Lubrication and maintenance

General

The choice of lubricant and lubrication method for a particular application depends primarily on the operating conditions, e.g. permissible temperatures or operating speeds, but may also be dictated by the lubrication of adjacent components (e.g. gear wheels).

For an adequate lubricant film to be formed between the rolling elements and raceways, only a very small amount of lubricant is required. With very small quantities, the hydrodynamic friction losses are small and operating temperatures can be kept down. Such minimal lubricant quantity can reliably be obtained using grease, and this method of lubrication is also becoming popular for spindle bearing arrangements. However, where speeds are very high, the bearings should be lubricated with oil, as the service life of grease is too short under such conditions.

Grease lubrication

Grease lubrication may be used for all bearings shown in this catalogue. The use of grease means that bearing arrangement design can be relatively simple because grease is more easily retained at the bearing position than oil, and it also contributes to sealing the bearing against contaminants and damp. Bear in mind that the grease applied should always be free from contaminants.

In most cases a lithium grease with a mineral oil base is suitable for high-precision bearings. Where demands for speed, temperature and service life are high, the use of bearing greases based on synthetic oils, e.g. the SKF grease LGLT 2 which has a diester oil base, have proved beneficial.

The lubrication of a bearing arrangement with a good quality grease in suitable quantity, permits relatively high speed operation without an excessive rise in temperature, compared with some other methods of lubrication. Grease lubricated bearing arrangements are therefore suitable for a wide speed range.

Grease selection
Lithium base greases with a mineral oil base are particularly suitable for the lubrication of rolling bearings and may also be used for high-precision bearings. These greases adhere well to the bearing surfaces and can generally be used in the temperature range –30 to +110 °C. This is sufficient for most applications. In cases involving special features (e.g. operating temperature below 50 °C or above 100 °C, very high or very low bearing speed, bearings subject to heavy load or shock loads, water resistance, compatibility) the following criteria can be adopted. First select consistency and base oil viscosity, check EP additives needed, then check for additional requirements (→ **Table 1**).

The method of selecting the required oil viscosity is explained in the section “Lubrication and maintenance” in the SKF General Catalogue or the SKF Interactive Engineering Catalogue.

The selection process is based on the elasto-hydrodynamic theory of lubrication (EHL). It is assumed that there is an abundant supply of oil to the contact to be lubricated (fully flooded inlet conditions). This is usually correct for oil lubrication, but for grease lubrication the situation can be quite different. In most grease lubricated bearings there is only a very minute amount of lubricant available in the actual contact between the rolling element and the raceway. This lubrication mode is called Starved Lubrication. The consequence of this is that with grease lubrication the lubricant film thickness is often much less than with oil lubrication and a correction has to be made when calculating the grease base oil viscosity.

From practical experience the following guidelines can be given.

If the calculation leads to a viscosity of more than 500 mm^2/s at 40 °C, then consider applying a correction factor for base oil viscosity of 0,5 or smaller. Greases with very high base oil viscosities should only be used in very special applications (typically very slowly rotating rolling bearings with continuous lubrication). If the calculated viscosity is lower, do not apply a correction

Consistency selection

Table 1

Consistency	Applications
NLGI 2 grade	Normal applications
NLGI 3 grade	Large bearings, vibration, high ambient temperatures, vertical shafts
NLGI 1 grade	Low ambient temperatures, oscillating applications

factor. If the required base oil viscosity is not high (typically below 20 mm^2/s at operating temperature), consider multiplying the calculated base oil viscosity by a factor 2.

Following from this empirical rule, selection of too high a grease base oil viscosity will impede the lubricant's access to the bearing contacts. And with increasing base oil viscosity, oil bleed also reduces. So with a very high base oil viscosity, the lubricant film actually becomes thinner instead of thicker.

Conversely, if the calculated oil viscosity is low, the lubricant film thickness can be increased by selecting a grease with a higher than calculated base oil viscosity.

Greases with EP additives may be used if bearings are subjected to heavy loads (e.g. C/P < 5), shock loads occur frequently, or if frequent start-up and shutdown occurs during the working cycle.

Use EP additive lubricants only if necessary. Certain EP additives are not compatible with some cage materials. Please consult the SKF application engineering service for further details.

Grease quantities

Bearings operating at high speeds, where it is desirable to keep the operating temperature low to ensure long grease life should be lubricated with small quantities of grease. In machine tool applications that mostly run at high speed the quantity should be lower than 30 % of the free space in a bearing. Freshly greased bearings should be run at low speed during a running-in phase so the grease will be evenly distributed within the bearing and excess grease can be ejected. If this running-in phase is neglected, risk of temperature peaking can lead to bearing failure later on.

From experience in the field, the most common filling quantities are about 10 % filling grade. Suggested quantities for high-precision bearings in machine tool applications are given in **Table 2**.

Pre-greased angular contact ball bearings may be delivered on request with the proper grease type and filling grade. Please consult SKF for availability and technical details.

Grease charges ➤

Table 2

Bearing bore diameter	Grease charge for bearings of series											
	719 CD 719 ACD 719 CX 719 ACX	70 CD 70 ACD 70 CX 70 ACX	72 CD 72 ACD 72 CX 72 ACX	719 CE 719 ACE	70 CE 70 ACE 70 CS	N 10	NN 30	NNU 49	BTM – A BTM – B	2344(00)	BSA 2 BSA 3[1]	BSD
mm	cm^3											
8	–	0,05	–	–	–	–	–	–	–	–	–	–
9	–	0,06	–	–	–	–	–	–	–	–	–	–
10	0,04	0,08	0,12	–	–	–	–	–	–	–	–	–
12	0,04	0,09	0,15	–	–	–	–	–	–	–	0,3	–
15	0,07	0,13	0,22	–	–	–	–	–	–	–	0,4	–
17	0,08	0,18	0,3	–	–	–	–	–	–	–	0,6	–
20	0,15	0,3	0,46	0,16	0,34	–	–	–	–	–	0,8	1
23,8	–	–	–	–	–	–	–	–	–	–	–	1,4
25	0,18	0,34	0,57	0,18	0,4	–	0,9	–	–	–	1	1,4
30	0,21	0,53	0,83	0,21	0,57	–	1,1	–	–	1,9	1,5	1,4
35	0,31	0,66	1,2	0,32	0,71	–	1,4	–	–	2,3	2	1,8
38,1	–	–	–	–	–	–	–	–	–	–	–	3,8
40	0,48	0,8	1,5	0,49	0,86	1,2	1,7	–	–	2,7	–	3,8 (1,6)[2]
44,4	–	–	–	–	–	–	–	–	–	–	–	5
45	0,54	1,1	1,8	0,55	1,1	1,3	1,9	–	–	3,2	–	5 (1,6)[2]
50	0,58	1,2	2,1	0,59	1,2	1,5	2,1	–	–	3,5	–	5
55	0,83	1,7	2,6	0,85	1,55	1,8	2,6	–	–	4,5	–	–
60	0,9	1,8	3,3	0,92	1,65	2	2,8	–	3,2	4,8	–	–
65	0,95	1,9	4,1	0,98	1,75	2,1	3	–	3,3	5,1	–	–
70	1,5	2,7	4,6	1,6	2,5	2,6	3,8	–	4,4	6	–	–
75	1,7	2,8	5	1,7	2,7	2,8	4	–		6,4	–	–
80	1,7	3,7	6	1,8	3,6	3,4	4,9	–	6,4	7,7	–	–
85	2,4	3,9	7,2	2,5	3,8	3,5	5,1	–	6,7	8,1	–	–
90	2,5	5	9	2,6	5	4,1	5,9	–	8,9	9,6	–	–
95	2,6	5,2	11	2,7	5,2	4,3	6,2			10	–	–
100	3,5	5,4	13	3,6	5,5	4,5	6,5	4,5	9,6	11	–	–
105	3,7	6,8	16	3,8	–	5,3	7,6	7		12	–	–
110	3,8	8,5	18	3,9	–	6,1	8,8	7,3	12	14	–	–
120	5,1	9	22	5,3	–	6,7	9,6	9	17	15	–	–
130	6,8	14	–	–	–	–	12	11	25,5	19	–	–
140	7,2	15	–	–	–	–	17	15	–	27	–	–
150	11	18	–	–	–	–	20	19	–	31	–	–
160	11	22	–	–	–	–	22	21	–	36	–	–
170	12	28	–	–	–	–	29	24	–	43	–	–
180	18	37	–	–	–	–	34	30	–	51	–	–
190	19	38	–	–	–	–	36	31	–	53	–	–
200	27	51	–	–	–	–	42	39	–	57	–	–
220	28	67	–	–	–	–	76	62	–	–	–	–
240	31	72	–	–	–	–	83	68	–	–	–	–
260	–	–	–	–	–	–	140	122	–	–	–	–
280	–	–	–	–	–	–	155	128	–	–	–	–

[1] Values for BSA 3 series are 1,7 times the values of BSA 2 series
[2] Values in brackets refer to smaller outside diameter where two different bearings have the same bore diameter (e.g. BSD 4072 C and BSD 4090 C)

Grease service life

Several methods are used to calculate the relubrication interval for grease lubricated bearings. However there are several important factors influencing the grease life, many of which are difficult to estimate. It is extremely complex to calculate precisely how long the grease can survive in a given application depending on the actual conditions. It is better to talk of grease life estimation, and the following data assists in making the best estimate. The graph (→ **Diagram 1 and Tables 3 and 4**) shows the theoretical relubrication interval t_f for high-precision bearings in various executions. The angular contact ball bearing curves refer to single bearings, so data for matched sets should be reduced depending on the arrangement as per **Table 3**.

For hybrid bearings the estimated grease service life can be obtained by multiplying the calculated value for the all-steel bearing by the factor given in **Table 4**.

Codes:

A = **A**ngular Contact Ball Bearings;
C = **C**ylindrical Roller Bearings;
T = Angular Contact **T**hrust Ball Bearings;
B = **B**all Screw Support Bearings.

Grease relubrication intervals guidelines

Diagram 1

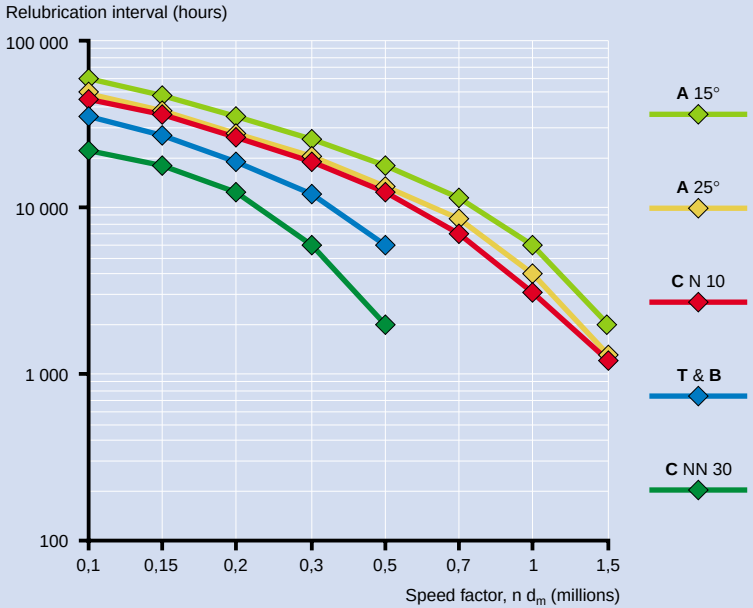


Table 3

Factors to calculate grease relubrication interval depending on bearing arrangement

Bearing design	Preload class Light	Medium	Heavy
A			
Set of 2	0,8	0,7	0,55
Set of 3	0,7	0,55	0,35
Set of 4	0,65	0,45	0,25
More	Contact SKF		
T, BTM series	1	–	0,5
B	As A	–	As A

Table 4

Ceramic material effect

$n\,d_m$ millions	A	T	C
0,5	3	3	3
0,7	3,5		3
1	3		3
1,5	2,8		2,5

Guidelines on correction factors for relubrication interval estimation

Table 5

Shaft position correction factor C_1	Vertical 0,5	Horizontal 1				
Bearing load correction factor C_2	C/P > 20 1	C/P > 10 0,7	C/P > 8 0,5	C/P > 5 0,3	C/P > 2 0,2	C/P >1 0,1
Reliability correction factor C_3	L_1 0,37	L_{50} 2				
Air flow-through correction factor C_4	Low 1	Moderate 0,3	Strong 0,1			
Damp & dust correction factor C_5	Low 1	Moderate 0,5	High 0,3	Very high 0,1		
Temperature correction factor C_6	40 °C > 1	55 °C > 1	70 °C 1	85 °C 0,5	100 °C 0,25	

The basic conditions for which **Diagram 1 page 66** has been drawn up are:

1) bearings are mounted on horizontal shafts
2) bearing operating temperature does not exceed 70 °C
3) a good quality lithium base grease is used
4) a relubrication interval at the end of which 90 % of the bearings are still reliably lubricated (L_{10} life).

The data from the graphs must then be multiplied by several factors related to the specific application data (➔ **Table 5 page 67**).

The relubrication interval then becomes

$$T_{relub} = t_f \times C_1 \times C_2 \times \ldots \times C_i$$

Other conditions such as the presence of water, cutting fluids, vibration etc may affect grease life.

Machine tool spindles often operate with working conditions that are not constant. If the speed spectrum is known and the lubrication interval for each speed is estimated, a total lubrication interval can be calculated with the following equation:

$$t_{f\,tot} = \frac{100}{\sum (a_i/t_{fi})}$$

where:
$t_{f\,tot}$ = total lubrication interval
a_i = part of the total cycle time at speed n_i, %;
t_{fi} = lubrication interval at speed n_i.

Changing the grease type

Where an alternative grease is considered for a certain application, its compatibility with the grease currently used should be checked first. **Tables 6 and 7** indicate compatibility of base oil and thickener type.

Before applying a new grease, the old one should be completely removed. Also, for a certain period during the early stages of running, regular checks, grease replacement and close monitoring of the bearings need to be made.

The above is based on grease composition and is an indication only, so in order to be certain, individual testing may be required.

The above procedure does not apply to PTFE thickener or silicone based greases for which bearings should be thoroughly washed (using appropriate solvents) before the new grease is applied. Always check that the new grease is suitable for the application.

Table 6

Compatibility of base oil types

	Mineral oil	Ester oil	Polyglycol	Silicone-methyl	Silicone-phenyl	Polyphenylether
Mineral oil	+	+	–	–	+	O
Ester oil	+	+	+	–	+	O
Polyglycol	–	+	+	–	–	–
Silicone-methyl	–	–	–	+	+	–
Silicone-phenyl	+	+	–	+	+	+
Polyphenylether	O	O	–	–	+	+

+ = compatible, – = incompatible, O = individual testing required

Compatibility of thickeners

Table 7

	Li soap	Ca soap	Na soap	Li complex soap	Ca complex soap	Na complex soap	Ba complex soap	Al complex soap	Clay	Polyurea
Li soap	+	O	–	+	–	O	O	–	O	O
Ca soap	O	+	O	+	–	O	O	–	O	O
Na soap	–	O	+	O	O	+	+	–	O	O
Li complex soap	+	+	O	+	+	O	O	+	–	–
Ca complex soap	–	–	O	+	+	O	–	O	O	+
Na complex soap	O	O	+	O	O	+	+	–	–	O
Ba complex soap	O	O	+	O	–	+	+	+	O	O
Al complex soap	–	–	–	+	O	–	+	+	–	O
Clay	O	O	O	–	O	–	O	–	+	O
Polyurea	O	O	O	–	+	O	O	O	O	+

+ = Compatible, – = Incompatible, O = Individual testing required

Running-in of greased bearings
A grease-lubricated bearing will initially run with a rather high frictional moment, and if the speed is high, the temperature rise can be considerable and excessive. The high frictional moment is due to churning of the grease, and it takes some time for the excess grease to work its way out of the contact zone and be forced away from the raceways. This can be minimized by applying a small quantity of grease and distributing the grease evenly on both sides of the bearing. Where possible, the adoption of spacers in between two adjacent bearings is also beneficial.

The time required to stabilize temperature depends on a number of factors – the type of grease, the grease charge, how the grease is applied to the bearings, the bearing type and internal design, and the running-in procedure. When properly run-in, the bearings work with minimal lubricant, giving the lowest frictional moment and temperature. The grease remaining at the sides of the raceways will act as a reservoir and the oil will bleed into the raceways, ensuring a safe lubrication for a long period of time.

Running-in can be done in several ways. The most common is to increase the speed in stages, waiting for the bearing temperature to stabilize before moving on to the next step. It is advisable to go one step more than the operating speed of the system, as this will ensure a lower temperature rise while operating. The temperature should be monitored during running-in to avoid large peaks which may later be detrimental to the grease life. As a general recommendation, the absolute temperature should be limited to approximately 60 – 65 °C. If running-in is done automatically on a test machine, it is preferable to set the machine with temperature alarms which will stop the spindle if the temperature rise exceeds fixed limits. Although easy to handle, automatic machines operating on time steps only, do not monitor the temperature that may go beyond acceptable levels.

Though widely used, the above procedure is time-consuming. Several hours may be required for a medium high spindle speed, as each step may take between 30 minutes and 1 – 2 hours before temperature stabilises. Total time for completing the running-in could be 8 – 10 hours.

The running-in time can be shortened considerably by using a few steps only and by starting at a speed approximately equal to 20 – 25 % of the bearing catalogue speed. This can significantly reduce the number of steps, but at each step the temperature increase may be very rapid. The temperature of the bearings must be carefully monitored with this procedure, and if possible should be measured on the bearing outer ring, using an automatic switch off when the temperature exceeds the above set limits.

After the outer ring has cooled down 5 – 10 °C the spindle is restarted at the same speed. The same procedure may have to be repeated several times, but the cycle time is just a few minutes. When a temperature peak, lower than the alarm limit, has been reached, the temperature will decrease rather rapidly and the bearing is then run-in at that particular speed.

Wherever possible and regardless of the procedure chosen, running-in should involve placing the spindle in rotation both clockwise and anti-clockwise.

Oil lubrication

General
Several methods of oil lubrication are available, which differ in specific characteristics. Oil lubrication is therefore recommended for many applications and can be adapted to suit the actual operating conditions and particular machine design. The most commonly used methods of oil lubrication are described below. For spindle bearing arrangements, the high operating speeds and requisite low operating temperatures generally necessitate the use of circulating oil lubrication with oil cooling, or the oil spot method. Depending on the method chosen, the following factors play an important part:

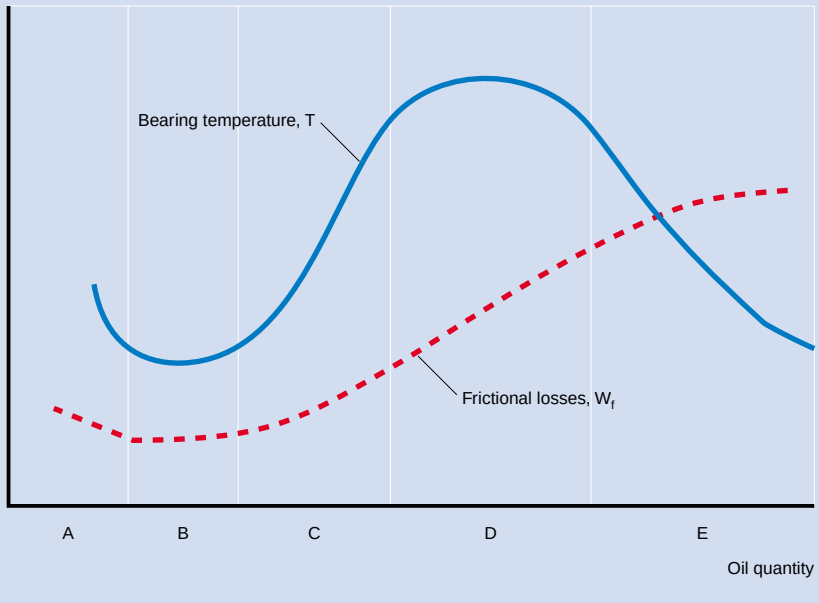
- quantity and viscosity of the oil
- speed and hydrodynamic friction loss which is a function of the speed
- permissible bearing temperature

The relationships between oil quantity, friction and bearing temperature are shown in **Diagram 2**. Where there is insufficient oil (region A), complete separation of rolling elements and raceways will not be achieved. Metallic contact will lead to increased friction and temperature, and finally to bearing wear. A cohesive, load-carrying oil film can only be formed if a greater quantity

Information on oil quantities for special applications may be obtained from SKF applications engineers.

Bearing temperature and frictional loss as a function of oil quantity

Diagram 2



of oil is available (region B). Here the condition is reached where friction and consequently temperature, are at a minimum. A further increase in oil quantity (region C) will result in increases in friction and temperature only until the quantity is such that an equilibrium is achieved between heat generation and heat loss, after which (region D) there is little change in temperature with oil quantity. If even more oil is added, the cooling effect predominates and the temperature starts to fall (region D-E). The conditions obtained with oil spot lubrication correspond to those of region B, and the conditions obtained with circulating oil lubrication where the oil is also cooled correspond to those of region E.

Common methods of oil lubrication
Several methods of oil lubrication are available, which differ in specific characteristics.

Oil bath lubrication
Oil bath lubrication is the simplest method of oil lubrication. The oil, which is picked up by the rotating components of the bearing, is distributed within the bearing and then flows back to the oil bath. Oil bath lubrication is particularly suitable for low speeds and enables design of relatively simple and economic bearing arrangements. At high speeds however, the bearings are supplied with too much oil, increasing friction within the bearing and causing the operating temperature to rise.

Circulating oil lubrication
With circulating oil lubrication, the lubricating oil is pumped to a position above the bearing where it runs down through the bearing. After the oil has passed through the bearing it is filtered and, if required, cooled before being returned to the bearing. Cooling the oil enables the operating temperature of the bearing to be kept at low level.

Oil drop lubrication
In this method, the bearing is supplied at given intervals with an accurately metered quantity of oil. The quantity supplied may be relatively small, so frictional losses at high speeds are small. However, it is not certain that the oil will penetrate the bearing at high speeds.

Oil jet lubrication
In the oil jet method a jet of oil under high pressure is directed at the side of the bearing. The velocity of the oil jet must be high enough (at least 15 m/s) so that at least some of the oil will penetrate the turbulence surrounding the rotating bearing.

This method is particularly efficient and is often used for high-speed bearing arrangements.

Oil mist lubrication
In oil mist lubrication, finely divided oil droplets are supplied to the bearing in a stream of compressed air. The air passing through the bearing serves to cool it and produces a slight excess pressure that enhances sealing. Minimum quantities of oil can be used however; in practice it is difficult to supply the bearing reliably with the very small quantities of oil involved. Oil mist is fairly costly and less and less accepted because polluting the immediate surroundings of the machine.

Oil spot (minimal) lubrication
With the oil spot method very small accurately metered quantities of oil are directed at each individual bearing by compressed air. The minimum quantity enables bearings to operate at lower temperatures than any other method of lubrication. The oil is supplied to the leads by a metering unit, it coats the inside surface of the leads and “creeps” along them. It is injected to the bearing via a nozzle. The compressed air serves to cool the bearing and also produces an excess pressure in the bearing arrangement that prevents contaminants from entering.

The speed rating for oil lubrication given in the product tables apply to oil spot lubrication.

When using the circulating oil, oil jet and oil spot methods it is necessary to ensure that the oil flowing from the bearing can leave the arrangement by adequately dimensioned ducts.

Circulating oil lubrication with additional cooling
This method is suitable for high-speed bearings provided an efficient oil cooling system and proper drainage ducts at both sides of the bearing are in place. Cooling the oil helps keeping bearing temperatures down, although the large quantities of oil and the increased frictional losses associated with the lubricant mean that more power is needed. Circulating oil lubrication with cooling system requires pumps and cooling devices and places considerable demands on the sealing. It is therefore relatively expensive.

The amount of oil required for circulating oil lubrication with additional cooling can be calculated approximately using

$$Q = (M_0 + M_1)\,(n\,\pi/\Delta T)10^{-6}$$

where
Q = requisite oil quantity, l/min
M_0 = load-independent frictional moment, see SKF General Catalogue, Nmm
M_1 = load-dependent frictional moment, see SKF General Catalogue, Nmm
n = bearing speed, r/min
ΔT = permissible increase in temperature of lubricating oil (difference between oil temperature before and after the bearing position), °C

The oil quantities determined using the above equation can be used as guideline values when selecting pumps and cooling arrangements. They are valid provided that the operating temperature of the shaft does not exceed ambient temperature by more than 30 °C; the bearing outer ring temperature may be 5 °C higher. With higher temperatures the formula may give wrong values, as the transfer of heat from the shaft is not considered. For accurate analysis, computer programs are available. Please consult SKF for details.

Table 8 gives guidelines for oil flow rate.

Table 8

Bearing bore diameter d		Oil flow rate	
over	incl.	low	high
mm		l/min	
–	50	0,3	1
50	120	0,8	3,6
120		1,8	6

Guidelines for oil flow rate

Oil spot lubrication

Oil spot lubrication enables reliable bearing lubrication to be achieved using extremely small quantities of oil.

Lower operating temperatures or higher speeds can be reached using this method of oil lubrication.

Guideline values for the oil quantity to be supplied to a bearing can be obtained from

$$Q = q\,d\,B \times 10^{-2}$$

where
Q = oil quantity, mm^3/h
d = bore diameter of bearing, mm
B = bearing width, mm

and q is a factor, see explanation below (→ **Table** 9).

The quantity varies depending on bearing type and design and this is determined by using different values of the factor q, i.e. should be 1 – 2 for roller bearings, and 2 – 5 for ball bearings. When speed is very high and angular contact ball bearings are used, the factor q will be much higher because of the pumping effect of angular contact ball bearings. For such cases a value of q between 10 and 20 should be used. Individual testing is always necessary however to optimise the conditions. Different bearing designs show varying sensitivity to oil quantity change, i.e. roller bearings are very sensitive to oil inlet changes while for ball bearings, the quantity can be changed substantially without any major effect on the bearing temperature rise.

Temperature rise and reliability with oil spot method depend to a large extent on the lubrication interval, i.e. the time in between two shots from the oil spot lubricator. Generally the lubrication interval is determined by the oil flow rate generated by each injector and the oil quantity supplied per hour. This interval can vary from one minute to one hour, with the most common interval being 15 – 20 minutes.

Pipes from the lubricator should be long enough, normally 1 – 5 m in length depending on the time interval between two subsequent shots. The air pressure should be 0,2 – 0,3 MPa, but must be increased when pipes are long to compensate for the pressure drop along the pipes.

To keep rise in temperature at the lowest possible level, good drainage must be designed. With horizontal spindles it is relatively easy to arrange drainage ducts at each side of the bearings involved. Where bearings with lubrication grooves are used, a drainage duct for the annular groove must also be considered.

For vertical shafts, the oil passing the upper bearings must be prevented from reaching the lower bearings which otherwise will receive too much lubricant. Drainage, together with a sealing device must be incorporated at the lower side of each bearing. An efficient seal should also be provided at the spindle nose to prevent lubricating oils reaching the workpiece.

Table 9

Bore code	q = 1	q = 2	q = 3	q = 5	q = 10	q = 15	q = 20
8	0,56	1,12	1,68	2,80	5,60	8,40	11,20
9	0,63	1,26	1,89	3,15	6,30	9,45	12,60
00	0,80	1,60	2,40	4,00	8,00	12,00	16,00
01	0,96	1,92	2,88	4,80	9,60	14,40	19,20
02	1,35	2,70	4,05	6,75	13,50	20,25	27,00
03	1,70	3,40	5,10	8,50	17,00	25,50	34,00
04	2,40	4,80	7,20	12,00	24,00	36,00	48,00
05	3,00	6,00	9,00	15,00	30,00	45,00	60,00
06	3,90	7,80	11,70	19,50	39,00	58,50	78,00
07	4,90	9,80	14,70	24,50	49,00	73,50	98,00
08	6,00	12,00	18,00	30,00	60,00	90,00	120,00
09	7,20	14,40	21,60	36,00	72,00	108,00	144,00
10	8,00	16,00	24,00	40,00	80,00	120,00	160,00
11	9,90	19,80	29,70	49,50	99,00	148,50	198,00
12	10,80	21,60	32,40	54,00	108,00	162,00	216,00
13	11,70	23,40	35,10	58,50	117,00	175,50	234,00
14	14,00	28,00	42,00	70,00	140,00	210,00	280,00
15	15,00	30,00	45,00	75,00	150,00	225,00	300,00
16	17,60	35,20	52,80	88,00	176,00	264,00	352,00
17	18,70	37,40	56,10	93,50	187,00	280,50	374,00
18	21,60	43,20	64,80	108,00	216,00	324,00	432,00
19	22,80	45,60	68,40	114,00	228,00	342,00	456,00
20	24,00	48,00	72,00	120,00	240,00	360,00	480,00
21	27,30	54,60	81,90	136,50	273,00	409,50	546,00
22	30,80	61,60	92,40	154,00	308,00	462,00	616,00
24	33,60	67,20	100,80	168,00	336,00	504,00	672,00
26	42,90	85,80	128,70	214,50	429,00	604,50	858,00
28	46,20	92,40	138,60	231,00	462,00	693,00	924,00
30	52,50	105,00	157,50	262,50	525,00	787,50	1 050,00
32	60,80	121,60	182,40	304,00	608,00	912,00	1 216,00
34	71,40	142,80	214,20	357,00	714,00	1 071,00	1 428,00
36	82,80	165,60	248,40	416,00	828,00	1 242,00	1 656,00
38	87,40	174,80	262,20	437,00	874,00	1 311,00	1 748,00
40	102,00	204,00	306,00	541,00	1 020,00	1 530,00	2 040,00
44	123,20	246,40	369,60	616,00	1 232,00	1 848,00	2 464,00
48	134,40	268,80	403,20	672,00	1 344,00	2 016,00	2 688,00

Factor q for oil spot lubrication and guidelines for angular contact ball bearings series 70

Position of the oil nozzles

Oil nozzles should be correctly positioned to avoid difficulties for the oil to enter the contact area between rolling element and raceway and so that it does not impinge on the cage. **Table** 10 gives the values of the diameters (from the shaft axis) where oil injection should take place for the most common bearing designs and series (➔ **fig** 1).

The data shown in **Table** 10 refers to bearings with their standard cage designs. For bearings fitted with other cages or bearing types not shown, SKF should be consulted.

Lubricating oils

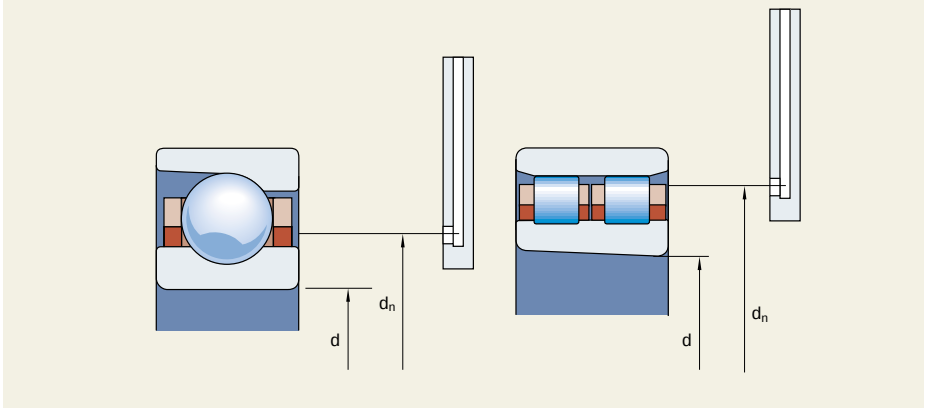
For the lubrication of high-precision bearings, high quality lubricating oils without additives should be considered. The requisite viscosity of the oil can be determined following the recommendations in the SKF Interactive Catalogue and is essentially a function of bearing size, speed and operating temperature.

The intervals at which the oil should be changed when using the oil bath, circulating oil and oil jet methods depend mainly on the operating conditions and the quantities of oil involved. Further information may be found in the SKF Interactive Catalogue, or obtained on request from the oil suppliers.

With oil spot lubrication systems there is no restriction as to oil type, and the oil viscosity may be much higher than for oil mist systems. Oils with 40 up to 100 mm^3/s viscosity at 40 °C are typically used, as are oils with EP additives that are preferable especially with roller bearings.

Where oil drop, oil mist or oil spot lubrication is applied, the oil is “lost”, i.e. it is only supplied to the bearing once.

Fig 1



Oil nozzles position

Oil nozzle position for different bearing design ➤

Table 10

Bore code	Bore diameter	Oil nozzle position d_n for bearings of series 719 CD, 719 ACD 719 CX 719 ACX	719 CE, 719 ACE	70 CD, 70 ACD 70 CX 70 ACX	70 CE, 70 ACE	72 CD, 72 ACD 72 CX 72 ACX	NNU 49	NN 30, N 10[1]
	mm	mm						
8	**8**	–	–	13,6	–	–	–	–
9	**9**	–	–	15,1	–	–	–	–
00	**10**	14,8	–	16,3	–	18,2	–	–
01	**12**	16,8	–	18,3	–	20,0	–	–
02	**15**	20,1	–	21,8	–	23,0	–	–
03	**17**	22,1	–	24,0	–	25,9	–	–
04	**20**	26,8	26,8	28,7	28,8	31,1	–	–
05	**25**	31,8	31,8	33,7	33,8	36,1	–	40,5
06	**30**	36,8	36,8	39,7	40,0	42,7	–	47,6
07	**35**	43,0	43,0	45,7	46,0	49,7	–	54,0
08	**40**	48,7	48,7	51,2	51,5	55,6	–	60,0
09	**45**	54,2	54,2	56,7	57,2	60,6	–	66,4
10	**50**	58,7	58,7	61,7	62,2	65,6	–	71,4
11	**55**	64,7	64,7	68,7	69,7	72,6	–	79,8
12	**60**	69,7	69,7	73,6	74,7	79,5	–	85,0
13	**65**	74,7	74,7	78,6	79,7	86,5	–	89,7
14	**70**	81,7	81,7	85,6	86,7	91,5	–	98,5
15	**75**	86,7	86,7	90,6	91,7	96,5	–	103,5
16	**80**	91,7	91,7	97,6	98,7	103,5	–	111,4
17	**85**	98,6	98,6	102,6	103,7	111,5	–	116,5
18	**90**	103,6	103,6	109,5	110,6	117,5	–	125,4
19	**95**	108,6	108,6	114,5	115,6	124,4	–	130,3
20	**100**	115,6	115,6	119,5	120,6	131,4	113,8	135,3
21	**105**	120,6	120,6	126,5	–	138,4	119,0	144,1
22	**110**	125,6	125,6	133,5	–	145,9	124,0	153
24	**120**	137,6	137,6	143,5	–	158,2	136,8	162,9
26	**130**	149,5	–	157,5	–	170,7	147,0	179,6
28	**140**	159,5	–	167,4	–	–	157,0	188
30	**150**	173,5	–	179,4	–	–	169,9	201,7
32	**160**	183,5	–	191	–	–	179,8	214,4
34	**170**	193,5	–	205,8	–	–	189,8	230,8
36	**180**	207,4	–	219,7	–	–	203,5	248,9
38	**190**	217,4	–	229,7	–	–	213,0	258,9
40	**200**	231,4	–	243,2	–	–	227,0	275,3
44	**220**	251,4	–	267,1	–	–	247	302,4
48	**240**	271,4	–	287	–	–	267,0	322,4
52	**260**	–	–	–	–	–	294,5	355,2
56	**280**	–	–	–	–	–	313,5	375,3

[1] For N 10 series equipped with TNHA cages please contact the SKF application engineering service

Maintenance

Bearings storage
Bearings can be stored in their original packages for years, provided relative humidity in the storage room does not exceed 60 % and there are no great fluctuations in temperature. Humidity and temperature must be controlled during storage, handling and transport if at all possible, particularly in tropical areas. Bearings should be kept in a vibration-free dry place where the relative humidity and temperature are reasonably constant.

Bearings that are not stored in their original packages should be well protected against corrosion and contamination.

Large rolling bearings should only be stored lying down, and preferably with support for the whole extent of the side faces of the rings. If kept in a standing position, the weight of the rings and rolling elements can give rise to permanent deformation because the rings are relatively thin-walled.

Lubricant storage
Most materials including oils and greases deteriorate with time. The art of good storage practice is to have materials always available when required, and to ensure stock turnover so that lubricants are used before any significant performance loss has occurred. Lubricant properties may vary considerably during storage due to exposure to air/oxygen, temperature, light, water and moisture, oil separation and presence of particles.

The recommended maximum storage time is 2 years for greases and 10 years for lubricating oils, assuming reasonable stock keeping practices and protection from excessive heat and cold are followed.

A lubricant in excess of the recommended shelf life is not necessarily unsuitable for service but it is advisable to check if it still meets the product requirements/specifications.

Dismounting and mounting

Dismounting

Proper maintenance of spindles is essential to their performance, and replacing the bearings in the correct way using the right tools is a major part of that maintenance.

Before disassembly begins, a suitable working area should be prepared and the proper tools made available.

The working area should be clean and away from areas where cutting and grinding operations take place. No traverses should pass over the working place. It is best to use a separate room if possible, one that is temperature controlled, dedicated to handling of accurate components and easy to keep clean.

Tool requirements differ depending on spindle design, but use of the correct tools makes the work easier and more efficient, and avoids damaging the components.

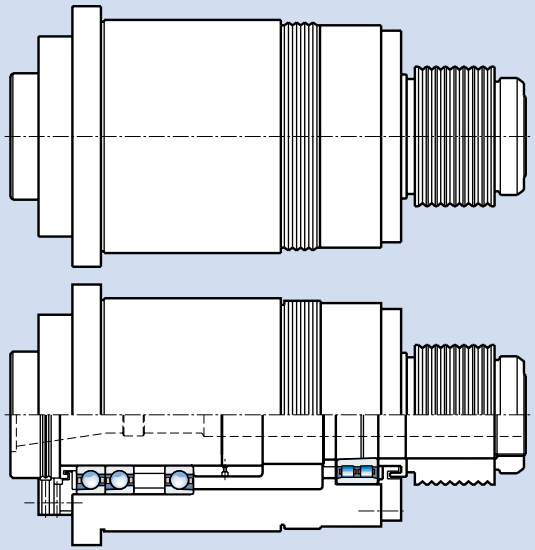
Information on mounting and dismounting tools can be found in the catalogue **SKF Maintenance and Lubrication Products.**

A detailed drawing of the spindle should also be available.

Dismounting a spindle with bearings arrangement 70 CD/TBT at the work side and NN 30 K at the drive side.
To illustrate the procedure for replacing the bearings of a spindle, one of the most common spindle designs is chosen here as an example. The spindle is a cartridge type and thus the complete spindle can be easily removed from the machine. The spindle is belt driven and the pulley is fitted directly at its rear. At the work side the spindle is equipped with a set of three angular contact ball bearings of series 70 ACD. At the drive side is a cylindrical roller bearing of series NN 30 K. This bearing is mounted on a tapered seating. The spindle has a flange at the nose that is common for lathe and milling spindles. Thus the bearings have to be removed and fitted from the drive side (➔ **fig 1 page 80**).

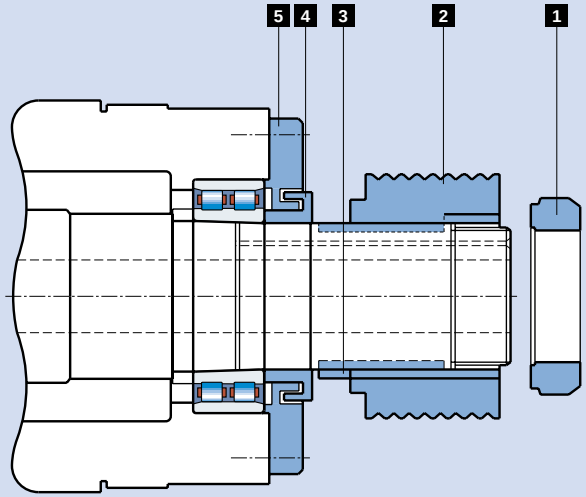
The bearings are lubricated with grease – the most common type of lubrication today. Clean the outside of the spindle before placing it on the worktable.

Fig 1



Spindle with bearing arrangement 70 CD/TBT at the work side and NN 30 K at the drive side

Fig 2



Removal of nut, pulley, sealing device and housing cover

Removing drive side bearings

Place the spindle on V-blocks or another arrangement, depending on external design of the spindle, for easy handling.

Unscrew the rear side nut (1), remove the pulley (2) with the keys (3) and the sealing device (4). Remove the housing cover (5) (➔ **fig** 2).

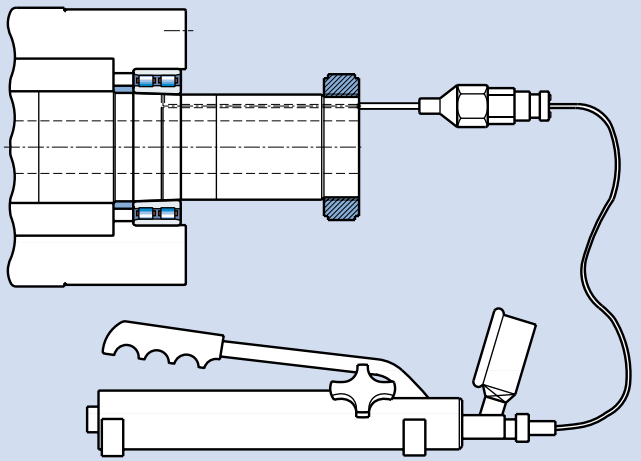
As the drive side bearing has a tapered bore and the shaft is provided with an oil duct and oil groove for this bearing, it is easily removed by using a hydraulic pump. (➔ **fig** 3).

1 Connect the nipple to the oil duct and then connect the oil pump to the nipple and tighten the release knob on the pump.
2 Put the rear side nut on the spindle to prevent the bearing falling off the spindle when released.
3 Inject the oil. The bearing **inner ring** will be released when the pressure is about 20 MPa. The spacer (6) (➔ **fig** 4 **page 82**) is taken away after removal of the shaft.

If the spindle is not provided with an oil duct and groove, removal of the bearing must wait until the shaft has been withdrawn from the housing.

Removing the drive side bearing: connect the nipple to the oil duct in the shaft

Fig 3



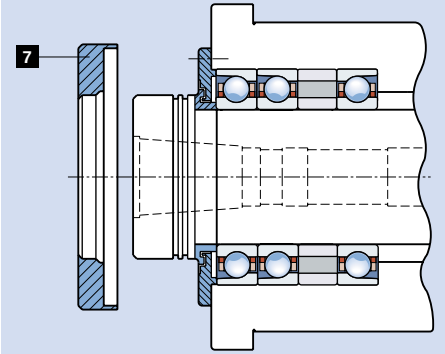
SKF pump 729124 for pressure up to 100 MPa

Withdrawal of the shaft from housing. The spindle is equipped with a reinforced labyrinth seal at the work side. Remove the external component (7) (➔ **fig** 5). Normally some force is necessary to withdraw the shaft from the cartridge, as in most cases the work side bearings have a certain interference fit. The required withdrawal force expressed in N for a set of three bearings can be estimated at 20 times the outside diameter of the bearing expressed in mm. Example: Bearing set 7020 ACD/P4ATBTB having an outside diameter of 150 mm will require a withdrawal force of approximately $20 \times 150 = 3\,000$ N.

If the bearings are to be used again, e.g. after being relubricated, great care must be taken to avoid damaging them during disassembly of the spindle. Blows to the shaft as a method of withdrawal must be avoided as this can easily create serious indentations on the raceways, making the bearings unfit for further use.

Use a puller to withdraw the shaft, rotating the shaft during withdrawal to minimise the risk of damage. If no suitable puller is available, one can be made according to the illustration (➔ **fig** 6). The bar through the shaft is threaded at both ends. Turn the nut at the work side of the spindle to withdraw the shaft and bearings. While doing this, rotate the shaft to avoid damage to the bearings if the fit is tight. If there is no bore through the shaft, arrange a suitable attachment of the bar to the spindle nose.

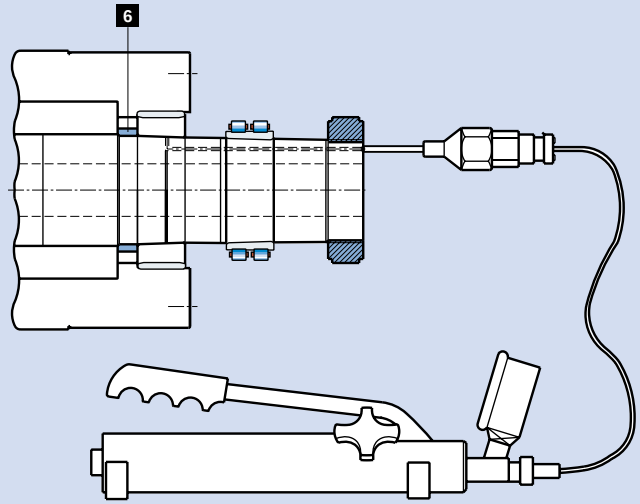
Fig 5



Withdrawal of the shaft from housing: removal of the external cover

Removing the drive side bearing: inject the oil until the inner ring is released

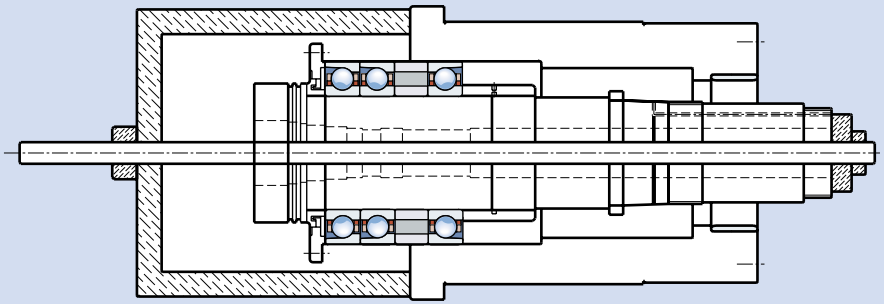
Fig 4



SKF pump 729124 for pressure up to 100 MPa

Removal of the shaft using a puller

Fig 6



Removing drive side bearings
If the drive side bearing has not already been removed by the oil injection method this should now be done.

A puller to grip the side face of the spacer (6) is used. Do not pull over the roller set as this could make the puller lose its grip, damaging the bearing and making it unfit for further use. The required pulling force expressed in N is about 300 times the bore diameter expressed in mm. As the seating is tapered the pulling force acts only to release the bearing (➔ **fig** 7).

The outer ring of the cylindrical roller bearing is still seated in the housing. Remove it using a puller, gripping the inner side-face of the ring. Do not put tools on the raceways if the bearings are to be used again. If you do not need to check the bearings, these may be left on site. Often the housing is provided with two diametrically opposed slots for gaining access with puller to the bearing inner side face (➔ **fig** 9).

Removing the locating device for the work side bearings
The locating device for the work side bearings in this example is a so-called stepped sleeve. It is kept on the shaft by a rather heavy interference fit and ensures very good accuracy of the spindle.

This method eliminates the need for threaded components and the inaccuracy that the threads can create. The stepped sleeve is fastened on the spindle by an interference fit that must be heavy enough to withstand the axial forces. The sleeve is mounted and dismounted by the oil injection method, thus ensuring careful handling of the components. The step that acts as a pressure surface during the mounting and dismounting operation can be sealed in two different ways; by the sleeve having two bore diameters, each with an interference fit on the shaft, or by using an O-ring. The latter system is somewhat easier from a manufacturing point of view.

Removal of the drive side bearing using a puller

Fig 7

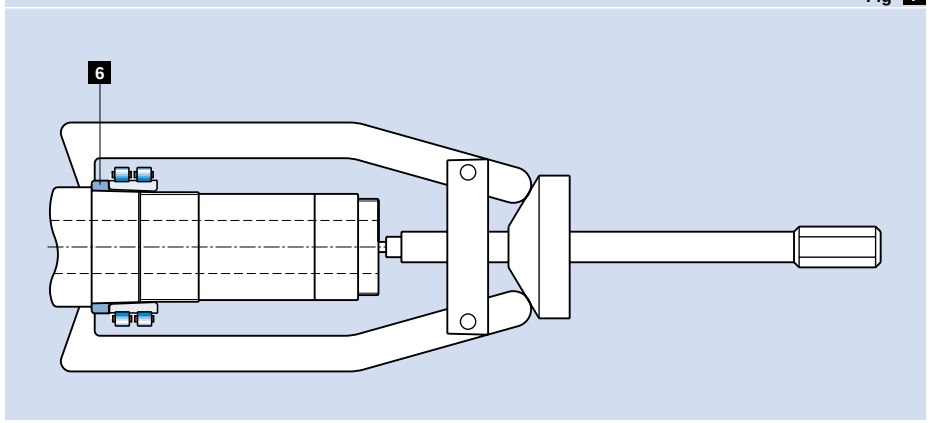


Fig 8

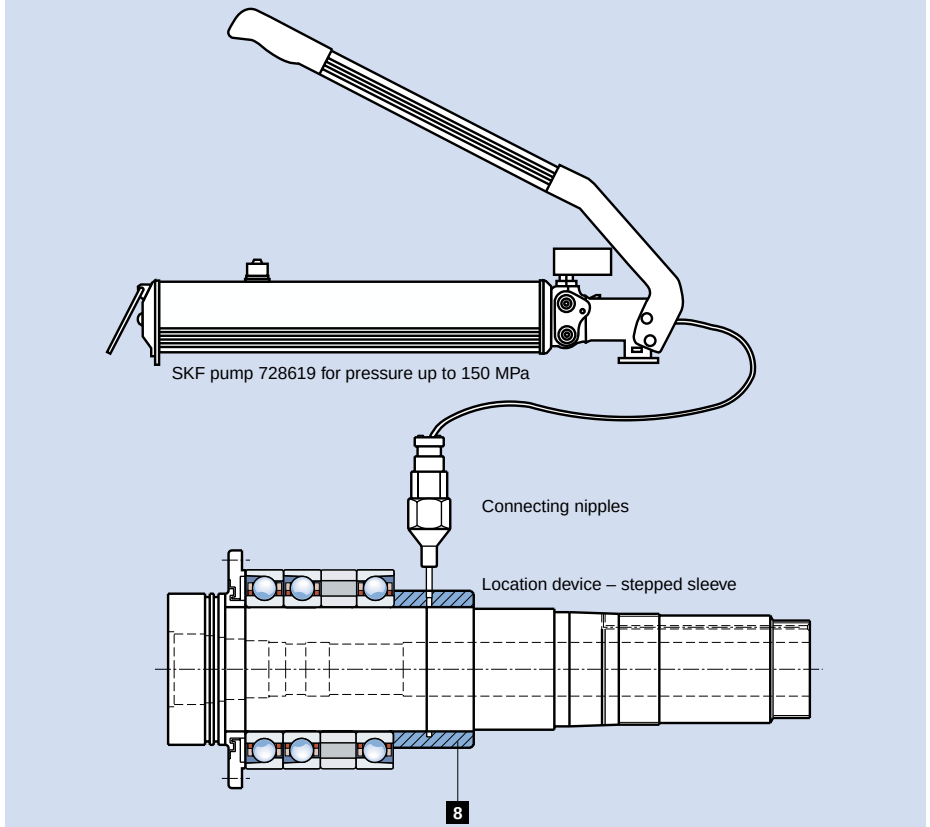
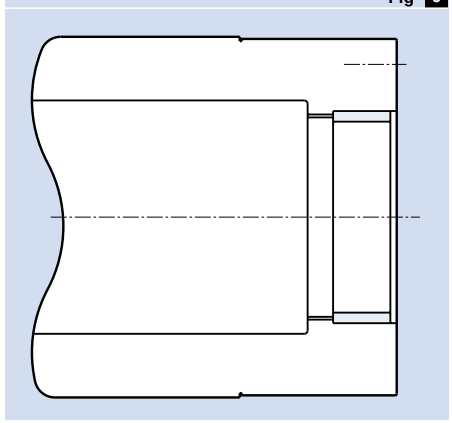


Fig 9



Removal of the outer ring of the cylindrical roller bearing

Removal of the locating device for the work side bearing

Removal of the stepped sleeve requires a hydraulic pump. (➔ **fig** 8 **page 85**). The pump should have a capacity of at least 100 MPa. The same injector as that for releasing bearings on tapered seatings can be used.

Connect the injector to the nipple that has been attached to the threaded hole of the sleeve. Put cloths around the shaft to dampen the sleeve when it comes loose. Be careful with all connections, as the oil pressure is high. Use only components that are certified for the pressure that can be delivered by the pump.

A suitable oil for dismounting and mounting the sleeve is the SKF fluid LHDF 900.

Start pumping and the sleeve will loosen when the pressure has reached a value of 60 – 100 MPa depending on size and actual fit.

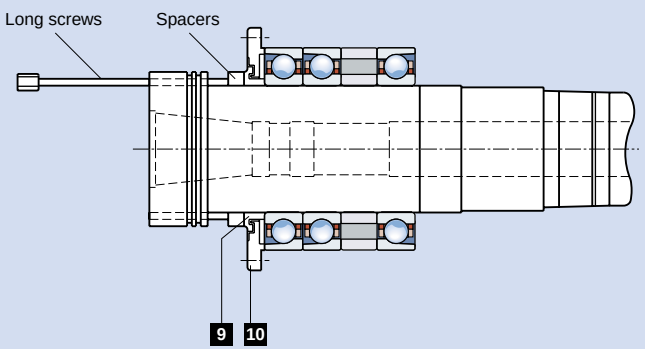
If the shaft or sleeve has been incorrectly manufactured, or if any of these components have been damaged, the sleeve may not come off completely. The oil will leak and there will be insufficient build up of pressure. If this is expected to happen the sleeve can be pushed off by hand while the oil is injected. If it remains stuck, use a powerful puller. Look for any potential damage to the components to avoid problems in the future.

Removal of work side bearings

The work side bearings normally have a light interference fit on the shaft. The withdrawal force to be expected is 70 times the bore diameter for a set of three bearings, force in N and diameter in mm. The spindle used in the example is provided with three threaded holes in the spindle nose for dismounting purposes. By using these holes and three long screws the bearings can easily be removed from the shaft without any risk of damage. Use spacers between the screw ends and the sealing/spacer component (9) to avoid damaging the surface of this component. If the bearings have to be pushed a long distance, spacers with different lengths can be made to avoid screws being too long. If the spindle nose is not provided with dismounting holes, a long puller gripping the cover (10) or preferably the labyrinth/spacer (9) can be used (➔ **fig** 10).

Removal of the external cover using three long screws

Fig 10



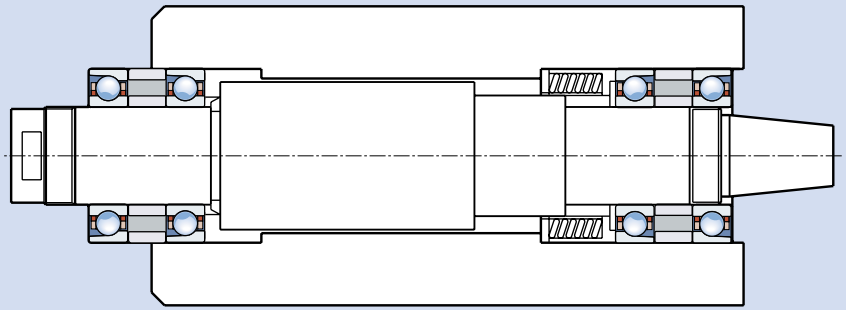
Dismounting spindles with other bearing arrangements

Spindle with tandem set of angular contact ball bearings at both sides (➔ fig 11)

A spindle having a tandem set of angular contact ball bearings at each side normally requires the drive side set to be withdrawn from the shaft at the same time as the shaft is withdrawn from the housing. The withdrawal force is then acting over the outer rings of this set and the shaft should be rotated during the withdrawal operation to minimise the risk of damage to the bearings.

Spindle with tandem arrangement of angular contact ball bearings at both sides

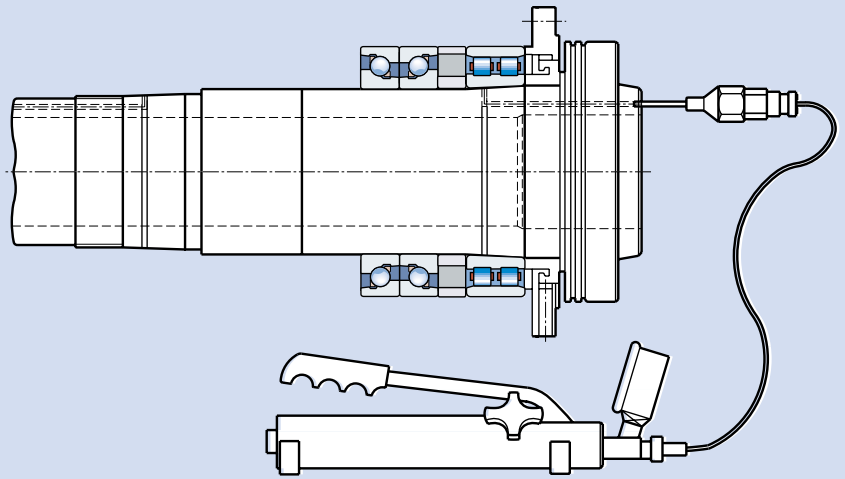
Fig 11



Spindle with cylindrical roller bearing and angular contact thrust ball bearing at work side
A cylindrical roller bearing at the work side of the spindle is dismounted in the same way as described for the drive side bearing. For removing this bearing, together with the angular contact thrust ball bearing generally present in this position, the system with screw should preferably be used if provided. Otherwise a puller is used. It should grip over the inner ring of the cylindrical bearing or over the shaft sealing-washer. Depending on the design, access may be difficult and it might be necessary to pull over the housing cover and the outer rings. Angular contact thrust ball bearings normally have only a light fit and the withdrawal force is minimal (→ **fig** 12).

Spindle with cylindrical roller bearing and angular contact thrust ball bearing at work side

Fig 12



SKF pump 729124 for pressure up to 100 MPa

Mounting

Mounting a spindle with bearings arrangement 70 CD/TBT at the work side and NN 30 K at the drive side
Before mounting the bearings, a grease of appropriate quality and quantity should be applied. For best performance of the spindle, a grease with synthetic base oil should be used. See section Grease lubrication under 'Lubrication' **page 62** for details.

The NN 30 K bearing at the drive side of the spindle should be adjusted to its final position by means of the spacer at the large end of the taper. This bearing should therefore be lubricated only when it is finally to be mounted.

For bearings that will be used at relatively high speeds, it is recommended that the rust-inhibiting compound is washed away and the bearings are dried before applying the grease. The properties of the grease are then better utilised. Special care must be taken to ensure that the bearings are not contaminated during washing.

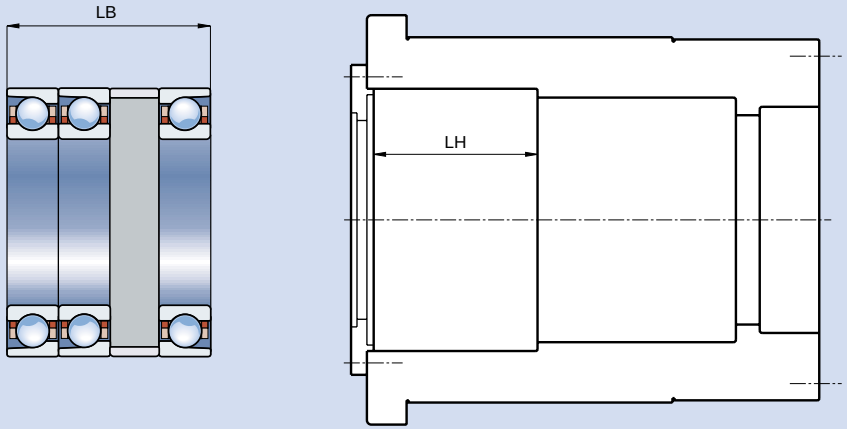
Using a syringe to apply the grease is a convenient way to get the proper quantity, and makes it easy for feeding inside the bearing. Distribute the grease evenly around the rolling element set.

Mounting the work side bearings

1 Check that the distance between the housing seating abutment and housing cover side face LH, is smaller than the total width of the bearing package LB, measured over the outer rings. If not there will be an axial play in the spindle. It may be recommended to have LH 10 – 15 µm smaller for a diameter around 100 mm. Too large a difference may induce deformations when tightening the screws (→ **fig** 13).

Mounting the work side bearings

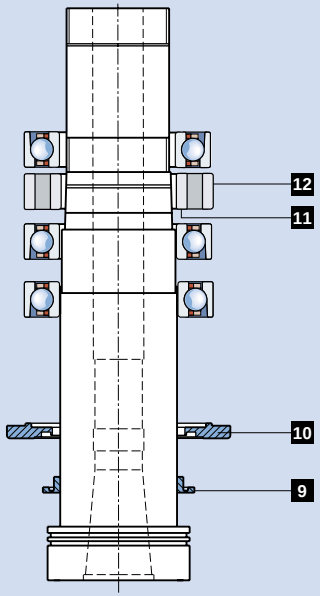
Fig 13



2 Particularly when large bearings are used, the spindle should preferably be kept in a vertical position during the mounting procedure (➔ **fig** 14).
3 Position the shaft labyrinth (9); it may have to be warmed slightly, and the housing cover (10). Be sure not to forget the housing cover!
4 Warm the bearings on a hot plate or induction heater; 20 – 30 °C above ambient is normally sufficient to allow the bearings to pass freely over the shaft.
5 Ensure that the bearings are facing in the correct direction. If using a matched set look for the markings on the outside diameter of the outer rings.
6 Align the rings so that the marking for the “thickest” part is at the same position for the inner rings as for the outer rings. This will ensure the best possible division of load between the bearings.
7 Smear the bearing seatings lightly with a thin oil.
8 Position the bearings on to the shaft, not forgetting the two spacers (11) and (12) between the second and third bearing. The shaft spacer may have to be warmed.

Mounting the work side bearings with the spindle in vertical position

Fig 14



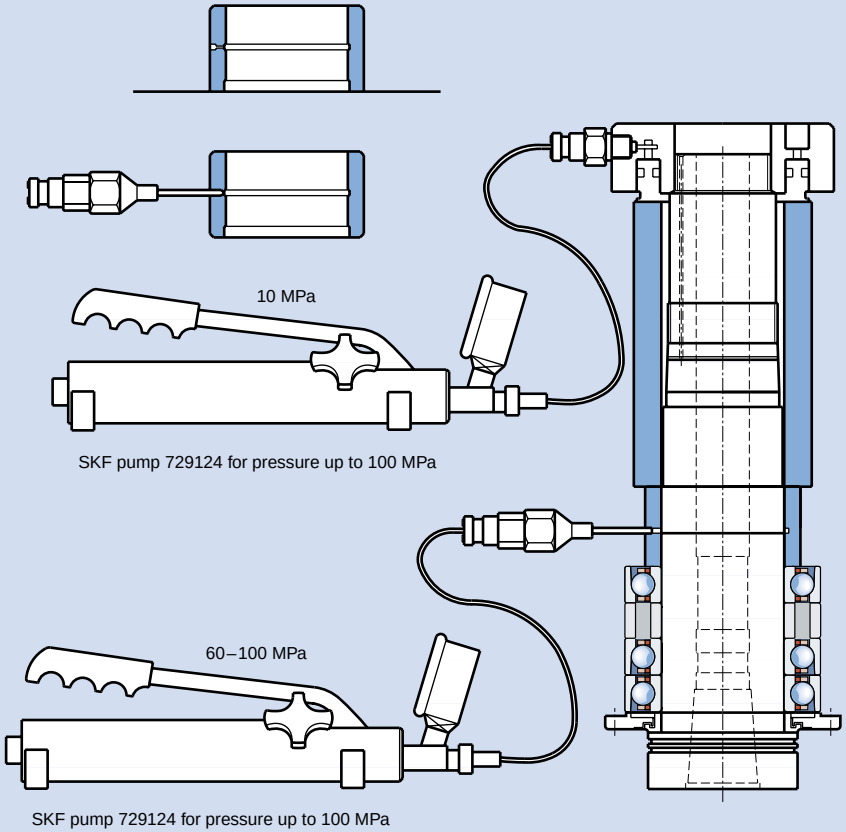
Do not forget to grease the bearings before mounting.

Mounting the stepped sleeve (➔ fig 15)

1 Heat the stepped sleeve to about 150 °C above ambient for smaller sizes (bore diameter ⩽ 60 mm) and about 110 °C for larger sizes, preferably with an induction heater.
2 When heated up, the sleeve can be easily handled by using the connecting nipple as a handle. Put the sleeve quickly on to the shaft so that the shaft does not heat up before the sleeve comes into contact with the bearings.
3 The sleeve should be allowed to cool down to room temperature before proceeding.
4 For final adjustment of the sleeve the oil injection method should be used.
5 Connect the nipples and the oil injector to the sleeve.
6 Provide for an arrangement that will apply an axial load over the sleeve-bearing system to withstand the force created by the oil injected to the sleeve, and to overcome the bearing fits and preload.

Mounting the stepped sleeve

Fig 15



7 Use a distance sleeve with length sufficient to cover the distance from the stepped sleeve to the position where the shaft is threaded – in this case the threads for the nut locating the drive side bearing. Use the nut to apply the necessary axial load or better still, use a hydraulic nut type SKF HMV E. A second oil pump must then be used. If the threads on the spindle do not match the threads of the hydraulic nut, a larger nut can be used and supported by the spindle nut. If there is no thread at all on the spindle, a support can be arranged, e.g. a washer that can be attached to the rear end of the spindle.

8 Tighten the nut or apply a pressure of about 30 MPa on the hydraulic nut. Inject oil to the stepped sleeve until it floats. There will probably be some leakage of oil. The required oil pressure is 60 – 100 MPa. While the sleeve is floating, tighten the nut or check that the required pressure for the hydraulic nut is still present.

9 The bearings must be protected from the pressure fluid as this can adversely affect the lubrication properties of the grease. Wrap a clean lint-free cloth around the sleeve where it contacts the bearing to absorb any pressure fluid that may leak out.

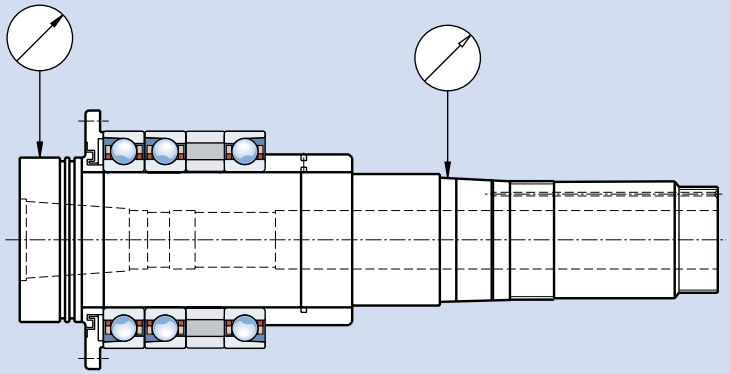
10 Release the pressure for the stepped sleeve, wait until the oil has drained and then release the axial load.

At this stage it is advisable to check the straightness of the spindle supported only by the work side bearings. Put the spindle on V-blocks as far apart as possible, and supporting the bearing outer rings. Rotate the shaft and measure the run-out at different positions of the shaft where it has suitable surfaces (→ **fig** 16).

If the values seem to be abnormal, a possible reason may be that the shaft has become misaligned due to spacers not having parallel side faces or shaft abutment/sleeve side face not being perpendicular to the bearing seating. Check by releasing the clamping load over the bearings.

Checking the shaft run-out

Fig 16



Mounting the drive side bearings

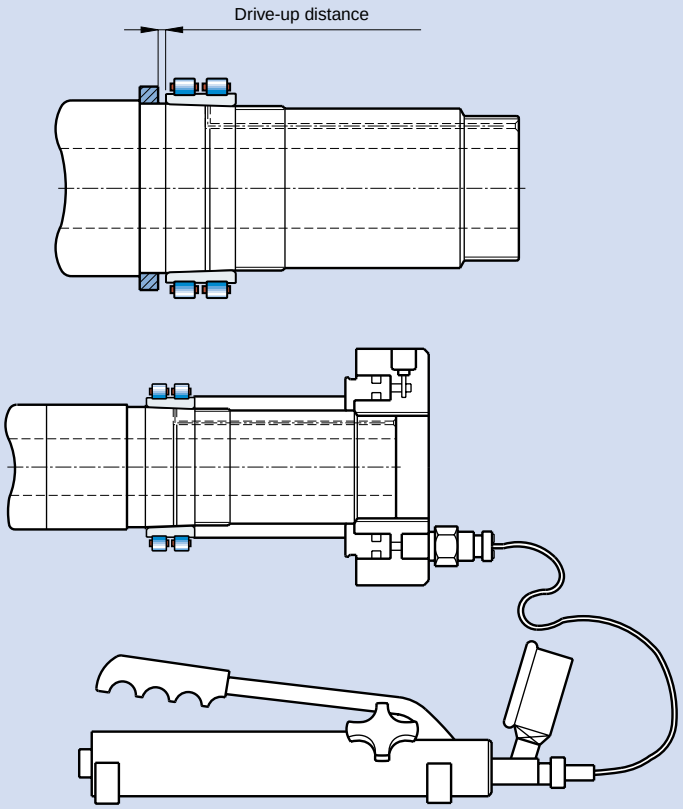
Drive side bearings with tapered bore should be adjusted to a suitable clearance or preload. The adjustment is made by pushing the inner ring up on the seating. The raceways will then expand. The final expansion determines the clearance or preload of the mounted bearing. The distance ring should have a width such that when the bearing is pushed up against the ring it will have the required clearance/preload (→ **fig** 17).

A special gauge, an internal clearance gauge, of series GB 30 is available for accurate adjustment of the clearance/preload of bearings of series NN 30 K. This type of gauge is generally used by machine tool manufacturers and can usually be hired from SKF for occasional use (→ **fig** 18 **page 94**).

1 The bearing outer ring need to be fitted in the housing. If the outer ring is to have a tight fit, the housing should be heated to between 10 and 30 °C above room temperature, e.g. in an oven or an oil

Mounting high-precision cylindrical roller bearings with GB 30 gauge

Fig 17



SKF pump 729124 for pressure up to 100 MPa

bath. The heated housing with outer ring should then be allowed to cool down to ambient temperature. All components as well as the gauges must have the same stable temperature during the measuring procedure (➔ **fig** 19).

2 The bore gauge need to be introduced into the outer ring raceway, and the indicator set to zero. The raceway diameter measured in this way is transferred to the GB 30 gauge (➔ **fig** 20).

3 The bore gauge is applied to the centre of the gauging zone of the GB 30 gauge. The screw of the latter gauge is then adjusted until the indicator of the bore gauge shows zero minus a correction factor. The correction factor is given in the instructions supplied with each GB 30 gauge (➔ **fig** 22 **page 96**).

4 The internal diameter of the envelope diameter gauge need to be reduced by the value of the desired clearance, or increased by the value of the desired preload using the adjustment screw. The indicator on the envelope diameter gauge need to be set to zero. The setting of the indicator should be left undisturbed from now on (➔ **fig** 25 **page 97**).

5 The tapered bearing seating on the spindle need to be lightly oiled with thin oil. The inner ring with roller and cage assembly need be lightly driven up on the tapered seating. The envelope diameter gauge need to be expanded using the adjustment screw and the gauge placed in position over the roller set. The adjustment screw need then to be turned in the opposite direction until the gauge, by virtue of its inherent resilience, is in contact with the roller set.

6 The inner ring is driven further up on to its seating until the indicator on the gauge again shows zero. The gauge need to be expanded using the adjustment screw and removed. The required drive-up force in N is 200 – 400 times the bearing bore diameter in mm.

7 The distance between the mounted inner ring and the shoulder on the spindle need to be measured using gauge blocks. Make

Fig 18



Fig 19

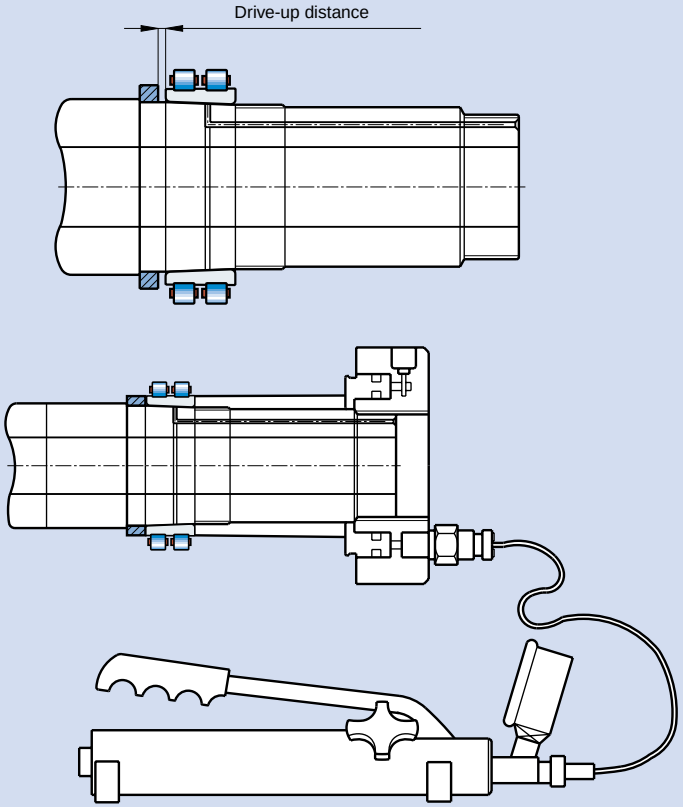


Fig 20



Pushing the inner ring up on the shaft until it firmly abuts the spacer ring

Fig 21



SKF pump 729124 for pressure up to 100 MPa

measurements at different positions for checking accuracy and misalignment. The difference should not normally be larger than 3 to 4 µm. The spacer ring need then to be machined so that its width corresponds to this dimension (➔ **fig** 23).

8 The inner ring with roller and cage assembly need to be withdrawn from the spindle. The finished spacer ring need to be pushed up on to the spindle until it abuts the shoulder. The inner ring need to then again be driven up until it firmly abuts the spacer ring. There should be no clearance between shoulder, spacer ring and inner ring (➔ **fig** 21 **page 95**).

9 The envelope diameter gauge need to be applied to the roller set as described under point 5. The indicator must again show zero (➔ **fig** 24).

10 The bearing need to be secured on the spindle. It is now time to apply the grease to the bearing. Put one string around each roller set and distribute it lightly over the rollers. Do not push any grease in between cage and inner ring shoulder. The spindle need to be slowly turned as the inner ring and cage assembly are inserted in the outer ring to prevent the raceways and rollers from being damaged. For preloaded bearing arrangements, the housing with outer ring need to be heated as under point 1. If the cylindrical roller bearing is to be mounted together with an angular contact thrust ball bearing, the cylindrical roller bearing outer ring need to be withdrawn from the housing and pushed over the inner ring with cage and roller assembly before the spindle is inserted. The spindle with the complete bearings is then inserted in the heated bearing housing. The housing cover need to be placed in position and the screws tightened.

Fig 22



Fig 23

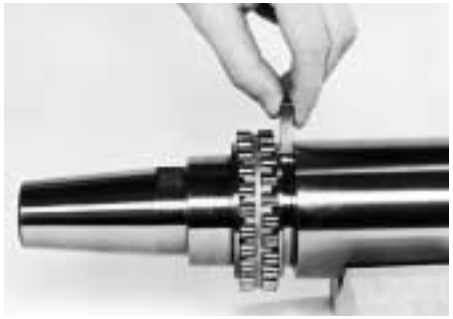


Fig 24



Mounting cylindrical roller bearings without GB-type gauges

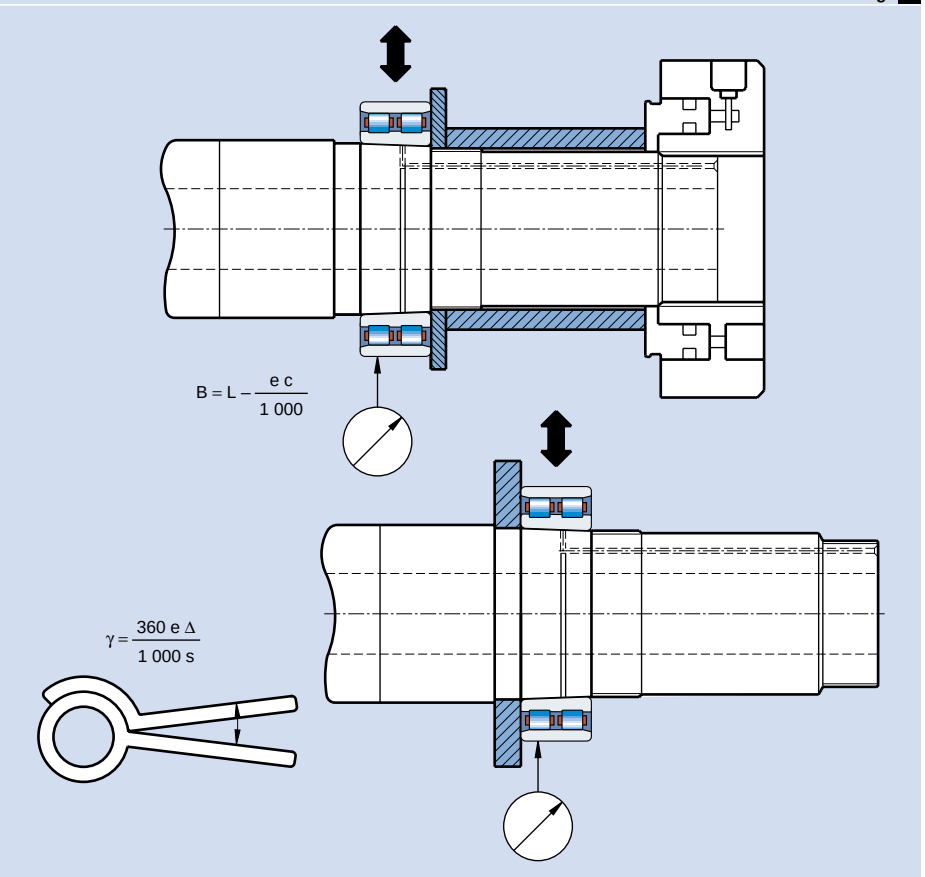
A GB-type gauge is normally used by manufacturers making many of the same kind of spindles. For occasional needs it may not be necessary to invest in this equipment, but to use other methods to adjust for the proper clearance/preload. Some methods used in practice are described later on.

Measuring clearance of cylindrical roller bearings with outer ring (➔ fig 25)

Oil the bearing seating lightly with a thin oil and drive up the inner ring so that it gets a firm seat. There must still be a clearance between roller sets and outer ring. It can be assumed that the clearance is reduced by 8 µm for each 0,1 mm axial drive up of the inner ring. As small bearings may have a rather small clearance, for instance 15 µm,

Measuring clearance of cylindrical roller bearings with outer ring

Fig 25



care must be taken not to push the inner ring too far up on the taper. The measuring principle is to use the outer ring of the bearing to measure the clearance at this initial position of the inner ring. The clearance is measured by moving the outer ring up and down. The total displacement is the clearance of the bearing at this particular position of the inner ring. During this operation it is important that the outer ring is moved perpendicular to the shaft. The outer ring should not be subjected to large forces as it can elastically deform and an erroneous value will be obtained.

To ensure that the outer ring is kept in the proper position when measuring,

Example
The clearance for bearing NN 3020 K was found to be 13 µm when the inner ring was driven up. The distance between bearing side face and abutment was 16,355 mm. The required preload is 3 µm. The inner diameter of the spindle is 51 mm and the outer diameter at the centre of the taper is 101,5 mm. The diameter ratio di/dm d_i/d_m is then 0,5 making factor =16.
The distance ring has to be adjusted to a width of

$B = 16{,}355 - 16 \times 16/1\,000 = 16{,}099$ mm

A tolerance of $\pm$ 0,005 mm is acceptable.

Compensation for interference fit

If the outer ring is to have an interference fit in the housing seating the raceway diameter will decrease. It can be assumed that the raceway diameter will decrease by 80 % of the diametric interference fit.

Example
The bearing outside diameter is 149,997 mm and the seating has a diameter of 149,992 mm. The raceway diameter will decrease by:

$0{,}8 \times (149{,}997 - 149{,}992) = 0{,}004$ mm

This value must be subtracted from c in the equation $B = L - e\ c/1\,000$. If the interference fit is not taken into account the bearing preload may be too high. Considering the interference fit, the distance ring width would then be

$B = 16{,}355 - 16 \times (16 - 4)/1\,000$

$= 16{,}163$ mm

If a threaded nut is used for driving up the inner ring assembly on the tapered seating, the angle through which the nut **need** to be turned for a given clearance reduction of the bearing can be calculated from the equation:

$\gamma = 360\ e\ \Delta/(1\,000\ s)$

its side face need to be supported. This can be done by using a disc that is placed either in the space for the distance ring or is clamped between the drive-up device and the inner ring. In the former case the disc must be provided with a slot to allow access to the shaft for measuring the distance between the inner ring side-face and the abutment This distance is needed for determining the exact width of the distance ring.

The distance is measured using gauge blocks as described earlier. When using the disc in this position it should not be too tightly clamped by the bearing inner ring as it must be rotated for measuring at different positions.

When the distance has been accurately measured the bearing is removed and the spacer is adjusted to the width giving the required preload or clearance of the bearing.

The width is calculated as follows:

$B = L - e\ c/1\,000$

For all details covering this equation, please see the high-precision cylindrical roller bearing/preloading bearing chapter **page 189**.

Other methods
If the requirement is not as stringent as for very accurate preload adjustments, methods other than those described above may be used. Exact preload adjustment is not critical when the speeds are low but clearance needs to be avoided especially if the cylindrical roller bearing is mounted at the work side of the spindle.

Using “feeling with the outer ring”
The inner ring is driven up together with the outer ring. During drive-up the outer ring should be rotated back and forth and as preloading starts the ring will be harder to rotate. The degree of resistance to rotation that corresponds to the bearing having been driven up to a suitable preload can be learned by experience. When the proper position has been achieved the distance between the inner ring side-face and the abutment needs to be measured with gauge blocks. The width of the distance ring should be equal to this distance. With this method as with the method previously described, if the outer ring is to have an interference fit in the housing, compensation should be made for this.

Inserting spindle shaft in the housing (→ fig 26)
The spindle with the work side bearings and the inner ring assembly of the drive side bearing can now be inserted in the housing. Heat the housing to 10 to 30 °C above ambient temperature to easily get the bearings into position. The spindle need to be turned slowly as the inner ring and roller and cage assembly are inserted in the outer ring, to prevent the raceways and rollers from becoming damaged.

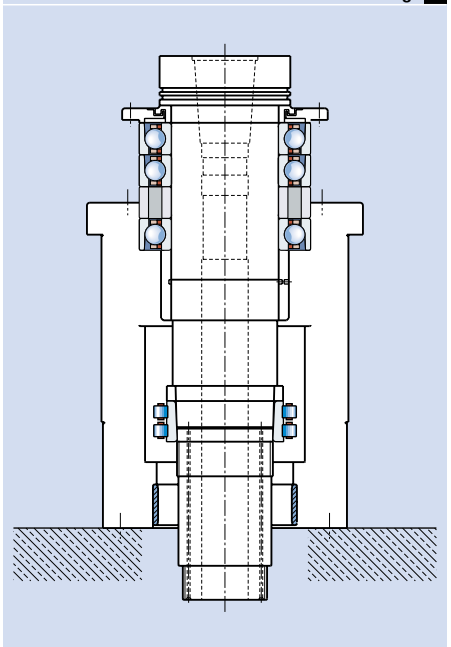
Alternatively the spindle with only the work side bearings can be inserted, and after securing the housing cover, the inner ring assembly of the drive side bearing is driven up on the seating.

Mount the sealing components and the pulley and tighten the nut.

Do not overtighten the screws for the housing covers – at the work side and at the drive side – as this may deform the housing seatings if the distance between the threads and the seatings is short.

Inserting spindle shaft in the housing

Fig 26



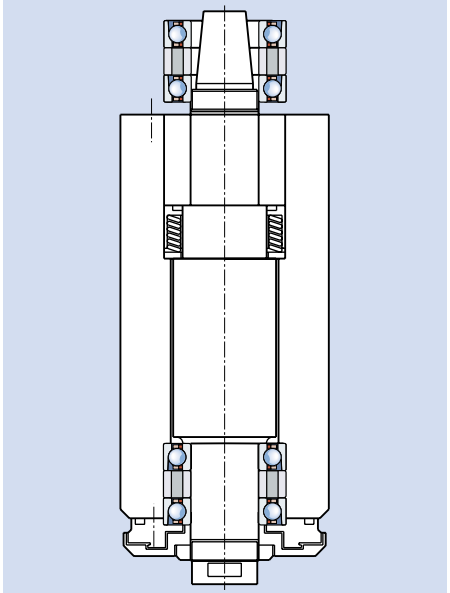
Mounting other bearings arrangements

Spindle with tandem set of angular contact ball bearings at both sides (→ fig 27)
The drive side bearings for this spindle must be mounted after the shaft with the work side bearings has been inserted in the housing. This type of arrangement must have a clearance between drive side bearing outer rings and housing seating as the bearings should be axially displaceable. Therefore the bearings can be mounted without heating the housing. However it may be preferred to heat the bearings somewhat to get them on to the shaft more easily. The outer rings are then also expanded and to avoid forcing the bearings into the housing it should also be somewhat heated.

Spindle with cylindrical roller bearing and angular contact thrust ball bearing at work side (→ fig 28)
A cylindrical roller bearing mounted together with an angular contact thrust ball bearing has to be inserted into the housing complete with the outer ring. The housing should then be heated to about 20 – 30 °C over ambient for easy introduction of the outer ring. The outside diameter of the angular contact thrust ball bearing has such tolerances that it will have a radial clearance in the housing even when the housing is not heated. The radial clearance between this bearing and the housing is necessary to ensure that only the cylindrical bearing is supporting the radial load.

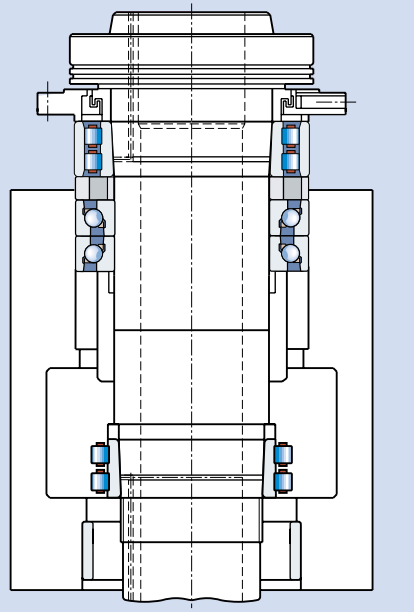
Spindle with tandem set of angular contact ball bearings at both sides

Fig 27



Spindle with cylindrical roller bearing and angular contact thrust ball bearing at work side

Fig 28



Checking running accuracy

When the spindle has been assembled the running accuracy needs to be checked. With the methods most generally used for checking machine tool spindles, radial and axial runouts are measured on suitable surfaces at the spindle nose. It has been found however, that the ovality and eccentricity of the spindle surfaces from which measurements are taken are usually so great that it is difficult to obtain a true indication of the eccentricity of the axis of rotation of the spindle.

A better method of measuring running accuracy (→ **fig** 29) is to use an accurately ground sphere soldered to a plinth which is fitted to the spindle nose. The gauging anvil of a micro-indicator or some similar instrument of corresponding accuracy is applied to the ball. The plinth holding the ball is secured to the spindle nose in such a way that by lightly tapping the ball it can be moved at right angles to the spindle axis. The minimum reading obtained when the spindle is rotated indicates the radial accuracy of the unloaded spindle. An indicator with a flat gauging anvil applied to the ball along the spindle axis gives a reading corresponding to the axial runout of the spindle.

If the raceway of the rotating inner ring is eccentric to the centre line of the shaft, then the shaft will have a runout that is twice the eccentricity (→ **fig** 31).

However in a truly circular raceway there will be a fixed rotational centre that is the centre of the raceway. In some applications the component that is to rotate can be centred to the true rotational centre and in such cases eccentricity errors may not be detrimental to the application. Grinding spindles are one example, where the grinding wheels can be dressed to concentricity with the axis of rotation.

If the positions of the maximum eccentricity of the shaft and bearings are known, the bearing inner rings can be positioned in such a way that the eccentricities compensate each other. SKF angular contact ball bearings have marks indicating the largest eccentricity of both the inner and outer rings i.e. the thickest part of the rings, at position (3) in **fig** 30. Having the eccentricities on opposite sides of the shaft centre line (→ **fig** 32 **page 104**) the runout of the centre line at the position indicated by the gauge will be:

$$\Delta = a/L \times (e_1 + e_2) + e_1$$

When the eccentricities are on the same side of the shaft centre line (→ **fig** 32 **page 104**) the runout will be:

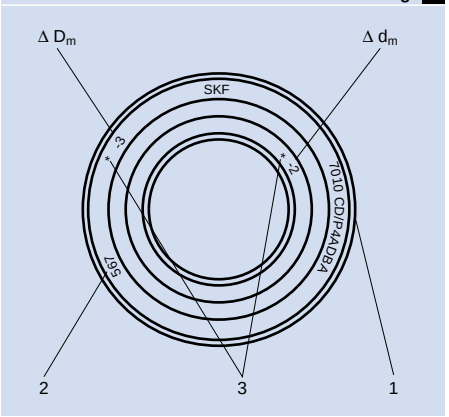
$$\Delta = a/L \times (e_1 - e_2) + e_1$$

The shaft wobble will be larger in the former case but it may give smaller runouts at positions between the bearings. However, it is easier to compensate for the runout in the latter case.

If the raceway is concentric with the bore but is oval or has other types of macro form deviations, then the rotational centre will change during one revolution. The runout will be approximately the same as the form deviation.

The non-repetitive runout must be considered for spindles with very exacting

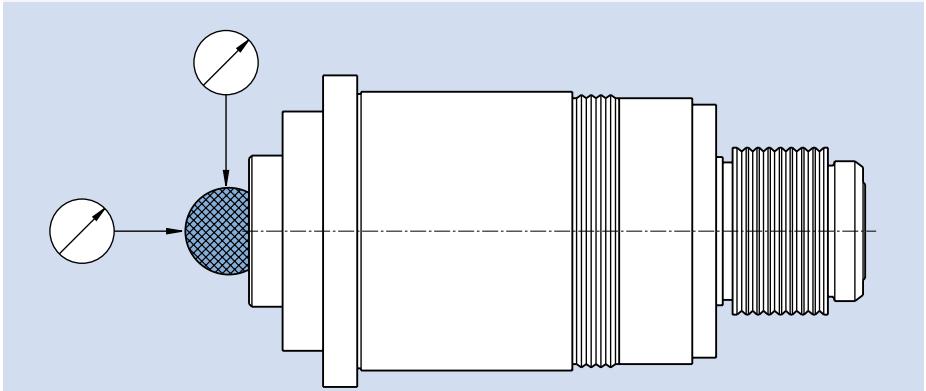
Fig 30



Markings on inner and outer rings for angular contact ball bearings: 1) designation; 2) serial number; 3) deviation from nominal diameter and position of the point of maximum eccentricity

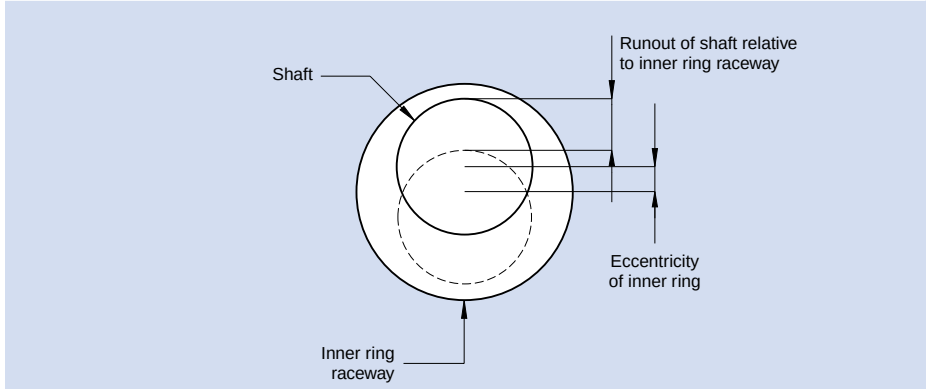
Checking running accuracy of the spindle

Fig 29



Runout of shaft relative to inner raceway

Fig 31



demands for accuracy, such as some types of grinding spindles and disk drive spindles. The non-repetitive runout cannot be compensated for by wheel dressing or by the control system of the machine, as may be the case for repetitive runout, and therefore very stringent demands are made on this type of runout, particularly fo disk drive spindles.

Fig 33 illustrates the principle of non-repetitive runout in comparison with repetitive runout.

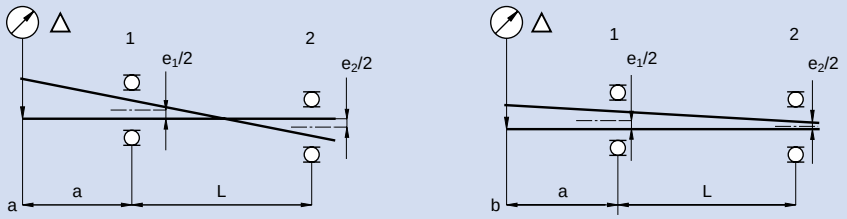
The difference in diameters of the rolling elements will influence the non-repetitive runout. A certain repetitiveness may be noticed if one rolling element (or a few adjacent to each other) happens to have a larger diameter than the others. There will then be a runout with a repetitiveness equal to the cage rotational frequency. At certain times the maximum runout due to rolling elements, and the maximum runout due to raceway will coincide and then give the total maximum runout of the bearing. If the cage speed is 40 % of the inner ring speed, a certain rolling element will coincide with a certain spot on the inner ring each fifth revolution.

As the lubricant plays an important role in the non-repetitive runout, it is required to be very clean. Applications for which non-repetitive runout must be considered are usually grease lubricated, for instance workhead spindles for grinding machines and disk drive spindles.

“Channelling” greases are often preferred, as once the grease has been overrolled and put aside, it stays more or less steady in its new position without disturbing the rolling elements at irregular intervals. A drawback with “channelling” grease can be that the oil bleeding is less efficient than with softer greases and this may influence the bearing life.

Calculating the shaft runout

Fig 32



Inspection

Cleaning of bearings

SKF precision bearings are supplied in a preserved condition. Normally the preservative with which new bearings are coated before leaving the factory needs not be removed from the bearings, just wiped off the outside surface and bore.

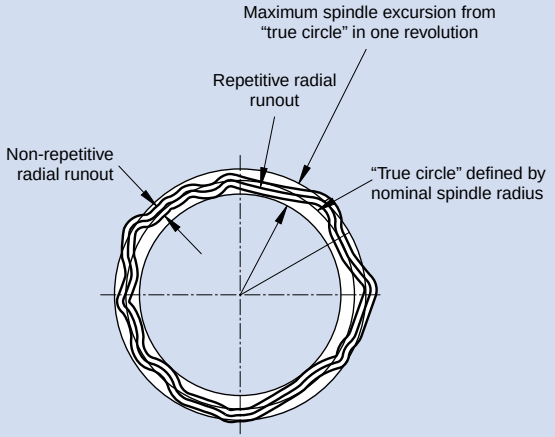
If, however, the bearing is to be grease lubricated and used at very high or very low temperature, or when the grease (for example a polyurea grease) is not compatible with the preservative, it is necessary to wash and carefully dry the bearing. Care should also be taken not to introduce contaminants into the bearing. Bearings contaminated because of improper handling (damaged packaging, etc.) should also be washed and dried before mounting. Washing of bearings being inspected during equipment servicing may also be necessary.

When removal of preservative is necessary, it is possible to remove most of the preservative by blowing the bearing with clean dry air. Precautions must be taken when pressurised air is used to remove material from the bearings to prevent physical injury or chemical contamination of personnel in the area of such operations.

When washing is necessary, the use of hydrocarbon solvents introduces several hazards (solvent flammability, health issues

Repetitive and non-repetitive runout

Fig 33



etc.) to the work place, and these must be addressed before washing operations commence.

When hydrocarbon washing is not possible, the use of aqueous solutions to clean bearings should only be considered when absolutely no other alternatives are possible. The washing solution should be maintained below 60 °C and the pH of the solution less than 12. The washing solution must be kept clean (see above) and preferably have a neutral pH that leaves either no residue or an oil-soluble residue on drying. If this is not available the washing fluid must be washed off the bearing, and the bearings completely dried as soon as possible. The bearings can be dried in hot dry air at temperatures up to 120 °C. It is also possible to use vacuum drying at lower temperatures. When this washing and drying procedure is completed the bearings must quickly be protected, usually by applying a coating of the lubricant to be used in the final operation.

Both the phenolic resin and the polyamide cages used by SKF are capable of withstanding the conditions detailed above.

If these washing conditions cannot be met (pH and temperature, especially when polymeric cages are employed), SKF strongly advises individual product testing to ensure the bearings are not affected by the procedure.

Inspection of bearings

If bearings have been disassembled from a spindle not because of a bearing damage, they may be fit for further use. Do not try to judge whether the bearings can be re-used until after they have been cleaned. Treat them as new.

Never spin a dirty bearing. Instead, rotate it slowly while washing. Wash with a suitable solvent (white spirit, paraffin etc.). Dry with a clean, lint-free cloth or with compressed, clean and moisture-free air, making sure that no bearing part starts rotating.

Examine the bearings closely to determine whether they are re-usable. Use a small mirror and a dental-type probe with a rounded point to inspect raceways, cage and rolling elements. Be alert for scratches, marks, streaks, cracks, discolorations, mirror-like surfaces, contact patterns and so on. Spin the bearing gently and listen to the sound.

An undamaged bearing can be remounted, but if it is not going to be used immediately after being cleaned, it needs to be oiled or greased to prevent corrosion.

If the bearings are separable, do not mix the components of different individual bearings.

Inspection of associated components

Before mounting new bearings, the spindle, housing and other components adjacent to the bearings need to be checked. This is particularly important if the spindle performance has not been satisfactory, even though the bearings have been found to be in good condition.

Inspection of shafts

The dimensions of cylindrical bearing seatings can be easily checked with a snap gauge dial indicator as shown in the picture (→ **fig** 34).

Special measuring equipment is needed to check the form and position tolerances. A precision measuring instrument like Talyrond is the most suitable equipment for checking the roundness of spindle bearing seatings, but is not available in most workshops. The principle of checking radial and axial run-out of a shaft is shown in the **fig** 35.

The bearing seatings and other external surfaces are checked with the spindle mounted in V-blocks.

Deviations are indicated by means of a dial gauge.

The spindle is located axially at one end by using a ball fitted into a centre hole and supported against a bracket. When checking shaft shoulders, the spindle is pressed hard against the ball and the bracket while it is rotated, and measuring is done with an indicator as shown in the illustration (→ **fig** 35).

Tapered bearing seatings can be checked by using ring gauges of series GRA 30. The gauging or reference face is at the large end of the taper bore and is used

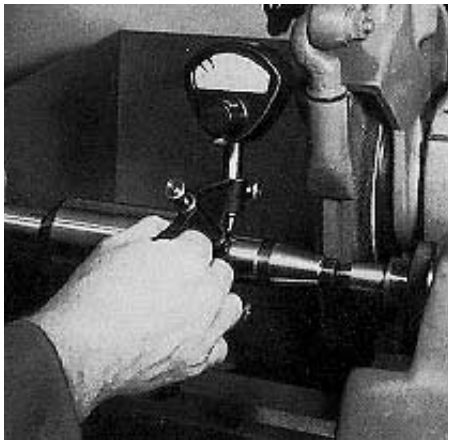
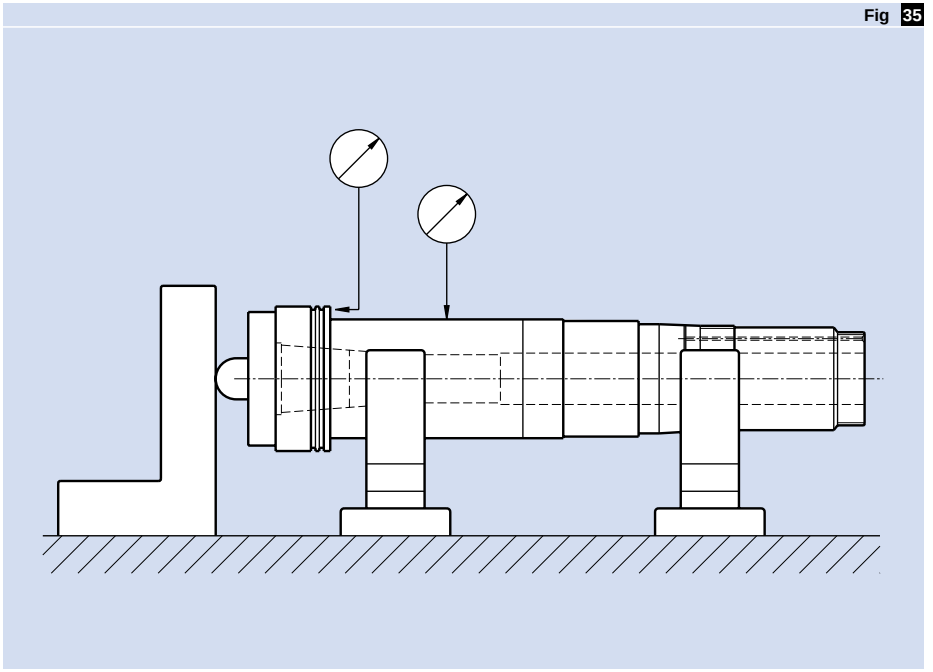


Fig 34

Checking the shaft radial and axial runout

Fig 35



to determine the position of the tapered seating relative to a reference surface on the shaft. This reference surface may be either in front of, or behind the gauging face of the ring gauge.

In addition to checking the position of the bearing seating and diameter, the ring gauge is used to check that the shaft shoulder is at right angles to the axis of the tapered seating by measuring the reference length R (➔ **fig** 36) at several diametrically opposed points. The form of the taper is checked by using marking blue. Applying just a very thin layer should give a marking coverage of at least 80 %.

As an alternative to the ring gauge, a taper gauge can be used. The taper gauge covers a range of sizes and may be an economical solution when dealing with spindles of different sizes.

Inspection of housings

The dimensions and form of the housing seatings can be checked with a bore indicator. To check the form, an attachment is fitted to the indicator to convert it into a three-point gauge. It is difficult to check the position of a shoulder relative to the housing seating in the absence of a reliable reference face. Quill housings of small spindles can usually be set up and rotated, thus making it possible to measure the axial run-out of the shoulder. This type of measurement does not necessarily prove that the shoulder is at right angles to the housing seating unless the housing seating axis coincides with the axis of rotation. Frequently there is no check made to ensure that both bearing seatings in a spindle housing have coinciding centre lines; the seatings are machined by the method giving the best results and that is all. If the centre lines do not coincide well the operating temperature may be too high, especially at high speeds. Checking the alignment may therefore be advisable. Measurements can be made by setting up the housing on a surface plate (➔ **fig** 38) so that the widest seating is parallel with the plate. The diameters of both seatings I and II are measured and the centre heights X_1 and X_2 and the difference $\Delta_X = X_1 - X_2$ are calculated.

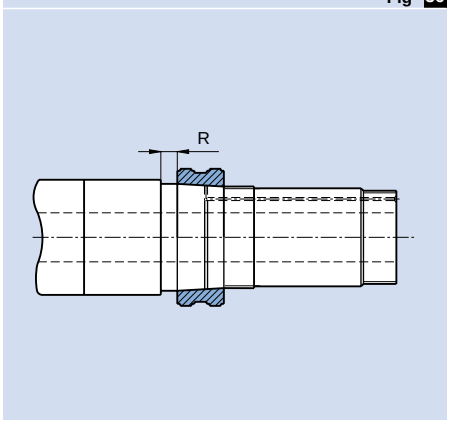
To check that bearing seating II is not misaligned relative to bearing seating I, dimension h is measured at two points; a and b between bearing seating II and the surface plate. The difference between the two read-off values should be less than or at the most equal to half any taper error of the bearing seating over the measuring distance m. m should be only a few millimetres less than the total width of the bearing seating. The housing is then rotated 90° on its axis and corresponding measurements are taken in the y direction. The eccentricity can then be calculated from the equation shown in **fig** 38.

Inspection of spacers

Spacers need to be checked for parallelity and flatness. For short spacers this can be done by measuring the width of the spacer at different positions while placed on a flat surface. Turning it upside down may indicate the flatness. Check if the two spacers between the bearings have the same width. This should normally be the case in order to achieve the right preload of the bearing set (➔ **fig** 37).

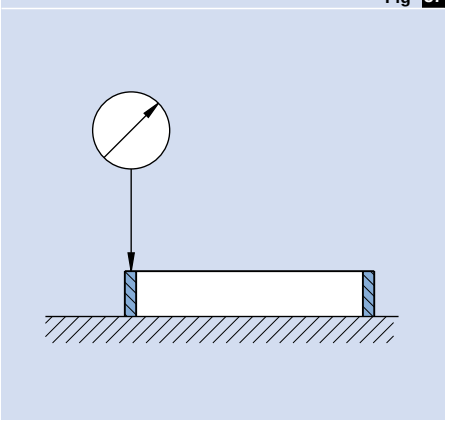
Measuring the reference length R

Fig 36



Inspection of spacers

Fig 37



Inspection of housings

Fig 38

