UNIVERSITY "POLITEHNICA" OF BUCHAREST

DEPARTMENT OF MACHINE ELEMENTS AND TRIBOLOGY

FACULTY OF ENGINEERING IN FOREIGN LANGUAGES

PROJECT

MECHANICAL TRASMISSION PROJECT

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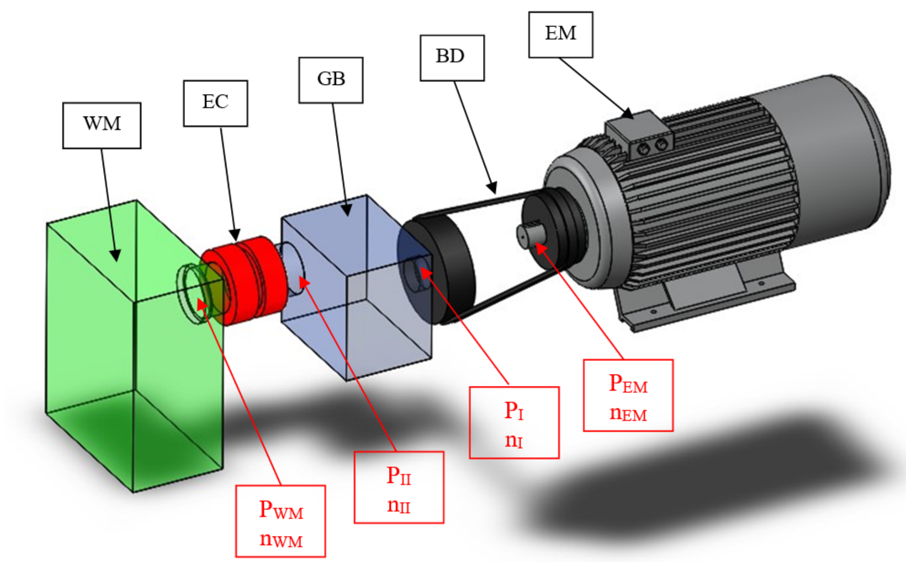
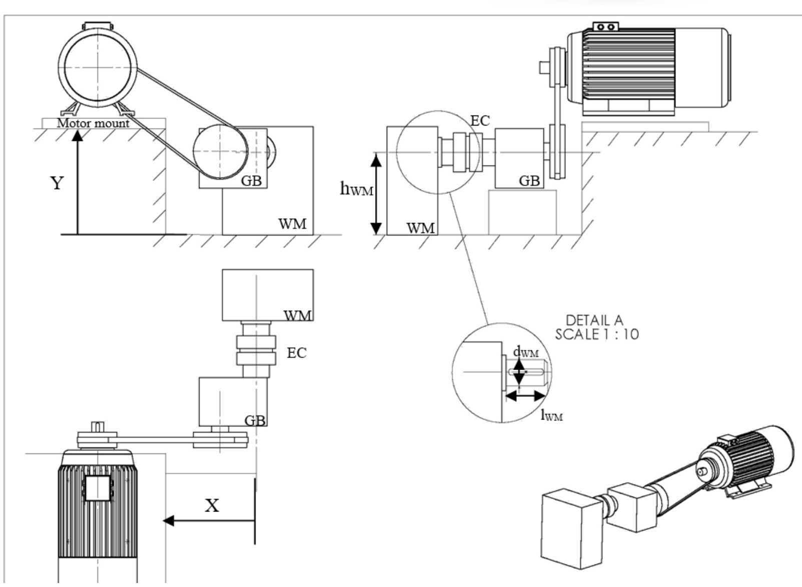
Design the power transmission sketched in figure below, considering it is used to drive a working machine (WM) with the service factor K_A=1.3,, that requires a nominal power rate P_{WM}= 12.3 kW at the constant speed n_{WM}=100.rpm. The transmission consists of:

- an AC electric motor (EM) with the nominal rotational speed n^{*}_{EM} =1500. rpm.
- a V-belt drive (BD).
- a double stage ***helical*** gear box (GB) foot mounted.
- an elastic coupling (EC).

- **Spatial arrangement of the transmission:**

- X=600mm, Y=-500mm, h_{WM}=500.mm, d_{WM}=.55.mm, l_{WM}=82.mm





Project Requirements

The project must contain a **Technical Report** with details of all the calculations performed and a **Graphical Part** .
Two versions of the power transmission will be presented:

i. a transmission made with a commercially available gear box.
   - assembly drawing of the transmission (a 3D model and 2D drawings with 2 views)

and

i. a custom made **one-stage gear box** which will include:
   - 2D - Main cross section - detail drawing.
   - Detail (execution) drawings of the shafts and gears.

For the second part of the project the one-stage gear box will be calculated for the following data:

- design arrangement: ....(V/H)
- transmission ratio: *iR*= ....

The gearbox is driven by the same electric motor with the same belt-drives, being capable to transmit the same power as the one selected for the first part of the project. Both gearboxes should have the same positioning with respect to the electric motor and V-belts.

Project Content

PART I

Ch. 1. POWER CALCULATION
Ch. 2. GEAR BOX AND ELECTRIC MOTOR SELECTION
Ch. 3. SCHEMATIC ARRANGEMENT OF THE TRANSMISSION
Ch. 4. DESIGN OF V-BELT DRIVE
3D-Assembly drawing

PART II

Ch. 1. Design of THE GEAR SET
Gear set dimensioning.
Gear set geometry.
Forces acting on the gear set.
Gear strength verification.
Ch. 2. Design of the SHAFTS
Sizing.
Diagrams bending and torsional moments.
Strength verification.
Hub-on-shaft joints.
Ch. 3. SELECTION AND CALCULATION OF ROLLING BEARINGS
Ch. 4. DESIGN OF THE GEAR BOX
2D - Main cross section - detail drawing
Detail (execution) drawings for the shafts and gears

Given data

Double stage helical gearbox

$n\_star_{EM} := 1500\ rpm$

$P_{WM} := 12.3\ kW$

$n_{WM} := 100\ rpm$

$K_A := 1.3$

$d_{WM} := 55\ mm$

$l_{WM} := 82\ mm$

$h_{WM} := 500\ mm$

$X := 600\ mm$

$Y := -500\ mm$

Mechanical efficiencies

Belt drive efficiency

$\eta_{BD} := 0.96$

Efficiency of the helical gear set

$\eta_{HG} := 0.98$

Churning efficiency

$\eta_C := 0.97$

Efficiency of a set of rolling bearings

$\eta_{RB} := 0.994$

The total efficiency for a 2 stage helical gearbox : $\eta_t := \eta_{BD} \cdot {\eta_{HG}}^2 \cdot {\eta_C}^2 \cdot {\eta_{RB}}^3 = 0.852$

Since the gearbox has 2 stages, the churning efficiency is doubled and since there are 3 pairs of bearings, one for each shaft, the bearing efficiency is multiplied 3 times over.

Electric motor selection

The electric motor is foot mounted.

Calculated power of the electric motor $P_{EM} := \dfrac{P_{WM}}{\eta_t} = 14.4\ kW$

Nominal power of the motor

$P_{nom} := K_A \cdot P_{EM} = 18.8\ kW$

The nominal power of the motor is 18.8 kW, so in order to drive the working machine it is needed a motor with a higher transmitted power and a rotational speed close to the nominal one (1500 rpm). The chosen motor, that of 22 kW power and 1470 rpm full load rotational speed will be enough for the intended work.

Table 1. Selected motor

TEFC, CLASS F, 40°C AMBIENT TEMP., IEC DESIGN N CONTINUOUS DUTY, S.F. 1.0 **400V/50HZ**

OUTPUT		FULL LOAD rpm	FRAME NO.	EFFICIENCY				POWER FACTOR				CURRENT		TORQUE				ROTOR GD2 kg-m2	APPROX. WEIGHT kg
HP	kW			FULL LOAD (%)	3/4 LOAD (%)	2/4 LOAD (%)	1/4 LOAD (%)	FULL LOAD (%)	3/4 LOAD (%)	2/4 LOAD (%)	1/4 LOAD (%)	FULL LOAD (A)	LOCKED ROTOR (A)	FULL LOAD N•m	LOCKED ROTOR %FLT	PULL UP %FLT	BREAK DOWN %FLT		
10	7.5	2920	132S	88.1	88.3	87.8	83.3	82.5	77.5	68.0	46.5	14.9	98	24.49	250	230	275	0.075	72.5
		1460	132M	88.7	89.6	89.5	85.3	84.0	78.5	67.0	44.5	14.5	112	48.98	275	200	305	0.142	79.0
		960	160M	87.2	88.2	87.7	82.4	82.0	77.0	66.5	44.5	15.1	105	74.50	210	195	260	0.363	110
		720	160L	86.0	86.0	84.0	76.0	70.0	61.0	49.0	30.0	18.0	105	99.33	210	180	300	0.586	146
15	11	2950	160M	89.4	89.3	88.1	82.5	90.5	87.5	80.5	61.5	19.6	172	35.56	230	180	305	0.154	110
		1465	160M	89.8	90.6	90.7	87.1	86.5	83.0	74.5	52.5	20.4	160	71.60	220	180	300	0.296	121
		965	160L	88.7	89.2	88.6	83.2	81.5	76.0	65.0	42.5	22.0	170	108.7	245	205	300	0.558	138
		720	180L	87.7	87.5	87.0	80.0	70.0	62.0	56.0	34.0	25.9	140	145.7	210	160	230	1.019	182
20	15	2930	160M	90.3	91.0	91.2	88.1	93.5	92.5	89.0	74.0	25.6	225	48.82	245	165	280	0.192	120
		1470	160L	90.6	91.3	91.2	88.1	86.5	82.5	73.5	51.5	27.6	220	97.30	220	185	300	0.427	138
		975	180L	89.7	90.4	90.2	86.7	82.5	77.5	67.5	46.0	29.3	220	146.7	210	195	300	1.337	205
		720	200L	89.0	90.0	91.0	87.0	77.0	71.5	64.5	41.0	31.6	165	198.7	185	140	205	1.749	275
25	18.5	2925	160L	90.9	91.5	91.7	88.6	93.0	91.5	88.0	73.0	31.6	290	60.31	260	185	310	0.237	137
		1475	180M	91.2	91.7	91.6	88.7	85.5	83.0	76.5	57.0	34.2	230	119.6	200	185	300	0.654	180
		975	200L	90.4	91.0	90.9	87.7	79.5	75.0	65.5	43.5	37.2	260	180.0	215	195	300	1.604	263
		735	225S	91.5	92.0	91.0	86.0	72.0	65.5	58.0	35.5	40.5	220	240.0	210	185	235	2.675	345
30	22	2930	180M	91.3	91.2	90.5	85.8	91.5	90.0	85.5	71.0	38.0	295	71.60	215	185	300	0.283	178
		1470	180L	91.6	92.4	92.2	89.3	85.5	83.5	77.5	58.0	40.5	270	142.7	195	155	250	0.770	199
		980	200L	90.9	91.4	91.8	88.0	81.0	77.5	68.5	44.0	43.1	300	214.1	210	180	255	1.912	283

rotational speed of the electric motor $n_{EM} := 1470\ rpm$

real power of the electric motor $P_m := 22\ kW$

Fig 1. Electric motor dimensioning

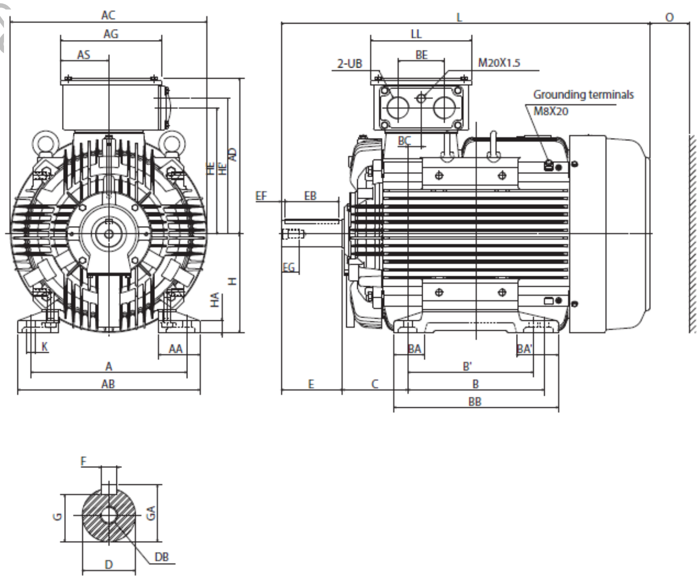


FIG. 6

Table 2. Motor dimensions

Dimension in mm

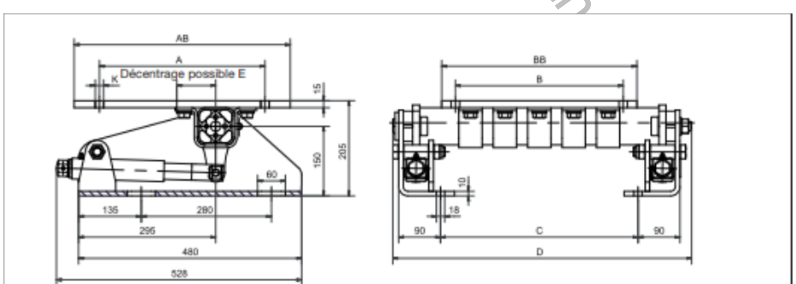
Output (kW) 2P	4P	6P	8P	FRAME SIZE	FIG. NO.	A	AA	AB	AC	AD	AG	AS	B	B'	BA	BA'	BB	BC	BE	C	H
0.75	0.55	0.37	—	80M	1	125	35	161	156	161	125	67.5	100	—	35.5	35.5	130	18.5	40	50	80
1.1	0.75	0.55	—																		
1.5	1.1	0.75	—	90S	2	140	40	180	176	171	125	67.5	100	—	33	33	125	36	40	56	90
2.2	1.5	1.1	—	90L		140	40	180	176	171	125	67.5	125	100	33	58	150	36	40	56	90
3	2.2	1.5	0.75	100L		160	40	200	196	191	147	78.5	140	—	43.5	43.5	176	21	50	63	100
—	3	—	1.1																		
4	4	2.2	1.5	112M		190	50	235	218	198.5	147	78.5	140	—	45.5	45.5	176	28	50	70	112
5.5	5.5	3	2.2	132S	3	216	63.5	259	258	216	147	78.5	140	—	59	59	184	6	50	89	132
7.5	—	—	—																		
—	7.5	4	3	132M		216	63.5	259	258	216	147	78.5	178	140	59	97	222	6	50	89	132
—	—	5.5	—																		
11 15	11	7.5	4 5.5	160M	4	254	71	300	317	271	193	91.5	210	—	46	46	256	38	89	108	160
18.5	15	11	7.5	160L		254	71	300	317	271	193	91.5	254	210	46	90	300	38	89	108	160
22	18.5	—-	—-	180M	5	279	72	330	354	297	193	91.5	241	—	57	57	292	34	89	121	180
—-	22	15	11	180L		279	72	330	354	297	193	91.5	279	241	57	95	330	34	89	121	180
30 37	30	18.5 22	15	200L		318	88	378	398	330	231	110.5	305	—	70	70	365	53	106	133	200
—	37	—	18.5	225SC		356	94	416	449	356	231	110.5	286	—	70	70	350	30.5	106	149	225

Dimension in mm

HA	HE	HE'	K	L	LL	O	UB1	SHAFT EXTENSION									BEARING	
								D	E	EB	EF	EG	F	G	GA	DB	DRIVE END	OPPOSITE DRIVE END
10	123.5	—	10	293	115	40	M25X1.5	19	40	32	4	16	6	15.5	21.5	M6	6204ZZC3	6204ZZC3
10	133.5	—	10	344.5	115	40	M25X1.5	24	50	40	5	19	8	20	27	M8	6205ZZC3	6205ZZC3
10	133.5	—	10	369.5	115	40	M25X1.5	24	50	40	5	19	8	20	27	M8	6205ZZC3	6205ZZC3
12	157	—	12	392	125	50	M25X1.5	28	60	50	5	22	8	24	31	M10	6206ZZC3	6206ZZC3
13	164.5	—	12	412.5	125	50	M32X1.5	28	60	50	5	22	8	24	31	M10	6306ZZC3	6306ZZC3
16	182	—	12	466	125	50	M32X1.5	38	80	70	5	28	10	33	41	M12	6308ZZC3	6306ZZC3
16	182	—	12	504	125	50	M32X1.5	38	80	70	5	28	10	33	41	M12	6308ZZC3	6306ZZC3
215	234.5	18	14.5	608	193	60	M40 x 1.5	42	110	100	5	36	12	37	45	M16	6309ZZC3	6307ZZC3
215	234.5	18	14.5	652	193	60	M40 x 1.5	42	110	100	5	36	12	37	45	M16	6309ZZC3	6307ZZC3
241	260.5	20	14.5	672	193	70	M40 x 1.5	48	110	100	5	36	14	42.5	51.5	M16	6311C3	6310C3
241	260.5	20	14.5	710	193	70	M40 x 1.5	48	110	100	5	36	14	42.5	51.5	M16	6311C3	6310C3

Motor mount

The motor mount is selected based on the frame size of the motor (180L).



Taille Moteur	A	B	K	AB	BB	C	D	E	F	Poids (kg)	Type
160 M	254	210	14	320	315	245	463	25	437	41	270-1
160 L	254	254	14								
180 M	279	241	14	350	350	245	463	72	452	43	270-2
180 L	279	279	14								
200 L	318	305	18	405	390	345	563	55	463	53	400
225 S	356	286	18	465	420	425	643	72	510	60	500
225 M	356	311	18	465	420	425	643	72	510	60	

The mounting distance on the motor mount, A, is the same as the distance between the holes for the screws on the feet of the motor as a result, the electric motor can surely be mounted on the selected motor mount.

Gearbox selection

The total transmission ratio: $i_{tot} := \frac{n_{EM}}{n_{WM}} = 14.7$

The power transmitted at the entrance of the gearbox: $P_1 := P_m \cdot \eta_{BD} = 21.12\ kW$

According to the type of gearbox given (2 stage helical gearbox),i have selected the SZN type gearbox.

Table 5. Selection of the gearbox

$n_1 = \frac{n_{EM}}{i_{BD}}$ $M_{t1} = 9.55 \cdot 10^6 \cdot \frac{P_1}{n_1}$

No.	i_{GB}	i_{BD}	n_1 [rpm]	P_1 [kW]	M_{t1} [Nm]
1	7.1	2.07	710	21.1	283.8
2	8	1.84	800	21.1	251.9
3	9	1.63	900	21.1	223.9
4	10	1.47	1000	21.1	201.5
5	11.2	1.31	1120	21.1	179.9
6	12.5	1.18	1250	21.1	161.2
7	14	1.05	1400	21.1	143.9
8	16	0.92	1600	21.1	125.9
9	18	0.82	1800	21.1	111.9
10	20	0.74	2000	21.1	100.8

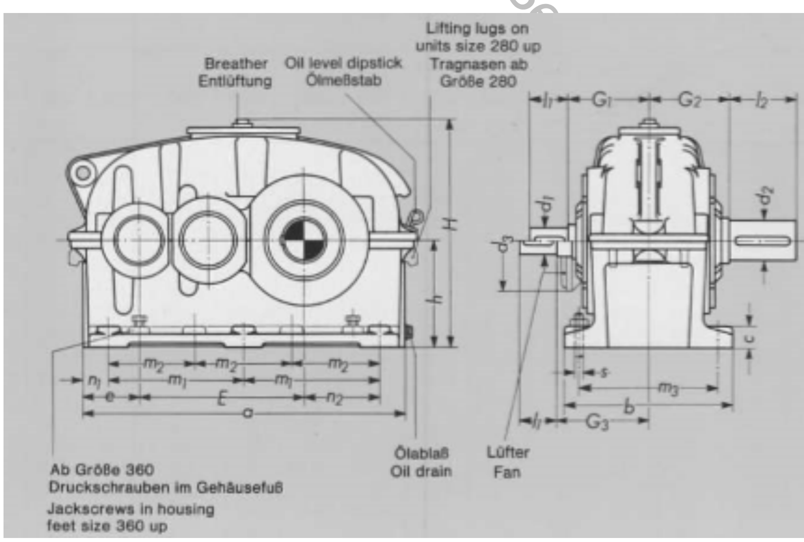
The gearbox which is selected has a transmission ratio of 10 and a unit size of 140, this gearbox was selected due to it's small unit size and the fact that it satisfies the requirements, moreover, the rotational speed, and the difference in size between the gearbox and the belt drive are convenient.

Table 6. Gearbox torque calculation

i_{GB}	M_{t1}	M_{t1}*K_A	n_{1GB}	Gear Unit Size				Gear Unit Size			
				112	125	140	160	112	125	140	160
	[Nm]	[Nm]	[rpm]	P_{1GB} [kW]				M_{t1GB} [Nm]			
8	251.9	327.4	1000	22.5	31	44	65	214.9	296.1	420.2	620.8
9	223.9	291.1	1000	21	27.5	40	58	200.6	262.6	382.0	553.9
10	201.5	262.0	1000	19	25	35	52	181.5	238.8	334.3	496.6

Starting from the gear unit size 140, and keeping in mind that the correction factor, the gearbox has a torque of higher value than the transmitted torque at the entrance of the gearbox, as a result the gear unit size 140 is the smallest unit size that meets the condition.
The transmission ratio is 10 because of the rotational speed at the entrance of the gearbox being closer to the rotational speed of the gearbox.

Fig 3. Gearbox dimensioning



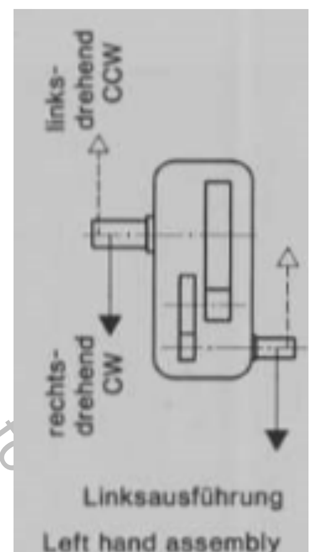


Table 7. Gearbox dimensions

Maße, Gewichte und Ölmengen — Dimensions, weights and oil quantities

Bauart Type SZN Größe Size	a	b	c	d_1	l_1	G_3 2)	d_1	l_1	G_3 2)	d_2	l_2	d_3 3)	E	e	G_1 2)	G_2	h	H	m_1
				Wellenenden1) i_N = 7,1–11,2			Shaft ends1) i_N = 12,5–20												
	mm	mm	mm	mm	mm	mm	mm	mm	mm	mm	mm	mm	mm	mm	mm	mm	mm	mm	mm
112	385	215	22	24 k6	40	–	22 k6	35	–	48 m6	80	–	192	75,5	105	110	125	268	160
125	425	235	25	28 m6	50	–	24 k6	40	–	55 m6	90	–	215	77,5	115	115	140	297	180

140	475	245	25	32 m6	60	140	28 m6	50	–	65 m6	105	202	240	85	125	125	160	335	200

Bauart Type SZN Größe Size	m_2 mm	m_3 mm	n_1 mm	n_2 mm	Fußschrauben Found. bolts s mm	Anzahl No. off	Gewicht Mittelwert Average weight kg	Öl-Höchstmenge Max. oil quantity Liter litres
112	–	180	32,5	85	M 12	6	53	3
125	–	200	32,5	100	M 12	6	72	4,3
140	–	210	37,5	112,5	M 12	6	100	6

Thermal calculation

Thermal calculation is performed to verify if the gearbox can function without overheating, in the case of overheating being present, a cooling system will be installed. The first calculation is computed at 30°C (ambient temperature) taking into account the factor f2 and will verify whether a cooling system is needed or not.

Table 8. Thermal factor f2

f_2 for ambient temperature					
for standard units without cooling $P_{G_1} \cdot f_2$	Ambient temperature in °C 10	20	30	40	50
f_2	0.89	1	1.14	1.33	1.6

Table 9. Thermal capacity

$i_{GB} := 10$

Wärme-Grenzleistungen / Thermal capacities	i_N	n_{1N}	112	125	140	160	180	200	225	250	280	320	360	400	450	500	560	630	710
			Getriebegrößen Gear unit sizes – Grenz-Leistungen P_{G_1} in kW Thermal capacities P_{G_1} in kW																
P_{G_1} ohne Kühlung without cooling	7,1	1500	29,5	39	46	62	74	90	115	140	180	225	280	360	450	570	710	890	1150
	10	1000	24,5	32	40	51	67	86	110	135	170	220	270	350	440	560	700	880	1150
	11,2	750	22	29	37	47	61	80	105	125	165	215	265	340	430	550	690	870	1100
	12,5	1500	23,5	33	41	53	68	87	110	135	175	220	270	350	430	550	700	870	1100
	↓	1000	20,5	27	35	45	57	77	100	125	165	210	260	340	420	540	680	860	1100
	20	750	18,5	24,5	32	40	53	70	90	115	150	195	250	320	410	520	670	850	1050

$f_2 := 1.14$

$P_{WM} \cdot f_2 = 14.022\ kW$ $\qquad$ $14.02\ kW < 40\ kW$

The thermal capacity of the selected gearbox is not exceeded by the working machine power.

The thermal calculation is done, the conclusion is that a cooling system won't be needed since the working machine power is much lower than the thermal capacity.

Coupling selection

$P_{WM}=12.3\ kW$ $\qquad$ $d_{WM}=55\ mm$ $\qquad$ $n_{WM}=100\ rpm$

diameter of gearbox output shaft: $d_{GB}:=28\ mm$

$$M_{t2}:=9.55\cdot 10^{6}\cdot\frac{P_{WM}}{n_{WM}}$$

$$M_{t2}:=1174.6\ N\cdot m$$

Rated torque: $\quad T_{KN}:=M_{t2}\cdot K_A=1527\ N\cdot m$

Based on the rated torque, the coupling can be selected

Fig 4. Coupling dimensioning

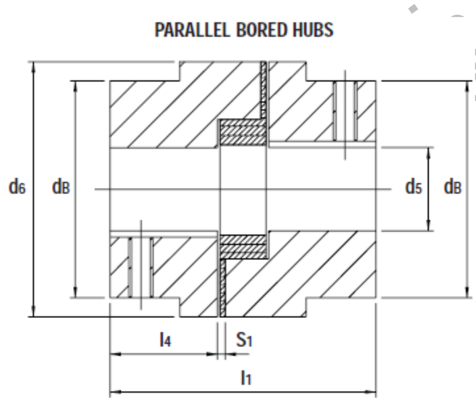


Table 10. Coupling selection

Dimensions and Technical Data - Parallel Bored Hubs

Coupling Size	Rating kW/ 100 rpm	Torque T_N Nm	l_1 mm	l_4 mm	d_6 mm	d_B mm	S_1 mm	Hub Bore d_5 Min mm	Hub Bore d_5 Max mm	Max Speed n_{max} rpm	Weight Unbored Hubs CI Kg	Weight Unbored Hubs Aluminium Kg	Moment of Inertia WR^2 CI kgm^2	Moment of Inertia WR^2 Aluminium kgm^2
P295	13.4	1280	238	95	237	162	3	30	95	2500	44.0	-	0.256	-

I have chosen the coupling of size P295 since it's torque of 1280Nm is higher than the rated torque that i have obtained, furthermore the bore dimensions interval is 30-95mm which satisfies the shaft diameters of 28-55mm plus, the coupling can be modified if needed

System arrangement

Fig 5. Location of components $X = 600\ mm$ $Y = -500\ mm$

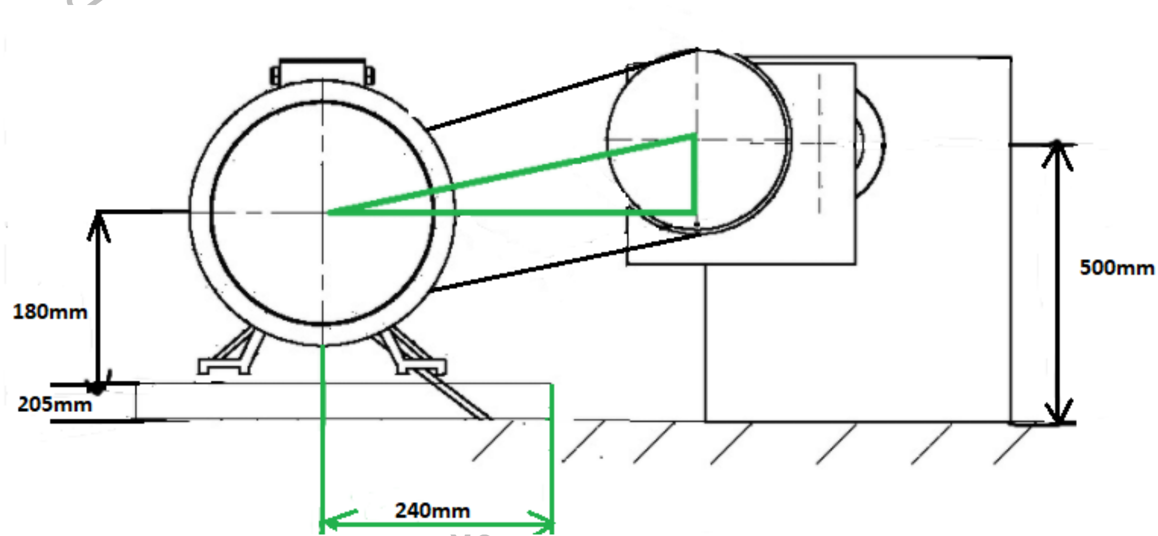


Fig 6. Top view of the system

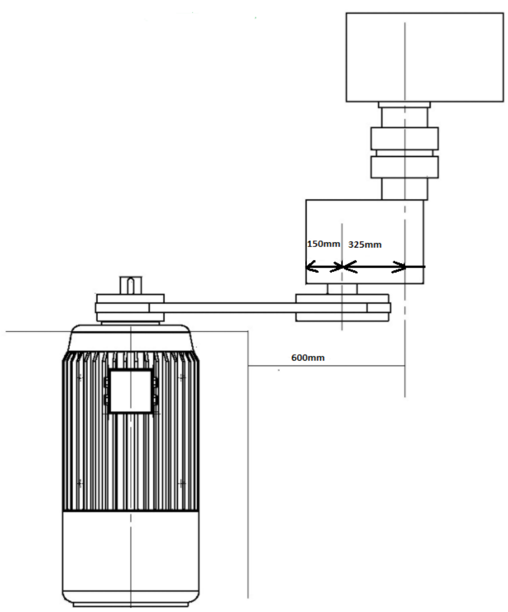
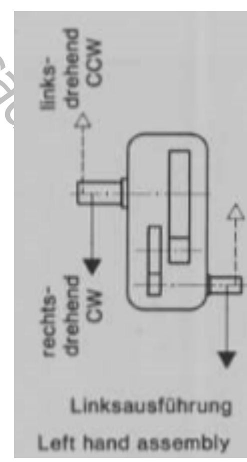
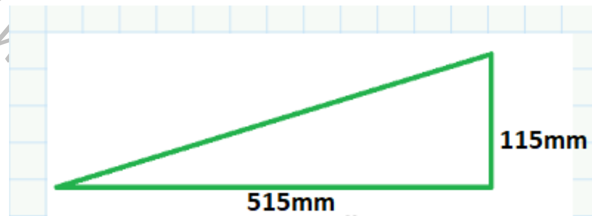


Fig 7. Assembly arrangement of the gearbox



The gearbox is left hand assembled so the input shaft will be the bigger one.
Since y is negative, the gearbox is above the electric motor.

Fig 8. Calculation of the center distance



Using the Pythagoras' Theorem, the center distance, A, was calculated as:

$$A := \sqrt{115^2 + 515^2} = 527.684 \qquad A := 527.68\ mm$$

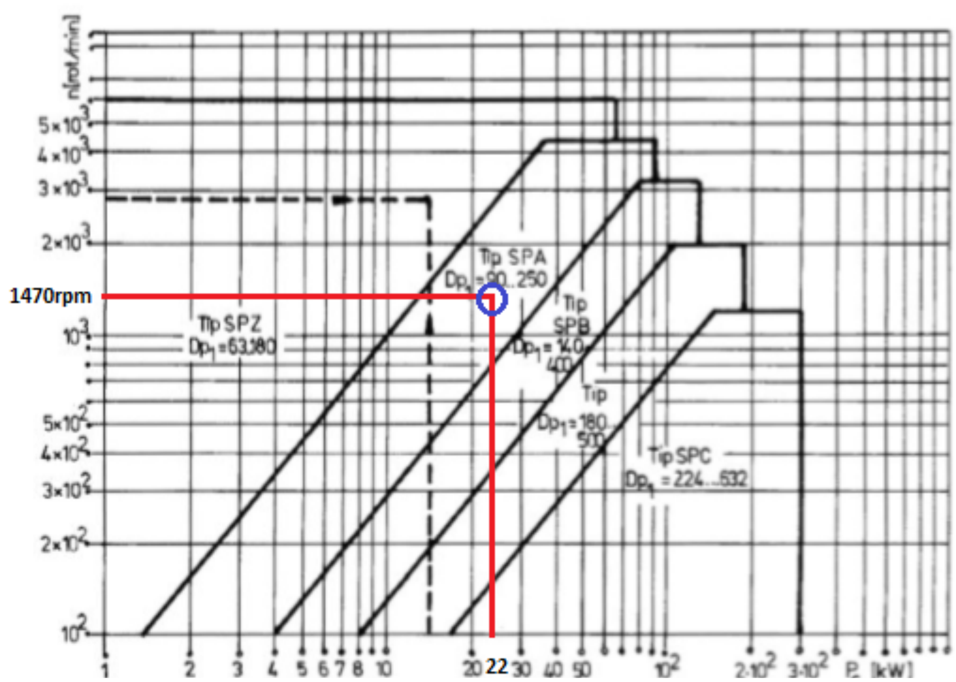
Belt transmission design

Input data:

$$P_1 := P_m = 22\ kW \qquad i_{BD} := 1.47 \qquad n_1 := n_{EM} = 1470\ rpm$$

$$K_A = 1.3 \qquad A_{init} := 527.68\ mm$$

Fig 9. Determination of the belt type



Based on the graph, the suggested belt type is SPA, but in order to get the best belt type, other iterations must be computed as well.

Table 11. Belt type iterations

DESIGN OF V-BELT TRANSMISSION

Nr.crt.	Input Data				Selected Data (from catalog)			Diameter of the sheave according to initial belt transmission ratio	Belt transmission ratio after the sheave standardization
	P1 [kW]	*n1 [rpm]*	*iBD [-]*	*Ainit [-]*	*V-Belt size*	*Dp1 [mm]*	*Dp2 [mm]*	*Dp2 [mm]*	*iBD [-]*
1	22	1470	1.47	527.68	SPA	150	224	220.5	1.49
2					SPA	100	150	147	1.5
3					SPA	132	200	194.04	1.52
4					SPA	118	180	173.46	1.53
5					SPB	160	236	235.2	1.48
6					SPB	170	250	249.9	1.47
7					SPB	190	280	279.3	1.47

Table 12. Number of belts

Belt Length (corresponding to product catalogs)	Effective centre distance (from catalogue)	Effective centre distance (calculated)	Power transmitted by one belt	Estimated nr. of belts	Final nr. of belts	Comment
Lp [mm]	*A [mm]*	*A [mm]*	*P0 [kW]*	*z0 [-]*	*z [-]*	
1600	505	504.905	7.24	4.36	5	
1400	503	503.029	3.52	9.08	11	
1600	538	538.174	5.93	5.30	6	
1600	565	565.101	4.88	6.42	8	
1800	588	587.754	9.95	3.34	4	
1800	569	568.726	11	3.03	3	
1800	529	528.949	13.05	2.57	3	

Table 13. Chosen belts of SPA type

Centre Distance SPA Wedge Belt Drives

Combined Arc and Belt Length Correction Factor					0,80	0,85			0,90		0,95		1,00			1,05		1,10			
Speed Ratio	Pitch Diameter of Pulleys		Power per SPA Belt (kW)		BELT LENGTH																Speed Ratio
	Driver	Driven	1440 rev/min	2880 rev/min	800	900	1000	1120	1250	1400	1600	1800	2000	2240	2500	2800	3150	3550	4000	4500	
1,39	*85	118	2,89	4,92	240	290	340	400	465	540	640	–	–	–	–	–	–	–	–	–	1,39
1,39	90	125	2,75	4,56	230	281	331	391	456	531	631	731	831	951	1081	1231	1406	1606	1831	2081	1,39
1,39	180	250	9,38	15,80	–	–	–	–	285	361	461	561	661	781	912	1062	1237	1437	1662	1912	1,39
1,39	95	132	3,14	5,26	221	271	321	381	446	521	621	721	822	942	1072	1222	1397	1597	1822	2072	1,39
1,40	*80	112	2,50	4,20	249	299	349	409	474	549	649	–	–	–	–	–	–	–	–	–	1,40
1,40	100	140	3,52	5,95	211	261	311	371	436	511	611	711	811	931	1061	1211	1386	1586	1811	2061	1,40
1,40	160	224	7,96	13,56	–	–	–	256	322	397	497	598	698	818	948	1098	1273	1473	1698	1948	1,40
1,40	200	280	10,77	17,87	–	–	–	–	–	321	421	521	622	742	872	1022	1197	1397	1623	1873	1,40
1,42	106	150	3,98	6,77	198	248	298	358	423	498	599	699	799	919	1049	1199	1374	1574	1799	2049	1,42
1,43	112	160	4,43	7,57	185	235	285	346	411	486	586	686	786	906	1036	1186	1361	1561	1786	2036	1,43
1,43	140	200	6,51	11,17	–	–	231	291	357	432	532	632	732	852	983	1133	1308	1508	1733	1983	1,43
1,44	125	180	5,41	9,28	–	209	259	319	384	460	560	660	760	880	1010	1160	1335	1535	1760	2010	1,44
1,47	90	132	2,75	4,56	225	275	325	385	450	525	625	725	825	945	1075	1225	1400	1601	1826	2076	1,47
1,47	*85	125	2,89	4,92	234	284	334	395	460	535	635	–	–	–	–	–	–	–	–	–	1,47
1,47	95	140	3,14	5,26	214	264	315	375	440	515	615	715	815	935	1065	1215	1390	1590	1815	2065	1,47
1,47	*80	118	2,50	4,20	244	294	344	404	469	544	644	–	–	–	–	–	–	–	–	–	1,47
1,49	150	224	7,24	12,39	–	–	–	264	329	405	$\boxed{505}$	605	705	825	956	1106	1281	1481	1706	1956	1,49

Table 14. Chosen belts of SPB type

Centre Distance SPB Wedge Belt Drives

Combined Arc and Belt Length Correction Factor					0,85		0,90		0,95			1,00		1,05					1,15		
Speed Ratio	Pitch Diameter of Pulleys		Power per SPB Belt (kW)		BELT LENGTH																Speed Ratio
	Driver	Driven	1440 rev/min	960 rev/min	1260	1410	1800	2020	2150	2280	2400	2990	3550	3800	4060	4310	4560	4820	7100	7990	
1,43	140	200	7,83	5,63	362	437	632	742	808	873	933	1228	–	–	–	–	–	–	–	–	1,43
1,43	280	400	21,65	15,67	71	160	361	472	538	603	663	959	1240	1365	1495	1620	1745	1875	–	–	1,43
1,47	170	250	11,00	7,87	298	373	569	679	744	809	869	1165	1445	1570	1700	–	–	–	–	–	1,47
1,47	190	280	13,05	9,34	257	333	529	639	705	770	830	1125	1405	1530	1660	1785	–	–	–	–	1,47
1,48	160	236	9,95	7,13	317	392	588	698	763	828	888	1184	1464	1589	–	–	–	–	–	–	1,48

The chosen belt is of type SPA, with the sheave diameters :

$$D_{p1} := 150\ mm \qquad\qquad D_{p2} := 224\ mm$$

This belt was selected due to the number of belts being below 6, and the sheaves diameter and width don't surpass the shaft length.

angle between the two sides of the belt: $\gamma := 2 \cdot \operatorname{asin}\left(\dfrac{D_{p2} - D_{p1}}{2 \cdot A_{init}}\right) = 8\ deg$

$$3 \cdot \left(D_{p1} + D_{p2}\right) = 1122\ mm \qquad\qquad D_{p2} < A_{init} < 3 \cdot \left(D_{p1} + D_{p2}\right)$$

angle of wrap on driving sheave: $\beta_1 := 180 - 8 = 172$

angle of wrap on driven sheave: $\beta_2 := 180 + 8 = 188$

Approx. belt length:

$$Lp_{init} := 2 \cdot A_{init} + \frac{\pi \cdot (D_{p1} + D_{p2})}{2} + \frac{(D_{p2} - D_{p1})^2}{4 \cdot A_{init}} = 1645 \ mm$$

belt length standardized from the catalogue: $L_p := 1600 \ mm$

Calculation of the effective center distance:

$$f(A) := 8 \cdot A^2 + (2 \cdot \pi \cdot (D_{p1} + D_{p2}) - 4 \cdot L_p) \cdot A + (D_{p2} - D_{p1})^2$$

guess value: $A := A_{init}$

$Given \qquad f(A) = 0$

$\mathbf{A := find(A)}$ $\qquad A := 527.68 \ mm$ $\qquad$ effective center distance

$A := 505 \ mm$ $\qquad$ effective center distance according to the catalogue

Table 15. Length correction factor

Working length, L_p.[mm]	Belt size			
	SPZ	SPA	SPB	SPC
1600	1.00	0.93	0.86	

length correction factor: $c_L := 0.93$

wrapping factor: $c_\beta := 1 - 0.003 \cdot (180 - 172) = 0.976$

power transmitted by one belt: $P_0 := 7.24 \ kW$

$$\nu := \frac{\pi \cdot D_{p1} \cdot n_1}{60000} \qquad \nu := 11.5 \ \frac{m}{s}$$

peripheral speed $\qquad \nu < 40 \ \frac{m}{s}$

$$z_0 := \frac{K_A \cdot P_1}{c_L \cdot c_\beta \cdot P_0} \qquad z_0 := 4.36$$

Table 16. V-belt load repartition factor

Number of belts, z_0	c_z
2…3	0.95
4….6	0.90
over 6	0.85

$c_z := 0.9$

$z := \frac{z_0}{c_z}$ $\qquad z := 5$

$f := 10^3 \cdot 2 \cdot \frac{\nu}{L_p}$ $\qquad f := 14.37\ Hz$ $\qquad$ bending frequency $\qquad f < 40\ Hz$ $\qquad$ it verifies

The bending frequency is within the limits shown in the Table 17.

$F_u := 10^3 \cdot \frac{P_1}{\nu}$ $\qquad F_u := 1913\ N$ $\qquad$ peripheral force

$F_0 := 1.5 \cdot F_u = 2870\ N$ $\qquad$ belt tensioning force

Table 17. V-belts groove dimensions

Groove size	Z	A	B	C
Belt size	SPZ	SPA	SPB	SPC
l_p	8.5	11	14	19
n_{min}	2.5	3.3	4.2	5.7
m_{min}	9	11	14	19
f	8 ± 1	$\boxed{10^{+2}_{-1}}$	12.5^{+2}_{-1}	17^{+2}_{-1}
e	12 ± 0.3	$\boxed{15 \pm 0.3}$	19 ± 0.4	22.5 ± 0.5
α	$38° \pm 1°$ $34° \pm 1°$	$38° \pm 1°$ $34° \pm 1°$	$38° \pm 1°$ $34° \pm 1°$	$38° \pm 30'$ $36° \pm 30'$
[illegible]	0.5	1.0	1.0	1.5

$f := 14.37\ hz$

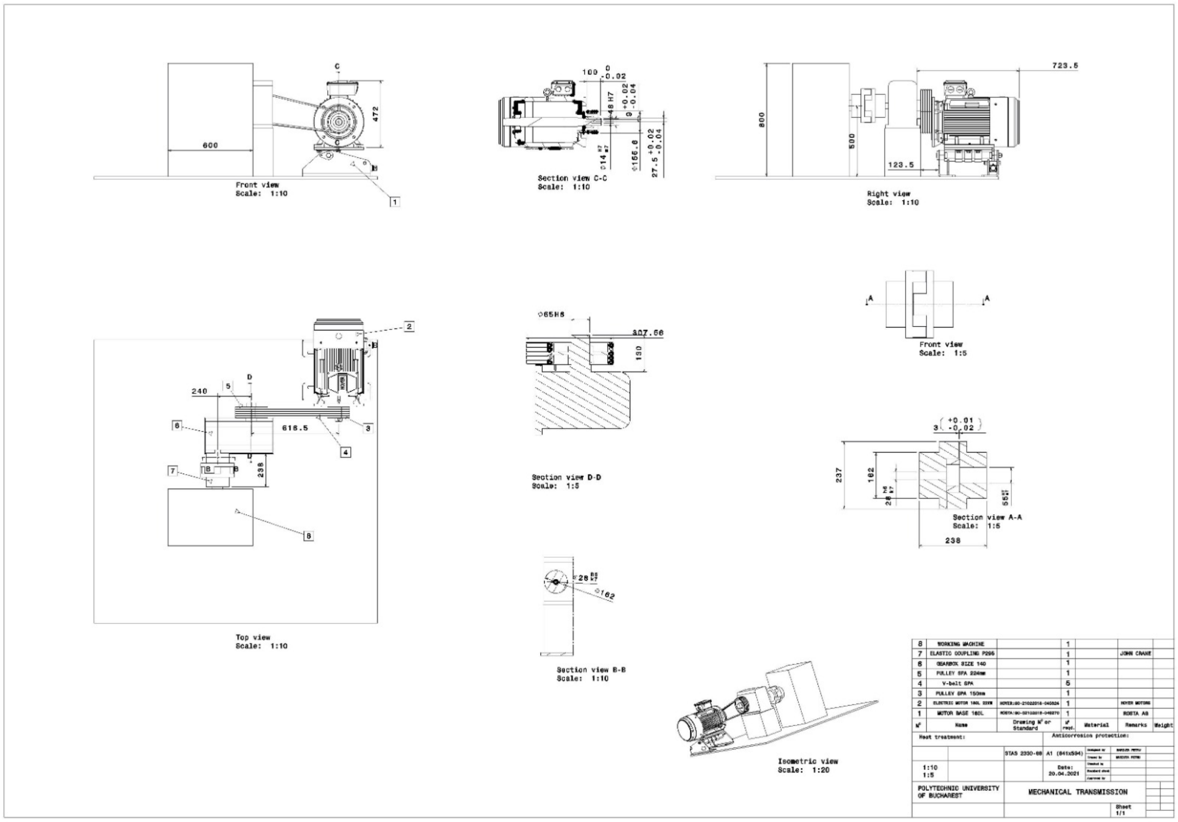
$e := 15$

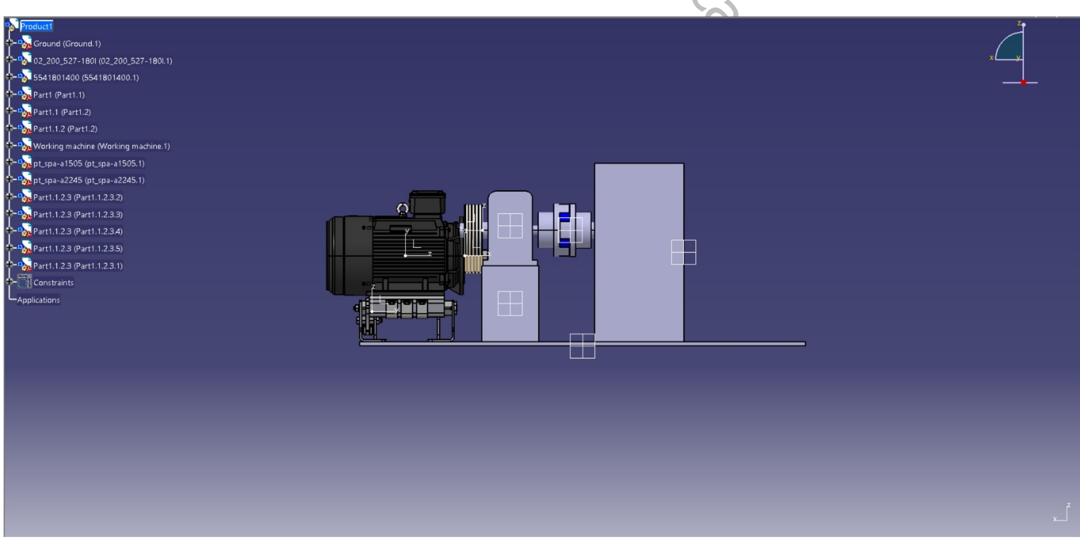
$B := (z-1) \cdot e + 2 \cdot f = 88.74$ $\qquad$ sheave width

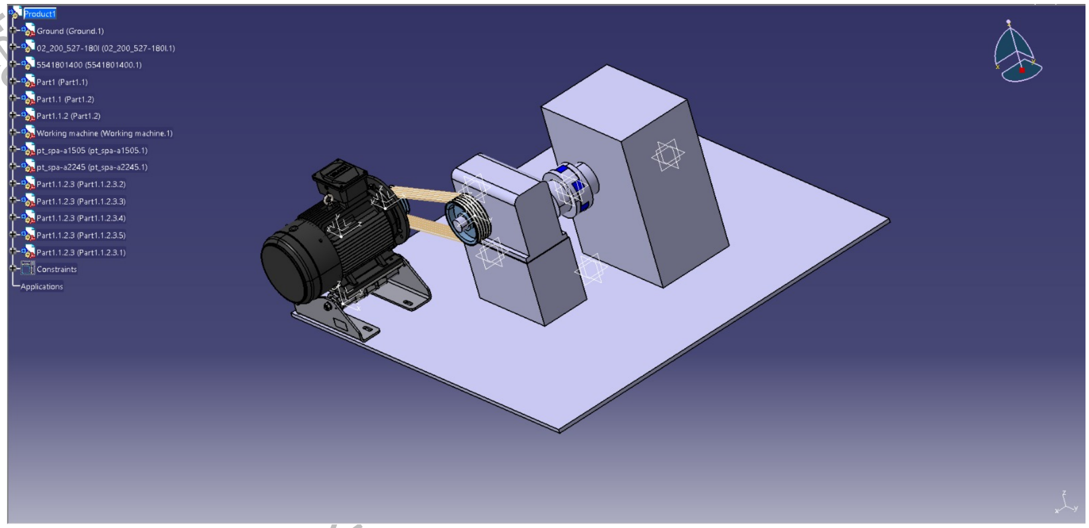
$l_2 := 105\ mm$ $\qquad$ length of the input end-shaft of the gearbox

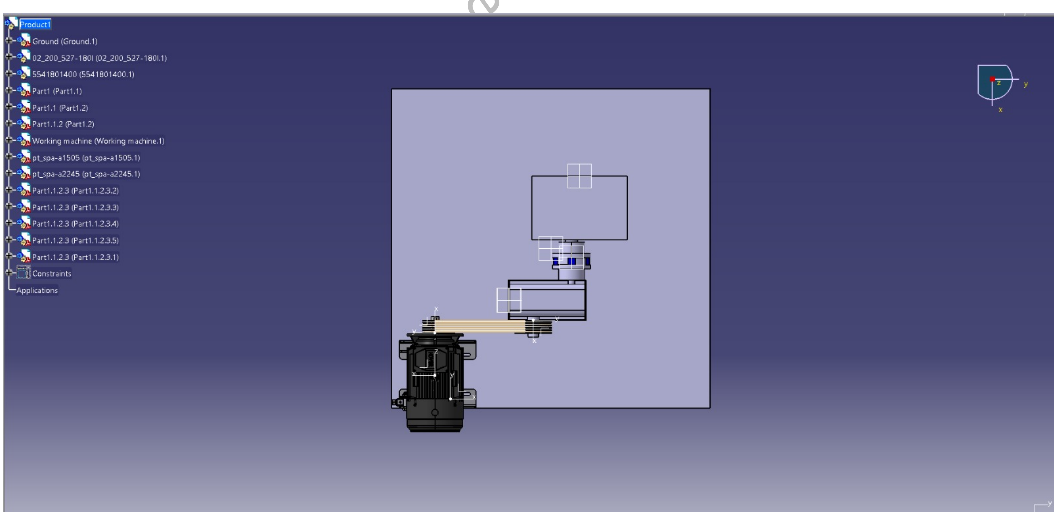
$B < l_1$ $\qquad$ the driven sheave can be mounted on the gearbox

Fig.10 Assembly






Product1
Ground (Ground.1)
02_200_527-180l (02_200_527-180l.1)
5541801400 (5541801400.1)
Part1 (Part1.1)
Part1.1 (Part1.2)
Part1.1.2 (Part1.2)
Working machine (Working machine.1)
pt_spa-a1505 (pt_spa-a1505.1)
pt_spa-a2245 (pt_spa-a2245.1)
Part1.1.2.3 (Part1.1.2.3.2)
Part1.1.2.3 (Part1.1.2.3.3)
Part1.1.2.3 (Part1.1.2.3.4)
Part1.1.2.3 (Part1.1.2.3.5)
Part1.1.2.3 (Part1.1.2.3.1)
Constraints
Applications


Product1
Ground (Ground.1)
02_200_527-180l (02_200_527-180l.1)
5541801400 (5541801400.1)
Part1 (Part1.1)
Part1.1 (Part1.2)
Part1.1.2 (Part1.2)
Working machine (Working machine.1)
pt_spa-a1505 (pt_spa-a1505.1)
pt_spa-a2245 (pt_spa-a2245.1)
Part1.1.2.3 (Part1.1.2.3.2)
Part1.1.2.3 (Part1.1.2.3.3)
Part1.1.2.3 (Part1.1.2.3.4)
Part1.1.2.3 (Part1.1.2.3.5)
Part1.1.2.3 (Part1.1.2.3.1)
Constraints
Applications

Tentative Gear Sizing

Input data

Horizontal helical gearbox   Clockwise rotation

$i_{GB} := 3.6 \quad i_{12} := i_{GB} \quad n_2 := 100\ rpm \quad n_{EM} := 1470\ rpm \quad P_2 := 12.3\ kW$

$P_{EM} := 22\ kW$ power of the electric motor

$n_1 := n_2 \cdot i_{GB} = 360\ rpm$ rotational speed at the entrance of the gearbox

$M_{t2} := 9.55 \cdot 10^6 \cdot \dfrac{P_2}{n_2} \qquad M_{t2} := 1174.6\ N \cdot m$

$K_A := 1.3$

$n_{EM\_s} := 1500\ rpm$ idle rotation speed of the electric motor

$X := 600\ mm \qquad Y := -500\ mm$

$d_{WM} := 55\ mm \qquad l_{WM} := 82\ mm \qquad h_{WM} := 500\ mm$

Total efficiency for a 1 stage gearbox

$\eta_{BD} := 0.96$ efficiency of the belt drive

$\eta_{HG} := 0.98$ efficiency of a helical gear set

$\eta_C := 0.97$ churning efficiency

$\eta_{RB} := 0.994$ efficiency of the rolling bearings

$\eta_t := \eta_{BD} \cdot \eta_{HG} \cdot \eta_C \cdot \eta_{RB}^{\ 2} = 0.902$ total efficiency

$P_1 := P_{EM} \cdot \eta_{BD} = 21.12\ kW$ power at the entrance of the gearbox

$M_{t1} := 9.55 \cdot 10^6 \cdot \dfrac{P_1}{n_1}$

$M_{t1} := 560266\ N \cdot mm$ torque at the entrance of the gearbox

Recommended values for typical steels used for gears

Material	Heat treatment	Hardness bulk [HB]	Hardness flank [HRC]	Ultimate stress, R_m [MPa]	Yield (Proof) stress, $R_{p0.2}$ [MPa]	Allowable bending stress number, σ_{0lim} [MPa]	Allowable stress number for contact stress, $\sigma_{H\,lim\,b}$ [MPa]
OLC 45	Through hardening	HB=2200÷2600MPa		620	360	0.04HB+140	0.15HB+200
	Flame or induction hardening	2000÷2600 MPa	50÷57			250÷350	20HRC+10
OLC 55	Through hardening	HB=2000÷3000MPa		730	420	0.04HB+140	0.15HB+200
	Flame or induction hardening	2000÷3000MPa	50÷57			250÷350	20HRC+10
41 MoCr11*	Through hardening	HB=2700÷3200MPa		950	750	0.04HB+155	0.18HB+200
	Flame or induction hardening	2700÷3200 MPa	50÷57			230÷290	20HRC+10
	Nitriding	2700÷3200 MPa	52÷60			250÷350	20HRC

The chosen gear material is OLC 45 **through hardening**.

<u>Material properties</u>

$HB := 2600\ MPa$

$HRC := 2600\ MPa$

$R_m := 620\ MPa$

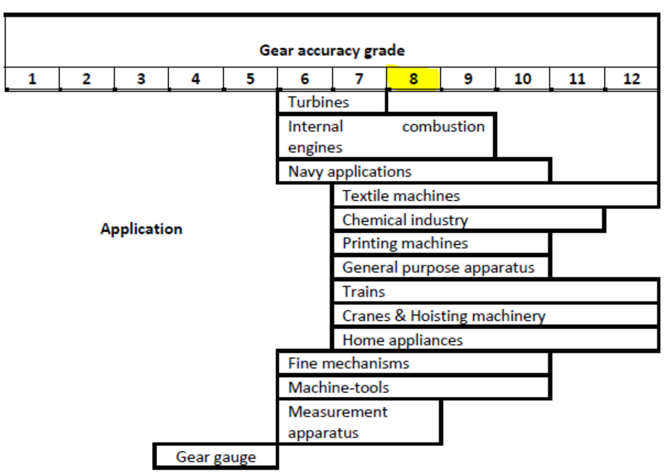
$R_{p0.2} := 360\ MPa$

$\sigma_{0lim} := 0.04 \cdot HB + 140\ MPa = 244\ MPa$

$\sigma_{H.limb} := 590\ MPa$

<u>Helix angle:</u> $\beta := 15$

Correlation between gear accuracy grade and applications



The gear accuracy grade is 8 according to the application of the gears (general purpose apparatus).

Reccomended values for relative width, Ψd, at helical gearset

Flank Hardness	Position of the pinion		Gear quality grade 5 - 6	7 - 8	9 - 10
Hardness of each gear HB≤3500 MPa	Between bearings	Symmetrical	1.5	1.3	0.8
		Non symmetrical	1.3	0.8	0.6
	Overhung		0.5	0.4	0.3
Hardness of each gear HB>3500MPa	Between bearings	Symmetrical	1.0	0.7	0.4
		Non symmetrical	0.7	0.4	0.3
	Overhung		0.4	0.3	0.2

Following the table, the relative width for the pinion placed symmetrically between the bearings (possible due to the gearbox having 1 stage) and of quality 8 is **1.3**.

$$\psi_d := 1.3$$

Determination of minimum center distance

$$K_H := 1 \cdot 10^5 \ \frac{N}{mm^2}$$ global factor for contact stress

$$a_{min} := \left(1 + i_{GB}\right) \cdot \sqrt[3]{\frac{K_H \cdot K_A \cdot M_{t1}}{\psi_d \cdot \sigma_{H.limb}^{\ 2}} \cdot \frac{\left(1 + i_{GB}\right)}{i_{GB}}} = 271.523 \ mm$$

The center distance has to be standardized according to a standard.

Standard values for center distances (STAS 6055 - 82)

I	II	I	II
50	56	160	180
63	71	200	225
80	90	250	280
100	112	315	355
125	140	400	450

standardized center distance $a_w := 280 \ mm$

Determination of the minimum module

$K_f := 2.2$ global factor for bending stress

$$m_{n.min} := \frac{K_f \cdot K_A \cdot M_{t1}}{\psi_d \cdot a_w^{\ 2} \cdot \sigma_{0lim}} \cdot \left(1 + i_{GB}\right)^2 = 1.363\ mm$$

The module has to be standardized according to a standard.

Standard values for module (STAS 822 - 82)

I	II	I	II
1			4.5
	1.125	5	
1.25			5.5
	1.375	6	
1.5			7
	1.75	8	
2			9
	2. 25	10	
2,5			11
	2.75	12	
3			14
	3.5	16	
4			18

Iterations

Computation of number of teeth based on different modules

m_n	z_1*	z_1	z_2*	z_2	a_0	i_{eff}	Δ_i	Profile shifting	0.3*m_n	a_w-a_0	1.3*m_n
1.5	78.39	79.00	284.4	283.00	281.08	3.58	0.49	-0.72	0.45	-1.08	1.95
1.75	67.19	68	244.8	243.00	281.72	3.57	0.74	-0.99	0.525	-1.72	2.275
2	58.80	59	212.4	213.00	281.60	3.61	0.28	-0.80	0.6	-1.60	2.6
2.25	52.26	53	190.8	191.00	284.18	3.60	0.10	-1.86	0.675	-4.18	2.925
2.5	47.04	48	172.8	173.00	286.00	3.60	0.12	-2.40	0.75	-6.00	3.25
3	39.20	40	144	143.00	284.18	3.58	0.69	-1.39	0.9	-4.18	3.9
3.5	33.60	34	122.4	123.00	284.44	3.62	0.49	-1.27	1.05	-4.44	4.55
4	29.40	30	108	106.00	281.60	3.53	1.85	-0.40	1.2	-1.60	5.2
4.5	26.13	26	93.6	93.00	277.20	3.58	0.64	0.62	1.35	2.80	5.85
5	23.52	24	86.4	84.00	279.52	3.50	2.78	0.10	1.5	0.48	6.5

The chosen module for the gears is 4.5 since the number of teeth for the pinion is less than the limit for through hardened gears.

Also, the profile shifting is positive, the selected module satisfies the first restriction and Δi <u>is less than 3%.</u>

<u>Restrictions</u>:

I. $0.3 \cdot m_n < a_w - a_0 < 1.3\ m_n$

II. $\Delta i = \frac{\left|i_{12\ eff} - i_{12}\right|}{i_{12}} 100 \leq 3\%$ (*strong condition* in principle)

III. z_1 and z_2 have no common dividers (*weak condition* i.e. only if possible).

$m_n := 4.5$

$$z_{1.s} := \frac{2 \cdot a_w \cdot \cos(\beta)}{(1 + i_{GB}) \cdot m_n} \qquad z_{1.s} := 26.13$$

$z_1 := 26$ ***teeth*** $\qquad z_1 < 35$

The number of teeth doesn't exeed the limit in the case of the selected treatment for the material of the gears (through hardened).

$z_{2.s} := z_1 \cdot i_{12} = 93.6$

$z_2 := 93 \qquad teeth$

Geometric calculation of the gearing

Input data

$m_n = 4.5$

$z_1 := 26$

$z_2 := 93$

$\beta := 15\ deg$

$a_w = 280\ mm$

$\psi_d = 1.3$

Basic rack

$\alpha_0 := 20\ deg$

$h_{star\_0a} := 1$

$h_{star\_0f} := 1.25$

$c_{star\_0} := 0.25$

Computation

$$\alpha_t := \operatorname{atan}\left(\frac{\tan(\alpha_0)}{\cos(\beta)}\right) = 20.6469\ deg$$

$$m_t := \frac{m_n}{\cos(\beta)} = 4.65874$$

$$a := m_t \cdot \frac{(z_1 + z_2)}{2} = 277.195 \qquad a := 277.195\ mm$$

$$\alpha_w := \operatorname{acos}\left(\frac{a}{a_w} \cdot \cos(\alpha_t)\right) = 22.12007\ deg$$

$$x_\Sigma := \frac{((\tan(\alpha_w) - \alpha_w) - (\tan(\alpha_t) - \alpha_t)) \cdot (z_1 + z_2)}{2 \cdot \tan(\alpha_t) \cdot \cos(\beta)} = 0.645$$

$$z_{e\_1} := \frac{z_1}{\cos(\beta)^3} = 28.8497$$

$$z_{e\_2} := \frac{z_2}{\cos(\beta)^3} = 103.1933$$

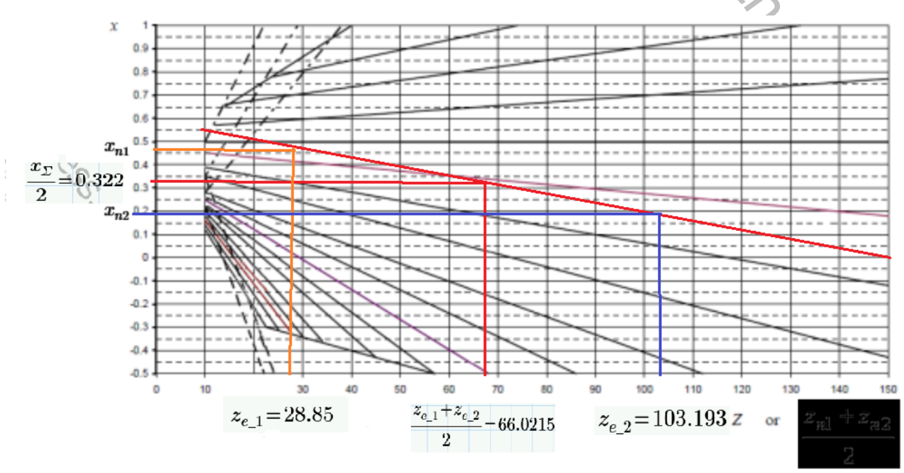
$$z_{e\_1} = 28.85$$

$$x_\Sigma$$

$$\frac{z_{e\_1} + z_{e\_2}}{2} = 66.0215 \qquad \frac{x_\Sigma}{2} = 0.322$$

$$z_{e\_2} = 103.193$$

Determination of shifting coefficient



Following the graph, the values for the shifting coefficients of the gear are:

$x_{n1} := 0.46$ $\quad\quad$ $x_{n2} := 0.19$

Reference diameters

$d_1 := m_t \cdot z_1 = 121.127 \quad mm$

$d_2 := m_t \cdot z_2 = 433.263 \quad mm$

Base diameters

$d_{b1} := d_1 \cdot \cos\left(\alpha_t\right) = 113.347 \quad mm \qquad d_{b1} := 113.347\ mm$

$d_{b2} := d_2 \cdot \cos\left(\alpha_t\right) = 405.435 \quad mm \qquad d_{b2} := 405.435\ mm$

Pitch diameters (rolling diameters)

$$d_{w1} := d_1 \cdot \frac{\cos\left(\alpha_t\right)}{\cos\left(\alpha_w\right)} = 122.353 \quad mm$$

$$d_{w2} := d_2 \cdot \frac{\cos\left(\alpha_t\right)}{\cos\left(\alpha_w\right)} = 437.647 \quad mm$$

Root diameters

$d_{f1} := d_1 - 2 \cdot m_n \cdot \left(h_{star\_0a} + c_{star\_0} - x_{n1}\right) = 114.017 \quad mm$

$d_{f2} := d_2 - 2 \cdot m_n \cdot \left(h_{star\_0a} + c_{star\_0} - x_{n2}\right) = 423.723 \quad mm$

Tooth depth

$a_w := 280 \quad mm$

$h := a_w - m_n \cdot c_{star\_0} - 0.5 \cdot \left(d_{f1} + d_{f2}\right) = 10.005 \quad mm$

Teeth shortening (tip depth modification)

$\Delta h := m_n \cdot \left(2 \cdot h_{star\_0a} + c_{star\_0}\right) - h = 0.12 \quad mm$

Outside diameters (tip diameters)

$d_{a1} := d_{f1} + 2 \cdot h = 134.027 \quad mm \qquad d_{a1} := 134.027\ mm$

$d_{a2} := d_{f2} + 2 \cdot h = 443.733 \quad mm \qquad d_{a2} := 443.733 \; mm$

$$\frac{\left(d_{a1} + \left(d_{a2}\right)\right)}{2} = 288.88 \; mm$$

Pressure angle on tip diameter

$\alpha_{a1} := \dfrac{d_{b1}}{d_{a1}} = 48.455 \; deg \qquad \alpha_{a2} := \dfrac{d_{b2}}{d_{a2}} = 52.351 \; deg$

Face width

$b_2 := d_1 \cdot \psi_d = 157.466 \quad mm \qquad b_2 := 146 \quad mm$

$b_1 := b_2 + m_n = 150.5 \; mm$

Geometric quality of the gearing

Pressure angle on tip diameter

$d_1 := 121.127 \; mm \qquad d_2 := 433.263 \; mm$

$$\alpha_{a\_t1} := \operatorname{acos}\left(\frac{d_1}{d_{a1}} \cdot \cos\left(\alpha_t\right)\right) = 32.253 \; deg$$

$$\alpha_{a\_t2} := \operatorname{acos}\left(\frac{d_2}{d_{a2}} \cdot \cos\left(\alpha_t\right)\right) = 23.979 \; deg$$

Arc tooth thickness on tooth diameter

$inv\left(\alpha_t\right) := \tan\left(\alpha_t\right) - \alpha_t$

$$S_{at1} := d_{a1} \cdot \left(\frac{\pi + 4 \cdot x_{n1} \cdot \tan\left(\alpha_0\right)}{2 \cdot z_1} + inv\left(\alpha_t\right) - inv\left(\alpha_{a\_t1}\right)\right) = 2.901 \; mm$$

$$S_{at2} := d_{a2} \cdot \left(\frac{\pi + 4 \cdot x_{n2} \cdot \tan\left(\alpha_0\right)}{2 \cdot z_2} + inv\left(\alpha_t\right) - inv\left(\alpha_{a\_t2}\right)\right) = 3.795 \; mm$$

$0.25 \cdot m_n = 1.125 \qquad S_{at1} > 0.25 \cdot m_n \quad S_{at2} > 0.25 \cdot m_n$

The condition for through hardened steel gear material is satisfied.

Diameters corresponding to the beginning of the involute profile

$$d_{l1} := d_{b1} \cdot \sqrt{1 + \left(\tan\left(\alpha_t\right) - \frac{2 \cdot \left(h_{star\_0a} - x_{n1}\right) \cdot \cos\left(\beta\right)}{z_1 \cdot \sin\left(\alpha_t\right) \cdot \cos\left(\alpha_t\right)}\right)^2} = 116.98\ mm$$

$$d_{l2} := d_{b2} \cdot \sqrt{1 + \left(\tan\left(\alpha_t\right) - \frac{2 \cdot \left(h_{star\_0a} - x_{n2}\right) \cdot \cos\left(\beta\right)}{z_2 \cdot \sin\left(\alpha_t\right) \cdot \cos\left(\alpha_t\right)}\right)^2} = 426.412\ mm$$

Diameters corresponding to the beggining of the active profile (first contact)

$a_w := 280\ mm$

$$d_{A1} := \sqrt{{d_{b1}}^2 + \left(2 \cdot a_w \cdot \sin\left(\alpha_w\right) - \sqrt{{d_{a2}}^2 - {d_{b2}}^2}\right)^2} = 117.387\ mm$$

$$d_{A2} := \sqrt{{d_{b2}}^2 + \left(2 \cdot a_w \cdot \sin\left(\alpha_w\right) - \sqrt{{d_{a1}}^2 - {d_{b1}}^2}\right)^2} = 428.712\ mm$$

$d_{l1} < d_{A1}$ $\quad$ $d_{l2} < d_{A2}$ $\quad$ The conditions are satisfied.

Profile contact ratio in frontal plane

$$\varepsilon_\alpha := \frac{\sqrt{{d_{a1}}^2 - {d_{b1}}^2} \cdot \cos\left(\beta\right)}{2 \cdot \pi \cdot m_n \cdot \cos\left(\alpha_t\right)} + \frac{\sqrt{{d_{a2}}^2 - {d_{b2}}^2} \cdot \cos\left(\beta\right)}{2 \cdot \pi \cdot m_n \cdot \cos\left(\alpha_t\right)} - \frac{a_w \cdot \sin\left(\alpha_w\right) \cdot \cos\left(\beta\right)}{\pi \cdot m_n \cdot \cos\left(\alpha_t\right)} = 1.497\ mm$$

$\varepsilon_\alpha := 1.497$ $\quad$ $\varepsilon_\alpha > 1.2$ $\quad$ The condition is fulfilled.

Overlap ratio (face contact ratio): $\quad$ $\varepsilon_\beta := \frac{b_2}{\pi \cdot m_n} \cdot \sin\left(\beta\right) = 2.673$

Total contact ratio (overall contact ratio): $\quad$ $\varepsilon_\gamma := \varepsilon_\alpha + \varepsilon_\beta = 4.17$

Measured parameters of the wheels

$$N_1 := \mathrm{ceil}\left(\frac{1}{\pi}\cdot\left(\frac{\sqrt{\left(z_1+x_{n1}\cdot\cos(\beta)\right)^2-\left(z_1\cdot\cos(\alpha_t)\right)^2}}{\cos(\alpha_t)\cdot\cos(\beta)^2}-2\cdot x_{n1}\cdot\tan(\alpha_0)-z_1\cdot inv(\alpha_t)\right)+0.5\right)$$

$N_1 = 5 \quad teeth$

$$N_2 := \mathrm{ceil}\left(\frac{1}{\pi}\cdot\left(\frac{\sqrt{\left(z_2+x_{n2}\cdot\cos(\beta)\right)^2-\left(z_2\cdot\cos(\alpha_t)\right)^2}}{\cos(\alpha_t)\cdot\cos(\beta)^2}-2\cdot x_{n2}\cdot\tan(\alpha_0)-z_2\cdot inv(\alpha_t)\right)+0.5\right)$$

$N_2 = 13 \quad teeth$

Base tangent length over N teeth

$$W_{N1} := \left(\pi\cdot\left(N_1-0.5\right)+\left(2\cdot x_{n1}\cdot\tan(\alpha_0)\right)+z_1\cdot inv(\alpha_t)\right)\cdot\left(m_n\cdot\cos(\alpha_0)\right)=63.006 \quad mm$$

$$W_{N2} := \left(\pi\cdot\left(N_2-0.5\right)+\left(2\cdot x_{n2}\cdot\tan(\alpha_0)\right)+z_2\cdot inv(\alpha_t)\right)\cdot\left(m_n\cdot\cos(\alpha_0)\right)=173.113 \quad mm$$

(Normal) Arc tooth thickness on reference diameter

$$S_{n1} := m_n\cdot\left(\frac{\pi}{2}+2\cdot x_{n1}\cdot\tan(\alpha_0)\right)=8.575 \quad mm \qquad S_{n1} := 8.575\ mm$$

$$S_{n2} := m_n\cdot\left(\frac{\pi}{2}+2\cdot x_{n2}\cdot\tan(\alpha_0)\right)=7.691 \quad mm \qquad S_{n2} := 7.691\ mm$$

(Normal) Chordal tooth thickness on reference diameter

$$S_{bar\_n1} := S_{n1}-\frac{S_{n1}^{\ 3}}{6\cdot d_1^{\ 2}}=8.568\ mm$$

$$S_{bar\_n2} := S_{n2}-\frac{S_{n2}^{\ 3}}{6\cdot d_2^{\ 2}}=7.691\ mm$$

Height at pitch chordal

$$h_{an1} := \frac{d_{a1}-d_1}{2}+\frac{S_{n1}^{\ 2}}{4\cdot d_1}\cdot\cos(\beta)^2=6.592\ mm$$

$$h_{an2} := \frac{d_{a2}-d_2}{2}+\frac{S_{n2}^{\ 2}}{4\cdot d_2}\cdot\cos(\beta)^2=5.267\ mm$$

Chordal tooth thickness

$$S_{bar\_c1} := S_{bar\_n1} \cdot \cos\left(\alpha_0\right)^2 = 7.566\ mm$$

$$S_{bar\_c2} := S_{bar\_n2} \cdot \cos\left(\alpha_0\right)^2 = 6.791\ mm$$

Constant chordal height

$$h_{bar\_cn1} := m_n \cdot \left(\left(h_{star\_0f} - \frac{\pi}{4} \cdot \sin\left(\alpha_0\right) \cdot \cos\left(\alpha_0\right)\right) + x_{n1} \cdot \cos\left(\alpha_0\right)^2\right) = 6.317 \quad mm$$

$$h_{bar\_cn2} := m_n \cdot \left(\left(h_{star\_0f} - \frac{\pi}{4} \cdot \sin\left(\alpha_0\right) \cdot \cos\left(\alpha_0\right)\right) + x_{n2} \cdot \cos\left(\alpha_0\right)^2\right) = 5.244 \quad mm$$

Nominal forces

$M_{t1} = 560.266\ N \cdot m$ input torque

Tangential forces

$$d_{w1} := 122.353\ mm$$

$$F_{t1} := \frac{2 \cdot M_{t1}}{d_{w1}} = 9158\ N \qquad F_{t2} := F_{t1}$$

Radial forces

$$F_{r1} := F_{t1} \cdot \tan\left(\alpha_t\right) = 3451\ N \qquad F_{r2} := F_{r1}$$

Axial forces

$$F_{a1} := F_{t1} \cdot \tan\left(\beta\right) = 2454\ N \qquad F_{a2} := F_{a1}$$

Normal force

$$F_n := \sqrt{F_{t1}^2 + F_{r1}^2 + F_{a1}^2} = 10090\ N$$

STRENGTH VERIFICATION OF THE GEAR SET

Calculation of surface durability

The requirement for a safe operation for the safety factor is:

$S_H > S_{HP} = 1.15$

Base helix angle $\beta_b := \operatorname{atan}\left(\frac{d_{b1}}{d_1} \cdot \tan(\beta)\right) = 14.076\ deg$

$a_0 := 277.195\ mm$

Working pressure angle (on working cylinder) $\alpha_{wt} := \operatorname{acos}\left(\frac{a_0}{a_w} \cdot \cos(\alpha_t)\right) = 22.12\ deg$

Single pair tooth contact factor $Z_H := \frac{1}{\cos(\alpha_t)} \cdot \sqrt{\frac{2 \cdot \cos(\beta_b)}{\tan(\alpha_{wt})}} = 2.335$

Elasticity factor $Z_E := 189.8\ \sqrt{MPa}$ since both gears are made of steel

Contact ratio factor for contact stress

$\varepsilon_\beta = 2.673 > 1$

It results that the contact ratio factor is $Z_\varepsilon := \sqrt{\frac{1}{\varepsilon_\alpha}} = 0.817$

Helix angle factor for contact stress $Z_\beta := \sqrt{\cos(\beta)} = 0.983$

Dynamic factor:

$v_{td} := \frac{\pi \cdot d_1 \cdot n_1}{60 \cdot 10^3}$ $v_{td} := 2.283\ \frac{m}{s}$ Peripheral speed on pitch diameter

Gear accuracy grade	Dynamic factor, K_v
5 or 6	$1 + \sqrt{v_{td}}/22$
7 or 8	$1 + \sqrt{v_{td}}/15$
9 or 10	$1 + \sqrt{v_{td}}/11$
11 or 12	$1 + \sqrt{v_{td}}/7$

Following the table, the dynamic factor is calculated as:

$$K_v := 1 + \frac{\sqrt{v_{td}}}{15\sqrt{\frac{m}{s}}} = 1.101$$

Transverse load factor for contact stress

$c_\gamma := 20$ stiffness of a pair of teeth in contact made of cast steel

$q_H := 2.75$ factor for normal pitch error according to the gear accuracy grade of 8

$$f_{pe} = 4 + 0.315 \cdot \left(m_n + 0.25 \cdot \sqrt{d_1}\right) \cdot q_H$$

$f_{pe} := 4 + 0.315 \cdot \left(4.5 + 0.25 \cdot \sqrt{121.127}\right) \cdot 2.75 = 10.282$ normal pitch error

$y_a := \frac{160\ MPa}{\sigma_{H.limb}} = 0.271$ run-in correction factor

$\varepsilon_\gamma = 4.17$ $\varepsilon_\gamma > 2$

$$b_w := d_1 \cdot \psi_d = 157.465\ mm$$

The total contact ratio is over 2, so it results that the transverse load factor is:

$$K_{H\alpha} = 0.9 + 0.4 \cdot \frac{c_\gamma \cdot \left(f_{pe} - y_a\right)}{\left(\frac{F_{t1}}{b_w}\right)} \cdot \sqrt{2 \cdot \frac{\left(\varepsilon_\gamma - 1\right)}{\varepsilon_\alpha}}$$

$$0.9 + 0.4 \cdot 20 \cdot \frac{(10.282 - 0.271)}{\frac{9158}{0.157465}} \cdot \sqrt{2 \cdot \left(\frac{4.17 - 1}{1.497}\right)} = 0.903$$

$K_{H\alpha} := 0.903$ $\frac{\varepsilon_\gamma}{\varepsilon_\alpha \cdot Z_\varepsilon^{\ 2}} = 4.17$

Face load factor for surface stress

Gear position related to the bearings	Gear accuracy grade	Surface hardened flanks	At least one gear is not hardened
Overhung	9 or 10	$1+\psi_d$	$1+0.5\ \psi_d$
Overhung	7 or 8	$1+0.7\psi_d$	$1+0.35\psi_d$
Between, non-symmetrical	9 or 10		
Overhung	5 or 6	$1+0.5\psi_d$	$1+0.25\psi_d$
Between, non-symmetrical	7 or 8		
Between, symmetrical	9 or 10		
Between, non-symmetrical	5 or 6	$1+0.3\psi_d$	$1+0.15\psi_d$
Between, symmetrical	7 or 8		
Between, symmetrical	5 or 6	$1+0.2\psi_d$	$1+0.1\psi_d$

According to the position of the gear on the shaft, symmetrically between the bearings, the face load factor is determined as follows:

$$K_{H\beta} := 1 + 0.3 \cdot \psi_d = 1.39$$

Lubricant factor for contact stress

$\sigma_{H.limb} < 590\ MPa$, it results that $C_{ZL} := 0.83$

Symbol	Kinematic viscosity at 50°C ν_{50} [cSt]	Viscosity index VI	Freezing point [°C]	Fire point [°C]
TIN 25 EP	21-26	60	-25	195
TIN 42 EP	37-45	60	-25	210
TIN 55 EP	50-57,5	60	-20	220
TIN 82 EP	82-90	60	-20	230
TIN 125 EP	130-140	60	-15	235
TIN 200 EP	200-220	70	-10	240
TIN 300 EP	230-300	70	0	255

The kinematic viscosity of the lubricant, according to the table, is
$\nu_{50} := 55$ cSt

$$Z_L := C_{ZL} + \frac{4 \cdot \left(1 - C_{ZL}\right)}{\left(1.2 + \frac{80}{\nu_{50}}\right)^2} = 0.927$$

Velocity factor for contact stress

$\sigma_{H.limb} = 590\ MPa$ $\sigma_{H.limb} < 850\ MPa$

It results that $C_{ZV} := 0.85$

$$Z_V = C_{ZV} + \frac{2 \cdot \left(1 - C_{ZV}\right)}{\sqrt{0.8 + \frac{32}{v_{td}}}}$$

$$v_{td} = 2.283\ \frac{m}{s}$$

$$0.85+2\cdot\left(\frac{1-0.85}{\sqrt{0.8+\frac{32}{2.283}}}\right)=0.928$$

$Z_V := 0.928$

Roughness factor for contact stress

$\sigma_{H.limb} < 850\ Mpa$ so $C_{ZR} := 0.15$

Flank roughness $R_a := 3.2$ μm

$R_z := 3 \cdot R_a = 9.6$ μm

$$R_{z100} := R_z \cdot \left(\frac{100\ mm}{a_w}\right)^{\frac{1}{3}} = 6.811$$

$$Z_R := \left(\frac{3}{R_{z100}}\right)^{C_{ZR}} = 0.884$$

Hardness ratio factor for contact stress

Both gears are surface hardened and finished $Z_W := 1$

Size factor for contact stress

$Z_X := 1$

Life factor for contact stress

$Z_N := 1$

Contact stress

$$\sigma_H := Z_H \cdot Z_E \cdot Z_\varepsilon \cdot Z_\beta \cdot \sqrt{\frac{F_{t1}}{d_1 \cdot b_w} \cdot K_A \cdot K_v \cdot K_{H\beta} \cdot K_{H\alpha} \cdot \frac{(1+i_{GB})}{i_{GB}}} = 374\ MPa$$

$\sigma_{Hlim} := \sigma_{H.limb} \cdot Z_L \cdot Z_V \cdot Z_R \cdot Z_W \cdot Z_X \cdot Z_N = 449\ MPa$

$S_H := \dfrac{\sigma_{Hlim}}{\sigma_H} = 1.201$ $\qquad S_H > 1.15$

The gears will perform well against pitting.

Calculation of tooth root cycling bending

For a safe operation, the following condition for the safety factor must be satisfied:

$S_F > S_{FP} = 1.25$

Basic allowable bending stress

$\sigma_{0lim} = 244\ MPa$

Face load distribution factor for root stress

$\dfrac{b_w}{h} = 15.739\ mm$ $\qquad \dfrac{b_w}{h} > 12$ $\qquad$, so $\qquad e := 1$

$K_{F\beta} := \left(K_{H\beta}\right)^e = 1.39$

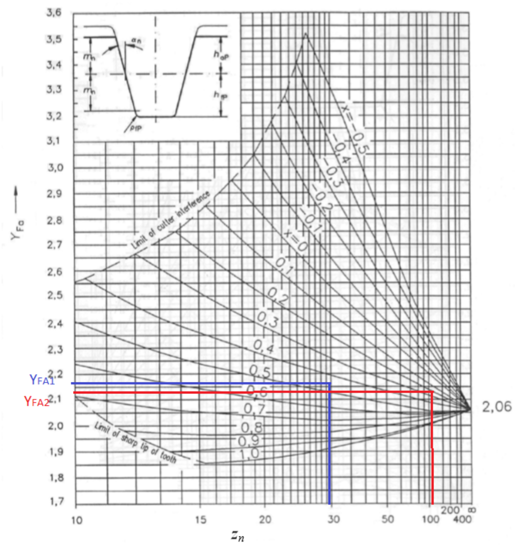
Transverse load factor for root stress

$K_{F\alpha} := K_{H\alpha} = 0.903$

Form factor

Thr form factor is determined using the table xx, taking into considerations the two values for the number of gear teeth and the shifting coefficients.

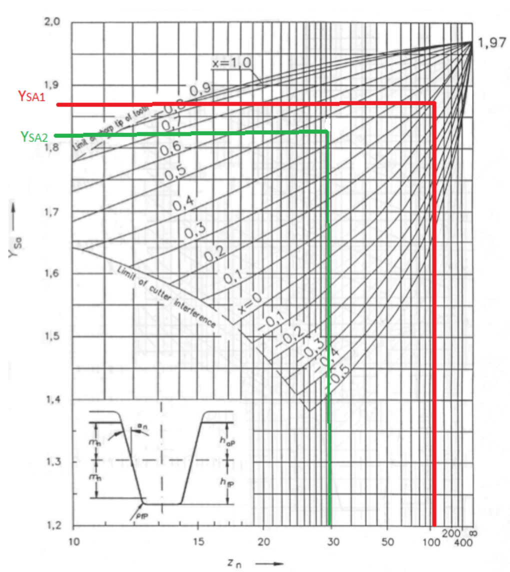
$z_{e\_1} = 28.85$ $\qquad x_{n1} = 0.46$ $\qquad z_{e\_2} = 103.193$ $\qquad x_{n2} = 0.19$



$$Y_{Fa1} := 2.17$$

$$Y_{Fa2} := 2.14$$

Stress correction factor



$$Y_{Sa1} := 1.87 \qquad Y_{Sa2} := 1.83$$

Contact ratio factor

$$\varepsilon_{\alpha n} := \frac{\varepsilon_\alpha}{\cos\left(\beta_b\right)^2} = 1.591$$

$$Y_\varepsilon := 0.25 + \frac{0.75}{\varepsilon_{\alpha n}} = 0.721 \qquad Y_\varepsilon < 0.75$$

Helix angle factor

$Y_\beta := 1 - \varepsilon_\beta \cdot \dfrac{\beta}{120} = 0.994$ $\qquad$ $Y_{\beta min} := 1 - 0.25 \cdot \varepsilon_\beta = 0.332$

$Y_\beta > Y_{\beta min}$ *The condition is fulfilled*

Life factor for tooth-root stress

Number of load cycles is higher than 10^7 $\quad N_1 > 10^7 \quad$ As a result:

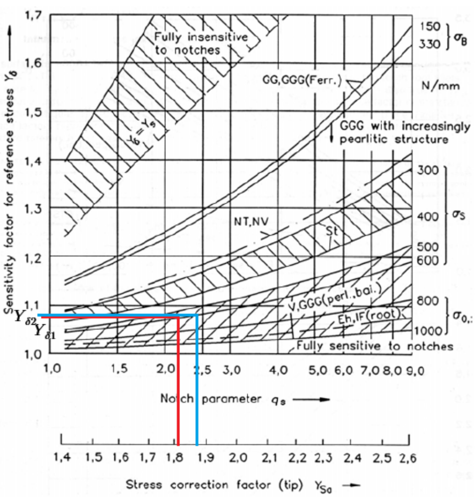
$Y_N := 1$

Roughness factor for root stress

$R_z = 9.6 \qquad 1 < R_z < 40\ \mu m$

$Y_R := 1.49 - 0.471 \cdot (R_z + 1)^{0.1} = 0.894$

Sensitivity factor for endurance limit



St – steel
V - Through hardening steel - through hardened
GG - grey cast iron
GGG (perl, bai, ferr) - nodular cast iron (perlitic, bainitic, ferritic structure)
GTS (perl) - black malleable cast iron (perlitic structure)
Eh - case-hardening steel, case hardened
IF - steel & GGG, flame or induction hardened
NT - nitriding steel, nitrided
NV (nitr) - Through hardening &case-hardening steel - nitrided
NV (nitrcar) - Through hardening &case-hardening steel - nitrocraburised

$Y_\delta := 1.07$

Size factor

$Y_X := 1$ since the module is under 5

Tooth root stress

$m_n := 4\ mm$

$$\sigma_{F1} := \frac{F_{t1} \cdot K_A \cdot K_v \cdot K_{F\alpha} \cdot K_{F\beta}}{m_n \cdot b_w} \cdot Y_{Fa1} \cdot Y_{Sa1} \cdot Y_{\varepsilon} \cdot Y_{\beta} = 75.999\ MPa$$

$$\sigma_{F2} := \frac{F_{t2} \cdot K_A \cdot K_v \cdot K_{F\alpha} \cdot K_{F\beta}}{m_n \cdot b_w} \cdot Y_{Fa2} \cdot Y_{Sa2} \cdot Y_{\varepsilon} \cdot Y_{\beta} = 73.346\ MPa$$

$$\sigma_{Flim} := \sigma_{0lim} \cdot Y_{\delta} \cdot Y_R \cdot Y_X \cdot Y_N = 233.296\ MPa$$

$$S_{F1} := \frac{\sigma_{Flim}}{\sigma_{F1}} = 3.07 \qquad S_{F2} := \frac{\sigma_{Flim}}{\sigma_{F2}} = 3.181$$

$$S_{F1} > S_{FP} = 1.25 \qquad S_{F2} > S_{FP}$$

The pinion and wheel bending is within normal parameters.

Shaft dimensioning

$\tau_{at} := 18\ MPa$ for shafts made of OLC material

$\tau_{at1} := \tau_{at}$ $\tau_{at2} := \tau_{at}$ since both shafts are made from the same material

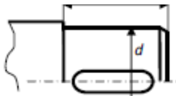
Diameters of the shafts

Minimum diameters

$$d_{min1} := \sqrt[3]{\frac{16 \cdot M_{t1} \cdot K_A}{\pi \cdot \tau_{at1}}} = 59.067\ mm \qquad d_{min2} := \sqrt[3]{\frac{16 \cdot M_{t2} \cdot K_A}{\pi \cdot \tau_{at2}}} = 75.598\ mm$$

d	Tolerances	Length, ℓ		d	Tolerances	Length, ℓ	
10	$j6\begin{pmatrix}+0,007\\-0,002\end{pmatrix}$	23	20	50	$k6\begin{pmatrix}+0,018\\+0,002\end{pmatrix}$	110	82
11	$j6\begin{pmatrix}+0,008\\-0,003\end{pmatrix}$			55	$m6\begin{pmatrix}+0,018\\+0,002\end{pmatrix}$		
12		30	25	56			
14				60		140	105
16		40	28	63			
18				65			
19	$j6\begin{pmatrix}+0,009\\-0,004\end{pmatrix}$			70			
20		50	36	71			
22				75	$m6\begin{pmatrix}+0,018\\+0,002\end{pmatrix}$	170	130
24				80			
25		60	42	85			
28				90			
30		80	58	95			
32	$k6\begin{pmatrix}+0,018\\+0,002\end{pmatrix}$			100	$m6\begin{pmatrix}+0,018\\+0,002\end{pmatrix}$	210	165
35				110			
38				120			
40		110	82	125	$m6\begin{pmatrix}+0,018\\+0,002\end{pmatrix}$	250	200
42	$k6\begin{pmatrix}+0,018\\+0,002\end{pmatrix}$			130			
45				140			
48				150			

d, [mm]	M_t, [N·m]	d, [mm]	M_t, [N·m]
10	1.85	50	515
11	2.56	55	730
12	3.55	56	775
14	6.00	60	975
16	9.75	63	1150
18	14.5	65	1280
19	17.5	70	1700
20	21.2	71	1800
22	29	75	2120
24	40	80	2650
25	46.2	86	3350
28	63	90	4120
30	87.5	95	4870
32	109	100	5800
35	150	110	8250
38	200	120	11200
40	236	125	14500
42	280	130	19000
45	355	140	24300
48	450	150	



Diameters of the shafts standardized according to the table

$d_{s1} := 60\ mm$ $d_{s2} := 80\ mm$

Calculation of the parallel key

PARALLEL KEY - Standard

d		Key				Keyway									
						b						t_1 (in the shaft)		t_2 (in the hub)	
		b	*h*	Length		nominal	Clearance (sliding free)		Normal fit		Pressed fit				
							on shaft N9	on hub D10	on shaft N9	on hub D10	on shaft & hub, P9	Nominal	Tolerances	Nominal	Tolerances
from	up to			from	up to		Tolerances								
22	30	8	7	18	90	8	+0.036 0	+0.098 +0.040	0 -0.036	±0.0180	-0.015 -0.051	4.0		3.3	
30	38	10	8	22	110	10						5.0		3.3	
38	44	12	8	28	140	12	+0.043 0	+0.120 +0.060	0 -0.043	±0.0215	-0.018 -0.061	5.0		3.3	
44	50	14	9	36	160	14						5.5		3.8	
50	58	16	10	45	180	16						6.0		4.3	
58	65	18	11	50	200	18						7	+0.2 0	4.4	+0.2 0
65	75	20	12	56	220	20	+0.052	+0.149	0	±0.0260	-0.022	7.5		4.9	
75	85	22	14	63	250	22						9		5.4	

Keyway calculation of the input shaft end shaft.

According to the table, the standard values for height and width of the key are

$b := 18\ mm$ $h := 11\ mm$

To verify that the key will resist the torque, we will take into account the contact and shear stresses when computing the length.

Contact stress

$\sigma_{ak} := 100\ MPa$ allowable limit

$d_0 := d_{s1}$ diameter of the input shaft

$$l_{effmin} := \frac{4 \cdot M_{t1} \cdot K_A}{h \cdot \sigma_{ak} \cdot d_0} = 44.142\ mm$$

Shear stress

$\tau_{af} := 65\ MPa$ allowable limit

$$l_{effmin} := \frac{2 \cdot M_{t1} \cdot K_A}{b \cdot \tau_{af} \cdot d_0} = 20.751\ mm$$

The highest effective length between the two stresses is 44.142*, which we will round to 44mm, as a result:*

$l_{effmin} := 44\ mm$

$l_{min} := l_{effmin} + b = 62\ mm$

$l_{min} := 90\ mm$ standard value

Keyway calculation of the output shaft end shaft

$d_{s2} = 80\ mm$

According to the table:

$b_{s2} := 22\ mm$ $h_{s2} := 14\ mm$

Contact stress

$\sigma_{ak} := 110\ MPa$

$d_0 := d_{s2}$

$$l_{effmin2} := \frac{4 \cdot M_{t2} \cdot K_A}{h_{s2} \cdot \sigma_{ak} \cdot d_0} = 49.577\ mm$$

Shear stress

$\tau_{af} := 70\ MPa$

$$l_{effmin2} := \frac{2 \cdot M_{t2} \cdot K_A}{b_{s2} \cdot \tau_{af} \cdot d_0} = 24.789\ mm$$

The highest effective length between the 2 calculated is chosen to be the final one, rounded to an integer:

$l_{effmin2} := 50\ mm$

$l_{min} := l_{effmin2} + b_{s2} = 72\ mm$

$l_{min} := 90\ mm$

Keyway of the driven gear

$d_{dg} := 120\ mm$

According to the table:

$b_3 := 32\ mm$ $h_3 := 19\ mm$

Contact stress

$\sigma_{ak} := 100\ MPa$

$d_0 := d_{dg}$

$$l_{effmin3} := \frac{4 \cdot M_{t1} \cdot K_A}{h_3 \cdot \sigma_{ak} \cdot d_0} = 12.778\ mm$$

Shear stress

$\tau_{af} := 60\ MPa$

$$l_{effmin3} := \frac{2 \cdot M_{t1} \cdot K_A}{b_3 \cdot \tau_{af} \cdot d_0} = 6.322\ mm$$

The highest effective length between the two computations is 12.778mm, we will round it to 13mm

$l_{effmin2} := 13\ mm$

$l_{min} := l_{effmin2} + b_3 = 45\ mm$

$l_{min} := 120\ mm$

Determination of forces acting on the shaft

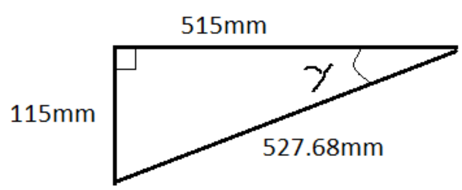
Peripheral force $F_u := 1913\ N$

Belt load min: $F_{0min} := 1.5\ F_u = 2870\ N$

Belt load max: $F_{0max} := 2 \cdot F_u = 3826\ N$

Shaft radial load: $S_a := 2 \cdot F_{0max} = 7652\ N$

Determination of γ angle



$$sin(\gamma) := \frac{115}{527.68} = 0.218$$

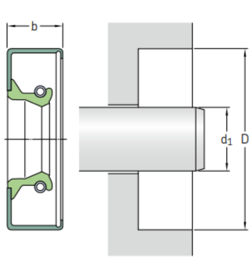
$$\gamma := \text{asin}(sin(\gamma)) = 12.588\ deg$$

Belt radial load $S_a \cdot sin(\gamma) = 1668\ N$

Belt tangential load $S_a \cdot \cos(\gamma) = 7468\ N$

Lip seal $F_{t1} = 9158.19\ N$

Radial shaft seals | Seals for general industrial applications | CRW1, CRWA1, CRWH1 and CRWHA1
d_1 **70–280** mm



Dimensions Shaft d_1	Bore D	Nominal seal width b	Design and lip material	Designation
mm			–	–
70 cont.	90	8	CRW1 R	**70×90×8 CRW1 R**
	90	10	CRW1 V	**70×90×10 CRW1 V**
	90	10	CRWHA1 P	**70×90×10 CRWHA1 P**
	92	11	CRWH1 R	**70×92×11 CRWH1 R**
	92	11	CRWH1 V	**70×92×11 CRWH1 V**
	95	10	CRW1 R	**70×95×10 CRW1 R**
	100	10	CRW1 R	**70×100×10 CRW1 R**
	105	10	CRW1 R	**70×105×10 CRW1 R**

$M_{t1} = 560.266 \ (N \cdot m)$

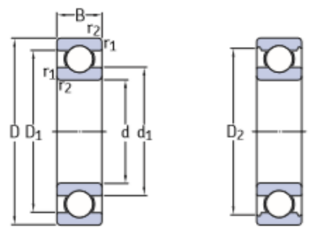
Width of the lip seal is $\quad b_{seal} := 10 \ mm$

Initial bearing selection

1.1

1.1 Single row deep groove ball bearings

d **75 – 80** mm



$F_{r1} = 3450.895 \ N$

$F_{a1} = 2453.93 \ N$

$d_1 = 0.121 \ m$

mm			kN		kN	r/min		kg	–	
75	95	10	12,5	10,8	0,585	–	4 000	0,15	▸ 61815-2RS1	–
	95	10	12,5	10,8	0,585	14 000	7 000	0,15	▸ 61815-2RZ	–
	95	10	12,5	10,8	0,585	14 000	8 500	0,15	▸ 61815	–
	105	16	24,2	19,3	0,965	13 000	8 000	0,36	▸ 61915	–
	105	16	24,2	22,4	0,965	–	3 600	0,37	61915-2RS1	–
	105	16	24,2	22,4	0,965	13 000	6 300	0,37	61915-2RZ	–
	115	13	30,2	27	1,14	12 000	7 500	0,46	▸ 16015	–
	115	20	41,6	33,5	1,43	12 000	10 000	0,74	6015 M	–
	115	20	41,6	33,5	1,43	12 000	7 500	0,65	▸ 6015	–
	115	20	41,6	33,5	1,43	–	3 400	0,67	▸ 6015-2RS1	6015-RS1
	115	20	41,6	33,5	1,43	12 000	6 000	0,67	6015-2RZ	6015-RZ
	115	20	41,6	33,5	1,43	12 000	6 000	0,68	▸ 6015-2Z	6015-Z
	130	25	68,9	49	2,04	10 000	9 500	1,4	6215 M	–
	130	25	68,9	49	2,04	10 000	6 700	1,2	▸ 6215	–
	130	25	68,9	49	2,04	–	3 200	1,2	▸ 6215-2RS1	6215-RS1
	130	25	68,9	49	2,04	10 000	5 300	1,25	▸ 6215-2Z	6215-Z
	160	37	119	76,5	3	9 000	5 600	3,05	▸ 6315	–
	160	37	119	76,5	3	9 000	5 600	3,7	6315 M	–
	160	37	119	76,5	3	–	2 800	3,15	▸ 6315-2RS1	6315-RS1
	160	37	119	76,5	3	9 000	4 500	3,15	▸ 6315-2Z	6315-Z
	190	45	153	114	4,15	8 000	5 000	5,8	6415	–
80	100	10	12,7	11,2	0,61	–	3 600	0,16	▸ 61816-2RS1	–
	100	10	12,7	11,2	0,61	13 000	8 000	0,15	▸ 61816	–
	110	16	25,1	20,4	1,02	–	3 400	0,4	▸ 61916-2RS1	–
	110	16	25,1	20,4	1,02	12 000	6 000	0,4	▸ 61916-2RZ	–
	110	16	25,1	20,4	1,02	12 000	7 500	0,38	▸ 61916	–
	125	14	35,1	31,5	1,32	11 000	7 000	0,61	▸ 16016	–

Shaft dimensions

$B := 16 \ mm \qquad$ width of the bearing

$b_{pulley} := 80 \ mm$

$$a := \frac{b_{pulley}}{2} + 10 \ mm + b_{seal} + \frac{B}{2} = 68 \ mm$$

$b_{...} = 157.465 \ mm \qquad b_{...} := 158 \ mm$

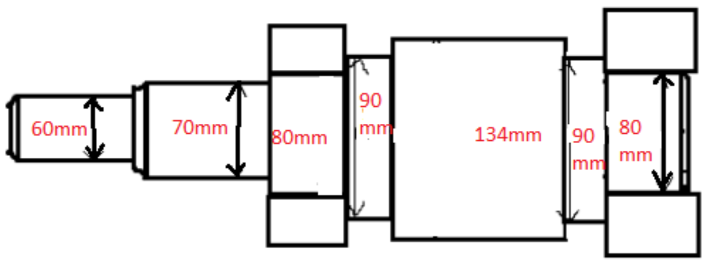
$$b := \frac{B}{2} + 10\ mm + \frac{b_w}{2} = 97\ mm$$

$$c := \frac{b_w}{2} + 10\ mm + \frac{B}{2} = 97\ mm$$

$$b_w = 0.158\ m$$

Selection of bearings

Input shaft sketch

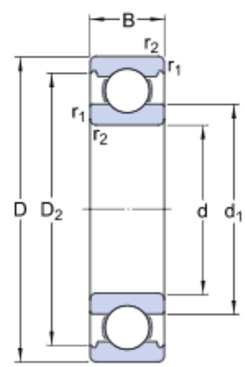


CALCULATION DATA

C	25.1 kN	Basic dynamic load rating
C_0	20.4 kN	Basic static load rating
P_u	1.02 kN	Fatigue load limit
	12 000 r/min	Reference speed
	7 500 r/min	Limiting speed
k_r	0.02	Minimum load factor
f_0	14	Calculation factor

DIMENSIONS

d	80 mm	Bore diameter
D	110 mm	Outside diameter
B	16 mm	Width
d_1	≈ 89.8 mm	Shoulder diameter
D_2	≈ 102.2 mm	Recess diameter
$r_{1,2}$	min. 1 mm	Chamfer dimension



$$F_{a1} = 2454\ N$$

$$n_1 = 360\ rpm$$

Forces on the bearings

$$H_A := 235\ N \qquad V_A := 5507\ N$$

$$H_B := 1547\ N \qquad V_B := -7197\ N$$

$$F_{rA} := \sqrt{{H_A}^2 + {V_A}^2} = 5512\ N$$

$$F_{rB} := \sqrt{{H_B}^2 + {V_B}^2} = 7361\ N$$

Ball bearing 61915

$C := 25.1\ kN$

$C_0 := 20.4\ kN$

$f_0 := 14$

$$\frac{f_0 \cdot F_{a1}}{C_0} = 1.684$$

$f_0 F_a/C_0$	e	X	Y
0,172	0,19	0,56	2,3
0,345	0,22	0,56	1,99
0,689	0,26	0,56	1,71
1,03	0,28	0,56	1,55
1,38	0,3	0,56	1,45
2,07	0,34	0,56	1,31
3,45	0,38	0,56	1,15
5,17	0,42	0,56	1,04
6,89	0,44	0,56	1

Calculation factors

Using the linear interpolation with the values from the tables, the calculation parameters are:

$X := 0.56$

$e := 0.317$

$Y := 1.388$

$\frac{F_{a1}}{F_{rA}} = 0.445$ $\qquad \frac{F_{a1}}{F_{rA}} > e$ $\qquad \frac{F_{a1}}{F_{rB}} = 0.333$ $\qquad \frac{F_{a1}}{F_{rB}} > e$

This results that X and Y keep their values

$V := 1$ $\qquad n_1 = 360\ rpm$

$P_A := X \cdot V \cdot F_{rA} + Y \cdot F_{a1} = 6492.781\ N$

$P_B := X \cdot V \cdot F_{rB} + Y \cdot F_{a1} = 7528.431\ N$

$L_A := \left(\frac{C}{P_A}\right)^3 = 57.8$ mil. rotations $\qquad L_B := \left(\frac{C}{P_B}\right)^3 = 37.1$ mil. rotations

$$L_{hA} := \frac{L_A \cdot 10^6}{n_1} = 425.7\ hr$$

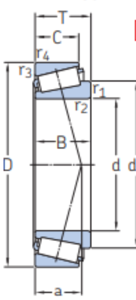
<u>Our result for the ball bearings falls short for the 15000 hours target, due to their life cycle being too low, we will try the tapered roller bearings next.</u>

$$L_{hB} := \frac{L_B \cdot 10^6}{n_1} = 273.1\ hr$$

Recalculation of the shaft dimensions

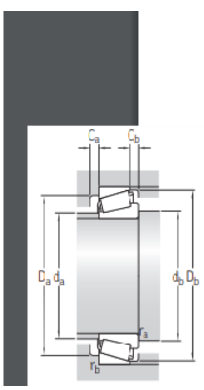
8.1 Metric single row tapered roller bearings

d **80 – 85** mm



Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation	Dimension series to ISO 355 (ABMA)
d	D	T	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	–
80	110	20	89,7	125	14	4 500	5 600	0,54	32916	2BC
	125	29	168	216	24,5	4 000	5 000	1,3	▸ 32016 X	3CC
	125	36	207	285	32	4 000	5 000	1,65	▸ 33016	2CE
	130	35	216	275	31	4 000	4 800	1,75	JM 515649/610	M 515600
	130	37	221	280	31	4 000	4 800	1,85	▸ 33116	3DE
	140	28,25	184	183	21,2	3 800	4 800	1,6	▸ 30216	3EB
	140	35,25	228	245	28,5	3 800	4 500	2,05	▸ 32216	3EC
	140	46	308	375	41,5	3 400	4 500	2,9	▸ 33216	3EE
	160	45	280	315	35,5	3 000	4 000	4	T7FC 080	7FC
	170	42,5	276	265	30,5	3 000	4 000	4,05	31316	7GB
	170	42,5	333	320	36,5	3 200	4 000	4,15	▸ 30316	2GB
	170	61,5	440	520	57	3 200	3 800	6,65	32316 B	5GD
	170	61,5	404	500	56	3 200	4 000	6,2	▸ **32316**	2GD

Dimensions and parameters of 33016 tapered bearing



Dimensions							Abutment and fillet dimensions									Calculation factors		
d	d_1 ≈	B	C	$r_{1,2}$ min.	$r_{3,4}$ min.	a	d_a max.	d_b min.	D_a min.	D_a max.	D_b min.	C_a min.	C_b min.	r_a max.	r_b max.	e	Y	Y_0
mm							mm									–		
80	94,8	20	16	1	1	19	86	88,5	102	102	106	4	4	1	1	0,35	1,7	0,9
	103	29	22	1,5	1,5	26	90	90	112	116	120	6	7	1,5	1,5	0,43	1,4	0,8
	102	36	29,5	1,5	1,5	25	90	89,5	112	116	119	6	6,5	1,5	1,5	0,28	2,1	1,1
	104	34	28,5	3	2,5	28	90	93	114	119	124	6	6,5	3	2,5	0,4	1,5	0,8
	105	37	29	2	1,5	30	89	91	114	121	126	6	8	2	1,5	0,43	1,4	0,8
	105	26	22	2,5	2	27	92	92	124	130	132	4	6	2,5	2	0,43	1,4	0,8
	106	33	28	2,5	2	30	91	92	122	130	134	5	7	2,5	2	0,43	1,4	0,8
	110	46	35	2,5	2	34	90	92	119	130	135	7	11	2,5	2	0,43	1,4	0,8
	125	41	31	3	3	53	94	93,5	121	148	152	5	14	3	3	0,88	0,68	0,4

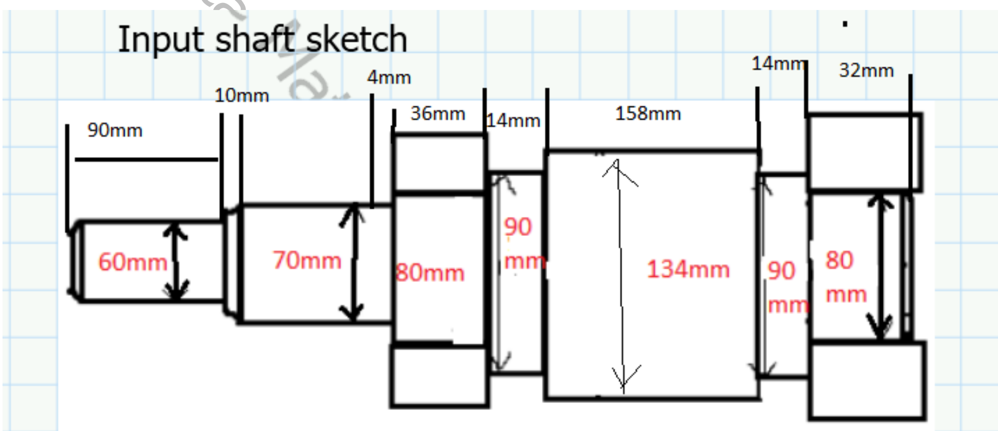
Calculation of the bearing life: Tapered roller bearing 33016

$B := 36\ mm$

$a := \frac{b_{pulley}}{2} + 10\ mm + b_{seal} + 4\ mm + \frac{B}{2} = 82\ mm$

$b := \frac{B}{2} + 14\ mm + \frac{b_w}{2} = 111\ mm$

$c := \frac{b_w}{2} + 14\ mm + \frac{B}{2} = 111\ mm$

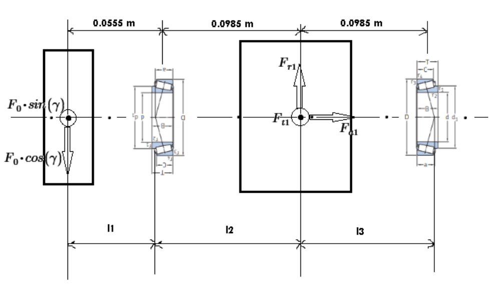


$b_{pulley} = 80\ mm$

"X" arrangement

$b_{seal} = 10\ mm$

Fig 24. Forces acting in an "X" arrangement of tapered bearings



$Y := 2.1 \qquad Y_A := Y \qquad Y_B := Y$

$C := 207\ kN \qquad C_0 := 285\ kN$

$X := 0.56 \qquad e := 0.28$

$d_{taper} := 80\ mm$

$D_{taper} := 140\ mm$

$T_{taper} := 39\ mm$

$a_{taper} := 25\ mm$

$a = 82\ mm \qquad b = 111\ mm \qquad c = 111\ mm$

$$l_1 := \frac{b_{pulley}}{2} + \left(a - \frac{b_{pulley}}{2} - \frac{B}{2}\right) + a_{taper} = 89\ mm$$

$$l_2 := \left(T_{taper} - a_{taper}\right) + \left(b - \frac{B}{2}\right) = 107\ mm \qquad l_3 := \left(c - \frac{B}{2}\right) + \left(T_{taper} - a_{taper}\right) = 107\ mm$$

$$H_A := 55\ N \qquad V_A := 5995\ N$$

$$H_B := 1727\ N \qquad V_B := -7685\ N$$

$$F_{rA} := \sqrt{H_A^{\ 2} + V_A^{\ 2}} = 5995\ N$$

$$F_{rB} := \sqrt{H_B^{\ 2} + V_B^{\ 2}} = 7877\ N$$

$$F_{asA} := 0.5 \cdot \frac{F_{rA}}{Y_A} = 1427\ N \qquad F_{asB} := 0.5 \cdot \frac{F_{rB}}{Y_B} = 1875\ N$$

$$F_{a1} = 2454\ N$$

$$R_a := F_{a1} + F_{asA} - F_{asB} = 2006\ N \qquad R_a > 0$$

<u>Direction of the axial resultant is from A to B, so bearing B supports the axial load.</u>

$$F_{aA} := F_{asA} = 1427\ N$$

$$F_{aB} := F_{asB} + F_{a1} = 4329\ N$$

$$\frac{F_{aA}}{F_{rA}} = 0.238 \qquad \frac{F_{aA}}{F_{rA}} < e$$ it results that X=1 and Y=0

$$\frac{F_{aB}}{F_{rB}} = 0.55 \qquad \frac{F_{aB}}{F_{rB}} > e$$ so X and Y >0

$$P_A := 1 \cdot F_{rA} + 0 \cdot F_{aA} = 5995\ N$$

$$P_B := X \cdot F_{rB} + Y \cdot F_{aB} = 13503\ N$$

$$L_A := \left(\frac{C}{P_A}\right)^{\frac{10}{3}} = 134032$$ mil rotations

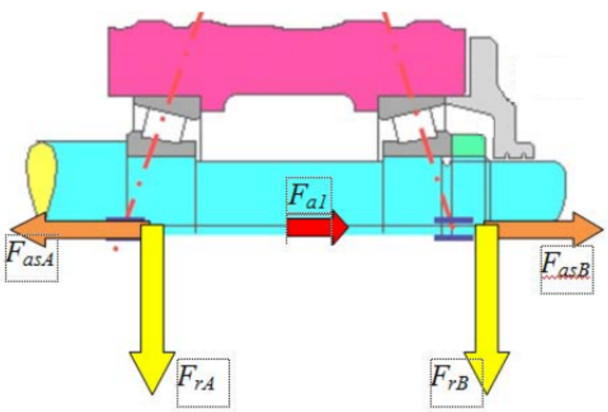
$$L_B := \left(\frac{C}{P_B}\right)^{\frac{10}{3}} = 8951$$ mil rotations

$L_{hA} := \frac{L_A \cdot 10^6}{n_1} = 987588\ hr$ $L_{hB} := \frac{L_B \cdot 10^6}{n_1} = 65951\ hr$

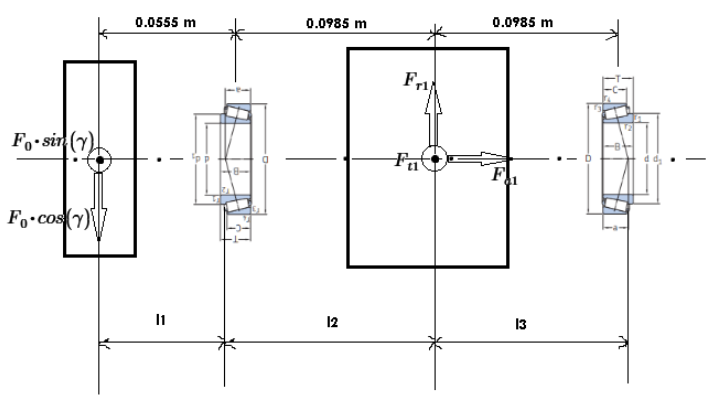
This time the results for the life cycle of the "X" arrangement are favorable, we will also compute the life cycle for the "O" arrangement in order to obtain the best design possible

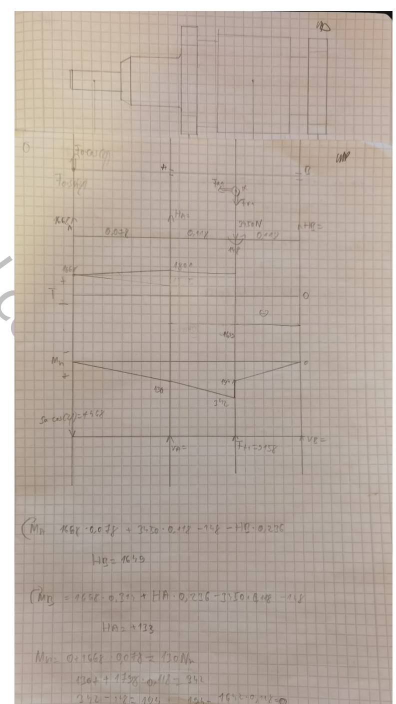
"O" arrangement

Forces acting in an "O" arrangement



Dimensions in the case of "O" arrangement





$$l_1 := \left(a - \frac{B}{2}\right) + T_{taper} - a_{taper} = 78\ mm$$

$$l_2 := a_{taper} + \left(b - \frac{B}{2}\right) = 118\ mm$$

$$l_3 := \left(c - \frac{B}{2}\right) + a_{taper} = 118\ mm$$

$H_A := 133\ N$ $\qquad$ $V_A := 5357\ N$

$H_B := 1649\ N$ $\qquad$ $V_B := -7047\ N$

$$F_{rA} := \sqrt{H_A^{\ 2} + V_A^{\ 2}} = 5359\ N$$

$$F_{rB} := \sqrt{H_B^{\ 2} + V_B^{\ 2}} = 7237\ N$$

$$F_{asA} := 0.5 \cdot \frac{F_{rA}}{Y_A} = 1276\ N$$

$$F_{asB} := 0.5 \cdot \frac{F_{rB}}{Y_B} = 1723\ N$$

$R_a := F_{a1} + F_{asB} - F_{asA} = 2901\ N$ $\qquad$ $R_a > 0$

Direction of the axial forces is from A to B. The bearing B will support the thrust load.

$$F_{aA} := F_{asA} = 1276\ N$$

$$F_{aB} := F_{asB} + F_{a1} = 4177\ N$$

$\frac{F_{aA}}{F_{rA}} = 0.238$ $\qquad$ $\frac{F_{aA}}{F_{rA}} < e$ $\qquad$ it results that X=1 and Y=0

$\frac{F_{aB}}{F_{rB}} = 0.577$ $\qquad$ $\frac{F_{aB}}{F_{rB}} > e$ $\qquad$ so X and Y >0

$P_A := 1 \cdot F_{rA} + 0 \cdot F_{aA} = 5359\ N$ $\qquad$ $P_B := X \cdot F_{rB} + Y \cdot F_{aB} = 12825\ N$

$L_A := \left(\frac{C}{P_A}\right)^{\frac{10}{3}} = 194857$ mil. rotations

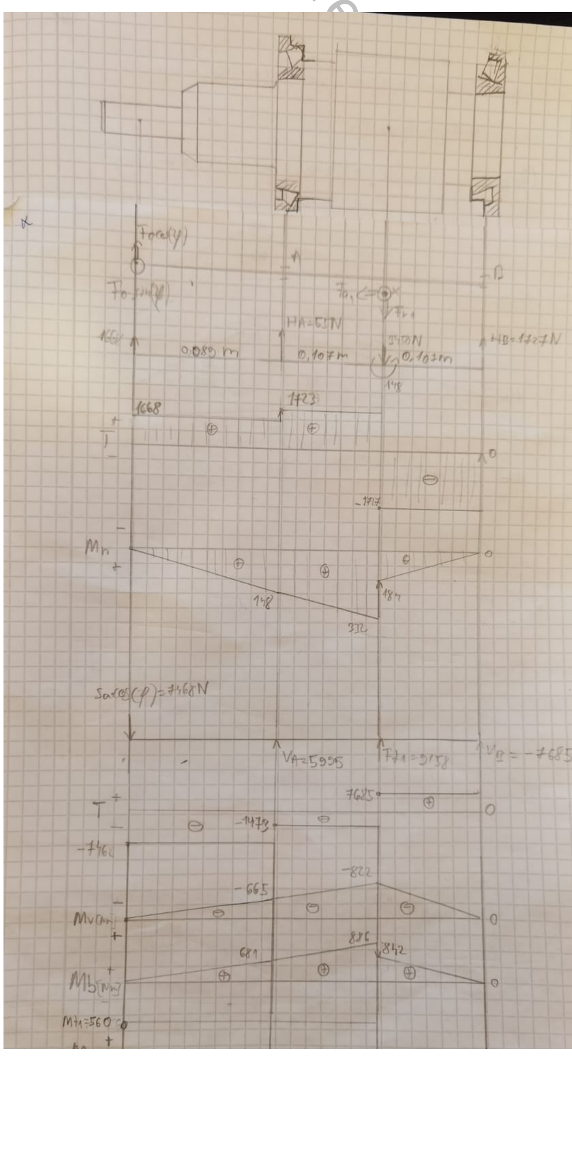
$$L_B := \left(\frac{C}{P_B}\right)^{\frac{10}{3}} = 10627 \qquad \text{mil. rotations}$$

$$L_{hA} := \frac{L_A \cdot 10^6}{n_1} = 1435763 \; hr$$

$$L_{hB} := \frac{L_B \cdot 10^6}{n_1} = 78300 \; hr$$

Both the "X" arrangement and the "O" arrangement surpass the 15000h limit which makes them good options, we will further continue and choose the "X" arrangement tapered bearings since the difference in life hours between the two arrangements is smaller in this case.

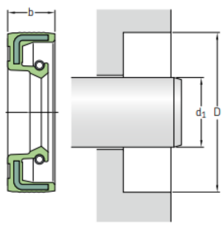
Final diagram of the input shaft



Selection of lip seal for the output shaft

Radial shaft seals | Seals for general industrial applications | HMS5 and HMSA10

d_1 **70–250** mm



[1] Designation to be followed by the design and material codes, indicating one of the four variants available for each dimension:

HMS5 RG	without auxiliary lip, nitrile rubber
HMS5 V	without auxiliary lip, fluoro rubber
HMSA10 RG	with auxiliary lip, nitrile rubber
HMSA10 V	with auxiliary lip, fluoro rubber
Example:	**6×16×5 HMSA10 RG**

[2] Design execution differs from the basic design and is indicated by a number, see also **page 102**.

Dimensions Shaft d_1	Bore D	Nominal seal width b	**Designation**[1]	**ISO / DIN**
mm			–	–
90	110	10	**90×110×10**	
	110	12	**90×110×12**	•
	115	12	**90×115×12**	

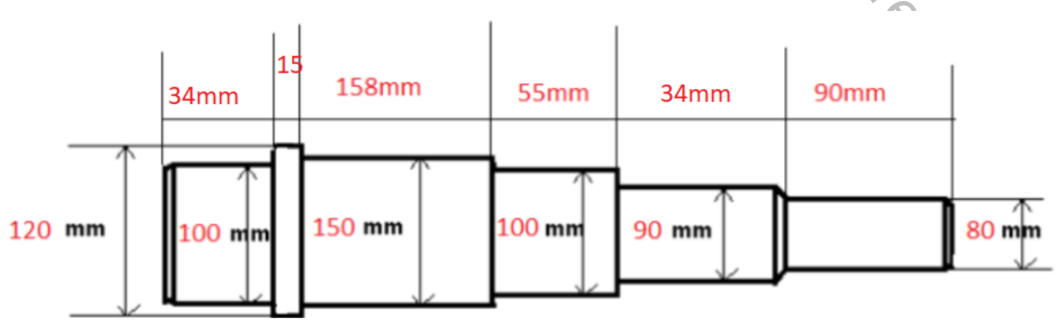
Width of the lip seal is

$$b_{seal} := 12\ mm$$

$d_{so} := 90\ mm$ diameter of the shoulder of the shaft where the lip seal is mounted

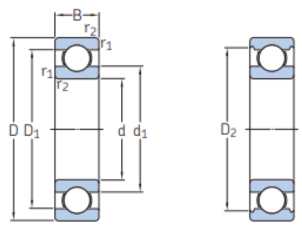
Selection of bearings for the output shaft

Output shaft sketch

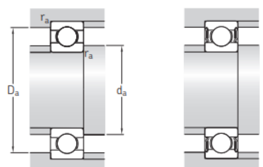


Selected ball bearing dimensions and parameters

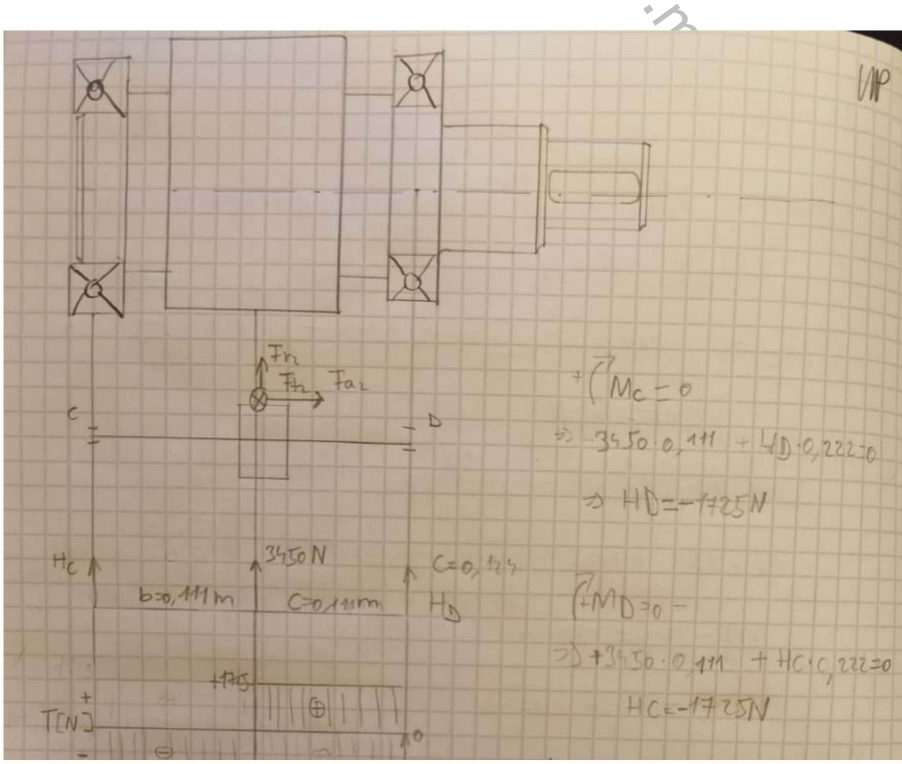
1.1 Single row deep groove ball bearings
d **90 – 100** mm

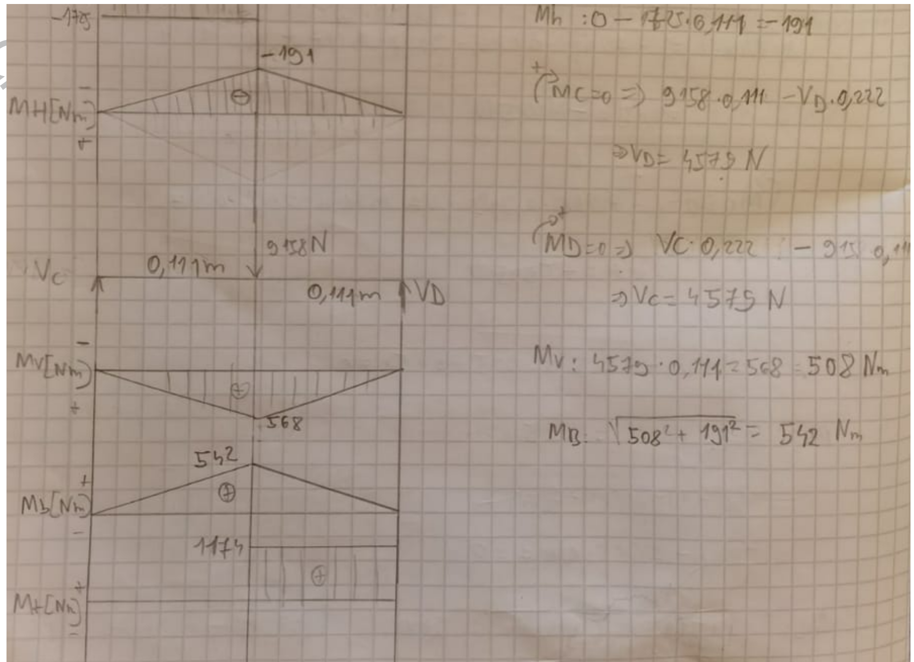


Principal dimensions			Basic load ratings		Fatigue load limit	Speed ratings		Mass	Designations	
			dynamic	static		Reference speed	Limiting speed[1)]		Bearing open or capped on both sides	capped on one side[1)]
d	D	B	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	
100	125	13	17,8	18,3	0,95	–	3 000	0,32	▸ **61820-2RS1**	–
	125	13	17,8	18,3	0,95	10 000	5 300	0,32	▸ **61820-2RZ**	–
	125	13	17,8	18,3	0,95	10 000	6 300	0,3	▸ **61820**	–
	140	20	42,3	41,5	1,63	9 500	6 000	0,83	**61920**	–
	150	16	46,2	44	1,7	9 500	5 600	0,94	▸ 16020	–
	150	24	63,7	54	2,04	9 500	7 500	1,45	6020 M	–
	150	24	63,7	54	2,04	9 500	5 600	1,25	▸ 6020	–
	150	24	63,7	54	2,04	–	2 600	1,3	▸ 6020-2RS1	6020-RS1
	150	24	63,7	54	2,04	9 500	4 500	1,3	▸ 6020-2Z	6020-Z
	180	34	127	93	3,35	7 500	4 800	3,2	▸ 6220	–
	180	34	127	93	3,35	7 500	7 000	3,8	6220 M	–
	180	34	127	93	3,35	–	2 400	3,3	▸ 6220-2RS1	6220-RS1



Dimensions						Abutment and fillet dimensions				Calculation factors	
d	d_1 ≈	d_2 ≈	D_1 ≈	D_2 ≈	$r_{1,2}$ min.	d_a min.	d_a max.	D_a max.	r_a max.	k_r	f_0
mm						mm				–	
100	108	–	–	120	1	105	107	120	1	0,015	13
	108	–	–	120	1	105	107	120	1	0,015	13
	108	–	–	120	1	105	–	120	1	0,015	13
	112	–	128	–	1,1	106	–	134	1	0,02	16
	116	–	134	–	1	105	–	145	1	0,02	17
	115	–	–	139	1,5	107	–	143	1,5	0,025	16
	115	–	–	139	1,5	107	–	143	1,5	0,025	16
	115	–	–	139	1,5	107	115	143	1,5	0,025	16
	115	–	–	139	1,5	107	115	143	1,5	0,025	16
	124	–	–	160	2,1	112	–	168	2	0,025	14
	124	–	–	160	2,1	112	–	168	2	0,025	14





$C := 127\ kN$ $\qquad C_0 := 93\ kN$ $\qquad f_0 := 14$

$H_C := -1725\ N$ $\qquad V_C := 4579\ N$

$H_D := -1725\ N$ $\qquad V_D := 4579\ N$

$$\frac{f_0 \cdot F_{a1}}{C_0} = 0.369$$

$f_0 F_a/C_0$	e	X	Y
0,172	0,19	0,56	2,3
0,345	0,22	0,56	1,99
0,689	0,26	0,56	1,71
1,03	0,28	0,56	1,55
1,38	0,3	0,56	1,45
2,07	0,34	0,56	1,31
3,45	0,38	0,56	1,15
5,17	0,42	0,56	1,04
6,89	0,44	0,56	1

$$F_{rC} := \sqrt{{H_C}^2 + {V_C}^2} = 4893\ N$$

$$F_{rD} := \sqrt{{H_D}^2 + {V_D}^2} = 4893\ N$$

After the interpolation, parameters are:

$X := 0.56$ $\qquad e := 0.222$ $\qquad Y := 1.97$

$$\frac{F_{a1}}{F_{rC}} = 0.502 \qquad \frac{F_{a1}}{F_{rC}} > e \qquad \frac{F_{a1}}{F_{rD}} = 0.502 \qquad \frac{F_{a1}}{F_{rD}} > e$$

X and Y remain the same.

$P_C := X \cdot V \cdot F_{rC} + Y \cdot F_{a1} = 7574\ N$

$P_D := X \cdot V \cdot F_{rD} + Y \cdot F_{a1} = 7574\ N$

$L_C := \left(\frac{C}{P_C}\right)^3 = 4713.745$ mil. rotations

$L_D := \left(\frac{C}{P_D}\right)^3 = 4713.745$ mil. rotations

$L_{hC} := \frac{L_C \cdot 10^6}{n_1} = 34732\ hr$ $\qquad$ $L_{hD} := \frac{L_D \cdot 10^6}{n_1} = 34732\ hr$

The life of the bearings for the output shaft are over the limit of 15000h, so the ball bearings are a good choice for the shaft.

Shaft strength verification

OLC 45 mechanical properties

Strength parameters of the material

$\sigma_r := 360\ MPa$ Yield stress (taken from Table 18)
$\sigma_c := 620\ MPa$ Ultimate tensile strength

Input shaft

Determination of $\beta_{K\sigma}$

For this parameter, it is considered the section of the shaft where is a stress concentration. In this case, it is the shoulder where the bearings are mounted.

$D := 80\ mm$ diameter of the shoulder of the shaft where the bearing is placed

$d := 70\ mm$ diameter of the shaft where the lip seal is placed

$t := D - d = 10\ mm$

$r := 4\ mm$ fillet of the shaft at the shoulder where is mounted the bearing

$$\frac{t}{r} = 2.5$$

$2 < \frac{t}{r} < 20$, it results that:

$$C_1 := 1.232 + 0.832 \cdot \sqrt{\frac{t}{r}} - 0.008 \cdot \frac{t}{r} = 2.528$$

$$C_2 := -3.813 + 0.968 \cdot \sqrt{\frac{t}{r}} - 0.260 \cdot \frac{t}{r} = -2.932$$

$$C_3 := 7.422 - 4.868 \cdot \sqrt{\frac{t}{r}} + 0.869 \cdot \frac{t}{r} = 1.898$$

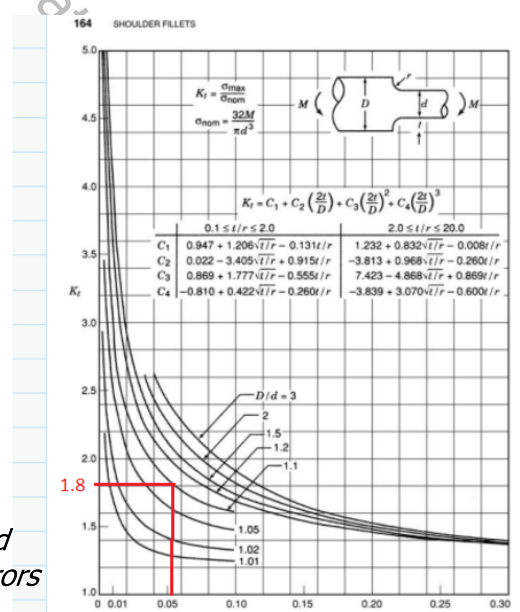
$$C_4 := -3.839 + 3.070 \cdot \sqrt{\frac{t}{r}} - 0.600 \cdot \frac{t}{r} = -0.485$$

$$\beta_{K\sigma} := C_1 + C_2 \cdot \left(\frac{2 \cdot t}{D}\right) + C_3 \cdot \left(\frac{2 \cdot t}{D}\right)^2 + C_4 \cdot \left(\frac{2 \cdot t}{D}\right)^3 = 1.905$$

__Graph determination of the bending parameter__

$$\frac{r}{d} = 0.057$$

$$\frac{D}{d} = 1.143$$



The value computed and that result from the graph are close in value,in this case, the selected value is the one calculated numerically, since errors can occur when selecting from the graph.

Determination of $\beta_{K\tau}$

The same section of the shaft is considered.

$D = 80\ mm$

$d = 70\ mm$

$t = 10\ mm$

$r = 4\ mm$

$$\frac{t}{r} = 2.5$$

$0.25 < \frac{t}{r} < 4$, it results that

$$C_1 := 0.905 + 0.783 \cdot \sqrt{\frac{t}{r}} - 0.075 \cdot \frac{t}{r} = 1.956$$

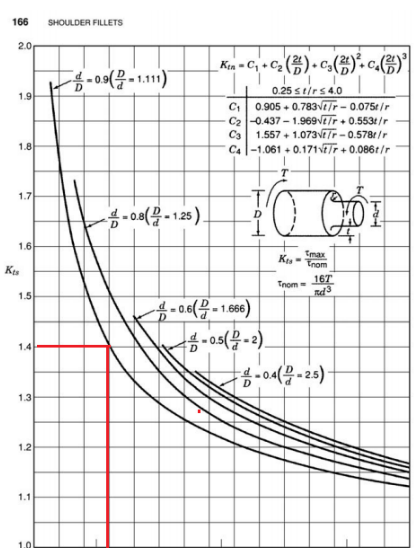
$$C_2 := -0.437 - 1.969 \cdot \sqrt{\frac{t}{r}} + 0.553 \cdot \frac{t}{r} = -2.168$$

$$C_3 := 1.557 + 1.073 \cdot \sqrt{\frac{t}{r}} - 0.578 \cdot \frac{t}{r} = 1.809$$

$$C_4 := -1.061 + 0.171 \cdot \sqrt{\frac{t}{r}} + 0.086 \cdot \frac{t}{r} = -0.576$$

$$\beta_{K\tau} := C_1 + C_2 \cdot \left(\frac{2 \cdot t}{D}\right) + C_3 \cdot \left(\frac{2 \cdot t}{D}\right)^2 + C_4 \cdot \left(\frac{2 \cdot t}{D}\right)^3 = 1.518$$

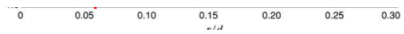
Graph determination of the torsion parameter



$$\frac{d}{D} = 0.875$$

$$\frac{r}{d} = 0.057$$

<u>The value from the graph is smaller than the one determined numerically. The selected value is once again the numerically computed one.</u>



Size factor

$d_{s1} := 60 \quad mm$ diameter of the input shaft

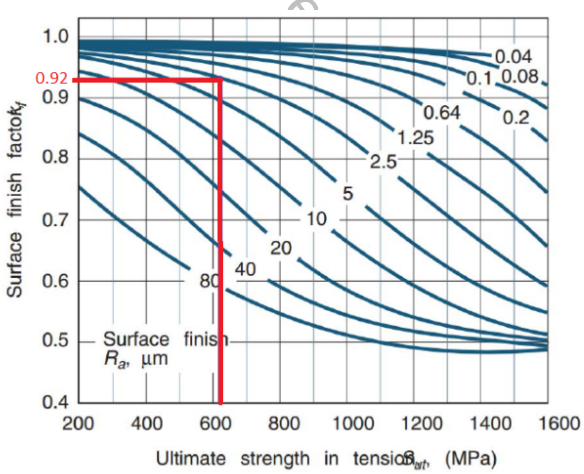
For a diameter of the shaft of 60mm, the size factor is calculated as follows:

$\varepsilon := 1.51 \cdot d_{s1}^{(-0.157)} = 0.794$

Surface finish factor

$\sigma_c = 620\ MPa$ Ultimate strength in tension

Surface finish factor



Considering a surface finish of 2.5 (typical for machined shafts), the surface finish factor has been determined to be:

$\gamma := 0.92$

Determination of the bending and torsion stresses

$\sigma_u := \sigma_c \qquad \sigma_Y := \sigma_r$

Stress parameters

Material	Bending		Traction		Torsion	
	σ_{-1}	σ_0	σ_{-1t}	σ_{0t}	τ_{-1}	τ_0
Carbon steel	min $0.44\,\sigma_u$ max $0.53\sigma_u - 4.1\cdot10^{-4}\,\sigma_u^2$ or $\dfrac{\sigma_u+\sigma_Y}{4}+50$ MPa	$(1.6\ldots1.8)\,\sigma_{-1}$	$0.315\sigma_u$ or $(0.7\ldots0.8)\,\sigma_{-1}$	$(1.5\ldots1.8)\,\sigma_{-1t}$	$(0.55\ldots0.58)\,\sigma_{-1}$	$(1.8\ldots2)\,\tau_{-1}$
Alloy steel	$0.4\,\sigma_u$		$0.26\,\sigma_u$ or $0.65\sigma_{-1}$			

According to the shaft material (OLC 45), which is a carbon steel, the bending stress is:

$$\sigma_{\_1} := \frac{\sigma_u + \sigma_Y}{4} + 50\ MPa = 295\ MPa$$

Torsion stress

$$\tau_{\_1} := 0.55 \cdot \sigma_{\_1} = 162.25\ MPa$$

Maximum stress

$$d_{s1} := 60\ mm$$

$M_{bmax} := 759\ N \cdot m$ maximum bending moment taken from the diagram

$$\sigma_{max} := \frac{32 \cdot M_{bmax}}{\pi \cdot {d_{s1}}^3} = 35.8\ MPa$$

$$\tau_{max} := \frac{16 \cdot M_{t1}}{\pi \cdot {d_{s1}}^3} = 13.2\ MPa$$

Bending fully reversible cycle

$$\sigma_{min} := -\sigma_{max}$$

$$\sigma_{min} = -35.792\ MPa$$

$$\sigma_v := \frac{2 \cdot \sigma_{max}}{2} = 35.792\ MPa$$

$$\sigma_m := \frac{\sigma_{max} + \sigma_{min}}{2} = 0\ MPa$$

Torsion - pulsating cycle

$$\tau_v := \frac{\tau_{max}}{2} = 6.605\ MPa$$

$$\tau_m := \frac{\tau_{max}}{2} = 6.605\ MPa$$ since τ_{min} is 0

$$\tau_c \cdot \tau_r = \tau_Y$$

$\tau_Y := 180\ MPa$ yield torsional strength

$$c_\sigma := \frac{1}{\frac{\beta_{K\sigma}}{\varepsilon \cdot \gamma} \cdot \frac{\sigma_v}{\sigma_{\_1}} + \frac{\sigma_m}{\frac{\sigma_r \cdot \sigma_c}{MPa}}} = 3.16 \qquad c_\tau := \frac{1}{\frac{\beta_{K\tau}}{\varepsilon \cdot \gamma} \cdot \frac{\tau_v}{\tau_{\_1}} + \frac{\tau_m}{\tau_Y}} = 8.246$$

$$c_1 := \frac{c_\sigma \cdot c_\tau}{\sqrt{{c_\sigma}^2 + {c_\tau}^2}} = 2.95 \qquad c_1 > 1$$

The safety coefficient higher than 1, which is a proof that the shaft can withstand the loads and work in safe conditions.

<u>Output shaft</u>

The shaft is made of the same material, OLC 45.

$$\sigma_r = 360\ MPa$$

$$\sigma_c = 620\ MPa$$

Calculation of $\beta_{K\sigma}$

$D := 100\ mm$
$d := 90\ mm$

$$t := D - d = 10\ mm$$

$$r := 4\ mm$$

$$\frac{t}{r} = 2.5$$

$2 < \frac{t}{r}\ 20$, so the parameters are:

$$C_1 := 1.232 + 0.832 \cdot \sqrt{\frac{t}{r}} - 0.008 \cdot \frac{t}{r} = 2.528$$

$$C_2 := -3.813 + 0.968 \cdot \sqrt{\frac{t}{r}} - 0.260 \cdot \frac{t}{r} = -2.932$$

$$C_3 := 7.423 - 4.868 \cdot \sqrt{\frac{t}{r}} + 0.869 \cdot \frac{t}{r} = 1.899$$

$$C_4 := -3.839 + 3.070 \cdot \sqrt{\frac{t}{r}} - 0.600 \cdot \frac{t}{r} = -0.485$$

$$\beta_{K\sigma} := C_1 + C_2 \cdot \left(\frac{2 \cdot t}{D}\right) + C_3 \cdot \left(\frac{2 \cdot t}{D}\right)^2 + C_4 \cdot \left(\frac{2 \cdot t}{D}\right)^3 = 2.013$$

Determination of $\beta_{K\tau}$

$D = 100\ mm$ $\quad$ $d = 90\ mm$

$t = 10\ mm$ $\quad$ $r = 4\ mm$

$\frac{t}{r} = 2.5$ $\quad$ $0.25 < \frac{t}{r} < 4$

$$C_1 := 0.905 + 0.783 \cdot \sqrt{\frac{t}{r}} - 0.075 \cdot \frac{t}{r} = 1.956$$

$$C_2 := -0.437 - 1.969 \cdot \sqrt{\frac{t}{r}} + 0.553 \cdot \frac{t}{r} = -2.168$$

$$C_3 := 1.557 + 1.073 \cdot \sqrt{\frac{t}{r}} - 0.578 \cdot \frac{t}{r} = 1.809$$

$$C_4 := -1.061 + 0.171 \cdot \sqrt{\frac{t}{r}} + 0.086 \cdot \frac{t}{r} = -0.576$$

$$\beta_{K\tau} := C_1 + C_2 \cdot \left(\frac{2 \cdot t}{D}\right) + C_3 \cdot \left(\frac{2 \cdot t}{D}\right)^2 + C_4 \cdot \left(\frac{2 \cdot t}{D}\right)^3 = 1.59$$

Size factor

$d_{s2} = 80\ mm$

The shaft diameter is between 51mm and 254mm, so the size factor is :

$$\varepsilon := 1.51 \cdot \left(\frac{d_{s2}}{mm}\right)^{-0.157} = 0.759$$

Surface finish factor

Since the material and the surface finish (machined) is the same as the input shaft, the surface finish factor is identical.

$$\gamma := 0.93$$

The bending and torsion stress are the same.

$\sigma_{\_1} = 295\ MPa$

$\tau_{\_1} = 162.25\ MPa$

Maximum stress calculation

$d_{s2} = 80\ mm$

$M_{bmax} := 542\ N \cdot m$

$$\sigma_{max} := \frac{32 \cdot M_{bmax}}{\pi \cdot {d_{s2}}^3} = 10.8\ MPa$$

$$\tau_{max} := \frac{16 \cdot M_{t2}}{\pi \cdot {d_{s2}}^3} = 11.7\ MPa$$

Bending fully reversible cycle

$\sigma_{min} := -\sigma_{max}$ $\qquad$ $\sigma_{min} = -10.783\ MPa$

$$\sigma_v := \frac{2 \cdot \sigma_{max}}{2} = 10.783\ MPa$$

$$\sigma_m := \frac{\sigma_{max} + \sigma_{min}}{} = 0\ MPa$$

$$\frac{}{2}$$

Torsion - pulsating cycle

$$\tau_v := \frac{\tau_{max}}{2} = 5.842\ MPa$$

$$\tau_m := \frac{\tau_{max}}{2} = 5.842\ MPa$$ τ_{min} is 0

$$\tau_c \cdot \tau_r = \tau_Y$$

$\tau_Y := 180\ MPa$ yield torsional strength

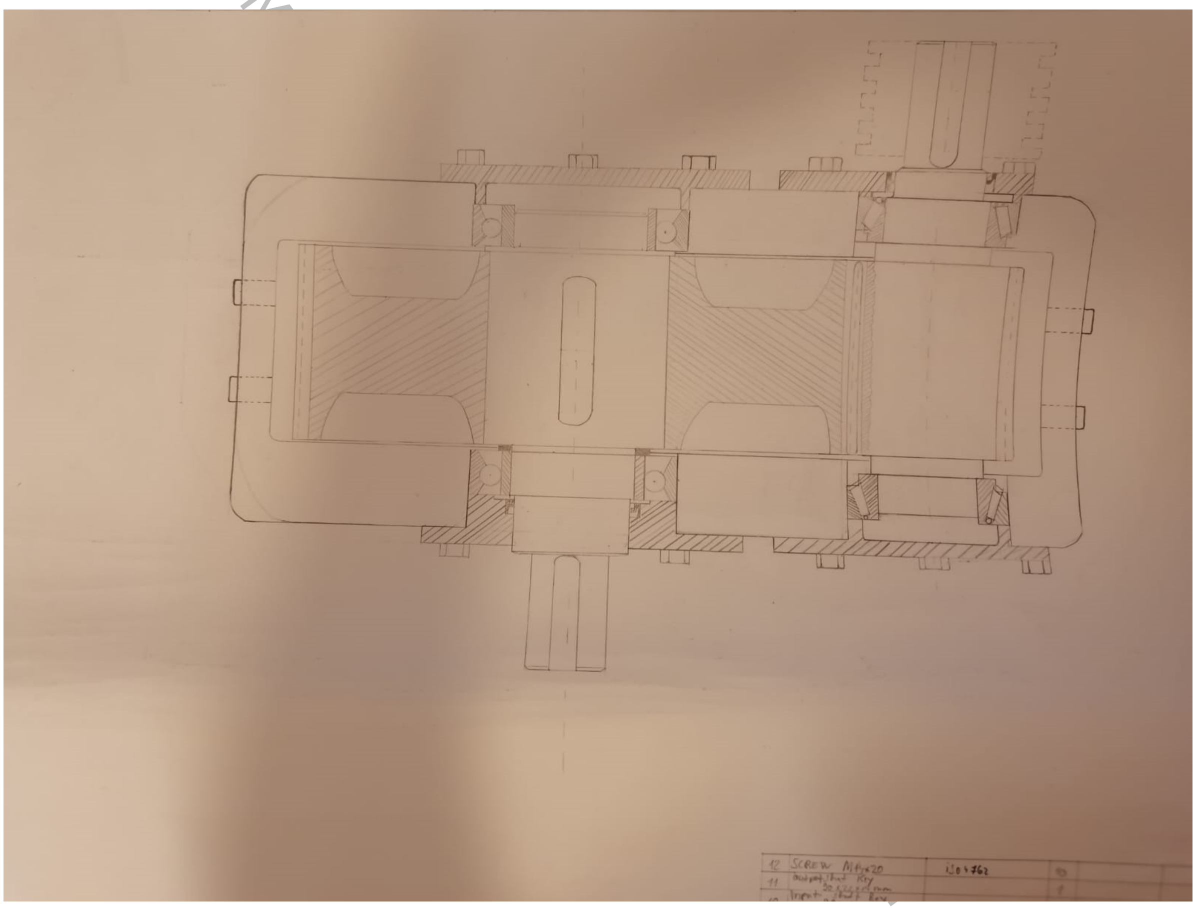
$$c_\sigma := \frac{1}{\frac{\beta_{K\sigma}}{\varepsilon \cdot \gamma} \cdot \frac{\sigma_v}{\sigma_{\_1}} + \frac{\sigma_m}{\frac{\sigma_r \cdot \sigma_c}{MPa}}} = 9.592 \qquad c_\tau := \frac{1}{\frac{\beta_{K\tau}}{\varepsilon \cdot \gamma} \cdot \frac{\tau_v}{\tau_{\_1}} + \frac{\tau_m}{\tau_Y}} = 8.806$$

$$c_2 := \frac{c_\sigma \cdot c_\tau}{\sqrt{{c_\sigma}^2 + {c_\tau}^2}} = 6.487 \qquad c_2 > 1$$

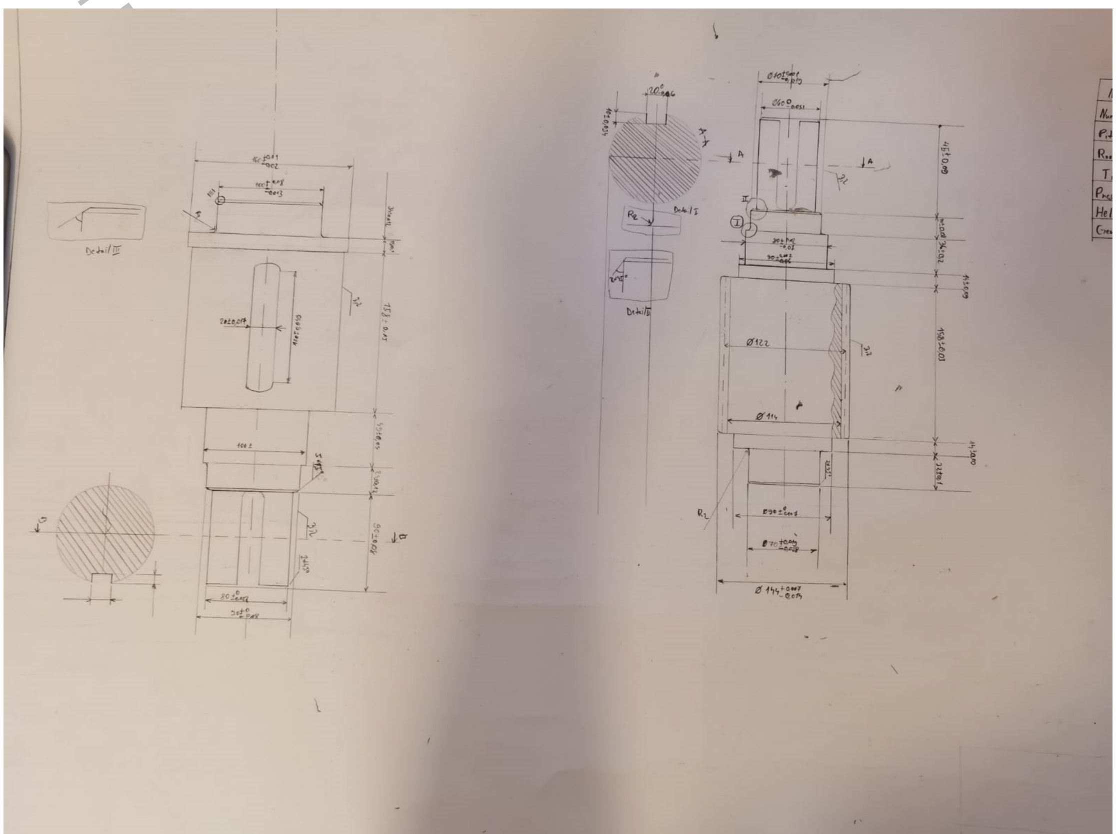
As expected, the safety factor for the output shaft is higher than the one for the input shaft. Although this value is too high, it still shows that the output shaft can work without issues and will not break.

Drawings of the gearbox and shafts

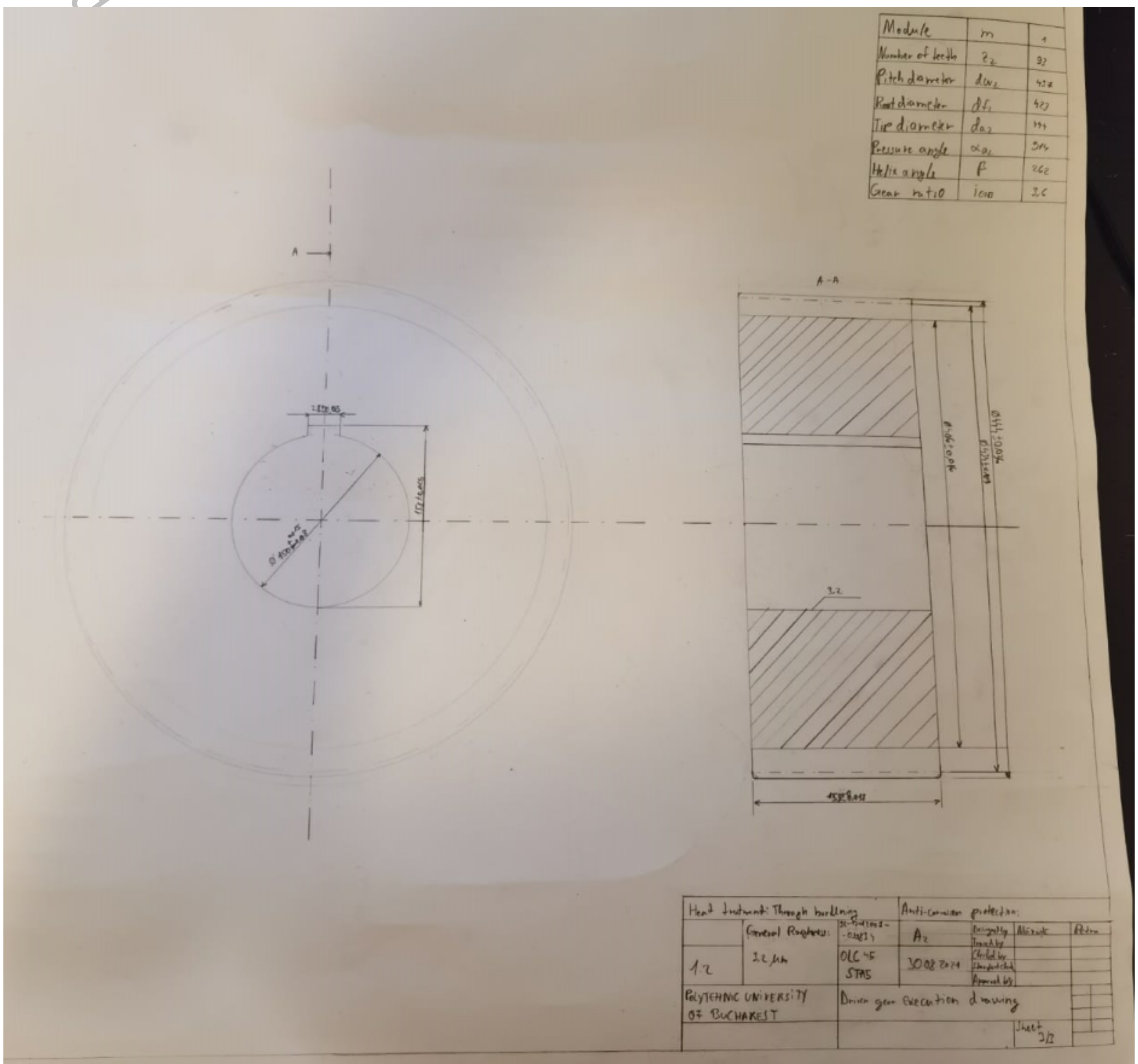
Main cross-section drawing of the gearbox



Detail drawing of the shafts



Driven gear execution drawing



Comments

In this project, the design of the mechanical transmission was separated into two parts, one that includes the design of the entire transmission with all components and the second part that covered the design of the details and the analysis of the gearbox. motor fed directly from a source to a working machine specially designed for the intended work.

The first part included the design of several components, such as an electric motor a belt drive to transfer power, speed and torque from the engine to a gearbox, a mounting gearbox that reduces speed and increases torque, a selected coupling to be able to transmit the output torque from the output shaft of the gearbox to the working machine and the working machine.

The first part also contained the creation of a 3D model of data transmission with coordinates for locating components, which deepened the understanding of the operation and design of a real transmission system.

The second part covered the calculation and detailed design of a single-speed helical gearbox, as opposed to the first part where the gearbox was double-speed helical and had just been selected from a catalog. The calculated gears will withstand good operation under the specified conditions, the shafts will not have problems under the same conditions, and the bearings will maintain optimal operation well above the required time limit.