FAG

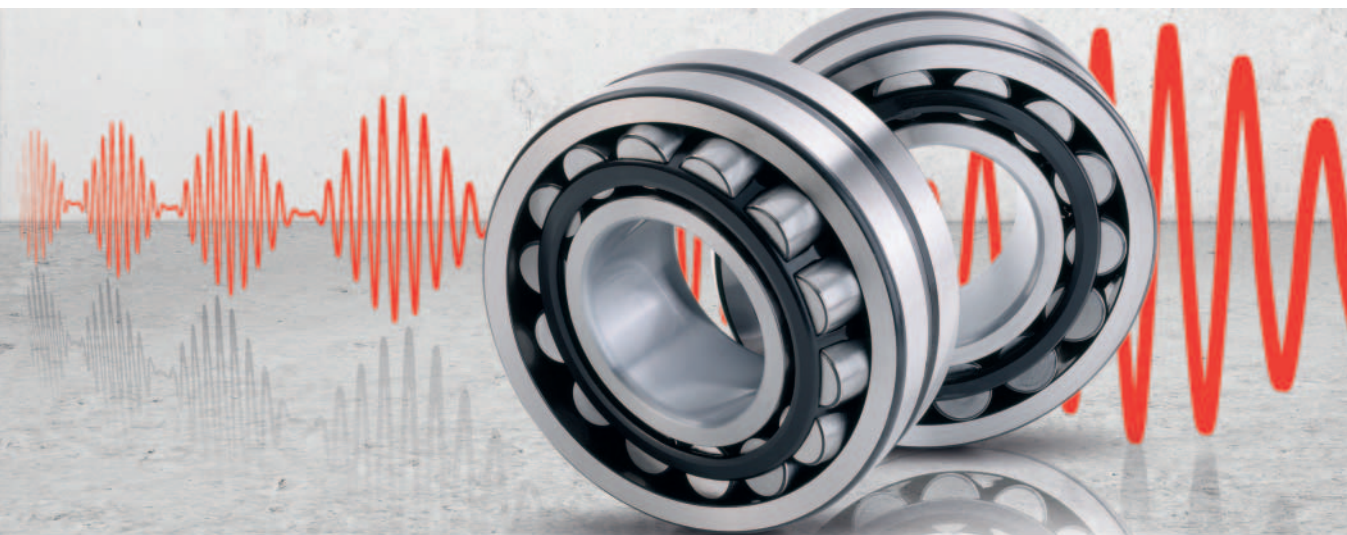# FAG Special Spherical Roller Bearings for Vibratory Machinery

SCHAEFFLER

Contents

FAG special spherical roller bearings for vibratory machinery

Features
Operating conditions
for bearings
in vibratory machinery

Vibratory screens used for grading, in other words the separation of solid materials according to grain size and other vibratory machinery such as road rollers and saw frames are among the machines subjected to the most severe stresses.

The rolling bearings fitted in the exciter units of these machines must support not only high loads and high speeds but also accelerations and centrifugal forces. In many cases, these applications involve adverse environmental conditions such as contaminated environments and excessive moisture.

The special spherical roller bearings developed by FAG are matched to the operating conditions in vibratory machinery and have proved highly successful in practical use.

In particular, the cages of the rolling bearings are subjected to stresses arising from high radial accelerations. In unfavourable cases, these may be overlaid by axial accelerations as well.

The rotating imbalance generates rotating shaft deflection and additional sliding motion within the bearings. This increases the friction and therefore the operating temperature of the bearings. FAG special spherical roller bearings can support dynamic angular misalignments of up to 0,15°. For larger misalignments, please consult Application Engineering at Schaeffler.

Basic designs of FAG special
spherical roller bearings

FAG special spherical roller bearings for vibratory machinery have main dimensions corresponding to dimension series 23 (DIN EN 616:1995-01, ISO 15). For the particular stresses occurring in vibratory machinery, we manufacture all the special spherical roller bearings described in this publication in accordance with specification T41A or T41D, see page 5.

Very high load carrying capacity is achieved through optimum use of bearing cross-section as a result of further development of spherical roller bearings of series 223..-E1-XL in X-life quality.
In the design for vibratory stresses, these bearings are supplied up to a bore diameter of 220 mm.

X-life spherical roller bearings 223..-E1-XL-T41A(D)

FAG spherical roller bearings of the E1 design have an inner ring without a rib and are characterised by very high load carrying capacity. This advantage is also offered by FAG special bearings for vibratory stress of the series 223..-E1-XL with the suffix T41A or T41D, *Figure 1*.

These are the FAG standard designs for bearings with a bore diameter of 40 mm up to and including 150 mm (bore code 08 to 30).

After extensive testing in rig and field tests, bearings of the X-life design 223..-E1-XL-T41A(D) have proved extremely successful in numerous practical applications.

The bearing has one sheet steel window cage half per row of rollers with high geometrical stability. The cage halves are supported via the cage guide ring in the outer ring. The guide ring is of a single piece design. All cage parts are subjected to a special surface hardening process.

X-life spherical roller bearings 223..-BE-XL-JPA-T41A

The bearings with a bore diameter of 160 mm to 220 mm (bore code 32 to 44) are also available in the described design, *Figure 1*. These bearings also correspond to our proven X-life standard. These larger bearings can be identified by the sheet steel cage JPA in the suffix.
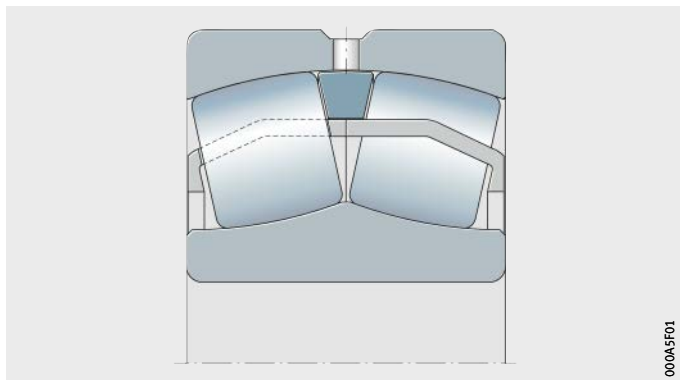

Figure 1
X-life designs 223..-E1-XL-T41A(D) and 223..-BE-XL-JPA-T41A of FAG special spherical roller bearings for vibratory machines

FAG special spherical roller bearings for vibratory machinery

Bearings with tapered bore

In special cases such as saw frames, bearings are also available with a tapered bore (taper 1:12). The ordering designations are respectively 223..-E1-XL-K-T41A and 223..-BE-XL-K-JPA-T41A.
They are available with a bore diameter of or larger than 160 mm.

Bearings with coated bore

In order to reduce or prevent fretting corrosion between the bearing bore and the shaft, we can for specific orders supply spherical roller bearings with a cylindrical bore coated with Durotect CK.
This ensures that the possibility of displacement (non-locating bearing function) between the bearing bore and shaft in response to thermal influences is maintained over and beyond a long period of operation.

The bearings with a coated bore correspond in their dimensions and tolerances to and are interchangeable with the FAG standard bearings for vibratory machinery.

For bearings 22317-E1-XL-T41D to 22330-E1-XL-T41D, the cylindrical bore coated with Durotect CK is the standard. Further information is given in our publication PPD, FAG Special Spherical Roller Bearings with Durotect CK Coating in the Bore.

For bearings outside this size range, a coated inner ring bore can be specified using the suffix J24BA in the ordering designation.

Ordering example for a bearing with a Durotect CK coated bore:
- within the standard range:
  22320-E1-XL-T41D
- outside the standard range:
  22316-E1-XL-J24BA-T41A.

Specification T41A (T41D)

FAG spherical roller bearings for vibratory machinery are manufactured in accordance with the specification T41A or T41D. This takes into consideration the particular requirements of the application. The specification defines, for example, the tolerances of bore and outside diameter, as well as the radial internal clearance of the bearings. The other tolerances are in accordance with tolerance class PN to DIN 620.

Tolerances for bearing bore and outside diameter

The specification T41A(D) prescribes a restriction of the bore tolerance to approx. the upper half of the normal tolerance. For the outside diameter, only the centre half of the normal tolerance zone is permissible. In bearings with a tapered bore, the reduced tolerance range applies to the outside diameter only, see tables. Through these measures, the sliding fit required for the inner ring is reliably achieved with the shaft tolerances g6 or f6 and the interference fit required for the outer ring is reliably achieved with the housing tolerance P6. For bearings with a coated bore, we recommend using the shaft tolerance f6. The inner ring does not have pure point load and the outer ring is subjected to circumferential load. Geometrical and positional tolerances of bearing seating surfaces, see table, page 6.

Inner ring tolerances

Bore d mm		Bore deviation $t_{\Delta dmp}$ μm	
over	incl.	max.	incl.
30	50	0	−7
50	80	0	−9
80	120	0	−12
120	180	0	−15
180	250	0	−18
250	315	0	−21

Outer ring tolerances

Outside diameter D mm		Outside diameter deviation $t_{\Delta Dmp}$ μm	
over	incl.	max.	incl.
80	150	−5	−13
150	180	−5	−18
180	315	−10	−23
315	400	−13	−28
400	500	−13	−30
500	630	−15	−35

FAG special spherical roller bearings for vibratory machinery

In order to achieve the required fit, the bearing seats and fit surfaces of the shaft and housing bore must conform to certain tolerances, see table.

Geometrical and positional tolerances of bearing seating surfaces

Bearing seating surface	Diameter tolerance	Roundness tolerance	Parallelism tolerance	Axial runout tolerance of abutment shoulder
Shaft	IT6 (IT5)	Circumferential load IT4/2	IT4	IT4
		Point load IT5/2	IT5	
Housing	IT7 (IT6)	Circumferential load IT5/2	IT5	IT5
		Point load IT6/2	IT6	

Radial internal clearance groups

The specification T41A(D) specifies Group 4 as the standard internal clearance group for all spherical roller bearings of vibrating screen design and it is therefore not necessary to indicate this explicitly. In this way, radial preloading of the bearings is prevented in the event of unfavourable interaction between the different influences such as fits, deformations, etc. This applies especially during the startup and running-in periods, during which the largest temperature differences occur between the inner and outer ring.

It is only necessary to consider a special radial internal clearance for spherical roller bearings in vibratory machinery in rare cases, for example if the material to be screened is hot or the bearing arrangement is subjected to excessive external heat. In special cases such as saw frames, bearings with an internal clearance other than Group 4 may be necessary. The suffix for the radial internal clearance, e.g. Group 3, must then be indicated explicitly. Bearings in this variant are available by agreement. Radial internal clearance values for FAG special spherical roller bearings, see table, page 8.

Reduction in radial internal clearance of bearings with tapered bore

Bearings with a tapered bore are mounted on a conical shaft seat or, using a sleeve, on a cylindrical shaft. The reduction in the radial internal clearance during mounting can be taken as an indication of the seating between the inner ring and shaft, see table, page 8.

FAG special spherical roller bearings for vibratory machinery

Reduction in radial internal clearance of FAG spherical roller bearings with tapered bore

Nominal bearing bore diameter d mm		Radial internal clearance before mounting Internal clearance group					
		Group N mm		Group 3 mm		Group 4 mm	
over	incl.	min.	max.	min.	max.	min.	max.
24	30	0,03	0,04	0,04	0,055	0,055	0,075
30	40	0,035	0,05	0,05	0,065	0,065	0,085
40	50	0,045	0,06	0,06	0,08	0,08	0,1
50	65	0,055	0,075	0,075	0,095	0,095	0,12
65	80	0,07	0,095	0,095	0,12	0,12	0,15
80	100	0,08	0,11	0,11	0,14	0,14	0,18
100	120	0,1	0,135	0,135	0,17	0,17	0,22
120	140	0,12	0,16	0,16	0,2	0,2	0,26
140	160	0,13	0,18	0,18	0,23	0,23	0,3
160	180	0,14	0,2	0,2	0,26	0,26	0,34
180	200	0,16	0,22	0,22	0,29	0,29	0,37
200	225	0,18	0,25	0,25	0,32	0,32	0,41
225	250	0,2	0,27	0,27	0,35	0,35	0,45
250	280	0,22	0,3	0,3	0,39	0,39	0,49
280	315	0,24	0,33	0,33	0,43	0,43	0,54
315	355	0,27	0,36	0,36	0,47	0,47	0,59
355	400	0,3	0,4	0,4	0,52	0,52	0,65
400	450	0,33	0,44	0,44	0,57	0,57	0,72
450	500	0,37	0,49	0,49	0,63	0,63	0,79
500	560	0,41	0,54	0,54	0,68	0,68	0,87
560	630	0,46	0,6	0,6	0,76	0,76	0,98
630	710	0,51	0,67	0,67	0,85	0,85	1,09
710	800	0,57	0,75	0,75	0,96	0,96	1,22
800	900	0,64	0,84	0,84	1,07	1,07	1,37
900	1 000	0,71	0,93	0,93	1,19	1,19	1,52
1 000	1 120	0,78	1,02	1,02	1,3	1,3	1,65
1 120	1 250	0,86	1,12	1,12	1,42	1,42	1,8
1 250	1 400	0,94	1,22	1,22	1,55	1,55	1,96

[1] Valid only for solid steel shafts and hollow shafts with a bore no larger than half the shaft diameter.
The following applies: Bearings with a radial internal clearance before mounting in the upper half of the tolerance range are mounted using the larger value for the reduction in radial internal clearance or the axial drive-up distance, while bearings in the lower half of the tolerance range are mounted using the smaller value for the reduction in radial internal clearance or the axial drive-up distance.

[2] The actual value of the radial internal clearance must not be smaller than the control value. In the case of bearings with a small diameter, this may be difficult to determine.

	Reduction in radial internal clearance[1]		Drive-up distance on taper 1:12[1]				Drive-up distance on taper 1:30[1]				Control value for radial internal clearance after mounting[2]		
			Shaft		Sleeve		Shaft		Sleeve		Group N	Group 3	Group 4
	mm		mm		mm		mm		mm		mm	mm	mm
	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	min.	min.
	0,015	0,02	0,3	0,35	0,3	0,4	–	–	–	–	0,015	0,02	0,035
	0,02	0,025	0,35	0,4	0,35	0,45	–	–	–	–	0,015	0,025	0,04
	0,025	0,03	0,4	0,45	0,45	0,5	–	–	–	–	0,02	0,03	0,05
	0,03	0,04	0,45	0,6	0,5	0,7	–	–	–	–	0,025	0,035	0,055
	0,04	0,05	0,6	0,75	0,7	0,85	–	–	–	–	0,025	0,04	0,07
	0,045	0,06	0,7	0,9	0,75	1	1,7	2,2	1,8	2,4	0,035	0,05	0,08
	0,05	0,07	0,7	1,1	0,8	1,2	1,9	2,7	2	2,8	0,05	0,065	0,1
	0,065	0,09	1,1	1,4	1,2	1,5	2,7	3,5	2,8	3,6	0,055	0,08	0,11
	0,075	0,1	1,2	1,6	1,3	1,7	3	4	3,1	4,2	0,055	0,09	0,13
	0,08	0,11	1,3	1,7	1,4	1,9	3,2	4,2	3,3	4,6	0,06	0,1	0,15
	0,09	0,13	1,4	2	1,5	2,2	3,5	4,5	3,6	5	0,07	0,1	0,16
	0,1	0,14	1,6	2,2	1,7	2,4	4	5,5	4,2	5,7	0,08	0,12	0,18
	0,11	0,15	1,7	2,4	1,8	2,6	4,2	6	4,6	6,2	0,09	0,13	0,2
	0,12	0,17	1,9	2,6	2	2,9	4,7	6,7	4,8	6,9	0,1	0,14	0,22
	0,13	0,19	2	3	2,2	3,2	5	7,5	5,2	7,7	0,11	0,15	0,24
	0,15	0,21	2,4	3,4	2,6	3,6	6	8,2	6,2	8,4	0,12	0,17	0,26
	0,17	0,23	2,6	3,6	2,9	3,9	6,5	9	5,8	9,2	0,13	0,19	0,29
	0,2	0,26	3,1	4,1	3,4	4,4	7,7	10	8	10,4	0,13	0,2	0,31
	0,21	0,28	3,3	4,4	3,6	4,8	8,2	11	8,4	11,2	0,16	0,23	0,35
	0,24	0,32	3,7	5	4,1	5,4	9,2	12,5	9,6	12,8	0,17	0,25	0,36
	0,26	0,35	4	5,4	4,4	5,9	10	13,5	10,4	14	0,2	0,29	0,41
	0,3	0,4	4,6	6,2	5,1	6,8	11,5	15,5	12	16	0,21	0,31	0,45
	0,34	0,45	5,3	7	5,8	7,6	13,3	17,5	13,6	18	0,23	0,35	0,51
	0,37	0,5	5,7	7,8	6,3	8,5	14,3	19,5	14,8	20	0,27	0,39	0,57
	0,41	0,55	6,3	8,5	7	9,4	15,8	21	16,4	22	0,3	0,43	0,64
	0,45	0,6	6,8	9	7,6	10,2	17	23	18	24	0,32	0,48	0,7
	0,49	0,65	7,4	9,8	8,3	11	18,5	25	19,6	26	0,34	0,54	0,77
	0,55	0,72	8,3	10,8	9,3	12,1	21	27	22,2	28,3	0,36	0,59	0,84

FAG special spherical roller bearings for vibratory machinery

Permissible radial acceleration

Since the centrifugal forces are supported against the outer ring, high acceleration forces are possible in FAG special spherical roller bearings for vibratory machinery, *Figure 2*.

Permissible radial acceleration values of FAG special spherical roller bearings for the dimension series 223 (n = operating speed, d_M = mean bearing diameter):

- $n \cdot d_M = 350\,000$ min^{-1} · mm
  Maximum possible values with optimum mounting conditions and oil lubrication, e.g. planetary gearbox

- $n \cdot d_M = 140\,000$ min^{-1} · mm
  Normal operating conditions for saw frames with grease lubrication

- $n \cdot d_M = 230\,000$ min^{-1} · mm to $300\,000$ min^{-1} · mm
  Normal operating conditions for vibrating screens with grease or oil lubrication.

g = gravitational acceleration
d = bearing diameter

① $n \cdot d_M = 350\,000$ min^{-1} · mm
② $n \cdot d_M = 140\,000$ min^{-1} · mm
③ $n \cdot d_M = 230\,000$ min^{-1} · mm
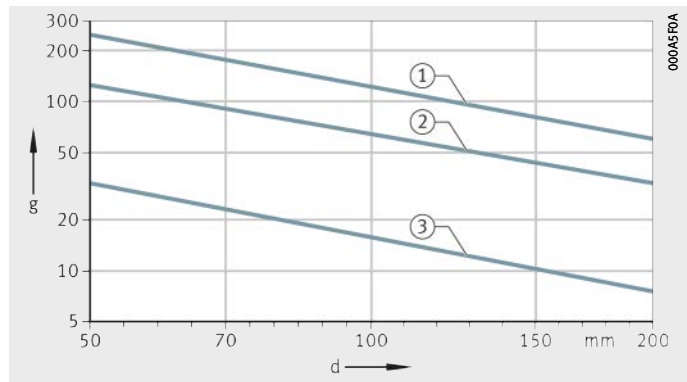   to $300\,000$ min^{-1} · mm

Figure 2
Permissible radial acceleration

Heat treatment

All FAG spherical roller bearings of series 223..-E1-XL-T41A(D) and 223..-BE-XL-JPA-T41A for vibratory stresses are heat treated such that they are dimensionally stable up to an operating temperature of +200 °C.

**Design and
safety guidelines**

Dimensioning of bearings   Vibrating screen bearings are normally designed for a basic rating life of between 10 000 hours and 20 000 hours.

When determining the equivalent dynamic load P of spherical roller bearings for vibratory stresses, the influences that cannot be precisely defined are taken into consideration by means of a safety factor f_z of 1,2 times the radial bearing load F_r. Based on practical experience, this gives sufficiently long running times. More precise calculations can be achieved by determining the expanded adjusted rating life L_{hnm} to ISO 281. The fatigue limit load C_{ur} required in this case is stated in the dimension tables.

Calculation methods   Methods for calculating the rating life include the following:
- basic rating life L_{10} and L_{10h} to ISO 281
- expanded rating life L_{nm} and L_{nmh} to ISO 281.

Basic rating life   The basic rating life L_{10} and L_{10h} is determined as follows:

$$L_{10} = \left(\frac{C}{P} \right)^p$$

$$L_{10h} = \frac{16\,666}{n} \cdot \left(\frac{C}{P} \right)^p$$

L_{10} 10^6 revolutions
The basic rating life in millions of revolutions that is reached or exceeded by 90% of a sufficiently large group of apparently identical bearings before the first evidence of material fatigue develops
C N
Basic dynamic load rating
P N
Equivalent dynamic bearing load for radial and axial bearings
p –
Life exponent;
for roller bearings: $p = 10/3$
for ball bearings: $p = 3$
L_{10h} h
The basic rating life in operating hours according to the definition for L_{10}
n min^{-1}
Operating speed.

FAG special spherical roller bearings for vibratory machinery

Equivalent dynamic bearing load

The equivalent dynamic load P is a calculated value.
This value is constant in magnitude and direction; it is a radial load for radial bearings and an axial load for axial bearings.

A load corresponding to P will give the same rating life as the combined load occurring in practice.

$$P = X \cdot F_r + Y \cdot F_a$$

P	N
Equivalent dynamic bearing load	
X	−
Radial factor given in the dimension tables or product description	
F_r	N
Radial dynamic bearing load	
Y	−
Axial factor given in the dimension tables or product description	
F_a	N
Axial dynamic bearing load.	

! This calculation cannot be applied to radial needle roller bearings, axial needle roller bearings and axial cylindrical roller bearings. Combined loads are not permissible with these bearings. For radial needle roller bearings under purely radial load, $P = F_r$, for axial needle roller bearings and axial cylindrical roller bearings under purely axial load, $P = F_a$.
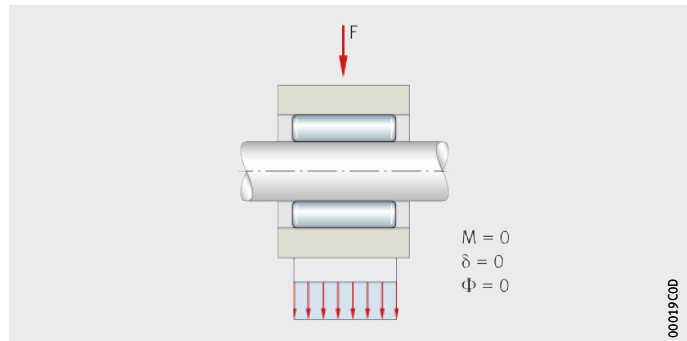
Influencing factors

Influencing factors can include, *Figure 3*:

- ▪ bearing alignment
- ▪ bearing loads
- ▪ operating clearance
- ▪ tilting and moment load
- ▪ lubrication and contamination.

F = load
M = torque
δ = radial deflection
Φ = tilting angle

Figure 3
Calculation model

Expanded rating life   The calculation of the expanded rating life L_{nm} and L_{nmh} was standardised for the first time in DIN ISO 281 Appendix 1. Since 2007, it has been standardised in the worldwide standard ISO 281. Computer-aided calculation to DIN ISO 281 Appendix 4 has been specified since 2008 in ISO/TS 16281 and standardised in DIN 26281.

The rating life L_{nm} and L_{nmh} is calculated as follows:

$$L_{nm} = a_1 \cdot a_{ISO} \cdot L_{10}$$

$$L_{nmh} = a_1 \cdot a_{ISO} \cdot L_{10h}$$

L_{nm} 10^6 revolutions
Expanded adjusted rating life to ISO 281

a_1 −
Life adjustment factor for a requisite reliability other than 90%
Requisite reliability 90% (L_{10m}) $a_1 = 1$
Requisite reliability 95% (L_{5m}) $a_1 = 0,64$
Requisite reliability 99% (L_{1m}) $a_1 = 0,25$

a_{ISO} −
Life adjustment factor for operating conditions

L_{10} 10^6 revolutions
basic rating life

L_{nmh} h
Expanded rating life in operating hours

L_{10h} h
Basic rating life in operating hours according to the definition for L_{10}.

The values for the life adjustment factor a_1 were redefined in ISO 281:2007 and differ from the previous data.

FAG special spherical roller bearings for vibratory machinery

Life adjustment factor a_{ISO}

The standardised method for calculating the life adjustment factor a_{ISO} essentially takes account of the following factors:

■ the load on the bearing
■ the lubrication conditions (viscosity and type of lubricant, speed, bearing size, additives)
■ the fatigue limit of the material
■ the type of bearing
■ the residual stress in the material
■ the ambient conditions
■ contamination of the lubricant.

The life adjustment factor a_{ISO} is calculated as follows:

$$a_{ISO} = f\left[\frac{e_C \cdot C_u}{P}, \kappa\right]$$

a_{ISO} —
Life adjustment factor for operating conditions

e_C —
Life adjustment factor for contamination
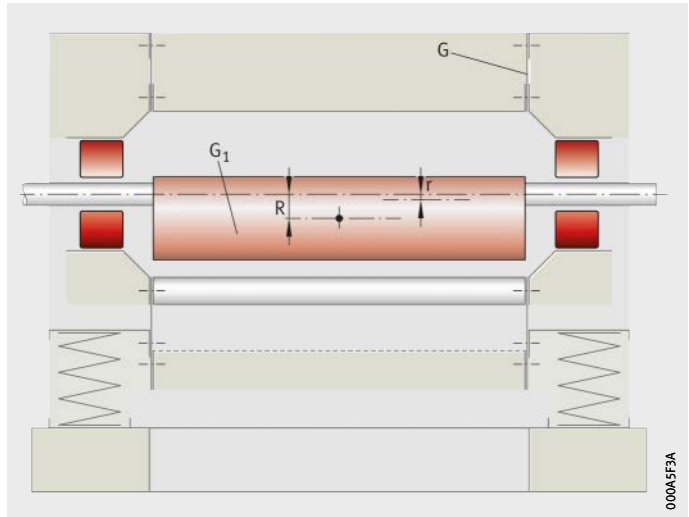
C_u N
Fatigue limit load

κ —
Viscosity ratio;
for $\kappa > 4$ a value of $\kappa = 4$ should be anticipated
for $\kappa < 0{,}1$ this calculation method cannot be used

P N
Equivalent dynamic bearing load.

Two bearing screen with circle throw

The schematic of an imbalance-type two bearing screen is shown in *Figure 4*.

G = screen box
G1 = exciter weight
R = distance between centre of gravity of exciter and bearing axis
r = vibration radius of screen box

Figure 4
Schematic of two bearing screen with circle throw

The radial bearing load imposed by the centrifugal force of the screen box is derived from the screen box weight, the vibration radius and the speed in accordance with the following equation:

$$F_r = \frac{1}{z} \cdot \frac{m}{10^3} \cdot r \cdot \omega^2 = \frac{1}{z} \cdot \frac{G}{g} \cdot r \left(\frac{\pi \cdot n}{30} \right)^2$$

F_r	kN
Radial bearing load	
m	kg
Screen box mass	
r	m
Vibration radius	
ω	1/s
Angular velocity	
G	kN
Screen box weight	
n	min^{-1}
Speed	
z	–
Number of bearings	
g	m/s^2
Gravitational acceleration; g = 9,81 m/s^2.	

FAG special spherical roller bearings for vibratory machinery

The vibration radius r in two bearing screens can be determined from the ratio of the screen box weight to the exciter weight. Since two bearing screens generally operate between the critical range approaching the static amplitude, it can be assumed that the common centroidal axis of the two masses of the screen box and exciter is maintained during rotation, *Figure 5*.

G = screen box weight
G1 = exciter weight
R = distance between centre of gravity of exciter and bearing axis
r = vibration radius of screen box

Figure 5
The vibration radius is determined by the ratio of the screen box weight to the exciter weight
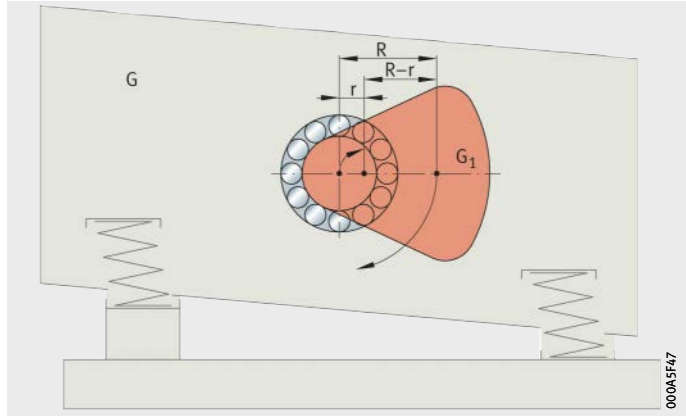
Based on this precondition:

$$G \cdot r = G_1 (R - r)$$

The vibration radius r is thus:

$$r = \frac{G_1 \cdot R}{G + G_1}$$

G   kN
Screen box weight
r   m
Vibration radius of screen box
G_1 kN
Exciter weight
R   m
Distance between centre of gravity of exciter and bearing axis
$G_1 \cdot R$ kN/m
Imbalance moment of exciter
$G + G_1$ kN
Total weight supported by springs.

If the vibration radius r is incorporated in the equation for calculation of the radial bearing load F_r, transformation gives:

$$F_r = \frac{1}{z} \cdot \frac{G_1}{g} \cdot \frac{R}{1 + \dfrac{G_1}{G}} \cdot \left(\frac{\pi \cdot n}{30} \right)^2$$

Example   Data for calculation:
- Screen box weight   G = 35 kN
- Vibration radius   r = 0,003 m
- Speed $n = 1\,200\ \text{min}^{-1}$
- Number of bearings   z = 2.

From the data, the radial bearing load is calculated as follows:

$$F_r = \frac{1}{2} \cdot \frac{35}{9,81} \cdot 0,003 \cdot \left(\frac{\pi \cdot 1\,200}{30} \right)^2 = 84,5\ \text{kN}$$

The equivalent dynamic bearing load required in order to determine the requisite basic dynamic load rating of the bearing is then:

$$P = 1,2 \cdot F_r = 1,2 \cdot 84,5 = 101\ \text{kN}$$

# FAG special spherical roller bearings for vibratory machinery

Two bearing screen with straight line motion

In principle, the exciter in a two bearing screen with straight line motion comprises two contra-rotating synchronous circular throw systems, *Figure 6*.

G = screen box weight

G_1 = exciter weight

R = distance between centre of gravity of exciter and bearing axis

r = vibration radius of screen box

Figure 6
Schematic of two bearing screen with straight line motion
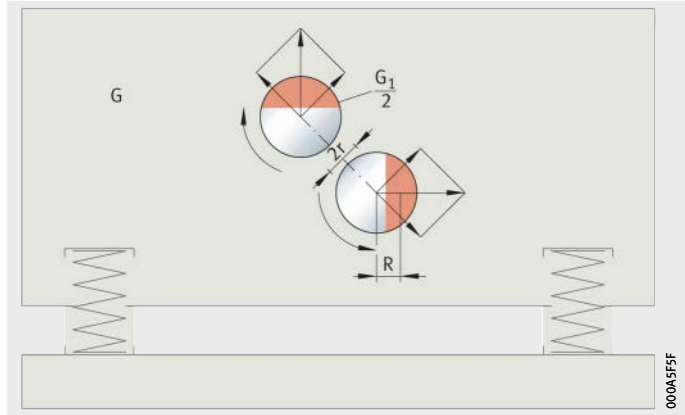
The forces are determined by resolving the rotating centrifugal force vectors of the imbalance shafts into two components, in the direction of the line connecting the two shafts and the direction perpendicular to this line. It can be seen that the components lying in the direction of the connecting line cancel each other out, whereas the perpendicular components add up, generating a harmonic pulsating inertia force that induces straight line vibration of the screen box. Since the so-called static amplitude is induced in the direction of vibration due to the supercritical operation and the common centroidal axis of the screen box and the imbalance masses does not vary during vibration, the bearing loads are as follows.

In the direction of vibration, the radial bearing load is as follows:

$$F_{r\,min} = \frac{1}{z} \cdot \frac{m}{10^3} \cdot r \cdot \omega^2 = \frac{1}{z} \cdot \frac{G}{g} \cdot r \cdot \left(\frac{\pi \cdot n}{30}\right)^2$$

$$= \frac{1}{z} \cdot \frac{G_1}{g} \cdot (R - r) \cdot \left(\frac{\pi \cdot n}{30}\right)^2$$

Perpendicular to the direction of vibration, there is a significantly higher radial bearing load:

$$F_{r\,max} = \frac{1}{z} \cdot \frac{G_1}{g} \cdot R \cdot \left(\frac{\pi \cdot n}{30}\right)^2$$

In contrast to a circle throw screen, in which the bearing load is constant, the bearing load in a straight line screen alternates twice during one revolution of the exciter shafts between $F_{r\,max}$ and $F_{r\,min}$. If the equation for calculating the minimum radial bearing load $F_{r\,min}$ with the equation for calculating the radial bearing load F_r, it can be seen that the minimum radial bearing load of a screen with straight line motion is exactly the same as the radial bearing load of a comparable circle throw screen.

For a straight line screen with a load varying according to a sinusoidal function, the radial bearing load F_r can be determined using the following equation:

$$F_r = 0{,}68 \cdot F_{r\,max} + 0{,}32 \cdot F_{r\,min}$$

Whereas the bearing load in a circle throw screen can be determined simply from data for the screen box screen box weight G, the vibration radius r and the speed n, these data only allow calculation of the minimum bearing load in a straight line screen. For more precise calculation, it is also necessary to know either the exciter weight G_1 or the distance R between the centres of gravity of the exciters and their bearing axes.

The missing value can be determined using the following equation:

$$G \cdot r = G_1 \cdot (R - r)$$

FAG special spherical roller bearings for vibratory machinery

Legend

$F_{r\,min}, F_{r\,max}, F_r$ kN
Radial bearing load

m   kg
Screen box mass

r   m
Vibration radius

ω 1/s
Angular velocity

G   kN
Screen box weight

n min^{-1}
Speed

z   –
Number of bearings

g m/s^2
Gravitational acceleration (g = 9,81 m/s^2)

G_1 kN
Exciter weight

R   m
Distance between the centres of gravity of the exciters and the corresponding bearing axes.

Example

Data for calculation:

- ▪ Screen box weight   G   = 33 kN
- ▪ Exciter weight G_1 = 7,5 kN
- ▪ Vibration radius   r   = 0,008 m
- ▪ Speed   n   = 900 min^{-1}
- ▪ Number of bearings   z   = 4.

The data can be used to determine the distance between the centres of gravity of the exciters and the corresponding bearing axes:

$$R = \frac{r \cdot (G + G_1)}{G_1} = \frac{0,008 \cdot (33 + 7,5)}{7,5} = 0,0432 \, m$$

Based on the equation for calculating the minimum radial bearing load $F_{r\,min}$ and the equation for calculating the maximum radial bearing load $F_{r\,max}$, this gives:

$$F_{r\,min} = \frac{1}{4} \cdot \frac{33}{9,81} \cdot 0,008 \cdot \left(\frac{\pi \cdot 900}{30}\right)^2 = 59,8 \, kN$$

$$F_{r\,max} = \frac{1}{4} \cdot \frac{7,5}{9,81} \cdot 0,0432 \cdot \left(\frac{\pi \cdot 900}{30}\right)^2 = 73,3 \, kN$$

Radial bearing load:

$$F_r = 0,68 \cdot 73,3 + 0,32 \cdot 59,8 = 69 \, kN$$

The equivalent dynamic bearing load required in order to determine the requisite basic dynamic load rating of the bearing is then:

$$P = 1,2 \cdot 69 = 83 \, kN$$

Eccentric screen

In contrast to a two bearing screen, the vibration radius of an eccentric screen is a function of the eccentricity of the shaft, *Figure 7*.

G = screen box weight
G_2 = weight of counterweights
r = eccentric radius of crankshaft

① Spherical roller bearing, normal design
② FAG special spherical roller bearing (design T41A)

Figure 7
Schematic of an eccentric screen
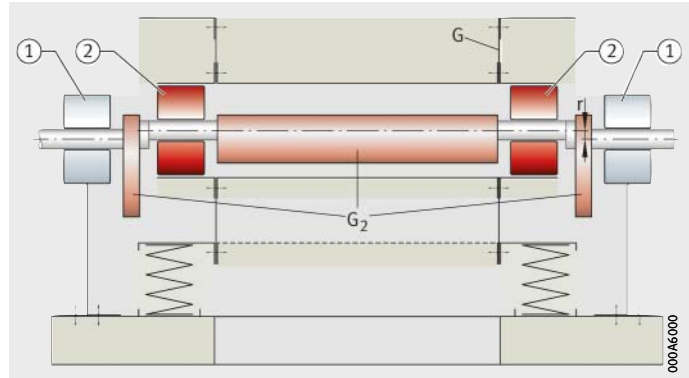
The bearing load for the two inner bearings is determined using the same equation as for the circle throw screen:

$$F_r = \frac{1}{z} \cdot \frac{G}{g} \cdot R \cdot \left(\frac{\pi \cdot n}{30} \right)^2$$

F_r kN
Radial bearing load

G   kN
Screen box weight

R   m
Distance between the centres of gravity of the exciters and the corresponding bearing axes

n min^{-1}
Speed

z   –
Number of bearings

g m/s^2
Gravitational acceleration (g = 9,81 m/s^2).

The influence of the support springs on the loading of the inner bearings can be regarded as negligible. The outer bearings of the eccentric screen are only lightly loaded, since the centrifugal force of the screen box during idling is compensated by the weight of the counterweights G_2. The load on these bearings is not constant; it varies according to a sinusoidal pattern due to the support springs on the screen box. In operation, the material in the box interferes with the balanced condition of the machine. This places additional load on the outer bearings. However, this additional load is very small. The selection of bearings is based on the shaft diameter. This results in bearings whose load carrying capacity is so high that a fatigue life calculation is unnecessary. Since these bearings do not undergo vibration, spherical roller bearings of the standard design are sufficient.

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Example   Data for calculation:
- ▣ Screen box weight G = 60 kN
- ▣ Eccentric radius r = 0,005 m
- ▣ Speed n = 850 min^{-1}
- ▣ Number of bearings z = 2.

From the data, the radial bearing load of the inner bearings is calculated as follows:

$$F_r = \frac{1}{2} \cdot \frac{60}{9,81} \cdot 0,005 \cdot \left(\frac{\pi \cdot 850}{30} \right)^2 = 121 \, kN$$

The equivalent dynamic bearing load required in order to determine the requisite basic dynamic load rating of the bearing is then:

$$P = 1,2 \cdot 121 = 145 \, kN$$

Nomogram for calculation of centrifugal force

Calculation of the centrifugal force of the imbalance masses or the centrifugal force of the screen box mass can be carried out in graphical form with the aid of a diagram, the so-called centrifugal force nomogram, *Figure 8*, page 24.

The centrifugal forces are determined according to the following equation:

$$F_{max} = \frac{G_1}{g} \cdot R \cdot \left(\frac{\pi \cdot n}{30} \right)^2$$

$$F_{min} = \frac{G_1}{g} \cdot (R - r) \cdot \left(\frac{\pi \cdot n}{30} \right)^2$$

$$F = \frac{G}{g} \cdot r \cdot \left(\frac{\pi \cdot n}{30} \right)^2$$

F_{max}, F_{min}, F kN
Centrifugal forces
G_1 kN
Imbalance mass weight
R m
Distance between centre of gravity of exciter and bearing axis
n min^{-1}
Speed
g m/s^2
Gravitational acceleration ($g = 9{,}81 \text{ m/s}^2$)
r m
Vibration radius
b m/s^2
Acceleration, *Figure 8*, page 24.

n [min^{-1}]	G$_1$, G [kN]	b [m/s^2]	F$_{max}$, F$_{min}$, F [kN]	R, R−r, r [mm]

① n = 1 200 min^{-1} ④ G = 35 kN
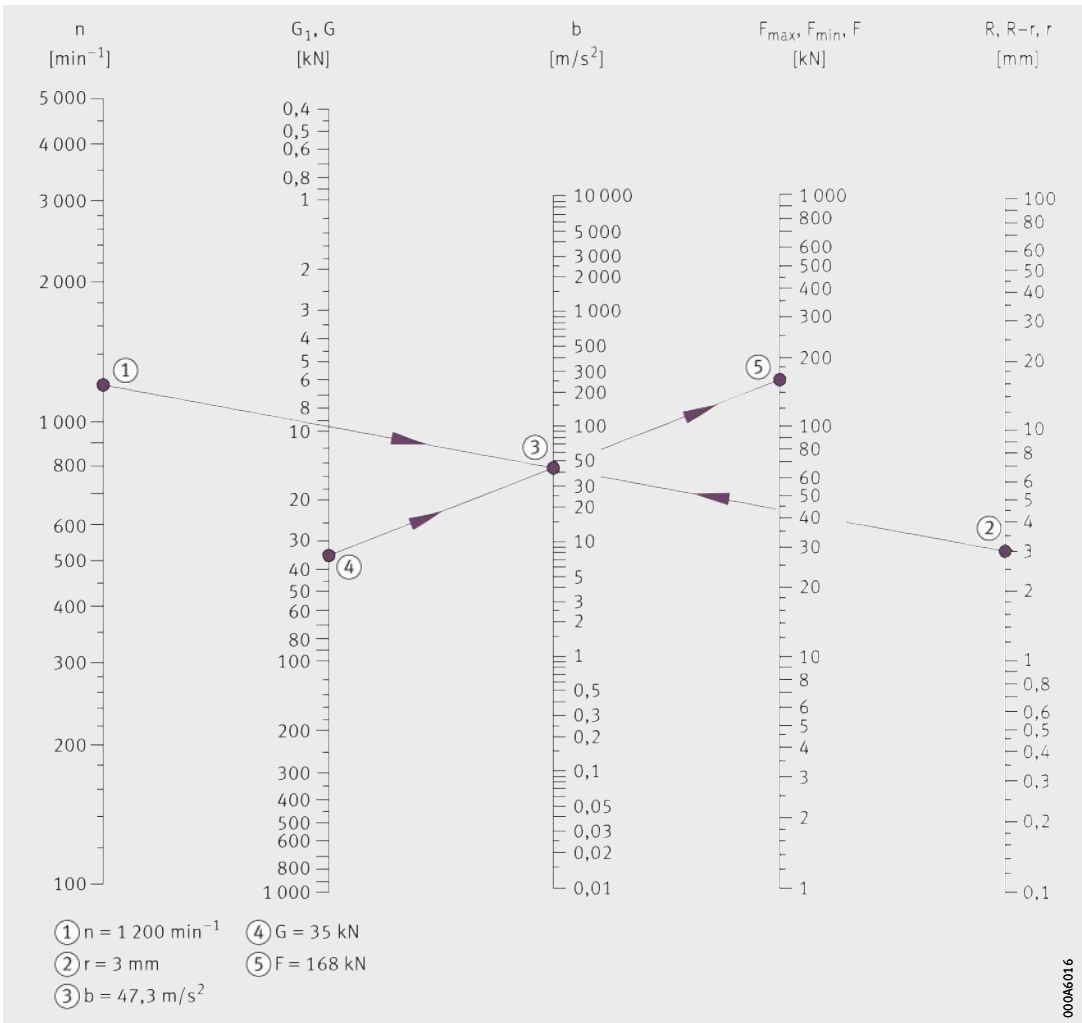② r = 3 mm   ⑤ F = 168 kN
③ b = 47,3 m/s^2

Figure 8
Nomogram for calculation of centrifugal force

Nomogram for calculation of basic load ratings

Calculation of the basic dynamic load rating C can also be carried out in graphical form with the aid of a diagram, the so-called basic load rating nomogram, *Figure 9*, page 26.

The following are required in order to calculate the basic dynamic load rating C:

- speed n in min^{-1}
- basic rating life L_h in h
- equivalent dynamic load P in kN.

In two bearing screens with circle throw and inner bearings with eccentric screens, the following applies:

$$P = 1,2 \cdot \frac{F}{z}$$

In two bearing screens with straight line motion, the following applies:

$$P = 1,2 \cdot \left(\frac{0,68 \cdot F_{max} + 0,32 \cdot F_{min}}{z} \right)$$

where:

- 1,2 = safety factor
- z   = number of bearings
- F   = centrifugal force.

n [min⁻¹]	C/P	Lₕ [h]	C [kN]	P [kN]

① n = 1 200 min⁻¹   ④ P = 100 kN
② Lₕ = 10 000 h   ⑤ C = 720 kN
③ C/P = 7,2
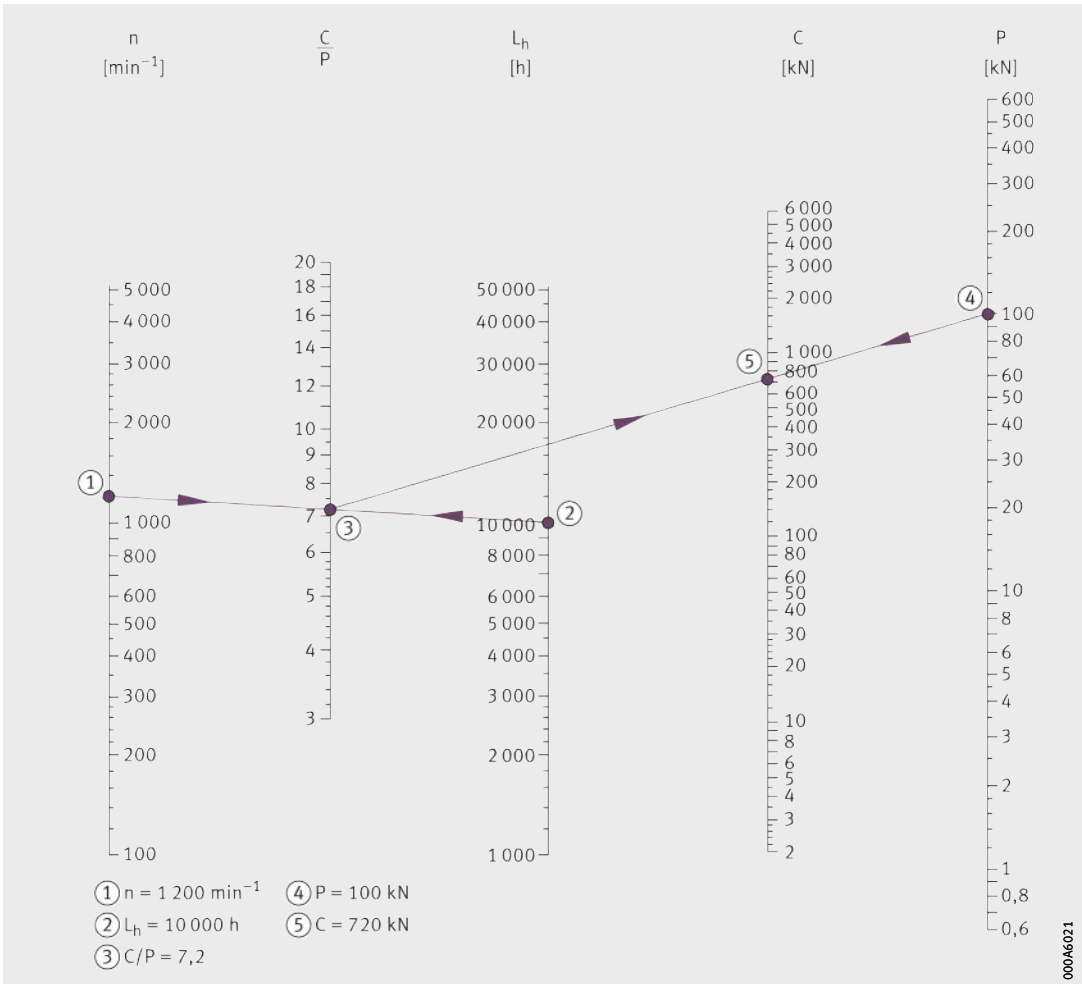
Figure 9
Nomogram for calculation
of basic load ratings

Design of bearing arrangements

The design of bearing positions is illustrated below by means of several examples.

Two bearing screen with circle throw

This section describes the bearing arrangements of two bearing screens with grease lubrication, oil sump lubrication and recirculating oil lubrication.

Grease lubrication

The schematic design of the bearing arrangement in a two bearing screen with circle throw and grease lubrication is shown in *Figure 10*. The imbalance shaft is supported in two FAG special spherical roller bearings 223..-E1-XL-T41A. The bearing on the drive side is fitted as a locating bearing while the opposing bearing is a non-locating bearing.

① Locating bearing
② Non-locating bearing
③ Flange of guard tube
④ Grease baffle
⑤ Grease collector pocket

Figure 10
Two bearing screen with circle throw
(grease lubrication)

000A6048
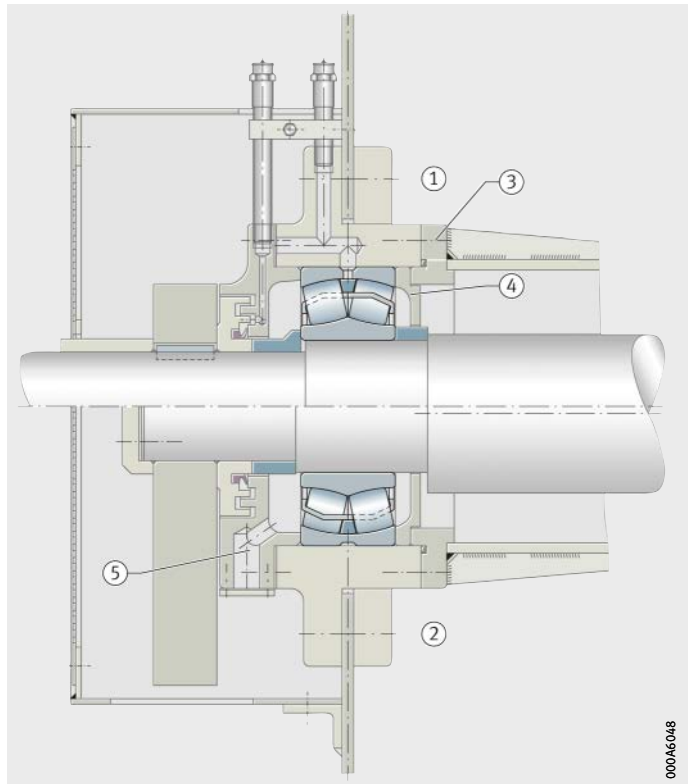
FAG special spherical roller bearings for vibratory machinery

After inspection of the adjacent parts, the bearing is then mounted in the housing bore. Smaller bearings can be pressed in while cold. For larger bearings, the housing is heated uniformly to the point where the interference between the bearing outer ring and housing bore is eliminated. As the housing cools down, the interference fit is achieved. The bearing and housing are then slid onto the shaft. For dismounting, it is easier to press the bearing out of the housing if the guard tube flange is replaced by a screw mounted ring, *Figure 10*, ①, page 27, with several extraction screws on its circumference.

A favourable option is to feed the grease as shown here via the circumferential groove and the lubrication holes in the bearing outer ring. In this way, the fresh grease is fed directly to the rolling and sliding surfaces of the rolling bearing, ensuring uniform lubrication of both rows of rollers. The fresh grease displaces the old, possibly contaminated grease from the interior of the bearing. On the inner side of the bearing arrangement, the old grease escapes via the gap in the grease baffle and collects in the guard tube. On the outer side, it collects at the grease collector pocket, from which it is periodically removed. The bearing is sealed against external influences by a labyrinth that can be relubricated and whose sealing action can be further increased by a V ring on the innermost labyrinth passage.

Oil sump lubrication

The schematic design of the bearing arrangement of a two bearing screen with circle throw and oil sump lubrication is shown in *Figure 11*. Sealing against the ingress of contaminant from outside is provided by a labyrinth filled with grease that can be relubricated. Egress of oil is prevented by a splash ring with an oil collector groove. On the bearing side, the sealing area is shielded by a flinger ring. In order to prevent the grease in the labyrinth entering the oil cavities, a V ring is fitted between the labyrinth and splash ring. The connecting hole in the lower section of the housing equalises the oil sump level between the two sides of the bearing. The oil level should be such that the lowest roller in the bearing is immersed to approximately half its diameter in oil when the bearing is stationary. At this level, there is an overflow hole that is closed off after the housing is filled. The oil outlet screw contains a small permanent magnet that draws wear particles out of the oil. In general, the shaft guard tube is used as an additional oil container.

① Locating bearing
② Non-locating bearing
③ Vent screw
④ Flinger ring
⑤ Oil overflow hole
⑥ Connecting hole
⑦ Oil outlet screw
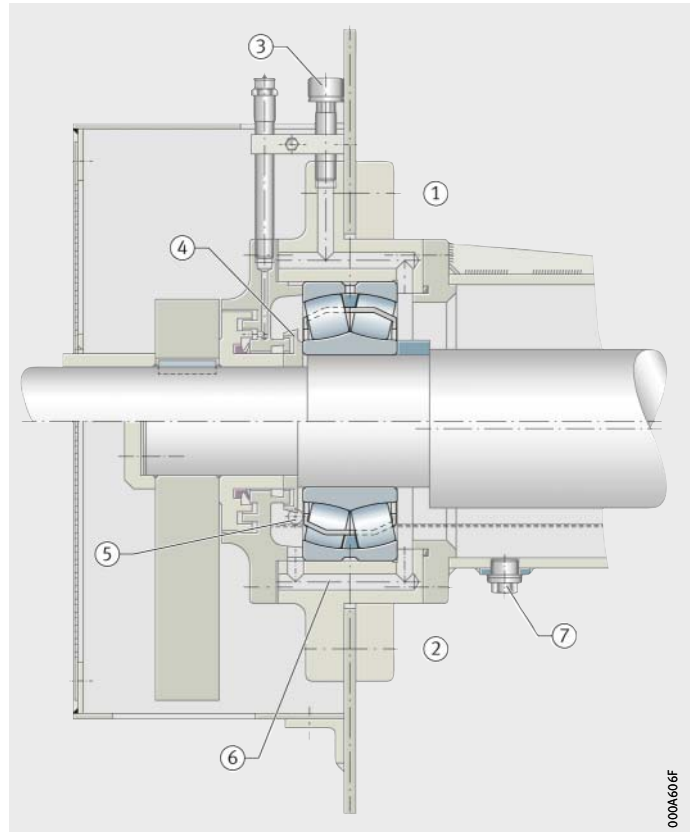
Figure 11
Two bearing screen with circle throw
(oil sump lubrication)

000A606F

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Recirculating oil lubrication

The design of the bearing arrangement with recirculating oil lubrication shown in *Figure 12* is similar to that of the bearing arrangement with oil sump lubrication, see page 29.
The connecting hole in the lower section of the housing equalises the oil level between the two sides of the bearing.

The sealing arrangement is taken from the oil sump lubrication.
The oil outlet hole is located at such a level that, even if the oil feed is interrupted, there is still an emergency oil reserve available.
The oil is fed via the lubrication groove and lubrication holes in the bearing outer ring. Oil filtration is absolutely essential, see page 41.

① Locating bearing
② Non-locating bearing

Figure 12
Two bearing screen with circle throw
(recirculating oil lubrication)

Two bearing screen with straight line motion
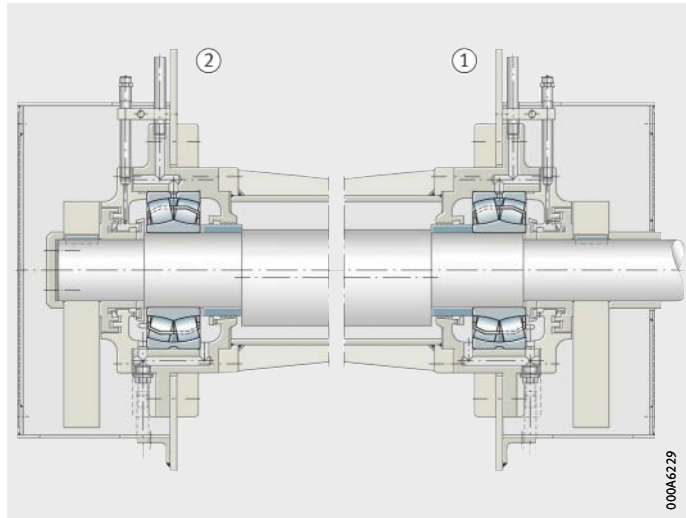
This section describes the bearing arrangement of a two bearing screen with oil injection lubrication.

Oil injection lubrication

The bearing arrangement of an exciter for a two bearing screen with straight line motion is shown in *Figure 13*. The two contra-rotating, synchronously geared imbalance shafts are fitted with FAG special spherical roller bearings 223..-E1-XL-T41A. The bearings on the gear side are fitted as locating bearings in order to prevent disruption to the gear cycling behaviour if length variations occur (temperature differences).

The bearings are lubricated by the oil thrown off by the gears and a flinger shield. The baffle plates on the lower halves of the housing end faces ensure that the oil level reaches approximately the centre of the lowest roller in the bearings.

The passage for the drive shaft is equipped with a splash ring seal and, in order to prevent ingress of contamination, with a labyrinth. A V ring can also be fitted between the labyrinth and splash ring. The oil level is just high enough that the lower gear and flinger shield are immersed in the oil sump. The oil level is monitored by lateral oil level indicators.

① Locating bearing
② Non-locating bearing
③ Baffle plates
④ Oil level indicator

Figure 13
Two bearing screen
with straight line motion
(oil splash lubrication)
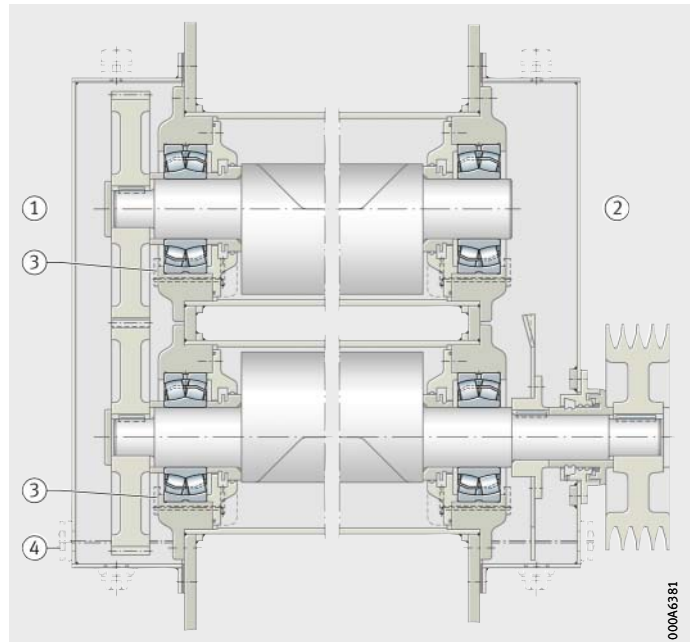
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Four bearing screen

This section describes the bearing arrangement of a four bearing screen with grease lubrication.

Grease lubrication

The eccentric shaft of a four bearing screen is shown in *Figure 14*, page 33. Since the stresses acting on the inner bearings are comparable with those acting on the bearings of a two bearing screen, these positions are fitted with FAG special spherical roller bearings of series 223..-E1-XL-T41A.

Although the interaction of the rotating screen box centrifugal force and the directionally constant spring forces does not give a pure point load on the inner ring, the fits selected are generally the same as for the two bearing screen.

The outer rings have a P6 fit in the housing, while the inner rings have a f6 or g6 fit on the shaft. One of the two inner bearings is fitted as a locating bearing, while the other is a non-locating bearing with an inner ring that can be displaced along the shaft.
In all other respects, the design of the inner bearing arrangement shown is identical to the bearing arrangement for a two bearing screen with grease lubrication.

Conditions are different in the outer bearings. In order to ensure that, if possible, imbalance forces are not transmitted to the foundations and the bearing load remains low, the imbalance moment of the screen box in the eccentric screen is compensated by means of counterweights. During idling, the outer bearings are only subjected to the forces exerted by the support springs. The support springs are preloaded to such an extent that the outer bearings are subjected to a sinusoidally pulsating but directionally constant radial load. Although the precisely balanced condition is disrupted during operation by the material in the box – the spring forces are overlaid by an uncompensated rotating centrifugal force – and the load direction may therefore vary within a certain angle, the bearing fits are determined on the assumption that the outer ring is subjected to point load.

A loose fit must therefore be selected for the outer rings in the housing bore. The inner rings are normally located on the shaft – as shown – using withdrawal sleeves. The bearing on the drive side is fitted as a locating bearing while the opposing bearing is a non-locating bearing with an outer ring capable of axial displacement.

Normal machining tolerances that have proved effective for the outer bearing seats are:

- shaft: H8/h9
  (shaft tolerance for withdrawal sleeve location)
- housing: H7.

Since the outer bearings do not undergo translation movement and are only subject to light loads, normal spherical roller bearings with a tapered bore and normal internal clearance can be selected.

① Locating bearing
② Non-locating bearing
③ Counterweight

Figure 14
Four bearing screen
(grease lubrication)
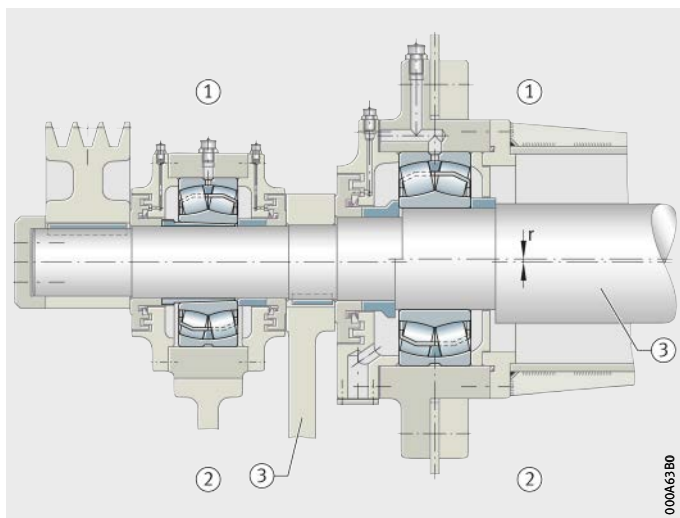
FAG special spherical roller bearings for vibratory machinery

Lubrication of bearings

Spherical roller bearings in vibratory machinery are subjected to very high operating loads and adverse environmental conditions.
The lubricant type, lubrication method and lubricant supply must be carefully selected and matched in order to fulfil the requirements for functional suitability and service life of the vibratory machinery bearings. Depending on the operating conditions, bearing size and particular requirements of the plant operator, lubrication using grease or oil can be selected.

Grease lubrication

In most vibratory machinery, the FAG special spherical roller bearings are lubricated using grease. Grease lubrication is normally used up to a speed parameter $n \cdot d_M = 300\,000$ $\text{min}^{-1} \cdot \text{mm}$ (n = operating speed, d_M = mean bearing diameter). Only greases that have been tested and proven should be used, see page 42. Any change of grease type should be avoided if possible.

For normal operating conditions in vibratory machinery, we recommend lithium soap greases with EP (extreme pressure) and anti-corrosion additives corresponding to penetration class 2. he minimum requirements described in DIN 51825 are not sufficient in this application. Instead, the suitability of greases for use in the rolling bearing must be demonstrated as is the case with, for example, the FAG rolling bearing greases Arcanol MULTITOP and LOAD400.

In applications with higher operating temperatures, for example in screens for hot materials or where the bearings may in special cases undergo considerable heating by the material in the box, special greases with high thermal stability should be used.

The base oil viscosity required is dependent on the operating conditions. The aim should be to achieve a viscosity ratio $\kappa = \nu/\nu_1 \geqq 2$. In this case, ν is the operating viscosity, ν_1 is the reference viscosity, see also Catalogue HR 1, Rolling Bearings. When rolling bearings are mounted, the internal cavities of the bearings must be filled with grease, see table, page 36. During the startup phase, the grease will distribute in the bearing and partly (approx. 30% of the grease quantity/bearing) fill the housing cavities V_G next to the bearing. These cavities may be filled with grease to a maximum of 50% once the grease distribution in the bearing is completed.

The grease fill quantity in the housing cavities H_G with a grease density (as is normal for most greases) of $\rho = 0{,}9$ g/cm^3 is:

$$H_G = \left(0{,}5 \cdot V_G\right) \cdot \rho - 0{,}3 \, \frac{\text{grease quantity}}{\text{bearing}}$$

H_G –
Grease fill quantity in housing cavity

V_G –
Volume of housing.

This prevents excessive working and the associated temperature development. The grease in the housing cavities acts as a reservoir and contributes to a longer operating life in accordance with the operating and the mounting conditions. It is recommended that relubrication should be carried out via the lubrication groove and the three lubrication holes that are present as standard in the outer ring of all FAG special spherical roller bearings.

This ensures uniform supply of lubricant to both rows of rollers. Where rolling bearings are relubricated from the side, the distance between the housing wall and the end face of the bearing should be as small as possible so that the grease can reach the bearing interior quickly and without losses. The grease outlet hole should be located on the opposite side of the bearing.

In bearing arrangements for vibratory machinery, it is advisable to relubricate the bearings with small quantities of grease at short intervals. The relubrication quantities are stated as a function of the bearing size and the speed, see table, page 36. These relubrication quantities relate to a relubrication interval of 50 operating hours and normal operating temperatures.

If continuous relubrication is carried out by means of a central lubricant supply system, the requisite grease quantity m_1 per hour and per bearing can be determined using the following equation:

$$m_1 = 0{,}00004 \cdot D \cdot B$$

m_1 g/h
Requisite grease quantity
D   mm
Bearing outside diameter
B   mm
Bearing width.

The labyrinth seals should be relubricated once per week, or more frequently if operating conditions are unfavourable (heavy exposure to dust, moisture, high operating temperature). The grease should be the same as that used in the rolling bearings.

FAG special spherical roller bearings for vibratory machinery

Initial greasing and relubrication quantities in g for spherical roller bearings 223 in vibratory machinery

Bore code	Initial grease quantity	Relubrication quantity at speed min⁻¹				
		500	600	700	800	900
08	16	5	5	5	5	5
09	22	5	5	5	5	5
10	27	5	5	5	5	5
11	43	5	5	5	5	5
12	50	5	5	5	5	5
13	56	5	5	5	5	5
14	76	5	5	5	5	5
15	91	5	5	5	5	5
16	100	5	5	5	10	10
17	130	5	5	10	10	10
18	145	10	10	10	10	10
19	180	10	10	10	10	10
20	185	10	10	10	10	15
22	270	10	10	15	15	20
24	330	15	15	20	25	30
26	420	15	20	20	25	35
28	525	20	25	30	35	45
30	630	25	30	40	50	65
32	725	25	35	45	60	80
34	870	30	40	55	80	110
36	1 000	35	50	65	90	120
38	1 200	45	65	90	130	–
40	1 400	50	70	100	150	–
44	1 700	70	105	160	–	–

1 000	1 200	1 400	1 600	1 800	2 000	2 200	2 400	2 600	2 800	3 000	3 200
5	5	5	5	5	5	5	5	5	5	5	5
5	5	5	5	5	5	5	5	5	5	10	10
5	5	5	5	5	5	5	10	10	10	10	15
5	5	5	5	5	10	10	10	10	15	15	20
5	5	5	5	10	10	10	10	15	15	–	–
5	5	10	10	10	10	15	15	20	–	–	–
5	10	10	10	15	15	20	25	–	–	–	–
5	10	10	10	15	20	25	–	–	–	–	–
10	10	10	15	20	25	–	–	–	–	–	–
10	10	15	20	25	35	–	–	–	–	–	–
10	15	20	25	30	40	–	–	–	–	–	–
15	15	25	35	45	–	–	–	–	–	–	–
15	20	30	40	–	–	–	–	–	–	–	–
20	30	50	70	–	–	–	–	–	–	–	–
35	55	85	–	–	–	–	–	–	–	–	–
40	65	–	–	–	–	–	–	–	–	–	–
60	100	–	–	–	–	–	–	–	–	–	–
90	–	–	–	–	–	–	–	–	–	–	–
100	–	–	–	–	–	–	–	–	–	–	–
140	–	–	–	–	–	–	–	–	–	–	–
–	–	–	–	–	–	–	–	–	–	–	–
–	–	–	–	–	–	–	–	–	–	–	–
–	–	–	–	–	–	–	–	–	–	–	–
–	–	–	–	–	–	–	–	–	–	–	–

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Oil lubrication

If the speeds are above the normal range for grease lubrication, in other words with a speed parameter $n \cdot d_M > 300\,000$ min$^{-1} \cdot$ mm, oil lubrication must be provided. Oil lubrication may also be necessary if there is heating by external sources or for reasons of maintenance. For lubrication of bearings, we recommend mineral oils or synthetic oils with EP (extreme pressure) and anti-corrosion additives, see page 42. Good quality multi-grade oils can also be used. The viscosity ratio should be $\kappa = \nu/\nu_1 \geqq 2$.

Oil sump lubrication (bath lubrication)

Oil sump lubrication is normally used up to a speed parameter $n \cdot d_M = 300\,000$ min$^{-1} \cdot$ mm; with frequent oil changes, it can be used up to $n \cdot d_M = 500\,000$ min$^{-1} \cdot$ mm. In this lubrication method, the lubricant is conveyed to the rolling contact points by any gears, the imbalance mass or the rolling elements themselves.
The oil level in the machine or bearing housing must be sufficiently high that the gears or imbalance masses are dipped in the oil and create a swirling effect. When the bearing is stationary, the lowest roller must be half immersed in the oil, *Figure 15* und table, page 39.

① Normal oil level
② Lowest oil level
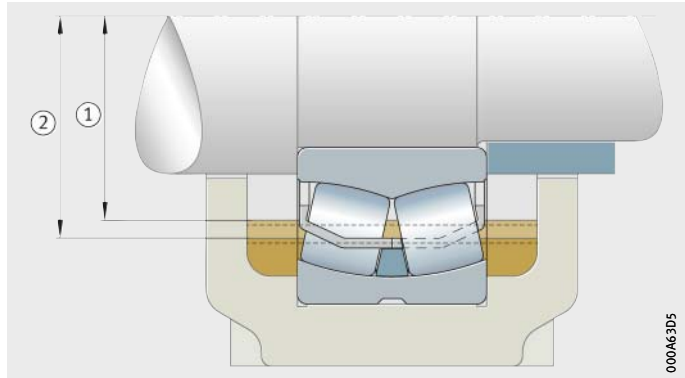
Figure 15
Determining the oil level
at standstill

Oil level at standstill

Bore code	Bearing series 223 Oil level	
	Normal mm	Low mm
08	31	34
09	35	38
10	39	42
11	42	46
12	46	50
13	50	54
14	54	59
15	58	62
16	62	67
17	66	71
18	69	74
19	72	78
20	78	84
22	86	94
24	93	101
26	100	109
28	107	117
30	115	125
32	122	133
34	129	140
36	137	149
38	144	156
40	152	165
44	168	182
48	182	195
52	196	211
56	212	228

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A sufficiently large oil quantity will extend the oil change interval. If the cavities in the housings are not sufficient, the shaft guard tube between the bearings can also be used as an oil reservoir or an additional container can be provided. The oil change interval is dependent on the contamination and the ageing condition of the oil.

Guide values for the oil quantity and oil change intervals as a function of the bearing bore are given in *Figure 16*.

We recommend regular oil inspection, since the results of such inspections will allow more precise determination of oil change intervals.

d = bearing bore
V = oil quantity

① Oil change interval
2 months to 3 months
② Oil change interval
10 months to 12 months

Figure 16
Oil quantity and oil change interval
as a function
of bearing bore diameter
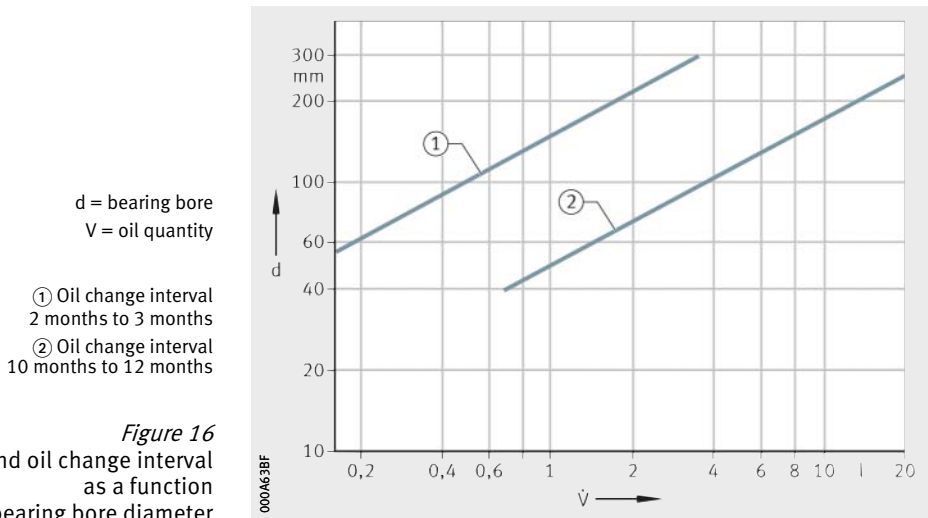
Further information ◾ TPI 176, Lubrication of Rolling Bearings.

Recirculating oil lubrication

If the speed parameter is higher than the permissible value for bath lubrication or where special conditions apply (increased heat dissipation required, insufficiently large oil cavities), recirculating oil lubrication must be used. The oil should be fed via the lubrication groove and lubrication holes in the bearing outer ring. Guide values for normal oil flow rates can be taken from *Figure 17*. In order to prevent oil backing up in the lubrication system, the cross-sections of the unpressurised return ducts must be adapted to the cross-sections of the feed ducts (4 to 5 times larger). In recirculating oil lubrication, it is absolutely essential that a filter is provided for retaining wear particles and contaminants in order to prevent impairment of the bearing operating life. Through evaluation of regular oil inspections, the oil change intervals can be matched more accurately to the operating conditions.

$\dot V$ = oil flow quantity
D = bearing outside diameter

Figure 17
Minimum oil flow rate
for spherical roller bearings
of series 223 in vibratory machinery
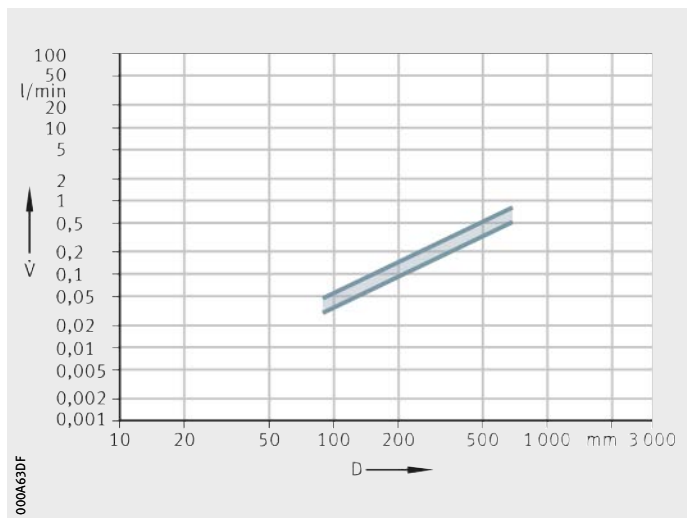
FAG special spherical roller bearings for vibratory machinery

Recommended lubricants

This section describes greases and oils for vibrating screen bearing arrangements.

Greases for vibrating screen bearing arrangements

Each delivery of the FAG rolling bearing greases Arcanol is subjected to comprehensive quality inspection. The quality of each batch can be clearly demonstrated and identified.

Greases for normal temperatures:
- Arcanol MULTITOP
- Arcanol LOAD400
- Arcanol LOAD220
- Arcanol VIB3.

Greases for high temperatures:
- Arcanol TEMP120.

In the case of greases that have not been subjected to our incoming goods inspection, we cannot make any statements regarding batch fluctuations, formulation changes or production influences.

Oils for vibrating screen bearing arrangements

If oils are to be used for this application, it must be demonstrated that the additives package is effective in the rolling bearing.
In principle, it is possible to use mineral oils and synthetic oils, with the exception of silicone oils. It is not advisable to use oils with viscosity index improvement agents.

Monitoring of vibrating screens

Vibration diagnosis is a reliable method for identifying the start of machine damage at an early stage. Schaeffler offers monitoring systems that can also be used under very aggressive environmental conditions.

Monitoring systems for vibrating screens detect machinery damage at a very early stage. Since the alarm is raised at an early stage, they help to prevent unplanned downtime. Measurement data can be retrieved on site or remotely at any time and assessed by the actual customer or by the Schaeffler Remote Service Center, *Figure 18* and *Figure 19*, page 44.

By monitoring the vibrating screens, the following system conditions, for example, can be identified:

- bearing damage
- loose parts
- broken springs
- overloads
- impact/contact.

Furthermore, the Schaeffler portfolio in the area of maintenance and quality assurance includes other products and services: from mounting, through plant monitoring to the introduction and implementation of preventive maintenance activities.

A wide range of mounting and alignment tools, measuring instruments and lubricants, as well as training courses makes maintenance work easier and work processes more efficient. Based on many years' experience and qualified experts, Schaeffler is the competent partner for customer-oriented solutions relating to the life cycle of rolling bearings.

FAG special spherical roller bearings for vibratory machinery

Figure 18
Vibrating screen

Figure 19
Schaeffler Remote Service Center
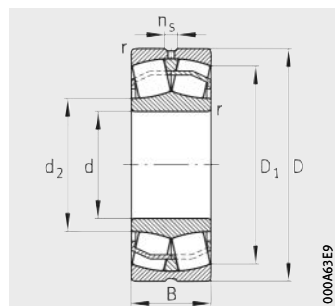
Further information
- www.schaeffler.com/services
- Or send an e-mail to
  industrial-services@schaeffler.com.

FAG special spherical roller bearings for vibratory machinery

With cylindrical bore
Series 223..-E1-XL-T41A(D)
Series 223..-BE-XL-JPA-T41A

X-life

Dimensions

Dimension table · Dimensions in mm									
Designation	Mass	Dimensions							
	m	d	D	B	r	n_s	D_1	d_2	
	≈ kg				min.		≈	≈	
22308-E1-XL-T41A	1,05	**40**	90	33	1,5	6,5	76	52,4	
22309-E1-XL-T41A	1,39	**45**	100	36	1,5	6,5	84,7	58,9	
22310-E1-XL-T41A	1,9	**50**	110	40	2	6,5	92,6	63	
22311-E1-XL-T41A	2,27	**55**	120	43	2	6,5	101,4	68,9	
22312-E1-XL-T41A	2,97	**60**	130	46	2,1	6,5	110,1	74,8	
22313-E1-XL-T41A	3,57	**65**	140	48	2,1	9,5	119,3	83,2	
22314-E1-XL-T41A	4,21	**70**	150	51	2,1	9,5	128	86,7	
22315-E1-XL-T41A	5,38	**75**	160	55	2,1	9,5	136,3	92,4	
22316-E1-XL-T41A	6,27	**80**	170	58	2,1	9,5	145,1	98,3	
22317-E1-XL-T41D	7,06	**85**	180	60	3	9,5	154,2	104,4	
22318-E1-XL-T41D	8,69	**90**	190	64	3	12,2	162,5	110,2	
22319-E1-XL-T41D	9,69	**95**	200	67	3	12,2	171,2	116	
22320-E1-XL-T41D	13,1	**100**	215	73	3	12,2	184,7	130,2	
22322-E1-XL-T41D	17,7	**110**	240	80	3	15	204,9	143,1	
22324-E1-XL-T41D	22,3	**120**	260	86	3	15	222,4	150,8	
22326-E1-XL-T41D	28	**130**	280	93	4	17,7	239,5	162,2	
22328-E1-XL-T41D	34,6	**140**	300	102	4	17,7	255,7	173,5	
22330-E1-XL-T41D	42,2	**150**	320	108	4	17,7	273,2	185,3	
22332-BE-XL-JPA-T41A	48,4	**160**	340	114	4	17,7	286,7	201,2	
22334-BE-XL-JPA-T41A	58,2	**170**	360	120	4	17,7	303,9	213,1	
22336-BE-XL-JPA-T41A	68,1	**180**	380	126	4	23,5	320,8	224,9	
22338-BE-XL-JPA-T41A	78,9	**190**	400	132	5	23,5	338,1	236,8	
22340-BE-XL-JPA-T41A	89,4	**200**	420	138	5	23,5	355,1	248,8	
22344-BE-XL-JPA-T41A	117	**220**	460	145	5	23,5	391,1	273,4	

Basic load ratings		Fatigue limit load	Limiting speed	Reference speed
dyn. C_r	stat. C_{0r}	C_{ur}	n_G	n_B
kN	kN	kN	min^{-1}	min^{-1}
156	149	13,1	7 600	5 500
187	183	16	6 800	5 000
229	223	20,1	6 300	4 800
265	260	23,9	5 800	4 500
310	310	28	5 400	4 200
350	365	32,5	5 000	3 800
390	390	36,5	4 800	3 700
445	450	40,5	4 500	3 550
495	510	45	4 250	3 400
540	560	49,5	4 100	3 200
610	630	55	3 850	3 000
670	700	59	3 700	2 800
810	920	75	3 300	2 380
950	1 070	90	3 000	2 130
1 080	1 170	102	2 850	2 000
1 250	1 370	116	2 650	1 820
1 460	1 630	131	2 420	1 660
1 640	1 850	147	2 290	1 520
1 680	1 990	157	2 250	1 420
1 870	2 220	173	2 130	1 320
2 060	2 460	190	2 030	1 230
2 220	2 650	207	1 940	1 160
2 440	2 950	225	1 830	1 080
2 800	3 400	265	1 690	950

Query for bearing calculation

(drawing attached: ❑ yes/❑ no)

Type (The schematics can be found in the section Two bearing screen with circle throw, page 15, to the section Eccentric screen, page 21)

❑ Circular screen ❑ Linear screen ❑ Eccentric screen

Load duty cycle/design data

Masses	kg	Screen box	m	_____
		Counterweights (exciter)	m_1	_____
Geometry	m	Vibration radius	r	_____
		Distance between centre of gravity of the exciter/bearing axis	R	_____
Speed	min^{-1}	Operating speed	n	_____
Operating time		Hours per day		_____
		One-shift or multi-shift operation		_____
Life	h	Minimum life required		_____

Bearing position/mounting data

Mounting location   ❑ Non-locating bearing   ❑ Locating bearing   ❑ Floating bearing arrangement

Bearing seat   ❑ Cylindrical   ❑ Conical

Number of rolling bearings z _____

Bearing seat diameter	mm	Shaft	_____	Fit	_____
		Housing	_____	Fit	_____
Other dimensions	mm	Maximum diameter		_____
		Minimum diameter		_____
		Width		_____

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Query for bearing calculation

(drawing attached: ❏ yes/❏ no)

Environmental influence

Ambient temperature	°C	
Material in box **(screen for hot materials)**	°C	
Operating temperature	°C	
Humidity	%	

Dust	❏ Heavy	❏ Moderate	❏ Little/none
Chemical influences	❏ Yes	❏ No	
Installation area	❏ Hall	❏ Outdoors	

Lubrication

❏ Grease	❏ Oil sump	❏ Oil recirculation	❏ Oil mist lubrication

Sealing

❏ Labyrinths that can be relubricated with V ring seals

❏ Oil splash rings and oil collector grooves

Other guidelines

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Every care has been taken to ensure the
correctness of the information contained
in this publication but no liability can
be accepted for any errors or omissions.
We reserve the right to make technical
changes.