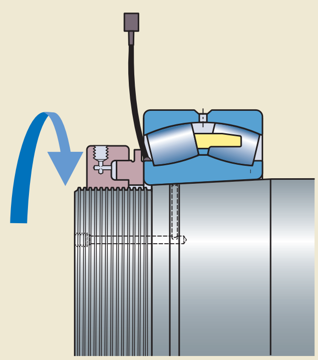
1 Clean and lubricate the shaft seating lightly with thin oil (→ **fig. 9.24**).
2 Place the bearing on its seating. Unpack the sensor cable and the HMV nut and its hydraulic pump (→ **fig. 9.25**).
3 Connect an oil injector to the shaft duct (→ **fig. 9.26**).

Fig. 9.24



Clean and lubricate the shaft seating lightly with thin oil

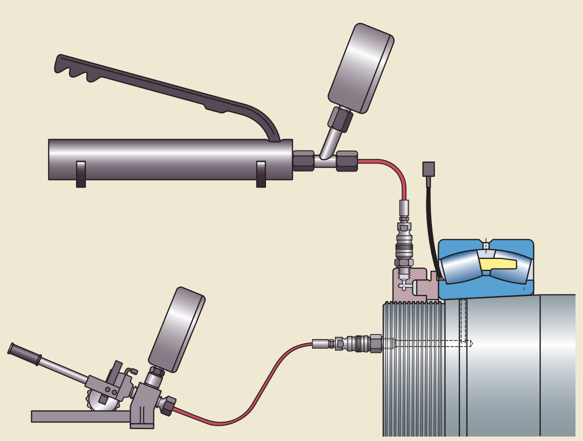
Fig. 9.25



Place the bearing on its seating. Unpack the sensor cable and the HMV nut and its hydraulic pump

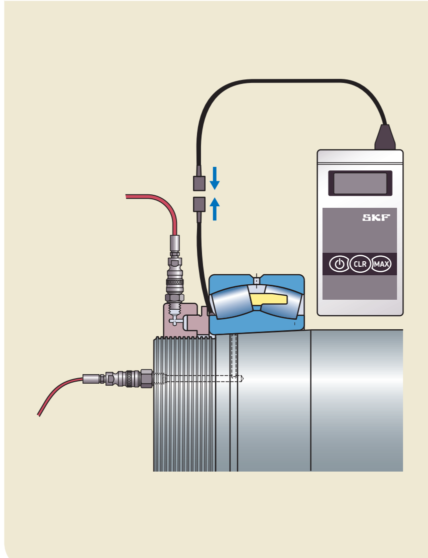
Connect an oil injector to the shaft duct

Fig. 9.26



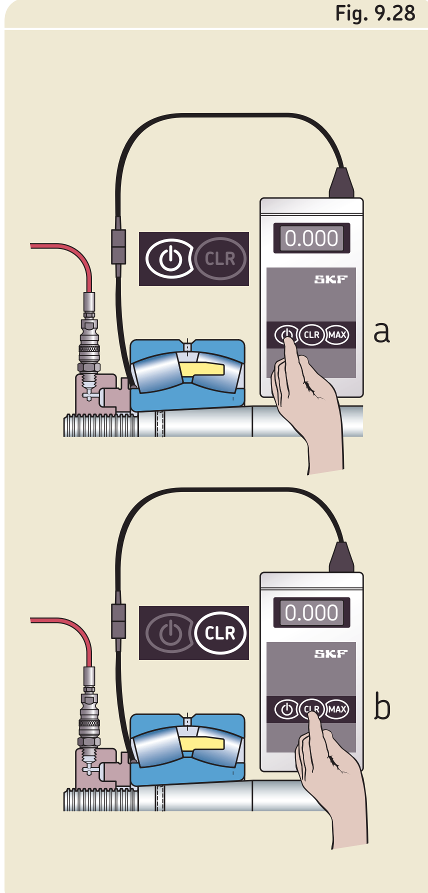
Connect the sensor cable to the indicator cable

Fig. 9.27



Switch on the indicator and zero the display

Fig. 9.28



4 Connect the sensor cable to the indicator cable (→ **fig. 9.27**).

5 Switch on the indicator and zero the display (→ **fig. 9.28**).

6 Initially, drive up the bearing slightly on its seating to achieve full contact between the surfaces. Then continue by pressurising the oil injector and finally driving up the bearing on its seating. When the indicator displays the required value, the drive-up is correct (→ **fig. 9.29**).

7 Release the pressure from the oil injector, but wait for some 20 minutes before releasing the oil pressure in the hydraulic nut (→ **fig. 9.30**). Note that for some very large size bearings and low ambient temperatures, it might be necessary to wait longer than 30 minutes. The time needed depends on the oil viscosity (thus journal temperature) and the contact surface conditions.

8 Cut off the cable close to the sensor with suitable wire cutters (→ **fig. 9.31**).

9 Mount and secure the locking nut (→ **fig. 9.32**).

Designations

- Bearings with sensors for mounting on a tapered shaft have the prefix ZE. The sensor is positioned on the small bore diameter side. Example: ZE 23084 CAK/W33.
- Bearings with sensors for mounting on a withdrawal sleeve have the prefix ZEB. The sensor is positioned on the large diameter side. Example: ZEB 23084 CAK/W33
- The hand-held indicator is supplied separately and has its own designation, SKF TMEM 1500.

Fig. 9.29

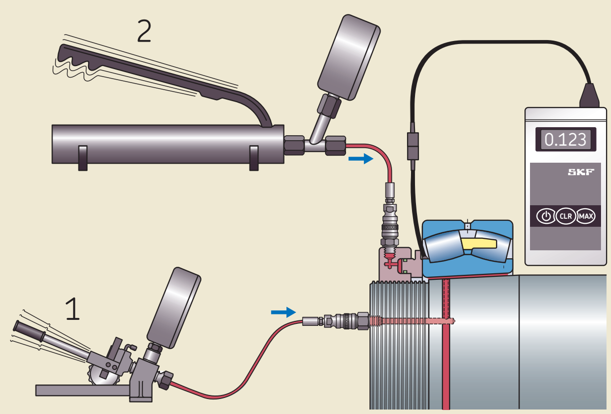
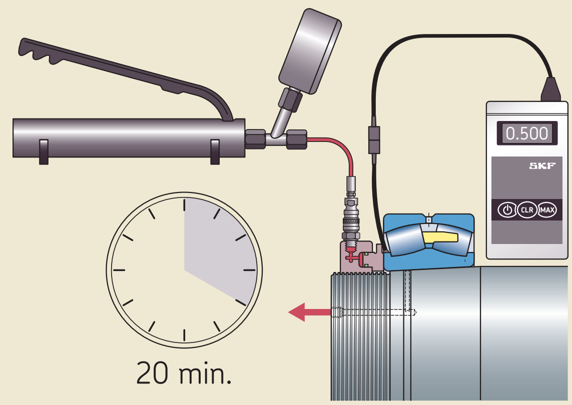
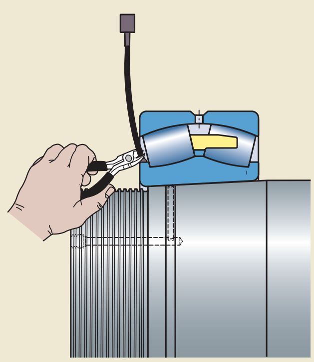


Fig. 9.30



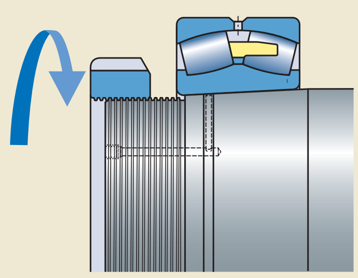
Drive up the bearing until the indicator displays the required value

Fig. 9.31



Cut off the cable close to the sensor

Fig. 9.32



Wait a certain amount of time depending on bearing size before removing hydraulic nut oil pressure

Mount and secure the locking nut

9

Dismounting

In most cases, drying cylinder bearings are mounted in non-split housings and dismounted without disconnecting the housing from the frame. The shaft should be centred in the housing (i.e. all the rollers in the bearing are unloaded) during dismounting.

First, the bearing inner ring is released from the tapered shaft using the SKF Oil Injection Method. The distance between the inner ring side face and the nut must be about twice the drive-up distance. If the distance is greater, there is a risk of raceway damage.

When the inner ring is released, the bearing can be dismounted from the housing.

It is often difficult to dismount a bearing from a non-split housing because it is hard to apply an axial dismounting force by hand and sometimes, especially if fretting corrosion has occurred, the outer ring tends to stick in the housing.

For spherical roller bearings the easiest way is to lift the shaft slightly, pull out the bearing and housing and when they are outside the machine, press out the bearing.

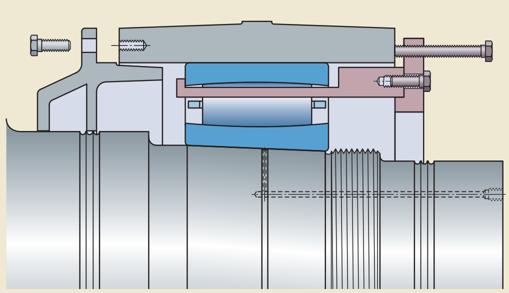
For spherical roller bearings mounted in the machine frame, special tools or methods may be needed.

As the dismounting force on a CARB toroidal roller bearing must be applied on the outer ring, the use of an extractor as shown in **fig. 9.33** is strongly recommended. The use of the extractor is possible due to the design of the CARB toroidal roller bearing with more space between rollers, cage and outer ring compared to a spherical roller bearing.

For more information about dismounting tools, please contact SKF. Please also contact SKF if you want mounting and dismounting services to be provided. Your local SKF company can offer the resources as well as the equipment for proper installation and cost savings by reducing the risk of secondary damage caused by improper handling of the bearings.

Extractor for easy dismounting of CARB toroidal roller bearings

Fig. 9:33



Condition monitoring

The aim of a condition monitoring system is to determine the condition of "wear" components and other functions that influence machine reliability. The advantage of condition monitoring using vibration analysis is that it acts as an early warning system. Early warning of machinery problems provides time for corrective actions and for planning maintenance actions, including bearing replacement.

Here are some examples of components and systems that are typically monitored with vibration analysis for asset reliability:

- bearings
- pumps
- gearboxes
- electric motors
- fans

Here are some examples of process components and systems that can also be with vibration analysis:

- felts
- nipped rolls/roll covers
- doctor blades (in tissue machine applications)

Seals are an example of "wear" components that are normally manually inspected.

Multi-parameter monitoring

SKF recommends multi-parameter monitoring for the most comprehensive, reliable and accurate approach to machinery monitoring and analysis. Collecting and analyzing various measurement types greatly increase lead time as well as the capability to diagnose bearing damage and other machinery problems.

By measuring a number of machinery parameters – from vibration acceleration, velocity and displacement to process parameters like speed, temperature, current, pressure and flow – users gain essential insights into a machine's condition. Advanced vibration analysis techniques, like SKF Acceleration Enveloping, enable analysts to take the guesswork out of maintenance by supplying the information needed to take action towards preventing unscheduled downtime.

Traditional vibration monitoring parameters (velocity, acceleration, and/or displacement) are essential for identifying and diagnosing many rotating machinery problems. Conditions such as imbalance, misalignment, looseness, etc. cause undue stress on mechanical components, which can then lead to damage (e.g. to bearings) and a reduction in machine efficiency. While traditional vibration parameters analysis can be effective indicators that bearing damage is present (especially in the later stages), they are not the earliest indicator of a rolling bearing problem.

SKF acceleration enveloping, an additional vibration measurement parameter that complements traditional vibration measurements, is very effective for early detection of impulsive machine faults. It enhances repetitive signals caused by impacts from a damaged bearing. This is extremely important particularly in the early stages when damage is just starting since impacts from small bearing surface damage generate signals that may go undetected amid general machine vibration. The use of acceleration enveloping enhances detection and diagnostic capabilities, allowing users to identify damage in specific bearings and/or gearbox components. SKF multi-parameter monitoring has been proven for more than 30 years to be especially effective for rolling bearing monitoring in all applications, even for very low rotation speeds (e.g. 10 r/min or lower).

Operator tools

Economical and easy-to-use handheld instruments provide a quick and basic indication of problem areas.

The SKF Machine Condition Advisor is a pocket-sized, go anywhere measurement device that provides an overall velocity vibration reading and automatically compares it to pre-programmed ISO guidelines. An alert or danger alarm displays when the measurement exceeds those guidelines. Simultaneously, an SKF Enveloped Acceleration measurement is taken and compared to established bearing vibration guidelines to verify conformity or indicate potential bearing damage. The SKF Machine Condition Advisor (→ **fig. 9.34**) also measures temperature using an infrared sensor to indicate uncharacteristic heat.

The hand-held product range also includes the SKF Machine Condition Detector (→ **fig. 9.35**) which is a probe that collects and compares operating data to provide advance warning of costly machine problems. Red, yellow and green lights indicate machine status instantly for assessment of bearing and lubrication problems.

Portable data collection

Portable data collectors, like the SKF Microlog Inspector and Wireless Machine Condition Detector (→ **fig. 9.36**), are rugged high-performance portable computers that replace paper inspection trails by documenting inspections with accurate, consistent and actionable information.

The Wireless Machine Condition Detector collects vibration, temperature and FFT data which is transferred to the SKF @ptitude Monitoring Suite for further analysis by the maintenance engineers.

The four-channel SKF Microlog Analyzer series includes route-based instruments that work with powerful SKF predictive maintenance software systems and stand-alone instruments that offer on-the-spot advice and signal analysis capabilities. Modular flexibility allows customers to customize their Analyzer to meet their business needs.

Fig. 9.34

SKF Machine Condition Advisor

Fig. 9.35

SKF Machine Condition Detector

Fig. 9.36

SKF Microlog Inspector and SKF Wireless Machine Condition Detector

Continuous monitoring

Online monitoring for round-the-clock bearing and machinery analysis offers significant advantages in a paper-making environment. With the SKF Multilog Online Systems IMx, DMx, and WMx, permanently installed sensors collect data from hard-to reach or more critical machine sections, providing continuous monitoring and eliminating the need for manual or walk-around data collection.

Fig. 9.37

SKF Multilog Online System IMx-S

Fig. 9.38

SKF Multilog Online System DMx

Fig. 9.39

SKF Multilog Online System WMx

The SKF Multilog Online System IMx (→ **fig. 9.37**) is the new standard for online monitoring of paper machines. It comes in two versions; IMx-S for field installation (includes industrial housing) or IMx-T for mounting in a 19" instrument cabinet. The modular design enables it to be used for other areas (e.g. pulp mills). Together with SKF @ptitude software, part of the SKF @ptitude Monitoring Suite, the SKF Multilog IMx provides a complete system for early fault detection and prevention, automatic recognition to be able to correct existing or impending conditions and advanced condition-based maintenance to improve machine reliability, availability and performance.

The SKF Multilog Online System DMx (→ **fig. 9.38**) provides vibration-based machinery protection and condition monitoring in a single device for use in both conventional and hazardous areas. The SKF Multilog DMx offers a four-channel vibration monitoring solution that for the first time enables the requirements of critical machinery monitoring, from transducer to dynamic data processing, to be fulfilled by an intrinsically safe device (e.g. for use on critical equipment in the mill's power house). The SKF Multilog DMx provides two distinct vibration monitoring capabilities in the same module:

- Machine protection measurements, in compliance with the API 670 standard, to react to a vibration alarm that may signify a looming catastrophic failure.
- Condition monitoring measurements to be used by sophisticated software to predict a potential catastrophic failure.

The SKF Multilog Online System WMx (→ **fig. 9.39**) is a compact, eight-channel, field mounted monitoring device that communicates using industry standard 802.11b/g wireless networking. The WMx collects acceleration, velocity, displacement, temperature and bearing condition data and automatically sends this to the SKF @ptitude Software for viewing, alarm evaluation, and analysis.

SKF @ptitude Analyst

SKF @ptitude Analyst is a comprehensive software solution with powerful diagnostic and analytical capabilities. It provides fast, efficient and reliable storage, analysis and retrieval of complex machine information and makes the information accessible. Its advantages are:

- One software program to manage machinery condition data from portable and online devices.
- One installation with limitless expansion capabilities.
- Easy to learn and use for novice or experienced users.
- Interconnectivity with other software programs and systems.
- Easy personalization for individual users.

When you select SKF @ptitude Analyst (→ **fig. 9.40**), you are immediately equipped to integrate and analyze data from the full range of SKF data collection devices. This enterprise-wide software platform enables operations, maintenance and reliability staff to view the integrated data and communicates the information to each department in a customized format to meet individual user needs. SKF @ptitude Analyst can incorporate data from other sources such as OPC servers and seamlessly interface to your SAP, Computerized Maintenance Management System (CMMS), Enterprise Resource Planning (ERP) or other information management systems.

SKF @ptitude Analyst's integrated platform forms the hub to share information, foster teamwork and facilitate consistent and reliable decision-making across functional departments. The addition of SKF @ptitude Decision Support automates reliability maintenance decision-making by identifying probable faults with an asset or process, then prescribing appropriate action.

SKF @ptitude Analyst software

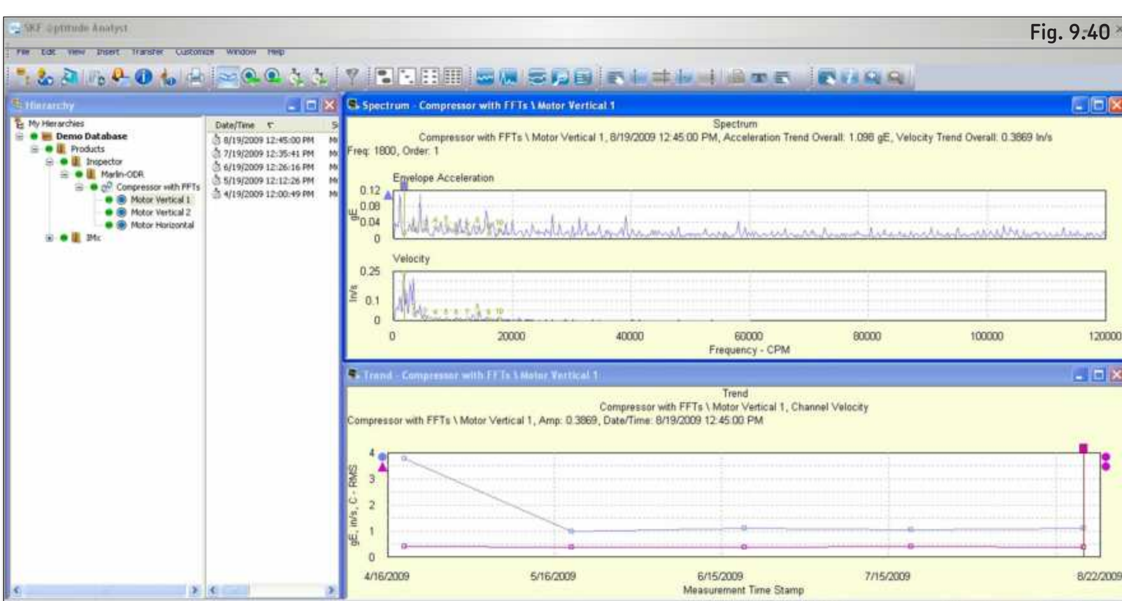


Fig. 9.40

Standstill precautions

Despite the use of water extractors, many oil lubrication systems have small quantities of free water especially in the forming and press section. Grease-lubricated bearings may also have free water in the lubricant.

The highest risk of water ingress probably occurs during the high-pressure cleaning of machine frames, wires and felts. Free water in the lubricant, especially water mixed with cleaning chemicals, will rapidly start a corrosion process on bearing surfaces when the machine is at standstill. There are some different ways of avoiding such corrosion:

- Never direct high-pressure spray nozzle towards the sealing gap.
- Keep the water outside the bearing by using more efficient seals.
- Remove free water by using more efficient water extractors on oil lubrication systems.
- Replace contaminated lubricant (grease or oil in a bath) with fresh and clean lubricant.

The last alternative is the most commonly used method for grease-lubricated bearings. Fresh grease is injected just before stopping the machine while the bearings are still rotating.

SKF Anti-corrosive agent LHRP 2

Fig. 9.41

The most common method with oil lubrication is to remove the water by increasing extractor capacity before stopping the machine, e.g. using a portable vacuum extractor. In some circulating oil systems, it is even possible to let the oil continue to circulate during standstill, which is of course an advantage.

If the machine is closed down for some time, oil analysis to determine water content is recommended. Oil circulation should not be stopped if the water content is above 200 ppm. When the oil circulation is stopped, a rust inhibiting agent should be sprayed into the bearings. SKF recommends SKF LHRP 2 (→ **fig. 9.41**) which provides long term indoor and outdoor protection to ferrous and non-ferrous surfaces. This product deposits a slightly greasy, thixotropic film. SKF LHRP 2 is compatible with mineral oils and PAO or PAO/ester oils which are commonly used in paper machines. It is not compatible with polyglycol oils that are often used in industrial gearboxes. For other types of oil, please contact SKF.

If a paper machine is stopped for months or years without knowing if it will be restarted, preventive maintenance to avoid bearing damage is often omitted. If the machine is to be put in operation again, it is often less costly to replace all small and medium bearings and check by visual and ultrasonic inspection if large size bearings can still be used or remanufactured or whether they should be replaced with new ones. Machines that are restarted after a very long period without maintenance tend to experience unplanned stops during the first months of operation due to bearings having been damaged by corrosion, lubrication issues and solid contamination. Lubrication issues can come from grease that has aged in the ducts or pipes, getting stiff. Solid contamination can come from oil feed steel pipes that have corroded.

How to store spare bearings

Unused bearings

Unused bearings should be stored in their original SKF packaging in an indoor low humidity storage area.

The storage area should:

- be clean
- be dry
- be free from air flow
- have air conditioning if the relative humidity in the area is >60% (peak at 65% accepted)
- be free from vibration
- have a constant temperature. Maximum temperature fluctuation: 3 °C/48 hours.

Table 9.2

Recommended maximum storage time for open bearings.

Relative air humidity	Temperature	Storage time
%	°C	years
60	20–25	10
75	20–25	5
75	35–40	3
Uncontrolled tropical conditions		1

Table 9.2 gives recommended maximum storage times (from packing date) for open bearings.

For lubricated (sealed) bearings, the time varies depending on the lubricant. As simple rule of thumb, SKF recommend 3 years maximum storage time after manufacturing date.

Used bearings

Paper mills have a number of spare rolls in their roll store. While suction, press and calendar rolls are regularly reground (→ **fig. 9.42**) and replaced in the machine, other types of rolls are only stored for security reasons. The latter type can be stored for a long time (→ **fig. 9.43**).

Rolls are normally pressure cleaned before being dismounted from the machine and then stored, often for several weeks, before regrinding. During this period, there is a great risk of corrosion if water is present in the housing.

When the roll has been dismounted from the machine, SKF's recommendations should be followed. However, experience shows that this often does not happen and corrosion occurs. Therefore, to be on the safe side, grease-lubricated bearings should be regreased after the cleaning procedure just before the machine is stopped. Note that the grease quantity must be large enough to press out all the contaminated grease. If the

Fig. 9.42

Thermo roll being reground and rotated on its main bearings cleaning of the rolls takes places outside the machine, it is important to cover the seals so that no water can enter through the seals. For oil-lubricated bearings, the housing should be flushed with fresh, clean oil with very good anti-corrosion properties. If possible, spray an anti-corrosion agent such as SKF LHRP 2 on and into the bearing. SKF LHRP 2 is compatible with mineral oils and PAO or PAO/ester oils which are commonly used in paper machines.

For bearings that are going to be stored for a lengthy period, SKF recommends the following:

- In the case of grease-lubricated bearings, dismount the bearing housings and remove all old grease from the bearings and the housings.
- Dismounting the bearings from the journals, if possible.
- Washing the bearings with a very thin oil.
- Inspecting the bearings to ascertain whether or not they can be used for further service on the machine.
- Remounting the bearing and the housings.
- Lubricate and protect the bearings against corrosion. This can be done with the same lubricant that is used in the machine.
- If possible, place the roll in a store where it is supported by the bearings and subjected to constant or regular rotations.
- Bearings that are to be stored under static conditions, either dismounted or on the journals, must be protected against corrosion. Grease-lubricated bearings should be packed with fresh grease with good anti-corrosion properties. Oil-lubricated bearings should be protected with rust inhibitor e.g. SKF LHRP 2 giving up to one year's protection. Thereafter, preservation needs to be renewed. For other types of oils, please contact SKF.

Fig. 9.43

Press roll and other rolls stored with bearings mounted

How to avoid transport damage

The best and only fully safe way to avoid bearing damage is to transport bearings in their original packaging and to only unpack them just before mounting. The transportation method used must not produce heavy vibrations that could damage the bearing by false brinelling. Large temperature variations during transportation should also be avoided to eliminate the risk of condensation.

When installed bearings are transported, the rolling elements (balls or rollers) should either be preloaded or completely free to move. When the rolling elements are preloaded, there is usually no risk of sliding and smearing. However, during transportation under bad conditions, small movements sometimes occur causing false brinelling.

Since the weight of rolling elements is relatively small, the contact load in the bearing with free rolling elements is usually not large enough to cause smearing even if the rolling elements are sliding in the bearing. However, during transportation under bad conditions, it might be enough to cause smearing.

The best way to reduce the risk of damage in installed spherical roller bearings is to axially preload one of the roller rows. The axial preloading should be around 10% of the C value. In the bearing, the rollers in one of the rows will be preloaded and in the other row completely free to move. This method cannot be used for CARB toroidal roller bearing because of the very small contact angle of these bearings. With axial preloading, there would be a risk of indentations. For CARB toroidal roller bearings, SKF recommends transportation with all rolling elements free to move. This is achieved by positioning the housing so that the radial internal clearance is the same in all positions of the bearing.

A thick clean plastic sheet can be placed between free rolling elements and rings.

CARB toroidal bearing mounted on an old drying cylinder as an upgrade

Bearing damage and failure

Bearing damage and failure in modern paper machines is generally the result of surface damage caused by:

- inadequate lubrication
- solid contamination
- water ingress
- improper handling during mounting or dismounting
- vibration
- the passage of electrical current

Standstill corrosion is perhaps the most common problem. Other problems such as subsurface white etching cracks (WEC) are less frequently seen though there have been a few cases, mainly on heavily-loaded press roll bearings, over the last few years.

Increasing speed on older machines can create several issues for bearings such as:

- Increased nip load and wire and felt tension. These changes can increase the load on bearings in some positions leading to reduced life. They can also reduce the load in other places and cause problems with rollers sliding rather than rolling and causing smearing damage.
- Increased steam temperatures causing lubrication issues on uninsulated journal bores.
- Inadequate lubrication.

When increasing machine speed, especially above the original design speed, many things need to be considered. Bearing type, size, operational clearance, fits and lubrication should be reviewed for all positions. When doing this, it should be noted that old machines tend to be more sensitive to water ingress as the bearing housings usually have less effective seal designs.

For more information, see SKF's handbook *Bearing damage and failure analysis*.

Root Cause Analysis

It is difficult if not impossible to undertake root cause failure analysis on a damaged bearing, particularly if it has run to failure (→ **fig. 10.1**), without information on the operating conditions, a lubricant analysis, the bearing assembly drawing and a mounting report.

Initial damage can give clues about the likely root causes. Often the initial damage is masked by the secondary damage such as spalls, ring fractures or heavy smearing which ultimately lead to catastrophic bearing failure. As such, SKF recommends dismounting damaged bearings before they fail.

Fig. 10.2 shows the development of damage on the inner ring of a spherical roller bearing:

1. The bearing exhibits incipient abrasive wear. This initial damage is a good indicator of fine solid contamination and/or too thin oil film thickness.
2. The first spall is detected using SKF acceleration enveloping. Due to the presence of the spall, you cannot be certain that the initial damage was caused by abrasive wear though it would still be the most likely cause.
3. Spalling has developed to such an extent that it can be detected by standard vibration monitoring.
4. Advanced spalling has developed leading to high vibration and noise together with an increase in operating temperature. By this stage it is difficult to determine the true root cause without additional information on operating conditions etc.
5. Severe damage occurs with fatigue fracture of the bearing inner ring.
6. Catastrophic failure with potential collateral damage.

Fig. 10.1

A bearing that is too heavily damaged for meaningful root cause failure analysis

The development of bearing damage

Fig. 10.2

It is important to dismount a damaged bearing as soon as realistically possible. This will increase the chances of determining the root cause allowing appropriate corrective actions to be taken, reduce the risk of repeat bearing failures and collateral damage and increase the possibilities to remanufacture the bearing.

While some bearings can operate for months with easily-detectable minor surface damage, in others the damage develops fast and the period between the first detectable damage and failure can be less than the interval between two vibration analysis readings taken with a handheld device. As such, continuous online temperature and vibration monitoring is worth considering in many cases.

Collecting information

It is very important to collect and document any relevant information when bearing damage or failure occurs to facilitate subsequent root cause failure analysis.

A short overview of what needs to be considered is summarized below.

General information

- Company and contact persons
- Machine – application description
- Problem description

Operating data

- Drawings and photographs of the application with adequate details to understand the shaft-bearing-housing system and bearing arrangement.
- Application data (speed, loads, temperature, lubricant method, lubricant used, lubricant renewal specifications, drive system, bearing designation, shaft and housing fits, sealing, required bearing life and time in operation).

Monitoring data

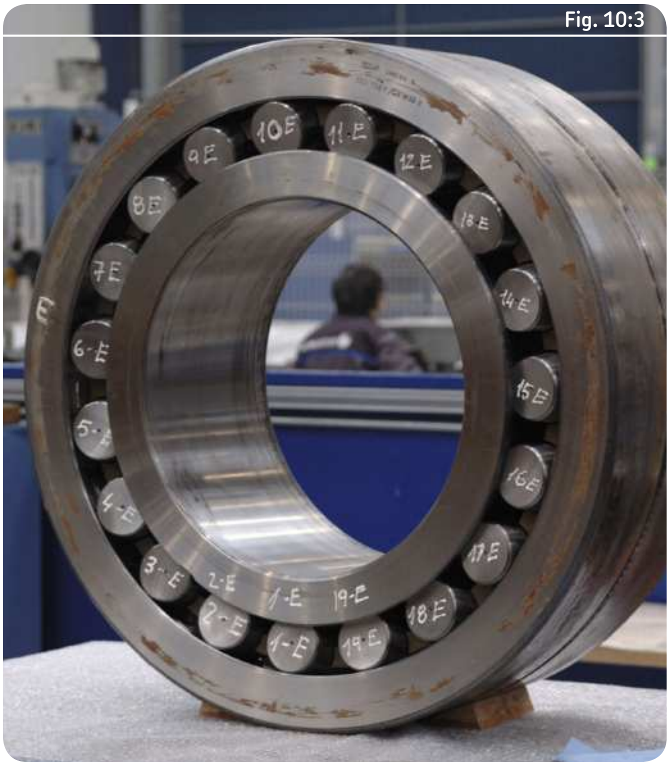
- Condition monitoring history (vibration levels, temperature readings, and sound recordings)
- Lubricant analysis history

During dismounting

- Document any visible damage on the equipment (loose bolts, scratches or damaged/worn adjacent parts including seals).
- Take lubricant samples from inside the bearing and the surrounding area. Store the samples in clean receptacles and label them.
- Mark the bearing ring mounting location in the machine (→ **fig. 10.3**) as well as its relative position against the shaft/housing.

Press roll bearing rings and rollers marked with "E" for the external side

Fig. 10:3



- Take photographs and write notes during the procedure.
- Dismount the bearing with care and document the dismounting method when damage cannot be avoided. If the bearing has to be cut, do it where there is no damage that could give clues about the root cause (→ **fig. 10.4a** and → **fig. 10.4b**). If possible, avoid flame cutting.
- Mark the bearing parts (→ **fig. 10.3**).
- Protect the bearing from dirt and moisture in an appropriate receptacle for further analysis by a SKF engineer. Do not clean the bearing unless the lubricant is contaminated by water or process water which could create additional corrosion damage.
- Check the bearing seats. Document the dimensions and appearance.

Tips for taking good photographs

Good photographs of any damage are essential. Not only can they can be shared with experts to help determine the root cause, but they can also be useful for increasing knowledge among mill workers and training new employees.

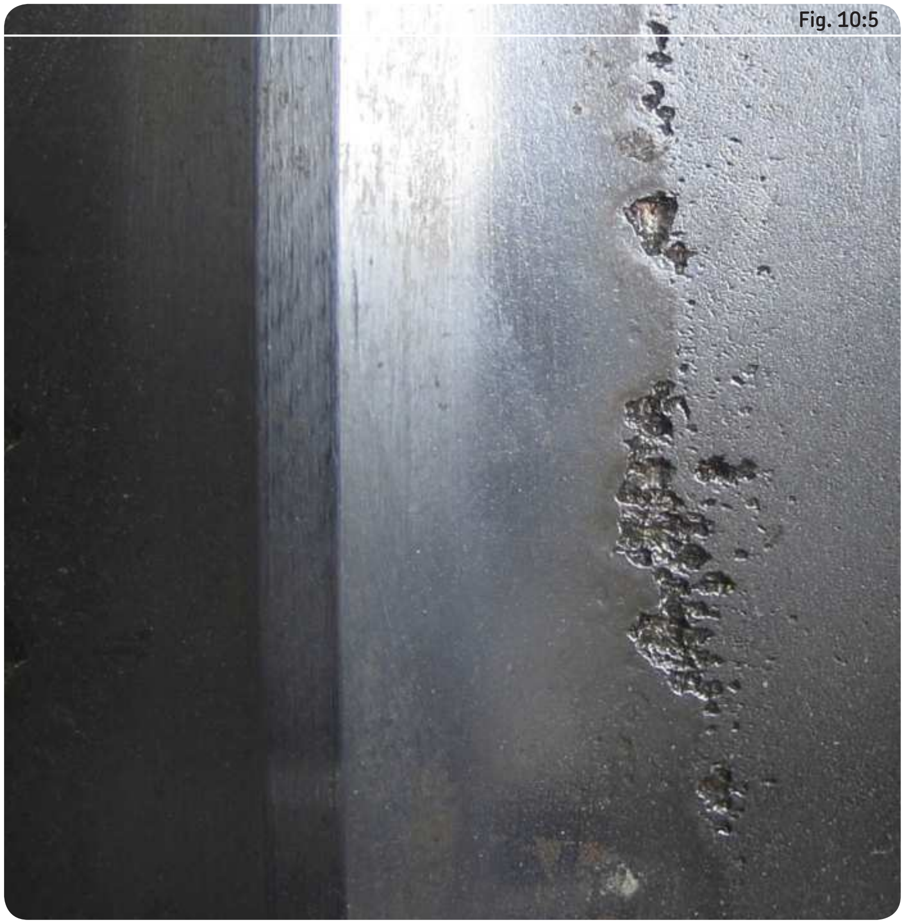
SKF receives many photographs that are not suitable for root cause failure analysis. Typical issues include out of focus images, heavy electronic noise and flash reflection. In such cases, it is difficult to judge the type of damage and identify the likely root cause and failure mode.

Lack of adequate light in mill workshops is often an issue. Cameras in automatic mode will either use flash or increase sensor sensitivity (ISO number) which creates electronic noise to avoid blurring due to movement of a handheld camera. Compare **fig. 10.5** which shows an image taken at ISO 80 and 1/5 second exposure and **fig. 10.6** which shows an image take at ISO 3200 with 1/100 second exposure. As you can see, there is significant electronic noise and detail loss on the latter. Given this, we recommend using a camera with a manual mode that allows settings to be adjusted to circumstances.

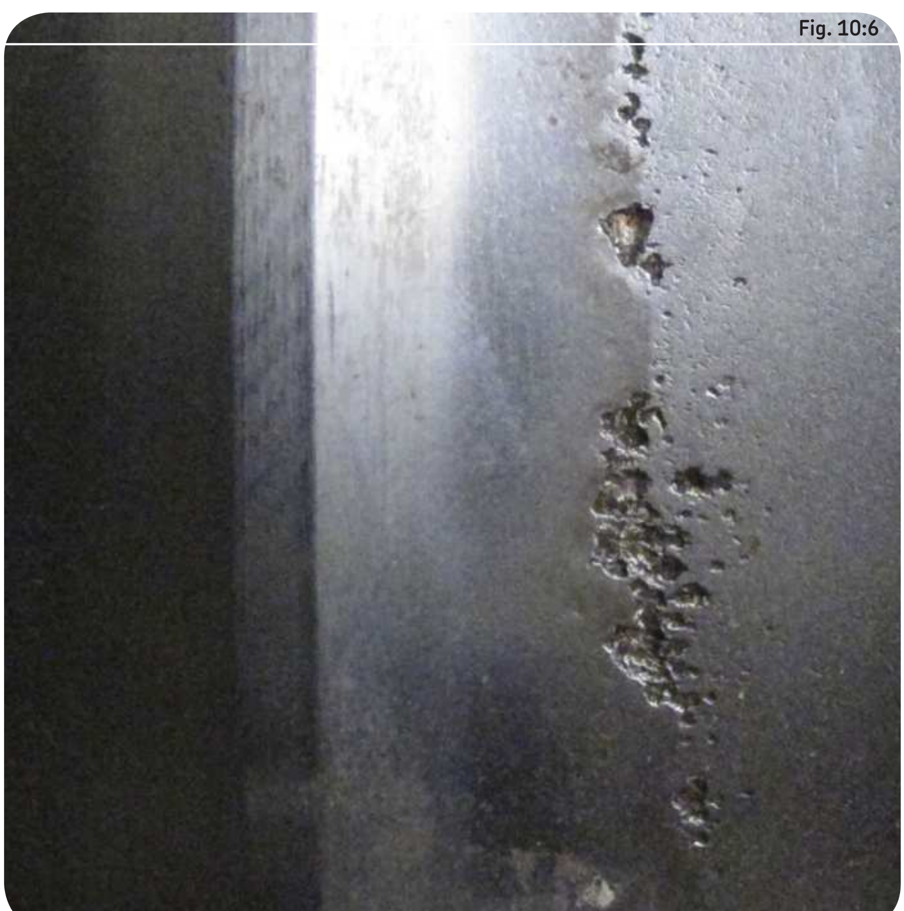
Fig. 10.4a and fig. 10.4b: A press roll bearing mounted on a journal is cut in a place without visible damage

Fig. 10:4a

Fig. 10:4b



Spalls and surface distress on the raceway of a felt roll bearing taken at f2.8 1/5 s and ISO 80



Spalls and surface distress on the raceway of the same felt roll bearing taken at f2.8 1/100s and ISO 3200

Tips:
- Take context photographs and detailed close-ups of the various parts and features that need to be documented.
- Do not use full auto mode or flash when taking photographs. Instead use manual mode to set exposure and sensor sensitivity (ISO number).
- Set the camera to take photographs in RAW format. If this is not possible, aim for JPEG with low compression.
- Use macro mode when doing close-up photographs.
- Set the camera to its native ISO. This is often the smallest ISO number.
- Use a tripod (→ **fig. 10.7**) and your camera's self-timer if the shutter speed is too low to avoid blur at low shutter speed.
- Take the photographs in an area where there are several light sources to reduce shadows.
- As the camera may not focus correctly on a steel surface due to lack of contrast, focus on a ruler (→ **fig. 10.8**) or another object such as a pencil or a finger placed next to the damage. The ruler has the advantage that it also indicates dimensions.
- After taking a photograph always look at it and zoom in to check that it is in focus.

Bearing damage investigations

Disassembling bearing components

Bearings should be disassembled with care to avoid additional damage. Some bearings such as SKF spherical roller bearings and SKF CARB toroidal bearing (except small dimension ones with glass fibre reinforced PA46 cages) can be disassembled in a non-destructive manner. This means that the majority of the SKF rolling bearings mounted on a paper machine can be disassembled without damaging them.

If a bearing component has to be damaged to allow disassembly, the component condition beforehand must be documented. Check first that the destructive disassembly will not remove clues about the cause of failure.

Tools for visual inspection

Visual inspection is the first step in bearing damage analysis. Experience shows that the naked eye is not adequate for this so a x8 magnifying glass (→ **fig. 10.9**) should be used. For most bearing damage investigations, this tool is good enough for an experienced engineer to determine the failure mode and root cause. If there are doubts about the failure mode, a x10 to x50 light optical microscope can be used (→ **fig. 10.10**).

More in-depth investigations with an electronic microscope are rarely needed. That said, they have been useful in determining some unexplained premature failures in recent years. White etching cracks (WEC) on press rolls, for example.

Using a tripod to avoid blur at low shutter speed

Fig. 10:7

Using a ruler so that the camera is able to focus properly

Fig. 10:8

Fig. 10:9

Using a x8 magnifying glass for visual inspection

Fig. 10:10

Using a light optical microscope for visual inspection

Investigation method

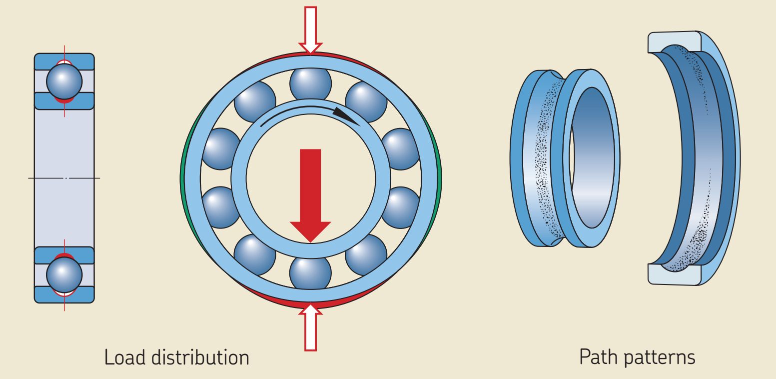
A common error is to focus only on the major damage like large spalls or fractures. As such, all bearing components and surfaces should be checked for clues that could explain the observed damage. Once the bearing is disassembled, the first step is to check the path pattern. Path patterns provide good clues about unexpected operating conditions.

For example, **fig. 10.11** shows possible oval clamping on a stationary outer ring. The outer ring has two diametrically opposed path patterns (load zone). A radially-pinched outer ring occurs for any one of the following reasons:

- The housing is not mounted on a flat surface.
- The housing is not rigid enough and deforms under load.
- The two halves of a split housing do not fit concentrically.
- Out of roundness of the housing seat.

In the case of two diametrically opposed path patterns, the bearing history should be checked. The outer ring might have been rotated 180° during a machine stop to increase bearing service life.

Fig. 10:11



Oval clamping of a stationary outer ring

Fig. 10:12

The fretting corrosion on the outer diameter of a shoe press bearing gives an indication of the load zone

Fretting corrosion in the bearing outer diameter can also give an indication on the load zone (→ **fig. 10.12**)

The second step is to search for damage on all bearing components to determine the failure mode. As a bearing can have several failure modes at the same time, all surfaces and components should be documented even if there is no damage.

Please note that if a bearing needs to be sent to SKF for investigation, the complete bearing should be sent.

For more information about path patterns and interpretation, see SKF's handbook *Bearing damage and failure analysis*.

Failures modes

The most common bearing failure modes in paper machines are:

- corrosion (mainly contact corrosion and etching)
- surface initiated fatigue (surface distress)
- indentation from debris
- abrasive wear
- premature subsurface fatigue
- adhesive wear
- fretting corrosion
- fracture (mainly as the final failure mode initiated by one or more of the types of damage listed above)

Less common are:
- electrical erosion
- overload deformation (during mounting)
- thermal cracking

The less common failures modes will not be covered in this handbook. More information on them can be found in the SKF handbook *Bearing damage and failure analysis*.

Subsurface initiated fatigue (→ **page 10:10**)
Premature subsurface failure (WEC) (→ **page 10:12**)
Surface initiated fatigue (surface distress) (→ **page 10:14**)
Abrasive wear (→ **page 10:16**)
Adhesive wear (smearing) (→ **page 10:20**)
Corrosion (→ **page 10:23**)
Indentations from debris (→ **page 10:25**)
Fracture (cracks) (→ **page 10:27**)
Fretting corrosion (→ **page 10:28**)

Subsurface initiated fatigue

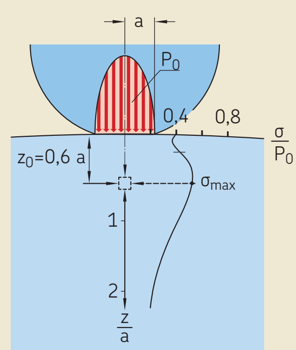
In a rotating bearing, cyclic stress induced changes occur in the microstructure beneath the contact surface of the raceway and rolling elements.

Consider the rotating inner ring of a radial bearing with radial load acting on it. As the ring rotates, one particular point on the raceway enters the loaded zone and continues through an area to reach a maximum load (stress) before it exits the loaded zone. As this happens, compressive and shear stresses occur (→ **fig. 10.13**).

Bearing steel always contains some defects/inclusions even in the highest quality steels. In small ball bearings, the stressed volume under the surface is small so it is unlikely that there is a defect of a sufficient size in the stressed volume. In this case, depending on the load, temperature and the number of stress cycles over a period of time, there is a build-up of residual stresses that cause the material to change from a randomly oriented grain structure to fracture planes.

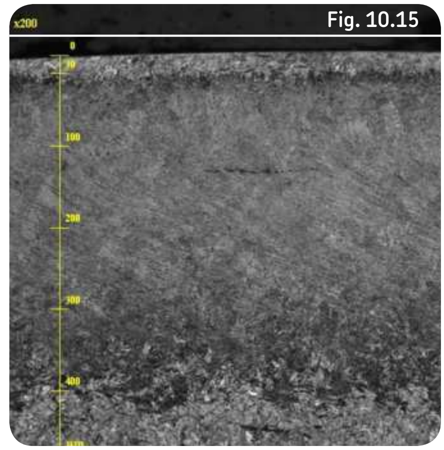
In these planes, so called subsurface micro cracks develop beneath the surface at the weakest location, around the zone of maximum shear stress, typically at a depth of 0,1 to 0,5 mm (→ **fig. 10.14** and **10.15**). The depth depends on the load, material, cleanliness, temperature and the microstructure of the steel. The crack finally propagates to the surface and a spall occurs (→ **fig. 10.16**).

Fig. 10.13



Compressive and shear stresses beneath a bearing raceway

Fig. 10.15

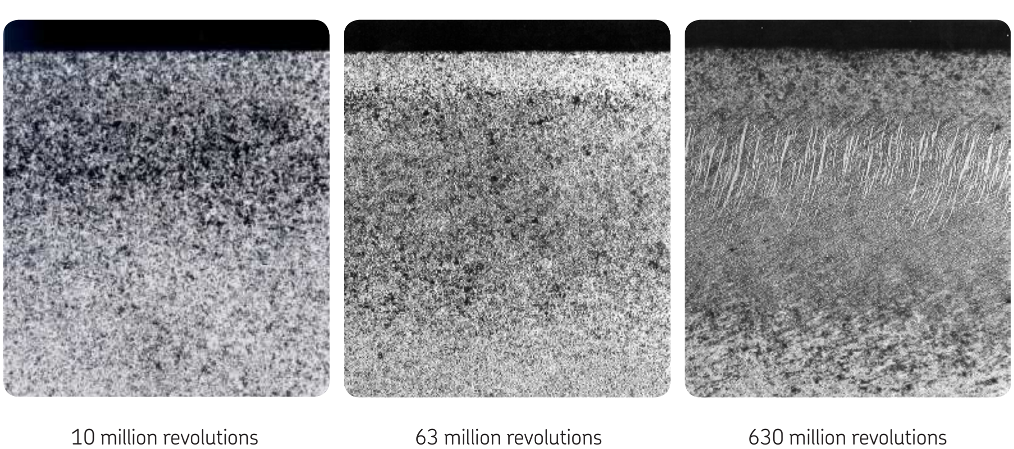


Crack development beneath the raceway surface

Fig. 10.16

Initial subsurface spalling on the inner ring of a deep groove ball bearing

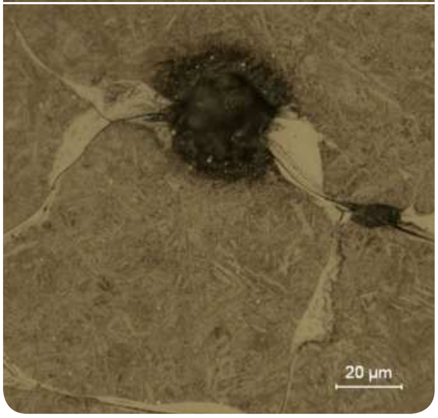
Fig. 10.14



Changes in the structure beneath the raceway surface of a highly-stressed, well-lubricated, small deep groove ball bearing over time

Butterfly i.e. micro cracks initiated at a defect/inclusion

Fig. 10.17



The bearing is damaged as soon as spalling occurs, but this does not mean that the bearing cannot remain in service. Spalling will gradually increase and noise and vibration levels will rise in the machine. The time between initial spalling and bearing failure depends on the operating conditions.

In larger ball or roller bearings where the stressed volume is much bigger, it is more likely to have a defect/inclusion (weakest link) of sufficient size in the stressed volume. The defect/inclusion acts as a stress raiser in the steel microstructure. At these weakest links, fatigue cracks are initiated and developed due to local damage caused by raised stresses under rolling contact loading. Micro cracks initiated by inclusions are also well known under the "butterfly" name (→ **fig. 10.17**). Theses cracks grow until they finally reach the surface creating a spall.

Fatigue micro cracks can propagate under the surface all the way around the bearing ring raceway before a spall occurs and will propagate very quickly when many micro cracks develop in the subsurface. This explains why no bearing defect is picked up by vibration analysis and yet the bearing fails soon afterwards with heavy spalling.

As bearings can have micro cracks under the surface, SKF recommends checking subsurfaces for micro-cracks with ultrasonic testing before remanufacturing them.

The majority of paper machine bearings fail due to surface damage rather than subsurface fatigue. This means that it is rare for a paper machine bearing to run until normal fatigue occurs. However, premature fatigue can occur when the bearing is subjected to higher loads than expected such as high residual axial load due to thermal elongation of the roll when the non-locating front spherical roller bearing cannot or does not displace axially or when a tapered roller bearing has to withstand high misalignment (→ **fig. 10.18**).

The inner ring of a tapered roller bearing on an oscillating breast roll that was subjected to high rotating misalignment. The bearing seat was not coaxial with the shaft.

Fig. 10.18

Premature subsurface failure (White Etching Cracks (WEC))

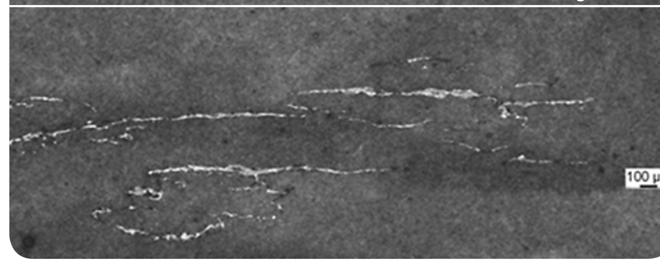
Premature subsurface failure is usually defined as having occurred when a large population of the same bearing design and size operating under the same conditions develop spalls before their calculated bearing rating life is achieved (→ **fig. 10.20**). A single instance of it happening is not sufficient to determine that premature subsurface failure has occurred (→ chapter 1: *General requirements and recommendations*).

Premature subsurface failures related to extensive micro cracks have (→ **fig. 10.19**):

- Many cracks growing in parallel where some cracks have white etching borders.
- Micro crack initiation and visible micro crack growth occurs, in contrast to normal subsurface fatigue, from many parallel stress raisers.

Premature subsurface failure is the result of unexpected loading conditions (including possible internal preloading), bearing seat deformations, weakened steel due to environmental reasons or a mix of these things. Causes of environmental weakening include water in the lubricant, electrical erosion, electrochemical erosion without micro pits, lubricant characteristics such as EP additives, oil film thickness etc.

Fig. 10.19



Microstructural features at a cross section from a region at 120 degrees in front of the spall on a press roll bearing

Premature subsurface failures are often referred to as white etching cracks, but it is important to understand that the white etching is a consequence of the problem rather than the root cause. White etching refers to the appearance of the altered steel microstructure when polishing and etching a micro section. The affected areas, consisting of ultra-fine nano-recrystallized carbide-free ferrite, appear white through a light optical microscope due to the low etching response of the material. The hardness of these areas is about 15% to 65% harder than the surrounding structural steel (→ **fig. 10.21**). Most research indicates that WEC occurs due to the rubbing of the two crack surfaces which transforms the material into ferritic steel.

The root cause that leads to premature subsurface failure is commonly related to the specific application conditions rather than the steel used in modern, high quality bearings though exceptions do exist. Therefore, root cause failure analysis should be undertaken before considering time-consuming and costly metallurgical analysis.

The bearings that may be subject to premature subsurface failure in paper machines are mainly heavily-loaded press roll

Heavy spall on shoe press bearing caused by WEC

Fig. 10.20

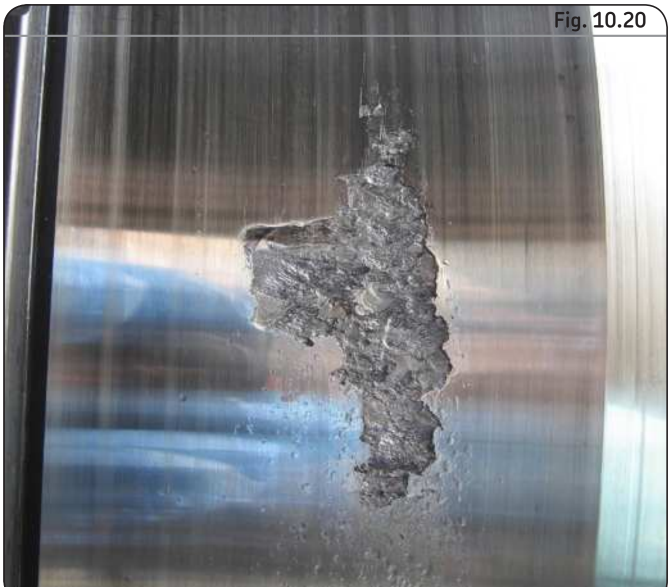
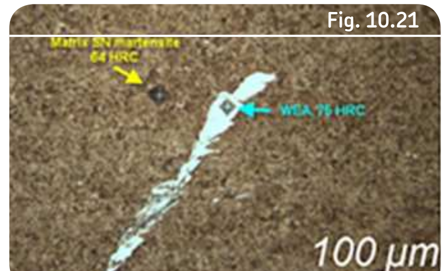


Fig. 10.21



White etching area in an SN stabilized martensitic matrix. The hardness is 75 HRC compared to the hardness of the matrix of 64 HRC

bearings. Proven solutions to this problem are to increase clearance to avoid heavy preloading during machine start up when the inner ring heats up more quickly than the outer ring and replacing front side spherical roller bearings with CARB toroidal bearings. Water content in the oil and aggressive oil additives also have a negative influence and should be avoided, where possible.

SKF black oxide coating on rings and rollers has also been used to extend the service life of bearings subjected to premature subsurface failure. Please contact your local SKF application engineering department for more information.

Surface initiated fatigue (surface distress)

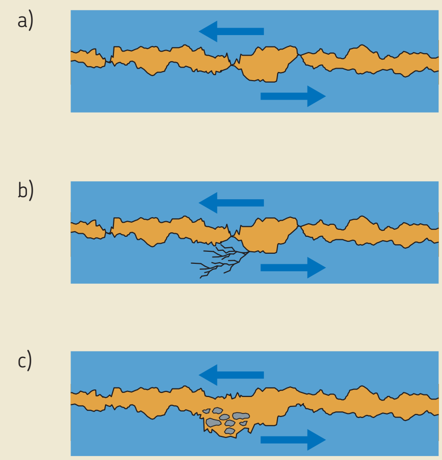
Basically, surface initiated fatigue is the result of damage to surface asperities in the rolling contact and is normally caused by inadequate lubrication. Whenever the oil film does not fully separate the rolling contact surfaces, there is a risk of surface initiated fatigue.

Inadequate lubrication can be caused by a number of different factors. If a surface is damaged, by the over-rolling of solid contaminants for example, it is likely that lubrication is no longer optimal as the lubricant film is reduced or becomes inadequate. This also occurs if the amount or type of lubricant is not appropriate for the application and the contact surfaces are not adequately separated.

Water in the lubricant can make it inadequate. This is a common cause of surface distress in rolling bearings mounted on paper machines. The resulting metal-to-metal contact causes the surface asperities to shear over each other which together with the micro slip that occurs between the rolling contact surface areas creates a burnished or glazed surface. After this, micro cracks may occur at the asperities followed by micro spalls before finally leading to surface initiated fatigue (→ **fig. 10.22**).

The risk increases if there is sliding in the rolling contact area. All rolling bearings show some micro slip, which is also sometimes known as micro sliding, in the contact area due to their specific geometry and elastic deformation of the rolling elements and raceway under load.

Fig. 10.22

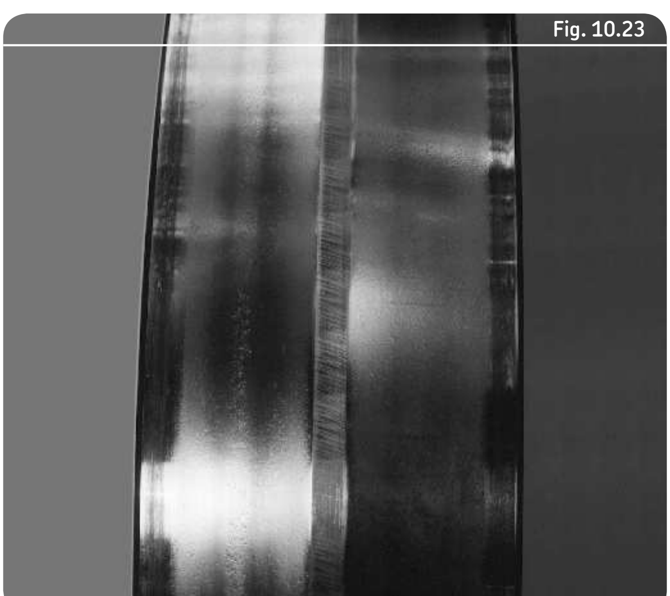


The mechanism of surface initiated fatigue

Another frequently overlooked cause of surface initiated fatigue is the use of EP additives. EP additives can become aggressive, especially at elevated temperatures, and can accelerate micro spalling. The presence of water can increase the negative effect of aggressive EP additives. Therefore, it is not recommended to use EP additives unless they are really necessary.

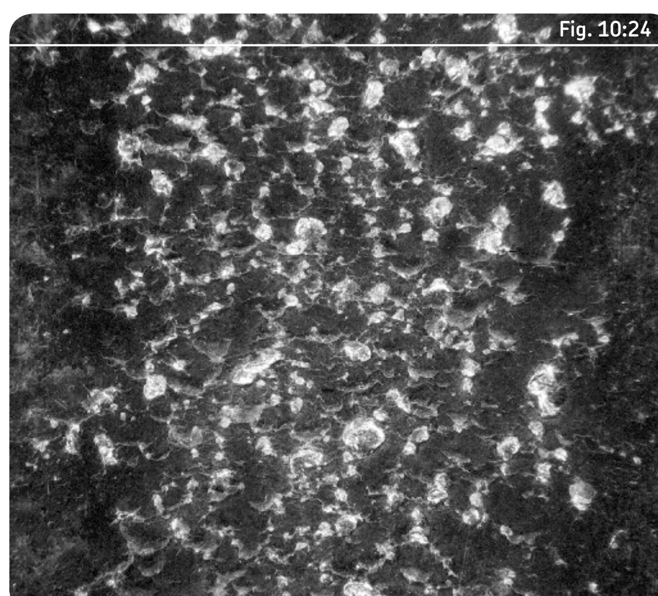
Surface initiated fatigue is generally the consequence of surface asperities coming into direct contact under mixed or boundary conditions (→ **fig. 10.22 a**). When the loading and the frictional forces reach a given magnitude, small cracks form on the surface (→ **fig. 10.22 b**). These cracks may develop in to micro spalls (→ **fig. 10.22 c**).

Fig. 10.23



Micro spalls on the inner ring raceway of a spherical roller bearing

Fig. 10:24



Highly magnified micro spalls and cracks on a raceway surface

Normally, these micro spalls are only a few microns in size and the surfaces looks dull and grey (→ **fig. 10.23**). Only under a microscope can the cracks and spalls be detected (→ **fig. 10.24**)

Fig. 10.25 shows the inner ring of a felt roll bearing that was contaminated by process water. The contamination level was enough to cause inadequate lubrication and for surface distress to develop, but not enough to cause corrosion.

To avoid surface distress, SKF recommends increasing oil film thickness and/or reducing solid contamination as a first step. The oil film thickness value, κ, should be between 2 and 4. However, in some applications like drying cylinders, Yankee cylinders and other heated rolls, the κ value can be below 1. In such cases, there is a special recommendation that can be found at the end of this chapter.

SKF recommends keeping the water content in the lubricant below 200 ppm.

Surface distress on a felt roll bearing contaminated with process water

Fig. 10.25

Abrasive wear

Abrasive wear means the progressive removal of material. Initially, most bearings experience some very light wear during the first period of their life corresponding to the path pattern.

Most of the time real abrasive wear occurs due to inadequate lubrication or the ingress of solid contaminants.

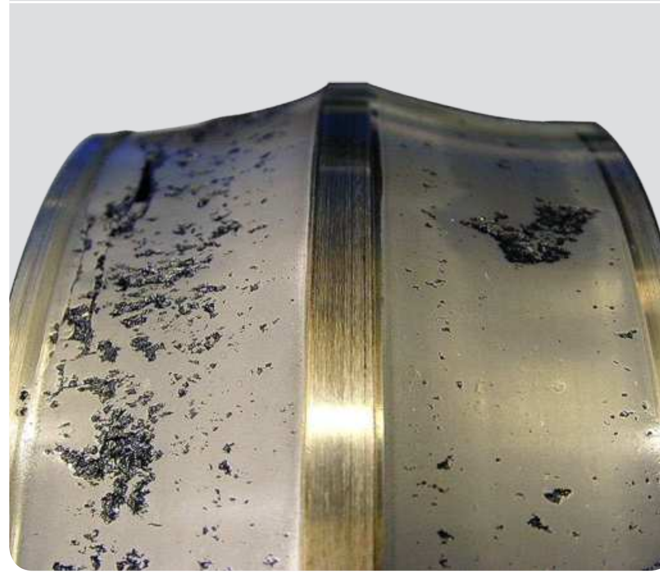
Abrasive wear is a degenerative process that eventually destroys the micro geometry of a bearing because wear particles further reduce the lubricant effectiveness. Abrasive wear particles can quickly wear down the raceway of a ring and rolling elements, as well as cage pockets.

Fig. 10.26 shows extreme abrasive wear on a spherical roller bearing inner ring mounted on a drying cylinder. Note that this bearing did not originally have flanges. In this case, the root cause was too low oil flow as a result of high temperature steam supplied to an uninsulated journal combined with excessive axial load on a non-locating bearing that was not moving in its housing to accommodate thermal elongation of the drying cylinder.

Fig 10.27 shows the inner ring of a spherical roller bearing where initial abrasive wear caused by contamination has developed into secondary damage in the form of surface distress and spalling.

Fig. 10.28 shows how micro slip occurs in spherical roller bearings which are the most common type of bearings used in paper machines. The roller surface speed will be highest in the middle of the roller where the diameter is greatest. The diameter at the end of the roller is smaller which means a lower surface speed. This difference in surface speed will lead to the micro slip in the rolling contact. In the points with true rolling, there is no micro slip and, thus, no wear. While in rolling contact, the distance slipped is short and long scratches will not be produced by contaminants although small particles will abrasively wear the raceway surfaces where micro slip occurs. **Fig. 10.29** shows a Yankee bearing where abrasive wear has led to fatigue spalling. Two ridges remain where true rolling has taken place. Where micro slip has occurred, there has been abrasive wear. When loading is concentrated at these ridges, they become overloaded and fatigued i.e. premature spalling occurs. The oil is almost certain to be equally contaminated in these true rolling zones, but there is negligible abrasive wear. The two bands where true rolling has occurred are clearly visible in the photo. Note that the bearing in the photograph is of an old design that is no longer produced by SKF.

Fig. 10.27

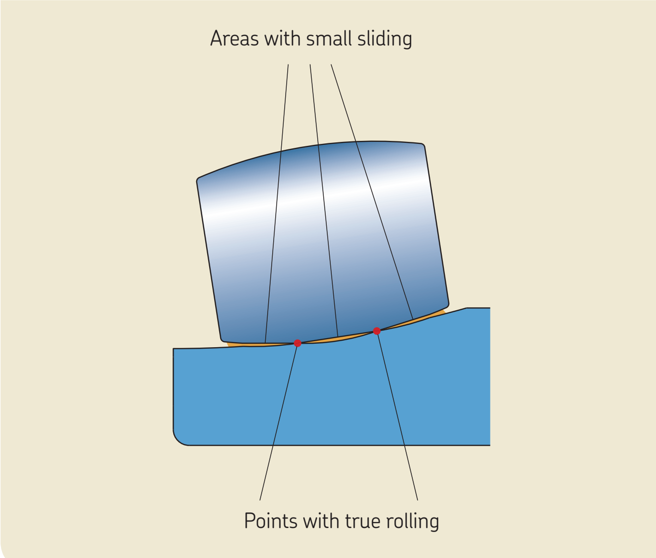


Spherical roller bearing inner ring. Abrasive wear develops into surface distress and spalling

Fig. 10.26

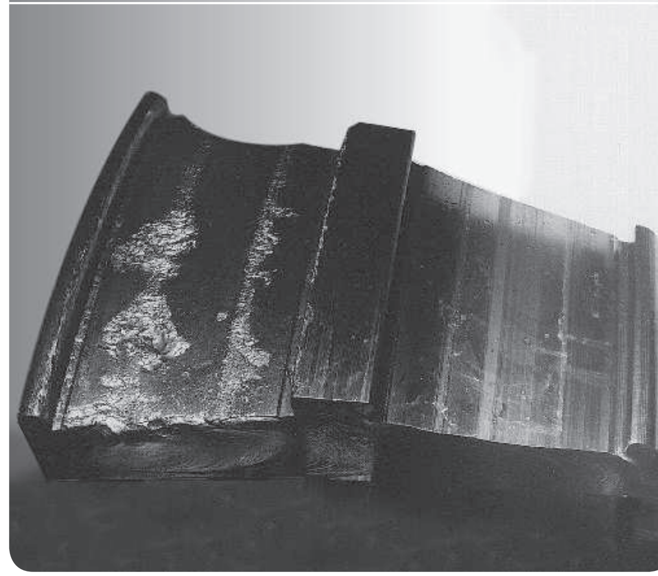
Extreme abrasive wear on the inner ring of a drying cylinder bearing

Fig. 10.28



Small sliding (micro slip) in spherical roller bearings

Fig. 10:29



Overload and spalling caused by wear

Under inadequate lubrication conditions, unhardened components such as cages may be the first to have signs of wear. Massive brass cages are very sensitive to poor lubrication and/or fine solid contamination. The pressed steel cages used for many SKF spherical roller bearings and SKF CARB toroidal bearings are less sensitive. This is not the case for all types of bearings. SKF E type spherical roller bearings and the spherical roller bearings for vibrating applications (i.e. VA405 or VA406 suffix) have hardened pressed steel cages which are even less sensitive to poor lubrication and/or fine solid contamination (→ **fig. 10.30**).

The test conditions for the cages shown in **fig. 10.30** were:

- Spherical roller bearing size: 22320
- C/P = 14 and 1 680 r/min
- Oil drip lubrication for the first 24 hours with the oil flow stopping after that
- Bearing run to failure or test stopped when temperature reaches 140 °C

Oil starvation test results with an SKF E type hardened pressed steel cage on the left

Fig. 10:30

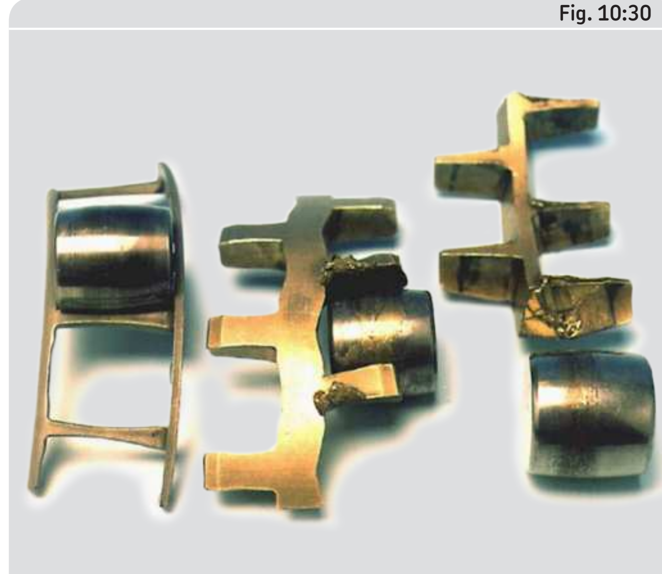


Fig. 10:31

Deep groove ball bearing cage with burrs due to abrasive wear due to a too long relubrication interval

Fig. 10:32

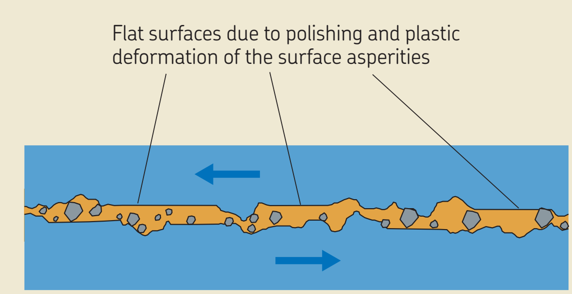
Polishing wear on the outer ring of a spherical roll bearing mounted on a felt roll in the dryer section

Fig. 10.31 shows a deep groove ball bearing cage with sharp edges due to abrasive wear due to a too long relubrication interval. Sharp burrs are created on the edges of the rolling element/cage contact which can injure the mechanic that handles the damaged bearing.

Polishing wear is a special form of abrasive wear. The raceway surfaces of new bearings are shiny but not highly reflective. Real mirror-like surfaces in a bearing that has been in service (→ **fig. 10.32**) result from inadequate lubrication caused by a thin oil film and particles that are acting as a polishing agent. This allows metal-to-metal contact which leads to abrasive wear and plastic deformation of the asperities (→ **fig. 10.33**). The surfaces may become extremely shiny, but this depends on the size of the particles, their hardness and the running time.

The process of polishing wear in a bearing

Fig. 10.33

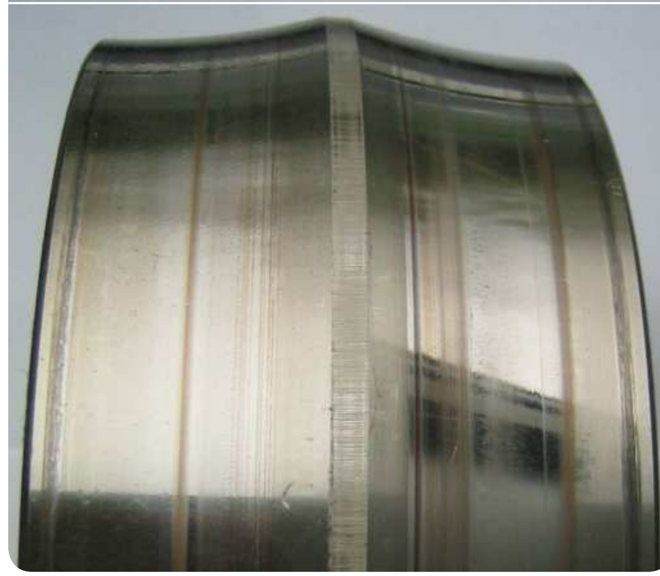


Mirror-like surfaces can be advantageous provided that abrasive wear and plastic deformation are confined to the asperities only. In some cases, polishing wear can go beyond the asperities to severely alter the shape of the raceway. This level of abrasive wear results from a combination of factors such as the viscosity of the oil being too low and the presence of an excessive amount of very small abrasive particles in the oil. Other factors can include a combination of low speed, heavy loads and an insufficient oil film. **Fig. 10.34** show polishing wear due to low speed and insufficient oil film due to a lack of grease base oil bleeding. The brown bands indicate high temperature that oxidized the lubricant. This particular case was a dryer section felt roll bearing at basement level (i.e. a lower temperature than under the hood) lubricated with a high temperature grease that didn't bleed enough oil at normal operating temperature.

To avoid abrasive wear, SKF recommends first to increase oil film thickness and/or reduce solid contamination. The oil film thickness value, κ, should be between 2 and 4. However, in some applications like drying cylinders, Yankee cylinders and other heated rolls, the κ value can be below 1. In such cases, there is a special recommendation that can be found at the end of this chapter.

Anti-wear (AW) additives in the lubricant may also improve the situation. Note that EP additives are not anti-wear additives and can be harmful to the bearing at higher temperatures. As such, SKF recommends using EP additives only when necessary.

Fig. 10:34



Polishing wear due to insufficient grease base oil bleeding

Adhesive wear (smearing)

Adhesive wear is a type of lubricant-related damage that occurs between two mating surfaces sliding relatively to each other (→ **fig. 10.35**).

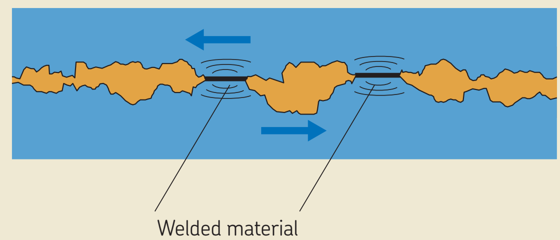
Adhesive wear is characterized by the transfer of material from one surface to another (smearing). It is typically accompanied by frictional heat which can sometimes temper or reharden the mating surfaces. The frictional heat produces local stress concentrations, which can cause cracking or spalling in the contact area.

Smearing is not common under normal bearing operating conditions. The relative sliding speed must be much higher than the micro slip induced by the bearing geometry and elastic deformation in the rolling contact area.

Smearing is generally caused by severe acceleration or too low loads (→ **fig. 10.36**). Smearing can also be to the result of inadequate lubrication and/or be the final stage of the bearing failure (→ **fig. 10.39**, **page 10:22**).

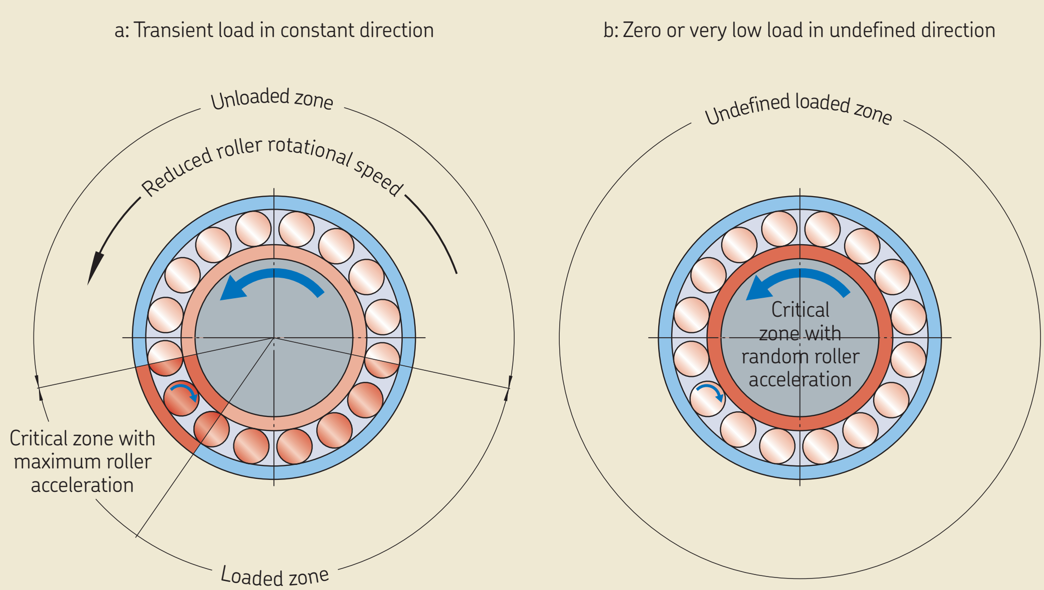
To roll without skidding or sliding, a roller needs to be squeezed between the inner ring and the outer ring. The roller has a certain mass and thus a certain inertia that makes it reluctant to accelerate. When exiting the loaded zone of the bearing, the roller slows down since there is friction against the cage and in the lubricant in the unloaded zone. When the roller enters the loaded

Fig. 10.35



The principle of adhesive wear

Fig. 10.36



The main root cause for roller sliding leading to smearing

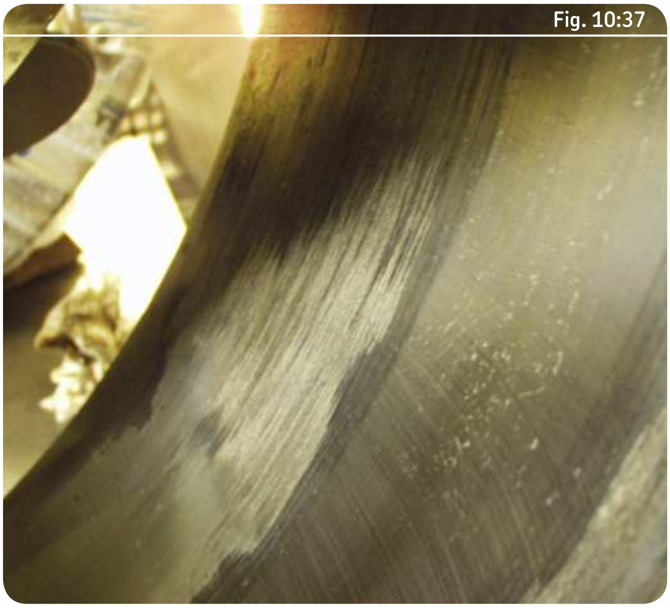


Fig. 10:37

Smearing marks in the entrance to the loaded zone

Fig. 10:38

Micro smearing marks on the rollers of a deflection-compensating press roll

zone again, it is squeezed between the rings and forced to accelerate. If there is a large difference between the speed of the rollers in the unloaded and loaded zones and if there is insufficient load in the loaded zone, it will accelerate more slowly than required and create smearing marks in the entrance to the loaded zone (→ **fig. 10.36 a** and **fig. 10.37**).

The other cause is when the load zone has very light roller loads or when the load zone isn't well defined due to the roll or cylinder weight being supported by the nip load rather than the bearings. This happens in some deflection-compensating press rolls as the radial load is carried by both the hydrostatic shoe bearings and the roller bearings. Depending on the oil pressure in the hydrostatic shoe bearings, roller bearings can face an undefined load situation with the possibility of zero load. As the load is low, there is a risk that that rollers are not always forced to roll at an adequate speed. Rollers slide, accelerate and decelerate in the insufficiently loaded load zone or in the undefined load zone (→ **fig. 10.36 b** and **10.38**). Note that this can also be an issue for bearings in some suction rolls, press rolls, calendar and – more rarely – wire and felt rolls. Such smearing due to light load generally leads to spalling, but it can also lead to catastrophic failure.

In case of inadequate lubrication, if energy is dissipated in the metal-to-metal contacts creating very high local temperature and smearing, the contact surfaces will get progressively rougher and rougher. As the rough surface leads to a decreased oil film with a lot of metal-to-metal contact, damage will get progressively worse. Smearing can lead to serious damage like blocked rollers due to a broken cage.

The best way to avoid smearing in normally loaded bearings is to increase the oil film thickness and/or use lubricant with good EP characteristics. If this does not resolve the problem, SKF recommends NoWear coating on the rollers (suffix L5DA) or SKF specific black oxide coating on rollers and raceways (suffix L4B).

For bearings that are not sufficiently loaded, SKF recommends NoWear coating on the rollers or reducing the number of rolling elements. It is not recommended to increase lubricant viscosity in order to increase oil film thickness since it would slow the rollers in the unloaded zone even more. Reducing the number of rolling elements in a bearing with high operating internal clearance should be approached with caution as there will be a negative influence on the radial run-out. Reducing the rolling element mass with ceramic rolling elements is a solution that is already used in spreader rolls. Reviewing lubricant choice or even using a smaller bearing, if possible, are other solutions.

Water content in the lubricant has also an important influence. SKF recommends keeping the water content in the lubricant below 200 ppm.

Fig. 10:39

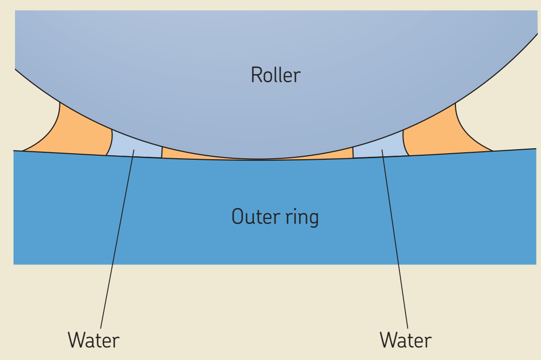
Rollers welded by smearing to the inner ring. This is the final stage after inadequate lubrication, abrasive wear and surface distress. Contamination and poor lubrication created heavy cage wear, the cage dropped and its bars got trapped between the rollers.

Corrosion

Corrosion is perhaps the most common reason for short service life in paper machine bearings. Bearing applications in paper machines are exposed to the ingress of water especially in the forming and press section of the machine. Water, either from the papermaking process itself or from machine cleaning, is very dangerous. The risk is highest in non-rotating bearings e.g. in the roll store or during a long machine stop. If a non-rotating bearing has free water in the lubricant, this water will accumulate at the bottom of the bearing. The concentration of the water will be highest at a certain distance from the rolling contact (→ **fig. 10.40**). The reason is that the free water in the oil, being heavier, will sink until it comes to a suitable gap between the roller and the raceway. Corrosion due to water or other aggressive liquid at and around the contact point between rollers and raceways is called contact corrosion, standstill corrosion or etching.

Fig. 10.41 shows corrosion on a spherical roller bearing inner ring under static conditions. The corrosion marks are separated by the same distance that separates two adjacent rollers. There is enough water in the lubricant to cause contact corrosion marks, but not enough to significantly corrode the other parts of the raceways that are protected by the lubricant. In roller bearings, severe corrosion and/or etching lead to spalls that propagate first in an axial direction. **Fig. 10.42** shows a suction press roll bearing that suffered from too much water in the lubricant.

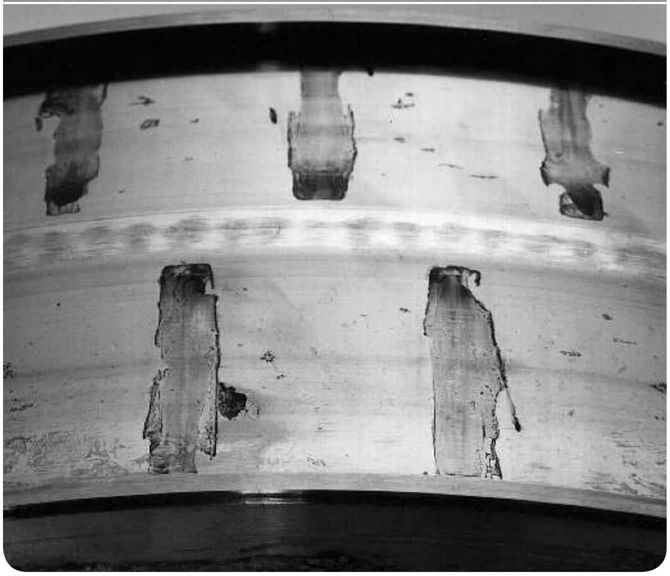
Fig. 10.40



Water concentration in a stationary bearing

Standstill corrosion under static conditions

Fig. 10:41



Suction press roll bearing with too much water in the lubricant

Fig. 10:42

The bearings in the dryer section which are subjected to high temperatures and aggressive chemicals (i.e. oil ageing) may develop a type of corrosion referred to as etching. Some EP additives have proven to be aggressive at high temperature.

Superficial contact corrosion can be removed by normal operating, just like polishing. Deeper contact corrosion usually leads to premature fatigue and extended spalling as the material is subjected to structural changes and the area of the load carrying surface is reduced to such an extent that overloading occurs. It relatively common for contact corrosion in a bearing to be removed by operation except for one or two deeper instances that lead to spalling (→ **fig. 10.43**)

The best way to avoid corrosion is to keep the lubricant free from water and to use lubricant with good rust-inhibiting additives.

Black oxide coating can reduce corrosion and etching damages, but is not an anti-corrosion coating.

Be aware that lubricant at operating temperature can contain dissolved water which becomes free when the machine cools down. As such, the lubricant should have less than 200 ppm water content before the machine stops.

Fig. 10:43

Spalling created by a single instance of deep-seated contact corrosion on a drying cylinder bearing. The spall has propagated first in the axial direction at the initial damage and then has propagated in the rolling direction (left to right).

Indentations from debris

Solid contaminants can be introduced into a bearing during mounting or via the lubricant or ineffective seals. Most bearings in paper mills are contaminated by hard particles during mounting or even during inadequate storage (→ **fig. 10.44** and **10.45**). If a circulating oil lubricated bearing is contaminated before going into operation, an oil filter with a good filter rating will not prevent damage. Unfortunately, the oil flow will not remove all the particles before the machine starts and one particle is enough to create a dent that will shorten the bearing service life.

Hard particles can also be the result of wear or damage of an adjacent component e.g. a gear.

When a solid contaminant is over-rolled by the rolling elements, it is pushed into the raceway and causes an indentation. The particle producing the indentation does not need to be hard. Even rather soft particles, when large enough, can be harmful.

Raised material around the edges of an indentation will initiate fatigue when subsequently over-rolled. When the fatigue level reaches a certain point, it leads to premature spalling originating at the back end of the indentation (**fig. 10.46**). The spall starts as a surface crack.

Fig. 10:44

A large black oxidized spherical roller bearing for a shoe press that has been stored out of its packaging and cleaned with dirty rags before mounting.

An example of inadequate storage in a paper mill

Fig. 10:45

Spalling starts at the back end of an indentation

Fig. 10.46

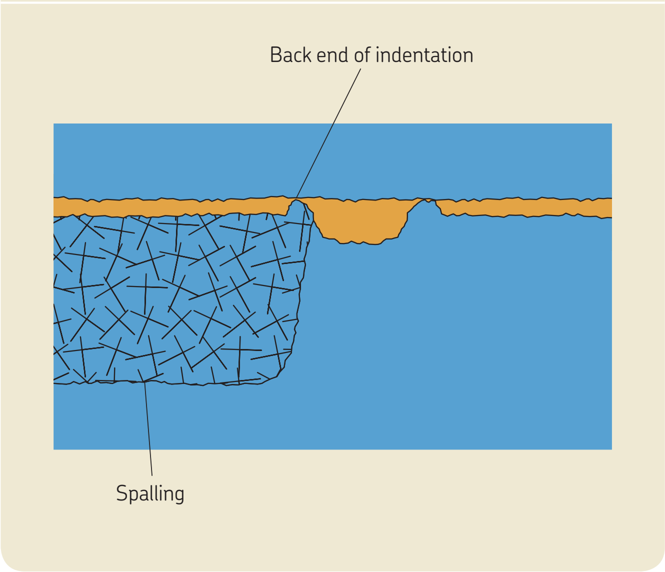


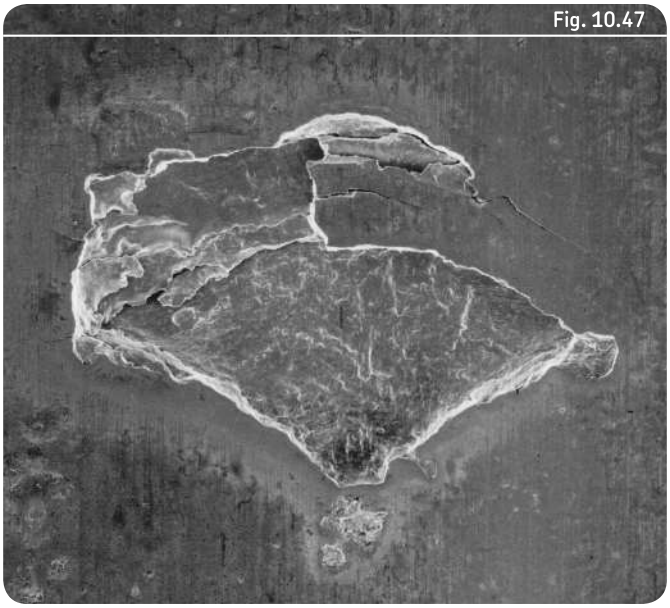
Fig. 10.47 shows spalling in a deep groove ball bearing resulting from an indentation. The over-rolling direction is from bottom to top. The V-shape is a typical sign of indentation damage in a bearing where the initial spalling opens up from back end of the indentation.

Fig. 10.48 clearly shows the consequences of indentation on a spherical roller bearing inner ring. The over-rolling direction is from right to left. A rather large and soft contaminant was trapped in a raceway and over-rolled. At the bottom of the dent, grinding lines are still visible. Also notice the raised rim, partly polished away, around the dent. To the left, behind the dent, there is a large black coloured spall where material has been detached. There are also some cracks where material is about to be detached.

The SKF Rating life method makes it possible to calculate the life reduction caused by solid contaminants. The most important data required for the calculation is the bearing type and size, rotational speed, bearing load, viscosity ratio and the contamination factor.

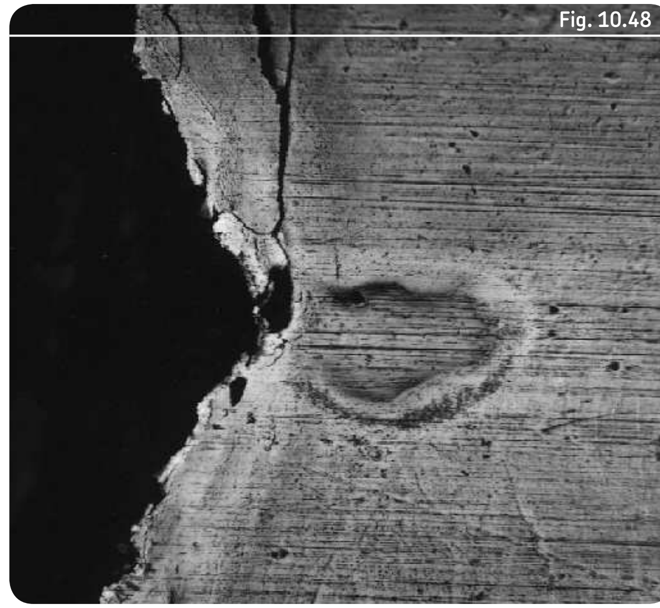
Spalling resulting from an indentation in the inner ring of a deep groove ball bearing

Fig. 10.47



Spalling resulting from an indentation in a spherical roller bearing

Fig. 10.48



Fracture (cracks)

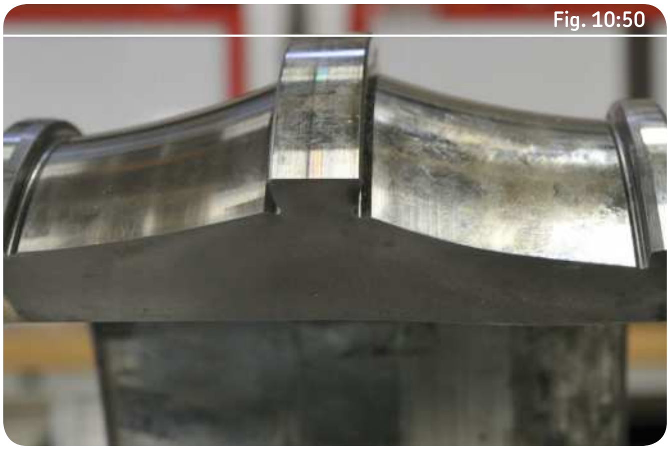
Normally some kind of initial damage is needed to cause ring fracture. Examples of initial damage are surface distress, indentations, smearing, fretting corrosion, corrosion (→ **fig. 10.49**), handling marks and mounting damage. Inadequate bearing inner ring heat treatment for heated cylinder applications is also a common cause (→ **fig. 10.50**). On heated cylinders, inner ring fracture often occurs during machine start up from cold.

The cracked inner ring of a drying cylinder bearing after ten years in service. The crack was caused by a deep spall. Too high drive-up during mounting, too thin an oil film, etching and contact corrosion led to fatigue and then spalling.

Fig. 10:49

The inner ring of a bearing with martensitic heat treatment after one year in service which cracked during start up. Note that there is no raceway surface damage.

Fig. 10:50



Fretting corrosion

Fretting corrosion occurs when there is relative movement between a bearing ring and its seat on a shaft or in the housing. Relative movement can be very small and occur due to the unique deformation of the rings under load, for example. The higher the load, the tighter the fit needs to be to avoid relative displacements. For bearings mounted with tight fits on very large bore hollow suction press roll journals (→ **fig. 2.15, page 2:10**), nip load will ovalise the journal and thus also the bearing inner ring. The tight fit in such applications must be higher than generally recommended to avoid relative movements between ring and shaft during rotation.

The relative movement may cause small particles of material to become detached from the bearing surface and its seat. These particles oxidize quickly when exposed to air and the result is iron oxide (→ **fig. 10.51**). Iron oxide is larger in volume than iron and steel. As a result of the fretting corrosion, the bearing may not be evenly supported, which can have a detrimental effect on the load distribution in the bearing. Iron oxide particles will also act as abrasive particles increasing wear rate and loss of the tight fit. Corroded areas also act as fracture notches.

Fretting corrosion appears as areas of rust on the outside surface of the outer ring or on the bore of the inner ring. The raceway path pattern could be heavily marked at corresponding positions. In some cases, fretting corrosion is actually secondary damage due to heavy spalling on the raceway.

Depending on the chemical reaction, corrosion could appear as:

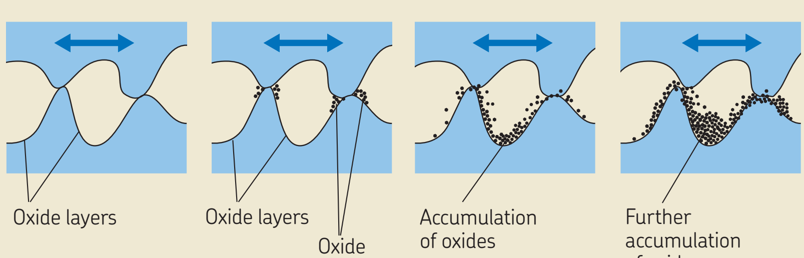
- red (hematite, Fe_2O_3)
- black (magnetite, Fe_3O_4)

Fig. 10:52

Fretting corrosion due to an insufficiently tight fit after felt tension increase

Fig. 10.52 shows fretting corrosion on a spherical roller bearing inner ring bore that was mounted on a felt roll in the dryer section. The bearing was mounted on an adapter sleeve. The fretting corrosion occurred on several bearings after replacing the drying cylinder gear drive with a silent drive which meant that felt tension was increased. The tight fit was not enough for the new operating conditions. The issue was solved by increasing the drive-up.

Fig. 10.51



Schematic view of different stages in development of fretting corrosion

Fig. 10.53 shows fretting corrosion mainly on one side of the outside diameter of the outer ring of a drying cylinder bearing. The fretting corrosion is due to the spall on the outer ring raceway seen on **fig. 10.43**, **page 10:24** in a corresponding position.

Fretting corrosion damages not only the bearing outer surfaces, but also the bearing seats on the journal or in the housing. A new bearing mounted on damaged seats due to heavy fretting corrosion will likely have its service life decreased. This is especially true when the bearing has to displace axially to cope with axial thermal elongation. Fretting corrosion increases friction between the bearing and its seat and thus increases residual axial load on the bearing.

To avoid fretting corrosion or to slow down the process, the tolerance (fit) should be adjusted or a special anti-fretting paste (e.g. SKF LGAF 3E) or coating (e.g. SKF specific black oxide) can be applied.

SKF does not recommend the use of specially formulated adhesives to avoid fretting corrosion.

Fretting corrosion on the left due to the spall seen on fig. 10.43, page 10:24

Fig. 10:53

How to avoid raceway and roller surface damage

As previously mentioned, bearings in paper machines are susceptible to different types of surface damage which is mainly caused by water ingress and insufficient lubrication. As such, damage is often a combination of two or more failure modes and it is difficult to propose a single action which can prevent the damage. SKF recommends following the guidelines in this handbook to reduce the risk of surface damage.

Table 10.1 summarizes suitable actions when different types of surface damage have occurred. More information can be found in SKF's handbook *Bearing damage and failure analysis*.

In extreme cases, like smearing due to low load in calenders, NoWear bearings can be a problem solver. Please contact SKF for further details.

Changes in running conditions, especially speed and temperature, result in oil film thickness (κ value) variations. So even if κ = 1 is considered as adequate lubrication, it is preferable to choose a lubricant to reach a κ value between 2 and 4.

Sometimes it is not possible to reach the recommended κ value. In such cases, SKF recommends checking the minimum acceptable κ value, called κ_{min}. The κ_{min} is based on field experience and is restricted to paper machines and oil lubrication only.

$$\kappa_{min} = n\, d_m / 80\,000$$

Where

κ_{min} = minimum value for κ

n = rotational speed, r/min

d_m = bearing mean diameter, mm

Recommended actions to avoid raceway and roller surface damage. Note that this table does not cover subsurface related failures.

Table 10.1

Recommended actions	**Type of damage** Abrasive wear	 Polishing wear	 Surface distress	 Adhesive wear at normal load	 Adhesive wear at load below minimum recommended	 Corrosion and etching	 Inner ring fracture
Increase oil film thickness (κ-value)	X	X	X	X			
Improve AW additives	X	X	X				
Improve EP additives			X	X	X		
Decrease activity of EP additives			X			X	
Improve water extraction			X	X		X	
Improve particle removal	X	X	X				
Improve rust inhibitor additives						X	
NoWear coating	(X)		(X)	X	X		
Reduce number of rolling elements					X		
Black oxide coating			(X)	(X)	(X)	(X)	
SKF new generation steel (WR) or case hardened inner ring (HA3)							X

(X): can be an option if other solutions are inadequate or impossible. For advice, please contact your local SKF application engineering department.

Notes:

1 If the calculated κ_{min} is below 0,25, κ_{min} = 0,25 should be chosen.

2 The equation for the κ_{min} value is based on the risk of surface distress and smearing which, in turn, are dependent on the bearing size, load and speed. Abrasive wear might occur even if κ is above κ_{min}.

3 The κ_{min} value cannot always be reached in drying and Yankee cylinders without journal insulation or with very high steam temperatures. In such cases, the bearings are subjected to a higher risk of surface distress. High filtration rate and lubricant cleanliness are extremely important in such cases.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Industrial bearing remanufacturing services from SKF

Extending the service life of bearings to reduce costs, downtime and environmental impact

Certain application conditions, like contamination or sporadic metal–to-metal contact in the rolling contact zone, can cause all sorts of damage to bearings. Wear, rust, indentations and microcracks, for example. As a result, the actual service life of a bearing is often shorter than its potential service life.

The alternative is to apply a controlled remanufacturing process before any major damage or bearing failure occurs. This can substantially prolong the service life of the bearing in question, reducing costs and lead times if a new bearing isn't available in time. And since it requires less energy than making a new bearing, it is better for the environment as well.

The typical candidates for bearing remanufacturing are large size bearings mounted in suction rolls, press rolls, calendar rolls and Yankee cylinders.

Benefits

The SKF professional bearing remanufacturing services available worldwide can bring advantages and benefits like:

- Reduced total life cycle cost.
- Extended bearing service life.
- Reduced machine downtime.
- Reduced environmental impact.
- Maintained stock of replacement bearings.
- Improved overall asset reliability.

Cost-benefit analysis shows that significant cost savings can be achieved by remanufacturing bearings though this is dependent on a number of factors such as bearing condition and size. In addition, up to 90 per cent less energy is needed to remanufacture a bearing compared to making a new one. Furthermore, SKF's remanufacturing services help protect the environment by responsible cleaning of used bearings and handling of waste.

How SKF remanufactures bearings

Experienced bearing analysts evaluate the bearings and define which remanufacturing process will be the most efficient for restoring them considering the application requirements. Through SKF's remanufacturing processes, relevant functional surfaces are repaired including, if necessary, the replacement of bearing components. As a consequence, the potential service life of the bearing can be fully exploited, as illustrated in **diagram 11.1**. A remanufactured bearing is not a new bearing, it is a bearing whose service life has been increased.

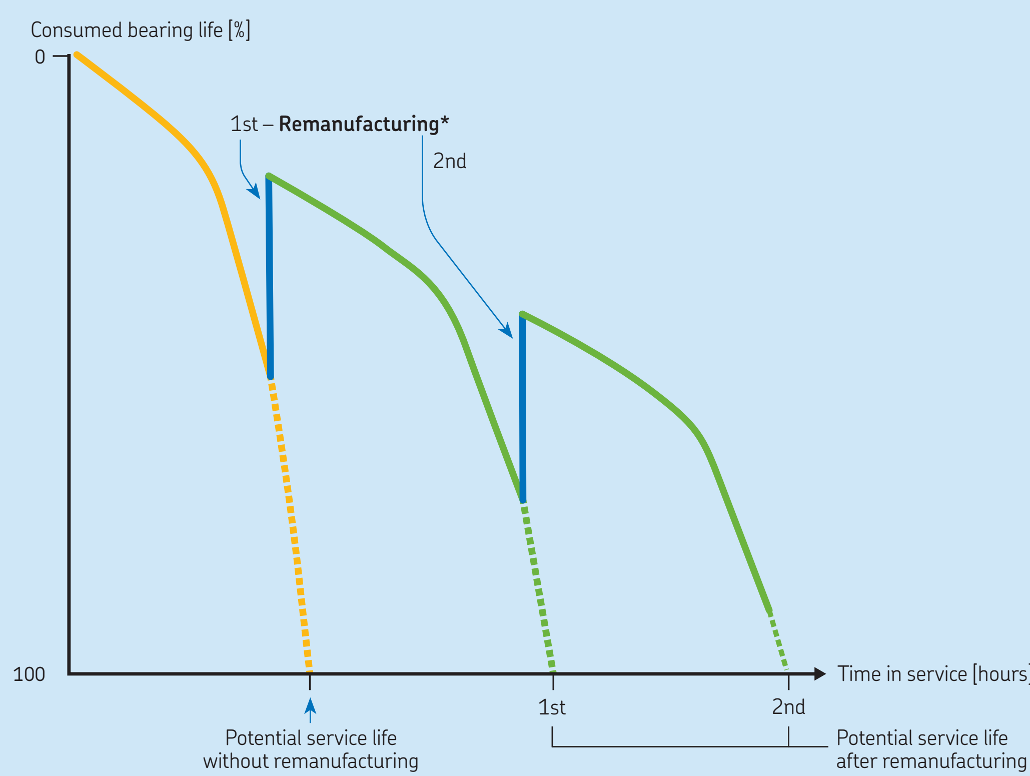
SKF applies the same manufacturing standards, processes, equipment, quality assurance, knowledge and competences used when making new bearings to its bearings remanufacturing service. This includes acceptance criteria that deliver high quality results even when extensive remanufacturing is needed.

To provide full traceability, SKF has developed and uses an advanced management system. Remanufactured bearings are given unique markings that allow them to be tracked during their subsequent life cycle.

In addition to standard remanufacturing, SKF can also remanufacture bearings to a new or higher specification. This can include mounting sensors and the provision of other enhancements such as integrated lubrication, sealing solutions and rework to other specifications.

Diagram 11.1

Remanufacturing: positive impact on service life



*) If possible

Remanufacturing increases the bearing service life

Overview of the SKF bearing remanufacturing process:

- Registration in the management system
- Pre-cleaning and pre-inspection
- Disassembly and components washing
- Component inspection (non-destructive testing, ultrasonic test is an option)
- Component measurement
- Repairability conclusion and report
- Quotation
- Component remanufacturing (e.g. sandblasting, vibro-finishing, polishing, grinding) (→ **fig. 11.6**, **page 11:5**)
- Component replacement if necessary (e.g. oversized rollers)
- Component washing, inspection and measuring after remanufacturing
- Assembly and final traceability marking
- Final inspection and final measurements (→ **fig. 11.1**)
- Cleaning and preservation
- Packaging and service reporting
- Documentation archiving

SKF recommends that bearings with unknown histories or that have been in operation for more than half of their calculated SKF rating lives undergo ultrasonic tests to check for subsurface micro cracks due to fatigue (→ **fig. 11.2**). Please note that ultrasonic testing is not standard and is undertaken on customer request.

Fig. 11:2

Checking for subsurface micro cracks with ultrasonic testing

Measurement of the wall thickness variation of the outer ring of a spherical roller bearing during final inspection

Fig. 11:1

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Remanufacturing timing

Good timing is essential to achieve the optimum balance between long service life and low remanufacturing costs (→ **diagram 11.2**)

The bearing shown in **fig. 11.3**, mounted in a thermo roll, had been dismounted as soon as vibration analysis indicated surface damage. It can be easily remanufactured by polishing.

In contrast, the corrosion on the bearing shown in **fig. 11.4**, mounted on a deflection compensated roll, was not detected until significant damage had occurred as the mill did not have condition monitoring. In this case, remanufacturing has involved grinding to remove the raceway damage and the replacement of the rollers with new oversized ones.

The shoe press bearing shown in **fig. 11.5** suffered catastrophic failure. Bearings with such heavy damage cannot be remanufactured.

Light surface standstill corrosion on a CARB toroidal roller bearing

Fig. 11:3

The earlier the bearing is dismounted after damage starts, the more likely the bearing can be remanufactured at low cost

Diagram 11.2

Early detection of bearing damage increases the possibility of remanufacturing and reduces cost

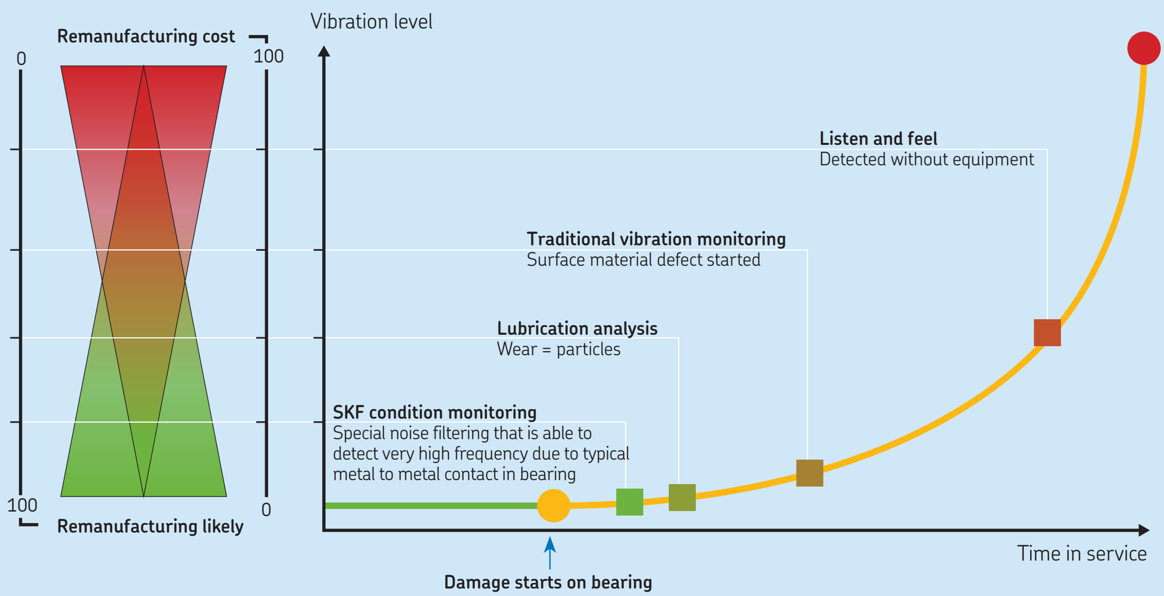

Fig. 11:4

Fig. 11:5

Corrosion on the inner ring of a spherical roller bearing due to process water ingress

Heavy spalling and a fracture on a spherical roller bearing

Preventing recurring damage

SKF's predictive maintenance expertise can help improve a remanufacturing programme by monitoring the condition of assets to aid decision making. In addition, SKF root cause failure analysis (RCFA) can identify the causes of problems and suggest corrective actions to prevent recurrence.

Grinding a bearing inner ring raceway to remove damage

Fig. 11:6

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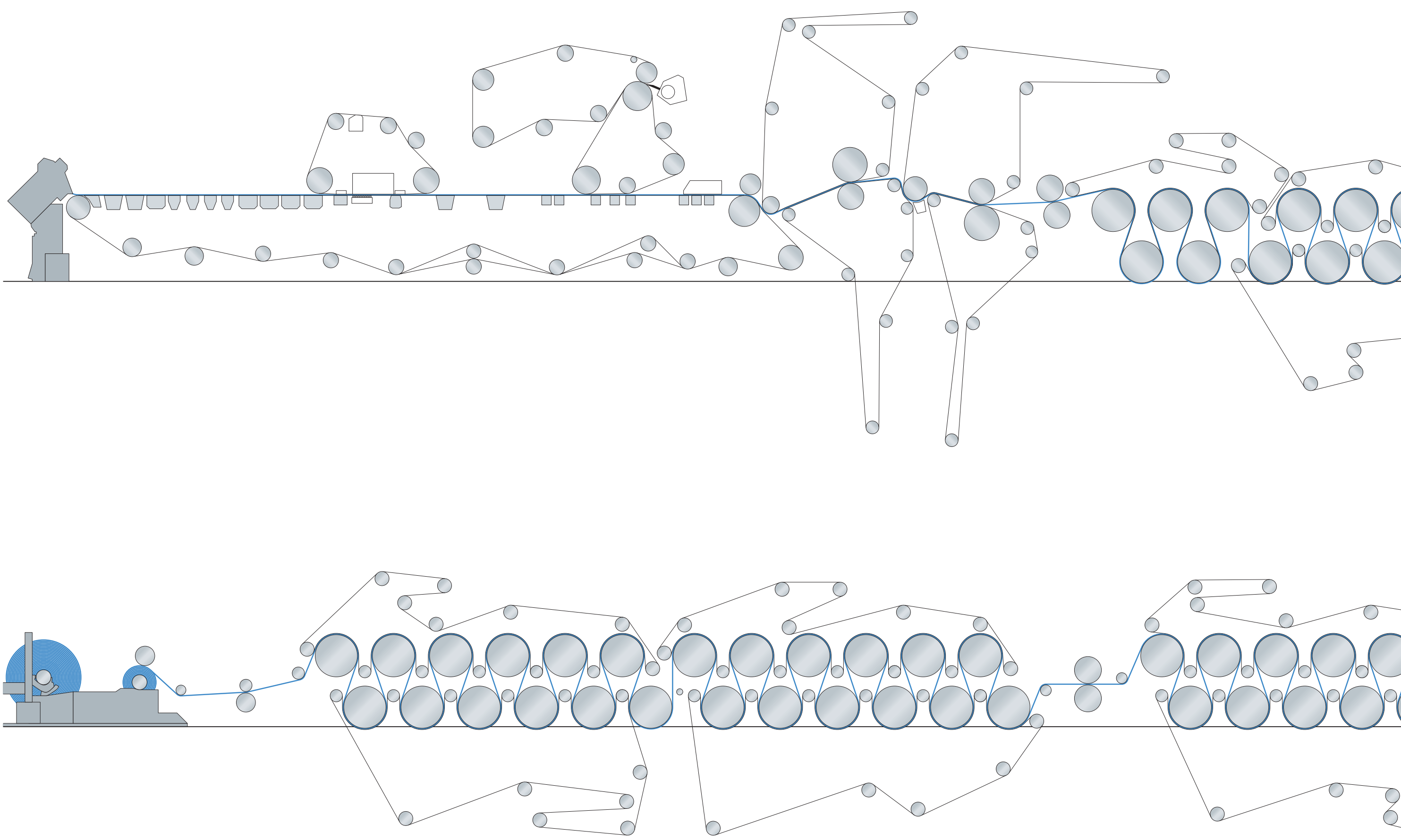

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