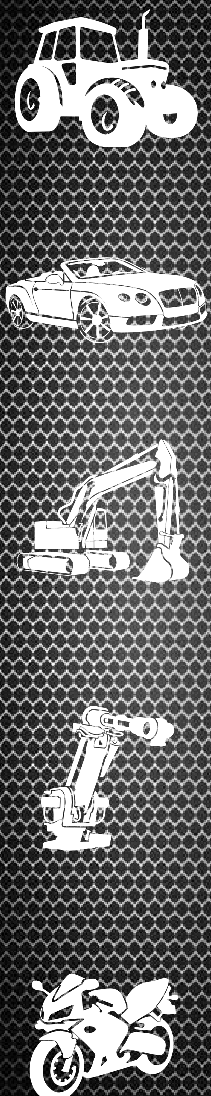
DUNLOP™

BEARINGS

Ball, Tapered and Roller Bearings



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8. Bearing applications

8.1 Arrangement of bearings

To locate rotary shaft you need at least two bearings that are located in certain distance from each other. Depending on the application method, location with axially free and axially guiding bearing is selected; prestressed location or floating arrangement of bearings. See figure 4.12 in chapter Bearing type selection for examples of bearing arrangements.

8.1.1 Location with axially free and axially guiding bearing

Axially guiding bearing on one shaft end brings besides radial load element also axial element in both direc-tions. For the above reason, it has to be secured both in the shaft and in the body. Axially free bearing in location compensates production inaccuracies in location and, first of all, changes in dimensions in operation due to increased temperatures. An ideal axially free bearing is roller bearing in N and NU design the rolling bodies of which can move on the raceway of bearing ring without guide flanges. Bearings of the other types, such as ball bearings and spherical-roller bearings, can be used as axially free only if one of bearing races is push-located.

Axially guide bearing guides shaft in axial direction and besides radial forces captures also axial forces. Selection of bearing type to be used as axially guide bearing depends on the size of axial load and on requi-rements for accuracy of shaft location. Double row angular-contact ball bearing ensures more accurate axial guidance than e.g. ball or spherical-roller bearing. Accurate axial guidance can be achieved also by a pair of tapered roller bearings which are used as axially guide bearing. At lower axial load even NUP cylindrical roller bearing can be used as axially guide bearing.

8.1.2 Symmetrical arrangement of bearings

This type of location suits mainly short shafts. It features shaft being guided in one direction by one bearing and in other direction by other bearing. Suitable bearings for this type of arrangement are all radial bearings that allow transfer of axial force at least in one direction. In this arrangement, prestressed bearings can be mounted (fig. 8.1).

8.1.3 Prestressed location

Location of prestressed bearing usually consists of symmetrically placed ball bearings with angular contact, or of tapered roller bearings. Prestress is achieved by use of springs. Such design compensates thermal dila-tation. It is used in case when idle bearings can be exposed to vibrations. Prestressed bearings can reduce noise level, especially in small electric motors.

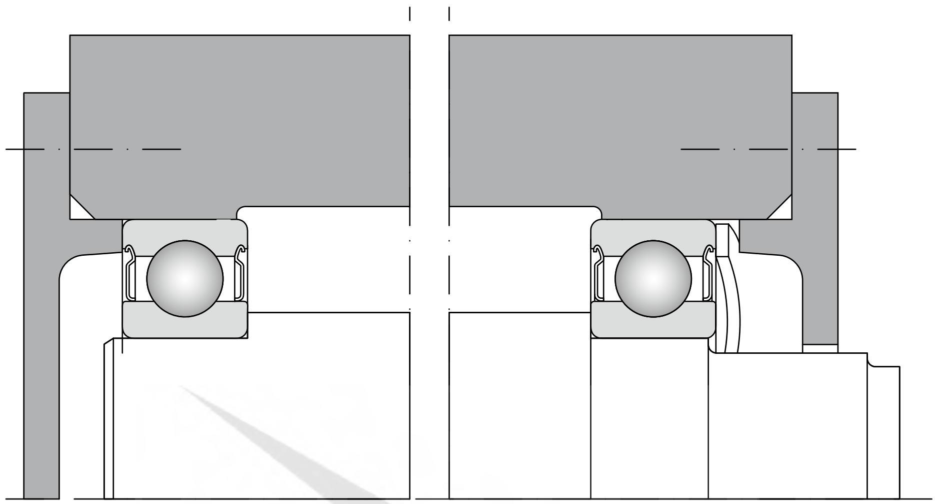

Fig. 8.1

Spring acts on outer race of one of the bearings whilst relevant outer race has to allow axial displacement in the body. Prestress remains practically constant even though the bearing axially moves due to thermal dilatation. Required prestress can be calculated using the below relation:

$$F = k \cdot d$$

F . . . . Prestress force (kN)

k . . . . coefficient, see next

d . . . . bearing hole diameter [mm]

Depending on design of electric motor, the coefficient may reach values of 0.005 up to 0.01.

If prestress is supposed to prevent bearing from getting damaged due to vibrations, it has to be set to higher level.

Then k = 0.02 has to be selected.

This method is however not suitable for locations that must feature high rigidity where the direction of acting load changes, or where shock load acts.

If certain optimum prestress value is exceeded, rigidity increases only insignificantly whilst friction and also service temperature in the bearing grow rapidly. This reduces durability of bearing since additional constant load acts on it. Informative relation between durability and prestress – clearance – is indicated in diagram in fig. 8.2.

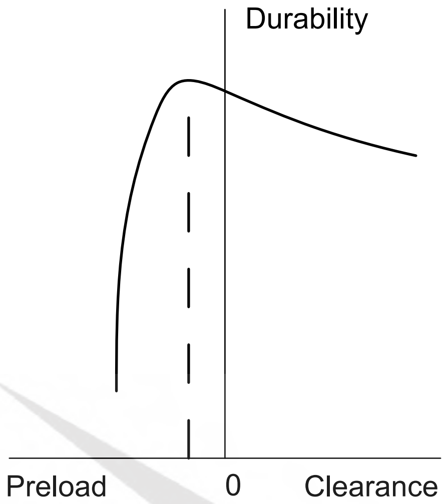

Fig. 8.2

8.2 Location design – General principles

Properties of bearings are fully utilised only when bearing races are supported along the entire circumference and width of raceways. Solid support surface can be of either cylindrical or tapered shape, in thrust bearings the surface is flat. Support surfaces must be manufactured to have adequate accuracy, and must not be provided with grooves, holes, etc. Besides that, bearing races must be reliably secured to prevent them from turning in the body or on the shaft.

Suitable radial security and adequate support can only be achieved if bearing rings are mounted with overlap. If however easy assembly and disassembly are required, alternatively axial transferability of axially free bearing, fixed location of the ring cannot be selected.

Where free location is chose, provisions must be adopted to avoid irrevocable wear during shifting the ring.

Rotating shaft or another component located in roller bearings is guided by them in radial and axial direction so that the principal condition of definiteness of its movement is achieved. If possible, the component should certainly be located, i.e. supported radially on two spots and axially in one spot.

Examples of such location are shown in figures 4.12. Most common location is such where the shaft is located radially in two bearings one of which locks it in axial direction. Guide (fixed) bearing transfers radial load and also axial load in both directions. Radial bearings are mostly used as guide. They are able to transfer combined load, e.g. single row ball bearings, double row angular-contact bearings, double row self--aligning ball bearing, double row spherical-roller bearings or single row angular-contact ball bearings and tapered roller bearings. The lastly mentioned two bearing types must be assembled in pairs. Free bearing only transfers radial load and must allow certain displacement of the shaft in axial direction in order to prevent occurrence of undesired prestress caused by external effects (thermal dilatation, production inaccuracy of connecting location components, etc.).

Axial displacement can be achieved by shifting between one of the body rings and machine components directly associated with the bearing, e.g. between the outer bearing ring and the bore in the body (fig. 4.12a, b), or directly in the bearing (fig. 4.12 c to h).

Locations where higher radial force and axial load in higher revolution frequency act should be solved by the bearings capturing only radial or axial forces, see fig. 8.3.

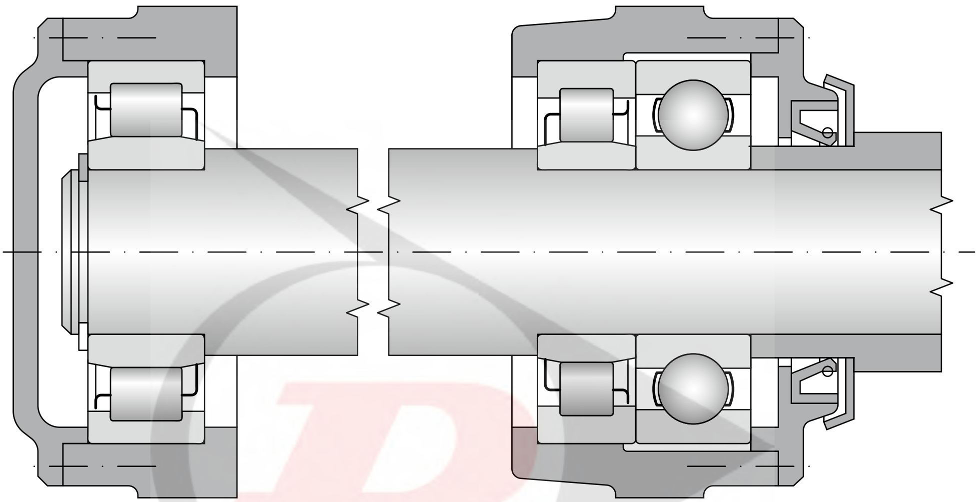

Fig. 8.3

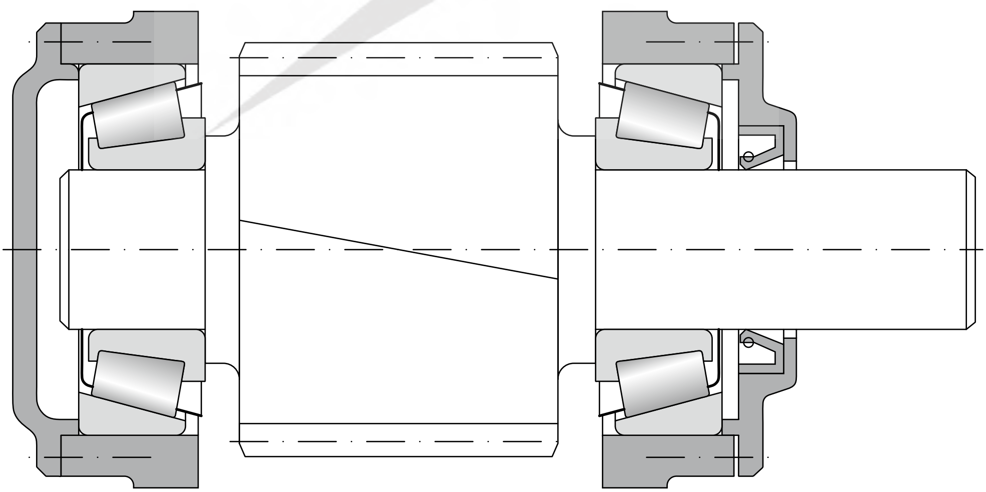

Fig. 8.4

In these cases, any of radial bearings can be used for radial guidance, and those radial bearings for axial guidance that feature the ability to transfer also axial load, alternatively a pair of these bearings or double direction thrust bearings or a pair of single direction thrust bearings. Condition is that axially guide bearings have to be located with radial clearance.

Another frequently used solution is location in two bearings the design of which allows capturing of both radial and axial load in both directions. Axial load is captured in turns by both bearings, always by the direc-tion in which forces act and, at the same time, they transfer also radial load. An example of such location is shown in figure 8.4.

In this case, a pair of single row tapered roller or single row angular-contact ball bearings is used as a well tested construction. Also other types of bearings that are able to transfer load in radial and axial direction at the same time can be used, e.g. single row bearings, alternatively single row cylindrical roller bearings in NJ design, etc.

Radial and axial security of bearing on journal and in body bore or in another part has direct connection with the overall design location arrangement. When selecting the method of fixation, the character and inten-sity of acting forces has to be considered particularly, as well as service temperature at the point of location and the material of connecting components.

When specifying the dimensions of connecting parts, the designer needs to consider also the assembly and disassembly method and maintenance actions, besides the type and dimensions of the bearing.

8.2.1 Radial security of bearings

Bearing is fixed in radial direction on fitted cylindrical surface on the surface of the journal and bore in the body. In some cases of fixation on journal, adapter or withdrawal sleeve is used; alternatively the bearing can be fixed directly on tapered journal.

Proper radial fixation of bearing on journal and in body is very important for utilisation of its loading capa-city and correct location function. In doing so, the following aspects need to be considered:

a) safe fixation and uniform support of rings

b) easy assembly and disassembly

c) displacement of free bearing in axial direction

In principle, both bearing rings should be fixed firmly since only this way their reliable support on the entire circumference and radial fixation against spinning can be achieved. To simplify assembly and disassembly or in order to shift free bearing, one of the rings can be located as sliding.

If proper radial fixation of bearing is selected, one needs to evaluate and consider the effect of the method of rotation and intensity of load.

Circumferential load

Circumferential load occurs when relevant bearing ring turns, and the direction of load does not change, or when the ring does not turn and the load rotates. The bearing ring circumference is loaded successively in one revolution. In this case, loaded ring must be always fixed with necessary overlap.

Spot load

Spot load occurs when the bearing ring stands and outer force is directed still in the same spot of the race-way, or when the ring and force rotate at the same revolution frequency. The ring to which the spot load acts can be located with clearance (mobile), if the conditions require so.

Uncertain way of loading

In case of uncertain way of loading, the ring is acted on by variable external forces the direction and change of load of which cannot be determined (e.g. unbalanced masses, shocks, etc.). Uncertain way of loading requires that both rings are located with overlap (firmly). Under this condition in majority cases of location bearings with increased radial clearance have to be selected.

Load intensity

The load directly affects selection of the size of overlap in location. The bigger the load of the bearing, the bigger overlap in location has to be selected. This particularly applies in cases of shock and vibration load of the bearing. Fixed location on journal or in bore of the body induces deformation of ring, which reduces radial clearance. To ensure the needed radial clearance in cases of fixed location, sometimes bearings with increased radial clearance have to be used. Final clearance after assembly depends on the type and size of the bearing. Therefore the size of needed overlap of fitted ring has to be considered by the type and size of the bearing. For bearings of smaller dimensions smaller overlaps are selected, and vice versa. Relatively smaller overlaps are used e.g. for ball bearings of the same bigness comparing to cylindrical roller, tapered roler or spherical roller bearings.

Material and design of connecting pieces

Designing and determination of tolerances of connecting parts must take into account the materials used, as well as the construction of the connecting pieces. Results of practical experiences reflect in the below stated charts. When bearings are mounted in bodies made of light metal alloys or on journals of hollow shafts, location with higher overlaps has to be selected.

Split bodies are not suitable for locations with big overlaps since they represent a risk of gripping the bearing in the dividing plane of the body.

Heating and warmth

Warmth generated in bearing may lead to release of overlap on the journal which may cause spinning the ring. An opposite case may occur in the body. Heating causes clearance adjustment which will limit up to eliminate axial displacement of the ring of free bearing in the body. Therefore we need to be very attentive to this factor when designing the location.

Accuracy of bearing surfaces

Accuracy of bearing surfaces in terms of tolerances and geometrical shapes is important since it may transfer to raceways of bearing rings. First of all, this has to be reflected in location designs which are highly focused on the running accuracy. Major share of inequality is transferred in thin profiles of bearing rings.

When normal accuracy level bearings are used, usually tolerances within the tolerance level IT6 are selected for the bearing surface on the journal, whilst for the bearing surface in the body the selected tolerance level is IT7.

For ball and cylindrical roller bearings of smaller dimensions, IT5 level can be used for the journal and IT6 for the bore in the body.

For bearings of higher accuracy levels, for locations with high accuracy requirements, e.g. machine tool spin-dles, the recommended least level is IT5 for the shaft, and at least IT6 for the body.

Table 8.1

Recommended accuracies of the shape of bearing surfaces for bearings			
Accuracy level of bearing	**Location place**	**Admissible deviation of cylindricality**	**Admissible frontal runout of support surfaces towards the axis**
P0, P6	shaft	IT5/2	IT3
	body	IT6/2	IT4
P5, P4	shaft	IT3/2	IT2
	body	IT4/2	IT3

Table 8.2

Basic tolerances IT2 to IT6						
Nominal diameter		**Tolerance level**				
over	**to**	**IT2**	**IT3**	**IT4**	**IT5**	**IT6**
mm		**µm**				
6	10	1,5	2,5	4	6	9
10	18	2	3	5	8	11
18	30	2,5	4	6	9	13
30	50	2,5	4	7	11	16
50	80	3	5	8	13	19
80	120	4	6	10	15	22
120	180	5	8	12	18	25
180	250	7	10	14	20	29
250	315	8	12	16	23	32
315	400	9	13	18	25	36
400	500	10	15	20	27	40

Allowed deviation of roundness and cylindricality and allowed frontal run out of bearing and support surfaces for bearings must be smaller against the axis than the scope of tolerance of the diameters of the journal and the bore. With increasing accuracy of the bearings used, also the requirements for the accuracy of bearing surfaces grow. The recommended accuracy values of the bearing surfaces shape for bearings are stated in chart 8.1, and general tolerances IT2 to IT6 in chart 8.2

Assembly and disassembly of bearing

If any of the rings is located with clearance (mobile), the assembly is easy. If the service conditions require that both rings are located with overlap, a suitable type of bearing has to be chosen, e.g. separable bearing (tapered, cylindrical, needle), or a bearing with tapered bore. Shaft journals for location of sleeves for bearing with tapered bore can be within the h9 or h10 tolerance, geometrical shape must be within the accuracy IT5 or IT7, depending on the complexity of location.

Axial displacement of free bearing races

At any service conditions the axial displacement of free bearing has to be ensured. If non-separable bearings are used, displacement of spot-loaded ring will be reached by locating with clearance (mobile location). In bodies made of light metal alloys the bore has to be sleeved with a steel sleeve, if outer ring is to be located with clearance. Reliable sliding ability in axial direction will be achieved if cylindrical roller bearings of N and NU designs or radial needle roller bearings are used in the location.

The recommended tolerances of journal and hole diameters of connecting pieces are for radial and axial bearings stated in charts 8.3 to 8.10.

Table 8.3

Tolerances of journal diameters for radial bearings (applies for full steel shafts)					
Service conditions	Examples of location	Journal diameter [mm]			Tolerance
		Ball bearings	Cylindrical roller, needle roller [1], tapered roller bearings	Spherical roller bearings	
Inner ring spot load					
Small and normal load Pr ≦ 0.15 Cr	Free wheel, pulleys, belt pulleys	All diameters			g6 [2]
Big impact load Pr > 0.15 Cr	Wheels of conveyance trolleys, tension pulleys				h6
Circumferential load of inner ring or uncertain way of loading					
Small and variable load Pr ≦ 0.07 Cr	Conveyers, fans	(18) to 100 (100) to 200	≦ 40 (40) to 140	- -	j6 k6
Normal and big load Pr > 0.07 Cr	General engineering, pumps, combustion engines transmissions, woodworking machines	≦ 18 (18) to 100 (100) to 140 (140) to 200	- ≦ 40 (40) to 100 (100) to 140 (140) to 200 > 200	- - ≦ 40 (40) to 65 (65) to 100 (100) to 140 > 140	j5 k5 (k6) [3] m5 (m6) [3] m6 n6 p6
Extremely big load, shocks heavy service conditions Pr > 0.15 Cr	Axle bearings of rail vehicles, traction motors rolling mills	- - -	(141) to 140 (140) to 500 > 500	(101) to 100 (100) to 500 > 500	n6 [4] p6 [4] r6 (p6) [4]
High location accuracy at small load Pr ≦ 0.07 Cr	Machine tools	≦ 18 (18) to 100 (100) to 200	- ≦ 40 (40) to 140 (140) to 200	- - - -	h5 [5] j5 [5] k5 [5] m5)
Axial load exclusively			all diameters		j6
Bearings with tapered bore and with adapter or withdrawal sleeve or dismantling sleeve					
All ways of loading	General locations, axle bearings of rail vehicles, Unexacting locations		all diameters		h9/IT5 h10/IT7

[1] Does not apply to needle bearings without rings
[2] For bearings tolerance f6 can be selected to ensure axial shift
[3] Tolerance in brackets is selected usually for single row tapered roller bearings or at low frequency revolutions where clearance diffusion does not have major significance.
[4] Bearings with increased radial clearance have to be used
[5] Tolerances for single row ball bearings of accuracy P5 and P4 are stated in chapter 12.2

Table 8.4

Tolerance of diameters of radial bearing body bores (applies to bodies of steel, alloy and cast steel)				
Service conditions	**Sliding ability of outer racew**	**Body**	**Examples of location**	**Tolerance**
Circumferential load of outer ring				
Big shock load Pr > 0.15 Cr Thin-walled elements	Does not slide	Single piece	Hubs with roller bearings, crank pin bearings	P7
Normal and big load Pr > 0.07 Cr	Does not slide		Hubs with roller bearings travelling wheels of cranes, crank shaft bearings	N7
Small and variable load Pr ≦ 0.07 Cr	Does not slide		Converyer rollers, tension pulleys	M7
Uncertain way of loading				
Big shock load Pr > 0.15 Cr	Does not slide		Traction motors	M7
Big and normal load Pr > 0,07 Cr	Usually does not slide	Single piece	Electromotors, pumps, fans, crank shafts	K7
Small and variable load Pr ≦ 0.07 Cr	Usually sliding		Electromotors, pumps, fans, crank shafts	J7
Accurate locations				
Small load Pr ≦ 0.07 Cr	Usually does not slide	Single piece	Roller bearings for machine tools, ball bearings for machine tools, small electromotors	K6 [1]
	Sliding			J6 [2]
	Slightly pushing			H6
Spot load of outer ring				
Optional load	Slightly pushing	Single piece or two piece	General engineering axle bearings of rail vehicles	H7 [3]
Small and normal load Pr ≦ 0.15 Cr	Slightly pushing	Single piece or two piece	General engineering less exacting mechanical eingineering	H8
			Paper machine drying cylinders, big electromotors	G7 [4]

1) For big load, stronger M6 or N6 tolerances are selected. For cylindrical roller bearings with tapered bore, tolerances K5 or M5 are selected.
2) Tolerances for single row ball bearings of accuracy P5 and P4 are stated in chapter 12.2
3) For bearings with outer diameter D < 250 mm with thermal difference between outer ring and body above 10 °C, tolerance G7 is selected
4) For bearings with outer diameter D > 250 mm with thermal difference between outer ring and body above 10 °C, tolerance F7 is selected.

Table 8.5

Tolerance of journal diameters for axial bearings				
Bearing type	**Way of loading**		**Journal diameter**	**Tolerances**
			[mm]	
Axial ball				j6
	Axial load exclusively		All diameters	
Axial spherical-roller				j6
	Current axial and radial load	Spot load of shaft ring	All diameters	j6
		Circumferential load of shaft ring or uncertain way of loading	≦ 200	k6
			(200) to 400	m6
			> 400	n6

Table 8.6

Tolerance of diameters of axial bearing body bores				
Bearing type	**Way of loading**		**Note**	**Tolerances**
Axial ball	Axial load exclusively		In common locations, the casing ring may feature clearance	H8
			Casing ring is mounte with radial clearance	-
Axial spherical-roller	Current axial and radial load	Spot load or uncertain way of loading of casing ring		H7
		Circumferential load		M7
		Circumferential load		

Table 8.7

Limit deviations of journal diameter tolerances																	
Nominal diameter of journal		**f6**		**g5**		**g6**		**h5**		**h6**		**j5**		**j6(js6)**		**k5**	
over	**to**	**upper**	**lower**	**upper**	**lower**	**upper**	**lower**	**upper**	**lower**	**upper**	**lower**	**upper**	**lower**	**upper**	**lower**	**upper**	**lower**
mm		**µm**															
1	3	-6	-12	-2	-6	-2	-8	0	-4	0	-6	2	-2	4	-2	4	0
3	6	-10	-18	-4	-9	-4	-12	0	-5	0	-8	3	-2	6	-2	6	1
6	10	-13	-22	-5	-11	-5	-14	0	-6	0	-9	4	-2	7	-2	7	1
10	18	-16	-27	-6	-14	-6	-17	0	-8	0	-11	5	-3	8	-3	9	1
18	30	-20	-33	-7	-16	-7	-20	0	-9	0	-13	5	-4	9	-4	11	2
30	50	-25	-41	-9	-20	-9	-25	0	-11	0	-16	6	-5	11	-5	13	2
50	80	-30	-49	-10	-23	-10	-29	0	-13	0	-19	6	-7	12	-7	15	2
80	120	-36	-58	-12	-27	-12	-34	0	-15	0	-22	6	-9	13	-9	18	3
120	180	-43	-68	-14	-32	-14	-39	0	-18	0	-25	7	-11	14	-11	21	3
180	250	-50	-79	-15	-35	-15	-44	0	-20	0	-29	7	-13	16	-13	24	4
250	315	-56	-88	-17	-40	-17	-49	0	-23	0	-32	7	-16	16	-16	27	4
315	400	-62	-98	-18	-43	-18	-54	0	-25	0	-36	7	-18	18	-18	29	4
400	500	-68	-108	-20	-47	-20	-60	0	-27	0	-40	7	-20	20	-20	32	5
500	630	-76	-120	-	-	-22	-66	-	-	0	-44	-	-	22	-22	-	-
630	800	-80	-130	-	-	-24	-74	-	-	0	-50	-	-	25	-25	-	-
800	1000	-86	-142	-	-	-26	-82	-	-	0	-56	-	-	28	-28	-	-
1000	1250	-98	-164	-	-	-28	-94	-	-	0	-66	-	-	33	-33	-	-

Table 8.8

Limit deviations of journal diameter tolerances																	
Nominal diameter of journal		k6		m5		m6		n6		p6		h9[1]		IT5	h10[1]		IT7
over	to	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower		upper	lower	
mm		μm															
1	3	6	0	6	2	8	2	10	4	12	6	0	-25	4	0	-40	10
3	6	9	1	9	4	12	4	16	8	20	12	0	-30	5	0	-48	12
6	10	10	1	12	6	15	6	19	10	24	15	0	-36	6	0	-58	15
10	18	12	1	15	7	18	7	23	12	29	18	0	-43	8	0	-70	18
18	30	15	2	17	8	21	8	28	15	35	22	0	-52	9	0	-84	21
30	50	18	2	20	9	25	9	33	17	42	26	0	-62	11	0	-100	25
50	80	21	2	24	11	30	11	39	20	51	32	0	-74	13	0	-120	30
80	120	25	3	28	13	35	13	45	23	59	37	0	-87	15	0	-140	35
120	180	28	3	33	15	40	15	52	27	68	43	0	-100	18	0	-160	40
180	250	33	4	37	17	46	17	60	31	79	50	0	-115	20	0	-185	46
250	315	36	4	43	20	52	20	66	34	88	56	0	-130	23	0	-210	52
315	400	40	4	46	21	57	21	73	37	98	62	0	-140	25	0	-230	57
400	500	45	5	50	23	63	23	80	40	108	68	0	-155	27	0	-250	63
500	630	44	0	-	-	70	26	88	44	122	78	0	-175	30	0	-280	70
630	800	50	0	-	-	80	30	100	50	138	88	0	-200	35	0	-320	80
800	1000	56	0	-	-	90	34	112	56	156	100	0	-230	40	0	-360	90
1000	1250	66	0	-	-	106	40	132	66	186	120	0	-260	46	0	-420	105

[1] In journals manufactured within tolerances h9 and h10 for bearings with adapter or withdrawal sleeve, the circularity and cylindricality deviations must not exceed the basic tolerance IT5 and IT7.

Table 8.9

Limit deviations of bore diameter tolerances															
Nominal diameter of bore		F7		G6		G7		H6		H7		H8		J6(Js6)	
over	to	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower
mm		μm													
6	10	28	13	14	5	20	5	9	0	15	0	22	0	5	-4
10	18	34	16	17	6	24	6	11	0	18	0	27	0	6	-5
18	30	41	20	20	7	28	7	13	0	21	0	33	0	8	-5
30	50	50	25	25	9	34	9	16	0	25	0	39	0	10	-6
50	80	60	30	29	10	40	10	19	0	30	0	46	0	13	-6
80	120	71	36	34	12	47	12	22	0	35	0	54	0	16	-6
120	180	83	43	39	14	54	14	25	0	40	0	63	0	18	-7
180	250	96	50	44	15	61	15	29	0	46	0	72	0	22	-7
250	315	108	56	49	17	69	17	32	0	52	0	81	0	25	-7
315	400	119	62	54	18	75	18	36	0	57	0	89	0	29	-7
400	500	131	68	60	20	83	20	40	0	63	0	97	0	33	-7
500	630	146	76	66	22	92	22	44	0	70	0	110	0	22	-22
630	800	160	80	74	24	104	24	50	0	80	0	125	0	25	-25
800	1000	176	86	82	26	116	26	56	0	90	0	140	0	28	-28
1000	1250	203	98	94	28	133	28	66	0	105	0	165	0	33	-33
1250	1600	235	110	108	30	155	30	78	0	125	0	195	0	39	-39

Table 8.10

Limit deviations of bore diameter tolerances															
Nominal diameter of bore		J7(Js7)		K6		K7		M6		M7		N7		P7	
over	to	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower
mm		µm													
6	10	8	-7	2	-7	5	-10	-3	-12	0	-15	-4	-19	-9	-24
10	18	10	-8	2	-9	6	-12	-4	-15	0	-18	-5	-23	-11	-29
18	30	12	-9	2	-11	6	-15	-4	-17	0	-21	-7	-28	-14	-35
30	50	14	-11	3	-13	7	-18	-4	-20	0	-25	-8	-33	-17	-42
50	80	18	-12	4	-15	9	-21	-5	-24	0	-30	-9	-39	-21	-51
80	120	22	-13	4	-18	10	-25	-6	-28	0	-35	-10	-45	-24	-59
120	180	25	-14	4	-21	12	-28	-8	-33	0	-40	-12	-52	-28	-68
180	250	30	-16	5	-24	13	-33	-8	-37	0	-46	-14	-60	-33	-79
250	315	36	-16	5	-27	16	-36	-9	-41	0	-52	-14	-66	-36	-88
315	400	39	-18	7	-29	17	-40	-10	-46	0	-57	-16	-73	-41	-98
400	500	43	-20	8	-32	18	-45	-10	-50	0	-63	-17	-80	-45	-108
500	630	35	-35	0	-44	0	-70	-26	-70	-26	-96	-44	-114	-78	-148
630	800	40	-40	0	-50	0	-80	-30	-80	-30	-110	-50	-130	-88	-168
800	1000	45	-45	0	-56	0	-90	-34	-90	-34	-124	-56	-146	-100	-190
1000	1250	52	-52	0	-66	0	-105	-40	-106	-40	-145	-66	-171	-120	-225
1250	1600	62	-62	0	-78	0	-125	-48	-126	-48	-173	-78	-203	-140	-265

8.2.2 Axial security of bearings

Inner bearing ring with cylindrical bore seated on journal with overlap (fixed location) is usually locked in axial direction using a adapter nut, terminal plate or snap ring whilst the other face is usually leaned by the shaft fitting. Adjacent components are used as support faces for inner rings and, if needed, spacer rings are inserted between this component and the inner ring of the bearing. Examples of axial fixation of bearing are shown in figure 8.5.

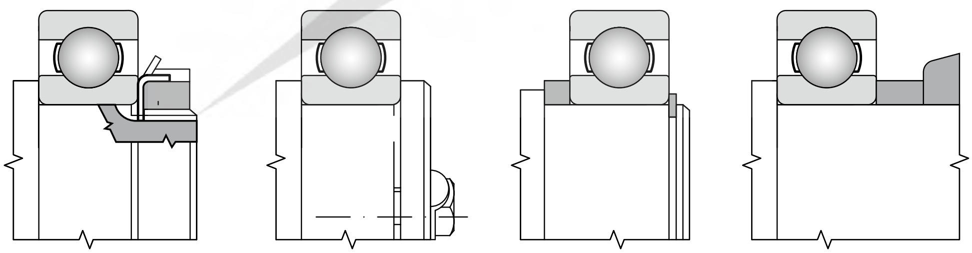

Fig. 8.5

Bearings with tapered bore mounted directly on tapered journal are usually secured with a safety nut screwed onto the thread on the shaft. If bearings are mounted on withdrawal sleeve, the inner ring must be supported, e.g. by a spacer ring. The spacer ring can form a part of labyrinth. The withdrawal sleeve is axially fixed with terminal plate or safety nut.

Examples of axial fixation of bearing with tapered bore directly on tapered journal or by means of adapter or withdrawal sleeve are shown in Fig. 8.6.

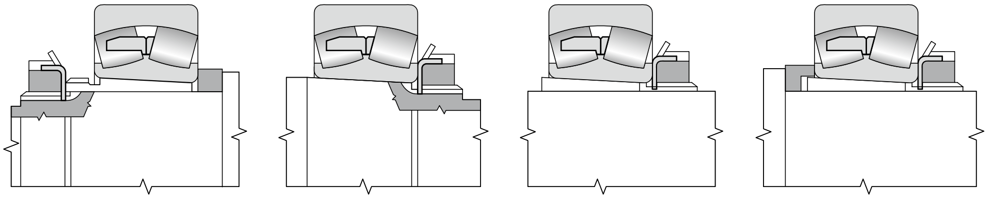

Fig. 8.6

Admissible axial load of bearings fixed by means of adapter sleeve on smooth shafts without the bearing leaning on shaft fitting is calculated by the below equation:

$$F_a = 3B \cdot d \qquad [N]$$

F_a . . . . . . admissible axial load of bearing [N]

B . . . . . . bearing width [mm]

d . . . . bearing hole diameter [mm]

If axial displacement of outer ring in body is not desirable, we can use a solution utilising the front support surface or seating surface of the bearing lid, nut or snap ring. Bearings with a groove for snap ring (NR) are less demanding in space, and their locking is simple.

Examples of solution are shown in Fig. 8.7.

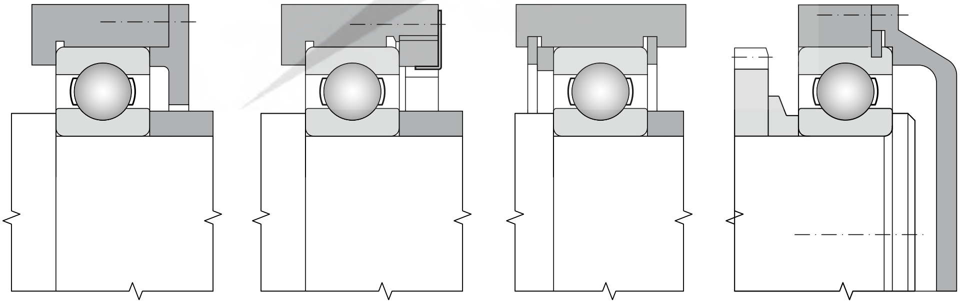

Fig. 8.7

Connecting dimensions for individual bearing types are stated in this publication in the chart section (chapter 12).

8.3 Seal

Sealing the bearing space is very important since harmful substances present in the proximity of the bearing affect it and often even put it out of service. Seal has also an opposite function – it prevents the grease from leaking out of the bearing and from the stowage compartment. For that reason, the seal has always to be designed considering the service conditions of the machine or equipment, lubrication method, maintenance options and economic aspects of production and use.

8.3.1 Contact-free sealing

This type of seal features only a tight gap between the non-rotary and rotary component which is sometimes filled with grease. In this design no wear due to friction occurs, and therefore this seal suits to use for highest circumferential speeds and high service temperatures. Examples of slotted seals are shown in fig. 8.8

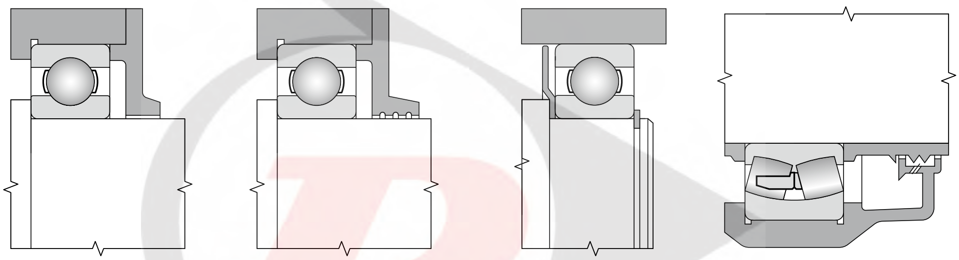

Fig. 8.8

Another very efficient seal is a labyrinth seal which can be used to enhance the packing effect by higher number of labyrinths or extension of sealing slots. See fig. 8.9. for examples of this seal.

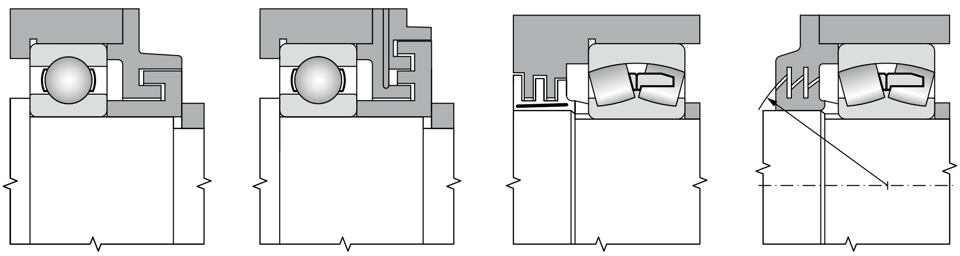

Fig. 8.9

8.3.2 Friction sealing

Friction sealing is made of elastic or soft but sufficiently solid and impermeable material that is inserted between the rotary and fixed component. Such seal is usually cheap and suits to various constructions. Disadvantage is sliding friction touching the surfaces which limits the use of it for high circumferential speeds.

The simplest is seal with a felt ring (fig. 8.10). It suits to service temperatures within -40 °C and +80 °C and to circumferential even to 7 $m.s^{-1}$, whilst the maximum required surface roughness of the sliding surface is R_a = 0.16, and minimum hardness 45 HRC or treatment by hard chromium plating. Dimensions of felt rings and grooves are solved by relevant national standards.

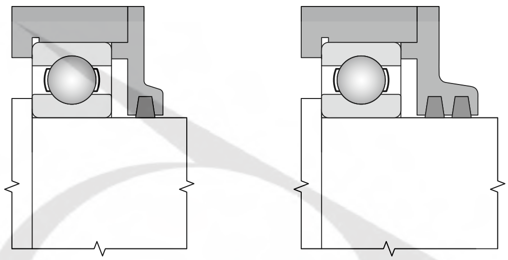

Fig. 8.10

A very frequent sealing method is sealing with shaft rings (fig. 8.11). Shaft rings are made of rubber or other suitable plastics, stiffened by metal stiffener. According by the material used they suit to service temperatures from -30 °C to +160 °C. Admissible circumferential speed depends on the roughness of the sliding surface roughness.

- to 2 $m.s^{-1}$ the roughness is max R_a = 0.8,
- to 4 $m.s^{-1}$ the roughness is max R_a = 0.4,
- to 12 $m.s^{-1}$ the roughness is max R_a = 0.2.

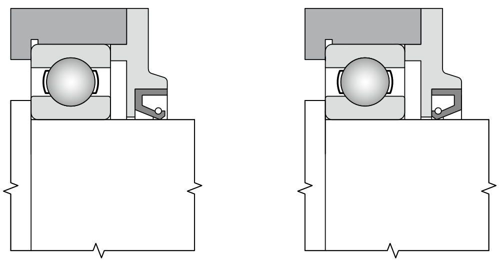

Fig. 8.11

Besides the stated most common sealing rings there are other friction seal designs that utilise specifically shaped sealing rings made of rubber, plastic, etc., or special elastic metal rings. This seal is either selected for locations with high demands on sealing the bearing space (bog contamination of ambient area, high temperature, effect of chemicals), or due to economic reasons in bulk and large lot production. Examples are shown in fig. 8.12.

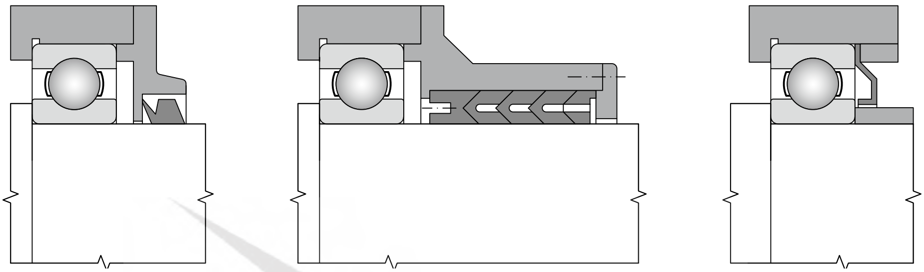

Fig. 8.12

8.3.3 Combined seals

Enhanced sealing effect is achieved by combination of contact-free and friction sealing. Such seals are reco-mmended for humid and contaminated environment. Example is shown in fig. 8.13.

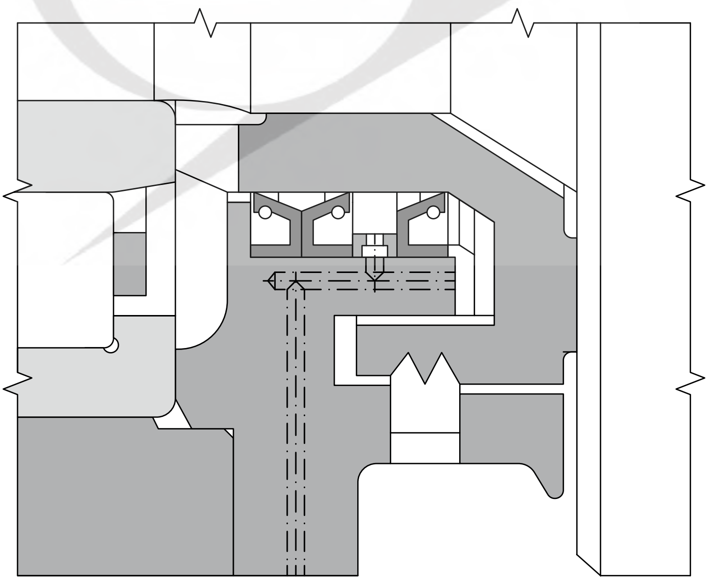

Fig. 8.13

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