# Politecnico di Torino

## Master Course in Automotive Engineering

## Thesis

# Test rig design of the rolling bearing

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# Abstract

Bearings are one of most important parts in the rotating mechanisms, which qualities directly influence the performances of machine tools. According to statistics, there are 30 percent of failures in the rotating mechanism to be relative to bearings. So, we must strictly check the quality of bearings before they are sold.

Some specific characteristics are desired in an endurance test system to achieve the control requirements of a life test series. An individual test run takes a long time; therefore, the test machine must be capable of running unattended without experiencing variation in the applied test parameters such as load(s), speed, lubrication conditions, and operating temperature. The basic test system that could also be subject to fatigue, such as load–support bearings, shafts, and load linkages, must be many times stronger than the test bearings so that test runs can be completed with the fewest interruptions from extraneous causes. In addition, the test system must be easy to maintain and should be capable of operating reliably and efficiently for years to ensure long-term compatibility of test results. Design simplicity is a key ingredient in meeting all these demands.

In this thesis, we mainly analyze some typical endurance test configurations and compare the different layouts to find out the lower power use configuration. Finally, we will present some rules about distance and power consumption.

# Index

# 1. Introduction of the bearing tester

Conducting an endurance test series on full-scale bearings is expensive because numerous test samples are required to obtain a useful experimental life estimate. The identification of simpler, less costly, life testing methods has therefore been a long-standing objective. The use of elemental test configurations offers a potential solution to this need. In this approach, a test specimen that has a simplified geometry (e.g., flat was her, rod, or ball) is used, and RC (rolling contact) is developed at multiple test locations. The aim is to extrapolate the life data generated in an element test to a real bearing application, thus saving calendar time and cost as compared with life data generated using full-scale bearing tests. This objective has historically not been achieved, generally because all of the operating parameters influencing fatigue life of rolling–sliding contacts were not reduced to stresses; rather, as is shown in Chapter 8 from [1], they were evaluated as life factors. The only stress directly evaluated in both element and full-scale bearing endurance testing has been the Hertz or normal stress acting on the contact. Lubrication, contamination, surface topography, and material effects have been evaluated as life factors. To be able to extrapolate the life data derived from element testing to full-scale bearing life data, it is necessary to evaluate both data sets from the standpoint of applied and induced stresses as compared with material strength.

# 2. Bearing

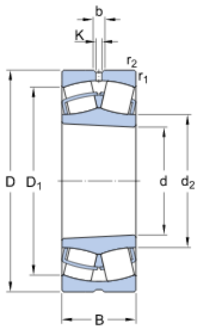
## 2.1. Selection of load and test bearings

We select three SKF rolling bearing as samples to analyze.

Reasons for choosing rolling bearing as test sample:

- Compared with ball bearings, rolling bearings can bear both axial and radial load
- Rolling bearings have higher radial load carrying capacity
- Accommodate axial displacement
- Specially used for vibratory applications

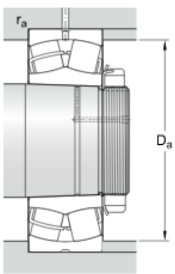
Dimensions




| | | | |
|---|---|---|---|
| d | | 140 | mm |
| D | | 250 | mm |
| B | | 68 | mm |
| $d_2$ | ≈ | 166 | mm |
| $D_1$ | ≈ | 216 | mm |
| b | | 11.1 | mm |
| K | | 6 | mm |
| $r_{1,2}$ | min. | 3 | mm |

Tapered bore, taper 1:12

Abutment dimensions



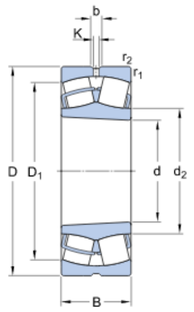

| | | | |
|---|---|---|---|
| $D_a$ | max. | 236 | mm |
| $r_a$ | max. | 2.5 | mm |

Figure 2.1: Load bearing (22228 CCK/W33)

## ▶ 23130 CCK/W33

Popular item
SKF Explorer

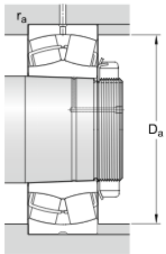
**Dimensions**




| | | | |
|---|---|---|---|
| d | | 150 | mm |
| D | | 250 | mm |
| B | | 80 | mm |
| $d_2$ | ≈ | 172 | mm |
| $D_1$ | ≈ | 216 | mm |
| b | | 11.1 | mm |
| K | | 6 | mm |
| $r_{1,2}$ | min. | 2.1 | mm |

Tapered bore, taper 1:12

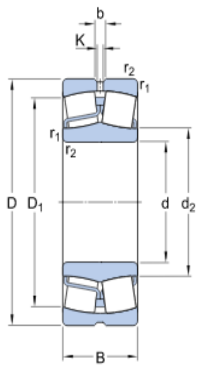
**Abutment dimensions**




| | | | |
|---|---|---|---|
| $D_a$ | max. | 238 | mm |
| $r_a$ | max. | 2 | mm |

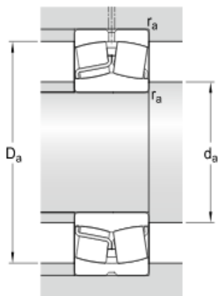
Figure 2.2: Load bearing (23130 CCK/W33)

**Dimensions**




| | | | |
|---|---|---|---|
| d | | 100 | mm |
| D | | 180 | mm |
| B | | 46 | mm |
| $d_2$ | ≈ | 118 | mm |
| $D_1$ | ≈ | 159 | mm |
| b | | 8.3 | mm |
| K | | 4.5 | mm |
| $r_{1,2}$ | min. | 2.1 | mm |

**Abutment dimensions**




| | | | |
|---|---|---|---|
| $d_a$ | min. | 112 | mm |
| $D_a$ | max. | 168 | mm |
| $r_a$ | max. | 2 | mm |

Figure 2.3: Test bearing (22220E)

### 2.2. Analysis of typical endurance test configurations

We consider four different layouts of test bearings among the five configurations shown in [2].

The first layout (Figure 2.4), two load bearings are distributed at both end of the shaft, while test bearing is at the middle of the shaft. Radial load and axial load are applied on the test bearing. The length of the shaft is 400 mm, the diameter of the shaft is 100 mm.

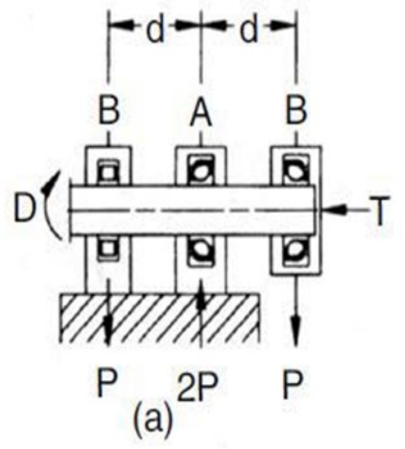



Figure 2.4: layout (a) from [2]

We use SKF calculator [3] to calculate the bearing power loss at the maximum load. It is very important to make sure test bearing is in a safety working condition. The rotating speed of shaft is constant 3000 rpm. Therefore, we design the radial load and axial load to make the test bearing in a safety working condition.

We design the radial load on the test bearing Fr = 100 kN, the axial load Fa = 10 kN. It is easy to get the load on the load bearing, radial load Fr = 50 kN, axial load Fa = 10 kN.

When using SKF calculator, we assume that operating temperature is 50 ℃, viscosity at 40 ℃ is 100 mm²/s. Power loss of test bearing (22220E) is 2700 W, while load bearing (22228 CCK/W33) is 3190 W.

The total power loss of these three bearings is

$P_{tot}$= 2700+3190×2 = 9080 W

We choose 36NiCrMo4 UNI EN 10083 as the shaft material (Rm =1000 MPa, $Rp_{0.2}$= 880MPa, $\sigma_{D-1}$= 450MPa). The surface roughness is 3.2 μm.

The fatigue limit of the component $\sigma^{C}_{D-1}$ (without notches) will be $\sigma^{C}_{D-1}=C_L C_S C_F \sigma_{D-1}$.

For the component loading mode effect $C_L$:

$C_L$=1, for plane bending

$C_L$=0.7, (experimentally 0.6÷0.8) for tension/compression

For the component scale effect $C_S$:

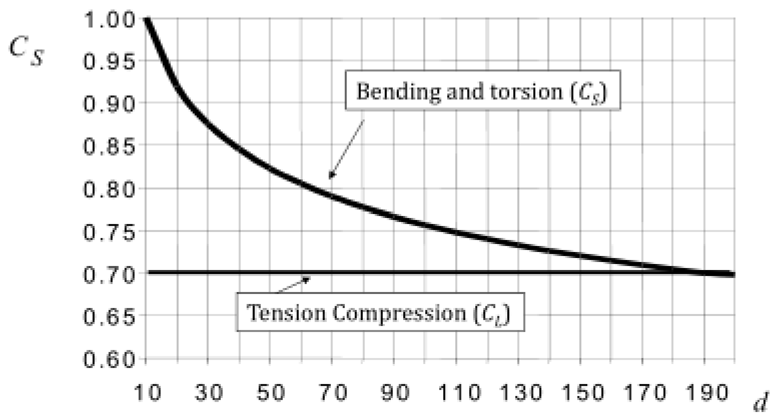



Figure 2.5: The Component Scale Effect from [5]

When the diameter of the shaft is 100 mm, the value of $C_S$ is 0.76. When the diameter of the shaft is 120 mm, the value of $C_S$ is 0.74. As is shown in figure 2.5.

For the component surface finish effect $C_F$:

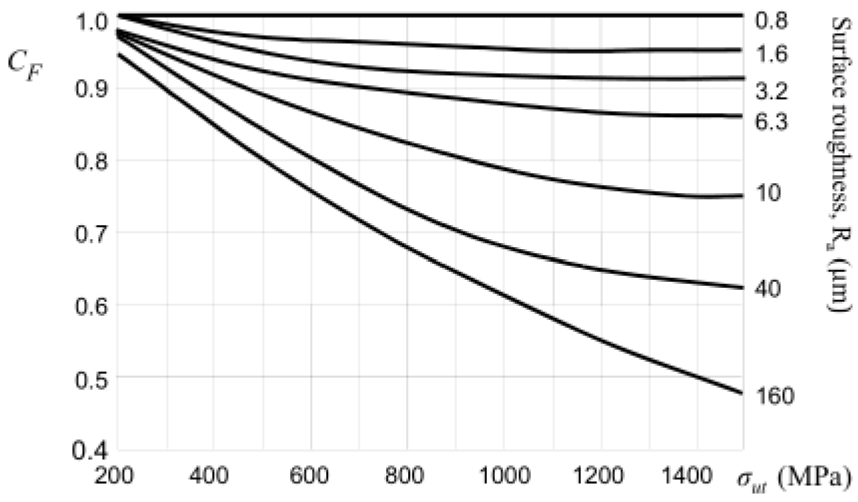



Figure 2.6: The Component Surface Finish Effect from [5]

(Rm=1000 MPa, the surface roughness Ra = 3.2 μm)

According to the diagram (figure 2.6), the value of $C_F$ is 0.92.

Figure 2.7 shows a diagram of torsion and bending moment distributed on the shaft and we choose section A as the dangerous section to calculate safety factor.

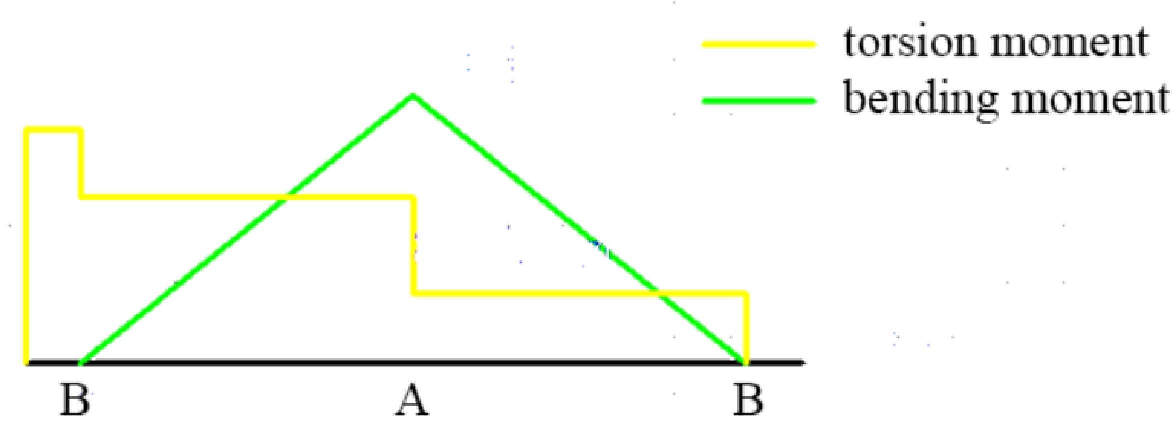



Figure 2.7: torsion and bending moment diagram of layout (a)

In section A, bending moment of inertia $W_F$:

$W_F = \frac{\pi D^3}{32} = \frac{\pi * 100^3}{32} = 98174.8\ mm^{-3}$

Torsion moment of inertia $W_T$:

$W_T = \frac{\pi D^3}{16} = \frac{\pi * 100^3}{16} = 196350\ Mm^{-3}$

Bending moment $M_F$=50*200=10000 Nm=$10^7$ Nmm

Torsion moment Mt=8602+6176.4=14778.4 Nmm

Normal load T=10*1000=10000 N

Stress of bending moment $\Sigma_{Mf} = \frac{Mf}{Wf}$ = 101.86 MPa

Stress of torsion moment $T_{Mt} = \frac{Mt}{Wt}$ =0.075 MPa

Normal stress $\Sigma_N = \frac{T}{\frac{\pi D^2}{4}}$=1.27 MPa

$\Sigma_a^{Mf}$=101.86 MPa

$\Sigma_m^{Mf}$=0

$T_a^{Mt}$=0

$T_m^{Mt}$=0.075 MPa

$\Sigma_a^N$=0

$\Sigma_m^N$=1.27 MPa

We consider normal load and torsion moment are constant. So, we just calculate $\Sigma_{D-1}^C$ Of bending moment:

$\Sigma_{D-1}^C$(Mf)=$\Sigma_{D-1} C_L C_S C_F$=314.6 MPa

Then we calculate the value of equivalent alternate and mean stress:

$$\Sigma_a^{eq}=\sqrt{(\sigma_a^{Mf}+\sigma_a^{N})^2+3*\mathrm{T}_a^{Mt\,2}}=101.86\text{ MPa}$$

$$\Sigma_m^{eq}=\sqrt{(\sigma_m^{Mf}+\sigma_m^{N})^2+3*\mathrm{T}_m^{Mt\,2}}=1.28\text{ MPa}$$

It is obvious that $\Sigma_m^{eq} << \Sigma_a^{eq}$, we can get safety factor:

$$SF=\frac{\Sigma_{D-1}^{C}}{\Sigma_a^{eq}}=3.09>1$$

The second layout (Figure 2.8), two load bearings are distributed at both end of the shaft, while test bearing is at the middle of the shaft. Just radial load is applied on the test bearing. The length of the shaft is 400 mm, the diameter of the shaft is 100 mm.

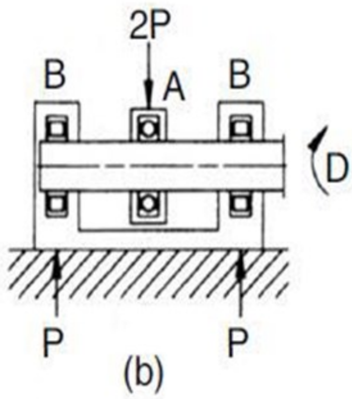



Figure 2.8: layout (b) from [2]

When using SKF calculator we keep rotating speed, operating temperature and viscosity constant.

We design the radial load on the test bearing Fr = 120 kN, the axial load Fa = 0 kN. It is easy to get the load on the load bearing, radial load Fr = 60 kN, axial load Fa = 0 kN.

Power loss of test bearing is 2690 W while load bearing is 2580 W.

The total power loss of these three bearings is

$P_{Tot}$=2690+2580×2=7850 W

The fatigue limit of the component $\Sigma^{C}_{D-1}$ (without notches) will be $\Sigma^{C}_{D-1}=C_L C_S C_F \Sigma_{D-1}$.

Figure 2.9 shows a diagram of torsion and bending moment distributed on the shaft and we choose section A as the dangerous section to calculate safety factor.

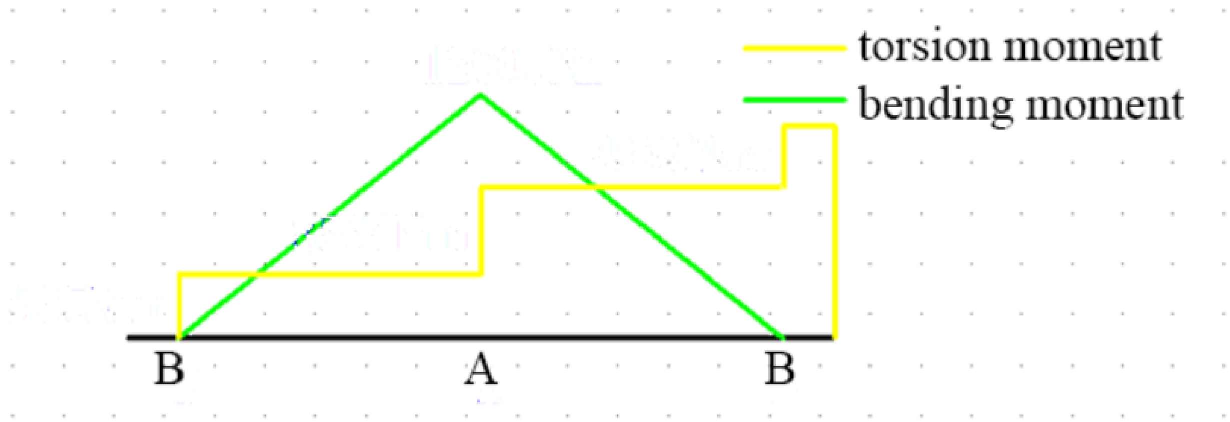



Figure 2.9: torsion and bending moment diagram of layout (b)

In section A, bending moment of inertia $W_F$:

$W_F = \frac{\pi D^3}{32} = \frac{\pi * 100^3}{32} = 98174.8 \text{ mm}^{-3}$

Torsion moment of inertia $W_T$:

$W_T = \frac{\pi D^3}{16} = \frac{\pi * 100^3}{16} = 196350 \text{ Mm}^{-3}$

Bending moment $M_F$ =60*200=12000 Nm=$1.2 \times 10^7$ Nmm

Torsion moment Mt=8564+4935.4=13499.4 Nmm

Normal load T=0 N

Stress of bending moment $\Sigma_{Mf} = \frac{Mf}{Wf}$ =122.23 MPa

Stress of torsion moment $\mathrm{T}_{Mt} = \frac{Mt}{Wt}$ =0.069 MPa

Normal stress $\Sigma_N = \frac{T}{\frac{\pi D^2}{4}}$=0 MPa

$\Sigma_a^{Mf}$=122.23 MPa

$\Sigma_m^{Mf}$=0

$\mathrm{T}_a^{Mt}$=0

$\mathrm{T}_m^{Mt}$=0.069 MPa

$\Sigma_a^N$=0

$\Sigma_m^N$=0 MPa

We consider normal load and torsion moment are constant. So, we just calculate $\Sigma_{D-1}^C$ Of bending moment:

$\Sigma_{D-1}^C$(Mf)=$\Sigma_{D-1} C_L C_S C_F$=314.6 MPa

Then we calculate the value of equivalent alternate and mean stress:

$$\Sigma_a^{eq}=\sqrt{\left(\sigma_a^{Mf}+\sigma_a^N\right)^2+3*\mathrm{T}_a^{Mt\,2}}=122.23 \text{ MPa}$$

$$\Sigma_m^{eq}=\sqrt{\left(\sigma_m^{Mf}+\sigma_m^N\right)^2+3*\mathrm{T}_m^{Mt\,2}}=0.12 \text{ MPa}$$

It is obvious that $\Sigma_m^{eq} << \Sigma_a^{eq}$, we can get safety factor:

$$SF=\frac{\Sigma_{D-1}^C}{\Sigma_a^{eq}}=2.57 > 1$$

The third layout (Figure 2.10), two test bearings are distributed at both end of the shaft, while two load bearing are distributed as the picture shown below. Radial load and axial load are applied on the test bearing. The length of the shaft is 700 mm (AB = 200 mm, BB = 300 mm). We design diameter of shaft 100 mm in test bearing and 125mm in load bearing. The radius of notch is 4 mm.

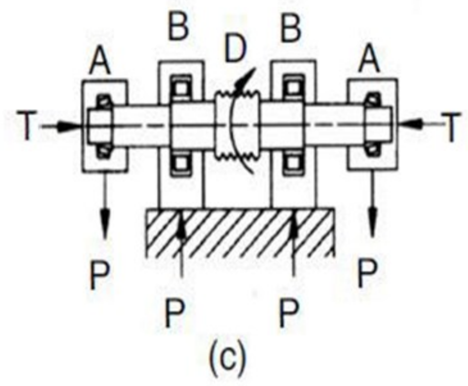



Figure 2.10: layout (c) from [5]

When using SKF calculator we keep rotating speed, operating temperature and viscosity constant.

We design the radial load on the test bearing Fr = 100 kN, the axial load Fa = 10 kN. It is easy to get the load on the load bearing, radial load Fr = 100 kN, axial load Fa = 10 kN.

Power loss of test bearing is 2700 W while load bearing is 4540 W.

The total power loss of these four bearings is

$P_{Tot}$ =2700×2+4540×2=14480 W

The fatigue limit of the component $\Sigma^{C}_{D\text{-}1}$ (with notches) will be $\Sigma^{C}_{D\text{-}1}=\frac{C_L C_S C_F \Sigma_{D-1}}{Kf}$.

Figure 2.11 shows a diagram of torsion and bending moment distributed on the shaft and we choose middle (diameter is 125 mm) as the dangerous section to calculate safety factor.

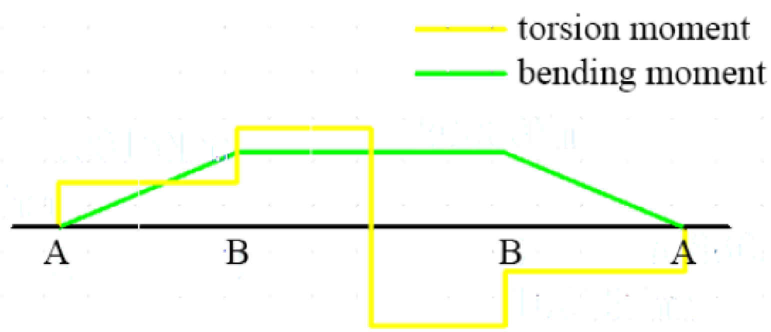



Figure 2.11: torsion and bending moment diagram of layout (c)

Bending moment of inertia $W_F$:

$W_F = \frac{\pi D^3}{32} = \frac{\pi * 125^3}{32} = 191747.6 \text{ mm}^{-3}$

Torsion moment of inertia $W_T$:

$W_T = \frac{\pi D^3}{16} = \frac{\pi * 100^3}{16} = 383495.2 \text{ Mm}^{-3}$

Bending moment $M_F$ =100*200=20000 Nm=$2 \times 10^7$ Nmm

Torsion moment Mt=8602+10012.6=18614.6 Nmm

Normal load T = 10000 N

Stress of bending moment $\Sigma_{Mf} = \frac{Mf}{Wf}$ =104.3 MPa

Stress of torsion moment $T_{Mt} = \frac{Mt}{Wt}$ =0.05 MPa

Normal stress $\Sigma_N = \frac{T}{\frac{\pi D^2}{4}}$=0.81 MPa

$\Sigma_a^{Mf}$= 104.3 MPa

$\Sigma_m^{Mf}$= 0

$T_a^{Mt}$= 0

$T_m^{Mt}$= 0.05 MPa

$\Sigma_a^{N}$= 0

$\Sigma_m^{N}$= 0.81 Mpa

We consider normal load and torsion moment are constant. So, we just calculate $\Sigma_{D-1}^{C}$ Of bending moment.

For fatigue stress concentration factor $K_f$=1+q*(Kt-1)

The value of $K_t$ Can be found in the diagram (Figure 2.12) as r/d = 0.04, D/d = 1.2. It is 2.05.

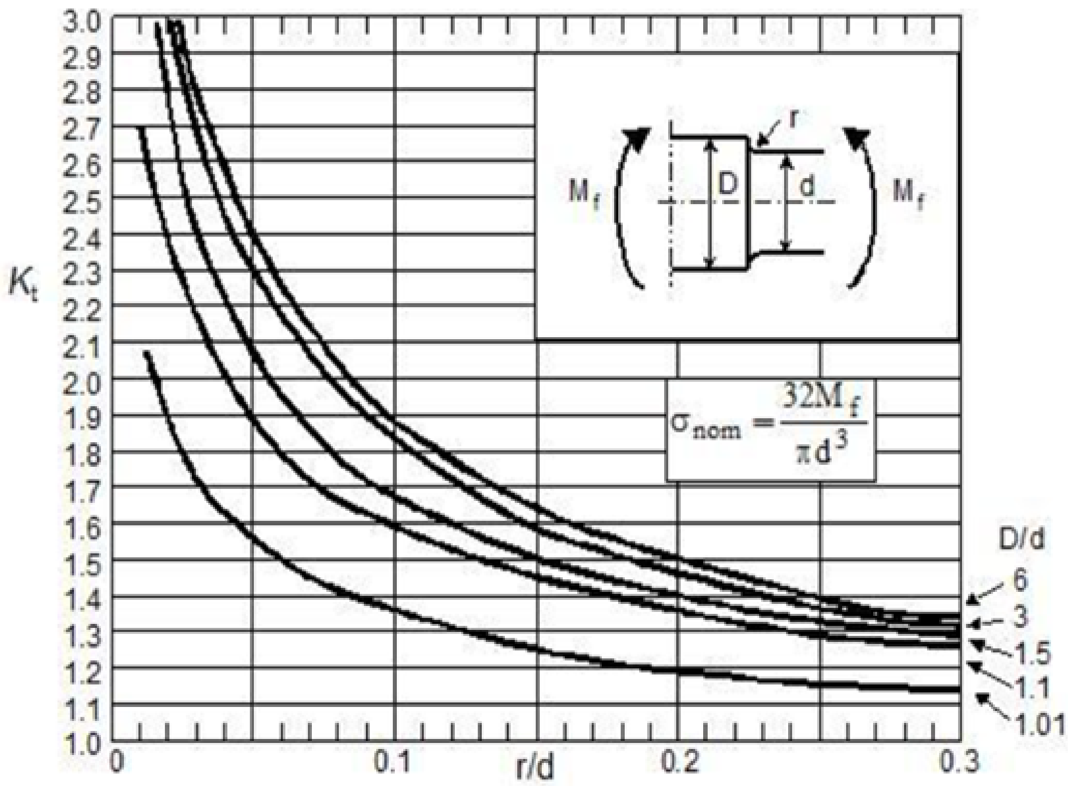


Figure 2.12: theoretical value from [5]

The value of q can be found in the diagram (figure 2.13) easily as r = 4 mm, that is q = 0.91.

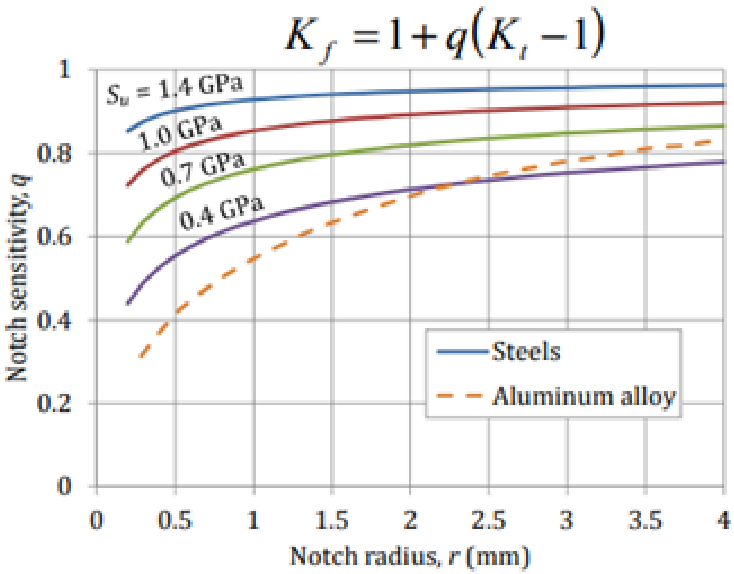



Figure 2.13: Notch sensitivity q depends on the material from [5]

So $K_f$ (bending)=1+0.91× (2.05-1) = 1.9555

$\Sigma^{C}_{D\text{-}1}=\frac{C_L C_S C_F \Sigma_{D-1}}{Kf}$= 147.68 MPa

Then we calculate the value of equivalent alternate and mean stress:

$\Sigma_a^{eq}=\sqrt{(\sigma_a^{Mf}+\sigma_a^{N})^2+3*\mathrm{T}_a^{Mt\,2}}$= 104.3 MPa

$\Sigma_m^{eq}=\sqrt{(\sigma_m^{Mf}+\sigma_m^{N})^2+3*\mathrm{T}_m^{Mt\,2}}$= 0.82 MPa

It is obvious that $\Sigma_m^{eq} \ll \Sigma_a^{eq}$, we can get safety factor:

SF=$\frac{\Sigma_{D-1}^{C}}{\Sigma_a^{eq}}$=1.42 > 1

The forth layout, just like the third one but the test bearing on the left is removed. Radial load and axial load are applied on the test bearing. The length of the shaft is 600 mm (AB = 200 mm, BB = 200 mm). We design diameter of shaft 100 mm in test bearing and 125 mm in load bearing. The radius of notch is 4 mm.

When using SKF calculator we keep rotating speed, operating temperature and viscosity constant.

We design the radial load on the test bearing Fr =100 kN, the axial load Fa = 10kN. It is easy to get the load on the left load bearing, radial load Fr = 100 kN, axial load Fa = 10kN. And the load on the right load bearing, radial load Fr = 200 kN, axial load Fa = 0 kN.

Power loss of test bearing is 2700 W, left load bearing is 4540 W, right load bearing is 8880 W.

The total power loss of these three bearings is

$P_{Tot}$=2700+4540+8880=16120 W

The fatigue limit of the component $\Sigma^{C}_{D\text{-}1}$ (with notches) will be $\Sigma^{C}_{D\text{-}1}=\frac{C_L C_S C_F \Sigma_{D-1}}{Kf}$.

Figure 2.14 shows a diagram of torsion and bending moment distributed on the shaft and we choose section of right load bearing (diameter is 120mm) as the dangerous section to calculate safety factor.

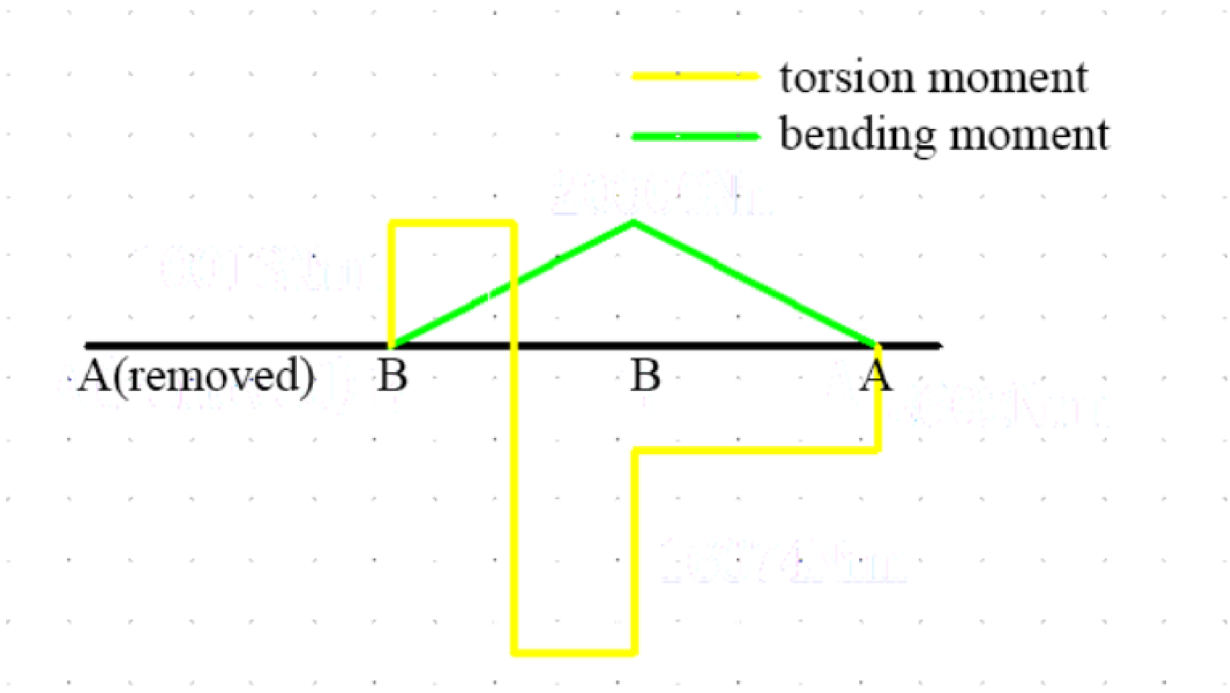



Figure 2.14: torsion and bending moment diagram of layout (d)

Bending moment of inertia $W_F$:

$W_F = \frac{\pi D^3}{32} = \frac{\pi * 125^3}{32} = 191747.6\ mm^3$

Torsion moment of inertia $W_T$:

$W_T = \frac{\pi D^3}{16} = \frac{\pi * 125^3}{16} = 383495.2\ mm^3$

Bending moment $M_F$ =100*200=20000Nm=$2 \times 10^7$ Nmm

Torsion moment Mt = 8602+10012.6 = 18614.6 Nmm

Normal load T =10000 N

Stress of bending moment $\Sigma_{Mf} = \frac{Mf}{Wf}$ =104.3 MPa

Stress of torsion moment $T_{Mt} = \frac{Mt}{Wt}$ = 0.05 MPa

Normal stress $\Sigma_N = \frac{T}{\frac{\pi D^2}{4}}$=0.81 MPa

$\Sigma_a^{Mf}$=104.3 MPa

$\Sigma_m^{Mf}$=0

$T_a^{Mt}$=0

$T_m^{Mt}$=0.05 MPa

$\Sigma_a^N$=0

$\Sigma_m^N$=0.81 MPa

We consider normal load and torsion moment are constant. So, we just calculate $\Sigma_{D-1}^C$ Of bending moment.

Just like the third layout, $K_F$(bending) = 1.9555

$\Sigma_{D-1}^{C} = \frac{C_L C_S C_F \Sigma_{D-1}}{Kf}$=147.68 MPa

Then we calculate the value of equivalent alternate and mean stress:

$\Sigma_a^{eq} = \sqrt{(\sigma_a^{Mf} + \sigma_a^N)^2 + 3 * T_a^{Mt\,2}}$=104.3 MPa

$\Sigma_m^{eq} = \sqrt{(\sigma_m^{Mf} + \sigma_m^N)^2 + 3 * T_m^{Mt\,2}}$=0.82 MPa

It is obvious that $\Sigma_m^{eq} << \Sigma_a^{eq}$, we can get safety factor:

$SF = \frac{\Sigma_{D-1}^C}{\Sigma_a^{eq}}$=1.42 > 1

Note: The cases above all are under the normal working condition. Then if one of the bearing is damaged, the power increased is equal to 10% of the total power.

### 2.3. Relationship between power consumption and the value of ratio γ

First, we assume the ratio γ:

Ratio $\gamma = \frac{Distance\ between\ test\ and\ load\ bearing}{Total\ length\ of\ shaft}$

There are two cases about the total length = 500 mm & 1000 mm.

According to the tests, we can find out the relationship between the distance of the total length and the power consumption:

| assume ratio γ=AB/L | | | | | |
|---|---|---|---|---|---|
| L=500mm | case1 | case2 | case3 | case4 | case5 |
| γ | 0.6 | 0.5 | 0.4 | 0.2 | 0.02 |
| Ptot(W) | 19520 | 16120 | 13790 | 11160 | 9850 |
| | | | | | |
| L=1000mm | case1 | case2 | case3 | case4 | case5 |
| γ | 0.3 | 0.25 | 0.2 | 0.1 | 0.01 |
| Ptot(W) | 12230 | 11650 | 11160 | 10370 | 9790 |

According to the table of test above, it is easy to draw the corresponding diagram.

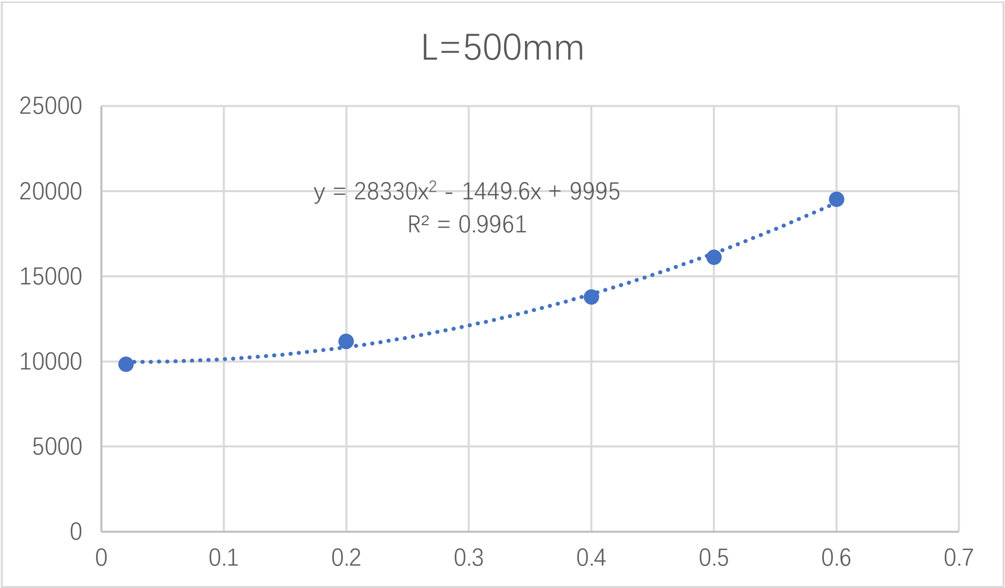



Figure 2.15: Power consumption & Ratio γ for 500 mm

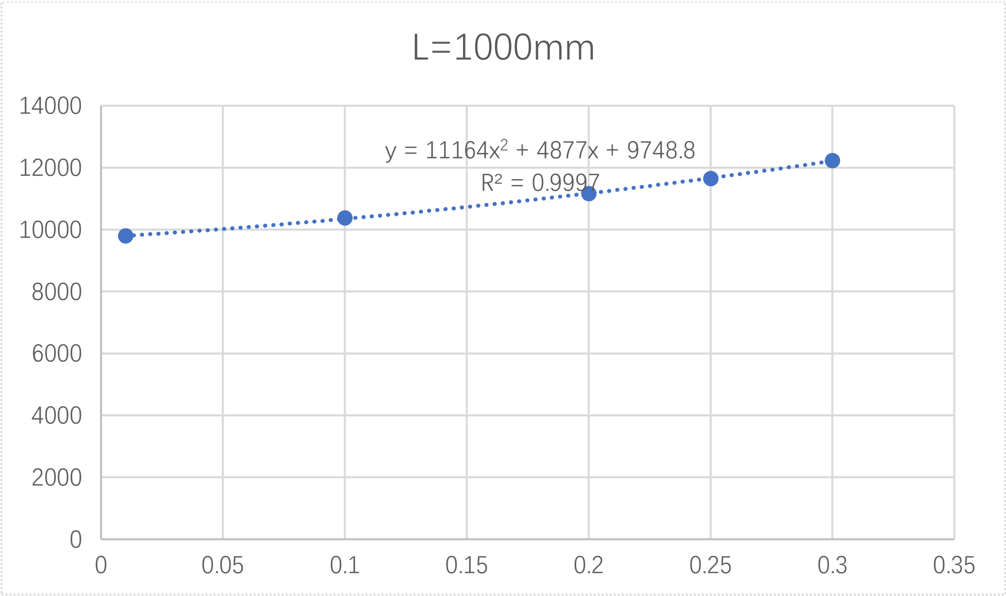



Figure 2.16: Power consumption & Ratio γ for 1000 mm

From the diagrams, if the ratio is lower, the power consumption is lower. This rule is the same in both two cases.

When L = 500 mm, the fitting curve can be expressed as

$P_{Tot} = 28330\gamma^2 - 1449.6\gamma + 9995$

$R^2 = 0.9961$, which indicates the goodness of the fitting since is very close to 1

When L = 1000 mm, the fitting curve can be expressed as

$P_{Tot} = 11164\gamma^2 + 4877\gamma + 9748.8$

$R^2 = 0.9997$, which indicates the goodness of the fitting since is very close to 1

So, if there is only one value is known (total length or power consumption), it is easy to obtain the corresponding value in the diagram.

# 3. Housing

### 3.1. Selection of housing

An SKF Bearing housing, together with appropriate SKF bearings constitute economic, interchangeable bearing units that meet the demand for designs that are easy to maintain.

SKF produces bearing housings in a wide range of designs and sizes that are based on experience collected in all industrial areas. Among others, SKF bearing housings have the following advantages

1. Large assortment of design and sizes
2. High quality of bearing housings design and manufacturing
3. Worldwide availability

According to the diameter of the shaft and the bearding, we choose the two housings with the shaft diameter equal to 125 mm, 100 mm. Below there are the two housings used in the process.

# SNL 528 TURU

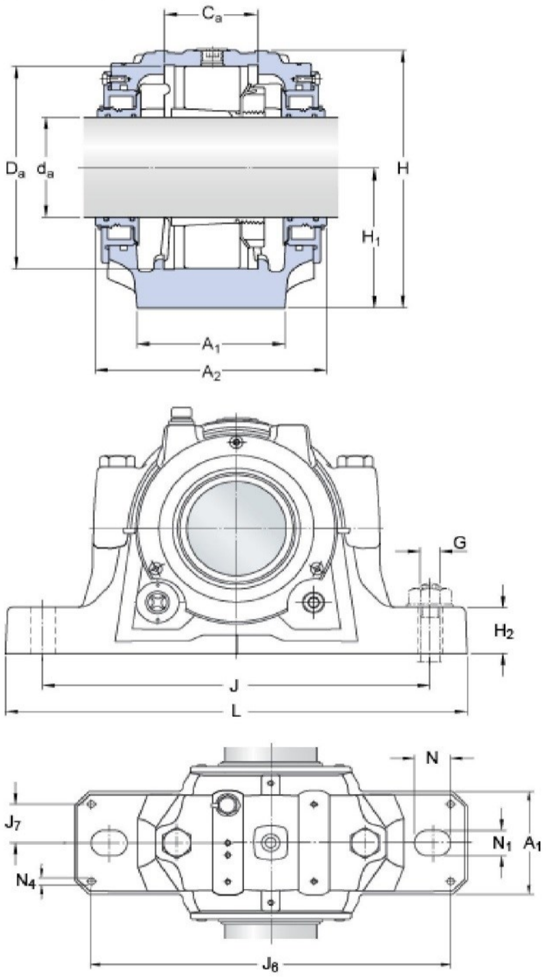
## Appropriate products

| | |
|---|---|
| Bearing (basic designation) | 22228 K |
| Adapter sleeve | H 3128 |
| Locating ring | 2 x FRB 15/250 |
| End cover | ASNH 528 R |

## Appertaining products

| | |
|---|---|
| Housing | SNL 528 RU |
| Oil seal | 2 x TSN 528 U |

## Dimensions




| | | | |
|---|---|---|---|
| $d_a$ | | 125 | mm |
| $C_a$ | | 98 | mm |
| $D_a$ | | 250 | mm |
| $A_1$ | | 150 | mm |
| $A_2$ | | 225 | mm |
| H | | 302 | mm |
| $H_1$ | | 150 | mm |
| $H_2$ | | 50 | mm |
| J | | 420 | mm |
| L | | 500 | mm |
| N | | 42 | mm |
| $N_1$ | | 35 | mm |

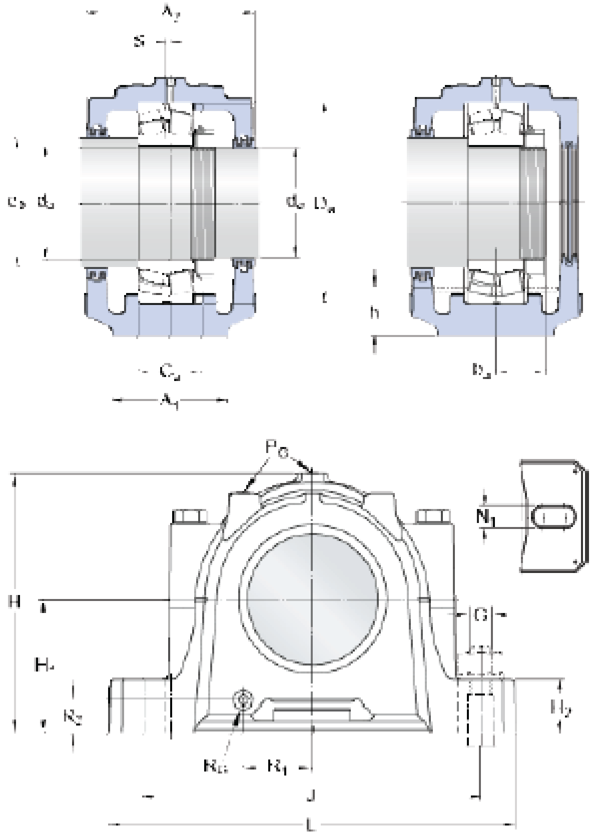
### Dowel pins

| | | | |
|---|---|---|---|
| $J_6$ | | 458 | mm |
| $J_7$ | | 54 | mm |
| $N_4$ | max. | 12 | mm |

Figure 3.1 SNL 528 TURU housing for the load bearing

## SAF 22220

### Dimensions



| | | | |
|---|---|---|---|
| $d_a$ | | 100 | mm |
| $d_b$ | | 114.3 | mm |
| $d_c$ | | 96.838 | mm |
| $D_a$ | | 180 | mm |
| $C_a$ | | 55.804 | mm |
| $A_1$ | | 111.125 | mm |
| $A_2$ | | 155.575 | mm |
| S | | 4.763 | mm |
| $b_a$ | | 44.45 | mm |
| H | | 217.488 | mm |
| $H_1$ | | 114.3 | mm |
| $H_2$ | | 44.45 | mm |
| J | min. | 295.275 | mm |
| J | max. | 333.375 | mm |
| L | | 387.35 | mm |
| $N_1$ | | 25.4 | mm |

Figure 3.2 SAF 22220 housing for the load bearing

# 4. Shaft

Here in this chapter, we will discuss the material, the dimensions and the properties of the shaft.

### 4.1. Build 3D model

After the bearings are determined, the diameters of the shaft can be easy to obtain. As for the length of the shaft, it is necessary to consider the total mass.

Before calculating the mass, we have to define the total length.

We choose the total length = 1000 mm as the model to study. Because when we define the total length, it is necessary to consider the length occupied by the rolling bearings width, the housings width, the connecting with the motor, the gaps between the adjacent housings. Figure 4.2, Figure 4.3

Consequently, we can draw it in Solidworks. Figure 4.1

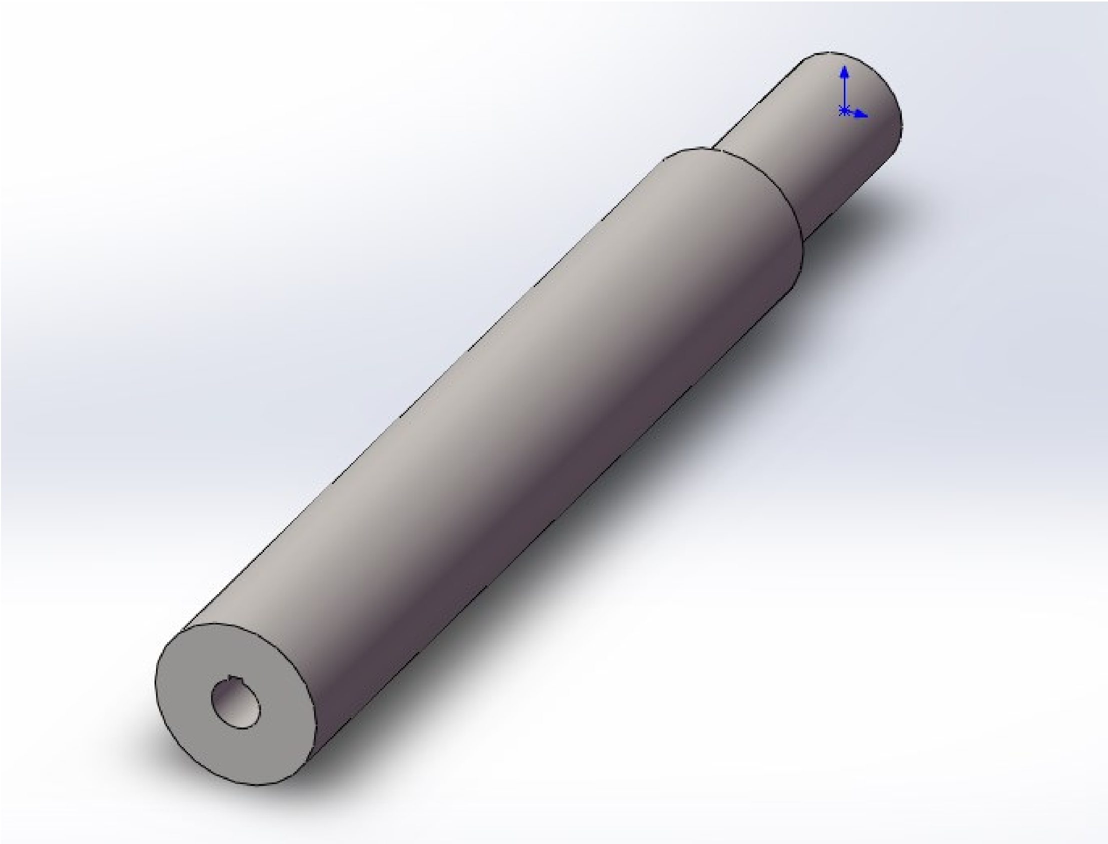


Figure 4.1 3D model of the shaft

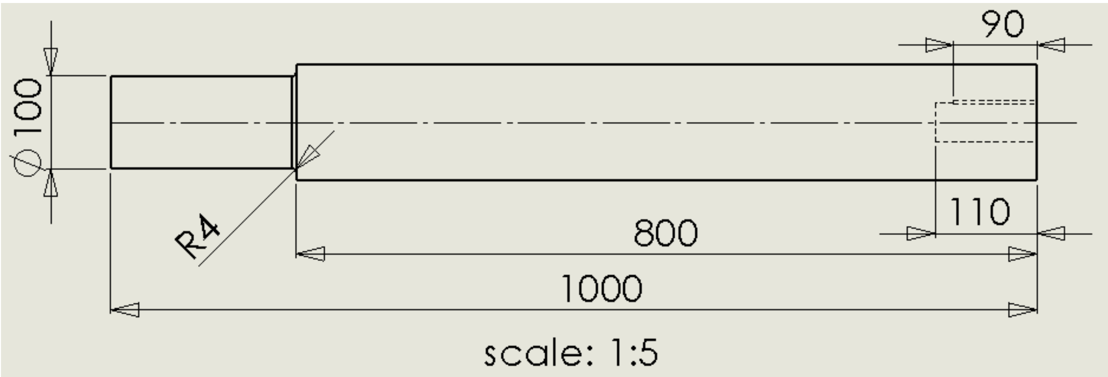



Figure 4.2 Length and diameters of the shaft

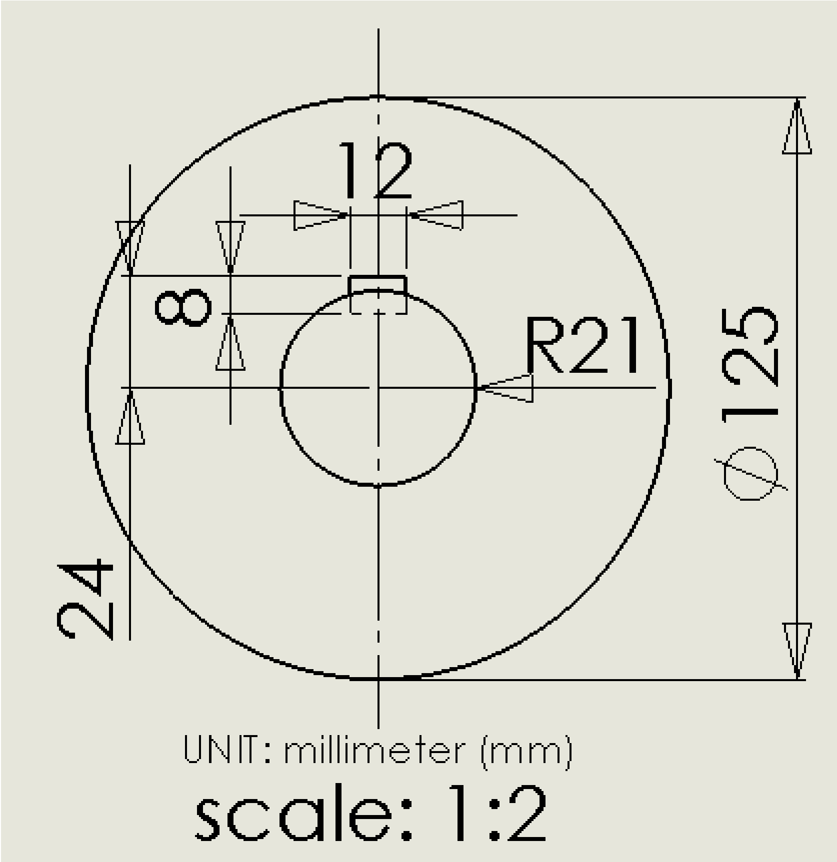



Figure 4.3 Section of the bushing connecting with the motor

### 4.2. Characteristics of the shaft material

As the material of the shaft has be chosen before, 36NiCrMo4 UNI EN 10083. Now here are some characteristics why this material is adopted for the shaft.

First of all, as with regular alloys, alloy steel has created the addition of alloying elements are added to iron along with carbon. The simplest steel is iron, which is alloyed with carbon. Alloy steel casting exporters look for other metals that would lend their specific properties to the iron-and-carbon blend. Some of the more common alloys used include manganese, silicon, boron, chromium, and nickel. Alloy steels come with improved properties as compared to regular or carbon steel – hot hardness, shine, wear resistance, harden ability and corrosion resistance. Alloy steel is used in all kinds of industrial applications, due to its unique properties. Alloy Steel casting's Unique properties or Top 4 benefits are as follows:

1. Improved strength

Alloy steels are much stronger and tougher than regular or carbon steel due adding of nickel, manganese, nickel, and copper. Since they contain less steel, they are much lighter than regular steel and are good for vehicles which in turn lead to fuel economy and less road damage. They are equally better than carbon steel and are smaller in size. The higher tensile strength and lighter weight also help in designing bridges so that the center spans can be longer and there is need of few supporting beams. Also used in the making of television transmitter masts, the extra strength allows the sections to be thinner and more stable to due to higher resistance to the wind.

2. Durability

Due to the addition of nickel, zirconium, cerium, and calcium, alloy steel is much more durable. Alloy steel can withstand heat and wear-and-tear better than other metals, and therefore has a much longer life span. Therefore, they are well-suited for streetlight poles, oil storage tanks, and earth moving equipment. They are also ideal for automobile and machinery parts due to the same reason. They are also resistant to high temperatures and are therefore helpful in heavy welding and pressure cutting.

3. Powerful, yet lightweight

As mentioned above, alloy steel is extremely light but is sturdy enough. Vehicle manufacturing factories choose alloy steel over regular or carbon steel to produce high-performance wheels. For example, drivers find it easier to handle, steering, cornering and accelerate in cars fitted with alloy wheels rather than steel wheels. Alloy wheels also offer a better braking performance and decrease the risk of brake failure to a large extent. The secret behind the better braking is the superior traction due to the reduction of the wheel hop with the alloy wheels. Alloy wheels are also good for tubeless, as compared to steel tires, and are completely airtight.

4. Corrosion and weather resistant

One of the most important properties of alloy steel is that it is weather- and corrosion resistant. Its anti-corrosion properties come by adding chromium, copper, and nickel to steel and carbon. Anti-corrosion is an essential property, especially if the structure is placed outside. Since it does not rust even in its “bare” condition, there is no need to repaint or re-galvanize it. Some of the applications here would include sculptures and other structures that are constantly exposed to harsh weather and less maintenance.

In the other hand, we talk about the different steel grade with different chemical elements and elements in different amount and effects on steel properties.

Steel in general is an alloy of carbon and iron, it does contain many other elements, some of which are retained from the steel making process, other elements are added to produce specific properties. We can see some common chemical elements used in 36NiCrMo4 UNI EN 10083 with important effects on steel properties.

Chromium (Cr)

Chromium is a powerful alloying element in steel. Cr presents in certain structural steels in small amounts. It is primarily used to increase hardenability of steel and increase the corrosion resistance as well as the yield strength of the steel material. For that reason, often occurs in combination with nickel and copper. Stainless steels may contain in excess of 12% chromium. The well-known “18-8” stainless steel contains 8 percent of nickel and 18 percent of chromium.

When the percent of chromium in the steel exceeds 1.1% a surface layer is formed that helps protect the steel against oxidation.

Molybdenum (Mo)

Molybdenum has effects similar to manganese and vanadium and is often used in combination with one or the other. This element is a strong carbide former and is usually present in alloy steels in amounts less than 1%. It increases hardenability and elevated temperature strength and also improves corrosion resistance as well as increased creep strength. It is added to stainless steels to increase their resistance to corrosion and is also used in high speed tool steels.

Nickel (Ni)

In addition to its favorable effect on the corrosion resistance of steel, Ni is added to steels to increase hardenability. Nickel enhances the low-temperature behavior of the material by improving the fracture toughness. The weldability of the steel is not decreased by the presence of this element. The nickel drastically increases the notch toughness of the steel.

Nickel is often used in combination with other alloying elements, especially chromium and molybdenum. It is a key component in stainless steels but at the low concentrations found in carbon steels. Stainless steels contain between 8% and 14% nickel.

One more reason Ni is added to an alloy is that it creates brighter portions in Damascus steels.

## 4.3. Calculation of mass

To calculate the mass, the value of density is important., it is easy to find out this material density according to Table 4.1 and Figure 4.4 below.

| T | E $10^{-5}$ | $\alpha$ $10^{6}$ | $\lambda$ | $\rho$ | C | R $10^{9}$ |
|---|---|---|---|---|---|---|
| Grade | MPa | 1/Grade | Watt/(m·Grade) | kg/m$^3$ | J/(kg·Grade) | Ohm·m |
| 20 | 2.15 | | 39 | 7850 | | 331 |
| 100 | 2.11 | 11.6 | 38 | | 490 | |
| 200 | 2.01 | 12.1 | 37 | | 506 | |
| 300 | 1.9 | 12.7 | 37 | | 522 | |
| 400 | 1.77 | 13.2 | 35 | | 536 | |
| 500 | 1.73 | 13.6 | 33 | | 565 | |
| 600 | | 13.9 | 31 | | | |
| 700 | | | 29 | | | |
| 800 | | | 27 | | | |
| T | E $10^{-5}$ | $\alpha$ $10^{6}$ | $\lambda$ | $\rho$ | C | R $10^{9}$ |

Table 4.1 Physical properties for grade 40KHN2MA (40XH2MA)

| USA | Germany | Japan | France | England | European | Italy | Spain | Bulgaria | Hungary | Poland | Romania | Czechia |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| - | DIN,WNr | JIS | AFNOR | BS | EN | UNI | UNE | BDS | MSZ | PN | STAS | CSN |
| 4340<br>9840<br>G43400<br>G43406<br>G98400<br>Gr.9840 | 1.6511<br>1.6565<br>1.7225<br>34CrNiMo6<br>36CrNiMo4<br>36NiCrMo4<br>40NiCrMo6<br>42CrMo4<br>G36CrNiMo4 | SNCM439<br>SNCM447 | 35NCD5<br>35NCD6<br>36NiCrMo4<br>40NCD3<br>42CD4TS | 34CrNiMo6<br>36NiCrMo4<br>708M40<br>816M40<br>817A37<br>817M37<br>817M40<br>818M40 | 1.6511<br>1.6582<br>34CrNiMo6<br>36CrNiMo4<br>40NiCrMo4KD | 35NiCrMo6KB<br>36NiCrMo4<br>38NiCrMo4<br>38NiCrMo4KB<br>40NiCrMo7 | 35NiCrMo4<br>36CrNiMo4<br>42CrMo4<br>F.1280 | 36CrNiMo4<br>40ChN2MA | 36CrNiMo4 | 36HNM<br>40HNMA | T35MoCrNi08 | 16341<br>16342 |

Figure 4.4 Equivalent steels for grade 40KHN2MA (40XH2MA)

Density ρ = 7850 $\left[\frac{\text{kg}}{m^3}\right]$

So, the total volume and the total mass of the shaft could be calculated simultaneously by using the function “Evaluate – Mass properties”.

These two values are:

Total volume = 11233420.227 $mm^3$

Total mass = 88.182 kg

# 5. Motor and Hydraulic cylinder

## 5.1. Selection standard of motor

In the previous chapter, the total power of the bearings has been calculated. The shaft has been designed according to the “case 3 (AB = 200 mm; BB = 800 mm), so the corresponding total power is 11160 W.

The most important technical data when we choose the motor in the test rig is output power. This power must be 1.5 times of the total power. Therefore, the power of motor is around 15 – 20 kW. Figure 5.4, Figure 5.5

According to the catalogue of motors of ABB company, the motor type (M2BAX 160 MLB) is the best choice for our requirements.

In addition, it is essential to consider the connecting method with the shaft. Below there is a 3D model of the motor, figure 5.1.

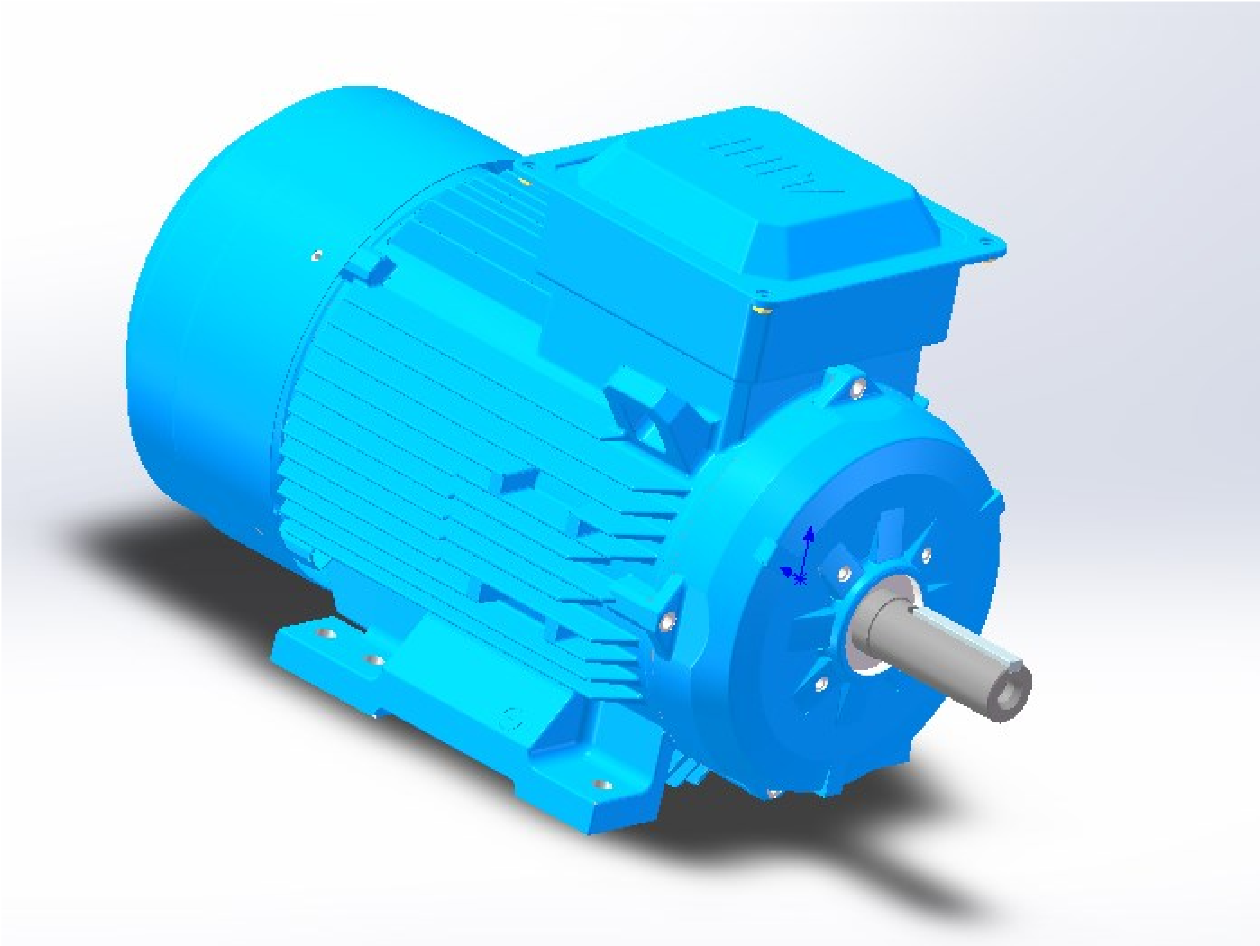


Figure 5.1: 3D model of the motor

## 5.2. Connecting method between motor and shaft: Bushing

During assembly, a key is inserted into machined keyways in the bushing and shaft to lock them together and prevent the shaft from rotating in the bushing. These keys are generally made from bar stock and come in square, rectangular, and tapered shapes. The Woodruff key, used primarily in machine and automotive applications, is shaped like a half-moon.

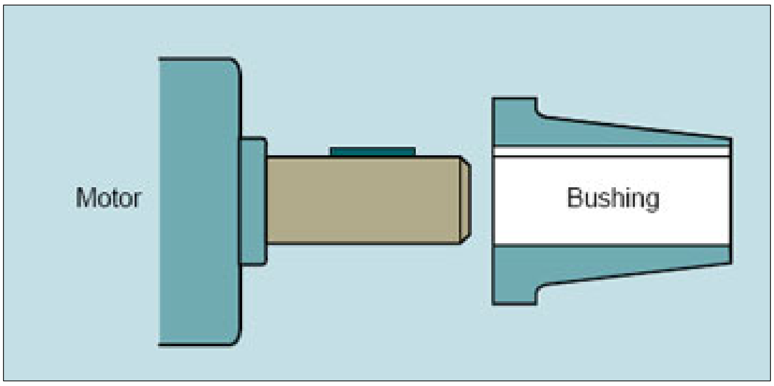



Figure 5.2: Bushing used in conventional mounting

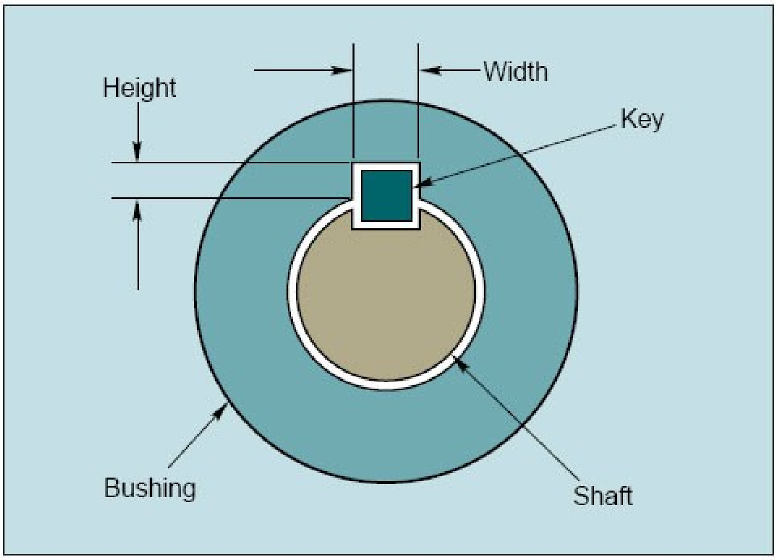



Figure 5.3: Keys fits snugly between bushing and shaft to lock them together radially

# 技术数据
# Technical data

## IE2
## 2P 380V 50HZ

### 三相全封闭鼠笼式电机的技术数据
### Technical data for totally enclosed squirrel cage three phase motors

IP 55 – IC 411 – 绝缘等级 F，温升等级 B
0.18-0.75kW， 符合 GB 25958-2010 的 3 级能效
0.75-355kW，符合 IEC 60034-30-1; 2014 的 IE2 效率等级及 GB 18613-2012 的 3 级能效
IP 55 - IC 411 - Insulation class F, temperature rise class B
0.18-0.75kW, Grade 3 according to GB 25958-2010
0.75-355kW, IE2 efficiency class according to 60034-30-1; 2014 , Grade 3 according to GB 18613-2012

| 输出 Output | 电机型号 Motor type | 产品代码 Product code | 转速 Speed | 效率 / Efficiency IEC 60034-2-1; 2007 | | | 功率因数 Power factor | 电流 Current | | 转矩 / Torque | | | 转动惯量 Moment of inertia | 重量 Weight | 声压等级 Sound pressure level, $L_{PA}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | 满载 load 100% | 3/4 负载 load 75% | 1/2 负载 load 50% | | $I_N$ | | $T_N$ | | | J=1/4 | | |
| kW | | | r/min | | | | cos φ | A | $I_S/I_N$ | Nm | $T_l/T_N$ | $T_B/T_N$ | $GD^2$kgm$^2$ | kg | dB |
| 3000 r/min = 2 极 / 2 poles | | | 380 V 50Hz | | | | | CENELEC- 设计 design | | | | | | | |
| 0.37 | M2BAX 71 MA | 3GBA 071 310-••CCN | 2769 | 73.5 | 73.0 | 70.3 | 0.84 | 0.91 | 4.9 | 1.26 | 2.5 | 3.0 | 0.00033 | 9 | 56 |
| 0.55 | M2BAX 71 MB | 3GBA 071 320-••CCN | 2790 | 75.5 | 75.3 | 73.1 | 0.83 | 1.33 | 5.2 | 1.86 | 2.8 | 3.2 | 0.00041 | 10 | 58 |
| 0.75 | M2BAX 80 MA | 3GBA 081 310-••CCN | 2797 | 77.4 | 76.5 | 76.2 | 0.86 | 1.71 | 5.3 | 2.51 | 2.7 | 3.9 | 0.00067 | 13 | 63 |
| 1.1 | M2BAX 80 MB | 3GBA 081 320-••CCN | 2821 | 79.6 | 80.2 | 79.8 | 0.87 | 2.41 | 5.2 | 3.67 | 2.8 | 3.7 | 0.00088 | 14 | 62 |
| 1.5 | M2BAX 90 SA | 3GBA 091 110-••CCN | 2876 | 81.3 | 80.6 | 78.6 | 0.84 | 3.34 | 6.7 | 4.93 | 2.8 | 3.5 | 0.00208 | 20 | 66 |
| 2.2 | M2BAX 90 LA | 3GBA 091 510-••CCN | 2882 | 83.2 | 83.6 | 82.9 | 0.88 | 4.57 | 7.1 | 7.25 | 2.8 | 3.4 | 0.00274 | 23 | 67 |
| 3 | M2BAX 100 LA | 3GBA 101 510-••CCN | 2894 | 84.6 | 85.2 | 84.1 | 0.93 | 5.79 | 7.1 | 9.90 | 2.6 | 3.4 | 0.00475 | 32 | 74 |
| 4 | M2BAX 112 MA | 3GBA 111 310-••CCN | 2878 | 85.8 | 87.0 | 86.9 | 0.94 | 7.54 | 7.0 | 13.3 | 2.4 | 3.2 | 0.00561 | 36 | 74 |
| 5.5 | M2BAX 132 SA | 3GBA 131 110-••CCN | 2908 | 87.0 | 86.5 | 84.8 | 0.89 | 10.8 | 7.6 | 18.0 | 2.3 | 3.8 | 0.01170 | 54 | 74 |
| 7.5 | M2BAX 132 SB | 3GBA 131 120-••CCN | 2905 | 88.1 | 87.7 | 86.7 | 0.88 | 14.7 | 8.1 | 24.6 | 2.7 | 4.0 | 0.01319 | 58 | 72 |
| 11 | M2BAX 160 MLA | 3GBA 161 410-••CCN | 2919 | 89.4 | 89.9 | 89.6 | 0.88 | 21.2 | 6.2 | 35.9 | 2.2 | 3.1 | 0.041 | 102 | 72 |
| 15 | M2BAX 160 MLB | 3GBA 161 420-••CCN | 2929 | 90.3 | 90.7 | 90.5 | 0.90 | 28.0 | 7.0 | 48.9 | 2.7 | 3.1 | 0.054 | 115 | 72 |
| 18.5 | M2BAX 160 MLC | 3GBA 161 430-••CCN | 2932 | 90.9 | 91.2 | 91.1 | 0.90 | 34.4 | 7.9 | 60.1 | 2.8 | 3.4 | 0.060 | 123 | 73 |
| 22 | M2BAX 180 MLA | 3GBA 181 410-••CCN | 2929 | 91.3 | 92.2 | 92.3 | 0.93 | 39.4 | 7.5 | 71.7 | 3.1 | 3.1 | 0.073 | 150 | 72 |
| 30 | M2BAX 200 MLA | 3GBA 201 410-••CCN | 2940 | 92.0 | 91.9 | 91.2 | 0.90 | 55.0 | 7.5 | 97.4 | 3.2 | 3.5 | 0.110 | 198 | 81 |
| 37 | M2BAX 200 MLB | 3GBA 201 420-••CCN | 2943 | 92.5 | 92.8 | 92.6 | 0.91 | 66.8 | 7.6 | 120 | 3.2 | 3.3 | 0.141 | 229 | 80 |
| 45 | M2BAX 225 SMA | 3GBA 221 210-••CCN | 2946 | 92.9 | 93.3 | 93.1 | 0.92 | 80.0 | 7.6 | 146 | 2.8 | 3.0 | 0.226 | 273 | 82 |
| 55 | M2BAX 250 SMA | 3GBA 251 210-••CCN | 2958 | 93.2 | 93.5 | 93.4 | 0.89 | 99.6 | 7.4 | 177 | 3.1 | 2.7 | 0.344 | 334 | 78 |
| 75 | M2BAX 280 SMD | 3GBA 281 240-••HCN | 2968 | 93.8 | 94.3 | 94.4 | 0.89 | 135 | 7.6 | 241 | 3.3 | 3.3 | 0.6 | 527 | 78 |
| 90 | M2BAX 280 SME | 3GBA 281 250-••HCN | 2971 | 94.1 | 94.5 | 94.6 | 0.91 | 158 | 7.2 | 290 | 3.6 | 3.1 | 0.7 | 576 | 76 |
| 110 | M2BAX 315 SMA | 3GBA 311 210-••CCN | 2981 | 94.3 | 94.0 | 92.8 | 0.84 | 211 | 6.7 | 352 | 2.0 | 2.9 | 1.2 | 767 | 78 |
| 132 | M2BAX 315 SMB | 3GBA 311 220-••CCN | 2979 | 94.6 | 94.5 | 93.6 | 0.86 | 247 | 6.9 | 423 | 2.1 | 2.8 | 1.4 | 827 | 78 |
| 160 | M2BAX 315 SMC | 3GBA 311 230-••CCN | 2979 | 95.1 | 95.0 | 94.3 | 0.90 | 284 | 6.8 | 512 | 2.1 | 2.7 | 1.7 | 917 | 78 |
| 200 | M2BAX 315 MLA | 3GBA 311 410-••CCN | 2978 | 95.0 | 95.1 | 94.4 | 0.89 | 359 | 7.1 | 641 | 2.6 | 2.6 | 2.1 | 1037 | 83 |
| 250 | M2BAX 355 SMA | 3GBA 351 210-••CCN | 2981 | 95.3 | 95.2 | 94.4 | 0.90 | 442 | 6.2 | 800 | 1.4 | 2.5 | 2.7 | 1329 | 83 |
| 315 | M2BAX 355 SMB | 3GBA 351 220-••CCN | 2978 | 95.4 | 95.4 | 94.8 | 0.89 | 563 | 6.5 | 1010 | 1.7 | 2.5 | 3.4 | 1469 | 83 |
| 355 [1] | M2BAX 355 SMC | 3GBA 351 230-••CCN | 2981 | 95.4 | 95.5 | 95.1 | 0.89 | 635 | 6.8 | 1137 | 1.9 | 2.4 | 3.6 | 1539 | 83 |

[1] 温升等级 F [1] temperature rise class F

产品代码中的两个圆点表示可选的安装方式、电压及频率代码（见订购信息一页）。

根据 IEC 60034-2-1 edition 2.0, 2014-06 的要求，给出效率值。
请注意，在测试方法未知时，这些数值没有可比性。
ABB 已经根据间接法计算出效率值，且根据测量得出杂散损耗（附加损耗）。

$I_s / I_N$ = 启动电流
$T_l / T_N$ = 转子堵转转矩
$T_b / T_N$ = 最大转矩

The two bullets in the product code indicate choice of mounting arrangements, voltage and frequency code (see ordering information page).

Efficiency values are given according to IEC 60034-2-1 edition 2.0, 2014-06.
Please note that the values are not comparable without knowing the testing method.
ABB has calculated the efficiency values according to indirect method, stray load losses (additional losses) determined from measuring.

$I_s / I_N$ = Starting current
$T_l / T_N$ = Locked rotor torque
$T_b / T_N$ = Breakdown torque

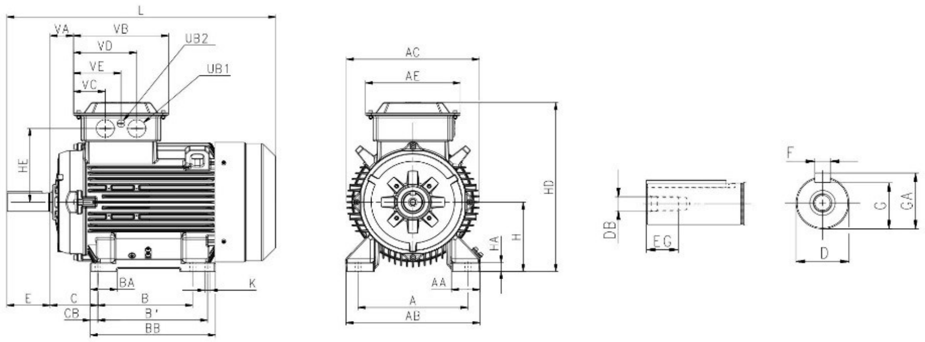
24 ABB Motors and Generators | 低压一般用途电机

Figure 5.4: Technical data of motors

# 外形图及外形尺寸
# Dimension drawings

## 机座号 160-250
## Frame size 160-250

### 底脚安装型电机 IM1001, B3
### Foot-mounted motor IM1001, B3




| 电机尺寸 Motor size | | 极数 Poles | A | AA | AB | AC | AE | B | B' | BA | BB | C | CB | D-tol. | DB | E | EG |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| M2BAX | 160ML[1] | 2-6 | 254 | 67 | 310 | 338 | 241 | 210 | 254 | 69 | 294 | 108 | 20 | 42-k6 | M16 | 110 | 36 |
| | 160ML[2] | 2-6 | 254 | 67 | 310 | 338 | 241 | 210 | 254 | 69 | 294 | 108 | 20 | 42-k6 | M16 | 110 | 36 |
| | 160ML[3] | 2-6 | 254 | 67 | 310 | 338 | 241 | 210 | 254 | 69 | 294 | 108 | 20 | 42-k6 | M16 | 110 | 36 |
| | 180ML[4] | 2-6 | 279 | 72 | 340 | 338 | 241 | 241 | 279 | 68 | 318 | 121 | 19 | 48-k6 | M16 | 110 | 36 |
| | 180ML[5] | 2-6 | 279 | 72 | 340 | 338 | 241 | 241 | 279 | 68 | 378 | 121 | 19 | 48-k6 | M16 | 110 | 36 |
| | 200ML[6] | 2-6 | 318 | 77 | 378 | 382 | 241 | 267 | 305 | 82 | 345 | 133 | 20 | 55-m6 | M20 | 110 | 42 |
| | 200ML[7] | 2-6 | 318 | 77 | 378 | 382 | 241 | 267 | 305 | 82 | 445 | 133 | 20 | 55-m6 | M20 | 110 | 42 |
| | 225SM | 2 | 356 | 91 | 435 | 414 | 262 | 286 | 311 | 69 | 351 | 149 | 20 | 55-m6 | M20 | 110 | 42 |
| | 225SM | 4-6 | 356 | 91 | 435 | 414 | 262 | 286 | 311 | 69 | 351 | 149 | 20 | 60-m6 | M20 | 140 | 42 |
| | 250SM | 2 | 406 | 98 | 480 | 462 | 262 | 311 | 349 | 72 | 392 | 168 | 22 | 60-m6 | M20 | 140 | 42 |
| | 250SM[8] | 4-6 | 406 | 98 | 480 | 462 | 262 | 311 | 349 | 72 | 392 | 168 | 22 | 65-m6 | M20 | 140 | 42 |
| | 250SM[9] | 4-6 | 406 | 98 | 480 | 462 | 262 | 311 | 349 | 72 | 437 | 168 | 22 | 65-m6 | M20 | 140 | 42 |

| 电机尺寸 Motor size | | 极数 Poles | F | G | GA | H | HA | HD | HE | K | L | UB1 | UB2 | VA | VB | VC | VD | VE |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| M2BAX | 160ML[1] | 2-6 | 12 | 37 | 45 | 160 | 23 | 413 | 188 | 14.5 | 586.5 | M40x1.5 | M16x1.5 | 59 | 241 | 81 | 161 | 120.5 |
| | 160ML[2] | 2-6 | 12 | 37 | 45 | 160 | 23 | 413 | 188 | 14.5 | 626.5 | M40x1.5 | M16x1.5 | 59 | 241 | 81 | 161 | 120.5 |
| | 160ML[3] | 2-6 | 12 | 37 | 45 | 160 | 23 | 413 | 188 | 14.5 | 683.5 | M40x1.5 | M16x1.5 | 59 | 241 | 81 | 161 | 120.5 |
| | 180ML[4] | 2-6 | 14 | 42.5 | 51.5 | 180 | 23 | 434 | 188 | 14.5 | 683.5 | M40x1.5 | M16x1.5 | 59 | 241 | 81 | 161 | 120.5 |
| | 180ML[5] | 2-6 | 14 | 42.5 | 51.5 | 180 | 23 | 434 | 188 | 14.5 | 743.5 | M40x1.5 | M16x1.5 | 59 | 241 | 81 | 161 | 120.5 |
| | 200ML[6] | 2-6 | 16 | 49 | 59 | 200 | 23 | 473 | 208 | 18.5 | 728 | M40x1.5 | M16x1.5 | 70 | 241 | 81 | 161 | 120.5 |
| | 200ML[7] | 2-6 | 16 | 49 | 59 | 200 | 23 | 473 | 208 | 18.5 | 828 | M40x1.5 | M16x1.6 | 70 | 241 | 81 | 161 | 120.5 |
| | 225SM | 2 | 16 | 49 | 59 | 225 | 23 | 539 | 228 | 18.5 | 824 | M63x1.5 | M16x1.7 | 79 | 262 | 83 | 179 | 131 |
| | 225SM | 4-6 | 18 | 53 | 64 | 225 | 23 | 539 | 228 | 18.5 | 854 | M63x1.5 | M16x1.8 | 79 | 262 | 83 | 179 | 131 |
| | 250SM | 2 | 18 | 53 | 64 | 250 | 23 | 585 | 248 | 24 | 882 | M63x1.5 | M16x1.9 | 72 | 262 | 83 | 179 | 131 |
| | 250SM[8] | 4-6 | 18 | 58 | 69 | 250 | 23 | 585 | 248 | 24 | 882 | M63x1.5 | M16x1.5 | 72 | 262 | 83 | 179 | 131 |
| | 250SM[9] | 4-6 | 18 | 58 | 69 | 250 | 23 | 585 | 248 | 24 | 927 | M63x1.5 | M16x1.5 | 72 | 262 | 83 | 179 | 131 |

**公差 Tolerance**

| | |
|---|---|
| A, B | ± 0.8 |
| D | ISO k6 ≤ ϕ 50 mm |
| | ISO m6 ≤ ϕ 50 mm |
| F | ISO h9 |
| H | +0, -0.5 |
| N | ISO j6 |
| C | ± 0.8 |

**附注 Footnotes**

M2BAX IE2:
1) MLB6 以外其余型号 All types except MLB6
2) MLB6
4) 所有型号 All types
6) 所有型号 All types
8) 所有型号 All types

M2BAX IE3:
1) MLA2, MLB2
2) MLA4, MLA6
3) MLC2, MLB4, MLB6
4) MLA2, MLA4
5) MLB4, MLA6
6) MLA6
7) MLA6 以外其余型号 All types except MLA6
9) 所有型号 All types

上表给出了主要尺寸（单位：mm）
如需图纸详情，请访问我们的网页
www.abb.com/motors&generators 或联系 ABB。

Above table gives the main dimensions in mm.
For detailed drawings please see our web-pages
www.abb.com/morors&generators or contact ABB.

38 ABB Motors and Generators | 低压一般用途电机

Figure 5.5: Dimension drawings of motors

## 5.3 Selection of Hydraulic cylinder

What Are Hydraulic Cylinders?

An actuation device that makes use of a pressurized hydraulic fluid is known as a hydraulic pump. This mechanism is used for producing linear motion and force in applications that transfer power. In other words, a hydraulic cylinder converts the energy stored in the hydraulic fluid into a force used to move the cylinder in a linear direction.

Function in bearing tester.

Bearing tester uses hydraulics for applying axial and radial loads to specimens.

Here I provide an example of hydraulic cylinder for the test rig.

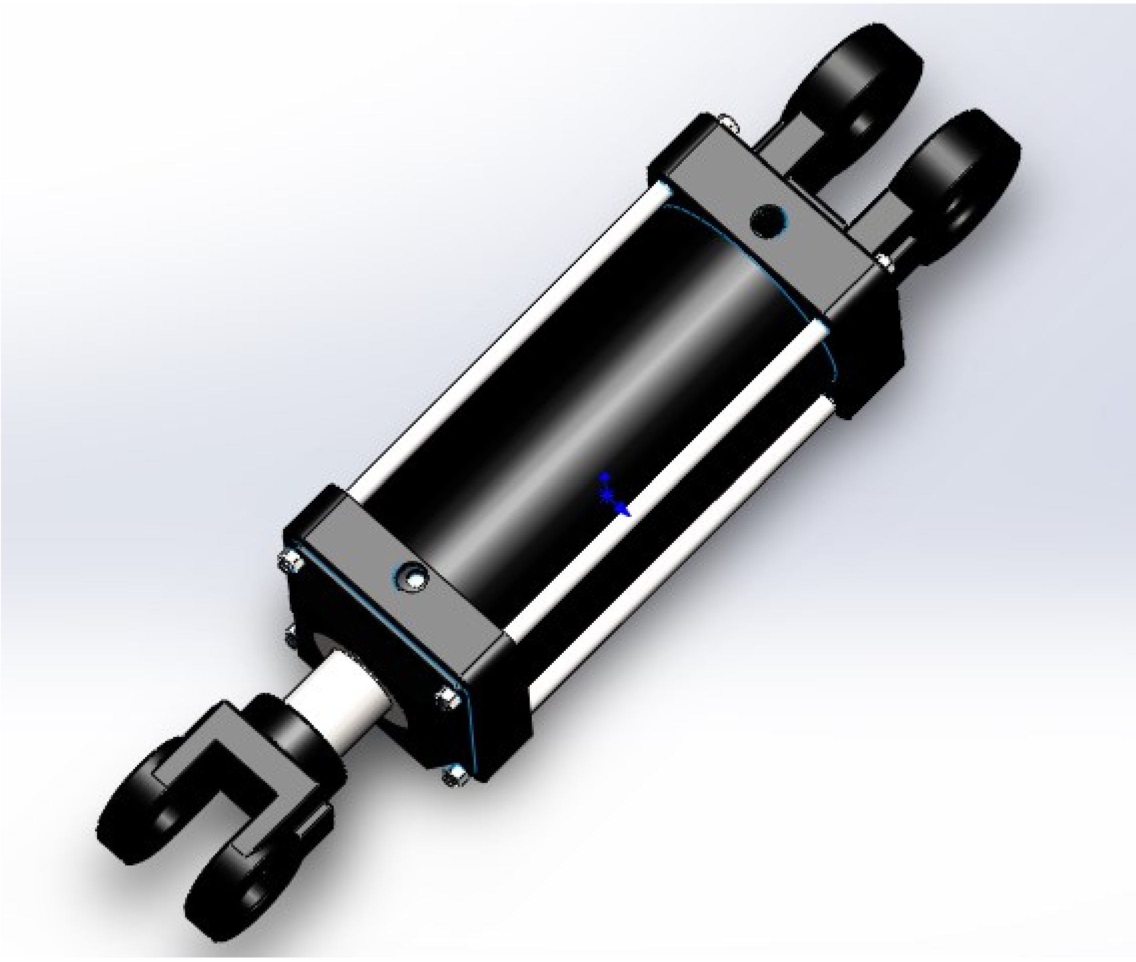


Figure 5.6: Hydraulic cylinder

# 6. Lubrication

The primary function of a lubricant is to lubricate the rolling and sliding contacts of a bearing to enhance its performance through the prevention of wear. This can be accomplished through various lubricating mechanisms such as hydrodynamic lubrication, elastic hydrodynamic lubrication (EHL), and boundary lubrication. The rolling/sliding contacts of concern are those between rolling element and raceway, rolling element and cage (separator), cage and supporting ring surface, and roller end and ring guide flanges.

In addition to wear prevention the lubricant performs many other vital functions. The lubricant can minimize the frictional power loss of the bearing. It can act as a heat transfer medium to remove heat from the bearing. It can redistribute the heat energy within the bearing to minimize geometrical effects due to differential thermal expansions. It can protect the precision surfaces of the bearing components from corrosion. It can remove wear debris from the roller contact paths. It can minimize the amount of extraneous dirt entering the roller contact paths, and it can provide a damping medium for separator dynamic motions.

No single lubricant or class of lubricants can satisfy all these requirements for bearing operating conditions from cryogenic to ultrahigh temperatures, from very slow to ultrahigh speeds, and from benign to highly reactive operating environments.

## 6.1. Selection Criteria

The selection of lubricants is based on their flow properties and chemical properties in connection with lubrication. Additional considerations, which sometimes may be of overriding importance, are associated with operating temperature, environment, and the transport or retention properties of the lubricant in the bearing.

According to the conditions of the selected bearings, such as: operating temperature, speed and environment etc. Liquid lubricant is best choice for the bearing lubrication.

### 6.2. Liquid Lubricants

Liquid lubricants are usually mineral oils; that is, fluids produced from petroleum-based stocks. They have a wide range of molecular constituents and chain lengths, giving rise to a large variation in flow properties and chemical performances. These lubricants are generally additive enhanced for both viscous and chemical performance improvement. Overall, petroleum-based oils exhibit good performance characteristics at relatively inexpensive costs.

Synthetic hydrocarbon fluids are manufactured from petroleum-based materials. They are synthesized with both narrowly limited and specifically chosen molecular compounds to provide the most favorable properties for lubrication purposes. Most synthetics have unique properties and are made from petroleum feedstocks, but they can be made from non-petroleum sources. Other "synthetic" fluids have unique properties and can be manufactured from non-petroleum-based oils. These include polyglycols, phosphate esters, dibasic acid esters, silicone fluids, silicate esters, and fluorinated ethers.

## 6.3. Advantages Over Other Lubricants

As compared to any other lubricant, in particular grease, a liquid lubricant provides the following advantages:

1. It is easier to drain and refill, a particular advantage for applications requiring short relubricating intervals.

2. The lubricant supply to the system can be more accurately controlled.

3. It is suitable for lubricating multiple sites in a complex system.

4. Because of its ability to be used in a circulating lubricant system, it can be used in higher temperature systems where its ability to remove heat is significant.

## 6.4. Lubrication Methods

Oil-air lubrication systems are appropriate for high-precision applications with very high operating speeds and requisite low operating temperatures.

The oil-air method (fig.6.1) also called the oil-spot method, uses compressed air to transport small, accurately metered quantities of oil as small droplets along the inside of feed lines to an injector nozzle, where it is delivered to the bearing (fig.6.2).

This minimum quantity lubrication method enables bearings to operate at very high speeds with relatively low operating temperature. The compressed air serves to cool the bearing and also produces an excess pressure in the bearing housing to prevent contaminants from entering. Because the air is only used to transport the oil and is not mixed with it, the oil is retained within the housing. Oil-air systems are considered to be environmentally safe, provided that any residual used oil is disposed of correctly.

For bearings used in sets, each bearing should be supplied by a separate injector. Most designs include special spacers that incorporate the oil nozzles.

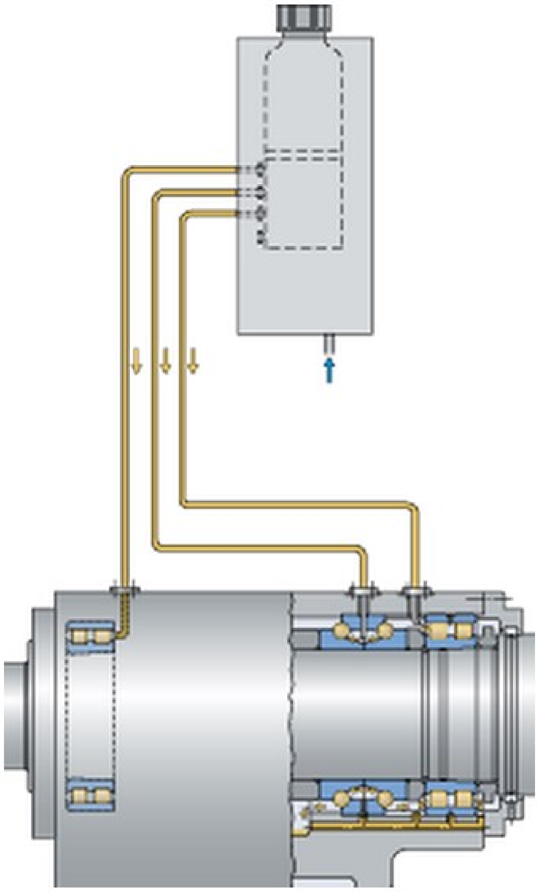


Figure 6.1: Oil-air from [8]

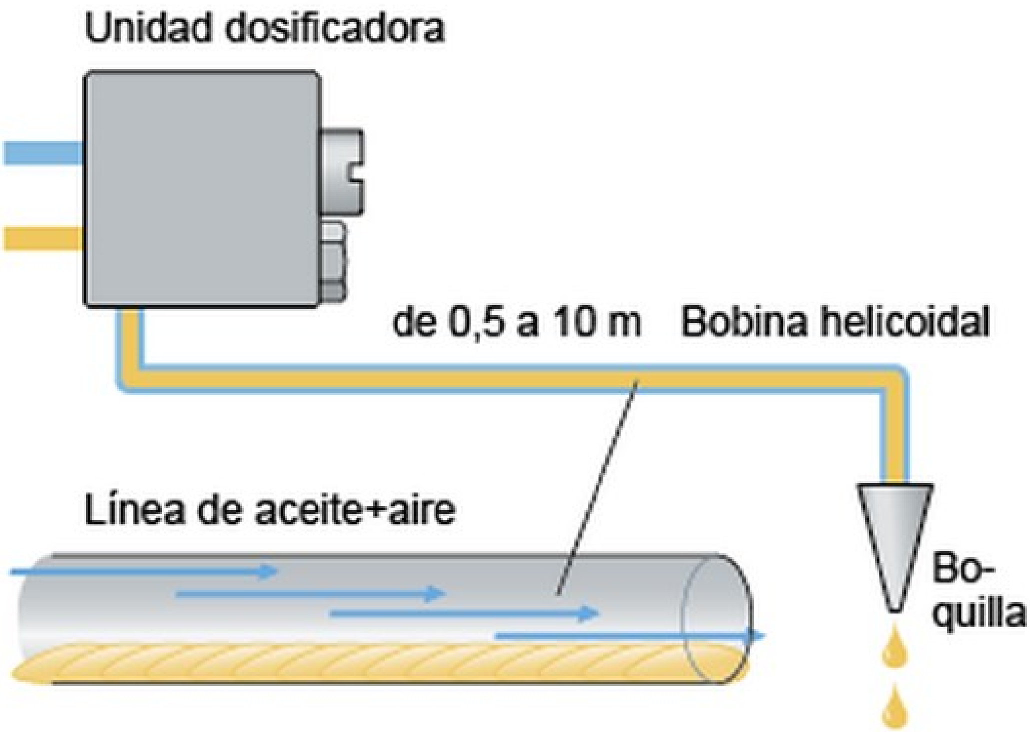



Figure 6.2: Oil-air from [8]

Guideline values for the oil quantity to be supplied to an angular contact ball bearing for high-speed operation can be obtained from:

Q = 1,3 dm

Guideline values for the oil quantity to be supplied to a cylindrical roller bearing or double direction angular contact thrust ball bearing can be obtained from:

Q = q d B / 100

Where

| | | |
|---|---|---|
| Q | = | oil flow rate [$mm^3$/h] |
| B | = | bearing width [mm] |
| d | = | bearing bore diameter [mm] |
| $d_m$ | = | bearing mean diameter [mm]<br>= 0,5 (d + D) |
| q | = | factor<br>= 1 to 2 for cylindrical roller bearings<br>= 2 to 5 for double direction angular contact thrust ball bearings |

Individual testing is, however, always recommended in order to optimize the conditions.

Different bearing designs show varying sensitivity to oil quantity changes. For example, roller bearings are very sensitive, whereas for ball bearings, the quantity can be changed substantially without any major rise in bearing temperature.

A factor influencing temperature rise and reliability of oil-air lubrication is the lubrication interval, i.e. the time in between two measures from the oil-air lubricator. Generally, the lubrication interval is determined by the oil flow rate generated by each injector and the oil quantity supplied per hour. The interval can vary from one minute to one hour, with the most common interval being 15 to 20 minutes.

Feed lines from the lubricator are 1 to 5 m in length, depending on the lubrication interval. A filter that prevents particles > 5 μm from reaching the bearings should be incorporated. The air pressure should be 0,2 to 0,3 MPa, but should be increased for longer runs to compensate for the pressure drop along the pipe's length.

To maintain the lowest possible operating temperature, ducts must be able to drain any excess oil away from the bearing. With horizontal shafts it is relatively easy to arrange drainage ducts on each side of the bearings. For vertical shafts the oil passing the upper bearing(s) should be prevented from reaching the lower bearings, which would otherwise receive too much lubricant. Drainage, together with a sealing device, should be incorporated beneath each bearing. An effective seal should also be located at the spindle nose to prevent lubricant from reaching the work piece.

The oil nozzles should be positioned so that oil can be introduced into the contact area between the rolling elements and raceways without interference by the cage. For the diameter (measured on the bearing) where oil injection should take place, refer to the product tables. For bearings equipped with alternative cages that are not listed, contact the SKF application engineering service.

The attainable speeds listed in the product tables for oil lubrication refer specifically to oil-air lubrication.

### 6.5. Guidelines for Use

In most applications pure petroleum oils are satisfactory as lubricants.

They must be free from contamination that might cause wear in the bearing, and should show high resistance to oxidation, gumming, and deterioration by evaporation. The oil must not promote corrosion of any parts of the bearing during standing or operation.

The friction torque in a liquid-lubricated bearing is a function of the bearing design, the load imposed, the viscosity and quantity of the lubricant, and the speed of operation. Only enough lubricant is needed to form a thin film over the contacting surfaces. Friction torque will increase with larger quantities and with increased viscosity of the lubricant.

Energy loss in a bearing depends on the product of torque and speed.

It is dissipated as heat, causing increased temperature of the bearing and its mounting structures. The temperature rise will always cause a decreased viscosity of the oil and, consequently, a decrease in friction torque from initial values. The overall heat balances of the bearing and mounting structures will determine the steady-state operating conditions.

It is not possible to give definite lubricant recommendations for all bearing applications. A bearing operating throughout a wide temperature range requires a lubricant with high viscosity index-that is, having the least variation with temperature. Very low starting temperatures necessitate a lubricant with a sufficiently low pouring point to enable the bearing to rotate freely on start-up. For specialized bearing applications involving unusual conditions, the recommendation of the bearing or lubricant manufacturer should be followed.

# 7. Vibration, Noise and Condition monitoring

## 7.1 Overview

In certain applications, the noise produced in operation is an important factor and can influence the bearing choice. SKF ball bearings are produced specifically for these applications.

In general, a rolling bearing does not generate noise by itself. What is perceived as "bearing noise" is in fact the audible effect of the vibrations generated directly or indirectly by the bearing on the surrounding structure. This is the reason why most of the time noise problems can be considered as vibration problems involving the complete bearing application.

Rolling element bearings can easily be affected by manufacturing errors, mounting defects or operational damage. This can lead to vibration, noise and even failure.

Where relevant, reference is also made to noise, sometimes resulting from excessive bearing vibration. A few common bearing applications in which noise and vibration are important are described.

Machine vibration or noise levels, whether excessive or not, are affected by bearings in three methods:

1. As a structural element defining in part a machine's stiffness;
2. As a generator of vibration by virtue of the way load distribution within the bearing varies cyclically;
3. As a vibration generator because of geometrical imperfections from manufacturing, installation or wear, and damage after continued use.

Detection of progressive bearing deterioration in operating machinery by vibration measurements has been in use for a long time and has become more economical and reliable in recent years. Some aspects of such machinery monitoring are considered.

**Significance of Vibration and Noise**

In many cases, objectionable airborne noise from a machine results from measurable vibration of machine components. Therefore, with respect to rolling bearings, the terms ‘‘noise’’ and ‘‘vibration’’ usually denote similar and related phenomena. Regardless of which seems to be more important in a particular application, noise and vibration may both be used as indicators in machines that have quality problems with bearings, machine components, or assembly methods when they are new, and as the first indication of an approaching need for repair or replacement after running for extended periods of time.

## 7.2 Classification of Vibration and Sound in Bearings

In rolling bearings, the varied types of vibration and sound are divided into the four categories in Table 7.1. Although there are many methods for classifying vibrations and sounds, here we only discuss about the classification system in this table.

| Generated frequency (frequency analysis) | | | Source | Countermeasures |
|---|---|---|---|---|
| FFT of original wave | | FFT after envelope (basic No.) | | |
| Radial (angular) direction | Axial direction | | | |
| $f_{RiN}$, $f_{MI}$ | $f_{AiN}$, $f_{AM}$ | — | Selective resonance of waviness (rolling friction) | Improve rigidity around the bearings, appropriate radial clearance, high-viscosity lubricant, high-quality bearings |
| $f_{RiN}$, $f_{MI}$ Natural frequency of cage | $f_{AiN}$, $f_{AM}$ | $Zf_c$ | Collision of rolling elements with inner ring or cage | Reduce radial clearance, apply preload, high-viscosity oil |
| ($\approx f_{R2N}$, $f_{R3N}$) | — | ? | Self-induced vibration caused by sliding friction at rolling surface | Reduce radial clearance, apply preload, change the grease, replace with countermeasured bearings |
| Natural frequency of cage | | $f_c$ | Collision of cage with rolling elements or rings | Apply preload, high-viscosity lubricant, reduce mounting error |
| Natural frequency of cage | | ? | Self-induced vibration caused by friction at cage guide surface | Change the grease brand, replace with countermeasured cage |
| Natural frequency of cage | | $Zf_c$ | Collision of cage and rolling element caused by grease resistance | Reduce radial clearance, apply preload, low-viscosity lubricant |
| $Zf_c$ | — | — | Displacement of inner ring due to rolling element passage | Reduce radial clearance, apply preload |
| $nZf_i \pm f_r$ ($nZ \pm 1$ peaks) | $nZf_i$ ($nZ$ peaks) | — | Inner ring raceway waviness, irregularity of shaft exterior | High-quality bearings, improve shaft accuracy |
| $nZf_c$ ($nZ \pm 1$ peaks) | $nZ_{fc}$ ($nZ$ peaks) | — | Outer ring raceway waviness, irregular bore of housing | High-quality bearings, improve housing bore accuracy |
| $2nf_b \pm f_c$ ($2n$ peaks) | $2nf_b$ ($2n$ peaks) | — | Rolling element waviness | High-quality bearings |
| $f_{RiN}$, $f_{MI}$ | $f_{AiN}$, $f_{AM}$ | $Zf_i$ | Nicks, dents, rust, flaking on inner ring raceway | Replacement and careful bearing handling |
| | | $Zf_c$ | Nicks, dents, rust, flaking on outer ring raceway | Replacement and careful bearing handling |
| | | $2f_b$ | Nicks, dents, rust, flaking on rolling elements | Replacement and careful bearing handling |
| $f_{RiN}$, $f_{MI}$ | $f_{AiN}$, $f_{AM}$ | Irregular | Entry of dirt and debris | Washing, improve sealing |
| Natural frequency of seal | | ($f_r$) | Self-induced vibration due to friction at seal contact area | Change the seal, change the grease |
| ? | ? | Irregular | Lubricant or lubricant bubbles crushed between rolling elements and raceways | Change the grease |
| $f_r$ | — | — | Irregular inner ring cross-section | High-quality bearings |
| $f_c$ | — | — | Ball variation in bearing, rolling elements non-equidistant | High-quality bearings |
| $f_r$ - $2f_c$ | — | — | Non-linear vibration due to rigid variation by ball variation | High-quality bearings |

$n$: Positive integer (1, 2, 3...)
$Z$: Number of rolling elements
$f_{RiN}$: Ring natural frequency in radial bending mode, Hz
$f_{MI}$: Natural frequency in the mode of angular vibration in inertia of outer ring-spring system, Hz
$f_r$: Rotation frequency of inner ring, Hz
$f_c$: Orbital revolution frequency of rolling elements, Hz
$f_{AiN}$: Ring natural frequency in axial bending mode, Hz
$f_{AM}$: Natural frequency in the mode of axial vibration in mass of outer ring-spring system, Hz
$f_i$: $f_i = f_r - f_c$, Hz
$f_b$: Rotation frequency of rolling element around its center, Hz

Table 7.1: Varied Types of Vibration and Sound in Rolling Bearings from [6]

In this classification system, the differences between categories are not absolute. The size of the sound or vibration of a structure, for example, is related to the manufacturing process. When sound is produced, it vibrates in most cases, and when it vibrates, it usually generates sound. Although vibration and sound almost always accompany each other, the problem of vibration and sound is usually characterized by one or the other. This is because the ability or ability of humans to hear sounds or feel vibrations depends on frequency. It is barely audible for low frequency sounds, while humans can't detect high frequency vibrations. Therefore, the low frequency problem is "vibration problem" and the high frequency problem is "noise problem". According to experience, the arbitrary boundary between vibration problem and noise problem is 1000 Hz. In other words, voice less than 1000 Hertz is considered a vibration and higher than 1000 Hertz is a sound or noise.

Types of main vibration and sound in bearings

i. Structural vibration and sound
ii. Vibration and sound related to bearing manufacturing
iii. Vibration and sound due to improper handling

i. Structural vibration and sound

In rolling bearings, vibration and sound will still occur even with the most progressive techniques used in manufacturing. Because the performance of the bearing is not degraded by such vibrations and sounds, they can be considered normal bearing characteristics.

Types of Structural vibration and sound:

- Race noise
- Click noise
- Squeal noise
- Cage noise
- Rolling element passage vibration

- Race noise

In rolling bearings field, race noise is the most basic sound. It is produced in all bearings shown as a smooth and continuous sound. We say it sounds like "sha---(sand is blew by wind)" In order to assess the bearing quality, sound size is exploited. Figure 7.1 gives the size of the race noise compared with familiar sounds. In this figure, even the loudest bearing # 6410 produces only about 1/100 of the normal volume. Clearly, the energy of the race noise is very limited. Race noise is characterized by:

:

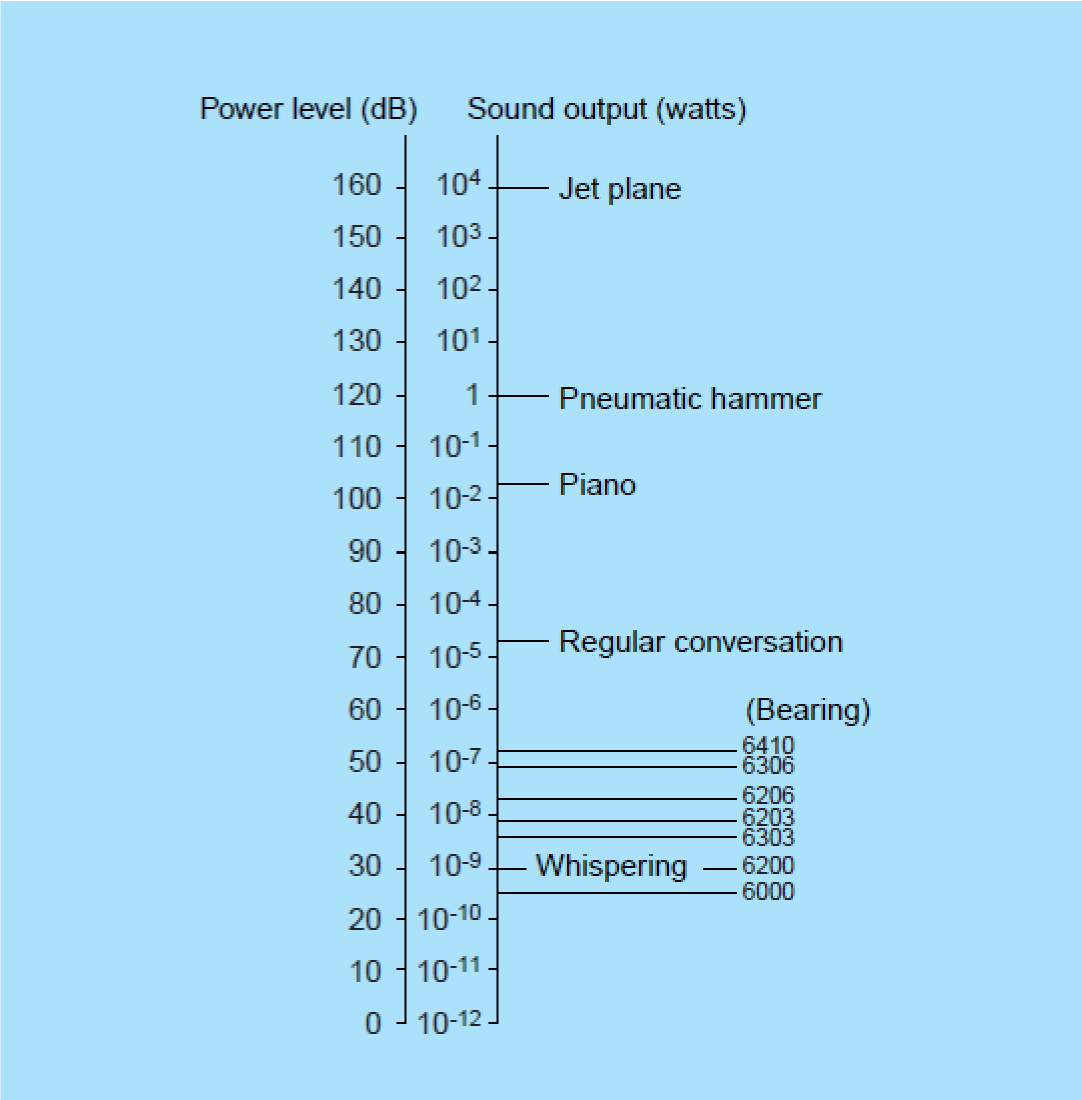



Fig. 7.1 Loudness range of race noise from [6]

(1) even if the rotation speed changes, the frequency will not vary. Its frequency is the natural frequency of the raceway ring, as shown in figure 7.2.

(2) the faster the speed, the sound will be louder.

(3) if the radial clearance decreases, the sound will increase.

(4) for lubricant, higher its viscosity, more decrement the sound will occur. In addition to the viscosity of the grease, the consistency, noise behavior is also influenced by shape and size of the soap fiber.

(5) the higher the rigidity of the shell, the smaller the sound.

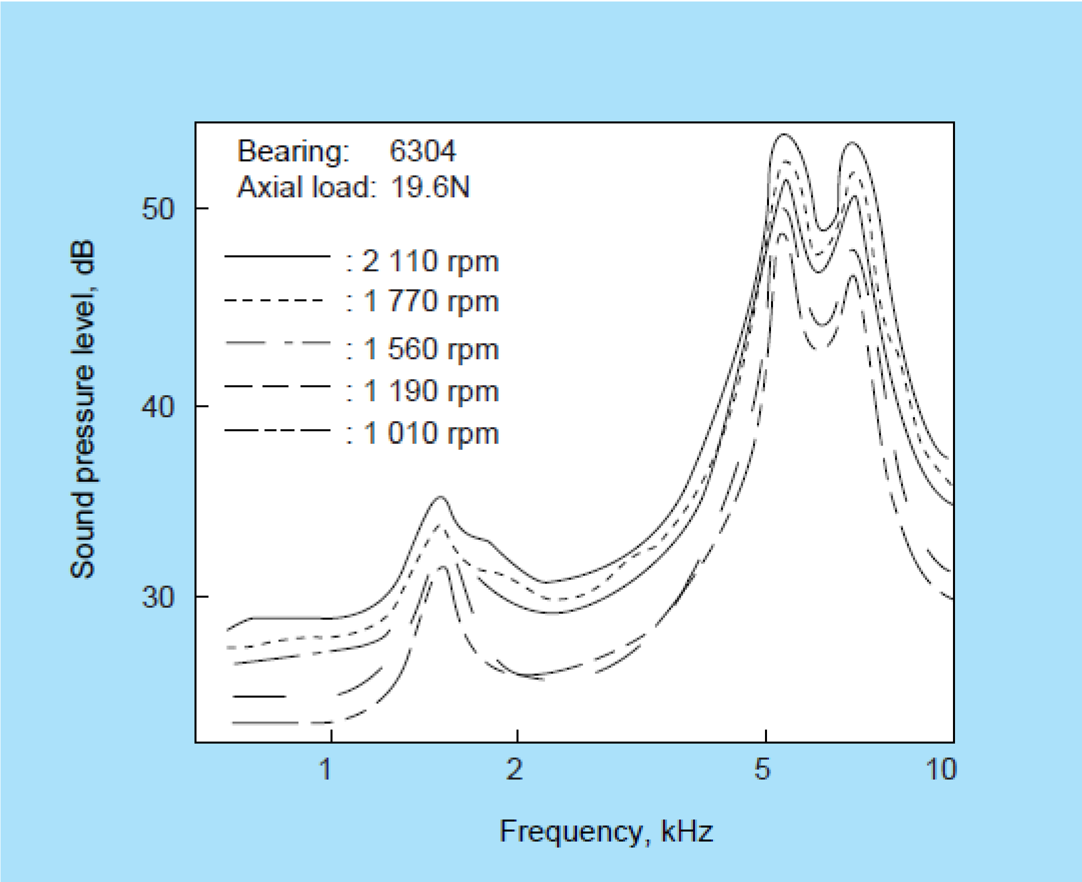



Fig. 7.2 Influence of rotation speed on race noise from [6]

Configuration errors will cause race noise and race noise isn't able to be avoided even if the most high-tech processing is used for processing of the raceway surfaces and the rolling elements. Because of this ripple, the contact during bearing operation between the raceway ring and the rolling element acts as a slightly fluctuating spring. On the raceway ring of the bearing this spring variation behaves like the exciting force. Hence, vibration and race noise are produced. The generation of racial noise can’t be avoided, and there is no special solution to eliminate it entirely. However, through improving the whole quality and accuracy of the bearing it can be minimized.

- Click noise

Click noise often occurs in relatively big bearings bearing radial load. It only happens at low speeds and after the speed exceeds a certain range it dies away. The simple similarity for this noise is "kata ---(like the voice of a clock)" The generation of this type of noise is considered to occur as shown in figure 7.3. When the bearing is running under the radial load, within the bearing, load areas and no-load areas coexist. Inside the bearing in the no-load area, exists some clearance – there is no contact between rolling elements and the inner ring, because of centrifugal force (Fc2) rolling elements impresses the outer ring. At low-level rpm, however, while gravity (W1) is larger than the centrifugal force, the rolling element fall and collisions arise with the internal ring. The impinging produces a clattering noise between the rolling elements and/or inner ring.

Preloading is a useful way to reduce click noise. Less benefit is obtained from reduction of radial clearance. Another strategy is to make use of lightweight material for manufacturing rolling elements.

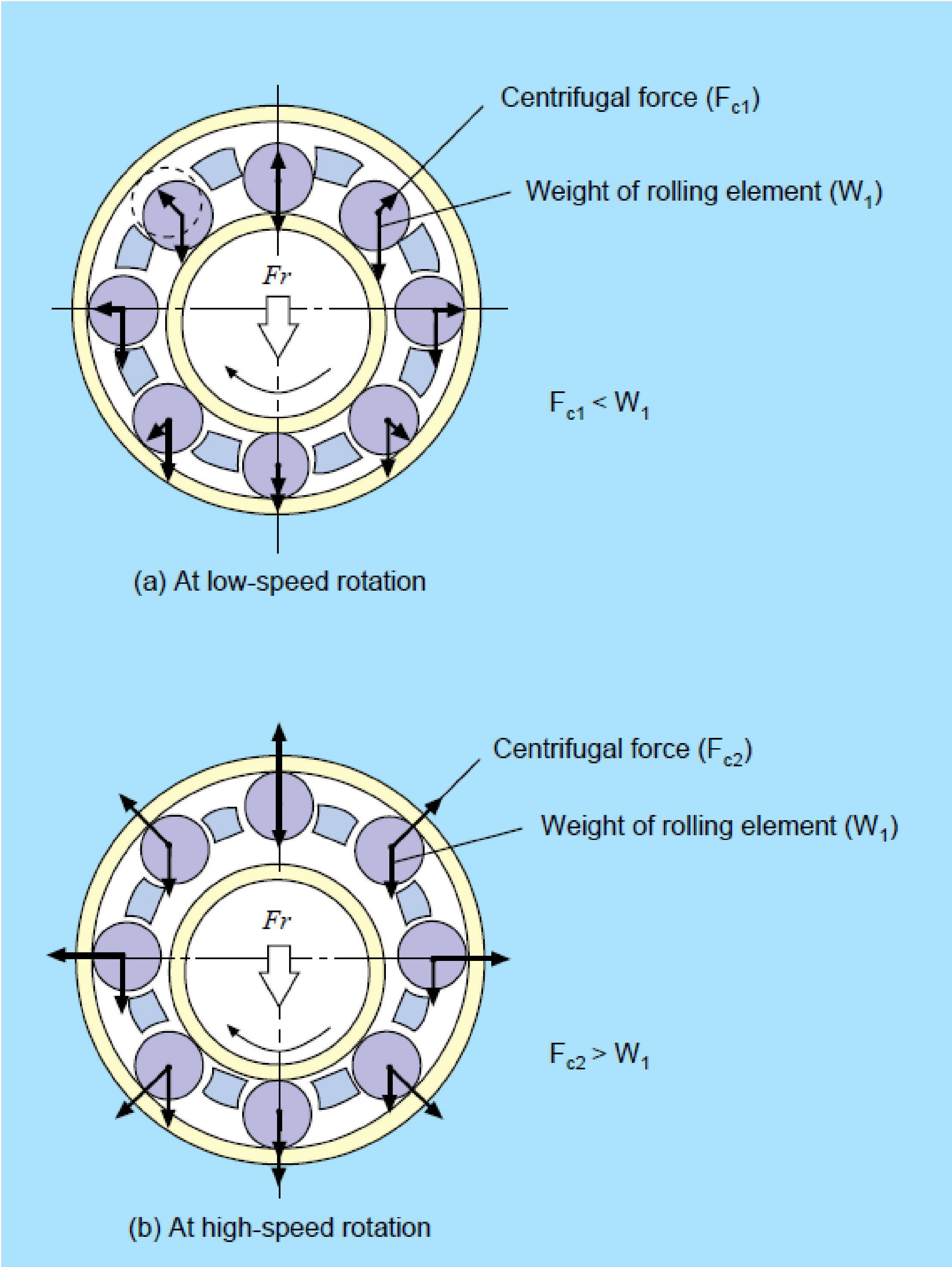



Fig. 7.3 Mechanism of rolling element falling noise from [6]

●Squeal noise

Squeal noise is a metallic noise that could be quite obvious sometimes. Its voice likes metal sliding over metal. Bearing temperatures do not clearly rise due to squeal noise and the bearing and grease life are not stricken. Practically, there are no other obstacles to bearing operation expect noise problems. The use of relatively large bearings under radial loads gives rise to produce “screaming” noise. It often appears in cylindrical roller bearings, may also be possible in ball bearings. Squealing noises are characterized by:

(1) large radial clearance leads easily to occur.

(2) mainly occurs in the case of grease lubrication, but oil lubrication almost impossibly results in squeal noise.

(3) more commonly in cold situation.

(4) it appears within a sure speed interval and when the bearing size becomes larger, it tends to be lower.

(5) its production is different and aleatory, just like incidence, the type and amount of grease and the operating conditions of the bearing have effect on it.

The friction between the outer ring and the rolling element is believed to be the reason of the squeal noise due to the lubrication and vibration of the outer ring used. The screaming noise is due to self-induction

Vibration related with lubrication. But there's a big question: which part of the outer ring is giving out the screaming voice? The commission is still discussing on the issue because someone believes the noise occurs in the loading zone, while others believe it happens in the no-load zone. Reduction of radial clearance and adopting rather narrow grooves in the bearing outer ring raceway lead to effective decrements for this noise.

- Cage noise

Cage noise can be classified into two main types: one is low frequency noise (" gaga ---screak") and the other one implying a cage colliding with a rolling element or bearing ring likes (" kacha-

--- clickety-clack"). "K noise" and "G noise" respectively represent " kacha---- clickety-clack" and " gaga ---screak" in shortening. K noise can occur in all kinds of bearing, and usually its size is relatively low. This noise has following features:

(1) it occurs with various cages manufacturing in different materials.

(2) only few lubrication methods can lead to cage noise, such as: grease and oil lubrication.

(3) if torque loads are impressed on the outer ring of the bearing, it is easy to arise.

(4) larger radial clearance makes it easily appear more frequently.

The collision of the running cage with the rolling element or raceway rings gives rise to K noise. Due to the gap existing between the components involved in collision, it is almost impossible to totally kill the K noise. But it can be limited by less installation errors.

G noise is a special kind of noise. Sliding friction between the guide surface of the cage and the bearing parts of the cage more easily causes this type of noise due to the self-generated vibration of the cage. Its volume often becomes greater.

Considering above discussion, a specially designed bearing satisfying the aim of limiting this kind of noises shown in figure 7.4. Figure 7.4 comparison of the behaviour of traditional bearings without effective strategy for limiting noise with the dedicatedly designed one. The bearing is installed on a spindle, and since at a lower range of temperature G noise can appear more frequently, hence, during the experiment process the environment temperature gradually goes down. For the dedicatedly designed bearing cage, even if the environment temperature is below 0 ° C, such a noise will not arise. Lubrication method of G noise produced is grease and G noise happens more frequently with relatively hard grease. Because of the better choice of lubricants, the dedicatedly designed bearing cages can effectively minimize G noise.

A cage noise with big volume ("gacha---") may happen during the first phase of rotation when a radial load is impressed onto a bearing with hard grease. While the rolling elements suddenly slow down due to grease resistance after leaving the load area, consequently, the collision generated by rolling elements with the cage tends to cause this noise. As time goes on, this noise is considered to be worthless and dies away.

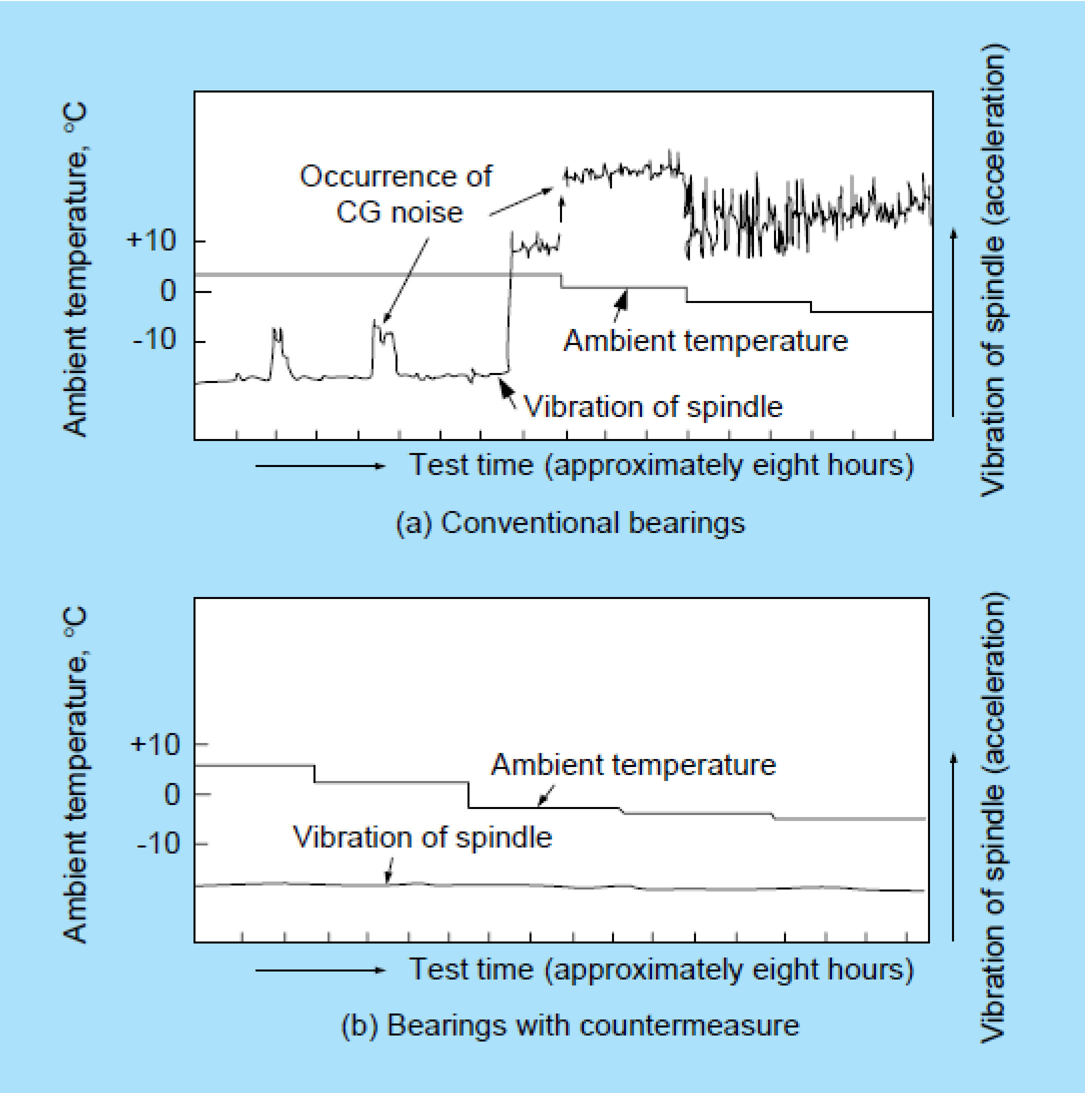



Fig. 7.4 Performance evaluation test of cage noise (G noise) from [6]

- Rolling element passage vibration

When rolling bearings operate under radial load, the rolling element passage vibration will cause troubles. When this vibration arises, the rotating center of the axis moves along un-down or left-right direction. The number of rolling elements, radial clearance and radial load have effects on the displacement of the vibration. This kind of vibration is related to the change in load distribution, which depends on the position of each rolling element in the load area.

Analyzing situations (a) and (b) in Fig. 5, the vertical displacement of the rotation axis is a little different. The axes also move laterally under the conditions in figure 5. The formula of the frequency of this type of vibration can be obtained from theoretical studies and experimental results:

$f = Z \cdot fc$ [Hz]

Z: Number of rolling elements

fc: Orbital revolution frequency of rolling elements [Hz]

Usually the rolling element passage vibration scarcely causes trouble because of its such tiny displacement. While in trouble, it is useful and effective to make the radial gap narrow or undergo a preload.

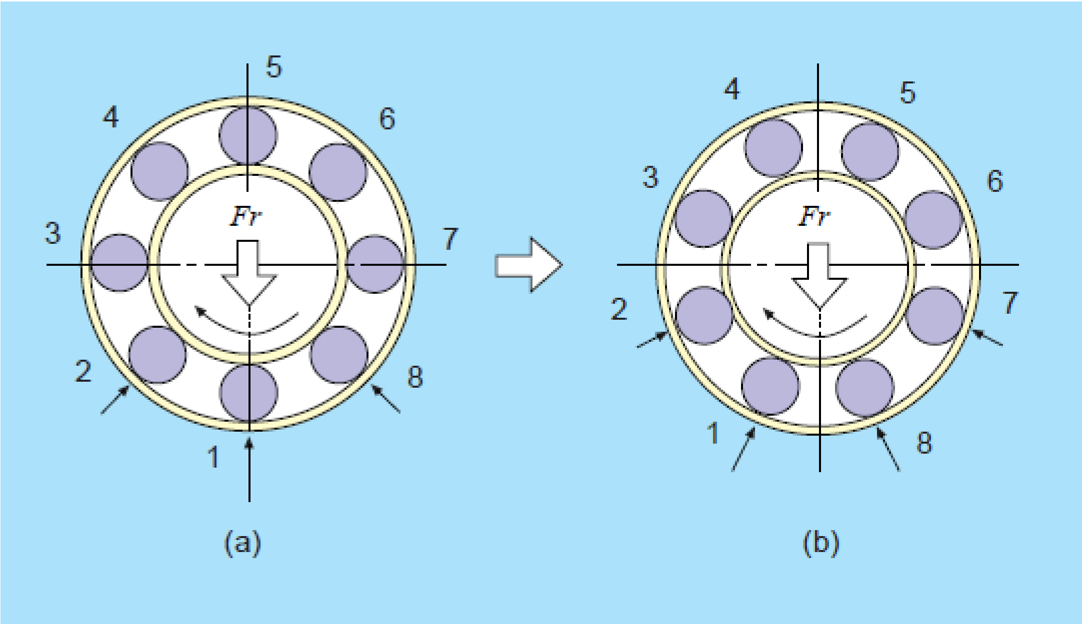



Fig. 7.5 Mechanism of vibration due to rolling elements passing from [6]

ii. Vibration and sound related to bearing manufacturing

Vibration and sound caused by bearing manufacturing, the most important one is generated by ripples, can't be ignored. Even with now day advanced manufacturing techniques, a ripple element can cause sound and vibration although considering negligible. Ripple noise may be a trouble only when the ripples are anomalous.

Different from other sounds produced by rolling bearings, the speed has effect on the frequency of ripple noise. This is a special feature of ripple noise that can be recognized from all kinds of noise. A ripple noise is jarring and has a certain frequency when the rotating speed is invariable. But when accelerations decelerations happen to the bearing, the ripple noise becomes rather jarring and the frequency is varying according with speed is varying. As the vibrations generated by the ripples of the raceway and/or rolling element are larger than a certain level, consequently this type of noise is troublesome.

Gustafsson has done basic researches about vibration produced by ripples. In his opinion, the ripple on inner and outer ring raceway surface belongs be a set of sinusoidal waves. In the presence of these ripples, he tested the force balance within the bearing and got the results of bearing vibration through calculations. As a result, the number of ripple upper limit tending to the vibration and the frequency of the vibration has been defined by means of a set of experiments. The findings are reflected in the section on the frequency of ripple noise in table 1. Ripples with a special number of peaks are responsible for such vibration, that is easy to find out from table 1. When there is a ripple with (nz-1) peaks on the internal ring, radial vibration appears, this situation is shown in figure 6.

Applying a constant axial load and the presence of a ripple only existing in one component is related to Gustafsson's analysis of the vibration. Because of his study, the research about the presence of ripples in two components and the applying a radial load have been done by other people. From these researches, we can find that an extra amount of ripple peaks also tends to vibrations. Although it has been demonstrated in many cases that the amount of vibration and noise generated by the number of peaks derived by Gustafsson and shown in table 1, but insufficient results have been collected to demonstrate this condition.

Reducing the circumferential corrugation on the finished surface of the bearing assembly can effectively decrease this vibration that may cause a problem. In addition, we can know that if there is a ripple on the surface of the shaft or the bearing housing, the ripple is mirrored onto the raceway surface of the bearing and consequently a vibration arises.

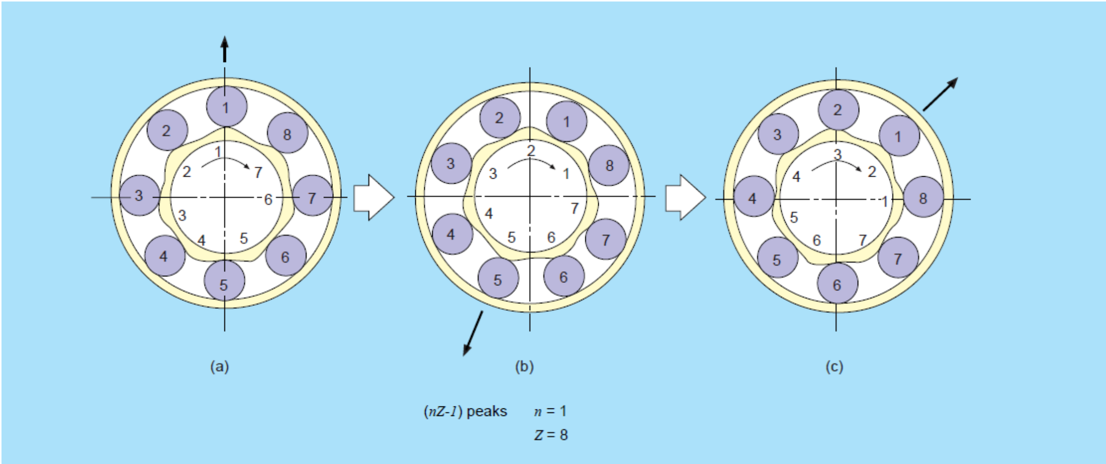



Fig. 7.6 Mechanism of vibration due to waviness of raceway or rolling element surface from [6]

iii. Vibration and sound due to improper handling

The surface of the main bearing parts is usually rather hard. However, if the bearing falls or withstands an impact over a time duration, it tends to create indentation on the surface of the bearing part. Vibrations or sounds can be produced by even slight shape flaws. On the side, just little pollution can make dents and flaws happen. Improper bearing handling is the major reason to cause the vibration and sound.

Type of vibration and sound because of improper handling

- Flaw noise
- Contamination noise

- Flaw noise

When we find imperfection like dents or rust is found on the surface of the finished raceway of the rolling bearing, it is easy to hear that when the bearing rotates pulsating machine-gun noise sounds. When a rolling element impacts a defect on the surface of the raceway generating this noise, we name it as flaw noise. We can find a thing from table 1, the frequency of flaw noise is identical with frequency of racial noise. The range of the whole spectrum goes up after flaw noise is produced. According to such a fact, haploid frequency analysis is not enough to recognize it from other noises. Comparing to other types of noise, defect noise has a specified generation period as shown in figure 7.7. The period of noise generation is constant under the condition of a constant velocity, but the period of noise generation costs more time with a reduction of speed.

The speed and the bearing's internal specifications together determine the period of noise generation. The fact as described in table 1, the location of defects on the bearing varies with varying the period of noise generation. Defects can be identified because of this feature, and, if they do exist, their location can be determined. To evaluate the producting period of this noise, we could employ an approach called envelope analysis.

If the ball bearing has blemishes on the ball, but not surely on the ball's rolling surface, the probability for noise occurring due to blemishes is not 100 percentage. Such noise generation and the feature of a period of noise generation always appears together.

noise is concealed and a reduction occurs to the corresponding noise range as high viscosity oils are employed. Besides, replacement of bearings should usually be done while noise within the rolling element is generated due to defects.

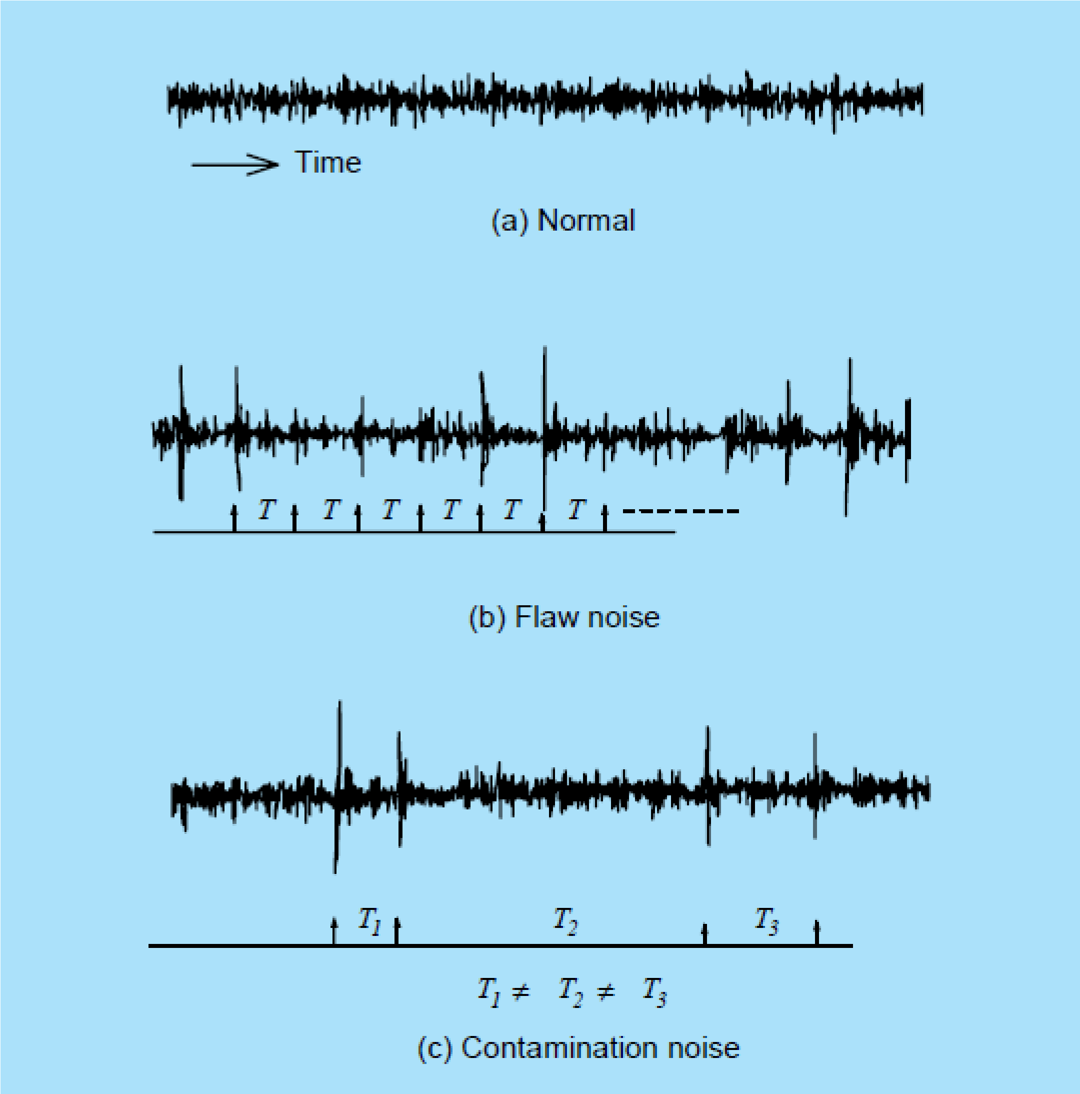



Fig. 7.7 Waveform of noise due to flaw and contamination from [6]

- Contamination noise

Due to incorrect operation or severe operating conditions, the entry of foreign particles into the bearing space will lead to contamination noise. The case usually occurs if foreign particles are trapped into the space between the rolling elements and the raceway surfaces. The order of pollution noise is various and its production is unconventional (figure 7). As the size of the bearing becomes smaller, the effect of foreign particles will be more important. The problem caused by contamination noise often happens if bearings showing small or very small dimensions.

Preventive strategies should be employed due to the entry of foreign particles which will not only generate noise, but also create indentations on the rolling surface of the bearing tending to decrease the bearing life.

### 7.3 Vibration measurement tools

**Machine monitoring**

We consider the abnormal vibration generated by various cases as the basic sign of a possible machine failure. The cases with imbalance, misalignment, components losses, worsening rolling element bearings and gear damage tend to cause such vibrations. By make use of vibration analysis devices and systems it is effective to help us detect many grave problems early, enabling us to carry out remedial activity in time.

**Noise and vibration tester**

The fundamental reasons, for example: local defects in corrugated bearing parts, rings and balls, or dirt particles within bearing space, will tend to noisy cases. Although low noise and low vibration are going to be more crucial than the basic conventional things of bearing such as stiffness, load capacity, speed limitation and service life which play key roles in cases.

The main function of bearing vibration device is to exam all these bearing quality problems.

Preventability of almost all problems and failures resulting from vibration analysis can be achieved through monitoring the instantaneous state of rolling bearings.

### 7.4 Condition monitoring

**Waviness Testing**

Many year ago, the ripple component check has been carried out. The check is employed to assess the degree of radial deviation from the true circle on the circumference of the component. This is achieved by rotating the part on the main shaft supported by fluid dynamics and using a touching sensor which is located perpendicularly to the surface of the part, as represented in figure 7.8. a stylus, which is called transducer, moves on a radial fluctuation and generates a voltage output whose value is proportional to the stylus movement or an instantaneous rate of varied amount of stylus movement.

The signal from the transducer is varying with the rate at which the pin measures the difference in movement. This ratio within a wide frequency range permits to undergo a regulare big speed during tests.

The voltage signal obtaining from the transducer is enlarged and converted from an analog signal into a digital signal for input into the computer which only accept the digital signal, and the spindle speed of the pin system is measured corresponding to the movement change.

A computer usually uses the bandpass filter for the signal to determine shape mistakes within a selected wavelength interval and to process frequency spectrum study in order to evaluate the corresponding amplitude at a certain frequency. In order to identify many components that have been inspected or rejected, and to give useful information about exact manufacturing process require the data is able to be compared to specifications.

FIGURE 7.8 Bearing component waviness tester. (Courtesy of Timken Company.) from [1]

Some kinds of defects on a component that is not able to be avoided are due to the inherent features of the component. Ripple testing for these defects may not be sufficient, because ripple testing may only obtain circumferential data from one or two axial locations on the component under test. Hence it is necessary to consider the device possessing a number of potential axis of rotation, called Bal ls. In consequent, a clearer guarantee for the final bearing quality is able to be achieved by the visual element check and vibration examination of the entire bearing.

**Vibration Testing**

If defects are not found in the wave-degree test, the component vibration test can be used to permit to check the damage that takes places during installation.

in vibration testing, detection is also useful and effective for some types of shaped problems. These kinds of geometrical issues include, such as, oversized rolling components, each intersecting groove in the raceway, or the raceway reaching the side of the raceway. On the other side, dirt or poor oil quality can make tests contaminative.

Figure 7.9 shows a vibration test device operated by hand specified for a relatively small bearings with an outside diameter up to 100mm. automatic versions for larger diameter bearings in production line implementation is similar to the equipment for the small size. testing station and vibration signal analysis device are combined together as the main components of the system for vibration test. a hydrodynamic spindle, an air cylinder for applying load to the bearing under test, and a regulable slider for locating the speed sensor, all these components together are combined into a test station. The spindle is driven by a motor belt mounted under the support. figure 7.10 clearly represents the schematic diagram of the system.

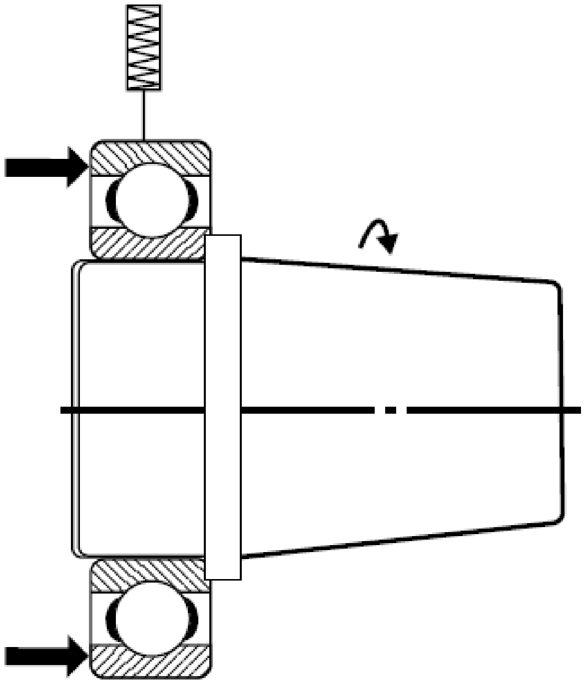


FIGURE 7.9 Bearing vibration tester: schematic of bearing loading and accelerometer on bearing outside diameter from [1]

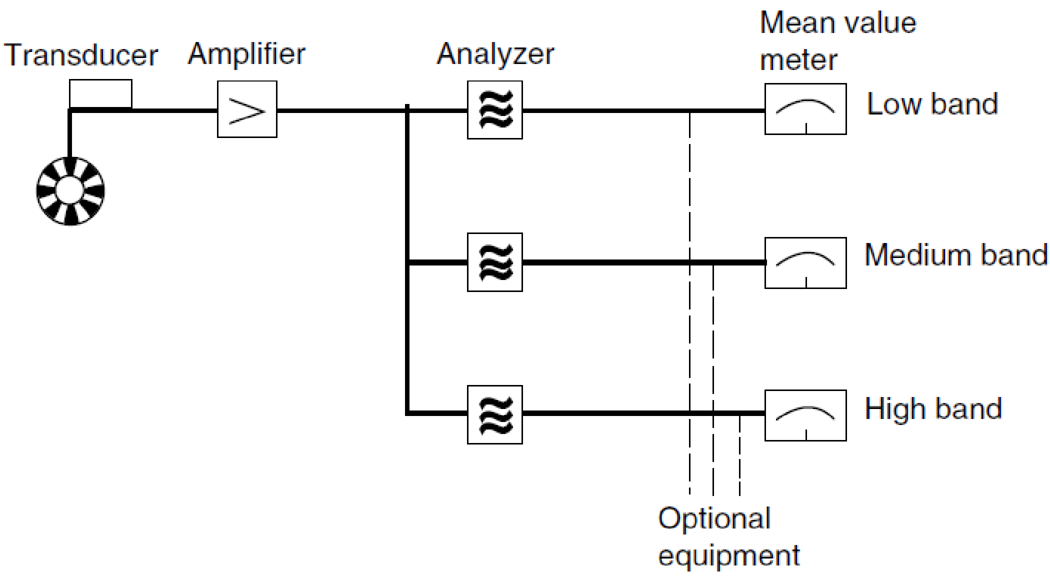



FIGURE 7.10 Schematic of bearing vibration measurements from [1]

The bearing's inner raceway is installed on a precision axis fixed to the spindle. On the side of the nonrotating outer ring there is the specified thrust load impressing on it.

The tip of the speed sensor is slightly loaded on the outside diameter of the outer ring. The loading tool is composed of a thin-walled steel rings molded into neoprene rings; The ring touches the side of the outer ring. The tool and load combination are flexible enough to allow the radial movement of the outer ring to occur when the ball rolls over the surface of the wave or the defects in the ball groove. The amplifier input is the voltage signal from the sensor, converted into a digital signal for input to the computer, which has a band-pass filter and displays the root-mean-square velocity of each frequency band. Slower rotational speeds are employed in large bearing test.

Other analytical manners, like the top check of asynchronous issues, can also be employed, often filtered, or with minimal impact on the time average results during using digital computer.

Over the years, bearing manufacturers and customers have successfully made use of elementary test techniques. Many improvements have been obtained during this period and the field of vibration measurement is also going to be developed further. These efforts include research into alternative transducer design and system calibration procedures, different methods for applying load to test bearings, the application of statistical methods in setting product specifications and analyzing test results, and complementary methods for implementing signal analysis.

The bearing effects on the vibration of the machine due to the inherent design features or defects of the ions and the deviation of the ideal operating geometry in the bearing have been clearly discussed above in this chapter. Such defects and shaped errors may occur during producing bearing components, during the assembly of the bearing into the machine or during the deterioration of the bearing during operation. Changing stiffness properties or acting as a force source that directly produces vibration are significant impacts on machine vibration by those defects or errors.

The fatigue failure of the bearing in the machine can be detected and the position of the failing bearing can be determined by using the condition monitoring.

**7.5 Reduced noise and vibration**

The importance of industry issue is growing little by little about excessive vibration and noise levels related to operating large fans. premature failure of device and costly maintenance, often including unplanned downtime and production losses is mainly caused by excessive vibration. Higher the vibration levels larger the energy consumption. In other way, higher the noise levels worse the working conditions.

Very smooth and quiet bearing operation can be achieved by a reduction of the noise and vibration during operating condition.

In order to precisely detect and find failures, it is useful to employ the high frequency technical analysis and measurement. Specific customer requirements can be achieved by spectrum mask which helps optimize bearing behaviors.

Based on experience results, experts can use the machine's measurements to detect faulty processing steps, such as lack of honing. The introduction of the global calibration system enables the vibration equipment to operate in accordance with international standards.

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