ME 4 2703 Product Design

Gear Box Project

by

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Bevel Gearbox for Mädler Gear Set

Article Numbers: 36115600 & 36115700

Designed for: $T_{per,P} = 3.4\ Nm$

$T_{per,G} = 6.8\ Nm$

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List of Symbols and Abbreviations

d_m	Pitch diameter [mm]
d_a	Outer diameter [mm]
δ	Pitch angle
α	Pressure angle
F_T	Force due to produced torque
F_A	Axial force
F_R	Radial force
L_1	Distance between bearing A and gear1
L_2	Distance between bearing B and gear1
L_3	Distance between bearing C and bearing D
L_4	Distance between bearing D and gear 2
$R_{A,B,C,D}$	Reaction forces due to bearings
$d_{shaft\ 1,2}$	Inner diameter of splined keyways on shaft
$D_{shaft\ 1,2}$	Outer diameter of splined keyways on shaft
n	Number of splined keyways on shaft
B	Length of each individual keyway
$n_{shaft1,2}$	Speed of shaft
C	Dynamic load capacity of bearing
$P_{A,B,C,D}$	Equivalent force on bearing
L	Bearing lifetime
d_w	Diameter of bolt head
l_k	Length between bolt and nut
d_h	Core bolt diameter
E	Youngs modulus
$\delta_{p,H,T,S,1,N}$	Resilience
ϕ	Operation load factor
F_{KR}	Clamping force
F_V	Pretension
α_k	Assembly factor
R_m	Tensile strength
$R_{p0.2}$	Yield strength
S_F	Safety factor

List of Figures

1 Free Body Diagram

1.1 Preliminary Design

The aim of the project is to design a gearbox for the Mädler gear set, articles 36115600 and 36115700. The shafts of the two gears are perpendicular to each other forming a bevel gear arrangement. The gears are enclosed in the housing which consists of holes for the input and output shafts. As the gearbox is used for power transmission and has rotating elements thus there are several bearings in the gearbox. The gearbox consists of several standardized as well as non-standardized machine elements.

The transfer of torque is done by splined shaft keyways between shafts and gears. The pinion (361115600) is the driven gear and hence is connected to the output shaft whereas the larger gear (36115700) is the driving gear and is connected to the input shaft.

The setup of the gearbox can be easily understood by the drawing below:

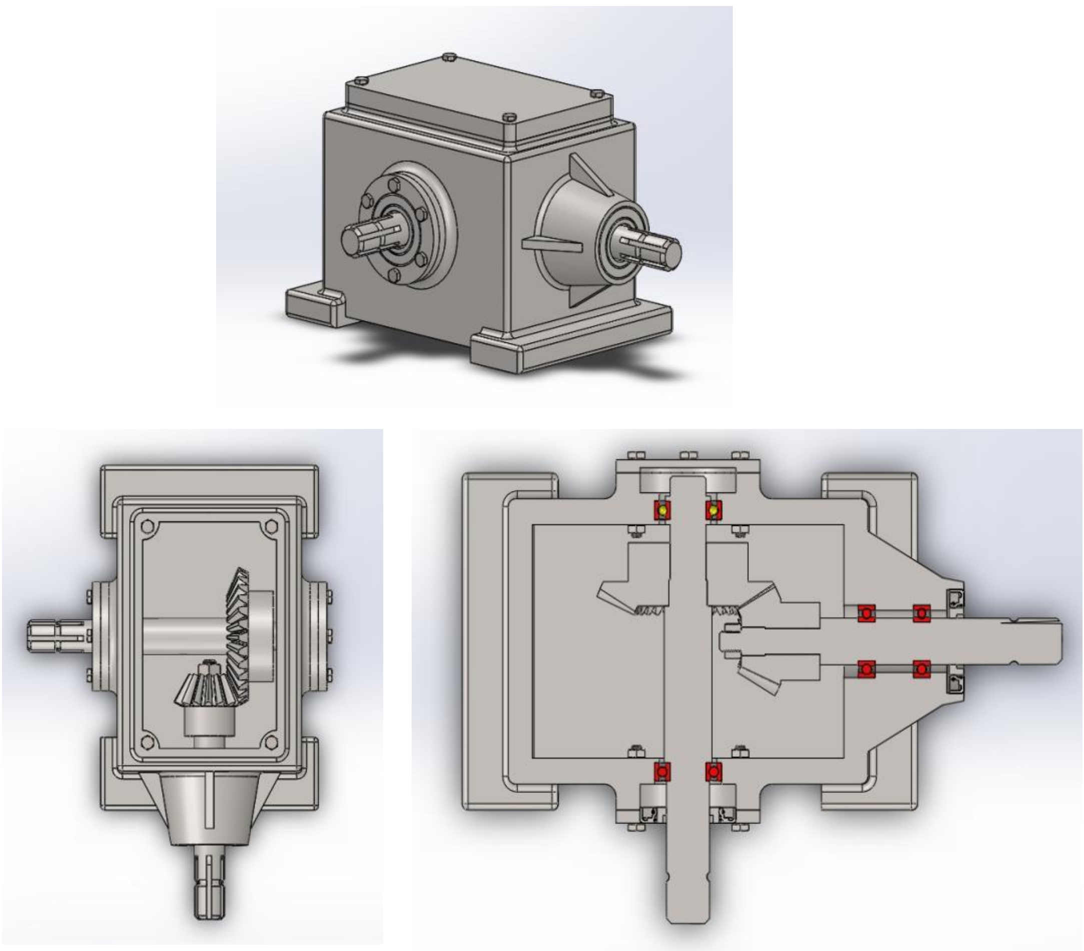


Figure 1: General Setup

1.2 Required Machine Elements

The major machine elements and their functions are:

- Housing- The gears are enclosed in a housing for proper functionality
- Gears- The gears are standardized elements, specifically, Mädler article numbers 36115600 and 36115700.
- Input and Output shafts- These shafts are connected to the gears and hence are important for power transmission. The dimensions of the shafts are chosen to ensure an interference fit of the shaft with the gears.
- Roller bearings- These elements facilitate the movement of the rotating elements.
- Bolted Fasteners- These support the radial loads due to the bearings
- Washers- Avoids loosening of the bolted fasteners over time.
- Housing lid- The lid of the housing is a separate element that can be opened regularly to facilitate maintenance.
- Sealing- To avoid leakage of lubricants

1.3 Calculation of Forces and Moments

1.3.1 Gears

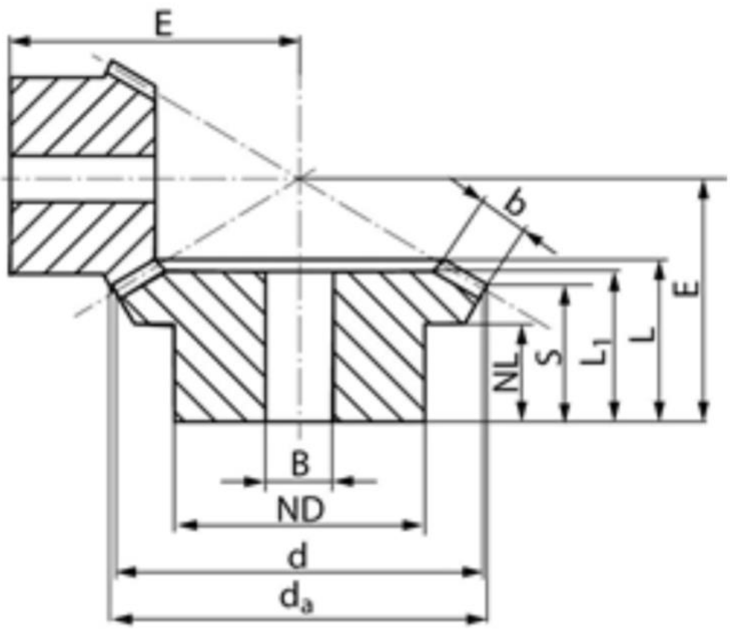


Figure 2: Gear Dimensions

By using Pythagoras theorem, we can calculate the approximate value of the pitch circle (d_{m1}) which is nothing but the mean of other and inner diameter:

$$d_{m1} = d_a - \sqrt{b^2 - (L - S)^2}$$

For Gear1:

d_a	b	S	L
77.3	17	21.6	28.1

$$d_{m1} = 61.59\ mm$$

For Gear2:

d_a	b	S	L
42.2	17	18.6	33.3

$$d_{m2} = 33.66\ mm$$

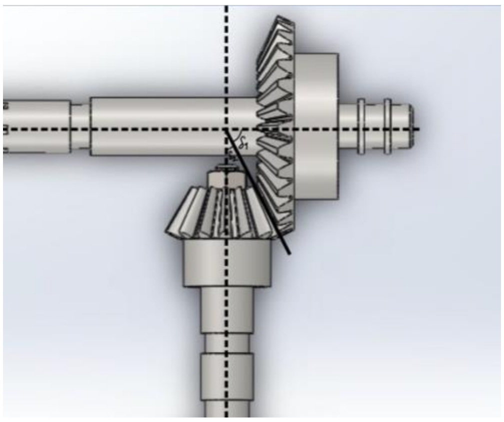


Figure 3: Pitch angles

By geometry as seen in the Figure above, it was found that:

$\delta_1 = 61.34°$

$\delta_2 = 28.66°$

The standard values for pressure angle vary from 15° and 22.5° so for the further calculations, the value of pressure angle (α) was assumed to be 20°.

δ_1	δ_2	α	d_{m1}	d_{m2}
61.34°	28.66°	20°	$61.59\ mm$	$33.66\ mm$

The force acting on the gear due the produced torqued $T_{per,Gear1}$

$$F_{T,Gear1} = \frac{T_{per,Gear1}}{d_{m1}/2}$$

$$F_{T,Gear1} = \frac{6.8\ Nm}{61.59\ mm/\ 2}$$

$$F_{T,Gear1} = 220.81\ N$$

Axial force on the gear 1:

$$F_{A,Gear1} = F_{T,Gear1} \cdot \tan\alpha \cdot \sin\delta_1$$

$$F_{A,Gear1} = \ 70.52\ N$$

Radial force acting on gear 1:

$$F_{R,Gear1} = F_{T,Gear1} \cdot \tan\alpha \cdot \cos\delta_1$$

$$F_{A,Gear1} = \ 33.82\ N$$

Axial, Radial and Tangential force on Gear 2 (pinion gear):

$$F_{A,Gear1} = \ F_{R,Gear2} = \ 33.82\ N$$

$$F_{R,Gear1} = F_{A,Gear2} = \ 70.52\ N$$

$$F_{T,Gear1} = F_{T,Gear2} = \ 220.81\ N$$

As the gears are at 90° to each the force will be same but opposite in direction and radial force for one gear will be axial force for another gear and vice versa.

1.3.2 Shaft 1

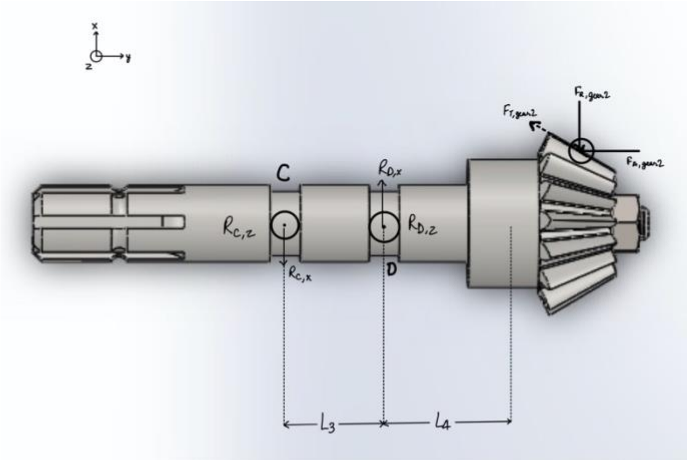


Figure 4:Shaft 1 (input) and Gear 1

Shaft 1 was connected to Gear 1 that is the pinion gear:

L_3	L_4
$23\ mm$	$32.94\ mm$

Taking moment at Bearing D in Z direction,

$\sum M_{D,Z} = 0$

$F_{R,Gear2} \cdot L_4 - F_{A,Gear2} \cdot \left(\frac{d_{m2}}{2}\right) - R_{C,x} \cdot L_3 = 0$

$R_{C,x} = -3.07\ N$

Taking Sum of forces in x direction

$\sum F_x = 0$

$R_{D,x} + R_{C,x} - F_{R,Gear2} = 0$

$R_{D,x} = 30.74\ N$

Taking moment at Bearing D in X direction,

$\sum M_{D,x} = 0$

$-F_{T,Gear2} \cdot L_4 + R_{C,z} \cdot L_3 = 0$

$R_{C,z} = 316.25\ N \ \otimes$

Taking Sum of forces in z direction

$\sum F_Z = 0$

$R_{D,z} - R_{C,z} - F_{T,Gear2} = 0$

$R_{D,z} = 537.07\ N\ \odot$

Total Radial reaction at bearing C:

$R_C = \sqrt{R_{C,z}^2 + R_{C,x}^2} = 316.27\ N$

Total Radial reaction at Bearing D:

$R_D = \sqrt{R_{D,z}^2 + R_{D,x}^2} = 537.95\ N$

1.3.3 Shaft 2

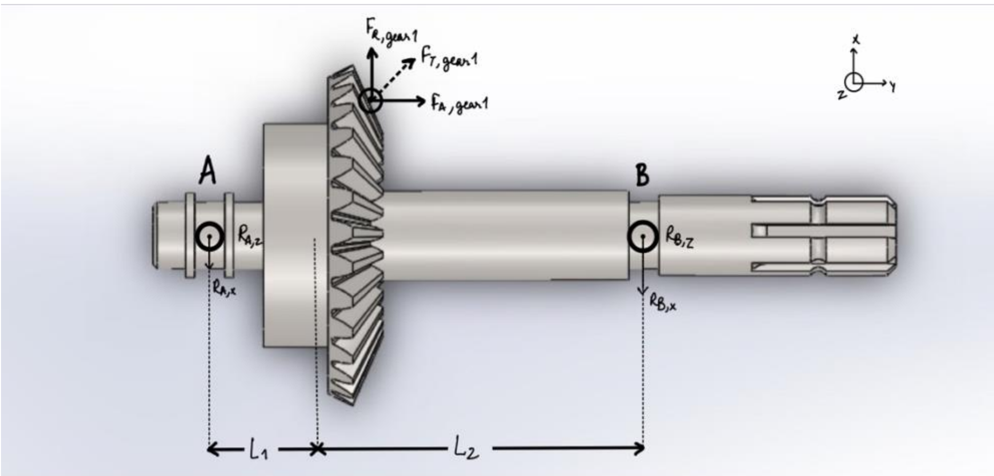


Figure 5: Shaft 2 (Output) and Gear 2

Shaft 2 was connected to Gear 2.

L_1	L_2
$28.74\ mm$	$80.44\ mm$

Taking moment at Bearing A in Z direction,

$\sum M_{A,Z} = 0$

$F_{R,Gear1} \cdot L_1 - F_{A,Gear1} \cdot \left(\frac{d_{m1}}{2}\right) - R_{B,x} \cdot (L_1 + L_2) = 0$

$R_{B,x} = 9.03N$

Taking Sum of forces in x direction

$\sum F_x = 0$

$-R_{A,x} - R_{B,x} + F_{R,Gear1} = 0$

$R_{A,x} = 61.49\ N$

Taking moment at Bearing A in x direction,

$\sum M_{A,x} = 0$

$F_{T,Gear1} \cdot L_1 - R_{B,z} \cdot (L_1 + L_2) = 0$

$R_{B,z} = 58.13\ N$ (Inside the paper) ⊗

Taking Sum of forces in z direction

$\sum F_Z = 0$

$R_{A,z} - R_{B,z} + F_{T,Gear1} = 0$

$R_{A,z} = 162.68\ N$ (Outside the paper)⊙

Total Radial reaction at bearing A:

$$R_A = \sqrt{R_{A,z}^2 + R_{A,x}^2} = 173.92\ N$$

Total Radial reaction at Bearing B:

$$R_B = \sqrt{R_{B,z}^2 + R_{B,x}^2} = 58.83\ N$$

Bearings A and C are locating bearings and Bearings B and D are non-locating bearings

Hence, the axial and radial forces on all bearing are given below:

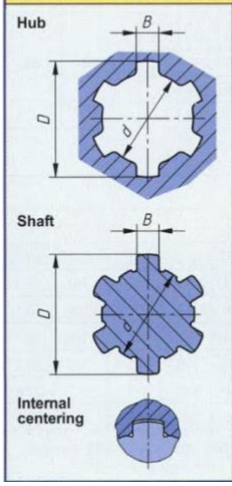
	Bearing A	Bearing B	Bearing C	Bearing D
Radial Force	$173.92\ N$	$58.83\ N$	$316.27\ N$	$537.95\ N$
Axial Force	$33.82\ N$	$0\ N$	$70.52\ N$	$0\ N$

2 Keyway Calculation

Shafts have been splined as shown in Figure 6 for smooth transfer of torque. Feather keyway is not used as this would add additional moving parts to the design.

2.1 Shaft 1

Splined shaft joints with straight flanks and internal centering cf. DIN ISO 14 (1986-12)



	Light series			Medium series				Light series			Medium series		
d	*N*[1]	*D*	*B*	*N*[1]	*D*	*B*	*d*	*N*[1]	*D*	*B*	*N*[1]	*D*	*B*
11	–	–	–	6	14	3	42	8	46	8	8	48	8
13	–	–	–	6	16	3.5	46	8	50	9	8	54	9
16	–	–	–	6	20	4	52	8	58	10	8	60	10
18	–	–	–	6	22	5	56	8	62	10	8	65	10
21	–	–	–	6	25	5	62	8	68	12	8	72	12
23	6	26	6	6	28	6	72	10	78	12	10	82	12
26	6	30	6	6	32	6	82	10	88	12	10	92	12
28	6	32	7	6	34	7	92	10	98	14	10	102	14
32	8	36	6	8	38	6	102	10	108	16	10	112	16
36	8	40	7	8	42	7	112	10	120	18	10	125	18

Tolerance classes for the hub					
Not heat treated dimensions			Heat treated dimensions		
B	*D*	*d*	*B*	*D*	*d*
H9	H10	H7	H11	H10	H7

Tolerance classes for the shaft			
Dimen.	Type of fit: Sliding fit	Transition fit	Press fit
B	d10	f9	h10
D	a11	a11	a11
d	f7	g7	h7

⇨ **Shaft (or hub) DIN ISO 14 – 6 x 23 x 26:** *N* = 6, *d* = 23 mm, *D* = 26 mm

[1] N number of splines

Figure 6: Splined Keyway Selection

$d_{shaft\ 1}$	$D_{shaft\ 1}$	n	B	$T_{per,P}$
$11\ mm$	$14\ mm$	6	$3\ mm$	$3400\ Nmm$

2. Berechnung[1]

Eine Berechnung von *Keilwellenverbindungen* ist bei ausreichendem Wellendurchmesser (maßgebend ist der Kerndurchmesser) und normalen Nabenabmessungen (s. **TB 12-1**) nicht erforderlich. Nur bei sehr kurzen Naben ist eine Nachprüfung der Flächenpressung an den „Keil"-Flächen zweckmäßig. Mit der Annahme, dass durch nicht zu vermeidende Herstellungsungenauigkeiten nur ≈75 % der „Keile" tragen, wird die *vorhandene mittlere Flächenpressung*

$$\boxed{p_m \approx \frac{2 \cdot T}{d_m \cdot L \cdot h' \cdot 0{,}75 \cdot n} \leq p_{zul}} \qquad (12.2)$$

T zu übertragendes Drehmoment; bei dynamischer Belastung $T = K_A \cdot T_{nenn}$, bei statischer Belastung $T = T_{max}$
K_A Anwendungsfaktor nach TB 3-5
d_m mittlerer Profildurchmesser aus $d_m = (D + d)/2$ mit D und d nach **TB 12-3a**
L Nabenlänge gleich tragende Keillänge
h' tragende Keilhöhe; unter Berücksichtigung der Fase f wird $h' = (D - d)/2 - 2 \cdot f \approx 0{,}4 \cdot (D - d)$
n Anzahl der Keile aus **TB 12-3a**
p_{zul} zulässige Flächenpressung des „schwächeren" Werkstoffes (meist Nabe). Anhaltswerte für p_{zul} nach **TB 12-1**

Hinweis: $L \leq 1{,}3 \cdot d$ wählen, siehe Hinweis zur Gleichung (12.1)

Bei der *Zahnwellenverbindung* kommt, wie bei der Keilwellenverbindung, eine Nachprüfung auf Flächenpressung nach Gl. (12.2) in Frage. Abweichend von den hierin benutzten Größen sind entsprechend den in DIN 5480 bzw. DIN 5481 und im **Bild 12-7** angegebenen zu setzen: für

$d_m = d_5 = d$ Teilkreisdurchmesser der Verzahnung
$h' \approx 0{,}5[d_{a1} - (d_{a2} + 0{,}16 \cdot m)]$ für die Passverzahnung mit Kerbflanken DIN 5480
$h' \approx 0{,}5(d_3 - d_1)$ für die Passverzahnung mit Evolventenflanken DIN 5481

(Werte aus **TB 12-4** oder den jeweiligen DIN-Normen entnehmen).

Figure 7: Splined Keyway Calculation

The permitted pressure of shaft 1: $C45 = 580\ \mathrm{N/mm^2}$

$$p_{per} = \frac{R_e}{S_F} = \frac{580\ N/mm^2}{1.5} = 386.67\ \mathrm{N/mm^2}$$

From Figure 7:

$$h' = 0.4\ (D - d) = 1.2\ mm$$

$$d_m = \frac{D + d}{2} = 12.5\ mm$$

The minimum required loaded length of the splined key:

$$L_{shaft\ 1} = \frac{2 \cdot T}{d_m \cdot h' \cdot n \cdot 0.75 \cdot p_{per}}$$

$$= 0.26\ mm$$

$$L_{shaft\ 1} \leq 1.3 \cdot d_{shaft\ 1}$$

$$0.26\ mm \leq 14.3\ mm$$

So, for the design $L_{shaft\ 1} = 10\ mm$ is chosen as it would be easier to manufacture.

To calculate average pressure:

$$p_{avg} = \frac{2 \cdot T}{d_m \cdot h' \cdot n \cdot 0.75 \cdot L_{shaft\ 1}}$$

$$= 10.08\ \mathrm{N/mm^2}$$

As $p_{avg} \leq p_{per}$, $L_{shaft\ 1}$ is the length of the splined key for which the design is safe.

Now the gear in which this splined shaft will be inserted is of the same material as that of the shaft, so no separate calculation is needed for the gear and splined keyway contact.

2.2 Shaft 2

Similar to shaft 1, calculation for shaft 2 is done.

$d_{shaft\ 2}$	$D_{shaft\ 2}$	n	B	$T_{per,G}$
$16\ mm$	$20\ mm$	6	$4\ mm$	$6800\ Nmm$

The permitted pressure of shaft 1:

$$p_{per} = \frac{R_e}{S_F} = \frac{580\ N/mm^2}{1.5} = 386.67\ \mathrm{N/mm^2}$$

From Figure 7:

$$h' = 0.4\,(D - d) = 1.6\ mm$$

$$d_m = \frac{D + d}{2} = 18\ mm$$

The minimum required loaded length of the splined key:

$$L_{shaft\ 1} = \frac{2 \cdot T}{d_m \cdot h' \cdot n \cdot 0.75 \cdot p_{per}}$$

$$= 0.27\ mm$$

$$L_{shaft\ 2} \leq 1.3 \cdot d_{shaft\ 2}$$

$$0.27 \leq 20.8\ mm$$

So, for the design $L_{shaft\ 2} = 10\ mm$ is chosen

To calculate average pressure with this

$$p_{avg} = \frac{2 \cdot T}{d_m \cdot h' \cdot n \cdot 0.75 \cdot L_{shaft\ 2}}$$

$$= 10.50\ \mathrm{N/mm^2}$$

As $p_{avg} \leq p_{per}$, $L_{shaft\ 2}$ is the length of the splined key for which the design is safe.

3 Bearing Calculation

From FBD calculation (chapter 1) of the gear system, it is determined that bearings will have to sustain both axial as well as radial loads. So deep groove single ball bearings have been selected as they can fairly sustain both of these types of loads and are also widely available. Also, Bearings A and C are locating bearings and Bearings B and D are non-locating bearings. The following Figure 8 from SFK manufacturer shows the reliability of these bearings.

Suitability of rolling bearings for industrial applications

Symbols

+++ excellent
++ good
+ fair
- poor
– – unsuitable

↔ double direction
← single direction
□ non-locating displacement on the seat
■ non-locating displacement within the bearing
✓ yes
✗ no

Bearing type	Load carrying capability			Misalignment		Arrangement				Suitable for					Design features			
	Radial load	Axial load	Moment load	Static misalignment	Dynamic misalignment (few tenths of a degree)	Locating	Non-locating	Adjusted	Floating	Long grease life	High speed	Low run-out	High stiffness	Low friction	Integral sealing	Separable ring mounting	Tapered bore	Standard housings and accessories available
Deep groove ball bearings A, B	+	+ ↔	A –, B +	–	– –	↔	□	✗	✓	A +++ B ++	A +++ B +	A +++ B ++	+	+++	A ✓	✗	✗	✗
Insert bearings A, B, C	+	+ ↔	– –	++	– –	↔	↔	✗	✗	+++	++	A, B + C ++	+	++	✓	✗	✗	✓
Angular contact ball bearings, single row	+1)	++ ←	– –	–	– –	✗	✗	✓	✗	++	++	+++	++	++	✓	✗	✗	✗
matched single row A, B, C	A, B ++ C ++1)	A, B ++ ↔ C ++ ←	A ++, B + C – –	A, C – –, B –	– –	A, B ↔ C ←	A, B □ C ✗	✗	✗	++	++	+++	++	++	✗	✗	✗	✗
double row A, B	++	++ ↔	++	– –	– –	↔	□	✗	✗	++	++	++	++	++	A ✓	B ✓	✗	✗
four-point contact	+1)	++ ↔	– –	– –	– –	↔1)	– –	– –	– –	+	+++	++	++	++	✗	✓	✗	✗
Self-aligning ball bearings	+	–	– –	+++	+2)	↔	□	✗	✓	+++	++	++	+	+++	✓	✗	✓	✓
Cylindrical roller bearings, with cage A, B	++	– –	– –	–	– –	✗	■	✗	✗	++	+++	+++	++	+++	✗	✓	✗	✗
A, B, C, D	++	A, B + ← C, D + ↔	– –	–	– –	A, B ← C, D ↔	A, B ■ ← C, D ✗	✗	A ✓ B, C, D ✗	++3)	+++	++	++	+++	✗	✓	✗	✗
full complement, single row A, B	+++	+ ←	– –	–	– –	←	A, B ←	✗	✓	–	+	+	+++	–	✗	A ✗ B ✓	✗	✗
full complement, double row A, B, C, D	+++	A – –, B + ← C + ↔	– –	–	– –	B ← C, D ↔	A ■ ↔ B ■ ←	✗	✗	–	+	+	+++	–	D ✓	✗	✗	✗

Figure 8: Bearing Type Selection

SKF. Generated from www.skf.com on 2021-07-01

61902 Deep groove ball bearings

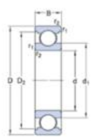
Popular item

Deep groove ball bearings

Bearing data
Tolerances,
Normal (metric), P6, P5, Normal (inch),
Radial internal clearance,
Matched bearing pairs, Stainless steel
d < 10 mm, Other bearings

Bearing interfaces
Seat tolerances for standard conditions,
Tolerances and resultant fits

Technical specification



DIMENSIONS

d	15 mm
D	28 mm
B	7 mm
d_1	≈ 18.8 mm
D_2	≈ 25.3 mm
$r_{1,2}$	min. 0.3 mm

SKF. Generated from www.skf.com on 2021-07-01

CALCULATION DATA

Basic dynamic load rating	C	4.36 kN
Basic static load rating	C_0	2.24 kN
Fatigue load limit	P_u	0.095 kN
Reference speed		56 000 r/min
Limiting speed		34 000 r/min
Calculation factor	k_r	0.02
Calculation factor	f_0	14.3

Figure 9: 61902 Deep groove ball bearing data from SKF website

Bearings have been selected from the SKF website according to the diameter of the shaft on which they are to be mounted. The Figure 9 show the data for the chosen Deep groove ball bearing.

As bearing will have to sustain both axial and radial loads, an equivalent force needs to be calculated which is given by:

$$P = F_{radial} \qquad \text{if } \frac{F_{axial}}{F_{radial}} \le e$$

$$P = X \cdot F_{radial} + Y \cdot F_{axial} \qquad \text{if } \frac{F_{axial}}{F_{radial}} > e$$

To obtain the calculation factors X and Y the following figure is referred:

	Single row and double row bearings			Single row bearings					
	Normal clearance			C3 clearance			C4 clearance		
$f_0 F_a/C_0$	e	X	Y	e	X	Y	e	X	Y
0,172	0,19	0,56	2,3	0,29	0,46	1,88	0,38	0,44	1,47
0,345	0,22	0,56	1,99	0,32	0,46	1,71	0,4	0,44	1,4
0,689	0,26	0,56	1,71	0,36	0,46	1,52	0,43	0,44	1,3
1,03	0,28	0,56	1,55	0,38	0,46	1,41	0,46	0,44	1,23

Figure 10: Calculation factors for bearing calculation

	Bearing A	Bearing B	Bearing C	Bearing D
Radial Force	$173.92\ N$	$58.83\ N$	$316.27\ N$	$537.95\ N$
Axial Force	$33.82\ N$	$0\ N$	$70.52\ N$	$0\ N$
F_A/F_R	0.19	0	0.22	0

	X	Y	e
Shaft 1 Bearings	0.56	1.99	0.22
Shaft 2 Bearings	0.56	1.99	0.22

For Bearing A:

$P_A = 173.92\ N$

For Bearing B:

$P_B = 58.83\ N$

For Bearing C:

$P_C = 317.45\ N$

For Bearing D:

$P_D = \ 537.95\ N$

Now, to check if the bearings are very lightly loaded as this can cause skidding and smearing of raceways or cage, F should be greater than 1 % of C, which is satisfied by all the bearings.

3.1 Dynamic Load capacity

The design is made for universal gearboxes and hence a lifetime of 8,000 hrs is taken according to the Figure 11

Gearboxes in general machine building

Mounting location	**Recommended rating life h**				**Operating life h**	
	Ball bearings		**Roller bearings**			
	from	**to**	**from**	**to**	**from**	**to**
Universal gearboxes	4 000	14 000	5 000	20 000	5 000	20 000
Geared motors	4 000	14 000	5 000	20 000	5 000	20 000
Large gearboxes, stationary	14 000	46 000	20 000	75 000	20 000	80 000

Figure 11: Recommended Ball Bearing Life

The design is made for an input speed of 1000 rpm (assumption). Now,

$Gear\ ratio\ (i) = 2$

$n_{shaft1} = \ 1000\ rpm = \frac{50}{3}\ rps$

$n_{shaft2} = \frac{n_{shaft1}}{i}$ AB

$= 500\ rpm = \frac{25}{3} rps$

Minimum dynamic load capacity of bearing so that it has $L_{10h} = 8{,}000$ hr:

$$C_{min} = F_{equivalent} \cdot \sqrt[p]{\frac{3600 \cdot n \cdot L}{a_1 \cdot 10^6}}$$

For ball bearings, p = 3 and a_1 for L_{10h} is 1.

So, values of C_{min} for each bearing are

	Bearing A	Bearing B	Bearing C	Bearing D
C_{min}	$1080.82\ N$	$365.60\ N$	$2485.55\ N$	$4212\ N$

As seen $C_{min} \leq C$ so the chosen bearing type satisfies this condition.

3.2 Bearing Lifetime

To get the bearing lifetime L_{10h} the following formula is used

$$L_{10h} = \frac{\left(\frac{C}{F}\right)^p \cdot 10^6}{3600 \cdot n}$$

	Bearing A	Bearing B	Bearing C	Bearing D
L_{10h}	5,25,157	14,568,807	43,179	8,873

As the required axial load and radial load is relatively small, L_{10h} are lower than the required lifetime of bearing. So, the chosen bearing will not fail for the design load.

Also, it is seen that the L_{10h} values of Bearing A and B are very high indicating that different bearing with lower value of C can be chosen so as to avoid over specification. The following bearing of W 61702 from SKF shown in Figure 12 could be used which give the following L_{10h}.

	Bearing A	Bearing B
$L_{10h,\ alternative}$	9,273	26,794

SKF. Generated from www.skf.com on 2021-07-02

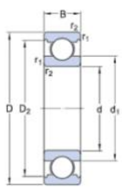
W 61702 Deep groove ball bearings

Deep groove ball bearings

Bearing data
Tolerances,
Normal (metric), P6, P5, Normal (inch),
Radial internal clearance,
Matched bearing pairs, Stainless steel d < 10 mm, Other bearings

Bearing interfaces
Seat tolerances for standard conditions,
Tolerances and resultant fits

Technical specification



DIMENSIONS

d	15 mm
D	21 mm
B	4 mm
d_1	≈ 16.86 mm
D_2	≈ 19.67 mm
$r_{1,2}$	min. 0.2 mm

CALCULATION DATA

Basic dynamic load rating	C	0.527 kN
Basic static load rating	C_0	0.29 kN
Fatigue load limit	P_u	0.012 kN
Reference speed		67 000 r/min
Limiting speed		40 000 r/min
Calculation factor	k_r	0.015
Calculation factor	f_0	8.4

Figure 12: Alternative Selection of Bearing for Bearing A and B

4 Bolted Fasteners

The fasteners used throughout the gear box are Hex Bolts ISO 4014 M4 x 30 x 14 - N - 8.8.

4.1 Bolted fasteners around shaft 1

The axial force will be transferred via the plate to the bolts. The chosen bolts should be able to sustain this force.

As 6 bolts are present, the load on the bolts will be

$$F_{B1} = \frac{F_{axial}}{6} = \ 11.75\ N$$

The following data is taken from Solidworks

d_w	l_k	d_h	E
$7\ mm$	$25\ mm$	$4\ mm$	$210000\ MPa$

The approach presented here bases on the work of Felix Rötscher (German engineer, 1873 – 1944, “Rötscher-Kegel”).

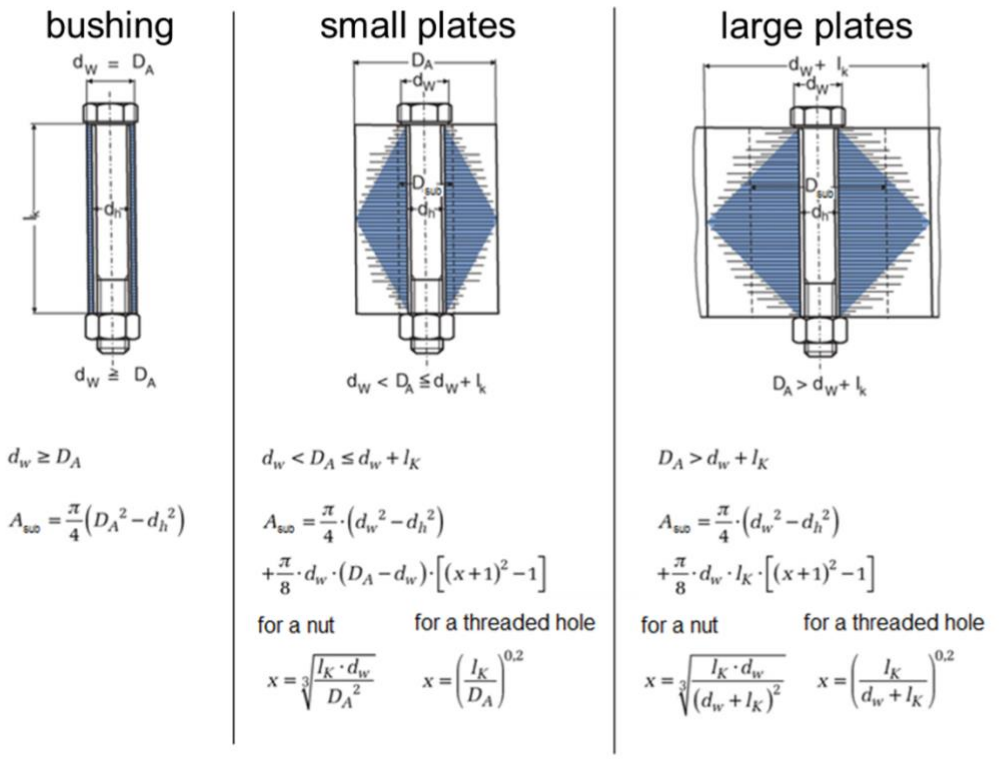


Figure 13: Bolt Area parameter Selection

The design value of D_A is selected to be $6\ mm$ as bushing is used.

As value of d_w is greater than D_A,

$A_{sub} = \frac{\pi}{4} \cdot (D_A^2 - d_h^2) = 15.70\ mm^2$

4.1.1 Resilience

The resilience of the plate δ_p:

$$\delta_p = \frac{l_k}{A_{sub} \cdot E} = 7.58 \cdot 10^{-6} mm/N$$

The spring constant of the plate:

$C_p = \delta_p^{-1} = 131926.12\ N/mm$

The resilience of bolt and nut is given by $\delta_s = \delta_H + \delta_1 + \delta_N + \delta_T$.

The resilience of bolt head:

$$\delta_H = \frac{l_H}{E \cdot A_N} = 0.607 \cdot 10^{-6} mm/N$$

where $A_N = \frac{\pi \cdot d^2}{4} = 12.56\ mm^2$

$l_H = 0.4 \cdot d$

$d = 4\ mm$ (Nominal diameter of the bolt- M4)

The resilience of bolt shank:

$$\delta_1 = \frac{l_1}{E \cdot A_1} = 14.62 \cdot 10^{-6} mm/N$$

where $\text{Core corss section} = A_1 = \frac{\pi \cdot d_1^2}{4} = 8.14\ mm^2$

$l_1 = l_k$

$d_1 = 3.22\ mm\ \ (core\ cross\ section\ diameter)$

The resilience of bolt thread:

$$\delta_T = \frac{l_T}{E \cdot A_3} = 1.05 \cdot 10^{-6} mm/N$$

where $\text{Loaded Cross Section} = A_3 = \frac{\pi \cdot d_3^2}{4} = 9.03\ mm^2$

$l_T = 0.5 \cdot d$

$$d_3 = \frac{d_1 + d_2}{2} = 3.39\ mm$$

Where $d_2 = nut\ thread\ section\ diameter = 3.56\ mm$

The resilience of bolt Nut:

$$\delta_N = \frac{l_N}{E \cdot A_N} = 0.607 \cdot 10^{-6} mm/N$$

where $A_N = \frac{\pi \cdot d^2}{4}$

$l_N = 0.4 \cdot d$

Now, $\delta_s = \delta_H + \delta_1 + \delta_N + \delta_T = 16.89 \cdot 10^{-6} mm/N$

The spring constant of the bolt:

$C_s = \delta_s^{-1} = 59206.63\ N/mm$

δ_p	δ_s	C_P	C_s
$7.58 \cdot 10^{-6} mm/N$	$16.89 \cdot 10^{-6} mm/N$	$131926.12\ N/mm$	$59206.63\ N/mm$

Now,

Operation Load factor = $\phi = \frac{\delta_P}{\delta_P + \delta_S} = 0.31$

4.1.2 Pretension

The residual clamping force is taken to be 100 N.

$F_{KR} = 100\ N$

$F_V \geq F_{KR} + F_{PB}$ where $F_{PB} = F_{B1}(1 - \phi) = 8.11\ N$

The pretension in the bolt: $F_V \geq 108.11\ N$

Under no consideration of settling,

$F_{V,assembly,min} = F_V = 108.11\ N$

The tightening factor or the assembly factor is conserved to be 1.2

$F_{V,assembly,max} = \alpha_k \cdot F_{V,assembly,min} = 129.73\ N$

4.1.3 Safety Check

The bolt used is M4 x 25 x 14 - N - 8.8 meaning

$R_m = 8 \cdot 100 \frac{N}{mm^2} = 800 \ \frac{N}{mm^2}$

$R_{p0.2} = R_m \cdot \frac{8}{10} = 640 \ \frac{N}{mm^2}$

To check if the bolt will be able to withstand the static external load F_{B1}

$\phi \cdot \frac{F_{B1}}{A_N} \leq 0.1 \cdot \frac{R_{p0.2}}{S_F}$

Where safety factor is taken as 1.5

$=> 0.29 \frac{N}{mm^2} \leq 42.66 \ N/mm^2$

Hence the bolts around shaft 2 will be able to sustain the axial force on shaft 1.

4.2 Bolted fasteners around shaft 2

Similar to bolted fasteners around shaft 1, the following are calculation for bolted fasteners around shaft 2.

6 bolts are present around shaft 2, the load on the bolts will be

$F_{B2} = \frac{F_{axial}}{6} = 5.64 \ N$

4.2.1 Resilience

As the bolts and the plate used around shaft 2 are same as shaft 1, the resilience will also be same.

δ_p	δ_s	C_P	C_s
$7.58 \cdot 10^{-6} mm/N$	$16.89 \cdot 10^{-6} mm/N$	$131926.12 \ N/mm$	$59206.63 \ N/mm$

Operation Load factor = $\phi = \frac{\delta_P}{\delta_P + \delta_S} = 0.31$

4.2.2 Pretension

The residual clamping force is taken to be 100 N. (Assumption)

$F_{KR} = 100 \ N$

$F_V \geq F_{KR} + F_{PB}$ where $F_{PB} = F_{B2}(1 - \phi) = 3.90 \ N$

The pretension in the bolt: $F_V \geq 103.90\ N$

Under no consideration of settling,

$F_{V,assembly,min} = F_V = 103.90\ N$

The tightening factor or the assembly factor is considered to be 1.2

$F_{V,assembly,max} = \alpha_k \cdot F_{V,assembly,min} = 124.68\ N$

4.2.3 Safety Check

The bolt used is M4 x 25 – 8.8 hence,

$R_m = 8 \cdot 100 \frac{N}{mm^2} = 800\ \frac{N}{mm^2}$

$R_{p0.2} = R_m \cdot \frac{8}{10} = 640\ \frac{N}{mm^2}$

To check if the bolt will be able to withstand the static external load F_{B1}

$$\phi \cdot \frac{F_{B2}}{A_S} \leq 0.1 \cdot \frac{R_{p0.2}}{S_F}$$

Where safety factor is taken as 1.5

$$=> 0.14 \frac{N}{mm^2} \leq 42.66\ N/mm^2$$

Hence the bolts around shaft 2 will be able to sustain the axial force on shaft 2.

5 Sealing

A custom sealing has been made in order to avoid lubricant leakage.

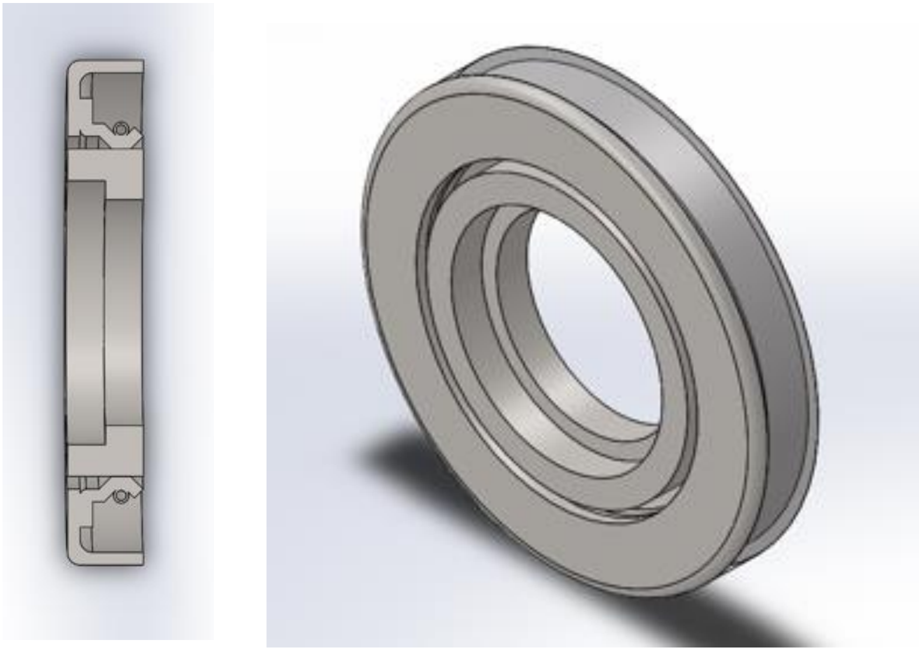


Figure 14: Sealing

6 Interference Detection

13 interferences have been detected due to presence of bolted fasteners.

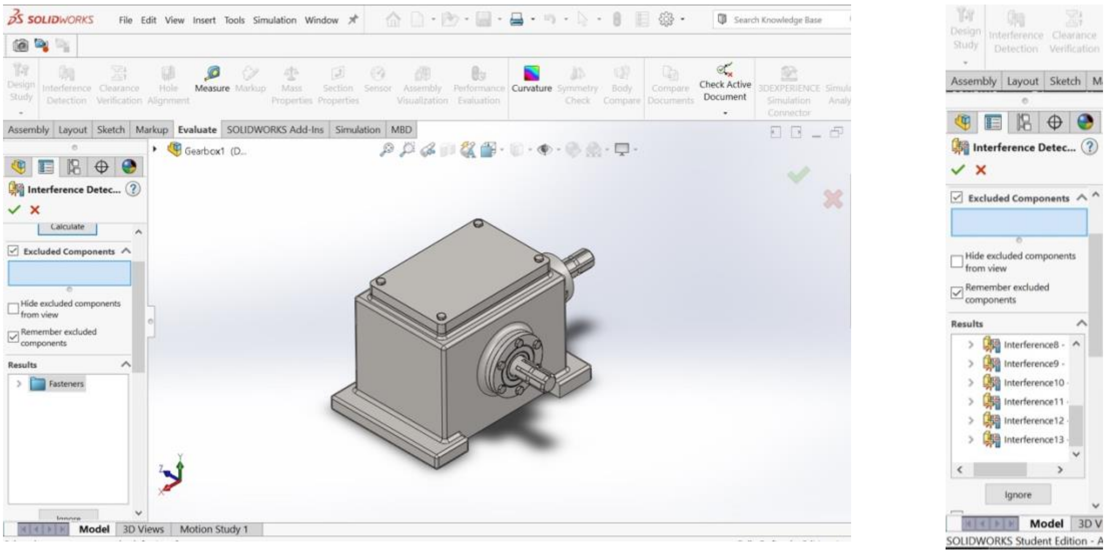
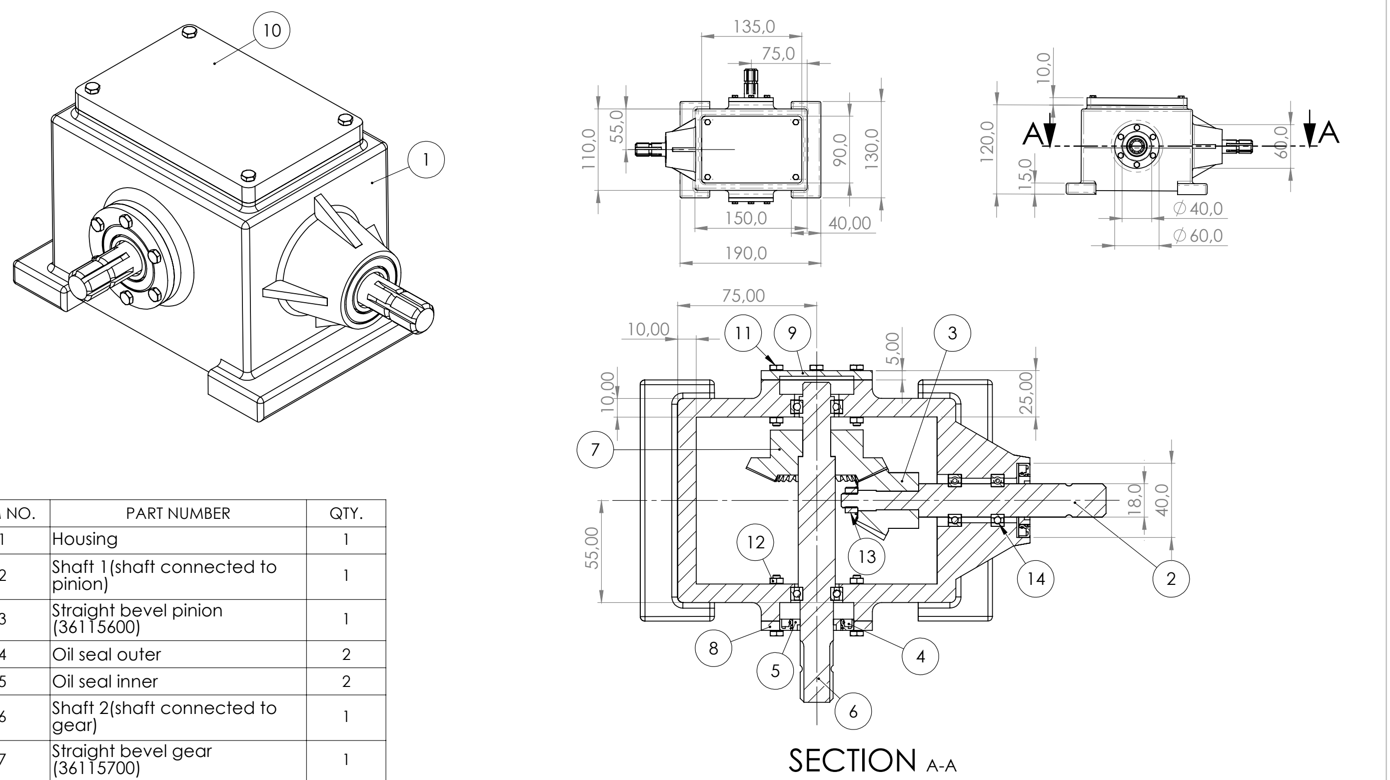


Figure 15: Interference Detection

7 Technical Drawings



general tolerances acc. to ISO 2768-mK
surface finish acc. to ISO 1302
edge finish acc. to ISO 13715
material: C45

ITEM NO.	PART NUMBER	QTY.
1	Housing	1
2	Shaft 1(shaft connected to pinion)	1
3	Straight bevel pinion (36115600)	1
4	Oil seal outer	2
5	Oil seal inner	2
6	Shaft 2(shaft connected to gear)	1
7	Straight bevel gear (36115700)	1
8	Cover with shaft 2	1
9	Closed cover	1
10	Housing lid	1
11	ISO 4014 - M4 x 25 x 14-N	16
12	ISO - 4032 - M4 - W - C	12
13	ISO - 4032 - M8 - W - C	1
14	Roller ball bearing(SKF_61902)	4

Fit Table	
Fit	Between parts
H7/n6	2 and 3, 6 and 7
H7/m6	1 and 11

faculty or department	technical reference	drawn by (last saved by)	drawing date	released by
Technology &Bionics		Tejasvi Rasgotra	02/07/2021	

HOCHSCHULE RHEIN-WAAL Rhine Waal University of Applied Sciences

title, additional title: Gearbox Assembly Drawing
projection method
scale

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drawing no. prefix: 22342-ME4.2703.47
type of document: part drawing
drawing no.: 200124-01-001
rev.: 00
date of issue: 02/07/2021
sheet: 1/1

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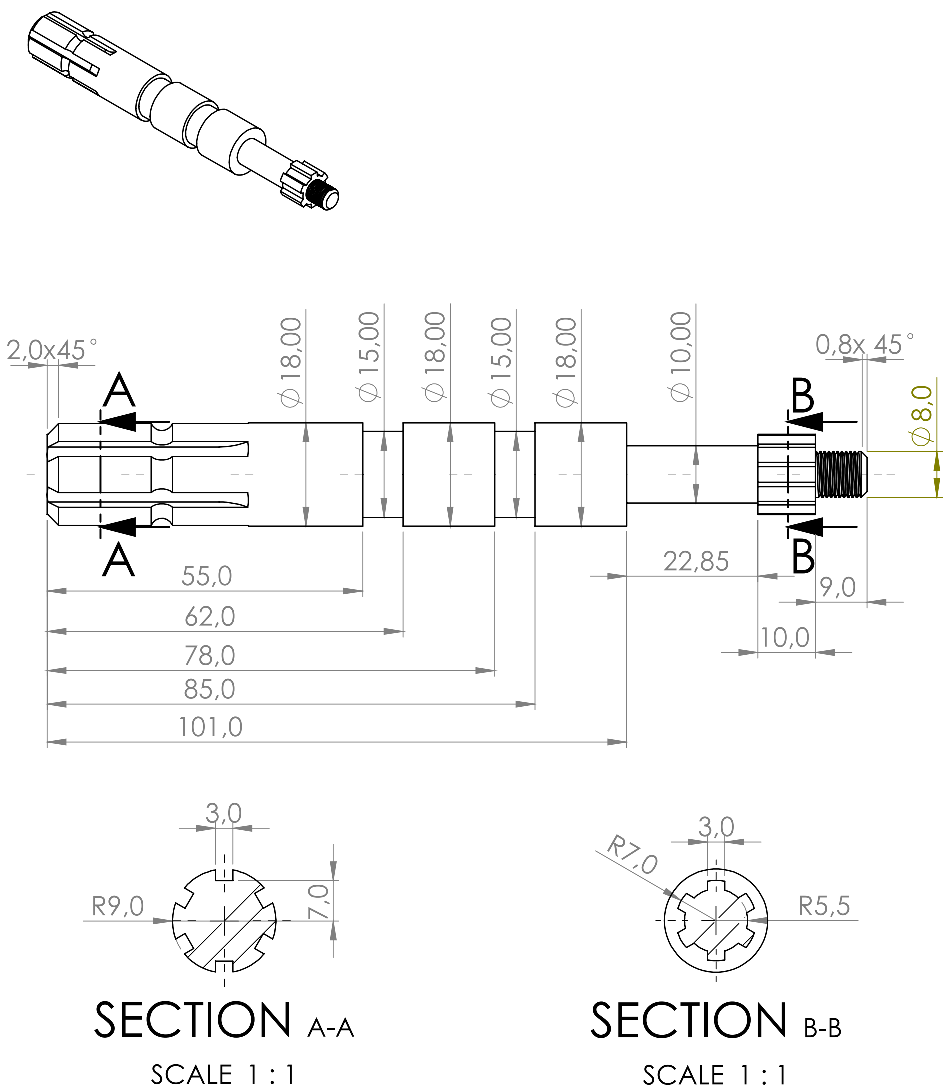

general tolerances acc. to ISO 2768- mK
surface finish acc. to ISO 1302
material: C45

edge finish acc. to ISO 13715

faculty or department	technical reference	drawn by (last saved by)	drawing date	released by
Technology & Bionics		Tejasvi Rasgotra	02/07/2021	

HOCHSCHULE RHEIN-WAAL
Rhine-Waal University of Applied Sciences

title, additional title	projection method	scale
Shaft 1		1:1

drawing no. prefix	type of document		
22342-ME4.2703.47	part drawing		
drawing no.	**rev.**	**date of issue**	**sheet**
200124-01-001	00	02/07/2021	1/1

8 References

1. Documents for “Product Design” course, Rhein Waal University of Applied Sciences
2. https://www.skf.com/uk/products/rolling-bearings/ball-bearings/deep-groove-ball-bearings
3. Documents of “Rolling Bearing Catalogue” from SKF official website
4. Shigley’s Mechanical Engineering Design, 10th Edition (Budynas & Nisbett, 2015).
5. Mechanical-and-Metal-Trades-Handbook, 3rd English Edition