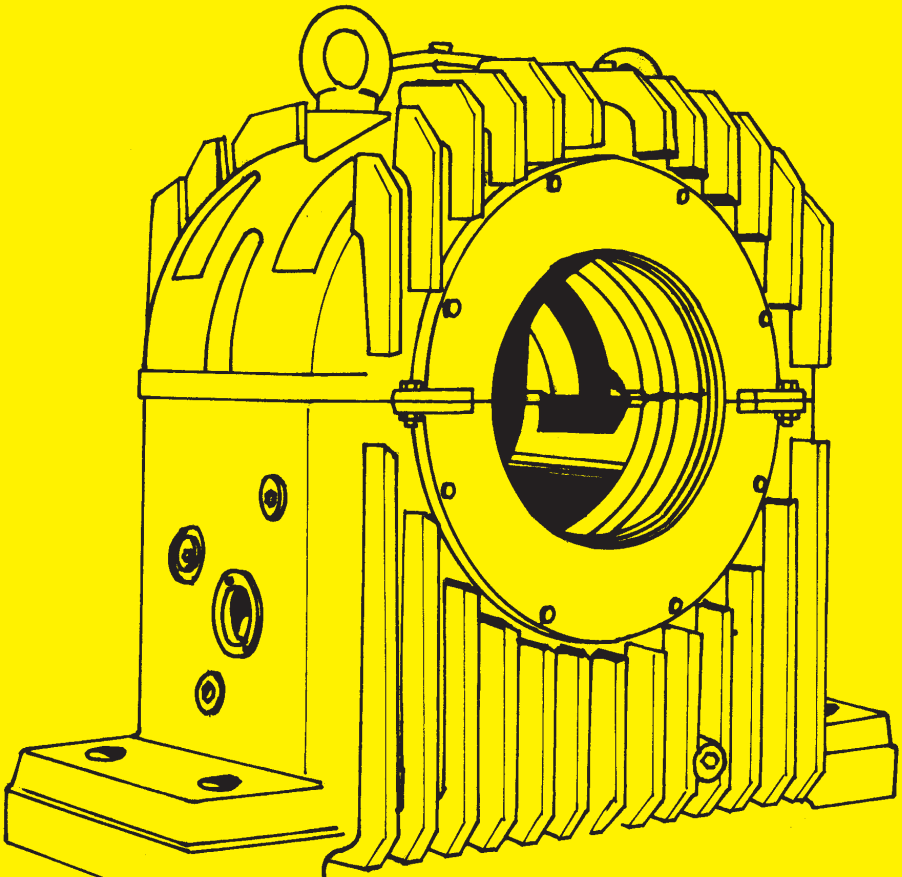
Manual for the application of RENK Slide Bearings



Content

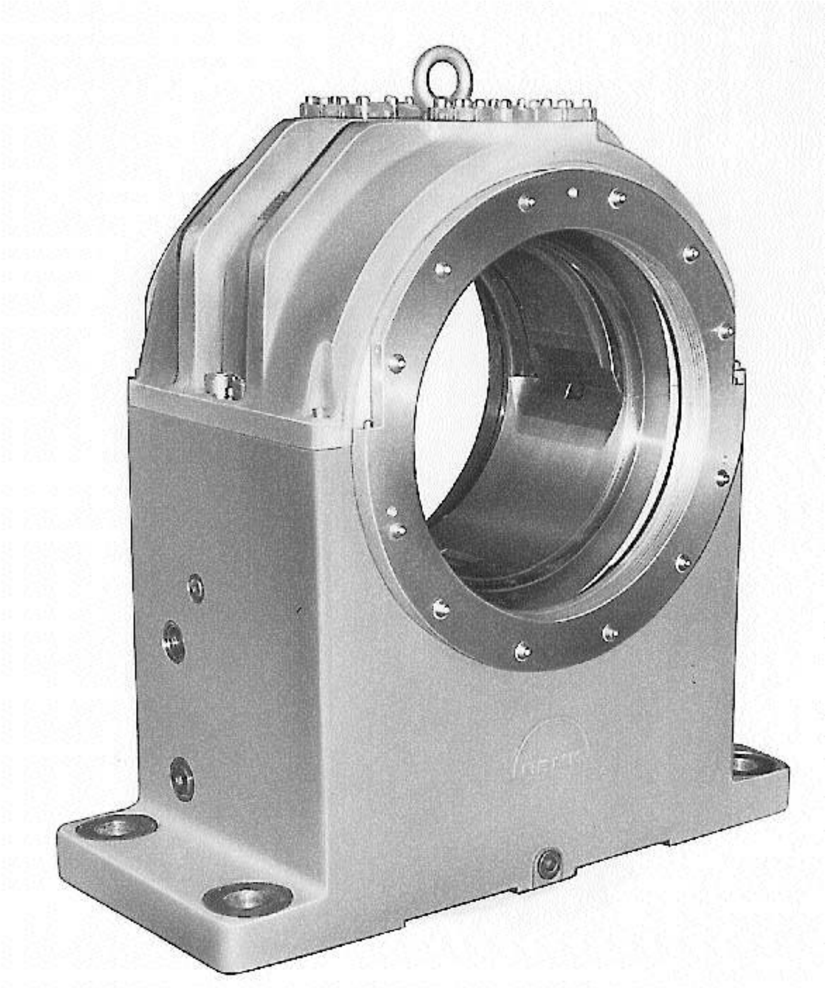


Figure 1 Slide Bearing SCNLB

RENK Slide Bearings are high quality machine parts manufactured on up-to-date machine tools. The designs are in accordance with the latest technological knowledge applied in conjunction with many years of practical experience.

The purpose of this publication is to inform project planners, designers and fitters of the contribution to assure that the RENK Slide Bearings give efficient and reliable service from the very beginning.

In addition to these guidelines, the „Instructions for Installation, Operation and Maintenance“, which we issue for all our standard bearings, describe all those details which have to be observed on account of design details.

1. Types of Bearing

The workshops of RENK Hannover are equipped for the series production of complete slide bearings with housings and ready-to-install bearing shells and thrust bearings. Also the production of slide bearings to customers drawings, large bearings and special bearings for practically all engineering applications is an area in which RENK has many years of experience.

Some of the best-known types of bearing for horizontal and vertical machines are briefly described below. According to the type and size of the bearing, the bearing shells consist of cast iron, cast steel or steel body with highgrade white metal lining.

1.1 Slide Bearing with Housings for Horizontal Shafts

1.1.1 Type M

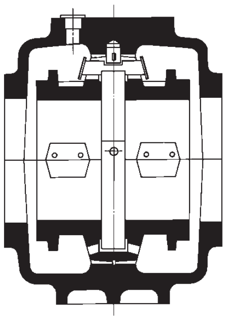


Figure 2 Slide bearing type M

Slide bearing in accordance with DIN 118 with natural cooling, water cooling or circulating oil lubrication, tilting two-component bearing shell (cast iron body).

For normal loads and uniform operation on transmissions, small fans, etc.

Diameter range: 50 to 180 mm.

1.1.2 Type I

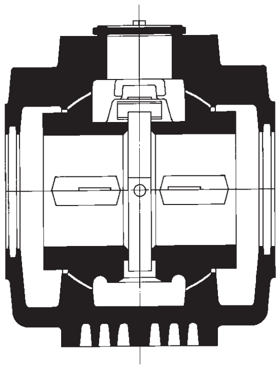


Figure 3 Slide bearing type I

Slide bearing with natural cooling, water cooling or circulating oil lubrication, spherically seated two-component shell (cast iron or cast steel body). For high loads and intermittent operation, e.g. on crushers and tube mill drives.

Diameter range: 100 to 520 mm.

1.1.1 Type E

Slide bearings of most modern conception with many variations to the unit composed system, with natural cooling, water cooling or cooling by circulated oil.
Spherically seated two-component shell (steel body).

Diameter range: 60 to 1250 mm.

Housings EG and ER: foot-mounted, main dimensions to DIN 31690 (fig. 4).

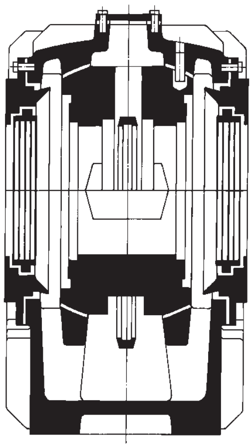


Figure 4 Slide bearing type ER (with loose oil ring)

Housing EF: flange-mounted (fig. 5).
Dimensions to DIN 31 693.

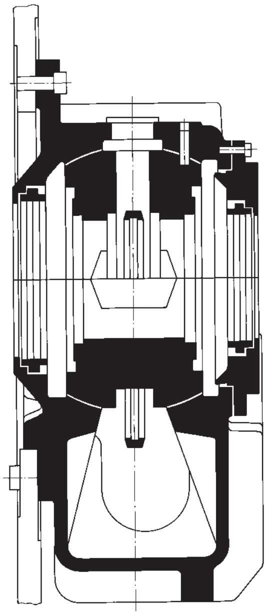


Figure 5 Slide bearing type EF (with loose oil ring)

Housing EM: centrally flange-mounted (fig. 6).
Dimensions to DIN 31 694.

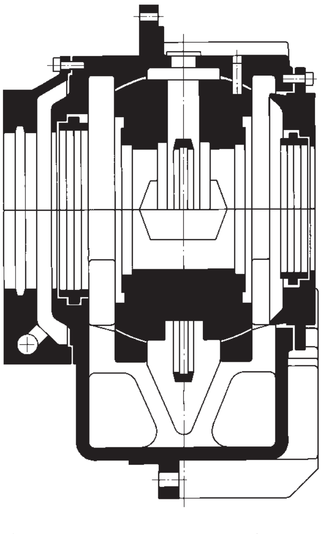


Figure 6 Slide bearing type EM (with loose oil ring)

Slide bearings type E are particularly suitable for electrical and turbo machines. The unit-composed system, however, assures almost universal application throughout the engineering industry.

1.1.3 Type SN

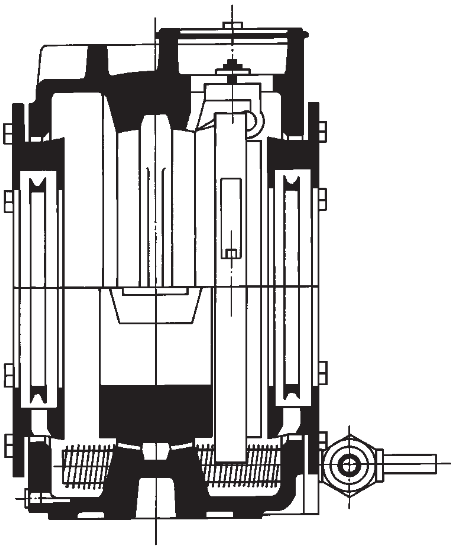


Figure 7 Slide bearing type SN

Propeller shaft bearing with natural cooling, water cooling or circulated oil lubrication, spherically seated two-component bearing shell (steel body).

Available as intermediate bearing (with bottom 180° shell) and aftermost bearing (with bottom and top shell 360°).

Diameter range: 140 to 900 mm.

1.1.5 Type DN

Marine thrust block with natural cooling, water cooling or circulating oil lubrication. The thrust is taken by tilting RS pads (see 1.4) and the shaft weight by one or two journal bearings.

Two-component bearing shell (steel body).

Diameter range: 140 to 850 mm.

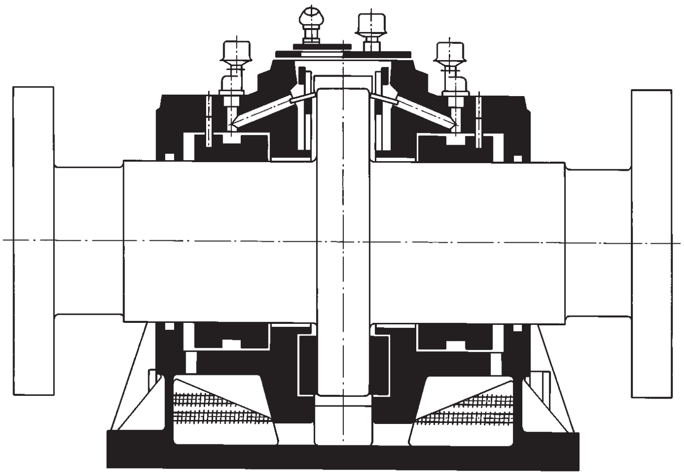


Figure 8 Slide bearing type DN

1.1.6 ROTRIX type

Trunnion bearing for tube mills with hydrostatic shaft jacking and hydrostatic jacking of the 120°-bearing shell in a spherical segment.
Two-component bearing shell (cast steel body).

Diameter range: 1250 to 2400 mm.

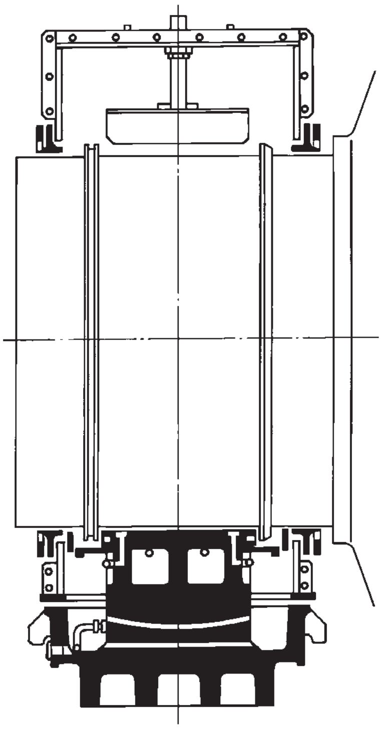


Figure 9 Slide bearing ROTRIX type

1.1.7 Type SH

Trunnion bearing for tube mills with tilting 180° bearing shell with optional hydrostatic jacking.
Two-component bearing shell (steel body).

Diameter range: 400 to 1000 mm.

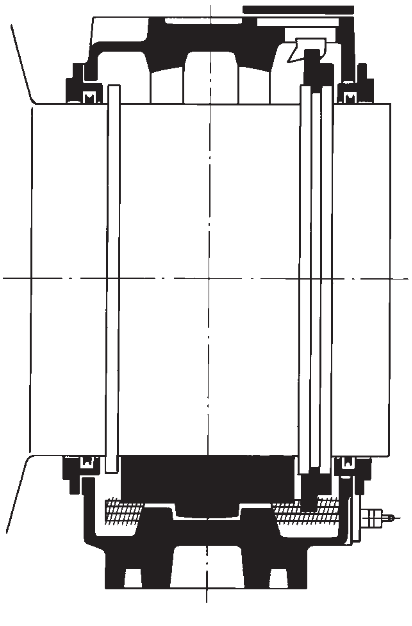


Figure 10 Slide bearing type SH

1.2 Slide Bearing with Housing for Vertical Shafts

1.2.1 Type VD

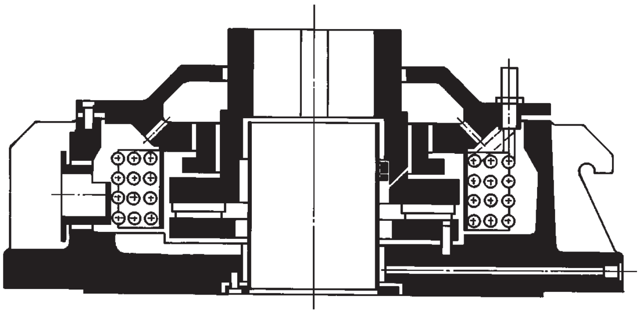


Figure 11 Slide bearing type VD

Vertical thrust bearing – main thrust directed downwards – limited speed range – with RD thrust pads (see 1.4) for small or short-time upward loads with white metal shoulder, 360° guide bearing, for natural cooling, water cooling or circulating oil lubrication.
Diameter range of shafts: 70 to 315 mm.

1.2.2 Type VF

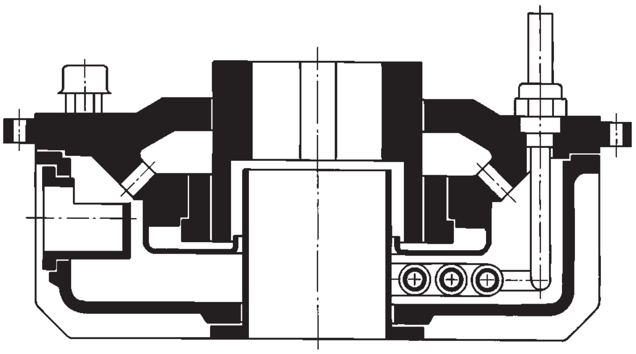


Figure 12 Slide bearing type VF

Vertical guide bearing, execution as above, but without thrust part.
Diameter range of shafts: 70 to 315 mm.

1.3 Insert Slide Bearing

1.3.1 Type G

Bearing shell for gear boxes or similar types of application, acting as non-locating or locating bearing; different shapes (adapted to the type of application) of radial part (cylindrical, 2-lobe or 4-lobe bore) and of thrust part (shoulders with taper lands, tilting pads).
Diameter range. 40 to 750 mm.

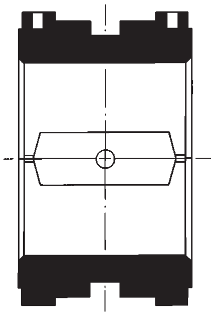


Figure 13 Bearing shell type G

1.3.2 Type EV

Vertical insert bearing for electric and turbo machines.

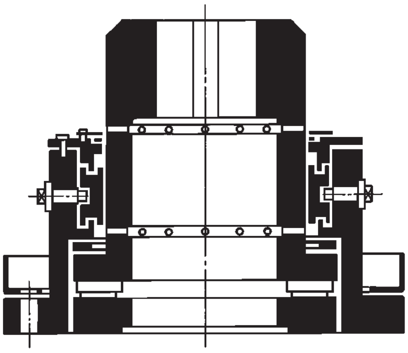


Figure 14 Slide bearing type EVE

As thrust bearing for one or two directions of thrust, with RD pads (see 1.4) combined with a guide bearing of adjustable tilting segments.

Diameter range of shafts: 70 to 500 mm.

1.3.3 Type EVF

Guide bearing as above, but without RD pads.

Diameter range of shafts: 70 to 500 mm.

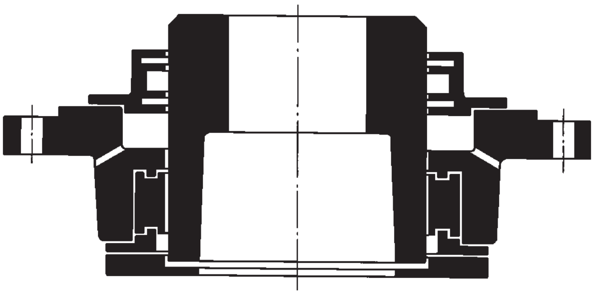


Figure 15 Slide bearing type EVF

1.4 RD and RS thrust bearings

RD and RS thrust bearings are a logical improvement of the segment type bearings. They replace in all fields the conventional segments.

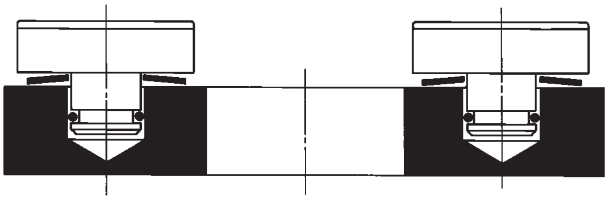


Figure 16 RD thrust bearing

RD thrust pads are cup spring-supported in the carrier (fig. 16), RS thrust pads are pivot-supported on the carrier (fig. 17).

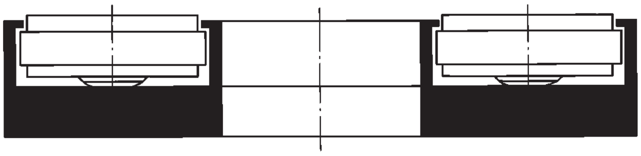


Figure 17 RS thrust bearing

2. Basic Principles

2.1 Hydrodynamic Lubrication

RENK slide bearings usually operate on the reliable principle of hydrodynamic lubrication. When the design and manufacturing conditions permit this type of lubrication, the slide bearings will fullfil all the requirements expected of them.

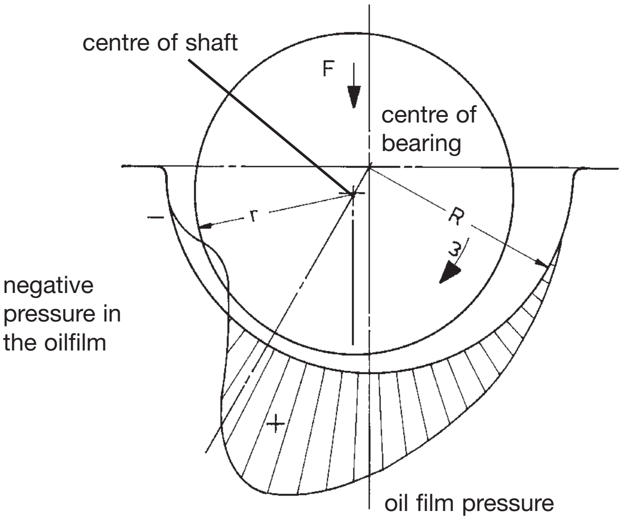


Figure 18 Pressure build-up in a plain cylindrical bearing

It should be noted that in plain cylindrical bearings (fig. 18) only, build-up of a carrying oil wedge is made possible by the clearance between bearing shell and shaft in eccentrical position (converging film). In thrust bearings the pressure build-up is made possible due to the clearance between shaft and bearing shell bore. As regards journal bearings with two- or four-lobe bores (fig. 21 and 22) or tilting pads (fig. 23) as well as thrust bearings with taper lands or pivoting thrust pads, it is achieved by design.

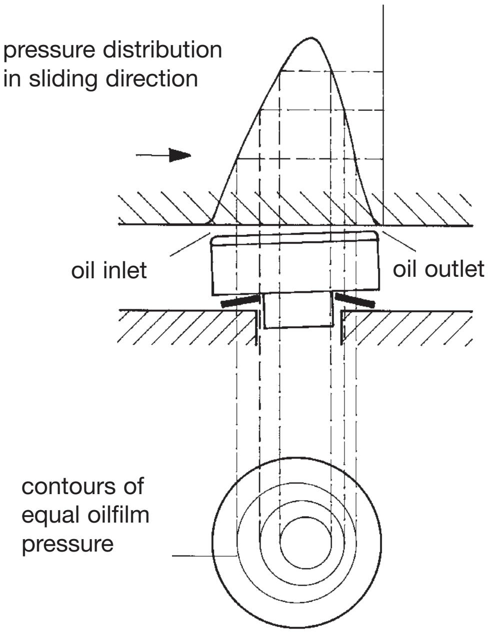


Figure 22 Pressure build-up at a tilting pad

When selecting the type of radial bearing, either plain cylindrical bore (fig. 23), two-lobe (fig. 24), four-lobe (fig. 25) or radial tilting segments (fig. 26), various factors have to be considered.

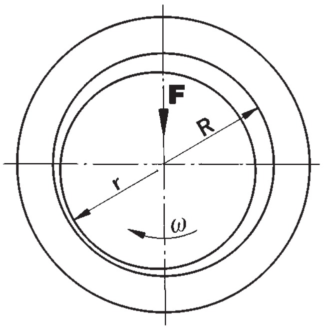


Figure 20 plain cylindrical bore

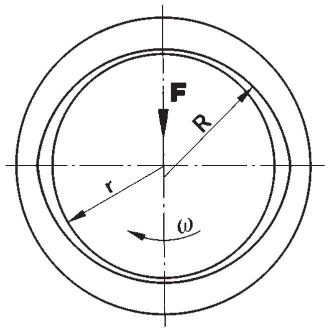


Figure 21 two-lobe bore (lemon shape)

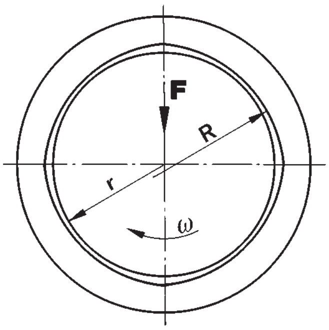


Figure 22 four-lobe bore

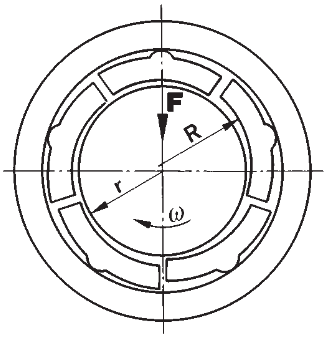


Figure 23 journal tilting pads

In the case of electrical machines, fans, compressors, turbines and gear-boxes a preliminary choice can be made by means of a simple criteria table (table 1). In many cases, the final decision can only be made on the basis of a vibration calculation taking ionto account the elasticity and damping values, shaft geometry, loads and mass moments of inertia as well as the characteristics of the foundation. Experiences gained with machines of the same or similar type are also of value.

Bearing type	cylindrical	two-lobe	four-lobe	tilting pads
Designation	C	Y	V	K
Type of radial bearing for electrical machines, fans, compressors and turbines				
Peripheral speed U [m/s] Specific load $\overline{p}$ [N/mm²]	0 . . . 30 0,1 . . . 4	25 . . . 75 0,1 . . . 3	25 . . . 125 0 . . . 2	15 . . . 150 0 . . . 2
Type of radial bearing for gear boxes				
Peripheral speed U [m/s] Specific load $\overline{p}$ [N/mm²]	0 . . . 35 0,1 . . . 5	28 . . . 63 0,1 . . . 4	45 . . . 100 0 . . . 3	63 . . . 160 0 . . . 3

Table 1 Selection of radial bearings

In the case of slide bearings with non-static load or a migrating load vector other additional criteria must also be taken into account.

Initially, the type of thrust bearings is selected according to the load which occurs:

1.) Smaller axial loads can be absorbed by plain whitemetal faces on the shoulders of the bearing shell (locating bearing). The specific load must not exceed 0,25 N/mm².

2.) Medium axial loads can be absorbed by taper land sections incorporated in the whitemetal lined faces of the shells. These taper land sections can be machined for one or both directions of rotation. In normal cases, the specific load should not exceed 2 N/mm² as this type of bearing is highly sensitive to edge loading and metal contact can occur at high loads because of small lubricant film thickness.

3.) High axial loads can be absorbed by tilting pads (preferably RD (RS) thrust pads. With this type of bearing, the specific load can be increased to 6 N/mm² during operation according to size and operating conditions.

The starting load should not exceed 2,0 ... 2,5 N/mm² either in the case of journal or thrust bearings as otherwise the working surfaces may wear and high initial torques may occur.

In each case the values of actual surface quality and accuracy of form have to be checked. Furthermore, the viscosity of the lubricant, compatibility of materials, duration and number of starts/stops and other parameters influence the selection of the admissible minimum oil film thickness.

Shaft diameter D or. mean sliding diameter of thrust part D_m [mm]	Sliding velocity of shaft U or mean sliding velocity of thrust part U_m [m/s]				
	... 0,3	> 0,3...3	> 3...10	> 10...30	> 30
24 ... 63	3	4	6	8	10
> 63 ... 160	4	5	8	11	14
> 160 ... 400	5	7	10	13	16
> 400 ... 1000	7	9	12	15	18
> 1000 ... 2500	9	12	15	18	21
> 2500	12	15	18	21	24

Table 2 Minimum admissible oil film thickness h_0 in µm for continuous operation

2.2 Slide Bearing Calculation

In the planning stage already, each of the bearings to be supplied by us will be selected or checked on the basis of hydrodynamic-thermic computer calculations, if customers state operating conditions. The values as for instance speed, load and load direction, oil viscosity and ambient temperature are basic figures for the calculation of the operational behaviour. We therefore strongly recommend to complete in detail our form „Enquiry for Slide Bearings“.

The safe operation of a slide bearing is indicated by bearing temperature and minimum oil film thickness. Admissible bearing temperatures are given in Section 6. Minimum oil film thicknesses for continuous operation are shown in table 2. The values in that table refer to shafts and thrust collars respectively, manufactured with a degree of accuracy of 5 ($h_0 < 10$ µm) or 10 (10 µm $\leq h_0 < 20$ µm) as per DIN 31 699.

2.3 Hydrostatic jacking

Within the range of application of RENK Slide Bearings there are cases for which supplementary hydrostatic jacking systems are recommended (e.g. frequent starts under high start-up load, operation at low speed or very long rundown periods).

Arrangement of the Lubricating Pockets in the Journal Bearing

The lubricating pockets are machined in that area of the lower half of the bearing shell, where the shaft – owing to the start-up load – tends to get into contact with the bearing shell (fig. 24-1). The dimensions of the pocket, which is placed on the imaginary line of contact, are determined by the bearing geometry and the radial load.

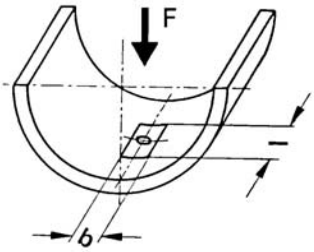


Figure 24-1 Lubricating pocket in a journal bearing

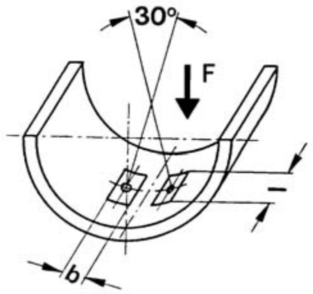


Figure 24-2 Execution with two lubricating pockets

As regards bearings with oil discs, where the radial working surfaces are vertically split, both halves have lubricating pockets in order to allow parallel lifting of the shaft. In special cases, e.g. for the four-lobe bearing, where the shaft gets into contact with the bearing shell in two areas, we alternatively recommend an arrangement consisting of two lubricating pockets in peripherial direction which are situated under the respective line of contact (fig. 24-3).

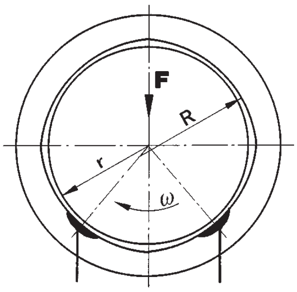


Figure 24-3 Arrangement of the Lubricating Pockets in the Four-Lobe-Bearing

Hydrostatic lifting cannot be compared with stationary hydrostatic film flubrication. Therefore the stability of the shaft position during lifting and holding is limited when operating the machine at low speed. In cases where lateral loads or other disturbances occur, the shaft may get unbalanced and displaced, which might result in wear. In these cases, it is possible to install two pockets which are situated symmetrically to the contact line of shaft and bearing shell (fig. 24-2). It must be noted, however, that in this case, as well as for the four-lobe bearing, the high-pressure pump unit has to be designed in such a way as to guarantee independant oil supply for the two lubricating pockets.

Output and Constant Supply Pressure

The high-pressure pumps for hydrostatic lifting units are constant volume systems, i.e. they supply a defined oil throughput. The maximum load carrying capacity of the system is determined by the maximum pump pressure, which is usually limited to 20 MPa (= 200 bar). The maximum pump pressure occurs during the starting process, due to the oil wedge being narrow when the bearing is still at rest. The beginning of the shaft lifting thus involves a remarkable pressure surge. As soon as the oil wedge gets wider as the shaft is further lifted, the pressure depending on the bearing geometry and the oil throughput, since the effectivly load-carrying surface is increased. The constant pump pressure for holding (up) the shaft should be adjusted to 10 MPa. Approximate values for the oil throughput required are given in table 3. For the critical operating mode, the minimum viscosity of the lubricating oil at normal operating temperature has to be taken into account. The lubricating pockets have to be designed according to the loads to maintain a constant pressure of ≤ 10 MPa.

Shaft dia. D [mm]	~ V [l/min]	
	20 mPa s (cP)* $\Psi \leq 2$ ‰	50 mPa s (cP)* $\Psi \leq 1{,}6$ ‰
80 ...90	0,25	
100 ... 110	0,5	
125 ... 140	0,75	
160 ... 180	1,25	0,5
200 ... 225	2,5	1
250 ... 335	5	2
400 ... 500		3
500 ... 630		3,5
630 ... 800		4
800 ... 1250		5
*corresponds at e.g. 60°C to:	ISO VG 46	ISO VG 150

Table 3 Oil throughput for high-pressure pump

Oil Supply for Hydrostatic Oil

The hydrostatic lifting unit mainly consists of E-motor, high pressure pump, oil tank and supply line. In case there are several lubricating pockets, oil supply lines for each of the pockets have to be installed. A misalignment of the shaft or an uneven load distribution might be sufficient to result in essentially higher oil throughput of the lubricating pocket with the wider oil wedge. For that reason, oil supply for two lubricating pockets has to be done either by two separate or one pump with a ratio control valve.

In the case of test runs with a smaller load, the calculated pressure will not occur (to be considered if switching monitors are incorporated in the pressure pipe). Initial starting torques (due to contact of solid masses) are considerably reduced particularly with heavy specific loads at rest. This is the case in trunnion bearings for tube mills and thrust bearings for vertical asynchronous motors.

A friction coeffizient of $\mu_0 = 0{,}1 \ldots 0{,}25$ should be used to calculate the starting torques for bearings without a hydrostatic jacking system. This friction coefficient varies considerably and may exceed a value of $\mu_0 = 0{,}25$ in individual cases, e.g. because of poor surface quality and also negligent assembly work.

The hydrostatic jacking device also prevents damage to the sliding surfaces during a prolonged slow-down time at a speed below the transitional speed. „Transitional speed“ means that number of revolutions when the region of hydrodynamic lubrication is left and mixed-film lubrication with metal-to-metal contact begins.

The high pressure pump unit for the hydrostatic jacking should be designed in accordance with table 3.

When selecting the pump it can be taken into account the quantity may deviate from the value given in the table by ± 25%. Because of the pressure surge occuring at start-up the pump unit has to be designed for approx. 200 bar.

Depending on service conditions and bearing geometry, a pressure of p = (2.4 … 6) (N/mm^2) in the oil pocket will adjust during operation ($\overline{p}$ = specific load).

Slowing-down of machines, particularly of those with self-contained lubrication, is often critical due to the fact that oil at operating temperature with a lower viscosity than in cold state is used for lubrication. Examples for machines with long slow-down times are blowers which are continuously turned by the stack flue and turbine generator sets which must be turned in order to avoid any heat distortion of the rotor.

2.4 Minimum Speeds

Frequently, slide-bearing supported machines or machine sets are operated at turning speeds for a longer period, in order to assure cooling without any distortion to the rotor. This applies to steam turbines, tube mills and hot-gas fans. In case there exists no hydrostatic jacking device, a sufficient oil film thickness must be assured by maintaining a certain minimum speed n_{min} .

From graph fig. 25 minimum speeds for oil viscosity ISO VG 46 and ambient temperatures = 40°C in function of shaft diameter D and specific load $\overline{p}$ can directly be read. The minimum thickness thus obtained is approx. 3...4 µm. A condition for safe operation at such a thin oil film thickness is excellent machining of the shaft, i.e. surface quality and tolerances of form, degree of accuracy 5 to DIN 31 699, in order to assure continuous operation without wear after a certain running-in period. If n_{min} is required to be < 10/min, it must always be checked that in case of self-lubricated bearings a sufficient oil supply is still assured. In such cases a pick-up ring or a circulation pump may be necessary. To limit the effects eventually caused by mixed-film lubrication, oils with MoS_2 additives may be chosen. Please always consult RENK in this respect.

For circulating oil lubrication the values indicated are based on an oil inlet temperature of 40°C.

If lubricating oils of viscosity ISO VG 68 will be used, the curve of the next lower specific load has to be selected (e.g. at existing $\overline{p}$ = 2,2 $N/mm^{2,}$, refer to $\overline{p}$ = 2,0 N/mm^2).

If lubricating oils of viscosity ISO VG 32 will be used, the curve of the next higher specific load has to be selected (e.g. at existing $\overline{p}$ = 2,2 N/mm^2 , refer to $\overline{p}$ = 2,5 N/mm^2).

3. Shafts and Thrust Collars

As often shafts and thrust collars are not supplied by RENK Werk Hannover we hereafter give details on their manufacture as well as a list of requirements which have to be met in order to assure safe operation of the bearings. Requirements with regard to geometry and surface quality of slide bearings are by no means lower to those applying to antifriction bearings.

3.1 Material and Hardness

The best material for the purpose is carbon steel (e.g. E295/ St 50-2, E335 / St 60-2 to DIN EN 10025 / DIN 17100; C 45 to DIN EN 10083-2). Constructional steel qualities (e.g. S235JR / St 37-2, S355JO / St 52-3 to DIN EN 10025 / DIN 17100) should not be used.

Several high-alloy steel qualities allow hydrodynamic operation only, if white metal, viscosity of oil and specific load are chosen in accordance. It is recommended to consult RENK Werk Hannover in any case.

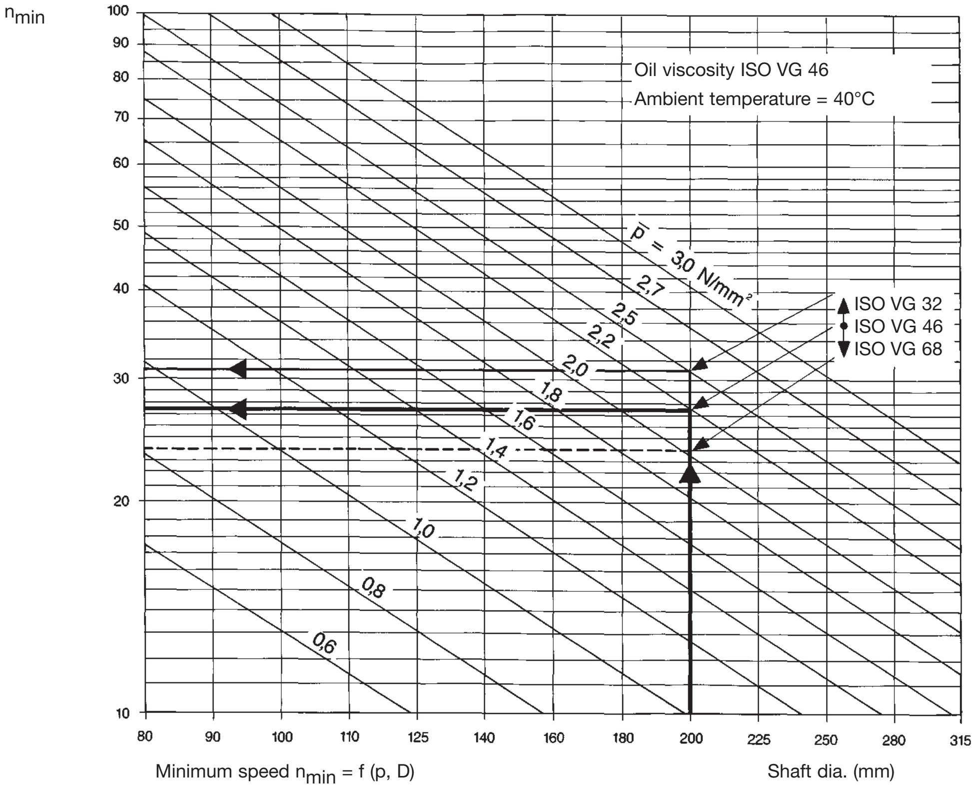


Figure 25

Example:

D = 200 mm
$\overline{p}$ = 2,2 N/mm^2
oil ISO VG 46
ambient temperature = 40°C
→ N_{min} = 27,5/min

If lubricating oil ISO VG 46
will be used instead of
ISO VG 32
N_{min} becomes = 31/min

With the use of ISO VG 68
N_{min} becomes = 24/min

3.2 Manufacturing Process

The most favourable method for making the radial bearing surfaces is by circular grinding. Special care must be taken to avoid spiral machining grooves at the seal areas after the finish-grinding (fig. 26). Spiral grooves would create a pumping effect in the seal of the bearing which could lead to oil leaks.

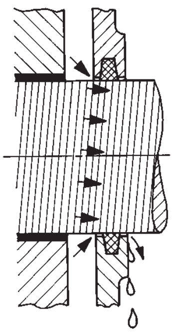
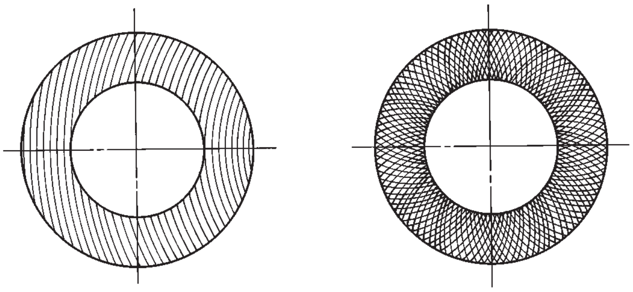


Figure 26 Shaft seal with improper machining grooves

Thrust collars can be forged on or machined from the solid or fitted separately. For shrunk-on collars the finish machining should be carried out together with the shaft.

With separately manufactured collars which are not finished machined on the shaft, the shaft shoulder must be manufactured with utmost care. The machining of the flat collar faces on a surface grinding machine should not be carried out in the longitudinal direction because there would be the danger of grinding groove formation which, as they would partly be at right angels to the running direction, could have a damaging influence (fig. 27). Here flat grinding on the rotating collar is recommended.



wrong right

Figure 27 Working surface of collars

In order to achieve very good working surfaces, separate collars can be lapped on a level plane or an a lapping machine with the aid of an appropriate paste.

It is essential that checking for flatness is carried out with a hair line straight edge and with a micrometer gauge for true running (fig. 28).

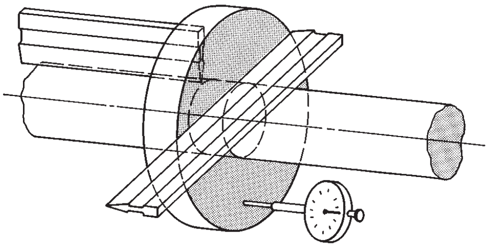


Figure 28 Checking the collars

3.3 Tolerances of form and position, surface quality

The tolerances of form and position for the radial part should at least conform to degree of accuracy 10 and for the thrust part to degree 20 as per DIN 31 699.

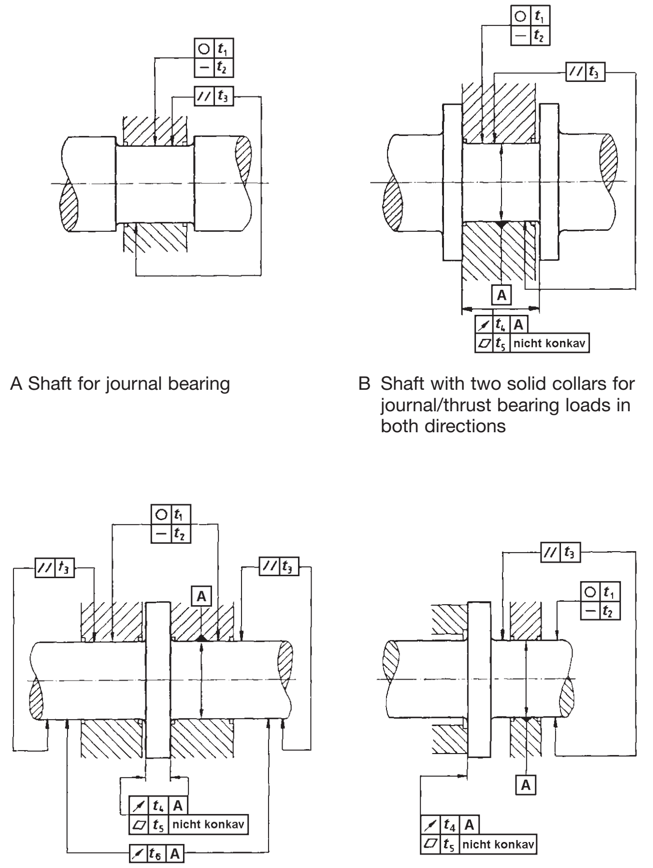
The recommended values are given below:

○ | t_1	t_1 =	Roundness (circularity), permitted deviation 0,006 mm
— | t_2	t_2 =	Straightness permitted deviation 0,010 mm
// | t_3	t_3 =	Parallelism permitted deviation 0,02 mm
↗ | t_4	t_4 =	Runout (axial) permitted deviation 0,012 mm
▱ | t_5	t_5 =	Flatness permitted deviation 0,012 mm
↗ | t_6	t_6 =	Runout (radial) permitted deviation 0,01 mm (no indication in DIN standards)

The former tolerance for cylindricity is no longer indicated, as it is not possible to check it with normal measuring means. It has been splitted in tolerance of roundness, straightness and parallelism.

Measuring methods and means are indicated in standard DIN 316 70, part 8.

The surface roughness may be maximum Ra = 0,63 µm in contrary to DIN 316 99).



A Shaft for journal bearing

B Shaft with two solid collars for journal/thrust bearing loads in both directions

C Shaft with solid collar for journal/thrust bearing loads in both directions

D Shaft with solid collar for journal/thrust bearing loads in one direction

Figure 29 Form and position tolerances

Extract from the RENK-Standard 124.31
Sleeve Bearing Fits
This Standard is in conformity with DIN 31 698, Issue April 1979

1. Scope

This standard applies to slide bearings for general engineering applicatins and to a mean relative clearance of ψ_m = 0,56 to 3,15 ‰.

It does not apply to slide bearing shells and bushes which, by virtue of their particular nature, are not measured across the diameter but across the wall thickness and which suffer dimensional changes as the result of compression.

2. Range of Application

Preferably, this standard is applied in the case of rotating machine parts and shafting; it can be applied accordingly for other ranges of applications.

3. Aim

This standard was prepared because with the ISO dimensions according to DIN 7160 and DIN 7161, it is not possible to form any clearance fits, which, based on the requirements of slide bearing technology, meet the mean relative bearing clearance which is practically the same in all nominal dimension ranges.

4. Further Valid Standards

DIN 7161	ISO dimensions for internal dimensions (bores), for nominal dimensions from 1 to 500 mm
DIN 7172 part 2	(draft standard) ISO tolerances and ISO dimensions for linear dimensions from 500 to 3150 mm; permissible deviations.

5. Classification of Fits

basic bore

Tolerance field H for bearing bore in accordance with DIN 7161 and DIN 7172 part 2 (draft standard). The tolerance field for the shaft corresponds to the mean relative bearing clearance ψ_m .

6. Mean Relative Bearing Clearance

The mean relative bearing clearance ψ_m in ‰ of a nominal dimension range is obtained from:

$$\psi_m = \frac{S_m}{D_m}$$

where:

S_m is the mean absolute bearing clearance in µm

$$= \frac{\text{maximum clearance + minimum clearance}}{2}$$

D_m is the arithmetic mean of the nominal dimension range in mm.

7. Tolerance Fields

7.1 Size

The size of the tolerance fileds is chosen in such a way that in the case of the same mean relative bearing clearance ψ_m a more or less the same maximum deviation form the relative bearing clearances is not exceeded within a tolerance field, from the minimum to the maximum nominal dimension range. The downward limit is governded by economic production methods.

The tolerance field for the shaft is 1 IT (basic tolerance in accordance with DIN 7151) smaller than the tolerance field for the corresponding bearing bore.

7.2 Position

The position of the tolerance field in relation to the zero line is determined by the mean relative bearing clearance ψ_m.

7.3 Number

On tolerance field is formed for each of the following ψ_m values:

0,56; 0,8; 1,12; 1,32; 1,6; 1,9; 2,24; 3,15

7.4 Symbols

The symbol for the relative bearing clearance is ψ.

Taking into account the limitations fo computer printers and typewriters, PSI is used instead of the Greek letter ψ .

8. Nominal Dimension Ranges

The nominal dimension ranges are more closly graduated than in DIN 7160 and DIN 7172 part 2 (draft standard) so that the maximum deviation from the mean relative bearing clearances ψ_m is kept within narrower limits.

9. Dimensions

The shaft dimenions are contained in the table 4 on page 13.

10. Minimum and Maximum Clearances

The minimum and maximum clearance between shaft and bearing bore which is necessary for calculating the slide bearings is contained in the table 4 on page 13 together with the shaft dimensions.

11. Example

Shaft fit 200 mm for a mean bearing clearance

ψ_m = 1,12 ‰: $\varnothing\ 200_{\text{PSI } 1,12}$

Nominal shaft range [mm]		Permissible deviations of the shaft 1) in µm for ψ_m in ‰								Maximum/minimum clearance between shaft dia. 2) 3) and bearing bore in µm for ψ_m in ‰							
over	up to	0,56	0,8	1,12	1,32	1,6	1,9	2,24	3,15	0,56	0,8	1,12	1,32	1,6	1,9	2,24	3,15
25	30	—	- 15 - 17	- 23 - 29	- 29 - 35	- 37 - 43	- 45 - 51	- 51 - 60	- 76 - 85	—	30 15	38 23	44 29	52 37	60 45	73 51	98 76
30	35	—	- 17 - 24	- 27 - 34	- 34 - 41	- 43 - 50	- 48 - 59	- 59 - 70	- 89 - 100	—	35 17	45 27	52 34	61 43	75 48	86 59	116 89
35	40	- 12 - 19	- 21 - 28	- 33 - 40	- 36 - 47	- 47 - 58	- 58 - 69	- 71 - 82	- 105 - 116	30 12	39 21	51 33	63 36	74 47	85 58	98 71	132 105
40	45	- 14 - 21	- 25 - 32	- 34 - 45	- 43 - 54	- 55 - 66	- 67 - 78	- 82 - 93	- 120 - 131	31 14	43 25	61 34	70 43	82 55	94 67	109 82	147 120
45	50	- 18 - 25	- 25 - 36	- 40 - 51	- 50 - 60	- 63 - 74	- 77 - 88	- 93 - 104	- 136 - 147	36 18	52 25	67 40	76 49	90 63	104 77	120 93	163 136
50	55	- 19 - 27	- 26 - 39	- 43 - 56	- 53 - 66	- 68 - 81	- 84 - 97	- 102 - 115	- 149 - 162	40 19	58 26	75 43	85 53	100 68	116 84	144 102	181 149
55	60	- 22 - 30	- 30 - 43	- 48 - 61	- 60 - 73	- 76 - 89	- 93 - 106	- 113 - 126	- 165 - 178	43 22	62 30	80 48	92 60	108 76	125 93	145 113	197 165
60	70	- 20 - 33	- 36 - 49	- 57 - 70	- 70 - 83	- 80 - 99	- 99 - 118	- 121 - 140	- 180 - 199	53 20	68 36	90 57	102 70	129 80	148 99	170 121	229 180
70	80	- 26 - 39	- 44 - 57	- 60 - 79	- 75 - 94	- 96 - 115	- 118 - 137	- 144 - 163	- 212 - 231	58 26	76 44	109 60	124 75	145 96	167 118	193 144	261 212
80	90	- 29 - 44	- 50 - 65	- 67 - 89	- 84 - 106	- 108 - 130	- 133 - 155	- 162 - 184	- 239 - 261	66 29	87 50	124 67	141 84	165 108	190 133	219 162	296 239
90	100	- 35 - 50	- 58 - 73	- 78 - 100	- 97 - 119	- 124 - 146	- 152 - 174	- 184 - 206	- 271 - 293	72 35	95 58	135 78	154 97	181 124	209 152	241 184	328 271
100	110	- 40 - 55	- 56 - 78	- 89 - 111	- 110 - 132	- 140 - 162	- 171 - 193	- 207 - 229	- 302 - 324	77 40	113 56	146 89	167 110	197 140	228 171	264 207	359 302
110	120	- 36 - 60	- 64 - 86	- 100 - 122	- 122 - 145	- 156 - 178	- 190 - 212	- 229 - 251	- 334 - 356	93 36	121 64	157 100	180 122	213 156	247 190	286 229	391 334
120	140	- 40 - 65	- 72 - 97	- 113 - 138	- 139 - 164	- 176 - 201	- 215 - 240	- 259 - 284	- 377 - 402	105 40	137 72	178 113	204 139	241 176	280 215	324 259	442 377
140	160	- 52 - 77	- 88 - 113	- 136 - 161	- 166 - 191	- 208 - 233	- 253 - 278	- 304 - 329	- 440 - 465	117 52	153 88	201 136	231 166	273 208	318 253	369 304	505 440
160	180	- 63 - 88	- 104 - 129	- 158 - 183	- 192 - 217	- 240 - 265	- 291 - 316	- 348 - 373	- 503 - 528	128 63	179 104	223 158	257 192	305 240	356 291	413 348	568 503
180	200	- 69 - 98	- 115 - 144	- 175 - 204	- 213 - 242	- 267 - 296	- 324 - 353	- 388 - 417	- 561 - 590	144 69	190 115	250 175	288 213	342 267	399 324	463 388	636 581
200	225	- 82 - 111	- 133 - 162	- 201 - 230	- 243 - 272	- 303 - 332	- 366 - 395	- 439 - 468	- 632 - 661	157 82	208 133	276 201	318 243	378 303	441 366	514 439	707 632
225	250	- 96 - 125	- 153 - 182	- 229 - 258	- 276 - 305	- 343 - 372	- 414 - 443	- 495 - 524	- 711 - 740	171 96	228 153	304 229	351 276	418 343	489 414	570 495	786 711
250	280	- 106 - 138	- 170 - 202	- 255 - 287	- 308 - 340	- 382 - 414	- 462 - 494	- 552 - 584	- 793 - 825	190 106	254 170	339 255	392 308	466 382	546 462	636 552	877 793
280	315	- 125 - 157	- 196 - 228	- 291 - 323	- 351 - 383	- 434 - 466	- 523 - 555	- 624 - 656	- 895 - 927	209 125	280 196	375 291	435 351	518 434	607 523	708 624	979 895
315	355	- 141 - 177	- 222 - 258	- 329 - 365	- 396 - 432	- 490 - 526	- 590 - 626	- 704 - 740	- 1009 - 1045	234 141	315 222	422 329	489 396	583 490	683 590	799 704	1102 1009
355	400	- 164 - 201	- 256 - 292	- 376 - 412	- 452 - 488	- 558 - 594	- 671 - 707	- 799 - 835	- 1143 - 1179	258 165	349 256	469 376	545 452	651 558	764 671	892 799	1236 1143
400	450	- 187 - 227	- 289 - 329	- 425 - 465	- 510 - 550	- 629 - 669	- 756 - 796	- 901 - 941	- 1287 - 1327	290 187	392 289	528 425	613 510	732 629	859 756	1004 901	1390 1287
450	500	- 215 - 255	- 329 - 369	- 481 - 520	- 576 - 616	- 709 - 749	- 851 - 891	- 1013 - 1053	- 1445 - 1485	318 215	432 329	584 481	679 576	812 709	954 851	1116 1013	1548 1445
500	560	- 240 - 284	- 367 - 411	- 537 - 581	- 643 - 687	- 791 - 835	- 950 - 994	- 1130 - 1174	- 1613 - 1657	354 240	481 367	651 537	757 643	905 791	1064 950	1244 1130	1727 1613
560	630	- 276 - 320	- 419 - 463	- 609 - 653	- 728 - 772	- 895 - 939	- 1074 - 1118	- 1276 - 1320	- 1852 - 1896	390 276	533 419	723 609	842 728	1009 895	1188 1074	1390 1276	1966 1852
630	710	- 310 - 360	- 471 - 521	- 685 - 735	- 819 - 869	- 1007 - 1057	- 1208 - 1258	- 1436 - 1486	- 2046 - 2096	440 310	601 471	815 685	949 819	1137 1007	1338 1208	1566 1436	2176 2046
710	800	- 358 - 408	- 539 - 589	- 781 - 831	- 932 - 982	- 1143 - 1193	- 1370 - 1420	- 1626 - 1676	- 2313 - 2363	488 358	669 539	911 781	1062 932	1273 1143	1500 1370	1756 1626	2443 2313
800	900	- 403 - 459	- 607 - 663	- 879 - 935	- 1049 - 1105	- 1287 - 1343	- 1542 - 1598	- 1831 - 1887	- 2605 - 2661	549 403	753 607	1025 879	1195 1049	1433 1287	1688 1542	1977 1831	2751 2605
900	1000	- 459 - 515	- 687 - 743	- 991 - 1047	- 1181 - 1237	- 1447 - 1503	- 1732 - 1788	- 2055 - 2111	- 2920 - 2976	605 459	833 687	1137 991	1327 1181	1593 1447	1878 1732	2201 2055	3066 2920
1000	1120	- 508 - 574	- 763 - 829	- 1102 - 1168	- 1314 - 1380	- 1611 - 1677	- 1929 - 1995	- 2289 - 2355	- 3254 - 3320	679 508	934 765	1273 1102	1485 1314	1782 1611	2100 1929	2460 2289	3425 3254
1120	1250	- 578 - 644	- 863 - 929	- 1242 - 1308	- 1479 - 1545	- 1811 - 1877	- 2166 - 2232	- 2569 - 2635	- 3647 - 3713	749 578	1034 863	1413 1242	1650 1479	1982 1811	2337 2166	2740 2569	3818 3647

1) The permissible deviations of shaft dimensions correspond to IT 4 above the graduation line, IT 5 between the graduation lines and IT 6 below the graduation line.

2) The maximum and minimum clearance for the shaft/bearing bore fit corresponds to IT 4/H 5 above the graduation line, IT 5/H6 between the graduation lines and IT 6/H7 below the graduation line.

Table 4

If the form and position tolerances cannot be maintained due to the large dimensions of the journal or due to manufacturing problems RENK Werk Hannover should be consulted regarding wider tolerances which they may be able to allow in the special case under consideration.

In case of extreme operating conditions (high degree of bearing utilization), the bearing manufacturer may possibly require degree of accuracy 5 (DIN 316 99).

The fitter is the last specialist to see shaft surfaces before shafts are fitted into the bearing. He should therefore pay very careful attention to surfaces being clean, smooth and even, true to dimensions and undamaged. Any necessary retouching work should be carried out during this assembly stage, though it may involve expenses and some delay. Nevertheless it could be much more cost-effective than a dismantling for trouble caused by overlooked or ignored defects during process-operation.

3.4 Dimensions, Bearing Clearances

In the case of M and I type slide bearings, the shafts are principally made according to the basic shaft system with tolerance fields h6 and h8 in accordance with DIN 7160. The necessary bearing clearance is achieved by the tolerance field of the bore. Please observe the existing literature in this respect. These types of bearings are used as secondary bearings and are normally not subject to high demands.

As far as the other types are concerned, the bearing bores are made according to the basic bore system specified in DIN 7161, with tolerance field H. The bearing clearance is included in the shaft tolerance. The shaft tolerances for eight different relative bearing clearances ψ_m can be obtained from DIN 31 698 (pages 12 and 13).

In case of normal operating conditions the recommendations as per table 5 apply for the selection of the mean bearing clearance ψ_m in function of the peripheral speed v and the bore diameter.

This table does not take into account any extraordinary factors such as, for example:

— high shaft temperature within the bearing in case of heat transfer through the shaft

— considerable elastic deformation through loading of the bearing

— particularly high or low viscosity lubricants

— thermal deformation or greatly varying expansion of journals and bearing shells

— limitation of the inclination of gear shafts.

For bearings with self-contained lubrication (oil ring), with natural or water cooling, a relative bearing clearance of ψ_m = 1,9 ‰ is recommended when the bearing is installed at 2-pole electric machines.

3.5 Determination of Cold-State Bearing Clearance in Case of Heat Transfer into the Bearing through the Shaft

Any external heat transferred into the bearing through the shaft changes to a certain extent the operating conditions compared with the calculation basics. An adaptation by reducing the shaft diameter is required.

The calculation method described hereafter has to be used:

1. determine mean bearing clearance ψ_m according to table 5.
   ψ_m = f (D, v and shape of bore)
2. determine absolute minimum and maximum clearance according to DIN 31 698, table 4.
3. determine decrease of clearance caused by thermal expansion of the shaft:
   Δ s = f (D and shaft temperature)
   assuming a mean bearing shell temperature of 75°C.
   In way of the bearings, the expected shaft temperature due to heat transfer during operation must not exceed:

   a) 80°C with natural cooling

   b) 100°C with water cooling

   c) 150°C with circulating oil

Shape of bore	nominal diameter [mm]	ψ_m [‰] for v [m/s]				
		... 3	> 3 ... 10	> 10 ... 25	> 25 .. 50	> 50 ... 125
plain cyclindrical bore	... 100	1,32	1,6	1,9	2,24	—
	> 100 ... 250	1,12	1,32	1,6	1,9	—
	> 250 ... 500	1,12	1,12	1,32	1,6	—
	> 500	0,8	1,12	1,32	1,32	—
two-lobe bore	... 100	—	—	1,6	1,9	1,9
	> 100 ... 250	—	—	1,32	1,6	1,9
	> 250	—	—	1,12	1,32	1,6
four-lobe bore		—	—	1,32	1,6	1,9
journal tilting pads with adjustable bearing clearance (EV)	... 100	—	0,8	1,12	1,12	1,32
	> 100 ... 250	—	0,8	0,8	1,12	1,12
	> 250 ... 500	—	0,56	0,8	0,8	1,12
	> 500	—	0,56	0,56	0,8	0,8
journal tilting pads with fixed bearing clearance	... 100	—	1,32	1,6	1,9	1,9
	> 100 ... 250	—	1,32	1,6	1,6	1,9
	> 250 ... 500	—	1,12	1,32	1,6	1,9
	> 500	—	1,12	1,32	1,6	1,6

The table shows bearings with two-lobe bores the mean vertical relative clearance.
Normally a clearance ratio of $\varphi = \Psi \text{ hor} / \Psi \text{ vert} \approx 2 \dots 2,75$ is selected.

Table 5 Selection of mean bearing clearance

4. The reduction of clearance Δ s allows together with DIN 31 698 the determination of the required cold-state bearing clearance which may result one, two or three stages higher.

This cold-state bearing clearance represents the basis for the selection of the shaft dimension.

Example

bearing bore diameter: D = 110 mm
expected shaft temperature in way of the bearing: 130°C

$\Delta T = (130 - 75)\ K = 55\ K,\ \gamma_1 = 11{,}1 \cdot 10^{-6}\ (\frac{mm}{mm \cdot K})$

$N = 1500\ min^{-1}$, $v = 8{,}6\ m/s$

1. mean bearing clearance ψ_m per table 5
   1,32 ‰
2. bearing clearance during operation and therefore basis for bearing calculation:
   $S_{min} = 110$ mm, $S_{max} = 167$ mm per table 4
3. $\Delta S = D \cdot \Delta T \cdot \gamma_1$
   $= 110 \cdot 55 \cdot 11{,}1 \cdot 10^{-6}$ mm
   $= 0{,}0671$ mm
   $= 67\ \mu m$

When manufacturing the shaft, the diameter has to be reduced by this figure.

$S_{min\ (kalt)} = 110 + 67 = 177\ \mu m$
$S_{max\ (kalt)} = 167 + 67 = 234\ \mu m$

The above calculated values for $S_{min\ (kalt)}$ and $S_{max\ (kalt)}$ correspond to approx. $\psi_m = 1{,}9$ ‰, i.e. an increase of the cold-state bearing clearance by two stages is necessary to compensate for the effect of heat transfer through the shaft in way of the bearing.
Permissible deviations of the shaft min. - 171 max. - 193 per table 4.

4. Whitemetals

For the running surfaces of the slide RENK use mainly high percentage tin alloys, in special cases lead alloys, according to the works standards.

The following three alloys are used:

RENKmetal therm 89	Tin-based whitemetal, leadfree and low in heavy metals
RENKmetal therm V80	Tin-based whitemetal, leadfree, with hardening and tempering alloying constituents
RENKmetal therm V6	Lead-based whitemetal with hardening and tempering alloying constituents

Several factors have to be taken into account when choosing the whitemetal.

First of all the maximum admissible temperature has to be taken into account. In the case of whitemetals „therm V6“ and „therm V80“ with lower softening points of 240°C and 235°C , respectively, the peak temperature in the whitemetal must not exceed 150°C. Higher temperatures would lead to plastic deformation of the whitemetal even with small loads.

Name		RENKmetal therm 89	RENKmetal therm V80	RENKmetal therm V6
Chemical composition in percentage [%]				
Alloying constituents	Sn	88 ... 90	79 .. 81	5 ... 7
	Pb	—	—	73 ... 79
	Sb	7 ... 8	11 ... 13	14 ... 16
	Cu	3 ... 4	5 ... 6	0,8 ... 1,2
	As	0,1	0,4 ... 0,6	0,3 ... 0,8
	Cd	—	1,0 ... 1,4	0,6 ... 1,2
	Ni	—	0,2 ... 0,4	0,2 ... 0,5
Admissible additives	Bi	0,08	—	0,08
	Pb	0,06	0,06	—
	Fe	0,01	0,01	0,01
	Al	0,005	0,005	0,005
	Zn	0,005	0,005	0,005
	other together	0,02	0,02	0,02
Technological properties				
Density ρ [kg/dm^3]		7,3	7,34	9,84
Modulus of elasticity E [N/mm^2]		56500	52500	29900
Poisson number ν [—]		≈ 0,33	≈ 0,33	≈ 0,44
0,2%-crushing yield point $R_{d0.2}$ [N/mm^2] at	20°C	47	80	46
	50°C	44	73	43
	100°C	27	48	26
Compressive / upsetting strength R_{dSch} [N/mm^2] at	20°C	48	70	46
	60°C	38	56	36
	100°C	28	38	24
Impact fatigue strength		very high	medium	low
Brinell hardness HB 10/250/180 at	20°C	22	35	25
	50°C	17	27	21
	100°C	11	17	14
Coefficient of linear expansion α_1 [10^{-6} K^{-1}]		23,9	20,2	24,7
Melting temperature [°C]		233 .. 360	235 ... 390	240 ... 400
Casting temperature [°C]		460	520	520

Table 6 Whitemetals, composition and physical properties

The peak temperatures specified are not the „admissible bearing temperatures“ specified in regulations and guidelines. These are usually considerably lower and refer to the so-called „mean bearing temperature“ or to measurement of the oil sump temperature only.

One advantage of the tin based whitemetals therm V 80 is the higher fatigue strength. Die höchste Dauerschlagbiegefestigkeit erreicht der Lagerwerkstoff „therm 89“.

Other criteria which have to be taken into account in individual cases are hardness, crusing yield point, compressive strength and upsetting strength.

5. Lubrication

5.1 General

RENK slide bearings for horizontal shafts can be operated as self lubricating bearings by means of oil rings of oil discs also with additional circulating oil if necessary or with circulating oil lubrication alone.

Oil discs are fitted to the shaft for up to 17,5 m/s peripheral speed at their outer diameter and oil rings are fitted for up to approx. 20 m/s These rings/discs are also often fitted as a precaution against failure of an external circulating oil supply; because with them the plant can be brought to a stop without fear of damage.

For emergency operation of bearings with lubrication by circulating oil, oil rings may be used for a peripheral shaft speed up to 26 m/s, oil discs up to a peripheral speed of 20 m/s at the disc’s outer diameter. It must be observed that in such a case, oil rings serve as a protection for the radial part only. Thrust bearings subjected to heavy loads require a special emergency oil supply.

Generally any branded mineral oil of low foaming tendency and good resistance to ageing can be used for lubrication purpose provided that it has the viscosity prescribed by RENK Hannover. If for any reason it is necessary to use an oil of different viscosity, RENK Werk Hannover should be informed because if the deviation in viscosity is considerable the bearing would have to be re-designed to different calculations.

In the design calculation of RENK slide bearings a certain operating temperature is taken as the basis. Therefore in order to attain the theoretical values in practical working, the oil must be of the viscosity prescribed by us. The viscosity is given for each of the different versions in the acknowledgement of order or in the instructions of assembly, operation and maintenance supplied.

We strongly recommend to follow our suggestions without exception and keep informed all those responsible for the initial start-up and those in subsequent day-to-day charge of the plant.

If additive-treated and synthetic oils are used, please ensure that these will not attack the whitemetal used (see acknowledgement of order) and that they will not coke if heated by immersion heaters.

5.2 Definition of Viscosity

German Standards Specifications (DIN 1342) define viscosity as follows:

> „Viscosity is the property of the flowing (mainly fluid or gaseous) medium of being able to take up the stress in any deformation, the stress being related to the deformation speed. The stress can also be regarded as the cause of a deformation speed.“

5.3 Viscosity/Temperature (V/T) Reaction of Lubricating Oil

The behaviour viscosity/temperature cannot be defined by simple regularities. It can be shown best in the form of graphs. In the Niemann V/T graph and DIN 51 519 (fig. 30) the dynamic viscosities of usual lubricants are entered in relation to the temperature.

From the graph it will be seen that the viscosity decreases considerably as the temperature rises.

5.4 Selection of Viscosity Classes of Lubricating Oils

As a general recommendation for the selection fo the ISO class of viscosity (ISO VG) the following table 7 was made up. The table does not take into account any unusual influences or requirements, such as

- high shaft temperature by heat transfer into the bearing
- high oil inlet temperature
- operation at turning speeds for long periods
- especially high or low ambient temperatures
- low power losses, in order to operate the bearing with self-contained lubrcation also at higher speeds

In critical cases or reasons of standardisation or in compliance with user’s requests other oil viscosities may be chosen. In each case it is necessary to confirm the selection with a bearing calculation.

$\overline{p}$ [N/mm^2]	ISO VG for v [m/s] =				
	... 3	> 3 ... 10	> 10 ... 25	> 25 ... 50	> 50 ...
... 1,25	68	46	46	32	32
> 1,25 ... 2,5	100	68	46	46	32
> 2,5	150	100	68	46	46

Table 7

v = sliding velocity of the shaft or mean sliding velocity of the collar

Attention!
For self-lubricated E-bearings (with loose oil ring) with natural or water cooling, lubricating oil ISO VG 32 is recommended for the use with 2-pole electric motors.

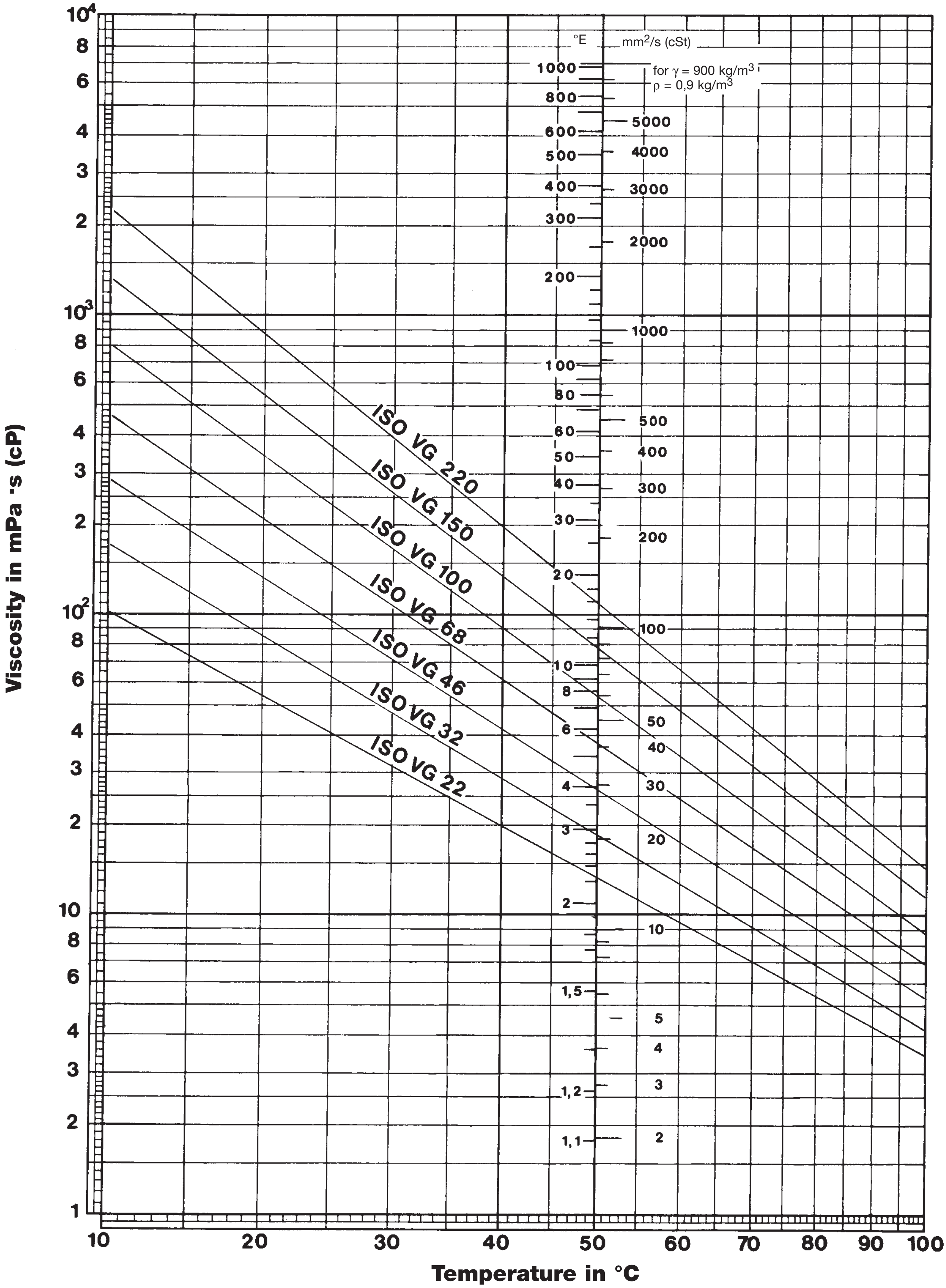


Figure 30 Viscosity-temperature-graph for mineral oils (with reference to G. Niemann and DIN 51 519)

6. Bearing Temperatures

6.1 Admissible Bearing Temperatures

Considering the operating life of lubricating oils and the heat resistance of white metals we make the following suggestions for admissible temperatures of our bearings:

a) measurements with standard thermoprobes in the oil flow or respectively in the oil bath:

T_{lim} = 80°C

b) measurement with standard thermoprobes in the loaded zone of the bearing shell:

T_{lim} = 90°C

c) measurements with good-quality thermoprobes which are carefully built in and have metallic contact for certain with the bearing shell in the loaded zone:

T_{lim} = 110°C

d) precise measurements with resistance thermometers (possibly also with thermocouples or other measuring instruments) in approx. 1...3 mm distance from the bond compound surface steel/whitemetal in the zone of maximum temperature:

T_{lim} = 125°C

If, due to hydrodynamic calculation, the peak temperatures are to be expected higher than 125°C bearing metals and lubricating oils should be carefully selected. In that case for bearings with high loadings and high speeds temperatures of up to

T_{lim} = 150°C

may be allowed.

If Installation or Survey Instructions or specific guidelines state maximum admissibe temperatures the relevant measuring points have to be considered.

Extremely low admissible bearing temperatures – as sometimes required – often result from measurements in the oil sump or at the oil outlet and cannot be compared with the temperatures taken in the loaded zone of the bearing.

6.2 Alarm and Shut-Down Temperatures

Basis for the initial setting of alarm and shut-down temperatures is the calculated operating temperature. However this calculation of operating temperature must take into consideration the highest possible site ambient, and/or oil inlet temperature, as well as the maximum speed and relevant loadings.

The alarm temperature should be set 10 K higher than the calculated bearing temperature and the shut-down-temperature approx. 20 K higher.

If, after start-up of the machine, the actual bearing temperatures are considerably higher than those calculated (e.g. by heat transfer to the bearings), the measured values will serve as the new basis for setting the alarm and shut-down temperatures, this however, only after having consulted RENK. The values should be rounded off to figures ending in 5s or 10s.

6.3 Temperature Measurements

Generally the following types of temperature measurement are used in engineering:

6.3.1 Measurements Using Liquid-Filled Thermometers

6.3.1.1 Bar and angle thermometers facilitating direct reading of the liquid column along a linear scale.

6.3.1.2 Needle thermometers, where the expansion of the liquid pressurises a Bourdon tube and where the expansion is transmitted to the shaft of the thermometer by means of a pinion.

Where a heat sensor and display unit are fitted separately, they are linked by means of a capillary tube. This must never be opened up, since it forms an integral part of the system.

6.3.2 Measurement using Electrical Thermometers

6.3.2.1 Thermocouples, as the name indicates, are elements, where the soldered junction of two wires made from different materials produces a voltage, the magnitude of which depends on the temperature, where this voltage can be read using a voltmeter calibrated in temperature units.

The advantage of thermocouples is that they are very small and that they are suitable for very high temperatures; their disadvantage is the larger measuring error compared with resistance thermometers and the need to use so-called compensated leads between thermocouples and measuring instruments, which must be made from the same materials as the thermocouple itself, since the junctions would otherwise represent another thermocouple in itself. Compensate leads can only be avoided if a measuring transducer is fitted directly to the thermocouple. Thermocouples age and change their measuring characteristics. This ageing cannot be predicted.

Resistance thermometers are frequently used, and for this reason they will be dealt with in detail here.

Since these are electrical or electronic methods, the description which follows – which is not intended to be scientifically accurate – is meant for easy understanding and relates specifically to the Pt 100 sensors in use in Europe.

The principle of temperature measuring using resistance thermometers is based on the measurement of the electrical resistance of a measuring resistance in the temperature sensor. The electrical resistance is a function of temperature i.e. the temperature-dependent change of the electrical resistance of the conductor (measuring resistor) is used for the determination of the temperature.

Materials with a large temperature coefficient are used for a measuring resistor. In addition great care is taken to ensure that the materials do not age in the temperature range in which they are used. For this reason platinum (Pt 100) is mainly used. „100“ means that this measuring resistance has a resistance of 100 Ohms (Ω) at a temperature of 0°C.

In accordance with DIN 43760 there must be a resistance ratio of $R_{100}/R_0 = 1,385$, i.e. at 100°C the resistance is 138,5 Ohms.

The characteristics curve of the measuring resistance is defined in accordance with the equation:

$$R_T = 100\,(1 + 3{,}90802 \cdot 10^{-3}\,T - 0{,}580195 \cdot 10^{-6} \cdot T^2)$$

where R_T [Ω] is the resistance at temperature T [°C].

For the display unit it is also possible to provide a digital display with LED (illuminated) or LCD (dark).

A more elaborate version of the display is the regulator (fig. 31). A regulator (generally an electronic unit) can be designed as a single, two or three point regulator. It is then possible to use these points as switching points. They actuate a limit switch once the set temperature has been reached. This contact can be used for any signalling and regulating purpose.

Some regulators are also provided with an output to which a temperature chart recorder can be connected.

It is possible to have a 2, 3 or 4-wire circuit between the temperature sensor and the display or regulator, in accordance with the circuit diagrams given on our type sheets.

To be able to use our standard thermoprobes the 3 and 4-wire circuit should begin in the connecting head. With the sensor lengths usually used in slide bearings a difference of approx. 0,1 K results directly at the measuring resistor which represents a figure below the measuring accuracy of the instrument.

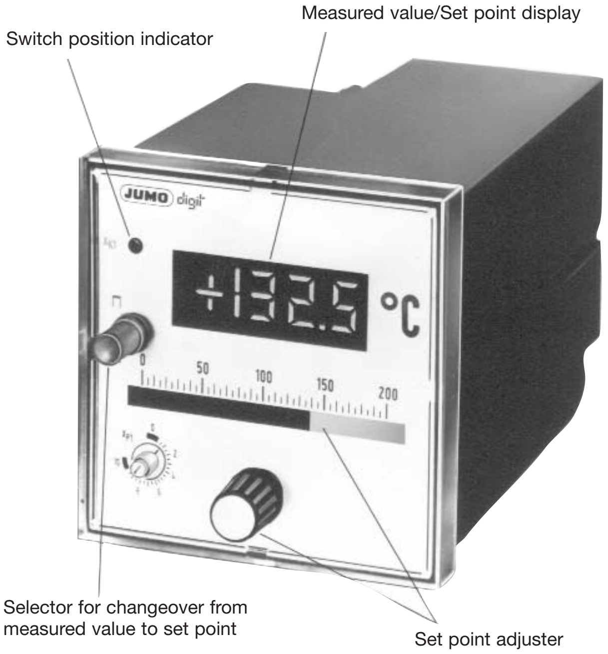


Figure 31 Regulator with digital display

Our scope of supply merely consists of the heat sensor which is supplied in the form of a flexible Pt 100 unit according to type specification RH 1016 (fig. 32), as a screw-in type resistance thermometer in accordance with type sheet RH 1015 (fig. 33) and as a sheathed resistance thermometer in accordance with type sheet RH 1036 (fig. 34).

These resistance thermometers are products which are manufactured specifically for our application by well-known German manufacturers. For this reason they are generally more suitable for the temperature measurement with slide bearings than standard products.

Display and regulating instruments do not normally form part of our scope of supply, since they are generally installed in a control panel which is central for the entire installation. However we shall be pleased to supply such units to special order.

Figure 32 Resistance thermometer Pt 100 flexible

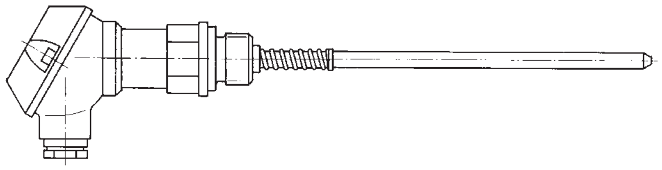


Figure 33 Screw-in type resistance thermometer Pt 100

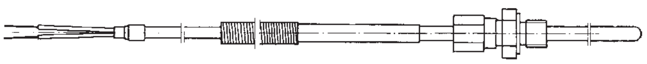


Figure 34 Sheathed resistance thermometer Pt 100

7. Heat Dissipation, Cooling

7.1 Radiation and Convection (Natural Cooling)

In many cases radiation and convection will be sufficient to limit the bearing temperature.

The temperature of the ambient air must be taken into account in the design stage. In most cases it will be impossible to protect the bearing from the ambient temperature. The ambient temperature has a direct influence on the heat-up of the bearing.

7.2. Forced-Air Convection Cooling

Forced air convection cooling is a better form of cooling. It is induced by the speed-up of air movement over the bearing housing by means of fans on the shaft or by separately installed blowers. Care must however be taken to see that no oil is drawn out of the seals by vacuum action. (Special seals or protections are available. See the section on „Seals“.)

If, in the case of slide bearings with natural cooling or forced convection cooling, the atmospheric temperature increases by ΔT_{amb} , then the bearing temperature will also increase at the same time.

The temperature rise ΔT_B in the slide bearing can be roughly estimated with the aid of the approximation formula

$$\Delta T_B = 0,7 \, . \, \Delta T_{amb}$$

7.3 Water Cooling

In cases of higher peripheral speeds where the oil supply by means of oil rings/discs is still sufficient, but the heat generated in the bearing or transferred through the shaft can no longer be dissipated through the housing surface, water cooling can be used.

Two kinds of water cooling have proved to be very efficient: Water cooling through cooling channels cast in the base and enclosed by a plate (Series Slide Bearings Type M and I) or cooling coils of smooth or finned tubing inserted into the oil sump (Series Slide Bearings Type E , EV, SN, DN and VD [fig. 35]).

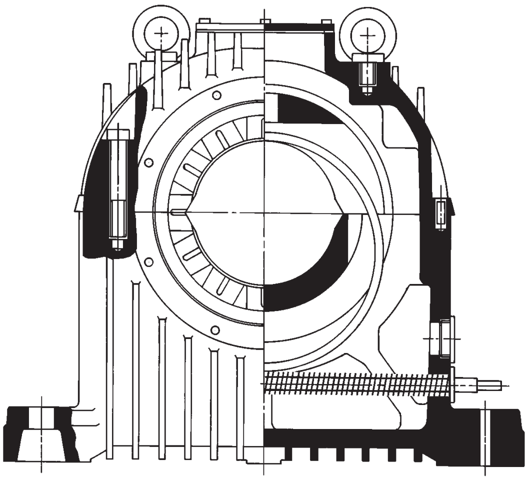


Figure 35 Bearing with water cooling

The discharge of the water should be pressureless. The cooling circuit must be drainable.

If seawater is to be used the materials for the cooling coils must be special for the purpose – such as CuZn20AAlF40 or CuNi10Fe. The channels cast in the base of the bearing are not seawater resistant.

CuNi30Fe, stainless steel or titanium are special materials used for certain specific applications.

The speed of water should not exceed 1,5 m/s in order to avoid damage by cavitation particularly in pipe bends, customers must fit the corresponding control valves.

7.4 Finned Housing Surface

The finning on the housing surface (fig. 36) is advantageous for backing up the two forms of cooling described.

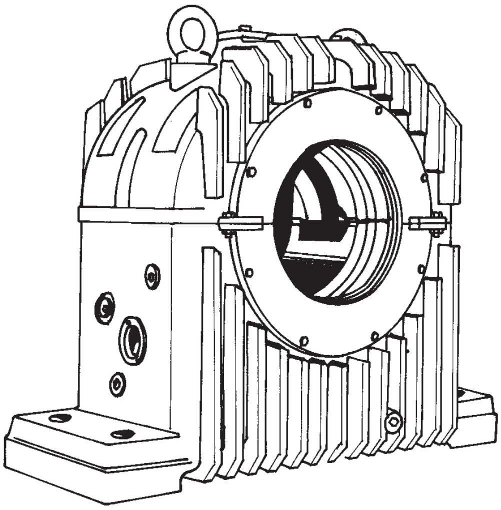


Figure 36 Finned bearing housing

7.5 External Oil Supply

An external oil supply (lubrication by oil circulation from an external system) is necessary when ring/disc lubrication If the bearing cannot function any more due to exceeding the admissible peripheral speed or when great frictions losses and/or heat transfer through the shaft are demanding external oil cooling. In general bearings should always be connected to oil supply systems where a system is already installed and contains oil suitable for the bearing. in such cases, the oil rings/discs sometimes may replace stand-by pumps.

Gear bearing shells are always supplied with oil from the oil circulating system of the gear box. Ensure compatibility of the mostly high-alloyed oils with the whitemetal. The precalculated oil quantity required by the bearing must be available.

Depending on the particular requirements, oil-supplying systems can be extremely varied in design (fig. 37). In principle such a system consists of an oil container capable of taking at least 5 times the throughput of the pump in one minute. To calculate the volume of the container it must also be taken into consideration that, depending on the length and cross section of the piping system, a more or less large quantity of oil is in circulation, and this must be accommodated in the container when the plant is stopped. Even when the plant is working there must still be an adequate quantity in the container as the recycled oil mostly carries air with it which must be separated before the oil is recirculated. A point to watch is that any coating applied within the container should be oilproof. Epoxy resin-based two pack paints have proved their suitability for this purpose.

Varnish dissolved in oil can cause damage!

According to the working requirements one or several pump units (i.e. pumps with their motors coupled to them) are mounted to the container.

For simple systems in which operation can be stopped on pump failure without a prolonged slow-down time or in which during slow-down the oil supply is assured by oil rings/discs, one pump unit would be sufficient.

For systems with prologend run-down times and those installed in plants which require to be in continuous operation and which have no rings/discs to keep the lubrication going, a second pump is provided as stand-by.

Where the safety requirements are stringent (e.g. for mine ventilators) a third pump unit is installed.

The main pump unit which ensures the oil supply during normal operation is supplied with electrical power from the mains. The second pump (and, where applicable, the third), as stand-by, must be driven by a source of energy which is independent of the mains (emergency 3-phase current, DC, compressed air).

If, during operation, the oil supply is assured by a shaft-driven pump, care must be taken that the motor-driven start-up pump does not switch off until the shaft-driven pump delivers an adequate quantitiy of oil of the required pressure.

The exclusive oil supply by means of a shaft-driven pump is only possible if during the starting phase the bearings can be supplied with lubricating oil by an oil ring/disc.

For plants operated solely on circulating oil ist must be assured that the lubricating oil pumps do not cut the supply off until the rotating masses come finally to a stop (this particularly applies to great mass inertia moments with prolonged slow-down times).

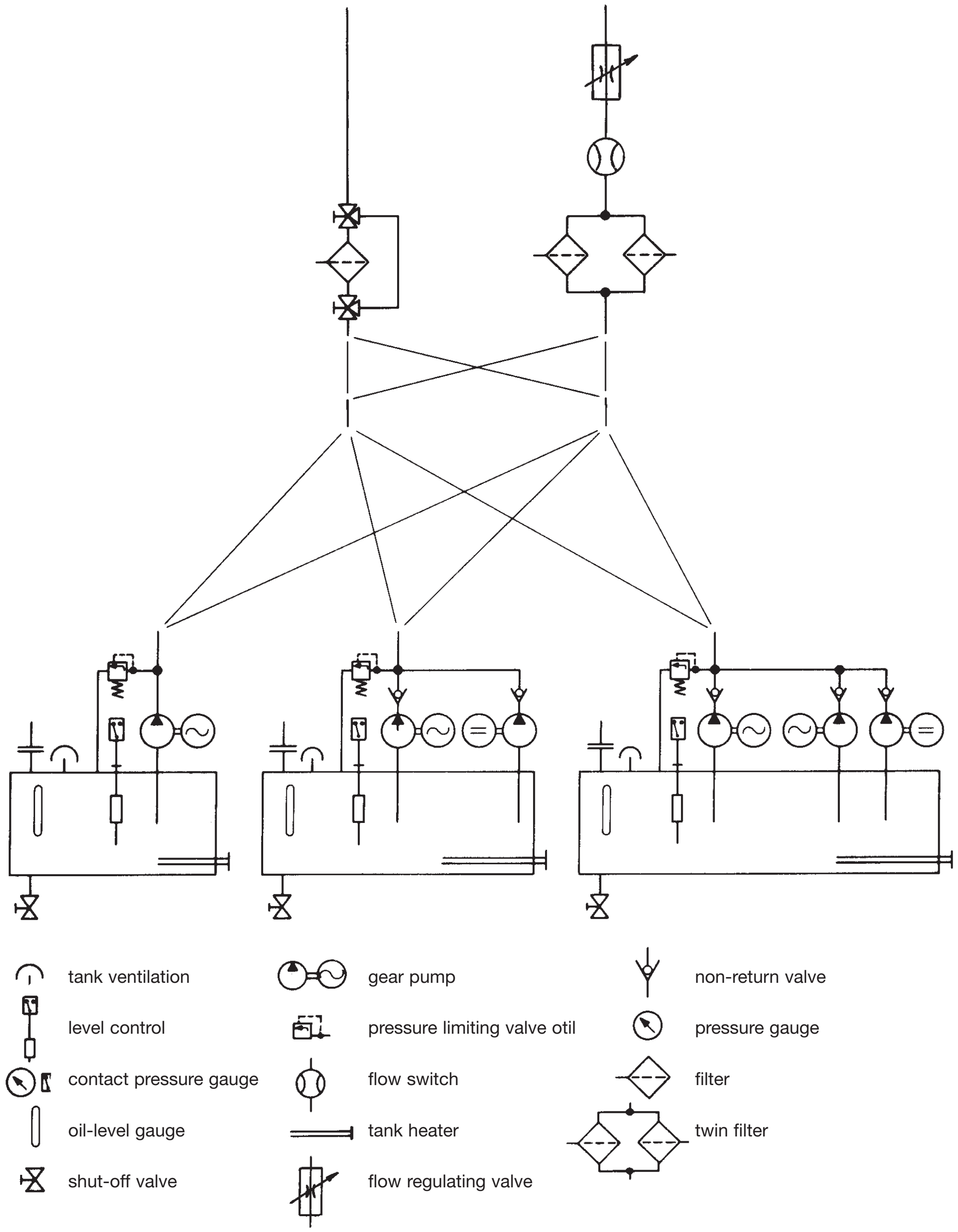


Figure 37 Oil supply system switch diagram for fluid media

Also take into account that directly coupled pumps deliver oil only up to a certain minimum speed. In this case the electric-motor driven pump must be switched on again for the slow-down or, if possible, the lubrication must be guaranteed by oil rings/discs.

Especially for highly loaded bearings see that the circulating oil system comes into operation before the plant starts up. This will avoid the danger of dry running.

7.6 Installation of the Oil Supply System (fig. 38)

The oil supply system should be installed in such a way that the pipelines to and from the bearings are of equal or nearly equal lengths.

The pipe bends, T-pieces etc. in the piping must be measured and added to make the total pipe-length. The viscosity of the oil must be noted in the calculation of pipeline resistances. The oil quantities must also be delivered in the case of a cold start (see V/T-Graph, page 18 for appropriate viscosity).

The difference in level between the bearings and the container should be such that the recycling pipelines can be laind at an inclination of approx. 15° (approx. height difference of 25 cm in 100 cm length).

Also, it is advantageous to begin the recycling pipeline with a greater inclination directly at the bearing.

If the oil supply system is placed out-of-door, it must be protected against atmospheric influences (heat, frost, rain, wind) in order to avoid fluctuations in temperature which could lead to failure of the equipment.

7.7 Pipelines

For inlet pipelines use precision steel tubes to DIN 2391 or steel tubes to DIN 2448 and for connection use dutting rings or annealed cast iron fittings. The pipelines may be bent and welded if required by the working conditions (vibrations which could cause leakage, safety rules and regulations etc.) The cross-section should be such that the flow speed does not exeed 1,5 m/s.

For the return lines steel piping to DIN 2448 is practically exclusively used due to the required large cross-section. Instead of pipe fittings use pipe bends of as greatest possible radii in order to reduce pipeline resistance. Junctions in return lines are to be tangential to the flow (fig. 38). The cross-sections should be selected in such a way that the flow speeds do not exceed 0,15 m/s, based on the full pipe cross section.

When the slope in the return lines, for design or constructional reasons, cannot be laid at an angle of inclination of 15° the cross-sections must be selected correspondingly greater (larger cross section just behind the bearing connection). Too slight a slope or/and to small a cross-section causes a damming-up effect which in turn can cause overflowing of the bearing and leakages.

It is inappropriate only to lay pipes vertically and horizontally for aesthetic reasons. Not only the difference in level of the pipe but also the actual angle of the pipe is to be regarded as the gradient.

Heat-treated (welded and hot bent) and/or internally rusted piping and/or piping very dirty on the inside must be pickled before pipeline laying.

> **Attention!**
> Wear protective clothing (rubber apron, rubber boots, rubber gloves) ! Wear eye-shielding glasses !

Check the current regulations governing the allowed concentration of alkalis and acids in drainwater. Before releasing any residue of either of the above solutions take steps to ensure that the valid Environmental Regulations are observed.

After installing the pipelines, rinse the whole of the oil circulation system to prevent dirt or impurities penetrating into the bearing and the pipe-fittings, using kerozene or rinsing oil. It is essential to remove all measuring gauges and fitments (e.g. pressure monitors, flowmeters) and to close-up the connections for them.

Never leave the slide bearing in rinsing-oil circulation otherwise dissolved particles of undesirable matter could enter the bearing and settle as a deposit in front of jets and pockets causing irreparable damage to the plant.

After rinsing clean the filters.

7.8 Oil Pumps

The oil supply systems are in nearly every case equipped with gear pumps. The direction of rotation of the motor generally determines the conveying direction of the pump, therefore check the electrical connections to the motor in order to obtain the desired direction of rotation. The direction of rotation is indicated by an arrow.

Before initial setting into operation fill the pump with oil. It must never be allowed to run in dry state.

Some pumps are fitted with overflow valves which return small excess quantities to the suction end. The maximum pressure can be set with these valves. Large quantities of oil should not be allowed to flow through these valves because the oil entering the circuit heats up considerably.

If the suction of the pump ceases, inspect all seals and gaskets of the pump and replace by new ones where necessary.

Suction heights of over 1,5 m are to be avoided (in long horizontally laid suction lines the pipeline resistance has to be added to the suction height).

Suction pipes of pumps placed at a distance from the bearings must be executed as short as possible and with minimum pipe resistance. Information about it will be furnished by the supplier of the pump.

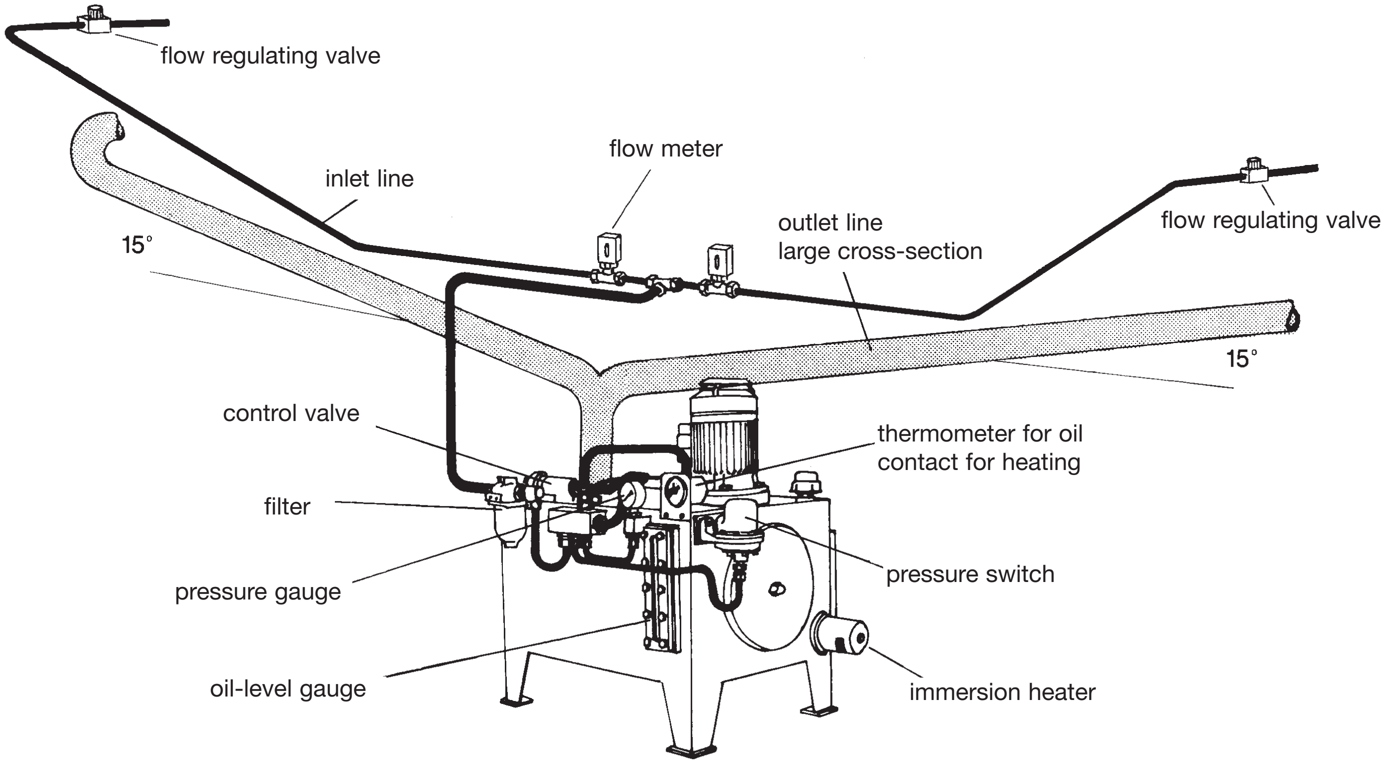


Figure 38 Diagramm of an oil supply system

7.9 Oil Coolers, Oil Filters

Oil coolers and oil filters causing pressure losses must always be fitted to the pressure end. Because of the large number of different types, only basic rules can be given here; otherwise please refer to the special fitting and operating instructions of the particular manufacturer.

The water is to be connected to the oil cooler so that it cannot freeze, even with the water circulation shut of (fit drainage pipes). It is not possible to maintain the circulation of the water because of the low oil temperature existing in this case.

Where hard water is used for oil cooling, the water system of the cooler must be decalcified from time to time.

In line with the operating conditions (degree of contamination of the oil) filters must be cleaned according to the instructions of the manufacturer.

In the majority of installations, twin change-over filters are used so that whilst one filter is in operation, the second filter can be cleaned without interrupting operation.

Mesh size of the filters to be 15 ... 25 µm.

7.10 Fittings and Monitoring Gauges

Fittings and monitoring gauges should be mounted according to manufacturer's instructions and (where applicable) incorporated into the electrical system as indicated in the circuit diagram.

Attention should be paid to the correct mounting position and flow direction, where rules for these are given.

Never incorporate into the line between pump and bearing any valves, taps or stopcocks which are not secured against unauthorised operation.

A good system should be equipped with the following monitoring devices:

1. Pressure switches or contact pressure gauges in front of filters and coolers.
2. Contamination indicator or differential pressure gauge on the filter.
3. Good temperature measuring systems with sensors incorporated into the bearing; (e.g. RENK screw-in probe Pt 100, RENK Pt 100 with flexible connections or embedded resistance thermometer Pt 100).
4. Temperature measuring equipment in the oil tank for measuring the oil temperature (not essential, can however be applied for controlling a heater and water cooler built into the tank).
5. Flow-meters (e.g. RENK flow meters NJ-MI) up-line of each bearing represent an important assembly aid. The normal methods used up to now of gauging the capacity with a measuring vessel (bucket) when the installation is stationary and cold, is no longer required. When using flow-meters, the installation is adjusted when cold and can then be re-adjusted without any problem once the operating temperature has been reached. The flow-meters subsequently serve to monitor the situation on the site and then indicate unauthorized changes in valve settings, etc., immediately.

With an additional switching unit they can be used as flow switches.

8. Protection against External Influences

8.1 Heat of radiation

Protection against heat of radiation is nearly always possible. In most cases an asbestos layered screen placed between the heat source and the bearing will be found to be sufficient. Heat of radiation can heat the housing considerably and will prevent dissipation of friction heat into the surrounding air.

8.2 Heat-Transfer through the Shaft

Heat transfer through the shaft must be avoided as far as possible. Therefore, cooling discs (fig. 39) which will prevent heat transfer into the bearing through the shaft have to be arranged in the shaft section between the area of heat absorption and the bearing area.

Attention: If the cooling disc is placed too close to the bearing, negative pressure is created in front of the shaft outlet which may lead to oil leakage.

For this reason fig. 39 shows a bolt-on baffle as per fig. 45 arranged on the side towards the cooling disc.

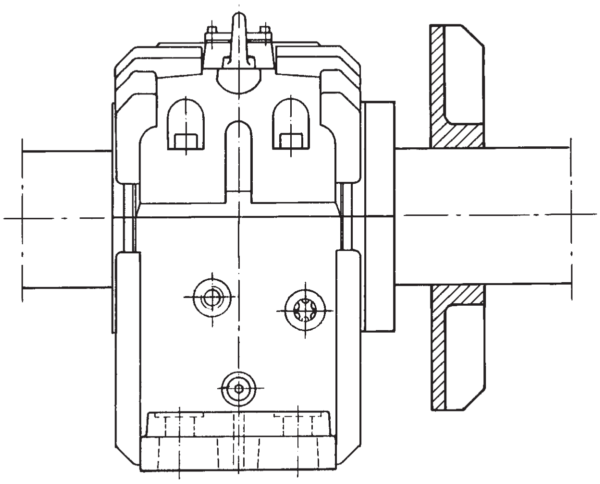


Figure 39 Arrangement of a cooling disc

8.3 Cold

Protection from cold is necessary, if the bearings are exposed to very low ambient temperatures because the viscosity of oil increases significantly.

Frequently it is necessary to start up machines with slide bearings in the open air or in unheated rooms at low ambient temperatures.

Heater bars can be installed in the oil reservoirs of slide bearings with oil circulating systems or in the sump of the slide bearing housings. Depending on their power, the oil will be heated up within a given period of time to a level at which it can be recirculated for example by means of a gear pump.

The specific heater loading depends on the type of oil and should not exceed 1.5 W/cm^2. With highly viscous and additive-treated oils this value needs to be reduced further, to avoid coking.

Depending on the distance of the oil supply system from the slide bearings, the feed and return lines need to be insulated, heating may possibly be required.

The installation of a heater in the bearing housing of explosion protected machinery not only requires the use of expensive special heater bars.

Temperature regulator and the necessary oil level control system need to be of an explosion-protected design and are therefore very elaborate.

Bearings with loose oil rings (E-bearings), aftermost and intermediate bearings of ships, trunnion bearings and idler bearings can be operated safely up to a viscosity η of 1000 mPas, whereas disc lubricated bearings (M bearings, I bearings) can be operated only at viscosities up to η = 800 mPas.

In the case of self-lubricated bearings with a loose oil ring it is important to prevent the oil ring from being braked too much. This risk only arises with viscosities above 1000 mPas. Fixed oil rings are not at risk through braking, but here the lubricating oil needs to penetrate to the oil pockets of the bearing through bores. When the viscosity is in excess of 800 mPas, this can no longer be ensured.

It is possible to avoid malfunctions through oil changes at the beginning of the cold and at the beginning of the hot season respectively („summer oil“ and „winter oil“). Thinner oils in a given range of oils generally have a lower pour point. (The pour point of a mineral oil is the temperature at which a sample, after cooling under specified conditions, no longer flows [DIN 51 597].)

Example 1

Oil ISO VG 100, pour point = – 18°C

Lowest permissible temperature for the oil:

T_{amb} = – 18°C + 10 K = – 8°C

Lowest permissible temperature for the oil disc bearing (η_{min} = 800 mPas)

= + 10°C

Permitted minimum start-up temperature:

= + 10°C

Example 2

Oil ISO VG 22, pour point = – 28°C

Lowest permissible temperature for the oil:

T_{amb} = – 28°C + 10 K = – 18°C

Lowest permissible temperature for the bearing with loose oil ring (η_{min} = 1000 mPas)

= – 25°C

Permitted minimum start-up temperatures:

= – 18°C

If, in certain cases, the temperature is below the admissible minimum start-up temperature, then it is necessary to contact RENK-Hannover.

8.4 Weather Protection

It is possible to design the bearing in such a way that the housing is sealed against influences of the weather. In order to achieve this, special measures are necessary which vary according to the type of bearing and also the speed of the shaft. However, due to fluctuations in temperature, these measures cannot prevent the formation of consideration inside the bearing. After some time, especially with self-lubricated bearings, this may cause the oil to emulsify if, for example, the bearing is exposed to solar irradation during the day and cools down considerably at night. The loading capacity of the oil film is reduced as a result.

Bright parts (shaft) may rust during down times.

Therefore, slide bearings which are operated outdoor should at least have a protective roof.

9. Shaft Seals

9.1 General

Shaft seals fitted to slide bearings are intended to

- prevent or limit to the least possible extent the leakage of oil or oil mist
- prevent the penetration of harmful quantities of impurities (sand, dust) and water

The seal type is selected by RENK Werk Hannover taking into consideration the oil throughput, i.e. the strain applied to the seal. With low oil throughput and low oil turbulences a simple sealing system may be adopted while high oil throughput and strong turbulences require a more elaborate sealing system.

Depending on site conditions, shaft seals have to respond to different requirements with regard to the penetration of impurities and water. If such data are known, RENK-Hannover is in a position to select the appropriate sealing system. In most of the cases, the user will indicate the required grade of protection (according to DIN 40050 or IEC-Publication 529).

The following types of seals are available standard items:

1. Labyrinth seal
1.1 Floating labyrinth seal (fig. 41)
1.2 Floating oil sealing rings (fig. 42)
1.3 Rigid labyrinth seal (fig. 43)
1.4 Labyrinth seal combined with dust flinger (fig. 44)
1.5 Labyrinth seal combined with bolt-on baffle (fig. 45)
2. Wind-back seal (fig. 46)
3. Soft packing (fig. 47)
4. Felt seal (DIN 5419)
5. Grease seal (fig. 48)
6. Flinger seal (fig. 49)

Special seals, such as air seals and contact seals are available for special requirements. Details on request.

The type of protection of the seal in accordance with DIN 40 050 is indicated underneath each figure.

9.2 Negative Pressure

Where negativ pressures occur at the seal, oil mist will be sucked out of the bearing. Such a local vacuum will also occur in case of strong air turbulences at the bearing (e.g. created by a coupling).

The bolt-on baffle as well as the dust flinger will counteract the emission of oil mist.

The labyrinth groove of the seal carrier for the dust flinger can also be filled with grease, so that the sealing gap is reduced to a minimum. The grease and in particular the grease collar thus formed can also trap dust.

The following table shows the values of admissible negative pressures in mm head of water which may occur in front of a specific type of seal or seal combination without causing undue leakages.

type 10 (fig. 41)	floating labyrinth seal	5 mm head of water
type 11 (fig. 44)	floating labyrinth seal combined with dust flinger	25 mm head of water
type 12 (fig. 45)	floating labyrinth seal combined with bolt-on baffle	15 mm head of water
type 20 (fig. 43)	rigid labyrinth seal	10 mm head of water
type 21	rigid labyrinth seal + dust flinger	30 mm head of water
type 22	rigid labyrinth seal + bolt-on baffle	20 mm head of water
	floating labyrinth seal + machine seal with smooth bore	25 mm head of water
(fig. 47)	floating labyrinth seal + machine seal with inserted hemp packing	100 mm head of water
	floating labyrinth seal + machine air seal (sealing air pressure = 300 mm head of water)	250 mm head of water

9.3 Types

9.3.1 Labyrinth Seals

Labyrinth seals are non-rubbing seals which can be used at high and highest speeds (over 100 m/s circumferential speed). They consist of a fiber reinforced, highly heat resistant material "RENKplastic therm P 50" or aluminium alloy.

9.3.2 Floating Labyrinth Seals

Their freely movable arrangement in the carrier (1) or in the bearing housing make them insensitive to radial shaft displacement.

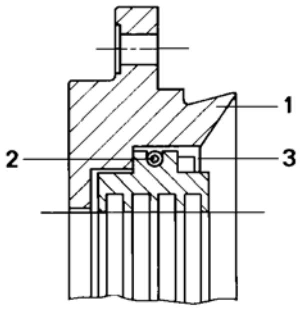


Figure 41 Floating labyrinth seal, type 10
protection IP 44

After the bearing housing has been installed and aligned on the base frame or foundation, the seal carrier can be fitted using non-hardening sealing compound (e.g. Curil T).

Afterwards, place garter spring (2), which is opened by twisting the lock, round the shaft at the sealing point and twist together again to close. The bottom half of the seal (recognisable by the oil recirculation bores), with the garter spring in position, can now be screwed into the bottom half of the carrier. When the top half of the seal and the garter spring have been fitted, turn the seal on the shaft. In this way it can be checked that there is no jamming as the result of distorsion and no subsequent overheating. Carefully eliminate any pressure points where necessary.

After this test, thinly coat the sides of the outer web (guide) of the bottom half of the seal with a nonhardening selaing compound (e.g. Curil T). When fitting the top half, make sure that the anti-rotation stopper (3) is always located in the upper locking groove.

Please take particular care when fitting the seal to ensure that it is not jammed, as this could ultimately result in its destruction.

9.3.3 Floating Oil Sealing Rings

Same as with floating labyrinth seals they are mobilely mounted in a seal carrier or in the housing.

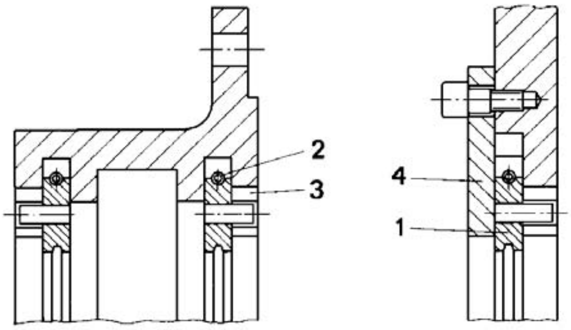


Figure 42 Floating oil sealing ring protection IP 44

a) radial installation

Radial installation (e.g. in the DUO-type seal) to be done according to the instructions given for floating labyrinth seals.

b) axial installation

The two-piece oil sealing ring (1) is placed round the shaft with the garter spring (2) in the groove at the outer diameter. Then the seal is pushed into the annular groove (3) (paying attention to the correct position of the anti-rotation pin) and is secured by the annular cover fastened with crews.

When using the seal on vertical shafts the sealing compound has not to be applied.

9.3.4 Rigid Labyrinth Seals

These seals directly mounted on the bearing housing with interposed sealing compound. When fitting, push lightly on the shaft from underneath and tighten the bolts.

Figure 43 illustrates a seal with two separate labyrinth systems. If there is only slight axial displacement, a small oil flinger can be arranged on the shaft between the two labyrinth systems.

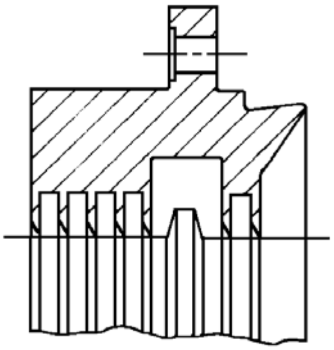


Figure 43 Rigid labyrinth seal, type 20 protection IP 44

As the shaft is „jacked“ by the oil film during operation, a rigid labyrinth seal also works without contact so that there is no wear.

However, the seals must be removed before bearings are fitted as otherwise they may be damaged by tilting.

9.3.5 Labyrinth Seal combined with Dust Flinger

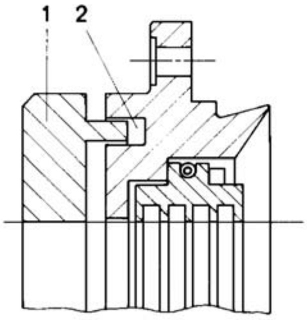


Figure 44 Labyrinth seal combined with dust flinger, type 11, protection IP 54

A flinger (1) is clamped on the shaft which engages in a groove (2) in the carrier. The purpose of this arrangement is to keep any negative pressure, which could draw oil mist out of the bearings, away from the actual seal. This also prevents dust, sand or water from penetrating into the bearing. The groove can also be filled with grease if necessary. The flinger which is clamped on must be aligned so that it does not touch the bottom of the groove in the carrier under any circumstances (consider the axial movement of the shaft).

9.3.6 Labyrinth Seal combined with Bolt-on Baffle

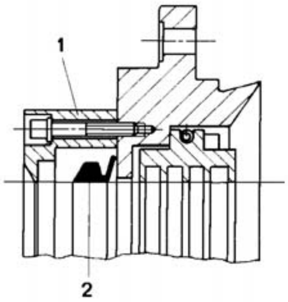


Figure 45 Floating labyrinth seal combined with bolt-on baffle, type 12, protection IP 55

The baffle is used both in cunjunction with the floating labyrinth seal (9.3.1.1) and the rigid labyrinth seal (9.3.1.3) if, for example, protection IP 55 is to be achieved. Sealing systems, grade of protection IP 56 are available in many diameter ranges, upon request.

9.3.7 Wind Back Seals

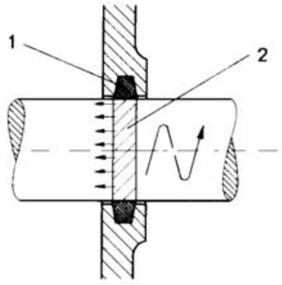


Figure 46 Wind back seal
protection IP 44

Wind back seals are graphitic synthetic tissue seals (6). The divided rings are placed in the carrier with the joint facing upwards. The spiral (7) woven into the sealing surface ensures that the oil is transported inwards when the shaft rotates.

9.3.8 Soft Packings

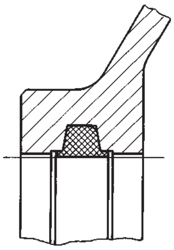


Figure 47 Soft packings
protection IP 44

In this context, soft packings are plaited sealing cords containing tallow, graphite and PTFE. These seals are ready for use. After inserting into the circular groove, the sealing cord is driven into the groove by means of a hammer of synthetic material. In connection with non-split seals (EF machine seals for example) the joints of the sealing cord must be located above the shaft. With split seals (EM machine for example), the sealing cord is driven into each half and cut to length with a sharp knife (at the joint).

Please consider our “Technical Information No. 64”.

9.3.9 Felt Seals (DIN 5419)

Felt seals are used in connection with shafts with low circumferential speeds and if the demands concerning leaktightness are not too high. Prior to installation, the halves of the seal have to be soaked in oil Please make sure that they are clean and not twisted.

9.3.10 Grease Seal

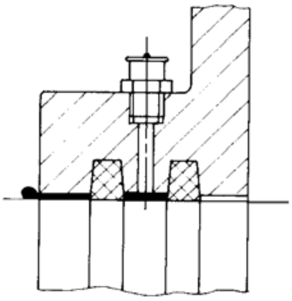


Figure 48 Grease seal
protection IP 54

These seals are used in extremely dusty or sandcharged environments.

Grease is injected through a lubricating nipple between two felt seals to form a collar around the shaft outlet which collects dirt or particles of dust and prevents the later from penetrating inside the bearing.

The grease has to be topped up according to the operating conditions. Under no circumstances must so much grease be injected that it penetrates inside the bearing and interferes with lubrication oil.

9.3.11 Flinger Seal

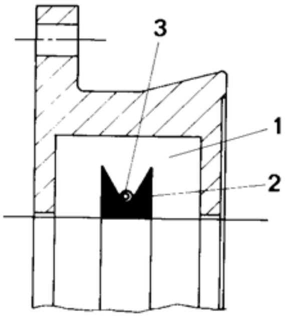


Figure 49 Flinger seal
protection IP 22

These seals are mainly used in propeller shaft bearings.

They make no contact whatsoever. An oil return chamber (1) houses a flinger (2) of an oil resistant elastomer mounted on the shaft. Oil creeping along the shaft into the chamber is deflexted by the flinger and returned to the bearing through the large drain holes.

Material for the flinger (2) is supplied in approximate length. During assembly it has to be put around the shaft and cut to length assuring its correct fit. Both ends have to be joined by an adhesive which should preferably be a cyanoacrylate adhesive, i.e. of quick-setting type.

The flinger is kept on the shaft by a garter spring (3) with twist lock.

When fixing the seal, the mounting flanges have to be sealed with non-hardening sealing compound (e.g. Curil T).

The length of the chambers was chosen to permit normal shaft extension. The flinger (2) has to be positioned in such a way that enough space is left between either side of the flinger and the walls of the chamber (1) (in order to prevent any contact) in case of axial shaft movement.

10. Mounting of Slide Bearings on Steel Structures (frames, motor shields, brackets)

10.1 Construction

The construction of steelwork supporting the slide bearing greatly influences the operational safety of the bearings as well as the vibration stability of the machine. The steelwork should be of nearly the same stiffness as the bearing housing. The errors mentioned hereunder lead to devitations from the rule:

a) Insufficient sheetmetal thickness (or wall thickness of the rolled steel). In the past, the plates used underneath pedestal bearings for instance, only had a thickness of 20 % of the bearing foot (see fig. 50).

The stiffness of such a structure is very low as the thin sheets bulge specially in the area of the fixing bolts of the bearing foot. This reduction of the total stiffness for the bearing system is lowering the critical speeds considerably. Thrust loads lead to heave tilting of the bearing housing and thus affect operational safety. Improvements can be achieved by simply chosing thicker plates.

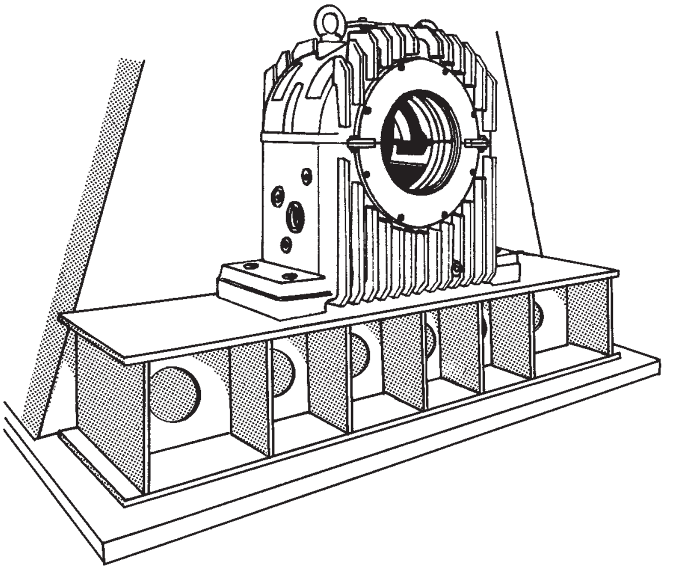


Figure 50

b) Support of slide bearings on thin high intermediate stands (possibly on concrete stands). The horizontal stiffness fo this type of intermediate stands is also very low, if not of truncated pyramid shape, i.e. base plates much larger than the bearing pedestal. Too small supports reduce stability and operational safety (see fig. 51).

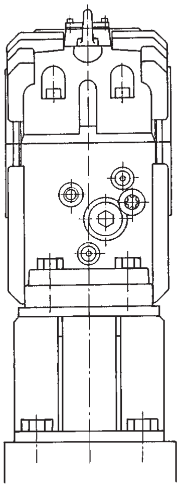


Figure 51

c) Fixation of slide bearings by means of anchor bolts; (Foundation bolts) reaching through a grouted-in intermediate plate (same size as bearing pedestal) or base plate into the concrete foundations. The module of elasticity of concrete compared to that of steel is very small such support may be of undue weakness. This can be compensated by intermediate plates or brackets larger than the bearing pedestal and by a greater number of bolts used for the fixation in the concrete foundation (see fig. 52).

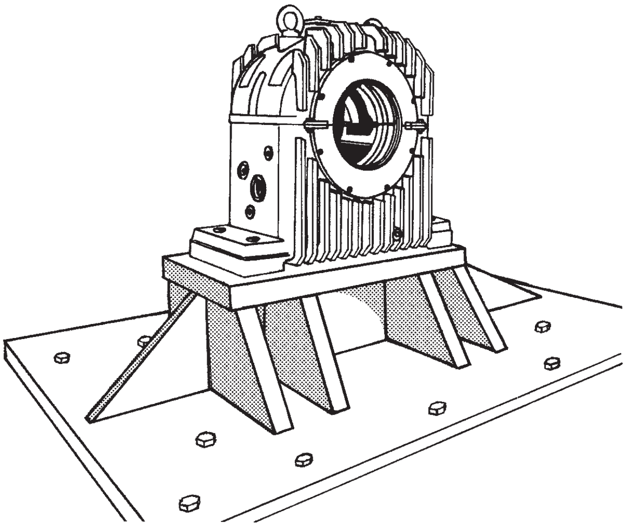


Figure 52

It may possibly happen that vibration excitation causes a steel structure to resonate. This can be corrected by changing the natural vibratons, by for instance arranging struts, gusset plates and other reinforcements.

10.2 Mounting of Insulated Bearings

Slide bearings on electrical machines should be insulated in order to prevent bearing currents. Under some circumstances, these bearing currents can result in the destruction of the whitemetal or cause the ignition of combustible gases and vapours through sparking.

Normally, it is only necessary to insulate **one** bearing.

If two bearings are insulated, one of them has to be equipped with an insulation checking device (please consider our “Technical Information No. 65).

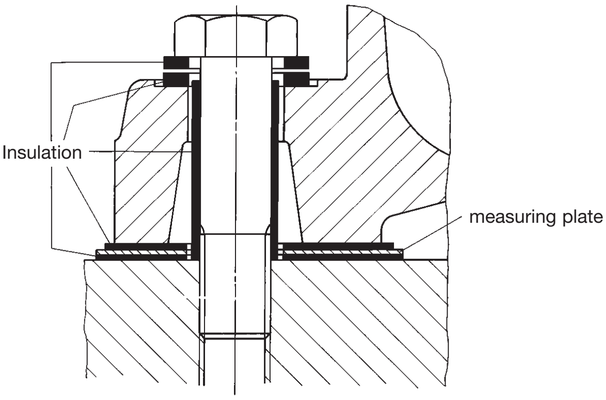


Figure 53 Foundation bolt, insulated for pedestal bearing

Generally, a pedestal bearing is insulated by means of insulating washers or insulating bushes (fig. 53). The graduated arrangement of the insulating washers makes it easier to keep clean and lengthens the creepage path.

During assembly, care must be taken to ensure that the insulation is not bridged by other parts. This applies in particular to pipes for circulating oil lubrication, metal conduits for electric cables, thermometers, etc. It must also be ensured that no bridging can take place during operation.

Flange-mounted bearings can be secured on the machine plate with the aid of an insulating intermediate flange. Care must also be taken here to ensure that the insulation is not bridged by the screws or pipes.

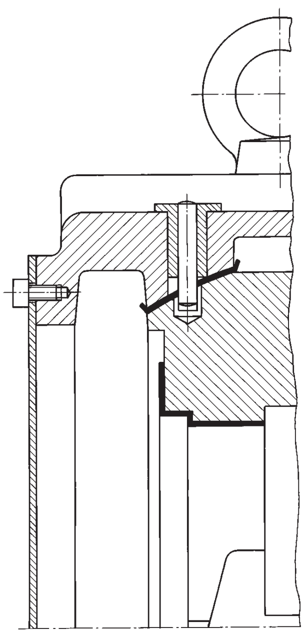


Figure 54 Insulation of the spherical seating

Upon request insulated E bearings can be supplied. In this case the spherical seating which accomodates the bearing shell is covered with a non-conductive plastic foil (fig. 54). The floating shaft seal consist of non-conductive material (see chapter 9.3.2).

Insulating screw connections are available for thermometers. With this feature it is no longer necessary to insulate pipes, etc.

11. Assembly and Initial Starting

The most important principle applying to any assembly work in connection with slide bearings is cleanliness!

Despite the attention given to careful packing, impurities cannot be prevented from entering the bearing during transportation to site. Therefore it is essential that all slide bearings should be thoroughly cleaned before mounting. Never use cotton or fibre rags because any threads left in could enter the oil circuit causing overheating and in extreme cases leading to destruction of the bearing.

Due to the manufacturing accuracy of RENK Slide Bearings spot-grinding or scraping of the working surface is no longer required. The former practise of scraping would destroy the bearing geometry which was calculated for safe operation.

Only minor corrections of the working surfaces are admissible (see chapter 11.8 „Checks after initial starting“).

The Special Instructions for Assembly, Operation and Maintenance issued for the individual types of bearing must be observed.

The check lists in which all important points are listed are a great help and must be checked by the fitter.

If for any reason the special instructions and check lists are not available contact RENK Hannover by letter or telephone.

Here we can give only general rules.

11.1 Installing the Shaft in the Bearing

The shaft must never be installed dry. According to the existing running-in conditions the following measures are recommended:

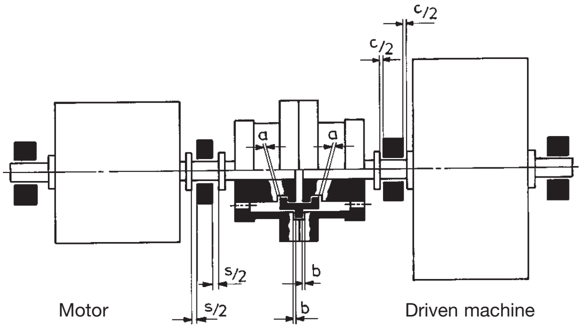
The working surfaces of the shaft must be coated with an oil film.

If there are to be long running-in periods and/or hard operating conditions we recommend a mixture of lubricating oil and molybdenum-disulphide (MoS_2) or treatment of the shaft with molybdenum-disulphide to the instructions of the suppliers.

11.2 Alignment of the Bearings

For aligment the requirements relating to the particular plant or machine are the first consideration. The spherically seated or tilting shells of our bearings can compensate assembly inaccuracies within limits but it is still advisable to align the bearing housing to the shaft in such a way that there is an even spacing to the end bore.

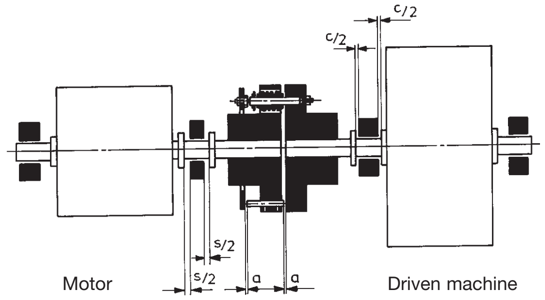
Gear coupling with limited end float



$s = 2 (a + b) + c + k$ [mm]

$a + b$ = normal to 2 mm

ELCO coupling with limited end float
Plate to limit the end float



$s = 2 a + c + k$ [mm]

s = end float iof motor bearing at drive end

s/2 = up to maximum of 6 mm possible (enquiry necessary)

a, b = maximum possible end float of coupling

c = maximum possible end float of locating bearing of driven machine

k ≥ 1 mm = safety margin
This safety margin should include elongation of the shaft between the locating bearing of the motor and the locating bearing of the machine or gearbox through heating to operating temperature.

Figure 55 Axial alignment of coupled machines

If couplings with limited end float are provided between a motor with locating bearing with considerable end float and a machine, then the axial alignment must be carried out very carefully (fig. 55).

The safety margin „k“ of at least 1 mm is to be provided in order to compensate for elongation of the shaft between the locating bearing of the motor and the locating bearing of the machine or gearbox through heating to operating temperature.

In this case one locating bearing acts as a guide for the whole shaft. It should be taken into consideration that thrust loads will be transmitted to this locating bearing via the coupling with limited end float.

11.3 Seals

Check end seals for quality before fitting. For details of fitting the different seals see chapter 9 „Shaft Seals“.

11.4 Lubrication Rings

Lubrication rings are not secondary but vital machine parts which are carefully manufactured and require care in handling. They must be protected against damage and deformation. The assembly joint screws of loose oil rings are to be firmly tightened using tools which fit exactly.

11.5 Oil Filling

Fill with an oil of the prescribed viscosity (see chapter 5 „Lubrication“). Follow the instructions with regard to oil level for each type of bearing.

11.6 Oil Supply System

Before setting the system into initial operation a functioning test of the oil supply must be carried out.

This to include:

Check of the pumps (flow and pressure); simulation of the failure of one pump for testing the necessary switch-over to the stand-by pump.

Check of all monitors and meters; function testing of electrical interlocking.

Check of the oil quantities fed to the bearing. (Compare with the Operating Instructions).

For additional information see chapter 7.5 „External Oil Supply“.

11.7 Checks before Starting

11.7.1 Bearings with Self-Contained Lubrication

Before starting check that the bearing is filled with oil according to instructions and that the oil-level is right. Check that lubrication ring and (where applicable) oil scraper have been incorporated.

Re-tighten bearing base bolts and bearing cover bolts.

11.7.2 Bearing with Circulating Oil Lubrication and/or Hydrostatic Jacking Device

Switch on oil supply system. Check to see that the bearings are properly supplied with oil.

11.7.3 Starting

After the initial start the installation should be continuously checked during several hours of operation.

Checks to include:

a) Bearing Temperature
This should rise evenly up to the settled permanent operating temperature. If the temperature does not stabilise below the bearing temperatures stated in chapter 6, the installation must be stopped and the cause be found.

If RENK Werk Hannover have stated operating temperatures and these cannot be reached (cold bearing), investigate for the reasons. A possible cause for instance may be a defective thermometer.

b) Oil-Tightness
The bearings should not loose oil at end bores, assembly joints, oil-level gauges, plugs etc. See that unused connection holes are closed – the plastic stoppers in these are not oil-tight! In bearings with self-contained lubrication, any oil loss can soon lead to destruction of the working surfaces.

c) Stability
Stability depends on many influences. If the bearing does not run quietly during initial starting it must be stopped immediately and the cause be found. Unbalance above a certain limit is admissible only, if already stated in the planning stage and duly considered in the design of the bearing.

11.8 Checks after Initial Starting

It is advisable to remove the bearing shells after a running period of 5 to 10 hours (test run at the manufacturer) for an inspection of the working surfaces for signs of damage (edge loading, striae, pressure marks). Remove these spots by careful scraping, taking care to remove only so much white metal that the geometry of the surfaces is not destroyed.

11.9 Oil Change Schedule (for mineral oils)

For self-lubricated bearings cleaning intervals with oil changes of approx. 8 000 operating hours are recommended (approx. 20 000 operating hours for bearings with oil circulating systems).

Shorter intervals may be necessary in case of frequent-start-ups, high oil temperatures or excessively high contamination due to external influences.

12. Corrosion and Transport Protection of Slide Bearings

12.1 Corrosion protection

Prior to despatch RENK applies corrosion protection to slide bearings.

The grade of protection depends on the duration and the location of storage. Approximate values may be taken from the table 9.

Location of storage	Duration	Grade of protection
indoors	18 months	1
	24 months	2
	48 months	3
in a shed	6 months	1
	12 months	2
	36 months	3
outdoor	6 months	2
	12 months	3

Table 9

The corrosion inhibitors used are the following:

a) grade 1:

Tectyl 511 M, non-hardening protective film.

b) grade 2:

Tectyl 846, dry film thickness approx. 0,04 bis 0,06 mm.

c) grade 3:

Tectyl 164, dry film thickness approx. 0,10 bis 0,15 mm.

If not otherwise stated in purchase orders, slide bearings will be supplied with grade of protection 1.

Though Tectyl 511 M is compatible with lubricants, it is recommended, however, to remove it prior to installation. Dirt particles may settle in the protective film, even if utmost care is taken during packing and transportation which would possibly affect the operational safety.

In case of grades of protection 2 and 3 Tectyl 511 M is applied to the spherical seating in the housing. All other surfaces susceptible to corrosion are protected with Tectyl 846 or Tectyl 164.

For the protection of machines with bearings and shafts in place, we recommend the following measures:

- apply or spray shells with Tectyl 511 M prior to installation
- apply or spray shaft in way of the bearings with Tectyl 511 M
- seal the bearing

  seal tapped holes with plugs, connecting flanges with blank flanges and gap between seal and shaft or between shaft and housing respectively with self-adhesive tape
- remove plexiglass cover or top sight glass and spray corrosion inhibitor which needs not to be removed (e.g. Tectyl 511 M or Valvoline) through the opening in the housing using a compressed air spray gun.

  Airless injectors are not suitable since they do not ensure adequate atomisation of the corrosion inhibitor.
- fit a bag with dessiccant (e.g. silica gel) below the plexiglass cover or the sight glass in order to prevent condensation
- seal the bearing again

If a period of six months is exceeded calculated from the time of conservation to the time of commissioning, then conservation must be repeated and a new bag of dessiccant needs to be introduced.

If it is known that a unit is to be stored for several years before being commissioned, dismantling of the bearing shells and conservation of individual components may be advisable.

12.2 Transport Protection

When transporting fully assembled machinery, care should be taken to prevent movement between shafts and bearing shells. As an example we quote electrical machinery here. If other machines are involved, a similar procedure should be adopted.

12.2.1 In the case of a machine with flanged bearings (fig. 56) the shaft is axially pressed against the locating bearings so that it abuts against it, using a clamping strap (or a cap) and a compression screw.

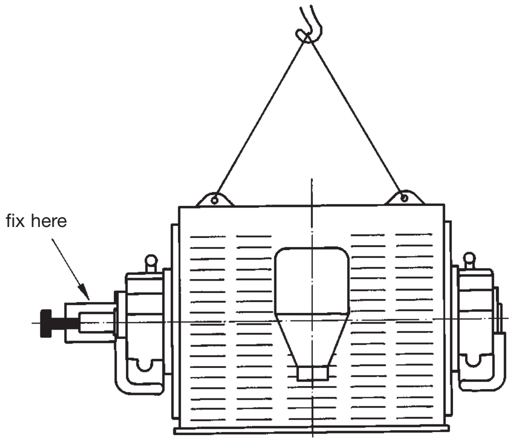


Figure 56 Transport of a machine with flanged bearings

12.2.2 In the case of a machine with pedestal bearings (fig. 57) the rotor is pulled against the sub-frame using (for example) timbers placed across the shaft and pressed firmly into the bearings in this way.

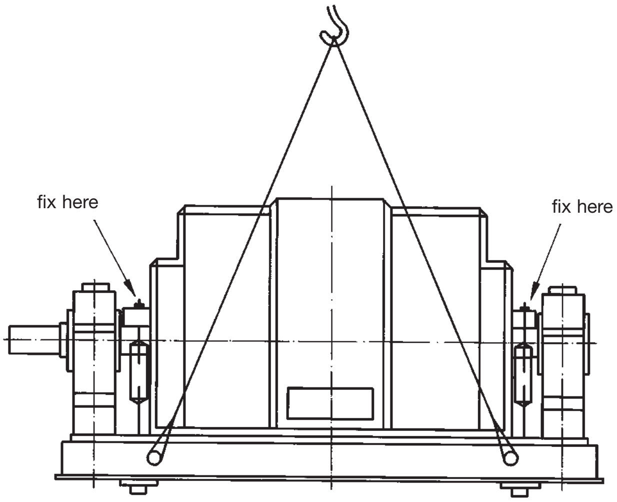


Figure 57 Transport of a machine with pedestal bearings

12.2.3 In the case of vertical machines with bearings with journal tilting pads (fig. 58), the radial pads firmly tightened onto the shaft. If no radial pads are provided, then proceed in accordance with 12.2.1 und 12.2.2 above.

Before transport, the working surfaces of the bearings must be well lubricated and a corrosion inhibitor should be applied in accordance with chapter 12.1.

Solid lubricants such as for example MoS_2 are not recommended, since under certain circumstances – despite the safety measures adopted – micro movements might lead to damage of the working surfaces which have a high grade finish.

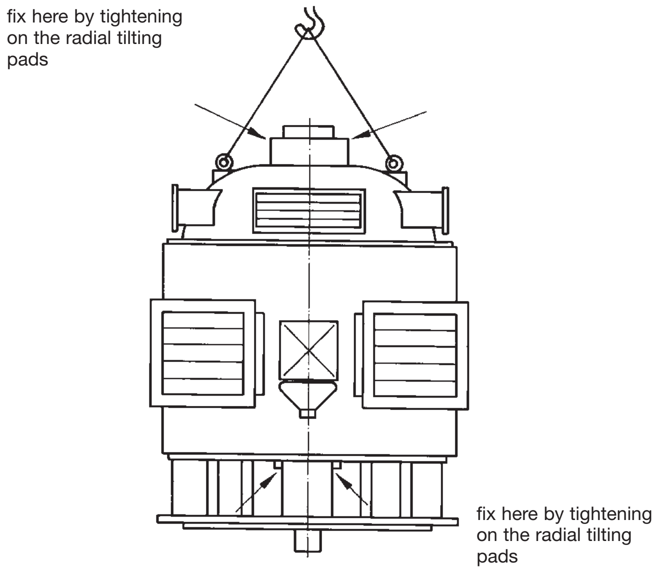


Figure 58 Transport of a vertical machine

For the transport of machinery the eye bolts fixed to the bearings must not be used. These are exclusively provided for the installation and dismantling of the bearings.

These recommendations apply to normal transport conditions. During extended transport overland (e.g. in developing countries), additional measures might be required. These might include the removal and separate packing of oil rings and, in the case of horizontal machines, the axial and radial fixing of the rotor.

How is it possible to check whether the above instructions have also been observed by the assembly personnel and by fitters who may have been brought in from outside the company?

All the important points which have to be checked by the fitter are indicated on the check list (pages 35 bis 38).

Signatures – not only of the fitter – provide some confirmation that the assembly work has been properly carried out.

On the other hand, a check list which has been properly filled in gives considerable assistance in tracing faults when damage occurs.

These check lists are available to our customers in German/English and German/French and are normally included in our consignments.

Additional lists are available at any time on request.

CHECKLISTE

für Gleitlager-Montagen

CHECK LIST

for the Installation of Slide Bearings

Auftr.-Nr. / Job-No. ☐☐☐☐☐☐

Besteller / Customer:

Montage bei Firma: / Installation effected at:

Montiert von Firma: /Installation effected by:

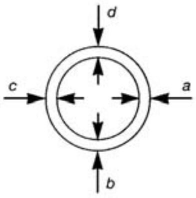
Name des verantwortlichen Monteurs: / Name of responsible fitter:

Lagertype (Bauart, Größe)
Type and size of bearing:

Pos. / Item ☐☐ ☐☐

1	**Kontrollen vor dem Einbau**	**Checks prior to installation**		
1.1	Ist die Lieferung vollständig	Completeness of the supply?	○	○
1.2	Ist die Lieferung ohne erkennbare Schäden? (Wenn nein, 1.2.1 bis 1.2.4 beantworten, gegebenenfalls Nachricht an RENK Hannover).	Is the supply free of visible faults? (In the negative, please answer points 1.2.1 to 1.2.4; if necessary, report to RENK Hannover).	○	○
1.2.1	Transportschaden	Defect caused during transport	○	○
1.2.2	Korrosionsschaden	Defect due to corrosion	○	○
1.2.3	Schaden durch Lagerung an Baustelle	Defect caused during storage on the site	○	○
1.2.4	Beschädigung während der Montage	Defect caused during installation	○	○
1.3	Lager sind mit Waschbenzin o.ä. gesäubert (Fasernde Lappen und Putzwolle wurden nicht verwendet).	Bearings have been cleaned with cleaning petrol (or the like) (fuzzy rags or cotton waste have not been used)	○	○
1.4	Prüfung der Wellen und Bunde	Checking of shafts and collars		
1.4.1	Keine Beschädigungen und fühlbare Riefen	No defects nor palpable striae		
	Welle	shaft	○	○
	Bunde	collar	○	○
1.4.2	Wellen-∅ im Bereich der Lagerschale (Ist-Maße)	Shaft diameter in way of the bearing shell (actual size)		
	von / from ☐☐☐☐ , ☐☐	bis / to ☐☐☐☐ , ☐☐ mm	○	
	von / from ☐☐☐☐ , ☐☐	bis / to ☐☐☐☐ , ☐☐ mm		○
1.4.3	Bundabstand	Spacing of collars	☐☐☐	☐☐ mm
1.4.4	Ebenheit der Bunde mit Haarlineal geprüft: In Ordnung	Flatness of collars (checked by means of a hair line ruler): all right	○	○
1.4.5	Rechtwinkligkeit der Bunde zur Welle mit Haarwinkel geprüft: In Ordnung	Perpendicularity of collars to shaft (checked by means of a hair line try square): all right	○	○
1.4.6	Die Welle ist im Bereich der Dichtungen frei von Riefen (auch Schleifriefen) mit erkennbarem Längenvorschub	The shaft is free of striae (also grinding striae) with perceptible longitudinal feed, in way of the seals	○	○
2	**Montage der Lager**	**Installation of the bearings**		
2.1	Die Lauffläche des Lagers ist vor dem Einlegen der Welle durch Einreiben, Einstreichen oder Einsprühen überzogen mit:	Prior to installing the shaft the working surface of the bearing has been coated or sprayed with:		
	Schmierstoff	lube-oil	○	○
	Schmierstoff-Graphit-Gemisch	oil-graphite mixture	○	○
	MoS$_2$-Schmierstoff-Gemisch	MoS$_2$-oil mixture	○	○

2.2 Ausrichten des Lagerkörpers zur Welle.
Spalt zwischen Welle und Gehäusebohrung
messen.



Alignment of bearing housing and shaft.
Control of the gap between shaft and bore of housing.

	Pos. Item	☐	☐
linke Seite on the left	a	☐,☐	a ☐,☐
	b	☐,☐	b ☐,☐
	c	☐,☐	c ☐,☐
	d	☐,☐	d ☐,☐
rechte Seite on the right	a	☐,☐	a ☐,☐
	b	☐,☐	b ☐,☐
	c	☐,☐	c ☐,☐
	d	☐,☐	d ☐,☐

2.3	Seitendichtungen sind gemäß Anweisung eingebaut	Installation of end seals according to instruction	○	○
2.4	Schmierstoffabstreifer bei Schmierung mit Festschmierring gemäß Anweisung eingebaut.	Installation of the oil scraper according to instruction (in case of lubrication with fixed oil ring	○	○
2.5	Schmierstofffüllung: (Nicht ausfüllen bei externer Schmierstoffversorgung)	Oil filling (not applicable for circulating oil lubrication)		
	Viskosität / Viscosity: ISO VG	bzw. / or ISO VG		
	Bezeichnung / Dennomination:	Fabrikat / Make:		
	Von RENK Hannover vorgeschriebene Viskosität: Viscosity prescribed by RENK Hannover:			
2.8	Nur bei wassergekühlten Lagern:	To be stated for water cooled bearings only:		
	Wasserkühlung angeschlossen	Water cooling has been connected	○	○
	Druck vor dem Lager (bar)	Pressure in front of the bearing (bar)		☐,☐
	Durchflussmenge (l/min)	Rate of flow (l/min)	☐	☐
	Wassereintrittstemperatur (°C)	Temperature at cooling water inlet (°C)	☐	☐
	Wasseraustrittstemperatur (°C)	Temperature at cooling water outlet (°C)	☐	☐

3 Inbetriebnahme

Putting into operation

3.1	Drehzahl (U/min) gemessen / nach Angabe	Speed (R. P. M.) measured / to indication ☐	○/○	○/○
3.2	Lagertemperatur (Beharrung) (°C)	Bearing temperature (during operation) (°C)	☐	☐
	gemessen in Schalenunterteil	taken in bottom half of the shell	○	○
	gemessen in Schalenoberteil	taken in top half of the shell	○	○
	gemessen im Schmierstoffsumpf	taken in oil sump	○	○
	wenn keine Thermometer eingebaut sind: am Gehäuse außen gemessen / geschätzt	if no thermometer is provided: taken on outside of housing / estimated	○/○	○/○
3.3	Umgebungstemperatur des Lagers gemessen / geschätzt	Ambient temperature of bearing measured / estimated ☐ °C ☐ °C	○/○	○/○
3.4	Lager sind schmierstoffdicht (wenn nicht gesonderten Bericht mit Angaben wo Schmierstoffaustritt ist und welche Mengen)	Bearings are oil tight (if not, report separately and state place and amount of leakage)	○	○
3.5	Die Anlage läuft ruhig. (wenn nicht, Ursache angeben)	The installation is operating without vibration. (if not, state reason)	○	
	Für Anlagen mit externer Schmierstoffversorgung Blatt 2 ausfüllen.	If plants with circulating oil lubrication are concerned, please fill in sheet 2.		

Montageort / Place of fitting .... Datum / Date ....

Unterschriften / Signatures

Monteur / Fitter ....

Montageleiter / Responsible on the site ....

Firma / Name of Company ....

Endabnehmer / Utilizer ....

Firma / Name of Company ....

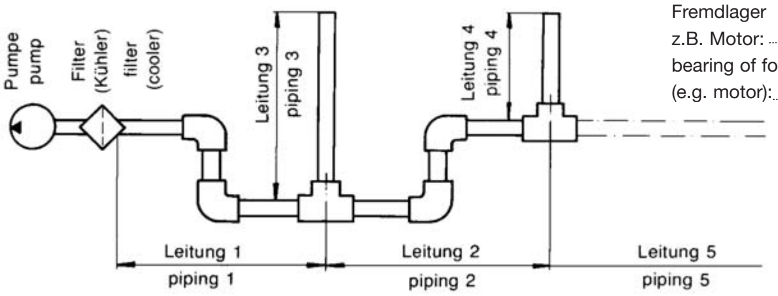
CHECK LISTE

CHECK LIST

Schmierstoffanlage

Oil circulation system

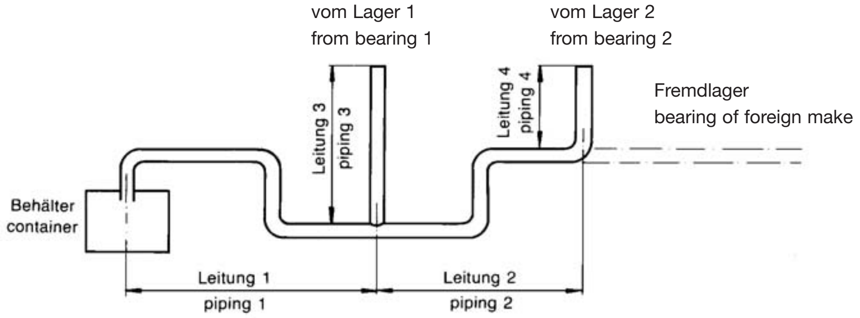
4.	**Montage der Schmierstoffanlage**	**Fitting of system**	
4.1	Der Schmierstoffbehälter ist gesäubert.	The oil container has been cleaned.	○
4.2	Druckleitungen verschraubt (Fittings, Schneidringverschraubung usw.)	Pressure pipes have been tightened up (fittings, taper-bush type pipe union, etc.)	○
	Druckleitungen verschweißt.	Pressure pipes have been welded.	○
	Lager 1: .... Liter/min	bearing 1: .... litres/min	
	Lager 2: .... l/min	bearing 2: .... l/min	



Fremdlager
z.B. Motor: .... Liter/min
bearing of foreign make
(e.g. motor):.. l/min

Leitung Piping	Innen-∅ Inside diameter	gestreckte Länge stretched length	Anzahl der Winkelverschrbg. Number of angle pipe unions	Rohrbogen bends	Höhendifferenz Anfang - Ende Difference of level start - end
1					
2					
3					
4					
5					

4.3	Rücklaufleitungen verschraubt (Fittings o.ä.)	Return pipes have been tightened up (fittings, etc.)	○
	Rücklaufleitungen verschweißt	Return pipes have been welded	○



Leitung Piping	Innen-∅ Inside diameter	gestreckte Länge stretched length	Anzahl der Winkelverschrbg. Number of angle pipe unions	Rohrbogen bends	Höhendifferenz Anfang - Ende Difference of level start - end
1					
2					
3					
4					

In Strömungsrichtung steigen = +
In Strömungsrichtung fallend = –

Direction of flow ascending = +
Direction of flow descending = –

4.4	Die Rohrleitungen sind vor dem Einbau gesäubert (geschweißte Leitungen gebeizt).	The pipings have been cleaned prior to their installation (welded pipings have been pickled).	○
4.5	Das gesamte Rohrleitungssystem is mindestens 1 Stunde mit Petroleum gespült. Lager war dabei nicht im Spülkreislauf, Instrumente waren nicht angebaut.	The piping system has been scavenged with mineral oil for at least one hour. The bearings were not in the scavenging circuit, instruments were not fitted.	○
4.6	Nach dem Spülen wurden die Filter gesäubert.	After scavenging, the filters have been cleaned.	○
4.7	Behälterheizung ist vorhanden und angeschlossen.	Container heating, if existing, has been connected.	○
4.8	Wasserkühlung ist vorhanden und angeschlossen.	Water cooling, if existing, has been connected.	○
4.9	Überwachungsinstrumente sind angeschlossen.	Monitoring instruments have been connected.	○
	Welche: Druckwächter	Pressure switch	○
	Kontaktmanometer	Contact pressure gauge	○
	Durchflusswächter	Flow rate indicator with warning switch	○
4.10	Schmierstofffüllung des Behälters:	Oil charge container:	
	Menge ………… Liter	Quantity ………… litres	
	Viskosität: ISO VG ………… bzw. …………	Viscosity: ISO VG ………… bzw. …………	
	Bezeichnung: …………	Denomination: …………	
	Fabrikat: …………	Make: …………	
5.	**Inbetriebnahme**	**Putting into operation**	
5.1	Funktionsprüfung der Pumpen bei stehender Anlage	Functioning test of the pumps when machine stationary	○
5.2	Betriebswerte:	Operating data:	

5.2.1

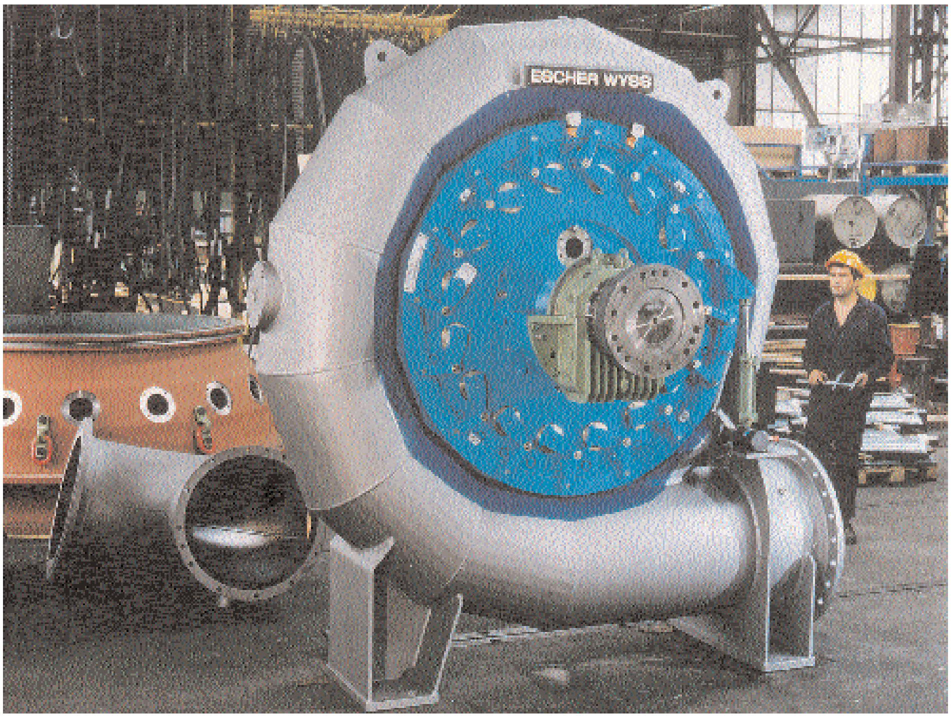
Schmierstoff Oil		Lager 1 Festlager bearing 1 locating bearing	Lager 2 Loslager bearing 2 non-locating bearing
Schmierstoffmenge Rate of flow	(l/min) (l/min)		
Schmierstoffdruck Oil pressure	(bar) (bar)		
Schmierstoffeintrittstemperatur Oil inlet temperature	(°C) (°C)		

5.2.2	Kühlwasser	Cooling water
	Druck vor dem Kühler: ………… bar	Pressure in front of the cooler: ………… bar
	Wassermenge: ………… l/min	Rate of water flow: ………… l/min
	Eintrittstemperatur: ………… °C	Water inlet temperature: ………… °C
	Austrittstemperatur: ………… °C	Water outlet temperature: ………… °C

Bemerkungen:	Remarks:

Die Angaben auf Blatt 2 werden durch die Unterschrift auf Blatt 1 bestätigt.
The statements made on this Check List are confirmed by the signatures on Check List for the installation of Slide Bearings.

Example of application

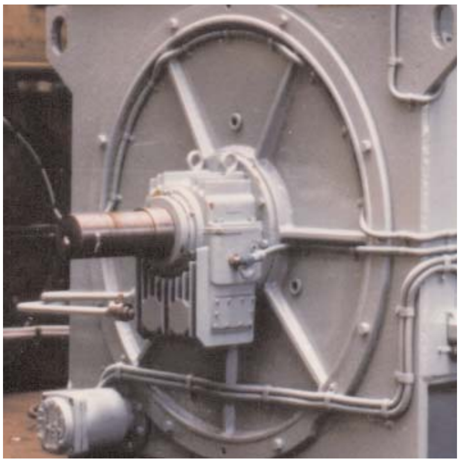


Horizontal spirally shaped Francis turbine for water power-station with RENK slide bearing EMZLA 18-180

(Photo: Sulzer Escher Wyss, Ravensburg, Zürich)

RENK E-type bearing
in a stand-by power generating set

(Photo: Jeumont-Schneider, Jeumont)

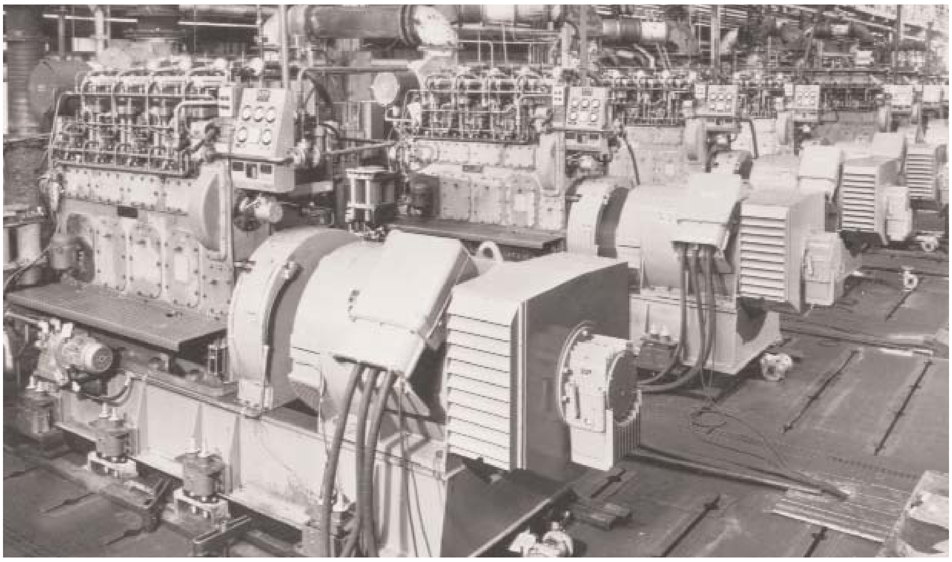


Asynchronous motor equipped with RENK slide bearing EFWLB 11-110

(Photo: AEG-Telefunken, Berlin)

Group Diesel engine/generator with RENK E-type bearings EF 18

(Photo: A. van Kaick, Ingolstadt)



Product Range

Plain bearings

Type E	for electrical machinery, fans, compressors, turbines	Catalogue no.	RH-1009
Type ZM	for electrical machinery	Catalogue no.	RH-1175
Type SC	Radial bearing, e.g. mounted to Diesel generators and rolling mill drives	Catalogue no.	RH-1149
Type SN	aftermost bearings and intermediate bearings for shipbuilding applications	Catalogue no.	RH-1202
Type HG	for hydrogenerators and electrical motors	Catalogue no.	RH-1189
Type WG	fabricated type for rolling mill drives	Catalogue no.	RH-1155
Type DN	marine thrust blocks	Catalogue no.	RH-1073
Type I	for tube mill (e.g. cement manufacture) transmission units	Catalogue no.	RH-1120
Type M	for general mechanical engineering applications	Catalogue no.	RH-1065
Type ROTRIX	trunnion bearing	Catalogue no.	RH-1089
Type SH	trunnion bearings tube mills	Catalogue no.	RH-1147

Vertical slide bearings and vertical bearing inserts

Type VT and VG	as complete thrust and guide bearings and guide bearings only	Catalogue no.	RH-1153
Type EV	vertical bearing inserts for electrical machinery, fans and pumps	Catalogue no.	RH-1021
Type G	Plain bearing shells	Catalogue no.	RH-1102
RD Thrust Bearing		Catalogue no.	RH-1025
RS Thrust pads		Catalogue no.	RH-1198

Special bearings for all branches of industry also to customer’s drawings.

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ELCO	flexible compression sleeve coupling	Catalogue no.	RH-1008
ELBI	flexible coupling for general mechanical engineering applications	Catalogue no.	RH- 076
AERO	pneumatically shifted friction clutch	Catalogue no.	RH-1118
	Centrifugal clutch	Catalogue no.	RH-1014
	Overrunning clutch	Catalogue no.	RH-1013
	Diaphragm coupling	Catalogue no.	RH-1224
	Special couplings and clutches		

Computer calculations available for plain bearings and couplings.

Sales Organisation

Headquarters and Manufacturing Plant

RENK AKTIENGESELLSCHAFT
Werk Hannover
Weltausstellungsallee 21
D-30539 Hannover
Phone: +49 (5 11) 86 01-0
Fax: +49 (5 11) 86 01-288
Email: info.hannover@renk.biz
Internet: www.renk.biz

Manufacturing Plant

OMEGA RENK BEARINGS PVT. LTD.
Anand Nagar, Raisen Road
Bhopal - 462 021
India
Phone: (91 -7 55) 5 28 45 61
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Assembly and Distribution Centers with Sales and Engineering Support

RENK Corporation
304, Tucapau Road
29334 Duncan S.C.
USA
Phone: (1-8 64) 4 33 00 69
Fax: (1-8 64) 4 33 06 36

COFICAL RENK MANCAIS DO BRASIL LTDA.
Rodovia BR-280,
km 54 - Em frente ao Parque Municipal de Exposições
CEP 89270-000 - Guaramirim - SC - Brasil
Phone: (55-4) 737 36 400
Fax: (55-4) 737 36 499

MAN B&W Diesel

MAN B&W (Japan) Ltd.
RH Division;
Fuji Building (Room 121)
3-2-3 Marunouchi
Chiyoda-ku, Tokyo 100-0005
Japan
Phone: (81-3) 3215 1310
Fax: (81-3) 3284 0867

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