Mounting and Maintenance of Rolling Bearings

Products · Services · Training

Mounting and Maintenance of Rolling Bearings

Products
Services
Training

Foreword

Industrial Service

This catalogue is aimed principally at maintenance managers and operators of plant in which rolling bearings and other rotating machine components play a critical role in determining the quality of products and processes. Those responsible for maintenance and production processes must be able to rely every day on the quality of their tools and the expertise of their service providers.

Within its Industrial Service concept, Schaeffler therefore offers high quality products, services and training, *Figure 1*.

Portfolio

This catalogue gives an overview of the portfolio:

- Mechanical maintenance
- Lubrication
- Condition monitoring
- Reconditioning.

The employees of Schaeffler worldwide will be pleased to help you select the ideal products, services and training courses, *Figure 1*.

Figure 1
Portfolio

Foreword

Saving on maintenance costs

Schaeffler Industrial Service is responsible for replacement parts and service business for end customers and sales partners in all significant industrial sectors. On the basis of innovative solutions, products and services relating to rolling and plain bearings, Schaeffler offers a comprehensive portfolio that covers all phases in the lifecycle of the rolling bearing and takes account of the total costs (TCO).

The aim is to help customers save on maintenance costs, optimise plant availability and avoid unforeseen machine downtime.

Schaeffler Industrial Service offers an individual and concept solution to each customer irrespective of the manufacturer involved.

Schaeffler has centres of competence all around the world. This means we can provide customers worldwide with products, services and training quickly and professionally. All service employees worldwide undergo a comprehensive training programme and are audited regularly by officially certified specialists. This ensures that services throughout the world conform to a uniformly high standard of quality.

The quality requirements are strongly influenced by a long history of high precision rolling bearing manufacture. The manufacture of the products and the provision of all services in this catalogue is proven in practice and is secured by a quality management system certified in accordance with ISO 9001:2015.

Sales partners

In order to achieve this objective, we have created a network of Schaeffler sales partners. This network makes it possible to support all end customers worldwide with the same high level of expertise.

Mounting Toolbox – mounting made easy

The Schaeffler Mounting Toolbox, *Figure 2*, brings together valuable knowledge relating to the mounting and dismounting of rolling bearings. In individual video sequences, the service experts present step by step the points that must be paid close attention for correct mounting, lubrication and alignment. The interface is a “Virtual Plant” and offers the user easy, rapid navigation. With just a few clicks of the mouse, it is possible to gain an overview of the tools and accessories as well as to select individual video sequences. Internet access is all that is needed to enter the “Virtual Plant” and watch the Schaeffler fitting personnel at work.

Link to Mounting Toolbox:
http://mtb.schaeffler.de

Figure 2
Mounting Toolbox

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000A325A

0009F96E

0009F980

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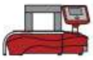

0008A942

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00019DD6

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- Publications
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Products: Mounting

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Mounting tool sets
Socket wrenches

FITTING-TOOL-ALU-10-50

000190A1

LOCKNUT-SOCKET

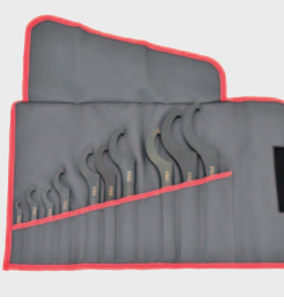
000A32EA

Hook and double hook wrenches
Hook wrenches

LOCKNUT-HOOK

000179C4

LOCKNUT-HOOK-KM0-16-SET



000190A3

Double hook wrenches

LOCKNUT-DOUBLEHOOK

0008AAB3

Double hook wrench set

LOCKNUT-DOUBLEHOOK-..-SET

00089FDE

Mechanical extractors

Two-arm extractors

Two-arm extractor set

PULLER-2ARM, PULLER-2ARM-SEPARATOR

PULLER-2ARM-SET

Three-arm extractors

PULLER-3ARM

Hydraulic extractors

PULLER-HYD

Three-section extraction plates

PULLER-TRISECTION

Mechanical mounting and dismounting

Features

These mechanical tools are designed for the mounting and dismounting of bearings. The mounting forces are transmitted by the form fit effect.

Mounting tool sets

The mounting tool sets are suitable for the simple mounting of rolling bearings with a bore of up to 50 mm, *Figure 1*. They can also be used for the mounting of sleeves, intermediate rings, seals and similar parts.

A mounting tool set contains mounting sleeves made from aluminium and mounting rings made from plastic.

Figure 1
Mounting tool set

An error frequently made during mounting is to transmit the mounting forces through the rolling elements and raceways.
This error can be avoided by driving the inner ring onto the shaft or driving the outer ring into the housing by applying hammer blows to an appropriate mounting sleeve. The precision parts are matched to each other, ensuring that the forces are uniformly transmitted to the end faces of the bearing rings.

Scope of delivery

Mounting tool set comprising 33 mounting rings for bearing bore 10 mm to 50 mm and outside diameter up to 110 mm
3 mounting sleeves
1 recoilless hammer, mass 1 kg
1 case

Ordering designation

FITTING-TOOL-ALU-10-50
Also available as individual parts.

Socket wrenches

Socket wrenches LOCKNUT-SOCKET are suitable for the simple tightening and loosening of locknuts on shafts, adapter sleeves and withdrawal sleeves. They require less space on the circumference of the nut than hook wrenches and allow the use of ratchets and torque wrenches, *Figure 2*.

Figure 2
Socket wrench and torque wrench

For increased reliability, socket wrenches should be secured using a locking pin and rubber ring. They therefore have a hole for the locking pin and a groove for the rubber ring. The locking pin and rubber ring are included in the scope of delivery.

Socket wrenches are available in sizes suitable for locknuts KM0 to KM20. Other sizes and special solutions are available by agreement.

Scope of delivery

1 socket wrench
1 locking pin
1 rubber ring

Ordering example: Socket wrench, suitable for locknut KM5

Ordering designation: **LOCKNUT-SOCKET-KM5**

Special sizes available by agreement.

Hook and double hook wrenches

These wrenches are used to move locknuts or extraction nuts for the mounting or dismounting of rolling bearings or withdrawal sleeves.

Hook wrenches

A hook wrench LOCKNUT-HOOK can be used to dismount not only bearings but also withdrawal sleeves with the aid of extraction nuts.

Hook wrenches are available in sizes suitable for locknuts KM0 to KM40, suitable for diameters from 16 mm to 245 mm.

Ordering example: Hook wrench, suitable for locknuts KM18, KM19 and KM20

Ordering designation: **LOCKNUT-HOOK-KM18-20**

These wrenches can be used for the mounting and dismounting of small bearings on shaft seats, adapter sleeves or withdrawal sleeves. In addition to the sizes stated here, other sizes are available by agreement.

Ordering example: Set comprising ten hook wrenches

Ordering designation: **LOCKNUT-HOOK-KM0-16-SET**

Hook wrenches can also be ordered as a set. The set comprises ten hook wrenches of sizes KM0 to KM16 in a roll-up pouch and is suitable for diameters from 16 mm to 100 mm.

Double hook wrenches

Double hook wrenches LOCKNUT-DOUBLEHOOK are intended for the mounting of spherical roller bearings and self-aligning ball bearings with a tapered bore, *Figure 3*. The individual wrenches are available as a set.

Figure 3
Double hook wrenches

The double hook wrench sets contain a torque wrench. This allows a precisely defined tightening torque to be achieved at the start of the mounting operation.

Double hook wrench sets are suitable for several sizes of locknuts. There is one set each for locknuts KM3 to KM8 and for locknuts KM9 to KM15. All the parts in the scope of delivery are also available individually.

Each double hook wrench is engraved with the torsion angles for the appropriate spherical roller bearings and self-aligning ball bearings. The drive-up distance and reduction in radial internal clearance can thus be precisely set.

Scope of delivery	Several double hook wrenches 1 torque wrench 1 mounting lever 1 user manual 1 case 1 mounting paste (20 g)
Ordering example	4 double hook wrenches, suitable for locknuts KM3 to KM8
Ordering designation	**LOCKNUT-DOUBLEHOOK-KM3-8-SET**
Ordering example	5 double hook wrenches, suitable for locknuts KM9 to KM15
Ordering designation	**LOCKNUT-DOUBLEHOOK-KM9-15-SET**

Mechanical extractors

Mechanical extractors can be used to dismount small and medium sized rolling bearings that are located with a tight fit on a shaft or in a housing. The bearing can be dismounted without damage if the extractor is in contact with the tightly fitted bearing ring.

In the case of mechanical extractors, the extraction force is normally applied by means of threaded spindles.

In addition to the two-arm and three-arm devices as well as a hydraulic pressure tool, special solutions are also possible.

For the dismounting of larger bearings, hydraulic extractors should be used, see page 26.

Two-arm and three-arm extractors

Two-arm and three-arm extractors, *Figure 4*, *Figure 5* and tables, page 25, are used for the extraction of complete rolling bearings or tightly fitted inner rings.

The two-arm extractor PULLER-2ARM and three-arm extractor PULLER-3ARM can also be used to extract other parts such as gears.

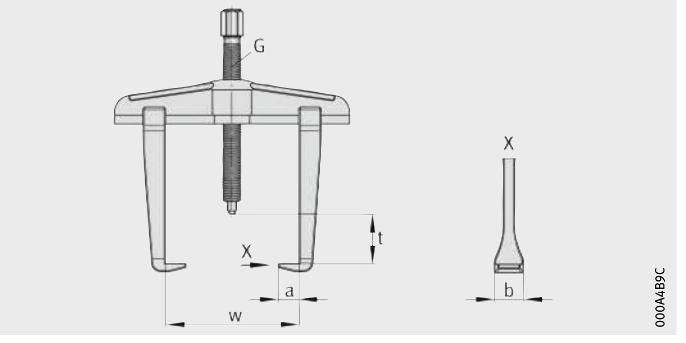


Figure 4
Two-arm extractor,
dimensions of gripper

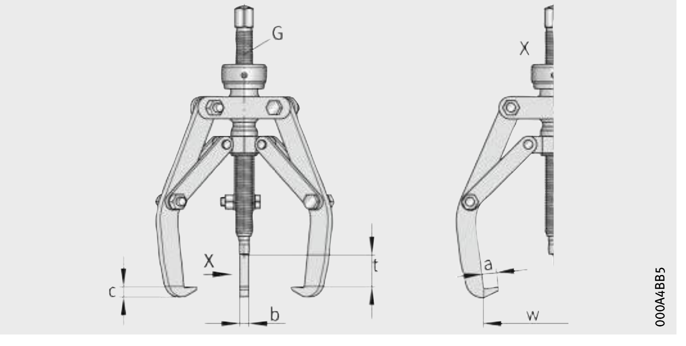


Figure 5
Three-arm extractor,
dimensions of gripper

Available two-arm extractors

Designation	Grip span w mm	Grip depth t mm	Dimensions a mm	 b mm	Extraction force kN
PULLER-2ARM90	90	100	15	22	30
PULLER-2ARM130	130	100	15	22	30
PULLER-2ARM160	160	150	24	30	50
PULLER-2ARM200	200	150	24	30	50
PULLER-2ARM250	250	200	32	36	75
PULLER-2ARM350	350	200	32	36	75
PULLER-2ARM-SEPARATOR45	45	65	2,5	12^{+1}	10
PULLER-2ARM-SEPARATOR90	90	100	2,5	14^{+1}	40
PULLER-2ARM-SEPARATOR150	150	150	2,5	28^{+1}	40

Available two-arm extractor set

Designation: **PULLER-2ARM-SET**
Two-arm extractors included
PULLER-2ARM130, PULLER-2ARM200, PULLER-2ARM350
Accessories included
Narrow extraction hook for size 130 and 200, tube of spindle grease, carry case

Available three-arm extractors

Designation	Grip span w mm	Grip depth t mm	Dimensions a mm	 b mm	Extraction force kN
PULLER-3ARM160	160	100	14^{+1}	15^{+1}	45
PULLER-3ARM230	230	165	19^{+1}	22^{+1}	100
PULLER-3ARM310	310	235	19^{+1}	22^{+1}	100
PULLER-3ARM430	430	240	20^{+2}	30^{+2}	150
PULLER-3ARM660	660	340	22^{+2}	34^{+2}	150

Further information

- TPI 216, Tools for the Mechanical Mounting and Dismounting of Rolling Bearings

Mechanical mounting and dismounting

Hydraulic extractors

Hydraulic extractors, see tables, are used where higher extraction forces are required.

These devices allow rolling bearings, gears, sleeves and many other shrink fitted parts to be quickly and easily dismounted.

For larger grip depths, the XL design or longer extraction arms are available as accessories.

Operating personnel can be protected by means of a safety grid or a safety cover.

The advantageous features of hydraulic extractors are as follows:

- parts under mechanical load made from high quality chromium-molybdenum steel
- smooth-running, chromium plated piston made from hardened and tempered steel
- stroke travel adjustable by means of standard adapter
- screw thread for setting of optimum grip depth
- simple centring by spring-loaded steel cone
- simple conversion to two-arm operation in case of insufficient space for three arms
- optimum operating position due to rotatable pump hand lever or separate pump.

Available hydraulic extractors with integral hand pump

Designation	Extrac-tion force	Grip span		Grip depth		Stroke length
		Standard	XL	Standard	XL	
	kN	mm	mm	mm	mm	mm
PULLER-HYD40	40	200	–	165	–	55
PULLER-HYD60(-XL)	60	200	260	165	210	82
PULLER-HYD80(-XL)	80	260	300	210	240	82
PULLER-HYD100(-XL)	100	250	280	185	210	82
PULLER-HYD120(-XL)	120	300	330	240	280	82
PULLER-HYD200(-XL)	200	360	380	275	330	82
PULLER-HYD250(-XL)	250	410	440	315	380	110
PULLER-HYD300(-XL)	300	540	540	405	610	110

Available hydraulic extractors with separate hand pump

Designation	Extrac-tion force	Grip span		Grip depth		Stroke length
		Standard	XL	Standard	XL	
	kN	mm	mm	mm	mm	mm
PULLER-HYD400(-XL)	400	580	1 000	420	635	125

Further information

- TPI 216, Tools for the Mechanical Mounting and Dismounting of Rolling Bearings

Three-section extraction plates

Three-section extraction plates PULLER-TRISECTION, see table, can be used with hydraulic and mechanical extractors.

These allow the extraction of complete bearings, tightly fitted inner rings and other components.

The load carrying capacity is matched to the maximum extraction force of the hydraulic extractors used in each case. In order to prevent damage to the bearing during extraction, the geometrical form of the three extraction segments means that they grip the bearing on the inner ring only.

The extraction plates can be fitted under the bearing with just a few movements.

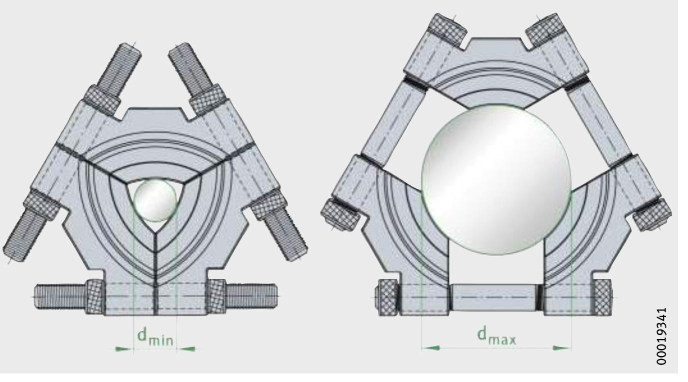


Figure 6
Maximum and minimum diameter of extraction plates, see table

Available three-section extraction plates

Designation	Dimensions		Recommended for extractor	
	d_{min} 1) mm	d_{max} 1) mm	PULLER-HYD	PULLER-3ARM
PULLER-TRISECTION-50	12	50	–	160
PULLER-TRISECTION-100	26	100	40, 60, 80, 100	230
PULLER-TRISECTION-160	50	160	80, 100, 120, 175, 200	310
PULLER-TRISECTION-260	90	260	175, 200, 250, 300	430
PULLER-TRISECTION-380	140	380	250, 300, 400	660

1) d_{min} and d_{max}, *Figure 6*.

Further information

- TPI 216, Tools for the Mechanical Mounting and Dismounting of Rolling Bearings

Product overview Hydraulic mounting and dismounting

Hydraulic nuts

HYDNUT

Hand pumps

Single stage

PUMP1000-0,7L

Twin stage

PUMP1000-4L, PUMP1000-8L, PUMP1600-4L, PUMP1600-8L, PUMP2500-4L, PUMP2500-8L

Mobile hydraulic unit

Tool set

TOOL-RAILWAY-AGGREGATE

TOOL-RAILWAY-AXLE

Hydraulic press

Sealing cap tool

TOOL-RAILWAY-SEALCAP-PRESS

TOOL-RAILWAY-SEALCAP

Connectors, accessories

Adapters and reduction nipples

PUMP.NIPPLE,
PUMP.ADAPTER

Rapid push fit coupling

PUMP1600.VALVE-NIPPLE,
PUMP1600.VALVE-SOCKET

Digital manometer

PUMP1000.MANO-DIGI

Manometer

PUMP1000.MANO-G1/2,
PUMP1600.MANO-G1/2

Sleeve connector

PUMP.SLEEVE-CONNECTOR

Hydraulic mounting and dismounting

Features

Hydraulic tools can be used to apply large forces. These tools are therefore particularly suitable for the mounting and dismounting of large bearings or parts with a tapered bore. Hydraulic nuts are used as a mounting tool. Pressure can be generated using hand pumps.

Software Mounting Manager

The program Mounting Manager gives assistance in selecting the correct mounting of bearings and offers the following options:

- It shows various mechanical and hydraulic mounting methods.
- It calculates the data required for mounting in relation to reduction in radial internal clearance, drive-up distance and start pressure.
- It gives advice on mounting.
- It generates a list of the accessories and tools required.

It also contains a library with references to publications giving further information and an electronic learning system.

Mounting method

Bearings with a tapered bore are mounted either directly on the tapered shaft or journal or by means of an adapter sleeve or withdrawal sleeve on the cylindrical shaft. The internal clearance is set either by measurement of the axial drive-up distance or by conventional means using feeler gauges.

Measurement of the axial drive-up distance

For measurement of the drive-up distance, a dial gauge is screwed into the hydraulic nut. The dial gauge is preloaded and the measurement sensor then precisely follows the displacement of the press ring. This value corresponds to the displacement of the rolling bearing on the tapered seat.

Measurement of the reduction in radial internal clearance

When the bearing is driven onto the tapered seat, the inner ring is expanded and the radial internal clearance is thereby reduced. This reduction in radial internal clearance is an indication of the tight fit of the bearing. Measurement is carried out using a feeler gauge.

Hydraulic nuts

Hydraulic nuts HYDNUT, *Figure 1* and table, are used to press components with a tapered bore onto their tapered seat. Presses are mainly used if the drive-up forces required cannot be applied using other accessories, e.g. shaft nuts or pressure screws.

Figure 1
Hydraulic nut with dial gauge

The main applications are as follows:

- Rolling bearings with a tapered bore can be mounted and dismounted.

  Bearings can be seated directly on a tapered shaft, an adapter sleeve or a withdrawal sleeve. The hydraulic nut can also be used for the dismounting of adapter or withdrawal sleeves.
- Components such as couplings, gears and ships' propellers can be mounted and dismounted.

Available hydraulic nuts

Designation	Design	Application
HYDNUT50..-E to **HYDNUT200..-E**	With metric fine pitch thread to DIN 13	Standardised adapter and withdrawal sleeves
HYDNUT205..-E to **HYDNUT1180..-E**	With trapezoidal thread to DIN 103	With metric dimensions
HYDNUT90-E-INCH to **HYDNUT530-E-INCH**	With inch size thread to ABMA "Standards for Mounting Accessories, Section 8, Locknut Series N-00"	Sleeves with inch dimensions
HYDNUT100-HEAVY to **HYDNUT900-HEAVY**	Increased capacity design with smooth bore	For high mounting forces, for example in shipbuilding

Hydraulic mounting and dismounting

Scope of delivery

The scope of delivery comprises the hydraulic nut, accessories and user manual, *Figure 2*.

① Hydraulic nut
② Knurled screws
③ Screw plug
④ Grub screws
⑤ Hand lever
⑥ Connection nipple
⑦ Replacement seals and red ring
⑧ User manual

Figure 2
Scope of delivery
Hydraulic nut

Scope of delivery

1	hydraulic nut
2	replacement seals
1	red ring
1	screw plug
3/5	grub screws
1	connection nipple
3	knurled screws
1	hand lever
1	user manual

Ordering designation **HYDNUT**

Further information

- TPI 196, FAG Hydraulic Nuts

Pressure generation devices

Pressure generation devices are available in various designs: from the hand pump via the mobile hydraulic unit to the hydraulic press, see tables.

Application Hand pump

Designation	Application
PUMP1000-0,7L	Mounting and dismounting of rolling bearings For driving hydraulic nuts up to HYDNUT395 or HYDNUT300-HEAVY
PUMP1000-4L	Mounting and dismounting of rolling bearings Mounting and dismounting of components such as ships' propellers For driving hydraulic nuts up to HYDNUT800
PUMP1600-4L	Mounting and dismounting of rolling bearings Mounting and dismounting of components such as rudder splines and rudder blades
PUMP2500-4L	Mounting and dismounting of bearings Mounting and dismounting of components such as gears and couplings

Application Mobile hydraulic unit

Designation	Application
TOOL-RAILWAY-AGGREGATE	Mounting and dismounting of tapered roller bearing units (TAROL)

Application Hydraulic press

Designation	Application
TOOL-RAILWAY-SEALCAP-PRESS	Mounting and dismounting of seals on tapered roller bearing units (TAROL)

Further information

- TPI 195, FAG Pressure Generation Devices.

Hydraulic mounting and dismounting

Hand pumps

Hand pumps have a single stage or twin stage pump with a manometer.

Single stage pump

The hand pump PUMP1000-0,7L has an oil container with a volume of 0,7 l. The maximum oil pressure is 1 000 bar, see table.

A digital manometer is available as an accessory.

Available single stage pump

Designation	Maximum oil pressure bar
PUMP1000-0,7L	1 000

Twin stage pump

The twin stage pumps, *Figure 3* and table, have a high delivery rate up to 50 bar and then switch automatically to the high pressure stage. This gives a high work rate.

Figure 3
Twin stage pump,
4-l oil container

Where there is an increased oil requirement, the twin stage pumps are available with an 8-l oil container (suffix 8L). If the type of mounting of the adapter or withdrawal sleeve requires a separate oil supply, a two-way valve is available (suffix D).

For pumps with an oil pressure of 1 000 bar and a connector, digital manometers are also available as accessories.

Available twin stage pumps

Designation	Maximum oil pressure bar
PUMP1000-4L	1 000
PUMP1600-4L	1 600
PUMP2500-4L	2 500

Mobile hydraulic unit

The mobile hydraulic unit, *Figure 4*, is used for the mounting and dismounting of tapered roller bearing units, also known as TAROL units. These units are used as wheelset bearings in rail vehicles such as goods wagons and passenger carriages.

The mobile unit has a valve-controlled, double direction pressure cylinder driven by a motor pump. The pressure cylinder is adjustable in height.

When making enquiries or placing orders, information on the power connection is required.

Figure 4
Mobile hydraulic unit

Ordering designation **TOOL-RAILWAY-AGGREGATE**

Tool set

Tool sets are produced for a specific application, *Figure 5*. When making enquiries or placing orders, information on the bearing type and installation drawings (shaft, housing, additional parts) are required.

Figure 5
Tool set

Ordering designation **TOOL-RAILWAY-AXLE**

Hydraulic press

The hydraulic press, *Figure 6*, is used for the mounting and dismounting of seals on tapered roller bearing units, also known as TAROL units. In addition, a matching tool set is required for each bearing type.

Figure 6
Hydraulic press

Ordering designation **TOOL-RAILWAY-SEALCAP-PRESS**

Sealing cap tool

Sealing cap tools are bearing-specific and include all the parts for mounting and dismounting of the seal. For dismounting, the parts required are an adapter ring, a punch and the appropriate press-out segments. For mounting of the new seal, a support and the appropriate press-in ring are supplied.

Ordering designation **TOOL-RAILWAY-SEALCAP**

Further information

- TPI 195, FAG Pressure Generation Devices
- TPI 156, Tapered Roller Bearing Units TAROL – Mounting, Maintenance, Repair

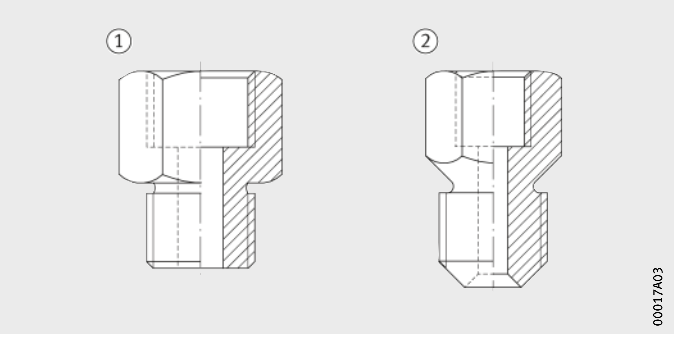
Connectors, accessories

Various connectors and accessories are available for use with the devices for hydraulic mounting and dismounting.

Adapters and reduction nipples

Adapters and reduction nipples are matched to the threads of high pressure hoses and pipes, *Figure 7* and tables.

Adapters and reduction nipples of type A (with sealing ring) are suitable for oil pressures up to 800 bar, *Figure 7*. Type B (with blade sealing) is suitable for oil pressures up to 2 500 bar, *Figure 7*.



① Type A
② Type B

Figure 7
Adapters and reduction nipples

Available adapters and reduction nipples

Designation	Designation
PUMP.NIPPLE-A-G1/4-G1/8	**PUMP.NIPPLE-A-G3/4-G1/8**
PUMP.NIPPLE-B-G1/4-G1/8	**PUMP.NIPPLE-B-G3/4-G1/8**
PUMP.NIPPLE-A-G1/4-G1/2	**PUMP.NIPPLE-A-G3/4-G1/4**
PUMP.NIPPLE-B-G1/4-G1/2	**PUMP.NIPPLE-B-G3/4-G1/4**
PUMP.NIPPLE-A-G1/4-G3/4	**PUMP.NIPPLE-A-G3/4-G3/8**
PUMP.NIPPLE-B-G1/4-G3/4	**PUMP.NIPPLE-B-G3/4-G3/8**
PUMP.NIPPLE-A-G1/4-M14	**PUMP.NIPPLE-A-M18×1,5-G1/4**
PUMP.NIPPLE-B-G1/4-M14	**PUMP.NIPPLE-A-M18×1,5-G3/8**
PUMP.NIPPLE-A-G1/4-M18×1,5	**PUMP.NIPPLE-A-M18×1,5-G3/8**
PUMP.NIPPLE-A-G3/8-G1/4	–
PUMP.NIPPLE-B-G3/8-G1/4	–

Available adapters

Designation	Designation
PUMP.ADAPTER-A-G1/4	**PUMP.ADAPTER-A-G3/4**
PUMP.ADAPTER-B-G1/4	**PUMP.ADAPTER-B-G3/4**

Rapid push fit coupling

A suitable connecting nipple is always included in the delivery of a hydraulic nut. Each hand pump with an oil pressure up to 1 600 bar is supplied with a rapid push fit coupling. The rapid push fit coupling allows rapid connection and disconnection of a hose and is suitable for oil pressures up to 1 600 bar, *Figure 8* and table.

After the coupling has been fitted, the high pressure hose must be secured to the connection point by means of a chain or cord.

1 Nipple
2 Socket

Figure 8
Rapid push fit coupling

Available nipple and socket

Designation	Threaded connector inch	Component
PUMP1600.VALVE-NIPPLE	G$^{1}/_{4}$	Nipple
PUMP1600.VALVE-SOCKET	G$^{1}/_{4}$	Socket

Manometer

In addition to the manometer with digital display, there are three analogue manometers with an indicator, see table.

When selecting a manometer, pay attention to the maximum oil pressure.

Available manometers

Designation	Threaded connector inch	Maximum oil pressure bar
PUMP1000.MANO-DIGI	G$^1/_4$	1 000
PUMP1000.MANO-G1/2	G$^1/_2$	1 000
PUMP1600.MANO-G1/2	G$^1/_2$	1 600

Sleeve connectors

Sleeve connectors can be used at pressures up to 800 bar.

The connector to the pump holder is G$^1/_4$. The connector to the consumer device is available in the sizes M6, M8, G$^1/_8$ and G$^1/_4$. For other thread sizes, a reduction nipple can be used.

Check the oil pressure using a manometer.

Ordering example: Sleeve connector with a connector G$^1/_8$ on the consumer device side

Ordering designation: **PUMP.SLEEVE-CONNECTOR-G1/8**

Product overview Thermal mounting, induction heating devices

Tabletop devices

HEATER50

HEATER100

HEATER200

Standalone devices

HEATER400

HEATER800

HEATER1600

Thermal mounting, induction heating devices

Features

Induction heating devices HEATER with mains frequency technology are used to heat rolling bearings and other components with a cylindrical bore where a tight fit on the shaft or in the housing is intended.

Adequate expansion of the bearings is achieved in most cases at +80 °C to +100 °C. During the heating operation, the maximum heating temperature must be observed. The temperature of rolling bearings must not normally exceed +120 °C, in order to prevent changes to the structure and hardness of the bearing. In all devices for heating, the temperature can be steplessly controlled.

!

Wear protective gloves during mounting and dismounting of heated parts.

Induction heating devices HEATER

The induction heating devices HEATER for rolling bearings up to a mass of 1 600 kg have been improved further in terms of their performance capability and safety compared with their predecessors. They can also be used to heat sealed and greased rolling bearings. In addition to the tabletop devices HEATER50 to HEATER200, the range also includes the standalone devices HEATER400 to HEATER1600 for larger rolling bearings.

The scope of delivery of the induction heating devices HEATER covers a basic setup, *Figure 1*.

1 Heating device
2 Slewing ledge
3 Temperature sensor
4 Lifting tool
5 User manual

Figure 1
Scope of delivery
Heating device HEATER200

The rolling bearing to be heated is either suspended from the ledge or is laid horizontally on the sliding table, *Figure 2*.

① Slewing ledge
② U-shaped core
③ Rolling bearing
④ Sliding table

Figure 2
Heating of rolling bearing

Advantages of FAG heating devices

The advantages of the induction heating devices are:

- very safe operation
- high reliability (TÜV certified)
- effective, energy-efficient heating (high efficiency level)
- uniform, controlled heating
- automatic demagnetisation
- simple operation
- high cost-effectiveness through selection of the device size most suitable for the particular application.

Operating modes

The induction heating devices can be operated in the following modes:

- temperature control
- time control
- ramp control
- delta-T control.

Thermal mounting, induction heating devices

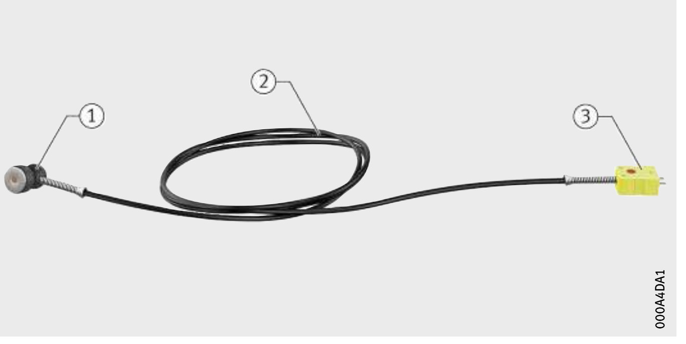
Accessories

The functional scope of an induction heating device can be extended by the use of accessories.

Temperature sensor

Two temperature sensors can be connected to each induction heating device. The sensor head of the temperature sensor is magnetic and is positioned on the component. The signal is fed via the cable and plug to the device, *Figure 3*.

The induction heating devices HEATER50 and HEATER100 are supplied with one temperature sensor. If two temperature sensors are used, it is possible to operate the heating method with delta-T control.



① Sensor head
② Cable
③ Plug

Figure 3
Temperature sensor

Ledge

Each induction heating device is supplied with one ledge.
This ledge has the same cross-section as the U-shaped core and allows maximum power to be achieved.

In order to heat rolling bearings of a smaller inside diameter, ledges with smaller cross-sections are available.

Adapter posts

For the tabletop devices HEATER50, HEATER100 and HEATER200, adapter posts are available. These are always placed in pairs on the U-shaped core and thus increase the inner height. With the aid of adapter posts, it is also possible to heat workpieces with a small inside diameter and a large outside diameter.

Further information

- TPI 200, FAG Heating Devices for the Mounting of Rolling Bearings

FAG Heating Manager

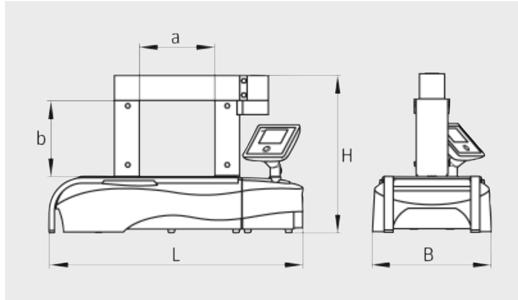
The software FAG Heating Manager is a user-friendly tool for selection of the optimum heating device for the heating of rolling bearings.

Once the rolling bearing to be heated has been selected, the bearing type, dimensions, mass and the suitable heating device are displayed, *Figure 4*.

Figure 4
FAG Heating Manager

Heating devices HEATER

Product range



Dimension table

Characteristics		Unit	**HEATER50** 
Operating voltage	U	VAC	110 to 230
Frequency	F	Hz	50 to 60
Power consumption	P	kVA	3
Current rating	I	A	13
Residual magnetism	H	A/cm	< 2
Operating duration	ED	%	100
Mass	m	kg	18
Length	L	mm	450
Width	B	mm	210
Height	H	mm	250
Dimension	a	mm	120
Dimension	b	mm	140
Maximum rolling bearing mass	m	kg	50
Maximum mass of other component	m	kg	40
Maximum width	b	mm	120
Minimum inside diameter [1]	d	mm	55
Minimum inside diameter with accessories	d	mm	10
Maximum inside diameter (lying flat)	d	mm	300
Maximum outside diameter	D	mm	400 (with **LEDGE-55**)

[1] When using the ledge included in the scope of delivery.

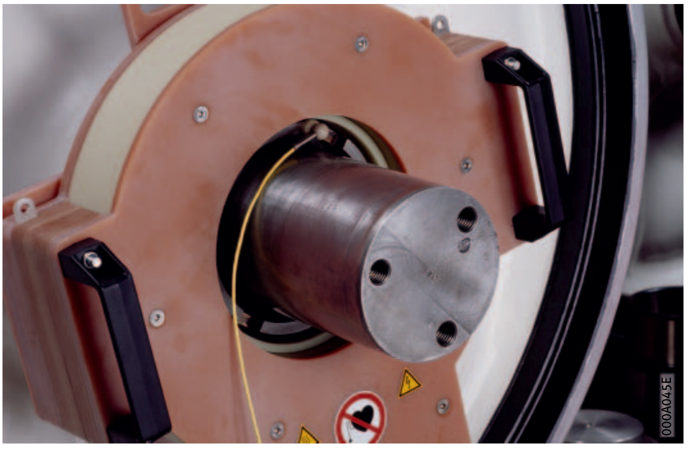
HEATER100	HEATER200	HEATER400	HEATER800	HEATER1600
				
110 to 230	400 to 480	400 to 480	400 to 480	400 to 480
50 to 60	50 to 60	50 to 60	50 to 60	50 to 60
3,7	8	12,8	25,2	40
16	20	32	63	100
< 2	< 2	< 2	< 2	< 2
100	100	100	100	100
35	86	157	280	650
540	695	850	1 080	1 500
275	330	420	500	800
310	370	950	1 250	1 600
180	210	300	430	690
180	210	330	490	700
100	200	400	800	1 600
80	150	300	600	1 200
180	210	330	400	650
70	100	120	150	220
15	20	35	50	90
400	500	900	1 400	1 900
500 (with **LEDGE-70**)	600 (with **LEDGE-100**)	1 000 (with **LEDGE-120**)	1 500 (with **LEDGE-150**)	2 000 (with **LEDGE-220**)

Thermal mounting and dismounting, medium frequency technology

Medium frequency technology

HEAT-INDUCTOR, HEAT-GENERATOR

HEAT-INDUCTOR



Thermal mounting and dismounting, medium frequency technology

Features

Induction units based on medium frequency technology are, in contrast to induction heating devices, suitable not only for thermal mounting but also for dismounting. Furthermore, they can be used for the heating of very large and heavy components.

Adequate expansion of the bearings is achieved in most cases at +80 °C to +100 °C. During the heating operation, the maximum heating temperature must be observed. The temperature of rolling bearings must not normally exceed +120 °C, in order to prevent changes to the structure and hardness of the bearing. In all devices for heating, the temperature can be steplessly controlled.

! Wear protective gloves during mounting and dismounting of heated parts.

Induction units with medium frequency technology

Due to their high flexibility and compact construction, these units can also be used for mobile operation. They can therefore be used, for example, at construction sites for wind turbines or for other large components that are difficult to transport.

Application

Examples of the use of medium frequency technology include:

- heating of medium-sized to large bearings for mounting and dismounting
- heating of housings prior to mounting of a bearing
- batch dismounting of bearing inner rings of cylindrical roller bearings and labyrinth rings, for example in the case of wheelset bearings in rail vehicles
- dismounting of bearing inner rings from traction motors in rail vehicles
- heating of large components, for example bearings or machine supports in wind turbines
- heating of roll rings and couplings, for example in steelworks
- loosening of shrink fit connections of gears.

Thermal mounting and dismounting, medium frequency technology

The units comprise a generator and an inductor that is positioned on the workpiece. Depending on the requirements, a rigid or flexible inductor is used. Depending on the application, flexible inductors are positioned in the bore or on the outside diameter of the workpiece, *Figure 1*. Flexible inductors are suitable for the heating of bearing inner rings or of large components such as machine supports in wind turbines. The length of the inductor is defined as a function of the dimensions of the workpiece.

Figure 1
The flexible inductor can be wrapped around the component

Flexible inductors

The flexible inductors are available in two designs that differ mainly in their geometrical characteristics but also in their maximum operating duration, see table.

Technical data on flexible inductors

Designation		Inductor HEAT-INDUCTOR	
		-..M-D15	-..M
Cooling system	–	Air cooling	
Length	m	12 – 16	12 – 40
Diameter	mm	approx. 18	approx. 20
Minimum bending radius	mm	80	150
Mass without plug	kg/m	approx. 0,6	approx. 1
Permissible temperature of work-piece surface	°C	+180	
Maximum temperature at push fit connector	°C	+90	
Maximum operating duration	–	≦ 10 min	∞
Connection of inductor and generator	–	Push fit connector	

The flexible inductors are available in various lengths, see table.

Ordering designations and lengths

Ordering designation	Length m
HEAT-INDUCTOR-12M-D15	12
HEAT-INDUCTOR-14M-D15	14
HEAT-INDUCTOR-16M-D15	16
HEAT-INDUCTOR-12M	12
HEAT-INDUCTOR-16M	16
HEAT-INDUCTOR-20M	20
HEAT-INDUCTOR-24M	24
HEAT-INDUCTOR-27M	27
HEAT-INDUCTOR-30M	30
HEAT-INDUCTOR-40M	40

Rigid inductors

Rigid inductors are particularly suitable for batch production, *Figure 2*. In such cases, the emphasis is less on flexibility and more on short set-up times and high process reliability.

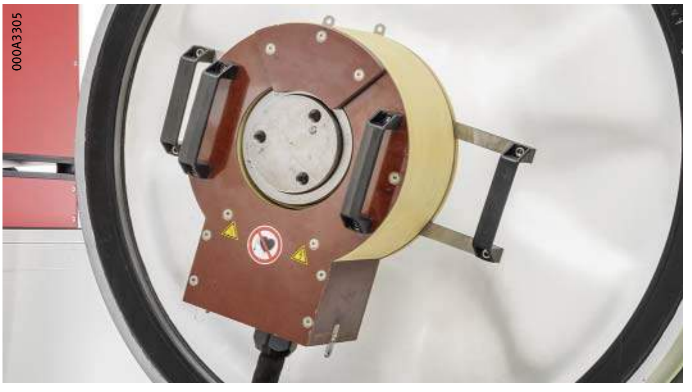


Figure 2
Rigid inductor for dismounting of wheelset bearings

Thermal mounting and dismounting, medium frequency technology

Generators

Compared with preceding models, the generators are of a significantly more compact and lighter design and are thus even more suitable for mobile operation. They are available in two performance variants and two voltage versions, *Figure 3* and tables, page 55.

Figure 3
Generators

Technical data of generators with voltage rating of 400 V

Designation		Generator HEAT-GENERATOR	
		20-2	40-2
Cooling	–	Open circuit ventilation	
Mains voltage	V	3×380 – 3×440	
Mains frequency	Hz	50 – 60	
Voltage tolerance	–	±10%	
Connector plug CEE	A	32	63
Line-side fuse protection	A	32	63
Effective power	kW	20[1)]	40[1)]
Output frequency	kHz	10 – 25	
Length of mains connection cable	m	5	
Width	mm	277	365
Depth (with mains connection cable)	mm	610	
Height (with grips)	mm	540	695
Mass	kg	30	55

1) Valid for voltage rating of 400 V.

Technical data of generators with voltage rating of 480 V

Designation		Generator HEAT-GENERATOR	
		20-2-480V	40-2-480V
Cooling	–	Open circuit ventilation	
Mains voltage	V	3×460 – 3×500	
Mains frequency	Hz	50 – 60	
Voltage tolerance	–	±10%	
Connector plug CEE	A	32	63
Line-side fuse protection	A	32	63
Effective power	kW	20[1]	40[1]
Output frequency	kHz	10 – 25	
Length of mains connection cable	m	5	
Width	mm	277	365
Depth (with mains connection cable)	mm	610	
Height (with grips)	mm	540	695
Mass	kg	30	55

[1] Valid for voltage rating of 480 V.

Digital control

Digital control is carried out by means of a 7″ TFT display and has the following characteristics:

- presentation of temperature patterns on the display
- storage and export of temperature patterns by means of an integrated temperature recorder
- separate registration for operator and service operator, with different access rights
- alarm functions for protection of the workpiece against overheating
  - temperature increase alert
  - temperature alarm on overshoot
- user languages: German and English
- remote access possible via an Ethernet interface.

Thermal mounting and dismounting, medium frequency technology

Advantages

The advantages of the heating device with medium frequency technology are as follows:

- suitable for mounting
- suitable for dismounting
- operating frequency from 10 kHz to 25 kHz
- efficiency of the generator higher than 90%
- low energy requirements
- short heating times
- time and temperature control as well as other operating modes
- automatic demagnetisation
- flexible and rigid inductors available
- inductors suitable for use either inside or outside the component
- lower mains connection power than heating devices with mains frequency
- almost silent
- air-cooled system.

Configuration

Each of the light and compact devices is designed for the specific application. It can be equipped, depending on the workpiece, with flexible or rigid inductors.

For enquiries, the following data are required:

- bearing dimensions, if possible with drawings
- representation of the adjacent construction
- data on the fit conditions
- description of the mounting process and its frequency
- power supply data
- ambient conditions
- your address.

Further information

- TPI 217, Induction Units with Medium Frequency Technology

Measurement and inspection

Feeler gauges

FEELER-GAUGE-100, FEELER-GAUGE-300

Taper gauges

MGK133

MGK132

Snap gauges

SNAP-GAUGE

Enveloping circle gauges

MGI21

MGA31

Visual inspection device

TOOL-RAILWAY-INSPECTION-DEVICE

Axial clearance gauge

Adapter set

TOOL-RAILWAY-CLEARANCE-BASIC

TOOL-RAILWAY-CLEARANCE.TOP

Measurement and inspection

Features

Feeler gauges and measurement gauges can be used to check the production of bearing seats and the mounting of bearings.

Feeler gauges

Feeler gauges FEELER GAUGE, see table, are used to measure the radial internal clearance, especially in mounting on tapered shaft seats and on adapter and withdrawal sleeves.

Available feeler gauges

Designation	Feeler length mm	Feeler thickness mm		
FEELER-GAUGE-100	100	0,03	0,08	0,14
		0,04	0,09	0,16
		0,05	0,1	0,18
		0,06	0,12	0,2
		0,07	–	–
FEELER-GAUGE-300	300	0,03	0,12	0,2
		0,04	0,13	0,25
		0,05	0,14	0,3
		0,06	0,15	0,35
		0,07	0,16	0,4
		0,08	0,17	0,45
		0,09	0,18	0,5
		0,1	0,19	–

Taper gauges

These gauges are used to inspect tapered bearing seats in production facilities. This is necessary to ensure a good match between the fit surfaces of the bearing and bearing seat. Gauges are available for different taper angles.

Taper gauge for taper 1:12 and 1:30

The taper gauge MGK133 is used for the measurement of external tapers 1:12 and 1:30 with a taper diameter of 27 mm to 205 mm.

The reproducibility of the measurement results is less than 1 μm.

The gauge rests on the workpiece with four hardened and polished pins. The position of the gauge on the taper is defined by these pins and a stop. The stop can be attached to either the front or back of the gauge.

The gauge has two movable measuring brackets. One of these is in contact with the small taper diameter, the other with the large taper diameter. There is a fixed spacing between the measuring brackets. The deviation of the taper diameter from the nominal value is displayed in both measurement planes by a precision indicator.

The gauge is set using a reference taper (available by agreement).

Ordering designation: Available by agreement

Taper gauge for taper angle 0° to 6°

The taper gauge MGK132 is used for the measurement of external tapers with a taper angle of 0° to 6° and a taper diameter of 90 mm to 360 mm.

The reproducibility of the measurement results is less than 1 μm.

The gauge rests on the workpiece with four hardened, ground and lapped ledges. The ledges form an angle of 90°. The position of the gauge on the taper is defined by a stop on the front or back of the gauge.

The measurement slide runs between the support ledges. A dial gauge in the housing acts against the measurement slide and displays the deviation of the taper diameter from the nominal value. The deviation of the taper from the nominal value is displayed by a precision indicator on the measurement slide.

The gauge is set using a reference taper (available by agreement).

Ordering designation: Available by agreement

Snap gauges

Snap gauges SNAP GAUGE, see table, can be used to check the diameter of cylindrical workpieces directly on the machine tool. The snap gauge is also used to set the enveloping circle gauge MGI21.

The snap gauge functions as a comparator gauge. It is set using shims. The deviation from the set value can then be determined.

Available snap gauges

Designation	Diameter range	
	min. mm	max. mm
SNAP-GAUGE-30/60	30	60
SNAP-GAUGE-60/100	60	100
SNAP-GAUGE-100/150	100	150
SNAP-GAUGE-150/200	150	200
SNAP-GAUGE-200/250	200	250
SNAP-GAUGE-250/300	250	300

Shims for numerous diameters are available as accessories.

Ordering example: Snap gauge for shaft diameter 120 mm
Ordering designation: **SNAP-GAUGE-100/150**

Ordering example: Shim for shaft diameter 120 mm
Ordering designation: **SNAP-GAUGE.MASTER120**

Enveloping circle gauges

Enveloping circle gauges, see table, can be used to set the radial internal clearance or preload of cylindrical roller bearings.

Available enveloping circle gauges

Designation	Design	For bearings	
		from	to
MGI21	For cylindrical roller bearings with separable inner ring	NNU4920-K	NNU4948-K
		NNU4920	NNU4948
MGA31	For cylindrical roller bearings with separable outer ring	NN3006-K	NNU3048-K
		N1006-K	N1048-K

Bearings with separable inner rings

The enveloping circle gauge MGI21 is used to measure, by means of two hardened and precision ground surfaces, the internal enveloping circle of a roller and cage assembly. One measurement surface is movable.

Before measurement, the gauge is set to the internal enveloping circle of the roller and cage assembly. This setting operation requires a snap gauge such as SNAP GAUGE.

After mounting of the outer ring together with the roller and cage assembly, the enveloping circle diameter is then determined using the gauge MGI in a comparative measurement.

In the case of a bearing with a tapered bore, the enveloping circle measurement is used to calculate its position on the tapered seat of the shaft. During mounting, the bearing is driven to this position. This results in the internal clearance or the preload.

In the case of bearings with a cylindrical bore, preground inner rings (suffix F12) are used and are finish ground to the required bearing diameter.

Ordering example: Enveloping circle gauge for cylindrical roller bearings NNU4920

Ordering designation: **MGI21-NNU4920**

Bearings with separable outer rings

The enveloping circle gauge MGA31 is used to measure, by means of two hardened and precision ground surfaces, the external enveloping circle of the roller and cage assembly.

The gauge is set to the raceway diameter of the mounted outer ring. This dimension is determined using a conventional internal gauge.

The tapered shaft with the premounted inner ring and roller and cage assembly is then inserted in the gauge. The shaft is driven axially by the hydraulic method until the required radial internal clearance or preload is achieved.

Ordering example: Enveloping circle gauge for cylindrical roller bearings NN3006-K

Ordering designation: **MGA31-NN3006**

Visual inspection device

In the reconditioning of wheelset bearings for rail vehicles (TAROL units), the bearing inner rings are subjected to visual examination after dismounting and cleaning. In order to check the condition of components, a device with a light and magnifying lens is used to visually assess the raceways, rings and all rolling elements.

Axial clearance gauge

The bearing is mounted on the gauge by means of the adapter set. The dial gauge is positioned on the end face of the outer ring and set to zero. By means of an eccentric mechanism, the bearing is raised by its inner ring and the axial clearance present can be read from the dial gauge.

Base device

The base device is suitable for all TAROL units. It comprises a frame and the measuring unit with a dial gauge.

Ordering example: Axial clearance gauge for TAROL units

Ordering designation: **TOOL-RAILWAY-CLEARANCE-BASIC**

Bearing-specific adapter set

The adapter set facilitates the precise positioning of the bearing on the base device.

Ordering example: Adapter set for TAROL unit F-578116.TAROL100/175-R-TVP

Ordering designation: **TOOL-RAILWAY-CLEARANCE.TOP-100/175**

Product overview Accessories

Transport and mounting tool

BEARING-MATE

Mounting paste

ARCANOL-MOUNTINGPASTE

Anti-corrosion oil

ARCANOL-ANTICORROSIONOIL

Accessories

Features

Accessories are used to assist in the storage, transport and mounting of rolling bearings.

Transport and mounting tool

The transport and mounting tool BEARING MATE, see table, is an accessory for the easy handling of medium-sized and large rolling bearings. It can also be used in the heating of bearings prior to mounting.

The tool comprises two handles and two steel strips. The steel strips are tightly clamped on the outer ring of the bearing. During the transport of spherical roller bearings and self-aligning ball bearings, tilting of the inner rings is prevented by the brackets supplied.

The bearing together with the tool is carried by either two people or by means of a crane. While it is being transported by crane, the bearing is suspended by the tool using the carrying slings and can be rotated into any position required.

During heating by means of an induction heating device HEATER, the bearing can remain in the tool. It expands to the same extent as the bearing. During heating by means of an induction device with medium frequency technology, the flexible inductor must not be in direct contact with the BEARING MATE.

The tool can be used for bearings up to a mass of 500 kg and at temperatures of up to +160 °C.

Available tools

Designation	Bearing outside diameter mm		Mass of tool kg
	from	to	
BEARING-MATE250-450	250	450	6,3
BEARING-MATE450-650	450	650	6,4
BEARING-MATE650-850	650	850	6,5

Ordering example: Transport and mounting tool for bearings with an outside diameter from 250 mm to 450 mm with two short brackets

Ordering designation: **BEARING-MATE250-450**

Accessories

Accessories, brackets 2 long brackets to prevent tilting of the inner rings of spherical roller bearings

Ordering designation **BEARING-MATE-LOCKBAR270**

Accessories, pack of small parts Pack of small parts

Ordering designation **BEARING-MATE.SERVICE-KIT**

Mounting paste

The mounting paste, see table, facilitates the sliding into place of bearing rings and prevents stick/slip effects, scoring, wear and fretting corrosion. It also gives protection against corrosion.

The operating temperature range is between –30 °C and +150 °C.

The paste is resistant to water, water vapour and many alkaline and acidic media.

Available mounting pastes

Designation	Container
ARCANOL-MOUNTINGPASTE-70G	Tube containing 70 g
ARCANOL-MOUNTINGPASTE-250G	Tube containing 250 g
ARCANOL-MOUNTINGPASTE-400G	Cartridge containing 400 g
ARCANOL-MOUNTINGPASTE-1KG	Can containing 1 kg

Anti-corrosion oil

This oil gives protection of bearings that have been unpacked. It also gives long term protection against corrosion of bright metallic surfaces, for example on devices and machinery, during storage indoors.

In general, it is not necessary to wash the anti-corrosion oil out of rolling bearings. It gives neutral behaviour towards conventional rolling bearing greases and oils.

The oil can be removed using alkaline solvents or neutral cleaning agents.

Ordering example Spray can containing 0,4 l

Ordering designation **ARCANOL-ANTICORROSIONOIL-400G**

Products: Lubrication

Products: Lubrication

Grease		Characteristic applications	Operating temperature °C		Continuous limit temperature °C	Thickener
			from	to		
Multi-purpose greases	MULTITOP	Ball and roller bearings in rolling mills Construction machinery Spinning and grinding spindles Automotive engineering	−50[1]	+140	+80	Lithium soap
	MULTI2	Ball bearings up to an outside diameter of 62 mm in large electric motors Agricultural and construction machinery Household appliances	−30	+120	+75	Lithium soap
	MULTI3	Ball bearings with an outside diameter of or more than 62 mm in large electric motors Agricultural and construction machinery Fans	−30	+120	+75	Lithium soap
High loads	LOAD150	Ball, roller and needle roller bearings Linear guidance systems in machine tools	−20	+140	+95	Lithium complex soap
	LOAD220	Ball and roller bearings in rolling mill plant Paper machinery Rail vehicles	−20	+140	+80	Lithium/calcium soap
	LOAD400	Ball and roller bearings in mining machinery Construction machinery Wind turbine main bearings	−40	+130	+80	Lithium/calcium soap
	LOAD460	Ball and roller bearings Wind turbines Bearings with pin cage	−40[1]	+130	+80	Lithium/calcium soap
	LOAD1000	Ball and roller bearings in mining machinery Construction machinery Cement plant	−30[1]	+130	+80	Lithium/calcium soap
High temperatures	TEMP90	Ball and roller bearings in couplings Electric motors Automotive engineering	−40	+160	+90	Polycarbamide
	TEMP110	Ball and roller bearings in electric motors Automotive engineering	−35	+160	+110	Lithium complex soap
	TEMP120	Ball and roller bearings in continuous casting plant Paper machinery	−30	+180	+120	Polycarbamide
	TEMP200	Ball and roller bearings in guide rollers for baking machinery Kiln trucks and chemical plant Piston pins in compressors	−30	+260	+200	PTFE
Special requirements	SPEED2,6	Ball bearings in machine tools Spindle bearings Rotary table bearings Instrument bearings	−40	+120	+80	Lithium complex soap
	VIB3	Ball and roller bearings in rotors for wind turbines (blade adjustment) Packaging machinery Rail vehicles	−30	+150	+90	Lithium complex soap
	FOOD2	Ball and roller bearings in applications with food contact (NSF-H1 registration, kosher and halal certification)	−30	+120	+70	Aluminium complex soap
	CLEAN-M	Ball, roller and needle roller bearings as well as linear guidance systems in clean room applications	−30	+180	+90	Polycarbamide
	MOTION2	Ball and roller bearings in oscillating operation Slewing rings in wind turbines	−40	+130	+75	Lithium soap

+++ Extremely suitable ++ Highly suitable + Suitable – Less suitable –– Not suitable

1) Measurement values according to Schaeffler FE8 low temperature test.

Base oil	Consis-tency NLGI	Base oil viscosity at +40 °C mm^2/s	Temperatures		Low friction, high speed	High load, low speed	Vibrations	Support for seals	Relubri-cation facility
			Low	High					
Partially synthetic oil	2	82	+++	++	++	+++	++	+	+++
Mineral oil	2	110	++	+	+	+	+	+	+++
Mineral oil	3	80	++	+	+	+	++	++	++
Mineral oil	2	160	+	++	–	+++	++	++	++
Mineral oil	2	245	+	+	–	+++	++	++	++
Mineral oil	2	400	+	+	–	+++	++	++	++
Mineral oil	1	400	++	+	–	+++	++	–	++
Mineral oil	2	1 000	+	+	––	+++	++	++	++
Partially synthetic oil	3	148	+++	++	+	+	+	++	++
Partially synthetic oil	2	130	+++	+++	++	+	+	+	+
Synthetic oil	2	400	++	+++	–	+++	+	++	+
Alkoxyfluoro oil	2	550	++	+++	––	++	+	+	+
Synthetic oil	2 – 3	25	+++	+	+++	––	–	+	+
Mineral oil	3	170	++	++	–	++	+++	++	–
Synthetic oil	2	150	++	–	+	+	+	+	+++
Ether oil	2	103	+++	+++	+	+	+	+	++
Synthetic oil	2	50	+++	+	–	++	+++	++	+

Lubricants

Features

A significant factor for the performance capability and life of a rolling bearing or linear unit is the selection of a suitable lubricant.

Rolling bearing greases Arcanol

Schaeffler has been investigating for decades which grease is the most suitable solution for which application. The Arcanol rolling bearing greases offer very good preconditions for favourable running behaviour of bearings and a long operating life and high operational security of the bearing arrangement. The lubricant range is graduated such that almost all areas of application are covered.

The areas of application of Arcanol greases were determined under widely differing operating conditions and with rolling bearings of all types by means of modern testing methods and testing systems.

In 2015 alone, Schaeffler used its own FE8 and FE9 test rigs to carry out more than 50 000 hours of testing, *Figure 1* and *Figure 2*, page 73. Arcanol rolling bearing greases have superior characteristics in all areas compared to normal greases.

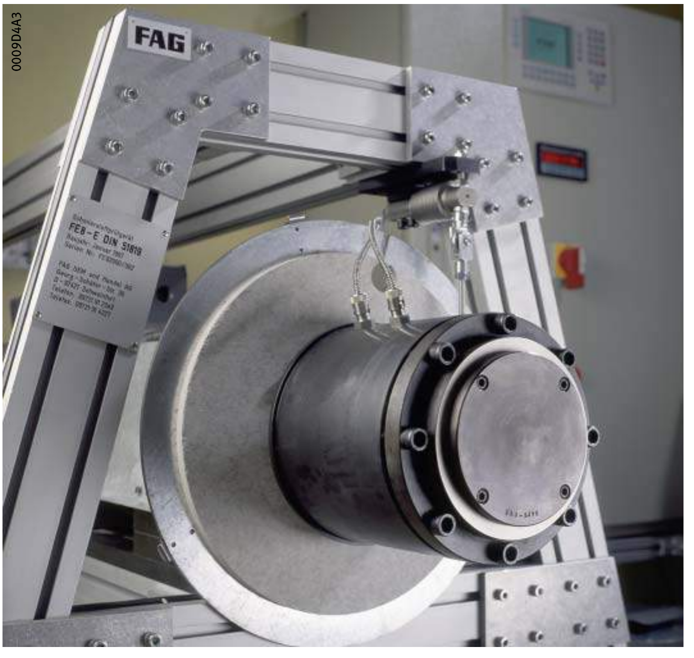


Figure 1
Test rig FE8

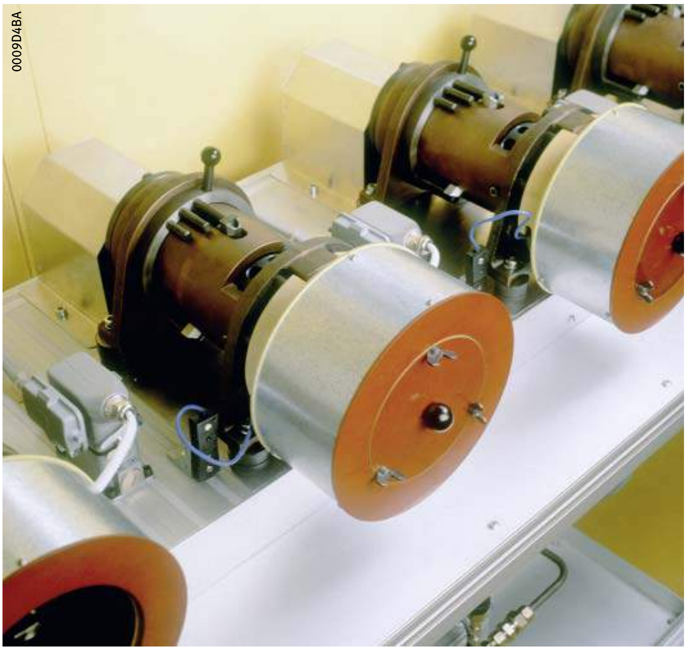


Figure 2
Test rig FE9

Based on the findings of the test rig runs, a range of greases has been developed that is subdivided into four groups:

- Multi-purpose greases:
  greases with a wide range of applications
- Heavy duty greases:
  greases suitable for high demands on load carrying capacity
- High temperature greases:
  greases that must withstand high application temperatures
- Special greases:
  greases that have been specially selected for a particular area of application.

Consistent product quality as a result of comprehensive quality inspection

Each delivery of Arcanol greases is subjected to comprehensive quality inspection. The quality of each batch can be clearly demonstrated and identified. In the in-house analysis laboratory, the chemical and physical characteristics of Arcanol greases are tested in accordance with strict test guidelines, thus ensuring the highest level of product quality.

Lubricants

Grease container sizes

Arcanol grease[1]	Tube		Cartridge	Can
	70 g	250 g	400 g	1 kg
MULTITOP	–	●	●	●
MULTI2	–	●	●	●
MULTI3	–	●	●	●
LOAD150	–	–	●	●
LOAD220	–	–	●	●
LOAD400	–	–	●	●
LOAD460	–	–	●	●
LOAD1000	–	–	–	–
TEMP90	–	–	●	●
TEMP110	–	–	●	●
TEMP120	–	–	●	●
TEMP200	●	–	–	●
SPEED2,6	–	●	●	●
VIB3	–	–	●	●
FOOD2	–	–	●	●
CLEAN-M	–	●	●	●
MOTION2	–	●	●	●
MOUNTINGPASTE	●	●	●	●

[1] Other containers are available by agreement.

Grease container sizes (continued)

Arcanol grease[1]	Bucket		Hobbock		Drum
	5 kg	12,5 kg	25 kg	50 kg	180 kg
MULTITOP	●	●	●	–	●
MULTI2	●	●	●	–	●
MULTI3	●	●	●	–	●
LOAD150	–	●	–	●	–
LOAD220	–	●	●	–	●
LOAD400	●	●	●	●	●
LOAD460	●	●	●	●	●
LOAD1000	●	–	●	●	●
TEMP90	●	–	●	–	●
TEMP110	–	–	–	●	–
TEMP120	●	–	●	–	–
SPEED2,6	–	–	●	–	–
VIB3	●	–	●	●	–
FOOD2	–	●	●	–	–
CLEAN-M	–	●	●	–	–
MOTION2	●	●	●	●	–

[1] Other containers are available by agreement.

Further information

- TPI 168, Rolling Bearing Greases Arcanol

Automatic relubrication devices

Lubricators

Lubrication systems

CONCEPT2

CONCEPT8

Lubrication system for spindle bearings

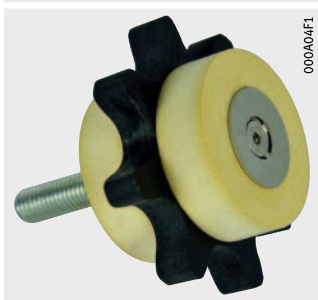
CONCEPT-PRECISION-GREASE

CONCEPT-PRECISION-OIL

Chain lubrication pinion

Lubrication gear

ARCALUB-X.CHAIN-PINION



ARCALUB-X.PINION

Pistol grease guns

ARCA-PUMP-BARREL.GUN-METER

Grease pumps

Drum pumps

Bearing-specific greasing tool

ARCA-PUMP-BARREL

TOOL-RAILWAY-GREASER

Lever grease guns

ARCA-GREASE-GUN

Lubrication devices

Features

Rolling bearings are automatically provided with the correct quantity of lubricant by lubricators and lubrication systems. This prevents the most frequent cause of rolling bearing failure: inadequate or incorrect lubrication. Approximately 90% of bearings are lubricated with grease. Relubrication with the correct quantity of grease at the appropriate intervals gives a significant increase in the life of rolling bearings. For manual relubrication, grease guns are suitable.

Automatic relubrication devices

Automatic relubrication devices convey fresh lubricant in the defined quantity at the correct time to the contact points of the rolling bearing. The devices adhere to the lubrication and maintenance intervals and prevent undersupply or oversupply of grease. Plant downtime and maintenance costs are reduced as a result.

The relubrication devices are matched to the bearing positions. They have a wide range of applications, for example on electric motors, pumps, compressors and fans, in linear systems, conveying equipment or machine tools.

Lubricator CONCEPT2

This lubricator of protection class IP54 has a very compact design. It has one or two pump bodies that can be individually controlled, depending on the design. This means it can supply one or two lubrication points with lubricant. LC units are available in the size 250 cm^3. The lubricator is supplied with voltage either from a battery or via a mains power pack, see table, page 79. It can work independently or can be optionally controlled by an external control system.

Advantages

The advantages of the lubricator are as follows:

- easy operation and good overview
- supply of up to two lubrication points
- facility for setting different lubrication intervals for each lubrication point
- supply of set lubricant quantity independent of temperature
- counterpressure measurement up to the lubrication point
- reliable piston pump as delivery pump
- low ongoing maintenance costs
- favourable price/performance ratio
- operating temperature from –20 °C to +70 °C
- battery or mains operation (DC 24 V) possible
- pressure build-up to 50 bar (mains operation) or 30 bar (battery operation)
- differentiated alarm messages
- simple coupling with machine operation possible
- suitable for control via an external control system.

Available lubricators CONCEPT2

Designation	Design
CONCEPT2-1P	Battery version with one outlet
CONCEPT2-2P	Battery version with two outlets
CONCEPT2-1P-24VDC	24-V version with one outlet
CONCEPT2-2P-24VDC	24-V version with two outlets

Available LC units

Designation	
ARCALUB-C2.LC250-MULTITOP	**ARCALUB-C2.LC250-TEMP90**
ARCALUB-C2.LC250-MULTI2	**ARCALUB-C2.LC250-TEMP110**
ARCALUB-C2.LC250-LOAD150	**ARCALUB-C2.LC250-TEMP120**
ARCALUB-C2.LC250-LOAD220	**ARCALUB-C2.LC250-TEMP200**
ARCALUB-C2.LC250-LOAD400	**ARCALUB-C2.LC250-SPEED2,6**
ARCALUB-C2.LC250-LOAD460	**ARCALUB-C2.LC250-MOTION2**
ARCALUB-C2.LC250-LOAD1000	**ARCALUB-C2.LC250-FOOD2**
–	**ARCALUB-C2.LC250-CLEAN-M**

Further information

LC units are also available by agreement with other greases.

Lubrication system CONCEPT8

This single-point and multi-point lubrication system offers high flexibility. It has one, two, three or four pump bodies that can be individually controlled, depending on the design. Each pump body has two outlets and, as a result, up to eight lubrication points can be flexibly provided with the required quantity of lubricant in the correct lubrication interval using just one lubrication system.

The lubrication system CONCEPT8 is designed for a wide variety of operating conditions. Designs for linear systems, the use of oils as lubricant or with an internal heating facility are also available, see table Available lubrication systems, page 81. Lubricant cartridges (LC units) provide the device with lubricant, see table Available LC units, page 81. LC units are available in the size 800 cm^3.

The lubrication system is supplied with voltage from a mains power pack. Coupling with machine operation is possible if the voltage supply to machine and lubrication system is coupled, then the relubrication interval will always be dependent on the number of operating hours.

Advantages

The advantages of the lubrication system are as follows:

- easy operation and good overview
- suitable for oil and grease up to NLGI 3
- supply of up to eight lubrication points
- supply of set lubricant quantity independent of temperature
- counterpressure measurement up to the lubrication point
- reliable piston pump as delivery pump
- favourable price/performance ratio
- operating temperature from –20 °C to +70 °C
- facility for setting different lubrication intervals and lubricant quantities for each pump body
- low operating voltage of DC 24 V
- pressure build-up to 70 bar
- differentiated alarm messages
- simple coupling with machine operation possible
- suitable for control via an external control system.

Available lubrication systems CONCEPT8

Designation	
CONCEPT8-1P	**CONCEPT8-1P-CC**
CONCEPT8-2P	**CONCEPT8-2P-CC**
CONCEPT8-3P	**CONCEPT8-3P-CC**
CONCEPT8-4P	**CONCEPT8-4P-CC**
CONCEPT8-1P-LIN	**CONCEPT8-1P-OIL**
CONCEPT8-2P-LIN	**CONCEPT8-2P-OIL**
CONCEPT8-3P-LIN	**CONCEPT8-3P-OIL**
CONCEPT8-4P-LIN	**CONCEPT8-4P-OIL**

LIN = for linear applications
CC = with internal heating facility
OIL = oil version

Available LC units

Designation	
ARCALUB-C8.LC800-MULTITOP	**ARCALUB-C8.LC800-TEMP90**
ARCALUB-C8.LC800-MULTI2	**ARCALUB-C8.LC800-TEMP110**
ARCALUB-C8.LC800-MULTI3	**ARCALUB-C8.LC800-TEMP120**
ARCALUB-C8.LC800-LOAD150	**ARCALUB-C8.LC800-TEMP200**
ARCALUB-C8.LC800-LOAD220	**ARCALUB-C8.LC800-SPEED2,6**
ARCALUB-C8.LC800-LOAD400	**ARCALUB-C8.LC800-VIB3**
ARCALUB-C8.LC800-LOAD460	**ARCALUB-C8.LC800-MOTION2**
ARCALUB-C8.LC800-LOAD1000	**ARCALUB-C8.LC800-FOOD2**
–	**ARCALUB-C8.LC800-CLEAN-M**

Further information

- LC units are also available by agreement with other greases or with oils
- Other accessories available by agreement

Lubrication devices

Minimal quantity lubrication devices

The compact lubrication systems allows very precise and efficient supply of lubricant to spindle bearings.

Grease lubrication system for spindle bearings

The grease lubrication system for spindle bearings is specially designed in terms of the delivery volume per stroke for the greasing of main spindles, see table.

Hoses filled with grease are connected to the outlets. These constitute the lubricant reservoir for relubrication. The cartridge only contains a pressure agent that is pumped into the hoses during delivery. The lubricant and pressure agent are separated from each other by a ball in the hose.

The lubricant is only subjected to pressure during the relubrication process in order to prevent separation of the grease.

Advantages

The advantages of the lubrication system are:

- optimum relubrication of the main spindles by means of very small quantities
- prevention of impermissible temperature increases
- suitable for bearings with different lubrication requirements
- simple coupling with machine operation possible
- favourable price/performance ratio.

Available grease lubrication system for spindle bearings

Designation	Outlet ducts Quantity	Delivery quantity per outlet duct cm^3/stroke
CONCEPT-PRECISION-GREASE	2	0,023

Oil lubrication system for spindle bearings

Spindle bearings run at high speeds. For this reason, pneumatic oil lubrication has previously been used for speed parameters above 1 600 000 mm/min. This requires extremely clean, dry compressed air. The high costs of this compressed air are not incurred with direct oil lubrication. The compressed air is replaced by a damper throttle element. This element gives an almost continuous delivery rate. Two systems are available, see table. One version has an internal oil tank, while the other is supplied via a connection adapter for an external oil tank.

Advantages

The advantages of the lubrication system are:

- optimum relubrication of main spindles by means of very small quantities at a constant delivery rate
- suitable for bearings with different lubrication requirements
- simple coupling with machine operation possible
- no compressed air costs for bearing lubrication
- no risk of spindle failure due to inadequate air cleanliness
- favourable price/performance ratio.

Available oil lubrication system for spindle bearings

Designation	Oil tank cm^3
CONCEPT-PRECISION-OIL-250	250
CONCEPT-PRECISION-OIL	–

Lubrication devices

Chain lubrication pinion

A chain lubrication pinion is used to supply chains with chain oil according to requirements and fully automatically. The rollers are made from open cell PU foam and transfer very small quantities of oil to the highest points on the chain links. From there, the oil passes between the inner and outer links and between the pins and rollers.

There is no coating of other surfaces of the chain with oil, which is unnecessary and in many case undesirable.

The chain pinion segments are made from special plastic and convert the linear (translation) movement of the chain into rotary motion (rotation) of the chain lubrication pinion. Despite the non-uniform surface of the chain links, the co-rotation of the plastic pinion ensures very quiet running of the lubrication pinion even at very high speeds.

Chain lubrication pinions are available for standard chains, simplex, duplex, triplex and also special chains.

Lubrication gear

A lubrication system comprises a lubrication gear and drive pinion or a lubrication gear and toothed rack. The lubrication gear gives automatic, continuous relubrication of the open tooth sets of the drive pinion or toothed rack.

A lubrication gear is connected to the drive pinion or toothed rack. The lubrication gear, made from open cell PU foam, stores the lubricant and transfers it in very small metering quantities to the tooth set in contact. This facilitates optimum supply to the tooth sets over very long periods and prevents both overlubrication as well as wear due to lubricant starvation.

A lubrication gear does not transmit either force or torque.

The following tooth sets are available:

- straight teeth
- helical teeth, helix angle up to 45°
- modulus: 2 to 30
- width: up to 700 mm.

Pistol grease gun

The pistol grease gun has a 4 digit digital counter that displays the lubricant quantity in grams. The specific mass of the lubricant can be set.

Ordering designation **ARCA-PUMP-BARREL.GUN-METER**

The features of the pistol grease gun are as follows:

- measurement range: 0,1 g to 1000 g
- display of gram counter: 4 digits
- display of total counter (kg): 4 digits
- maximum operating pressure: 600 bar
- burst pressure: 1000 bar
- maximum operating temperature: +60 °C
- tolerance: ±3% of displayed value
- battery life: 24 months
- inlet: z type swivel joint G $^1/_4$
- outlet: nozzle tube with 4 jaw nozzle
- mass: 1,7 kg.

Grease pumps

Grease pumps are driven by pneumatic or manual means.

Drum pumps

Drum pumps ARCA-PUMP-BARREL, see table, are pneumatically driven and suitable for delivering large quantities of grease under high pressure over long distances. Drum pumps can be used either as delivery pumps for individual greasing stations or as a supply pump for central lubrication systems.

Available drum pumps

Designation	Pump ratio	Delivery rate at 6 bar g/min	Air consump-tion l/min	Suitable for container sizes kg
ARCA-PUMP-BARREL-25-S	70:1	1 100	150	25
ARCA-PUMP-BARREL-50-S	70:1	1 100	150	50
ARCA-PUMP-BARREL-180-S	70:1	1 100	150	180

The following accessories are available for the drum pumps: drum cover (dust cover), follower plate, high pressure delivery hoses and pistol grease guns.

Bearing-specific greasing tools

In the reconditioning of wheel bearing sets for rail vehicles (TAROL units), rapid and uniform greasing can be achieved by means of bearing-specific greasing tools. The tool is connected to a drum pump that supplies the appropriate grease quantity.

Ordering example: Greasing tool for bearing F-561775

Ordering designation: **TOOL-RAILWAY-GREASER-F-561775**

Further information

- TPI 156, Tapered Roller Bearing Units TAROL – Mounting, Maintenance, Repair

Lever grease gun and reinforced hose

The lever grease gun, see table, can be used to manually relubricate rolling bearings via lubrication nipples.

The container on the lever grease gun can be filled with 500 g loose grease or with a 400-g cartridge. The cartridge must conform to DIN 1284 (diameter 53,5 mm, length 235 mm).

The lever grease gun is connected to the lubrication nipple via a reinforced hose. The reinforced hose must be ordered separately, see table. The connector thread is G$^1/_8$. The reinforced hose has a hydraulic grip coupling for connection to the taper type lubrication nipple in accordance with DIN 71412.

Alternatively, the reinforced hose can be fitted with a connector for cylindrical lubrication nipples in accordance with DIN 3404.

In place of the hydraulic grip coupling, slide couplings for button head lubrication nipples in accordance with DIN 3404 or other nozzles can be connected. These connectors are available from normal trade outlets.

Available lever grease guns

Designation	Maximum delivery pressure bar	Delivery quantity per stroke cm^3
ARCA-GREASE-GUN	800	2

Available reinforced hoses

Designation	Length mm	Connector
ARCA-GREASE-GUN.HOOK-ON-HOSE	300	Cylindrical lubrication nipples with head 16 mm in accordance with DIN 3404
ARCA-GREASE-GUN.HOSE	300	Taper type lubrication nipples in accordance with DIN 71412

FAG

Products: Condition Monitoring

Products: Condition Monitoring

Product overview Alignment

Belt pulley alignment device
Top-Laser SMARTY2
Belt tension measuring device
Top-Laser TRUMMY2

LASER-SMARTY2	LASER-TRUMMY2
 00019CE3	 00019C83

Shaft alignment device
Top-Laser EQUILIGN
Shims
Top-Laser SHIM

LASER-EQUILIGN	LASER-SHIM
 00019C84	 00019D92

Alignment

Features

These products assist in the alignment of shafts and belt pulleys and the checking of belt tension values.

Belt pulley alignment device FAG Top-Laser SMARTY2

The FAG Top-Laser SMARTY2 is a line laser for the alignment of belt pulleys and chain sprockets with a diameter of more than 60 mm.

The alignment of belt pulleys and chain sprockets reduces wear and energy losses in tension drives, their bearings and seals. Less heat is generated and the lifetime and reliability of the machines is increased.

The features of the line laser are as follows:

- The parallelism and angular errors of the two pulleys are displayed.
- Alignment can be carried out on both horizontally and vertically mounted belt pulleys.
- Alignment is significantly more rapid and more precise than with conventional methods.
- Alignment can be carried out by one person working alone.
- The measuring device is attached to the pulleys by magnetism.

The laser can be clearly seen on the target marks. Once the machine is adjusted such that the laser beam coincides with the slots in the target marks, it is correctly aligned.

The target marks are available in an optical, *Figure 2*, page 94, and an electronic design, *Figure 1* and *Figure 3*, page 94. In the case of electronic target marks, the adjustment values are displayed in real time on the digital display. Angular errors are shown in degrees, while the parallelism offset is shown in millimetres.

Caution ⚠ Do not look into the laser beam or point the laser beam into another person’s eyes.

Figure 1
Electronic target mark

All the parts are supplied in a lined case, *Figure 2*.

Scope of delivery

1 emitter
2 optical target marks, attached by magnetism
1 battery
1 lined case

Ordering designation **LASER-SMARTY2**

① Emitter
② Optical target mark
③ Battery
④ Lined case

Figure 2
Scope of delivery
FAG Top-Laser SMARTY2

Replacement part 1 optical target mark, attached by magnetism

Ordering designation **LASER-SMARTY2.TARGET**

Accessories

1 electronic target mark, attached by magnetism
1 case

Ordering designation **LASER-SMARTY2.TARGET-DIGITAL**

① Electronic target mark
② Case

Figure 3
Scope of delivery
FAG Top-Laser TARGET-DIGITAL

Belt tension measuring device FAG Top-Laser TRUMMY2

The robust, handy FAG Top-Laser TRUMMY2 is an optical-electronic manual measuring instrument for belt tension (strand force).

The correct belt tension is an essential prequisite for achieving the maximum life of the belt drive. In addition, this gives reduced wear of the drive components, lower energy costs and increased cost-efficiency.

The FAG Top-Laser TRUMMY2 comprises a cableless measurement probe for direct connection, a measurement probe with cable for difficult to access locations and a manual control device that displays the relevant measurables for belt tension as a frequency in Hz or force in N.

Caution ⚠ Do not look into the laser beam or point the laser beam into another person’s eyes.

The simple and reliable user instructions are given in several languages.

All the parts of the belt tension measuring device are supplied packed in a case, *Figure 4*.

Scope of delivery

1 manual control device
1 measurement probe for direct connection
1 measurement probe with cable
1 case

Ordering designation **LASER-TRUMMY2**

① Manual control device
② Measurement probe, direct connection
③ Measurement probe, cable connection
④ Case

Figure 4
Scope of delivery
FAG Top-Laser TRUMMY2

The belt tension measuring device should be calibrated at least every 2 years. The FAG Top-Laser TRUMMY2 should be sent to us for this purpose.

Service Calibration

Ordering designation **LASER-TRUMMY.CALI-CHECK**

Application

Before calculating the belt tension, the belt mass and length must be entered. Vibration of the belt is then induced. The device measures the natural frequency by means of clock pulse light and uses this to determine the belt tension, *Figure 5*. This technique is less prone to disruptive influences in comparison with measurement using sound waves.

① Belt
② TRUMMY2, cableless measurement probe

Figure 5
Measurement

Shaft alignment device FAG Top-Laser EQUILIGN

The FAG Top-Laser EQUILIGN, *Figure 6*, is an alignment system for coupled and decoupled shafts in motors, pumps, ventilators and gearboxes with rolling bearings.

The advantages of the system are:

- simple mounting
- error-free handling even by untrained personnel using step-by-step display on the manual control device
- automatic tolerance checking. A symbol indicates when the shafts are correctly aligned
- more precise alignment than with conventional methods
- rapid, simple measurement by means of Active Clock measurement mode
- robust control device. Watertight and insensitive to contamination in accordance with IP 65
- user interface in 20 languages
- easy generation of reports
- real time display of displacement in all axes.

Caution ⚠ Do not look into the laser beam or point the laser beam into another person’s eyes.

Figure 6
Shaft alignment device
FAG Top-Laser EQUILIGN

All the parts of the shaft alignment device are supplied packed in a case, *Figure 7*.

Scope of delivery

1 manual control device
1 emitter and receiver including cable 2 m long
1 reflector
5 batteries
1 Allen key
1 cable for connecting USB memory stick to device
1 cable for connecting device to PC via USB port
2 brackets
2 chains, 300 mm long
4 posts, 115 mm long
1 tape measure
1 case

Ordering designation **LASER-EQUILIGN**

① Manual control device
② Emitter/receiver
③ Reflector
④ Batteries, LR6 (AA) DC 1,5 V, 5 pieces
⑤ Allen key, 4 mm
⑥ Cable for USB memory stick
⑦ Cable for PC
⑧ Bracket
⑨ Chain, 300 mm long
⑩ Post, 115 mm long
⑪ Tape measure
⑫ Case

Figure 7
Scope of delivery
FAG Top-Laser EQUILIGN

Replacement parts

Designation	Description	Scope of delivery Quantity
LASER-EQUILIGN-DEVICE	Manual control device	1
LASER-EQUILIGN.TRANS	Emitter/receiver with cable	1
LASER-EQUILIGN.REFLECT	Reflector	1
LASER-EQUILIGN.USB-CABLE	Cable for USB memory stick, 2 m long	1
LASER-EQUILIGN.PC-CABLE	Cable for PC, 2 m long	1
LASER.BRACKET	Bracket	2
LASER.CHAIN300-SET	Chain, 300 mm long	2
LASER.POST115-SET	Post, 115 mm long	4
LASER.TAPE	Tape measure, 1 m long	1
LASER-EQUILIGN.CASE	Case	1

Comprehensive range of accessories

A comprehensive range of accessories is available in order to expand the possible applications of the base device FAG Top-Laser EQUILIGN.

The accessories can be ordered as a set in a handy, robust case or as individual parts.

Accessories, individual parts

Designation	Description	Scope of delivery Quantity
LASER.CHAIN600-SET	Chain, 600 mm long	2
LASER.CHAIN1500-SET	Chain, 1 500 mm long	2
LASER.POST150-SET	Post, 150 mm long	4
LASER.POST200-SET	Post, 200 mm long	4
LASER.POST250-SET	Post, 250 mm long	4
LASER.POST300-SET	Post, 300 mm long	4
LASER.BRACKET-MAGNET	Magnetic holder including 2 posts, 150 mm long	1

Accessories, set

Designation	Description	Scope of delivery Quantity
LASER.ACCESS-SET	Chain, 600 mm long	2
	Chain, 1 500 mm long	2
	Post, 150 mm long	4
	Post, 200 mm long	4
	Post, 250 mm long	4
	Post, 300 mm long	4
	Magnetic holder including 2 posts, 150 mm long	2
	Case	1

① Chains, 600 mm
② Chains, 1 500 mm
③ Posts, 150 mm
④ Posts, 200 mm
⑤ Posts, 250 mm
⑥ Posts, 300 mm
⑦ Magnetic holder
⑧ Case

Figure 8
Accessories, set

Alignment

Before alignment is carried out, any soft foot must be eliminated. FAG Top-LaserEQUILIGN clearly indicates the soft foot. Each individual screw foot connection is loosened and the device is monitored to see if it displays any changes between the foot screwed firmly into place and the loosened foot. The soft foot can then be eliminated using shims. This eliminates any tendency towards vibration and bearing damage as a result of housing deformation. During measurement, at least three positions are approached at different angles. These must be measured at an angle of at least 90°. The intelligent control system prevents incorrect usage here. The actual condition of the subassembly is then displayed, *Figure 9*.

① Display of actual condition
② Foot screw connection
③ Direction of vertical displacement
④ Direction of horizontal displacement

Figure 9
Alignment

Once the foot screw connections have been loosened, the vertical misalignment is first eliminated by means of shims. FAG Top-Laser EQUILIGN shows the displacement in real time. This means that the user can monitor on the display how the measurement results change as soon as the subassembly is moved. Horizontal adjustment is then carried out until the symbol with the thumb pointing upwards is displayed. Once the foot screw connections are tightened, the shafts are aligned.

Shims FAG Top-Laser SHIM

Shims FAG Top-Laser SHIM are used to eliminate vertical misalignment or soft feet.

These shims are made from corrosion-resistant high grade steel and are available in seven thicknesses (0,05 mm, 0,1 mm, 0,2 mm, 0,5 mm, 0,7 mm, 1 mm, 2 mm) and in four sizes (dimension c = 15 mm, 23 mm, 32 mm, 44 mm), *Figure 10* and table, page 103.

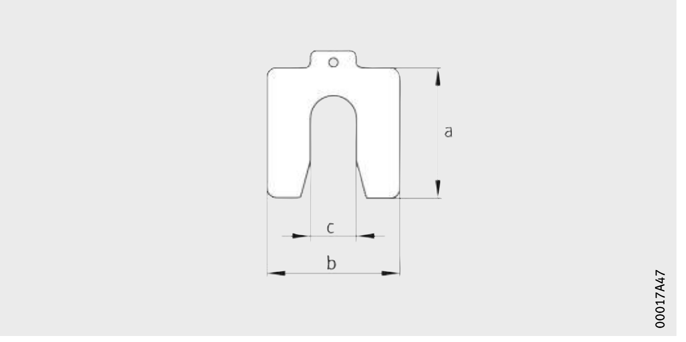


Figure 10
Shim, dimensions

Scope of delivery Basic set	Case 360 shims: 20 shims each in 3 sizes (dimension c = 15 mm, 23 mm, 32 mm) and 6 thicknesses (0,05 mm to 1 mm) 1 extraction hook
Ordering designation	**LASER.SHIM-SET**
Replacement parts	As replacement parts, we can supply 10 shims each in one of the 4 sizes and one of the 7 thicknesses stated above.
Ordering example 1	10 shims of dimension c = 15 mm and thickness 0,2 mm
Ordering designation	**LASER.SHIM15×0,20**
Ordering example 2	10 shims of dimension c = 44 mm and thickness 0,1 mm
Ordering designation	**LASER.SHIM44×0,10**

Available shims

Designation	Mass	Dimensions in mm			
	m g	a	b	c	Thickness
LASER.SHIM15×0,05	11	55	50	15	0,05
LASER.SHIM15×0,10	22	55	50	15	0,1
LASER.SHIM15×0,20	44	55	50	15	0,2
LASER.SHIM15×0,50	110	55	50	15	0,5
LASER.SHIM15×0,70	155	55	50	15	0,7
LASER.SHIM15×1,00	220	55	50	15	1
LASER.SHIM15×2,00	440	55	50	15	2
LASER.SHIM23×0,05	21	75	70	23	0,05
LASER.SHIM23×0,10	42	75	70	23	0,1
LASER.SHIM23×0,20	84	75	70	23	0,2
LASER.SHIM23×0,50	210	75	70	23	0,5
LASER.SHIM23×0,70	295	75	70	23	0,7
LASER.SHIM23×1,00	420	75	70	23	1
LASER.SHIM23×2,00	840	75	70	23	2
LASER.SHIM32×0,05	29	90	80	32	0,05
LASER.SHIM32×0,10	58	90	80	32	0,1
LASER.SHIM32×0,20	115	90	80	32	0,2
LASER.SHIM32×0,50	290	90	80	32	0,5
LASER.SHIM32×0,70	410	90	80	32	0,7
LASER.SHIM32×1,00	580	90	80	32	1
LASER.SHIM32×2,00	1 160	90	80	32	2
LASER.SHIM44×0,05	53	125	105	44	0,05
LASER.SHIM44×0,10	106	125	105	44	0,1
LASER.SHIM44×0,20	212	125	105	44	0,2
LASER.SHIM44×0,50	530	125	105	44	0,5
LASER.SHIM44×0,70	742	125	105	44	0,7
LASER.SHIM44×1,00	1 050	125	105	44	1
LASER.SHIM44×2,00	2 100	125	105	44	2

Further information

- TPI 182, FAG Alignment Tools – Top-Laser:
  SMARTY2 · TRUMMY2 · EQUILIGN · SHIM

Vibration measuring device Detector III

DETECT3-KIT, DETECT3-KIT-RFID, DETECT3.BALANCE-KIT

000190B1

Online monitoring system SmartCheck SmartQB

SMART-CHECK

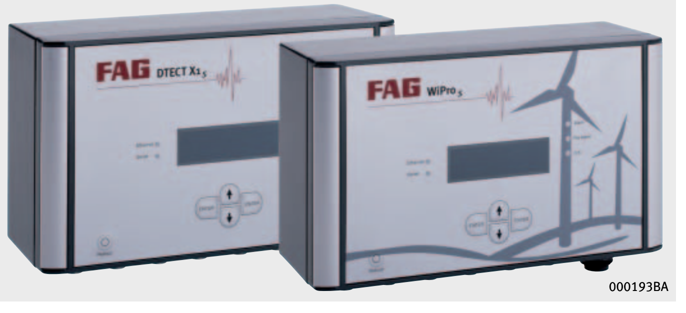
00017A65

SMART-QB

0009F9B6

Online monitoring system DTECT X1$_S$ WiPro$_S$

DTECTX1-S, DTECTX1-S-WIPRO



000193BA

Online monitoring system ProCheck

PRO-CHECK

000179B9

Vibration diagnosis

Features

Vibration diagnosis is the most reliable method for identifying the start of machine damage at an early stage. Unbalance and misalignment defects can be detected accurately, as well as rolling bearing damage and gear tooth defects.

FAG vibration measuring devices help in planning maintenance work, increasing bearing life and reducing costs. As a result, plant availability is increased and the risk of unplanned downtime is reduced.

Monitoring devices – offline and online

In the field of offline monitoring devices (regular monitoring), Schaeffler offers the FAG Detector III.

The online monitoring systems (continuous measurement) include FAG SmartCheck, FAG DTECT $X1_S$, FAG $WiPro_S$ and FAG ProCheck.

In order to achieve optimum networking to plant control facilities or monitoring centres, all online systems have versatile communication options as standard.

Worldwide service

On all devices relating to condition monitoring, Schaeffler offers a worldwide service – from the customer hotline to customer-specific service contracts.

Vibration measuring device FAG Detector III

FAG Detector III is a handy, easy to use vibration measuring device. Preinstalled standard configurations in accordance with DIN ISO 10816 make this a Plug-and-Play solution and allow authoritative information on the machinery condition, entirely without time-consuming training or system configuration.

This allows, for example, rapid inspection of ventilators, pumps, electric motors, compressors or vacuum pumps. All the user must do is start the measurement process by pressing a few buttons and wait until it is completed. The device evaluates the measurement results and presents the results, with self-explanatory symbols, on the device display, *Figure 1*.

① Value OK
② Prealarm
③ Main alarm

Figure 1
Symbols in the device display

The system also has the following features:

- non-contact temperature measurement
- speed detection
- route function
- report generator.

Analysis software

For more detailed analysis, the PC software Trendline is available free of charge and includes comprehensive functions.

These include the Viewer, which gives the user a wide range of tools for data evaluation. The integrated rolling bearing database, containing approximately 20 000 bearings from various manufacturers, facilitates easier and more efficient analysis of the measured data. Since the damage frequencies can be incorporated in the measurement results, simple damage analysis is possible.

Automatic detection of measurement points

The automatic detection of measurement points through the use of RFID technology gives error-free and precise identification of the measurement points on a measurement route. FAG Detector III identifies the measurement points by means of RFID tags on the machinery. With the proven FAG Detector III, mobile vibration and temperature monitoring is thus quicker, simpler and more reliable.

The functionality of automatic measurement point detection is not available worldwide.

Balancing function

A further special feature of FAG Detector III is the balancing function. For this purpose, the optionally available Balancing Kit is required. This makes it possible to not only detect but also eliminate unbalances.

The results of the balancing process are also transferred to the Trendline software for evaluation.

Ordering examples	The vibration measuring device FAG Detector III is available in two variants, with the balancing function available for ordering in a upgraded version.
Scope of delivery Base device	1 base device with rechargeable battery 1 accelerometer, attached by magnetism, and sensor cable 1 infrared temperature sensor 1 charger with worldwide compatibility 1 PC data cable (serial and USB) 1 user manual 1 protective bag with holder for temperature sensor 1 Trendline PC software free of charge 1 case
Ordering designation	**DETECT3-KIT**
Scope of delivery Device with automatic measurement point detection	As DETECT3-KIT 1 RFID reader (integrated) 5 RFID tags for identifying the measurement point
Ordering designation	**DETECT3-KIT-RFID**
Scope of delivery Upgrade level with balancing function	1 accelerometer, attached by magnetism, and sensor cable 1 trigger sensor, optical 1 trigger sensor, induction 1 reflective mark for trigger sensor 1 cable for trigger sensor, 10 m 1 magnetic holder for trigger sensor 1 extension for magnetic holder 1 balance 1 dongle for activation of balancing function 1 case
Ordering designation	**DETECT3.BALANCE-KIT**
Accessories	Sensor extension cables with a length of 5 m or 15 m are available by agreement. The charging dock, mounting pads and additional RFID tags are available by agreement.

Online monitoring system FAG SmartCheck

FAG SmartCheck is a compact, innovative, modular online measuring system for continuous monitoring of machinery and process parameters on a decentralised basis. It can be used on assemblies where such monitoring was previously too cost-intensive.

FAG SmartCheck is suitable, for example, for early detection of rolling bearing damage, unbalances and misalignments on:

- electric and geared motors
- vacuum and fluid pumps
- ventilators and fans
- gearboxes and compressors
- spindles and machine tools
- separators and decanters.

Plug-and-Play system

FAG SmartCheck is ready for immediate use. When supplied, it already has a set of key data that facilitate general, reliable machinery monitoring.

In addition, predefined configuration templates are available for monitoring of items such as fans, pumps and bearings. These can easily be matched to individual requirements. Due to the integrated bearing database of FAG and INA standard bearings, data configuration and later analysis are simpler. The system has an independent teaching mode that identifies the alarm thresholds.

Parameters monitored

In addition to the standard parameters of vibration and temperature, it is possible to record other classic operating parameters such as pressure or flow rate. All parameters can be correlated with each other and included in the alarm configuration.

The data are recorded and analysed centrally by the system. The current machine condition can be displayed directly on the device or transferred to any control facility as required. It is only necessary to integrate FAG SmartCheck in the existing network structure.

Mitsubishi control system

General communication with controllers can be carried out through connection of the analogue and digital interfaces with the controller. The communication protocol SLMP has been implemented specially for Mitsubishi controllers of the L and Q series. This allows direct transfer of the measurables status and gives information, for example, on rolling bearing damage, unbalance, misalignments or temperature fluctuations that can be notified in plain text to the operator by means of the controller.

Access via the Internet

FAG SmartCheck has an intuitive user concept designed as a Web interface. It is therefore possible to access the system via the Internet using any standard Internet browser. The Web interface can be used to configure the system and view current data.

Remote monitoring

The data can be transferred to other locations by remote access and analysed there by the operator or external service providers such as the Schaeffler vibration experts. This is particularly important for customers who still have little experience of data analysis or wish to outsource this function.

Online monitoring system FAG SmartQB

FAG SmartQB is an easy way to get started in Condition Monitoring. It monitors the vibrations in electric motors, pumps and fans. Commissioning can be carried out by an employee who does not have specific knowledge in the field of vibration diagnosis. The 7″ display shows user-friendly plain text messages, *Figure 2*.

The features of FAG SmartQB are:

- suitable for machinery with fixed and variable speeds from 100 min^{-1} to 15 000 min^{-1}
- preconfigured for up to six sensors
- touch panel with plain text messages
- minimal installation work using 1 cable technique (Power over Ethernet)
- live display of current values
- trend pattern of damage development
- user interface in 16 languages
- RJ45 Ethernet interface for service technicians.

Figure 2
Online monitoring system
FAG SmartQB

The scope of delivery comprises three parts, *Figure 3*.

Scope of delivery

1 housing with sensor unit FAG SmartQB with 7″ touch panel
1 FAG SmartQB sensor 1
1 Ethernet cable, 10 m long

Ordering designation

SMART-QB

① Housing
② Sensor
③ Ethernet cable

Figure 3
Scope of delivery
Online monitoring system
FAG SmartQB

Replacement parts

Designation	Description	Scope of delivery Quantity
SMART-QB.SENSOR-1	Sensor 1	1
SMART-CHECK.CABLE-ETH-P-M12-RJ45-10M	Ethernet cable, 10 m long	1

Installation and commissioning

In addition, installation and commissioning are exceptionally simple. Any industrial electrician can install the system and, without prior knowledge of vibration technology, can carry out commissioning within five minutes.

The touch display gives personnel all relevant information, from mounting through recommended actions in the case of errors. At first startup, the customer selects one of 16 languages and, where necessary, replaces the preset contact details for technical support from Schaeffler with his own information.

After selection of the machine (electric motor, pump or fan) to which the FAG SmartQB Sensor is attached and the category “variable speed machine” or “constant speed machine” and input of the individual machine name, the FAG SmartQB automatically selects the best measurement configuration and the system is immediately ready for teach-in mode. This runs automatically.

A maximum of six sensors can be connected to one FAG SmartQB. Each sensor can monitor a different machine. New sensors can also be added using the menu just as easily as in first installation.

After commissioning, the FAG SmartQB shows relevant information for each sensor on the display.

Examples include:

- alarm status
- vibration values
- defect frequency
- maximum values
- mean values
- trend patterns.

Defect causes

The Condition Monitoring system can detect a total of five defect causes and output these on the display:

- bearing damage
- unbalance
- friction/cavitation
- increases in temperature
- significant changes in the vibration patterns.

Due to the automated allocation of defects, the maintenance personnel no longer require knowledge of vibration technology. Maintenance measures and any ordering of replacement parts as necessary can be immediately initiated through defect allocation.

Market sectors

FAG SmartQB is typically used in the following sectors:

- cement
- paper
- steel
- water management
- machinery and equipment building
- repair shops for electric motors, pumps and fans.

FAG SmartQB is optimised for use in these sectors and is supplied already configured. Due to the automated defect allocation, maintenance measures and any ordering of replacement parts as necessary can be immediately initiated.

Comprehensive range of accessories

An extensive range of accessories is available to expand the possible applications of the monitoring system FAG SmartQB, see table and *Figure 4*. The accessories can be ordered as individual items.

Accessories, individual parts

Designation	Description	Scope of delivery Quantity
SMART-QB.SENSOR-2	Sensor 2	1
SMART-QB.SENSOR-3	Sensor 3	1
SMART-QB.SENSOR-4	Sensor 4	1
SMART-QB.SENSOR-5	Sensor 5	1
SMART-QB.SENSOR-6	Sensor 6	1
SMART-CHECK.CABLE-ETH-P-M12-RJ45-20M	Ethernet cable, 20 m long	1
SMART-CHECK.CABLE-ETH-P-M12-RJ45-30M	Ethernet cable, 30 m long	1
SMART-QB.LAMP	Lamp incl. cable 2×10 m long	1

1 Sensor 2
2 Sensor 3
3 Sensor 4
4 Sensor 5
5 Sensor 6
6 Ethernet cable
7 Cable for lamp
8 Lamp

Figure 4
Accessories for FAG SmartQB

Online monitoring system FAG DTECT X1$_S$

FAG DTECT X1$_S$ is a versatile online system for the monitoring of rotating components and elements in the machinery and plant industry. Typical applications can be found, for example, in the steel, raw materials, paper and marine industries.

The system gives early, reliable detection of possible damage and thus helps to prevent unplanned and expensive downtime.
The risk of possible production shutdowns is reduced. As a result, the capacity utilisation of machinery and plant is increased.

Versatile system

The system can be tailored to customer-specific requirements by means of the software.

The base device has 8 measurement channels. All conventional acceleration, speed and travel sensors can be attached.

Due to its compact size and robust housing (protection class IP 67), it is suitable for a wide range of monitoring applications. It has standardised connectors allowing easy installation on machinery and plant.

Remote monitoring

Defects and damage can be detected on machinery without the need for a diagnosis expert on site. The data can be transferred to other locations by remote access and analysed there, for example by Schaeffler vibration experts.

Further information

- TPI 170, FAG DTECT X1$_S$

Online monitoring system FAG WiPro$_S$

FAG WiPro$_S$ allows online monitoring of wind farms – onshore and offshore. The system gives early and reliable detection of possible machine damage. This helps to prevent unplanned downtime and expensive secondary damage. Due to its small size, it can easily be accommodated in small spaces such as the nacelle of a wind turbine.

Versatile system

FAG WiPro$_S$ is equipped with a signal processor and evaluates all measurement signals internally. Due to the intelligent linking of expert knowledge with information from the turbine, it is possible to keep the transferred data volume very small. This is particularly important where a large number of turbines must be continuously monitored over a long period.

Remote monitoring

The automatic messaging function by means of TCP/IP, wifi modem (optional), landline modem or DSL router allows efficient worldwide monitoring. The data can be transferred to other locations by remote access and analysed there, for example by Schaeffler vibration experts.

Further information

- WL 80 373, Flyer FAG WiPro$_S$

Online monitoring system FAG ProCheck

FAG ProCheck is a versatile online monitoring system. It can be used to prevent unplanned downtime and for quality control. The system offers a high level of functionality and is available in a range of variants – from an 8 channel to a 16 channel system.

Parameters monitored

FAG ProCheck continuously records data on vibration, temperature and other process parameters and subsequently carries out their evaluation. As a result, incipient damage and its causes can be detected at a very early stage and the appropriate countermeasures can be introduced in good time. This gives a considerable reduction in operating costs.

In addition, FAG ProCheck offers the possibility of correlating a large number of analogue and digital input and output signals to the vibration data. These channels also allow simple communication with higher level systems such as process control systems.

Versatile system

Due to its extremely robust and compact design, this system is ideally suited for use in almost all industrial segments. The system can be used in steelworks, paper machinery, cement plants or in the oil and gas industry.

Remote monitoring

Defects and damage can be detected on machinery without the need for a diagnosis expert on site. The data can be transferred to other locations by remote access and analysed there, for example by Schaeffler vibration experts.

Explosion-protected variant

An explosion-protected version of FAG ProCheck is available by agreement. In this version, a specially pressure-encapsulated housing prevents the system coming into contact with an explosive atmosphere. This is because wherever flammable gases, vapours, fluids or dust occur, the presence of oxygen and an ignition source can rapidly cause an explosion.

Further information

- TPI WL 80-69, FAG ProCheck State of the Art Machine Monitoring for Maximum Availability

Customer-specific solutions

The display, which is individually tailored to the customer's requirements, gives a user interface that allows a rapid overview of the condition of the plant. Depending on the complexity of the plant, this display can be arranged on several levels.

Other monitoring systems

Other monitoring systems for the requirements of specific sectors are available by agreement.

Grease sensor GreaseCheck

GREASE-CHECK

Particle sensor Wear Debris Check

WEAR-DEBRIS-CHECK

Monitoring of lubricants

Features

The operating life of the lubricant is the decisive value for the bearing life. Depending on the application, either a grease or particle sensor can be used for monitoring. The lubricant can be topped up or replaced in good time before damage occurs.

Grease sensor FAG GreaseCheck

The grease sensor has a diameter of 5 mm and is inserted in a hole in the housing as close as possible to the rolling bearing. The sensor is positioned in the lubricant. This grease sensor optically measures the water content, the extent of grease deterioration and the grease temperature directly in the bearing arrangement. This information is transferred by cable to the evaluation unit, *Figure 1*. The evaluation unit generates an analogue signal and a digital system that gives the user rapid and simple information on the condition of the grease.

① Grease sensor
② Electronic evaluation system

Figure 1
Grease sensor and electronic evaluation system

In the past, bearings were regreased as a function of time. The grease quantities and lubrication intervals were calculated numerically. If the grease sensor is used, regreasing can be carried out as a function of condition.

Advantages

The grease sensor facilitates:

- lubrication appropriate to needs
- lower lubricant costs
- prevention of unplanned downtime
- lower maintenance costs
- lower equipment costs.

Further information

- TPI 234, Condition Monitoring of Greases in Rolling Bearings

Particle sensor FAG Wear Debris Check

Particle sensors of this type can be used to determine wear at an early stage in heavily loaded industrial gearboxes on the basis of particles in the oil. The debris particles that can indicate a failure can be detected in the oil several months in advance. Through monitoring of particles in the lubricant, damage is detected at an early stage. This helps to prevent secondary damage and downtime periods. The particle sensor is installed in an ancillary flow of the recirculating lubrication system in the gearbox ahead of the filter or in a separate circuit.

Typical applications for the FAG Wear Debris Check can be found, for example, in gearboxes in raw material extraction plant, planetary gearboxes in wind turbines or in ship propulsion systems.

The features of the particle sensor are as follows:

- monitoring of the number of particles in the oil
- differentiation of the particles into ferrous and non-ferrous metals
- classification of the particles according to size
- possible integration in an online monitoring system for linking of oil particle and vibration data.

Where oil and vibration monitoring facilities are combined, damage in gearboxes with recirculating oil lubrication can be detected at an early stage and the source can be determined. In this way, it is possible to prevent production shutdowns or secondary damage.

Further information

- WL 80 366, Flyer FAG Wear Debris Check

Services

Services

Features

Services

Features

Schaeffler offers, irrespective of the manufacturer of the bearing arrangement, a wide range of services relevant to the lifecycle of a rolling bearing: starting with mounting and progressing through maintenance to the reconditioning of rolling bearings.

During the operational phase, the Schaeffler experts provide support through services in the fields of condition monitoring and corrective maintenance. Companies that wish to build up their knowledge in the areas of rolling bearings and condition monitoring also have access to the Schaeffler training and consultancy portfolio on site, centrally or online. Our e-learning portfolio on the Internet provides an introduction to this field. In this way, customers benefit from the expertise of a leading supplier of rolling and plain bearings.

Mounting and dismounting

The Schaeffler Industrial Service experts offer mounting and dismounting services for rolling bearings that are applicable across industrial sectors. They have detailed knowledge and extensive experience in all industrial sectors.

The experts in the Industrial Service function are trained and skilled personnel who can provide reliable, rapid and competent assistance. The services are provided worldwide, either on site at your location or in Schaeffler workshop facilities.

Mounting and dismounting services

The mounting and dismounting services, *Figure 1*, page 125, include:

- mounting and dismounting of rolling bearings, plain bearings and bearing systems of all types by experts available worldwide
- instruction in the use of mounting tools
- installation of lubricators
- measurement and condition analysis of bearing arrangements
- problem solving and preparation of concept solutions
- design and manufacture of special tools
- rental of tools (only available in Europe)
- emergency service
- training courses on products and mounting
- certification of mounting and dismounting processes.

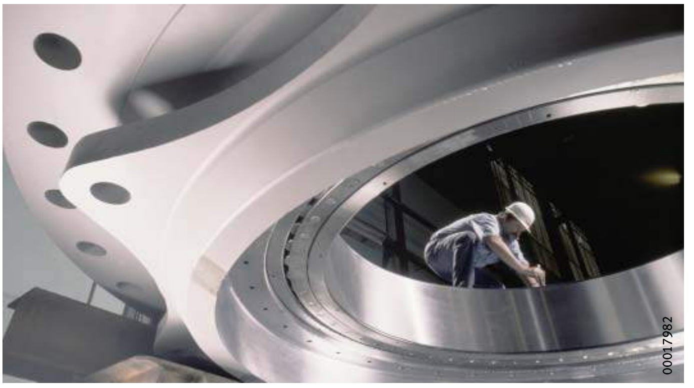


Figure 1
Mounting service

Advantages

The mounting services give the following advantages:

- readily available service worldwide
- correct mounting or dismounting through precise preparation
- professional mounting and dismounting using special high-quality tools
- increased plant availability and productivity as a result of reduced unplanned downtime
- correct use of bearings of all types as a result of customer training
- minimum time outlay by means of training on site.

Rental of tools

European customers who require special mounting and dismounting tools or measuring equipment only infrequently can rent these from Schaeffler on a weekly basis.

Our service includes:

- rapid delivery to the installation site
- rental costs including shipment costs
- checked quality products in keeping with the latest technological developments
- delivery of the tools, including all add-on parts
- user manuals available in several languages.

If one of our qualified experts in the Industrial Service function is commissioned to carry out the particular activity, rental costs are not generally incurred.

Services

Qualification

Mounting errors are the cause of approximately 25% of all premature bearing failures. In addition to basic knowledge of rolling bearings, it is particularly important to have theoretical and practical knowledge of the correct mounting and dismounting procedures in order to achieve a long bearing operating life.

In order to ensure that the training received by mounting personnel is as close to reality as possible, Schaeffler offers individual training on mounting and dismounting processes.

Here, information on handling rolling bearings correctly and avoiding errors during mounting and dismounting is imparted by Schaeffler experts. This is carried out with direct reference to the specific application and the individual circumstances of the customer. A practical demonstration of the mounting and dismounting process is then provided, which also covers adherence to the necessary processes and specifications.

Finally, the training participants must put their acquired knowledge to the test. Only then do they receive application-specific certification from Schaeffler.

Lubrication

In more than half of all cases, inadequate lubrication is the cause of unplanned machine downtime. The life of rotating machine elements can be significantly extended by the use of greases appropriate to the different operating and environmental conditions as well as the definition of and adherence to lubrication intervals and quantities.

Services

Services relating to lubrication, *Figure 2*, include:

- selection of lubricants and lubrication systems
- mounting and commissioning of lubricators and lubrication systems CONCEPT
- preparation of lubrication and maintenance plans
- lubrication point management
- consultancy on lubricants
- lubricant investigations and tests.

Advantages

The Schaeffler lubrication service helps to:

- prevent failures of rotating components
- increase productivity
- reduce lubrication costs.

An extensive selection of high quality Arcanol rolling bearing greases is available, *Figure 2*. These greases were specially tested and selected for use in rolling bearings.

Figure 2
Supporting the lubrication service: the large grease selection

Condition Monitoring

The malfunction-free and optimised operation of complex machinery and plant can only be achieved by means of condition-based maintenance. Schaeffler uses various processes for this purpose.

Vibration measurements

The measurement of vibrations is a proven method here. Vibrations reveal damage in machinery at a very early stage.

The vibration expert can assess the condition of the machine without the need for any dismounting work. A large proportion of possible causes of damage can thus be detected and assessed with little work. As a result, damaged components can be replaced as part of planned downtime.

Depending on the type of machine and its importance for the production process, condition monitoring can be carried out by means of either continuous (online) monitoring or monitoring at regular intervals (offline).

Continuous monitoring

For production-critical machinery, continuous monitoring, *Figure 3*, page 129, by means of vibration diagnosis is indispensable in many cases. Investment in such monitoring systems often pays for itself after a few months due to the reduced failure costs. Depending on the area of application, Schaeffler offers a wide range of solutions, including single channel standalone solutions for smaller equipment and medium-sized systems with up to 16 channels that can be extended on a modular basis.

In addition to giving advice on selecting the right system, Schaeffler also implements monitoring of the machine. This includes not only hardware selection but also system configuration and, where necessary, its integration into existing systems. Schaeffler prepares condition reports at regular intervals defined in agreement with the customer. If any anomalies are found, a recommendation for action is issued.

The customer can decide whether to carry out plant monitoring himself or to enlist the services of Schaeffler for online monitoring. Due to the communication options of the monitoring systems, remote analysis can be carried out by the Schaeffler experts.

Figure 3
Continuous monitoring

Regular monitoring

The failure of so-called “B” or “C” category plant items not does lead directly to downtime and does not therefore lead to expensive secondary damage. In the case of such machine parts, regular monitoring is generally recommended as a more economical option. The Schaeffler experts can assist in identifying the most economically appropriate solution between cost-effective continuous monitoring and regular monitoring.

In this type of monitoring, machinery is examined and assessed by vibration analysis at regular intervals, for example every four weeks. This regularity gives more in-depth knowledge of the normal condition of the machine. Deviations can thus be detected.
For the monitoring concept, the selection of measurement points and monitoring accessories as well as the measurement interval play a decisive role. If deviations occur during measurement or if trends are to be investigated, the data can be sent to the Schaeffler Diagnosis Centre. Vibration experts will then analyse the data and prepare a diagnosis report. Through working with the Schaeffler experts, customers can build up their own know-how in analysis.

If no personnel are available for data logging, Schaeffler can also offer support in data logging. Experts from Schaeffler then carry out regular measurements on site.

Services

Troubleshooting

Where malfunctions occur on a machine, defects and weaknesses must be detected and rectified very quickly. Based on many years of experience with different sectors and applications, the Schaeffler diagnosis experts are well versed in such troubleshooting tasks. Various types of information are fed into the analysis. These include earlier measurement records or repair reports. If no measurement records are available, the diagnosis experts orient themselves to the specific task through observation, perusing the machine documentation and holding discussions with the machine operators.

Problems or malfunctions in machine operation often become apparent through changes in vibration behaviour, unusual temperature patterns or similar phenomena. Where the diagnosis experts carry out measurements on the machine themselves, the measurement method is selected as a function of the specific machine and the type of malfunction. The Schaeffler diagnosis experts are familiar with all analytical techniques, from vibration measurement to torque analysis or endoscopy. As a result, they can quickly identify malfunctions and prepare proposed solutions. The investigation is closed out by a handover discussion between the diagnosis experts and all relevant employees on site. In addition to the results of the investigation, the recommended countermeasures are discussed in particular. These results are then documented with the recommendations in a measurement report.

Modal analysis

Modal analysis is a particular form of vibration diagnosis. This method does not examine individual components of a machine but the machine as a whole. The aim is to determine the overall vibration behaviour of the machine. A model of the machine is created on the computer and a large number of measurement points are defined. The machine is then specifically excited to vibration using an impulse hammer. Based on parallel measurement of the excitation and resulting machine vibrations at the various measurement points, a vibration model of the machine can be determined by the computer and presented in three dimensions.

Modal analysis has a wide variety of possible applications:

- Determination of natural frequencies or resonance frequencies:
  - Due to design-related factors such as mass and rigidity, each machine has one or more natural frequencies. If the speed of the motor in a machine is within the range of a natural frequency, extreme vibrations may occur in the machine. With the aid of modal analysis, the Schaeffler vibration experts can submit recommendations for design improvements to the machine.
- Detection of the “soft spot” in a machine:
  - If high levels of vibration occur during commissioning or after a technical modification of a machine, this may be due to a so-called “soft spot”. This is defined as a rigidity problem, often caused by a poor quality connection between two machine parts, for example at a screw connection. For analysis, the measurements are used to produce an animation showing the movements of the machine. Showing the movement of the individual machine parts in relation to each other quickly leads to the “soft spot” in the machine. A joint discussion can be held to prepare proposals for improving the design of the machine.

Endoscopy

Digital, optical endoscopes can be used to examine the interior of a machine, *Figure 4*, to determine the extent of damage. The images can be stored as a digital photograph or video and form the basis of diagnosis by Schaeffler experts. The condition of individual components such as rolling bearings and gear teeth can be assessed.
If the bearings inspected are Schaeffler products, the customer also has access to the knowledge of the Schaeffler application engineers. These experts will prepare a detailed damage analysis and submit proposals for improvement.

Figure 4
Endoscopy

Thermography

Thermography is one of the most important non-destructive diagnostic techniques, *Figure 5*. Many technical problems manifest themselves in the form of heat generation, which can be detected with the aid of a high resolution infrared camera. The major advantage of thermography is the rapid, non-contact collection of temperature data. If a photograph is taken at the same time, the temperature patterns present in a machine part can be assigned on site.

Thermography can be used for assessment in relation to numerous objects, such as in the case of:

- non-contact monitoring of rolling bearings during operation
- thermal monitoring of processes.

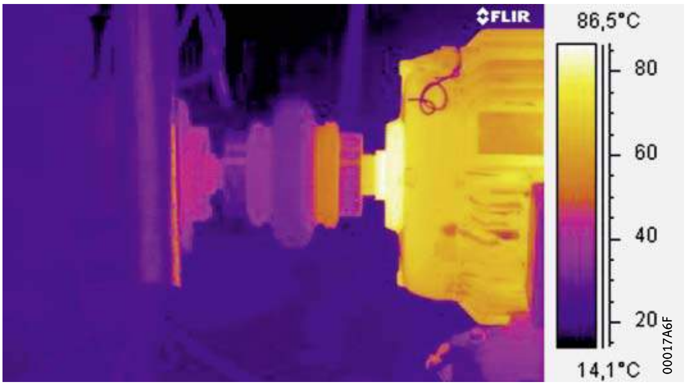


Figure 5
Thermography® FLIR systems

Approval inspection of new plant

The combination of different diagnostic techniques enables the Schaeffler experts to carry out assessment of new plant irrespective of the manufacturer. Frequently occurring installation defects can thus be detected in the initial operation phase. These defects can include: misalignment of motors, pumps or fans as well as incorrect electrical connections in switch cabinets. The Schaeffler experts check the most common problems and record the actual status. Where necessary, corrective measures can then be requested from the plant manufacturer or operator in good time. If such defects remain undetected over the period of the warranty, their removal and the secondary damage may incur considerable costs and downtime.

Tools and measuring devices

High-quality tools and measuring devices facilitate high quality of work. Our tools and measuring devices can be repaired in many cases. Regular maintenance and calibration leads to an extended service life and ensures good measurement results.

Repair

If damage occurs, the Schaeffler Repair Service offers rapid and reliable assistance.

The advantages include:

- repair by qualified skilled personnel
- use of original replacement parts
- safety inspection after any repair.

Maintenance and calibration

A mounting defect will often cause a major reduction in service life. If measuring devices are used in the mounting of rolling bearings, the service life of the bearing is directly dependent on the measurement accuracy.

Sensitive measuring devices such as enveloping circle gauges or alignment devices, as well as some tools, must undergo maintenance and calibration at certain intervals. This ensures consistently high quality of operating equipment and also extends the service life. The devices are checked by service experts from Schaeffler, repaired where necessary, subjected to maintenance and then recalibrated. Their suitability for problem-free use is confirmed by means of a certificate. Where necessary and if available, a suitable loan device can be provided.

Corrective maintenance

Once a machine problem has been diagnosed, it should be eliminated as quickly as possible. Two of the most frequent problems, namely unbalance in pumps and fans as well as misalignment of machine components to each other, can be corrected directly by the Schaeffler experts.

Balancing

Unbalance is one of the most frequent defects that lead to unexpected failure of rotating machine elements. Correct balancing gives a decisive increase in the life of rotating machine parts. This increases the productivity and availability of the machine. The Schaeffler experts reduce to a normal level any unbalance that occurs, for example, as a result of contamination, wear and repairs. They can detect and eliminate the causes on machinery operating at a speed of 40 min^{-1} to 10 000 min^{-1}. Typical examples of such machines include pumps, ventilators, compressors, turbines and motors. Schaeffler offers not only a detailed analysis of the causes of the problem but also the elimination of unbalance.

Alignment

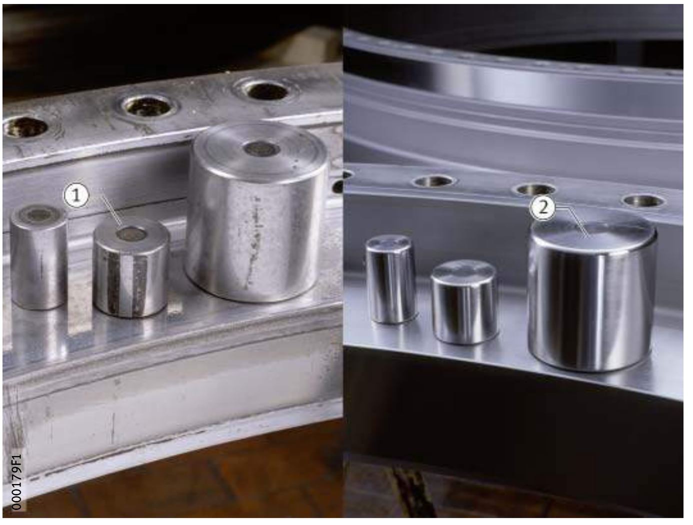
Many machines consist of different components, such as electric motors and pumps. After installation, repair or overhaul, the components of such machines must be aligned with each other, *Figure 6*. If this is carried out incorrectly or not at all, this results in high loads being placed on the bearings as well as increased energy demand and wear.

In addition to laser alignment systems, Schaeffler offers alignment of machinery as a service. Where necessary, the service technician from Schaeffler will take the necessary laser alignment system to the customer and carry out alignment of the machine in accordance with the manufacturer’s specifications. The work is then documented.

Figure 6
Alignment

Rolling bearing reconditioning

It is often the case that new rolling bearings are fitted although the existing bearings could be restored to as-new condition by means of appropriate reconditioning. In many cases, reconditioning of rolling bearings is significantly more cost-effective than using new bearings, *Figure 7*.



① Before reconditioning
② After reconditioning

Figure 7
Rolling bearing raceway and rollers before and after reconditioning

Advantages

Reconditioning is carried out irrespective of manufacturer and is thus not restricted to products from Schaeffler Technologies. Before reconditioning, the condition of the bearings can be assessed on site in consultation with experts from the Global Technology Network.

The advantages for the customer are:

- reductions in life cycle costs (LCC = Life Cycle Costs)
- increases in operating life
- savings in material and energy costs
- reductions in inventory costs
- high flexibility through short lead times
- feedback on the characteristics and frequencies of damage.

Quality

Schaeffler performs reconditioning of rolling bearings to uniform standards throughout the world. All sites apply identical processes and guidelines. Schaeffler rolling bearings are processed in accordance with the original drawings. In the case of all bearings, work is carried out using only original components and original replacement parts. High quality reconditioning is achieved as a result of our comprehensive knowledge of rolling bearings.

Market sectors Reconditioning of rolling bearings is of particular interest if these are used in machinery or vehicles in the following market sectors:

- raw material extraction and processing
- metal extraction and processing
- pulp and paper
- railways.

Dimensions Reconditioning and, where required, modification can be carried out on rolling bearings with an outside diameter D of 100 mm to 4 500 mm. Please contact us for information on reconditioning or modifying bearings with other outside diameters.

Overview The operations necessary in reconditioning are dependent on the condition of the rolling bearing. In order to allow a reliable statement of the work required, the rolling bearing must be disassembled, cleaned and then carefully examined.

Beyond this requalifying process (Level I), which is always necessary, further reconditioning steps may be appropriate, *Figure 8*.

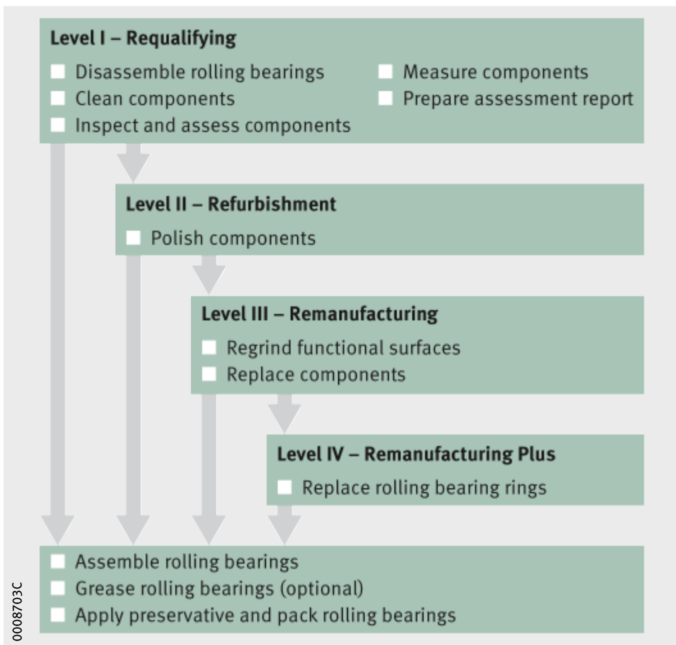


Figure 8
Level I to Level IV

Services

Worldwide reconditioning

Schaeffler offers the reconditioning of rolling bearings at several locations worldwide, *Figure 9*.

Figure 9
Reconditioning locations

Technical consultancy

Companies that wish to change to the concept of condition-based maintenance are supported by Schaeffler with training, attendance during the introductory phase, ongoing advice during the period of use and service contracts.

Condition-oriented maintenance

In condition-oriented maintenance, machinery and plant no longer undergo maintenance work on the basis of failures or times but on their assessed condition. In partnership with the customer, the advisory experts of Schaeffler prepare plans that give recommendations for action on the basis of the results of condition monitoring. These recommendations result in targeted maintenance measures and thus to reduced costs.

Service concepts for plant manufacturers and operators

Services are not "off the peg" products and the requirements vary according to the machine and the knowledge on site. Due to the wide portfolio of Schaeffler services, local certified Schaeffler employees can weave together the right package of Schaeffler training and services together with in-house activities. The scope is enormous and dependent on the prior knowledge and usable work capability as well as the requirements for quality of monitoring.

The following four examples show the extent of the scope and how widely service concepts can vary. Starting from the customer requirements, the Schaeffler experts prepare a concept to meet the needs and assist in its implementation.

Example 1:
Instruction

Company A has employees with experience in the field of condition diagnosis.

In this case, it is sufficient to instruct the employees of the company in the use of the systems and accompany them while making the initial steps. In difficult cases, assistance can be requested from the Schaeffler experts. These will help in the analysis and formulation of measures.

Example 2:
Teaching

Company B would like to build up its own knowledge in the field of condition diagnosis.

Condition diagnosis is a complex subject. Building up knowledge therefore takes time. In such cases, Schaeffler offers a two-year programme, after which even customers without prior knowledge can themselves monitor the condition of their machinery.
The support given by the Schaeffler experts is progressively reduced in stages and the customer’s employees use their new knowledge directly in their daily work.

Example 3:
Outsourcing

Company C would like to completely outsource the area of condition diagnosis.

Schaeffler offers complete packages under which the entire service is provided by Schaeffler. This begins with the initial operation of systems and progresses through continuous monitoring to complete leasing of the hardware, such that no initial costs are incurred by the customer. Such monitoring packages are very attractive, for example, to operators of wind turbines.

Example 4:
Offering a service

Company D is a plant manufacturer and would like to offer condition monitoring as a service itself.

In this case, Schaeffler acts as a subcontractor for the mostly portable monitoring systems, as a trainer to the service employees of the plant manufacturer and as an expert team. The expert team always moves into action when machines show sets of characteristics that cannot be clearly assessed by the employees of the plant manufacturer. The plant manufacturer can thus offer its customers a highly qualified monitoring service without having to establish its own experts.

TCO approach

The Schaeffler approach to reducing total costs (TCO, Total Cost of Ownership) is a practically based concept for plant operators (MRO) and plant manufacturers (OEM). The aim is to reduce operating costs associated with rolling bearings and increase plant availability on a continuing basis. The costs incurred are objectively and measurably compared with the benefits to be accrued by means of appropriate parameters.

In the case of MRO, Schaeffler works as a partner with employees from production and maintenance facilities to optimise and conserve the value of plant. With the user, Schaeffler identifies specifically named problems and submits technical approaches to solutions. Furthermore, Schaeffler provides proposals for cost reductions that are based on experience within specific sectors and spanning different sectors. This analysis covers plant components, operations and qualification measures as well as their practical implementation.

Schaeffler is also interested in advising OEMs on product design with favourable maintenance-related characteristics, in order to achieve a significant contribution to reductions in operating costs as early as the plant design phase.

Training courses

Training courses

Rolling bearing technology
Training example

TRAINING-BEARING-BASIC-TECH-2

Mounting of rolling bearings
Training example

TRAINING-BEARING-MOUNTING-PRACT

Machinery monitoring
Training example

TRAINING-CM-D3-BASIC

Training courses

Features

Appreciating the use of rolling bearings, linear guidance systems and plain bearings as indispensable elements in thousands of applications requires the necessary understanding of these machine elements. Schaeffler Technologies offers numerous theory and practical training courses for its extensive product portfolio. Starting from product knowledge, it is possible to become familiar with subject areas such as mounting and dismounting of rolling bearings using the optimum tools and the condition monitoring of bearing arrangements, especially through the use of noise, vibration and torque measurements.

Systematic learning processes, in conjunction with the appropriate methods, help the training participant to discover the world of bearing arrangements. With the support of machine building engineers and technicians who have undergone didactic and methodological training, there is no longer any obstacle to effective learning progress.

Target groups

Our training courses cover the information needs of employees in a very wide variety of areas of activity within a company. In this way, the technically oriented employee, whether a designer, fitter or maintenance person can find the right training course, in the same way as the employee in business administration, for example in purchasing.

Basic training on rolling bearings, plain bearings and linear guidance systems

All target groups are considered using a uniform training approach. In general, the initial steps are provided by basic training covering the different characteristics, features and types of rolling bearings, plain bearings and linear guidance systems as well as their combination to form systems, extending all the way to mechatronic units. Application examples reflect the selection criteria and the customer benefits achieved.

Rolling bearing theory

These product-oriented training courses are followed by modules covering rolling bearing theory as well as selected applications. Rolling bearing theory conveys the necessary knowledge on subjects such as bearing clearance, load distribution in the bearing, rating life and lubrication.

Workshops on applications

In workshops, the participants consider applications from practice, such as the bearing arrangements in a machine tool or a shaft bearing arrangement. All process steps are covered, from bearing selection and bearing calculation through to mounting. We also offer workshops in the field of mechatronics (principles and products).

Training courses

Conditioning monitoring by means of vibration analysis

Furthermore, we offer a certification training course with examination in accordance with DIN ISO 18436-2. Knowledge of condition monitoring by means of vibration analysis is certified by an independent body that is accredited by the DAkkS. This certificate is recognised at an international level.

Mounting and dismounting

Numerous training modules cover the mounting and dismounting of rolling bearings and linear guidance systems. Based on perception and exercises, the participant gains the mounting knowledge and skills that he will require in practice. Our training courses on mounting cover a large number of applications. Mounting exercises on individual products are followed by work on more complex systems such as gearboxes or wheelsets.

Vibration analysis and measuring systems

The possibilities for plannable and economical design of maintenance work on machines, plant and rolling bearings are communicated to the training participant in appropriate courses. Our trainers communicate the theoretical principles of vibration analysis, the practical use of measurement systems and the handling of configuration and analysis software. The theoretical knowledge acquired is consolidated by means of practical exercises.

Teachers and trainers

For the further training of teachers and trainers, Schaeffler Technologies offers a basic course and an advanced course each once per year. The three-day basic course includes product training and the principles of rating life calculation. A one-day mounting training course can be booked to follow on directly. The three-day advanced course covers the rolling bearing design of a gearbox shaft according to the catalogue and Schaeffler calculation software such as ***medias*** interchange and BEARINX-online.

Training locations

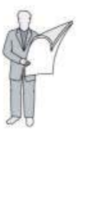
Schaeffler has its own training centres worldwide. Qualified presenters with considerable experience ensure practice-oriented knowledge transfer in various languages at local sites. Alternatively, we will be pleased to train your employees at your location.

Training centres worldwide

The Schaeffler Technology Centers – Training (training centres with headquarters in Eltmann, Germany) offer theoretical and practical training courses in their modern training facilities. All product and services portfolios of Schaeffler Technologies are covered in detail. Starting from the principles and progressing to more detailed special knowledge, training courses communicate knowledge of rolling bearing theory, mounting and dismounting as well as all levels of condition monitoring and mechatronics.

Quality assurance

Through continuous market monitoring and exchange of experience, Schaeffler is in a position to continuously improve its training courses. What is particularly important to us is the ideas and suggestions that we receive through feedback from our training participants. Certification of the Schaeffler Technical Training Centres in accordance with ISO 9001:2008 underlines our continuous aspiration to increasing quality. After training, each training course attendee receives a certificate.



Training courses

Training courses, standard and individual

The standard programme of training courses is sufficient in most cases to achieve acquisition of the knowledge necessary for day-to-day work. Upon customer request, Schaeffler also offers individually tailored training courses. In these cases, customers can themselves define the key components of the content. The standard training programme for the areas of rolling bearing technology and mounting as well as condition monitoring is already comprehensive; an excerpt is shown in the table.

Excerpt from the Schaeffler training portfolio

Training courses	
Rolling bearing technology and mounting	Basic training: Rolling bearing technology
	Mounting of rotary bearings
	Basic training: Mounting of rolling bearings (in gearboxes)
	Practical training: Mounting of rolling bearings (using large rolling bearings)
	Linear – Products and applications
	Mounting of linear bearings
	Rolling bearing failures: Identifying causes – Optimising operation
	Mounting and maintenance of rolling bearings for rail vehicle maintenance personnel (customer location)
	Maintenance of spindle bearings
Condition Monitoring	Detector III: Entry level, design, machine diagnosis, balancing
	SmartCheck: Introduction, design
	Vibration condition monitoring Category 1 and 2 in accordance with DIN ISO 18436-2 with certification
	DTECT $X1_s$
	ProCheck
	Administrator 4
Mechatronics	Basic training: Mechatronics
Special training	Training completely oriented to the application

Training material

Literature on the correct mounting of bearings is readily available, however there is a general lack of appropriate equipment on which apprentices can practise in as practical a sense as possible. As a result, trainers from the Schaeffler training workshops have developed a mounting cabinet and a mounting cross for practising the mounting and dismounting of rolling bearings.

Mounting cabinet

The mounting cabinet is used in the basic course. The aim of this rolling bearing course is to communicate knowledge on the selection of the correct bearing, correct mounting and dismounting and the maintenance of bearing positions. Material from the mounting cabinet is, however, also used to provide instruction on individual contents with the aid of various mounting sets, *Figure 1*.

Figure 1
Basic course:
Mounting cabinet

Mounting cross

In order to provide professional training courses on the correct mounting and dismounting of rolling bearings, Schaeffler has developed the so-called mounting cross, *Figure 2*. This piece of equipment allows the expert to communicate the correct handling procedure visually, using a variety of different bearings, and under realistic conditions.

Figure 2
Mounting cross

Ordering designation	Title
Mounting	
FAG	Flyer: FAG Heating Devices
FIM	Flyer: FAG Induction Units with Medium Frequency Technology
TPI 156	Tapered Roller Bearing Units TAROL – Mounting, Maintenance, Repair
TPI 180	FAG Tools for Thermal Dismounting
TPI 195	FAG Pressure Generation Devices
TPI 196	FAG Hydraulic Nuts
TPI 200	FAG Heating Devices for Mounting of Rolling Bearings
TPI 216	Tools for Mechanical Mounting and Dismounting of Rolling Bearings
TPI 217	Induction Units with Medium Frequency Technology
MH 1	Mounting Handbook – Mounting of Rotary Bearings
WL 80 112	Mounting and Dismounting of Rolling Bearings (Tools, Devices, Methods)
Lubrication	
OFC	Flyer: FAG CONCEPT2
OFG	Flyer: FAG CONCEPT Precision Grease
OFO	Flyer: FAG CONCEPT Precision Oil
TPI 168	Rolling Bearing Greases Arcanol
TPI 176	Lubrication of Rolling Bearings
TPI 252	Automatic Relubrication Devices
WL 80382	Flyer: FAG CONCEPT8
Training	
WL 80 386	Certified Vibration Expert to DIN ISO 18436-2

Ordering designation	Title
Condition Monitoring	
FQB	Flyer: FAG SmartQB
SI SK001	Flyer: FAG SmartCheck Service Kit
TPI 170	FAG DTECT X1$_S$ – Continuous Monitoring of Machinery and Equipment
TPI 182	FAG Alignment Tools – Top-Laser: SMARTY2 · TRUMMY2 · EQUILIGN · SHIM
TPI 214	FAG SmartCheck
TPI 216	Tools for Mechanical Mounting and Dismounting of Rolling Bearings
TPI 217	Induction Units with Medium Frequency Technology
TPI 234	Condition Monitoring of Greases in Rolling Bearings
TPI 252	Automatic Relubrication Devices
TPI WL 80-64	FAG Detector III – The Solution for Monitoring and Balancing
TPI WL 80-69	FAG ProCheck – State of the Art Machinery Monitoring for Maximum Availability
WL 80 362	Flyer: FAG ProCheck
WL 80 366	Flyer: FAG Wear Debris Check
WL 80 373	Flyer: FAG WiPro$_S$
WL 80 375	Flyer: FAG SmartCheck Starter Kit
WL 80 378	Flyer: FAG Top-Laser EQUILIGN
WL 80 380	Flyer: FAG GreaseCheck
Rolling Bearing Reconditioning	
TPI 207	Reconditioning and Repair of Rolling Bearings

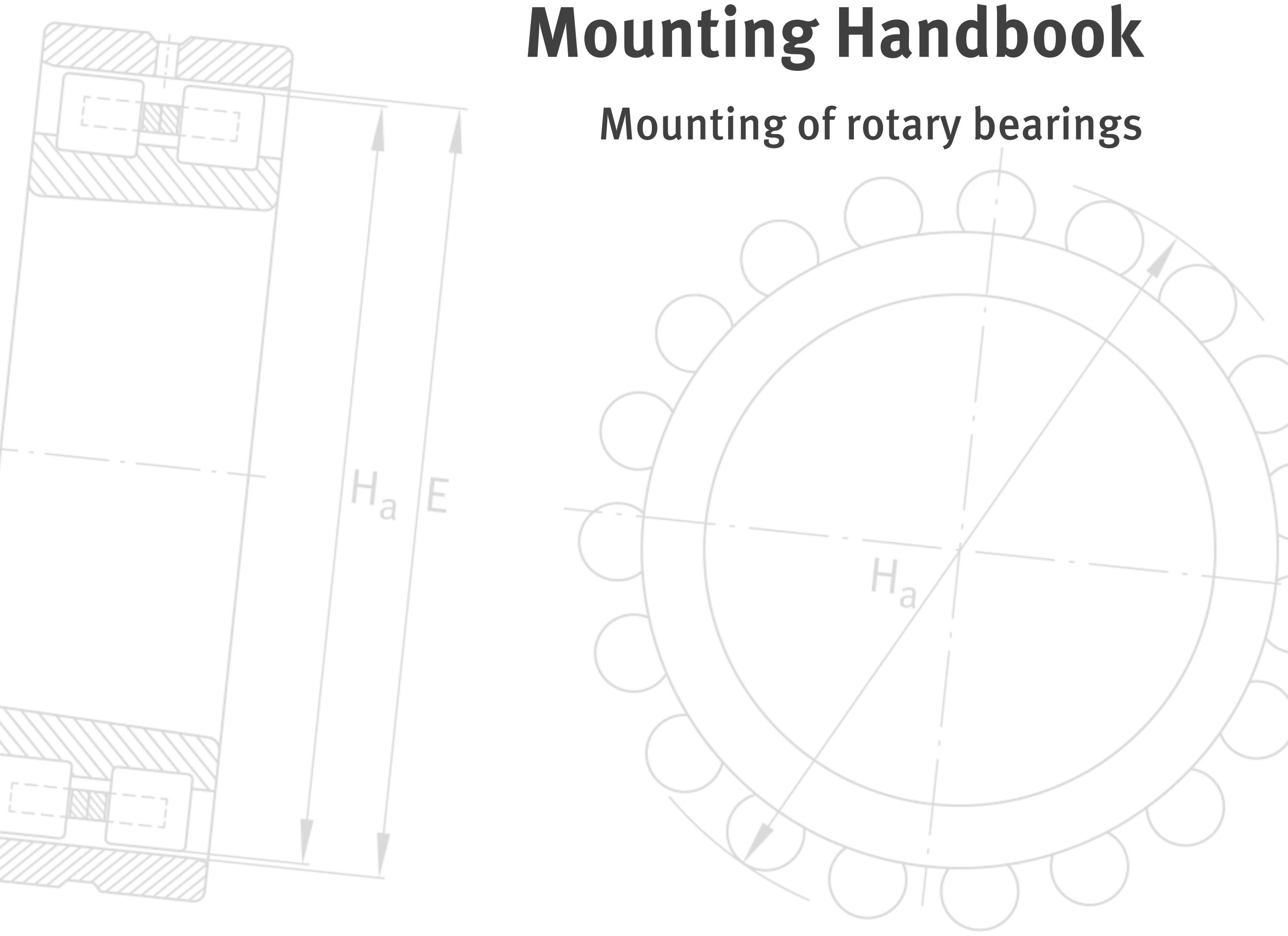
SCHAEFFLER

Mounting Handbook

Mounting of rotary bearings

Mounting Handbook

Mounting of rotary bearings



Foreword

Schaeffler is a leading worldwide supplier of rolling bearings, accessories specific to bearings and comprehensive maintenance products and services. Schaeffler has approximately 100 000 catalogue products manufactured as standard, providing an extremely wide portfolio that gives secure coverage of applications from all existing industrial market sectors.

Catalogue MH 1, Mounting Handbook

Rolling bearings are high quality goods and must therefore be handled with care. The use of suitable equipment as well as care and cleanliness in mounting and dismounting make a significant contribution to increasing the availability and lifetime of rolling bearings. With the aid of the diverse portfolio of products and services, operating life and performance capability of production plant can be increased and overall costs can be reduced.

This Catalogue MH 1 gives important guidelines on the correct handling of rotary bearing arrangements in mounting, dismounting and maintenance. Further information on the various bearing types, tools and methods can be found in the corresponding product brochures. If you have any other questions on the subject of bearing arrangements, the employees of Schaeffler worldwide will be pleased to assist you.

Catalogue HR 1, Rolling Bearings

Catalogue HR 1 describes the rolling bearings in accordance with DIN ISO that are required for original equipment manufacture, distribution and the aftermarket, specific rolling bearing accessories and further rolling bearing types and design variants.

It shows which products can be considered for a bearing arrangement, the factors that must be taken into consideration in the design, the tolerances required on the adjacent construction and how the bearing arrangement is sealed. It gives detailed information on the calculation of bearing rating life, on temperatures and loads, on the lubricants that are most suitable for the bearing arrangement and, last but not least, on the correct mounting and maintenance of the products.

Catalogue IS 1, Mounting and Maintenance

Catalogue IS1 is aimed principally at maintenance managers and operators of plant in which rolling bearings and other rotating machine components play a critical role in detemining the quality of products and processes. Those responsible for maintenance and production processes must be able to rely every day on the quality of their tools and the expertise of their service providers.

This catalogue gives an overview of the portfolio:

- mounting
- lubrication
- condition monitoring
- services.

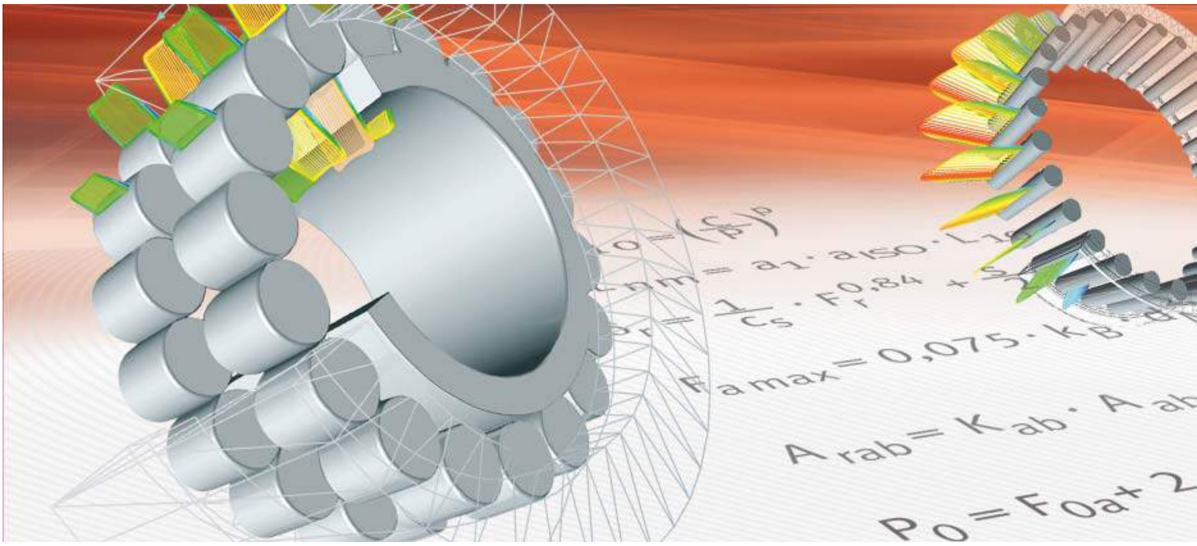
Foreword

Global Technology Network

Schaeffler offers its diverse portfolio of products and services worldwide. In the Global Technology Network, Schaeffler combines its local competence in the regions with the knowledge and innovative strength of its experts worldwide under one philosophy. With our local centres of competence under the name “Schaeffler Technology Center”, we bring our portfolio of services and our engineering and service expertise directly to your area. Through this combination, you will experience optimum support anywhere in the world and, thanks to our bundled knowledge, innovative and customised solutions of the highest quality. This makes it possible to achieve sustainable reductions in the overall costs of your machinery and plant and thus improvements in efficiency and competitiveness.

Figure 1
Our portfolio

Contents



Technical principles

Bearing types
Bearing arrangements
Fits
Internal clearance and operating clearance
Geometrical and positional tolerances
Safety guidelines
Preparations for mounting and dismounting
Dimensional and geometrical inspection
Lubrication
Storage of rolling bearings
Seals
Bearing housings

Technical principles

Technical principles

Bearing types

Rolling bearings

The task (function) of rotary rolling bearings is to guide parts that are movable in relation to each other and support them relative to the adjacent structure. They support forces and transmit these into the adjacent construction. In this way, they perform support and guidance tasks and thus form the connection between stationary and moving machine elements.

The function "Support" comprises the transmission of forces and moments between parts moving relative to each other.

The function "Guidance" principally comprises defining to an appropriate (normally high) accuracy the position of parts moving relative to each other.

Principal requirements placed on bearings

Technical implementation is oriented to the two principal requirements:

- Function must be ensured and fulfilled for as along as possible.
- The resistance to motion (bearing friction) should be as low as possible in order to reduce the energy required for motion (energy efficiency).

Bearing types

Rolling bearing types

An overview of typical bearing types for rotary motion is shown in the following diagram, *Figure 1*.

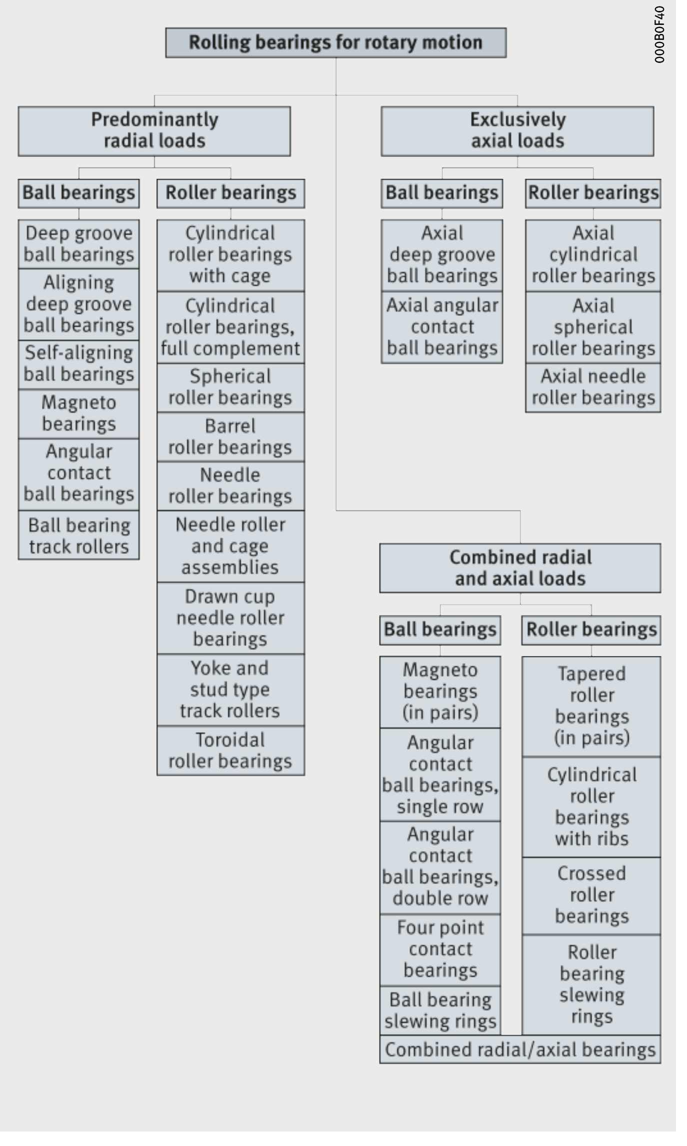


Figure 1
Overview
of rolling bearing types

Bearing arrangements

The guidance and support of a rotating shaft requires at least two bearings arranged at a certain distance from each other. Depending on the application, a decision is made between a locating/non-locating bearing arrangement, an adjusted bearing arrangement and a floating bearing arrangement.

Locating/non-locating bearing arrangement

On a shaft supported by two radial bearings, the distances between the bearing seats on the shaft and in the housing frequently do not coincide as a result of manufacturing tolerances. The distances may also change as a result of temperature increases during operation. These differences in distance are compensated in the non-locating bearing. Examples of locating/non-locating bearing arrangements are shown in *Figure 1* to *Figure 7*, page 14.

Non-locating bearings

Ideal non-locating bearings are cylindrical roller bearings with cage N and NU or needle roller bearings, *Figure 1* ②, ④. In these bearings, the roller and cage assembly can be displaced on the raceway of the bearing ring without ribs.

All other bearing types, for example deep groove ball bearings and spherical roller bearings, can only act as non-locating bearings if one bearing ring has a fit that allows displacement, *Figure 2*. The bearing ring subjected to point load therefore has a loose fit; this is normally the outer ring, see page 20.

① Deep groove ball bearing as locating bearing and cylindrical roller bearing NU as non-locating bearing

② Axial angular contact ball bearing ZKLN as locating bearing and needle roller bearing NKIS as non-locating bearing

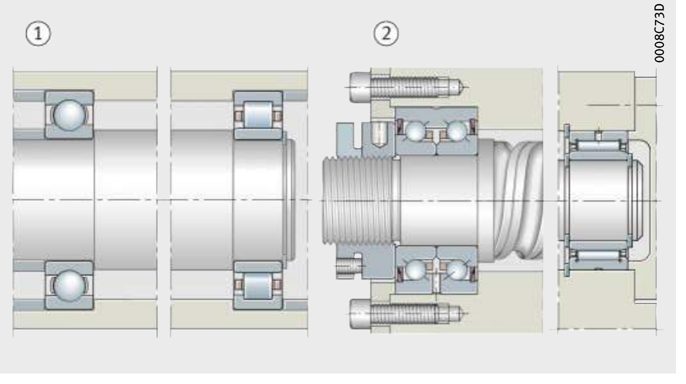


Figure 1
Locating/non-locating bearing arrangements

① Deep groove ball bearings as locating and non-locating bearings

② Spherical roller bearings as locating and non-locating bearings

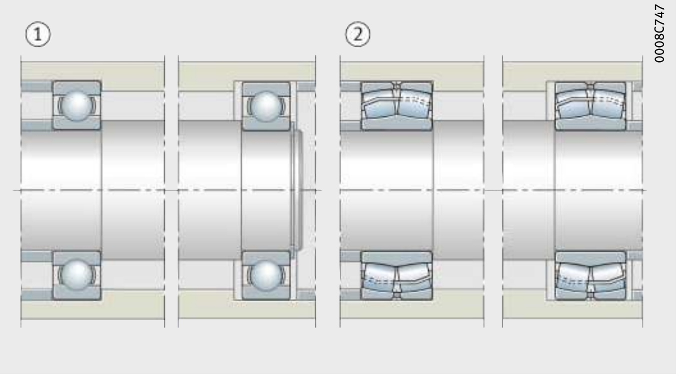


Figure 2
Locating/non-locating bearing arrangements

Locating bearings

The locating bearing guides the shaft in an axial direction and supports external axial forces. In order to prevent axial preload, shafts with more than two bearings have only one locating bearing. The type of bearing selected as a locating bearing depends on the magnitude of the axial forces and the accuracy with which the shafts must be axially guided.

A double row angular contact ball bearing, *Figure 3* ①, for example, will give closer axial guidance than a deep groove ball bearing or a spherical roller bearing. A pair of symmetrically arranged angular contact ball bearings or tapered roller bearings, *Figure 4*, used as locating bearings will provide extremely close axial guidance.

In gearboxes, a four point contact bearing is sometimes fitted directly adjacent to a cylindrical roller bearing to give a locating bearing arrangement, *Figure 3* ②. The four point contact bearing, without radial support by the outer ring, can only support axial forces. The radial force is supported by the cylindrical roller bearing.

① Double row angular contact ball bearing as locating bearing and cylindrical roller bearing NU as non-locating bearing

② Four point contact bearing and cylindrical roller bearing as locating bearing and cylindrical roller bearing NU as non-locating bearing

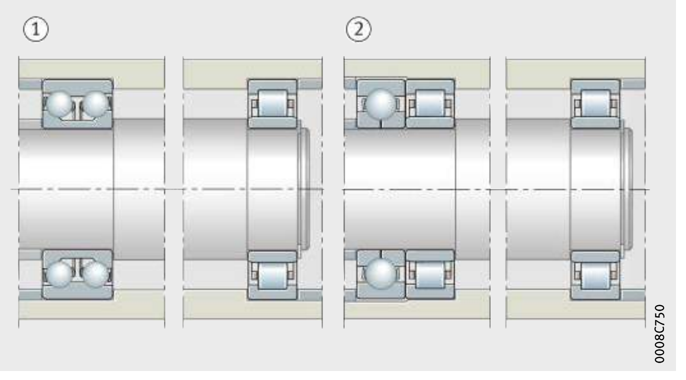


Figure 3
Locating/non-locating bearing arrangements

If a lower axial force is present, a cylindrical roller bearing with cage NUP can also be used as a locating bearing, *Figure 4* ②.

① Two tapered roller bearings as locating bearing and cylindrical roller bearing NU as non-locating bearing

② Cylindrical roller bearing NUP as locating bearing and cylindrical roller bearing NU as non-locating bearing

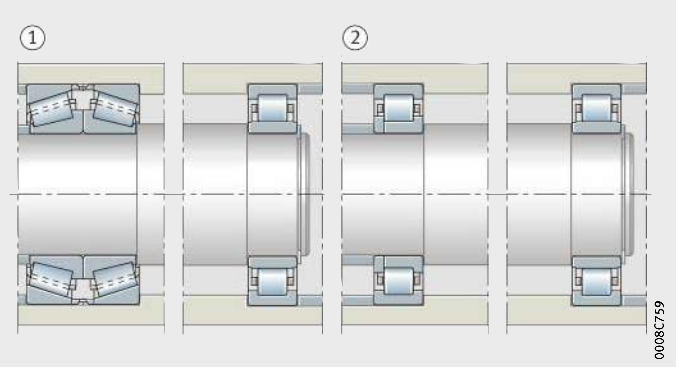
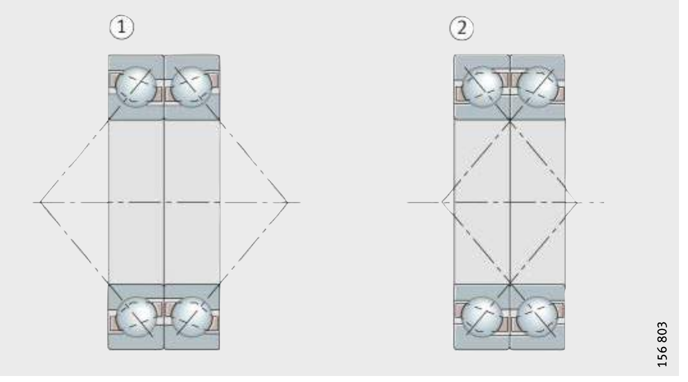


Figure 4
Locating/non-locating bearing arrangements

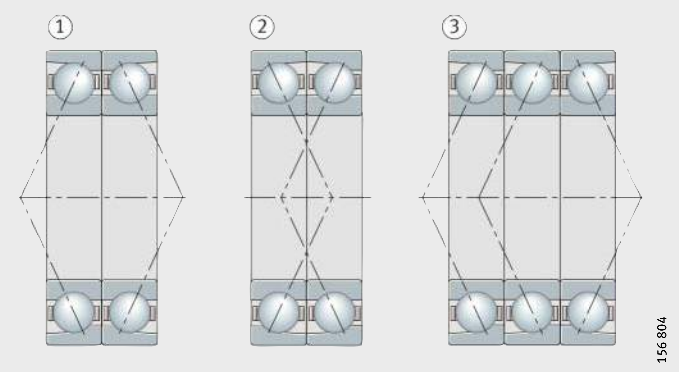
There are particular advantages in using angular contact ball bearings of the universal design, *Figure 5*. The bearings can be fitted in pairs in any O or X arrangement without shims. Angular contact ball bearings of the universal design are matched such that, in an X or O arrangement, they have a low axial internal clearance (design UA), zero clearance (UO) or slight preload (UL).



Pair of angular contact ball bearings of universal design
1 O arrangement
2 X arrangement

Figure 5
Locating bearing arrangements

Spindle bearings of the universal design UL, *Figure 6*, have slight preload when mounted in an X or O arrangement (designs with higher preload are available by agreement).

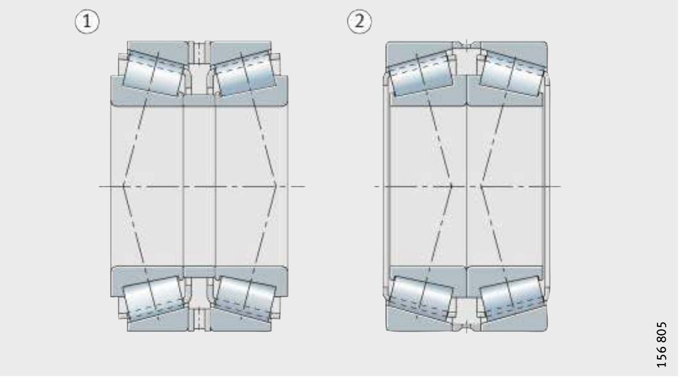


Spindle bearings of universal design
1 O arrangement
2 X arrangement
3 Tandem O arrangement

Figure 6
Locating bearing arrangements

No adjustment or setting work with matched pairs of tapered roller bearings

Mounting is also made easier with a matched pair of tapered roller bearings as a locating bearing (313..-N11CA), *Figure 7* ②. They are matched with appropriate axial internal clearance so that no adjustment or setting work is required.



Pair of tapered roller bearings
① O arrangement
② X arrangement

Figure 7
Locating bearing arrangements

Adjusted bearing arrangement

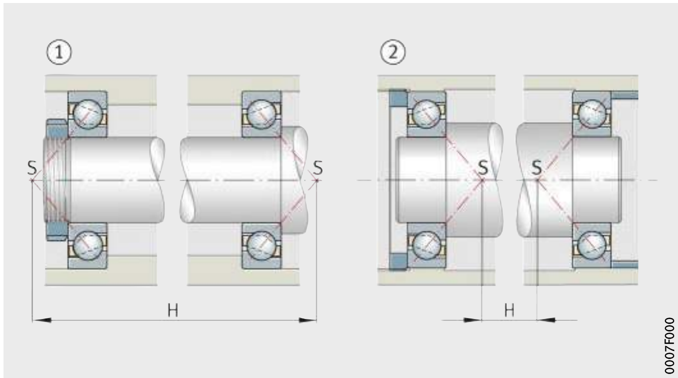
These bearing arrangements normally consist of two symmetrically arranged angular contact ball bearings or tapered roller bearings, *Figure 8*, page 15. During mounting, one bearing ring is displaced on its seat until the bearing arrangement achieves the required clearance or the necessary preload.

Area of application

Due to this adjustment facility, the adjusted bearing arrangement is particularly suitable where close guidance is required, for example in pinion bearing arrangements with spiral toothed bevel gears, in spindle bearing arrangements in machine tools or within the rotor bearing arrangement of a wind turbine.

X and O arrangement

A fundamental distinction is drawn between the O arrangement, *Figure 8* ①, and the X arrangement, *Figure 8* ②, of the bearings. In the O arrangement, the cones and their apexes S formed by the pressure lines point outwards; in the X arrangement, the cones point inwards. The support distance H, in other words the distance between the apexes of the contact cones, is larger in the O arrangement than in the X arrangement. The O arrangement therefore gives the lower tilting clearance.



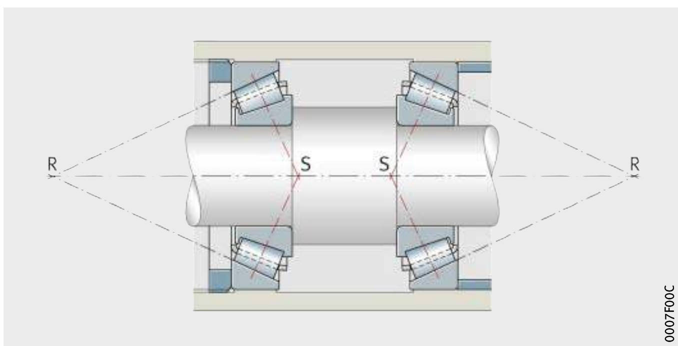
S = apexes of the contact cones
H = support distance

Angular contact ball bearings
① O arrangement
② X arrangement

Figure 8
Adjusted bearing arrangement

Influence of thermal expansion in X and O arrangements

When setting the axial internal clearance, thermal expansion must be taken into consideration. In the X arrangement, *Figure 9*, a temperature differential between the shaft and housing always leads to a reduction in the internal clearance (assuming the following preconditions: shaft and housing of identical material, inner ring and complete shaft at identical temperature, outer ring and complete housing at identical temperature).



S = apexes of the contact cones
R = roller cone apexes

Figure 9
Adjusted tapered roller bearings in X arrangement

In the O arrangement, a distinction is drawn between three cases:

- The roller cone apexes R, i.e. the intersection points of the extended outer ring raceway with the bearing axis, coincide: the internal clearance set is maintained, *Figure 10* ①.
- The roller cone apexes R overlap and there is a short distance between the bearings: the axial internal clearance is reduced, *Figure 10* ②.
- The roller cone apexes R do not meet and there is a large distance between the bearings: the axial internal clearance is increased, *Figure 11*.

S = apexes of the contact cones
R = roller cone apexes

① Intersection points coincide
② Intersection points overlap

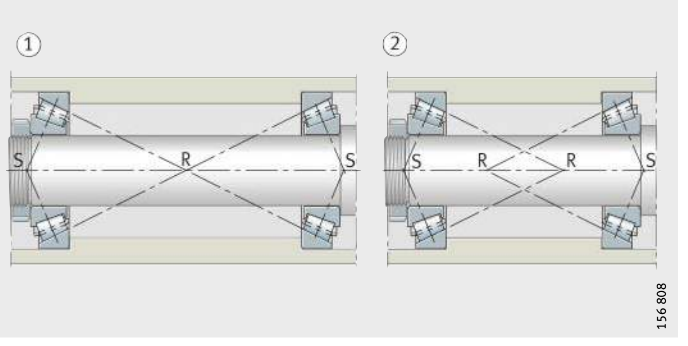


Figure 10
Adjusted tapered roller bearings in O arrangement

S = apexes of the contact cones
R = roller cone apexes

Intersection points do not overlap

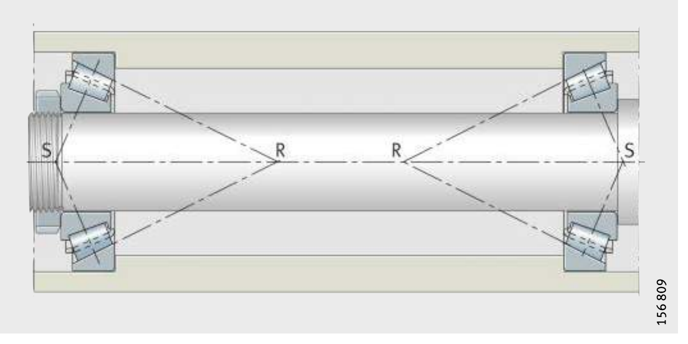
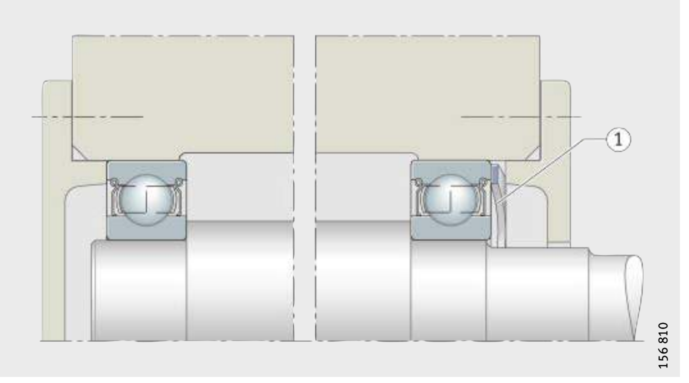


Figure 11
Adjusted tapered roller bearings in O arrangement

Elastic adjustment

Adjusted bearing arrangements can also be achieved by preloading using springs, *Figure 12* ①. This elastic adjustment method compensates for thermal expansion. It can also be used where bearing arrangements are at risk of vibration while stationary.



Deep groove ball bearing preloaded by means of spring washer
① Spring washer

Figure 12
Adjusted bearing arrangement with spring washer

Floating bearing arrangement

The floating bearing arrangement is an economical solution where close axial guidance of the shaft is not required, *Figure 13*. Its construction is similar to that of the adjusted bearing arrangement.

In the floating bearing arrangement, however, the shaft can be displaced in relation to the housing to the extent of the axial clearance s. The value s is defined as a function of the required guidance accuracy such that the bearings are not axially stressed even under unfavourable thermal conditions.

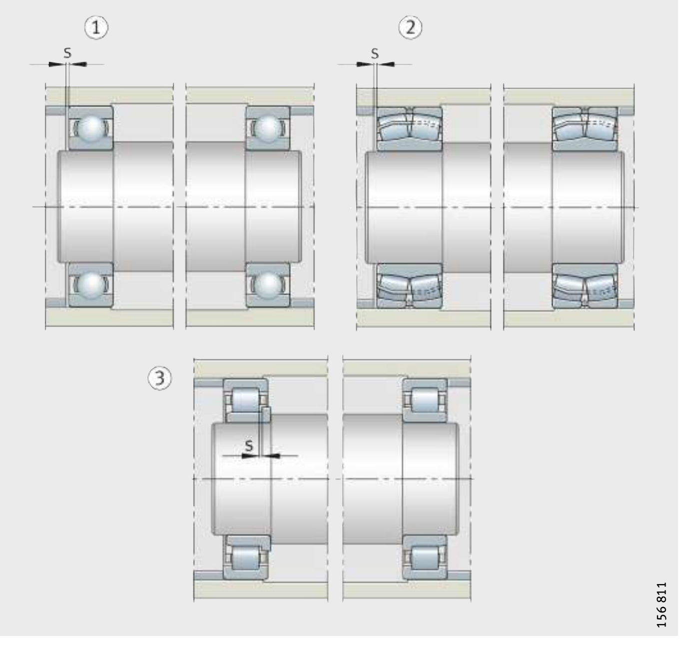
Suitable bearings

Suitable bearing types for the floating bearing arrangement include deep groove ball bearings, self-aligning ball bearings and spherical roller bearings.

In both bearings, one ring, usually an outer ring, has a fit that allows displacement.

In floating bearing arrangements and cylindrical roller bearings with cage NJ, the length compensation takes place within the bearings. The inner and outer rings can have tight fits, *Figure 13* ③.

Tapered roller bearings and angular contact ball bearings are not suitable for a floating bearing arrangement, since they must be adjusted in order to run correctly.



s = axial clearance

① Two deep groove ball bearings
② Two spherical roller bearings
③ Two cylindrical roller bearings NJ

Figure 13
Floating bearing arrangements

Fits

Criteria for selection of fits

Rolling bearings are located on the shaft and in the housing in a radial, axial and tangential direction in accordance with their function. Radial and tangential location is normally achieved by force locking, i.e. by tight fits on the bearing rings. Axial location of the bearings is normally achieved by form fit.

The following must be taken into consideration in the selection of fits:

- The bearing rings must be well supported on their circumference in order to allow full utilisation of the load carrying capacity of the bearing.
- The bearings must not creep on their mating parts, otherwise the seats will be damaged.
- One ring of the non-locating bearing must adapt to changes in the length of the shaft and housing and must therefore be capable of axial displacement.
- The bearings must be easy to mount and dismount.

Good support of the bearing rings on their circumference requires a tight fit. The requirement that rings must not creep on their mating parts also requires firm seating. If non-separable bearings must be mounted and dismounted, a tight fit can only be achieved for one bearing ring.

In cylindrical roller bearings N, NU and needle roller bearings, both rings can have tight fits, since the length compensation takes place within the bearing and since the rings can be fitted separately.

As a result of tight fits and a temperature differential between the inner and outer ring, the radial internal clearance of the bearing is reduced. This must be taken into consideration when selecting the internal clearance.

If materials other than cast iron or steel are used for the adjacent construction, the modulus of elasticity and the differing coefficients of thermal expansion of the materials must also be taken into consideration to achieve rigid seating.

For aluminium housings, thin-walled housings and hollow shafts, a closer fit should be selected if necessary in order to achieve the same force locking as with cast iron, steel or solid shafts.

Higher loads, especially shocks, require a fit with larger interference and narrower geometrical tolerances.

Seats for axial bearings

Axial bearings, which support axial loads only, must not be guided radially (with the exception of axial cylindrical roller bearings which have a degree of freedom in the radial direction due to flat raceways). This degree of freedom is not present in the case of groove-shaped raceways and must be achieved by a loose fit for the stationary washer. A rigid seat is normally selected for the rotating washer.

Where axial bearings also support radial forces, such as in axial spherical roller bearings, fits should be selected in the same way as for radial bearings.

The contact surfaces of the mating parts must be perpendicular to the axis of rotation (axial runout tolerance to IT5 or better), in order to ensure uniform load distribution over all the rolling elements.

Fits

Conditions of rotation

The conditions of rotation indicate the motion of one bearing ring with respect to the load direction and are expressed as either circumferential load or point load, see table.

Point load

If the ring remains stationary relative to the load direction, there are no forces that displace the ring relative to its seating surface.
This type of loading is described as point load.

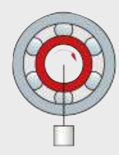
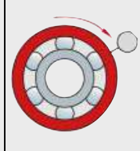
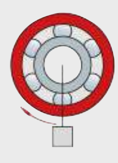
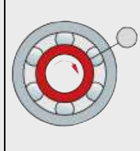
There is no risk that the seating surface will be damaged and a loose fit is possible.

Circumferential load

If forces are present that displace the ring relative to its seating surface, every point on the raceway is subjected to load over the course of one revolution of the bearing. A load with this characteristic is described as a circumferential load.

As damage to the bearing seating surface can occur, a tight fit should be used.

Conditions of rotation

Conditions of motion	Example	Schematic	Load case	Fit
Rotating inner ring Stationary outer ring Constant load direction	Shaft with weight load		Circumferential load on inner ring	Inner ring: tight fit necessary Outer ring: loose fit permissible
Stationary inner ring Rotating outer ring Load direction rotates with outer ring	Hub bearing arrangement with significant imbalance		and Point load on outer ring	
Stationary inner ring Rotating outer ring Constant load direction	Passenger car front wheel, track roller, (hub bearing arrangement)		Point load on inner ring	Inner ring: loose fit permissible Outer ring: tight fit necessary
Rotating inner ring Stationary outer ring Load direction rotates with inner ring	Centrifuge, vibrating screen		and Circumferential load on outer ring	

Tolerance zones

The ISO tolerances are defined in the form of tolerance zones. They are determined by their position relative to the zero line (= tolerance position) and their size (= tolerance grade, see ISO 286-1:1988). The tolerance position is indicated by letters (upper case for housings, lower case for shafts). A schematic illustration of the most common rolling bearing fits is shown in *Figure 1*.

$t_{\Delta Dmp}$ = tolerance for bearing outside diameter

$t_{\Delta dmp}$ = tolerance for bearing bore

① Zero line
② Housing bore
③ Shaft diameter
④ Loose fit
⑤ Transition fit
⑥ Tight fit

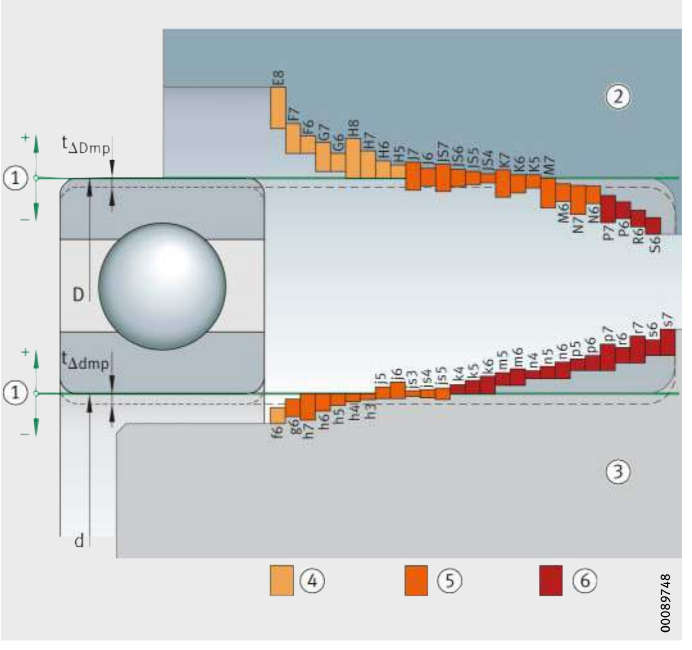


Figure 1
Fits for rolling bearings

The fit interference or fit clearance for shafts and housings is dependent on the specific bore diameter, see table, page 138, and table, page 146.

Radial internal clearance

The radial internal clearance applies to bearings with an inner ring and is determined on the unmounted bearing. It is defined as the amount by which the inner ring can be moved in a radial direction from one extreme position to the other in relation to the outer ring, *Figure 1*.

The groups are defined in DIN 620-4 or ISO 5753-1 respectively and are described in DIN 620-4 by means of symbols comprising the letter C and a numeral. ISO 5753-1 designates the groups by means of “Group” and a numeral, *Figure 1* and table.

CN, C2, C3, C4, C5 = radial internal clearance groups in accordance with DIN 620-4

Group N, 2, 3, 4, 5 = radial internal clearance groups in accordance with ISO 5753-1

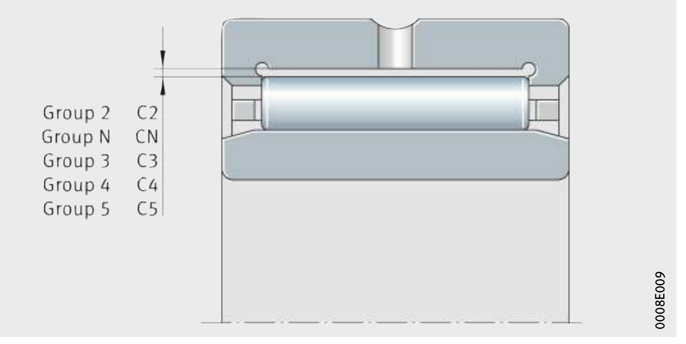


Figure 1
Radial internal clearance

Radial internal clearance groups

Internal clearance group to		Description	Application
DIN 620-4	ISO 5753-1		
CN	Group N	Normal radial internal clearance Group N is not included in bearing designations	For normal operating conditions with shaft and housing tolerances, see page 138
C2	Group 2	Internal clearance < Group N	For heavy alternating loads combined with swivel motion
C3	Group 3	Internal clearance > Group N	For bearing rings with press fits and large temperature differential between the inner and outer ring
C4	Group 4	Internal clearance > Group 3	
C5	Group 5	Internal clearance > Group 4	

The radial internal clearance of a bearing is dependent on the specific bore diameter and the type, see tables starting on page 172.

The internal clearance of spherical roller bearings, cylindrical roller bearings and toroidal roller bearings is normally determined using feeler gauges in a vertical position, *Figure 7*, page 73. It is important that the rings are centred relative to each other and the rollers within the bearing are correctly aligned. This can be achieved, for example, by rotating the bearing several times.

When the internal clearance is measured before mounting of the bearing, the specified radial internal clearance tolerance of the specific bearing should be obtained. In order to determine the actual internal clearance, a feeler gauge is then passed between the roller and bearing raceway.

In the case of multiple row bearings, the radial internal clearance must be measured simultaneously over all rows of rollers.

A measurement blade is used first that is somewhat thinner than the minimum value of the initial internal clearance. When passing the blade between the raceway and roller, it must be carefully moved back and forth. This operation must be carried out with measurement blades of increasing thickness until a certain resistance is detected. In the case of particularly large or thin-walled bearings, elastic deformation of the rings can influence the internal clearance determined.

Measurement is always carried out in the load-free zone.
During mounting, the radial internal clearance should be measured continuously until the specified value is achieved.

The radial internal clearance should be determined at approx. +20 °C. In the case of particularly thin-walled bearing rings, elastic deformation of the rings can influence the internal clearance determined.

In spherical roller bearings, the radial internal clearance must be measured simultaneously over both rows of rolling elements, *Figure 2*. It can only be ensured that the inner ring is not laterally offset relative to the outer ring when the internal clearance values are identical for both rows of rollers. Due to the width tolerance of the rings, alignment of the end faces cannot be taken as a reliable indicator.

s_r = radial internal clearance

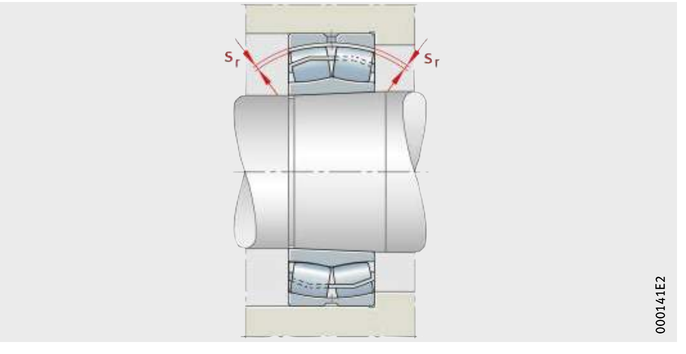


Figure 2
Radial internal clearance of a spherical roller bearing

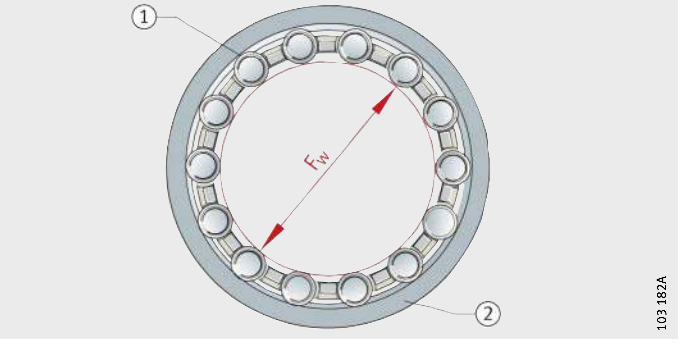
Internal clearance and operating clearance

In the case of cylindrical roller bearings, the inner ring and outer ring can be mounted individually. If the inner ring can be separated from the bearing, the expansion of the inner ring can be measured using an external micrometer instead of the reduction in radial internal clearance, *Figure 3*.

Figure 3
Measurement of expansion using external micrometer

Enveloping circle

For bearings without an inner ring, the enveloping circle F_w is used. This is the inner inscribed circle of the needle rollers in clearance-free contact with the outer raceway, *Figure 4*. When the bearings are unmounted, it is in the tolerance zone F6 (except in the case of drawn cup needle roller bearings).



F_w = enveloping circle diameter

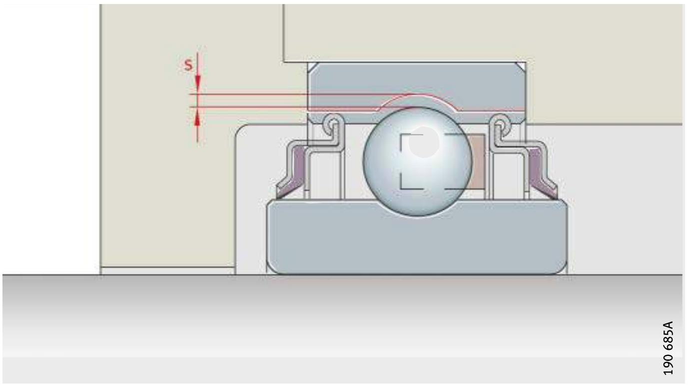
1 Needle roller
2 Outer raceway

Figure 4
Enveloping circle

Operating clearance

The operating clearance is determined on a mounted bearing still warm from operation. It is defined as the amount by which the shaft can be moved in a radial direction from one extreme position to the other, *Figure 5*.

The operating clearance is derived from the radial internal clearance and the change in the radial internal clearance as a result of interference fit and thermal influences in the mounted condition.



s = operating clearance

Figure 5
Operating clearance

Operating clearance value

The size of the operating clearance is dependent on the bearing operating and installation conditions. A larger operating clearance is, for example, necessary if heat is transferred via the shaft, the shaft undergoes deflection or if misalignment occurs.

An operating clearance smaller than Group N should only be used in special cases, for example in high precision bearing arrangements.

The normal operating clearance is achieved with the internal clearance Group N or, in the case of larger bearings, predominantly with Group 3 if the recommended shaft and housing tolerances are observed, see page 138.

Calculation of operating clearance

The operating clearance is derived from:

$$s = s_r - \Delta s_P - \Delta s_T$$

s µm
Radial operating clearance of mounted bearing warm from operation

s_r µm
Radial internal clearance

Δs_P µm
Reduction in radial internal clearance due to fit

Δs_T µm
Reduction in radial internal clearance due to temperature.

Reduction in radial internal clearance due to fit

The radial internal clearance is reduced due to the fit as a result of expansion of the inner ring and contraction of the outer ring:

$$\Delta s_P = \Delta d + \Delta D$$

Δd µm
Expansion of the inner ring
ΔD µm
Contraction of the outer ring.

Expansion of the inner ring

The expansion of the inner ring is calculated as follows:

$$\Delta d \approx 0{,}9 \cdot U \cdot d/F \approx 0{,}8 \cdot U$$

U µm
Theoretical interference of the fitted parts with firm seating. The theoretical oversize of the fitted parts with a firm seating is determined from the mean deviations and the upper and lower deviations of the tolerance zones of the fitted parts reduced by $^1/_3$ of their acceptable value. This must be reduced by the amount by which parts are smoothed during fitting
d mm
Bore diameter of the inner ring
F mm
Raceway diameter of the inner ring.

!

For very thin-walled housings and light metal housings, the reduction in the radial internal clearance must be determined by mounting trials.

Contraction of the outer ring

The contraction of the outer ring is calculated as follows:

$$\Delta D \approx 0{,}8 \cdot U \cdot E/D \approx 0{,}7 \cdot U$$

E mm
Raceway diameter of the outer ring
D mm
Outside diameter of the outer ring.

Reduction in radial internal clearance due to temperature

The radial internal clearance can alter considerably if there is a substantial temperature difference between the inner ring and outer ring.

$$\Delta s_T = \alpha \cdot d_M \cdot 1000 \cdot \left(\vartheta_{IR} - \vartheta_{AR}\right)$$

Δs_T µm
Reduction in radial internal clearance due to temperature
α K^{-1}
Coefficient of thermal expansion of steel: $\alpha = 0{,}000011\ K^{-1}$
d_M mm
Mean bearing diameter (d + D)/2
ϑ_{IR} °C, K
Temperature of the inner ring
ϑ_{AR} °C, K
Temperature of the outer ring (usual temperature difference between inner and outer ring: 5 K to 10 K).

!

Where shafts start up quickly, a larger radial internal clearance should be used since adequate thermal compensation between the bearing, shaft and housing does not occur in this situation.

Δs_T can, in this case, be significantly higher in this case than for continuous operation.

Axial internal clearance

The axial internal clearance s_a is defined as the amount by which one bearing ring can be moved relative to the other, without load, along the bearing axis, *Figure 6*.

s_a = axial internal clearance
s_r = radial internal clearance

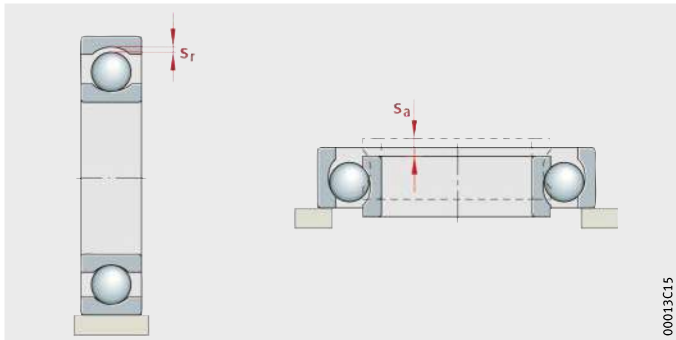


Figure 6
Axial internal clearance in comparison with radial internal clearance

In various bearing types, the radial internal clearance s_r and the axial internal clearance s_a are dependent on each other. Guide values for the correlation between the radial and axial internal clearance are shown for some bearing types in the table.

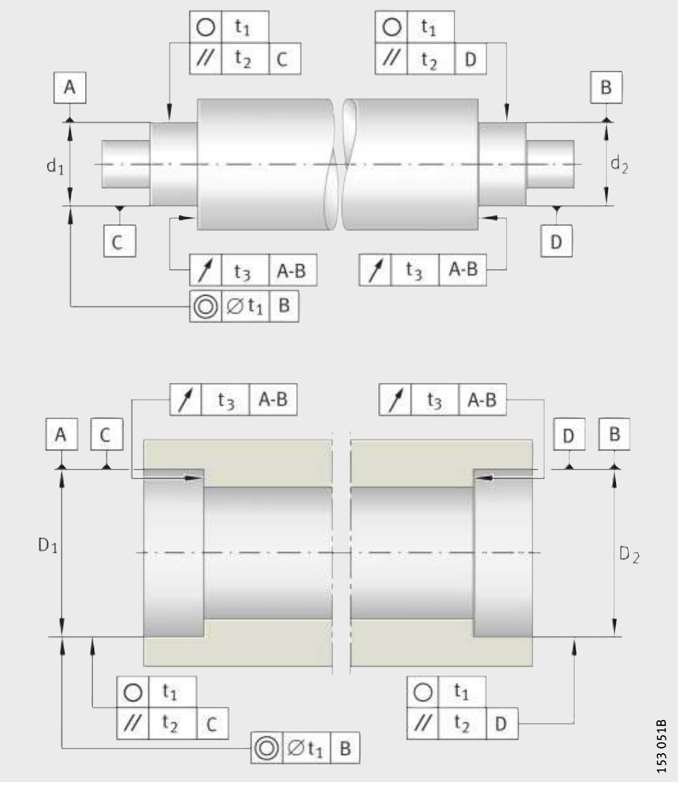
Correlation between axial internal clearance and radial internal clearance

Bearing type			Ratio between axial and radial internal clearance s_a/s_r
Self-aligning ball bearings			$2{,}3 \cdot Y_0$
Spherical roller bearings			$2{,}3 \cdot Y_0$
Tapered roller bearings	single row, arranged in pairs		$4{,}6 \cdot Y_0$
	matched pairs (N11CA)		$2{,}3 \cdot Y_0$
Angular contact ball bearings	double row	series 32 and 33	1,4
		series 32..-B and 33..-B	2
	single row	series 72..-B and 73..-B, arranged in pairs	1,2
Four point contact bearings			1,4

Axial internal clearance for double row FAG angular contact ball bearings and FAG four point contact bearings, see tables starting on page 182.

Geometrical and positional tolerances of bearing seating surfaces

In order to achieve the required fit, the bearing seats and fit surfaces of the shaft and housing bore must conform to certain tolerances, *Figure 1* and table, page 29.



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

Figure 1
Geometrical and positional tolerances

Accuracy of bearing seating surfaces

The degree of accuracy for the bearing seat tolerances on the shaft and in the housing is given in the table, page 29.

Second bearing seat

The tolerances for a second bearing seat on the shaft (d_2) or in the housing (D_2) (expressed in terms of coaxiality to DIN ISO 1101) must be based on the angular adjustment facility of the specific bearing. Misalignments due to elastic deformation of the shaft and housing must be taken into consideration.

Housings

For split housings, the joints must be free from burrs.

The accuracy of the bearing seats is determined as a function of the accuracy of the bearing selected.

Guide values for geometrical and positional tolerances of bearing seating surfaces

Bearing tolerance class to ISO 492	Bearing tolerance class to DIN 620	Bearing seating surface	Fundamental tolerance grades: Diameter tolerance	Roundness tolerance t_1	Parallelism tolerance t_2	Total axial runout tolerance of abutment shoulder t_3
Normal 6X	PN (P0) P6X	Shaft	IT6 (IT5)	Circumferential load IT4/2	IT4/2	IT4
				Point load IT5/2	IT5/2	
		Housing	IT7 (IT6)	Circumferential load IT5/2	IT5/2	IT5
				Point load IT6/2	IT6/2	
5	P5	Shaft	IT5	Circumferential load IT2/2	IT2/2	IT2
				Point load IT3/2	IT3/2	
		Housing	IT6	Circumferential load IT3/2	IT3/2	IT3
				Point load IT4/2	IT4/2	
4	P4 P4S[1] SP[1]	Shaft	IT4	Circumferential load IT1/2	IT1/2	IT1
				Point load IT2/2	IT2/2	
		Housing	IT5	Circumferential load IT2/2	IT2/2	IT2
				Point load IT3/2	IT3/2	
	UP[1]	Shaft	IT3	Circumferential load IT0/2	IT0/2	IT0
				Point load IT1/2	IT1/2	
		Housing	IT4	Circumferential load IT1/2	IT1/2	IT1
				Point load IT2/2	IT2/2	

ISO fundamental tolerances (IT grades) to ISO 286-1:1988.

[1] Not included in DIN 620.

Geometrical and positional tolerances

Roughness of bearing seats

The roughness of the bearing seats must be matched to the tolerance class of the bearings. The mean roughness value Ra must not be too high, in order to maintain the interference loss within limits. Shafts should be ground and bores should be precision turned. Guide values are given in the table.

The bore and shaft tolerances and permissible roughness values are also given in the design and safety guidelines in the product sections. The guide values for roughness correspond to DIN 5425-1.

Guide values for surface quality of bearing seats

Diameter of bearing seat d (D) mm		Recommended mean roughness value Ra and roughness classes for ground bearing seats Diameter tolerance corresponding to [1] μm			
over	incl.	IT7	IT6	IT5	IT4
–	80	1,6	0,8	0,4	0,2
80	500	1,6	1,6	0,8	0,4
500	1 250	3,2[2]	1,6	1,6	0,8

[1] Values for IT grades in accordance with DIN ISO 286-1:2010-11.

[2] For mounting of bearings using the hydraulic method, a value Ra = 1,6 μm must not be exceeded.

Safety guidelines

Guidelines on the mounting of rolling bearings

In the mounting and dismounting of rolling bearings, important safety guidelines must be observed so that these activities can be carried out safely and correctly. The purpose of this mounting manual is to assist the fitter in mounting rolling bearings safely and correctly.

The objective of the safety guidelines is:

- to prevent personal injury or damage to property that may be caused by errors in mounting
- to facilitate, through correct mounting, a long operating life of the mounted bearing.

General safety regulations

The mounting and dismounting of rolling bearings normally involves high forces, pressures and temperatures. Due to these risk factors, mounting and dismounting of rolling bearings should only be carried out by qualified personnel.

Qualification of personnel

A person defined as qualified personnel:

- is authorised to perform mounting of the rolling bearings and adjacent components
- has all the knowledge necessary for mounting and dismounting of the components
- is familiar with the safety regulations.

Personal protective equipment

Personal protective equipment is intended to protect operating personnel against health hazards. This comprises safety shoes, safety gloves and if necessary a protective helmet and these must be used in the interests of personal safety.

Depending on the mounting location and on the machine or equipment in which the rolling bearings are to be mounted, it may be necessary to use additional personal protective equipment.
The applicable regulations relating to occupational safety must be observed.

Safety guidelines

Safety specifications

In order to prevent the occurrence of personal injury or damage to property during mounting, the following safety specifications must be observed.

Fundamental specifications

The mounting area must be kept free of trip hazards.

Heavy components such as the upper and lower housing sections, seals, covers and rolling bearings must be secured to prevent toppling or falling.

When heavy components are being set down and fitted together, particular attention must be paid to the limbs in order to prevent crushing.

Mounting and maintenance work of all types may only be carried out when the machine or equipment is at a standstill.

Lubricants

The lubricants used for greasing may contain components that are hazardous to health. A safety data sheet exists for each lubricant that describes the hazards.

Avoid direct bodily contact with the lubricant and use protective gloves.

Environmental hazards

Depending on the ambient conditions, safety risks may be present at the mounting location that are not associated directly with the rolling bearing but must be taken into consideration in mounting of the rolling bearing. These may include dusts that are hazardous to health or working at a considerable height. Furthermore, the machine or equipment in which the rolling bearing is mounted may be a source of hazards, for example as a result of movable machinery or equipment parts.

Before starting mounting work, a local safety engineer must be consulted. All safety specifications that are applicable to the mounting location and the machine or equipment affected by the mounting work must be observed.

Disposal

Any cloths soaked with grease or solvents, excess grease, packaging material and any other waste generated in connection with mounting and dismounting must be disposed of by environmentally acceptable methods. The applicable legal regulations must be observed.

Transport specifications

In order to prevent the occurrence of personal injury or damage to property during transport, the following transport specifications must be observed.

Before transport, secure rolling bearings against swivelling out or falling apart, *Figure 1*.

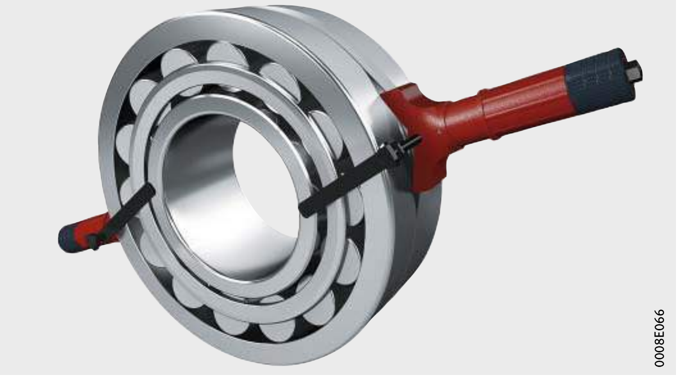


Figure 1
Secure lifting tool with protection against swivelling out

Lifting of heavy components must be carried out using suitable technical accessories. The mounting personnel must be familiar with correct usage of the accessories and must observe all safety specifications relating to the handling of suspended loads.

The following must be observed:

- Do not remain below or within the swivel range of suspended loads.
- Use only lifting gear and tackle that is approved and has sufficient load capacity.
- Do not draw unprotected lifting tackle under load across sharp edges, avoid kinking or twisting.
- Never leave suspended loads unsupervised.

Preparations for mounting and dismounting

Working conditions

Before the mounting and dismounting of rolling bearings, all preparations must be made for a problem-free work process.

Based on the workshop drawing, it is necessary to become familiar with the structure of the design and the sequence in which the individual parts are joined together. Before starting mounting work, a program should be prepared of the individual work operations and clarity should be established on the necessary heating temperatures, the forces for fitting and removal of the bearings and the grease quantity required.

For more extensive work, a mounting manual should be available that precisely describes all relevant work. The manual should also contain details on means of transport, mounting equipment, measurement tools, type and quantity of lubricant and a precise description of the mounting procedure.

Guidelines for mounting

The following guidelines must always be taken into account:

- Keep the mounting area clean and free from dust.
- Protect bearings from dust, contaminants and moisture. Contaminants have a detrimental influence on the running and operating life of rolling bearings.
- Before mounting work is started, familiarise yourself with the design by means of the final assembly drawing.
- Before mounting, check whether the bearing presented for mounting corresponds to the data in the drawing.
- Check the housing bore and shaft seat for dimensional, geometrical and positional accuracy and cleanliness.
- Check that no edges are present which could hamper the mounting of bearing rings on the shaft or in the housing bore. A lead chamfer of 10° to 15° is advantageous in this case.
- Wipe away any anti-corrosion agent from the seating and contact surfaces, wash anti-corrosion agent out of tapered bores.
- Cylindrical seating surfaces of the bearing rings should be rubbed with a very thin layer of Arcanol mounting paste.
- Do not cool the bearings excessively. Moisture due to condensation can lead to corrosion in the bearings and bearing seats.
- After mounting, supply the rolling bearings with lubricant.
- Check the correct functioning of the bearing arrangement.

Handling of rolling bearings before mounting

The anti-corrosion agents in bearings with an oil-based preservative are compatible and miscible with oils and greases having a mineral oil base. Compatibility should be checked if synthetic lubricants or thickeners other than lithium or lithium complex soaps are used. If there is an incompatibility, the anti-corrosion oil should be washed out before greasing, especially in the case of lubricants with a PTFE/alkoxyfluoroether base and thickeners based on polycarbamide. If in doubt, please contact the relevant lubricant manufacturer. In washing out, there is a risk that contamination will be introduced into the bearing.

The anti-corrosion oil should be washed off the seating and locating surfaces (especially in the case of tapered bearing bores) before mounting in order to ensure a secure fit.

In the thermal mounting of bearings, the maximum permissible temperature of the anti-corrosion agent must be observed.

Prior to mounting, wash used and contaminated bearings carefully with kerosene agent and then oil or grease them immediately afterwards.

Rolling bearings must not be machined subsequently.
For example, it is not permissible to introduce lubrication holes, grooves, ground surfaces or the like, since this can liberate stresses in the rings that lead to premature destruction of the bearing.
There is also a risk in this case that swarf or grinding dust will penetrate the bearing.

When washing out bearings, the highest possible cleanliness must be ensured.

Cleanliness during mounting

Rolling bearings must be protected under all circumstances against contamination and moisture, since the ingress of even very small particles into the bearing can damage the running surfaces.
For this reason, the mounting area must be dry and free from dust. For example, it must not be located in the vicinity of grinding machines. The use of compressed air must be avoided. Cleanliness of the shaft and housing as well as all other parts must be ensured. Castings must be free from moulding sand. After cleaning, a protective coating should be applied to the inner housing surfaces that will prevent very small particles from coming loose during operation. Any anti-corrosion coatings and colour residues must be carefully removed from the bearing seats on the shaft and in the housing.
In the case of turned parts, it must be ensured that burrs are removed and all sharp edges are broken.

Adjacent parts

All parts of the bearing arrangement must be checked for dimensional and geometrical accuracy before assembly.

For example, correct running of a rolling bearing can be impaired by non-compliant bearing seating tolerances, out-of-round housings and shafts and misaligned locating surfaces and this can lead to premature failure.

Measurement of bearing seat

A significant work operation for successful mounting of bearings is the prior measurement of the components used. Various measuring devices are used here. In all measurements, it must be ensured that the measuring device is at approximately the same temperature as the parts to be measured.

Cylindrical seating surfaces

The dimensional accuracy of cylindrical seating surfaces and their roundness should be checked with the aid of micrometers at various measurement points, *Figure 1* and *Figure 4*, page 37.

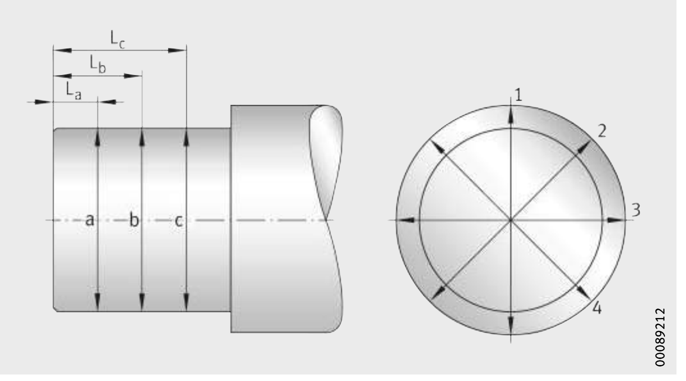
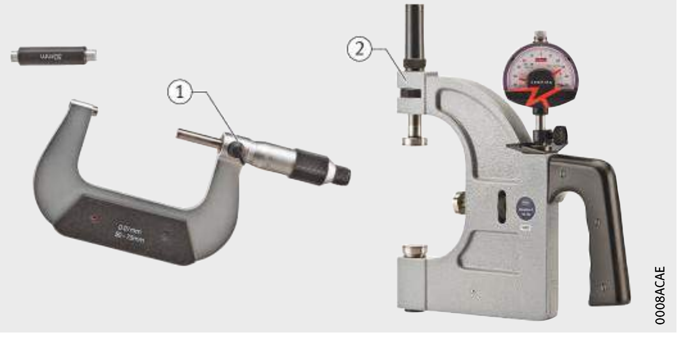


Figure 1
Checking the cylindricity of a shaft

Secure positioning and correct measurement of cylindrical seating surfaces is ensured by the snap gauge, *Figure 2*. The master disc is marked with the diameter to which the gauge must be set.



① External micrometer
② Snap gauge

Figure 2
Gauge for measurement
of shaft diameters

The measurement of bores is carried out using either conventional internal micrometers or so-called comparator gauges, *Figure 3*. The master ring shown is used for calibration of the measurement tool.

① Internal micrometer
② Comparative gauge with master ring

Figure 3
Gauge for measurement of bores

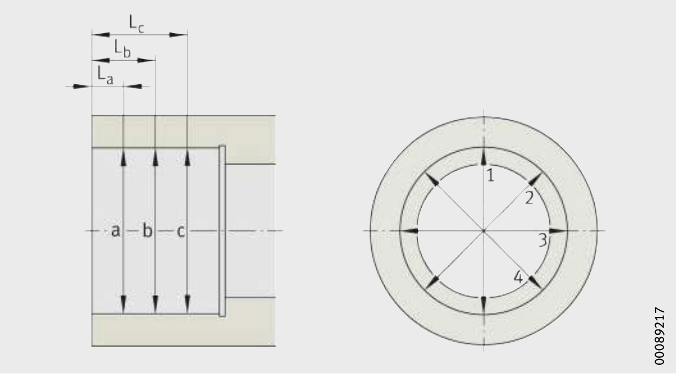


Figure 4
Checking the cylindricity of a housing

Tapered seating surfaces

In order to ensure firm seating of the inner ring on the shaft, the taper of the shaft must match precisely the taper of the inner ring bore.

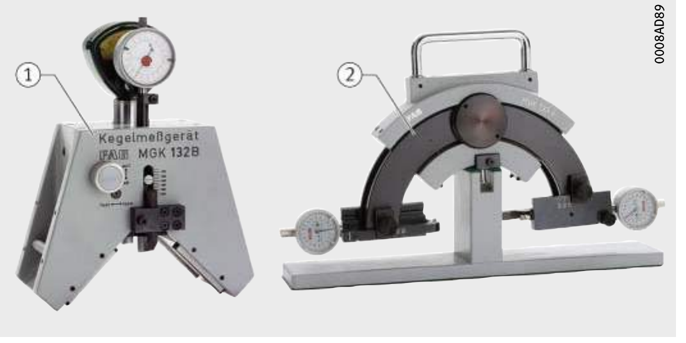
The taper of rolling bearing rings is standardised. For most bearing series, it is 1:12. Depending on the requirements and the bearing width, bearings with a taper 1:30 are possible.

The simplest gauge for small, tapered bearing seats is the taper ring gauge, *Figure 5*. By means of the touching method, it can be determined whether the shaft and ring gauge match, while corrections are made until the ring gauge is in contact over its whole width.

The inner rings of bearings should not be used as ring gauges.

Figure 5
Touching with a taper ring gauge

For precise checking of tapered shaft seating surfaces, Schaeffler has developed the taper gauges FAG MGK 133 and FAG MGK 132, *Figure 6*. The taper and diameter of the bearing seat are measured precisely using a comparator taper or segment. Both devices are easy to use, since it is not necessary to remove the workpiece from the machining equipment for measurement.



① Taper gauge FAG MGK 132
② Taper gauge FAG MGK 133

Figure 6
Taper gauges FAG MGK 132 and FAG MGK 133

The taper gauge FAG MGK 133 is used to measure tapers shorter than 80 mm. Depending on the device size, the outside diameter of the taper can be between 27 mm and 205 mm.

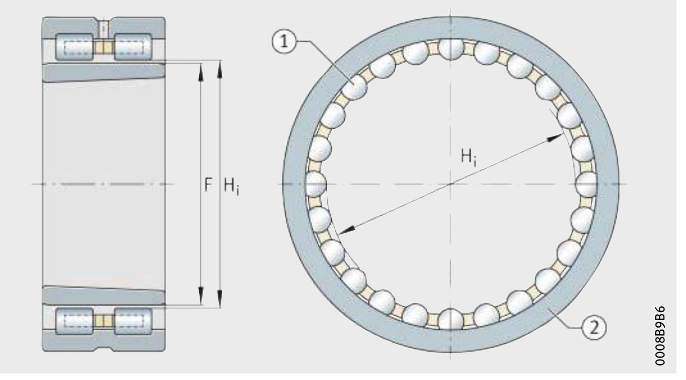
The taper gauge FAG MGK 132 is suitable for taper lengths of 80 mm or larger and taper diameters from 90 mm to 820 mm.

Enveloping circle

The radial internal clearance of a mounted cylindrical roller bearing is determined by the difference between the roller enveloping circle diameter and the raceway diameter of the ribless ring.

Enveloping circle gauge FAG MGI 21

In the case of cylindrical roller bearings with a separable inner ring NNU49SK, the radial internal clearance or preload is determined by the difference between the diameter of the inner enveloping circle H_i and the raceway F. The internal enveloping circle is defined as the circle inscribed internally by all rollers when they are in contact with the outer ring raceway, *Figure 7*.



H_i = inner enveloping circle
F = raceway diameter

① Rolling element
② Outer ring

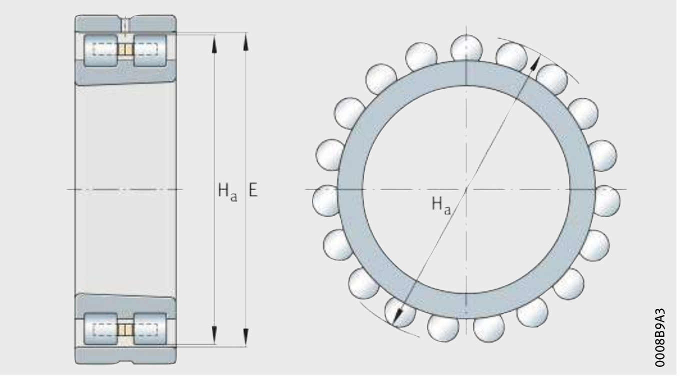
Figure 7
Inner enveloping circle of cylindrical roller bearings NNU49SK (separable inner ring)

The internal enveloping circle is measured using MGI 21; in conjunction with a snap gauge, the radial internal clearance of the mounted bearing can be determined, *Figure 8*. The dimension for the enveloping circle diameter is transferred to the snap gauge. The enveloping circle gauge FAG MGI 21 is used for cylindrical roller bearings with a separable inner ring, such as FAG NNU49SK.

Figure 8
Enveloping circle gauge FAG MGI 21

Enveloping circle gauge FAG MGA 31

In the case of cylindrical roller bearings with a separable outer ring NN30ASK, the radial internal clearance or preload is determined by the difference between the diameter of the raceway E and the outer enveloping circle H_a. The outer enveloping circle is defined as the circle inscribed externally by all rollers when they are in contact with the inner ring raceway, *Figure 9*.



E = raceway
H_a = external enveloping circle

Figure 9
External enveloping circle of cylindrical roller bearings NN30ASK (separable outer ring)

The external enveloping circle is measured using MGA 31; in conjunction with a snap gauge, the radial internal clearance of the mounted bearing can be determined, *Figure 10*.

The dimension for the raceway diameter is transferred using the bore gauge to the enveloping circle gauge. The enveloping circle gauge FAG MGA 31 is used for cylindrical roller bearings with a separable outer ring, such as FAG NN30ASK.

Figure 10
Enveloping circle gauge FAG MGA 31

The two opposing steel segments of the enveloping circle gauge are the measuring surfaces. One segment is rigidly attached to the device while the other is capable of radial motion; this motion is transferred to the precision dial indicator.

For measurement, the bearing outer ring must be mounted in the housing. Once the diameter of the outer ring raceway has been determined using the bore gauge, the dimension is transferred to the enveloping circle gauge.

The inner ring, which is held together with the roller and cage assembly by the cage, is first slid onto the tapered shaft seat with form fit. The enveloping circle gauge is then positioned on the roller and cage assembly and the inner ring is pressed into place until the precision dial indicator shows the required dimension.

Plus values indicate preload, while minus values indicate radial internal clearance; the value zero gives a clearance-free bearing.

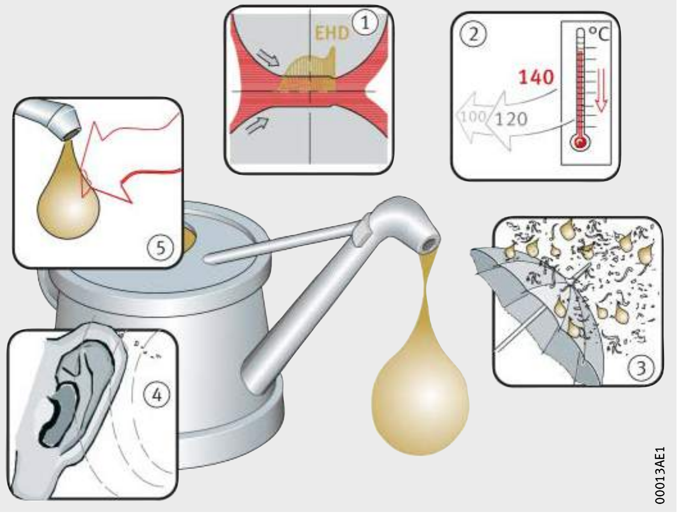
Principles

Lubrication and maintenance are important for the reliable operation and long operating life of rolling bearings.

Functions of the lubricant

The lubricant should, *Figure 1*:

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



① Formation of a lubricant film capable of supporting loads
② Heat dissipation in the case of oil lubrication
③ Sealing of the bearing against external contaminants in the case of grease lubrication
④ Damping of running noise
⑤ Protection against corrosion

Figure 1
Functions of the lubricant

Selection of the type of lubrication

It should be determined as early as possible in the design process whether bearings should be lubricated using grease or oil.

The following factors are decisive in determining the type of lubrication and quantity of lubricant:

- the operating conditions
- the type and size of the bearing
- the adjacent construction
- the lubricant feed.

Criteria for grease lubrication

In the case of grease lubrication, the following criteria must be considered:

- very little design work required
- the sealing action
- the reservoir effect
- long operating life with little maintenance work (lifetime lubrication possible in certain circumstances)
- in the case of relubrication, the provision of collection areas for old grease and feed ducts
- no heat dissipation by the lubricant
- no rinsing out of wear debris and other particles.

Criteria for oil lubrication

In the case of oil lubrication, the following criteria must be considered:

- good lubricant distribution and supply to contact areas
- dissipation of heat possible from the bearing (significant principally at high speeds and/or loads)
- rinsing out of wear debris
- very low friction losses with minimal quantity lubrication
- more demanding requirements in terms of feed and sealing.

Under extreme operating conditions (such as very high temperatures, vacuum, aggressive media), it may be possible to use special lubrication methods such as solid lubricants in consultation with the engineering service.

Design of lubricant feed lines

The feed lines and lubrication holes in the housings and shafts, *Figure 2* and *Figure 3*, must:

- lead directly to the lubrication hole in the rolling bearing
- be as short as possible
- be equipped with a separate feed line for each bearing.

Ensure that the feed lines are filled, *Figure 2*; the feed line should be bled if necessary.

Follow the guidelines provided by the manufacturers of the lubrication equipment.

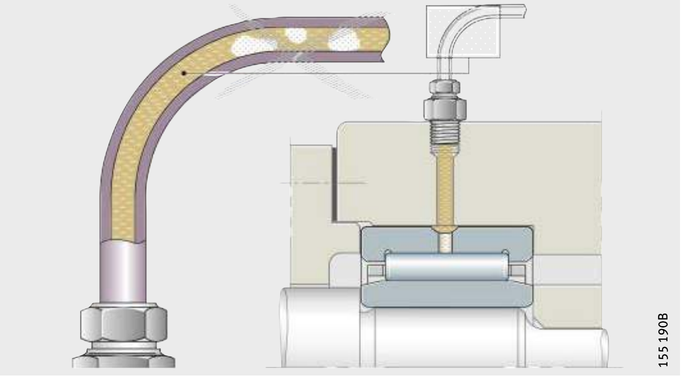


Figure 2
Lubricant feed lines

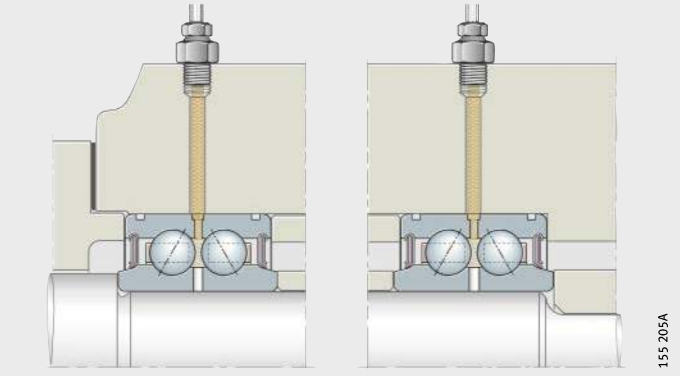


Figure 3
Arrangement of feed lines to more than one bearing on a shaft

Lubrication

Greases

The optimum bearing operating life can be achieved if suitable lubricants are selected. Account must be taken of application-related influencing factors such as bearing type, speed, temperature and load. In addition, attention must be paid to environmental conditions, the resistance of plastics, legal and environmental regulations as well as costs.

Specification according to DIN or the design brief

Greases K standardised in accordance with DIN 51825 should be used in preference. However, this standard only formulates minimum requirements for greases. This means that greases in one DIN class may exhibit differences in quality and may be suitable to varying degrees for the specific application. As a result, rolling bearing manufacturers frequently specify greases by means of design briefs that give a more detailed description of the profile of requirements for the grease.

Initial greasing and new greasing

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

- Fill the bearings such that all functional surfaces definitely receive grease.
- Fill any housing cavity adjacent to the bearing with grease only to the point where there is still sufficient space for the grease displaced from the bearing. This is intended to avoid co-rotation of the grease. If a large, unfilled housing cavity is adjacent to the bearing, sealing shields or washers as well as baffle plates should be used to ensure that an appropriate grease quantity (similar to the quantity that is selected for the normal degree of filling) remains in the vicinity of the bearing. A grease filling of approx. 90% of the undisturbed free bearing volume is recommended. This is defined as the volume in the interior of the bearing that does not come into contact with rotating parts (rolling elements, cage).
- In the case of bearings rotating at very high speeds, such as spindle bearings, a smaller grease quantity is generally selected (approx. 60% of the undisturbed free bearing volume or approx. 30% of the total free bearing volume), in order to aid grease distribution during starting of the bearings.
- The sealing action of a gap seal is improved by the formation of a stable grease collar. This effect is supported by continuous relubrication.
- If the correct degree of filling is used, favourable friction behaviour and low grease loss will be achieved.
- If there is a pressure differential between the two sides of the bearing, the flow of air may drive the grease and the released base oil out of the bearing and may also carry contamination into the bearing. In such cases, pressure balancing is required by means of openings and holes in the adjacent parts.

- Bearing rotating at low speeds ($n \cdot d_M < 50\,000\ \text{min}^{-1} \cdot \text{mm}$) and their housings must be filled completely with grease. The churning friction occurring in this case is negligible. It is important that the grease introduced is held in the bearing or vicinity of the bearing by the seals and baffle plates. The reservoir effect of grease in the vicinity of the bearing leads to an increase in the lubrication interval. However, this is conditional on direct contact with the grease in the bearing (grease bridge). Occasional shaking will also lead to fresh grease moving into the bearing from its environment (internal relubrication).
- If a high temperature is expected in the bearing, the appropriate grease should be supplemented by a grease reservoir that has a surface as large as possible facing the bearing and that dispenses oil. The favourable quantity for the reservoir is two to three times the normal degree of filling. The reservoir must be provided either on one side of the bearing or preferably to an identical extent on both sides.
- Bearings sealed on both sides using sealing washers or sealing shields are supplied with an initial greasing. The grease quantity normally introduced fills approx. 90% of the undisturbed free bearing volume. This filling quantity is retained well in the bearing even in the case of high speed parameters ($n \cdot d_M > 400\,000\ \text{min}^{-1} \cdot \text{mm}$). In the case of higher speed parameters, please consult Schaeffler. A higher degree of filling in sealed bearings will lead to higher friction and continuous grease loss until the normal degree of filling is restored. If the egress of grease is hindered, a considerable increase in torque and temperature must be anticipated. Bearings with a rotating outer ring also receive less grease (50% of the normal filling).

- In the case of higher speed parameters, the bearing temperature may settle at a higher value, in some cases over several hours, if the grease quantity during the starting phase has not been set correctly, *Figure 4*. The temperature is higher and the increased temperature is present for longer, the more the bearings and the cavities adjacent to the bearings are filled with grease and the more difficult it is for grease to escape freely. A remedy is a so-called interval running-in process with appropriately determined standstill periods for cooling. If suitable greases and grease quantities are used, equilibrium is achieved after a very short time.

Deep groove ball bearing, freshly greased

M_r = frictional torque
t = time
ϑ = temperature

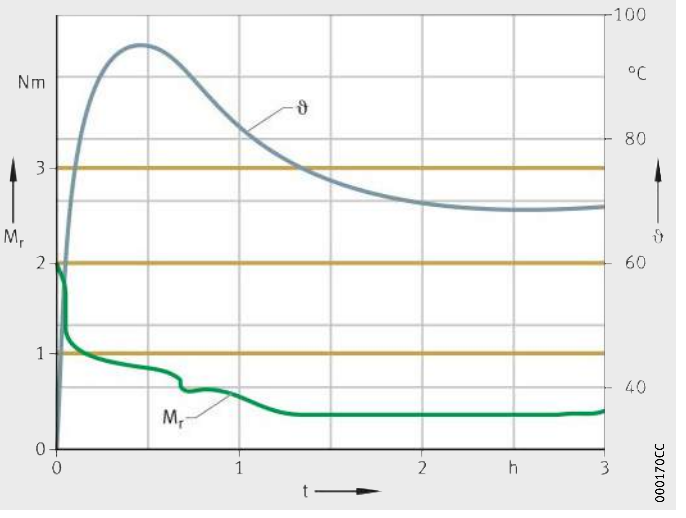


Figure 4
Frictional torque and temperature

Arcanol rolling bearing greases

Rolling bearing greases under the name Arcanol are subjected to 100% quality inspection, *Figure 5*. The inspection methods at Schaeffler are among the most demanding in the market. As a result, Arcanol rolling bearing greases fulfil the highest quality requirements.

The different greases cover almost all applications. They are developed by experienced application engineers and are produced by the best manufacturers in the market. Different greases are used depending on the particular application, see table, page 190.

Figure 5
Analysis of the thermal behaviour of greases

Oils

For the lubrication of rolling bearings, mineral oils and synthetic oils are essentially suitable, see table, page 190. Oils with a mineral oil base are used most frequently. These mineral oils must fulfil at least the requirements according to DIN 51517 (lubricating oils).

Special oils, which are often synthetic oils, are used where extreme operating conditions are present. The resistance of the oil is subjected to particular requirements under challenging conditions involving, for example, temperature or radiation. The effectiveness of additives in rolling bearings has been demonstrated by well-known oil manufacturers. For example, wear protection additives are particularly important for the operation of rolling bearings in the mixed friction range.

Further information

- Further information on the storage, miscibility and selection of lubricants can be found in TPI 176, Lubrication of Rolling Bearings.

Storage of rolling bearings

Corrosion protection and packaging

The performance capability of modern rolling bearings lies at the boundaries of what is technically achievable. Not only the materials but also the dimensional accuracies, tolerances, surface quality values and lubrication are optimised for maximum function.
Even the slightest deviations in functional areas, for example as a result of corrosion, can impair the performance capability.

In order to realise the full performance capability of rolling bearings, it is essential to match the corrosion protection, packaging, storage and handling to each other. They are optimised by Schaeffler as part of the process of preserving all the characteristics of the product at the same time. In addition to protection of the surfaces against corrosion, other important characteristics include emergency running lubrication, friction, lubricant compatibility, noise behaviour, resistance to ageing and compatibility with rolling bearing components (brass cage, plastic cage, elastomer seal).
Corrosion protection and packaging are matched by Schaeffler to these characteristics. The bearings must be stored in their original packaging for as long as possible.

Storage conditions

The basic precondition for storage is a closed storage room in which no aggressive media of any sort may have an effect, such as exhaust from vehicles or gases, mist or aerosols of acids, alkalis or salts. Direct sunlight must also be avoided. The bearings must be stored lying flat, not standing.

The storage temperature should be as constant as possible and the humidity as low as possible. Jumps in temperature and increased humidity lead to condensation.

The following conditions must be fulfilled:

- frost-free storage at a minimum temperature of +5 °C (secure prevention of hoarfrost formation, permissible up to 12 hours per day down to +2 °C)
- maximum temperature +40 °C (prevention of excessive run-off of anti-corrosion oils)
- relative humidity less than 65% (with changes in temperature, up to 70% permissible for up to 12 hours per day).

! The temperature and humidity must be continuously monitored.

Storage periods

Rolling bearings should not be stored for longer than 3 years. This applies both to open and to greased rolling bearings with sealing shields or washers. In particular, greased rolling bearings should not be stored for too long, since the chemical-physical behaviour of greases may change during storage. Even if the minimum performance capacity remains, the safety reserves of the grease may have diminished. In general, rolling bearings can be used even after their permissible storage period has been exceeded if the storage conditions during storage and transport were observed. If the storage periods are exceeded, it is recommended that the bearing should be checked for corrosion, the condition of the anti-corrosion oil and where appropriate the condition of the grease before it is used.

Seals

Classification of seals

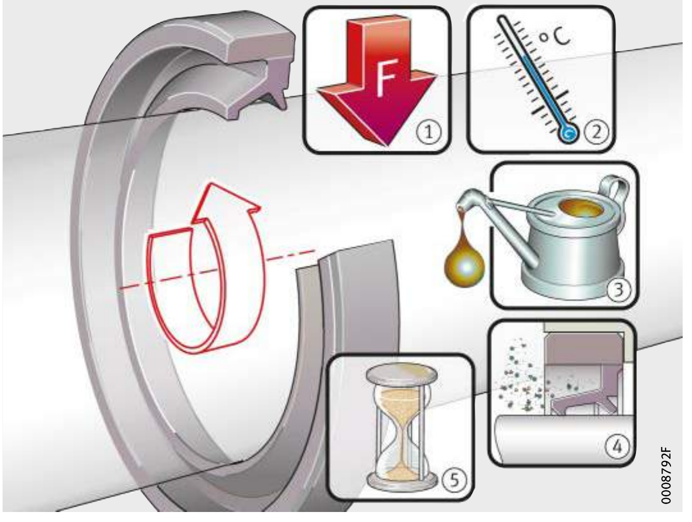
Seals play a decisive role in protecting bearings against contamination. If inadequate seals are used, contaminants can penetrate the bearing or an unacceptably large quantity of lubricant may escape from the bearing. Bearings that are contaminated or running dry will fail long before they reach their fatigue rating life.

Non-contact and contact seals

A basic distinction is made between contact and non-contact seals in the adjacent construction and the bearing.

Non-contact seals

Non-contact seals include gap seals, labyrinth seals, baffle plates and sealing shields. When fitting these types of seal, particular attention must be paid to the size of the seal gap after fitting and during operation. The resulting seal gap in operation is decisively influenced by external factors such as temperature differences, loads and deformations, *Figure 1*.



① Load
② Temperature differences
③ Lubricants
④ Contaminants
⑤ Ageing

Figure 1
Influences
on the operating life of seals

In grease lubrication of the bearing, the seal gaps formed must be filled with the same grease that is used within the bearing arrangement. An additional grease collar on the outside of the seal will protect the bearing against contamination.

Contact seals

Contact seals include felt rings, V rings or rotary shaft seals with one or more lips. They are normally in contact with the running surface under radial contact force. The contact force should be kept small in order to avoid an excessive increase in frictional torque and temperature. The frictional torque and temperature as well as the wear of the seal are also affected by the lubrication condition at the running surface, the roughness of the running surface and the sliding velocity. Correct fitting of the seal has a decisive influence on the possible operating life of the bearing.

Sealed bearings

Sealed rolling bearings are fitted with different seal concepts depending on the specific bearing type and series.

In the case of almost all bearings that are already sealed at the time of delivery, removal of the seal should be avoided. If a prefitted seal does not function correctly, the entire bearing must be replaced.

Sealed bearings must not be heated in an oil bath, and the heating temperature must not exceed +80 °C.

Mounting space and boundary conditions for a sealing position

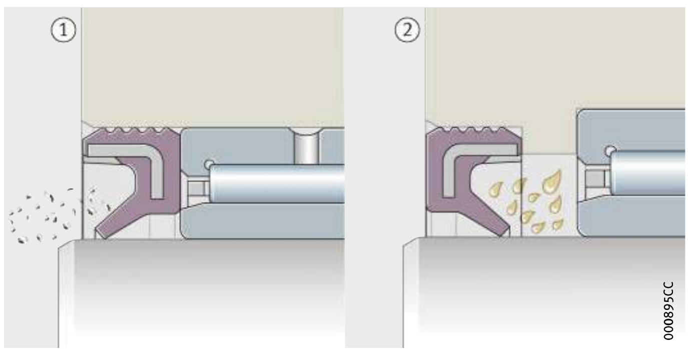
This section describes the mounting space and boundary conditions of sealing rings and rotary shaft seals (RWDR).

Mounting space

In order to achieve the optimum sealing action of a seal, the mounting space must in particular be modelled appropriately. This is carried out using, for example, DIN 3760 Rotary shaft seals and DIN 3761 Rotary shaft seals for vehicles. Design of the shaft and the bore at a sealing position is carried out in particular with the aid of DIN 3761-2. The data on mounting space that relate in this case only to rotary shaft seals, can also be carried over to sealing rings.

In general, the following fundamental rules apply:

- The adjacent construction should be designed such that the seal lips are not constrained in an axial direction.
- Sealing rings must be handled and fitted correctly. It is only in this way that long, problem-free sealing function is ensured.
- The mounting position of the seal lip must be observed, *Figure 2*.



① Seal lip facing outwards
② Seal lip facing inwards

Figure 2
Fitting in accordance with the function of the seal

Seal running surface

Seal running surfaces are an important factor for the life of a seal.

Characteristics of seal running surfaces

Seal running surface	Surface roughness	Minimum hardness
Sliding surface for radial seals (sealing for rotary motion)	Ra = 0,2 μm – 0,8 μm	600 HV or 55 HRC
	Rz = 1 μm – 4 μm	
	$Rz_{1\,max}$ ≦ 6,3 μm	
Sliding surface for rods and piston seals (sealing for axial motion)	Ra = 0,05 μm – 0,3 μm	600 HV or 55 HRC
	Rmr(0) 5% Rmr(0,25×Rz) 70%	
	$Rz_{1\,max}$ ≦ 2,5 μm	
Contact surfaces (static sealing)	R ≦ 1,6 μm	–
	Rz ≦ 10 μm	
	$Rz_{1\,max}$ ≦ 16 μm	

Guidelines for mounting

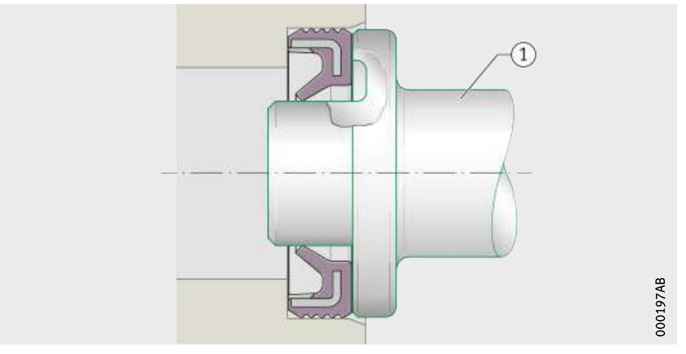
Irrespective of the type or form of the seal, it must always be ensured that it is not damaged during fitting. It must also be ensured in the mounting of directly sealed bearings that the prefitted sealing washer is not damaged or deformed in any way.

Mounting of seals

The axial adjacent construction should be designed such that the seal lips are not constrained in an axial direction.

Seals are fitted correctly as follows:

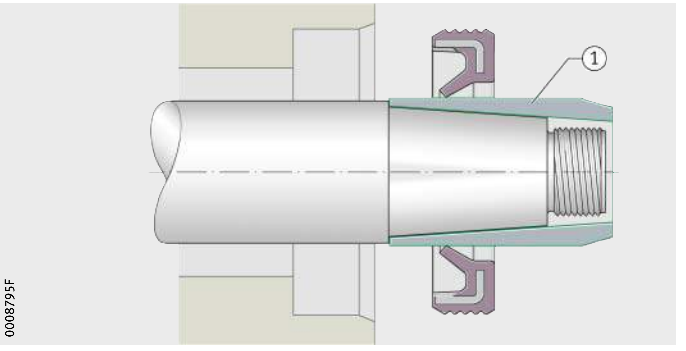
- A seal lip facing outwards protects the bearing against the ingress of dust and contamination, *Figure 2*, ①, page 54.
- A seal lip facing inwards prevents the egress of lubricant from the bearing, *Figure 2*, ②, page 54. In the case of sealing rings SD, the side with the protective lip is the marked side. It should be relubricated from inside, so the lip must face outwards.
- The running surface on the shaft and seal lip must be oiled or greased. This reduces the frictional energy during initial movement. In the case of sealing rings with an encased reinforcing ring – sealing ring G – the outside surface should be oiled before pressing in. This makes it easier to fit the seal in the housing.
- Press sealing rings carefully into the housing bore using a pressing device and a suitable pressing tool, *Figure 3*.



① Pressing tool

Figure 3
Fitting using a pressing tool

- Ensure that the seal lip is not damaged. Cover any sharp shaft edges, slots, teeth or threads by means of fitting sleeves, *Figure 4*.
- Fit sealing rings in such a way that the pressing-in force is applied as close as possible to the outside diameter. Sealing rings SD have an oversized outside diameter. This gives firm seating once the rings are pressed into the housing bore. The rings will adopt their correct geometrical form once fitted in the bore.



1 Fitting sleeve

Figure 4
Fitting using a fitting sleeve

- Fit sealing rings perpendicular to the shaft axis and the housing bore, *Figure 5*.

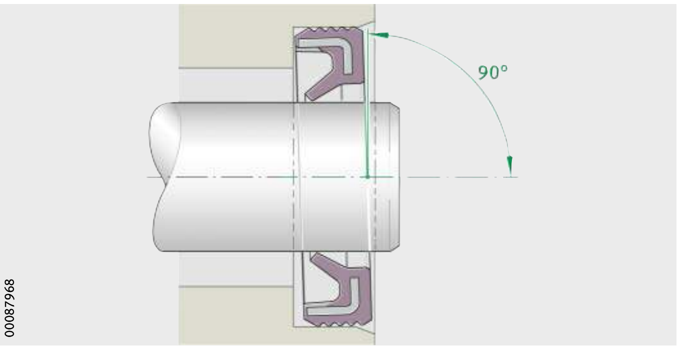


Figure 5
Perpendicularity –
position of sealing ring
relative to shaft axis/housing bore

Do not exceed the maximum perpendicularity between the sealing ring and the shaft axis once fitted, see table. A larger deviation will influence the sealing action.

- In the case of sealing rings SD, the space between the seal lip and protective lip must be filled with grease, *Figure 6*.
- After fitting, allow the sealing rings to run in and check the sealing function. Slight leakage (forming a grease or liquid film) is desirable in order to lubricate the contact surface for the seal lips.

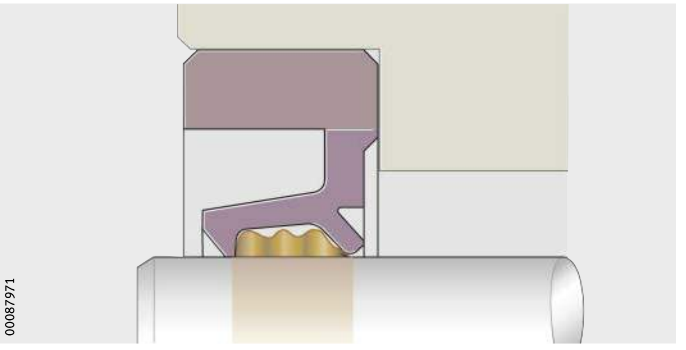


Figure 6
Grease filling between seal lip and protective lip

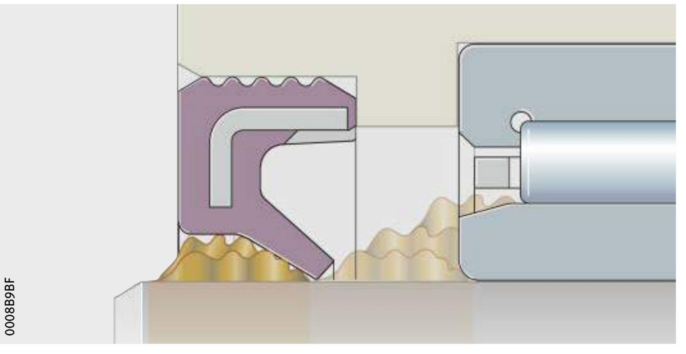


Figure 7
Grease collar for supporting sealing action

Maximum perpendicularity deviation

Shaft diameter d mm	Maximum deviation mm
d < 25	0,1
d ≧ 25	0,2

Seals

Fitting of O rings

For an O ring, correct placement in the groove is very important.

In order to prevent damage to the O ring during fitting, sharp edges should be avoided. A lead chamfer will not only eliminate a sharp edge but will also give easier pressing-in of the O ring. The lead chamfer should be in the range between 10° and 20°.

The following guidelines must be observed:

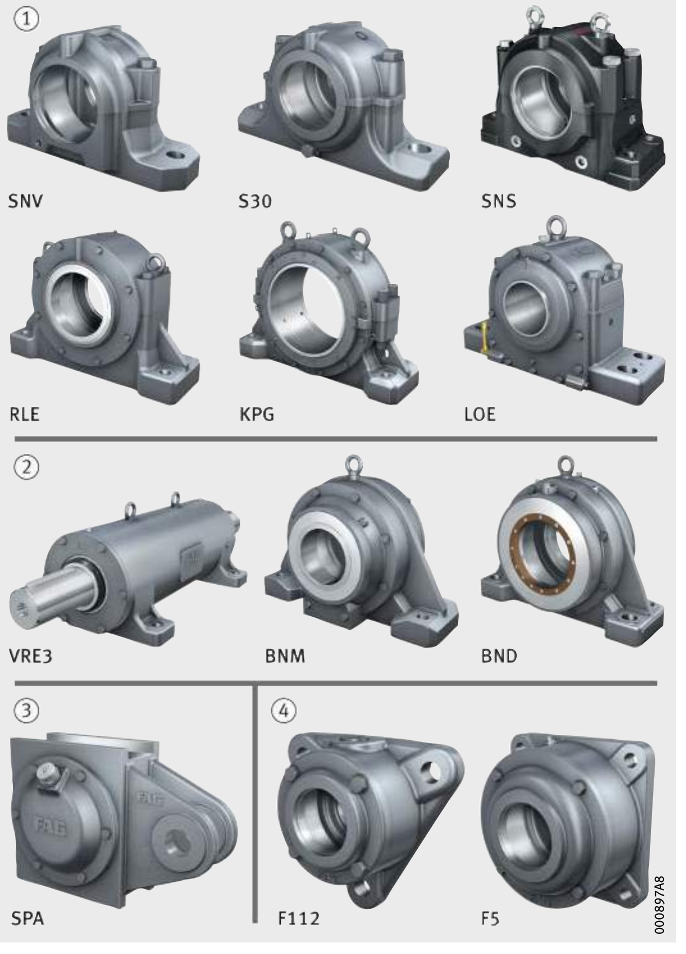
- Before fitting, the cord size and inside diameter of the O ring must be checked.
- The seal position must be clean and free from particles.
- The O ring must not be adhesive bonded in the groove under any circumstances. Alternatively, a fitting grease can be used if chemical compatibility has been established.
- During fitting, the O ring must not be forced over sharp edges, threads, grooves and undercuts.
- The use of sharp or pointed tools is not permitted.
- As a result of fitting, the O ring must not be elongated by more than 5% to 6%.
- During fitting, the inside diameter must not be stretched by more than 50%.
- It must be ensured that the O ring is not fitted in a rotated position during fitting.
- For the removal of an O ring, a special removal tool should always be used.

Removal of seals

Once a seal contact has been broken, for example in the removal of a cover or a rotary shaft seal, the seal must be replaced. Since the seal was already in seal contact due to initial fitting and underwent deformation as a result, the integrity of the seal cannot be ensured if it is used again. Furthermore, most seals are in any case heavily deformed or even destroyed during removal. During removal, it must be ensured that the seal running surface is not damaged.

Housing types

Housings are normally designed as plummer block housings (split or unsplit) or as flanged housings. However, a large number of special housings are also used across a wide range of different applications. They are made predominantly from flake graphite case iron or cast steel and, together with the associated bearing and seals, form a complete unit.



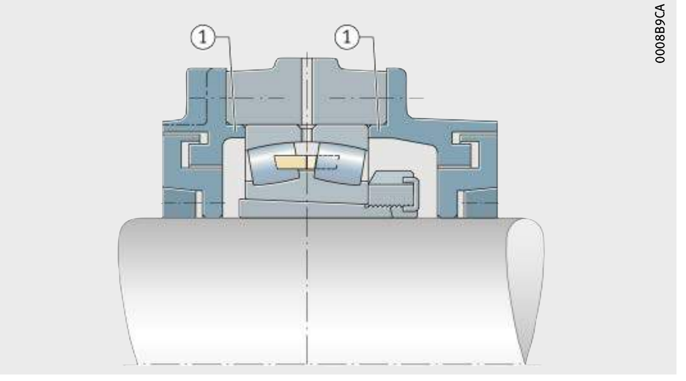
1 Split plummer block housings
2 Unsplit plummer block housings
3 Unsplit SPA housing
4 Flanged housings

Figure 1
Bearing housings

Housings in locating bearing design and non-locating bearing design

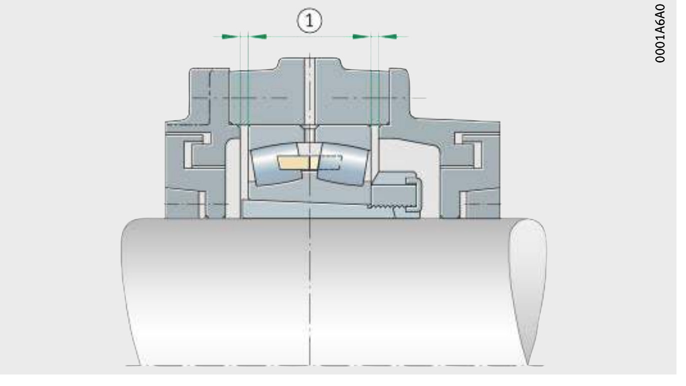
In this housing concept for the implementation of a locating or non-locating bearing arrangement, the housing must be ordered as necessary in a locating bearing design or a non-locating bearing design. This applies to the housings RLE, KPG, KPGZ, LOE, BNM, BND and SPA.

In the case of the locating bearing design, the bearings are axially clamped between the covers on the housings, *Figure 2*. In the case of the non-locating bearing design, the covers have shorter centring collars. As a result, the bearing can be axially displaced, *Figure 3*.



1 Centring collars on covers for axial location of the bearing

Figure 2
Housing
in locating bearing design



1 Bearing can be axially displaced

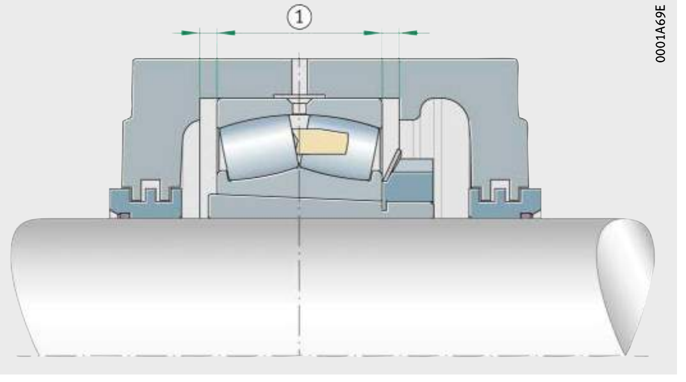
Figure 3
Housing
in non-locating bearing design

Housings with locating rings

In the case of many housings, the bearing seats are designed such that the bearing is capable of axial displacement and therefore acts as a non-locating bearing, *Figure 4*.

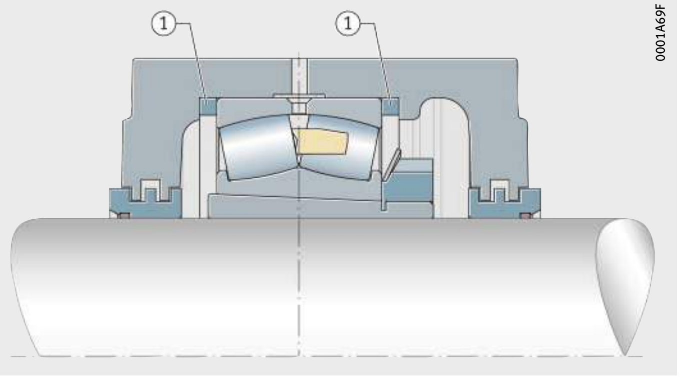
In this housing concept for the implementation of a locating bearing arrangement, a so-called locating ring is used, *Figure 5*. This applies to the housings SNV, S30, SNS and F5.

Once locating rings are inserted, the bearings are axially located. The locating rings are generally inserted in the housing on both sides of the bearing. Normally, an even number of locating rings is specified in order to achieve concentric seating of the bearing in the housing. In some cases, a single locating ring is sufficient.



1 Bearing can be axially displaced

Figure 4
Non-locating bearing arrangement, no locating bearing rings inserted



1 Locating rings give axial location of the bearing

Figure 5
Locating bearing arrangement, as a result of inserted locating rings

Housing seals

The rolling bearings normally used in bearing housings are spherical roller bearings, barrel roller bearings and deep groove ball bearings, which do not have their own sealing arrangement. The bearing position must therefore be sealed by means of the housing. In order to seal the housing against the shaft, contact seals, non-contact seals and combinations of these are available, depending on the operating conditions. These seals can also be ordered in a split or unsplit design.

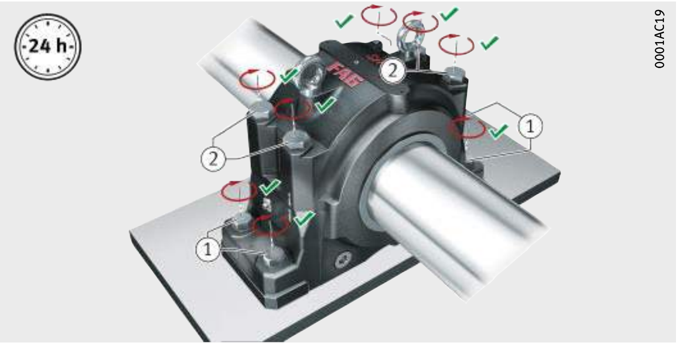
Mounting

For most series of housings from Schaeffler, mounting manuals are available. In some cases, there are also manuals relating to specific applications. Correct mounting has a decisive influence on the achievable bearing life.

Special features in the mounting of housings

In the mounting of housings, attention must be paid to the following:

- The mounting dimensions and critical dimensions must be checked before starting work on mounting.
- The upper and lower sections must not be transposed with parts of other housings.
- Before mounting, all lubrication holes must be cleaned.
- The screws must be dry and free of lubricants.
- A thin coating of mounting paste must be applied to the housing bore.
- In the case of split bearings, first the foot screws and then the cover screws must be tightened to the required torque.
- The specified maximum lubricant quantity must not be exceeded.
- After mounting, precise alignment and the tightening torque of the screws must be checked again and corrected as necessary, *Figure 6*.



① Foot screws
② Connecting screws

Figure 6
Checking of tightening torques

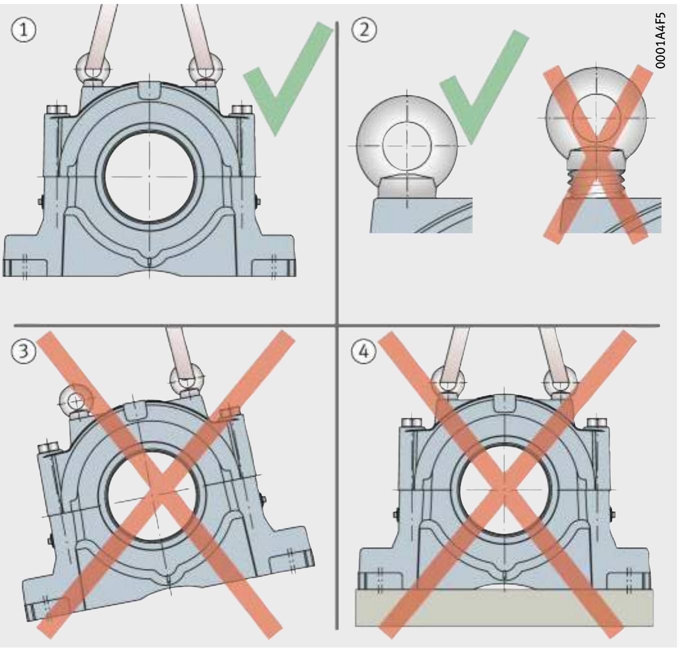
Eye bolts

On many housings, the housing body is provided with one or two eye bolts to DIN 580. These are intended as locating points for mounting and dismounting of the housing. The load carrying capacity of the eye bolts allows lifting of the housing including, in many cases, a bearing fitted in the housing, but without a shaft. Further relevant information is given in the description of the specific housing.

Correct usage of eye bolts on the housing body

Specifications for the use of eye bolts as locating points, *Figure 7*:

- Eye bolts must always be screwed fully into the housing.
- If several eye bolts are provided on the housing body, all the eye bolts must be used simultaneously as locating points.
- Only use eye bolts for lifting the housing and, if permitted for this housing, the bearing fitted in the housing. The eye bolts must not be subjected to additional load as a result of parts attached to the housing.



① Correct usage of eye bolts as locating points
② Screw in eye bolts completely
③ Always use all eye bolts simultaneously
④ Do not apply additional load as a result of attached parts

Figure 7
Correct usage of eye bolts on the housing body

Surface quality of the mounting surface

The requirements for the surface on which the housing is to be mounted are as follows:

- sufficiently robust to withstand the static and dynamic loads occurring in operation over the long term
- surface roughness Ra ≦ 12,5
- flatness tolerance to DIN EN ISO 1101 of IT7, measured across the diagonal
- free from colour.

A difference in level between the mounting surfaces of bearing housings will lead to misalignment of the shaft, *Figure 8*.

ψ = misalignment of shaft
h = difference in level between mounting surfaces

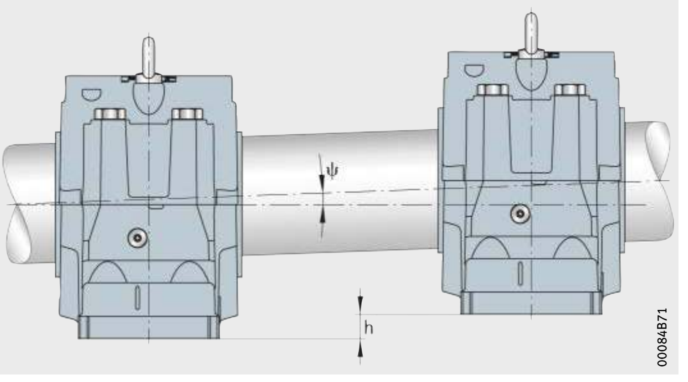


Figure 8
Misalignment of the shaft

The permissible misalignment is dependent on the housing and seal variant. Differences in level must be compensated such that the permissible misalignment is not exceeded. Levelling shims can be used for this purpose.

In addition, it must be ensured that the bearings mounted can compensate the misalignments present.

Tightening torques for connecting screws

In the case of split housings, the necessary tightening torque of the connecting screws for the upper and lower housing section must be determined in accordance with Schaeffler Catalogue GK 1, Bearing Housings. The tightening procedure should be carried out in stages and in a crosswise sequence.

Tightening torques for foot screws

Foot screws are used for screw mounting the housing to the mounting surface. They are not included in the scope of delivery of the housings.

The following table contains tightening torques for metric coarse pitch threads in accordance with DIN 13, DIN 962 and DIN ISO 965-2 as well as head contact dimensions in accordance with DIN EN ISO 4014, DIN EN ISO 4017, DIN EN ISO 4032, DIN EN ISO 4762, DIN 6912, DIN 7984, DIN 7990 and DIN EN ISO 8673.

The maximum tightening torques are valid with 90% utilisation of the yield stress of the screw material 8.8 and a friction factor of 0,14. We recommend that foot screws should be tightened to approx. 70% of these values, see table.

Tightening torques for foot screws with metric thread in accordance with DIN 13, DIN 962 and DIN ISO 965-2

Nominal screw diameter	Maximum tightening torque Nm	Recommended tightening torque Nm
M6	11,3	8
M8	27,3	20
M10	54	35
M12	93	65
M16	230	160
M20	464	325
M24	798	550
M30	1 597	1 100
M36	2 778	1 950
M42	3 991	2 750
M48	6 021	4 250
M56	9 650	6 750
M64	14 416	10 000
M72	21 081	14 500
M80	29 314	20 500
M90	42 525	29 500
M100	59 200	41 000

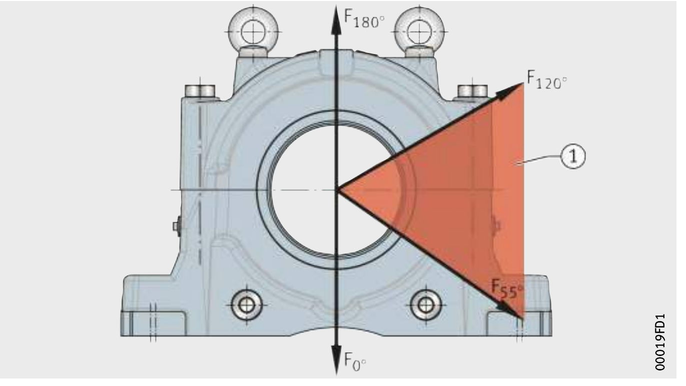
Horizontal location

In the case of plummer block housings, it may be necessary to supplement the foot screws by additional horizontal location of the housing.

Such horizontal location is necessary if one of the following conditions is fulfilled:

- The load angle is between 55° and 120°, *Figure 9*.
- Axial load is present.

Depending on the housing, the location may be implemented by means of stops in the load direction or pins.



1 Load angle range within which horizontal location of the housing is necessary

Figure 9
Load directions on a plummer block housing

Mounting of rolling bearings

Mounting methods
Mounting of special types

Mounting of rolling bearings

Mounting methods

Mounting of special types

Mounting

Due to the different types and sizes of rolling bearings, they cannot all be mounted using the same method. A distinction is made between mechanical, hydraulic and thermal methods.

In the mounting of non-separable bearings, *Figure 1*, the mounting forces must always be applied to the ring with a tight fit. Any forces applied to the ring with a loose fit would be transmitted by the rolling elements, which could cause damage to the raceways and rolling elements. Heating of the housing causes expansion of the bearing seat and thus makes the mounting process considerably easier.

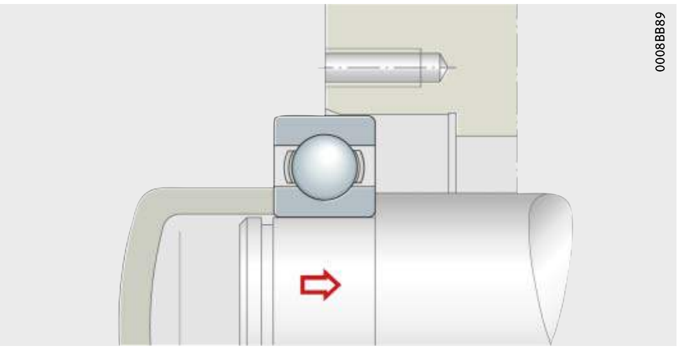


Figure 1
Mounting
of a non-separable bearing

In the case of separable bearings, *Figure 2*, mounting is simpler; both rings can be mounted individually. Rotating the ring during mounting gives a screwdriver effect that will help to avoid scraping marks.

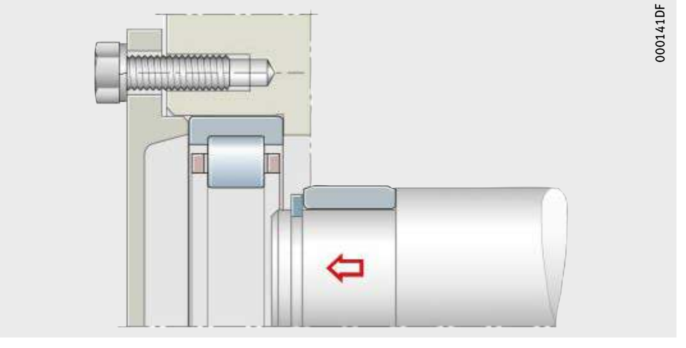


Figure 2
Mounting
of a separable bearing

Mechanical mounting

Smaller rolling bearings can often be mounted by purely mechanical means if the bearing seat is not too tight. It must be ensured, however, that the forces applied in this case do not cause damage to the bearings or their seating surfaces. The use of suitable tools and compliance with certain specifications is helpful in avoiding this.

Mounting of cylindrical seats

Bearings up to a bore diameter of approx. 80 mm can be pressed onto the shaft where a cylindrical seat is present. It is recommended that a mechanical or hydraulic press is used in this case, *Figure 3*.

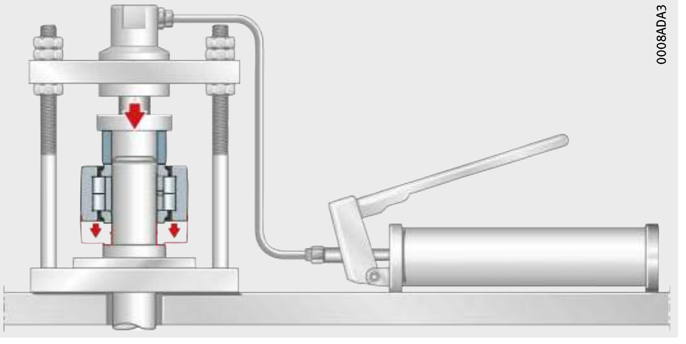


Figure 3
Mounting by means of a hydraulic press

If a press is not available, bearings with a bore diameter up to 50 mm can also be driven onto the shaft by means of light hammer blows if the fits are not too tight. Since the hardened bearing rings are sensitive to impact load, it is recommended that aluminium mounting sleeves and plastic mounting rings are used, in which case the mounting forces are transmitted by means of form fit. This method can also be used for the mounting of sleeves, intermediate rings, seals and similar parts, *Figure 4*.

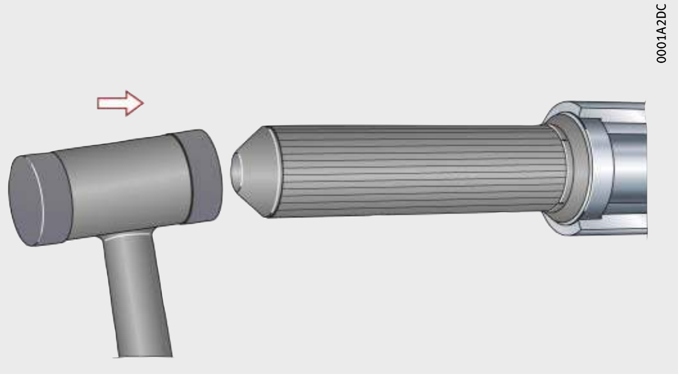
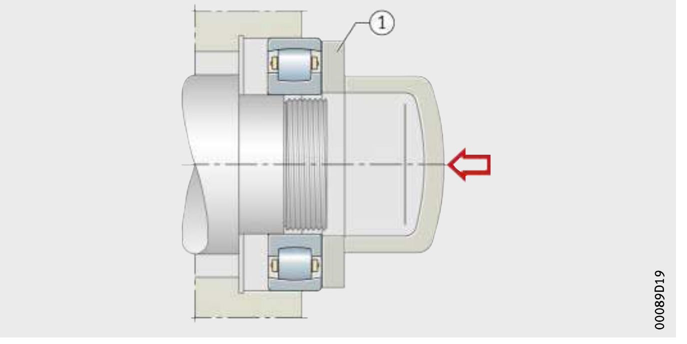


Figure 4
Mounting using mounting sleeve

When selecting the dimensions of the mounting sleeve or mounting ring, it must be ensured that the mounting forces are applied over the largest possible circumference but without creating any risk that the cage or rolling elements will be damaged.

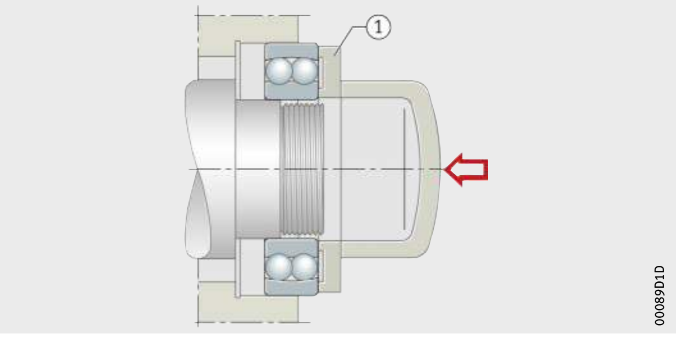
If a bearing is to be simultaneously pressed onto the shaft and into the housing, a disc must be used that is in contact with both bearing rings; this prevents tilting of the outer ring in the housing, *Figure 5*.



1 Mounting disc

Figure 5
Simultaneous pressing into place using mounting disc

In some bearings, the rolling elements or bearing cage project at the sides. In this case, a recess must be produced in the disc by means of turning, *Figure 6*.



1 Mounting disc

Figure 6
Pressing into place of self-aligning ball bearings with adapted mounting disc

If very tight fits are specified, even small bearings should be heated for mounting, see page 78.

In the case of housings made from light metal or with a press fit, the seating surfaces can be damaged if the outer ring is pressed into the housing bore. In this case, the housing must be heated.

Mounting of tapered seats

Bearings with a tapered bore are mounted either directly on the tapered shaft or journal or by means of an adapter sleeve or withdrawal sleeve on the cylindrical shaft.

Before mounting, the bearing bore and the seating surfaces on the shaft and sleeve must be cleaned. No mounting paste or similar lubricant should be used. A layer of lubricant would reduce the friction and thus facilitate mounting; in operation, however, the lubricant is gradually squeezed out of the fit joint. As a result, the tight fit is lost and the ring or sleeve begins to creep, causing fretting of the surfaces.

When the bearing is slid onto the taper, the inner ring is expanded and the radial internal clearance is thus reduced. The reduction in radial internal clearance is therefore valid as a measure of the seating of the inner ring.

The reduction in radial internal clearance is determined by the difference in the radial internal clearance before and after mounting of the bearing. The radial internal clearance must first be measured before mounting; during pressing onto the taper, the radial internal clearance must be checked continuously until the required reduction in internal clearance and thus the necessary tight fit is achieved, *Figure 7*.

Figure 7
Measurement of radial internal clearance using feeler gauges

In the case of sealed bearings, the radial internal clearance is not measured.

Instead of the reduction in radial internal clearance, the axial drive-up distance on the taper can be measured. In the case of the normal taper 1:12 of the inner ring bore, the drive-up distance corresponds to approximately fifteen times the reduction in radial internal clearance. The factor 15 takes into consideration that the interference of the fit surfaces acts only to the extent of 75% to 80% as expansion of the inner ring raceway.

If neither the reduction in radial internal clearance nor the drive-up distance can be reliably determined, the bearing should if possible be mounted outside the housing. The bearing may only be pressed into place so far that it can still be rotated easily and the outer ring can easily be swivelled by hand. The fitter must be able to sense when the located bearing still runs freely.

If a dismounted bearing is mounted again, it is not sufficient to move the retaining nut to its earlier position. After longer periods of operation, the fit loosens again since the thread undergoes settling and the seating surfaces become smoothed. The reduction in radial internal clearance, the drive-up distance or the expansion must therefore also be measured in this case.

In order to press the bearing onto the tapered seat or press in a withdrawal sleeve, mechanical or hydraulic presses are used. The type of mounting to be selected in the individual case is dependent on the mounting conditions.

Hook wrenches

Hook wrenches can be used to tighten and loosen locknuts (precision locknuts) on shafts, adapter sleeves or withdrawal sleeves, *Figure 8*. Hook wrenches can be used to mount small and medium-sized rolling bearings on tapered shaft seats, adapter sleeves or withdrawal sleeves. If no torque value is specified, jointed hook wrenches, jointed pin wrenches and jointed face wrenches can be used for locknuts and precision locknuts.

Figure 8
Hook wrench

Small bearings with an adapter sleeve are slid onto the tapered seat of the sleeve by means of the adapter sleeve nut and a hook wrench, *Figure 9*.

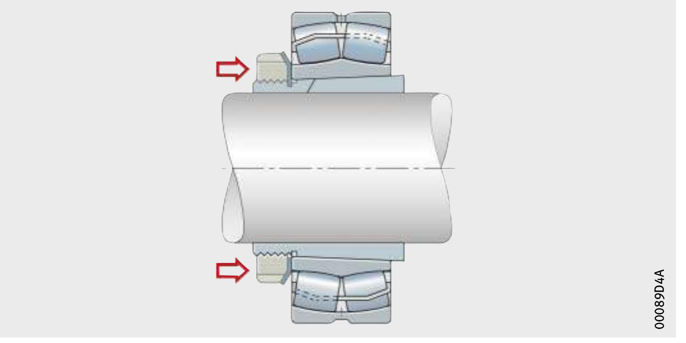


Figure 9
Pressing a spherical roller bearing onto an adapter sleeve using the adapter sleeve nut

Small withdrawal sleeves are pressed into the gap between the shaft and inner ring bore using a locknut, *Figure 10*.

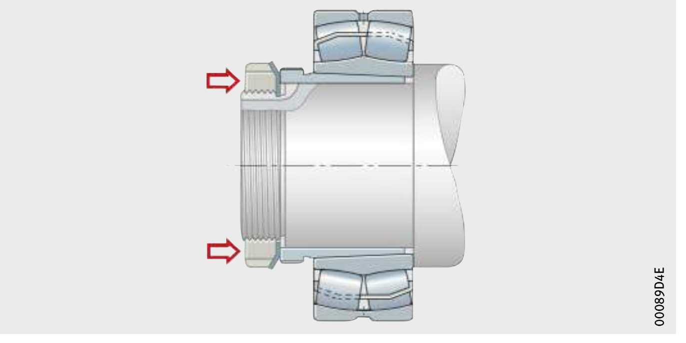


Figure 10
Pressing into place a withdrawal sleeve using the shaft nut

Double hook wrenches

Double hook wrenches are intended for the mounting of smaller self-aligning ball bearings and spherical roller bearings on adapter sleeves. They contain torque wrenches for precisely determining the initial mounting position before the bearing is pushed into place.

Each double hook wrench is engraved with torsion angles so that the drive-up distance and reduction in radial internal clearance can be precisely set, *Figure 11*.

Measurement of the radial internal clearance is difficult especially in the case of smaller self-aligning ball bearings and spherical roller bearings. If the bearing is mounted in a housing, it is not possible to measure the radial internal clearance in some cases.

As a result, measurement is often dispensed with and the radial internal clearance is estimated in approximate terms by means of the method normally used in the past. In this case, the rolling bearing is pressed onto the adapter sleeve until the outer ring can still be freely rotated and slight resistance is felt under swivelling.

With the method we recommend, the radial internal clearance can be set very accurately. The radial internal clearance is reduced in two stages. First, the locknut is lightly tightened to a specified tightening torque. This gives a precisely defined initial position and the radial internal clearance is then set very accurately in the second stage.

The locknut is then tightened by a defined angle. The radial internal clearance has now been reduced by the recommended 60% to 70%.

Figure 11
Mounting
by means of double hook wrench

Shaft nuts with pressure screws

In the case of larger bearings, considerable forces are required in order to tighten the nuts. In such cases, mounting is made easier by means of the shaft nut with pressure screws shown in *Figure 12*. A spacer ring should be inserted between the nut and sleeve in order to prevent damage to the sleeve.

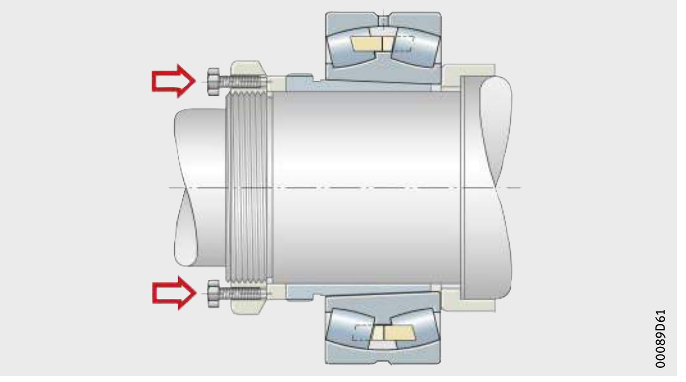


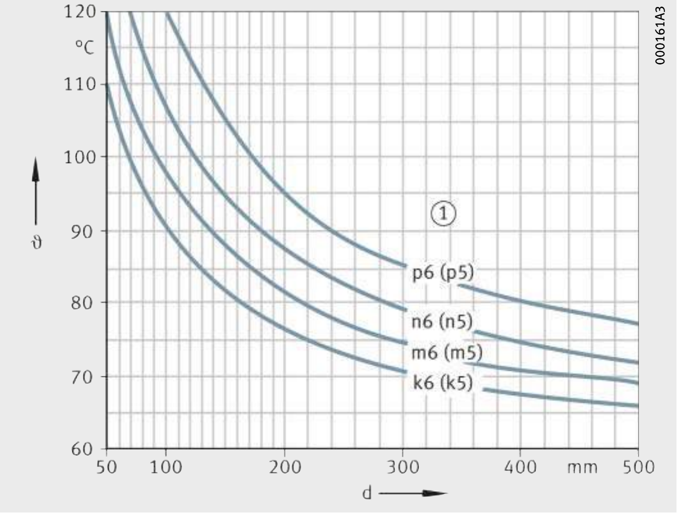
Figure 12
Mounting by means of shaft nut with pressure screws

In order to prevent tilting of the bearing or sleeve, the nut is first tightened only to the point where the nut and mounting ring are fully in contact. The pressure screws are made from quenched and tempered steel and uniformly distributed over the circumference – their quantity is based on the forces required – and are tightened uniformly in a circular sequence until the necessary reduction in radial internal clearance is achieved. Since the taper connection is self-locking, the device can then be removed and the bearing secured by means of its own retaining nut. The principle can also be applied to bearings that are located on an adapter sleeve or directly on a tapered journal.

For the mounting of large bearings, it is advisable to use hydraulic methods in order to slide the bearing into place or press in the sleeve. Further information on this procedure can be found on page 83.

Thermal mounting

Bearings with a cylindrical bore should be heated before mounting if a tight fit on the shaft is intended and the effort required for pressing into place by mechanical means is too great. The temperature required for mounting is shown in *Figure 13*. The data are valid for maximum fit interference, a room temperature of +20 °C and an excess temperature safety margin of 30 K.



ϑ = heating temperature
d = bearing diameter

① Shaft tolerance

Figure 13
Temperature

When heating the bearings, the temperature must be precisely monitored. An excessive temperature differential between the individual components can lead to distortion with the bearing and thus to damage. In the case of all non-separable rolling bearings, such as spherical roller bearings, it must be noted that the radial internal clearance present is taken up relatively quickly by the temperature difference between the bearing components and the rolling elements may be pressed into the raceway of the colder components. In addition, the heating must not normally exceed +120 °C, in order to prevent changes to the structure and hardness of the bearing. Furthermore, the maximum temperature of the preservative must be observed.

Bearings with cages made from glass fibre reinforced polyamide and bearings that are sealed or have already been greased can be heated to max. +80 °C but not, however, in an oil bath.

After heating, the parts are slid in a single movement, rapidly and without tilting, up to the stop on the seat. While sliding onto the shaft, slight rotation to give a screwdriver effect helps to achieve prompt mounting. Protective gloves should be worn when mounting the heated parts, *Figure 14*.

Where tight fits are present in the housing, in other words where there is circumferential load on the outer ring, the housing can also be heated for the purposes of support.

Figure 14
Sliding into place
of heated bearing parts

Once the inner ring has been slid in place, it must immediately be secured against its axial abutment point and held under tension until it has cooled, so that it then remains in full contact. There must not be any gap between two rings positioned adjacent to each other.

Induction heating device

Rolling bearings can be brought to mounting temperature quickly, safely and above all cleanly by the use of induction heating devices operating by the transformer principle. The devices are used mainly in batch mounting work.

The heating devices can be used to heat rolling bearings of all types, including greased and sealed bearings. The smallest heating device is used for bearings with a bore of 10 mm or larger, *Figure 15*. The maximum bearing mass for the heating device shown here, for example, is 50 kg.

Figure 15
Small heating device

The operating range of the largest heating device starts at a bore of 90 mm, *Figure 16*. The heaviest workpiece mass can be up to 1 600 kg.

Figure 16
Large heating device

After the heating process, the bearing is automatically demagnetised. Further details on induction heating devices are given in TPI 200, FAG Heating Devices for Mounting of Rolling Bearings.

Heating plate

Heating plates are used to heat rolling bearings or small machine parts by means of contact heat. It must be ensured, however, that the entire bearing is heated uniformly.

A ring or disc is placed between a heating plate without temperature control and the inner ring of a bearing with a polyamide cage.

Oil bath

With the exception of sealed, greased bearings and high precision bearings, rolling bearings of all sizes and types can be heated in an oil bath. For heating, a clean oil with a flash point above +250 °C must be used. A thermostatic controller is advisable (temperature +80 °C to +120 °C). In order that the bearings are heated uniformly and no deposits of contamination occur within them, they should be laid on a grid or suspended in the oil bath. After heating, the oil must be allowed to drain off thoroughly and all fit and locating surfaces must be carefully wiped.

!

With this method, please note the risk of accidents, environmental pollution by oil vapour and flammability of hot oil.

Heating cabinet

Safe, clean heating of rolling bearings can be carried out in a heating cabinet. The temperature is controlled by means of a thermostat and is therefore precisely maintained. There is almost no possibility of bearing contamination. The disadvantage is that heating by means of hot air takes a relatively long time and is comparatively intensive in terms of time and energy.

Medium frequency technology

With the aid of FAG devices using medium frequency technology, it is possible to heat very large and heavy bearings as well as other components of shrink fit connections by inductive means in order to achieve joining and loosening. Due to its compact construction, the device can also be used for mobile operation.

The heating device comprises the medium frequency generator and an inductor, *Figure 17*, page 82. Depending on the application, this can be of a flexible or rigid design. The flexible version is similar to a cable that is placed either within the bore or on the outside of the workpiece. Flexible inductors are suitable for workpieces of various sizes and various shapes and can be used for long periods, depending on their design, at heating temperatures up to +180 °C or +300 °C.

Figure 17
FAG medium frequency heating device

In batch production applications, where large quantities of identical components are mounted, flexibility is less important than reduced setup times and increased process reliability. Rigid inductors are suitable for this task. In this design, the coil is fitted in a housing matched to the workpiece and can thus be placed quickly and easily in the heating zone. Rigid inductors are also suitable, in contrast to the flexible variant, for small components.

! In the heating of non-separable bearings, the outer ring must be heated first so that the internal clearance is maintained and damage to the bearing is prevented.

! The devices are designed for the specific application. Please contact the application experts at Schaeffler.

Advantages

- Versatile application possibilities by means of flexible inductors.
- Ease of transport – usable anywhere.
- Rapid, energy-efficient operation.
- Short heating times and high productivity.

Further information

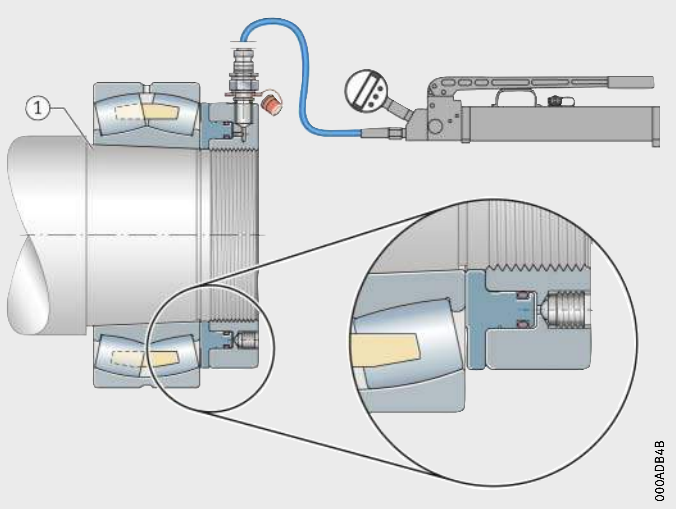
- TPI 217, Induction Units with Medium Frequency Technology.

Hydraulic mounting

Hydraulic tools can be used to apply large forces. These methods are therefore particularly suitable for the mounting of large bearings with a tapered bore. Hydraulic nuts are used as a mounting tool. Pressure can be generated using oil injectors, hand pumps or hydraulic units.

Hydraulic nuts

Hydraulic nuts are used to press components with a tapered bore onto their tapered seat, *Figure 18* and *Figure 19*, page 84.
These tools are mainly used if the drive-up forces required cannot be applied using other accessories, e.g. shaft nuts or pressure screws. Hydraulic nuts comprise a press ring and an annular piston.
Depending on size, the nut thread is a metric precision thread or a trapezoidal thread. Inch size designs are also available.
The oil connector, irrespective of size, is always designed as G$^{1}/_{4}$.
The necessary drive-up distance is checked by means of a dial gauge.



① Mounting on a tapered seat

Figure 18
Mounting
of a spherical roller bearing
using a hydraulic nut

Mounting methods

① Pressing into place on an adapter sleeve

② Pressing into place of a withdrawal sleeve

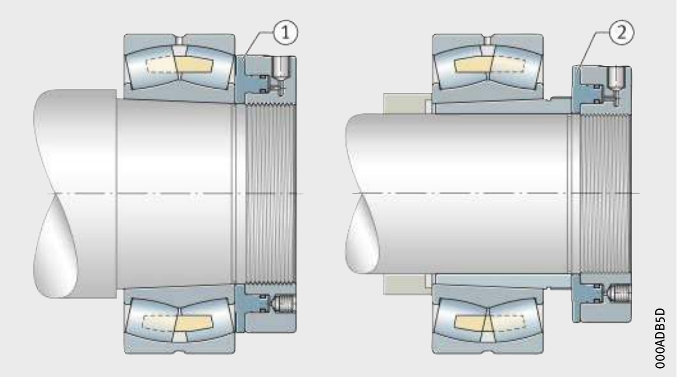


Figure 19
Mounting of a spherical roller bearing using a hydraulic nut

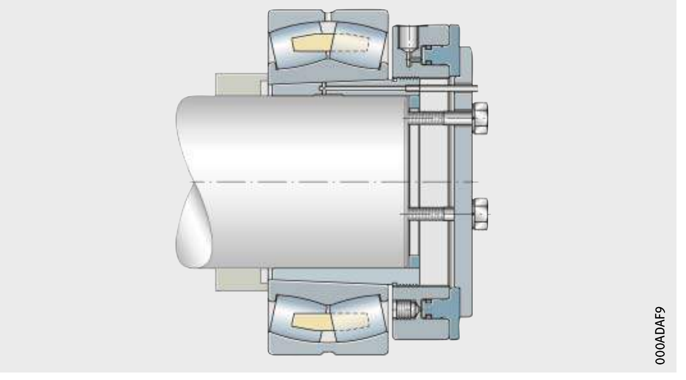


Figure 20
Mounting of a spherical roller bearing with a withdrawal sleeve and pressure plate (using the hydraulic method)

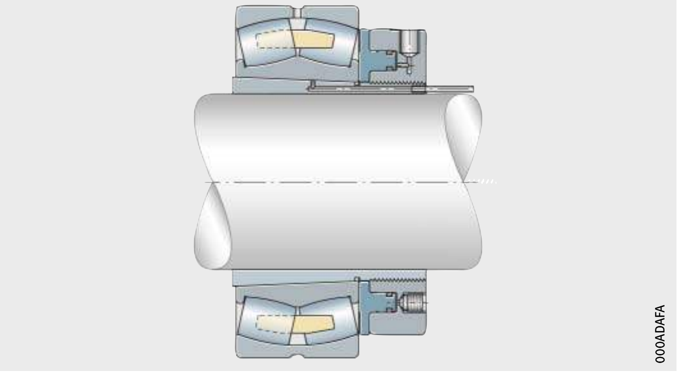


Figure 21
Mounting of a spherical roller bearing with a withdrawal sleeve and support ring (using the hydraulic method)

Oil pressure method

In the oil pressure method, oil is pressed between the fit surfaces, *Figure 22*. This method is particularly suitable for the mounting of large bearings with a tapered bore on a conical shaft or on an adapter or withdrawal sleeve. The oil film substantially neutralises the contact between the fit parts, so they can be displaced relative to each other with the application of little force and without the risk of surface damage. Fretting corrosion is loosened by means of paraffin or rust-dissolving additives in the oil.

① Cylindrical seating surface
② Tapered seating surface

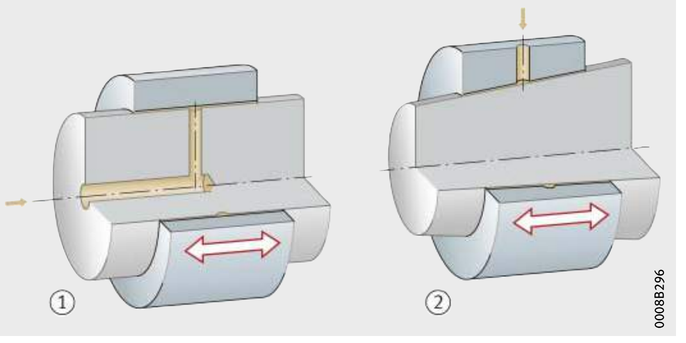
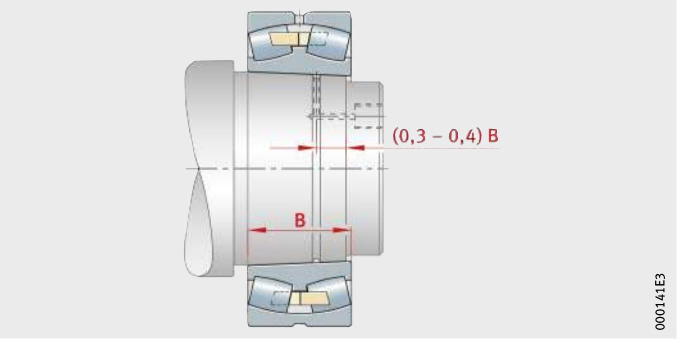


Figure 22
Principle of hydraulic mounting: creation of a fluid film between the fit surfaces

In both cases, oil grooves and feed ducts as well as threaded connectors for the pressure generation devices must be provided. The width of the oil groove is dependent on the bearing width, *Figure 23*, page 86. Further design guidelines are given in the FAG publication WL 80102, Hydraulic Method for the Mounting and Dismounting of Rolling Bearings. Hydraulic nuts are used as a mounting tool. Pressure can be generated using hand pumps and hydraulic units. In isolated cases, oil injectors can also be used.

A mounting plate prevents damage to the sleeve or bearing ring. When pressing in the withdrawal sleeve, *Figure 25*, page 86, the oil connector is guided by the shaft nut. The drive-up distance of the bearing or withdrawal sleeve is determined in accordance with the necessary reduction in radial internal clearance. In order to measure the radial internal clearance, the bearing must be disconnected from the oil pressure.

Once the oil pressure has been disconnected, it will take between 10 minutes and 30 minutes until the oil has completely escaped from the fit joint. During this time, the axial preload must continue to act. After this point, the mounting device (nut with pressure screws or hydraulic nut) is removed and the shaft or sleeve nut is screwed into place and secured.



B = bearing width

Figure 23
Recommended width for oil grooves

Tapered shaft

If the bearing is seated directly on a tapered shaft, the oil is pressed between the fit surfaces, while the bearing is simultaneously pressed onto the taper by means of screws or a nut. The reduction in radial internal clearance or the axial drive-up distance is measured at this time, *Figure 24*.

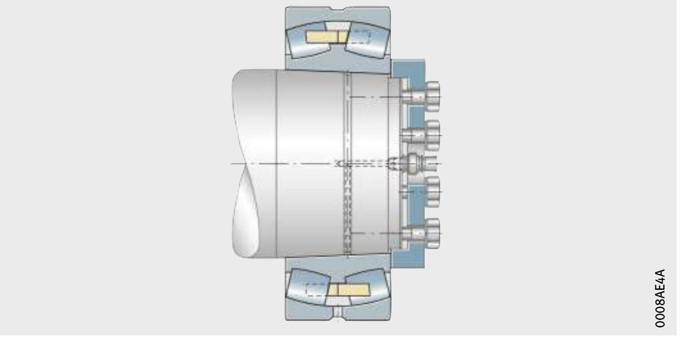


Figure 24
Bearing seat
on the shaft

Withdrawal sleeve

If the bearing is seated on the withdrawal sleeve, the oil is pressed between the fit surfaces, while the withdrawal sleeve is pressed into the bearing bore by means of screws or a nut. The oil is fed through the shaft nut. The reduction in radial internal clearance is measured at this time, *Figure 25*.

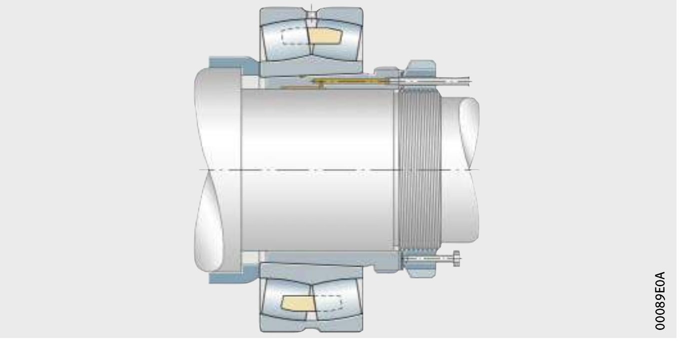
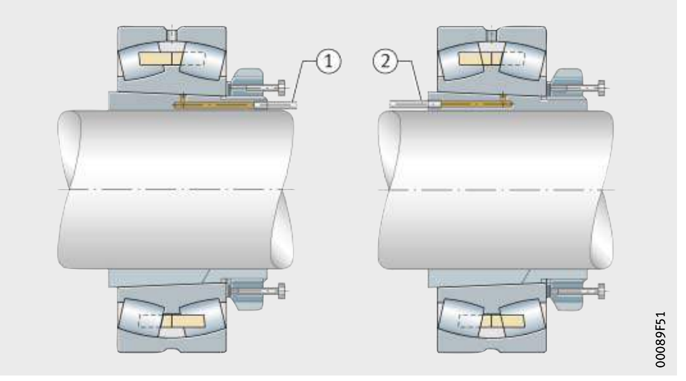


Figure 25
Bearing seat
on the withdrawal sleeve

Adapter sleeve

If the bearing is seated on the adapter sleeve, the oil is pressed between the fit surfaces, while the bearing is pressed onto the adapter sleeve by means of screws or a nut. The reduction in radial internal clearance is measured at this time, *Figure 26*.



① Oil connector on threaded side
② Oil connector on taper side

Figure 26
Bearing seat
on the adapter sleeve

Hand pump

In hydraulic mounting, the oil pressure is normally built up by means of a hand pump, *Figure 27*.

A machine oil of moderate viscosity is used as a pressure fluid. For mounting, it is recommended that the oil used should be as thin-bodied as possible and have a viscosity of ≈ 75 mm^2/s at +20 °C (nominal viscosity 32 mm^2/s at +40 °C), in order that the oil can easily escape from the fit joint without leaving a residue after mounting.

Figure 27
FAG hand pump set

Mounting of special types

Features

Selection of the suitable mounting method is based not only on the bearing type but also on the adjacent construction and the relevant dimensions, see page 194. In the case of some rolling bearing types, attention must be paid during mounting to particular features or a particular procedure must be applied, which is discussed in detail below. Further details are given in the product-specific catalogues and brochures. The decisive factor for correct mounting is, however, the mounting manual relating to the application.

Mounting of angular contact ball bearings and tapered roller bearings

Angular contact ball bearings and tapered roller bearings are always mounted in pairs. The axial internal clearance and thus also the radial internal clearance of two bearings adjusted against each other is set at the time of mounting. The magnitude of the internal clearance or preload is based on the operational requirements. Angular contact ball bearings of the universal design can be mounted directly adjacent to each other in any arrangement required.

High loads and high speeds lead to an increase in temperature in the bearing position. As a result of thermal expansion, the internal clearance set at the time of mounting may change during operation. Whether the internal clearance will increase or decrease depends on the arrangement and size of the bearings, the materials of the shaft and housing and the spacing between the two bearings.

If the shaft requires the closest possible guidance, the internal clearance is set in steps. A test run must be carried out after each new adjustment and the temperature must be checked.
This ensures that the internal clearance does not become too small, leading to an excessive increase in running temperature.
During the test runs, the bearing arrangement will “settle” such that the internal clearance undergoes hardly any further change.

The guide value for the correct bearing temperature at moderate to high speed and moderate load can be taken as follows: if there is no heating due to an external source, a correctly adjusted bearing arrangement may reach a temperature of approx. +60 °C to +70 °C during the test run; the temperature should decrease after between two and three hours of operation, however, especially in the case of grease lubrication, once the excess grease has been driven out of the bearing interior and the churning work has decreased.

Bearings exposed to vibrations at low speed, are mounted free from clearance or even with preload, otherwise there is a risk that the rolling bearings will impact against the roller raceways. Angular contact ball bearings and tapered roller bearings are adjusted against each other on the shaft by means of locknuts, *Figure 1*, shims, *Figure 2*, or ring nuts in the housing.

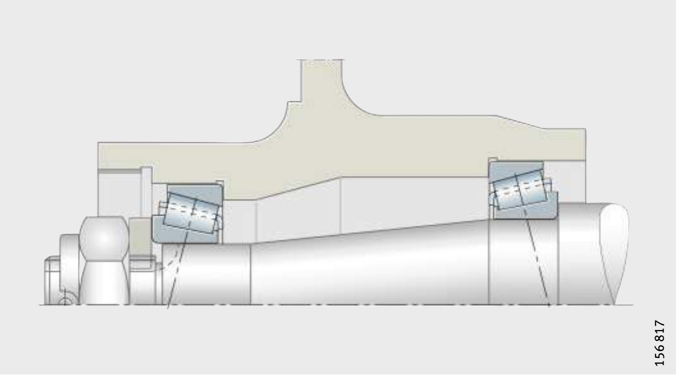


Figure 1
Adjustment of tapered roller bearings of a freewheel using the kingpin nut

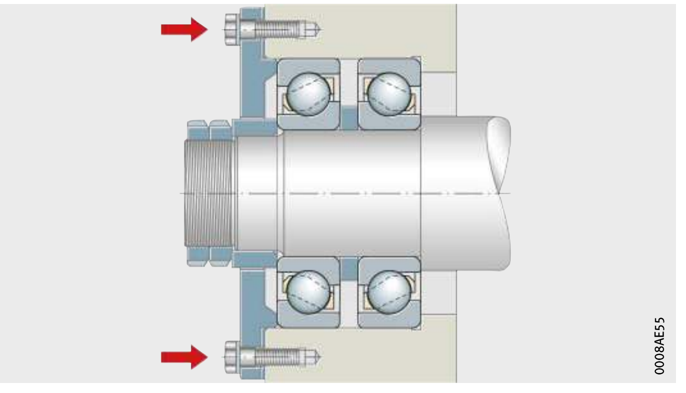


Figure 2
Axial location of an angular contact ball bearing pair – setting of internal clearance using shim

The axial internal clearance or preload of an adjustable bearing arrangement is set – starting from a clearance-free state – by loosening or tightening of the locknut or by insertion of calibrated plates. The axial internal clearance and preload can be converted into revolutions of the locknut with the aid of the thread pitch.

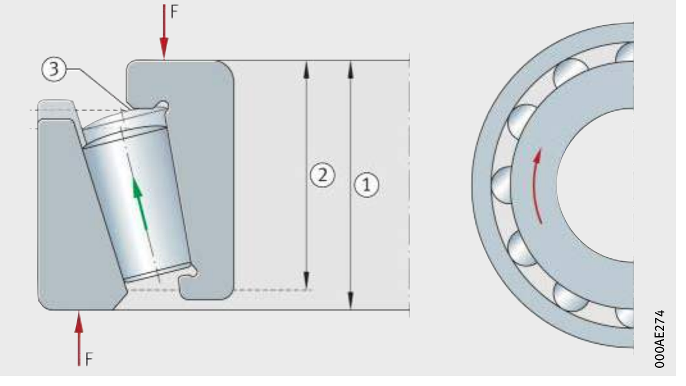
Mounting of special types

The transition from internal clearance to preload is sought during the adjustment process by rotating the shaft continuously by hand and simultaneously checking the possible movement of the shaft by means of a dial gauge.

The correct setting can be found more easily using a torque wrench. The locknut is tightened to the specified torque as a function of the bearing size. The necessary internal clearance is then achieved by reversing the locknut by approx. $^1/_{12}$ of a revolution.

In the case of tapered roller bearings, it must be ensured that the rolling elements are in contact with the guide rib during mounting. This prevents any increase in the operating clearance of the rolling bearing due to subsequent settling effects. In order to achieve this, the bearing must be screwed in by several revolutions during mounting. As a result, the rolling elements creep out of their undefined initial position into their required contact position on the guide rib. This process is known as “screwing in”. After this process, contact between the rollers and the rib must be checked, for example with the aid of a feeler gauge.

Tilting of the rings relative to each other must be avoided.



① Section height before screwing in
② Section height after screwing in
③ Gap

Figure 3
Screwing-in process

In the case of matched pairs and multiple-row tapered roller bearings, *Figure 4* and *Figure 5*, the axial internal clearance is determined by the width of the intermediate ring.

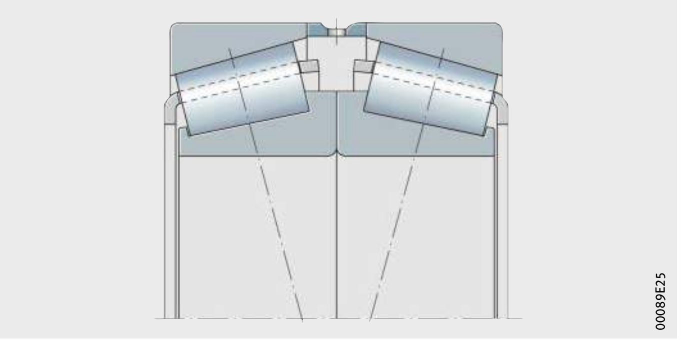


Figure 4
Matched tapered roller bearings in X arrangement (suffix N11CA)

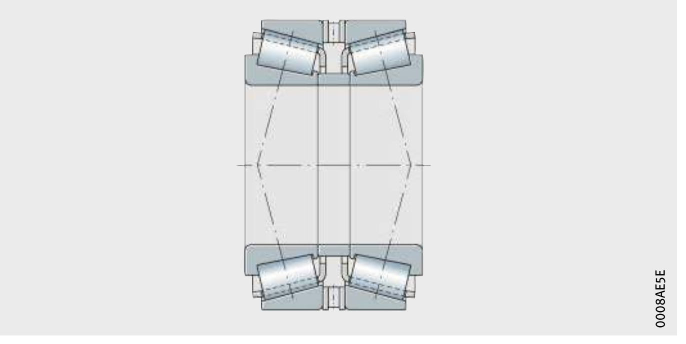
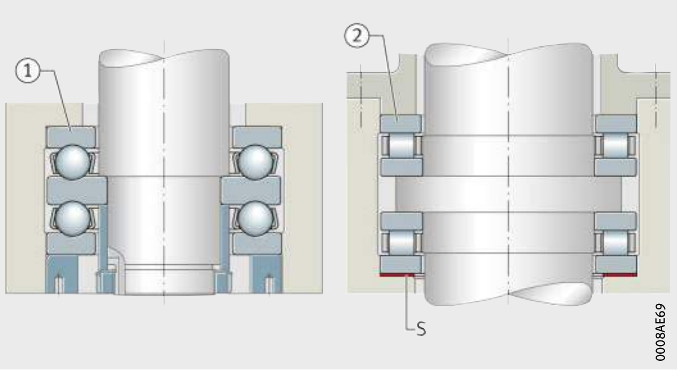


Figure 5
Double row tapered roller bearing in O arrangement

Mounting of axial bearings

In the case of axial bearings, the shaft locating washers have a transition fit normally and a tight fit only in exceptional cases, while the housing locating washers always have a loose fit.
In double direction axial bearings, the central washer is axially clamped, *Figure 6*.



① Double direction axial deep groove ball bearing adjusted free from clearance
② Axial cylindrical roller bearing preloaded by means of shim S

Figure 6
Axial bearings adjusted free from clearance

Mounting of machine tool bearing arrangements

In the case of machine tool spindles, setting the correct internal clearance is particularly important since the quality of workpieces produced on the machine are dependent on this factor. In order that the operating clearance or preload required by the designer can be precisely set during mounting of the bearings, Schaeffler has developed its own gauges.

Super precision bearings

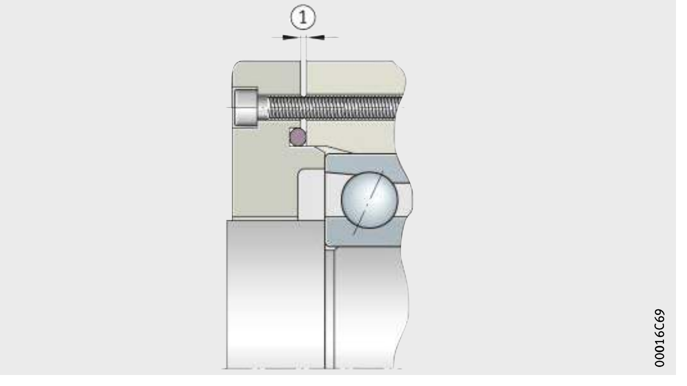
Super precision bearings include:

- spindle bearings
- super precision cylindrical roller bearings
- axial angular contact ball bearings.

Matching operations

In order to maintain optimum performance or achieve precise positioning of the spindle in relation to the housing, it is often necessary to carry out special matching operations on the components. This applies, for example, to the covers used to clamp the bearings. Before clamping, a gap should be present, *Figure 7*.

Matching of the intermediate rings may be advisable in the case of high speed spindles, in order to compensate the influence of fit and ring expansion on the preload.



1 Gap before tightening of end cover screws

Bearing bore d $\leqq$ 100 mm: 0,01 mm to 0,03 mm

Bearing bore d $>$ 100 mm: 0,02 mm to 0,04 mm

Figure 7
Matching of end cover (recommendation)

Greasing

The preservative applied to FAG super precision bearings is such that it is not necessary to wash out the bearings before greasing. The setting of the grease quantity places high requirements on the greasing and measurement equipment used. It is recommended that bearings already greased and sealed from Schaeffler are used.

Greasing must be carried out under extremely clean conditions.

In the case of bearings with grease lubrication, a grease distribution cycle must be carried out on the bearings before the test run on the spindle.

Axial clamping of inner rings

Spindle bearing sets are generally clamped on the shaft using shaft nuts. Nuts with axial bores are to be used in preference over locknuts for tightening on the shaft, since they minimise the air turbulence that occurs at high speeds.

The contact faces of the nuts should be ground in a single clamping operation together with the thread. The recommended maximum axial runout tolerance is 2 μm.

In order to prevent impairment of the runout during the clamping operation, the clamping inserts should be ground together with the thread and the axial face.

Values for axial clamping of inner rings on the shaft using a precision nut are given in Catalogue SP 1, Super Precision Bearings.

In order to eliminate or reduce settling effects, the nut should first be tightened to three times the stated torque, loosened and then finally tightened to the nominal torque. The retaining screws should then be fully tightened in accordance with the manufacturer's data.

Mounting procedure for cylindrical roller bearings

Cylindrical roller bearings with a tapered bore are mounted with clearance, clearance-free or with preload.

The precise procedure for the mounting and dismounting of super precision bearings is given in the relevant mounting and maintenance manual for the specific bearing and in Catalogue SP 1, Super Precision Bearings.

Mounting of special types

Mounting of rotary table bearings

Axial/radial bearings as well as axial angular contact ball bearings are ready-to-fit high precision bearings for high precision applications with combined loads. They can support radial loads, axial loads from both sides and tilting moments without clearance and are particularly suitable for bearing arrangements with high requirements for running accuracy, such as rotary tables, face plates, milling heads and reversible clamps.

Mounting of these units is very simple due to the fixing holes in the bearing rings. The bearings are radially and axially preloaded after mounting.

High precision bearings for combined loads include:

- axial/radial bearings YRT, RTC, $\text{YRT}_{\text{Speed}}$
- axial angular contact ball bearings ZKLDF
- axial/radial bearings YRT with integral angular measuring system YRTM.

Further information

- TPI 103, High Precision Bearings for Combined Loads, Mounting and Maintenance Manual
- MON 36, Series YRTSM and YRTM
- MON 20, High Precision Bearings for Combined Loads, Mounting and Maintenance Manual.

Mounting of screw drive bearings ZKLF, ZKLN, ZKRN, ZARF, ZARN

Bearings for screw drives include:

- double row axial angular contact ball bearings for screw mounting ZKLF
- double row axial angular contact ball bearings not for screw mounting ZKLN
- single row axial angular contact ball bearings BSB, 7602, 7603
- angular contact ball bearing unit TZKLR
- double row and triple row axial angular contact ball bearings ZKLFA, DKLFA
- needle roller/axial cylindrical roller bearings for screw mounting DRS, ZARF
- needle roller/axial cylindrical roller bearings not for screw mounting ZARN.

Mounting of these bearings is described in detail in TPI 100, Bearings for Screw Drives.

Mounting of toroidal roller bearings

For toroidal roller bearings, the procedure is fundamentally the same as for other standard bearings. An overview of the methods and tools recommended as a function of the bearing diameter is given in the table Mounting and dismounting methods for rolling bearings, page 194, in this Mounting Handbook. Further details are given below on recommendations for mounting and dismounting.

Measurement of radial internal clearance

The tight, tapered fit of a ring is often determined by the change in the radial internal clearance. Before and after mounting of the bearing, the radial internal clearance of the bearing must be determined by means of feeler gauges. It must be ensured in this case that the two bearing rings are concentrically aligned to each other. The operating clearance required is generally set by means of the axial displacement of the two rings relative to each other.

Free space on end faces and mounting dimensions

In the axial location of FAG toroidal roller bearings, the degrees of freedom in relation to axial displacement and tilting must be taken into consideration. Any possible contact with retainers or the bearing environment must be avoided. On the one hand, the requisite value for the depth of the free space $C_{a\ req}$ must be maintained, which ensures axial displacement of the shaft in the housing, *Figure 8*.

$$C_{a\ req} = C_a + 0{,}5 \cdot \left(\delta_{ax} + s_\varphi\right)$$

$C_{a\ req}$ mm
Requisite value for the depth of the free space

C_a mm
Minimum value for depth of free space in the case of bearing rings without offset, see TPI 232, Toroidal Roller Bearings TORB

δ_{ax} mm
Change in shaft length due to temperature

s_φ mm
Reduction in axial displacement facility as a result of tilting.

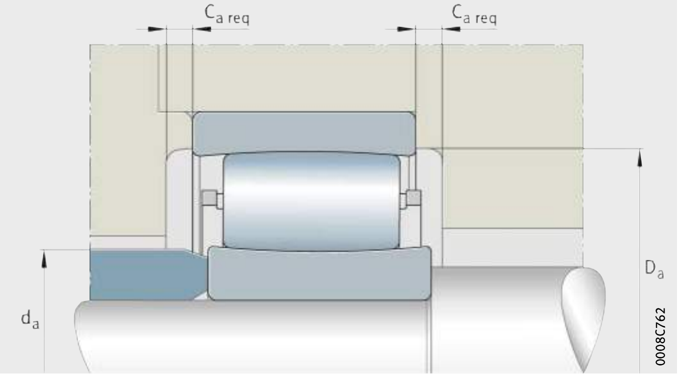


Figure 8
Free spaces in the housing

On the other hand, the appropriate mounting dimensions D_a and d_a must be observed. In those cases where the axial retainers or mounting nuts on the outside diameter do not match the specified mounting dimensions, the use of spacer nuts is necessary.

Mounting of special types

Axial positioning of the bearing

Toroidal roller bearings are normally mounted with the inner and outer ring centred relative to each other, where these are used to set the required internal clearance. Since the radial internal clearance in the bearing is reduced by axial displacement of the rings relative to each other, the required radial clearance can be achieved by offset of the rings. More detailed information on calculation of the reduction in radial internal clearance is given in TPI 232, Toroidal Roller Bearings TORB. Larger axial displacements, which are caused by large temperature fluctuations or other influences, must be counteracted by positioning of the rings relative to each other during mounting. In the case of oscillating bearing arrangement systems, it must be ensured that the axial displacement caused by the oscillation always occurs on the same side relative to the centre of the bearing. Crossing the centre of the bearing is only permissible when starting up an application.

Guidelines for mounting

During mounting, it must be ensured that the two bearing rings are not offset relative to each other. Horizontal mounting is recommended in all cases. If vertical mounting is absolutely necessary, appropriate devices must be used that hold the two bearing rings concentric to each other.

If the bearing is mounted both on a shaft and in a housing at the same time, the mounting pressure must be applied via both the bearing inner ring and the bearing outer ring, in order to prevent tilting.

Mounting of TAROL bearings

The tapered roller bearing units TAROL are used for the bearing arrangements of wheelsets on rail vehicles such as freight wagons and passenger carriages, *Figure 9*. These are compact, ready to fit, greased, sealed and axially adjusted rolling bearings that are pressed onto the shaft journal in a single operation. If the shaft journal diameter is within the specified tolerance, the press fit of the bearing will give the required axial internal clearance.

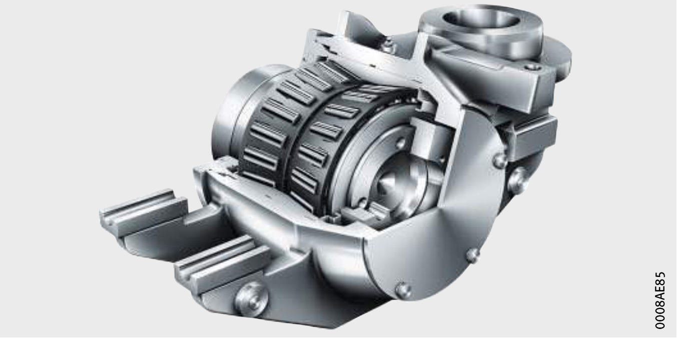


Figure 9
Wheelset bearing for rail vehicles

For the mounting and dismounting of these bearings, Schaeffler recommends the use of a mobile hydraulic unit, *Figure 10*. This unit has a valve-controlled, height-adjustable and double direction pressure cylinder driven by a motor pump. The relevant tool sets are matched to the bearing and the mounting situation. Comprehensive information on the product described and the precise procedure for mounting and dismounting is given in TPI 156, Tapered Roller Bearing Units TAROL – Mounting, maintenance, repair.

Figure 10
Mobile hydraulic unit

The mounting of bearings is subject not only to the specification of the Association of American Railroads (AAR) but also to the mounting specifications of the AAR in their valid issue. These can be found mainly in the Sections G, G-II, H and H-II of the "Manual of Standards and Recommended Practices".

Mounting of four-row tapered roller bearings

Four-row tapered roller bearings are special bearings for rolling mills and comprise solid bearing rings and tapered roller and cage assemblies, *Figure 11*. These bearings are separable and are generally mounted in the chock. The chock with the bearing is then slid onto the journal. This requires either a loose cylindrical fit of the inner ring on the journal or a bearing with a tapered bore that is mounted on a tapered shaft journal.

Figure 11
Four-row tapered roller bearing

Comprehensive information on the mounting and dismounting of four-row tapered rolling bearings is given in publication WL 80154, Four-Row Tapered Roller Bearings, Mounting Manual.

Mounting of needle roller bearings

Needle roller bearings with machined rings are mounted in accordance with the same perspectives as cylindrical roller bearings. Bearings mounted adjacent to each other must have the same radial internal clearance so that the load is distributed uniformly.

Needle roller bearings with ribs

Needle roller bearings with ribs are single or double row units comprising machined outer rings with ribs, needle roller and cage assemblies and removable inner rings.

Replacement of inner rings

In the case of the standard bearings, the inner rings are matched to the enveloping circle tolerance F6 and can be interchanged with each other (mixed use) within the same accuracy class.

! In needle roller bearings with ribs, the inner ring is not self-retaining.

Radial and axial location

Needle roller bearings with an inner ring are radially located by means of a fit on the shaft and in the housing. In order to prevent axial creep of the bearing rings, they must be located by means of form fit, *Figure 12*.

The abutment shoulders (shaft, housing) should be sufficiently high and perpendicular to the bearing axis. The transition from the bearing seat to the abutment shoulder must be designed with rounding to DIN 5418 or an undercut to DIN 509. The minimum values for the chamfer dimensions r in the dimension tables must be observed.

The overlap between the snap rings and the end faces of the bearing rings must be sufficiently large, *Figure 12*.

The maximum chamfer dimensions for the inner rings in accordance with DIN 620-6 must be taken into consideration.

NA49

(1) Snap rings
(2) Abutment shoulders

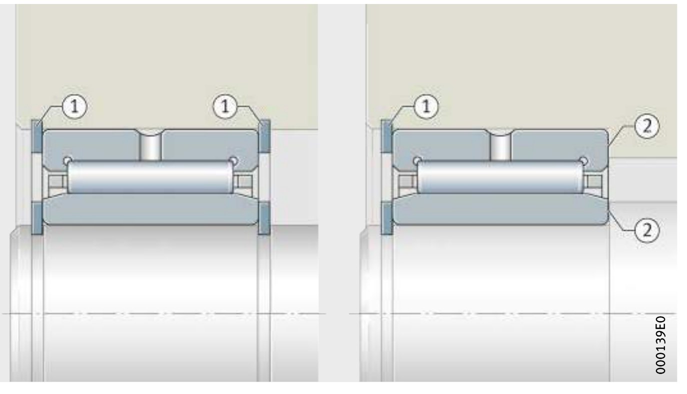


Figure 12
Axial location of bearing rings

Needle roller bearings without ribs

These single or double row units comprise machined outer rings without ribs, needle roller and cage assemblies and removable inner rings. Since the bearings are not self-retaining, the outer ring, needle roller and cage assembly and inner ring can be mounted separately from each other.

Replacement of inner rings

!

In needle roller bearings without ribs, the inner ring is not self-retaining.

The outer ring and the needle roller and cage assembly are matched to each other and must not be interchanged during fitting with components from other bearings of the same size.

In the standard bearings, the inner rings are matched to the enveloping circle tolerance F6 and can be interchanged with each other (mixed use) within the same accuracy class.

Radial and axial location

Needle roller bearings with an inner ring are radially located by means of a fit on the shaft and in the housing. In order to prevent axial creep of the bearing rings, they must be located by means of form fit, *Figure 13*.

The abutment shoulders (shaft, housing) should be sufficiently high and perpendicular to the bearing axis. The transition from the bearing seat to the abutment shoulder must be designed with rounding to DIN 5418 or an undercut to DIN 509. The minimum values for the chamfer dimensions r in the dimension tables must be observed.

The overlap between the snap rings and the end faces of the bearing rings must be sufficiently large, *Figure 13*.

The maximum chamfer dimensions for the inner rings in accordance with DIN 620-6 must be observed.

NAO..-ZW-ASR1
RNAO

① Snap rings
② Abutment shoulders

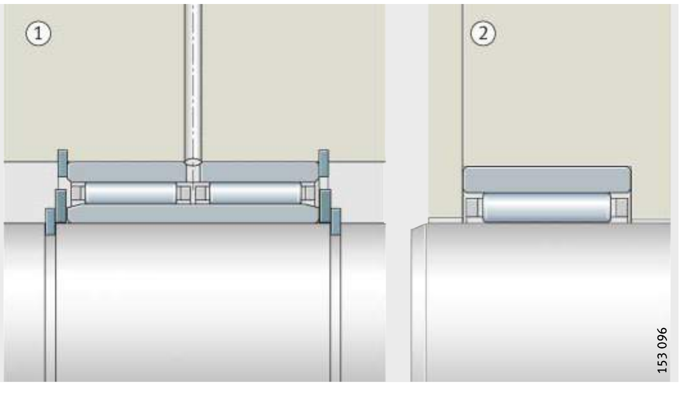


Figure 13
Axial location of bearing rings

Aligning needle roller bearings

The bearings comprise drawn outer cups, plastic support rings with a concave inner profile, outer rings with a spherical outside surface, needle roller and cage assemblies and removable inner rings.

Radial and axial location

Aligning needle roller bearings are firmly seated in the housing bore. No further axial location is required. The bore can therefore be produced easily and economically.

Replacement of inner rings

In the standard bearings, the inner rings are matched to the enveloping circle tolerance F6 and can be interchanged with each other (mixed use) within the same accuracy class.

In aligning needle roller bearings, the inner ring is not self-retaining.

Mounting using pressing mandrel

Due to the drawn outer cup, the bearings must be mounted using a special pressing mandrel, see page 103. The marked side of the bearing should be in contact with the flange of the mandrel. A toroidal ring on the mandrel holds the bearing securely on the mandrel.

Combined needle roller bearings

These series comprise radial needle roller bearings and a rolling bearing component capable of supporting axial loads. They can support high radial forces as well as axial forces in one direction, while NKIB can support axial forces from both directions, and are used as locating or semi-locating bearings.

The bearings are available in the following designs:

- needle roller/axial deep groove ball bearings
- needle roller/axial cylindrical roller bearings
- needle roller/angular contact ball bearings.

The tight fits of combined needle roller bearings lead to relatively large press-in forces. This must be noted especially in the case of needle roller/axial deep groove ball bearings and needle roller/axial cylindrical roller bearings with dust caps, where the roller and cage assembly of the axial bearing must not be removed. These bearings must be pressed in. It is advantageous to heat the housing.

Combined needle roller bearings must be pressed into the housing, *Figure 14*.

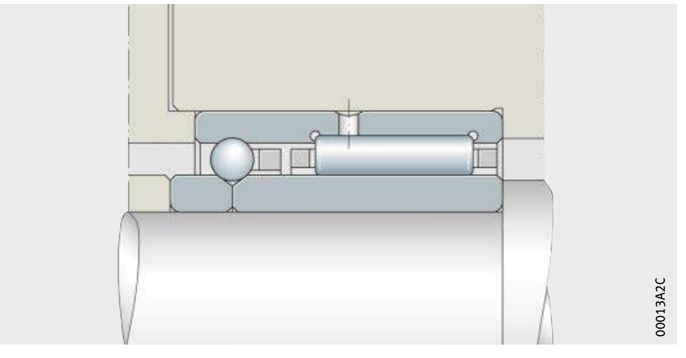


Figure 14
Mounting of combined needle roller bearings (needle roller/angular contact ball bearing)

Mounting of special types

Replacement of inner rings

In the standard bearings of series NKIA and NKIB, the inner rings are matched to the enveloping circle tolerance F6 and can be interchanged with each other (mixed use) within the same accuracy class.

Combined needle roller bearings are not self-retaining.

Radial and axial location

Bearings with an inner ring are radially located by means of a fit on the shaft and in the housing. The axial abutment shoulders (shaft, housing) should be sufficiently high and perpendicular to the bearing axis. The transition from the bearing seat to the abutment shoulder must be designed with rounding to DIN 5418 or an undercut to DIN 509. The minimum values for the chamfer dimensions r in the dimension tables must be observed.

The overlap between the snap rings and the end faces of the bearing rings must be sufficiently large.

The maximum chamfer dimensions for the inner rings in accordance with DIN 620-6 must be taken into consideration.

In order to prevent lateral creep of the bearing rings, they must be located by means of form fit. For locating bearings and for bearings with a split inner ring, axial abutment of the bearing rings on both sides is particularly important.

Mounting of drawn cup needle roller bearings

Due to their thin-walled outer ring, drawn cup needle roller bearings adopt their exact form as a result of tight housing fits, eliminating the need for lateral location.

Special mounting mandrels are used for pressing in drawn cup needle roller bearings. The mandrel normally rests against the stamped end face of the bearing, which is hardened in the case of smaller bearings. Even if the bearing is pressed in via an unhardened rib, this will not lead to deformations or jamming of the needle roller and cage assembly as long as the mandrel is correctly dimensioned.

Radial and axial location

Drawn cup needle roller bearings are located in the housing bore by means of a press fit. They are pressed into the bore and require no further axial retainers.

Mounting using pressing mandrel

The bearings should be installed using a special pressing mandrel, *Figure 15*. The shoulder of the pressing mandrel must rest against the end face of the bearing. This is indicated by the designation.

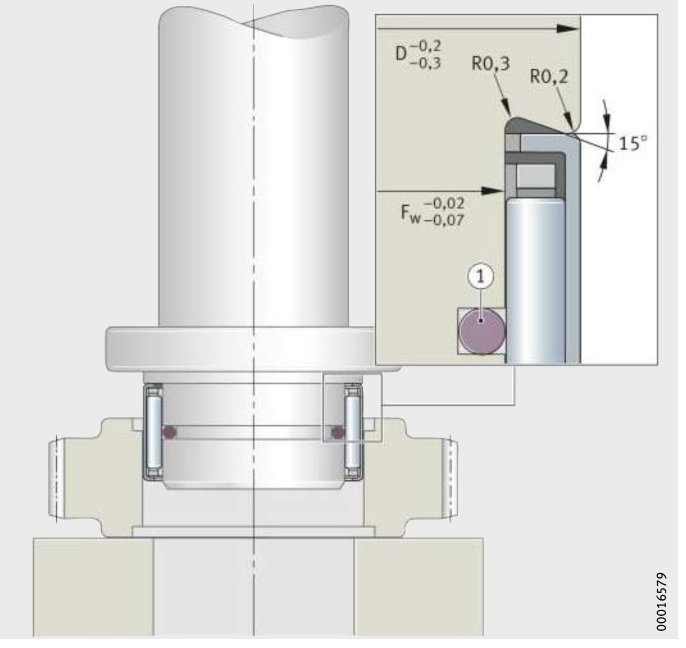
A toroidal ring should be used to retain the bearing. The length and oversize of the ring must be matched by the customer to the dimensions and mass of the bearing.

If grease lubrication is to be used, the bearings should be lubricated with grease before mounting.

! Drawn cup bearings must not be tilted while they are being pressed in.

The forces occurring during pressing-in are dependent on several factors. Mounting must be carried out so that the bearing rib on the end face is not deformed.

If the application requires a mounting procedure different from the one described, mounting trials must be carried out in order to ensure that the bearings can be mounted correctly and without causing damage.



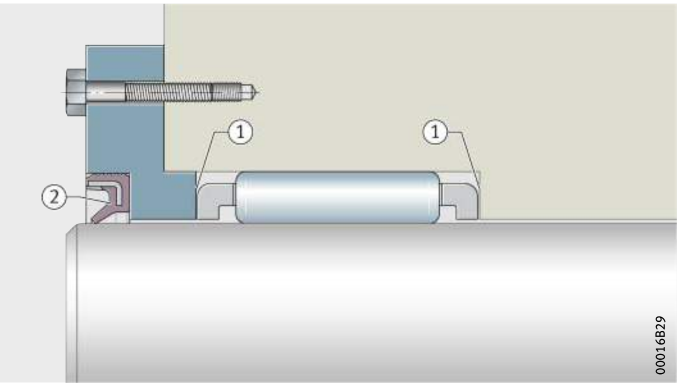
① Toroidal ring

Figure 15
Mounting using pressing mandrel

Mounting of needle roller and cage assemblies

Needle roller and cage assemblies are either slid onto the shaft and the parts are then inserted together into the housing, or needle roller and cage assemblies are slid into the housing and the shaft is then inserted. Mounting is carried out without load and using a screw-driver type motion.

Needle roller and cage assemblies can be laterally guided on the shaft or on the housing, *Figure 16*.



① Guidance on housing
② Guidance on shaft

Figure 16
Guidance of needle roller and cage assemblies

The spacing between the lateral contact running surfaces of the cage must be sufficiently large (tolerance H11), in order to avoid jamming of the needle roller and cage assemblies.

The radial internal clearance of a bearing arrangement with needle roller and cage assemblies is based on the machining tolerances of the hardened and ground raceways on the shaft and in the housing. Needle roller and cage assemblies arranged adjacent to each other must contain needle rollers of the same grade.

Mounting of rope sheave bearings

Before the bearing is pressed into the rope sheave, it is recommended that the bearing seating surfaces should be lightly oiled or alternatively rubbed or sprayed with dry lubricant. In order to prevent bearing damage and inaccurate seating of the bearings, the pressing-in operation should be carried out on a suitable device with control of force and travel.

In order to facilitate the pressing-in operation, the rope sheave can be heated. Appropriate tools for steel rope sheaves are available from Schaeffler.

Guidelines for mounting

During the mounting of cylindrical roller bearings SL04, the mounting forces should be applied only to the bearing ring to be mounted, *Figure 17*.

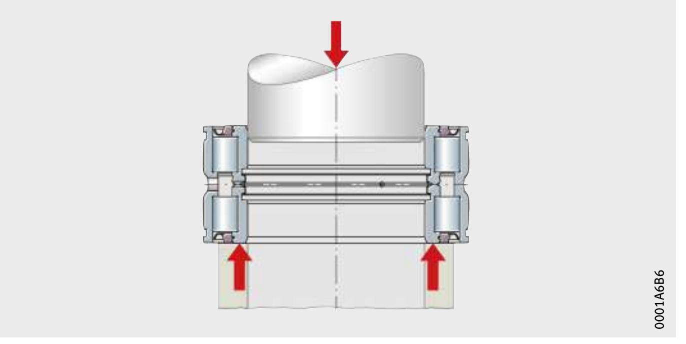


Figure 17
Application of mounting forces

Mounting forces must not be directed through the cylindrical rollers, *Figure 18*. During dismounting of the bearings, the dismounting forces must not be directed through the fasteners on the split inner ring.

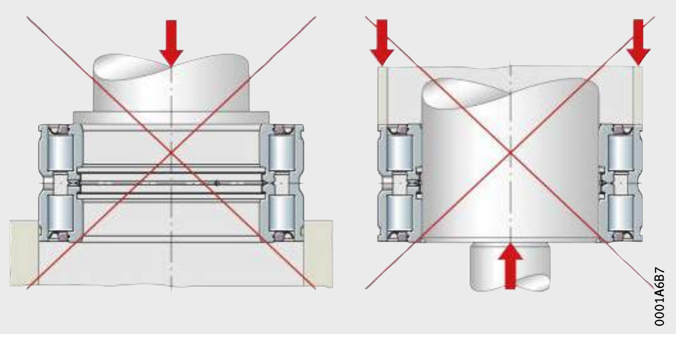
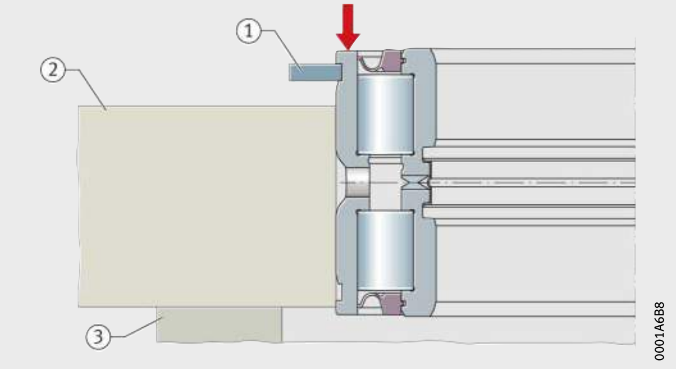


Figure 18
Impermissible mounting or dismounting methods

Mounting with a premounted retaining ring

If bearings with a premounted retaining ring are pressed into the rope sheave, this must be always carried out with monitoring of force (or alternatively with monitoring of travel), *Figure 19*.



1 Retaining ring
2 Rope sheave
3 Surface for support of mounting forces

Figure 19
Mounting with a premounted retaining ring

Mounting of special types

Mounting of track rollers

Track rollers are precision machine elements. These products must be handled very carefully before and during mounting. Their trouble-free operation depends largely on the care taken during mounting.

The seating surfaces of the bearing rings must be lightly oiled or rubbed with solid lubricant.

After mounting, the bearings must be supplied with lubricant. Finally, the correct functioning of the bearing arrangement must be checked.

Mounting of yoke type track rollers

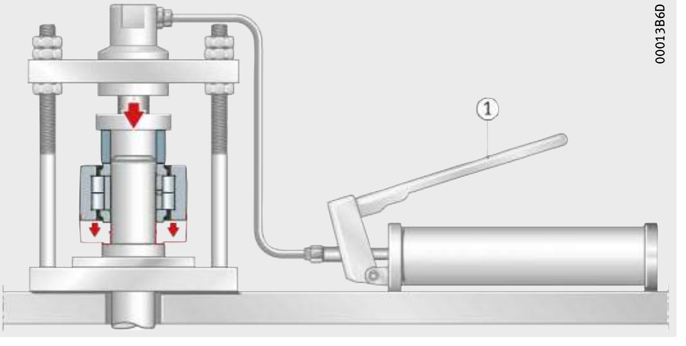
If the tolerances are unfavourable, the yoke type track roller should be pressed onto the shaft using a mounting press, *Figure 20*.
The inner ring must be mounted such that the pressing-in force is distributed uniformly over the end face of the inner ring.

!

Yoke type track rollers RSTO and STO are not self-retaining. The outer ring and the needle roller and cage assembly are matched to each other and must not be interchanged during mounting with components from other bearings of the same size.

Lubrication hole

The bearings must be mounted such that the lubrication hole is positioned in the unloaded zone. For yoke type track rollers PWTR and NNTR, there is no need for defined positioning of the lubrication hole.

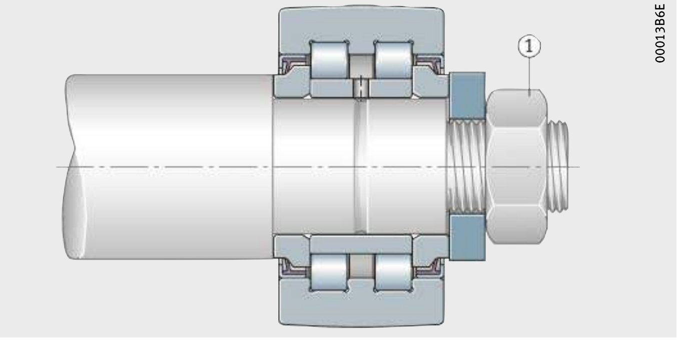


① Mounting press

Figure 20
Mounting of yoke type track roller using a mounting press

Axial location

Yoke type track rollers NUTR, PWTR and NNTR must be axially clamped in place, *Figure 21*.



① Hexagon nut

Figure 21
Axial location

Mounting of stud type track rollers

Stud type track rollers should be mounted using a mounting press if possible (similar to *Figure 20*, page 106).

Blows on the flange of the roller stud must be avoided.

Drive fit lubrication nipples for stud type track rollers

Stud type track rollers are supplied with loose drive fit lubrication nipples that must be pressed in correctly before mounting of the bearings, *Figure 22*.

Only the lubrication nipples supplied may be used, see table.

If relubrication is to be carried out via the locating bore, the axial lubrication holes in the stud type track roller must be closed off using the lubrication nipples before mounting, *Figure 22*.

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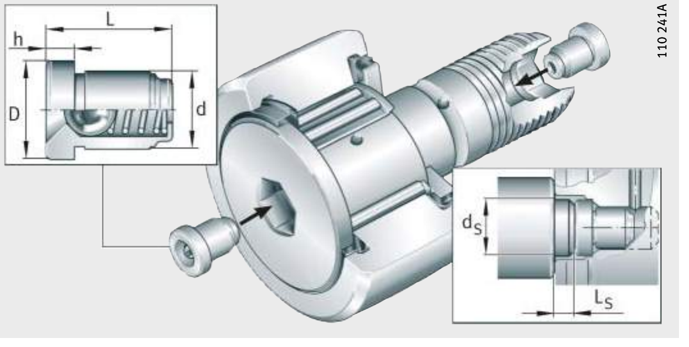


Figure 22
Stud type track roller with drive fit lubrication nipple and dimensions for pressing mandrel

Drive fit lubrication nipples

Lubrication nipple	Dimensions in mm						Suitable for outside diameter D
	D	d	L	h	d_S $\pm 0{,}1$	L_S	
NIPA1	6	4	6	1,5[1)]	–	–	16, 19
NIPA1×4,5	4,7	4	4,5	1	4,5	5	22 – 32
NIPA2×7,5	7,5	6	7,5	2	7,5	6	35 – 52
NIPA3×9,5	10	8	9,5	3	10	9	62 – 90

1) Projection of lubrication nipple.

Axial location of stud type track rollers

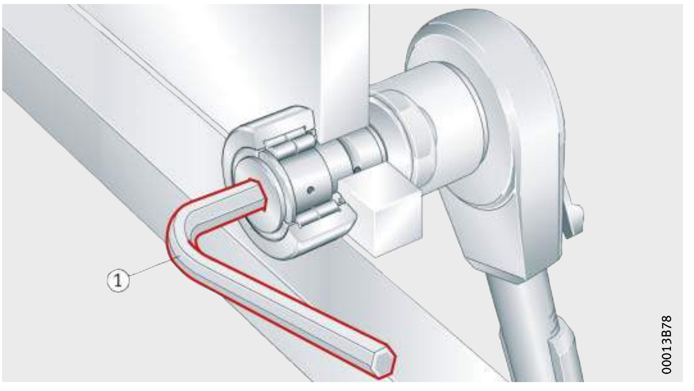
Stud type track rollers must be axially secured using a hexagon nut.

The slot or hexagonal socket on the end of the roller stud can be used to hold the bearing by means of a key while tightening the fixing nut and to adjust the eccentric collar, *Figure 23*.

If heavy vibration occurs, self-locking nuts to DIN 985 or special locking washers can be used.

The tightening torque stated for the fixing nuts must be observed. It is only in this way that the permissible radial load can be ensured. If this cannot be adhered to, an interference fit is required.

For self-locking nuts, a higher tightening torque must be observed; the advice given by the nut manufacturer must be followed.



① Allen key

Figure 23
Holding the bearing using an Allen key

Stud type track rollers with eccentric collar

The highest point on the eccentric collar is indicated on the roller stud side.

Commissioning and relubrication

Stud type track rollers have a lubrication hole for relubrication:

- on the flange side of the roller stud
- on the thread-side end face,
  from an outside diameter of 22 mm
- on the shank of the roller stud,
  from an outside diameter of 30 mm with an additional lubrication groove.

Stud type track rollers with an eccentric collar cannot be relubricated via the stud. The eccentric collar covers the lubrication hole.

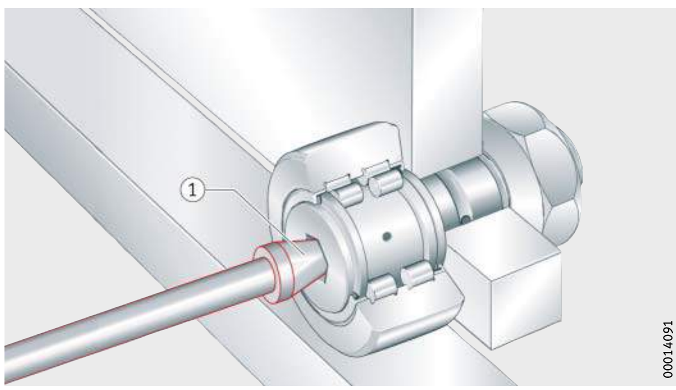
For lubrication, only grease guns with needle point nozzles may be used that have an opening angle ≦ 60°, *Figure 24*.

Before commissioning, the lubrication holes and feed pipes must be filled with grease in order to ensure protection against corrosion; lubrication can be carried out at the same time.

Lubrication will be more difficult if a rolling element is located over the radial lubrication hole. Relubrication should therefore be carried out with the bearing still warm from operation and rotating if safe to do so, before the bearing comes to rest if safe to do so and before extended breaks in operation.

The grease used for relubrication must be the same as that used for initial greasing. If this is not possible, the miscibility and compatibility of the greases must be checked.

Relubrication should continue until a fresh collar of grease appears at the seal gaps. The old grease must be able to leave the bearing unhindered.



(1) Needle point nozzle, opening angle ≦ 60°

Figure 24
Relubrication using grease gun

Dismounting of rolling bearings

Dismounting methods

Dismounting of rolling bearings

Dismounting methods

Dismounting of rolling bearings

Dismounting methods

In order to prevent damage during the dismounting of bearings, various dismounting methods are used depending on the bearing size and type of application that facilitate the reuse of components. In general, a distinction is made in the dismounting of bearings between mechanical, thermal and hydraulic methods. Before dismounting is actually carried out, the mounting drawings and any instructions for mounting and dismounting must be carefully checked. In case of doubt, the Schaeffler expert team is available to provide advice and assistance.

Mechanical dismounting

In the mechanical method, special extractors are normally used. It must be ensured above all that the extraction tool is positioned on the ring that has the tighter fit, otherwise the rolling elements will press into the raceways of the bearing, *Figure 1*. Furthermore, there is a risk of fracture in the case of thin-walled outer rings.
In the case of non-separable bearings with a sliding seat on the shaft or housing, this adjacent component should if possible be removed before dismounting of the bearing. The force that must be used in pressing the ring off is normally considerably greater than the force used in pressing the ring on, since the ring becomes fixed in place over the course of time. Dismounting can be difficult even in the case of rings with a loose fit if fretting corrosion has formed after long periods of operation.

!

The following must therefore be observed:

- Avoid direct blows on the bearing rings.
- Do not direct dismounting forces through the rolling elements.

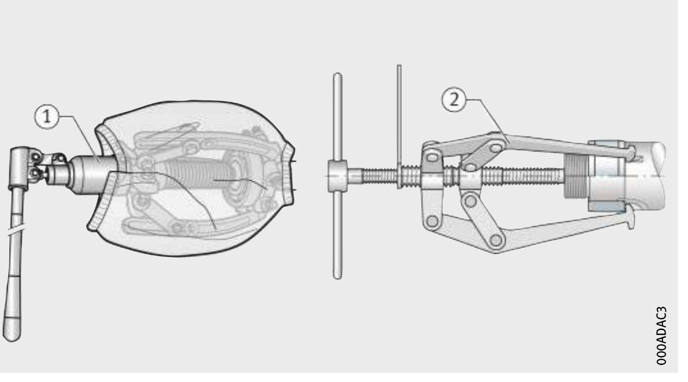
Figure 1
Dismounting
by means of extraction device

If it is not possible to avoid carrying out extraction via the rolling elements, a collar made from unhardened steel is placed around the outer ring (its thickness should be greater than $^1/_4$ of the height of the bearing cross-section). This applies in particular to rolling bearings with a small cross-sectional height and small contact angle, such as tapered and spherical roller bearings. However, the bearings cannot be subsequently reused.

The rings of separable bearings can be dismounted individually.

Dismounting of cylindrical seats

The extraction of small bearings is normally carried out using mechanical extraction devices, *Figure 2*, or hydraulic presses, *Figure 3*, that are in contact either with the ring with a tight fit itself or with the contact parts, such as those on the labyrinth ring. These are available with a mechanical spindle and hydraulic cylinder if higher forces are required.



1 Extraction process using a trisection plate
2 Extraction device with three adjustable arms and extraction slot in inner ring

Figure 2
Extraction device for rolling bearings

A press can also be used to assist in dismounting. It must be ensured in this case that the bearing is abutted on its inner ring in order to avoid damage. In the pressing operation, the shaft is withdrawn from the bearing.

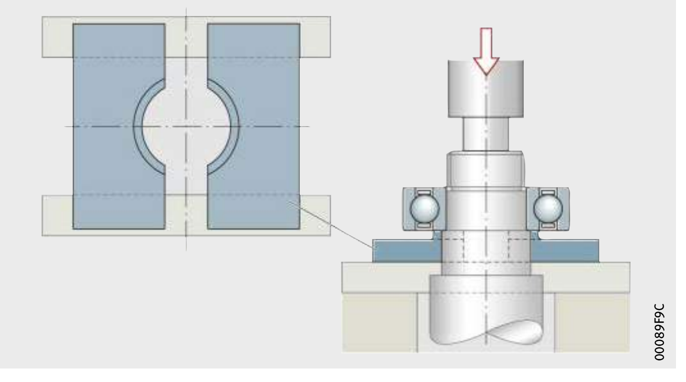


Figure 3
Dismounting using a press

Dismounting is made much easier if the design includes extraction slots so that the extraction tool can be applied directly to the bearing ring with a tight fit. A further alternative to dismounting of bearings is the use of extraction screws, *Figure 10*, page 119.

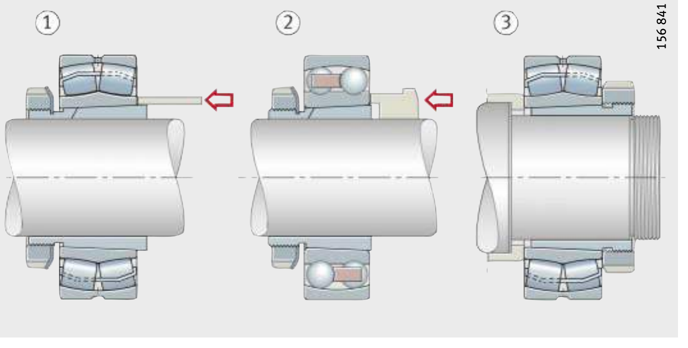
Internal extractors

If the shaft has already been dismounted, the bearing can also be removed from the housing by means of an internal extractor.
The gripping segments of the extractor are spread when the threaded spindle is tightened. The lip of the jaws is pressed against the back of the bearing inner ring bore. With the aid of a countersupport or an impact extractor, the bearing is then withdrawn using the internal extractor. As a result, it is not generally possible to reuse the bearing.

Dismounting of tapered seats

If bearings are mounted directly on a tapered shaft seat or on an adapter sleeve, the locking action of the shaft or adapter sleeve nut must be loosened first. The nut must then be unscrewed by at least the amount of the drive-up distance. The inner ring is then driven off the sleeve or shaft, for example using a metal drift or impact block, *Figure 4* ①, ②. An impact block avoids the risk of slipping.

Bearings located using withdrawal sleeves are dismounted using a withdrawal nut, *Figure 4* ③.



① Metal drift
② Impact block
③ Withdrawal nut

Figure 4
Dismounting of bearings

The dismounting of large bearings located using a withdrawal sleeve requires considerable force. In this case, locknuts with additional pressure screws can be used, *Figure 4*. A disc must be inserted between the inner ring and the pressure screws in order to prevent damage to the bearing.

Thermal dismounting

In thermal dismounting, the bearing ring to be dismounted is heated within a very short space of time, leading to its expansion.
As a result, the fit on the bearing seat is neutralised and any possible adhesion as a result of fretting corrosion is overcome.

Heating of the bearing ring should not be carried out using a direct flame, since this can cause damage to the components.

Heating rings

Heating rings made from light metal can be used for dismounting the inner rings of cylindrical roller bearings that have no ribs or only one rigid rib. The rings are heated on an electric heating plate, depending on their tight fit or interference, to between +200 °C and +300 °C pushed over the bearing ring to be removed and clamped in place. Once the press fit on the shaft has been neutralised, both rings are removed together.

After removal, the bearing ring must immediately be released from the heating ring in order to prevent overheating.

Heating rings are particularly advantageous for the occasional extraction of medium-sized bearing rings. Each bearing size requires a specific heating ring.

Medium frequency technology

With the aid of FAG devices using medium frequency technology, it is possible to heat very large bearings and components of shrink fit connections by inductive means in order to achieve loosening.

The FAG medium frequency heating device comprises the medium frequency generator and an inductor. Depending on the application, this can be of a flexible or rigid design. The flexible version is similar to a cable. It must be ensured in this case that the winding is applied directly to the component with fit.

Where a rolling bearing has a tight fit on a shaft, for example, the inductor must be applied directly to the inner ring. Through energy-efficient heating, the workpiece to be loosened is heated very quickly and undergoes expansion, so the press fit can be loosened. Flexible inductors are suitable for workpieces of various sizes and various shapes and can be used for long periods at heating temperatures up to +150 °C.

Figure 5
Dismounting of bearing inner rings using flexible inductor

In the batch dismounting of identical components such as wheelset bearings on rail vehicles, flexibility is less important than reduced setup times and increased process reliability. Rigid inductors are suitable for this task. In this design, the coil is fitted in a housing matched to the workpiece and can thus be placed quickly and easily in the heating zone. Rigid inductors are also suitable, in contrast to the flexible variant, for small components.

! The devices are designed for the specific application. Please contact the application experts at Schaeffler.

Advantages

Advantages in the loosening of shrink fit connections:

- batch dismounting of bearing rings and labyrinth rings
- rapid dismounting of gears and couplings
- simple heating of large and heavy components (such as machine supports, housings, shafts, ...).

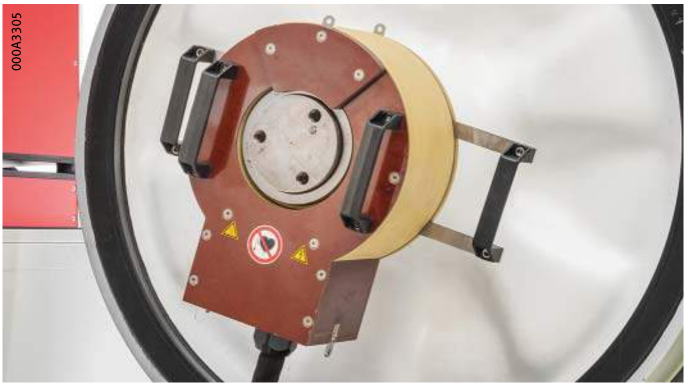


Figure 6
Dismounting of bearing inner rings of wheelset bearings (rail vehicles) using rigid inductor

Further information

- TPI 217, Induction Units with Medium Frequency Technology.

Hydraulic dismounting

In the oil pressure method, oil is pressed between the fit surfaces. The oil film substantially neutralises the contact between the fit parts, so they can be displaced relative to each other with the application of little force and without the risk of surface damage, see page 83.

The oil pressure method is suitable for dismounting in the case of both tapered and cylindrical seats. In both cases, oil grooves and feed ducts as well as threaded connectors for the pressure generation devices must be provided. Large adapter and withdrawal sleeves have appropriate grooves and holes.

For the dismounting of bearings with a tapered bore that are located directly on the shaft, injectors are sufficient as pressure generation devices. For bearings with cylindrical bores and where adapter and withdrawal sleeves are present, a pump must be used, *Figure 27*, page 87.

For dismounting, the same oils are used as for mounting, which means oils with a viscosity of approx. 75 mm^2/s at +20 °C (nominal viscosity 32 mm^2/s at +40 °C). Fretting corrosion can be dissolved by rust-dissolving additives in the oil.

Dismounting of cylindrical bearing bore

In the dismounting of bearings with a cylindrical bore, the oil pressure method is normally only used in a supporting function for mechanical tools. The specific extraction device is first placed on the ring with fit and oil under pressure is then pumped into the oil grooves, *Figure 7*.

This neutralises the fit and the bearing can be removed, for example by means of a mechanical extractor.

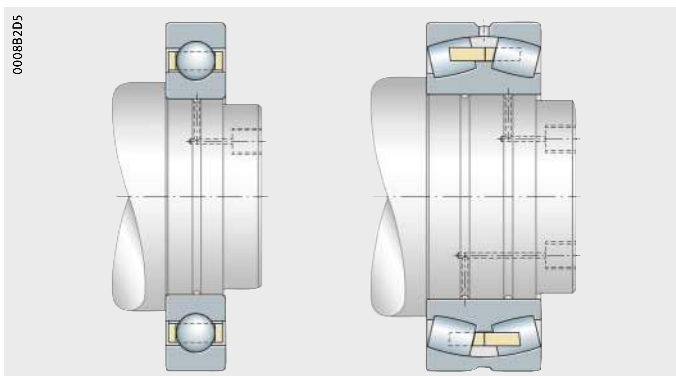


Figure 7
Hydraulic dismounting of cylindrical seat

Dismounting of rolling bearings

If there are no grooves and ducts in the shaft, for example for reasons of strength, the oil can be pressed between the fit surfaces from the end face of the inner ring. A sealed contact ring is located at the front end of the interference fit, through which the oil is pressed into the fit joint. A container fixed to the end of the shaft makes it possible to press oil between the fit surfaces until the end of the extraction process. If it is not possible to fit such a container, a very stiff oil with a viscosity of 320 mm^2/s (cSt) at +40 °C must be used. With an oil of such stiffness, the oil film remains in the fit joint for up to 5 minutes. This time is sufficient for extraction of the bearing.

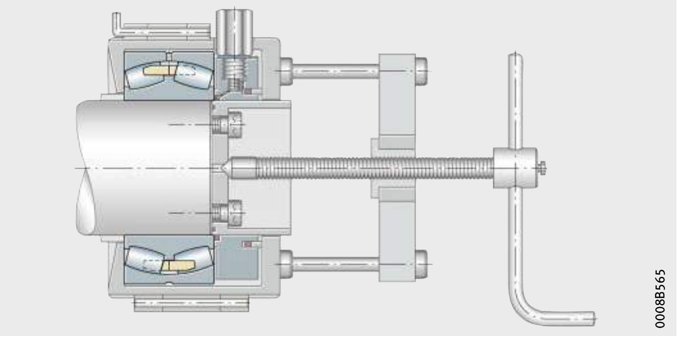


Figure 8
Special device for extraction from a shaft without oil grooves

Dismounting of tapered bore

In the extraction of bearings located on a tapered shaft journal, a withdrawal sleeve or an adapter sleeve, it is only necessary to press oil between the fit surfaces.

! The interference fit becomes loose abruptly. Due to the risk of accidents, axial movement of the rolling bearing or withdrawal sleeve during dismounting must be restricted by a shaft nut, adapter sleeve nut or a stop, *Figure 9*.

Figure 9
Hydraulic dismounting of tapered seat

Dismounting is sometimes made more difficult by fretting corrosion. The use of a rust-dissolving hydraulic fluid is recommended, especially in the case of bearings that are dismounted after a long period of operation. In difficult cases, removal of the withdrawal sleeve can be supported by the extraction nut, *Figure 10*. If pressure screws are present in the withdrawal sleeve nut, an intermediate ring must be inserted in order that the extraction forces do not act directly on the rib of the rolling bearing ring.

Dismounting of a withdrawal sleeve:
① Using a nut and pressure screws
② Using a hydraulic nut

Dismounting of a spherical roller bearing from the withdrawal sleeve:
③ Using the hydraulic method

Dismounting of a spherical roller bearing on an adapter sleeve:
④ Using the hydraulic method

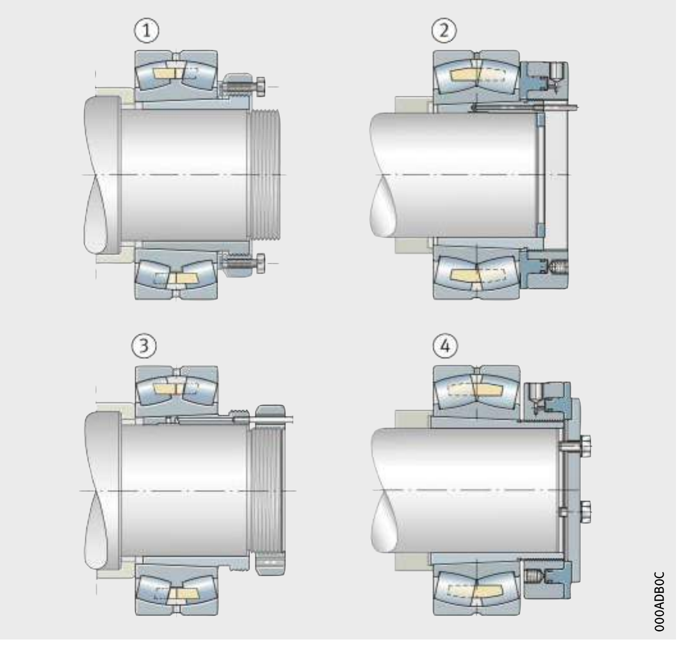


Figure 10
Dismounting of a withdrawal sleeve and spherical roller bearing

Services

Services

Features

Services

Features

Schaeffler offers, irrespective of manufacturer, a wide range of services relevant to the lifecycle of a rolling bearing: starting with mounting and progressing through maintenance to the reconditioning of rolling bearings.

During the operational phase, the Schaeffler experts provide support through services in the fields of condition monitoring and corrective maintenance. Companies that wish to build up their knowledge in the areas of rolling bearings and condition monitoring also have access to the Schaeffler training and consultancy portfolio on site, centrally or online. Our e-learning portfolio on the Internet provides the first steps into this field. In this way, customers benefit from the expertise of a leading supplier of rolling and plain bearings.

Mounting and dismounting

The Schaeffler Industrial Service experts offer mounting and dismounting services for rolling bearings that are applicable across industrial sectors. They have detailed knowledge and extensive experience in all industrial sectors, *Figure 1*.

Figure 1
Mounting service provided by Schaeffler experts

The experts in the Industrial Service function are trained and skilled personnel who can provide reliable, rapid and competent assistance. The services are provided either at the customer's location or in the Schaeffler workshop facilities.

The mounting and dismounting services include:

- mounting and dismounting of rolling bearings and bearing systems of all types
- measurement and condition analyses
- problem solving and preparation of concept solutions
- design and manufacture of special tools
- rental of tools
- emergency service
- training courses on products and mounting
- certification of mounting and dismounting processes.

Advantages

The mounting services give the following advantages:

- rapid availability worldwide of experts in bearing arrangement technology with extensive experience in almost every application
- rapid mounting or dismounting by means of professional preparation and implementation
- increased plant availability and productivity as a result of reduced unplanned downtime
- optimisation of mounting and dismounting processes
- professional mounting and dismounting using special high-quality tools
- training and awareness measures for employees relating to the correct handling of bearings of all types.

Rental of tools

Customers who require special mounting and dismounting tools or measuring equipment only infrequently can rent these from Schaeffler for a fee.

Our service includes:

- prompt rental in Europe
- free-of-charge, rapid delivery to the installation site
- checked quality products in keeping with the latest technological developments
- delivery of the tools, including all add-on parts
- user manuals available in several languages.

If one of our qualified experts in the Industrial Service function is commissioned to carry out the particular activity, rental costs are not generally incurred.

Certification

Approximately 25 percent of all premature bearing failures can be attributed to mounting errors. In order to achieve a long bearing operating life, it is particularly important to have not only basic knowledge of rolling bearings but also theoretical and practical knowledge of their correct mounting and dismounting. In order to ensure that the training received by mounting personnel is as close to reality as possible, Schaeffler offers certification of individual mounting and dismounting processes, *Figure 2*.

Figure 2
Theoretical training

Here, information on the correct handling of rolling bearings and the avoidance of mounting errors is imparted by our rolling bearing experts. This is carried out with direct reference to the specific application and the individual circumstances of the customer.

A practical demonstration of the mounting and dismounting process is then provided, which also covers adherence to the necessary processes and regulations.

Finally, the training participants must put their acquired knowledge to the test. Only then do they receive application-specific certification from Schaeffler.

Reconditioning of rolling bearings

It is often the case that new rolling bearings are fitted although the existing bearings could be restored to as-new condition by means of appropriate reconditioning. In many cases, reconditioning of rolling bearings is significantly more cost-effective than using new rolling bearings, *Figure 3*.

① Before reconditioning
② After reconditioning

Figure 3
Rolling bearing raceway and rollers before and after reconditioning

Advantages

The advantages for the customer are as follows:

- reductions in life cycle costs (LCC = Life Cycle Costs)
- increases in operating life
- savings in material and energy costs
- reductions in inventory costs
- high flexibility through short lead times
- feedback on the characteristics and frequencies of damage.

The operations necessary in reconditioning are dependent on the condition of the rolling bearing. In order to allow a reliable statement of the work required, the rolling bearing must be disassembled, cleaned and then carefully examined. Beyond this requalifying process (Level I), which is always necessary, there are three further reconditioning levels, *Figure 4*.

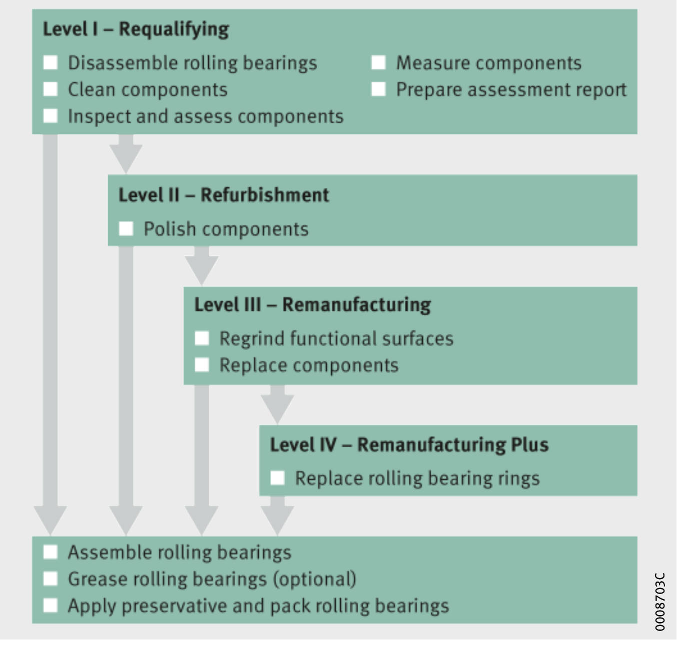


Figure 4
Reconditioning levels

Quality

Schaeffler performs reconditioning of rolling bearings to uniform standards throughout the world. All sites apply identical processes and guidelines. Schaeffler rolling bearings are processed in accordance with the original drawings. In the case of all bearings, work is carried out using only original components and original replacement parts. High quality reconditioning is achieved as a result of our comprehensive knowledge of rolling bearings.

Market sectors

Reconditioning is carried out irrespective of manufacturer and is thus not restricted to products of Schaeffler Technologies. Before reconditioning, the condition of the bearings can be assessed on site in consultation with experts from the Global Technology Network.

Reconditioning of rolling bearings is of particular interest if these are used in machinery or vehicles in the following market sectors:

- raw material extraction and processing
- metal extraction and processing
- pulp and paper
- railways.

Dimensions

Reconditioning and, where required, modification can be carried out on rolling bearings with an outside diameter D of 100 mm to 4 500 mm. Please contact us for information on reconditioning or modification of bearings with other outside diameters.

Training courses

The operating life of rolling bearings is determined to a substantial degree by their correct mounting and dismounting. Appreciating the use of rolling bearings, linear guidance systems and plain bearings as indispensable elements in thousands of applications requires the necessary understanding of these machine elements. Schaeffler has its own training centres worldwide certified to ISO 9001, *Figure 5*.

Figure 5
Training course at the Eltmann site

Training courses on mounting and dismounting generally comprise a theoretical part and a practical part. Thorough knowledge is communicated, for example, on the mounting and dismounting of rolling bearings using the optimum tools and on the condition monitoring of bearing arrangements, especially through the use of noise, vibration and torque measurements.

In general, the initial steps are provided by basic training covering the different characteristics, features and types of rolling bearings, plain bearings and linear guidance systems as well as their combination to form systems, extending all the way to mechatronic units. Application examples reflect the selection criteria and the customer benefits achieved. These product-oriented training courses are followed by modules covering rolling bearing theory as well as selected applications. Rolling bearing theory conveys the necessary knowledge on subjects such as bearing clearance, load distribution in the bearing, rating life and lubrication. In workshops, the participants concentrate on applications, for example the bearing arrangements in a machine tool or a shaft bearing arrangement. All process steps are covered, from bearing selection and bearing calculation through to mounting, *Figure 6*, page 129. We also offer workshops in the field of mechatronics.

Figure 6
Training course on mounting of rolling bearings

Several training modules cover the mounting and dismounting of rolling bearings and linear guidance systems. Based on perception and exercises, the participant gains the mounting knowledge and skills that he will require in practice. Our training courses on mounting cover a large number of applications. Mounting exercises on individual products are followed by work on more complex systems such as gearboxes, rail wheelsets or machine tools. The possibilities for plannable and economical design of maintenance work on machines, plant and rolling bearings are communicated to the training participant in appropriate courses.

Mounting cabinet

Literature on the correct mounting of bearings is readily available, however there is a general lack of appropriate equipment on which apprentices can practise in as practical a sense as possible. The trainers from the Schaeffler training workshops therefore compiled a basic course, *Figure 7*, page 130.

Figure 7
Basic course: Mounting cabinet

The aim of this rolling bearing course is to communicate knowledge on the selection of the correct bearing, correct mounting and dismounting and the maintenance of bearing positions. It is divided into two parts.

The theoretical part communicates basic knowledge on rolling bearing technology, illustrating the subject areas of technical drawing, technical calculation and technical theory by means of state-of-the-art media. In the practical part, the basic skills in the mounting and dismounting of common types of bearings are practised with the aid of exemplary simplified mating parts (shafts, housings).
Various methods and tools are used here.

The learning content comprises smaller learning stages and is available in various languages. They correspond in their full scope to the degree of difficulty that is currently required in vocational training. On the basis of this basic course, training can also be given on individual content by means of various mounting sets, *Figure 8*.

Figure 8
Exercises
using the mounting cabinet

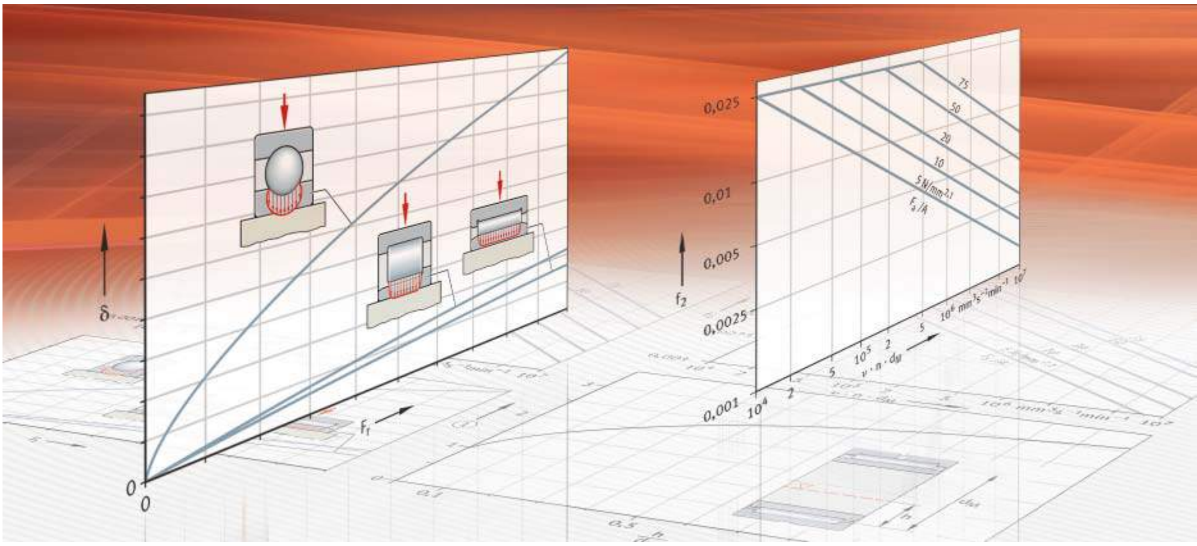
Mounting cross

In order to provide professional training courses on the correct mounting and dismounting of rolling bearings, Schaeffler has developed the so-called mounting cross, *Figure 9*. This piece of equipment allows the skilled trainer to communicate the correct handling procedure visually, using a variety of different bearings, and under realistic conditions. The degree of difficulty corresponds to the basic training of persons who work on a regular basis with rolling bearings.

Figure 9
Training equipment:
Mounting cross

The mounting cross is of a modular structure and can be supplemented and expanded by a large number of different exercises. The initial configuration contains the basic tools required, the mounting cross itself and four different exercises on the most frequently used bearing types. Each one of the exercises contains the bearings, adjacent parts and tools required. Mechanical, thermal and hydraulic methods are communicated.

The training documents enclosed give precise explanations of the correct procedure and the correct use of bearings and tools. The necessary safety measures and alternative procedures are also explained.



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Tables

Dimension and tolerance symbols for radial rolling bearings in accordance with ISO 492:2014

Dimension symbol	Tolerance symbol	Description for radial bearings in accordance with ISO 492:2014	Old term in accordance with ISO 1132-1:2000
		Width	
B	–	Nominal inner ring width	Nominal inner ring width
	t_{VBs}	**Symmetrical rings** range of two-point sizes of inner ring width	Variation of inner ring width
		Asymmetrical rings range of minimum circumscribed sizes of inner ring width, between two opposite lines, obtained from any longitudinal section which includes the inner ring bore axis	
	$t_{\Delta Bs}$	**Symmetrical rings** deviation of a two-point size of inner ring width from its nominal size	Deviation of a single inner ring width
		Asymmetrical rings, upper limit deviation of a minimum circumscribed size of inner ring width, between two opposite lines, in any longitudinal section which includes the inner ring bore axis, from its nominal size	
		Asymmetrical rings, lower limit deviation of a two-point size of inner ring width from its nominal size	
C	–	Nominal outer ring width	Nominal outer ring width
	t_{VCs}	**Symmetrical rings** range of two-point sizes of outer ring width	Variation of outer ring width
		Asymmetrical rings range of minimum circumscribed sizes of outer ring width, between two opposite lines, obtained from any longitudinal section which includes the outer ring outside surface axis	
	$t_{\Delta Cs}$	**Symmetrical rings** deviation of a two-point size of outer ring width from its nominal size	Deviation of a single outer ring width
		Asymmetrical rings, upper limit deviation of a minimum circumscribed size of outer ring width, between two opposite lines, in any longitudinal section which includes the outer ring outside surface axis, from its nominal size	
		Asymmetrical rings, lower limit deviation of a two-point size of outer ring width from its nominal size	
C_1	–	Nominal outer ring flange width	Nominal outer ring flange width
	t_{VC1s}	Range of two-point sizes of outer ring flange width	Variation of outer ring flange width
	$t_{\Delta C1s}$	Deviation of a two-point size of outer ring flange width from its nominal size	Deviation of a single outer ring flange width

Dimension and tolerance symbols for radial rolling bearings in accordance with ISO 492:2014 (continued)

Dimension symbol	Tolerance symbol	Description for radial bearings in accordance with ISO 492:2014	Old term in accordance with ISO 1132-1:2000
		Diameter	
d	–	Nominal bore diameter of a cylindrical bore or at the theoretical small end of a tapered bore	Nominal bore diameter
	t_{Vdmp}	Range of mid-range sizes (out of two-point sizes) of bore diameter obtained from any cross-section of a cylindrical bore	Variation of mean bore diameter
	$t_{\Delta dmp}$	**Cylindrical bore** deviation of a mid-range size (out of two-point sizes) of bore diameter in any cross-section from its nominal size	Deviation of mean bore diameter in a single plane
		Tapered bore deviation of a mid-range size (out of two-point sizes) of bore diameter at the theoretical small end from its nominal size	
	t_{Vdsp}	Range of two-point sizes of bore diameter in any cross-section of a cylindrical or tapered bore	Variation of single bore diameter in a single plane
	$t_{\Delta ds}$	Deviation of a two-point size of bore diameter of a cylindrical bore from its nominal size	Deviation of a single bore diameter
d_1	–	Nominal diameter at the theoretical large end of a tapered bore	Diameter at the theoretical large end of a basically tapered bore
	$t_{\Delta d1mp}$	Deviation of a mid-range size (out of two-point sizes) of bore diameter at the theoretical large end of a tapered bore from its nominal size	Deviation of mean bore diameter in a single plane at the theoretical large end of a basically tapered bore
D	–	Nominal outside diameter	Nominal outside diameter
	t_{VDmp}	Range of mid-range sizes (out of two-point sizes) of outside diameter obtained from any cross-section	Variation of mean outside diameter
	$t_{\Delta Dmp}$	Deviation of a mid-range size (out of two-point sizes) of outside diameter in any cross-section from its nominal size	Deviation of mean outside diameter in a single plane
	t_{VDsp}	Range of two-point sizes of outside diameter in any cross-section	Variation of outside diameter in a single plane
	$t_{\Delta Ds}$	Deviation of a two-point size of outside diameter from its nominal size	Deviation of a single outside diameter
D_1	–	Nominal outside diameter of outer ring flange	Nominal outside diameter of outer ring flange
	$t_{\Delta D1s}$	Deviation of a two-point size of outside diameter of outer ring flange from its nominal size	Deviation of a single outside diameter of outer ring flange
		Tapered bore	
SL	–	Taper slope is the difference between nominal diameters at the theoretical large end and small end of a tapered bore ($SL = d_1 - d$)	–
	$t_{\Delta SL}$	Deviation of taper slope of a tapered inner ring bore from its nominal size ($\Delta SL = \Delta d1mp - \Delta dmp$)	–
α	–	Frustum angle of tapered inner ring bore (description based on ISO 1119)	–

Dimension and tolerance symbols for radial rolling bearings in accordance with ISO 492:2014 (continued)

Dimension symbol	Tolerance symbol	Description for radial bearings in accordance with ISO 492:2014	Old term in accordance with ISO 1132-1:2000
		Width of assembled bearing	
T	–	Nominal assembled bearing width	Assembled bearing width
	$t_{\Delta Ts}$	Deviation of minimum circumscribed size of assembled bearing width from its nominal size	Deviation of the actual (assembled) bearing width
T_1	–	Nominal effective width of inner subunit assembled with a master outer ring	Effective width of the inner subunit assembled with a master outer ring
	$t_{\Delta T1s}$	Deviation of minimum circumscribed size of effective width (inner subunit assembled with a master outer ring) from its nominal size	Nominal effective width of outer ring assembled with a master inner subunit
T_2	–	Effective width of outer ring assembled with a master inner subunit	Effective nominal width size of the outer ring, paired with an inner master unit
	$t_{\Delta T2s}$	Deviation of minimum circumscribed size of effective width (outer ring assembled with a master inner subunit) from its nominal size	Deviation of the actual effective width of outer ring assembled with a master inner subunit
T_F	–	Nominal assembled flanged bearing width	–
	$t_{\Delta TFs}$	Deviation of minimum circumscribed size of assembled flanged bearing width from its nominal size	–
T_{F2}	–	Nominal effective width of flanged outer ring assembled with a master inner subunit	–
	$t_{\Delta TF2s}$	Deviation of minimum circumscribed size of effective width (flanged outer ring assembled with a master inner subunit) from its nominal size	–
		Running accuracy	
	t_{Kea}	Circular radial run-out of outer ring outside surface of assembled bearing with respect to datum, i.e. axis, established from the inner ring bore surface	Radial run-out of outer ring of assembled bearing
	t_{Kia}	Circular radial run-out of inner ring bore of assembled bearing with respect to datum, i.e. axis, established from the outer ring outside surface	Radial run-out of inner ring of assembled bearing
	t_{Sd}	Circular axial run-out of inner ring face with respect to datum, i.e. axis, established from the inner ring bore surface	Perpendicularity of inner ring face with respect to the bore
	t_{SD}	Perpendicularity of outer ring outside surface axis with respect to datum established from the outer ring face	Perpendicularity of outer ring outside surface with respect to the face
	t_{SD1}	Perpendicularity of outer ring outside surface axis with respect to datum established from the outer ring flange back face	Perpendicularity of outer ring outside surface with respect to the flange back face
	t_{Sea}	Circular axial run-out of outer ring face of assembled bearing with respect to datum, i.e. axis, established from the inner ring bore surface	Axial run-out of outer ring of assembled bearing
	t_{Sea1}	Circular axial run-out of outer ring flange back face of assembled bearing with respect to datum, i.e. axis, established from the inner ring bore surface	Axial run-out of outer ring flange back face of assembled bearing
	t_{Sia}	Circular axial run-out of inner ring face of assembled bearing with respect to datum, i.e. axis, established from the outer ring outside surface	Axial run-out of inner ring of assembled bearing

Dimension and tolerance symbols for axial rolling bearings in accordance with ISO 199:2014

Dimension symbol	Tolerance symbol	Description for axial bearings in accordance with ISO 199:2014	Old term in accordance with ISO 1132-1:2000
		Diameter	
d	–	Nominal bore diameter of shaft washer, single-direction bearing	Nominal bore diameter of shaft washer
	$t_{\Delta dmp}$	Deviation of a mid-range size (out of two-point sizes) of shaft washer bore diameter in any cross-section from its nominal size	Deviation of mean bore diameter in a single plane
	t_{Vdsp}	Range of two-point sizes of shaft washer bore diameter in any cross-section	Variation of single bore diameter in a single plane
d_2	–	Nominal bore diameter of central shaft washer, double-direction bearing	Nominal bore diameter of central shaft washer
	$t_{\Delta d2mp}$	Deviation of a mid-range size (out of two-point sizes) of central shaft washer bore diameter in any cross-section from its nominal size	Deviation of mean bore diameter in a single plane
	t_{Vd2sp}	Range of two-point sizes of central shaft washer bore diameter in any cross-section	Variation of single bore diameter in a single plane
D	–	Nominal outside diameter of housing washer	Nominal outside diameter of housing washer
	$t_{\Delta Dmp}$	Deviation of a mid-range size (out of two-point sizes) of housing washer outside diameter in any cross-section from its nominal size	Deviation of mean outside diameter in a single plane
	t_{VDsp}	Range of two-point sizes of housing washer outside diameter in any cross-section	Variation of outside diameter in a single plane
		Height	
T	–	Nominal assembled bearing height, single-direction bearing	Nominal bearing height
	$t_{\Delta Ts}$	Deviation of minimum circumscribed size of assembled bearing height from its nominal size, single-direction bearing	Deviation of the actual bearing height
T_1	–	Nominal assembled bearing height, double-direction bearing	Nominal bearing height
	$t_{\Delta T1s}$	Deviation of minimum circumscribed size of assembled bearing height from its nominal size, double-direction bearing	Deviation of the actual bearing height
	t_{Se}	**Axial cylindrical roller bearings:** range of two-point sizes of thickness between housing washer raceway and the back face	Variation in thickness between housing washer raceway and back face
		Axial ball bearings: range of minimum spherical sizes between the raceway and the opposite back face of the housing washer	
	t_{Si}	**Axial cylindrical roller bearings:** range of two-point sizes of thickness between shaft washer raceway and the back face	Variation in thickness between shaft washer raceway and back face
		Axial ball bearings: range of minimum spherical sizes between the raceway and the opposite back face of the shaft washer	

Shaft and housing fits

Shaft fits

	Nominal shaft diameter in mm				
	over **3** incl. **6**	**6** **10**	**10** **18**	**18** **30**	**30** **50**
	Deviation of bearing bore diameter in µm (normal tolerance)				
	Δ_{dmp} 0 −8	0 −8	0 −8	0 −10	0 −12
	Shaft deviation, fit interference or fit clearance in µm				
e7	−20 −32	−25 −40	−32 −50	−40 −61	−50 −75
e8	−20 −38	−25 −47	−32 −59	−40 −73	−50 −89
f6	−10 −18	−13 −22	−16 −27	−20 −33	−25 −41
f7	−10 −22	−13 −28	−16 −34	−20 −41	−25 −50
g5	−4 −9	−5 −11	−6 −14	−7 −16	−9 −20
g6	−4 −12	−5 −14	−6 −17	−7 −20	−9 −25
h5	0 −5	0 −6	0 −8	0 −9	0 −11
h6	0 −8	0 −9	0 −11	0 −13	0 −16
j5	+3 −2	+4 −2	+5 −3	+5 −4	+6 −5
j6	+6 −2	+7 −2	+8 −3	+9 −4	+11 −5
js3	+1,25 +1,25	+1,25 +1,25	+1,5 +1,5	+2 −2	+2 −2
js4	+2 −2	+2 −2	+2,5 +2,5	+3 −3	+3,5 +3,5
js5	+2,5 −2,5	+3 −3	+4 −4	+4,5 −4,5	+5,5 −5,5
js6	+4 −4	+4,5 −4,5	+5,5 −5,5	+6,5 −6,5	+8 −8
k3	+2,5 0	+2,5 0	+3 0	+4 0	+4 0
k4	+5 +1	+5 +1	+6 +1	+8 +2	+9 +2
k5	+6 +1	+7 +1	+9 +1	+11 +2	+13 +2
k6	+9 +1	+10 +1	+12 +1	+15 +2	+18 +2

50	**65**	**80**	**100**	**120**	**140**	**160**	**180**	**200**	**225**	**250**	**280**
65	**80**	**100**	**120**	**140**	**160**	**180**	**200**	**225**	**250**	**280**	**315**
0	0	0	0	0	0	0	0	0	0	0	0
−15	−15	−20	−20	−25	−25	−25	−30	−30	−30	−35	−35
−60	−60	−72	−72	−85	−83	−85	−100	−100	−100	−110	−110
−90	−90	−107	−107	−125	−125	−125	−146	−146	−146	−162	−162
−60	−60	−72	−72	−85	−85	−85	−100	−100	−100	−110	−110
−106	−106	−126	−126	−148	−148	−148	−172	−172	−172	−191	−191
−30	−30	−36	−36	−43	−43	−43	−50	−50	−50	−56	−56
−49	−49	−58	−58	−68	−68	−68	−79	−79	−79	−88	−88
−30	−30	−36	−36	−43	−43	−43	−50	−50	−50	−56	−56
−60	−60	−71	−71	−83	−83	−83	−96	−96	−96	−108	−108
−10	−10	−12	−12	−14	−14	−14	−15	−15	−15	−17	−17
−23	−23	−27	−27	−32	−32	−32	−35	−35	−35	−40	−40
−10	−10	−12	−12	−14	−14	−14	−15	−15	−15	−17	−17
−29	−29	−34	−34	−39	−39	−39	−44	−44	−44	−49	−49
0	0	0	0	0	0	0	0	0	0	0	0
−13	−13	−15	−15	−18	−18	−18	−20	−20	−20	−23	−23
0	0	0	0	0	0	0	0	0	0	0	0
−19	−19	−22	−22	−25	−25	−25	−29	−29	−29	−32	−32
+6	+6	+6	+6	+7	+7	+7	+7	+7	+7	+7	+7
−7	−7	−9	−9	−11	−11	−11	−13	−13	−13	−16	−16
+12	+12	+13	+13	+14	+14	+14	+16	+16	+16	+16	+16
−7	−7	−9	−9	−11	−11	−11	−13	−13	−13	−16	−16
+2,5	+2,5	+3	+3	+4	+4	+4	+5	+5	+5	+6	+6
+2,5	+2,5	−3	−3	−4	−4	−4	−5	−5	−5	−6	−6
+4	+4	+5	+5	+6	+6	+6	+7	+7	+7	+8	+8
−4	−4	−5	−5	−6	−6	−6	−7	−7	−7	−8	−8
+6,5	+6,5	+7,5	+7,5	+9	+9	+9	+10	+10	+10	+11,5	+11,5
−6,5	−6,5	−7,5	−7,5	−9	−9	−9	−10	−10	−10	−11,5	−11,5
+9,5	+9,5	+11	+11	+12,5	+12,5	+12,5	+14,5	+14,5	+14,5	+16	+16
−9,5	−9,5	−11	−11	−12,5	−12,5	−12,5	−14,5	−14,5	−14,5	−16	−16
+5	+5	+6	+6	+8	+8	+8	+10	+10	+10	+12	+12
0	0	0	0	0	0	0	0	0	0	0	0
+10	+10	+13	+13	+15	+15	+15	+18	+18	+18	+20	+20
+2	+2	+3	+3	+3	+3	+3	+4	+4	+4	+4	+4
+15	+15	+18	+18	+21	+21	+21	+24	+24	+24	+27	+27
+2	+2	+3	+3	+3	+3	+3	+4	+4	+4	+4	+4
+21	+21	+25	+25	+28	+28	+28	+33	+33	+33	+36	+36
+2	+2	+3	+3	+3	+3	+3	+4	+4	+4	+4	+4

Shaft fits
(continued)

Nominal shaft diameter in mm				
over	**315**	**355**	**400**	**450**
incl.	**355**	**400**	**450**	**500**
Deviation of bearing bore diameter in μm (normal tolerance)				
Δ_{dmp}	0 −40	0 −40	0 −45	0 −45
Shaft deviation, fit interference or fit clearance in μm				
e7	−125 −182	−125 −182	−135 −198	−135 −198
e8	−125 −214	−125 −214	−135 −232	−135 −232
f6	−62 −98	−62 −98	−68 −108	−68 −108
f7	−62 −119	−62 −119	−68 −131	−68 −131
g5	−18 −43	−18 −43	−20 −47	−20 −47
g6	−18 −54	−18 −54	−20 −60	−20 −60
h5	0 −25	0 −25	0 −27	0 −27
h6	0 −36	0 −36	0 −40	0 −40
j5	+7 −18	+7 −18	+7 −20	+7 −20
j6	+18 −18	+18 −18	+20 −20	+20 −20
js3	+6,5 −6,5	+6,5 −6,5	+7,5 −7,5	+7,5 −7,5
js4	+9 −9	+9 −9	+10 −10	+10 −10
js5	+12,5 −12,5	+12,5 −12,5	+13,5 −13,5	+13,5 −13,5
js6	+18 −18	+18 −18	+20 −20	+20 −20
k3	+13 0	+13 0	+15 0	+15 0
k4	+22 +4	+22 +4	+25 +5	+25 +5
k5	+29 +4	+29 +4	+32 +5	+32 +5
k6	+40 +4	+40 +4	+45 +5	+45 +5

500 560	560 630	630 710	710 800	800 900	900 1 000	1 000 1 120	1 120 1 250
0 −50	0 −50	0 −75	0 −75	0 −100	0 −100	0 −125	0 −125
−145 −215	−145 −215	−160 −240	−160 −240	−170 −260	−170 −260	−195 −300	−195 −300
−145 −255	−145 −255	−160 −285	−160 −285	−170 −310	−170 −310	−195 −360	−195 −360
−76 −120	−76 −120	−80 −130	−80 −130	−86 −142	−86 −142	−98 −164	−98 −164
−76 −146	−76 −146	−80 −160	−80 −160	−86 −176	−86 −176	−98 −203	−98 −203
−22 −51	−22 −51	−24 −56	−24 −56	−26 −62	−26 −62	−28 −70	−28 −70
−22 −66	−22 −66	−24 −74	−24 −74	−26 −82	−26 −82	−28 −94	−28 −94
0 −29	0 −29	0 −32	0 −32	0 −36	0 −36	0 −42	0 −42
0 −44	0 −44	0 −50	0 −50	0 −56	0 −56	0 −66	0 −66
–	–	–	–	–	–	–	–
+22 −22	+22 −22	+25 −25	+25 −25	+28 −28	+28 −28	+33 −33	+33 −33
–	–	–	–	–	–	–	–
–	–	–	–	–	–	–	–
+14,5 −14,5	+14,5 −14,5	+16 −16	+16 −16	+18 −18	+18 −18	+21 −21	+21 −21
+22 −22	+22 −22	+25 −25	+25 −25	+28 −28	+28 −28	+33 −33	+33 −33
–	–	–	–	–	–	–	–
–	–	–	–	–	–	–	–
+29 0	+29 0	+32 0	+32 0	+36 0	+36 0	+42 0	+42 0
+44 0	+44 0	+50 0	+50 0	+56 0	+56 0	+66 0	+66 0

Shaft and housing fits

Shaft fits
(continued)

	Nominal shaft diameter in mm				
	over **3** incl. **6**	**6** **10**	**10** **18**	**18** **30**	**30** **50**
	Deviation of bearing bore diameter in µm (normal tolerance)				
Δ_{dmp}	0 −8	0 −8	0 −8	0 −10	0 −12
	Shaft deviation, fit interference or fit clearance in µm				
m5	+9 +4	+12 +6	+15 +7	+17 +8	+20 +9
m6	+12 +4	+15 +6	+18 +7	+21 +8	+25 +9
n5	+13 +8	+16 +10	+20 +12	+24 +15	+28 +17
n6	+16 +8	+19 +10	+23 +12	+28 +15	+33 +17
p6	+20 +12	+24 +15	+29 +18	+35 +22	+42 +26
p7	+24 +12	+30 +15	+36 +18	+43 +22	+51 +26
r6	+23 +15	+28 +19	+34 +23	+41 +28	+50 +34
r7	+27 +15	+34 +19	+41 +23	+49 +28	+59 +34
s6	+27 +19	+32 +23	+39 +28	+48 +35	+59 +43
s7	+31 +19	+38 +23	+46 +28	+56 +35	+68 +43
Shaft tolerances for adapter sleeves and withdrawal sleeves					
h7/ $\frac{IT5}{2}$	0 −12 *2,5*	0 −15 *3*	0 −18 *4*	0 −21 *4,5*	0 −25 *5,5*
h8/ $\frac{IT5}{2}$	0 −18 *2,5*	0 −22 *3*	0 −27 *4*	0 −33 *4,5*	0 −39 *5,5*
h9/ $\frac{IT6}{2}$	0 −30 *4*	0 −36 *4,5*	0 −43 *5,5*	0 −52 *6,5*	0 −62 *8*
h10/	0 −48 *6*	0 −58 *7,5*	0 −70 *9*	0 −84 *10,5*	0 −100 *12,5*

The cylindricity tolerance t_1 (values printed in *italic*) relates to the radius (DIN ISO 1101).
When measuring the shaft diameter, the tolerance values should be doubled.
For general machine building, the values h7 or h8 should be used in preference.

50 **65**	**65** **80**	**80** **100**	**100** **120**	**120** **140**	**140** **160**	**160** **180**	**180** **200**	**200** **225**	**225** **250**	**250** **280**	**280** **315**
0 −15	0 −15	0 −20	0 −20	0 −25	0 −25	0 −25	0 −30	0 −30	0 −30	0 −35	0 −35
+24 +11	+24 +11	+28 +13	+28 +13	+33 +15	+33 +15	+33 +15	+37 +17	+37 +17	+37 +17	+43 +20	+43 +20
+30 +11	+30 +11	+35 +13	+35 +13	+40 +15	+40 +15	+40 +15	+46 +17	+46 +17	+46 +17	+52 +20	+52 +20
+33 +20	+33 +20	+38 +23	+38 +23	+45 +27	+45 +27	+45 +27	+51 +31	+51 +31	+51 +31	+57 +34	+57 +34
+39 +20	+39 +20	+45 +23	+45 +23	+52 +27	+52 +27	+52 +27	+60 +31	+60 +31	+60 +31	+66 +34	+66 +34
+51 +32	+51 +32	+59 +37	+59 +37	+68 +43	+68 +43	+68 +43	+79 +50	+79 +50	+79 +50	+88 +56	+88 +56
+62 +32	+62 +32	+72 +37	+72 +37	+83 +43	+83 +43	+83 +43	+96 +50	+96 +50	+96 +50	+108 +56	+108 +56
+60 +41	+62 +43	+73 +51	+76 +54	+88 +63	+90 +65	+93 +68	+106 +77	+109 +80	+113 +84	+126 +94	+130 +98
+71 +41	+73 +43	+86 +51	+89 +54	+103 +63	+105 +65	+108 +68	+123 +77	+126 +80	+130 +84	+146 +94	+150 +98
+72 +53	+78 +59	+93 +71	+101 +79	+117 +92	+125 +100	+133 +108	+151 +122	+126 +80	+130 +84	+146 +94	+150 +98
+83 +53	+89 +59	+106 +71	+114 +79	+132 +92	+140 +100	+148 +108	+168 +122	+126 +80	+130 +84	+146 +94	+150 +98
0 −30 *6,5*	0 −30 *6,5*	0 −35 *7,5*	0 −35 *7,5*	0 −40 *9*	0 −40 *9*	0 −40 *9*	0 −46 *10*	0 −46 *10*	0 −46 *10*	0 −52 *11,5*	0 −52 *11,5*
0 −46 *6,5*	0 −46 *6,5*	0 −54 *7,5*	0 −54 *7,5*	0 −63 *9*	0 −63 *9*	0 −63 *9*	0 −72 *10*	0 −72 *10*	0 −72 *10*	0 −81 *11,5*	0 −81 *11,5*
0 −74 *9,5*	0 −74 *9,5*	0 −87 *11*	0 −87 *11*	0 −100 *12,5*	0 −100 *12,5*	0 −100 *12,5*	0 −115 *14,5*	0 −115 *14,5*	0 −115 *14,5*	0 −130 *16*	0 −130 *16*
0 −120 *15*	0 −120 *15*	0 −140 *17,5*	0 −140 *17,5*	0 −160 *20*	0 −160 *20*	0 −160 *20*	0 −185 *23*	0 −185 *23*	0 −185 *23*	0 −210 *26*	0 −210 *26*

Shaft fits
(continued)

	Nominal shaft diameter in mm			
	over 315 incl. 355	over 355 incl. 400	over 400 incl. 450	over 450 incl. 500
Deviation of bearing bore diameter in μm (normal tolerance)				
Δ_{dmp}	0 –40	0 –40	0 –45	0 –45
Shaft deviation, fit interference or fit clearance in μm				
m5	+46 +21	+46 +21	+50 +23	+50 +23
m6	+57 +21	+57 +21	+63 +23	+63 +23
n5	+62 +37	+62 +37	+67 +40	+67 +40
n6	+73 +37	+73 +37	+80 +40	+80 +40
p6	+98 +62	+98 +62	+108 +68	+108 +68
p7	+119 +62	+119 +62	+131 +68	+131 +68
r6	+144 +108	+150 +114	+166 +126	+172 +132
r7	+165 +108	+171 +114	+189 +126	+195 +132
s6	+165 +108	+171 +114	+189 +126	+195 +132
s7	+165 +108	+171 +114	+189 +126	+195 +132
Shaft tolerances for adapter sleeves and withdrawal sleeves				
h7/ $\frac{IT5}{2}$	0 –57 *12,5*	0 –57 *12,5*	0 –63 *13,5*	0 –63 *13,5*
h8/ $\frac{IT5}{2}$	0 –89 *12,5*	0 –89 *12,5*	0 –97 *13,5*	0 –97 *13,5*
h9/ $\frac{IT6}{2}$	0 –140 *18*	0 –140 *18*	0 –155 *20*	0 –155 *20*
h10/	0 –230 *28,5*	0 –230 *28,5*	0 –250 *31,5*	0 –250 *31,5*

The cylindricity tolerance t_1 (values printed in *italic*) relates to the radius (DIN ISO 1101).
When measuring the shaft diameter, the tolerance values should be doubled.
For general machine building, the values h7 or h8 should be used in preference.

500 **560**	**560** **630**	**630** **710**	**710** **800**	**800** **900**	**900** **1 000**	**1 000** **1 120**	**1 120** **1 250**
0 −50	0 −50	0 −75	0 −75	0 −100	0 −100	0 −125	0 −125
+55 +26	+55 +26	+62 +30	+62 +30	+70 +34	+70 +34	+82 +40	+82 +40
+70 +26	+70 +26	+80 +30	+80 +30	+90 +34	+90 +34	+106 +40	+106 +40
+73 +44	+73 +44	+82 +50	+82 +50	+92 +56	+92 +56	+108 +66	+108 +66
+88 +44	+88 +44	+100 +50	+100 +50	+112 +56	+112 +56	+132 +66	+132 +66
+122 +78	+122 +78	+138 +88	+138 +88	+156 +100	+156 +100	+186 +120	+186 +120
+148 +78	+148 +78	+168 +88	+168 +88	+190 +100	+190 +100	+225 +120	+225 +120
+194 +150	+199 +155	+225 +175	+235 +185	+266 +210	+276 +220	+316 +250	+326 +260
+220 +150	+225 +155	+255 +175	+265 +185	+300 +210	+310 +220	+355 +250	+365 +260
+220 +150	+225 +155	+255 +175	+265 +185	+300 +210	+310 +220	+355 +250	+365 +260
+220 +150	+225 +155	+255 +175	+265 +185	+300 +210	+310 +220	+355 +250	+365 +260
0 −70 *14,5*	0 −70 *14,5*	0 −80 *16*	0 −80 *16*	0 −90 *18*	0 −90 *18*	0 −105 *21*	0 −105 *21*
0 −110 *14,5*	0 −110 *14,5*	0 −125 *16*	0 −125 *16*	0 −140 *18*	0 −140 *18*	0 −165 *21*	0 −165 *21*
0 −175 *22*	0 −175 *22*	0 −200 *25*	0 −200 *25*	0 −230 *28*	0 −230 *28*	0 −260 *33*	0 −260 *33*
0 −280 *35*	0 −280 *35*	0 −320 *40*	0 −320 *40*	0 −360 *45*	0 −360 *45*	0 −420 *52,5*	0 −420 *52,5*

Housing fits

	Nominal housing bore diameter in mm				
over incl.	**6** **10**	**10** **18**	**18** **30**	**30** **50**	**50** **80**
	Deviation of bearing outside diameter in μm (normal tolerance)				
Δ_{Dmp}	0 −8	0 −8	0 −9	0 −11	0 −13
	Housing deviation, fit interference or fit clearance in μm				
D10	+98 +40	+120 +50	+149 +65	+180 +80	+220 +100
E8	+47 +25	+59 +32	+73 +40	+89 +50	+106 +60
F7	+28 +13	+34 +16	+41 +20	+50 +25	+60 +30
G6	+14 +5	+17 +6	+20 +7	+25 +9	+29 +10
G7	+20 +5	+24 +6	+28 +7	+34 +9	+40 +10
H5	+6 0	+8 0	+9 0	+11 0	+13 0
H6	+9 0	+11 0	+13 0	+16 0	+19 0
H7	+15 0	+18 0	+21 0	+25 0	+30 0
H8	+22 0	+27 0	+33 0	+39 0	+46 0
J6	+5 −4	+6 −5	+8 −5	+10 −6	+13 −6
J7	+8 −7	+10 −8	+12 −9	+14 −11	+18 −12
JS4	+2 −2	+2,5 −2,5	+3 −3	+3,5 −3,5	+4 −4
JS5	+3 −3	+4 −4	+4,5 −4,5	+5,5 −5,5	+6,5 −6,5
JS6	+4,5 −4,5	+5,5 −5,5	+6,5 −6,5	+8 −8	+9,5 −9,5
JS7	+7,5 −7,5	+9 −9	+10,5 −10,5	+12,5 −12,5	+15 −15

80 120	120 150	150 180	180 250	250 315	315 400	400 500	500 630	630 800	800 1 000	1 000 1 250	1 250 1 600
0 –15	0 –18	0 –25	0 –30	0 –35	0 –40	0 –45	0 –50	0 –75	0 –100	0 –125	0 –160
+260 +120	+305 +145	+305 +145	+355 +170	+400 +190	+440 +210	+480 +230	+540 +260	+610 +290	+680 +320	+770 +350	+890 +390
+126 +72	+148 +85	+148 +85	+172 +100	+191 +110	+214 +125	+232 +135	+255 +145	+285 +160	+310 +170	+360 +195	+415 +220
+71 +36	+83 +43	+83 +43	+96 +50	+108 +56	+119 +62	+131 +68	+146 +76	+160 +80	+176 +86	+203 +98	+235 +110
+34 +12	+39 +14	+39 +14	+44 +15	+49 +17	+54 +18	+60 +20	+66 +22	+74 +24	+82 +26	+94 +28	+108 +30
+47 +12	+54 +14	+54 +14	+61 +15	+69 +17	+75 +18	+83 +20	+92 +22	+104 +24	+116 +26	+133 +28	+155 +30
+15 0	+18 0	+18 0	+20 0	+23 0	+25 0	+27 0	–	–	–	–	–
+22 0	+25 0	+25 0	+29 0	+32 0	+36 0	+40 0	+44 0	+50 0	+56 0	+66 0	+78 0
+35 0	+40 0	+40 0	+46 0	+52 0	+57 0	+63 0	+70 0	+80 0	+90 0	+105 0	+125 0
+54 0	+63 0	+63 0	+72 0	+81 0	+89 0	+97 0	+110 0	+125 0	+140 0	+165 0	+195 0
+16 –6	+18 –7	+18 –7	+22 –7	+25 –7	+29 –7	+33 –7	–	–	–	–	–
+22 –13	+26 –14	+26 –14	+30 –16	+36 –16	+39 –18	+43 –20	–	–	–	–	–
+5 –5	+6 –6	+6 –6	+7 –7	+8 –8	+9 –9	+10 –10	–	–	–	–	–
+7,5 –7,5	+9 –9	+9 –9	+10 –10	+11,5 –11,5	+12,5 –12,5	+13,5 –13,5	–	–	–	–	–
+11 –11	+12,5 –12,5	+12,5 –12,5	+14,5 –14,5	+16 –16	+18 –18	+20 –20	+22 –22	+25 –25	+28 –28	+33 –33	+39 –39
+17,5 –17,5	+20 –20	+20 –20	+23 –23	+26 –26	+28,5 –28,5	+31,5 –31,5	+35 –35	+40 –40	+45 –45	+52 –52	+62 –62

Housing fits
(continued)

	Nominal housing bore diameter in mm				
	over **6** incl. **10**	**10** **18**	**18** **30**	**30** **50**	**50** **80**
	Deviation of bearing outside diameter in µm (normal tolerance)				
Δ_{Dmp}	0 −8	0 −8	0 −9	0 −11	0 −13
	Housing deviation, fit interference or fit clearance in µm				
K4	+0,5 −3,5	+1 −4	0 −6	+1 −6	+1 −7
K5	+1 −5	+2 −6	+1 −8	+2 −9	+3 −10
K6	+2 −7	+2 −9	+2 −11	+3 −13	+4 −15
K7	+5 −10	+6 −12	+6 −15	+7 −18	+9 −21
M6	−3 −12	−4 −15	−4 −17	−4 −20	−5 −24
M7	0 −15	0 −18	0 −21	0 −25	0 −30
N6	−7 −16	−9 −20	−11 −24	−12 −28	−14 −33
N7	−4 −19	−5 −23	−7 −28	−8 −33	−9 −39
P6	−12 −21	−15 −26	−18 −31	−21 −37	−26 −45
P7	−9 −24	−11 −29	−14 −35	−17 −42	−21 −51
R6	−16 −25	−20 −31	−24 −37	−29 −45	−35 −54
S7	−20 −29	−25 −36	−31 −44	−38 −54	−47 −66

80 120	120 150	150 180	180 250	250 315	315 400	400 500	500 630	630 800	800 1 000	1 000 1 250	1 250 1 600
0 −15	0 −18	0 −25	0 −30	0 −35	0 −40	0 −45	0 −50	0 −75	0 −100	0 −125	0 −160
+1 −9	+1 −11	+1 −11	0 −14	0 −16	0 −17	0 −20	–	–	–	–	–
+2 −13	+3 −15	+3 −15	+2 −18	+3 −20	+3 −22	+2 −25	–	–	–	–	–
+4 −18	+4 −21	+4 −21	+5 −24	+5 −27	+7 −29	+8 −32	0 −44	0 −50	0 −56	0 −66	0 −78
+10 −25	+12 −28	+12 −28	+13 −33	+16 −36	+17 −40	+18 −45	0 −70	0 −80	0 −90	0 −105	0 −125
−6 −28	−8 −33	−8 −33	−8 −37	−9 −41	−10 −46	−10 −50	−26 −70	−30 −80	−34 −90	−40 −106	−48 −126
0 −35	0 −40	0 −40	0 −46	0 −52	0 −57	0 −63	−26 −96	−30 −110	−34 −124	−40 −145	−48 −173
−16 −38	−20 −45	−20 −45	−22 −51	−25 −57	−26 −62	−27 −67	−44 −88	−50 −100	−56 −112	−66 −132	−78 −156
−10 −45	−12 −52	−12 −52	−14 −60	−14 −66	−16 −73	−17 −80	−44 −114	−50 −130	−56 −146	−66 −171	−78 −203
−30 −52	−36 −61	−36 −61	−41 −70	−47 −79	−51 −87	−55 −95	−78 −122	−88 −138	−100 −156	−120 −186	−140 −218
−24 −59	−28 −68	−28 −68	−33 −79	−36 −88	−41 −98	−45 −108	−78 −148	−88 −168	−100 −190	−120 −225	−140 −265
−44 −66	−56 −81	−61 −86	−68 −97	−85 −117	−97 −133	−113 −153	−150 −194	−175 −225	−210 −266	−250 −316	−300 −378
−64 −86	−85 −110	−101 −126	−113 −142	−149 −181	−179 −215	−219 −259	–	–	–	–	–

Normal tolerances for FAG radial bearings (excluding FAG tapered roller bearings)

Normal tolerances for FAG radial bearings, excluding tapered roller bearings.

Inner ring tolerances

Bore d mm		Bore deviation $t_{\Delta dmp}$ µm Deviation		Variation t_{Vdsp} µm Diameter series			Variation t_{Vdmp} µm	Runout t_{Kia} µm	Deviation of inner ring width $t_{\Delta Bs}$ µm Deviation				Variation t_{VBs} µm
				9	0, 1	2, 3, 4			normal		modified[1]		
over	incl.	upper	lower	max.	max.	max.	max.	max.	upper	lower	upper	lower	max.
0,6[2]	2,5	0	–8	10	8	6	6	10	0	–40	0	–	12
2,5	10	0	–8	10	8	6	6	10	0	–120	0	–250	15
10	18	0	–8	10	8	6	6	10	0	–120	0	–250	20
18	30	0	–10	13	10	8	8	13	0	–120	0	–250	20
30	50	0	–12	15	12	9	9	15	0	–120	0	–250	20
50	80	0	–15	19	19	11	11	20	0	–150	0	–380	25
80	120	0	–20	25	25	15	15	25	0	–200	0	–380	25
120	180	0	–25	31	31	19	19	30	0	–250	0	–500	30
180	250	0	–30	38	38	23	23	40	0	–300	0	–500	30
250	315	0	–35	44	44	26	26	50	0	–350	0	–500	35
315	400	0	–40	50	50	30	30	60	0	–400	0	–630	40
400	500	0	–45	56	56	34	34	65	0	–450	0	–	50
500	630	0	–50	63	63	38	38	70	0	–500	0	–	60
630	800	0	–75	–	–	–	–	80	0	–750	0	–	70
800	1 000	0	–100	–	–	–	–	90	0	–1 000	0	–	80
1 000	1 250	0	–125	–	–	–	–	100	0	–1 250	0	–	100
1 250	1 600	0	–160	–	–	–	–	120	0	–1 600	0	–	120
1 600	2 000	0	–200	–	–	–	–	140	0	–2 000	0	–	140

[1] Only for bearings manufactured specifically for use as matched pairs.
[2] This diameter is included in the group.

Outer ring tolerances[1]

Outside diameter D mm		Deviation of outside diameter $t_{\Delta Dmp}$ µm Deviation		Variation					Runout
				t_{VDsp} µm				t_{VDmp}[2] µm	t_{Kea} µm
				Open bearings Diameter series			Bearings with sealing shields or sealing washers		
				9	0, 1	2, 3, 4			
over	incl.	upper	lower	max.	max.	max.	max.	max.	max.
2,5[3]	6	0	−8	10	8	6	10	6	15
6	18	0	−8	10	8	6	10	6	15
18	30	0	−9	12	9	7	12	7	15
30	50	0	−11	14	11	8	16	8	20
50	80	0	−13	16	13	10	20	10	25
80	120	0	−15	19	19	11	26	11	35
120	150	0	−18	23	23	14	30	14	40
150	180	0	−25	31	31	19	38	19	45
180	250	0	−30	38	38	23	–	23	50
250	315	0	−35	44	44	26	–	26	60
315	400	0	−40	50	50	30	–	30	70
400	500	0	−45	56	56	34	–	34	80
500	630	0	−50	63	63	38	–	38	100
630	800	0	−75	94	94	55	–	55	120
800	1 000	0	−100	125	125	75	–	75	140
1 000	1 250	0	−125	–	–	–	–	–	160
1 250	1 600	0	−160	–	–	–	–	–	190
1 600	2 000	0	−200	–	–	–	–	–	220
2 000	2 500	0	−250	–	–	–	–	–	250

1) Δ_{Cs}, Δ_{C1s}, V_{Cs} and V_{C2s} are identical to Δ_{Bs} and V_{Bs} for the inner ring of the corresponding bearing (table Tolerance class Normal Inner ring, page 152).

2) Applies before assembly of the bearing and after removal of internal and/or external snap rings.

3) This diameter is included in the group.

Normal tolerances for FAG tapered roller bearings in metric sizes

The main dimensions conform to ISO 355 and DIN 720, the dimensional and running tolerances conform to ISO 492:2014. These values are only valid for bearings in metric sizes.

Width tolerance to tolerance class Normal

Single row tapered roller bearings 302, 303, 313, 322, 323, T2EE, T4CB, T4DB, T5ED and T7FC correspond to the tolerance class Normal.

Bearings 320, 329, 330, 331 and 332 for shaft diameters over 200 mm have width tolerances to the tolerance class Normal. Bearings for shaft diameters < 200 mm have width tolerances to the tolerance class 6X, see table, page 157.

Inner ring tolerances

Bore d mm		Bore deviation $t_{\Delta dmp}$ µm		Variation t_{Vdsp} µm	Variation t_{Vdmp} µm	Runout t_{Kia} µm
over	incl.	max.	min.	max.	max.	max.
–	10	0	−12	12	9	15
10	18	0	−12	12	9	15
18	30	0	−12	12	9	18
30	50	0	−12	12	9	20
50	80	0	−15	15	11	25
80	120	0	−20	20	15	30
120	180	0	−25	25	19	35
180	250	0	−30	30	23	50
250	315	0	−35	35	26	60
315	400	0	−40	40	30	70
400	500	0	−45	45	34	80
500	630	0	−60	60	40	90
630	800	0	−75	75	45	100
800	1 000	0	−100	100	55	115
1 000	1 250	0	−125	125	65	130
1 250	1 600	0	−160	160	80	150
1 600	2 000	0	−200	200	100	170

Width tolerances

Bore		Deviation of inner ring width		Width deviation					
d		$t_{\Delta Bs}$		$t_{\Delta Ts}$		$t_{\Delta T1s}$		$t_{\Delta T2s}$	
mm		μm		μm		μm		μm	
over	incl.	max.	min.	max.	min.	max.	min.	max.	min.
–	10	0	–120	+200	0	+100	0	+100	0
10	18	0	–120	+200	0	+100	0	+100	0
18	30	0	–120	+200	0	+100	0	+100	0
30	50	0	–120	+200	0	+100	0	+100	0
50	80	0	–150	+200	0	+100	0	+100	0
80	120	0	–200	+200	–200	+100	–100	+100	–100
120	180	0	–250	+350	–250	+150	–150	+200	–100
180	250	0	–300	+350	–250	+150	–150	+200	–100
250	315	0	–350	+350	–250	+150	–150	+200	–100
315	400	0	–400	+400	–400	+200	–200	+200	–200
400	500	0	–450	+450	–450	+225	–225	+225	–225
500	630	0	–500	+500	–500	–	–	–	–
630	800	0	–750	+600	–600	–	–	–	–
800	1 000	0	–1 000	+750	–750	–	–	–	–
1 000	1 250	0	–1 250	+900	–900	–	–	–	–
1 250	1 600	0	–1 600	+1 050	–1 050	–	–	–	–
1 600	2 000	0	–2 000	+1 200	–1 200	–	–	–	–

Normal tolerances

Outer ring tolerances

Outside diameter D mm		Deviation of outside diameter $t_{\Delta Dmp}$ µm		Variation		Runout
				t_{VDsp} µm	t_{VDmp} µm	t_{Kea} µm
over	incl.	max.	min.	max.	max.	max.
–	18	0	–12	12	9	18
18	30	0	–12	12	9	18
30	50	0	–14	14	11	20
50	80	0	–16	16	12	25
80	120	0	–18	18	14	35
120	150	0	–20	20	15	40
150	180	0	–25	25	19	45
180	250	0	–30	30	23	50
250	315	0	–35	35	26	60
315	400	0	–40	40	30	70
400	500	0	–45	45	34	80
500	630	0	–50	60	38	100
630	800	0	–75	80	55	120
800	1 000	0	–100	100	75	140
1 000	1 250	0	–125	130	90	160
1 250	1 600	0	–160	170	100	180
1 600	2 000	0	–200	210	110	200
2 000	2 500	0	–250	265	120	220

The width tolerance Δ_{Cs} is identical to Δ_{Bs} for the inner ring of the same bearing.

Width tolerance to tolerance class 6X

Tapered roller bearings 320, 329, 330, 331 and 332 for shaft diameters up to 200 mm and inch size bearings with the code KJ have restricted width tolerances to the tolerance class 6X.

Width tolerances

Bore		Deviation of inner ring width		Width deviation							
d		$t_{\Delta Bs}$		$t_{\Delta Cs}$		$t_{\Delta Ts}$		$t_{\Delta T1s}$		$t_{\Delta T2s}$	
mm		µm		µm		µm		µm		µm	
over	incl.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.
–	10	0	–50	0	–100	+100	0	+50	0	+50	0
10	18	0	–50	0	–100	+100	0	+50	0	+50	0
18	30	0	–50	0	–100	+100	0	+50	0	+50	0
30	50	0	–50	0	–100	+100	0	+50	0	+50	0
50	80	0	–50	0	–100	+100	0	+50	0	+50	0
80	120	0	–50	0	–100	+100	0	+50	0	+50	0
120	180	0	–50	0	–100	+150	0	+50	0	+100	0
180	250	0	–50	0	–100	+150	0	+50	0	+100	0
250	315	0	–50	0	–100	+200	0	+100	0	+100	0
315	400	0	–50	0	–100	+200	0	+100	0	+100	0
400	500	0	–50	0	–100	+200	0	+100	0	+100	0

Normal tolerances

Restricted tolerance class 5

Tapered roller bearings with restricted tolerances correspond to the tolerance class 5 to ISO 492:2014.

Inner ring tolerances

Bore d mm		Bore deviation $t_{\Delta dmp}$ µm		Variation		Runout
				t_{Vdsp} µm	t_{Vdmp} µm	t_{Kia} µm
over	incl.	max.	min.	max.	max.	max.
–	10	0	–7	5	5	5
10	18	0	–7	5	5	5
18	30	0	–8	6	5	5
30	50	0	–10	8	5	6
50	80	0	–12	9	6	7
80	120	0	–15	11	8	8
120	180	0	–18	14	9	11
180	250	0	–22	17	11	13
250	315	0	–25	19	13	13
315	400	0	–30	23	15	15
400	500	0	–35	28	17	20
500	630	0	–40	35	20	25
630	800	0	–50	45	25	30
800	1 000	0	–60	60	30	37
1 000	1 250	0	–75	75	37	45
1 250	1 600	0	–90	90	45	55

Width tolerances

Bore d mm		Deviation of inner ring width $t_{\Delta Bs}$ µm		Deviation of bearing width $t_{\Delta Ts}$ µm	
over	incl.	max.	min.	max.	min.
–	10	0	–200	+200	–200
10	18	0	–200	+200	–200
18	30	0	–200	+200	–200
30	50	0	–240	+200	–200
50	80	0	–300	+200	–200
80	120	0	–400	+200	–200
120	180	0	–500	+350	–250
180	250	0	–600	+350	–250
250	315	0	–700	+350	–250
315	400	0	–800	+400	–400
400	500	0	–900	+450	–450
500	630	0	–1 100	+500	–500
630	800	0	–1 600	+600	–600
800	1 000	0	–2 000	+750	–750
1 000	1 250	0	–2 000	+750	–750
1 250	1 600	0	–2 000	+900	–900

Outer ring tolerances

Outside diameter D mm		Deviation of outside diameter $t_{\Delta Dmp}$ µm		Variation t_{VDsp} µm	 t_{VDmp} µm	Runout t_{Kea} µm
over	incl.	max.	min.	max.	max.	max.
–	18	0	–8	6	5	6
18	30	0	–8	6	5	6
30	50	0	–9	7	5	7
50	80	0	–11	8	6	8
80	120	0	–13	10	7	10
120	150	0	–15	11	8	11
150	180	0	–18	14	9	13
180	250	0	–20	15	10	15
250	315	0	–25	19	13	18
315	400	0	–28	22	14	20
400	500	0	–33	26	17	24
400	500	0	–38	30	20	30
500	630	0	–45	38	25	36
630	800	0	–60	50	30	43
800	1 000	0	–80	65	38	52
1 000	1 250	0	–100	90	50	62
1 250	1 600	0	–125	120	65	73

Normal tolerances for FAG tapered roller bearings to ANSI/ABMA

Tapered roller bearings of series K are manufactured as standard with normal tolerances based on ANSI/ABMA.
Exception: series KJ = 6X.
The width Δ_{Bs} and radial runout correspond to the tolerance class Normal to ISO 492:2014.
The bore and outside diameters of bearings in inch sizes have plus tolerances.

Inner ring tolerances

Bore d mm		Bore deviation $t_{\Delta dmp}$ µm		Runout t_{Kia} µm
over	incl.	max.	min.	
10	18	13	0	15
18	30	13	0	18
30	50	13	0	20
50	81	13	0	25
81	120	25	0	30
120	180	25	0	35
180	305	25	0	50
305	400	50	0	50

Width tolerances

Bore d mm		Deviation of inner ring width (in relation to bore) $t_{\Delta Bs}$ µm		Deviation of bearing width $t_{\Delta Ts}$ µm	
over	incl.	max.	min.	max.	min.
10	50	0	−120	+200	0
50	81	0	−150	+200	0
81	102	0	−200	+200	0
102	120	0	−200	+350	−250
120	180	0	−250	+350	−250
180	250	0	−300	+350	−250
250	305	0	−350	+350	−250
305	315	0	−350	+375	−375
315	400	0	−400	+375	−375

Outer ring tolerances

Outside diameter D mm		Deviation of outside diameter $t_{\Delta Dmp}$ µm		Runout t_{Kea} µm
over	incl.	max.	min.	
18	30	+25	0	18
30	50	+25	0	20
50	81	+25	0	25
81	120	+25	0	35
120	150	+25	0	40
150	180	+25	0	45
180	250	+25	0	50
250	305	+25	0	50
305	400	+50	0	50

Normal tolerances for axial bearings

The normal tolerances for axial bearings correspond to ISO 199, DIN 620-3.

Bore diameter tolerances for shaft locating washers

Bore d mm		Bore deviation $t_{\Delta dmp}$ µm Tolerance class				Variation t_{Vdp} µm Tolerance class	
		Normal, 6 and 5 Deviation		4 Deviation		Normal, 6 and 5	4
over	incl.	upper	lower	upper	lower	max.	max.
–	18	0	–8	0	–7	6	5
18	30	0	–10	0	–8	8	6
30	50	0	–12	0	–10	9	8
50	80	0	–15	0	–12	11	9
80	120	0	–20	0	–15	15	11
120	180	0	–25	0	–18	19	14
180	250	0	–30	0	–22	23	17
250	315	0	–35	0	–25	26	19
315	400	0	–40	0	–30	30	23
400	500	0	–45	0	–35	34	26
500	630	0	–50	0	–40	38	30
630	800	0	–75	0	–50	56	–
800	1 000	0	–100	0	–	75	–
1 000	1 250	0	–125	0	–	95	–

Outside diameter tolerances for housing locating washers

Outside diameter D mm		Deviation of outside diameter $t_{\Delta Dmp}$ µm Tolerance class				Variation t_{VDp} µm Tolerance class	
		Normal, 6 and 5 Deviation		4 Deviation		Normal, 6 and 5	4
over	incl.	upper	lower	upper	lower	max.	max.
10	18	0	−11	0	−7	8	5
18	30	0	−13	0	−8	10	6
30	50	0	−16	0	−9	12	7
50	80	0	−19	0	−11	14	8
80	120	0	−22	0	−13	17	10
120	180	0	−25	0	−15	19	11
180	250	0	−30	0	−20	23	15
250	315	0	−35	0	−25	26	19
315	400	0	−40	0	−28	30	21
400	500	0	−45	0	−33	34	25
500	630	0	−50	0	−38	38	29
630	800	0	−75	0	−45	55	34
800	1 000	0	−100	–	–	75	–
1 000	1 250	0	−125	–	–	75	–
1 250	1 600	0	−160	–	–	120	–

Variation of washer thickness for shaft and housing locating washers

Bore d mm		Variation				
		t_{Si} µm				t_{Se} µm
		Tolerance class Normal	Tolerance class 6	Tolerance class 5	Tolerance class 4	Tolerance class Normal, 6, 5, 4
over	incl.	max.	max.	max.	max.	
–	18	10	5	3	2	Identical to t_{Si} for the shaft locating washer of the corresponding bearing
18	30	10	5	3	2	
30	50	10	6	3	2	
50	80	10	7	4	3	
80	120	15	8	4	3	
120	180	15	9	5	4	
180	250	20	10	5	4	
250	315	25	13	7	5	
315	400	30	15	7	5	
400	500	30	18	9	6	
500	630	35	21	11	7	
630	800	40	25	13	8	
800	1 000	45	30	15	8	
1 000	1 250	50	35	18	9	

Normal tolerances

Tolerances for nominal bearing height

Tolerances: see table. The corresponding dimension symbols are shown in *Figure 1*.

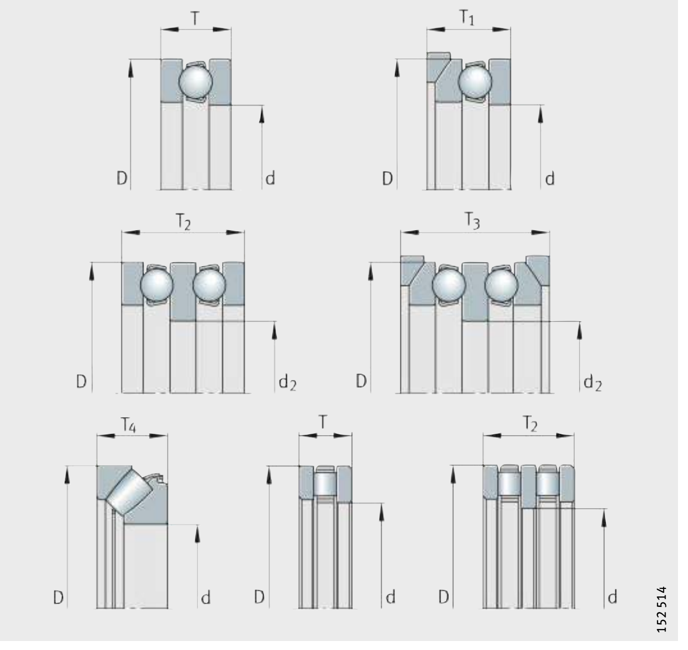


Figure 1
Tolerances for nominal bearing height

Tolerances for nominal bearing height

Bore d mm		T Deviation µm		T_1 Deviation µm		T_2 Deviation µm		T_3 Deviation µm		T_4 Deviation µm	
over	incl.	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower
–	30	20	–250	100	–250	150	–400	300	–400	20	–300
30	50	20	–250	100	–250	150	–400	300	–400	20	–300
50	80	20	–300	100	–300	150	–500	300	–500	20	–400
80	120	25	–300	150	–300	200	–500	400	–500	25	–400
120	180	25	–400	150	–400	200	–600	400	–600	25	–500
180	250	30	–400	150	–400	250	–600	500	–600	30	–500
250	315	40	–400	200	–400	350	–700	600	–700	40	–700
315	400	40	–500	200	–500	350	–700	600	–700	40	–700
400	500	50	–500	300	–500	400	–900	750	–900	50	–900
500	630	60	–600	350	–600	500	–1 100	900	–1 100	60	–1 200
630	800	70	–750	400	–750	600	–1 300	1 100	–1 300	70	–1 400
800	1 000	80	–1 000	450	–1 000	700	–1 500	1 300	–1 500	80	–1 800
1 000	1 250	100	–1 400	500	–1 400	900	–1 800	1 600	–1 800	100	–2 400

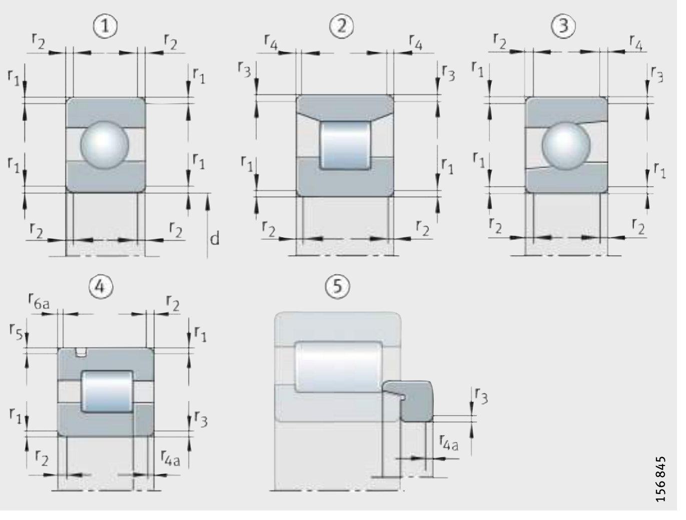
Chamfer dimensions for radial bearings (excluding tapered roller bearings)

The chamfer dimensions correspond to DIN 620-6.

The minimum and maximum values for the bearings are given in the table, page 166.

In the case of drawn cup needle roller bearings with open ends HK, drawn cup needle roller bearings with closed end BK and aligning needle roller bearings PNA and RPNA, the chamfer dimensions deviate from DIN 620-6.

Chamfer dimensions for tapered roller bearings: see table, page 167, for axial bearings see table, page 171.



1 Symmetrical ring cross-section with identical chamfers on both rings
2 Symmetrical ring cross-section with different chamfers on both rings
3 Asymmetrical ring cross-section
4 Annular slot on outer ring, bearing with rib washer
5 L-section ring

Figure 1
Chamfer dimensions for radial bearings excluding tapered roller bearings

Chamfer dimensions

Limit values for chamfer dimensions of radial bearings to DIN 620-6 (excluding tapered roller bearings)

Nominal chamfer dimension r[1)] mm	Nominal bearing bore diameter d mm		Chamfer dimension			
			r_1 to r_{6a} mm	r_1, r_3, r_5 mm	r_2, r_4, r_6[2)] mm	r_{4a}, r_{6a} mm
	over	incl.	min.	max.	max.	max.
0,05	–	–	0,05	0,1	0,2	0,1
0,08	–	–	0,08	0,16	0,3	0,16
0,1	–	–	0,1	0,2	0,4	0,2
0,15	–	–	0,15	0,3	0,6	0,3
0,2	–	–	0,2	0,5	0,8	0,5
0,3	–	40	0,3	0,6	1	0,8
	40	–	0,3	0,8	1	0,8
0,5	–	40	0,5	1	2	1,5
	40	–	0,5	1,3	2	1,5
0,6	–	40	0,6	1	2	1,5
	40	–	0,6	1,3	2	1,5
1	–	50	1	1,5	3	2,2
	50	–	1	1,9	3	2,2
1,1	–	120	1,1	2	3,5	2,7
	120	–	1,1	2,5	4	2,7
1,5	–	120	1,5	2,3	4	3,5
	120	–	1,5	3	5	3,5
2	–	80	2	3	4,5	4
	80	220	2	3,5	5	4
	220	–	2	3,8	6	4
2,1	–	280	2,1	4	6,5	4,5
	280	–	2,1	4,5	7	4,5
2,5	–	100	2,5	3,8	6	5
	100	280	2,5	4,5	6	5
	280	–	2,5	5	7	5
3	–	280	3	5	8	5,5
	280	–	3	5,5	8	5,5
4	–	–	4	6,5	9	6,5
5	–	–	5	8	10	8
6	–	–	6	10	13	10
7,5	–	–	7,5	12,5	17	12,5
9,5	–	–	9,5	15	19	15
12	–	–	12	18	24	18
15	–	–	15	21	30	21
19	–	–	19	25	38	25

1) The nominal chamfer dimension r is identical to the smallest permissible chamfer dimension r_{min}.

2) For bearings with a width of 2 mm or less, the values for r_1 apply.

Chamfer dimensions for tapered roller bearings

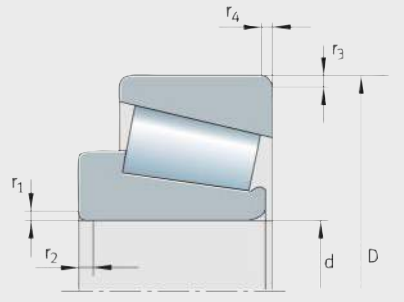


Figure 2
Chamfer dimensions for metric tapered roller bearings

Limit values for chamfer dimensions of tapered roller bearings

Nominal chamfer dimension r[1)] mm	Nominal dimension of bearing bore, outside diameter d, D mm		Chamfer dimension		
			r_1 to r_4 mm	r_1, r_3 mm	r_2, r_4 mm
	over	incl.	min.	max.	max.
0,3	–	40	0,3	0,7	1,4
	40	–	0,3	0,9	1,6
0,6	–	40	0,6	1,1	1,7
	40	–	0,6	1,3	2
1	–	50	1	1,6	2,5
	50	–	1	1,9	3
1,5	–	120	1,5	2,3	3
	120	250	1,5	2,8	3,5
	250	–	1,5	3,5	4
2	–	120	2	2,8	4
	120	250	2	3,5	4,5
	250	–	2	4	5
2,5	–	120	2,5	3,5	5
	120	250	2,5	4	5,5
	250	–	2,5	4,5	6
3	–	120	3	4	5,5
	120	250	3	4,5	6,5
	250	400	3	5	7
	400	–	3	5,5	7,5
4	–	120	4	5	7
	120	250	4	5,5	7,5
	250	400	4	6	8
	400	–	4	6,5	8,5
5	–	180	5	6,5	8
	180	–	5	7,5	9
6	–	180	6	7,5	10
	180	–	6	9	11

1) The nominal chamfer dimension r is identical to the smallest permissible chamfer dimension r_{min}.

Chamfer dimensions

Chamfer dimensions for tapered roller bearings in metric sizes

The limit values for chamfer dimensions r are only valid for tapered roller bearings in metric sizes to ISO 582:1995.

Limit values for chamfer dimensions

Nominal chamfer dimension r[1) mm	Nominal dimension of bearing bore, outside diameter d, D mm		Chamfer dimension		
			r_1 to r_4 mm	r_1, r_3 mm	r_2, r_4 mm
	over	incl.	min.	max.	max.
0,3	–	40	0,3	0,7	1,4
	40	–	0,3	0,9	1,6
0,6	–	40	0,6	1,1	1,7
	40	–	0,6	1,3	2
1	–	50	1	1,6	2,5
	50	–	1	1,9	3
1,5	–	120	1,5	2,3	3
	120	250	1,5	2,8	3,5
	250	–	1,5	3,5	4
2	–	120	2	2,8	4
	120	250	2	3,5	4,5
	250	–	2	4	5
2,5	–	120	2,5	3,5	5
	120	250	2,5	4	5,5
	250	–	2,5	4,5	6
3	–	120	3	4	5,5
	120	250	3	4,5	6,5
	250	400	3	5	7
	400	–	3	5,5	7,5
4	–	120	4	5	7
	120	250	4	5,5	7,5
	250	400	4	6	8
	400	–	4	6,5	8,5
5	–	180	5	6,5	8
	180	–	5	7,5	9
6	–	180	6	7,5	10
	180	–	6	9	11

1) The nominal chamfer dimension r is identical to the smallest permissible chamfer dimension r_{min}.

Chamfer dimensions for FAG tapered roller bearings to ANSI/ABMA

The limit values for chamfer dimensions r are only valid for tapered roller bearings based on ANSI/ABMA.

Limit values for chamfer dimensions r_{max} for the inner ring

Nominal bearing bore diameter d mm		Chamfer dimension	
		r_1 mm	r_2 mm
over	incl.		
–	50,8	+0,4	+0,9
50,8	101,6	+0,5	+1,25
101,6	254	+0,65	+1,8

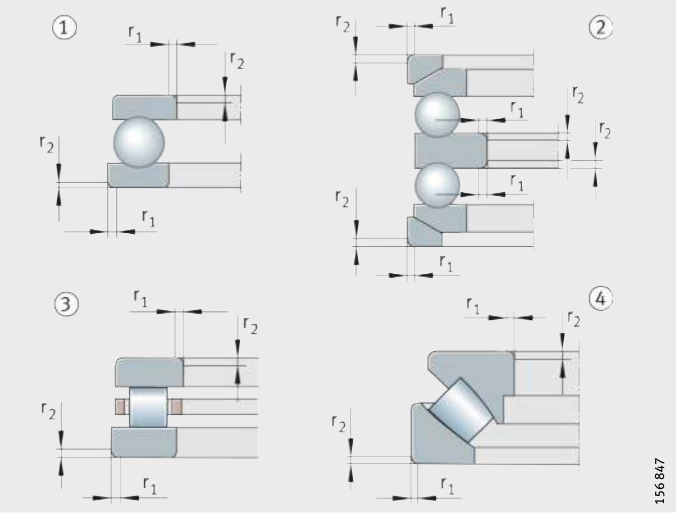
Limit values for chamfer dimensions r_{max} for the outer ring

Nominal outside diameter D mm		Chamfer dimension	
		r_3 mm	r_4 mm
over	incl.		
–	101,6	+0,6	+1,05
101,6	168,3	+0,65	+1,15
168,3	266,7	+0,85	+1,35
266,7	355,6	+1,7	+1,7

Chamfer dimensions for axial bearings

Minumum and maximum values for metric tapered roller bearings: *Figure 3* and table. The table corresponds to DIN 620-6.

In the case of axial deep groove ball bearings, the tolerances for the chamfer dimensions are identical in both axial and radial directions.



① Single direction axial deep groove ball bearing with flat housing locating washer
② Double direction axial deep groove ball bearing with spherical housing locating washers and seating washers
③ Single direction axial cylindrical roller bearing
④ Single direction axial spherical roller bearing

Figure 3 Chamfer dimensions for axial bearings

Limit values for chamfer dimensions of axial bearings

Chamfer dimension		
r[1)] mm	r_1, r_2 mm	
	min.	max.
0,05	0,05	0,1
0,08	0,08	0,16
0,1	0,1	0,2
0,15	0,15	0,3
0,2	0,2	0,5
0,3	0,3	0,8
0,6	0,6	1,5
1	1	2,2
1,1	1,1	2,7
1,5	1,5	3,5
2	2	4
2,1	2,1	4,5
3	3	5,5
4	4	6,5
5	5	8
6	6	10
7,5	7,5	12,5
9,5	9,5	15
12	12	18
15	15	21
19	19	25

1) The nominal chamfer dimension r is identical to the smallest permissible chamfer dimension r_{min}.

Radial internal clearance

Radial internal clearance of FAG deep groove ball bearings

The radial internal clearance corresponds to the internal clearance group Group N to ISO 5753-1, DIN 620-4.

Standardised bearings with increased internal clearance have the suffix C3. Special bearings with the radial internal clearance Group 3 or Group 4 are indicated in the dimension tables.

Radial internal clearance of FAG deep groove ball bearings with cylindrical bore

Bore d mm		Radial internal clearance							
		Group 2 µm		Group N µm		Group 3 µm		Group 4 µm	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.
1,5	6	0	7	2	13	8	23	–	–
6	10	0	7	2	13	8	23	14	29
10	18	0	9	3	18	11	25	18	33
18	24	0	10	5	20	13	28	20	36
24	30	1	11	5	20	13	28	23	41
30	40	1	11	6	20	15	33	28	46
40	50	1	11	6	23	18	36	30	51
50	65	1	15	8	28	23	43	38	61
65	80	1	15	10	30	25	51	46	71
80	100	1	18	12	36	30	58	53	84
100	120	2	20	15	41	36	66	61	97
120	140	2	23	18	48	41	81	71	114
140	160	2	23	18	53	46	91	81	130
160	180	2	25	20	61	53	102	91	147
180	200	2	30	25	71	63	117	107	163
200	225	2	35	25	85	75	140	125	195
225	250	2	40	30	95	85	160	145	225
250	280	2	45	35	105	90	170	155	245
280	315	2	55	40	115	100	190	175	270
315	355	3	60	45	125	110	210	195	300
355	400	3	70	55	145	130	240	225	340
400	450	3	80	60	170	150	270	250	380
450	500	3	90	70	190	170	300	280	420
500	560	10	100	80	210	190	330	310	470
560	630	10	110	90	230	210	360	340	520
630	710	20	130	110	260	240	400	380	570
710	800	20	140	120	290	270	450	430	630
800	900	20	160	140	320	300	500	480	700
900	1 000	20	170	150	350	330	550	530	770
1 000	1 120	20	180	160	380	360	600	580	850
1 120	1 250	20	190	170	410	390	650	630	920
1 250	1 400	30	200	190	440	420	700	680	990
1 400	1 600	30	210	210	470	450	750	730	1 060

Radial internal clearance of FAG self-aligning ball bearings

The radial internal clearance is Group N to ISO 5753-1, DIN 620-4.

Radial internal clearance of FAG self-aligning ball bearings with cylindrical bore

Bore d mm		Radial internal clearance			
		Group N µm		Group 3 µm	
over	incl.	min.	max.	min.	max.
–	6	5	15	10	20
6	10	6	17	12	25
10	14	6	19	13	26
14	18	8	21	15	28
18	24	10	23	17	30
24	30	11	24	19	35
30	40	13	29	23	40
40	50	14	31	25	44
50	65	16	36	30	50
65	80	18	40	35	60
80	100	22	48	42	70
100	120	25	56	50	83
120	140	30	68	60	100
140	160	35	80	70	120

Bearings with a tapered bore have the internal clearance group Group 3 to ISO 5753-1, DIN 620-4.

Radial internal clearance of FAG self-aligning ball bearings with tapered bore

Bore d mm		Radial internal clearance			
		Group N µm		Group 3 µm	
over	incl.	min.	max.	min.	max.
18	24	13	26	20	33
24	30	15	28	23	39
30	40	19	35	29	46
40	50	22	39	33	52
50	65	27	47	41	61
65	80	35	57	50	75
80	100	42	68	62	90
100	120	50	81	75	108
120	140	60	98	90	130
140	160	65	110	100	150

Radial internal clearance

Radial internal clearance of FAG barrel roller bearings

The radial internal clearance corresponds to the internal clearance group Group N to ISO 5753-1, DIN 620-4.

Bearings with a tapered bore have the internal clearance group Group 3 to ISO 5753-1, DIN 620-4.

Radial internal clearance of FAG barrel roller bearings with cylindrical bore

Bore d mm		Radial internal clearance							
		Group 2 µm		Group N µm		Group 3 µm		Group 4 µm	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.
–	30	2	9	9	17	17	28	28	40
30	40	3	10	10	20	20	30	30	45
40	50	3	13	13	23	23	35	35	50
50	65	4	15	15	27	27	40	40	55
65	80	5	20	20	35	35	55	55	75
80	100	7	25	25	45	45	65	65	90
100	120	10	30	30	50	50	70	70	95
120	140	15	35	35	55	55	80	80	110
140	160	20	40	40	65	65	95	95	125
160	180	25	45	45	70	70	100	100	130
180	225	30	50	50	75	75	105	105	135
225	250	35	55	55	80	80	110	110	140
250	280	40	60	60	85	85	115	115	145
280	315	40	70	70	100	100	135	135	170
315	355	45	75	75	105	105	140	140	175

Radial internal clearance of FAG barrel roller bearings with tapered bore

Bore d mm		Radial internal clearance							
		Group 2 µm		Group N µm		Group 3 µm		Group 4 µm	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.
–	30	9	17	17	28	28	40	40	55
30	40	10	20	20	30	30	45	45	60
40	50	13	23	23	35	35	50	50	65
50	65	15	27	27	40	40	55	55	75
65	80	20	35	35	55	55	75	75	95
80	100	25	45	45	65	65	90	90	120
100	120	30	50	50	70	70	95	95	125
120	140	35	55	55	80	80	110	110	140
140	160	40	65	65	95	95	125	125	155
160	180	45	70	70	100	100	130	130	160
180	225	50	75	75	105	105	135	135	165
225	250	55	80	80	110	110	140	140	170
250	280	60	85	85	115	115	145	145	175
280	315	70	100	100	135	135	170	170	205
315	355	75	105	105	140	140	175	175	210

Radial internal clearance of FAG cylindrical roller bearings

The radial internal clearance of bearings with a cylindrical bore normally corresponds to the internal clearance group Group N to ISO 5753-1, DIN 620-4.

Radial internal clearance of FAG cylindrical roller bearings with cylindrical bore

Bore d mm		Radial internal clearance							
		Group 2 µm		Group N µm		Group 3 µm		Group 4 µm	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.
–	10	0	25	20	45	35	60	50	75
10	24	0	25	20	45	35	60	50	75
24	30	0	25	20	45	35	60	50	75
30	40	5	30	25	50	45	70	60	85
40	50	5	35	30	60	50	80	70	100
50	65	10	40	40	70	60	90	80	110
65	80	10	45	40	75	65	100	90	125
80	100	15	50	50	85	75	110	105	140
100	120	15	55	50	90	85	125	125	165
120	140	15	60	60	105	100	145	145	190
140	160	20	70	70	120	115	165	165	215
160	180	25	75	75	125	120	170	170	220
180	200	35	90	90	145	140	195	195	250
200	225	45	105	105	165	160	220	220	280
225	250	45	110	110	175	170	235	235	300
250	280	55	125	125	195	190	260	260	330
280	315	55	130	130	205	200	275	275	350
315	355	65	145	145	225	225	305	305	385
355	400	100	190	190	280	280	370	370	460
400	450	110	210	210	310	310	410	410	510
450	500	110	220	220	330	330	440	440	550
500	560	120	240	240	360	360	480	480	600
560	630	140	260	260	380	380	500	500	620
630	710	145	285	285	425	425	565	565	705
710	800	150	310	310	470	470	630	630	790
800	900	180	350	350	520	520	690	690	860
900	1 000	200	390	390	580	580	770	770	960
1 000	1 120	220	430	430	640	640	850	850	1 060
1 120	1 250	230	470	470	710	710	950	950	1 190
1 250	1 400	270	530	530	790	790	1 050	1 050	1 310
1 400	1 600	330	610	610	890	890	1 170	1 170	1 450
1 600	1 800	380	700	700	1 020	1 020	1 340	1 340	1 660
1 800	2 000	400	760	760	1 120	1 120	1 480	1 480	1 840

Radial internal clearance of FAG cylindrical roller bearings with tapered bore

Bore d mm		Radial internal clearance							
		Group 2 µm		Group N µm		Group 3 µm		Group 4 µm	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.
–	10	5	15	30	55	40	65	50	75
10	24	5	15	30	55	40	65	50	75
24	30	5	15	35	60	45	70	55	80
30	40	5	15	40	65	55	80	70	95
40	50	5	18	45	75	60	90	75	105
50	65	5	20	50	80	70	100	90	120
65	80	10	25	60	95	85	120	110	145
80	100	10	30	70	105	95	130	120	155
100	120	10	30	90	130	115	155	140	180
120	140	10	35	100	145	130	175	160	205
140	160	10	35	110	160	145	195	180	230
160	180	10	40	125	175	160	210	195	245
180	200	15	45	140	195	180	235	220	275
200	225	15	50	155	215	200	260	245	305
225	250	15	50	170	235	220	285	270	335
250	280	20	55	185	255	240	310	295	365
280	315	20	60	205	280	265	340	325	400
315	355	20	65	225	305	290	370	355	435
355	400	25	75	255	345	330	420	405	495
400	450	25	85	285	385	370	470	455	555
450	500	25	95	315	425	410	520	505	615
500	560	25	100	350	470	455	575	560	680
560	630	30	110	380	500	500	620	620	740
630	710	30	130	435	575	565	705	695	835
710	800	35	140	485	645	630	790	775	935
800	900	35	160	540	710	700	870	860	1 030
900	1 000	35	180	600	790	780	970	960	1 150
1 000	1 120	50	200	665	875	865	1 075	1 065	1 275
1 120	1 250	60	220	730	970	960	1 200	1 200	1 440
1 250	1 400	60	240	810	1 070	1 070	1 330	1 330	1 590
1 400	1 600	70	270	920	1 200	1 200	1 480	1 480	1 760
1 600	1 800	80	300	1 020	1 340	1 340	1 660	1 660	1 980
1 800	2 000	100	320	1 120	1 480	1 480	1 840	1 840	2 200

Bearings with a tapered bore frequently have a radial internal clearance Group 3 or Group 4 to DIN 620-4 (ISO 5753-1).

Radial internal clearance of FAG toroidal roller bearings

The radial internal clearance of toroidal roller bearings corresponds to the internal clearance groups in accordance with ISO 5753-1.

Radial internal clearance of FAG toroidal roller bearings with cylindrical bore

Bore d mm		Radial internal clearance Group 2 µm	
over	incl.	min.	max.
18	24	15	30
24	30	15	35
30	40	20	40
40	50	25	45
50	65	30	55
65	80	40	70
80	100	50	85
100	120	60	100
120	140	75	120
140	160	85	140
160	180	95	155
180	200	105	175
200	225	115	190
225	250	125	205
250	280	135	225
280	315	150	240
315	355	160	260
355	400	175	280
400	450	190	310
450	500	205	335
500	560	220	360
560	630	240	400
630	710	260	440
710	800	300	500
800	900	320	540
900	1 000	370	600
1 000	1 120	410	660
1 120	1 250	450	720
1 250	1 400	490	800
1 400	1 600	570	890
1 600	1 800	650	1 010

Group N µm		Group 3 µm		Group 4 µm		Group 5 µm	
min.	max.	min.	max.	min.	max.	min.	max.
25	40	35	55	50	65	65	85
30	50	45	60	60	80	75	95
35	55	55	75	70	95	90	120
45	65	65	85	85	110	105	140
50	80	75	105	100	140	135	175
65	100	95	125	120	165	160	210
80	120	120	160	155	210	205	260
100	145	140	190	185	245	240	310
115	170	165	215	215	280	280	350
135	195	195	250	250	325	320	400
150	220	215	280	280	365	360	450
170	240	235	310	305	395	390	495
185	265	260	340	335	435	430	545
200	285	280	370	365	480	475	605
220	310	305	410	405	520	515	655
235	330	330	435	430	570	570	715
255	360	360	485	480	620	620	790
280	395	395	530	525	675	675	850
305	435	435	580	575	745	745	930
335	475	475	635	630	815	810	1 015
360	520	510	690	680	890	890	1 110
390	570	560	760	750	980	970	1 220
430	620	610	840	830	1 080	1 070	1 340
490	680	680	920	920	1 200	1 200	1 480
530	760	750	1 020	1 010	1 330	1 320	1 660
590	830	830	1 120	1 120	1 460	1 460	1 830
660	930	930	1 260	1 260	1 640	1 640	2 040
720	1 020	1 020	1 380	1 380	1 800	1 800	2 240
800	1 130	1 130	1 510	1 540	1 970	1 970	2 460
890	1 250	1 250	1 680	1 680	2 200	2 200	2 740
1 010	1 390	1 390	1 870	1 870	2 430	2 430	3 000

Radial internal clearance

Radial internal clearance of FAG toroidal roller bearings with tapered bore

Bore d mm		Radial internal clearance Group 2 µm	
over	incl.	min.	max.
18	24	15	35
24	30	20	40
30	40	25	50
40	50	30	55
50	65	40	65
65	80	50	80
80	100	60	100
100	120	75	115
120	140	90	135
140	160	100	155
160	180	115	175
180	200	130	195
200	225	140	215
225	250	160	235
250	280	170	260
280	315	195	285
315	355	220	320
355	400	250	350
400	450	280	385
450	500	305	435
500	560	330	480
560	630	380	530
630	710	420	590
710	800	480	680
800	900	520	740
900	1 000	580	820
1 000	1 120	640	900
1 120	1 250	700	980
1 250	1 400	770	1 080
1 400	1 600	870	1 200
1 600	1 800	950	1 320

Group N µm		Group 3 µm		Group 4 µm		Group 5 µm	
min.	max.	min.	max.	min.	max.	min.	max.
30	45	40	55	55	70	65	85
35	55	50	65	65	85	80	100
45	65	60	80	80	100	100	125
50	75	70	95	90	120	115	145
60	90	85	115	110	150	145	185
75	110	105	140	135	180	175	220
95	135	130	175	170	220	215	275
115	155	155	205	200	255	255	325
135	180	180	235	230	295	290	365
155	215	210	270	265	340	335	415
170	240	235	305	300	385	380	470
190	260	260	330	325	420	415	520
210	290	285	365	360	460	460	575
235	315	315	405	400	515	510	635
255	345	340	445	440	560	555	695
280	380	375	485	480	620	615	765
315	420	415	545	540	680	675	850
350	475	470	600	595	755	755	920
380	525	525	655	650	835	835	1 005
435	575	575	735	730	915	910	1 115
470	640	630	810	800	1 010	1 000	1 230
530	710	700	890	880	1 110	1 110	1 350
590	780	770	990	980	1 230	1 230	1 490
670	860	860	1 100	1 100	1 380	1 380	1 660
730	960	950	1 220	1 210	1 530	1 520	1 860
810	1 040	1 040	1 340	1 340	1 670	1 670	2 050
890	1 170	1 160	1 500	1 490	1 880	1 870	2 280
970	1 280	1 270	1 640	1 630	2 060	2 050	2 500
1 080	1 410	1 410	1 790	1 780	2 250	2 250	2 740
1 200	1 550	1 550	1 990	1 990	2 500	2 500	3 050
1 320	1 690	1 690	2 180	2 180	2 730	2 730	3 310

Axial internal clearance of double row FAG angular contact ball bearings

The main dimensions of the bearings conform to DIN 628-3.

The dimensional and running tolerances of the bearings correspond to tolerance class 6 to DIN 620-2, ISO 492:2014.

Double row angular contact ball bearings of the basic design have the normal axial internal clearance (CN). Bearings are available by agreement with an axial internal clearance larger (C3) or smaller (C2) than normal.

Bearings with a split inner ring are intended for higher axial loads. In general, they have a tighter fit than unsplit bearings. Their normal internal clearance corresponds approximately to the internal clearance group C3 for unsplit bearings.

Axial internal clearance to DIN 628-3 of FAG angular contact ball bearings with unsplit inner ring

Bore d mm		Axial internal clearance							
		C2 µm		CN µm		C3 µm		C4 µm	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.
–	10	1	11	5	21	12	28	25	45
10	18	1	12	6	23	13	31	27	47
18	24	2	14	7	25	16	34	28	48
24	30	2	15	8	27	18	37	30	50
30	40	2	16	9	29	21	40	33	54
40	50	2	18	11	33	23	44	36	58
50	65	3	22	13	36	26	48	40	63
65	80	3	24	15	40	30	54	46	71
80	100	3	26	18	46	35	63	55	83
100	120	4	30	22	53	42	73	65	96
120	140	4	34	25	59	48	82	74	108

Axial internal clearance of FAG angular contact ball bearings with split inner ring

Bore d mm		Axial internal clearance					
		C2 µm		CN µm		C3 µm	
over	incl.	min.	max.	min.	max.	min.	max.
24	30	8	27	16	35	27	46
30	40	9	29	18	38	30	50
40	50	11	33	22	44	36	58
50	65	13	36	25	48	40	63
65	80	15	40	29	54	46	71

Axial internal clearance of FAG four point contact bearings

The axial internal clearance corresponds to the internal clearance group CN to DIN 628-4.

Axial internal clearance of FAG four point contact bearings

Bore d mm		Axial internal clearance							
		C2 µm		CN µm		C3 µm		C4 µm	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.
18	40	30	70	60	110	100	150	140	190
40	60	40	90	80	130	120	170	160	210
60	80	50	100	90	140	130	180	170	220
80	100	60	120	100	160	140	200	180	240
100	140	70	140	120	180	160	220	200	260
140	180	80	160	140	200	180	240	220	280
180	220	100	180	160	220	200	260	240	300
220	260	120	200	180	240	220	300	280	360
260	300	140	220	200	280	260	340	320	400
300	355	160	240	220	300	280	360	–	–
355	400	180	270	250	330	310	390	–	–
400	450	200	290	270	360	340	430	–	–
450	500	220	310	290	390	370	470	–	–
500	560	240	330	310	420	400	510	–	–
560	630	260	360	340	450	430	550	–	–
630	710	280	390	370	490	470	590	–	–
710	800	300	420	400	540	520	660	–	–
800	900	330	460	440	590	570	730	–	–
900	1 000	360	500	480	630	620	780	–	–

Reduction in radial internal clearance

Reduction in radial internal clearance of FAG cylindrical roller bearings with tapered bore

Nominal bearing bore diameter d mm		Radial internal clearance before mounting Internal clearance group					
		Group N mm		Group 3 mm		Group 4 mm	
over	incl.	min.	max.	min.	max.	min.	max.
24	30	0,035	0,06	0,045	0,07	0,055	0,08
30	40	0,04	0,065	0,055	0,08	0,07	0,095
40	50	0,045	0,075	0,06	0,09	0,075	0,105
50	65	0,05	0,08	0,07	0,1	0,09	0,12
65	80	0,06	0,095	0,085	0,12	0,11	0,145
80	100	0,07	0,105	0,095	0,13	0,12	0,155
100	120	0,09	0,13	0,115	0,155	0,14	0,18
120	140	0,1	0,145	0,13	0,175	0,16	0,205
140	160	0,11	0,16	0,145	0,195	0,18	0,23
160	180	0,125	0,175	0,16	0,21	0,195	0,245
180	200	0,14	0,195	0,18	0,235	0,22	0,275
200	225	0,155	0,215	0,2	0,26	0,245	0,305
225	250	0,17	0,235	0,22	0,285	0,27	0,335
250	280	0,185	0,255	0,24	0,31	0,295	0,365
280	315	0,205	0,28	0,265	0,34	0,325	0,4
315	355	0,225	0,305	0,29	0,37	0,355	0,435
355	400	0,255	0,345	0,33	0,42	0,405	0,495
400	450	0,285	0,385	0,37	0,47	0,455	0,555
450	500	0,315	0,425	0,41	0,52	0,505	0,615
500	560	0,35	0,47	0,455	0,575	0,56	0,68
560	630	0,38	0,5	0,5	0,62	0,62	0,74
630	710	0,435	0,575	0,565	0,705	0,695	0,835
710	800	0,485	0,645	0,63	0,79	0,775	0,935
800	900	0,54	0,71	0,7	0,87	0,86	1,03
900	1 000	0,6	0,79	0,78	0,97	0,96	1,15
1 000	1 120	0,665	0,875	0,865	1,075	1,065	1,275
1 120	1 250	0,73	0,97	0,96	1,2	1,2	1,44
1 250	1 400	0,81	1,07	1,07	1,33	1,33	1,59

[1] Valid only for solid steel shafts and hollow shafts with a bore no larger than half the shaft diameter.
The following applies: Bearings with a radial internal clearance before mounting in the upper half of the tolerance range are mounted using the larger value for the reduction in radial internal clearance or the axial drive-up distance, while bearings in the lower half of the tolerance range are mounted using the smaller value for the reduction in radial internal clearance or the axial drive-up distance.

[2] The actual value of the radial internal clearance must not be smaller than the control value. In the case of bearings with a small diameter, this may be difficult to determine.

Reduction in radial internal clearance[1]		Drive-up distance on taper 1:12[1]				Control value for radial internal clearance after mounting[2]		
mm		Shaft mm		Sleeve mm		Group N mm	Group 3 mm	Group 4 mm
min.	max.	min.	max.	min.	max.	min.	min.	min.
0,015	0,02	0,3	0,35	0,3	0,4	0,02	0,025	0,035
0,02	0,025	0,35	0,4	0,35	0,45	0,02	0,025	0,04
0,025	0,03	0,4	0,45	0,45	0,5	0,02	0,03	0,045
0,03	0,035	0,45	0,55	0,5	0,65	0,02	0,035	0,05
0,035	0,04	0,55	0,6	0,65	0,7	0,025	0,04	0,07
0,04	0,045	0,6	0,7	0,65	0,8	0,03	0,05	0,075
0,045	0,055	0,7	0,85	0,8	0,95	0,045	0,065	0,085
0,055	0,065	0,85	1	0,95	1,1	0,045	0,07	0,095
0,06	0,075	0,9	1,2	1,1	1,3	0,05	0,075	0,105
0,065	0,085	1	1,3	1,3	1,5	0,06	0,08	0,11
0,075	0,095	1,2	1,5	1,4	1,7	0,065	0,09	0,125
0,085	0,105	1,3	1,6	1,6	1,8	0,07	0,1	0,14
0,095	0,115	1,5	1,8	1,7	2	0,075	0,105	0,155
0,105	0,125	1,6	2	1,9	2,3	0,08	0,125	0,17
0,115	0,14	1,8	2,2	2,2	2,4	0,09	0,13	0,185
0,13	0,16	2	2,5	2,5	2,7	0,095	0,14	0,195
0,14	0,17	2,2	2,6	2,6	2,9	0,115	0,165	0,235
0,15	0,185	2,3	2,8	2,8	3,1	0,135	0,19	0,27
0,16	0,195	2,5	3	3,1	3,4	0,155	0,215	0,31
0,17	0,215	2,7	3,4	3,5	3,8	0,18	0,24	0,345
0,185	0,24	2,9	3,7	3,6	4,2	0,195	0,26	0,38
0,2	0,26	3,1	4,1	3,9	4,7	0,235	0,305	0,435
0,22	0,28	3,4	4,4	4,3	5,3	0,26	0,35	0,495
0,24	0,31	3,7	4,8	4,8	5,5	0,3	0,39	0,55
0,26	0,34	4,1	5,3	5,2	6,2	0,34	0,44	0,62
0,28	0,37	4,4	5,8	5,7	7	0,385	0,5	0,7
0,31	0,41	4,8	6,4	6,3	7,6	0,42	0,55	0,79
0,34	0,45	5,3	7	0,3	8,3	0,47	0,62	0,85

Reduction in radial internal clearance

Reduction in radial internal clearance of FAG spherical roller bearings with tapered bore

Nominal bearing bore diameter d mm		Radial internal clearance before mounting Internal clearance group Group N mm		Group 3 mm		Group 4 mm	
over	incl.	min.	max.	min.	max.	min.	max.
24	30	0,03	0,04	0,04	0,055	0,055	0,075
30	40	0,035	0,05	0,05	0,065	0,065	0,085
40	50	0,045	0,06	0,06	0,08	0,08	0,1
50	65	0,055	0,075	0,075	0,095	0,095	0,12
65	80	0,07	0,095	0,095	0,12	0,12	0,15
80	100	0,08	0,11	0,11	0,14	0,14	0,18
100	120	0,1	0,135	0,135	0,17	0,17	0,22
120	140	0,12	0,16	0,16	0,2	0,2	0,26
140	160	0,13	0,18	0,18	0,23	0,23	0,3
160	180	0,14	0,2	0,2	0,26	0,26	0,34
180	200	0,16	0,22	0,22	0,29	0,29	0,37
200	225	0,18	0,25	0,25	0,32	0,32	0,41
225	250	0,2	0,27	0,27	0,35	0,35	0,45
250	280	0,22	0,3	0,3	0,39	0,39	0,49
280	315	0,24	0,33	0,33	0,43	0,43	0,54
315	355	0,27	0,36	0,36	0,47	0,47	0,59
355	400	0,3	0,4	0,4	0,52	0,52	0,65
400	450	0,33	0,44	0,44	0,57	0,57	0,72
450	500	0,37	0,49	0,49	0,63	0,63	0,79
500	560	0,41	0,54	0,54	0,68	0,68	0,87
560	630	0,46	0,6	0,6	0,76	0,76	0,98
630	710	0,51	0,67	0,67	0,85	0,85	1,09
710	800	0,57	0,75	0,75	0,96	0,96	1,22
800	900	0,64	0,84	0,84	1,07	1,07	1,37
900	1 000	0,71	0,93	0,93	1,19	1,19	1,52
1 000	1 120	0,78	1,02	1,02	1,3	1,3	1,65
1 120	1 250	0,86	1,12	1,12	1,42	1,42	1,8
1 250	1 400	0,94	1,22	1,22	1,55	1,55	1,96

[1] Valid only for solid steel shafts and hollow shafts with a bore no larger than half the shaft diameter.
The following applies: Bearings with a radial internal clearance before mounting in the upper half of the tolerance range are mounted using the larger value for the reduction in radial internal clearance or the axial drive-up distance, while bearings in the lower half of the tolerance range are mounted using the smaller value for the reduction in radial internal clearance or the axial drive-up distance.

[2] The actual value of the radial internal clearance must not be smaller than the control value. In the case of bearings with a small diameter, this may be difficult to determine.

Reduction in radial internal clearance[1)]		Drive-up distance on taper 1:12[1)]				Drive-up distance on taper 1:30[1)]				Control value for radial internal clearance after mounting[2)]		
		Shaft		Sleeve		Shaft		Sleeve		Group N	Group 3	Group 4
mm		mm		mm		mm		mm		mm	mm	mm
min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	min.	min.
0,015	0,02	0,3	0,35	0,3	0,4	–	–	–	–	0,015	0,02	0,035
0,02	0,025	0,35	0,4	0,35	0,45	–	–	–	–	0,015	0,025	0,04
0,025	0,03	0,4	0,45	0,45	0,5	–	–	–	–	0,02	0,03	0,05
0,03	0,04	0,45	0,6	0,5	0,7	–	–	–	–	0,025	0,035	0,055
0,04	0,05	0,6	0,75	0,7	0,85	–	–	–	–	0,025	0,04	0,07
0,045	0,06	0,7	0,9	0,75	1	1,7	2,2	1,8	2,4	0,035	0,05	0,08
0,05	0,07	0,7	1,1	0,8	1,2	1,9	2,7	2	2,8	0,05	0,065	0,1
0,065	0,09	1,1	1,4	1,2	1,5	2,7	3,5	2,8	3,6	0,055	0,08	0,11
0,075	0,1	1,2	1,6	1,3	1,7	3	4	3,1	4,2	0,055	0,09	0,13
0,08	0,11	1,3	1,7	1,4	1,9	3,2	4,2	3,3	4,6	0,06	0,1	0,15
0,09	0,13	1,4	2	1,5	2,2	3,5	4,5	3,6	5	0,07	0,1	0,16
0,1	0,14	1,6	2,2	1,7	2,4	4	5,5	4,2	5,7	0,08	0,12	0,18
0,11	0,15	1,7	2,4	1,8	2,6	4,2	6	4,6	6,2	0,09	0,13	0,2
0,12	0,17	1,9	2,6	2	2,9	4,7	6,7	4,8	6,9	0,1	0,14	0,22
0,13	0,19	2	3	2,2	3,2	5	7,5	5,2	7,7	0,11	0,15	0,24
0,15	0,21	2,4	3,4	2,6	3,6	6	8,2	6,2	8,4	0,12	0,17	0,26
0,17	0,23	2,6	3,6	2,9	3,9	6,5	9	5,8	9,2	0,13	0,19	0,29
0,2	0,26	3,1	4,1	3,4	4,4	7,7	10	8	10,4	0,13	0,2	0,31
0,21	0,28	3,3	4,4	3,6	4,8	8,2	11	8,4	11,2	0,16	0,23	0,35
0,24	0,32	3,7	5	4,1	5,4	9,2	12,5	9,6	12,8	0,17	0,25	0,36
0,26	0,35	4	5,4	4,4	5,9	10	13,5	10,4	14	0,2	0,29	0,41
0,3	0,4	4,6	6,2	5,1	6,8	11,5	15,5	12	16	0,21	0,31	0,45
0,34	0,45	5,3	7	5,8	7,6	13,3	17,5	13,6	18	0,23	0,35	0,51
0,37	0,5	5,7	7,8	6,3	8,5	14,3	19,5	14,8	20	0,27	0,39	0,57
0,41	0,55	6,3	8,5	7	9,4	15,8	21	16,4	22	0,3	0,43	0,64
0,45	0,6	6,8	9	7,6	10,2	17	23	18	24	0,32	0,48	0,7
0,49	0,65	7,4	9,8	8,3	11	18,5	25	19,6	26	0,34	0,54	0,77
0,55	0,72	8,3	10,8	9,3	12,1	21	27	22,2	28,3	0,36	0,59	0,84

Reduction in radial internal clearance

Reduction in radial internal clearance of FAG toroidal roller bearings with tapered bore

Nominal bearing bore diameter d mm		Radial internal clearance before mounting Internal clearance group					
		Group N mm		Group 3 mm		Group 4 mm	
over	incl.	min.	max.	min.	max.	min.	max.
24	30	0,035	0,055	0,050	0,065	0,065	0,085
30	40	0,045	0,065	0,060	0,080	0,080	0,100
40	50	0,050	0,075	0,070	0,095	0,090	0,120
50	65	0,060	0,090	0,085	0,115	0,110	0,150
65	80	0,075	0,110	0,105	0,140	0,135	0,180
80	100	0,095	0,135	0,130	0,175	0,170	0,220
100	120	0,115	0,155	0,155	0,205	0,200	0,255
120	140	0,135	0,180	0,180	0,235	0,230	0,295
140	160	0,155	0,215	0,210	0,270	0,265	0,340
160	180	0,170	0,240	0,235	0,305	0,300	0,385
180	200	0,190	0,260	0,260	0,330	0,325	0,420
200	225	0,210	0,290	0,285	0,365	0,360	0,460
225	250	0,235	0,315	0,315	0,405	0,400	0,515
250	280	0,255	0,345	0,340	0,445	0,440	0,560
280	315	0,280	0,380	0,375	0,485	0,480	0,620
315	355	0,315	0,420	0,415	0,545	0,540	0,680
355	400	0,350	0,475	0,470	0,600	0,595	0,755
400	450	0,380	0,525	0,525	0,655	0,650	0,835
450	500	0,435	0,575	0,575	0,735	0,730	0,915
500	560	0,470	0,640	0,630	0,810	0,800	1,010
560	630	0,530	0,710	0,700	0,890	0,880	1,110
630	710	0,590	0,780	0,770	0,990	0,980	1,230
710	800	0,670	0,860	0,860	1,100	1,100	1,380
800	900	0,730	0,960	0,950	1,220	1,210	1,530
900	1 000	0,810	1,040	1,040	1,340	1,340	1,670
1 000	1 120	0,890	1,170	1,160	1,500	1,490	1,880
1 120	1 250	0,970	1,280	1,270	1,640	1,630	2,060
1 250	1 400	1,080	1,410	1,410	1,790	1,780	2,250
1 400	1 600	1,200	1,550	1,550	1,990	1,990	2,500
1 600	1 800	1,320	1,690	1,690	2,180	2,180	2,730

1) Valid only for solid steel shafts and hollow shafts with a bore no larger than half the shaft diameter.
The following applies: Bearings with a radial internal clearance before mounting in the upper half of the tolerance range are mounted using the larger value for the reduction in radial internal clearance or the axial drive-up distance, while bearings in the lower half of the tolerance range are mounted using the smaller value for the reduction in radial internal clearance or the axial drive-up distance.

2) The actual value of the radial internal clearance must not be smaller than the control value. In the case of bearings with a small diameter, this may be difficult to determine.

Reduction in radial internal clearance[1) mm		Drive-up distance on taper 1:12[1) Shaft mm		Drive-up distance on taper 1:30[1) Shaft mm		Control value for radial internal clearance after mounting[2) Group N mm	Group 3 mm	Group 4 mm
min.	max.	min.	max.	min.	max.	min.	min.	min.
0,010	0,017	0,24	0,29	0,61	0,72	0,025	0,035	0,048
0,014	0,021	0,30	0,34	0,76	0,84	0,031	0,041	0,059
0,018	0,028	0,37	0,42	0,91	1,04	0,033	0,046	0,062
0,024	0,035	0,46	0,50	1,14	1,24	0,036	0,054	0,075
0,030	0,046	0,55	0,61	1,37	1,53	0,045	0,065	0,090
0,040	0,056	0,67	0,73	1,68	1,83	0,056	0,080	0,114
0,049	0,069	0,79	0,89	1,98	2,23	0,066	0,093	0,131
0,060	0,083	0,91	1,05	2,29	2,62	0,075	0,105	0,147
0,072	0,095	1,04	1,21	2,59	3,02	0,083	0,123	0,170
0,081	0,107	1,16	1,36	2,90	3,41	0,089	0,137	0,193
0,090	0,121	1,28	1,52	3,20	3,81	0,100	0,150	0,204
0,101	0,134	1,43	1,68	3,58	4,20	0,109	0,162	0,226
0,113	0,151	1,59	1,88	3,96	4,69	0,123	0,177	0,249
0,126	0,168	1,77	2,08	4,42	5,19	0,129	0,186	0,273
0,142	0,188	1,98	2,31	4,95	5,78	0,138	0,203	0,292
0,160	0,211	2,23	2,59	5,56	6,47	0,155	0,221	0,329
0,180	0,238	2,50	2,90	6,25	7,26	0,170	0,251	0,357
0,203	0,268	2,81	3,26	7,01	8,15	0,178	0,279	0,382
0,225	0,300	3,11	3,66	7,78	9,14	0,210	0,300	0,430
0,250	0,335	3,48	4,05	8,69	10,13	0,220	0,325	0,465
0,285	0,375	3,90	4,52	9,76	11,31	0,245	0,355	0,505
0,320	0,420	4,39	5,08	10,98	12,69	0,270	0,380	0,560
0,360	0,475	4,94	5,71	12,35	14,27	0,310	0,425	0,625
0,405	0,535	5,55	6,42	13,88	16,05	0,325	0,460	0,675
0,450	0,605	6,16	7,21	15,40	18,03	0,360	0,490	0,735
0,505	0,670	6,89	8,00	17,23	20,00	0,385	0,545	0,820
0,565	0,750	7,69	8,95	19,21	22,37	0,410	0,580	0,880
0,630	0,840	8,60	9,98	21,50	24,94	0,450	0,640	0,940
0,720	0,940	9,82	11,16	24,55	27,90	0,480	0,685	1,050
0,810	1,070	11,04	12,74	27,60	31,85	0,510	0,705	1,110

FAG rolling bearing greases Arcanol – chemical/physical data

Grease		Characteristic applications	Operating temperature °C from	to	Continuous limit temperature °C	Thickener
Multi-purpose greases	MULTITOP	Ball and roller bearings in rolling mills Construction machinery Spinning and grinding spindles Automotive engineering	–50[1]	+140	+80	Lithium soap
	MULTI2	Ball bearings up to an outside diameter of 62 mm in small electric motors Agricultural and construction machinery Household appliances	–30	+120	+75	Lithium soap
	MULTI3	Ball bearings with an outside diameter of or more than 62 mm in large electric motors Agricultural and construction machinery Fans	–30	+120	+75	Lithium soap
High loads	LOAD150	Ball, roller and needle roller bearings Linear guidance systems in machine tools	–20	+140	+95	Lithium complex soap
	LOAD220	Ball and roller bearings in rolling mill plant Paper machinery Rail vehicles	–20	+140	+80	Lithium/calcium soap
	LOAD400	Ball and roller bearings in mining machinery Construction machinery Wind turbine main bearings	–40	+130	+80	Lithium/calcium soap
	LOAD460	Ball and roller bearings Wind turbines Bearings with pin cage	–40[1]	+130	+80	Lithium/calcium soap
	LOAD1000	Ball and roller bearings in mining machinery Construction machinery Cement plant	–30[1]	+130	+80	Lithium/calcium soap
High temperatures	TEMP90	Ball and roller bearings in couplings Electric motors Automotive engineering	–40	+160	+90	Polycarbamide
	TEMP110	Ball and roller bearings in electric motors Automotive engineering	–35	+160	+110	Lithium complex soap
	TEMP120	Ball and roller bearings in continuous casting plant Paper machinery	–30	+180	+120	Polycarbamide
	TEMP200	Ball and roller bearings in guide rollers for baking machinery Kiln trucks and chemical plant Piston pins in compressors	–30	+260	+200	PTFE

Continued on next page.

+++ Extremely suitable. ++ Highly suitable. + Suitable. – Less suitable. –– Not suitable.

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

Base oil	Consistency NLGI	Base oil viscosity at +40 °C mm²/s	Temperatures Low	Temperatures High	Low friction, high speed	High load, low speed	Vibrations	Support for seals	Relubrication facility
Partially synthetic oil	2	82	+++	++	++	+++	++	+	+++
Mineral oil	2	110	++	+	+	+	+	+	+++
Mineral oil	3	80	++	+	+	+	++	++	++
Mineral oil	2	160	+	++	–	+++	++	++	++
Mineral oil	2	245	+	+	–	+++	++	++	++
Mineral oil	2	400	+	+	–	+++	++	++	++
Mineral oil	1	400	++	+	–	+++	++	–	++
Mineral oil	2	1 000	+	+	––	+++	++	++	++
Partially synthetic oil	3	148	+++	++	+	+	+	++	++
Partially synthetic oil	2	130	+++	+++	++	+	+	+	+
Synthetic oil	2	400	++	+++	–	+++	+	++	+
Alkoxyfluoro oil	2	550	++	+++	––	++	+	+	+

FAG rolling bearing greases Arcanol – chemical/physical data

Grease		Characteristic applications	Operating temperature °C		Continuous limit temperature °C	Thickener
			from	to		
Special requirements	SPEED2,6	Ball bearings in machine tools Spindle bearings Rotary table bearings Instrument bearings	–40	+120	+80	Lithium complex soap
	VIB3	Ball and roller bearings in rotors for wind turbines (blade adjustment) Packaging machinery Rail vehicles	–30	+150	+90	Lithium complex soap
	FOOD2	Ball and roller bearings in applications with food contact (NSF-H1 registration, kosher and halal certification)	–30	+120	+70	Aluminium complex soap
	CLEAN-M	Ball, roller and needle roller bearings as well as linear guidance systems in clean room applications	–30	+180	+90	Polycarbamide
	MOTION2	Ball and roller bearings in oscillating operation Slewing rings in wind turbines	–40	+130	+75	Lithium soap

+++ Extremely suitable. ++ Highly suitable. + Suitable. – Less suitable. –– Not suitable.

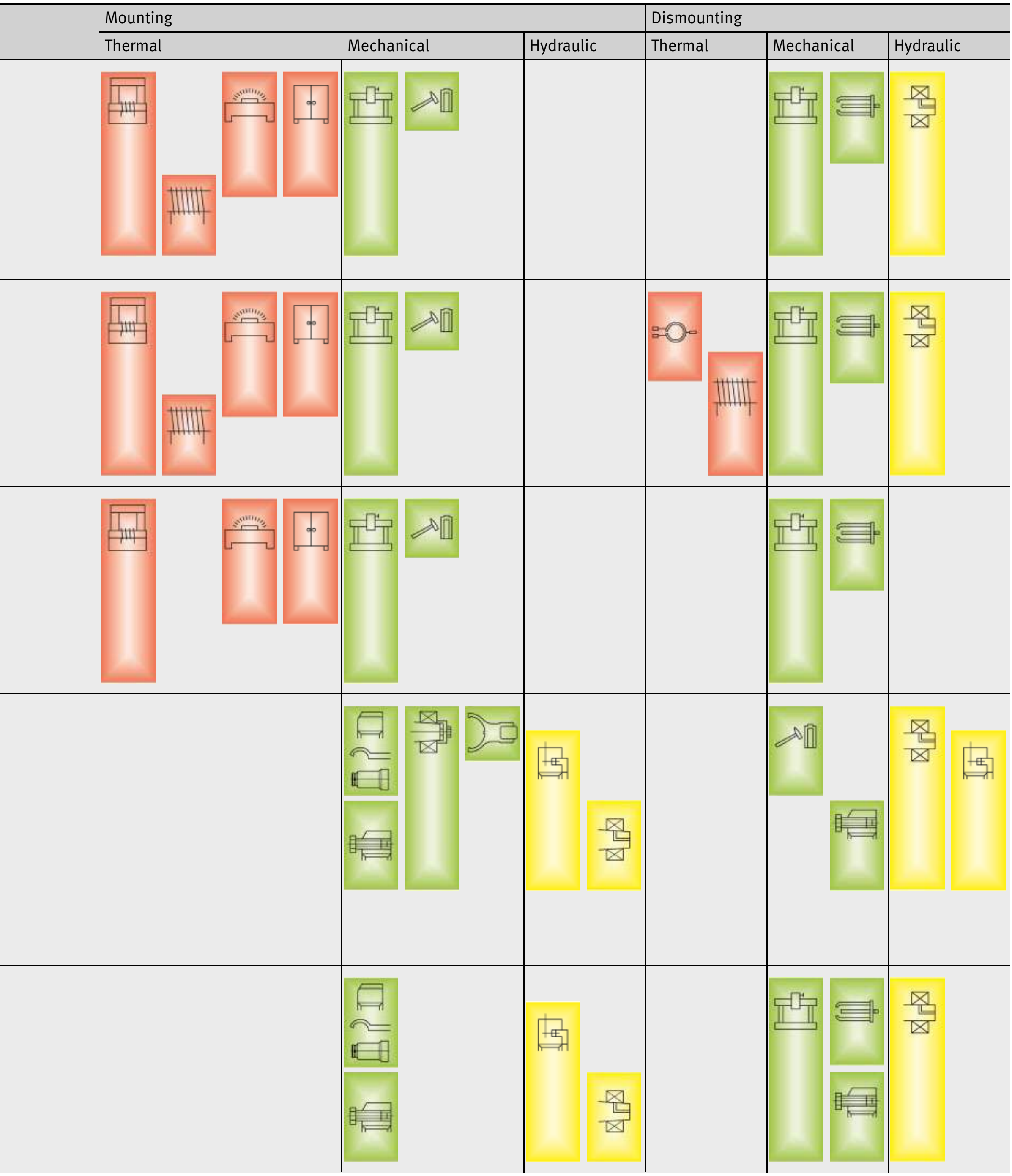
Base oil	Consistency NLGI	Base oil viscosity at +40 °C mm²/s	Temperatures Low	Temperatures High	Low friction, high speed	High load, low speed	Vibrations	Support for seals	Relubrication facility
Synthetic oil	2 – 3	25	+++	+	+++	––	–	+	+
Mineral oil	3	170	++	++	–	++	+++	++	–
Synthetic oil	2	150	++	–	+	+	+	+	+++
Ether oil	2	103	+++	+++	+	+	+	+	++
Synthetic oil	2	50	+++	+	–	++	+++	++	+

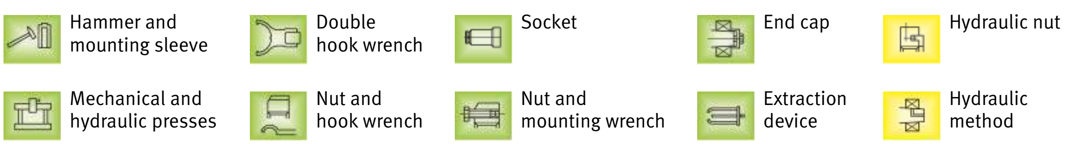
Mounting and dismounting methods for rolling bearings

Bearing type		Bearing bore	d mm
Deep groove ball bearings Angular contact ball bearings Spindle bearings Four point contact bearings Self-aligning ball bearings	Tapered roller bearings Barrel roller bearings Spherical roller bearings Toroidal bearings	Cylin-drical	< 80 80 – 200 > 200
Cylindrical roller bearings Needle roller bearings		Cylin-drical	< 80 80 – 200 > 200
Axial deep groove ball bearings Axial angular contact ball bearings Axial cylindrical roller bearings Axial spherical roller bearings		Cylin-drical	< 80 80 – 200 > 200
Self-aligning ball bearings Self-aligning ball bearings with adapter sleeve Toroidal bearings Barrel roller bearings Barrel roller bearings with adapter sleeve Spherical roller bearings Spherical roller bearings with adapter sleeve Spherical roller bearings with withdrawal sleeve Adapter sleeve	Withdrawal sleeve	Tapered	< 80 80 – 200 > 200
Cylindrical roller bearings, double row		Tapered	< 80 80 – 200 > 200

Symbols

- Induction heating device
- Heating cabinet
- Heating ring
- Heating plate
- Medium frequency technology


Mounting
Dismounting
Thermal
Mechanical
Hydraulic
Thermal
Mechanical
Hydraulic


Hammer and mounting sleeve
Double hook wrench
Socket
End cap
Hydraulic nut
Mechanical and hydraulic presses
Nut and hook wrench
Nut and mounting wrench
Extraction device
Hydraulic method

Measurement record

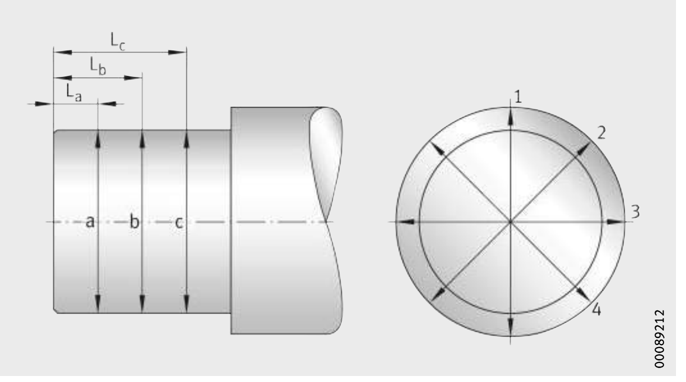


Figure 1
Shaft

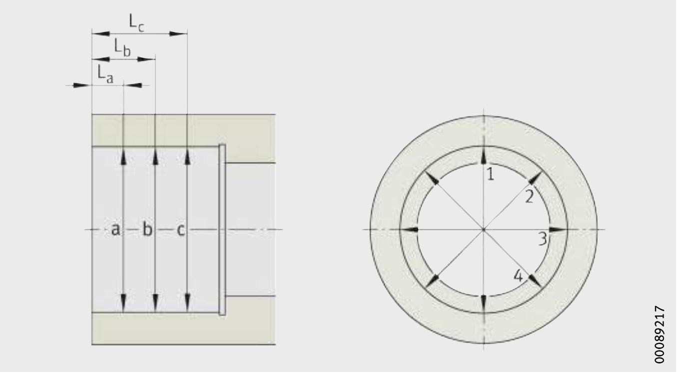


Figure 2
Housing

Measurement record for shaft

Spacing [mm]	L_a	L_b	L_c
Diameter [mm]	a	b	c
1			
2			
3			
4			
Mean value (1 + 2 + 3 + 4)/4			

Measurement record for housing

Spacing [mm]	L_a	L_b	L_c
Diameter [mm]	a	b	c
1			
2			
3			
4			
Mean value (1 + 2 + 3 + 4)/4			

INA FAG

Lubrication of Rolling Bearings

Principles
Lubrication methods
Lubricant selection and testing
Storage and handling

SCHAEFFLER

LuK INA FAG

Foreword

Schaeffler Group

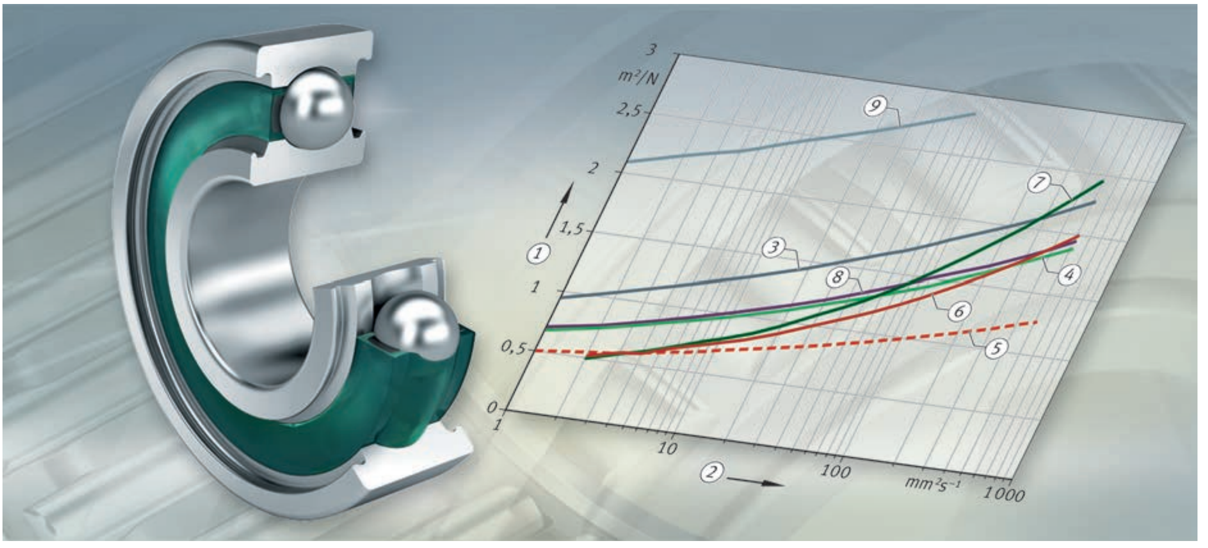
The Schaeffler Group with its brands INA and FAG is a leading worldwide supplier of rolling bearings, spherical plain bearings, plain bearings, linear products, accessories specific to bearings and comprehensive maintenance products and services. It has approximately 40 000 catalogue products manufactured as standard, providing an extremely wide portfolio that gives secure coverage of applications from all 60 designated industrial market sectors.

Research and development

As a company looking to the future, we are especially active in the field of research and development. The key areas in this respect include not only research into fundamental principles, materials technology, tribology and calculation but also extensive inspection and test methods as well as activities to optimise manufacturing technology. This is oriented towards ensuring the continuous development, improvement and application of our products in the long term. We carry out research and development on a global basis. Our development centres are linked with each other worldwide and are thus in a position to exchange current information on a very short timescale as well as access and communicate the most recent data. This ensures that a uniform level of knowledge and information is available worldwide.

This publication gives a comprehensive overview of the lubrication of rolling bearings.

Lubrication of rolling bearings



Lubricant in the rolling bearing

Lubricant in the rolling bearing

Principles

The functions of lubrication

The main function of the lubrication of rolling bearings is to prevent or reduce contact between rolling and sliding surfaces. As a result, friction and wear are kept to a low level.

Types of lubrication

A distinction is made between physical and chemical lubrication.

Physical lubrication

Lubricant is conveyed into the contact areas of rolling bearings and adheres to the surfaces of parts rolling against each other.
The lubricant thus separates the contact surfaces and prevents metal-to-metal contact.

Chemical lubrication

If a lubricant film is not formed that can fully support loads, some areas of the surfaces are not separated by the lubricant film. Operation with low levels of wear is possible even in such cases if tribomechanical reaction layers are formed between the additives in the lubricant and the rolling element or bearing ring.

Lubricant can be supported not only by additive reactions but also by the thickener in the grease and solid lubricants that are added to the oil or grease. In special cases, it is possible to lubricate rolling bearings with solid substances only.

Rolling and sliding motion

On the contact surfaces of rolling bearings, not only rolling motion but also sliding motion occurs and this is dependent on the bearing type. This sliding motion is caused by elastic deformations of the parts rolling against each other, the curved geometry of the rolling surfaces and the kinematics of certain bearing types, such as axial cylindrical roller bearings.

In pure sliding motion, the forces and pressures are in general significantly lower than in the rolling area. This case occurs in the rolling bearing between the cage and rolling elements or between the roller end faces and rib surfaces.

Other functions

Other functions of the lubricant are as follows:

- anti-corrosion protection
- heat dissipation from the bearing (in the case of oil lubrication)
- flushing out of wear particles and contaminants (recirculating oil lubrication with filtration of the oil)
- support for the sealing effect of bearing seals (grease collar, pneumatic oil lubrication).

Lubrication and friction regimes

The friction and lubrication behaviour and the achievable life of the rolling bearing are dependent on the lubrication regime and the resulting friction regime.

The possible lubrication regimes are delineated in the Stribeck curve, *Figure 1*.

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

μ = friction
v = speed

① Boundary lubrication
② Mixed lubrication
③ Full lubrication

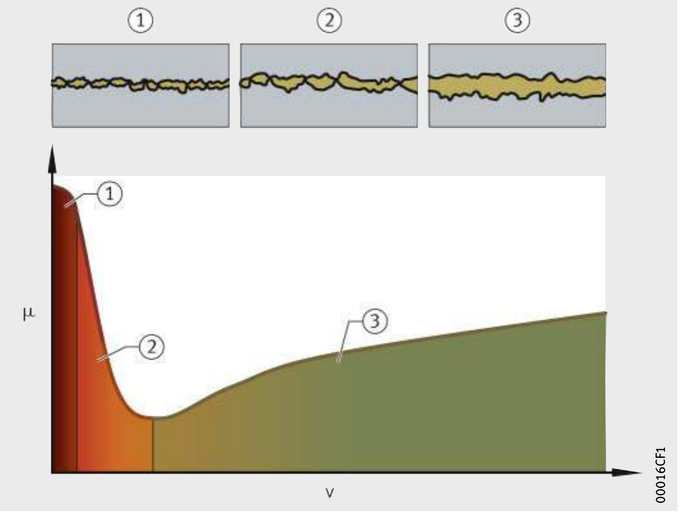


Figure 1
Stribeck curve

Boundary lubrication

Fluid friction is present only partially. In this case, the lubricant film thickness is negligible. This regime is caused by an insufficient quantity of lubricant, an inadequate operating vicosity value or relative motion. Predominantly, solid body contact occurs.

If the lubricant contains suitable additives, reactions occur between the additives and the metallic surfaces in the solid body contacts under conditions of high pressure and high temperature. Reaction products are formed that can provide lubrication and form a thin boundary layer.

Mixed lubrication

If the lubricant film thickness is too small, solid body contact occurs partially. As a result, so-called mixed friction is present.

Full lubrication

The surfaces moving relative to each other are separated completely or almost completely by a lubricant film. Almost pure fluid friction is present. For long term operation, it is desirable to achieve this lubrication regime.

The theory of lubrication

The life of rolling bearings is influenced by the lubricant film. There are two physical theories that describe the lubricant film formed by oil.

Hydrodynamic lubrication

The lubricant is conveyed into the narrowing lubrication gap by the motion of the contact surfaces relative to each other.
Due to the extremely high pressure in the immediate contact zone, the lubricant here has extremely high viscosity for a short period and facilitates separation of the contact surfaces, *Figure 4*, page 11.

Elastohydrodynamic lubrication (EHD theory)

This expands on the theory of hydrodynamic lubrication and takes account of the elastic deformation of the bodies in contact with each other. The theory is used specifically for the lubrication regime in the rolling contact.

Lubricant film thickness

Practical experience and tests have shown that a lubricant film thickness of only a few tenths of a micron is sufficient to separate the contact surfaces from each other.

The lubricant film thickness is determined by:

- the lubricant characteristics
- the macrogeometry and microgeometry of the contact surfaces
- the speed of the contact surfaces relative to each other.

!

The physical theory takes account only of the lubrication regime in the rolling contact. It does not cover the lubrication conditions at the other contact surfaces with higher sliding friction components, for example between the rolling element and the cage pocket.
In the selection of lubricant, it is therefore necessary to take account not only of the EHD theory but also practical experience and the complete lubrication regime in the bearing as well as any possible additive reactions.

Furthermore, it does not take account of the fact that the profile geometry of the surfaces has an influence on the lubrication regime. It is therefore not sufficient simply to compare the theoretical lubricant film thickness with the roughness of the surfaces.

Minimum load

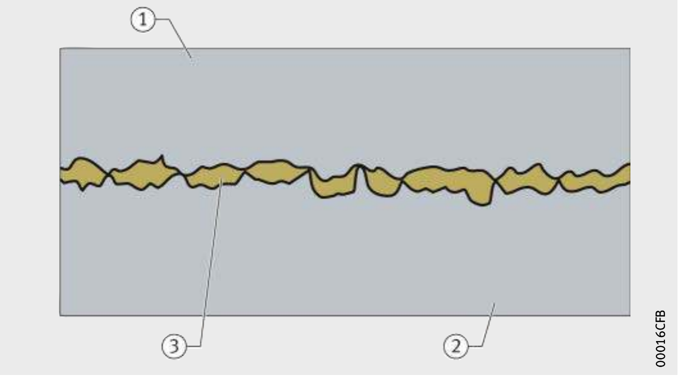
In order to ensure that the rolling elements undergo rolling motion correctly, a minimum load is necessary. The guide value as a function of the bearing type is given by the ratio $C_0/P = 60$.

Comparable lubrication regime in grease lubrication

The thickener contained in greases has an influence on formation of the lubricant film and on protection against wear. This effect has been demonstrated in practice but cannot yet be defined in theoretical terms. In order to allow an estimate of a comparable lubrication regime, current practice is based on calculation using only the base oil data.

Additive reactions

Additives can trigger chemical reactions that are not defined by the theory of lubrication. Lubrication on the basis of additive reactions falls within the scope of boundary lubrication, *Figure 2*.

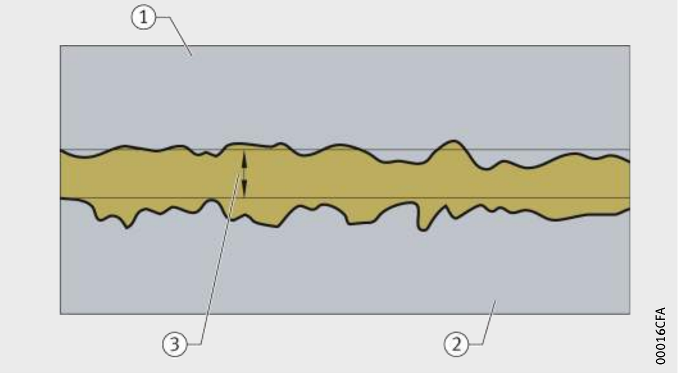


1 Rolling element
2 Raceway
3 Effective lubricant film

Figure 2
Boundary lubrication

Lubrication by means of reaction layers

Very high pressures and temperatures at the solid body contacts can lead to reactions between additives and metallic surfaces. Reaction products are formed that can provide lubrication and form a thin boundary layer in the nanometre range (lubrication by means of reaction layers). This can lead to complete separation of the surfaces and is comparable in its effect with EHD full lubrication, *Figure 3*.



1 Rolling element
2 Raceway
3 Effective lubricant film

Figure 3
Full lubrication

Additives can also trigger undesirable side effects.
These frequently occur as a result of reactions with the bearing materials or the reaction of several additives with each other.

Viscosity

In order that a lubricant film capable of supporting load can be formed at the contact surfaces between rolling elements and raceways, the oil must exhibit a certain viscosity.

Oil viscosity is subject to limits in terms of function. At higher speeds, these limits come into effect as a result of:

- increasing mechanical power losses, especially with higher idling friction
- higher bearing temperatures
- poor conveyability of highly viscous oils.

The influence of temperature

The viscosity of an oil decreases with increasing temperature. It is therefore important that the nominal viscosity is present at the operating temperature. If the operating temperature is known, the corresponding ISO VG class can be derived from diagrams, *Figure 2*, page 24. If the operating temperature is not known on the basis of experience, it can be determined, see section Operating temperature, page 50.

The influence of pressure

The viscosity changes with increasing pressure. Under high load, the pressure values at the rolling contact calculated according to Hertz are up to 40 000 bar and, in the entry zone, they are up to 7 000 bar.

If the influence of temperature in the high pressure range is not taken into account, the viscosity in the lubrication gap can be estimated:

$$\eta = \eta_0 \cdot e^{\alpha p}$$

η mPa · s
Dynamic viscosity at pressure

η_0 mPa · s
Dynamic viscosity at normal pressure

e = 2,7182 –
Euler's number

α m^2/N
Pressure/viscosity coefficient of the fluid

p N/m^2
Pressure.

Pressure/viscosity behaviour

The pressure/viscosity behaviour describes the change in the viscosity of an oil at different pressures. This change is quantified by the pressure/viscosity coefficient α.

In standard calculations, the α values of paraffin-based mineral oils are normally used. These are the basis for the a_{ISO} diagram, *Figure 4*.

The pressure/viscosity behaviour of a lubricant is influenced significantly by the type of base oil, its moecular structure and its additive package. In many cases, precise values are not available for individual lubricants. In practice, however, the significant difference between mineral and synthetic oils should at least be taken into consideration by means of representative values, for example in calculations of the lubricant film thickness.

Source: FVA Research Project no. 400

α = pressure/viscosity coefficient
ν = kinematic viscosity

① Mineral oils
② PAO/E
③ Polyglycol oils (soluble in water)
④ Polyglycol oils (not soluble in water)
⑤ Diester mixture
⑥ Hydrocrack oils
⑦ Fluorinated hydrocarbons

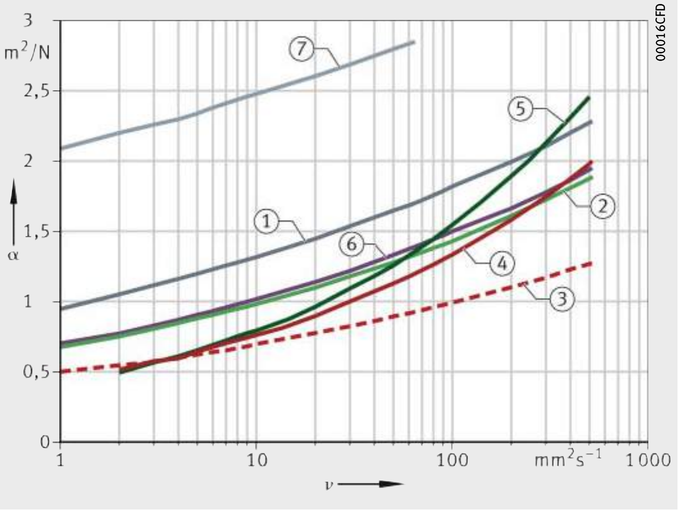


Figure 4
Pressure/viscosity behaviour α_{P2000}

α values

The α values were determined under quasistatic conditions, *Figure 4*. At the rolling contact, the pressure conditions change quickly and the high pressure normally has an effect for a very short time only. The effects of these time influences are not taken into consideration.

For the purposes of checking, lubricant film thicknesses were calculated using the α values and the values were compared with measured lubricant film thicknesses. Very good agreement was found at rolling pressures up to p_{max} = 14 000 bar, while the pressures were correspondingly lower in the entry zone. This was also confirmed for synthetic oils by more recent measurements of the lubricant film thickness in the rolling bearing and on the two disc test rig. As a result, these α values can generally be used for rolling bearings.

The lubricant film in oil lubrication

In order to assess the lubrication regime, it is assumed that a lubricant film is formed between the rolling and sliding surfaces supporting load. The lubricant film between the rolling surfaces can be described in theoretical terms by means of elastohydrodynamic lubrication. The lubrication conditions at the sliding contact, for example between the roller end face and rib in tapered roller bearings, can be adequately described by means of the hydrodynamic lubrication theory, since lower pressures occur at the sliding contacts than in the rolling contacts.

Minimum lubricant film thickness

The minimum lubricant film thickness h_{min} for EHD lubrication is calculated using the formulae for point and linear contact according to Hamrock and Dowson, *Figure 5*, formulae.

p_0 = Hertzian pressure
2b = pressure surface axis according to Hertz

① Entry side
② Exit side
③ Deformation of the roller
④ Lubricant film
⑤ Deformation of the raceway
⑥ Hertzian pressure distribution
⑦ EHD pressure distribution

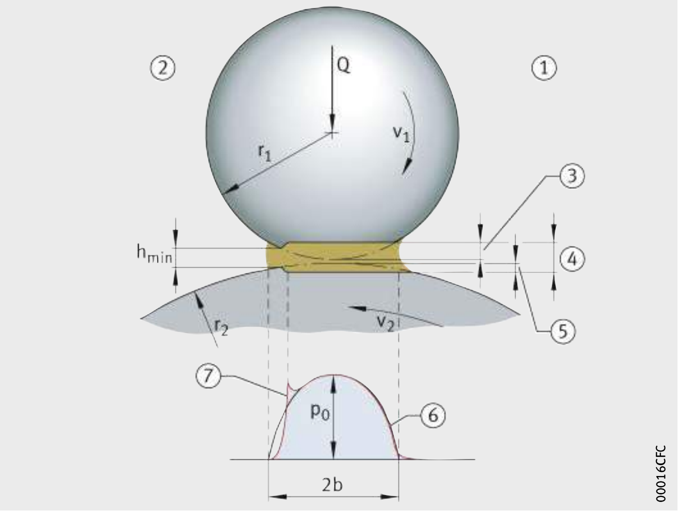


Figure 5
Lubricant film at the rolling contact

The influence of pressure is taken into consideration in calculation of the lubrication regime in accordance with the EHD theory by means of the pressure/viscosity coefficient α, *Figure 4*, page 11.

The formulae show the major influence of the rolling velocity v, the dynamic viscosity η and the pressure/viscosity coefficient α on the minimum lubricant film thickness h_{min}. The load Q has little influence, since the viscosity increases with increasing load and the contact surfaces become larger as a result of elastic deformations.

The calculated lubricant film thickness can be used to check whether a sufficiently strong lubricant film is formed under the conditions present.

Lubricant film thickness in line contact

Calculation according to Dowson:

$$h_{min} = \frac{2,65 \cdot \alpha^{0,54} \cdot (\eta \cdot v)^{0,7}}{\left(\frac{1}{r_1} + \frac{1}{r_2}\right)^{0,43} \cdot \left(\frac{Q}{L}\right)^{0,13}} \cdot \left(\frac{E}{1 - \left(\frac{1}{m}\right)^2}\right)^{-0,03}$$

Lubricant film thickness in point contact

Calculation according to Hamrock and Dowson:

$$h_{min} = \frac{3,63 \cdot \alpha^{0,49} \cdot (\eta \cdot v)^{0,68}}{\left(\frac{1}{r_1} + \frac{1}{r_2}\right)^{0,466} \cdot Q^{0,073}} \cdot \left(\frac{E}{1 - \left(\frac{1}{m}\right)^2}\right)^{-0,117} \cdot \left(1 - e^{-0,68k}\right)$$

h_{min} mm
Minimum lubricant film thickness

α mm^2/s
Pressure/viscosity coefficient

η mPa · s
Dynamic viscosity

v m/s
$v = (v_1 + v_2)/2$, mean cumulative roller velocity
v_1 = rolling element velocity
v_2 = velocity at inner or outer contact

E N/mm^2
Modulus of elasticity ($E = 2,08 \cdot 10^5$ N/mm^2 for steel)

r_1 mm
Radius of rolling element

r_2 mm
Radius of inner or outer ring raceway

Q N
Rolling element load

L mm
Gap length, effective roller length

1/m –
Poisson's constant (1/m = 0,3 for steel)

e = 2,7182 –
Euler's number

k –
k = a/b, ratio of pressure surface semiaxes.

In general, the minimum thickness of the lubricant film should be between one and several tenths of a micron. Under favourable circumstances, it is possible to achieve several microns.

Nominal viscosity

In day-to-day practice, it is too cumbersome to design the nominal oil viscosity through calculation of the lubricant film thickness. Instead, the nominal viscosity is determined by means of the viscosity ratio $\kappa = \nu/\nu_1$, see section Viscosity ratio, page 22. The operating viscosity ν is the kinematic viscosity of the lubricant at the operating temperature. The reference viscosity ν_1 is a function of the bearing size and speed. The reference and operating viscosity can be derived from diagrams, *Figure 2*, page 24.

Density

The density ρ of mineral oils is a function of temperature, *Figure 6*. The trend can be determined for an oil of a different density if the density ρ at +15 °C is known.

ρ = density
ϑ = temperature

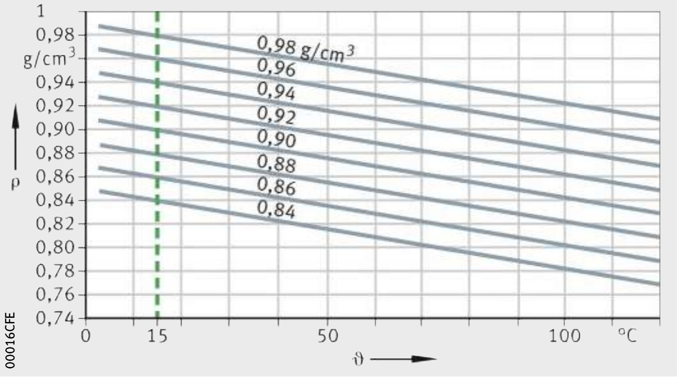


Figure 6
Influence of temperature on the density of mineral oil

The lubricant film in grease lubrication

In the case of greases, bearing lubrication is performed mainly by the base oil that is released in small quantities over time by the thickener. The principles of the EHD theory also apply to grease lubrication.

Viscosity ratio

In order to determine the viscosity ratio $\kappa = \nu/\nu_1$ (ν = kinematic viscosity of the lubricant at operating temperature, ν_1 = reference viscosity of the lubricant), the operating viscosity ν of the base oil is used, see section Viscosity ratio, page 22.
At low κ values in particular, the thickener and additives contribute to effective lubrication.

Lubricant film thickness

The effect of the grease thickener becomes clear if the film thickness is measured as a function of the running time. At the start of bearing running, the film thickness formed in the contact area as a function of the bearing type is significantly greater than the theoretically possible value of the base oil. As a resul, rolling bearings with grease can be provided with adequate lubricant in the long term. Changes in the grease and displacement of the grease can quickly reduce the film thickness, *Figure 7*.

t = running time

① = (grease film thickness)/(base oil film thickness)

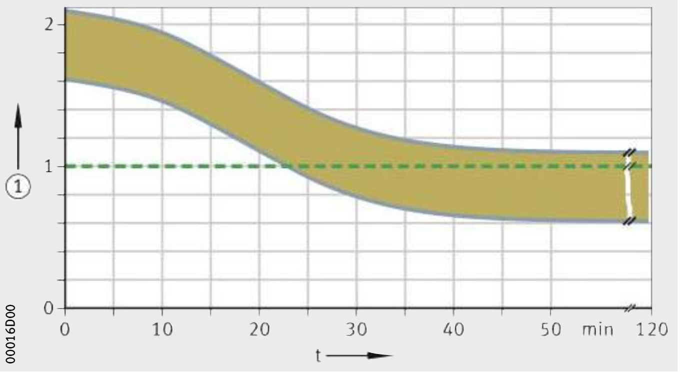


Figure 7
Ratio of grease film thickness to base oil film thickness

So-called “lubricant starvation” is a special case of undersupply. High overrolling frequencies distribute the grease in the bearing, which means that there is less grease at the rolling contact. As a result, the lubricant film thickness is smaller than the theoretically possible value. Nevertheless, bearings lubricated with grease can still achieve a sufficiently long life even under such conditions.

Grease selection

Correct grease lubrication is particularly important in the case of bearings with high proportions of sliding motion and bearings subjected to heavy loads. Under high load, the lubrication capability of the thickener and the additive package are of particular importance.

In grease lubrication, the amount of lubricant playing an active role in the lubrication process is very small. Grease of normal consistency is largely displaced from the rolling contact and is deposited laterally or exits the bearing arrangement through the seals. The grease remaining on the raceway surfaces and laterally in or on the bearing continuously releases the nominal small quantity of oil and, in some cases, thickener as well for lubrication of the functional surfaces. The effective lubricant quantity between the rolling contact surfaces is sufficient for lubrication under moderate load over an extended period.

Solid lubricants

Solid lubricants such as graphite and molybdenum disulphide, which are applied as a thin layer to the functional surfaces, can prevent metal-to-metal contact. However, a layer of this type continues to adhere for an extended period only at low circumferential speeds and low pressures. The lubrication of solid body contacts can also be improved by solid lubricants in oils or greases.

Thickeners

Thickeners and agents in the grease support lubrication through the formation of boundary layers, with the result that no reduction in life is anticipated. In order to achieve long lubrication intervals, it is advisable that the grease should release precisely the amount of oil that is required for lubrication of the bearing. The release of oil can thus continue over a long period. Greases with a highly viscous base oil have a reduced oil release rate. If these are used, a good lubrication regime can only be achieved if the bearing and housing are filled to a large extent or if relubrication is carried out at short intervals. Certain types of thickener additionally have a the effect of forming boundary layers during operation in the mixed friction range.

The release of oil is dependent on:

- the thickener (type, quantity and consistency)
- the additives
- the type of base oil
- the viscosity of the base oil
- the size of the surface releasing oil
- the temperature
- the mechanical strain on the grease.

Special lubricants

As a supplement to pure oil or grease lubrication, lubrication using special lubricants may be advisable in special applications.

Compound lubrication

Solid lubricant compounds that are applied as a thin layer to the functional surfaces can prevent metal-to-metal contact. They comprise a combination of solid lubricants such as molybdenum disulphide, graphite or PTFE and a binder with good high temperature stability. The bearings are filled with the paste-like compound and this is then hardened by the effect of heat. During operation, the compound rotates with the cage. However, a layer of this type continues to adhere for an extended period only at low circumferential speeds and low pressures.

Compound lubrication is a transfer type lubrication, which means that there is ongoing erosion of the hardened compound which is then deposited on the balls and raceway surfaces. Tests have shown that there is a steep reduction in the life of such bearings with increasing speed. In contrast to oil or grease lubrication, the influence of load or temperature is less pronounced.

Compound lubrication is used, for example, in the high temperature range $>$ +250 °C, for example in kiln car bearing arrangements or in areas with strong chemical or physical influences such as vacuum.

Polymer lubrication

Other special lubricants are so-called polymer lubricants, *Figure 8*. These comprise a porous carrier material, frequently polymers such as polyethylene, and a flowable grease or oil. The carrier material can be seen as a type of sponge that holds flowable grease or oil, releasing this under load.

One possible area of application is in bearing arrangements with slewing operation, low speeds multi-row bearings mounted vertically.

Figure 8
Polymer lubricant in a ball bearing

Load carrying capacity and rating life

The Schaeffler Group introduced the "Expanded calculation of the adjusted rating life" in 1997. This method was standardised for the first time in DIN ISO 281 Appendix 1 and has been a constituent part of the international standard ISO 281 since 2007.

As part of the international standardisation work, the life adjustment factor a_{DIN} was renamed as a_{ISO} but without any change to the calculation method.

Fatigue theory as a principle

The basis of the rating life calculation in accordance with ISO 281 is the fatigue theory developed by Lundberg and Palmgren, which always gives a final rating life.

However, modern, high quality bearings can exceed by a considerable margin the values calculated for the basic rating life under favourable operating conditions. Ioannides and Harris have developed a further model of fatigue in rolling contact that expands on the theory by Lundberg and Palmgren and gives a better description of the performance capability of modern bearings.

The method "Expanded calculation of the adjusted rating life" takes account of the following influences:

- the bearing load
- the fatigue limit of the material
- the extent to which the surfaces are separated by the lubricant
- the cleanliness in the lubrication gap
- the additive package in the lubricant
- the internal load distribution and frictional conditions in the bearing.

The influencing factors, particularly those relating to contamination, are very complex. A great deal of experience is required in order to arrive at an accurate assessment. Further advice should therefore be sought from the Schaeffler Group engineering service.

The tables and diagrams can give only guide values.

Further information is also given in Catalogue HR1, Rolling Bearings.

Dimensioning of rolling bearings

The required size of a rolling bearing is dependent on the demands made on its:

- rating life
- load carrying capacity
- operational reliability.

Dynamic load carrying capacity and operating life

The dynamic load carrying capacity is described in terms of the basic dynamic load ratings. The basic dynamic load ratings are based on DIN ISO 281.

The fatigue behaviour of the material determines the dynamic load carrying capacity of the rolling bearing.

The dynamic load carrying capacity is described in terms of the basic dynamic load rating and the basic rating life.

The fatigue life is dependent on:

- the load
- the operating speed
- the statistical probability of the first appearance of failure.

The basic dynamic load rating C applies to rotating rolling bearings. It is:

- a constant radial load C_r for radial bearings
- a constant, concentrically acting axial load C_a for axial bearings.

The basic dynamic load rating C is that load of constant magnitude and direction which a sufficiently large number of apparently identical bearings can endure for a basic rating life of one million revolutions.

Calculation of the rating life

The methods for calculating the rating life are:

- the basic rating life L_{10} and L_{10h} according to ISO 281
- the adjusted rating life L_{na} according to DIN ISO 281:1990 (no longer a constituent part of ISO 281)
- the expanded adjusted rating life L_{nm} according to ISO 281.

Load carrying capacity and rating life

Basic rating life

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

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

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

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

C N
Basic dynamic load rating

P N
Equivalent dynamic bearing load for radial and axial bearings, see section Equivalent operating values, page 35

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

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

n min^{-1}
Operating speed.

Equivalent dynamic bearing load

The equivalent dynamic load P is a calculated value. This value is constant in magnitude and direction; it is a radial load for radial bearings and an axial load for axial bearings.

A load corresponding to P will give the same rating life as the combined load occurring in practice.

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

P N
Equivalent dynamic bearing load

X –
Radial factor given in the dimension tables or product description

F_r N
Radial dynamic bearing load

Y –
Axial factor given in the dimension tables or product description

F_a N
Axial dynamic bearing load.

This calculation cannot be applied to radial needle roller bearings, axial needle roller bearings and axial cylindrical roller bearings. Combined loads are not permissible with these bearings.

Equivalent values for non-constant loads or speeds: see section Equivalent operating values, page 35.

Adjusted rating life

The adjusted rating life L_{na} can be calculated if, in addition to the load and speed, other influences are known such as:

- special material characteristics
- lubrication
  or
- if a requisite reliability other than 90% is specified.

This calculation method was replaced in ISO 281:2007 by the calculation of the expanded adjusted rating life L_{nm}.

$$L_{na} = a_1 \cdot a_2 \cdot a_3 \cdot L_{10}$$

L_{na} 10^6 revolutions
Adjusted rating life for special material characteristics and operating conditions with a requisite reliability of (100 – n) %

a_1 –
Life adjustment factor for a requisite reliability other than 90% In ISO 281:2007, the values for the life adjustment factor a_1 were redefined, see table

a_2 –
Life adjustment factor for special material characteristics
For standard rolling bearing steels: $a_2 = 1$

a_3 –
Life adjustment factor for special operating conditions; in particular for the lubrication regime, *Figure 1*, page 22

L_{10} 10^6 revolutions
Basic rating life.

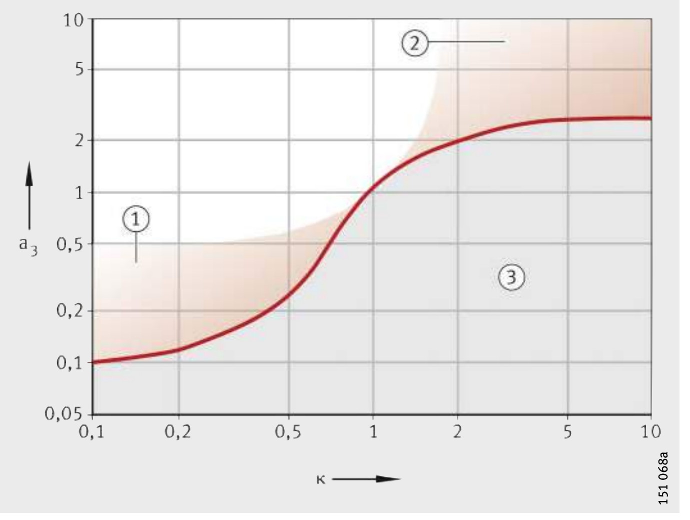
Life adjustment factor a_1

Requisite reliability %	Expanded adjusted rating life L_{nm}	Life adjustment factor a_1
90	L_{10m}	1
95	L_{5m}	0,64
96	L_{4m}	0,55
97	L_{3m}	0,47
98	L_{2m}	0,37
99	L_{1m}	0,25
99,2	$L_{0,8m}$	0,22
99,4	$L_{0,6m}$	0,19
99,6	$L_{0,4m}$	0,16
99,8	$L_{0,2m}$	0,12
99,9	$L_{0,1m}$	0,093
99,92	$L_{0,08m}$	0,087
99,94	$L_{0,06m}$	0,08
99,95	$L_{0,05m}$	0,077

The values for the life adjustment factor a_1 were redefined in ISO 281:2007 and differ from the previous data.

Life adjustment factor a_3

In order to determine the life adjustment factor a_3, the viscosity ratio κ must first be determined, see section Viscosity ratio.



a_3 = life adjustment factor
κ = viscosity ratio

① Good cleanliness and suitable additives
② Very high cleanliness and low load
③ Contaminants in the lubricant

Figure 1
Life adjustment factor a_3

Viscosity ratio

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

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

κ –
Viscosity ratio
ν mm^2s^{-1}
Kinematic viscosity of the lubricant at operating temperature
ν_1 mm^2s^{-1}
Reference viscosity of the lubricant at operating temperature.

At values of $\kappa = 4$ and above, full lubrication is present, i.e. the partners are not in contact.

At $\kappa \geqq 4$ and very high cleanliness as well as moderate load, rolling bearings can be fatigue-resistant. Experience shows that, at values of $\kappa = 2$ and above, a lubricant film fully capable of supporting load can be anticipated.

At values of $\kappa = 1$ and above as well as good cleanliness, a life corresponding approximately to the basic rating life can be achieved.

If κ is in the range between 0,4 and 1, a reduction in the basic rating life can be anticipated and the regime can be described as moderate mixed friction.

At $\kappa < 0,4$, mixed friction is present. If undoped lubricants are used in this case, wear must additionally be anticipated. If the lubricant contains suitable anti-wear additives, however, separation in the contact area may also be achieved by the reaction layers formed by the additives. Through this chemical lubrication, it is also possible to achieve low-wear operation, see also section Taking account of EP additives in the lubricant, page 25.

The reference viscosity ν_1 is determined from the mean bearing diameter d_M and the operating speed n, *Figure 2*, page 24.

Alternatively, the reference viscosity ν_1 can also be calculated using the following formulae:

$$\nu_1 = 45000 \cdot n^{-0,83} \cdot d_M{}^{-0,5} \qquad \text{for } n < 1000 \text{ min}^{-1}$$

$$\nu_1 = 4500 \cdot n^{-0,5} \cdot d_M{}^{-0,5} \qquad \text{for } n \geqq 1000 \text{ min}^{-1}$$

n $\quad$ min^{-1}
Operating speed

d_M $\quad$ mm
Mean bearing diameter (d + D)/2.

The nominal viscosity of the oil at +40 °C is determined from the required operating viscosity ν and the operating temperature ϑ, *Figure 2*, page 24. The diagram and the formula $\kappa = \nu/\nu_1$ can also be used as an approximation for synthetic oils such as PAO.
In this case, the higher viscosity index in comparison with mineral oils is compensated by means of a higher pressure/viscosity coefficient. In the case of greases, the operating viscosity of the base oil is the decisive factor.

In the case of heavily loaded bearings with a high proportion of sliding motion, the temperature in the contact area of the rolling elements may be up to 20 K higher than the temperature measured on the stationary ring (without any influence from external heat).

ν_1 = reference viscosity
d_M = mean bearing diameter
ϑ = operating temperature
n = operating speed

① Viscosity $mm^2 \cdot s^{-1}$ at +40 °C

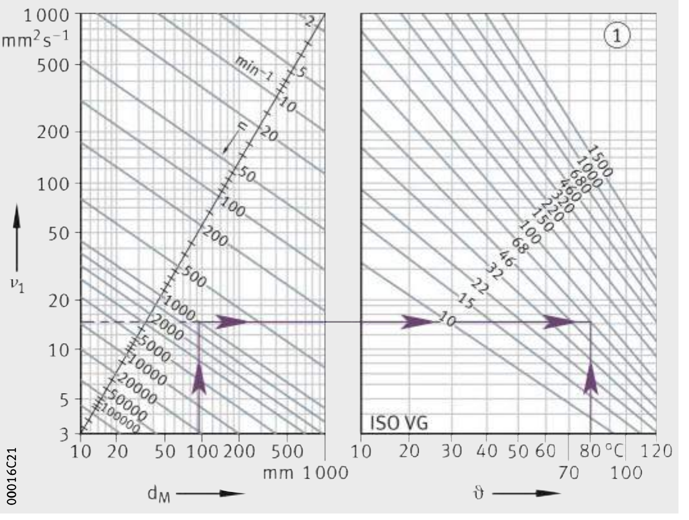


Figure 2
V/T diagram for mineral oils

The curves presented are valid for a lubricant density of $\rho = 0{,}89\ g/cm^3$ at a temperature of +20 °C.

For lubricants of a different density, the viscosity ratio can be determined using the following formula:

$$\kappa = \frac{\nu}{\nu_1} \cdot \left(\frac{\rho}{0{,}89\ g / cm^3} \right)^{0{,}83}$$

κ –
Viscosity ratio

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

ρ g/cm^3
Density

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

Expanded adjusted rating life

The calculation of the expanded adjusted rating life L_{nm} was standardised in DIN ISO 281 Appendix 1. Since 2007, it has been standardised in the worldwide standard ISO 281. Computer-aided calculation in accordance with DIN ISO 281 Appendix 4 has been specified since 2008 in ISO/TS 16 281.

L_{nm} is calculated as follows:

$$L_{nm} = a_1 \cdot a_{ISO} \cdot L_{10}$$

L_{nm} 10^6 revolutions
Expanded adjusted rating life to ISO 281

a_1 –
Life adjustment factor for a requisite reliability other than 90%, see table, page 21
The values for the life adjustment factor a_1 were redefined in ISO 281:2007 and differ from the previous data

a_{ISO} –
Life adjustment factor for operating conditions

L_{10} 10^6 revolutions
Basic rating life, see page 20.

Life adjustment factor a_{ISO}

The standardised method for calculating the life adjustment factor a_{ISO} essentially takes account of:

- the load on the bearing
- the lubrication conditions (viscosity and type of lubricant, speed, bearing size, additives)
- the fatigue limit of the material
- the type of bearing
- the residual stress in the material
- the ambient conditions
- contamination of the lubricant, see section Contaminants in the lubricant.

$$a_{ISO} = f\left[\frac{e_C \cdot C_u}{P}, \kappa\right]$$

a_{ISO} –
Life adjustment factor for operating conditions,
Figure 3, page 26 to *Figure 6*, page 27. Alternatively, a_{ISO} can also be calculated using the formulae according to DIN ISO 281:2009

e_C –
Life adjustment factor for contamination, see table, page 28

C_u N
Fatigue limit load, according to dimension tables

κ –
Viscosity ratio, see page 22
For $\kappa > 4$, calculation should be carried out using $\kappa = 4$.
For $\kappa < 0,1$, this calculation method cannot be used.

P N
Equivalent dynamic bearing load.

Taking account of EP additives in the lubricant

In accordance with ISO 281, EP additives can be taken into consideration in the following way:

- At a viscosity ratio $\kappa < 1$ and a contamination factor $e_C \geqq 0,2$, a value $\kappa = 1$ can be used in calculation in the case of lubricants with EP additives that have proven effective. Under severe contamination (contamination factor $e_C < 0,2$), the effectiveness of the additives under these contamination conditions must be proven. The effectiveness of the EP additives can be demonstrated in the actual application or on a rolling bearing test rig FE 8 to DIN 51 819-1.

  If the EP additives are proven effective and calculation is carried out using the value $\kappa = 1$, the life adjustment factor must be restricted to $a_{ISO} \leqq 3$. If the value a_{ISO} calculated for the actual κ is greater than 3, this value can be used in calculation.

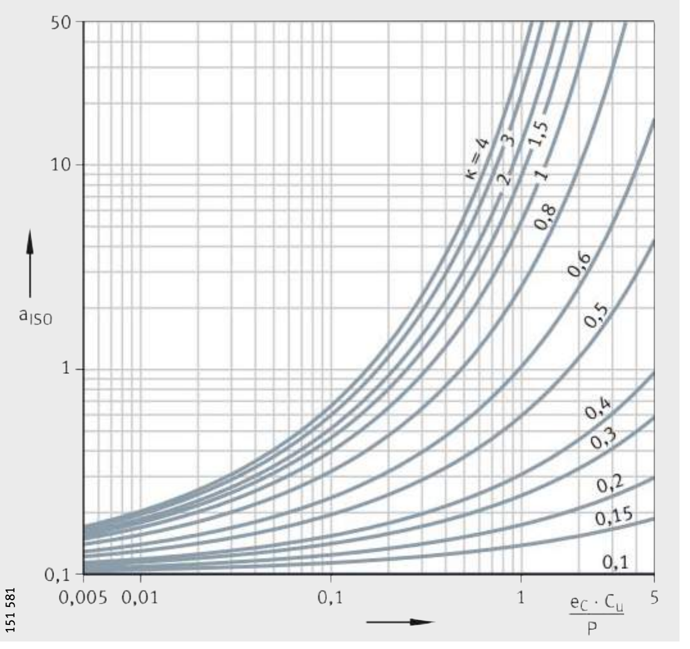


Figure 3
Life adjustment factor a_{ISO} for radial roller bearings

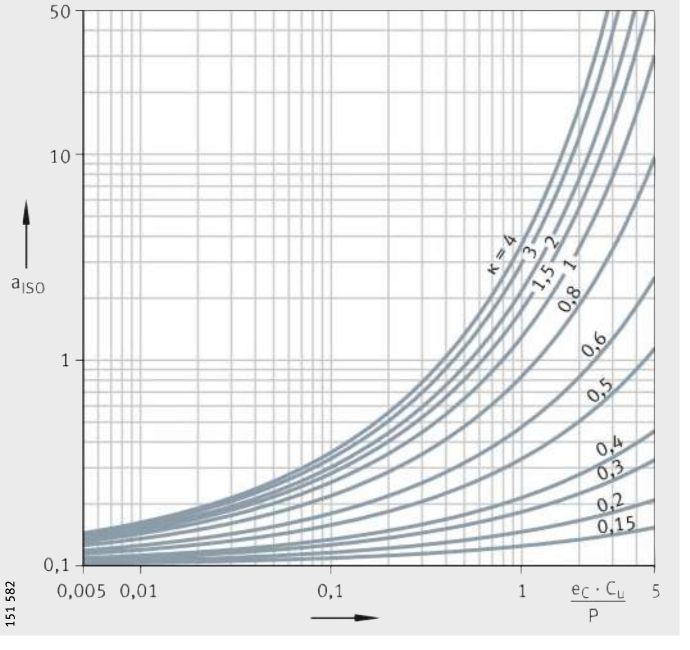


Figure 4
Life adjustment factor a_{ISO} for axial roller bearings

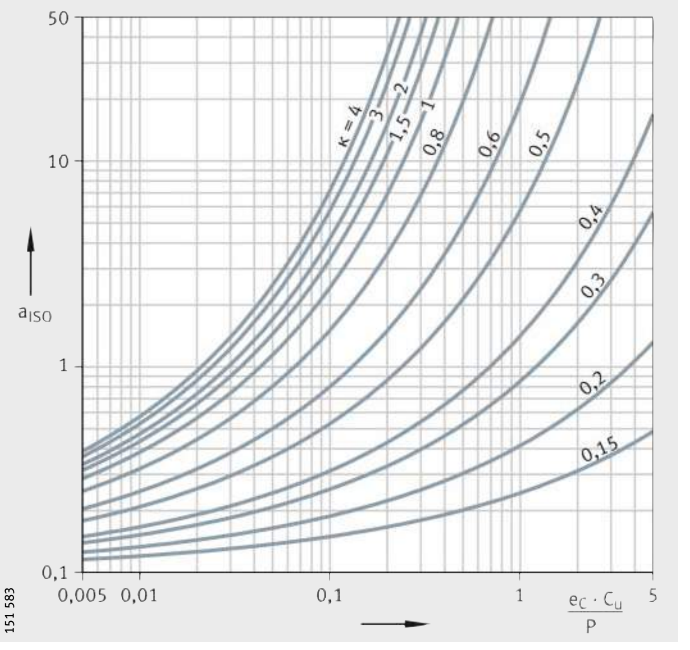


Figure 5
Life adjustment factor a_{ISO}
for radial ball bearings

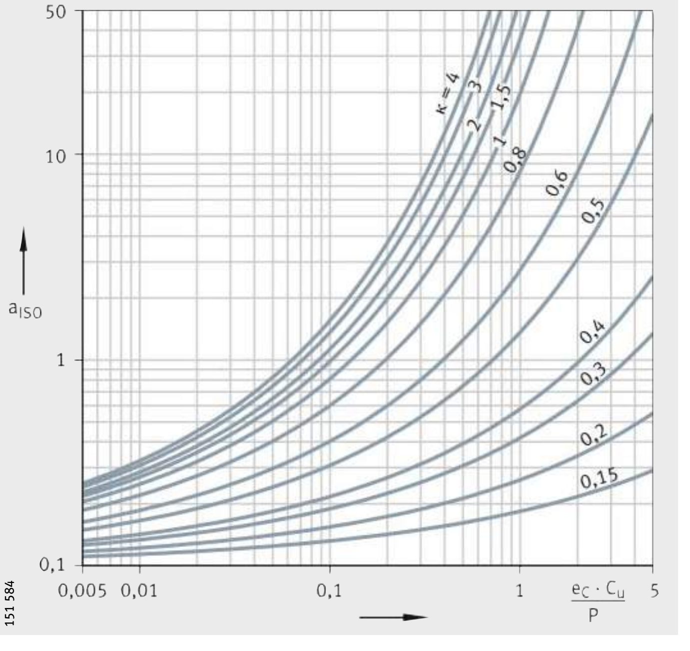


Figure 6
Life adjustment factor a_{ISO}
for axial ball bearings

Fatigue limit load

The fatigue limit load C_u is defined in accordance with ISO 281 as the load at which the most heavily loaded rolling element reaches the fatigue limit.

Life adjustment factor for contamination

If particles in the lubricant are subjected to overrolling, this can lead to plastic deformations of the raceway. Localised areas of high stress may then occur which lead to a reduction in the fatigue life. This influence of contaminants in the lubrication gap on the rating life is taken into consideration by the life adjustment factor for contamination e_C, see table.

The rating life is reduced by solid particles in the lubrication gap and is dependent on:

- the type, size, hardness and quantity of particles
- the relative lubricant film thickness
- the bearing size.

Due to the complex nature of the interaction between these influencing factors, only an approximate guide value can be attained. The values in the tables are valid for contamination by solid particles (factor e_C). They do not take account of other contamination such as that caused by water or other fluids.

Under severe contamination ($e_C \to 0$), the bearings may fail due to wear. In this case, the operating life is substantially less than the calculated life.

Factor e_C

Contamination	Factor e_C	
	$d_M < 100$ mm[1]	$d_M \geqq 100$ mm[1]
Extreme cleanliness - Particle size within lubricant film thickness - Laboratory conditions	1	1
High cleanliness - Oil filtered through extremely fine filter - Sealed, greased bearings	0,8 to 0,6	0,9 to 0,8
Standard cleanliness - Oil filtered through fine filter	0,6 to 0,5	0,8 to 0,6
Slight contamination - Slight contamination of oil	0,5 to 0,3	0,6 to 0,4
Typical contamination - Bearing is contaminated by wear debris from other machine elements	0,3 to 0,1	0,4 to 0,2
Heavy contamination - Bearing environment is heavily contaminated - Bearing arrangement is insufficiently sealed	0,1 to 0	0,1 to 0
Very heavy contamination	0	0

[1] d_M = mean bearing diameter $(d + D)/2$.

Detailed calculation of the contamination factor

For the following types of lubrication, the life adjustment factor e_c can be determined by means of diagrams or formulae (see DIN ISO 281:2009):

- recirculating oil lubrication with continuous oil filtration before entry into the bearing (online filtration)
- oil bath lubrication or recirculating oil lubrication with intermittent or one-off oil filtration (offline filtration)
- grease lubrication.

! It is recommended that detailed calculation of the contamination factor e_C is used when calculating the adjusted reference rating life in accordance with DIN ISO 281 Appendix 4. For calculation of the adjusted rating life L_{nm} nach ISO 281, the values in the tables should be used in preference.

In order to achieve the calculated bearing rating life, the bearings must be operated both from the beginning and after oil changes under the assumed conditions. It is therefore important to clean the bearings and the application thoroughly before mounting. It is also important to filter the oil before it is introduced into the system. The filter used for this purpose should be at least as effective as the filter in the system itself.

Recirculating oil lubrication with online oil filtration

For recirculating oil lubrication with continuous oil filtration, the contamination factor e_C can be determined by means of diagrams, *Figure 7*, page 30 to *Figure 10*, page 30. The diagram to be used is selected on the basis of the filter retention rate $\beta_{x(c)}$ according to ISO 16889 and the oil cleanliness code according to ISO 4406. The index (c) is the particle size in μm according to ISO 1171, see section Filtration values, page 141.

e_c = contamination factor
κ = viscosity ratio

D_{pw} = pitch circle diameter

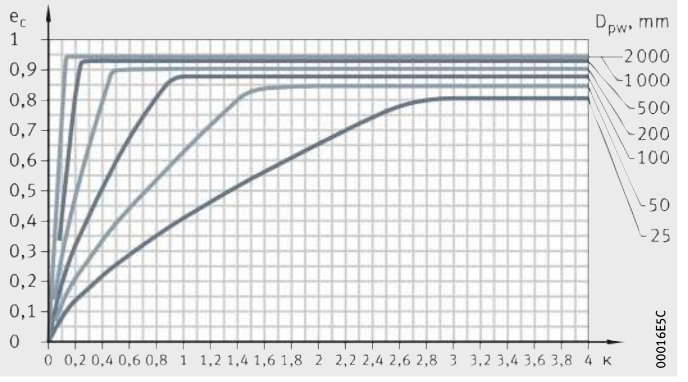


Figure 7
Filter retention rate $\beta_{6(c)} = 200$

e_c = contamination factor
κ = viscosity ratio

D_{pw} = pitch circle diameter

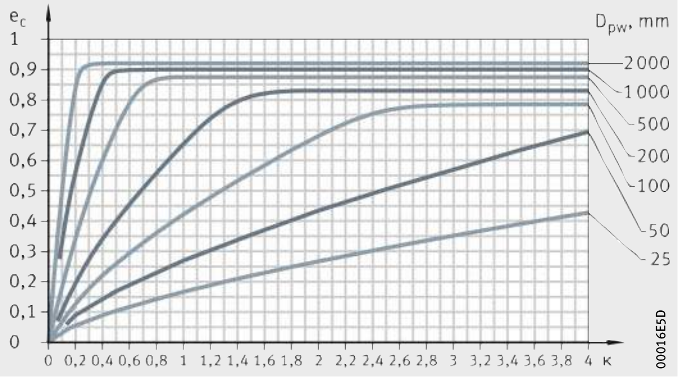


Figure 8
Filter retention rate $\beta_{12(c)} = 200$

e_c = contamination factor
κ = viscosity ratio

D_{pw} = pitch circle diameter

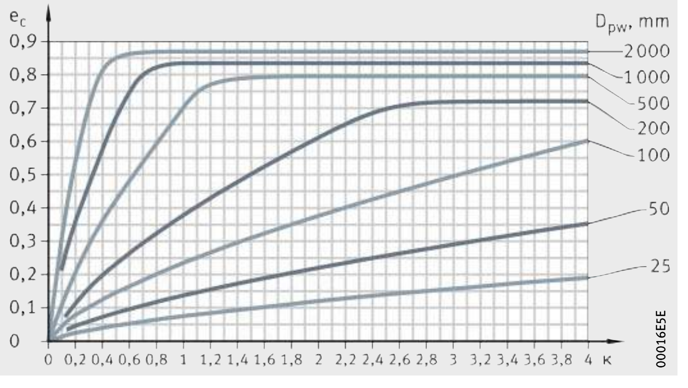


Figure 9
Filter retention rate $\beta_{25(c)} \geqq 75$

e_c = contamination factor
κ = viscosity ratio

D_{pw} = pitch circle diameter

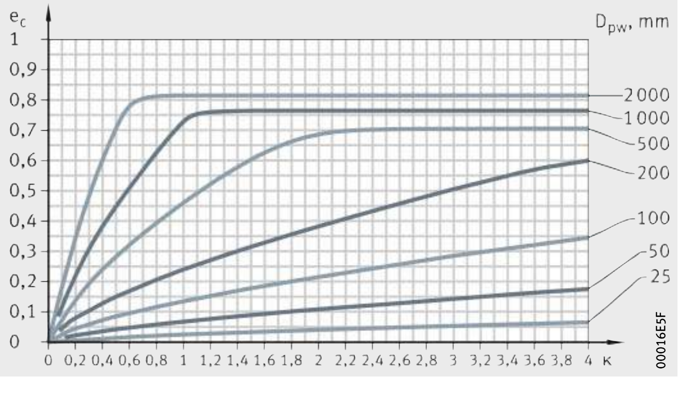


Figure 10
Filter retention rate $\beta_{40(c)} \geqq 75$

Oil bath lubrication or recirculating oil lubrication with offline oil filtration

For oil bath lubrication or recirculating oil lubrication with offline filtration, the contamination factor e_C can be determined by means of diagrams, *Figure 11* to *Figure 15*, page 32. The diagram to be used is selected on the basis of the oil cleanliness code according to ISO 4406.

e_c = contamination factor
κ = viscosity ratio

D_{pw} = pitch circle diameter

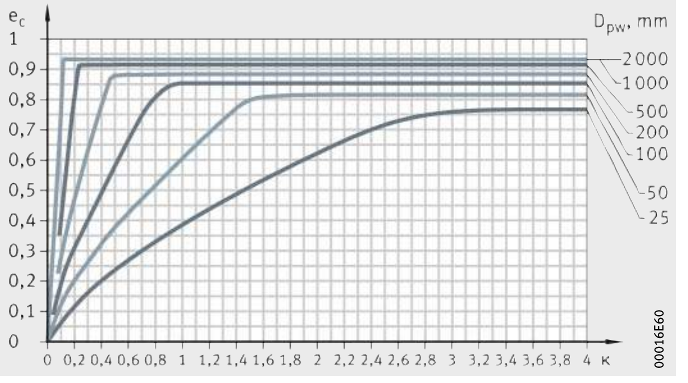


Figure 11
Oil cleanliness code –/13/10 according to ISO 4406

e_c = contamination factor
κ = viscosity ratio

D_{pw} = pitch circle diameter

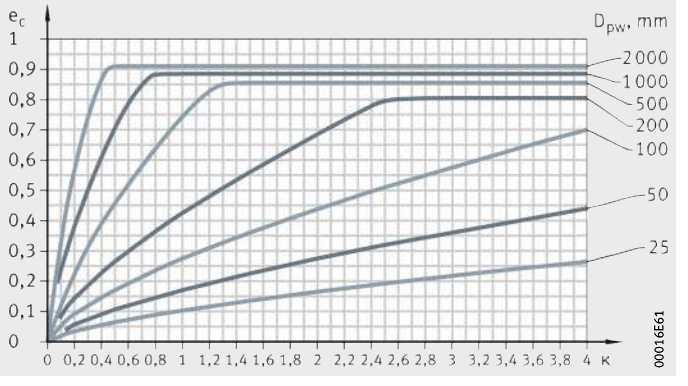


Figure 12
Oil cleanliness code –/15/12 according to ISO 4406

e_c = contamination factor
κ = viscosity ratio

D_{pw} = pitch circle diameter

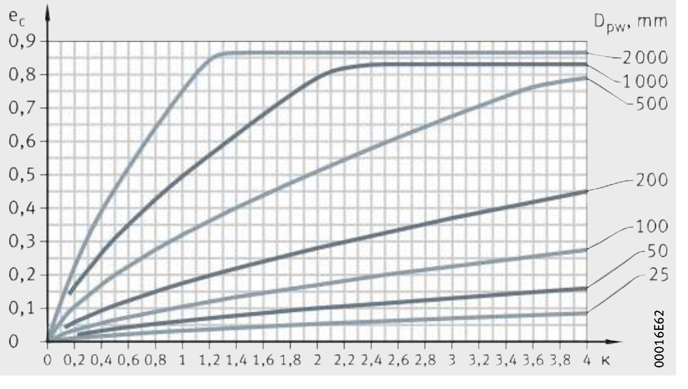
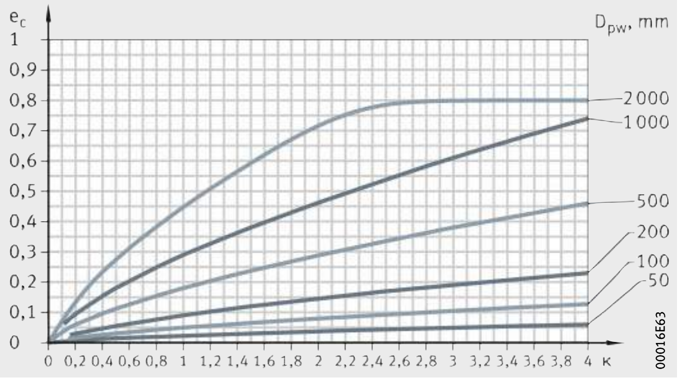


Figure 13
Oil cleanliness code –/17/14 according to ISO 4406

e_c = contamination factor
κ = viscosity ratio

D_{pw} = pitch circle diameter

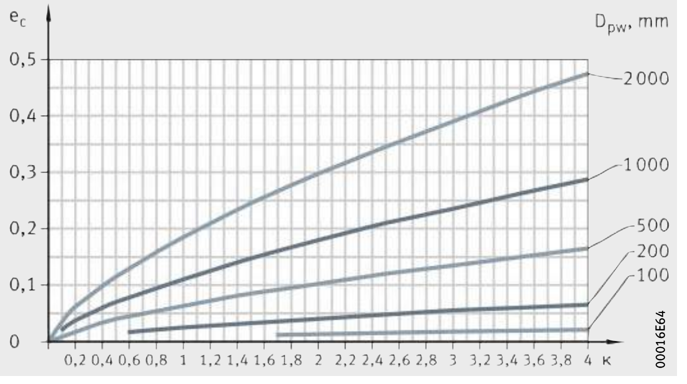
Figure 14
Oil cleanliness code –/19/16
according to ISO 4406



e_c = contamination factor
κ = viscosity ratio

D_{pw} = pitch circle diameter

Figure 15
Oil cleanliness code –/21/18
according to ISO 4406



Grease lubrication

For grease lubrication, the contamination factor e_C can be determined by means of diagrams, *Figure 16* to *Figure 20*, page 34. The diagram to be used is based on the operating conditions, see table.

Operating conditions	Diagram
High cleanliness Very clean mounting with careful flushing Very good sealing Continuous relubrication or short relubrication intervals	*Figure 16*
Bearings with effective sealing Greased for life	
Standard cleanliness Clean mounting with flushing Good sealing Relubrication in accordance with manufacturer's guidelines	*Figure 17*
Sealed bearings (for example, with sealing washers) Greased for life	
Slight to typical contamination Clean mounting Moderate sealing Relubrication in accordance with manufacturer's guidelines	*Figure 18*
Heavy contamination Mounting under workshop conditions Bearing and application not washed to appropriate standard Poor sealing Relubrication interval longer than manufacturer's guidelines	*Figure 19*
Severe contamination Machine in contaminated environment Inadequate sealing Long relubrication intervals	*Figure 20*

e_c = contamination factor
κ = viscosity ratio

D_{pw} = pitch circle diameter

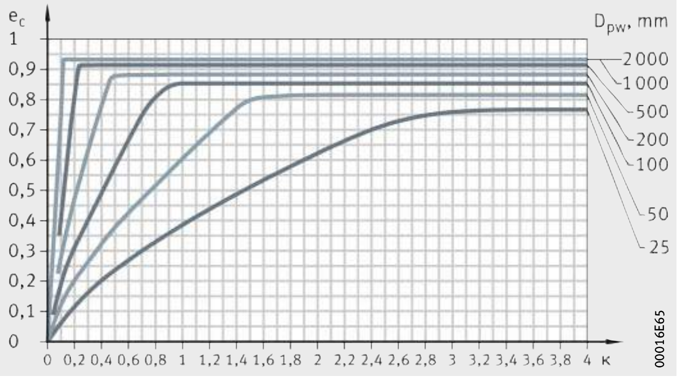


Figure 16
High cleanliness

e_c = contamination factor
κ = viscosity ratio

D_{pw} = pitch circle diameter

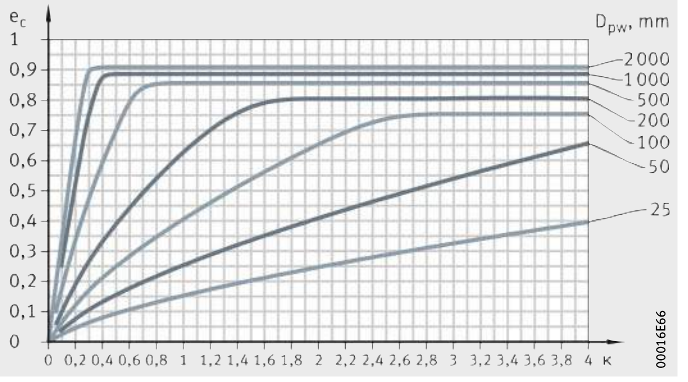


Figure 17
Standard cleanliness

e_c = contamination factor
κ = viscosity ratio

D_{pw} = pitch circle diameter

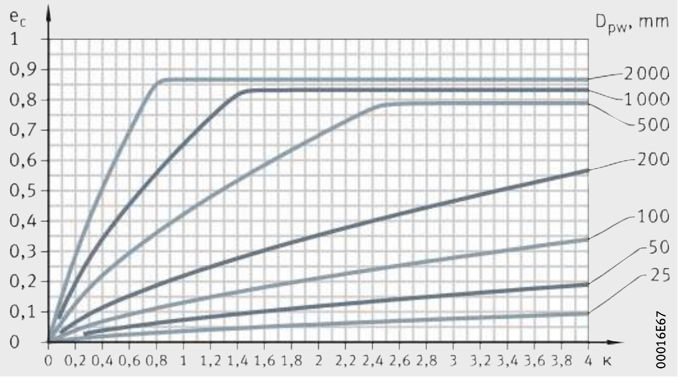


Figure 18
Slight to typical contamination

e_c = contamination factor
κ = viscosity ratio

D_{pw} = pitch circle diameter

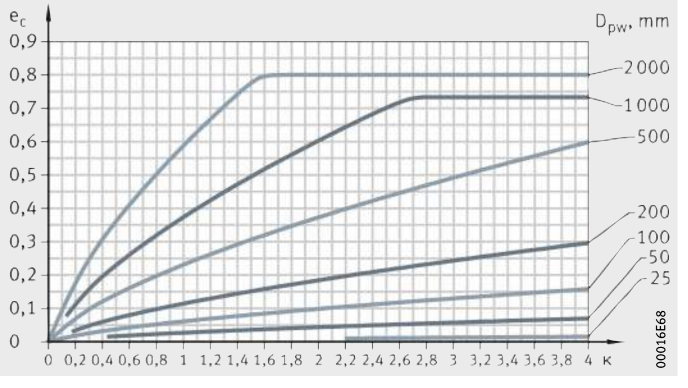


Figure 19
Heavy contamination

e_c = contamination factor
κ = viscosity ratio

D_{pw} = pitch circle diameter

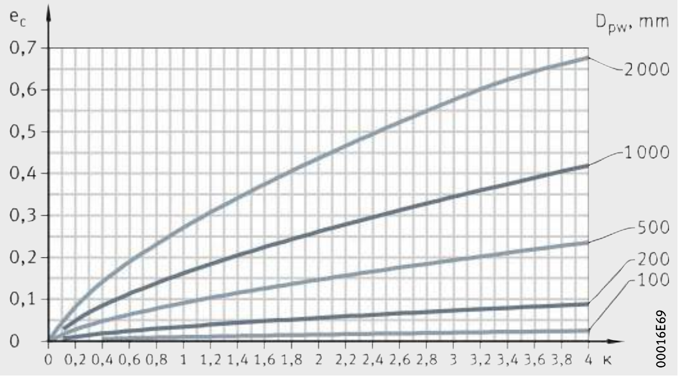


Figure 20
Severe contamination

Equivalent operating values

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

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

Variable load and speed

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

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

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

Variation in steps

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

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

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

Variable load at constant speed

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

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

Load varying in steps and constant speed

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

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

Constant load at variable speed

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

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

Constant load with speed varying in steps

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

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

Oscillating bearing motion

The equivalent speed is calculated as follows:

$$n = n_{osc} \cdot \frac{\varphi}{180°}$$

The formula is valid only if the angle of oscillation is greater than twice the angular pitch of the rolling elements. If the angle of oscillation is smaller, there is a risk of false brinelling.

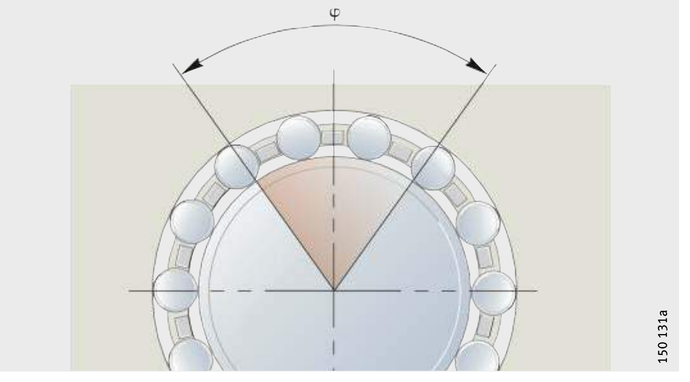


Figure 21
Angle of oscillation φ

Symbols, units and definitions

n — min^{-1}
Mean speed

T — min
Time period under consideration

P — N
Equivalent bearing load

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

a_i, $a(t)$ — –
Life adjustment factor a_{ISO} for current operating condition,
see section Life adjustment factor a_{ISO}, page 25

n_i, $n(t)$ — min^{-1}
Bearing speed for a particular operating condition

q_i — %
Duration of operating condition as a proportion of the total operating period;
$q_i = (\Delta t_i/T) \cdot 100$

F_i, $F(t)$ — N
Bearing load for a particular operating condition

n_{osc} — min^{-1}
Frequency of oscillating motion

φ — °
Angle of oscillation, *Figure 21*.

Friction and increases in temperature

Friction

The friction in a rolling bearing is made up of several components, see table. Due to the large number of influencing factors, such as dynamics in speed and load, tilting and skewing resulting from installation, actual frictional torques and frictional energy may deviate significantly from the calculated values. If the frictional torque is an important design criterion, please consult the Schaeffler engineering service.

Frictional component and influencing factor

Frictional component	Influencing factor
Rolling friction	Magnitude of load
Sliding friction of rolling elements Sliding friction of cage	Magnitude and direction of load Speed and lubrication conditions, running-in condition
Fluid friction (flow resistance)	Type and speed Type, quantity and operating viscosity of lubricant
Seal friction	Type and preload of seal

The idling friction is dependent on the lubricant quantity, speed, operating viscosity of the lubricant, seals and the running-in condition of the bearing.

Heat dissipation

Friction is converted into heat. This must be dissipated from the bearing. The equilibrium between the frictional energy and heat dissipation allows calculation of the thermally safe operating speed n_{ϑ}.

Heat dissipation by the lubricant

if oil lubrication is used, some of the heat is dissipated by the oil. Recirculating oil lubrication with additional cooling is particularly effective.

Grease does not give dissipation of heat.

Heat dissipation via the shaft and housing

Heat dissipation via the shaft and housing is dependent on the temperature difference between the bearing and the surrounding structure, *Figure 1*.

Any additional adjacent sources of heat or thermal radiation must be taken into consideration.

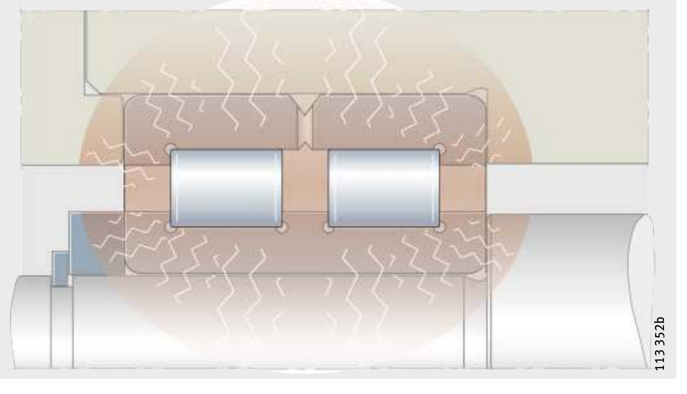


Figure 1
Temperature distribution between bearing, shaft and housing

Friction and increases in temperature

Determining the friction values

For this process, the speed and load must be known. The type of lubrication, lubrication method and viscosity of the lubricant at operating temperature are other factors necessary for calculation.

$$M_R = M_0 + M_1$$

Total frictional torque M_R (calculation of axially loaded cylindrical roller bearings, see page 43):

Frictional power N_R:

$$N_R = M_R \cdot \frac{n}{9\,550}$$

Frictional torque as a function of speed for $\nu \cdot n \geqq 2\,000$:

$$M_0 = f_0 \cdot \left(\nu \cdot n\right)^{2/3} \cdot {d_M}^3 \cdot 10^{-7}$$

Frictional torque as a function of speed for $\nu \cdot n < 2\,000$:

$$M_0 = f_0 \cdot 160 \cdot {d_M}^3 \cdot 10^{-7}$$

Frictional torque as a function of load for cylindrical roller bearings:

$$M_1 = f_1 \cdot F \cdot d_M$$

Frictional torque as a function of load for ball bearings, tapered roller bearings and spherical roller bearings:

$$M_1 = f_1 \cdot P_1 \cdot d_M$$

M_R Nmm
Total frictional torque

M_0 Nmm
Frictional torque as a function of speed

M_1 Nmm
Frictional torque as a function of load

N_R W
Frictional power

n min^{-1}
Operating speed

f_0 –
Bearing factor for frictional torque as a function of speed, *Figure 2*, page 39 and tables from page 40 to page 42

ν mm^2s^{-1}
Kinematic viscosity of lubricant at operating temperature
In the case of grease, the decisive factor is the viscosity of the base oil at operating temperature

d_M mm
Mean bearing diameter (d + D)/2

f_1 –
Bearing factor for frictional torque as a function of load, see tables from page 40 to page 42

F_r, F_a N
Radial load for radial bearings, axial load for axial bearings

P_1 N
Decisive load for frictional torque.
For ball bearings, tapered roller bearings and spherical roller bearings, see page 42.

Bearing factors

The bearing factors f_0 and f_1 are mean values from series of tests and correspond to the data according to ISO 15 312.

They are valid for grease lubrication applied to bearings after running-in. In the freshly greased state, the bearing factor f_0 can be two to five times higher.

If oil bath lubrication is used, the oil level must reach the centre of the lowest rolling element. If the oil level is higher, f_0 may be up to three times the value given in the table, *Figure 2*.

f_0 = bearing factor
h = oil level
d_M = mean bearing diameter (d +D)/2

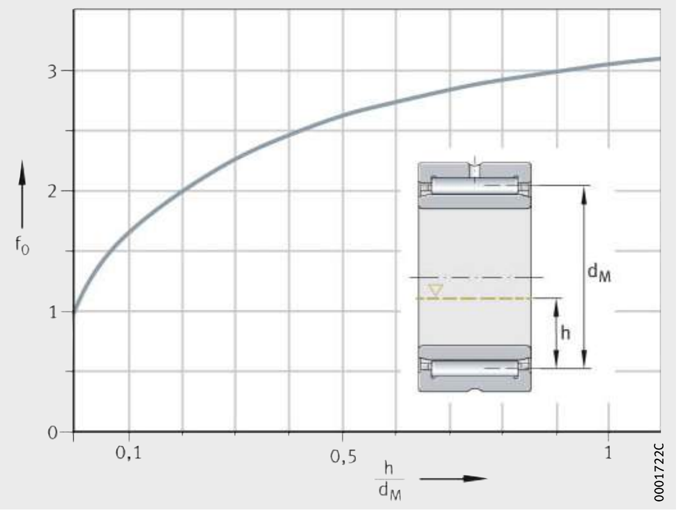


Figure 2
Increase in the bearing factor, as a function of the oil level

Bearing factors for needle roller bearings, drawn cup needle roller bearings, needle roller and cage assemblies

Series	Bearing factor f_0		Bearing factor f_1
	Grease, pneumatic oil	Oil bath, recirculating oil	
NA48	3	5	0,0005
NA49	4	5,5	
RNA48	3	5	
RNA49	4	5,5	
NA69	7	10	
RNA69			
NKI, NK, NKIS, NKS, NAO, RNO, K	$(12 \cdot B)/(33 + d)$	$(18 \cdot B)/(33 + d)$	
HK, BK	$(24 \cdot B)/(33 + d)$	$(36 \cdot B)/(33 + d)$	
HN	$(30 \cdot B)/(33 + d)$	$(45 \cdot B)/(33 + d)$	

Bearing factors for cylindrical roller bearings, full complement

Series	Bearing factor f_0		Bearing factor f_1
	Grease, pneumatic oil	Oil bath, recirculating oil	
SL1818	3	5	0,00055
SL1829	4	6	
SL1830	5	7	
SL1822	5	8	
SL0148, SL0248	6	9	
SL0149, SL0249	7	11	
SL1923	8	12	
SL1850	9	13	

Bearing factors for cylindrical roller bearings with cage

Series	Bearing factor f_0		Bearing factor f_1
	Grease, pneumatic oil	Oil bath, recirculating oil	
LSL1923	1	3,7	0,00020
ZSL1923	1	3,8	0,00025
2..-E	1,3	2	0,00030
3..-E			0,00035
4			0,00040
10, 19			0,00020
22..-E	2	3	0,00040
23..-E	2,7	4	0,00040
30	1,7	2,5	0,00040

Bearing factors for axial roller bearings

Series	Bearing factor f_0		Bearing factor f_1
	Grease, pneumatic oil	Oil bath, recirculating oil	
AXK, AXW	3	4	0,0015
811, K811	2	3	
812, K812			
893, K893			
894, K894			

Bearing factors for combined bearings

Series	Bearing factor f_0		Bearing factor f_1
	Grease, pneumatic oil	Oil bath, recirculating oil	
ZARN, ZARF	3	4	0,0015
NKXR	2	3	
NX, NKX	2	3	$0{,}001 \cdot (F_a/C_0)^{0{,}33}$
ZKLN, ZKLF	4	6	
NKIA, NKIB	3	5	0,0005

Bearing factors for tapered roller bearings

Series	Bearing factor f_0		Bearing factor f_1
	Grease, pneumatic oil	Oil bath, recirculating oil	
302, 303, 320, 329, 330, T4CB, T7FC	2	3	0,0004
313, 322, 323, 331, 332, T2EE, T2ED, T5ED	3	4,5	

Bearing factors for axial and radial spherical roller bearings

Series	Bearing factor f_0		Bearing factor f_1
	Grease, pneumatic oil	Oil bath, recirculating oil	
213	2,3	3,5	$0{,}0005 \cdot (P_0/C_0)^{0{,}33}$
222	2,7	4	
223	3	4,5	$0{,}0008 \cdot (P_0/C_0)^{0{,}33}$
230, 239			$0{,}00075 \cdot (P_0/C_0)^{0{,}5}$
231	3,7	5,5	$0{,}0012 \cdot (P_0/C_0)^{0{,}5}$
232	4	6	$0{,}0016 \cdot (P_0/C_0)^{0{,}5}$
240	4,3	6,5	$0{,}0012 \cdot (P_0/C_0)^{0{,}5}$
241	4,7	7	$0{,}0022 \cdot (P_0/C_0)^{0{,}5}$
292..-E	1,7	2,5	0,00023
293..-E	2	3	0,00030
294..-E	2,2	3,3	0,00033

Bearing factors for deep groove ball bearings

Series	Bearing factor f_0		Bearing factor f_1
	Grease, pneumatic oil	Oil bath, recirculating oil	
618, 618..-2Z, (2RSR)	1,1	1,7	$0{,}0005 \cdot (P_0/C_0)^{0{,}5}$
160	1,1	1,7	$0{,}0007 \cdot (P_0/C_0)^{0{,}5}$
60, 60..-2RSR, 60..-2Z, 619, 619..-2Z, (2RSR)	1,1	1,7	
622..-2RSR	1,1	–	$0{,}0009 \cdot (P_0/C_0)^{0{,}5}$
623..-2RSR	1,1	–	
62, 62..-2RSR, 62..-2Z	1,3	2	
63, 63..-2RSR, 63..-2Z	1,5	2,3	
64	1,5	2,3	
42..-B	2,3	3,5	$0{,}0010 \cdot (P_0/C_0)^{0{,}5}$
43..-B	4	6	

Friction and increases in temperature

Bearing factors for angular contact ball bearings

Series	Bearing factor f_0		Bearing factor f_1
	Grease, pneumatic oil	Oil bath, recirculating oil	
70..-B, 70..-B-2RS	1,3	2	$0{,}001 \cdot (P_0/C_0)^{0{,}33}$
718..-B, 72..-B, 72..-B-2RS			
73..-B, 73..-B-2RS	2	3	
30..-B, 30..-B-2RSR, 30..-B-2Z	2,3	3,5	
32..-B, 32..-B-2RSR, 32..-B-2Z, 32			
38..-B, 38..-B-2RSR, 38..-B-2Z			
33..-B, 33..-B-2RSR, 33, 33..-DA	4	6	

Bearing factors for self-aligning ball bearings

Series	Bearing factor f_0		Bearing factor f_1
	Grease, pneumatic oil	Oil bath, recirculating oil	
12	1	2,5	$0{,}0003 \cdot (P_0/C_0)^{0{,}4}$
13	1,3	3,5	
22	1,7	3	
23	2	4	

Bearing factors for four point contact bearings

Series	Bearing factor f_0		Bearing factor f_1
	Grease, pneumatic oil	Oil bath, recirculating oil	
QJ2, QJ3	2,7	4	$0{,}001 \cdot (P_0/C_0)^{0{,}33}$

Bearing factors for axial deep groove ball bearings

Series	Bearing factor f_0		Bearing factor f_1
	Grease, pneumatic oil	Oil bath, recirculating oil	
511, 512, 513, 514, 532, 533	1	1,5	$0{,}0012 \cdot (F_a/C_0)^{0{,}33}$
522, 523, 524, 542, 543	1,3	2	

Decisive load for ball bearings, tapered roller bearings and spherical roller bearings

Bearing type	Single bearing P_1	Bearing pair P_1
Deep groove ball bearings	$3{,}3 \cdot F_a - 0{,}1 \cdot F_r$	–
Angular contact ball bearings, single row	$F_a - 0{,}1 \cdot F_r$	$1{,}4 \cdot F_a - 0{,}1 \cdot F_r$
Angular contact ball bearings, double row	$1{,}4 \cdot F_a - 0{,}1 \cdot F_r$	–
Four point contact bearings	$1{,}5 \cdot F_a + 3{,}6 \cdot F_r$	–
Tapered roller bearings	$2 \cdot Y \cdot F_a$ or F_r, use the larger value	$1{,}21 \cdot Y \cdot F_a$ or F_r, use the larger value
Spherical roller bearings	$1{,}6 \cdot F_a/e$ if $F_a/F_r > e$ $F_r \{1 + 0{,}6 \cdot [F_a/(e \cdot F_r)]^3\}$ if $F_a/F_r \leqq e$.	

If $P_1 \leqq F_r$, then $P_1 = F_r$.

Cylindrical roller bearings under axial load

In cylindrical roller bearings under axial load, sliding friction between the end faces of the rolling elements and the ribs on the rings leads to an additional frictional torque M_2.

The total frictional torque is therefore calculated as follows:

$$M_R = M_0 + M_1 + M_2$$

$$M_2 = f_2 \cdot F_a \cdot d_M$$

$$A = k_B \cdot 10^{-3} \cdot d_M^{\,2,1}$$

M_R Nmm
Total frictional torque

M_0 Nmm
Frictional torque as a function of speed

M_1 Nmm
Frictional torque as a function of radial load

M_2 Nmm
Frictional torque as a function of axial load

f_2 –
Factor as a function of the bearing series, *Figure 3* and *Figure 4*, page 44

F_a N
Axial dynamic bearing load

d_M mm
Mean bearing diameter (d + D)/2

A –
Bearing parameter according to formula

k_B –
Factor as a function of the bearing series, see table, page 44.

The bearing factors f_2 are subject to wide scatter. They are valid for recirculating oil lubrication with an adequate quantity of oil. The curves must not be extrapolated, *Figure 3* and *Figure 4*, page 44.

Cylindrical roller bearings of standard design

f_2 = bearing factor
ν = operating viscosity
n = operating speed
d_M = mean bearing diameter
$\nu \cdot n \cdot d_M$ = operating parameter
F_a = axial dynamic bearing load
A = bearing parameter

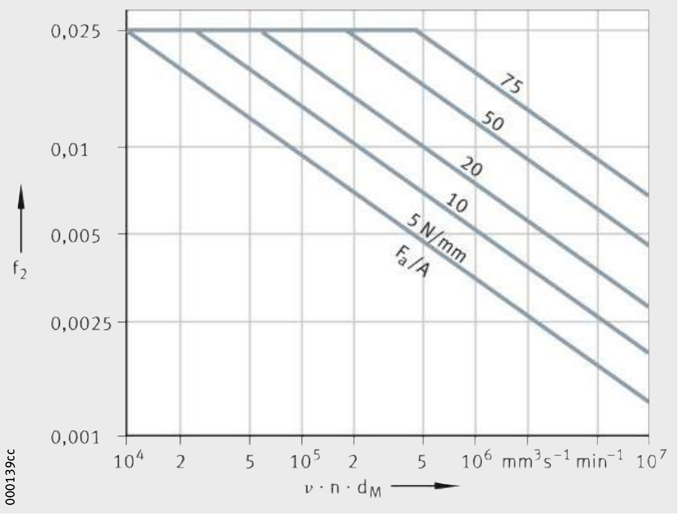


Figure 3
Bearing factor f_2,
as a function of operating parameter

Cylindrical roller bearings of TB design

f_2 = bearing factor
ν = operating viscosity
n = operating speed
d_M = mean bearing diameter
$\nu \cdot n \cdot d_M$ = operating parameter
F_a = axial dynamic bearing load
A = bearing parameter

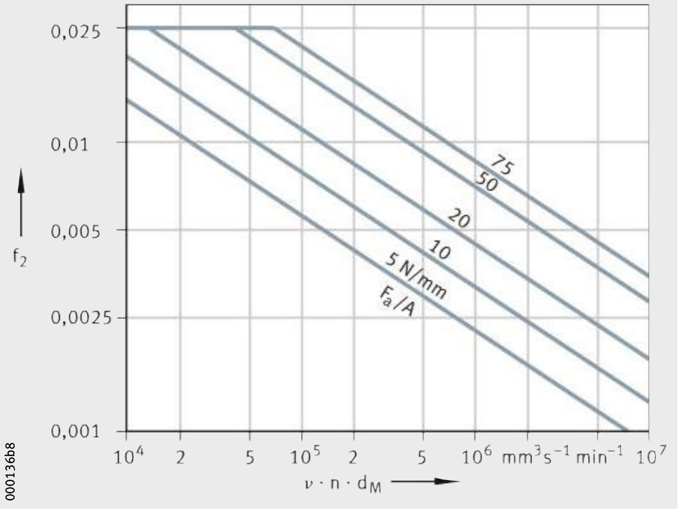


Figure 4
Bearing factor f_2,
as a function of operating parameter

Bearing factor k_B

Bearing series	Factor k_B
SL1818, SL0148	4,5
SL1829, SL0149	11
SL1830, SL1850	17
SL1822	20
LSL1923	28
SL1923	30
NJ2..-E, NJ22..-E, NUP2..-E, NUP22..-E	15
NJ3..-E, NJ23..-E, NUP3..-E, NUP23..-E	20
NJ4	22

Speeds

On the basis of DIN 732-1, calculation of the thermal reference speed n_B has been standardised in ISO 15 312. The calculation of reference speeds has been matched to this standard.

Thermal reference speed

The thermal reference speed n_B is used as an ancillary value when calculating the thermally safe operating speed n_ϑ. This is the speed at which, under defined reference conditions, a bearing operating temperature of +70 °C is reached.

Reference conditions

The reference conditions are based on the usual operating conditions of the most significant bearing types and sizes.

They are defined in ISO 15 312 as follows:

- mean ambient temperature $\vartheta_{Ar} = +20$ °C
- mean bearing temperature at the outer ring $\vartheta_r = +70$ °C
- load on radial bearings $P_{1r} = 0,05 \cdot C_{0r}$
- load on axial bearings $P_{1a} = 0,02 \cdot C_{0a}$

operating viscosities (axial bearings according to DIN 732-1)
For radial bearings, the reference speeds are approximately the same for oil and grease lubrication:

  - radial bearings: 12 mm^2s^{-1} (ISO VG 32)
  - axial bearings: 24 mm^2s^{-1} (ISO VG 68)

- heat dissipation via the bearing seating surfaces, see section Bearing seating surface $A_r \leqq 50\,000$ mm^2 and section Bearing seating surface $A_r > 50\,000$ mm^2.

Bearing seating surface $A_r \leqq 50\,000$ mm^2

Radial bearings have a heat dissipation value $q_r = 0,016$ W/mm^2.

Axial bearings have a heat dissipation value $q_r = 0,02$ W/mm^2.

Bearing seating surface $A_r > 50\,000$ mm^2

For radial bearings, the heat dissipation in W/mm^2 is:

$$q_r = 0,016 \cdot \left(\frac{A_r}{50\,000}\right)^{-0,34}$$

For axial bearings, the heat dissipation in W/mm^2 is:

$$q_r = 0,020 \cdot \left(\frac{A_r}{50\,000}\right)^{-0,16}$$

A_r mm^2
Bearing seating surface

q_r W/mm^2
Heat dissipation.

Limiting speed

The limiting speed n_G is based on practical experience and takes account of additional criteria such as smooth running, sealing function and centrifugal forces.

The limiting speed must not be exceeded even under favourable operating and cooling conditions.

Thermally safe operating speed

The thermally safe operating speed n_ϑ is calculated according to E DIN 732:2008. The basis for the calculation is the heat balance in the bearing, the equilibrium between the frictional energy as a function of speed and the heat dissipation as a function of temperature. When conditions are in equilibrium, the bearing temperature is constant.

The permissible operating temperature determines the thermally safe operating speed n_ϑ of the bearing. The preconditions for calculation are correct fitting, normal operating clearance and constant operating conditions.

The calculation method is not valid for:

- sealed bearings with contact seals, since the maximum speed is restricted by the permissible sliding speed at the seal lip
- yoke and stud type track rollers
- aligning needle roller bearings
- axial deep groove and axial angular contact ball bearings.

Bearings with special cages (such as TBH, T9H) are capable, due to their cage designs, of speeds that are higher than those calculated according to this method.

The limiting speed n_G must always be observed.

Calculation of the thermally safe operating speed

The thermally safe operating speed n_ϑ is a product of the reference speed n_B and the speed ratio f_n:

$$n_\vartheta = n_B \cdot f_n$$

The speed ratio is derived from *Figure 1*, page 48:

$$k_L \cdot f_n^{5/3} + k_P \cdot f_n = 1$$

In the normal range $0{,}01 < k_L < 10$ and $0{,}01 < k_P < 10$, f_n can be calculated using an approximation formula:

$$f_n = \frac{490{,}77}{1 + 498{,}78 \cdot k_L^{0{,}599} + 852{,}88 \cdot k_P^{0{,}963} - 504{,}5 \cdot k_L^{0{,}055} \cdot k_P^{0{,}832}}$$

Heat dissipation via the bearing seating surfaces $\dot{Q}_S$, *Figure 2*, page 48:

$$\dot{Q}_S = k_q \cdot A_r \cdot \Delta\vartheta_A$$

Heat dissipation by the lubricant $\dot{Q}_L$:

$$\dot{Q}_L = 0{,}0286 \frac{\text{kW}}{\text{l/min} \cdot \text{K}} \cdot \dot{V}_L \cdot \Delta\vartheta_L$$

Total dissipated heat flow $\dot{Q}$:

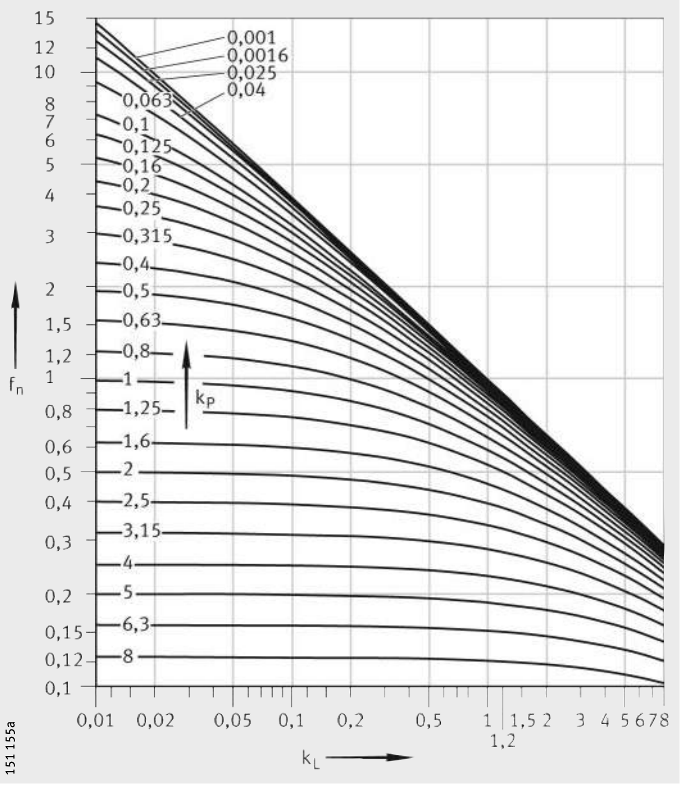
$$\dot{Q} = \dot{Q}_S + \dot{Q}_L - \dot{Q}_E$$

Lubricant film parameter k_L:

$$k_L = 10^{-6} \cdot \frac{\pi}{30} \cdot n_B \cdot \frac{10^{-7} \cdot f_0 \cdot (\nu \cdot n_B)^{\frac{2}{3}} \cdot d_M^{\,3}}{\dot{Q}}$$

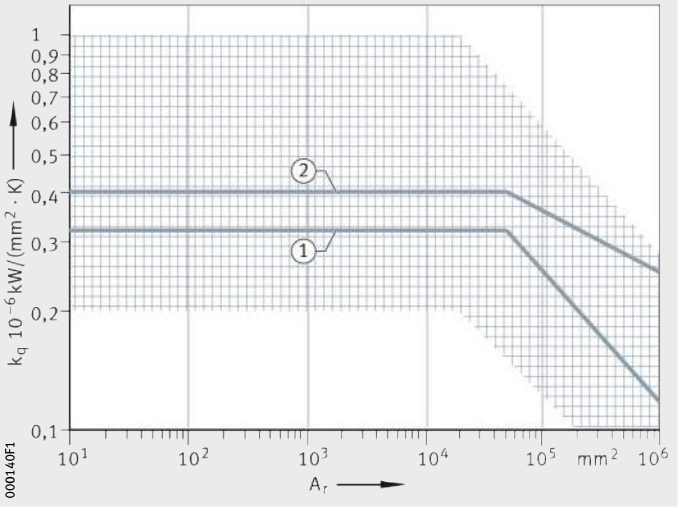
Load parameter k_P:

$$k_P = 10^{-6} \cdot \frac{\pi}{30} \cdot n_B \cdot \frac{f_1 \cdot P_1 \cdot d_M}{\dot{Q}}$$



f_n = speed ratio
k_L = lubricant film parameter
k_p = load parameter

Figure 1
Speed ratio



k_q = heat transfer coefficient
A_r = bearing seating surface

① Reference condition for radial bearings
② Reference condition for axial bearings

Figure 2
Heat transfer coefficient,
as a function
of the bearing seating surface

Symbols, units and definitions

A_r mm^2

Bearing seating surface
for radial bearings: $A_r = \pi \times B \times (D + d)$
axial bearings: $A_r = \pi/2 \times (D^2 - d^2)$
tapered roller bearings: $A_r = \pi \times T \times (D + d)$
axial spherical roller bearings: $A_r = \pi/4 \times (D^2 + d_1{}^2 - D_1{}^2 - d^2)$

Symbols, units and definitions continued

B mm
Bearing width

d mm
Bearing bore diameter

d_1 mm
Outside diameter of shaft locating washer

D mm
Bearing outside diameter

D_1 mm
Inside diameter of housing locating washer

d_M mm
Mean bearing diameter (D + d)/2

f_0 –
Bearing factor for frictional torque as a function of speed, see section Bearing factors, page 39

f_1 –
Bearing factor for frictional torque as a function of load, see section Bearing factors, page 39

f_n –
Speed ratio, *Figure 1*, page 48

k_L –
Lubricant film parameter

k_P –
Load parameter

k_q 10^{-6} kW/(mm^2 · K)
Heat transfer coefficient of bearing seating surface, *Figure 2*, page 48
It is dependent on the housing design and size, the housing material and the mounting position
In normal applications, the heat transfer coefficient of bearing seating surfaces up to 25 000 mm^2 is between 0,2 · 10^{-6} kW/(mm^2 · K) and 1,0 · 10^{-6} kW/(mm^2 · K)

n_ϑ min^{-1}
Thermally safe operating speed

n_B min^{-1}
Reference speed according to dimension tables

P_1 N
Radial load for radial bearings, axial load for axial bearings

q_r W/mm^2
Heat flow density

$\dot{Q}$ kW
Total dissipated heat flow

$\dot{Q}_E$ kW
Heat flow due to heating by external source

$\dot{Q}_L$ kW
Heat flow dissipated by the lubricant

$\dot{Q}_S$ kW
Heat flow dissipated via the bearing seating surfaces

T mm
Total width of tapered roller bearing

$\dot{V}_L$ l/min
Oil flow

$\Delta\vartheta_A$ K
Difference between mean bearing temperature and ambient temperature

$\Delta\vartheta_L$ K
Difference between oil input temperature and oil output temperature

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

Friction and increases in temperature

Operating temperature

The operating temperature of a bearing arrangement increases after startup. Once an equilibrium has been achieved between heat generation and heat dissipation, the temperature remains constant.

This equilibrium temperature ϑ_B can be calculated using the formulae for the heat flow generated by the bearing $\dot{Q}_{bearing}$ and the heat flow dissipated to the environment $\dot{Q}_S$. It is heavily dependent on the temperature difference between the bearing, adjacent parts and environment.

If the data K_t and q_{LB} required here are known (possibly as a result of tests), the thermal balance can be used to derive the equilibrium temperature ϑ_B.

Generated heat flow

Heat flow generated by bearing friction:

$$\dot{Q}_{bearing} = N_R = M_R \cdot \frac{n}{9550} = 1{,}047 \cdot 10^{-4} \cdot n \cdot M_R$$

Dissipated heat flow

Heat flow dissipated to the environment:

$$\dot{Q}_S = k_q \cdot A_r \cdot \Delta\vartheta_A$$

Additional dissipated heat flow

In the case of recirculating oil lubrication, the oil additionally dissipates heat. The dissipated heat flow $_L$ can be determined in the case of normal mineral oils using $\rho = 0{,}89\ g/cm^3$:

$$\dot{Q}_L = 0{,}0286 \cdot \frac{W}{l/min \cdot K} \cdot \dot{V}_L \cdot \Delta\vartheta_L$$

Equilibrium temperature

The equilibrium temperature of the bearing is determined by equating the heat introduced and heat dissipated ($_{bearing}$ = $_S$ + $_L$) and resolving it by ϑ_B:

$$\vartheta_B = \frac{1{,}047 \cdot 10^{-4} \cdot n \cdot M_R - 0{,}0286 \cdot \frac{W}{l/min \cdot K} \cdot \dot{V}_L \cdot \Delta\vartheta_L}{k_q \cdot A_r} + \vartheta_U$$

$\dot{Q}_{bearing}$ W
Heat flow due to bearing friction

N_R W
Frictional power

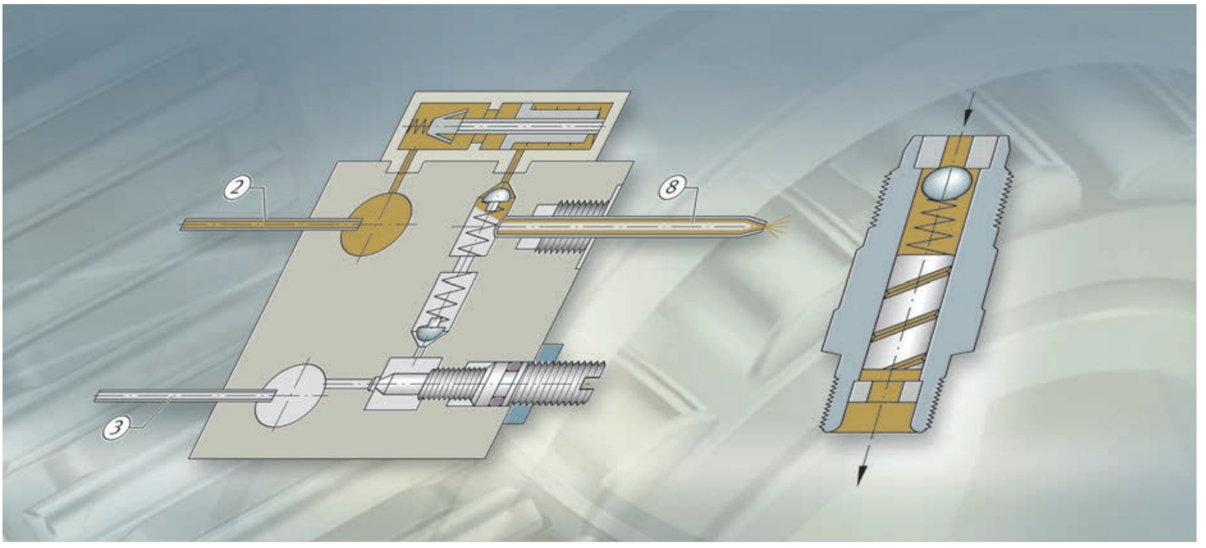
M_R Nmm
Total frictional torque

n min^{-1}
Operating speed

ϑ_B °C
Operating temperature

ϑ_U °C
Ambient temperature.

The temperature prediction derived from such a calculation is relatively imprecise, since the values to be inputted are generally not known to a precise degree. A secure basis can only be obtained if the equilibrium temperature is determined in a test run.



Lubrication methods

Lubrication methods

Lubrication methods

Lubrication methods

When a machine is being designed, the method to be used for lubrication of the rolling bearings should be defined as early as possible. A decision may be made in favour of grease or oil lubrication, or in special cases lubrication by means of solid substances.

Grease lubrication

Grease lubrication is used in approx. 90 % of all rolling bearing arrangements.

The advantages of grease lubrication include:

- very little design work required
- sealing action supported by the grease
- long operating life with maintenance-free lubrication, eliminating the requirement for lubrication devices
- suitability for speed parameters $n \cdot d_M \leqq 2{,}6 \cdot 10^6 \text{ min}^{-1} \cdot \text{mm}$
- longer emergency running phase in case of lubrication supply failure
- low frictional torque.

Under normal operating and environmental conditions, lifetime lubrication (lubrication for life) is often possible.

If high demands are present, for example in terms of speed, temperature and load, it will be necessary to plan for relubrication at appropriate time intervals. In this case, it is necessary to provide inlet and outlet ducts for grease as well as a collection chamber for used grease. If short relubrication intervals are to be used, it may also be necessary to provide a grease pump and a regulator for the grease quantity.

Oil lubrication

Oil lubrication presents itself as a sensible option if adjacent machine elements are already supplied with oil or if heat is to be dissipated by the lubricant. Heat dissipation may be necessary if high speeds or loads are present or if the bearing arrangement is subjected to heating by an external source.

If minimal quantity lubrication is used, small quantities of oil can be metered precisely. This can be achieved by means of oil drop lubrication, oil pulse lubrication or pneumatic oil lubrication. This offers the advantage that splash losses are prevented and bearing friction is kept to a low level. The use of air as a carrier allows targeted feed and flow, giving support to the sealing arrangement.

Oil injection lubrication can be used to achieve targeted supply to all the contact points in rapidly rotating bearings as well as good cooling.

Selection of the lubrication method

When selecting a method for lubricating bearings, attention must be paid to:

- operating conditions
- running behaviour
- running noise
- friction
- temperature
- operational reliability (security against premature failure as a result of wear, fatigue, corrosion or damage due to media introduced from the environment, such as water or sand)
- costs for installation and maintenance of the lubrication system.

In order to achieve high operational reliability, the lubricant supply to the bearings must be free from disruptions and the lubricant must continuously reach all the functional surfaces. An adequate quantity of lubricant is not achieved at all times with all lubrication methods. A reliable supply is facilitated by a monitored and continuous oil feed. Where oil sump lubrication is used and there are high requirements for operational reliability, the oil level must be checked on a regular basis.

Bearings lubricated with grease have sufficient operational reliability if the relubrication intervals or the grease operating life (in the case of bearing arrangements lubricated for life) are not exceeded.

Where lubrication methods with relubrication at short intervals are used, the operational reliability depends on the reliability of the supply devices.

Common lubrication methods are shown in the table Lubrication methods, page 60.

Examples from practice

Lubrication methods are subdivided into methods for individual supply and central supply. Which variant is used is based on the number of lubrication points to be supplied.

Individual supply

If there are only a few lubrication points or they are situated some distance from each other, individual supply should be used in preference. In this case, lubrication with grease is recommended. This can be carried out either manually using a grease gun or by means of automatic MOTION GUARD lubricators. These dispense their fill quantity to the lubrication point over an adjustable time period.

For further information, see TPI 168, Arcanol Rolling Bearing Greases.

Central supply

If lubricant is to be supplied to a large number of lubrication points, possibly with differing lubricant demand, central supply or a central lubrication system is a sensible option. This can be a consumption lubrication system or a recirculating lubrication system.

Consumption lubrication systems

In the case of consumption lubrication systems, the lubricant is fed to the lubrication point once only and the quantities fed are normally small. Consumption lubrication systems are suitable for oils and flowable greases of the NLGI grades 0, 00 or 000 (NLGI: National Lubricating Grease Institute), see section NLGI grade, page 66.

They generally comprise a pump installed in the storage container, the controller (operating on a time-controlled or cycle-controlled basis), the metering valves, the feed lines to the metering valves and the feed lines from the metering points to the lubrication points.

The lubricant demand for each bearing position can be set specifically by means of the metering quantity for each metering valve (5 mm^3 to 1 000 mm^3, in various stages) and the pump pulse.
If greasing is to be carried out using greases with higher consistency of the NLGI grade 1, 2 or 3, special central lubrication systems must be used. These can be so-called twin-line, progressive line or multi-line systems. The lubricant demand for each bearing position can be specifically adjusted by means of appropriate metering elements in these systems as well.

A special type of consumption lubrication system is the pneumatic oil lubrication system, *Figure 1* and *Figure 2*, page 57.

! The conveying capability of the grease must be clarified with the equipment manfacturer.

Principle applied
by Woerner GmbH & Co. KG, Wertheim

1 Oil line
2 Air line
3 Pneumatic oil line
to the lubrication point

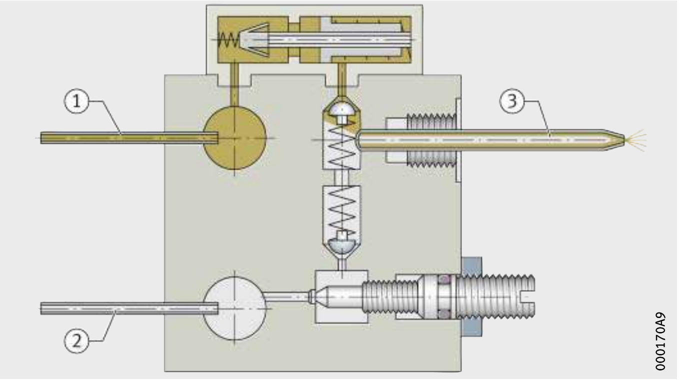


Figure 1
Pneumatic oil lubrication

① Time-controlled oil pump
② Oil line
③ Air line
④ Oil/air mixing unit
⑤ Oil metering unit
⑥ Air metering unit
⑦ Mixing chamber
⑧ Pneumatic oil line to the lubrication point

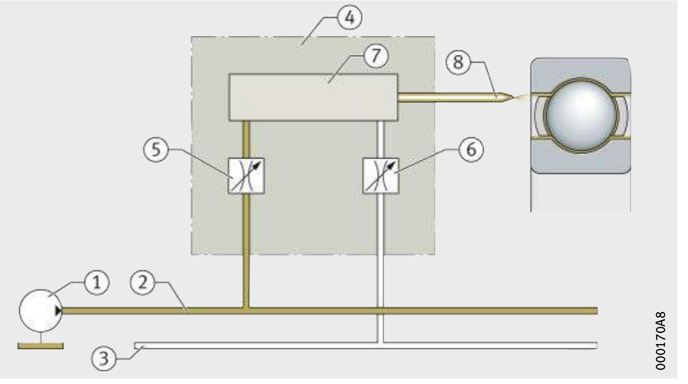


Figure 2
Schematic of pneumatic oil lubrication

On a periodic basis, a metering valve feeds a certain oil quantity into a mixing valve and it is exposed there to a continuous air flow. As a result, the metered oil quantity is very finely distributed on the wall of the lubrication point line and the lubrication point is continuously supplied with very small quantities of lubricant. The frictional torque does not increase during relubrication and splash losses are minimised. The requisite length and diameter of the lubrication point line as well as the air pressure to be selected must be agreed in consultation with the equipment manufacturer. Pneumatic oil lubrication has superseded the oil mist lubrication used previously, since it gives smoother feed and can be set accurately.

Recirculating lubrication systems

In contrast to consumption lubrication systems, recirculating lubrication systems feed the lubricant to the lubrication point several times. It is, however, only used for oil lubrication. A schematic of a recirculating lubrication system is shown in *Figure 3*.

In addition, it is recommended that an oil level monitoring system on the storage container, devices for fitration and cooling of the oil and a manometer are fitted. Depending on the oil viscosity and ambient temperature, heating of the storage container may be necessary.

① Storage container
② Oil pump
③ Pressure control valve
④ Electric oil level monitoring device
⑤ Cooling system
⑥ Thermometer
⑦ Manometer
⑧ Filter
⑨ Metering element
(flow control valve or choke valve)
⑩ Lubrication point
⑪ Oil return lines

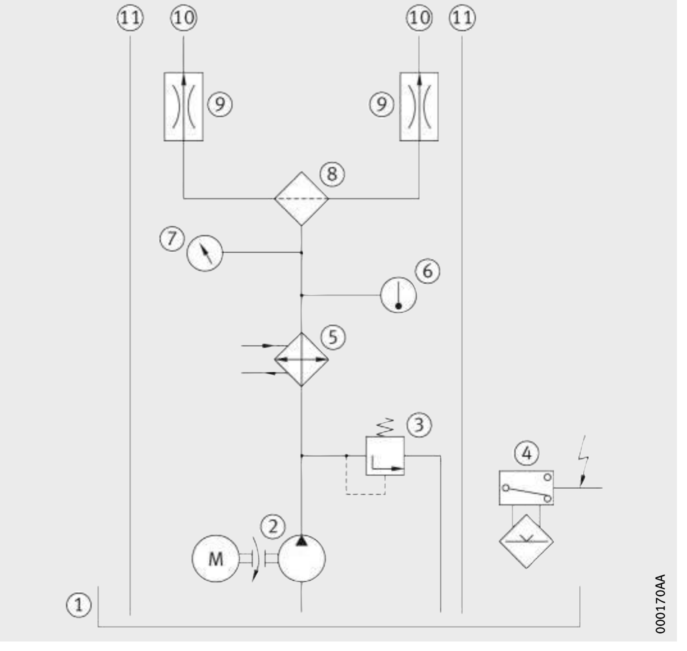


Figure 3
Schematic of a recirculating oil lubrication system

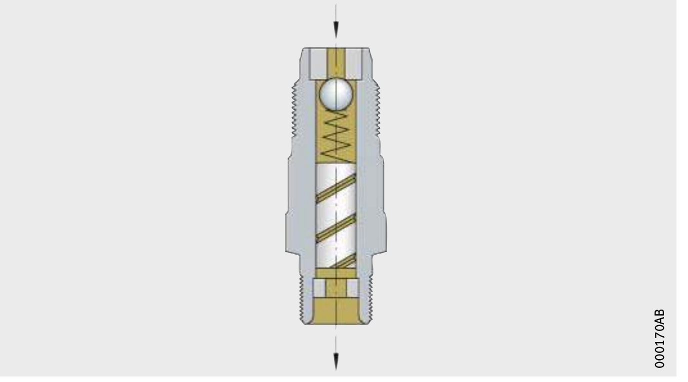


Figure 4
Choke valve

Metering elements can also feed the lubrication points with larger quantities of oil in the range of several litres per minute, *Figure 4*. As a result, heat can be dissipated from the bearing. The maximum oil quantity that can flow through the bearing is dependent on the viscosity and temperature of the oil as well as the inlet and outlet cross-sections. It must always be taken into consideration that larger quantities of oil in bearings and transmissions can lead to splash losses. In cases where the speeds are not very low, these can be considerable and may lead, for example, to increased oil temperatures.

Lubrication methods

Lubricant		Lubrication method	Equipment for the lubrication method
Solid lubricant		Lifetime lubrication	–
Grease		Lifetime lubrication	–
		Relubrication	Manual grease gun Grease pump Automatic relubrication systems
		Spray lubrication	Consumption lubrication system[3)]
Oil	Larger quantities	Oil sump lubrication	Dipstick Standpipe Level monitoring device
		Recirculating oil lubrication	Recirculating lubrication system[3)]
		Oil injection lubrication	Recirculating lubrication system[3)] with injection nozzles[5)]
	Minimal quantities	Oil pulse lubrication, oil drop lubrication	Consumption lubrication system[3)] Oil dropper device Oil spray lubrication system
		Pneumatic oil lubrication	Pneumatic oil lubrication system[7)]

1) Depending on the bearing type and mounting conditions.

2) Depending on the rotational speed and grease type.

3) Recirculating lubrication systems have an oil return system.
Consumption lubrication systems have simultaneously time-controlled metering valves with a small delivery quantity (5 mm^3/stroke – 10 mm^3/stroke).
Further information: see section Consumption lubrication systems, page 56.

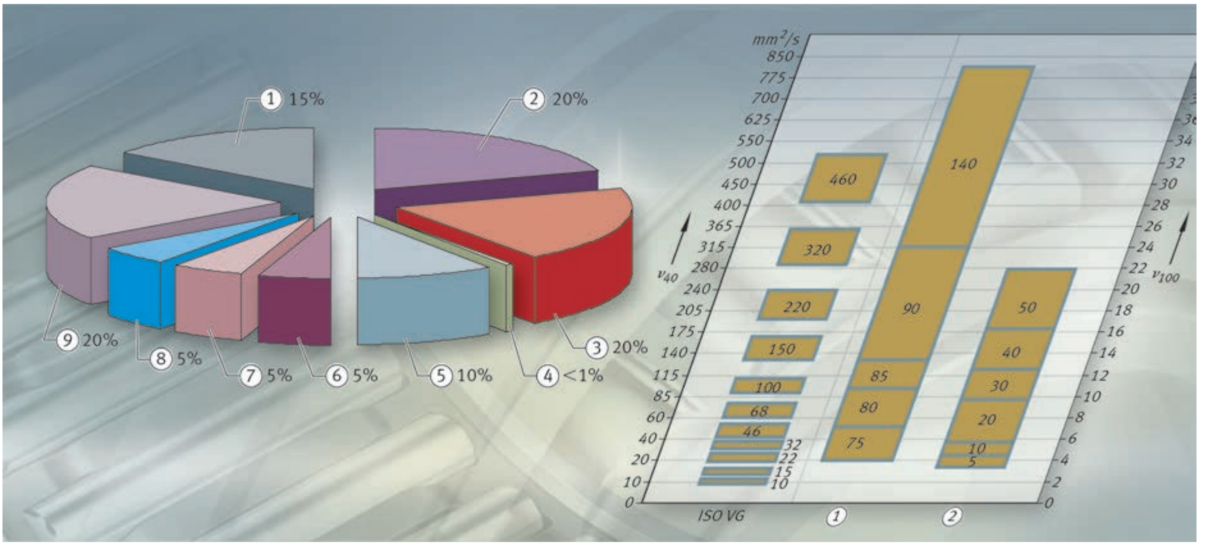
4) Depending on the oil viscosity.

5) Design of the nozzles, *Figure 20*, page 116.

6) Depending on the oil viscosity and oil quantity.

7) A pneumatic oil lubrication system comprises a pump, container, feed lines, volumetric pneumatic oil metering distributor, nozzles, controller and compressed air supply.

Design measures	Achievable speed parameter[1)] $n \cdot d_M$ $min^{-1} \cdot mm$	Suitable bearing types	Operating behaviour
–	≈ 1 500	Predominantly deep groove ball bearings	–
–	≈ $0,5 \cdot 10^6$ For suitable special greases and bearings: ≈ $2,6 \cdot 10^6$	All bearing types[2)]	Special greases have: - low friction - favourable noise behaviour
- Feed holes - Possibly grease quantity regulator - Collection chamber for used grease			
- Feed via pipes or holes - Collection chamber for used grease			
- Housing with sufficient oil volume - Overflow holes - Connector for monitoring devices	≈ $0,5 \cdot 10^6$	All bearing types	In general: - high bearing friction due to splash losses - good cooling action - noise damping[4)] In the case of recirculating oil and oil injection lubrication: - additional removal of wear particles
- Sufficiently large holes for oil inlet and outlet	≈ $1,5 \cdot 10^6$		
- Oil inlet via directed nozzles - Oil outlet via sufficiently large holes	Tested to: ≈ $0,5 \cdot 10^6$		
- Outlet holes	Depending on environmental conditions[1) 6)]: ≈ $2,6 \cdot 10^6$	All bearing types	In general: - noise damping[4)] - friction[6)]
- Possibly outlet holes			



Lubricant selection

Lubricant selection

Lubricant selection

Selection of the correct lubricant is decisive for reliable function of the bearing. Failure statistics show that a significant proportion of premature failures are related directly or indirectly to the lubricant used. The principal issues that should be highlighted in this connection are unsuitable lubricants (20%), aged lubricants (20%) and lubricant starvation (15%), *Figure 1*.
For further information on the subject of contamination, see section Contaminants in the lubricant.

1 Lubricant starvation
2 Unsuitable lubricant
3 Aged lubricant
4 Material and production defects
5 Unsuitable bearing selection
6 Secondary damage
7 Mounting defects
8 Liquid contaminants
9 Solid contaminants

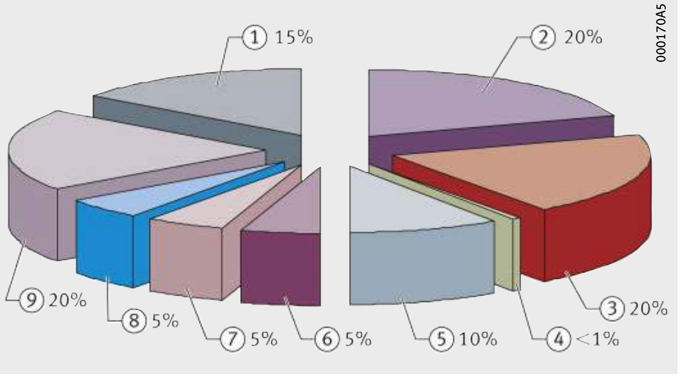


Figure 1
Causes of failure for rolling bearings
Source: Antriebstechnik, *93*

Requirements for lubricants

Manufacturers of rolling bearings recommend lubricants that fulfil the specifications for rolling bearing lubricants. Minimum requirements are stated in standards. If the correct selection is made, they facilitate reliable lubrication for a wide range of speeds and loads. The Schaeffler Group places requirements on the lubricants used that extend beyond the minimum requirements.

Lubricant testing

Lubricants for the mixed friction range operating under high load or with low operating viscosity at high temperature are assessed in accordance with their friction and wear behaviour. Wear can only be prevented here if separating boundary layers are formed in the contact zones. For testing of the lubricants, FE8 test rigs in accordance with DIN 51819 are used.

!

In the case of mineral oils with high levels of additives (hypoid oils, synthetic oils), attention must be paid to their compatibility with the seal and bearing materials, especially with the cage material.

Greases

The optimum bearing operating life can be achieved if suitable lubricants are selected. Account must be taken of application-related influencing factors such as bearing type, speed, temperature and load. In addition, attention must be paid to environmental conditions, the resistance of plastics, legal and enironmental regulations as well as costs.

Specification according to DIN or the design brief

Type K greases standardised in accordance with DIN 51825 should be used in preference. However, this standard only formulates minimum requirements for greases. This means that greases in one DIN class may exhibit differences in quality and may be suitable to varying degrees for the specific application. As a result, rolling bearing manufacturers frequently specify greases by means of design briefs that give a more detailed description of the profile of requirements for the grease.

Characteristics

The characteristics of a grease are fundamentally dependent on:

- the base oil type
- the base oil viscosity (which is responsible for the formation of the lubricant film)
- the thickener (which is relevant to shear strength)
- the additive package.

The thickeners normally used are metal soaps or metal complex soaps. Organic or polymer thickeners such as polycarbamide are increasing in importance.

PTFE is used as a solid lubricant for lubrication in the high temperature range (continuous temperature $>$ +150 °C) or media resistance. Inorganic thickeners such as bentonite are only used to a lesser extent in modern greases.

As a base oil, mineral oils or synthetic oils are used. It is important that synthetic oils are differentiated according to their type (polyalphaolefin, polyglycol, ester, fluoro oil), since these possess very different characteristics.

In addition, greases contain additives. A distinction is made between additives that have an effect on the oil itself (oxidation inhibitors, viscosity index improvers, detergents, dispersants) and additives that have an effect on the bearing or the metal surface (anti-wear additives, corrosion inhibitors, friction value modifiers).

Greases are predominantly classified in terms of their principal components, namely thickener and base oil. An overview of the most important grease types is given in the table Greases, page 84.

Greases are produced in various consistencies. These are defined as NLGI grades, which are determined by means of worked penetration in accordance with ISO 2137. The higher the NLGI grade, the harder the grease. For rolling bearings, greases of NLGI grades 1, 2 and 3 are used in preference.

NLGI grade

Consistency NLGI grade	Penetration 0,1 mm	Consistency
000	445 to 475	Fluid
00	400 to 430	
0	355 to 385	Semi-fluid
1	310 to 340	
2	265 to 295	Soft
3	220 to 250	
4	175 to 205	Firm
5	130 to 160	
6	85 to 115	Hard

The influence of bearing type

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

Bearings with point contact

In ball bearings, each overrolling motion at the rolling contact places strain on only a relatively small volume of grease. In addition, the rolling kinematics of ball bearings exhibit only relatively small proportions of sliding motion. The specific mechanical strain placed on greases in bearings with point contact is therefore significantly less than in bearings with line contact. Typically, greases with a base oil viscosity ISO VG 68 to ISO VG 100 are used.

These should always contain so-called antioxidants (AO). However, this is in any case customary in the case of modern greases.

Bearings with line contact

Roller bearings with line contact place higher requirements on the grease. Not only is a larger grease quantity at the contact subjected to strain, but sliding and rib friction is also to be expected.
This prevents the formation of a lubricant film and would therefore lead to wear. As a countermeasure, greases for bearings with line contact exhibit higher base oil viscosity (ISO VG 150 to 460 and, in special cases, even higher). In addition, anti-wear additives (EP) are recommended. The consistency is normally NLGI 2.

The influence of speed

As in the case of rolling bearings, greases have a maximum speed parameter $n \cdot d_M$. The speed parameter of the bearing should always be a good match for the speed parameter of the grease, *Figure 2*.

In the case of a grease, this is dependent on the type and proportion of the thickener, the base oil type and the proportion of base oil. These data can be found in the technical data sheets for the greases. Typically, greases for high speeds have a low base oil viscosity and are based on ester oil. They are also suitable for low temperatures.

Greases for low speeds have a higher base oil viscosity and are frequently used as heavy duty greases. The speed parameter of a grease is not a material parameter but is dependent on the bearing type and the required minimum running time.

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

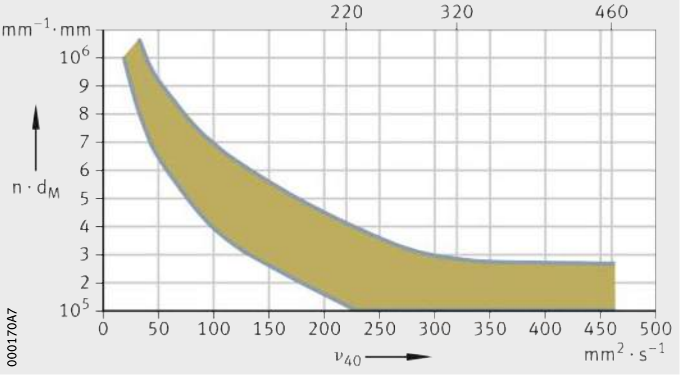


Figure 2
Speed parameter for greases

For selection of the suitable grease, an initial guide can be given as follows:

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

Lubricant selection

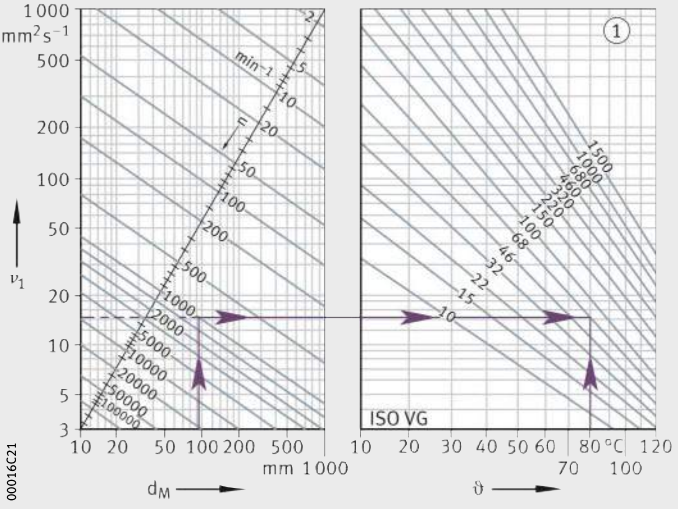
Base oil viscosity

In addition to the speed, the base oil viscosity has a direct influence on formation of the lubricant film. In normal cases, the base oil viscosity should therefore be selected such that good lubrication conditions are present under the operating regime ($\kappa > 1$).
The required operating viscosity or corresponding ISO VG grade can be determined in approximate terms from the diagram.
The input values are not only the speed and the mean bearing diameter but also the temperature, since this has a significant influence on the viscosity, *Figure 3*.

In the example below, the base oil viscosity is determined for a bearing with the following values:

- mean bearing diameter d_M = 100 mm
- speed n = 1000 min^{-1}
- operating temperature ϑ = 80 °C.

This gives a viscosity ratio of κ = 1. The result is a minimum required viscosity of ISO VG 68.



ν_1 = reference viscosity
d_M = mean bearing diameter
ϑ = operating temperature
n = operating speed

① Viscosity $mm^2 \cdot s^{-1}$ at +40 °C

Figure 3
V/T diagram for mineral oils

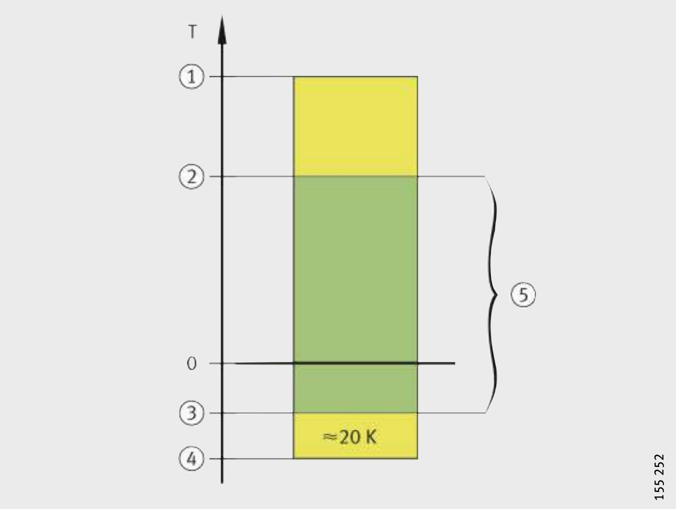
The influence of temperature

The temperature range of the grease must correspond to the range of possible operating temperatures in the rolling bearing. Grease manufacturers state this range for type K rolling bearing greases in accordance with DIN 51825.

The operating temperature range is dependent on the thickener type, the proportion of thickener, the base oil type, the proportion of base oil, the production quality and the production process.

The stability at high temperature is dependent principally on the production quality and the production process.

It is generally recommended that greases should be used in accordance with the bearing temperature normally occurring in the standard operating range, in order to achieve reliable lubrication and an acceptable grease operating life, *Figure 4*.



T = operating temperature

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

Figure 4
Operating temperature range

Upper operating temperature

The upper operating temperature of a type K grease is determined using a rolling bearing test rig FE9 in accordance with DIN 51821. At the upper operating temperature, a 50% failure probability rate (B_{50} or F_{50}) of at least 100 hours must be achieved in this test.

This shows clearly that a grease should not be used for an extended period at its upper operating temperature, since the grease operating life is then relatively short.

Dropping point

Data sheets for greases also state the dropping point, which is determined in accordance with ISO 2176. The dropping point is defined as the temperature at which slowly heated grease passes from a semi-solid to a liquid state and the first drop of grease falls from the standardised dropping point nipple. The dropping point is fundamentally dependent on the type of thickener and less so on the base oil. When this temperature is reached, the structure of the thickener undergoes an irreversible change. Grease that has softened beyond its dropping point, will not regain its original performance capability once it has undergone subsequent cooling. Greases in rolling bearings should therefore always be operated at temperatures significantly below their dropping point. The upper operating temperature of a lithium soap grease with a mineral oil base is therefore approx. 50 K below its dropping point.

Due to the thickener structure of PTFE, bentonite and gel greases, they do not have a dropping point. The upper continuous limit temperature $T_{upperlimit}$ must not be exceeded if a reduction in the grease operating life due to temperature is to be avoided.

Lower operating temperature

The lower operating temperature of a type K grease is defined by means of the flow pressure in accordance with DIN 51805. This is the pressure that is required in order to press a stream of grease through a defined nozzle.

This makes it possible to state whether the grease can still be moved at the low temperature. This is important, for example, in the case of central lubrication systems. For type K greases, the flow pressure at the lower operating temperature must be less than 1400 mbar.

The flow pressure cannot be used, however, to derive any statement about suitability at low temperatures in rolling bearings. In addition to the lower operating temperature of a grease, therefore, the low temperature frictional torque is also determined in accordance with ASTM D 1478 or IP 186/93. In this case, the friction behaviour of a greased ball bearing at low temperature is tested. At the lower operating temperature, the starting torque must not exceed 1 000 Nmm and the running torque must not exceed 100 Nmm.
The grease is not damaged by low temperatures but it does become stiff. This becomes apparent through an increase in starting torque, but also leads to slippage and thus to greater noise and wear.
The grease undergoes relatively rapid heating through the churning work and, as a result, becomes capable of lubrication again as soon as the low temperature is not present isothermally, such as in cold store applications. At low temperatures, greases release very little base oil. This can result in inadequate supply to the rolling contact and thus to mixed or boundary friction. It is therefore recommended that greases are not used below the lower continuous limit temperature $T_{lowerlimit}$. The upper continuous limit temperature $T_{upperlimit}$ must not be exceeded if a reduction in the grease operating life due to temperature is to be avoided. These data can be found in the table Greases, page 126 and the table Arcanol rolling bearing greases, page 128.

Greases for the low temperature range

Greases for the low temperature range (below –20 °C) should exhibit a lower operating temperature that is sufficiently low. The guide value can be taken as: at least 20 K lower than the expected ambient temperature, *Figure 4*, page 69.

In many cases, greases with a low base oil viscosity (ISO VG 10 to ISO VG 32) are used. As a base oil, consideration can be given to polyalphaolefin or diester oil, frequently in conjunction with a lithium soap thickener. Before use, however, the resistance of plastic materials must be tested.

Greases for the high temperature range

In order to prevent a reduction in the grease operating life as a result of high temperatures (> +140 °C), the bearing temperature during operation must be continuously below the upper continuous limit temperature $T_{upperlimit}$, *Figure 4*, page 69.

If this is not known, it is recommended that the grease used should have an upper operating temperature at least 20 K higher than the bearing temperature. In the high temperature range, greases based on synthetic oils should be used, since these have higher thermal resistance compared to greases based on mineral oil. Ester oils are used predominantly in this case. At continuous temperatures above +150 °C, alkoxyfluoro oils offer the longest life. If these oils or greases are used, all the components must be absolutely free from hydrocarbons, in order to prevent reactions due to incompatibility. This also has an influence on the preservative to be used with the bearings.

Lubricants based on alkoxyfluoro oils are inferior to other standard lubricants, however, in terms of their lubrication effect and their anti-wear protection in the normal temperature range.

The speed stability of such greases is also lower (normally $n \cdot d_M < 350\,000\ \text{min}^{-1} \cdot \text{mm}$). The base oil viscosity of high temperature greases is normally above ISO VG 100 in order to keep vapourisation losses to a low level. Typical thickener types for the high temperature range are polycarbamide (normally in conjunction with easter oils) and PTFE (a a very high temperature grease in conjunction with alkoxyfluoro oil).

! Before these special greases are used, the resistance of non-ferrous and light metals as well as of plastic materials must be tested where these may come into contact with the lubricant. If the bearing will be continuously relubricated, for example by means of a central lubrication system or individual metering units, normal greases can be used in the high temperature range. The grease operating life is, however, correspondingly shorter at these high temperatures and this must be compensated by means of short relubrication intervals. Greases must be selected that will not undergo hardening or caking during their dwell time in the bearing. This would hinder the exchange of grease and lead in extreme cases to jamming of the bearing. A sufficiently large space must also be provided to accommodate used grease when it is pressed out.

The influence of load

For a load ratio $C/P < 10$ or $P/C > 0,1$, greases are recommended that have higher base oil viscosity and in particular anti-wear additives (EP). These additives form a reaction layer on the metal surface that gives protection against wear. Such greases are identified in accordance with DIN 51825 by KP. Their use is also recommended for bearings with an increased proportion of sliding motion (including slow running) or line contact as well as under combined loads (radial, axial). Greases with solid lubricants such as PTFE or molybdenum disulphide should be used in preference for applications in the boundary or mixed friction range (chemical lubrication). The solid lubricant particle size must not exceed 5 μm. Silicone lubricants have a low load carrying capacity that cannot be compensated by an appropriate additive package and may therefore be used only under very low loads $P \leqq 3\%\ C$.

The influence of water and moisture

Moisture can enter the bearing from outside if the application is operated in a damp environment, for example outdoors. Water may condense within the bearing if there are rapid temperature changes between warm and cold. This may occur in particular if there are large cavities in the bearing or housing. Water can cause severe damage to the grease or bearing. This is due to ageing or hydrolysis, interruption of the lubricant film and not least corrosion. Barium and calcium complex soap greases have proved favourable here since they have good water resistance or act to repel water. The anti-corrosion effect of a grease is also influenced by additives. This is tested using the SKF Emcor method in accordance with ISO 11007 or DIN 51805. Type K greases in accordance with DIN 51825 must exhibit a corrosion rating < 1. For further information, see section Liquid contaminants, page 143.

The influence of oscillations, shocks and vibrations

Oscillation loads can have a considerable effect on the structure of thickeners in greases. If mechanical stability is not sufficient, changes in consistency may occur. This leads to softening, deoiling on an isolated basis but also hardening of the grease with a corresponding reduction in lubrication capability. It is therefore recommended that a grease should be selected whose mechanical stability has been tested accordingly. The options here are the expanded worked penetration, the Shell Roller Test in accordance with ASTM D 1831 and a test run on the FAG AN42 test rig.

Under shock-type strain or very high load, it is advantageous to use greases of consistency grade NLGI 1 to NLGI 2 with a high base oil viscosity (ISO VG 460 to ISO VG 1 500). Due to their high base oil viscosity, these greases form a comparatively thick, elastohydrodynamic lubricant film that gives damping of shocks. However, the disadvantage of greases with a high base oil viscosity is that, due to the low oil release rate, it must be ensured that the lubricant is present to an effective extent at the contact by a high fill level or shorter term relubrication. If very small swivel angles and vibrations are present, there is a danger of so-called false brinelling. In order to counteract this wear mode, which has not so far been fully researched, the use of special lubricants and in special cases also coatings has proved to be advantageous. The decisive factor here is the correct combination of the base oil and thickener type, base oil viscosity, consistency, additive package and, as appropriate, solid lubricants. Effectiveness can only be checked by means of appropriate tests. In addition, more frequent relubrication is advisable.

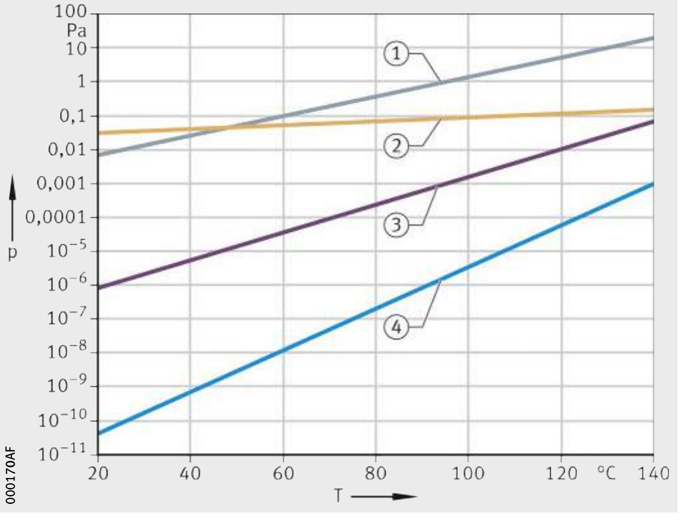
The influence of nuclear radiation

If a grease is exposed to nuclear radiation (for example in nuclear power stations, in certain non-destructive testing methods or in the medical sector), the grease must exhibit appropriate resistance to radiation. The decisive factor here is the amount of energy to which the grease is exposed, in other words the radiation dose. This is stated in J/kg or Gray. It is immaterial in this case whether the radiation is present at low intensity over a long period or high intensity over a short period. A grease is classified as resistant to radiation if it can withstand a larger amount of energy than will act over its lifetime. The consequences of a radiation dose can include not only accelerated ageing but also gas emission as well as changes in the base oil viscosity, consistency and dropping point.

Standard metal soap greases exhibit a radiation resistance of approx. 500 000 J/kg to 3 000 000 J/kg. Based on current knowledge, greases with a high radiation resistance are aromatic polycarbamide greases with a polyphenylether oil base, exhibiting a value of up to 5 000 000 J/kg. In the disposal of such lubricants, attention must be paid to the fact that the greases may have become radioactive. The influence of radiation on the grease operating life cannot be quantified on the basis of current technology.

The influence of vacuum

In vacuum applications, there is a danger of vapourisation of lubricant components. This is dependent on the negative pressure and the temperature. This not only reduces the performance capability of the lubricant but also impairs the vacuum. In addition to the base oil type, the viscosity also has an influence on the vapour pressure behaviour. Whether an oil or grease is suitable can be determined, as a function of vacuum and temperature, using the vapour pressure curve of the base oil, *Figure 5*.



p = vapour pressure
T = temperature

① Mineral oil (ISO VG 68)
② PAO (ISO VG 46)
③ Ester oil (ISO VG 100)
④ PFPE (ISO VG 460)

Figure 5
Vapour pressures curves of various oils

If it is not possible to use oils and greases due to the combination of negative pressure and temperature, solid lubricants such as PTFE or MoS_2 can be used. Graphite is not suitable for vacuum applications. In addition to the lubricants, the suitability of plastics and elastomers must be tested in the case of vacuum applications.

The influence of seals

If hard contaminant particles penetrate the bearing, this will lead not only to increased noise but also to wear. This should be prevented by appropriate sealing of the bearing. The grease can assist this sealing effect by forming a stable collar on the seal. More solid greases are most suitable in this case. Greases that are too soft will tend to favour the escape of grease. In addition, a so-called barrier grease with high base oil viscosity and consistency can be used for sealing. Special seals are available with a reservoir that is filled with such a grease. For further information, see section Solid foreign matter, page 138.

The influence of mounting position and adjacent components

Even where an axis of rotation is vertical or inclined, lubricant must remain at the lubrication point. In addition to appropriate seals, flowing away of the grease can be prevented by using a more viscous grease. If several lubrication points are located close together, unintentional contact can occur. Attention must therefore be paid to compatibility of the lubricants with each other. Where possible, the optimum solution is to use only one grease. It must be ensured that the lubricant is compatible with the cage and seal material. Particular attention must be paid to the use of greases based on synthetic oils. Incompatibility can lead to embrittlement or swelling, with eventual destruction of the plastic. Meaningful results can be obtained through the use of appropriate storage tests.

The influence of legal and environmental regulations

In the foodstuffs sector, the use of greases with appropriate authorisation is specified. A worldwide standard is approval in accordance with the NSF (National Sanitary Foundation) H1 or H2, listed in the so-called White Book™.

A lubricant with the code H1 (Food Grade Lubricant) may be used where occasional, technically unavoidable contact with foodstuffs cannot be eliminated. This means that the grease must be non-toxic, rapidly broken down by the organism and neutral in terms of both odour and taste. Such lubricants frequently comprise aluminium complex soap thickeners and polyalphaolefins or medicinal white oils as a base oil. Recently, authorisation to H1 has been granted to greases with other thickener types such as PTFE or calcium sulphonate complex soaps. H2 lubricants are intended for general use within the foodstuffs industry where no foodstuff contact occurs. Furthermore, there are lubricants that fulfil individual religious rules, such as Jewish (kosher) or Islamic (halal).

Greases with biological degradability must be provided where the lubricant can pass directly into the environment, see section Lubricants with biological degradability, page 88.
Greases must conform to the appropriate legal regulations relating to prohibited substances.

Lubricating oils

For the lubrication of rolling bearings, mineral oils and synthetic oils are essentially suitable, see table. Oils with a mineral oil base are used most frequently. These mineral oils must fulfil at least the requirements according to DIN 51517 (lubricating oils).

Special oils, which are often synthetic oils, are used where extreme operating conditions are present. The resistance of the oil is subjected to particular requirements under challenging conditions involving, for example, temperature or radiation. The effectiveness of additives in rolling bearings has been demonstrated by well-known oil manufacturers, see table, page 78. For example, anti-wear additives are particularly important for the operation of rolling bearings in the mixed friction range.

Base oils and their typical characteristics

Base oil Abbreviation	Operating temperature		Viscosity/ temperature index	Compatibility with elastomers	Notable feature	Price ratio
	Upper °C	Lower °C				
Mineral oil Min	+120	–20	100	Good	Most frequently used base oil type, “naturally uncontaminated” due to origin as a product of nature	1
Polyalphaolefin PAO, SHC	+150	–40	160	Good	Widely used synthetic oil type, including use for lubricants with foodstuff approval	6
Polyglycol PG	+150	–40	220	Moderate	Critical in aluminium contacts, normally not miscible with mineral oil, PAO, ester	4 to 10
Ester E	+180	–60	180	Moderate to poor	Also suitable as mixture with PAO and mineral oil, in some cases with good biological degradability	4 to 10
Silicone oil Si	+200	–60	500	Very good	Steel/steel contacts tend to undergo fretting, extremely low surface tension, “spreading”; LABS[1)]	40 to 100
Alkoxyfluoro oil PFAE, PFPE	+250	–30	160	Good	Rolling bearings must be free from hydrocarbons, not miscible with other oils	200 to 800

1) LABS: Substance impairing wetting by coating.

Lubricant additives and their effect

Additive type		Function
Extreme pressure additives	EP	Improved pressure absorption behaviour Reduction in wear through formation of reaction layer
Friction modifiers	FM	Modified friction under mixed and boundary friction
Anti-wear protection	AW	Reduction in mild adhesive/abrasive wear under mixed friction
Corrosion inhibitors	KI	Protection of metal surfaces against corrosion
Ageing inhibitors	OI	Delay in oxidation breakdown of lubricant
Adhesion additives		Improved adhesion of lubricant to surface
Detergents and dispersants		Improved contaminant separation and transport behaviour of lubricant
VI improvers		Improved (reduced) viscosity/temperature interdependence
Foam inhibitors		Prevention of stable foam formation
Pourpoint reducers		Reduced solidification point

Recommended oil viscosity

The achievable life and security against wear increase with increasing separation of the contact surfaces by a lubricant film. Since the lubricant film thickness increases with oil viscosity, an oil with a higher operating viscosity ν should be selected where possible. A very long life can be achieved with a viscosity ratio $\kappa = \nu/\nu_1 = 2$ to 4. With increasing viscosity, however, the lubricant friction increases. Problems may occur with feed and removal of oil at low and even at normal temperatures.

! The oil selected must be sufficiently viscous that, on the one hand, the longest possible fatigue life is achieved but, on the other hand, the power loss due to increased friction is kept as low as possible. It must be ensured that the bearings are provided with sufficient oil at all times.

Operating viscosity

In individual cases, the preferred level of operating viscosity cannot be achieved because:

- The oil selection is determined by other components in the machine, which require a thin-bodied oil
- A sufficiently flowable oil is to be used for recirculating lubrication in order to dissipate contaminants and heat from the bearing
- Higher temperatures or very low circumferential viscosity are present at some times and the operating viscosity that can be achieved with the most viscous suitable oil is below the required viscosity.

In such cases, an oil with lower than recommended viscosity may be used. The oil must then, however, contain effective additives and its suitability for lubrication must be demonstrated by means of a rolling bearing test. Depending on the deviation from the nominal value, a reduction in fatigue life and the symptoms of wear on the functional surfaces must then be anticipated, as will be demonstrated by the calculation of the achievable life.

Common viscosity classes in accordance with ISO and SAE, *Figure 6*.

Viscosity grades in accordance with ISO and SAE

ν_{40} = viscosity at +40 °C
ν_{100} = viscosity at +100 °C

① Transmission oil to SAE classification
② Engine oil to SAE classification

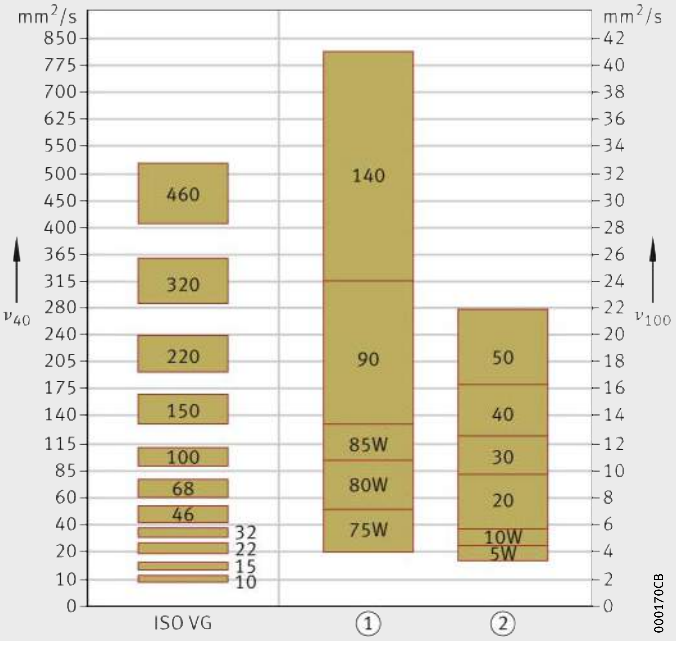


Figure 6
Viscosity grades

Lubricant selection

Viscosity grades ISO VG

Viscosity grade ISO VG	Midpoint viscosity mm^2/s	Limits of kinematic viscosity at 40 °C mm^2/s	
		min.	max.
2	2,2	1,98	2,42
3	3,2	2,88	3,52
5	4,6	4,14	5,06
7	6,8	6,12	7,48
10	10	9,00	11
15	15	13,5	16,5
22	22	19,8	24,2
32	32	28,8	35,2
46	46	41,4	50,6
68	68	61,2	74,8
100	100	90	110
150	150	135	165
220	220	198	242
320	320	288	352
460	460	414	506
680	680	612	748
1 000	1 000	900	1 100
1 500	1 500	1 350	1 650
2 200	2 200	1 980	2 420
3 200	3 200	2 880	3 520

Selection of oil in accordance with operating conditions

In order to select the correct oil for the specific application, the operating conditions must first be analysed precisely.

Normal operating conditions

Under normal operating conditions (atmospheric pressure, temperature max. +100 °C in oil sump lubrication and +150 °C in recirculating oil lubrication, load ratio $C/P > 10$, speed up to permissible speed), undoped oils but preferably oils with inhibitors (anti-corrosion and anti-ageing, code L in accordance with DIN 51502) can be used. If the stated viscosity recommendations cannot be fulfilled, oils with effective anti-wear additives must be provided.

High speed parameters

An oxidation-resistant oil with low foaming tendency and a favourable viscosity/temperature behaviour (V/T behaviour) is advantageous if high circumferential speeds are present ($k_f \cdot n \cdot d_M > 500\,000 \text{ min}^{-1} \cdot \text{mm}$). Suitable synthetic oils with good V/T behaviour include esters and polyalphaolefins (PAO), since the viscosity of these oils decreases less sharply with increasing temperature. In the starup phase, when the temperature is normally low, this avoids high splashing friction and thus heating; at the higher equilibrium temperature, there is still sufficient viscosity present to ensure lubrication.

High loads

If the bearings are subjected to high loads ($C/P < 10$) or the operating viscosity ν is lower than the reference viscosity ν_1, oils with anti-wear additives should be used (code P in accordance with DIN 51502).

Anti-wear additives reduce the harmful effects of metallic contact occurring at various points. The suitability of anti-wear additives varies and is normally heavily dependent on temperature. Their effectiveness can only be assessed by means of testing in the rolling bearing (for example test rig FE 8).

High temperature

In the case of oils for high operating temperatures, not only the operating temperature limit but also the viscosity/temperature behaviour is particularly important. This behaviour can be assessed using so-called V/T diagrams, which are made available individually by the lubricant manufacturers. Selection should be made on the basis of the oil characteristics.

Lubricant selection

Selection of oil in accordance with oil characteristics

The different types of oil have particular characteristics. On the basis of these characteristics, the most suitable oil can be selected.

Mineral oils

Mineral oils can only be used up to +120 °C. Depending on the temperature and dwell time in the hot range, ageing products are formed that impair the lubrication effect and are deposited as solid residues (oil carbon) in the bearing or vicinity of the bearing.

Esters (diesters and sterically hindered esters)

Esters are thermally stable (–60 °C to +180 °C), have a favourable V/T behaviour, show low volatility and are therefore highly suitable for use at high speed parameters and high temperature. They are normally miscible with mineral oils. In the presence of water, esters undergo various reactions depending on their type. Some types saponify and break down into their components, which is mainly the case if they contain alkaline additives.

Poly (alkylene) glycols

Poly(alkylene) glycols have a favourable V/T behaviour and are suitable for use at high and low temperatures (–40 °C to +150 °C). They are for the most part not water-soluble and cannot be mixed at all with mineral oils, have a lower pressure/viscosity coefficient than other oils and can attack seals and coatings in the housing as well as cages, such as those made from aluminium. Due to their high oxidation resistance, it is possible to increase the oil change intervals in high temperature operation to between 2 and 5 times the intervals normally used with mineral oil.

Polyalphaolefins (PAO)

Polyalphaolefins are synthetically produced hydrocarbon compounds (also known as SHC for Synthetic Hydro Carbon). They have a favourable V/T behaviour, can be used over a wide temperature range (–50 °C to +150 °C) and have good oxidation resistance, which means they can achieve a lifetime several times longer compared to similarly viscous mineral oils under the same conditions. They are also miscible in any ratio with mineral oils.

Silicone oils (phenyl methyl siloxane)

Silicone oils can be used at extreme temperatures (–60° to +200 °C), have a favourable V/T behaviour, low volatility and high thermal stability. They have low load carrying capacity ($C/P \geqq 30$) and low anti-wear capability.

Alkoxyfluoro oils

Alkoxy fluoro oils are resistant to oxidation and water, are very expensive in comparison with mineral oil products, have a higher pressure/viscosity coefficient and a higher density than mineral oils of the same viscosity. Their operating temperature range extends from –30 °C to +250 °C.

When changing to another oil type, attention must be paid to compatibility, see section Miscibility of lubricants, page 130. In general, a change to an oil with lower performance capability should not be made.

Fire resistant hydraulic fluids

Fire resistant hydraulic fluids occupy a special category. For reasons of safety, they have been used for many years in underground workings in mining, on ships, in aircraft and industrial plant with a fire risk. The reasons for their increasing include fire safety, availability and price.

Fire resistant hydraulic fluids must fulfil defined requirements in relation to fire resistance, occupational hygiene and ecological compatibility. The various fluid groups are defined in the 7th Luxembourg Report, see table Fire resistant hydraulic fluids, page 86.

Compatibility with elastomers

Where fire resistant pressure fluids act on hose or seal materials, physical or chemical interactions may occur. These can lead to changes in the volume, strength or elasticity characteristics of plastics (cages, sealing shields) and elastomers. If manufacturers' data are not available, resistance investigations should be carried out before they are used. The test methods and criteria described in the 6th and 7th Luxembourg Report or in CETOP RP 81 H are to be taken as authoritative here. In these tests, defined test speciments are stored in the fluid to be tested for 168 hours at temperatures from +60 °C to +100 °C.

Application examples

The fluid types HFA-E and HFA-S with up to 99 vol. % water are predominantly used in chemical plant, hydraulic presses and in hydraulic mine face supports.

Fluids of type HFC with up to 45 vol. % water are normally used in processing machines such as hydraulic loaders, hammer drills and printing machinery.

The synthetic HFD fluids are used in stage loaders, cableway machines, belt conveyors, hydrostatic couplings, pumps and in printing machinery.

Lubricant selection

Greases

Type of grease			Characteristics1)	
Thickener		Base oil	Temperature range	Dropping point
Type	Soap		°C	°C
Normal	Lithium	Mineral oil	–35 to +130	+170 to +200
		PAO	–60 to +150	+170 to +200
		Ester	–60 to +130	+190
Complex	Aluminium	Mineral oil	–30 to +160	+260
	Barium		–30 to +140	+220
	Calcium		–30 to +140	+240
	Lithium		–30 to +150	+240
	Aluminium	PAO	–60 to +160	+260
	Barium		–40 to +140	+220
	Calcium		–60 to +160	+240
	Lithium		–40 to +180	+240
	Barium	Ester	–40 to +130	+200
	Calcium		–40 to +130	+200
	Lithium		–40 to +180	+240
		Silicone oil	–40 to +180	+240
Bentonite	–	Mineral oil	–20 to +150	–
		PAO	–50 to +180	–
Polycarbamide	–	Mineral oil	–25 to +160	+250
		PAO	–30 to +170	+250
		Ester	–40 to +180	+250
PTFE	–	Alkoxyfluoro oil	–50 to +250	–

Definition of the symbols:
+++ Very good
++ Good
+ Moderate
– Poor.

1) The data represent mean values.

Water resistance	Pressure resistance	Price ratio	Suitability for rolling bearings	Special notes
+++	+	1	+++	Multi-purpose grease
+++	++	4 to 10	+++	For lower and higher temperatures For high speeds
++	+	5 to 6	+++	For low temperatures For high speeds
+++	+	2,5 to 4	+	Multi-purpose grease
++	++	4 to 5	+++	Multi-purpose grease Resistant to vapour
++	++	0,9 to 1,2	+++	Multi-purpose grease Hardening tendency
++	++	2	+++	Multi-purpose grease
+++	++	10 to 15	+	Wide temperature range Easy to move
+++	+++	15 to 20	+++	High speed
+++	+++	15 to 20	+++	For lower and higher temperatures Suitable for high speeds
++	+++	15	+++	Wide temperature range
++	++	7	+++	For higher speeds
+++	++	7	+++	For moderate load
++	+	10	+++	Particularly wide temperature range
++	–	20	++	For low loads only
+++	+	2 to 6	+	For higher temperatures at low speeds
+++	+	12 to 15	+	Wide temperature range
+++	++	3	+++	For higher temperatures at moderate speeds
+++	+++	10	+++	High temperature grease Good long term action
+++	++	10	+++	For high and low temperatures
+++	++	100 to 150	+++	For very high and low temperatures

Fire resistant hydraulic fluids

Fluid group	Composition of fluid
HFA-E	Oil-in-water emulsion with an emulsion oil content of max. 20 vol. %, normal content 1 vol. % to 5 vol. %
HFA-S	Fluid concentrates dissolved in water, normal content not more than 10 vol. %. **Microemulsions** are HFA fluids that are produced from a concentrate by mixing with water. The finely dispersed concentrate droplets have a diameter between 2 μm and 25 μm. The so-called **water thickeners**, also described as thickened HFA fluids, are highly viscous polymer solutions whose long molecular chains form a mechanical matrix that prevents free movement of the water molecules. As a result, such fluids have viscosity at operating temperature within the range of mineral oil. The difficulty with these products at present lies in achieving sufficient anti-wear protection together with high shear stability, since the additives that promote these characteristics have a reciprocal negative effect.
HFB	Water-in-oil emulsion, in general with 40 vol. % water; not used in Germany, used almost exclusively in British mining operations
HFC	Aqueous polymer solution (polyglycols) with approx. 40 vol. % water
HFD	HFD fluids are synthetic, water-free pressure fluids that most closely resemble the tribological behaviour of mineral oil. Due to their major ecological disadvantages, they are only used in individual power drives, where an increased level of protection of maintenance and operating personnel as well as measures against fluid loss must be provided.
HFD-R	Based on phosphoric acid esters
HFD-S	Based on chlorinated hydrocarbons
HFD-T	Based on mixture of phosphoric acid esters and chlorinated hydrocarbons
HFD-U	Based on other compounds

ISO VG grade	Normal operating temperature range °C	Fire resistance	Density T = 15 °C g/cm^3	Standards and specifications
Not defined	+5 to +55	Very good	Approx. 1	DIN 24320
32 46 68 100	+5 to +60	Good	0,92 to 1,05	VDMA 24317
15 22 32 46 68 100	–20 to +60	Very good	1,04 to 1,09	
15 22 32 46 68 100	–20 to +150	Good	1,1 to 1,45	

Special applications

Lubricants with biological degradability

Lubricants with biological degradability must be used in preference or specified where the lubricant can pass directly into the environment. This is the case, for example, in agricultural equipment applications, railway switch points, in treatment plants or sluices with loss lubrication. However, there are no universally recognised directives here and regulations and laws are often only valid locally.

Biological degradability of oils is increasingly determined in accordance with OECD 301 A-F. For type K greases in accordance with DIN 51821, CEC-L-33-T-82 or the so-called Zahn-Wellen test is applied. In this case, the lubricant must exhibit biological degradation of at least 60% (OECD) or 80% (CEC) after 21 days. In addition, an environmentally friendly lubricant must conform in terms of component toxicity to the German Chemicals Act and the water pollution hazard class (WGK). This information can be found in the safety data sheet of the lubricant.

Restricted temperature range

Lubricants based on vegetable oils and mixtures of such oils with inter-esterified products are only suitable for a restricted temperature range and low to normal loads for simple subassemblies with relubrication. If central lubrication systems are used, their suitability must be checked.

Greases based on synthetic oils, predominantly ester oils, have in contrast a performance capacity of around the same level as normal greases. Attention must be paid to the resistance of any plastics used.

Ceramic and hybrid bearings

In comparison with steel, ceramics have a lower specific mass, higher heat resistance, lower thermal expansion, good chemical resistance, higher rigidity, high specific heat capacity, lower thermal conductivity and are also antimagnetic and electrically insulating.

The ceramic material used in rolling bearings is hot pressed silicon nitride ($Si3N4$). This is used predominantly for the rolling elements. The associated rings are made, depending on the application, from various steels. In combination with special steels, such rolling bearings have particular resistance to corrosion, achieve increased fatigue life under unfavourable lubrication conditions and are significantly less sensitive to wear if there is inadequate separation of the contact surfaces.

Lower strain on lubricant

Under identical loads, rolling bearings with ceramic rolling elements have higher contact pressures but significantly smaller contact surfaces than in conventional rolling bearings. These are significantly easier to supply with lubricant and, due to the smaller deformation, have significantly smaller proportions of sliding motion. Despite the higher pressure, the strain placed on the lubricant is thus lower and, especially in the case of grease lubrication, a significant increase in operating life is achieved. The smaller contact surfaces and the more favourable sliding characteristics lead to lower friction and, as a result, to lower temperatures. The preferred areas of use are rolling bearings at very high speeds in a wide temperature range as well as rolling bearings that must achieve very long running times with grease lubrication, and bearings under other extreme operating conditions.

Depending on the application, various lubricants are required for the lubrication of such hybrid bearings (steel rings and ceramic rolling elements). Hybrid bearings used in aircraft engines are lubricated using approved ester oils.

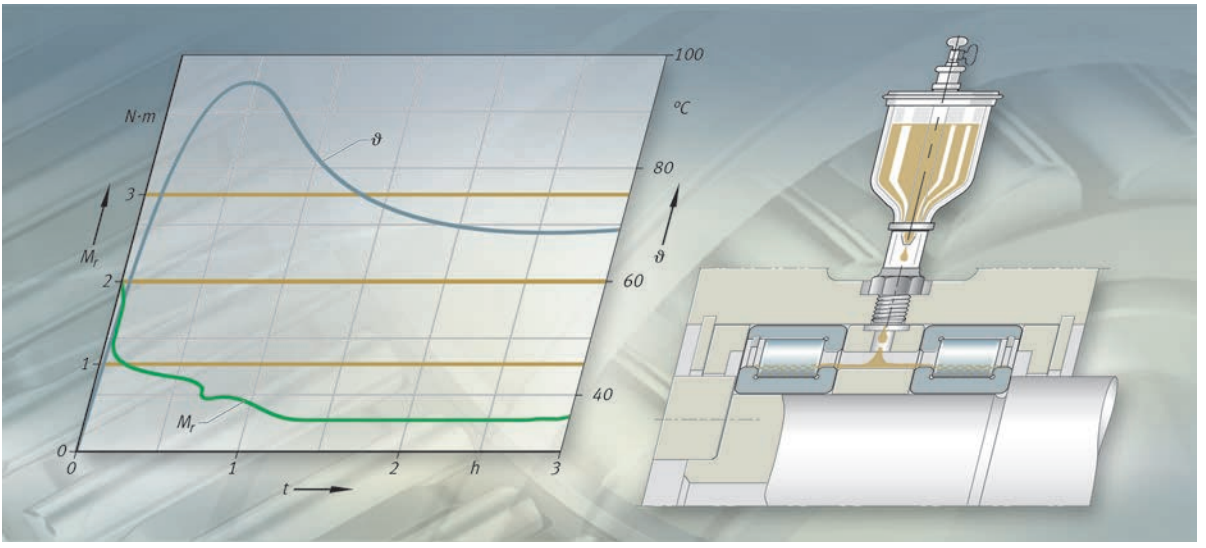
Hybrid spindle bearings

Hybrid spindle bearings are also used very widely and are made available in some cases with seals and greasing.
For grease-lubricated spindle bearings, greases with ester base oils and special additives are used. In general, strongly polar oils with additives matched to the steel/ceramic material pair are necessary.

In tests with spindle bearings, the use of hybrid spindle bearings has made it possible to achieve a lubrication interval or grease operating life that is longer by a factor of 2 to 3.

Full ceramic bearings

Full ceramic bearings are used only rarely. While the demands on the lubricant are low, mounting of these bearings is a considerable technical challenge. Due to the different thermal expansion and the high sensitivity to tensile stresses, it is difficult to achieve location of the bearings on shafts and in housings. Furthermore, the high price restricts their use to those cases where it is absolutely necessary.



The supply of lubricant to bearings

The supply of lubricant to bearings

The supply of lubricant to bearings

The lubricant quantity actually required by a rolling bearing is extraordinarily small. Due to the operational reliability of the bearing arrangement, however, it is normally estimated at a higher value in practice. However, too much lubricant in the bearing can lead to damage. If excess lubricant cannot escape, the splashing or churning work will lead to temperatures at which the lubricant may be impaired or even destroyed.

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

- selection of the correct lubricant quantity and distribution in the bearing
- attention to the operating life of the lubricant
- appropriate addition of lubricant or lubricant replacement
- targeted design of the bearing position
- the necessary devices and lubrication method, see table, page 60.

The supply of grease

In the case of grease lubrication, little or no work on devices is normally required in order to lubricate the bearings adequately. If the bearings fitted do not have an initial greasing carried out by the manufacturer, the bearings are frequently greased by hand when they are mounted. In many cases, this is assisted by the use of injection syringes or grease guns.

A selection of specific rolling bearing greases is shown in table Greases, page 126.

Initial greasing and new greasing

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

- Fill the bearings such that all functional surfaces definitely receive grease.
- Fill any housing cavity adjacent to the bearing with grease only to the point where there is still sufficient space for the grease displaced from the bearing. This is intended to avoid co-rotation of the grease. If a large, unfilled housing cavity is adjacent to the bearing, sealing shields or washers as well as baffle plates should be used to ensure that an appropriate grease quantity (similar to the quantity that is selected for the normal degree of filling) remains in the vicinity of the bearing. A grease filling of approx. 90% of the undisturbed free bearing volume is recommended. This is defined as the volume in the interior of the bearing that does not come into contact with rotating parts (rolling elements, cage).

- In the case of bearings rotating at very high speeds, such as spindle bearings, a smaller grease quantity is generally selected (approx. 60% of the undisturbed free bearing volume or approx. 30% of the total free bearing volume), in order to aid grease distribution during starting of the bearings.
- The sealing action of a gap seal is improved by the formation of a stable grease collar. This effect is supported by continuous relubrication.
- If the correct degree of filling is used, favourable friction behaviour and low grease loss will be achieved.
- If there is a pressure differential between the two sides of the bearing, the flow of air may drive the grease and the released base oil out of the bearing and may also carry contamination into the bearing. In such cases, pressure balancing is required by means of openings and holes in the adjacent parts.
- Bearing rotating at low speeds ($n \cdot d_M < 50\,000\ \text{min}^{-1} \cdot \text{mm}$) and their housings must be filled completely with grease.
  The churning friction occurring in this case is negligible. It is important that the grease introduced is held in the bearing or vicinity of the bearing by the seals and baffle plates. The reservoir effect of grease in the vicinity of the bearing leads to an increase in the lubrication interval. However, this is conditional on direct contact with the grease in the bearing (grease bridge). Occasional shaking will also lead to fresh grease moving into the bearing from its environment (internal relubrication).
- If a high temperature is expected in the bearing, the appropriate grease should be supplemented by a grease reservoir that has a surface as large as possible facing the bearing and that dispenses oil. The favourable quantity for the reservoir is two to three times the normal degree of filling. The reservoir must be provided either on one side of the bearing or preferably to an identical extent on both sides.

- Bearings sealed on both sides using sealing washers or sealing shields are supplied with an initial greasing. The grease quantity normally introduced fills approx. 90% of the undisturbed free bearing volume. This filling quantity is retained well in the bearing even in the case of high speed parameters ($n \cdot d_M > 400\,000\ min^{-1} \cdot mm$). In the case of higher speed parameters, please consult Schaeffler. A higher degree of filling in sealed bearings will lead to higher friction and continuous grease loss until the normal degree of filling is restored. If the egress of grease is hindered, a considerable increase in torque and temperature must be anticipated. Bearings with a rotating outer ring also receive less grease (50% of the normal filling).
- In the case of higher speed parameters, the bearing temperature may settle at a higher value, in some cases over several hours, if the grease quantity during the starting phase has not been set correctly, *Figure 1*. The temperature is higher and the increased temperature is longer, the more the bearings and the cavities adjacent to the bearings are filled with grease and the more difficult it is for grease to escape freely. A remedy is a so-called interval running-in process with appropriately determined standstill periods for cooling. If suitable greases and grease quantities are used, equilibrium is achieved after a very short time.

Deep groove ball bearing, freshly greased

M_r = frictional torque
t = time
ϑ = temperature

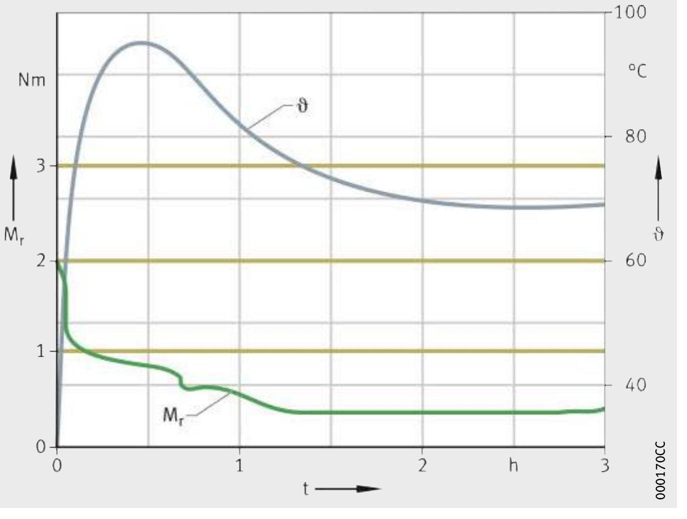


Figure 1
Frictional torque and temperature

Grease operating life

The grease operating life describes the period over which the grease is capable, without relubrication, of lubricating the bearing to an adequate extent. Once the grease operating life has been reached, function of the bearing is only conditionally possible and the bearing will fail relatively quickly as a result of lubricant failure. The grease operating life is therefore a decisive value if it is shorter than the calculated bearing life. It applies where rolling bearings cannot be relubricated.

The factors influencing the grease operating life are:

- the grease quantity and distribution
- the type of grease (thickener, base oil, additives)
- the production process of the grease
- the bearing type and size
- the magnitude and type of load
- the speed parameter
- the bearing temperature
- the mounting conditions.

Determination by testing

The grease operating life is determined by means of tests on a rolling bearing test rig (FE9) and on component test rigs. Such tests on lubrication must be carried out several times and evaluated by statistical methods.

Through statistical evaluation, it is possible to differentiate correctly on the basis of experience between different greases.
For assessment of a grease, both the 10% value as well as the 50% value for Weibull failure probability are necessary.

Calculation of the grease operating life

A guide value for the grease operating life t_{fG} can be determined in approximate terms using the following formula:

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

t_f h
Basic grease operating life

K_T –
Correction factor for increased temperature

K_P –
Correction factor for increased load

K_R –
Correction factor for oscillation

K_U –
Correction factor for environmental influences

K_S –
Correction factor for vertical shaft.

The supply of lubricant to bearings

The values determined are guide values only, since the determination is based on statistical principles. It is assumed that operating conditions are constant and that a suitable lubricant is present in a sufficient quantity. This is rarely the case in practice. As a result, the calculation model cannot supply precise values and almost no account is taken of other influences such as thermal conduction or contaminants.

Guidelines on calculating the grease operating life:

- In the case of combined bearings, the radial bearing and axial bearing must be calculated separately. The shorter grease operating life is then taken as the defining value.
- If the outer ring rotates, there may be a reduction in the grease operating life.
- In the case of yoke and stud type track rollers, angular defects may occur. The effect of the rotating outer ring is already taken into consideration in the bearing type factor k_f.

The grease operating life cannot be determined using the method described in the following cases:

- The grease can flow out of the rolling bearing
  - there is excessive vapourisation of the base oil
  - the bearing is not sealed
  - the axial bearing has a horizontal axis of rotation
- Air is sucked through the rolling bearing during operation
  - risk of increased grease oxidation
- Combined rotary and linear motion is present
  - the grease is distributed over the whole stroke length
- Contamination, water or other fluids enter the bearings
- There is no type factor for the bearings.

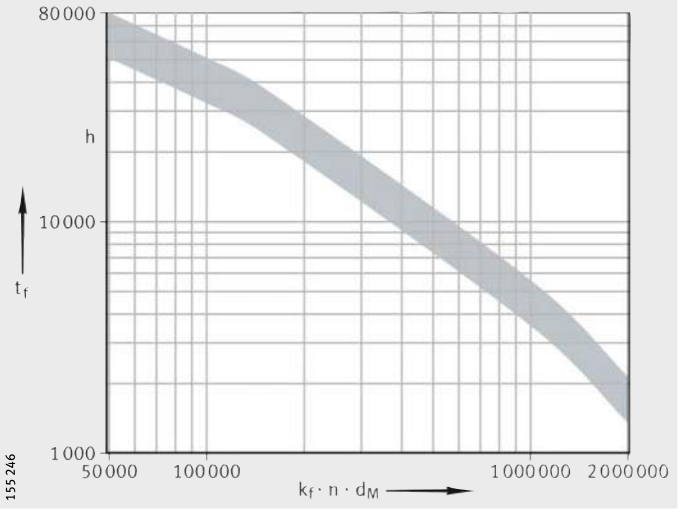
If the grease operating life is longer than 3 years, the lubricant manufacturer must be consulted.

Basic grease operating life

The basic grease operating life t_f is dependent on the bearing-specific speed parameter $k_f \cdot n \cdot d_M$. It is determined using *Figure 2* and table, page 98.

The basic grease operating life in accordance with *Figure 2* is valid in the following cases:

- greases with a proven performance capability for bearings, table Greases, page 126
- bearing arrangements where the bearing temperature is lower than the upper comtinuous limit temperature of the grease $T_{upperlimit}$
- a load ratio of $C_0/P \geqq 20$
- constant speed and load
- load in the main direction (radial in radial bearings, axial in axial bearings)
- radial bearings with a horizontal axis of rotation
- a rotating inner ring
- bearing arrangements without disruptive environmental influences.



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

Figure 2
Basic grease operating life t_f

k_f –
Bearing type factor, see table, page 98

n
Operating speed or equivalent speed

d_M mm
Mean bearing diameter (d + D)/2.

Factor k_f as a function of bearing type

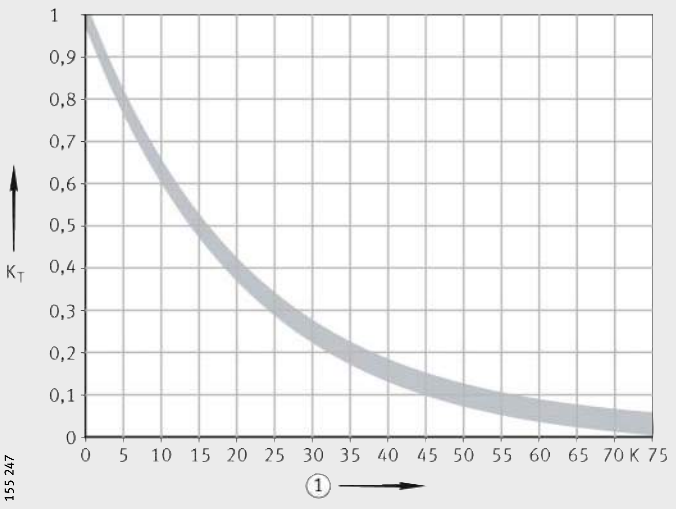
Bearing type	Factor k_f
Deep groove ball bearings, single row	1
Deep groove ball bearings, double row	1,5
Angular contact ball bearings, single row	1,6
Angular contact ball bearings, double row	2
Four point contact bearings	1,6
Self-aligning ball bearings	1,45
Axial deep groove ball bearings	5,5
Axial angular contact ball bearings, double row	1,4
Cylindrical roller bearings, single row, with constant axial load	3,25
Cylindrical roller bearings, single row, with alternating axial load or without axial load	2
Cylindrical roller bearings, double row (not valid for NN30)	3,5
Cylindrical roller bearings, full complement	5,3
Tapered roller bearings	4
Barrel roller bearings	10
Spherical roller bearings without central rib	8
Spherical roller bearings with central rib	10,5
Needle roller and cage assemblies, needle roller bearings	3,6
Drawn cup needle roller bearings with open ends, drawn cup needle roller bearings with closed end	4,2
Yoke and stud type track rollers, with cage or full complement cylindrical roller set	20
Yoke and stud type track rollers with full complement needle roller set	40
Ball bearing track rollers, single row	1
Ball bearing track rollers, double row	2
Yoke type track rollers PWTR, stud type track rollers PWKR	6
Cylindrical roller bearings LSL, ZSL	3,1
Crossed roller bearings	4,4
Axial needle roller bearings, axial cylindrical roller bearings	58
Insert bearings, housing units	1

Correction factor for increased temperature

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

As a rule of thumb, the following applies: an increase in temperature of 15 K will reduce the grease operating life by half. In the case of high grade greases. however, this effect is only pronounced above the so-called upper continuous limit temperature $T_{upperlimit}$. If the bearing temperature is above $T_{upperlimit}$, the reduction in the grease operating life due to temperature must be determined, *Figure 3*.

This diagram must not be used if the bearing temperature is higher than the upper operating temperature of the grease used, see table Greases, page 126 and table Arcanol rolling bearing greases, page 128. If necessary, another grease must be selected.



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

Figure 3
Temperature factor

Correction factor for increased load

Under higher bearing load, greases are subjected to greater strain. As a function of the load ratio C_0/P and the bearing type, this influence can be taken into consideration using the factor K_P, *Figure 4*.

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

①, ②, ③, ④: see table

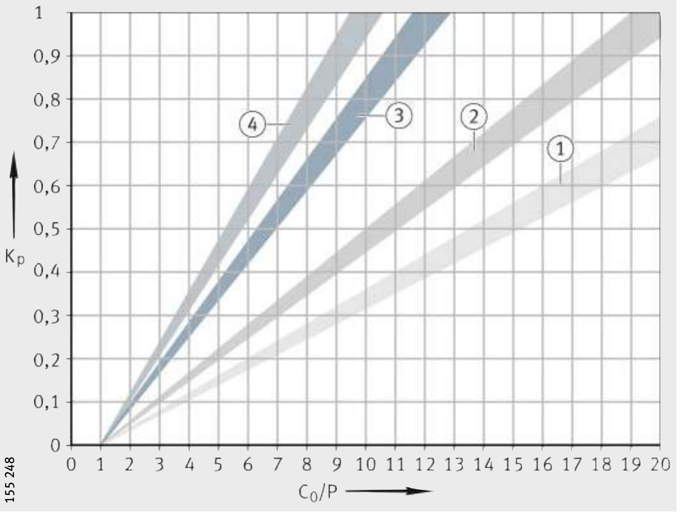


Figure 4
Load factor

Load factor K_P

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

[1] Curves, *Figure 4*.

Oscillation factor

Oscillating movements place a higher strain on the grease than continuously rotating bearings. The strain is placed continuously on the same grease volume, since no new grease can be drawn into the lubrication contact. As a result, the grease at the contact becomes depleted. In order to reduce fretting corrosion, the lubrication interval should be shortened. The reduction-inducing influence can be taken into consideration using the oscillation factor K_R, *Figure 6*. This is active starting from an angle of oscillation $\varphi < 180°$, *Figure 5* and *Figure 6*.

φ = angle of oscillation

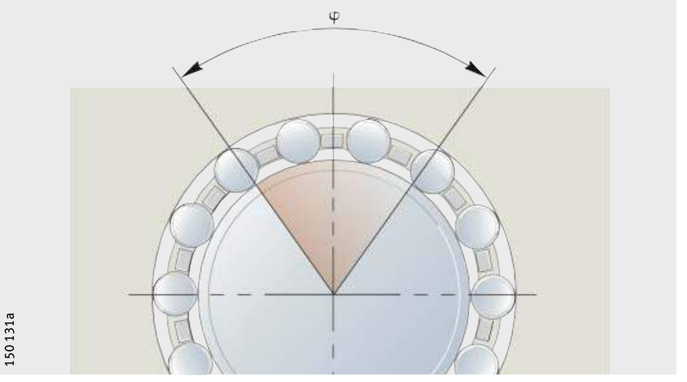


Figure 5
Angle of oscillation

K_R = oscillation factor
φ = angle of oscillation

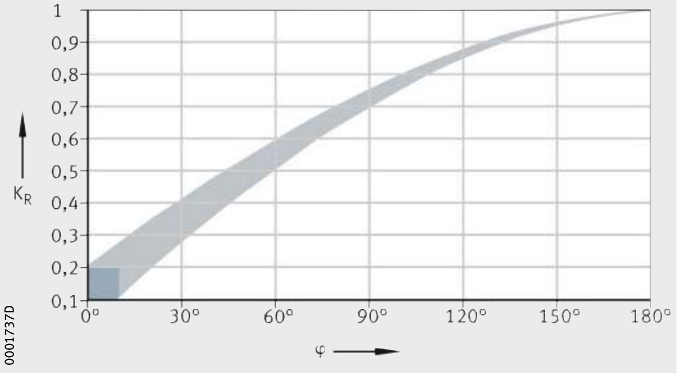


Figure 6
Oscillation factor

Environment factor

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

It does not take account of extreme environmental influences such as water, aggressive media, contamination, nuclear radiation and extreme vibrations such as those occurring in vibratory machines.

In relation to contamination, the influence of contamination on rating life calculation must also be noted, see section Load carrying capacity and rating life, page 18.

Environment factor

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

Factor for vertical shaft

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

Factor

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

Relubrication interval

Where rolling bearings are suitable for relubrication, regular relubrication is recommended in order to ensure the reliable function of the bearings.

Experience shows that, as a guide value, the relubrication interval t_{fR} for most applications can be calculated as follows:

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

t_{fR} h
Guide value for relubrication interval

t_{fG} h
Guide value for grease operating life, see page 95.

After this time, the grease in the bearing is used up to the extent that addition or replacement is necessary. Once the grease operating life is reached, the grease is in such a condition that it can no longer be simply pressed out of the bearing. For organisational and economic reasons, the lubrication intervals should be matched to the maintenance periods that are required in operational terms. Experience shows that relubrication intervals longer than one year should not be recommended, since they are frequently forgotten.

Relubrication should also be carried out before and after extended periods without operation, in order to achieve anti-corrosion protection in the bearing and to facilitate restarting with fresh grease.

The relubrication procedure should be carried out while the bearing is warm from operation and slowly rotating, in order to ensure good grease distribution. Old grease must be allowed to leave the bearing unhindered.

Relubrication and relubrication intervals

Relubrication or a grease change is necessary if the grease operating life is shorter than the expected bearing life.

Relubrication can be carried out as follows. Relubrication can often still be carried out using lever grease guns and lubrication nipples. Increasing importance is being attached to greasing systems such as the automatic lubricator Motion Guard and also central lubrication systems and grease spraying equipment. It is important that the used grease can be displaced by the new grease so that grease is replaced but overlubrication does not occur.

Special types of relubrication

Under certain environmental and application conditions, or where required by the adjuacent construction, special types of relubrication are necessary.

Addition of grease

Addition of grease should only be carried out if the used grease cannot be removed during relubrication (no free cavities in the housing, no grease outlet hole, no grease valve). The grease quantity supplied should be restricted in order to prevent overlubrication.

The supply of lubricant to bearings

Increased relubrication

Increased relubrication is necessary if there are large free cavities in the housing, grease regulators, grease outlet holes or grease valves are present or in the case of low speeds corresponding to $n \cdot d_M \leqq 100\,000\ \text{min}^{-1} \cdot \text{mm}$. In such cases, there is only a slight increase in temperature due to grease churning friction.
Generous relubrication improves the replacement of used grease by fresh grease and supports sealing against dust and moisture.
Where possible, relubrication should be carried out with the bearing warm from operation and rotating.

Grease replacement

Where lubrication intervals are long, the aim should be to achieve grease replacement. Substantial replacement of used grease by fresh grease is achieved with the aid of a larger relubrication quantity. A large relubrication quantity is necessary principally if the used grease has already been damaged due to higher temperature.
In order to remove as much used grease as possible by means of the "flushing effect", relubrication is carried out using a quantity that is up to three times as large as the normal relubrication quantity.
Suitable greases can be recommended by lubricant manufacturers.
A uniform supply of grease around the bearing circumference will aid grease replacement. Relevant design examples are shown in *Figure 7*, page 106 to *Figure 14*, page 111. The precondition for substantial replacement of used grease by fresh grease is that the used grease can escape freely or a sufficiently large cavity is made available for collection of the used grease.

Very short relubrication intervals

Very short relubrication intervals (daily or even shorter) are applicable if extreme strains are present. In such cases, the use of a grease pump or lubricators is justified.

Support for seals by escaping grease

The seals can be supported by escaping grease if relubrication is carried out using small quantities at short intervals.
The relubrication quantity per hour can be between half and several times as large as the grease quantity that will fit in the free bearing interior.

Relubrication at high temperature

At high temperature, grease lubrication is possible with economical grease that is stable for only short periods or with expensive grease that has good temperature stability. For the greases stable for short periods, relubrication corresponding to 1% to 2% of the free bearing cavity per hour has proved effective for lubrication.
In the case of greases with good temperature stability, significantly smaller relubrication quantities are sufficient.

! During relubrication, it must be ensured that there is no impermissible mixing of lubricant, see section Miscibility of lubricants, page 130.

Arcanol rolling bearing greases

A selection of lubricants in various container sizes is included in the Schaeffler range under the name Arcanol. Each of these greases is subjected to a comprehensive series of tests before it is accepted into the range. These are carried out not only in the lubricant laboratory but also and principally on test rigs, where the grease must demonstrate its suitability in various rolling bearing types and under defined conditions.

The greases are tested in rolling bearings in relation to operating life, friction behaviour and wear on the FE8 test rig (DIN 51819) and FE9 test rig (DIN 51821). If the results fulfil the requirements of the Schaeffler specifications, the grease is accepted into the Arcanol range.

Each batch supplied of these greases is first tested in order to ensure uniform quality. It is only after the incoming goods testing has been completed successfully that approval is given to fill containers using the grease as Arcanol. The greases are sold via the Business Division Industrial Aftermarket of the Schaeffler Group. Technical data sheets and safety data sheets can be requested.

The Arcanol grease range is graduated such that many areas of application can be covered using this selection of greases. The individual rolling bearing greases therefore differ in their possible applications and specific key data, table, page 128.

Examples of grease lubrication

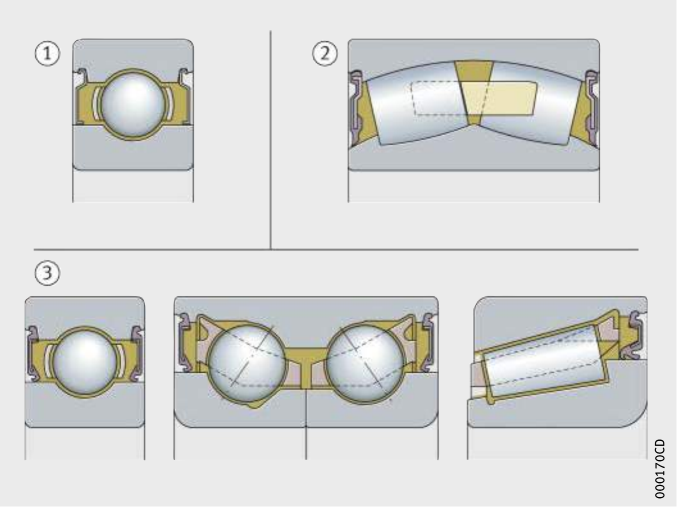
There are various possibilities for supplying a rolling bearing with grease. The method used is based on the requirements of the specific bearing arrangement.

Sealed bearings

Rolling bearings that are sealed and filled with grease during manufacture facilitate simple adjacent constructions, *Figure 7*. Sealing shields or sealing washers are provided, depending on the application, as single seals or in addition to a further outer seal. Contact type sealing washers increase the bearing temperature as a result of seal friction. Sealing washers and non-contact seals form a gap relative to the inner ring and do not therefore influence friction.

Deep groove ball bearings sealed on both sides are filled with a lithium soap grease of consistency grade 2 or 3, where the softer grease is used for small bearings.

The grease quantity introduced fills approx. 90% of the undisturbed free bearing volume, *Figure 7*. It is determined such that, under normal operating and environmental conditions, a long operating life will be achieved. The grease is distributed during a short running-in phase and settles to a large extent in the undisturbed part of the free bearing cavity, in other words on the inner sides of the washers. No significant co-rotation is found after this time and the bearing runs with low friction. Once the running-in phase is complete, the friction is only 30% to 50% of the starting friction.

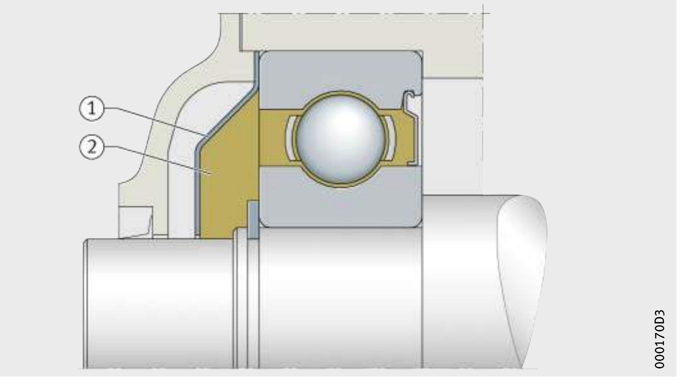


① Design with sealing shields
② Design with non-contact sealing washers
③ Design with contact type sealing washers

Figure 7
Sealed bearings

Bearings sealed on one side with baffle plate

The deep groove ball bearing is sealed on one side, while a baffle plate with a grease reservoir is arranged on the other side, *Figure 8*. The bearing thus has a larger grease quantity in the vicinity of the bearing but not in the bearing itself. At high temperature, the grease reservoir releases oil intensively and over the long term to the deep groove ball bearing. As a result, long running times are achieved without the occurrence of additional lubricant friction. Suitable greases can be recommended by agreement by the Schaeffler engineering service.



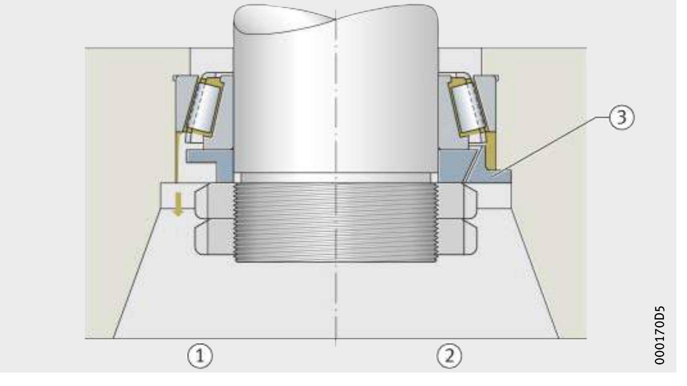
1 Baffle plate
2 Grease reservoir

Figure 8
Bearing sealed on one side with baffle plate

Vertically arranged bearings with baffle plate

Where bearings have a pumping action or bearing arrangements have a vertical shaft, a baffle plate prevents the grease from flowing out of the bearing at all or not as quickly, *Figure 9*. In the case of bearing types that have higher proportions of sliding motion and a pronounced pumping effect in particular (for example tapered roller bearings), an outer baffle plate is advantageous if not always sufficient at higher circumferential speeds.

Short relubrication intervals are a further measure for ensuring supply of grease.

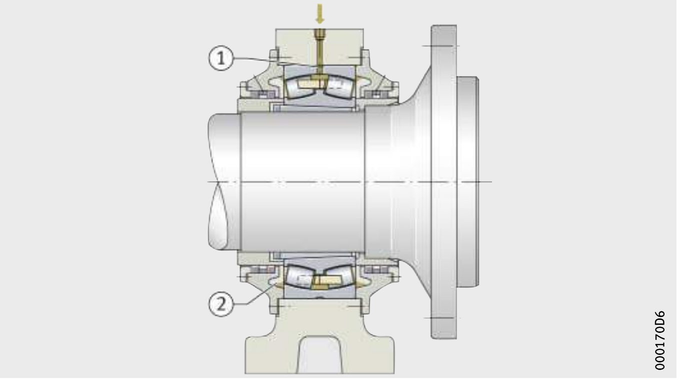


1 Incorrect
2 Correct
3 Baffle plate

Figure 9
Bearing with vertical arrangement and baffle plate

Lubrication hole in the outer ring

The grease is pressed into the bearing interior via a lubrication groove and lubrication holes in the bearing outer ring, *Figure 10*. Due to the direct and symmetrical feed of the grease, a uniform supply to both rows of rollers is achieved. On both sides, sufficiently large cavities for collection of the used grease or openings for the escape of grease must be provided.



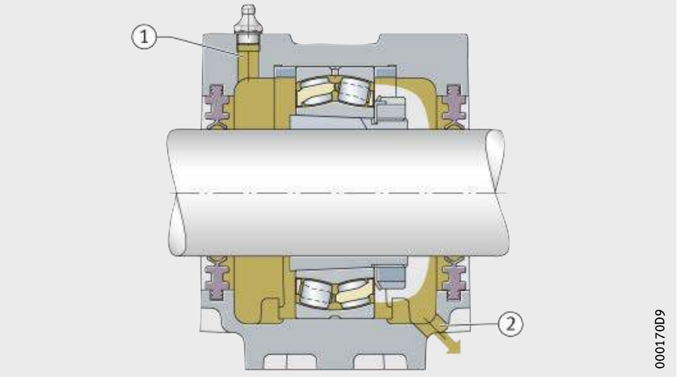
① Lubrication groove with lubrication holes
② Cavity for grease collection

Figure 10
Relubrication via the lubrication groove in the outer ring

Spherical roller bearings

The spherical roller bearing is relubricated from the side, *Figure 11*. During relubrication, grease is intended to exit on the opposing side. Grease back-up may occur if large quantities are used frequently for relubrication and there is resistance to the escape of grease. This can be remedied by a grease outlet hole or a grease valve.

During the startup phase, the movement of grease leads to a temperature increase (approx. 20 K to 30 K above the equilibrium temperature), which may last for one or more hours. The type and consistency of grease have a strong effect on the temperature behaviour.



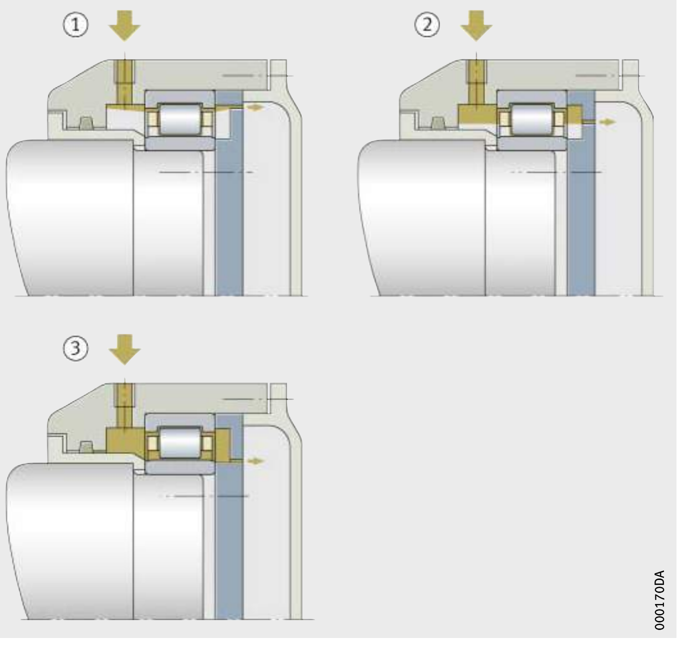
① Lubrication groove
① Grease outlet hole

Figure 11
Relubrication of a spherical roller bearing

Grease quantity regulator

A grease quantity regulator conveys excess grease to the exterior through a narrow gap between the housing and a regulating washer rotating with the shaft, *Figure 12*. Where long relubrication intervals, higher circumferential speeds and an easily movable grease are present, there is a risk that only a small quantity of grease will remain in the bearing on the side with the regulating washer. This can be remedied by moving the gap between the rotating regulating washer and the stationary outer part towards the shaft.

In a normal grease quantity regulator with a gap on the outside, there is a strong pumping action. A moderate pumping action is achieved if the gap is arranged approximately on the pitch circle diameter of the bearing. If the gap is on the inside, practically no pumping action is achieved, the washer acts as a baffle plate and retains the grease in the bearing.

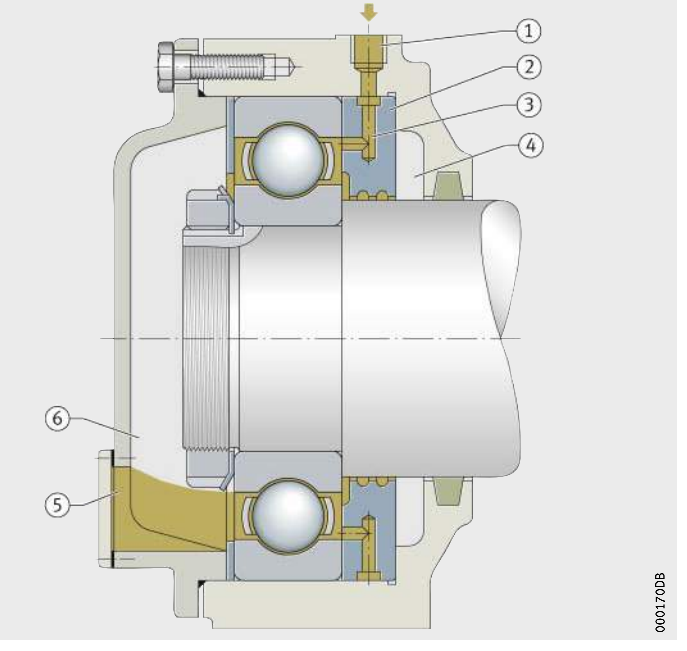


① Gap arranged on the outside
② Gap arranged on the pitch circle diameter
③ Gap arranged on the inside

Figure 12
Pumping action due to regulating washer

Targeted relubrication from the side

A washer with holes allows targeted lubrication from one side, *Figure 13*. During relubrication, the grease passes through the hole in the washer directly into the gap between the cage and outer ring. The grease displaced during relubrication collects in the free space, which must be emptied from time to time via an opening.
The chamber on the right side of the bearing is filled with grease at the time of mounting. This is intended to improve sealing.
During relubrication while stationary, good replacement of used grease by fresh grease is achieved if the holes are arranged around the circumference of the disc such that the grease is distributed uniformly around the circumference of the bearing. The holes located in the area of the filling hole must therefore be further from each other than the diametrically positioned holes. This gives uniform flow resistance and the relubrication grease pushes the used grease uniformly out of the bearing. The replacement of used grease by fresh grease is promoted by large relubrication quantities.

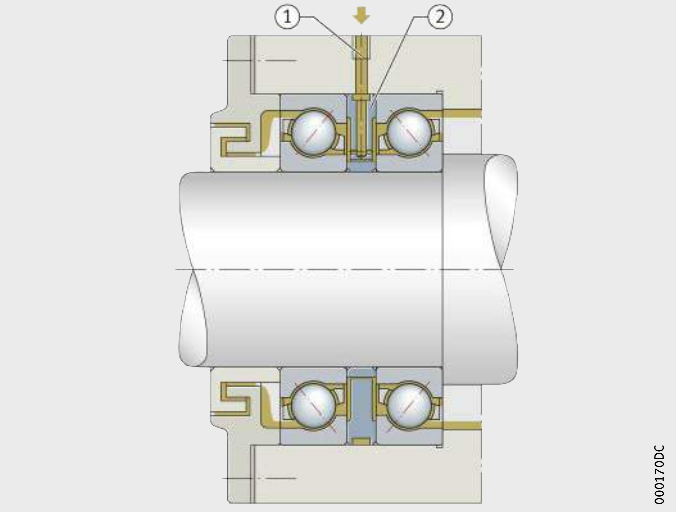


1 Filling hole
2 Washer
3 Hole
4 Chamber
5 Opening
6 Free space

Figure 13
Targeted relubrication from the side

Bearing pairs

The pair of angular contact ball bearings is supplied with fresh grease via lubrication holes. These are located in the washer that is fitted between the bearings, *Figure 14*. This prevents grease back-up caused by grease being fed to the small diameter. The centrifugal force directs it outwards to the larger diameter. This effect only occurs in bearings with an asymmetrical cross-section and thus in angular contact ball bearings and tapered roller bearings.
If a bearing pair with a symmetrical cross-section is lubricated from the centre, a regulating washer or exit opening should be arranged next to each individual bearing. It is important that the escape resistance at each point is approximately the same. If this is not the case, the grease will tend to move towards the side with the lowest escape resistance. There is then a risk of lubricant undersupply on the opposing side.



1 Lubrication hole
2 Washer

Figure 14
Lubrication
of a bearing pair from the centre

Summary

The examples show that correct guidance of grease is normally costly. It is preferable that these costs are expended in the case of expensive machines or difficult operating conditions such as higher speed, load or temperature. In these cases, the replacement of used grease must be ensured and overlubrication must be prevented.

In a normal application, such costs are not necessary. This is shown by operationally reliable bearings with a lateral grease buffer. These grease buffers on both sides of the bearing gradually release oil for lubrication of the contact surfaces and offer additional protection against contamination of the bearing interior. In general, it is also the case that relubrication of bearings represents a source of defects. For example, contamination can enter the bearing from outside through relubrication. Lifetime lubrication should always be used in preference to relubrication.

The supply of oil

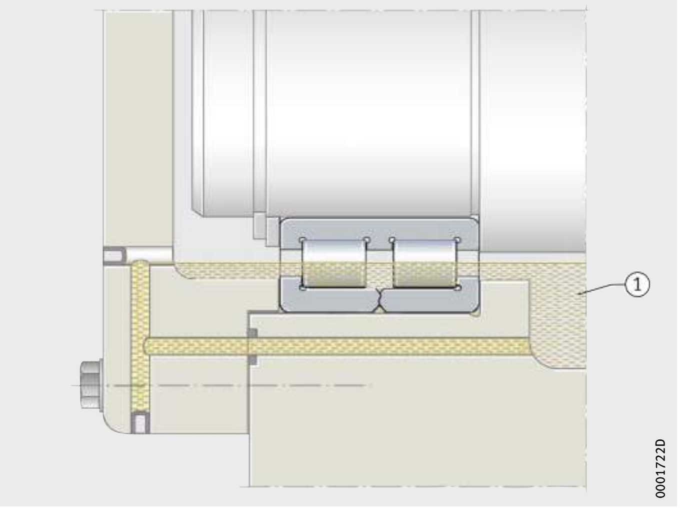
If oil bath lubrication is not provided, the bearing positions must be supplied with oil by means of devices. The cost of devices depends on the lubrication method selected. Oil is supplied by pumps where lubrication is carried out using larger and smaller quantities. Pneumatic oil systems and central oil lubrication systems can be used for lubrication with small and very small quantities.
Oil metering is carried out by means of metering elements, throttles and nozzles. For the most common lubrication systems, see section Lubrication methods, page 52.

Oil bath lubrication

In oil bath lubrication (sump lubrication), part of the bearing is located in an oil sump. Where the axis of rotation is horizontal, the oil level should be measured such that the lowest rolling element is half-immersed or completely immersed in the oil when stationary, *Figure 15*.

When the bearing is rotating, some of the oil is picked up by the rolling elements and cage and thus distributed over the circumference. In bearings with an asymmetrical cross-section, which convey oil, oil return ducts must be provided due to the pumping effect so that recirculation can be achieved. If the oil level is higher than the lowest rolling element, the splashing friction especially at high circumferential speeds will lead to increased bearing temperature and often also to foaming. Speed parameters $n \cdot d_M < 150\,000 \text{ min}^{-1} \cdot \text{mm}$ also facilitate a higher oil level.
If it is unavoidable that a rolling bearing is located completely in oil, for example where the axis of rotation is vertical, the frictional torque can be two to three times higher than with a normal oil level.

The maximum speed parameter in the case of oil lubrication is normally $n \cdot d_M = 300\,000 \text{ min}^{-1} \cdot \text{mm}$ and, with frequent oil changes, $500\,000 \text{ min}^{-1} \cdot \text{mm}$. At a speed parameter of or higher than $n \cdot d_M = 300\,000 \text{ min}^{-1} \cdot \text{mm}$, the bearing temperature will often be above +70 °C. In oil bath lubrication, the oil level should be checked regularly.



1 Oil sump

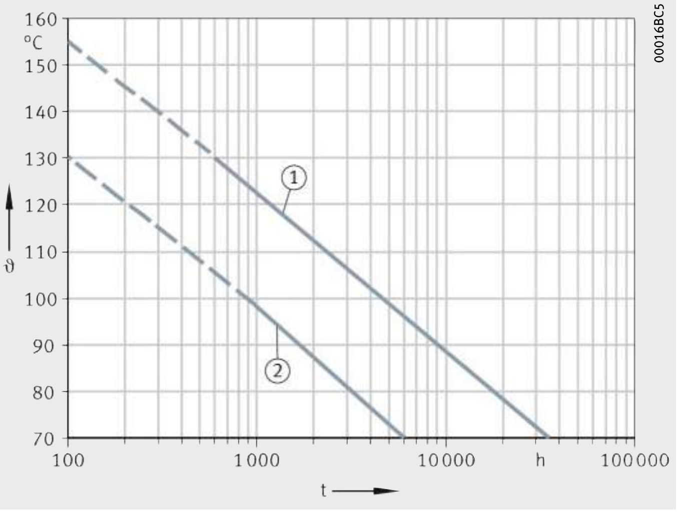
Figure 15
Oil bath lubrication

Oil change interval

The oil change interval is dependent on the contamination, the ageing condition and the additive consumption of the oil. Under normal conditions, oil changes intervals should be observed, *Figure 16*.

The precondition is that contamination due to foreign matter and water must remain low. Housings with small oil quantities require frequent oil changes, especially in the case of bearings that are lubricated together with gears. An early oil change is often undertaken due to the increasing quantity of solid and liquid contaminants. The permissible quantities of solid contaminants are based on the size and hardness of the particles.

The laboratories of the Schaeffler Group carry out investigations into the condition and lubrication capability of oils. Ageing is promoted by oxygen, metal debris (acting as a catalyst) and high temperatures. The ageing status can be assessed by the change in the acid number NZ and the saponification number VZ. In critical cases, the oil change interval should be defined on the basis of repeated oil investigations. It is recommended that the acid number NZ, the saponification number VZ, the quantity of solid foreign matter, the water content and the viscosity of the oil should be determined first after 1 to 2 months and later at longer intervals depending on the result. The bearing life is drastically reduced when constantly in contact with even a low water content. The degree of ageing and contamination can be estimated on an approximate basis by 1 drop each of fresh and used oil on blotting paper. Significant differences in colour indicate considerable ageing or contamination, see section Contaminants in the lubricant, page 136.



Source: ExxonMobil

ϑ = continuous oil bath temperature
t = oil change interval

① Synthetic gearbox oils
② Mineral gearbox oils

Figure 16
Oil change intervals

Recirculating lubrication

In recirculating relubrication, the oil passes through the bearings, is directed into a collection container and is then fed back into the bearings, *Figure 17*. Wear particles and contaminants have a negative effect on the achievable life, see section Load carrying capacity and rating life, page 18.
It is therefore absolutely essential to provide a filter in order to separate out the wear particles and contaminants.

1 Filter
2 Pump
3 Cooling system

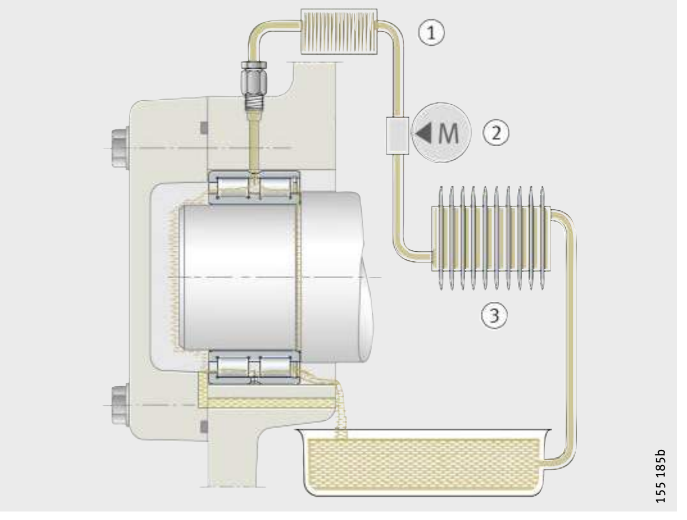


Figure 17
Recirculating oil lubrication

Oil recirculation quantity

The recirculation quantities which, at viscosity ratios $\kappa = \nu/\nu_1$ from 1 to 2,5, give a moderate bearing flow resistance can be taken from the diagram, *Figure 18*.

$\dot{V}$ = oil quantity
D = bearing outside diameter

1 Increasing oil quantity required for heat dissipation
2 No heat dissipation necessary

a = oil quantity sufficient for lubrication
b = upper limit for bearings of symmetrical design
c = upper limit for bearings of asymmetrical design
a_1; b_1; c_1: D/d > 1,5
a_2; b_2; c_2: D/d ≦ 1,5

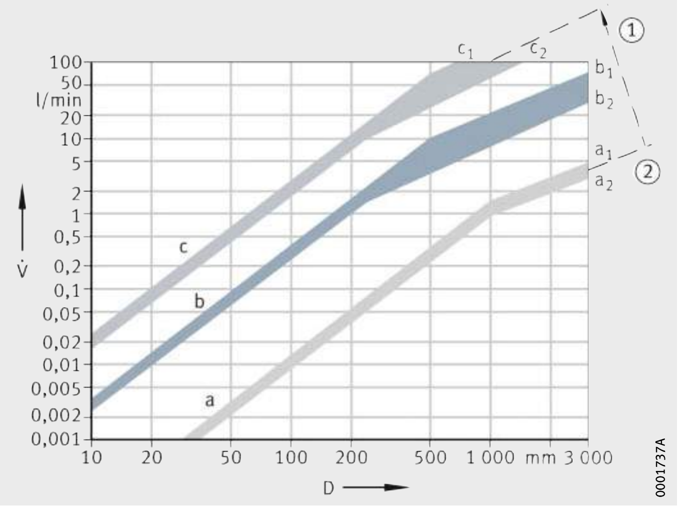


Figure 18
Oil quantities

Operating conditions

The recirculation quantities are matched to the operating conditions:

- Lubrication of the bearings requires only a very small quantity of oil. In comparison, the quantities stated as sufficient for lubrication (*Figure 18*, line a) are large. These oil quantities are recommended in order to ensure that all contact surfaces are still reliably supplied with oil even if the feed of oil to the bearing is unfavourable, in other words if feed is not directly into the bearing. The minimum quantities stated are used for lubrication if a low level of friction is required. The temperature level achieved in this case is comparable with that in oil bath lubrication.
- If heat dissipation is required, larger oil quantities are necessary (*Figure 18*, line b). Since each bearing provides some resistance to the flow of oil, there are also upper limits for the oil quantities.
- For bearings with an asymmetrical cross-section such as angular contact ball bearings, tapered roller bearings or axial spherical roller bearings, larger throughput quantities are permissible (*Figure 18*, line c) than for bearings with a symmetrical cross-section. This is due to the fact that bearings with an asymmetrical cross-section provide less resistance to the oil flow due to their pumping action.

The stated limits are based on the precondition of unpressurised feed and back-up of oil on the feed side of the bearing as far as just below the shaft. The oil quantity that must be provided in individual cases in order to maintain an adequately low bearing temperature is dependent on the conditions of heat input and dissipation. Values higher than those in region c according to *Figure 18* are not advisable. The correct oil quantity can be determined by temperature measurement during commissioning of the machine and then regulated accordingly.

Injection lubrication

With increasing circumferential speed, bearings with a symmetrical cross-section provide increasing resistance to the oil flow. If larger recirculation quantities are planned, the oil is injected specifically into the gap between the cage and bearing ring in the case of rolling bearings rotating at high speeds. With oil injection, smaller splashing losses occur.

Normal oil quantities can be determined as a function of the speed parameter and bearing size, *Figure 19*, page 116. Furthermore, the nozzle diameter can be determined, *Figure 20*, page 116. The back-up of oil ahead of the bearing is prevented by injecting oil at points that allow free entry into the bearing. If the outlet ducts ahead of and after the bearing arrangement are adequately dimensioned, this will ensure that the oil not consumed by the bearing and flowing through the bearing can escape freely.

Injection lubrication

$\dot{V}$ = volume flow of oil (oil quantity)
$n \cdot d_M$ = speed parameter
d_M = mean bearing diameter

① d_M = 150 mm
② d_M = 100 mm
③ d_M = 50 mm

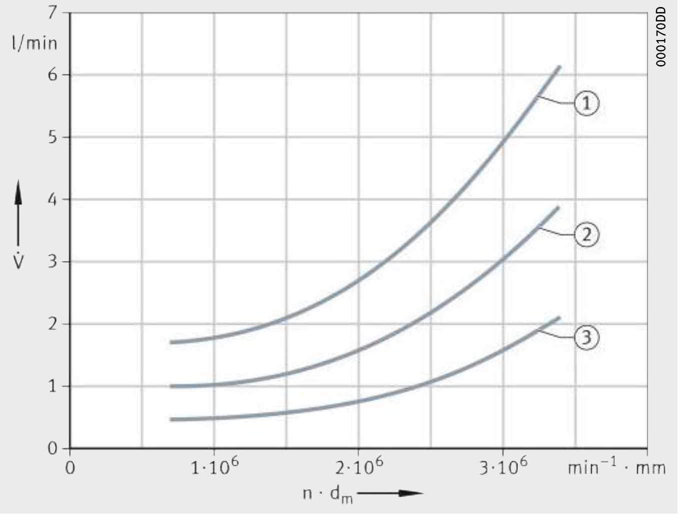


Figure 19
Oil quantities

$d_M \leqq 50$ mm: 1 nozzle
50 mm $\leqq d_M \leqq 100$ mm: 2 nozzles
$d_M \geqq 100$ mm: 3 nozzles

D = nozzle diameter
$n \cdot d_M$ = speed parameter
d_M = mean bearing diameter

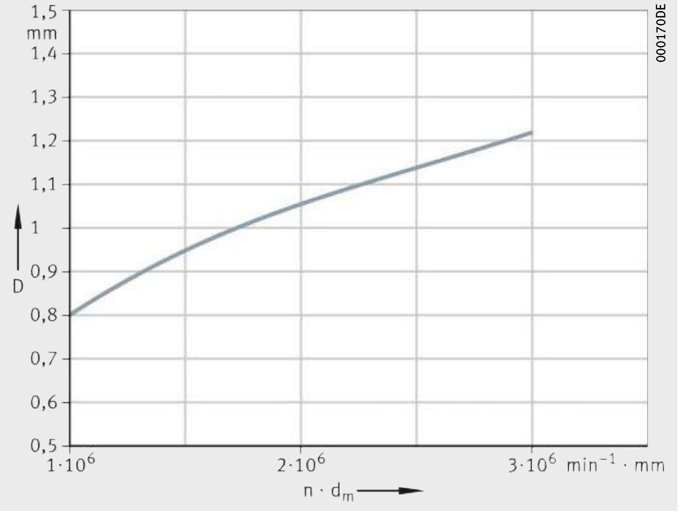


Figure 20
Nozzle diameter

Pressure loss and injection speed

In the range of high circumferential speeds, which are normal for injection lubrication, the oils that have proven effective are those with which an operating viscosity $\nu = 5\ mm^2/s$ to $10\ mm^2/s$ is achieved ($\kappa = 1$ to 4). The diagrams give, as a function of pressure drop, nozzle diameter and operating viscosity, the oil quantity and the jet velocity, *Figure 21*.

These data are derived from tests. The oil flow rate through the rapidly rotating bearing decreases with increasing speed. It increases with increasing injection velocity, for which 30 m/s is the advisable upper limit.

Nozzle length L = 8,3 mm

$\dot{V}$ = volume flow of oil (oil quantity)
v = jet velocity
Δp = pressure drop
D = nozzle diameter in mm

① Operating viscosity $\nu = 7{,}75\ mm^2/s$
② Operating viscosity $\nu = 15{,}5\ mm^2/s$

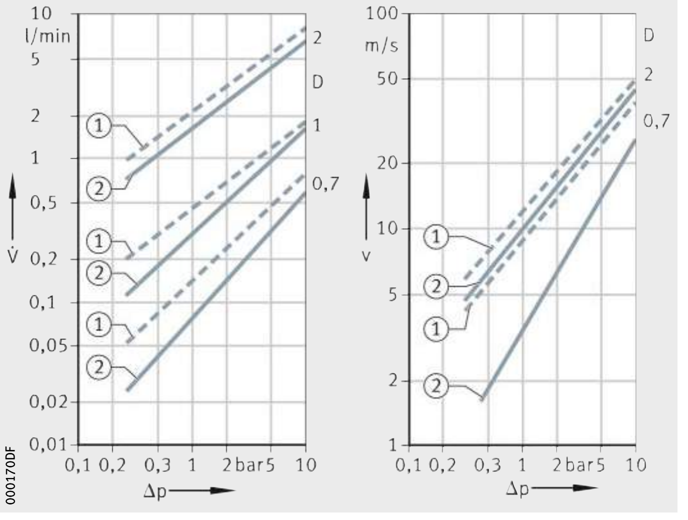


Figure 21
Pressure loss and injection speed

Design considerations

Rolling bearings must be provided with lubricant as soon as the machine is switched on. In the case of recirculating oil lubrication, the pump should therefore start up before the bearing starts to move. An oil sump provided in addition to the recirculating lubrication system also contributes to operational security, since oil can be supplied from the sump for at least a certain period if the pump fails. At low temperatures, the recirculating oil quantity can initially be reduced to the quantity necessary for lubrication until the oil in the container has heated up. The assists in the design of the recirculation system (pump drive, oil return system).

If lubrication is carried out using a larger oil quantity, outlet ducts must be provided in such a way as to prevent oil back-up that leads, mainly at high circumferential speeds, to significant power losses. The required diameter of the outlet line is dependent on the viscosity of the oil and the drop angles of the discharge pipes.

Diameter of outlet line

For oils with an operating viscosity of up to 500 mm^2/s, the diameter of the outlet line can be stated approximately in mm:

$$d_a = (15\ldots25) \cdot m^{0,5}$$

d_a	mm
Free diameter of outlet line	
m	l/min
Oil throughput quantity.	

For more precise dimensioning in the drop region of the outlet line from 1% to 5%, the diameter is as follows:

$$d_a = 11,7 \cdot \left(\frac{m \cdot \nu}{G}\right)^{0,25}$$

d_a	mm
Free diameter of outlet line	
m	l/min
Oil throughput quantity	
ν	mm^2/s
Operating viscosity	
G	%
Drop.	

Fill quantity of oil container

$$M = m \cdot \frac{60\,\text{min}}{z}$$

M	l
Fill quantity of oil container	
m	l/min
Oil throughput quantity	
z	–
Circulation parameter.	

The fill quantity of the oil container is based on the oil throughput. In general, the fill quantity is selected such that circulation occurs approx. z = 3 to 8 times per hour.

At a low circulation parameter, contaminants are easily deposited in the oil container, the oil can be cooled and does not age so quickly. At a high circulation parameter, there is a risk of excessive foaming, see section Foaming behaviour, page 147.

Minimal quantity lubrication

Minimal quantity lubrication is defined as the supply of lubricant to all contacts in that quantity which both ensures lubrication and also generates the least possible lubricant friction. Minimal quantity lubrication can be carried out with grease as well as with oil.

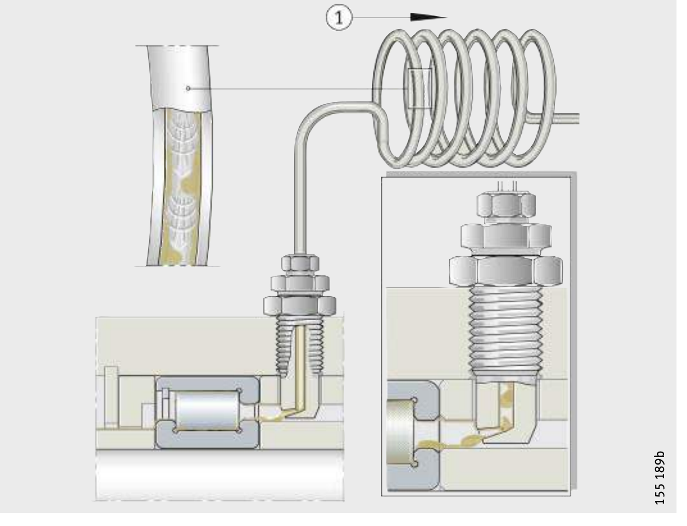
Minimal grease quantities

Lifetime lubrication with grease is the optimum minimal quantity lubrication. With extrapolation to the total running time, reliable lubrication of a small electric motor bearing will require the consumption of only 0,05 mm^3/h of base oil. Relubrication with a very small quantity is already common practice for machine tool bearings. In this case, speed parameters of up to $2 \cdot 10^6$ $min^{-1} \cdot mm$ require the supply of quantities of 0,1 cm^3 at short intervals of 2 hours and longer. It is important that the grease used remains consistent under the supply conditions. For this type of lubrication, the Schaeffler Group makes its own bearing designs available. The used grease can be collected in a reservoir or transported outside.

Minimal oil quantities

In the case of oil, minimal quantities are defined as oil supply in the range of a few cm^3/h or mm^3/h. This is possible if no dissipation of heat from the bearing is necessary. Oil can be metered in very small quantities at intervals. For transport and equalisation of the oil supply, the oil is added in metered quantities of $\geqq 5$ mm^3 to a continuous air flow, *Figure 22*.

This prevents flooding of the contact points and creates a quasicontinuous pneumatic oil flow. The air flow should be in quantities of approx. 2 m^3/h.



1 To the pneumatic oil unit

Figure 22
Pneumatic oil lubrication

Frictional torque and bearing temperature

The example of a double row cylindrical roller bearing shows how, in the case of minimal quantity lubrication, the frictional torque and bearing temperature change as a function of the oil throughput quantity, *Figure 23* and *Figure 24*.

In particular, it can be seen that the double row cylindrical roller bearing with ribs on the outer ring is particularly sensitive to over-lubrication. More suitable designs in this case are double row cylindrical roller bearings with ribs on the inner ring, for example NN30, or single row cylindrical roller bearings of the series N10 and N19. The minimum friction and temperature (start of full lubrication) is already achieved at an oil quantity of 0,01 mm^3/min to 0,1 mm^3/min. As the oil quantity increases to 10^4 mm^3/min, the bearing temperature increases. It is only when an even larger oil quantity is used that the bearing temperature decreases as a result of heat dissipation.

Double row cylindrical roller bearing NNU4926
Speed $n = 2\,000\ min^{-1}$
Radial bearing load $F_r = 5$ kN
Oil viscosity $\nu = 32\ mm^2/s$ at 40 °C

M = frictional torque
$\dot{V}$ = volume flow of oil (oil quantity)

① Maximum frictional torque occurring
② Minimum frictional torque occurring

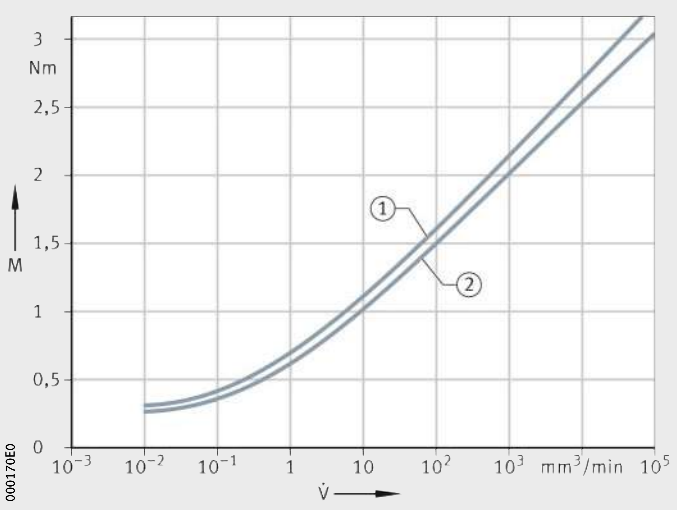


Figure 23
Frictional torque

Double row cylindrical roller bearing NNU4926
Speed $n = 2\,000\ min^{-1}$
Radial bearing load $F_r = 5$ kN
Oil viscosity = 32 mm^2/s at 40 °C

T = bearing temperature
= volume flow of oil (oil quantity)

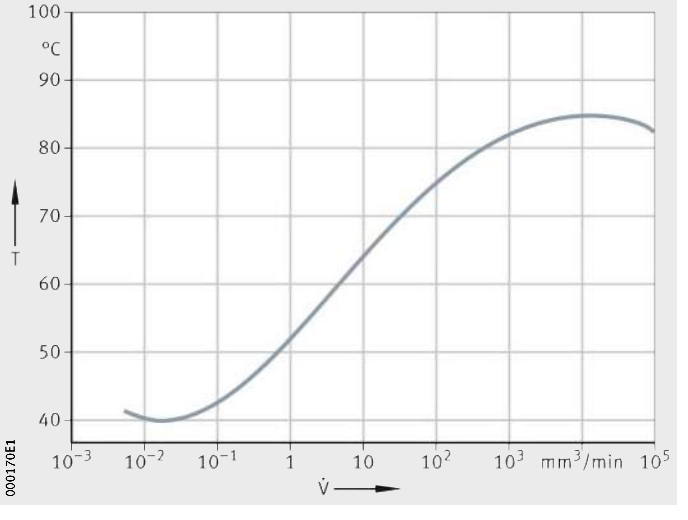


Figure 24
Bearing temperature

Bearing type

The oil quantity required for adequate supply is heavily dependent on the bearing type. Bearings with a pumping action in the flow direction thus require a relatively large oil quantity. In contrast, the oil requirement of double row bearings without pumping action is extremely small if the oil is fed between the rows of rollers. The oil is prevented from flowing out by the rotating rolling elements.

The precondition for lubrication with very small quantities is that the small oil quantity gives sufficient coating of all contact surfaces in the bearing and, in particular, the sliding contact surfaces such as rib and cage guidance surfaces, which are particular challenging in terms of lubrication technology. In machine tool bearing arrangements with ball bearings and cylindrical roller bearings, oil supply directly into the bearing and, in the case of angular contact ball bearings in the pumping direction, has proved effective.

The oil quantities in minimal quantity lubrication are stated for various bearing types as a function of bearing size, contact angle (pumping behaviour) and the speed parameter, *Figure 25*, page 122.

In bearings with pumping action, the oil quantity should be increased as a function of the speed, since the minimum oil requirement and the pumping action increases with increasing speed.

In bearings with contact between ribs and the end faces of rollers, such as tapered roller bearings, an additional oil feed directly to the roller end faces – opposing the pumping direction – has proved favourable. The precondition for extremely low oil quantities are a reliable feed between the cage and inner ring as well as high dimensional accuracy of the adjacent parts. The viscosity of the oil should, in the case of an extremely small oil quantity, conform to a viscosity ratio $\kappa = \nu/\nu_1$ = 8 to 10 and contain suitable agents.

In contrast, the uniform feed of a large oil quantity or the pulse-type feed even of small quantities leads, in the case of radial cylindrical roller bearings, and especially at high circumferential speeds, to a spontaneous increase in lubricant friction and non-uniform heating of the bearing rings. In bearings with a small radial internal clearance, for example in machine tool bearing arrangements, this can lead to failure of the bearings as a result of radial tensioning.

$\dot{V}$ = volume flow of oil (oil quantity)
d = bore diameter

① Angular contact ball bearings, axial angular contact ball bearings
② Spindle bearings
③ Single row and double row cylindrical roller bearings
④ Cylindrical roller bearings with ribs on the inner ring
⑤ Cylindrical roller bearings with ribs on the outer ring

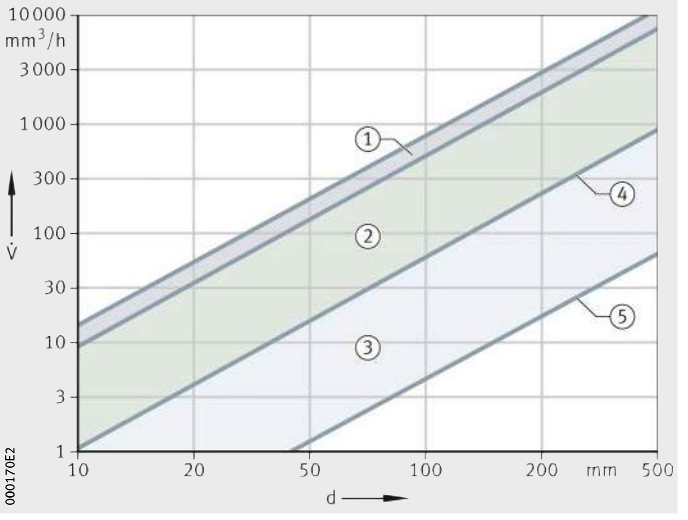


Figure 25
Oil quantities
in minimal quantity lubrication

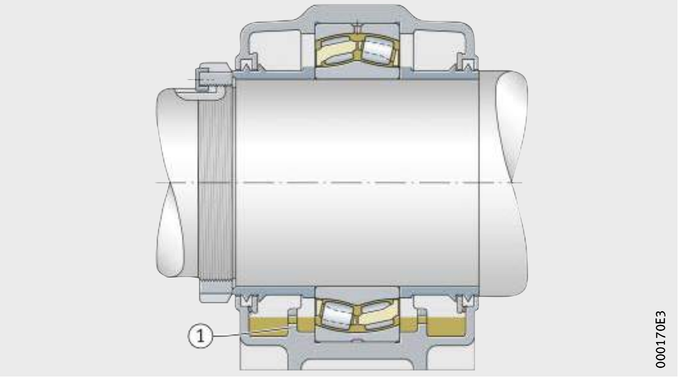
Regions in the diagram

Region	Bearing type	Contact angle α °	Speed parameter $n \cdot d_M$ $min^{-1} \cdot mm$
	Angular contact ball bearings	40	≦ 800 000
	Axial angular contact ball bearings	60 to 75	
		90	
	Spindle bearings	15 to 25	$\leqq 2 \cdot 10^6$
	Single row and double row cylindrical roller bearings	–	–
Line	with ribs on the inner ring	–	$\leqq 10^6$
Line	with ribs on the outer ring	–	≦ 600 000

Examples of oil lubrication

Bearing housing with oil partitions

In larger housings with a correspondingly large oil content, the oils sump is divided up by partitions with through holes, *Figure 26*. This prevents the entire oil quantity from coming into motion, principally at higher circumferential speeds. Contaminants settle in the adjacent chambers and are not continually stirred up.

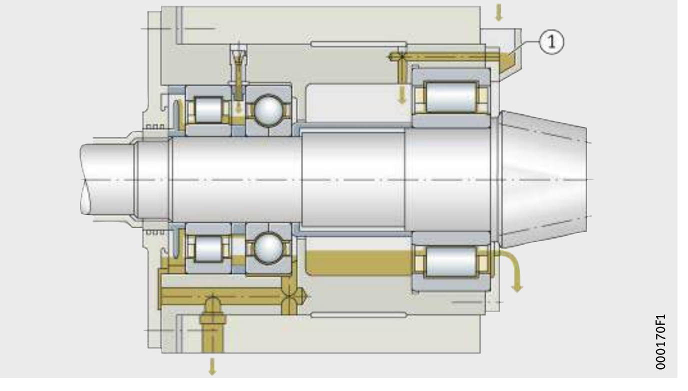


① Oil partition

Figure 26
Bearing housing with oil partitions

Splash oil feed through collector pocket

In gearboxes, the oil splashing off gears is often sufficient for lubrication of the rolling bearings, *Figure 27*. It must be ensured here that the splash oil reaches the bearings under all operating conditions. In the example, splash oil is collected in a pocket above the cylindrical roller bearing and fed to the bearing via holes.
In the lower area, a baffle plate is located next to the cylindrical roller bearing. This ensures that a minimum oil sump is always present in the bearing and the bearing is already lubricated at startup.

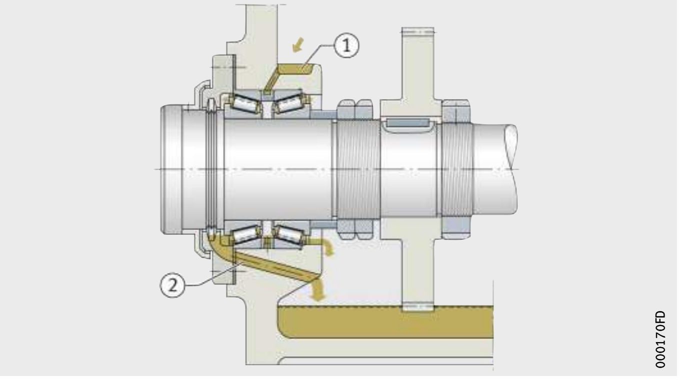


① Oil feed via collector pocket

Figure 27
Splash oil feed through collector pocket

Bearings with pumping action

As in the case of all bearings with an asymmetrical cross-section, tapered roller bearings have a pumping action, *Figure 28*. This pumping action, which is heavily dependent on the circumferential speed, can be utilised in recirculating oil lubrication. The outlet holes should be designed such that no oil back-up occurs adjacent to the bearing.

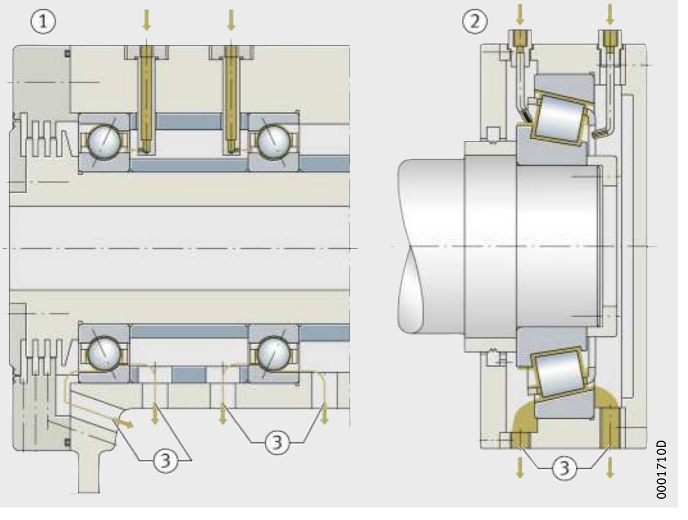


① Oil feed via collector pocket
② Outlet hole

Figure 28
Increasing oil recirculation

Oil injection lubrication

In oil injection lubrication, the oil is injected between the cage and inner ring, *Figure 29*. Oil back-up ahead of and after the bearings is prevented by oil outlet ducts. If the bearings have a pumping action, injection is carried out on the side with the smaller raceway diameter. In the case of tapered roller bearings rotating at very high speeds, injection is additionally carried out to the roller end faces on the other side. This counteracts undersupply of lubricant between the rib and end faces of the rollers.



① Angular contact ball bearing
② Tapered roller bearing
③ Outlet holes

Figure 29
Oil injection lubrication

Drip feed oil lubrication

Drip feed oil lubrication can be used on bearings running at high speeds, *Figure 30*. The oil quantity required is dependent on the bearing size, the bearing type, the speed and the load. The guide value for the oil quantity is between 3 drops/min and 50 drops/min for each rolling element raceway (one drop weighs approx. 0,025 g).

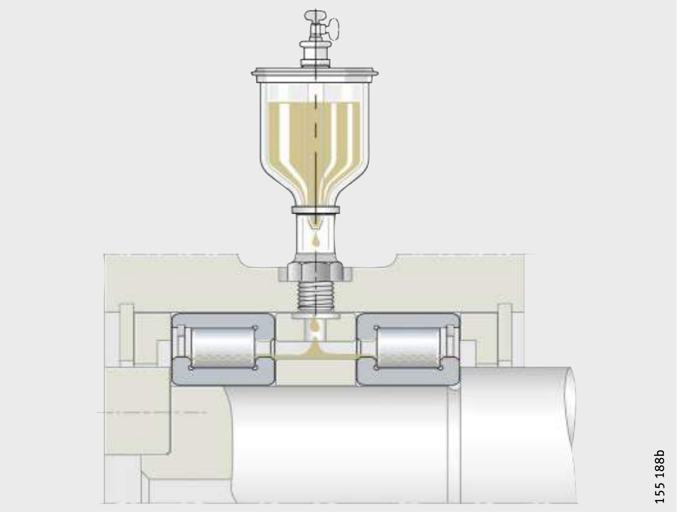


Figure 30
Drip feed oil lubrication

The lines carrying lubricant:

- should lead directly to the lubrication hole of the rolling bearing
- should be as short as possible
- should be provided for each bearing (individual line)
- must be filled (lines should be bled if necessary)
- must be designed taking account of the guidelines available from the lubrication system manufacturer.

The supply of lubricant to bearings

Greases

Desig- nation	Classification	Type of grease
GA01	Ball bearing grease for T < +180 °C	Polycarbamide Ester oil
GA02	Ball bearing grease for T < +160 °C	Polycarbamide SHC
GA13	Standard ball bearing and insert bearing grease for D > 62 mm	Lithium soap Mineral oil
GA14	Low-noise ball bearing grease for D ≦62 mm	Lithium soap Mineral oil
GA15	Low-noise ball bearing grease for high speeds	Lithium soap Ester oil
GA22	Free-running grease with low frictional torque	Lithium soap Ester oil
L014	Initial greasing for insert bearings for low temperatures	Gel Ester oil
L086	Initial greasing for insert bearings for wide temperature range and low loads	Sodium complex soap Silicone oil
L069	Insert bearing grease for wide temperature range	Polycarbamide Ester oil
GA08	Grease for line contact	Lithium complex soap Mineral oil
GA26	Standard grease for drawn cup roller clutches	Calcium/lithium soap Mineral oil
GA28	Screw drive bearing grease	Lithium soap Ester oil
GA11	Rolling bearing grease resistant to media for temperatures up to +250 °C	PTFE Alkoxyfluoroether
GA47	Rolling bearing grease resistant to media for temperatures up to +140 °C	Barium complex soap Mineral oil

[1] GA.. stands for **G**rease **A**pplication Group.., based on Grease Spec 00.

[2] The upper continuous limit temperature $T_{upperlimit}$ must not be exceeded if a temperature-induced reduction in grease operating life is to be avoided.

[3] Dependent on bearing type.

[4] Operating temperature range determined not according to DIN 51 825 but to MIL specification.

Operating temperature range °C	Upper continuous limit temperature $T_{upperlimit}$ 2) °C	NLGI grade	Speed parameter $n \cdot d_M$ $min^{-1} \cdot mm$	ISO VG grade (base oil) 3)	Designation	Recommended Arcanol grease for relubrication
–40 to +180	+115	2 to 3	600 000	68 to 220	**GA01**	–
–40 to +160	+85	2 to 3	500 000	68 to 220	**GA02**	–
–30 to +140	+75	3	500 000	68 to 150	**GA13**	**MULTI3**
–30 to +140	+75	2	500 000	68 to 150	**GA14**	**MULTI2**
–50 to +150	+70	2 to 3	1 000 000	22 to 32	**GA15**	–
–50 to +120	+70	2	1 000 000	10 to 22	**GA22**	–
–54 to +204 4)	+80	1 to 2	900 000	22 to 46	**L014**	–
–40 to +180	+115	3	150 000	68 to 150	**L086**	–
–40 to +180	+120	2	700 000	68 to 220	**L069**	–
–30 to +140	+95	2 to 3	500 000	150 to 320	**GA08**	**LOAD150**
–20 to +80	+60	2	500 000	10 to 22	**GA26**	–
–30 to +160	+110	2	600 000	15 to 100	**GA28**	**MULTITOP**
–40 to +250	+180	2	300 000	460 to 680	**GA11**	**TEMP200**
–20 to +140	+70	1 to 2	350 000	150 to 320	**GA47**	–

The supply of lubricant to bearings

Arcanol rolling bearing greases

Arcanol grease	Designation to DIN 51825	Classification
MULTI2	K2N-30	Low-noise ball bearing grease for D $\leqq$ 62 mm
MULTI3	K3N-30	Standard ball bearing/insert bearing grease for D $>$ 62 mm
SPEED2,6	KE3K-50	Standard spindle bearing grease
MULTITOP	KP2N-40	Universal high performance grease
TEMP90	KP2P-40	Low-noise rolling bearing grease, up to 160 °C
TEMP110	KE2P-40	Universal grease for higher temperatures
TEMP120	KPHC2R-30	Grease for high temperatures and high loads
TEMP200	KFK2U-40	Rolling bearing grease for T $>$ 150 °C to 250 °C
LOAD150	KP2N-20	Multi-purpose grease for automotive applications, high performance grease for line contact
LOAD220	KP2N-20	Heavy duty grease, wide speed range
LOAD400	KP2N-20	Grease for high loads, shocks
LOAD460	KP1K-30	Grease for high loads, vibrations, low temperatures
LOAD1000	KP2N-20	Grease for high loads, shocks, large bearings
FOOD2	KPF2K-30	Grease with foodstuffs approval
VIB3	KP3N-30	Grease for oscillating motion
BIO2	KPE2K-30	Grease with rapid biodegradability
CLEAN-M	KE2S-40	Clean room grease, grease resistant to radiation
MOTION2	–	High performance grease paste for oscillating applications and plain bearing arrangements

[1] With EP additive.

Type of grease Thickener Base oil	Operating temperature range °C	Upper continuous limit temperature $T_{upperlimit}$ °C	NLGI grade	Speed parameter $n \cdot d_M$ $min^{-1} \cdot mm$	Kinematic viscosity	
					at 40 °C mm^2/s	at 100 °C mm^2/s
Lithium soap Mineral oil	−30 to +140	+75	2	500 000	100	10
Lithium soap Mineral oil	−30 to +140	+75	3	500 000	80	8
Polycarbamide PAO + ester oil	−50 to +120	+80	2, 3	2 000 000	22	5
Lithium soap Mineral oil + ester oil[1)]	−40 to +150	+80	2	800 000	85	12,5
Calcium soap + polycarbamide PAO[1)]	−40 to +160	+90	2	500 000	130	15,5
Lithium complex soap Ester oil	−40 to +160	+110	2	600 000	150	19,8
Polycarbamide PAO + ester oil[1)]	−35 to +180	+120	2	300 000	460	40
PTFE Alkoxyfluoroether	−40 to +260	+200	2	300 000	400	35
Lithium complex soap Mineral oil	−20 to +140	+90	2	500 000	160	15,5
Lithium/calcium soap[1)] Mineral oil	−20 to +140	+80	2	500 000	220	16
Lithium/calcium soap[1)] Mineral oil	−25 to +140	+80	2	400 000	400	28
Lithium/calcium soap[1)] Mineral oil	−30 to +130	+80	1	400 000	400	25
Lithium/calcium soap[1)] Mineral oil	−20 to +140	+80	2	300 000	1 000	42
Aluminium complex soap White oil	−30 to +120	+70	2	500 000	192	17,5
Lithium complex soap Mineral oil	−30 to +150	+90	3	350 000	170	13,5
Lithium/calcium soap[1)] Ester oil	−30 to +120	+80	2	300 000	58	10
Polycarbamide Ether	−40 to +200	–	2	–	103	–
Solid lubricants Synthetic	−45 to +110	–	2	–	130	–

Miscibility of lubricants

Miscibility of greases and oils

Care must be taken when mixing different lubricants On the one hand, oils or the base oils of greases and their thickeners may not be compatible, see tables. On the other hand, the effect of additives and the performance capability of lubricant mixtures cannot be estimated without appropriate testing.

Mixing of greases should be avoided. If this is not possible, it is recommended that the substances to be mixed have:

- the same base oil type
- a compatible thickener type
- similar base oil viscosities (the difference must be no more than one ISO VG grade)
- the same consistency (NLGI grade).

Miscibility of base oils

	Mineral oil	PAO	Ester oil	Polyglycol oil	Silicone oil	Alkoxyfluoro oil
Mineral oil	+	+	+	–	o	–
PAO	+	+	+	–	o	–
Ester oil	+	+	+	o	–	–
Polyglycol oil	–	–	o	+	–	–
Silicone oil	o	o	–	–	+	–
Alkoxyfluoro oil	–	–	–	–	–	+

Compatibility of different thickener types

	Lithium soap	Lithium complex	Sodium complex	Calcium complex	Aluminium complex
Lithium soap	+	+	–	+	–
Lithium complex	+	+	o	+	o
Sodium complex	–	o	+	o	o
Calcium complex	+	+	o	+	o
Aluminium complex	–	o	o	o	+
Barium complex	+	o	o	o	o
Bentonite	–	–	–	0	–
Polycarbamide	–	o	o	o	–
PTFE	+	+	+	+	+

Definition of the symbols:

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

Compatibility of different thickener types (continued)

	Barium complex	Bentonite	Polycarba-mide	PTFE
Lithium soap	+	–	–	+
Lithium complex	o	–	o	+
Sodium complex	o	–	o	+
Calcium complex	o	o	o	+
Aluminium complex	o	–	–	+
Barium complex	+	+	o	+
Bentonite	+	+	–	+
Polycarbamide	o	–	+	+
PTFE	+	+	+	+

Definition of the symbols:
+ Mixing generally non-critical
o Miscible in individual cases, but checking should be carried out
– Mixing not permissible

Before mixing, the lubricant manufacturer must always be consulted. Even if the preconditions are fulfilled, the performance capability of the mixed grease may be impaired. Relubrication should only be carried out using greases of comparable performance capability. If a different grease grade is to be used, the previous grease must first be flushed out as far as this is permitted by the design. Further relubrication should be carried out after a shortened period. If incompatible greases are mixed, this can leading to considerable structural changes. Substantial softening of the grease mixture may also occur.

Checking of miscibility

Definite statements on miscibility and compatibility can only be obtained by means of suitable tests, *Figure 1*. The cost of the testing required is relatively high. Alternatively, tests can be carried out on the basis of specific applications.

DH = return strokes in accordance with DIN ISO 2137

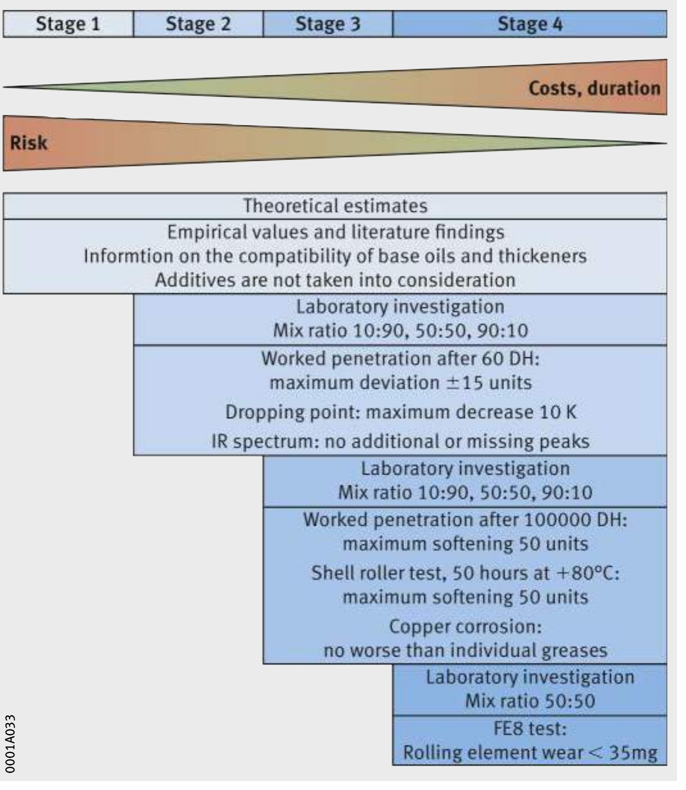


Figure 1
Checking of the miscibility of two greases

Lever grease gun

In difficult operating conditions or aggressive environments, rolling bearings must be frequently relubricated via lubrication nipples. This can be carried out easily, cleanly and quickly using lever grease guns.

The devices available from Schaeffler conform to DIN 1283 and the gun can filled either with loose grease or using a cartridge in accordance with DIN 1284.

Motion Guard

When automatic lubricators are used for controlled relubrication, a sufficient quantity of fresh grease is continuously supplied to the contact points of the rolling bearing. This extends the lubrication and maintenance intervals and shortens the downtime of the plant.

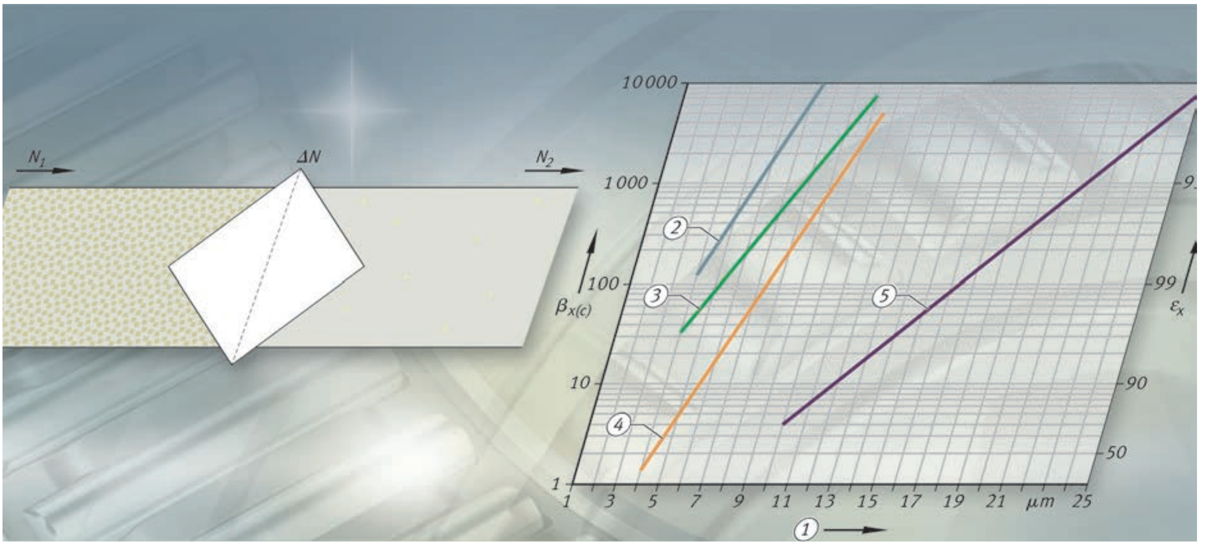
There are three series of lubricator:

- Motion Guard COMPACT
  - Single-point lubrication system comprising an activation screw and housing, filled with 120 cm^3 grease
- Motion Guard CHAMPION
  - Single-point lubrication system comprising an LC unit (Lubricant Cartridge) with a volume of 120 cm^3 or 250 cm^3 and a battery set
- Motion Guard CONCEPT 6
  - Single-point lubrication system with distributor, expandable to multi-point lubrication system, cartridge with volume of 250 cm^3 or 500 cm^3.

Condition monitoring

Condition monitoring on the basis of vibrations is currently the most reliable method for detecting damage at an early stage.
A distinction is made here between offline and online monitoring. In offline monitoring, machinery is examined by vibration analysis at regular intervals, for example every four weeks. In online monitoring, the machine condition is subject to continuous monitoring.
Both methods work on the basis of signals and facilitate assessment of the condition of equipment and components.
Imbalance and misalignment defects can be detected accurately, as well as rolling bearing damage and gear tooth defects. Depending on the priority and location of the machinery, the operator must decide which method of condition monitoring is most suitable for his requirements.

In the field of condition monitoring, Schaeffler Technologies AG & Co. KG offers a comprehensive product portfolio, ranging from simple vibration monitors to complex monitoring systems for a large number of measurement points. Vibration measuring devices help to detect incipient damage to rotating components at an early stage. As a result, unplanned downtime can be prevented and maintenance costs can be reduced. As necessary, the Business Division Industrial Aftermarket (IAM) of Schaeffler can advise on the selection of suitable monitoring methods. Further product information is given in Catalogue IS 1.



Contaminants in the lubricant

Contaminants in the lubricant

Contaminants in the lubricant

Contaminants in the lubricant

In practice, there are hardly any lubrication systems that are completely free of contaminants. Contaminants that are common to applications are already taken into consideration in determining the fatigue life and operating life, since the calculation methods are based on results from practice and tests. If a level of contamination higher than in the normal application is unavoidable, this will lead to reduced running times or premature failures. If the level of cleanliness is particularly good, however, longer running times can be achieved.

Due to the production processes, all lubricants already contain a certain proportion of contaminants. The minimum requirements for lubricants defined in DIN standards include limit values for permissible contamination in production. In the delivered condition, lubricants contain additional contaminants from the containers.
At initial mounting, contaminants often also enter the bearing due to inadequate cleaning of machine parts and oil lines.
During operation, the bearing may become contaminated due to inadequate sealing as a result of open points in the lubrication system (oil container, pump). Contaminants may also enter the bearing during maintenance, for example due to contamination on the lubrication nipple or on the nozzle of the grease gun as well as during greasing by hand.

When determining the harmful influence of contaminants, the following are particularly important for all lubricants:

- the type and hardness of foreign matter
- the concentration of foreign matter in the lubricant
- the particle size of the foreign matter.

Solid foreign matter

Solid foreign matter leads to wear and premature fatigue. The higher the hardness of the overrolled particles (for example iron swarf, grinding swarf, moulding sand, corundum) and the smaller the bearings, the greater the reduction in the life. A correlation between the indentation diameter and the relative life can be derived on the basis of artificially created indentations, *Figure 1*, page 139.

For information on taking account of hard contaminants as an influence on reduced life, see section Load carrying capacity and rating life, page 18.

Hard particles cause abrasive wear in rolling bearings, especially at points with high proportions of sliding motion. This occurs, for example, in the contact area of the roller end face and rib in tapered roller bearings or at the raceway ends of rollers in axial cylindrical roller bearings. The wear increases as the hardness of the particles increases. It also increases in an approximately proportional manner with the concentration of the particles in the lubricant and the particle size. Wear also occurs with extremely small particles. The permissible size depends on the particular application.

Artificially created indentation

L_{rel} = relative life

① Length of contact ellipse
② Indentation diameter = 0,3 mm
③ Indentation diameter = 0,1 mm
④ No indentation

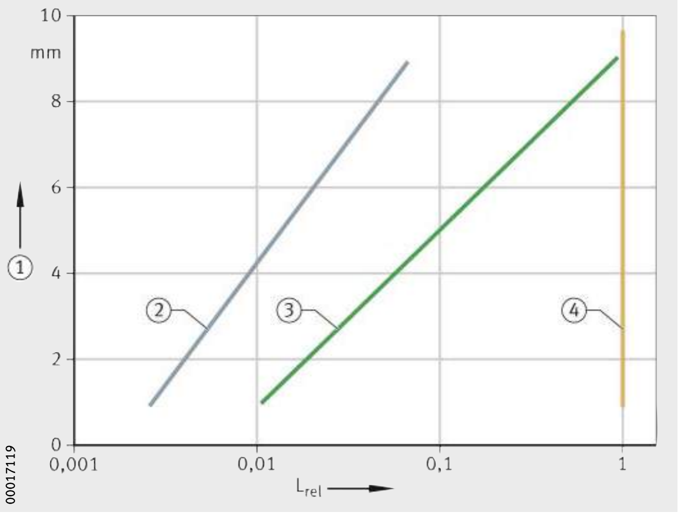


Figure 1
Influence
of indentation diameter on life

Reduction in the concentration of foreign matter

The concentration of foreign matter is reduced by:

- clean lubricants
- effective sealing
- thorough cleaning of parts adjacent to the bearing
  - cleanliness during mounting
  - cleaning during oil lubrication prior to commissioning
  - filtration of the oil through filters of appropriate mesh size
  - sufficiently short grease change intervals.

Classification of contaminants in accordance with ISO 4408:1999 can be used to define the degree of contamination and the oil cleanliness code, see table, page 140. The filtration values are matched to the requirements for the oil cleanliness code.

Contaminants in the lubricant

Classification of contaminants in accordance with ISO 4406

Number of particles per 100 ml						ISO code		
> 4 μm		> 6 μm		> 14 μm				
over	incl.	over	incl.	over	incl.			
4 000 000	8 000 000	500 000	1 000 000	64 000	130 000	23	20	17
2 000 000	4 000 000	250 000	500 000	32 000	64 000	22	19	16
1 000 000	2 000 000	130 000	250 000	16 000	32 000	21	18	15
500 000	1 000 000	64 000	130 000	8 000	16 000	20	17	14
250 000	500 000	32 000	64 000	4 000	8 000	19	16	13
130 000	250 000	16 000	32 000	2 000	4 000	18	15	12
64 000	130 000	8 000	16 000	1 000	2 000	17	14	11
32 000	64 000	4 000	8 000	500	1 000	16	13	10
16 000	32 000	2 000	4 000	250	500	15	12	9
8 000	16 000	1 000	2 000	130	250	14	11	8
4 000	8 000	500	1 000	64	130	13	10	7
2 000	4 000	250	500	32	64	12	9	6
1 000	2 000	130	250	16	32	11	8	5
500	1 000	64	130	8	16	10	7	4

Consumption lubrication systems

Consumption lubrication systems must be equipped with a suction filter and a pressure filter. The pressure filter must be located directly after the lubricant pump. If the bearing is additionally lubricated by spray lubrication, filtration and dewatering must also be provided for the air. The mesh size of the air filter should be approx. 5 μm.

Recirculating lubrication systems

Recirculating lubrications systems have a suction and pressure filter in the main or ancillary flow or in both flows. In addition, they are equipped with a filter for the return flow. The design of the oil container size also has an influence on the degree of contamination in the oil to be pumped. A large minimum oil quantity prevents oil being sucked or stirred up from the base of the container, since contamination collects on the base. Furthermore, a large container influences the cooling rate of the oil and thus possible water condensation, see section Liquid contaminants, page 143.

Filtration values

In design of the filters, not only the mesh size but also the filtration ratio $\beta_{x(c)}$ in accordance with ISO 16889 must be taken into consideration. This indicates the proportion of particles that are retained, *Figure 2*.

N_1 = number of particles ahead of the filter
N_2 = number of particles after the filter
ΔN = number of particles remaining in the filter

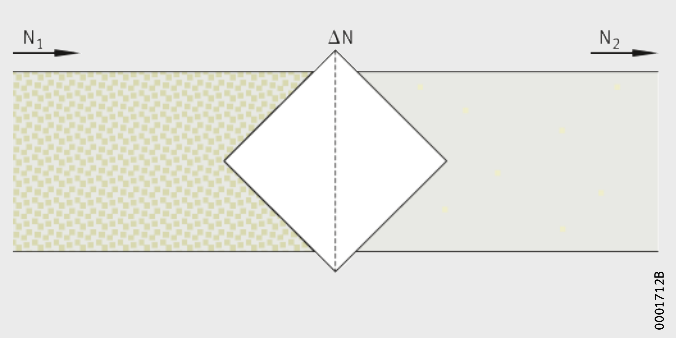


Figure 2
Filtration ratio

Filtration ratio

The filtration ratio $_{x(c)}$ can be determined using the following formula:

$$\beta_{x(c)} = \frac{N_1}{N_2}$$

Separation efficiency ε_x

The separation efficiency ε_x can be determined using the following formula:

$$\varepsilon_x = -\frac{\Delta N}{N_1} = -\frac{\beta_{x(c)} - 1}{\beta_{x(c)}}$$

$\beta_{x(c)}$ –
Filtration ratio

$\mathrm{Index}_{(c)}$ –
Values determined or measured in accordance with ISO 16889

N_1, N_2 –
Number of particles ahead of and after the filter

ε_x %
Separation efficiency

ΔN –
Number of particles remaining in the filter $\Delta N = N_1 - N_2$.

Filtration ratio and separation efficiency

Filtration ratio $\beta_{x(c)}$	Separation efficiency ε_x %
1	0
2	50
10	90
5	98,67
100	99
200	99,5
1 000	99,9
10 000	99,99

Hydraulic filters with glass filter elements achieve, in accordance with ISO 16889, filtration ratios $\beta_{x(c)}$ of more than 1 000.
This corresponds to a separation efficiency of 99,9%.

! The $\beta_{x(c)}$ value (separation efficiency 99,5%) should not be less than 200, since very good filtration will increase the operating life of the bearing. At the same time, attention must however be paid to the price/performance ratio, since the costs of components (pump and filter) are higher in the case of very good filtration. Monitoring of filters must not be omitted under any circumstances. This prevents the entire quantity of contamination entering the line system and the bearings if the filter is destroyed.

The filter size determines the number of particles (filtration ratio, separation efficiency) as a function of the particle size, *Figure 3*.

$\beta_{x(c)}$ = filtration ratio
ε_x = separation efficiency

① Particle size
② Filter size H3SL
③ Filter size H6SL
④ Filter size H10SL
⑤ Filter size H20SL

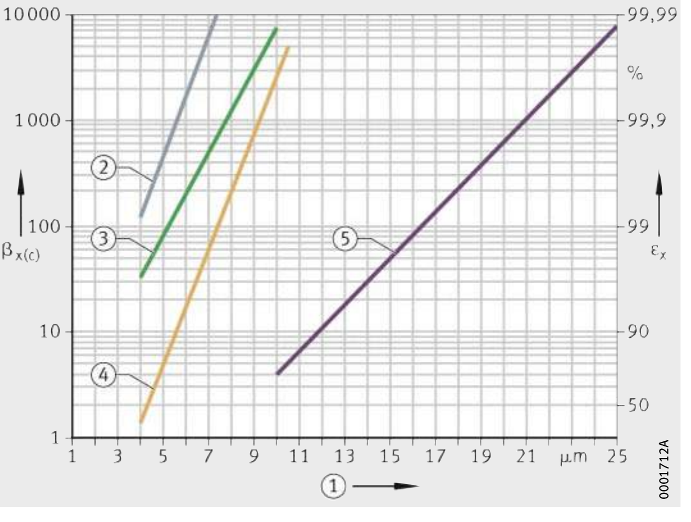


Figure 3
Number of particles and particle size according to ISO 4406

Liquid contaminants

The harmful effect of liquid contaminants in the lubricant is often seriously underestimated. Even pure water without additional aggressive media has very high potential for damage in rolling bearings.

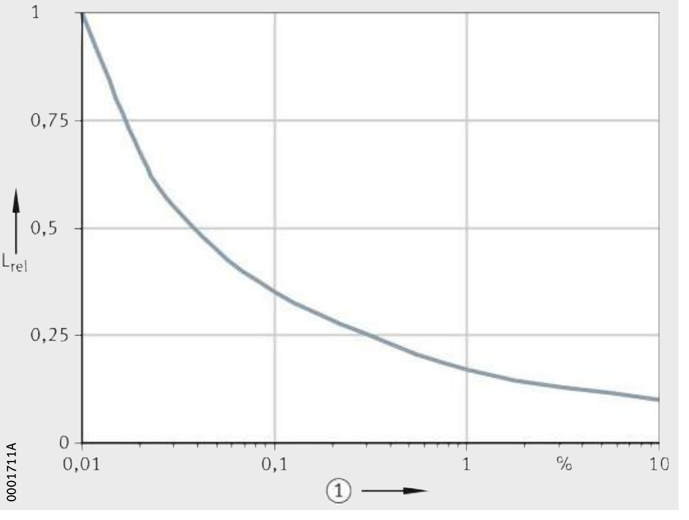
The potential for damage is divided into the following categories:

- reduction in the fatigue running time
- cause of wear
- acceleration of lubricant ageing and formation of residues
- corrosion.

The damage mechanisms occur individually or in combination and are dependent on the lubricant type, bearing material and the free quantity of wear carried in with the lubricant. They can lead to functional incapacity or can completely destroy the bearing.

The influence of water in oils

With an increasing water content, the relative life decreases, *Figure 4*.



Schematic representation

L_{rel} = relative life

1 Water content

Figure 4
The influence of water on the relative life

Contaminants in the lubricant

Depending on the composition of the lubricant and the bearing type, a detectable impairment of the fatigue running time can be expected with a water content of or greater than approx. 200 ppm. Due to the mixing effect, the viscosity of the lubricant is reduced. The actual cause of damage is, however, the occasional overrolling of water microdroplets occurring under high pressure and the associated punctures in the lubricant film. With an increasing water content, the number of microdroplets increases and, starting at a certain concentration, the size of the microdroplets also increases.
As a result, the probability of overrolling increases. In parallel, a chemical reaction occurs between the water, lubricant and material that additionally impairs the surface.

Starting at approximately the same concentration of approx. 300 ppm, there is a rapid increase in the tendency of oils at high temperatures to form residues in the form of sludge, paint or coking. The ageing of the base oils is accelerated and additives and water are precipitated or their effect is blocked. In addition to the disruption by deposits of the distributor, feed and outlet systems and the blocking of filter elements, the lubrication capability itself is reduced.

At water contents over 1 000 ppm, a different damage mechanism comes into play, depending on the oil composition, prior to fatigue. Even before failure due to material ruptures, the bearing functional surfaces undergo wear. The wear progresses, depending on the strain, without negative effects until the function of the bearings is finally disrupted. However, rapid surface spalling and destruction of the bearings is also possible. The bearing type has a significant influence here. High contact pressures and high proportions of sliding motion will promote damage. A even more severe influence is exerted by the oil type and the additives. The scatter of the tolerable water quantity can, depending on the lubricant, be up to a power of ten.

If free water is present, there is an increased risk of corrosion. If bearings are at a standstill, water infiltrates the bearing surfaces protected by the anti-corrosion additives. This effect is supported by the capillary action in the narrowing gap between the rolling element and raceway and occurs there first. This leads to harmful rust formation on the materials, which are not normally corrosion-resistant. If these corrosion scars are overrolled, early fatigue will occur. The bearing surfaces are completely destroyed if the free water is not removed. These damage mechanisms occur if water is continuously present in the stated order of magnitude. Water content present for part of the time also has high potential for damage but this is difficult to quantify. Water vapourises from lubricant at low temperatures. When water enters and exits continuously due to cooling and heating, this causes considerable damage to the oil and also has effects on the rolling bearings. This is the case, for example, when condensation is formed in oil containers during operational shutdown and in vapourisation at operating temperature.

The influence of water in greases

In grease, water causes structural changes depending on the thickener type. There is a risk that the greases will undergo considerable softening. The damage mechanisms are comparable with those in oils. Greases have the advantage that contaminated lubricant does not necessarily enter the contact and does not flow in when water is vapourised. If there is ingress of water, the grease change interval must be shortened in accordance with the quantity of water present. The action of the grease in supporting sealing is applied in labyrinth lubrication. Aggressive substances such as acids, bases or solvents lead to major changes in the chemical/physical key data and principally to lubricant ageing and corrosion. If such contaminants are expected, the compatibility data from lubricant manufacturers must be considered. At points that are not protected from the lubricant, corrosion will occur sooner or later depending on the aggressiveness of the contaminant and destroy the surface.

Contaminants in the lubricant

Gaseous contaminants

Oils can, depending on the base oil type, dissolve considerable quantities of gases (in general air).

Dissolved air in oil

The determining parameters are principally pressure and temperature. The degree of refinement, viscosity and additives exert only an subordinate influence. In principle, the law formulated by Henry/Dalton applies to dissolved gases: under normal conditions (+20 °C, 1013 mbar) mineral oils can dissolve 7 vol. % to 9 vol. % of air. This corresponds to approx. 1% to 2% of oxygen in the oil. The solubility of air in oil increases correspondingly with increasing pressure.

The method in accordance with ASTM D 2779 facilitates calculation of the solubility of various gases in mineral oil products. The method in accordance with ASTM D 3827 contains a calculation method that is also valid for synthetic oils.

Finely distributed air in oil

In addition to dissolved air, oils can also contain finely distributed air quantities (dispersed phase).

The problems in technical plant as a result of these air-in-oil dispersions include:

- cavitation damage
- increase in temperature due to impaired thermal conductivity capacity and lower oil flow
- more rapid oil ageing due to oxidation and cracking (splitting of hydrocarbon molecules)
- wear of parts under high strain due to smaller oil film thickness
- blocking of filters.

Dispersed air in oil leads to an increase, even if a small increase, in viscosity. As a guide value, 10 vol. % air in oil leads to an increase in viscosity of approx. 15%.

Air release

The air release capacity is determined in accordance with DIN 51381. Under defined test conditions, the time is measured in minutes that is required for the release of air bubbles to a proportion of less than 0,2 vol. %. In practice, the air release capacity can be improved by low recirculation rates and thus long dwell times in the oil storage container. Special design of the inlets in the oil tank and appropriate guide plates allow more rapid escape of small gas bubbles.

The air release capacity (LAV) of mineral oils is essentially determined by:

- oil viscosity
  - the higher the viscosity, the worse the LAV
- oil temperature
  - the LAV improves with increasing temperature
- the presence of additives
  - additives that reduce the surface tension of the oil will reduce the LAV (ageing products)
- solid and liquid contaminants.

The influence of small gas bubbles in oil on lubricant film formation has not yet been researched adequately. For example, there are no precise findings on the size up to which gas bubbles are overrolled in rolling contacts and metal-to-metal contact then occurs. Theoretical analyses show that passage through the lubrication gap can be substantially ruled out. This is due to the size of the gas bubbles.

Foaming behaviour

If gas (air) and oil are actively mixed, a substantially stable foam can form on the surface. If the foam breaks down in a sufficiently short time, hardly any problems will occur. If stable foaming occurs, there may be delivery problems in oil pumps. An oil foam is also strongly compressible. Great care must be taken in the use of foam inhibitors, since the introduction of these additives impairs the air release capacity. Careful handling of the oil (filtration, degassing, water separation, cooling) and the selected of appropriate oils will help to avoid problems in practice. This applies particularly in equipment with a relatively large oil volume such as paper machinery or wind turbines.

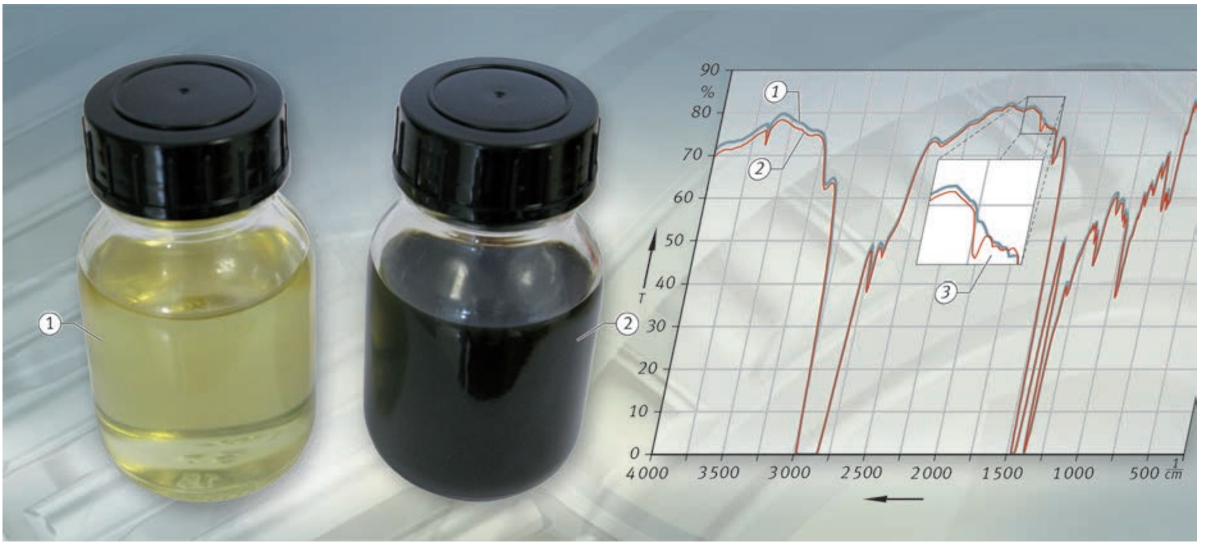
Cleaning of contaminated bearings

All parts that are removed from undamaged original packaging are very clean and do not require cleaning. Cleaning in this case would very probably impair the original condition. Parts that have become contaminated as a result of environmental influences can be cleaned using petroleum ether, petroleum, spirit, dewatering fluids, aqueous, neutral and also alkaline cleaning agents. It must be noted that petroleum, petroleum ether, spirit and dewatering fluids are flammable and alkaline agents are corrosive. The washing process should be carried out using brushes, paint brushes or lint-free cloths.

After washing, the parts must be:

- additionally cleaned using a very clean rinsing medium appropriate to the washing chemicals
- subsequently dried
- protected immediately using preservation in order to prevent corrosion.

Compatibility of the preservation with the lubricant used must be observed. If the bearings contain resinous oil or grease residues, precleaning by mechanical means followed by longer softening with an aqueous, strongly alkaline cleaning agent is recommended.



Lubricant testing

Lubricant testing

Sensory and analytical testing

The condition of a sample of used lubricant can be assessed using sensory and analytical lubricant tests. Correct sampling is always a precondition here. In sensory lubricant testing, the condition is determined on the basis of the optical appearance of the lubricant sample. Analytical methods are comparative methods, in which the corresponding data for an unused reference sample must always be known. On the basis of deviations from this reference, conclusions can be drawn regarding the condition of the used sample. The values determined always represent the condition of the used sample at that time.

Sensory lubricant testing

Sensory lubricant testing is defined as simple testing of the sample in relation to colour, odour, lubrication effect and, in the case of greases, the consistency of the sample. The optical appearance gives initial guide values on the condition of the sample.

Odour and colour

A pungent odour from a used sample can indicate the presence of ageing products and therefore ageining of the lubricant.

Lubricants age change in colour due to use. The darkening of a sample can indicate, for example, thermal influences or contaminants or, in the case of engine oils, soot for example, *Figure 1*.

If an oil is cloudy, this may in turn indicate the ingress of water. However, changes in colour may occur as a result of short term operation or storage of the sample in air or with exposure to light. This is not critical initially. For a uniform description of the colour tone, the RAL colour system is suitable.

① Fresh engine oil
② Used engine oil

Figure 1
Change in colour of an engine oil

Lubrication effect and consistency

Once some experience has been gained, it can be determined by rubbing a sample between two fingers whether it essentially has a lubrication effect. In addition, it is often possible to detect solid lubricants in this manner.

Greases that have been subjected to heavy use or are heavily contaminated will frequently exhibit higher consistency (resistance). This can be explained by a reduced base oil content (for example, through heavy oil loss during operation) or the occurrence of ageing products. Sensory testing can give initial guide values here.

In general, it is not possible to make a reliable statement on the condition of the sample on the basis of sensory testing alone. This can only be achieved by means of suitable analytical methods in the laboratory. Furthermore, sensory testing is subjective and thus dependent on the observer. The results may therefore vary from one observer to another.

Analytical lubricant testing

A large number of analytical test methods are available. Selecting which methods are used is based on the relevant application of the sample or on the direction of the investigation, for example towards symptoms of ageing or wear. In many cases, only a limited sample quantity will be available, however, as a result of which it will be necessary to prioritise the test methods.

Element content

The element content of a lubricant sample can be determined, for example, by means of optical ICP emission spectrometry (ICP-OES). ICP-OES stands for Inductively Coupled Plasma Optical Emission Spectrometry. The sample is first digested in a suitable solvent by means of microwaves in order to release the chemical compounds. The molecules of the sample are then broken up in an argon plasma and the atomic ions are excited in order to promote light emission. The wavelength emitted is used to identify the element. The intensity of the light emission also indicates the concentration of the relevant element. If the element content of a fresh sample is present as a reference for a lubricant sample, statements can be made about any changes. This includes, for example, the decomposition of certain additive elements or the contamination of the sample by inorganic substances.

A further method for determining elements is X-ray fluorescence analysis (RFA). In this non-destructive method, the atoms of a sample are excited by energy-intensive X-rays to emit fluorescent radiation. The energy (wavelength) of the fluorescent radiation from the sample is characteristic of the relevant atom, while the intensity of the radiation indicates the concentration.

Infrared spectroscopy

A lubricant is an organic substance with functional groups of differing structure. Irradiation using infrared light excites the molecules (or more precisely their bonds) to undergo oscillation, which leads to the absorption of energy. In transmitted light infrared spectroscopy, it is measured which part of the infrared light is absorbed by the sample. This gives a characteristic spectrum for the sample. Infrared spectroscopy can therefore be used with little associated work to give a statement on the structure of a lubricant covering, in the case of greases, the base oil and thickener type. It can normally be implemented using only a very small sample quantity, see table, page 156. If ageing products occur in the lubricant as result of use, these will form characteristic bands as appropriate in the infrared spectrum. Through comparison with the infrared spectrum of a corresponding fresh reference, it is possible to draw conclusions from any deviations on the condition of a sample. As a result, infrared spectroscopy is a powerful instrument in the chemical analysis of organic substances, *Figure 2*.

T = transmission
$\tilde{\nu}$ = wave number

① Fresh mineral oil
② Used mineral oil
③ Typical ageing

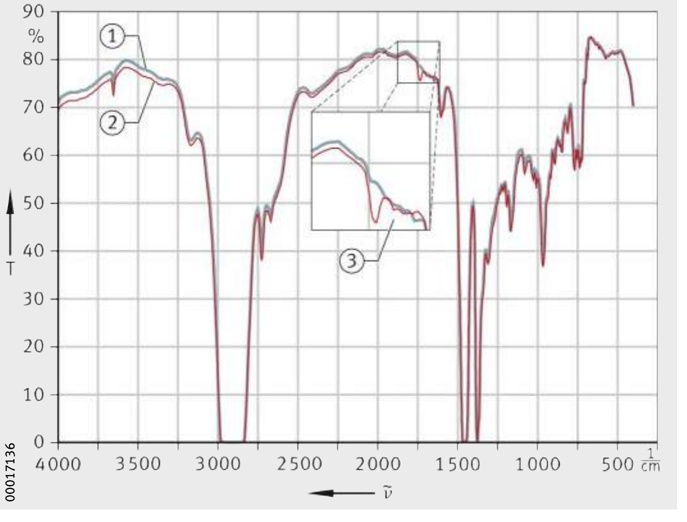


Figure 2
Infrared spectroscopy

Proportion of solids

The lubricant probe is digested using a suitable solvent in an ultrasonic bath and the solution is then filtered off. Filters of different pore sizes are available for this purpose. After drying, the filter is evaluated in both quantitative and qualitative (optical) respects. Detectable particles can, as necessary, be subjected to further investigation in order to determine the origin and effect of the particles. For example, infrared spectroscopy facilitates statements on organic chemistry and energy-dispersive X-ray spectroscopy (EDX) facilitates statements on inorganic chemistry.

Water content

Even a small ingress of water or moisture into the lubricant can cause severe damage through corrosion or breakdown of the lubricant film, see section Liquid contaminants, page 143.

The Karl-Fischer method allows the water content of a lubricant sample to be determined by means of titration. Comparison with the water content of a fresh sample gives a statement on any water ingress. It must be noted here that this investigation can only show the water content present at the time. Between the actual ingress of water, taking of the sample and the investigation, at least part of the water may have vapourised from the lubricant. This will be promoted by increased temperature, negative pressure or storage of the sample in an open vessel.

Viscosimetry

Oils are subjected to mechanical load during operation. As a result, the molecular chains may be broken, leading to a reduction in viscosity. The viscosity may also be reduced as a result of thinning with a low viscosity fluid such as petrol. On the other hand, the viscosity may be increased by ageing products or contaminants such as soot. A very common method for determining viscosity is Ubbelohde viscosimetry. The method uses the capillary action of the oil and gives a statement on the kinematic viscosity.
A further method is rotational viscosimetry, for example by means of a Stabinger viscosimeter. These devices determine the dynamic viscosity and density of a lubricant. The kinematic viscosity is calculated and also outputted. Both methods can be used for Newtonian fluids. For non-Newtonian fluids (structurally viscous fluids such as greases), rheology can give statements on the apparent viscosity. In the case of these substances, viscosity is a function of time, shear rate and temperature. For guide values on the minimum quantity required in lubricant testing, see table, page 156.

Minimum quantities for lubricant testing

Statement	Method	Minimum quantity required	
		Grease ≈g	Oil ≈ ml
Colour	RAL colour code	–	–
Lubrication effect	Finger test	–	Not applicable
Identification Ageing Contamination (qualitative, quantitative)	Infrared spectroscopy	0,1	5 (capillaries) 1 (window)
Water content	Karl- ischer titration (indirect method)	0,5 to 1	2
Ageing Strain Shear Thinning (for example with fuel)	Viscosimetry	Not applicable	6
Iron content or element content	Emission spectroscopy (ICP-OES)	0,1	15
	or X-ray fluorescence analysis (RFA)	Not applicable	10
Contaminant content (particles > 1 μm or > 11 μm depending on filter) with optical filter evaluation	Filtration	0,5	10
Material determination	Energy-dispersive X-ray analysis (EDX)	[1]	[1]
Consistency	Worked penetration	500	Not applicable

[1] Element analysis of filtrate or contaminant residue.

Mechanical-dynamic testing

In most cases, rolling bearings are components that are subjected to the highest mechanical and dynamic loads. The core functions of the relevant application are very often dependent on their reliable function. Continuous progress in technical development, economic optimisation and performance increase leads in the rolling bearing in particular to concentrations of load that must be withstood, not least by the lubricant as a design element.

Selecting or developing an optimum lubricant for the relevant application is only possible if such lubricants can be tested, under conditions matching practical use, in relation to characteristics that are decisive in terms of function.

In contrast to purely substance-based characteristics, which are determined in the classical chemistry laboratory, the functional behaviour of a lubricant in a rolling bearing can, so far, only be tested in the rolling bearing itself.

These mechanical-dynamic tests are not a simulation of the actual application. Instead, they are a model of individual functions of the lubricant, such as its anti-wear capacity, in the bearing. The tests are thus the basis for assessing the performance of oils and greases. They are carried out on different bearing types and under conditions that reflect the limits of the actual applications that are relevant to failure.

Accelerated test method, element tests, tribometer

Ideally, tests should allow rapid results and incur only low test costs.

For example, the following test methods are widely used:

- VKA (Shell four ball test machine)
- pin-on-disc test rig
- Almen-Wieland test machine
- SRV test machine
- two-disc test rig.

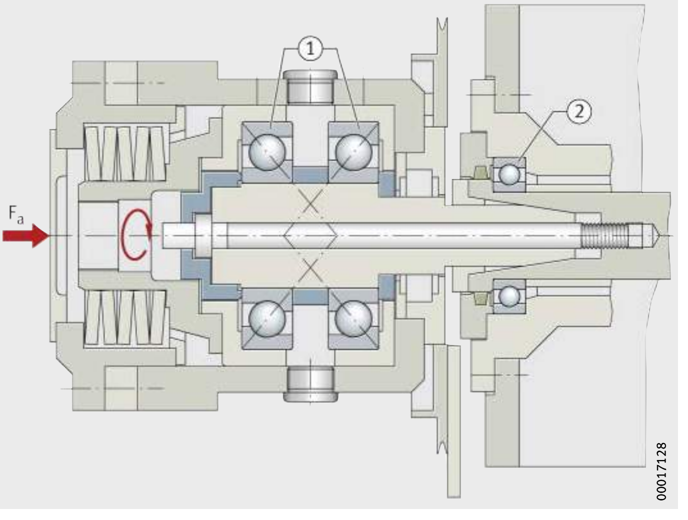
Some of these methods are used in the development of lubricants in order to allow rapid and economical determination of changes in characteristics between individual development models or development stages. The correlation of these test methods with actual applications is not, however, sufficient in order to assess the performance of lubricants in the rolling bearing. For performance tests on oils and greases in rolling bearings, see section Rolling bearing test devices, page 158 and section Special tests for specific applications, page 166.

Rolling bearing test devices

There are a large number of rolling bearing test devices whose function is to test bearings under defined conditions. The objective is always to select test conditions such that it is possible with the shortest possible test period to draw a conclusion as to the performance capability in the actual application. It must be ensured that the lubricant is not subjected to excessive load, which would falsify the result of the test. For the tribological assessment of lubricants in rolling bearings, some methods have proved particularly suitable and, in the case of FE8 and FE9 testing, have been standardised in accordance with DIN. These machines are used worldwide for lubricant assessment.

Test rig FE8

The FE8 test rig is used to determine the anti-wear behaviour of lubricants, *Figure 1*. It can be used to test oils and greases. It is designed such that bearings with either point contact or line contact can be used. It is possible to test angular contact ball bearings, tapered roller bearings and axial cylindrical roller bearings. The axial cylindrical roller bearing is only used for oil testing, while the other bearing types can be used in grease or oil tests. The test rig is standardised in accordance with DIN 51819. The test results are applied in various requirement standards, such as in the gearbox oil standard DIN 51517.



F_a = axial load

① Test bearing
② Ancillary bearing

Figure 1
Test rig FE8

Load range Depending on the requirements of the application, axial loads from 5 kN to 100 kN can be applied. The speed range extends from 7,5 min^{-1} to 4 500 min^{-1} (in the special version up to 6 000 min^{-1}). Not all load/speed duty cycles can be run in the test.
There are load/speed duty cycles that are appropriate and for which comparative results are presented. In this test method, a very wide range of lubrication regimes can be tested under the same conditions, from extreme mixed friction through moderate mixed friction to a full load-bearing lubricant film. With the axial cylindrical roller bearing, extreme mixed friction and sliding can be generated.

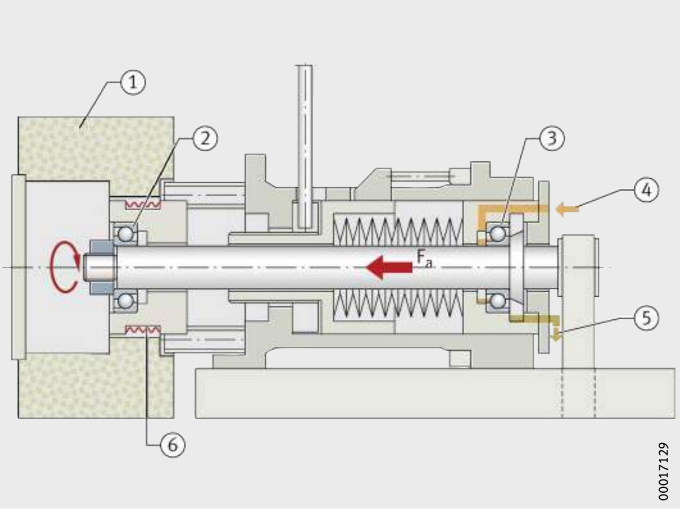
Special version A special version includes a preheating container. This addition makes it possible to conduct special tests for paper machinery and wind energy gearboxes. The oil is contaminated with distilled water or process water before it is fed into the preheating container, passes through the container in a sheet metal cascade device and then reaches the test head. Oil deposits may be formed in the tempered preheating container (depending on the type of testing, at +100 °C or +120 °C). In addition to the anti-wear protection by the oil, these deposits and the filterability of the oil are assessed.

Test rig FE9

The test rig FE9 is used to determine the high temperature suitability of greases for rolling bearings, *Figure 2*. In each case, an angular contact ball bearing is filled with a defined quantity of grease, subjected to axial load and run at a particular speed. Depending on the thickener and base oil, the test is carried out at +100 °C up to max. +250 °C. It is possible to carry out the test on open bearings (method A), but cases with sealing washers on both sides or with a grease reservoir can also be applied.

The test normally used, which is also standardised in DIN 51821-2, is A/1500/6000. In this case, testing is carried out on the open bearing with an axial load of 1 500 N at a speed of 6 000 min^{-1}.

Five tests can always be carried out simultaneously on the five test heads of the test rig. The running time until failure due to increased friction is determined. Based on the five failure times, a Weibull evaluation is used to determine the statistical test running times B10 and B50. Application standards such as DIN 51825 define the running times that must be achieved at particular temperatures. For example, DIN 51825 specifies that the minimum running time F50 (B50) must be greater than 100 hours. If this value is achieved, the test temperature can be stated as the upper operating temperature limit.



F_a = axial load

1 Insulation
2 Test bearing
3 Ancillary bearing
4 Oil inlet
5 Oil outlet
6 Heating system

Figure 2
Test head FE9

Test rig A2

This test rig is used to determine the anti-wear protection of oils, *Figure 3*. The design is similar to that of the test rig FE8. The main difference is that testing is carried out using oil sump lubrication. For this reason, only very small quantities of oil are required. The wear behaviour of the axial cylindrical roller bearing is also assessed. In addition, the result for adhesion capacity or transport behaviour of the oil is influenced since the rolling elements are only immersed in the oil sump and must therefore transport the oil over almost the entire circumference.

The test runs at a speed of 11 min^{-1} under an axial load of 51,5 kN.

Temperatures up to +160 °C are possible.

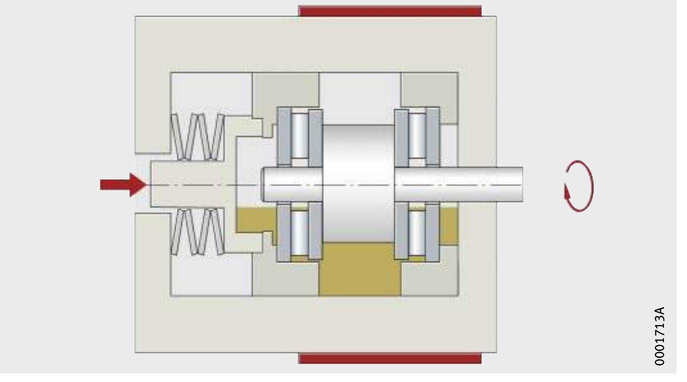
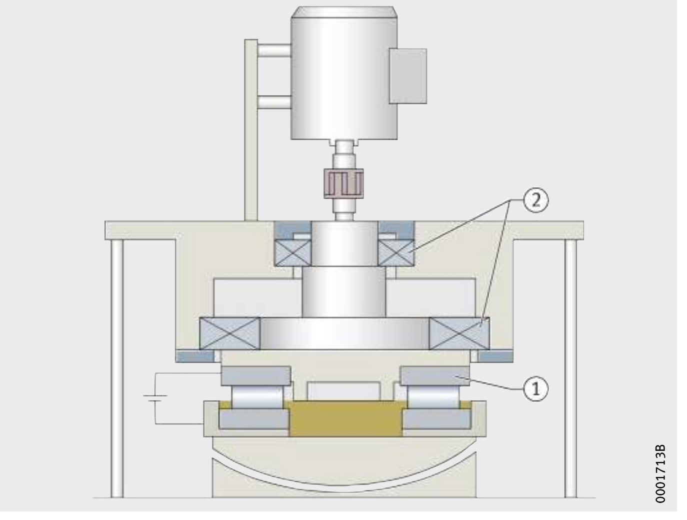


Figure 3
Test head A2

Test rig LFT

This test rig is used to assess the anti-wear protection of oils and lubricant film formation, *Figure 4*. The test bearing used is an axial cylindrical roller bearing with a plastic cage. Testing is carried out with a vertical shaft. The test bearing is immersed partially with the rolling elements in oil. The wear and contact stress are measured and these are used as indicators of lubricant film formation. Speeds between 10 min^{-1} and 4 000 min^{-1} are possible at an axial load of 0,5 kN to 100 kN.



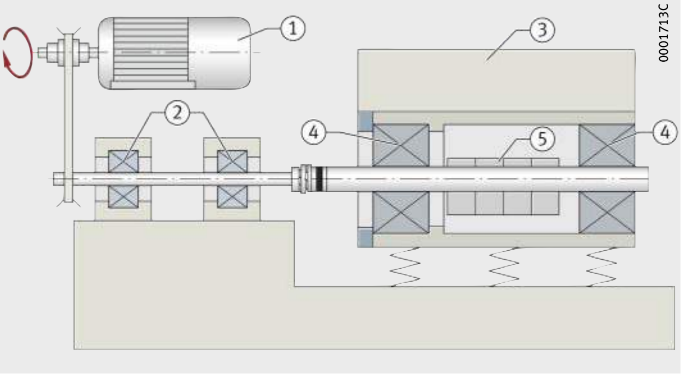
① Test bearing
② Ancillary bearing

Figure 4
Test rig LFT

Test rig AN42

Greases may undergo softening during operation. Greases at particular risk here include those in wheelset bearings for rail vehicles. Whenever shocks and high frequency vibrations act over an extended period act on a grease, softening of the grease must be expected. This is tested by Schaeffler under conditions close to practice in accordance with an internal company test method on a converted vibratory screen, *Figure 5*.

The shaft has an imbalance in order to generate vibrations. It is possible, at speeds from 1 000 min^{-1} to 2 000 min^{-1}, to achieve acceleration of up to 15 g. The change in consistency of the grease in the test bearing is measured after a running time of 96 hours.



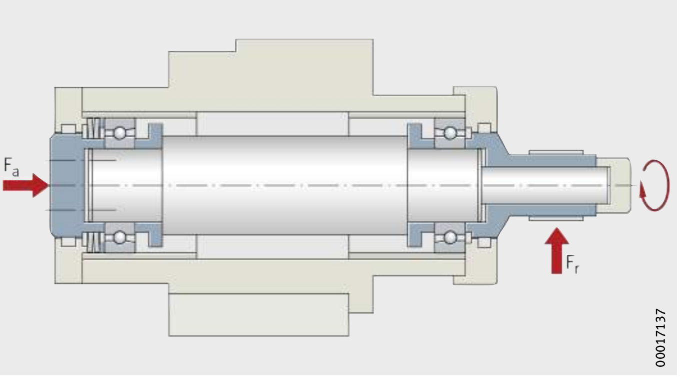
① Electric motor
② Ancillary bearing
③ Oscillating mass
④ Test bearing
⑤ Imbalance

Figure 5
Test rig AN42

Test rig WS22

Greases for high speeds place particular requirements on the thickener systems and base oils. The suitability of such greases is tested on specially developed high speed test rigs such as the WS22 (spindle test rig), *Figure 6*.

Two high precision spindle bearings under low axial and radial load rotate at speeds up to 60 000 min^{-1}. This corresponds to a speed parameter of 2 000 000 $min^{-1} \cdot mm$. This test is carried out with a rotating inner ring. The temperature on the stationary outer ring and the time to failure are determined.



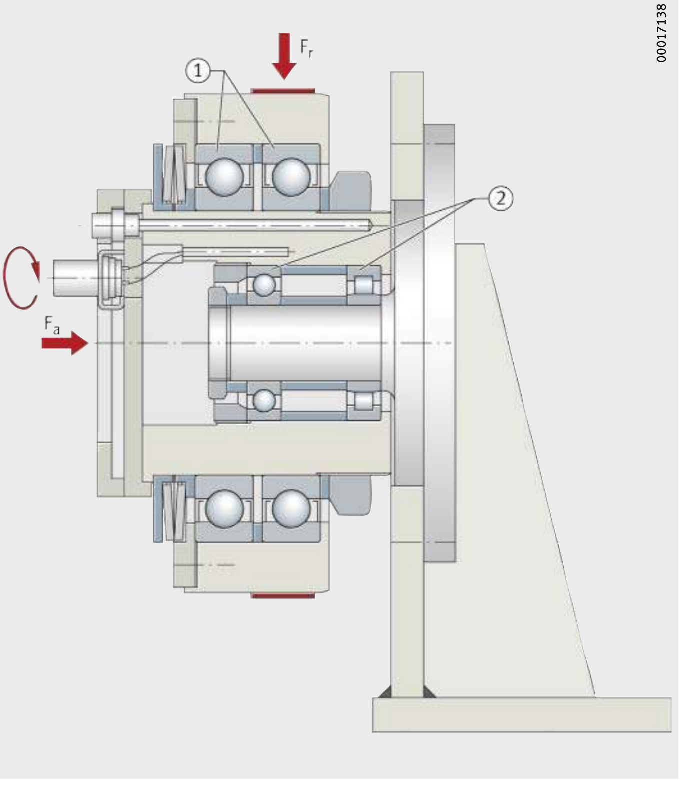
F_a = axial load
F_r = radial load

Figure 6
Test rig WS22
(spindle test rig)

Test rig WS10

In applications with a rotating outer ring, especially in bearing arrangements running at high speeds, additional requirements are placed on the grease. The test rig WS10 was developed in order to test these requirements, *Figure 7*.

The drive system is by means of a belt driving the bearing housing. The radial load is generated by the belt tension and weight force. The axial load is applied by means of disc springs.
The maximum outer ring speed is 4 000 min^{-1}. Speed parameters up to 650 000 $min^{-1} \cdot mm$ are possible.



F_a = axial load
F_r = radial load

① Test bearing
② Ancillary bearing

Figure 7
Test head WS10
(rotating outer ring)

Special tests for specific applications

FE8 paper machinery testing

This test models the environmental conditions of the dry section in a paper machine. The method is based on the test rig FE8 in the special version with a preheating container, see section Test rig FE8, page 158. Operating conditions close to practice are selected for this test.

For paper machinery testing, these are:

- speed 750 min^{-1}
- axial load 20 kN
- temperature 140 °C at the outlet of the preheating container
- running time 500 hours
- addition of synthetic process water.

Wind energy 4 stage test

In the case of this test method, an attempt was made to model all the damage mechanisms then known in wind energy gearboxes. The objective is to identify unsuitable lubricants at an early stage through these four tests.

Stage 1 Anti-wear protection under extreme mixed friction

The test carried out here is based on a requirement similar to that on type CLP gearbox oils in accordance with DIN 51517 on the test rig FE8. The difference here is that a higher axial load of 100 kN is used instead of 80 kN. This test is carried out in the extreme mixed friction range and takes account not only of the requirements on rolling element wear but also the occurrence of rippling and other surface damage.

Test conditions:

- speed 7,5 min^{-1}
- axial load 100 kN
- temperature +80 °C
- running time 80 hours.

Stage 2 Fatigue under moderate mixed friction

Testing is carried out on the test rig FE8 under moderate mixed friction conditions. Test bearings F-562831 with a plastic cage are used here. This cage differs from the screwed together plastic cage previously used in that it has two fewer pockets. The test load was therefore reduced from 100 kN to 90 kN. After a running time of 800 hours, there must be no rippling or pitting.

Test conditions:

- speed 75 min^{-1}
- axial load 90 kN
- temperature +70 °C
- running time 800 hours.

Stage 3
Additive reactions under EHD conditions

Testing is carried out under EHD conditions on the test rig L11. The aggressiveness of the additives is tested. After a running time of 700 hours, there must be no failures in the test bearings 6206 that were caused by additives. Since the required test running time is a multiple of the calculated bearing life, it is still possible that failures will occur that were not caused by the lubricant.

Test conditions:

- speed 9 000 min^{-1}
- radial load 8,5 kN
- running time 700 hours.

Stage 4
Oil behaviour at increased temperature and with addition of water

In many applications, gearboxes are subjected to moisture by the ambient conditions or condensation forms in the gearbox. This moisture can lead to additive reactions that, in conjunction with high temperatures, can cause formation of residues and filtration problems. At the same time, moisture has a negative effect on the lubricant film and precipation reactions of the additives can lead to unfavourable reaction layers in the bearing. In Stage 4, an attempt is made to model these aspects in a single test. The test rig FE8 with the preheating container already described is used here.

Test conditions:

- speed 750 min^{-1}
- axial load 60 kN
- temperature +100 °C at the outlet of the preheating container
- running time 600 hours.

Assessment is carried out on achieving the running time of 600 hours without bearing failure, formation of residues, any occurrence of filter blockage or rolling element wear.

Storage and handling

Storage and handling

Storage

Storage of rolling bearings

The performance capability of modern rolling bearings lies at the boundaries of what is technically achievable. Not only the materials but also the dimensional accuracies, tolerances, surface quality values and lubrication are optimised for maximum function. Even the slightest deviations in functional areas, for example as a result of corrosion, can impair the performance capability.

In order to realise the full performance capability of rolling bearings, it is essential to match the anti-corrosion protection, packaging, storage and handling to each other. Anti-corrosion protection and packaging are components of the product. They are optimised by Schaeffler as part of the process of preserving all the characteristics of the product at the same time. In addition to protection of the surfaces against corrosion, other important characteristics include emergency running lubrication, friction, lubricant compatibility, noise behaviour, resistance to ageing and compatibility with rolling bearing components (brass cage, plastic cage, elastomer seal). Anti-corrosion protection and packaging are matched by Schaeffler to these characteristics. The precondition here is the use of storage that is normal for high quality goods.

Storage conditions

The basic precondition for storage is a closed storage room in which no aggressive media of any sort may have an effect, such as exhaust from vehicles or gases, mist or aerosols of acids, alkalis or salts. Direct sunlight must also be avoided.

The storage temperature should be as constant as possible and the humidity as low as possible. Jumps in temperature and increased humidity lead to condensation.

The following conditions must be fulfilled:

- frost-free storage at a minimum temperature of +5 °C
  (secure prevention of hoarfrost formation,
  permissible up to 12 hours per day down to +2 °C)
- maximum temperature +40 °C
  (prevention of excessive run-off of anti-corrosion oils)
- relative humidity less than 65%
  (with changes in temperature,
  permissible up to max. 12 hours per day up to 70%).

! The temperature and humidity must be continuously monitored.

Storage periods

Rolling bearings should not be stored for longer than 3 years. This applies both to open and to greased rolling bearings with sealing shields or washers. In particular, greased rolling bearings should not be stored for too long, since the chemical-physical behaviour of greases may change during storage. Even if the minimum performance capacity remains, the safety reserves of the grease may have diminished. In general, rolling bearings can be used even after their permissible storage period has been exceeded if the storage conditions during storage and transport were observed. If the storage periods are exceeded, it is recommended that the bearing should be checked for corrosion, the condition of the anti-corrosion oil and where appropriate the condition of the grease before it is used.

Storage of Arcanol rolling bearing greases

Storage conditions

The applicable storage conditions for rolling bearing greases are the same as those for rolling bearings. The precondition is always that the Arcanol rolling bearing grease is stored in closed, completely filled original containers.

Storage periods

If the storage conditions are fulfilled, Arcanol rolling bearing greases can be stored in their closed original container without loss of performance for a maximum of 3 years. As in the case of rolling bearings, the permissible storage period should not be seen as a rigid limit. Rolling bearing greases are mixtures of oil, thickener and additives. Such mixtures of liquid and solid substances do not have unlimited stability but can rather be described as being in a metastable state. During storage, their chemical-physical characteristics may change and they should therefore be used up as soon as possible. If there is any doubt when using older greases, it is recommended that random sample checking of chemical-physical characteristics should be carried out to determine any changes in the grease. It is therefore not possible to state storage periods for containers that have been opened. If containers are to be stored after opening, the grease surface should always be brushed flat, the container should be sealed airtight and it should be stored such that the empty space is upwards. High temperatures should be avoided in all cases.
For checking of older greases, Schaeffler can provide assistance as a service for risk assessment covering further storage or use.

Measures to be taken after opening the original packaging

The packaging is a component of the anti-corrosion protection of rolling bearings. Rolling bearings should always remain in their original packaging until immediately before mounting. Once the original packaging has been opened, there is an increased risk of corrosion due to humidity and particles that reach the steel surface. If rolling bearings must be removed from their packaging before they are ultimately used, they must always be covered and stored in conditions of the lowest possible humidity. Protection can be provided either by reusing the original packaging or by means of a similar polyethylene or polypropylene film.

Follow-on preservation

If the anti-corrosion oil is removed from the steel surface, it is recommended that follow-on preservation should be used in the form of anti-corrosion oil, VCI paper or VCI film, depending on the type of lubrication to be used subsequently (VCI = Volatile Corrosion Inhibitor). Lubricants with a mineral oil base are compatible with practically all conventional anti-corrosion oils, such as Schaeffler Arcanol Anticorrosion Oil, see section Mixing of anti-corrosion oil with grease.

Manual handling of rolling bearings

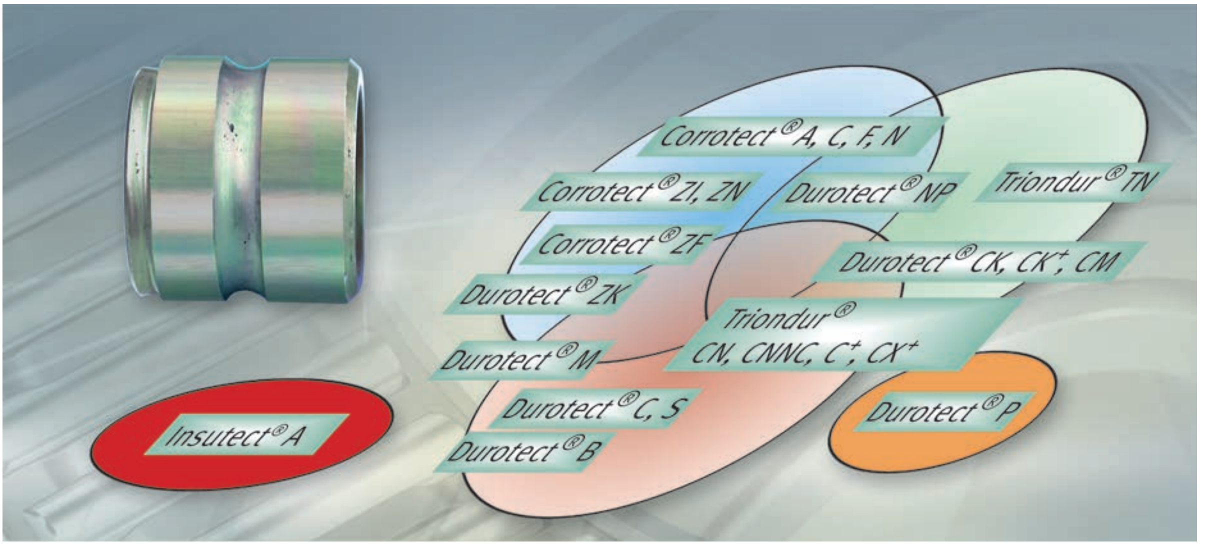
Rolling bearings should not be touched with bare hands, since fingermarks may remain on the steel surface that lead to an increased risk of corrosion at these points. Furthermore, some people may have an allergic reaction to mineral oil products. The wearing of gloves is strongly recommended.

Washing out of rolling bearings

If rolling bearings must be washed out in order to remove anti-corrosion oil or an existing greasing, the dry rolling bearings are then at extremely high risk of corrosion, see section Cleaning of contaminated bearings, page 148. Relubrication is necessary immediately.

Mixing of anti-corrosion oil with grease

Small proportions of anti-corrosion oil are compatible with practically all greases based on mineral oil. The quantity of anti-corrosion oil should be max. 8% of the lubricant quantity. This excludes greases that contain bentonite as a thickener. These may be softened or in extreme cases liquified by the anti-corrosion additives. Greases with a synthetic oil base are not generally compatible with anti-corrosion oils based on mineral oil. In this case, anti-corrosion protection must be provided by means of VCI paper, VCI film or an anti-corrosion oil matched to the specific lubricant together with appropriate packaging (VCI = Volatile Corrosion Inhibitor).



Dry running and media lubrication
Coatings

Dry running and media lubrication
Coatings

Dry running and media lubrication

Dry running of rolling bearings is necessary if the use of oil or grease is not facilitated or not permitted as a result of extreme conditions, such as vacuum or extreme temperatures. In certain applications, such as pumps or compressors, it may be advantageous if the bearings run in the ambient medium.

In conditions of moderate load and speed, well lubricated and sealed bearings show no significant wear even after long operating periods. If no lubricant film giving adequate separation is present under dry running or media lubrication, however, various damage modes may occur. These include adhesive and abrasive wear, hot running, fatigue and corrosion.

Bearing optimisation

In order to prevent or delay such damage, optimisation of bearings in relation to material, surface, geometry and lubrication is necessary. A significant proportion of this is covered by the selection of suitable materials for rolling bearing rings, rolling elements and cages.

High performance steels with coatings

In order to assess corrosion resistance, steels and coatings are subjected to a standardised salt spray test in accordance with DIN EN ISO 9227, *Figure 1* and *Figure 2*.

1 Cronitect®
2 440C steel

Figure 1
Corrosion resistance, comparison after 24 h salt spray test

1 Cronitect®
2 440C steel

Figure 2
Corrosion resistance, comparison after 500 h salt spray test

The figures show clearly the excellent corrosion resistance of the high performance steels Cronidur® and Cronitect® in comparison with the classic corrosion-resistant steels. While the typical rolling bearing steel X46Cr13 shows clear signs of corrosion after only approx. 6 hours, Cronidur® and Cronitect® are up to 200 times more resistant compared to the corrosion-resistant steels and coatings, *Figure 3*.

t = corrosion resistance in hours

1 X46Cr13
2 X105CrMo17 (440C)
3 Durotect® CK (Protect A), 2 μm
4 Durotect® CK+ (Protect B), 2 μm
5 Durotect® CM, 2 μm
6 Durotect® CM, 50 μm, up to 360 h possible, depending on post-treatment
7 Corrotect®, free from Cr(VI)
8 Corrotect® C, containing Cr(VI)
9 Cronitect®

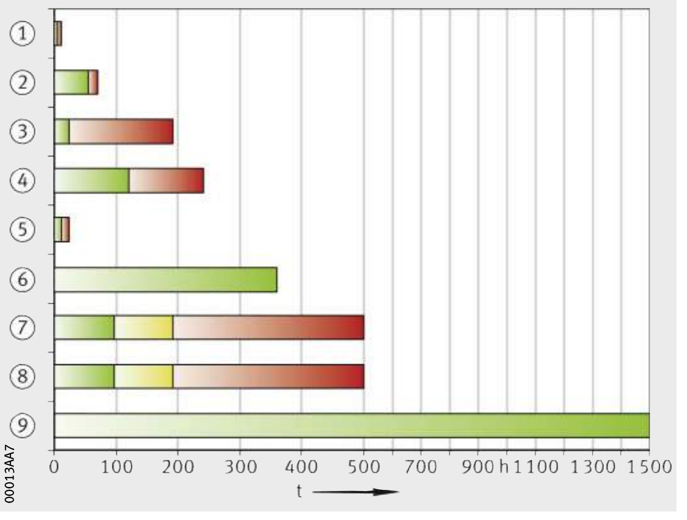


Figure 3
Corrosion resistance
Comparison: corrosion-resistant steels, coatings, Cronitect®

High temperature plastics

High temperature plastics such as PEEK (polyether ether ketone) offer high temperature resistance. The wear resistance is additionally optimised through the targeted selection of filler materials and is significantly better than that of the standard material PA66, with glass fibre reinforcement, that is frequently used for cages, *Figure 4*.

1 Volume loss
2 Plastic cage and steel rolling elements

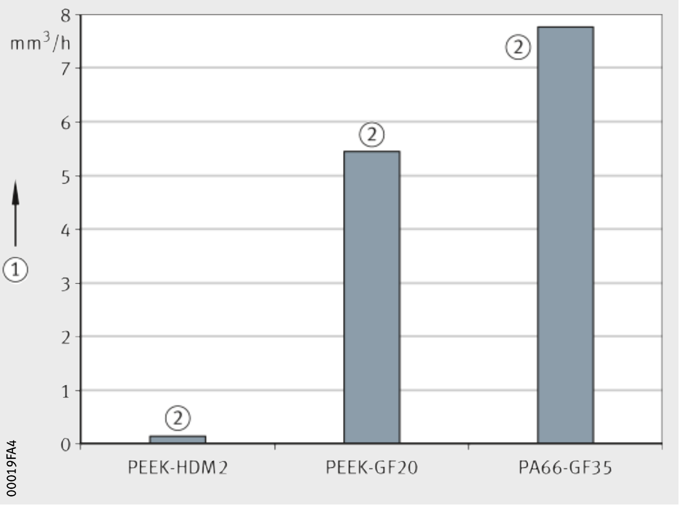


Figure 4
Wear of various cage materials in dry running

Polyether ether ketone is a partially crystalline material that is highly resistant even at high temperatures to chemicals as well as organic and inorganic fluids, see table. PEEK is highly suitable for rolling bearing cages, end pieces of linear guidance systems and tyres on track rollers.

Resistance of PEEK cages in cleaning agents

Medium	Max. chloride concentration mg/l	Max. concentration	Temperature + °C	Resistant
Sodium hydroxide solution NaOH	500	5%	90	Yes
Phosphoric acid H_3PO_4	200	5%	90	
Nitric acid HNO_3	200	5%	90	
Sulphuric acid H_2SO_4	150	1,5%	60	
Peracetic acid (Aseptic) (Aseptic)	100 5 5	500 mg/l 2 000 mg/l 4 000 mg/l	40 60 60	
Monobromoacetic acid or monochloroacetic acid	100	1% mixed with each of 1%: H_3PO_4, HNO_3, H_2SO_4	30	
NaOH + NaOCl Chloralkaline cleaner	300	5%	70	
Sodium hypochlorite NaOCl	300	300 mg/l active chlorine	60 20	
Hot water	100	–	125	
Steam approx. 0,5 bar	100	–	110	
Ozone	80	3 mg/l	30	

Ceramics

Ceramic has become firmly established as an important group of materials for rolling bearing components. Since this material has a range of excellent characteristics, rolling elements made from silicon nitride Si_3N_4 are used with increasing frequency in combination with coatings, special materials or for very specific application requirements.

Due to the tribological characteristics of the ceramic/steel material pair, the wear resistance is significantly higher than that of a steel/steel pair. In combination with the highly wear-resistant high performance steels Cronidur® and Cronitect® in particular, a long bearing operating life is achieved with ceramic rolling elements under conditions of dry running or media exposure.

Wear resistance

Coatings can be used to improve not only the corrosion resistance but also the wear resistance of surfaces.

Wear results from a dry running test on a standard angular contact ball bearing in comparison with an angular contact ball bearing with an optimised material selection are shown in *Figure 5* and *Figure 6*.

The test conditions were as follows:

- bearing type: ACBB 7205-B
- speed n = 1 000 min^{-1}
- Hertzian pressure p_H = 1 350 N/mm^2
- lubrication: dry running
- temperature: room temperature.

Bearing rings: 100Cr6
Balls: 100Cr6
Cage: PA66-GF25

① Surface contour at start of test
② Surface contour at end of test

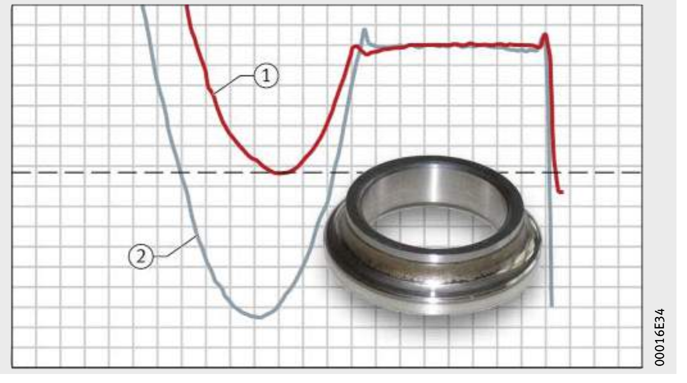


Figure 5
Dry running test with standard material

Bearing rings: Cronitect®
Balls: Si_3N_4
Cage: PEEK-HDM2

① Surface contour at start of test
② Surface contour at end of test

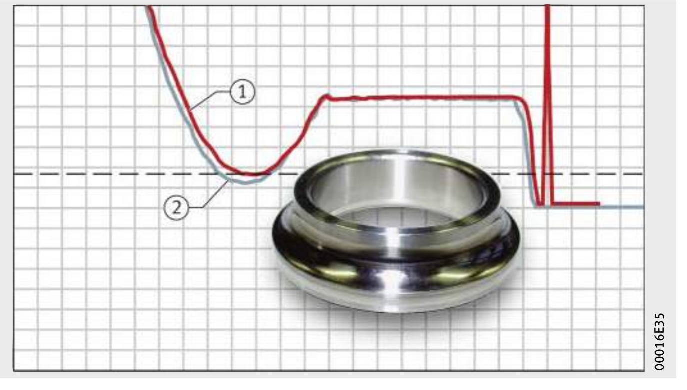


Figure 6
Dry running test with optimised material

Which steels are used, whether a coating is more advisable or whether corrosion-resistant steels are better in technical terms or more cost-effective, is fundamentally dependent on the relevant application.

Coatings

For many years, the Schaeffler Group has been a leader in the field of innovative surface and coating technology. With the aid of special processes, the functionality of surfaces has been optimised for many areas of application. The focus is on product characteristics such as wear resistance, sliding behaviour and reductions in friction, lustre, optics, electrical and thermal conductivity and insulation as well as anti-corrosion protection. Under the brand names Corrotect®, Triondur®, Durotect® and Insutect®, Schaeffler offers successfully coated components, *Figure 1*.

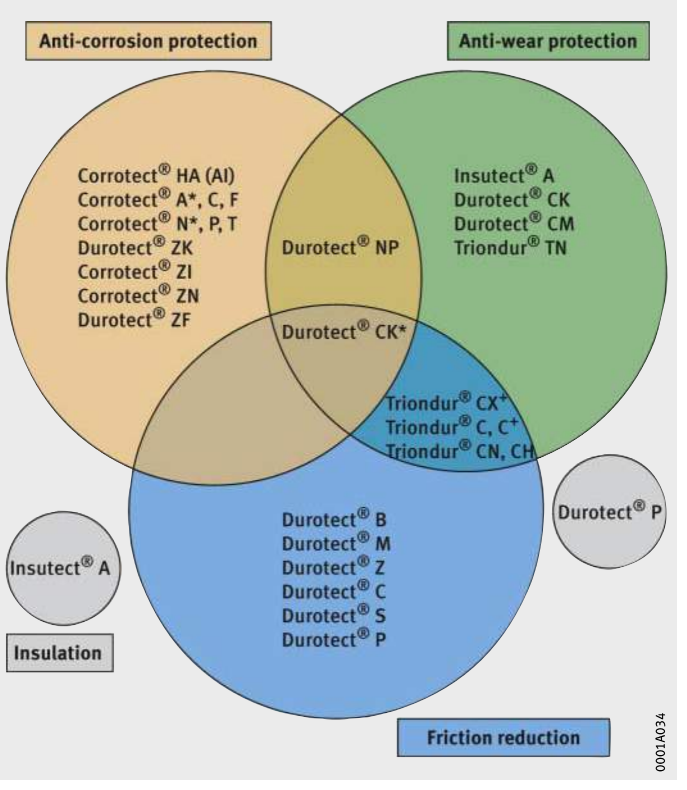


Figure 1
Coating systems and their areas of application

Development centre for surface technology

With its development centre for surface technology, the Schaeffler Group has created further conditions and possibilities for the successful implementation in practice of theoretical approaches to optimised products. The portfolio of the surface technology centre includes all stages and capacities, from the development of application-oriented coating systems through to volume production. Coating development is carried out on equipment which is equivalent to that in the production plants, so that the transfer to volume production can be carried out in a rapid and robust manner.
With the surface technology centre, development engineers at Schaeffler now have access to equipment and capacity that opens up new possibilities, especially in the combination of various processes and materials.

Use of coatings

Bearings and precision components from the Schaeffler Group offer high performance capacity and a long operating life. They provide the user with thoroughly developed and economical solutions for the large majority of requirements. Nevertheless, operating conditions sometimes occur that are beyond the limits of the standard designs. In such cases, one of the very wide range of coatings available can be a solution to the task of increasing the operating life of a component.

Coatings are used in the following cases:

- for anti-corrosion protection and the prevention of fretting corrosion
- for the reduction of wear, particularly under mixed friction, the improvement of friction behaviour and the prevention of slippage damage
- as insulation in order to prevent the passage of current.

Types of coatings

Coatings are applied to the surfaces of components without thermochemical diffusion taking place between the coating and the base material. In the Schaeffler Group, a large number of coatings are used. They are applied by a wide variety of methods and give widely differing advantages for the component. They should always be individually matched to the mounting situation. In many cases, it is sufficient to coat only one of the components in rolling contact or only a part thereof. The table Coating systems, page 186 gives an overview of the coatings used at Schaeffler, arranged according to the main areas of use. The features, advantages and benefits are given for each type of coating. Specific applications and references are presented in detail in TPI 186.

Examples

Protection against corrosion and fretting corrosion

For preventing corrosion and fretting corrosion, the coating systems Corrotect® $A^+$and N have proved effective. The coatings have a silver iridescent appearance.

They comprise a Zn/Fe alloy with thick layer passivation and coating thicknesses between 2 μm and 5 μm.

Figure 2
Coated and uncoated part after 24 h in the salt spray test

Advantages

The coating systems offer very good corrosion protection (in the salt spray test in accordance with DIN EN ISO 9227, at least 192 hours without base metal corrosion for rack coating without heat treatment), *Figure 2*. They are thus an economical, alternative method of cathodic anti-corrosion protection.

Application The coatings are particularly suitable for smaller bearings and bearing adjacent parts where high corrosion resistance is required, for example drawn cup needle roller bearings and thin-walled components in large quantities. These are found, for example, in the agricultural equipment sector or industry, *Figure 3* and *Figure 4*.

Figure 3
Transport line in drinks production

Figure 4
Combine harvester

Protection against wear, friction and slippage damage

For the reduction of wear, friction and slippage damage, the coating systems **Triondur® C, C⁺, CX⁺** (PVD/(PA)CVD hard material layers) have proved effective. The coatings are anthracite to black in colour and have a hardness of more than 1000 HV, *Figure 5*.

They comprise multi-layer, amorphous, hydrogen-containing carbon layers that are doped with metals or non-metals. They have a smooth surface structure and a thickness of 0,5 μm to 4 μm.

Figure 5
Tappet with Triondur® CX⁺

Advantages

Friction in the dry state between the coating and steel is up to 80% lower in comparison with a steel/steel combination. The low friction value is favourable particularly in cases of adhesive wear.
The coating is characterised by high wear resistance in the mixed friction range. The high hardness also contributes high wear resistance. The coatings are ideally suited in conditions of slippage damage and for large bearing components.

Applications These coatings are suitable for applications such as:

- barrel rollers in spherical roller bearings for paper calenders
- cages with a coated outside diameter in the printing industry
- valve train components such as tappets, *Figure 5*, page 184
- high precision components for diesel injection systems, *Figure 6*.

Figure 6
Control pistons for diesel injectors

Coatings

Coating systems

Designation of coating system	Comment	Main function		
		Anti-corrosion protection	Anti-wear protection	Friction reduction
Corrotect® A		■		
Corrotect® N	CT004	■		
Corrotect® P	Paint	■		
Corrotect® ZK	Zinc CT010 – CT013	■		
Corrotect® ZI	Zinc-iron CT020 – CT023	■		
Corrotect® ZN	Zinc-nickel CT030 – CT033	■		
Corrotect® ZF	Cr(VI)-free CT100	■		
Durotect® NP	Chemical nickel CT200 – CT205	■	■	
Durotect® HA	Hard anodising (Al)	■	■	
Durotect® CK	(Protect A in linear sector) Columnar hard chomium coating CT230		■	
Durotect® CK+	Columnar hard chromium coating + mixed chromium oxide CT231	■	■	■
Durotect® CM	Microcracked hard chromium coating CT220 – CT224		■	
Durotect® B	Mixed iron oxide CT240			■
Durotect® M	Manganese phosphate CT260 – CT261			■
Durotect® Z	Zinc phosphate CT250 – CT251			■
Durotect® C	Copper CT270			■
Durotect® S	Silver CT271			■
Durotect® P	Polymer-based coating CT700 – CT702			■
Insutect® A	Aluminium oxide			
Triondur® CN	Cr_XN CT400 – CT404			
Triondur® CNN	CrN/CrC CT405 – CT408			
Triondur® C	a-C:H:Me CT420			
Triondur® C+	a-C:H CT450 – CT479			
Triondur® CX+	a-C:H:X CT480 – CT509			
Triondur® TN	TiN CT415 – CT419			
Triondur® CH	ta-C CT520 – CT529			

Additional function	Main area of application Special feature
	Automotive, belt drives, selector shafts, Cr(VI)-free
	Automotive, belt drives, detents, Cr(VI)-free
	Automotive, belt drives
	Industrial, Automotive
	Industrial, Automotive, belt drives, bearing components, screws
	Industrial, Automotive, belt drives, bearing components, screws
	Industrial, Automotive, chassis engineering, bearing components, screws
Current insulation	
Slight anti-corrosion protection, slight reduction in friction	
Slight anti-corrosion protection, slight reduction in friction	
Improved running-in behaviour, reduced slippage damage, slight anti-corrosion protection	Industrial, Automotive, bearing components
Improved running-in behaviour, slight anti-corrosion protection, emergency running lubrication	Aerospace, bearing components
Temporary anti-corrosion protection, protection against fretting corrosion	Industrial, Aerospace, linear guidance systems, bearing components
Emergency running lubrication	Industrial
Emergency running lubrication	Linear guidance systems, bearing components
	Industrial, bearing rings
	Current insulation, Industrial, rail vehicles, electric motors
	Automotive, valve train components
	Automotive, valve train components
Reduced slippage damage	Industrial, Automotive, bearing components, rolling bearings, engine components
	Industrial, Automotive, engine components, bearing components
Minimal friction in valve train	Automotive, valve train components, bearing components
	Aerospace, bearing components
	Automotive

Industrial Service

Industrial Service

Industrial Aftermarket

Portfolio

Schaeffler Industrial Aftermarket (IAM) is responsible for replacement parts and service business for end customers and sales partners in all significant industrial sectors. On the basis of innovative solutions, products and services relating to rolling and plain bearings, the service function of Industrial Aftermarket offers a comprehensive portfolio that covers all phases in the lifecycle of the rolling bearing and takes account of the total costs (TCO), *Figure 1*.

Figure 1
Industrial Aftermarket

The portfolio in the area of maintenance and quality assurance extends from mounting, through plant monitoring, to the introduction and implementation of preventive maintenance measures.

In more than half of all cases, inadequate lubrication is the cause of unplanned machine downtime. The life of rotating machine elements can be significantly extended by the use of greases appropriate to the different operating and environmental conditions as well as the definition of and adherence to lubrication intervals and quantities.

Mounting Toolbox

In the Mounting Toolbox, Schaeffler brings together valuable knowledge relating to the lubrication, mounting and dismounting of rolling bearings. Videos show the points that must be paid close attention for correct lubrication, mounting and alignment. In the "Virtual Plant" of the Mounting Toolbox you can watch the fitting personnel at work, *Figure 2*.

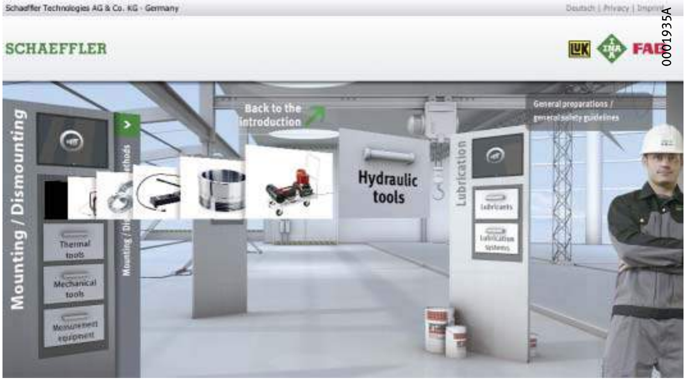


Figure 2
Mounting Toolbox

Services

Services relating to lubrication include:
- selection of lubricants and lubrication systems
- lubrication of bearing positions
- preparation of lubrication and maintenance plans
- lubrication point management
- consultancy on lubricants
- lubricant investigations and tests.

Advantages

The Schaeffler lubrication service helps to:
- prevent failures of rotating components
- increase productivity
- reduce lubrication costs.

Further information

Condition monitoring of grease and oil is a reliable method that is also used in rolling bearing lubrication. In this area, Schaeffler offers innovative products that help to securely prevent damage and downtime.

Grease sensor FAG GreaseCheck

In the past, bearings were regreased as a function of time. The grease quantities and lubrication intervals were calculated numerically. Through the use of the grease sensor FAG GreaseCheck, regreasing can be carried out as a function of condition.

By means of the grease sensor, the following parameters are optically measured directly in the bearing arrangement:

- water content
- turbidity
- thermal and mechanical wear
- temperature.

This information is transferred by cable to the evaluation unit, *Figure 1*. The evaluation unit generates an analogue signal that gives the user rapid and simple information on the condition of the grease.

The individual benefits are as follows:

- lubrication appropriate to needs
- lower grease costs
- prevention of unplanned downtime
- lower maintenance costs
- lower equipment costs.

① Grease sensor
② Electronic evaluation system

Figure 1
Grease sensor
FAG GreaseCheck

Oil sensor FAG Wear Debris Check

FAG Wear Debris Check is an oil sensor that monitors the quantity of wear particles in fluids and classifies these according to size and material. The oil sensor is installed either in an ancillary flow of the recirculating lubrication system in the gearbox ahead of the filter or in a separate circuit.

Typical applications for the FAG Wear Debris Check can be found, for example, in gearboxes in raw material extraction plant and in the steel industry, planetary gearboxes in wind turbines or in ship propulsion systems.

The features of the oil sensor are as follows:

- monitoring of the number of particles in the oil
- differentiation of the particles into ferrous and non-ferrous metals
- classification of the particles according to size
- possible integration in an online monitoring system for linking of oil particle and vibration data.

Where oil and vibration monitoring facilities are combined, damage in gearboxes with recirculating oil lubrication can be detected at an early stage and the source can be determined. In this way, it is possible to prevent production shutdowns or secondary damage.

Relubrication systems

Lubricators and lubrication systems automatically provide bearings with the correct quantity of lubricant. This prevents the most frequent cause of rolling bearing failure: inadequate or incorrect lubrication. Approximately 90% of bearings are lubricated with grease. Relubrication with the correct quantity of grease at the appropriate intervals gives a significant increase in the life of bearings.

Single-point lubricators

Lubricators convey fresh grease in the defined quantity at the correct time to the contact points of the rolling bearing.

The devices adhere to the lubrication and maintenance intervals and prevent undersupply or oversupply of grease. Plant downtime and maintenance costs are reduced as a result.

Lubricators have the following advantages:

- individually configured, precise supply to each bearing position
- fully automatic, maintenance-free operation
- reduced personnel costs compared to manual relubrication
- different dispensing times can be selected
- pressure buildup to max. 25 bar.

Small lubrication systems

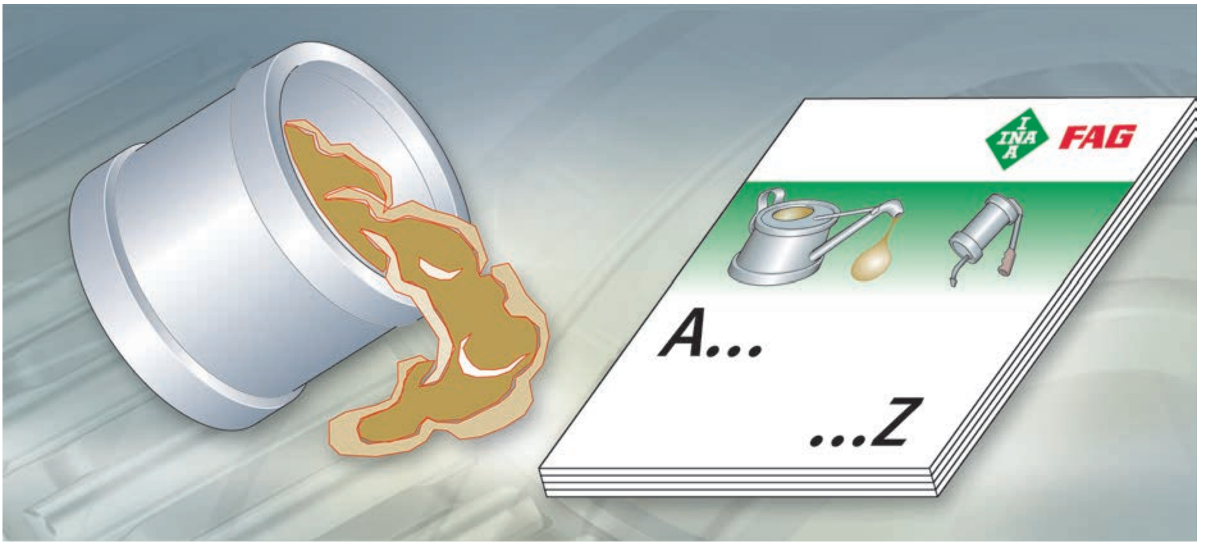
A single-point or multi-point lubrication system can supply lubrication points precisely and irrespective of temperature. The dispensing times can be set variably.

CONCEPT8

This single-point and multi-point lubrication system can grease up to eight lubrication points. The grease cartridges are available in the size 800 cm^3. The lubrication system controls the greasing of the lubrication points independently of the machine. The voltage supply for the drive of the lubrication system is provided by a mains power pack.

The advantages of the lubrication system are as follows:

- suitable for oil and grease up to NLGI 3
- reliable piston pump as delivery pump
- operating temperature from –20 °C to +70 °C
- low operating voltage of 24 VDC
- pressure buildup to max. 70 bar.



Lubrication lexicon

Lubrication lexicon

A

Acid number NZ
Indicator of the ageing of a mineral oil. It indicates how much mg potassium hydroxide is required to neutralise the free acids that are contained in 1 g of oil. In the case of doped oils, the acid number in the fresh condition is normally above zero due to the presence of agents. A change in the acid number compared to the new condition should not exceed a value of 2.

Additive
Substance soluble in oil that is added to lubricants in order to improve their characteristics through chemical or physical effects (for example the EP effect, viscosity/temperature behaviour, solidification point, flowability, oxidation resistance, foaming).

Adhesive oil
Highly viscous, tough and sticky lubricant that is normally used in predissolved form.

Ageing
Undesirable chemical changes in mineral and synthetic lubricants that occur during use and storage. They are initiated by reaction with oxygen (formation of peroxides, hydrocarbon radicals). This oxidation is accelerated by heat, light and the catalytic influences of metals and other contaminants. Acids and sludge are formed. Anti-ageing products, so-called antioxidants (AO), delay ageing.

Agent
See section Additive, page 197.

Aluminium complex soap grease
Grease based on aluminium complex soaps with good water resistance and, through the use of high pressure additives, a high pressure capacity. Depending on the base oil, they can be used at up to approx. +160 °C.

Analysis data
Data that describe the physical and chemical characteristics of lubricants. These include: density, flash point, viscosity, solidification point, dropping point, penetration, acid number and saponification number. Within a certain scope, they allow conclusions to be drawn as to useability.

Antioxidant (AO)
Agent that leads to a considerable delay in lubricant ageing.

Anti-wear additive
Additive intended to reduce wear in the mixed friction range. A distinction is made between mildly acting additives (such as aliphatic acids, aliphatic oils), high pressure agents (such as sulphur, phosphorus and zinc compounds) and solid lubricants (such as graphite, PTFE, molybdenum disulphide).

Arcanol
Selected greases that are sold by Schaeffler Group Industrial. The operating limits of individual greases have been determined using the most modern test methods (for example FE8 and FE9 test rigs) under various operating conditions and with rolling bearings of various types. They can be used to fulfil almost all requirements placed on the lubrication of bearings, see table, page 128.

Aromatic compound
Unsaturated hydrocarbon compound with a ring-shaped molecular structure (for example benzene, toluene, naphthalene). Aromatic compounds have poor viscosity/temperature behaviour and have an unfavourable influence on the oxidation resistance of lubricants.

Ash content
The content of non-combustible residues in a lubricant, normally metal oxides. The ash may have different origins. These include agents dissolved in the oil, graphite, molybdenum sulphide as well as soaps and thickeners in greases. Fresh, undoped mineral oil raffinates must be complete free from ash. Used oil will also contain undissolved metal soaps that are formed in operation, as well as non-combustible residues of contaminants, such as wear debris from bearing parts and seals. The ash contact can occasionally be used to determine approaching bearing damage.

ASTM
Abbreviation for American Society for Testing and Materials. An institute whose activities include the formulation of the American mineral oil standards.

ATF
Abbreviation for Automatic Transmission Fluid. Special oils that are matched to the requirements present in automatic gearboxes.

B

Barium complex soap grease
Grease comprising barium complex soaps and mineral oils or synthetic oils. Barium complex soap greases are water-repellent, have high working stability and the lubricant film has high load carrying capacity.

Base oil
The oil contained in the grease is described as the base oil. The oil proportion varies according to the thickener and intended purpose of the grease. The proportion and viscosity of the base oil change the penetration and friction behaviour of the grease.

Bentonite
Inorganic thickener that is used in the production of temperature-resistant greases with good cold characteristics. It is included in the class of minerals (aluminium silicate).

Bleeding
The base oil in the grease becomes separate from the thickener.

Bright stock
Highly viscous, refined oil residue derived from vacuum distillation. It is used as a mixing component for oils and improves the lubrication behaviour.

C

Calcium soap grease
Greases comprising calcium soaps and mineral oils. They have good water resistance and are therefore frequently used as a sealant grease against water. Since they offer hardly any resistance to corrosion, they must contain agents to give anti-corrosion protection. Due to their limited temperature range of –20 °C to +50 °C, they are not used widely.

Centipoise (cP)
Unit commonly used in earlier times for dynamic viscosity: 1 cP = 1 mm^2/s.

Centistoke (cSt)
Unit commonly used in earlier times for kinematic viscosity: 1 cSt = 1 mm^2/s.

Cold behaviour
See section Solidification point, page 209 and section Flow pressure, page 201.

Colour of oils
This allows conclusions to be drawn on the operating life. Fresh oil can be more or less dark. As a result, ageing can only be detected by comparing the oil to be examined with a sample of the fresh oil. A dark colour may also occur, however, if the oil is contaminated by dust, soot or wear debris. This is possible with even very small quantities.

Combustion point
The lowest temperature, in relation to a defined pressure, at which the vapours from a fluid heated to a uniform extent will, after ignition by a flame, continue to burn for a period of five seconds (DIN ISO 2592).

Complex grease
Greases based on the metal soaps of high-molecular aliphatic acids. They also contain metal salts derived from low-molecular organic acids. These salts form complexes with the acids that have more favourable characteristics than simple soap greases (temperature limits, behaviour in the presence of water, anti-corrosion protection, pressure absorption capacity).

Consistency
See section Penetration, page 206.

Copper strip test
Method for the qualitative testing of the corrosion effect of mineral oils (DIN EN ISO 2160) and greases (DIN 51811) on copper.

Co-rotation
Transport of the grease by rotating parts. Grease clumps repeatedly enter the space between the rolling element and raceways, leading to an increase in the undesirable worked penetration. At high speeds, a grease must therefore be selected that does not tend to undergo co-rotation. Co-rotation is influenced by the thickener, penetration, temperature and bearing type.

D

Decompression behaviour This allows statements on the suitability of greases for use in central lubrication systems (DIN 51816-2).

Demulsifying ability The separation capacity of oils from oil/water mixtures.

Density Mass per volume of mineral oil products in relation to 20 °C. It has the symbol ρ and is stated in g/cm^3. The density is dependent on the chemical structure of the oil. In oils with the same origin, it increases with increasing viscosity and the increasing degree of refining. Density alone is no measure of quality.

Deposit Lubricant residues such as soot and contaminant particles that occur as a result of ageing of the oil, excessively long oil change intervals and mechanical wear under the strong influence of heat. They settle in the oil sump, in the bearings, in filters and in the lubricant feeds and can endanger the operational reliability.

Detergent Agent with the ability to separate residues and clean deposits from surfaces to be relubricated.

Dispersant An agent in oil that holds solid contaminants in a very finely distributed suspension until they are filtered out or removed by means of an oil change.

Dispersion A system of finely distributed, non-soluble substances in a liquid or gas, e.g. an emulsion or suspension.

Dispersion greasing A method for introducing lubricant into the rolling bearing. The rolling bearing is immersed in a dispersion bath (dispersant and grease). After vapourisation of the dispersant, a lubricant layer with a thickness of 1 μm to 100 μm remains on the bearing surface. This method reduces the friction but also the grease operating life.

Distillate Hydrocarbon mixture derived from the distillation of crude oil.

DLC Diamond-like carbon coatings are protective carbon layers resembling diamond. They essentially comprise a highly crosslinked amorphous carbon matrix with various proportions of sp^2 and sp^3 orbital bonds as well as various contents of embedded oxygen. The tribological behaviour of DLC layers is more similar to the behaviour of graphite.

Doped lubricants Oils or greases that contain one or more agents to improve specific characteristics, see section Additive, page 197.

Dropping point Guide value for the upper operating temperature of a grease.

The grease is heated under standardised test conditions in accordance with DIN ISO 2176. The temperature is determined at which, when a nipple is opened, the sample flows and falls to the base of the test pipe.

Dynamic viscosity See section Viscosity, page 211.

E

Emcor method
Testing of the anti-corrosion characteristics of rolling bearing greases in accordance with DIN 51802.

Emulsifiability
The tendency of an oil to form an emulsion with water.

Emulsifier
Substance that acts on the emulsifiability of oils.

Emulsion
A mixture of fluids that are not normally soluble in each other. An emulsifier is generally used when mixing mineral oils with water.

EP additive
Oils or greases that contain Extreme Pressure agents in order to prevent wear and fretting.

Ester
Compound produced by chemical means between acids and alcohols with the release of water. They can be used to produce synthetic oils, whose characteristics are defined by the molecular structure of the ester. Esters of higher alcohols with bivalent aliphatic acids form so-called diester oils. Ester oils comprising multivalent alcohols and various organic acids have particularly high thermal stability.

F

Flash point
The lowest temperature at which, under specified test conditions, so much oil vapour is released that the oil/air mixture ignites for the first time in the presence of a flame. It is included with the key data for an oil and is standardised in accordance with DIN ISO 2592. The flash point has hardly any significance in relation to the tribological assessment.

Flowable grease
Grease of semi-fluid to paste-like consistency of the NLGI grades 000, 00 and 0. In order to increase the pressure absorption capacity, they may contain extreme pressure additives (EP) or solid lubricants. They are normally used for gearbox lubrication.

Flow pressure
The flow pressure gives information on the consistency of a grease and indicates its flow behaviour. In accordance with DIN 51805, this is the pressure that is required in order to press a stream of grease through a standardised nozzle. In accordance with DIN 51825, it determines the lower operating temperature.

Four ball test machine (VKA)
Device for testing of lubricants with high pressure and anti-wear agents, standardised in accordance with DIN 51350.

In order to assess the high pressure additives, four balls are arranged in a pyramid. The upper ball rotates and is subjected to a force until the balls weld together. The welding force measured is the so-called VKA value.

In order to assess the anti-wear additives, the same test is performed at a defined test force for one hour. The impression diameter of the three static balls is then measured and used as the wear parameter.

G

Gearbox grease
See section Flowable grease.

Gearbox oil
Oil for gearboxes that are used predominantly in the industrial sector. These are standardised in accordance with DIN 51509 and DIN 51517 (oils, type C, CL, CLP). In the automotive sector, gearbox oils are classified in accordance with SAE.

Gel grease
Inorganic thickener type, normally silica gel. The thickener comprises very finely distributed solid particles whose surface can absorb oil. Gel greases have a wide operating temperature range and are resistant to water. They are less suitable for high speeds and loads.

Grease
Consistent mixture of thickener and base oil. A distinction is made between different type of grease. Metal soap greases comprise metal soaps as thickeners and oils. Soap-free greases bind the oil using inorganic gel formers or organic thickeners. Synthetic greases comprise organic or inorganic thickeners and synthetic oils. For the selection of greases, see table Greases, page 84.

Grease operating life
The period between startup and failure of a bearing as a result of lubricant failure, see section Grease operating life, page 95. The grease operating life is dependent on the grease quantity, grease type (thickener, base oil, additives), bearing type, bearing size, magnitude and type of load, speed parameters and bearing temperature. It can be estimated if the operating conditions are known.

The grease operating life is also described as a lubrication interval. It must not be confused with the relubrication interval, see section Relubrication interval, page 207.

H

Hard layer
A particularly hard, oxidation-resistant and chemically resistant layer. It comprises an oxide, nitride, carbide, carbonitride or carboxynitride of an element of the main group 4, 5 or 6 of the periodic table, such as TiN, CrN.

HD oil
Heavy duty oils are engine oils that are matched to the considerable requirements in internal combustion engines by means of additives.

High pressure additive
See section EP additive, page 201.

Homogenisation
The final phase of grease production. In order to achieve a uniform structure and very fine distribution of the thickener, the grease is subjected to strong shearing. This is carried out in a special machine, the so-called homogeniser.

Hydraulic fluid
Pressure fluid for force transmission and control in hydraulic plant. It is standardised in accordance with DIN 51524 and comprises mineral oil with a low solidification point. It is resistant to ageing, thin, non-foaming and fire-resistant.

Hydraulic oil
See section Hydraulic fluid.

Hypoid oil
High pressure oil with EP additives for hypoid gearboxes that are mainly used for final drives in motor vehicles.

I

Inhibitor Agent that delays certain reactions in a lubricant.
Inhibitors are used in preference to combat ageing and corrosion processes in lubricants.

ISO VG See section Viscosity classification, page 211.

K

Key data See section Analysis data, page 197.

Kinematic viscosity See section Viscosity, page 211.

L

Lithium soap grease Greases based on lithium soap. They are characterised by good water resistance and a wide operating temperature range.
They contain oxidation and corrosion inhibitors as well as extreme pressure additives (EP). Due to their good characteristics, lithium soap greases are used widely for the lubrication of rolling bearings. The operating limits of normal lithium soap greases are at –35 °C and +130 °C.

Lubricant additive See section Additive, page 197.

Lubrication interval See section Grease operating life, page 202.

M

Mechanical-dynamic lubricant testing
Test methods for the investigation of rolling bearings under conditions close to operation (operating conditions and environmental conditions). The lubricant is assessed by observing the behaviour of the test element and lubricant during testing and their condition after testing. The results of model test devices can only be applied to rolling bearings under certain conditions. Mehods are therefore used in preference that include rolling bearings as test elements.

MIL-Spezifikation
Minimum requirement of US armed forces for operating materials to be supplied. Although these are aimed only at the military sector, these specifications have also found use in the civil sector.
Engine and machine manufacturers have in some cases the same minimum requirements for lubricants. There are valid as a quality indicator.

Mineral oil
Oil derived from crude oil that is processed by distillation and refining for lubrication purposes. In chemical terms, it predominantly comprises hydrocarbons.

Miscibility of oils
Statement as to whether different oils are miscible with each other. This is not always possible with different grades and manufacturers. The exception is HD engine oils, which can be mixed with each other in almost all cases. If fresh oils are mixed with used oils, there is a risk that sludge will be precipitated. In order to exclude this possibility, it is recommended that samples should be mixed in a glass beaker in advance.

Multigrade oil
Engine oils and gearbox oils with an improved viscosity/temperature behaviour. In comparison with single grade oils, the multigrade oil is not too thick and low temperatures and not too thin at high temperatures.

N

NLGI
Abbrevation for the National Lubricating Grease Institute in the USA. Greases are subdivided into grades defined by the NLGI, see section Penetration, page 206.

Nominal viscosity
See section Viscosity, page 211.

Normal oil
Oil of the class L-AN in accordance with DIN 51501, which is used if no particular requirements are present.

O

Oil separation The tendency of a grease to release oil in the case of extended storage or at increased temperature. Long term lubrication requires the long term release of a small quantity of oil that must, however, be large enough to ensure supply to all contact surfaces.
The oil separation is defined in accordance with DIN 51817.

Oil, type B Dark mineral oils containing bitumen with good adhesion capacity, in accordance with DIN 51513.

Oil, type C, CL, CLP Gearbox oils for recirculating lubrication, in accordance with DIN 51517.

Oil, type CG Slideway oils.

Oil, type K Refrigerator oils in accordance with DIN 51503.

Oil, type N Normal oils in accordance with DIN 51501.

Oil, type T Steam turbine and regulator oils in accordance with DIN 51515-1.

Oil, type V Air compressor oils in accordance with DIN 51506.

Oil, type Z Steam cylinder oils in accordance with DIN 51510.

Operating viscosity Kinematic viscosity (see section Viscosity, page 211) of an oil at operating temperature. It has the symbol ν. The operating viscosity can be determined with the aid of a viscosity/temperature diagram. For mineral oils with an average viscosity/temperature behaviour, *Figure 2*, page 24.

Oxidation See section Ageing, page 197.

P

Passivation

The formation of a covering layer that prevents or considerably slows down the corrosion of the metallic base material.
Electroplating methods are used such as thick layer passivation, yellow and black chromate passivation.

Penetration

Indicator of the deformability of a grease.

It is determined by dropping a standardised brass cone from a defined height into a container filled with grease. The penetration depth after a period of 5 s is then measured. The measured value is stated in 0,1 mm.

The National Lubricating Grease Institute has subdivided the measurement values into penetration grades (NLGI grades) 000 to 6, see table NLGI grade, page 66. Greases for rolling bearings are normally in the consistency grades 1 to 3. This subdivision is used worldwide and is standardised in accordance with DIN 51818.

The consistency of greases changes as a result of mechanical load. A distinction is made between static penetration and worked penetration.

Pour point

See section Solidification point, page 209.

Pressure/viscosity behaviour

The influence of pressure on the viscosity of an oil. With increasing pressure, the vicosity of mineral oils increases, *Figure 4*, page 11.

R

Radiation Influence on the operating life of lubricants, for example as a result of radioactive substances. The energy dose is stated in Gray (Gy) (1 Gy = 1 J/kg). The equivalent dose is stated in Sievert (Sv) (1 Sv = 1 J/kg). In addition to the SI units, the older units Rad (rd) and Rem (rem) are still commonly used in some cases (1 rd = 1 rem). Conversion: 1 Gy = 100 rd and 1 Sv = 100 rem.

Radioactivity See section Radiation, page 207.

Raffinate Product occurring as a result of refining.

Recirculating lubrication Lubrication method in which the oil is repeatedly fed to the friction point and becomes effective.

Reference viscosity Kinematic viscosity (see section Viscosity, page 211) of an oil, as allocated to a defined lubrication condition. It has the symbol ν_1.
The reference viscosity can be determined with the aid of the mean bearing diameter and the speed, *Figure 2*, page 24.
The so-called viscosity ratio κ of the operating viscosity ν to the reference viscosity ν_1 allows an assessment of the lubrication condition ($\kappa = \nu/\nu_1$).

Refining Method for the purification of distillates in the production of oils. Refining improves the ageing resistance of oils.
Unstable compounds in which nitrogen, oxygen or metal salts may be embedded are separated out. The most important refining methods include sulphuric acid refining (sulphuric acid raffinate) and solvent refining (solvent raffinate).

Refrigerator oil Oil that is subjected in refrigerators to the effect of the refrigerant. These are subdivided into groups in accordance with the refrigerants. Their minimum requirements are standardised in DIN 51524.

Relubrication interval The period during which a bearing is relubricated. The relubrication interval should be defined as shorter than the grease operating life.

S

SAE

Abbreviation for Society of Automotive Engineers. Various standards and classifications, in particular the SAE classification for engine oils are derived from this association of US American automotive engineers and are used worldwide, see section SAE classification.

SAE classification

Viscosity classes for engine oils in accordance with SAE, used in the vehicle sector. A comparison of viscosities from SAE and ISO VG is possible, *Figure 6*, page 79.

Saponification number (VZ)

Indicator of the ligated and free acids in one gram of grease.
It indicates how many milligrams of the acid regulator potassium hydroxide are required in order to neutralise the free and ligated acids in one gram of oil and saponify the esters present.

The saponification number indicates the change in the oil in unused and used mineral oils with and without additives.

Seal behaviour

Organic seal materials show behaviour that differs from oils and greases. In some cases, seals undergo swelling, shrinking or embrittlement or even dissolve. The operating temperature and composition of the lubricant as well as the effective duration have a major influence. Information on the resistance of seals is provided by their manufacturers and, as appropriate, by the lubricant manufacturers.

Silicone oil

Synthetic oils used under special operating conditions.
They have more favourable key data than mineral oils but inferior lubrication characteristics and a lower pressure absorption capacity, see table Base oils and their typical characteristics, page 77.

Sludge formation

Precipations of mineral oil products that are deposited as sludge. These are oxidation products and polymerisates that are formed through the influence of air and water.

Sodium soap grease

No longer commonly used.

Solid foreign matter

Non-soluble, foreign contaminants in n-heptance or in solvent mixtures in accordance with DIN 51813. Solid foreign matter in oils are determined in accordance with DIN 51592 E and, in greases and solvent mixtures, in accordance with DIN 51813.

Solid lubricant

Substances suspended or directly added in oils and greases that reduce friction. The most well known of these are graphite, PTFE and molybdenum disulphide.

Solidification point The lowest temperature of a mineral oil at which a sample still flows when cooled under certain conditions.

Solvate Mineral oils refined using solvent. Also known as solvent raffinate.

Specification Military and company specifications for lubricants, that define the physical and chemical characteristics and test methods.

Spindle oil Thin oils with a viscosity of approx. 10 mm^2/s to 68 mm^2/s at +40 °C.

Static penetration Penetration measured at +25 °C of a grease sample that was not pretreated in the grease shaper.

Steam turbine oil Highly refined, ageing resistant oils used for the lubricaion of gearboxes and bearings in steam turbines. The oils are available in both doped (EP) and undoped form. They are designated in accordance with DIN 51515-1 as oil, type T.

Stick/slip additive Agent added to lubricants to prevent jolting sliding motion, for example on the guideways of machine tools.

Suspension Solid bodies finely distributed in fluids, for example non-soluble agents in oils.

Swelling behaviour The influence, for example on the form and structure of rubber and elastomers due to the effect of lubricants (DIN 53521).

Synthetic oil Synthetic oils are produced by chemical synthesis of molecules. Polymerisation leads to polyalphaolefins (PAO) or polyalkylene glycols (PAG) or condensation reactions lead to esters. Synthetic oils have advantages in comparison with mineral oils at particularly low or particularly high operating temperatures. They are, however, significantly more expensive.

T

Thickener The component of greases that retains the base oil in the grease. The most frequent thickeners are metal soaps (such as Li-, Ca-, Na- 12-hydroxy stearate) or compounds of the type polycarbamide, PTFE and Mg-Al layered silicate (bentonite).

Thixotropy The characteristic of a lubricant in becoming temporarily softer/thinner under mechanical effects such as stirring or kneading. Greases behave in a thixotropic manner when their viscosity decreases as a result of mechanical strain and increases again when at rest. Preservative oils, especially those with additives, also behave thixotropically.

Toughness See section Viscosity, page 211.

V

Vapourisation loss

The loss in mass of an oil at higher temperatures through vapourisation. It may be equivalent to increased oil consumption and can change the characteristics of the oil.

Viscosity

A fundamental physical characteristic of oils. It indicates the inner friction of a fluid. In a physical sense, it is the resistance opposing the reciprocal displacement of the adjacent layers of a fluid.

A distinction is made between the dynamic viscosity η and the kinematic viscosity ν. The kinematic viscosity is the dynamic viscosity relative to the density ρ. The relationship $\eta = \rho \cdot \nu$ applies.

For dynamic viscosity, the SI units Pa · s and mPa · s are used. These replace the units Poise P and Centipoise cP that were commonly used in earlier times. Conversion: 1 cP = 10^{-3} Pa · s.

For kinematic viscosity, the SI units m^2/s and mm^2/s are used. These replace the unit Centistoke cSt that was commonly used in earlier times.

The viscosity decreases with increasing temperature and increases with decreasing temperature, see section Viscosity/temperature behaviour (V/T behaviour), page 211. For each viscosity value, the reference temperature must therefore be stated. The nominal viscosity is the kinematic viscosity at +40 °C, see section Viscosity classification, page 211.

Viscosity/temperature behaviour (V/T behaviour)

The change in viscosity with temperature. A favourable V/T behaviour is defined as one where the viscosity of an oil does not change considerably with temperature. See also section Viscosity index VI.

Viscosity classification

Subdivision of fluid industrial lubricants according to their viscosity (ISO 3448 and DIN 51519). There are 20 viscosity grades defined (in the range from 2 mm^2/s to 3 200 mm^2/s at +40 °C), see table Viscosity grades ISO VG, page 80.

Viscosity index improver

Additives that are dissolved in the oil and improve the viscosity/temperature behaviour. At high temperatures they induce higher viscosity, while at low temperatures they improve the flow behaviour.

Viscosity index VI

Indicator of the viscosity/temperature behaviour of an oil. See also section Viscosity/temperature behaviour (V/T behaviour).

Viscosity ratio

See section Reference viscosity, page 207.

W

Water content

The quantity of water contained in an oil.

Water reduces the lubrication capability, since the lubricant film is interrupted by water drops. It accelerates ageing and leads to corrosion.

The water content is determined by means of distillation or a settling test. In the settling test, water settles on the base of the test tube due to its higher specific gravity. Emulsions must first be heated.
In order to confirm a lower water content, the crackle test is used. The oil in the test tube is heated over a flame. If traces of water are present, a crackling noise will be heard. Further information on the influence of water on lubricants, see section Liquid contaminants, page 143.

Water resistance

The ability of a grease not to change its characteristics under the influence of water. This is determined by means of a static test in accordance with DIN 51807. It is tested whether and to what extent static, distilled water has an effect on a grease not subjected to load at various temperatures. The result only represents a description of characteristics and does not permit any conclusions as to the water resistance of the grease in practice.

Water separation capacity

The ability of an oil to separate from water. Testing is carried out in accordance with DIN 51589.

Worked penetration

Penetration measured at +25 °C of a grease sample that was preworked in the grease shaper (DIN 51804-2 and DIN ISO 2137).

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