VILNIUS GEDIMINAS TECHNICAL UNIVERSITY
Department of Mechatronics, Robotics and Production of Digital Press

Machine Elements
Course Project

Belt Conveyor Drive

Lecturer:
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Team 13
MRfuc-19

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Vilnius 2021

Team 13

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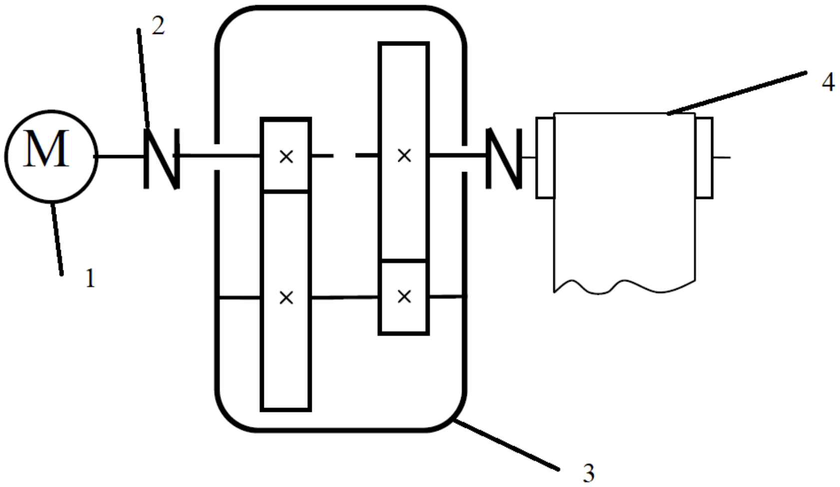
Belt conveyor drive:



1 – Engine;
2 - Elastic coupling;
3 - Gearbox;
4 - Conveyor belt.
5 - Drum;

I - motor shaft;
II - high-speed gearbox shaft;
III – intermediate gearbox shaft;
IV - low-speed gearbox shaft;
V - Shaft of the working machine.

Input data:

Belt pulling force F, kN	8.5
Belt speed V, m / s	0.4
Drum diameter D, mm	100
Drive service life Lw, years	8
Work hours per day, Lh	12
Machine operating mode:	C - medium

Data 2021.02.04

Project task: Chain conveyor drive Team No. 13

Student (Team Members): Kaan Unlu	Group

Project realization schedule:

Part	Date	*Possible Points*	*Earned Points*
I part	2021.02.25 <u>Drawings: Kinematical scheme. Sketches of machine.</u> 1. Functions, design requirements and evaluation criteria. 2. Kinematical calculation. Engine selection. 3. Power, torque transferred by shafts.	10	
II part	2021.03.18 <u>Drawings: Sketches of machine. Primary drawings of drives.</u> 4. Drives calculations. 5. Primary calculation of shaft diameter. 6. Selection of couplings, clutches or brakes.7. Elementary selection of bearings.8.Composition of machine.	10	
III part	2021.04.08 <u>Drawings: Drawings of drives. Sketches of reducer, machine composition.</u> 9. Calculation of shafts according acting loads. 10. Selection of bearings according dynamical loads. 11. Selection of tolerances and accuracy for bearings. 12. Selection and control calculations of key and spline joints, other fasteners.	10	
IV part	2021.04.22 <u>Drawings: Drawings. Specifications.</u> 13. Selection of tolerances, roughness, shape variations for surfaces of elements.14. Shaft calculation for durability. 15.Lubrication and sealing.	10	
	Total	30 (max)	
	1. Written Report (Drawings 30; Report 20) 2. Oral Report	50 20	
	TotalTotal	100	

Deadline 2021.04.30 (The report needs to be presented for lecturer 3 days before date of defense).

2021.05.06 (2021.05.13) project presentation

dr Eug. Jurkonis

Abstract

The belt conveyor drive is widely used in different industries, here a medium drive designed by using given data and adding some assumptions and regulations.

Design process consist of motor, shafts, gears, bearings, couplings, keys and corresponding calculations were made to make sure the device has a rational safety factor.

All drawings and tolerances of parts have been attached to the report.

Contents

1 Introduction

Belt conveyors are the dominant form of conveyor systems. They are used to transport materials or products over level or slightly inclined areas. In-line helical gear boxes, worm boxes, and bevel-helicals are used to power conveyor belt systems. In mining operations, a more heavy duty shaft mounted gearbox is ideal for moving large amounts of raw materials.

The main component of conveyor is travelling vertically-closed belt (so-called conveyor belt). Depending on the roller type, the belt shape may be flat or trough. The upper loaded and lower empty belt strands are supported by rollers. Progressive conveyor belt motion is imparted by a drive drum which is driven by a motor through a gearbox. Continuous tension is provided by means of a take-up. Load is fed onto the belt through a charging device and discharged though a drive drum funnel or at any point along the conveyor by means of special discharging devices.

A conveyor gearbox is the heart of any conveyance system. Conveyor belt systems are the most used method of moving a substantial amount of goods. The gearbox is the center of the conveyor system that pulls the various belts, chains, and rollers to move the products. Therefore, if the gearbox fails, the entire system fails.

1.1 Review of known structures

Conveyors drive systems can be mainly categorized to drives with gearbox and gearless ones. Since this course consist of gear selection and calculations and according to given instructions as Figure 1-1, the design will include gearbox.

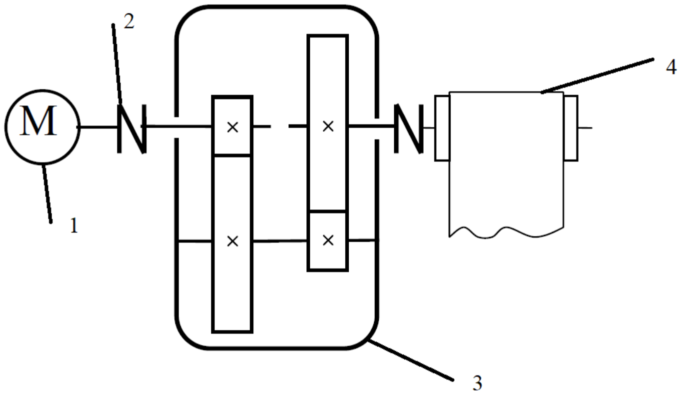


Figure 1-1: Given schematic of projects

In practice the design is usually a compact design and gearbox and drum axes are perpendicular to each other and worm gears or bevel gears are used to achieve this as shown in following figure.

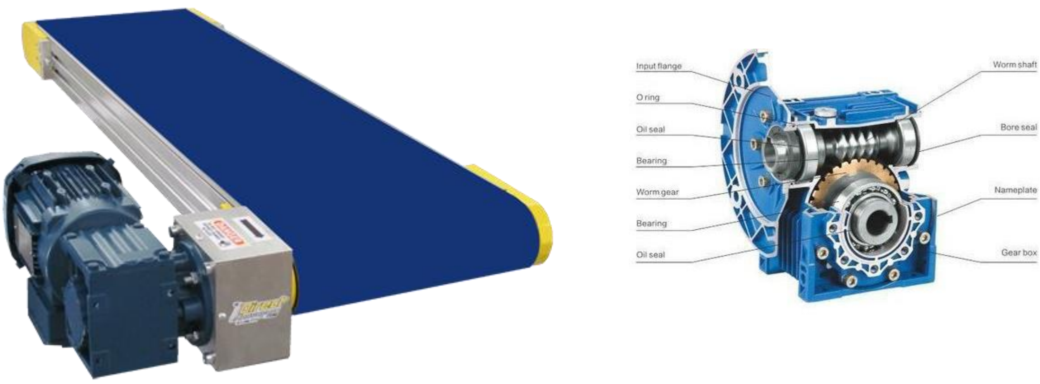


Figure 1-2: Conveyor drive system with worm gearbox

In given task linear gearbox is used which can have different steps to lower rotation speed of motor to increase the torque which transmits to belt drum. These type of drives are mainly used in heavy systems like heavy mining industries and can have different types of gearbox like 2 stage or 4 stage reducers.

Conveyor drive systems consist of other components like the brakes, couplings, hydraulic or elastic couplings, suspension system and etc. which are selected and used if required by the industry or environmental conditions.

Figure 1-3: Heavy duty linear conveyor drive system

2 Functions, design requirements and evaluation criteria

2.1 Function

The “Belt Conveyor Drive” is used to transmit power from an electromotor using a gearbox to a drum with reduced speed to rotate the drum.

2.2 Design Requirements

2.2.1 General information

The given requirements from customer are as following

Table 2-1: Design requirements stated in course description

Belt pulling force F, kN	8.5
Belt speed V, m / s	0.4
Drum diameter D, mm	100
Drive service life Lw, years	8
Work hours per day, Lh	12
Machine operating mode:	C - medium

By using this data we can find that drum’s angular speed and torque are:

$$\omega = {}^{v}/_{r} = {}^{0.4}/_{0.05} = 8\, rad/s \qquad 2\text{-}1$$

$$T = F.r = 8.5 * 1000 * 0.05 = 425\, N.m \qquad 2\text{-}2$$

So the required power will be,

$$P = \omega.T = 8 * 425 = 3.4\, kW \qquad 2\text{-}3$$

2.2.2 Cost

- Cost to purchase

This kind of drive with 4 kW power and around 70 rpm can be purchased for around 750 Euros.

Figure 2-1: Sample purchasable drive acquired from https://www.js-technik.de/en/products/gear-reducer-geared-motors/helical-gear-reducer-motors/6354/helical-gear-motor-js-mv-273-112m-4kw-75rpm?c=177

- Cost to use

Aside servicing like oil refilling and mounting costs, this product will use 4 kWh of electricity which varies for different locations but for typical EU industries 5 cents per kWh will adds up to 860 Euros annually.

- Cost to repair

Repairs can take place if a case like faulty gearbox had occurred where usually it will be replaced with new one for about half the price of product itself.

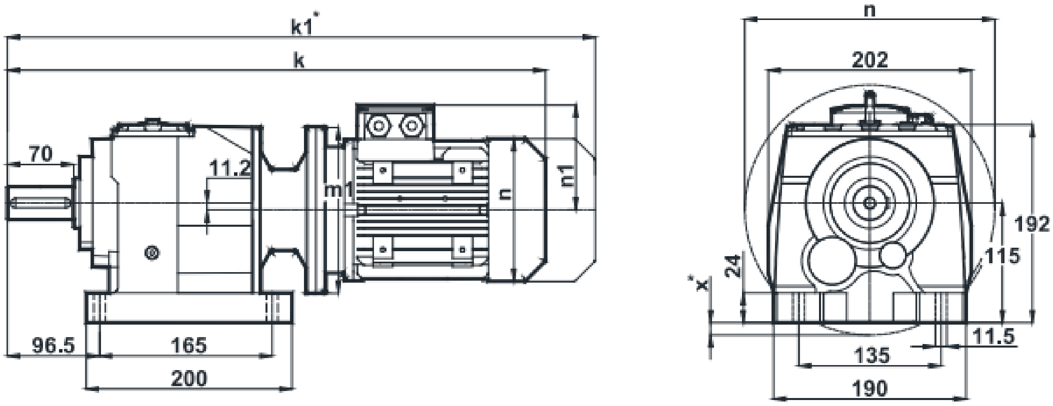
The motor's fan's price is around 150 Euros which is most likely to be in need of repairs.

2.2.3 Aesthetics

It is better be a one color finished product; so drum, motor, gearbox housing and couplings will have the same color like industrial anti corrosion color blue (RAL5010).

2.2.4 Geometry

The initial estimation of size of the machine can be obtained from the purchasable one which is as show in figure below its overall size is about 20x25x75 cm.



IEC	112M/B5 112M/B14
k	648
k1	752.5
n / n1	215 / 158
m1 (B5)	250
m1 (B14)	160
x (B5)	22

Figure 2-2: Overall size of conveyor drive

2.2.5 Physical characteristics

- Weight

  The overall weigh of product is about 60kg
- Surface texture

  The surface is not required to be fully polished except shafts.

2.2.6 Performance characteristics

- Accuracy

  The accuracy is mostly important in assembling parts and straightness of shafts.

  Assembled parts are bearings and gears on to the shafts and bearings into the housing.
- Reproducibility

  Because of usage of commercially available and pretty standard elements with widely used manufacturing technics it can be reproduced easily.
- Speed

  The drum will rotate with 8rad/s velocity.
- Acceleration and Deceleration

  It is not indicated in requirements of project so it will depend on the weight on belts during start or stop.

2.2.7 Inputs

- Energy consumption

  Energy consumption is around 4kWh of electricity.
- Labor

  If it is not connected to an automated circuit or it needs to be operated by a person to turn it on and off if needed.

2.2.8 Outputs

- Power

  This product will rotate a conveyor belt drum with diameter of 10mm with tension force of 8.5kN with angular velocity of 8rad/s by using a gearbox and transmitting electromotor power to it.

Some pollution due to bearings lubrication will be produced.

2.2.9 Manufacturing considerations

- Difficulty of making

  This drive system is made by typical mechanical and electromechanical parts so there is not any major difficulty but assembling bearings and shafts to have minimum misalignment and vibration can make the product more reliable
- Manufacturing Techniques

  Shaft will be manufactured using CNC lathe and milling machines,
  The housing will be a casted and machined using CNC.
  Gears will be manufactured using hub machines.
  Bearings, couplings and electro motor will be purchased.

2.2.10 Environmental requirements

- Operating temperature range

  Since the temperature for electromotor efficiency is important so the environment temperature shouldn't exceed 50 ° C.
- Resistance to corrosion

  The outer color of product is anti-corrosion but shafts should be covered with oil in time periods to avoid corrosion.

2.2.11 User requirements

- Ease of use

  The product is only needs to be powered on after mounting so there isn't any difficulty in usage.

2.2.12 Regulatory & licensing considerations

- Meets government rules

  It has very little pollution so government rules won't be a problem but electrical frequency of motor should be chosen as location's frequency.
- It doesn't need any patents or licenses.

2.2.13 How does it hold up?

- Service requirements

  It is required to refill gearbox oil and bearings lubricant in periodical manner as instructed by manufacturer. As a thumb rule the servicing every 6 months is enough.
- Ease of repair

  The housing can be opened and repaired if needed and parts are connected with couplings so it can be disassembled and repaired without moving whole parts.
- Lifespan

  As indicated in project requirement the life span is about 8 years.

2.2.14 Acoustic characteristics

The motor's sound is about 65 dB

2.3 Evaluation Criteria

2.3.1 Safety

The housing shouldn't allow direct contact with moving parts. The electromotor and its connection shouldn't be without electrical contact with other parts and earth cable should be connected properly.

2.3.2 Performance

The performance should be as required; 8rad/s angular velocity and 425Nm torque for drum.

2.3.3 Ease of manufacture

Manufacturing should be possible by usual methods and should be assembled easily

2.3.4 Ease of service or replacement of components

The components should be standard parts than can be replaced if it is needed and there should be an easy way to disassemble the component and install it without disassembling other parts.

2.3.5 Ease of operation

It should have the option to connect to automation circuit to act in synchronized with other components of system.

2.3.6 Low initial cost

Initial cost shouldn’t exceed 1000 euros.

2.3.7 Low operating and maintenance costs

The maintenance cost shouldn’t exceed 10 percent of initial cost annually.

2.3.8 Small size and low weight

The weight of product should be less than 100 kg and the length of it shouldn’t be more than 1 meters.

2.3.9 Low noise and vibration; smooth operation

The noise level should be less than 75 dB for industrial environment at day time.

2.3.10 Use of readily available materials and purchased components

The materials and components should be available in the country or near the usage location, especially replicable parts and highly worn components should be available to reduce unavailability time of device.

2.3.11 Use of both uniquely designed parts and commercially available components

The parts with high risk of fault should be commercially available component that can be replaced easily.

2.3.12 Appearance that is attractive and appropriate to the application

The appearance is better to be in one color and without rough edges.

3 Kinematical calculation & Engine selection

Required data shown in Table 2-1 so the drum angular rotation is 8rad/s and the required torque is 425Nm.

We find the efficiency of system to choose appropriate motor for the device. The efficiency is calculated by multiplying components efficiencies.

$$\eta = \eta_0 \eta_1 \eta_2 \eta_3 \eta_4 \qquad \text{3-1}$$

Where η_3 (here =0.95) is belt drum's efficiency, η_{1-2} (here = 0.96) is gear couplings and η_0 & η_4 are the efficiency of elastic couplings (here = 0.98).

So $\eta = 0.96^2 * 0.95 * 0.98^2 = 0.84$. Therefore the required power of electromotor is:

$$P_{eng} = \frac{T\omega}{\eta} = \frac{425 * 8}{0.84} = 4.04\ kW \qquad \text{3-2}$$

Since the angular velocity is going to reduce to 8rad/s (=76.4 rpm) then we assume that each stage reduce about 4 times so the rotation speed of motor should be about 1220 rpm.

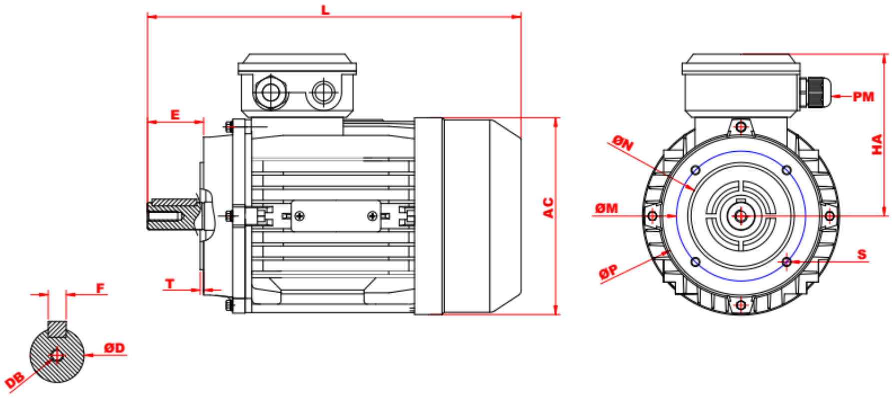
We chose 1500 rpm (1440) motors so we will increase the transmission ratio of gears.

$$n_{eng}/n_{exit} = i_1 i_2 = \frac{1440}{76.4} = 18.85 \qquad \text{3-3}$$

So each ratio is equal to 4.34 which is a little more than recommended value.

JS TECHNIK

MX3-112M1-4, 4 KW, 4 polig, B14 IE3



IEC 34-30

Maßblatt Drehstrommotor / dimensions three phase motor IM B14 [mm]

Typ / type	AC	ØD	DB	E	F	HA	PM	L	ØP	ØN	ØM	ØS
MX3-112M1-4	220	28	M10	60	8	172	M25*1.5	408	160	110	130	M8

Technische Daten / technical data

Typ / type	MX3-112M1-4	
Leistung / power	[kW]	4,00
Drehzahl / speed	[upm]	1440
Anzahl der Pole/ number of poles		4
Nennstrom / rated current (400V)	[A]	7,95
Leistungsfaktor / power factor	[cos φ]	0,82
Wirkungsgrad / efficiency	[100%] [75%] [50%]	88,6 87,8 83,9
Schutzart / degree of protection	[IP]	IP 55
Drehmoment / torque	[Nm]	26,5
Betriebsart / type of duty		S1
Iso. Klasse		F
max. Temperaturanstieg / temperature rise	[K]	≦70
Frequenz / frequency	[Hz]	50/60
Spannung / voltage	400/690 [Δ/Y] 50Hz 460/796 [Δ/Y] 60Hz	
Motorschutz / motor protection	3x PTC Kaltleiter / Thermistors	

Anzugs- zu Nennstrom / starting current	[Ia/In]	6,5
Kipp- zu Nennmoment / pull-out torque	[MK/Mn]	2,3
Anzugs- zu Nennmoment / starting torque	[Ma/Mn]	2,2
Trägheitsmoment / moment of inertia	[Kgm^2]	0,01508
Lager / bearing	[DE] [NDE]	6306 6306
max. Radialbelastung / max. radial force	[N]	1300
Schallleistungspegel / noise level	[dB]	65
Umgebungstemperatur / ambient temperature	[°C]	≦40
Gewicht / weight	[kg]	34
Aufstellungshöhe altitude	[m]	≦1000

4 Power, torque transmitted by shafts

Power transmission and torques are functions of efficiency and transmission ratio.

$$P_i = P_{i-1} * \eta_i \quad \text{4-1}$$

$$\omega_i = \omega_{i-1}/i_i \quad \text{4-2}$$

$$T_i = P_i/\omega_i \quad \text{4-3}$$

According to these formulas, Table 4-1 obtained by using project's input and requirements.

Table 4-1: Calculated power, angular velocity and momentum for shafts

Shaft	η_i	i_i	P_i (W)	ω_i (rad/s)	n_i (rpm)	T_i (Nm)
Drum Shaft	0.95*0.98=0.93	1	3360	8.0	76	425.0
Gear Shaft 3	0.96	4.34	3613	8.0	76	451.2
Gear Shaft 2	0.96	4.34	3763	34.7	332	108.3
Gear Shaft 1	0.98	1	3920	150.8	1440	26.0
Motor Shaft	1	1	4000	150.8	1440	26.5

- Motor Shaft

$$P_1 = 4000\,W$$

$$n_1 = 1440\,rpm \;=>\; \omega_1 = 150.8\;{}^{rad}/_{s}$$

$$T_1 = {}^{P_1}/_{\omega_1} = {}^{4000}/_{150.8} = 26.5\,N.m$$

- Gear Shaft 1

$$P_2 = P_1.\eta_2 = 4000 * 0.98 = 3920\,W$$

$$\omega_2 = {}^{\omega_1}/_{i_2} = 150.8\;{}^{rad}/_{s}$$

$$T_2 = {}^{P_2}/_{\omega_2} = {}^{4000}/_{150.8} = 26.0\,N.m$$

- Gear Shaft 2

$$P_3 = P_2.\eta_3 = 3920 * 0.96 = 3763\ W$$

$$\omega_3 = {}^{\omega_2}/_{i_3} = {}^{150.8}/_{4.34} = 34.7\ {}^{rad}/_{s}$$

$$T_3 = {}^{P_3}/_{\omega_3} = {}^{3763}/_{34.7} = 108.3\ N.m$$

- Gear Shaft 3

$$P_4 = P_3.\eta_4 = 3763 * 0.96 = 3613\ W$$

$$\omega_4 = {}^{\omega_3}/_{i_4} = {}^{34.7}/_{4.34} = 8\ {}^{rad}/_{s}$$

$$T_4 = {}^{P_4}/_{\omega_4} = {}^{3613}/_{8} = 451.2\ N.m$$

- Drum Shaft

$$P_5 = P_4.\eta_5 = 3613 * 0.93 = 3360\ W$$

$$\omega_5 = {}^{\omega_4}/_{i_5} = {}^{8}/_{1} = 8\ {}^{rad}/_{s}$$

$$T_5 = {}^{P_5}/_{\omega_5} = {}^{3360}/_{8} \cong 425.0\ N.m$$

5 Drive calculations

Calculations made by using MITCalc. Material for Gears and shafts are Alloy structural steel 42 CrV 6 tooth face hard.

5.1 Parameters of the chosen material

Density	ρ=7870 kg/m^3
Young's Modulus (Modulus of Elasticity)	E = 206 Gpa
Tensile Strength, Ultimate	R_m=980 MPa
Tensile Strength, Yield	$R_{p0.2}$ =850 MPa
Poison's Ratio	0,3
Contact Fatigue Limit	S_{Hlim}=1160 MPa

Bending Fatigue Limit	S_{Flim}=528 MPa
Tooth Hardness - Side	VHV=600 HV
Tooth Hardness - Core	JHV=315 HV
Base Number of Load Cycles in Contact	N_{Hlim}=1,00E+08
Wohler Curve Exponent for Contact	qH=10
Base Number of Load Cycles in Bend	NFlim=3,00E+06
Wohler Curve Exponent for Bend	qF=6

5.2 Common Coefficients for safety calculation

Dynamic factor KV (max. value)	5.00
Face load factor contact stress KHbeta (max)	5.00
Reversals of the load (factor YA)	Without reversals (YA=1)
Oil type (ZL)	Synthetic Oil
Used / Recommended lubricant viscosity	v50
Tooth roughness (factor ZR)	Ra 0.4
Roughness in the tooth root fillets (factor YR)	Ra 1.6
Coefficient of one-off overloading	KAS=2,00
Coefficient of safety (contact/bend)	S_H=1,30 S_F=1,60

5.3 First Gears

Basic dimensions of gearing

Number of teeth Pinion / Gear	z	15	65	
Face width (Pinion / Gear)	b	47	45	[mm]
Normal module	mn	3.5		[mm]
Transverse module	mt	3.5000		[mm]
Circular pitch	p	10.996		[mm]
Transverse circular pitch	pt	10.996		[mm]
Base circular pitch	ptb	10.332		[mm]
Center distance (pitch)	a	140.0000		[mm]
Center distance (production)	av	140.0000		[mm]
Center distance (working)	aw	140.0000		[mm]
Pressure angle	α	20.0000		[°]
Transverse pressure angle	αt	20.0000		[°]
Pressure angle at the pitch cylinder	αwn	20.0000		[°]

Transverse pressure angle at the pitch cylinder	αwt	20.0000		[°]
Helix angle	β	0.00		[°]
Base helix angle	β b	0.0000		[°]
Tip diameter	da	59.5000	234.5000	[mm]
Reference diameter	d	52.5000	227.5000	[mm]
Base diameter	db	49.3339	213.7801	[mm]
Root diameter	df	43.7500	218.7500	[mm]
Operating pitch diameter	dw	52.5000	227.5000	[mm]
Addendum	ha	3.5000	3.5000	[mm]
Dedendum	hf	4.3750	4.3750	[mm]
Tooth thickness on the tip diameter	sna	2.2974	2.7639	[mm]
Tooth thickness on the tip diameter (transverse)	sta	2.2974	2.7639	[mm]
Tooth thickness on the pitch diameter	sn	5.4978	5.4978	[mm]
Tooth thickness on the pitch diameter (transverse)	st	5.4978	5.4978	[mm]
Tooth thickness on the root diameter	sb	5.2336	8.5467	[mm]
Unit tooth thickness on the tip diameter	sa*	0.6564	0.7897	[modul]
Common for the gearing				
Theoretical single stiffness	c'_{th}	16.243		[N/(μm*mm)]
Stiffness of a tooth pair (single stiffness)	c'	8.032		[N/(μm*mm)]
Meshing stiffness per unit face width	$c_{\gamma\alpha}$	11.883		[N/(μm*mm)]
Application factor	KA	1.750		
Dynamic factor	KV	1.113		
Number of cycles	NK	3.02E+09	6.98E+08	
For pitting safety calculation				
Face load factor (contact stress)	$K_{H\beta}$	1.740		
Transverse load factor (contact stress)	$K_{H\alpha}$	1.036		
Total factor of additional loads	KH	3.512		
Elasticity factor	ZE	189.81		
Zone factor	ZH	2.495		
Helix angle factor	Z_β	1.000		
Contact ratio factor	Z_ε	0.887		
Work hardening factor	ZW	0.925	1.000	
Size factor	ZX	1.000	1.000	
Lubricant factor	ZL	1.184	1.184	
Peripheral speed factor	ZV	0.974	0.974	

Roughness factor affecting surface durability	ZR	0.951	0.951
Life factor for contact stress	ZNT	0.887	0.934
Single pair tooth contact factor	ZB/ZD	1.155	1.000

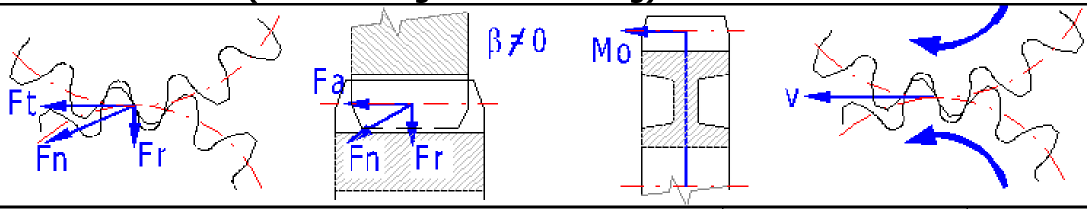
For bending safety calculation

Face load factor (root stress)	$K_{F\beta}$	1.583	
Transverse load factor (root stress)	$K_{F\alpha}$	1.036	
Total factor of additional loads	KF	3.196	
Helix angle factor	Y_{β}	1.000	
Rim thickness factor	YB	1.000	1.123
Deep tooth factor	YDT	1.000	1.000
Notch sensitivity factor	Y_{δ}	0.990	0.997
Size factor	YX	1.000	1.000
Tooth-root surface factor	YR	1.004	1.004
Alternating load factor	YA	1.000	
Production technology factor	YT	1.000	
Life factor for bending stress	YNT	0.871	0.897
Stress correction factor	YST	1.000	1.000
Form factor (bending)	YF	1.785	1.325
Stress correction factor	YS	1.733	2.068
Stress correction factor for gears with notches in fillets	YSg	1.932	2.422

Stress and safety coefficients

Safety coefficient for surface durability	SH	1.60	2.10	
Safety coefficient for bending durability	SF	7.68	7.67	
Safety in contact in one-time overloading	SHst	2.60	3.00	
Safety in bending in one-time overloading	SFst	15.68	16.94	
Variability coefficient for calculation of probability of a failure	vH/vF	0.06	0.1	
Probability of a failure	P	0.00		[%]
Nominal contact stress	Sigma H0	301.71		[MPa]
Contact stress	Sigma H	652.94	565.43	[MPa]
Pitting stress limit	Sigma HG	1042.55	1187.50	[MPa]
Permissible contact stress	Sigma HP	801.96	913.46	[MPa]
Nominal tooth-root stress	SigmaF 0	18.62	19.35	[MPa]
Tooth-root stress	SigmaF	59.49	61.84	[MPa]
Tooth-root stress limit	SigmaF G	457.11	474.11	[MPa]
Permissible bending stress	SigmaF P	285.69	296.32	[MPa]

Force conditions (forces acting on the toothing)



Tangential force	Ft	990.37		[N]
Normal force	Fn	1053.93		[N]
Axial force	Fa	0.00		[N]
Radial force	Fr	360.47		[N]
Bending moment	Mo	0.00	0.00	[Nm]
Peripheral speed on the pitch diameter	v | vmax	3.96	< 8	[m/s]
Tangential load per unit tooth width / Unit load	wt | wt*	38.51	11.00	[N/mm | MPa]

According to Safety coefficients, chosen material and gears sizes satisfies requirements.

5.4 Second Gears

Basic dimensions of gearing

Number of teeth Pinion / Gear	z	15	65	
Face width (Pinion / Gear)	b	61	59	[mm]
Normal module	mn	4.5		[mm]
Transverse module	mt	4.5000		[mm]
Circular pitch	p	14.137		[mm]
Transverse circular pitch	pt	14.137		[mm]
Base circular pitch	ptb	13.285		[mm]
Center distance (pitch)	a	180.0000		[mm]
Center distance (production)	av	180.0000		[mm]
Center distance (working)	aw	180.0000		[mm]
Pressure angle	α	20.0000		[°]
Transverse pressure angle	αt	20.0000		[°]
Pressure angle at the pitch cylinder	αwn	20.0000		[°]
Transverse pressure angle at the pitch cylinder	αwt	20.0000		[°]
Helix angle	β	0.00		[°]
Base helix angle	β b	0.0000		[°]
Tip diameter	da	76.5000	301.5000	[mm]

Reference diameter	d	67.5000	292.5000	[mm]
Base diameter	db	63.4293	274.8601	[mm]
Root diameter	df	56.2500	281.2500	[mm]
Operating pitch diameter	dw	67.5000	292.5000	[mm]
Addendum	ha	4.5000	4.5000	[mm]
Dedendum	hf	5.6250	5.6250	[mm]
Tooth thickness on the tip diameter	sna	2.9538	3.5536	[mm]
Tooth thickness on the tip diameter (transverse)	sta	2.9538	3.5536	[mm]
Tooth thickness on the pitch diameter	sn	7.0686	7.0686	[mm]
Tooth thickness on the pitch diameter (transverse)	st	7.0686	7.0686	[mm]
Tooth thickness on the root diameter	sb	6.7289	10.9886	[mm]
Unit tooth thickness on the tip diameter	sa*	0.6564	0.7897	[modul]
Unit correction	dY	0.0000		[modul]
Total unit correction	x1+x2	0.0000		[modul]
Addendum modification coefficient	x	0.0000	0.0000	[modul]
Common for the gearing				
Theoretical single stiffness	c'_{th}	16.243		[N/(μm*mm)]
Stiffness of a tooth pair (single stiffness)	c'	12.512		[N/(μm*mm)]
Meshing stiffness per unit face width	$c_{\gamma\alpha}$	18.510		[N/(μm*mm)]
Application factor	KA	1.750		
Dynamic factor	KV	1.023		
Number of cycles	NK	6.97E+08	1.61E+08	
For pitting safety calculation				
Face load factor (contact stress)	$K_{H\beta}$	1.498		
Transverse load factor (contact stress)	$K_{H\alpha}$	1.022		
Total factor of additional loads	KH	2.740		
Elasticity factor	ZE	189.81		
Zone factor	ZH	2.495		
Helix angle factor	Z_β	1.000		
Contact ratio factor	Z_ε	0.887		
Work hardening factor	ZW	0.906	1.000	
Size factor	ZX	1.000	1.000	
Lubricant factor	ZL	1.222	1.222	
Peripheral speed factor	ZV	0.951	0.951	
Roughness factor affecting surface durability	ZR	0.958	0.958	

Life factor for contact stress	ZNT	0.934	0.983
Single pair tooth contact factor	ZB/ZD	1.155	1.000

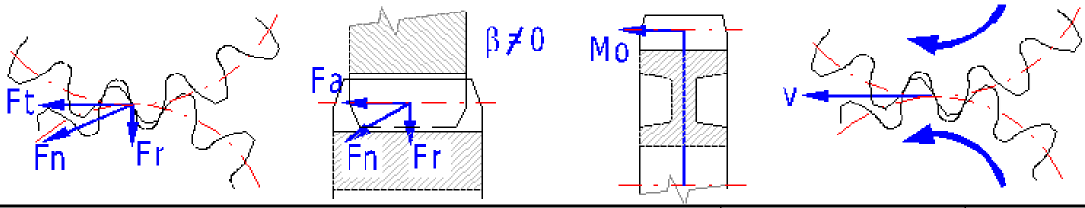
For bending safety calculation

Face load factor (root stress)	$K_{F\beta}$	1.400	
Transverse load factor (root stress)	$K_{F\alpha}$	1.022	
Total factor of additional loads	KF	2.561	
Helix angle factor	Y_{β}	1.000	
Rim thickness factor	YB	1.000	1.000
Deep tooth factor	YDT	1.000	1.000
Notch sensitivity factor	Y_{δ}	0.990	0.997
Size factor	YX	1.000	1.000
Tooth-root surface factor	YR	1.004	1.004
Alternating load factor	YA	1.000	
Production technology factor	YT	1.000	
Life factor for bending stress	YNT	0.897	0.923
Stress correction factor	YST	1.000	1.000
Form factor (bending)	YF	1.785	1.325
Stress correction factor	YS	1.733	2.068
Stress correction factor for gears with notches in fillets	YSg	1.932	2.422

Stress and safety coefficients

Safety coefficient for surface durability	SH	1.37	1.83	
Safety coefficient for bending durability	SF	5.09	5.76	
Safety in contact in one-time overloading	SHst	2.12	2.45	
Safety in bending in one-time overloading	SFst	10.08	11.01	
Variability coefficient for calculation of probability of a failure	vH/vF	0.06	0.1	
Probability of a failure	P	0.26		[%]
Nominal contact stress	Sigma H0	418.18		[MPa]
Contact stress	Sigma H	799.33	692.20	[MPa]
Pitting stress limit	Sigma HG	1092.51	1269.56	[MPa]
Permissible contact stress	Sigma HP	840.39	976.59	[MPa]
Nominal tooth-root stress	SigmaF 0	36.13	33.10	[MPa]
Tooth-root stress	SigmaF	92.52	84.76	[MPa]
Tooth-root stress limit	SigmaF G	470.75	488.26	[MPa]
Permissible bending stress	SigmaF P	294.22	305.16	[MPa]

Force conditions (forces acting on the toothing)



Tangential force	Ft	3207.20		[N]
Normal force	Fn	3413.03		[N]
Axial force	Fa	0.00		[N]
Radial force	Fr	1167.32		[N]
Bending moment	Mo	0.00	0.00	[Nm]
Peripheral speed on the pitch diameter	v | vmax	1.17	< 8	[m/s]
Tangential load per unit tooth width / Unit load	wt | wt*	95.13	21.14	[N/mm | MPa]

According to Safety coefficients, chosen material and gears sizes satisfies requirements.

6 Elementary calculation of shafts diameters

Since the actual shaft diameters not known yet an initial calculation of shafts estimates the diameter of shafts.

This equation used to find shaft diameters.

$$d \geq \sqrt[3]{\frac{T}{0.2[\tau]}} \qquad \textit{6-1}$$

Where d=diameter of the shaft mm,

T=Torque N.mm,

[τ] – allowed shear stress, MPa. Usually [τ] can be taken 20 MPa.

- Drum Shaft

$$d \geq \sqrt[3]{\frac{425 * 1000}{0.2 * 20}} = 47.36$$

Therefore, the dimeter chosen to be **48mm**

- Gear Shaft 3

$$d \geq \sqrt[3]{\frac{451.2 * 1000}{0.2 * 20}} = 48.3$$

Therefore, the dimeter chosen to be **50mm**

- Gear Shaft 2

$$d \geq \sqrt[3]{\frac{108.3 * 1000}{0.2 * 20}} = 30.0$$

Therefore, the dimeter chosen to be **30mm**

- Gear Shaft 1

$$d \geq \sqrt[3]{\frac{26.0 * 1000}{0.2 * 20}} = 18.7$$

Therefore, the dimeter chosen to be **20mm**

- Motor Shaft

  The dimeter of chosen motor shaft is **28mm**

7 Selection and control calculations of couplings, clutches, brakes.

Since calculated shaft diameter needed for the first gear shaft is 20 and EM's shaft diameter is 28mm, desired coupling is 20/28 and with torque limit of 26.5 N.m

In the catalogue used from http://browneuropeltd.com/couplings.htm and it has a model specially made for common EMs and it is 24/28 for the 27N.m AC motor.

BROWN EUROPE

RX TORSIONALLY FLEXIBLE COUPLING

RX couplings for standard IEC Metric AC Motors

This table pre-supposes normal operation and the selections are made using a 92º shore (yellow) spider.

AC Motor Size	Motor Output 3000 rpm kW	Nm	RX Size	Safety factor with TK MAX	Motor Output 1500 rpm kW	Nm	RX Size	Safety factor with TK MAX	Motor Output 1000 rpm kW	Nm	RX Size	Safety factor with TK MAX	Motor Output 750 rpm kW	Nm	RX Size	Safety factor with TK MAX	Motor Shaft 3000 ≤ 1500
56	0.09 0.12	0.32 0.41	9	19 14	0.06 0.09	0.43 0.64	9	14 9.4	0.037 0.045	0.43 0.52	9	14 12					9 x 20
63	0.18 0.25	0.62 0.86	14	24 17	0.12 0.18	0.88 1.3	14	17 11.5	0.06 0.09	0.72 1.1	14	21 14					11 x 23
71	0.37 0.55	1.3 1.9		12 7.9	0.25 0.37	1.8 2.5		8.3 6.0	0.18 0.25	2.0 2.7		7.5 5.6	0.09 0.12	1.4 1.8	14	11 8.3	14 x 30
80	0.75 1.1	2.5 3.7	19/24	8.0 5.4	0.55 0.75	3.7 5.1	19/24	5.4 3.9	0.37 0.55	3.9 5.8	19/24	5.1 3.4	0.18 0.25	2.5 3.5	19/24	8.0 5.7	19 x 40
90 S	1.5	5.0		4.0	1.1	7.5		2.7	0.75	8.0		2.5	0.37	5.3		3.8	24 x 50
90 L	2.2	7.4		2.7	1.5	10		2.0	1.1	12	24/28	5.8	0.55	7.9		2.5	
100 L	3	9.8	24/28	7.1	2.2 3	15 20	24/28	4.7 3.5	1.5	15		4.7	0.75 1.1	11 16	24/28	6.4 4.4	28 x 60
112 M	4	13		5.4	4	27		2.6	2.2	22		3.2	1.5	21		3.3	

8 Elementary selection of bearings

8.1 Selection Criteria

8.1.1 Available space

Space available for bearings can get up to 50 millimeters in outer diameter for smaller shaft (25mm diameter) and up to 80mm for bigger shafts (35-55mm) and Available width of bearings can go up to 17mm

8.1.2 Loads

Loads acting on shafts will be from 350 N (input shaft) to 1100 N (output shaft). Therefore, on output shafts bearings should be stronger.

8.1.3 Misalignment

Misalignment will be medium because drives designed so it would have as less shock as possible.

8.1.4 Precision

Bearings should precisely fit into the sockets as well as shafts should precisely fit into bearing bore.

8.1.5 Speed

Speed varies from 76rpm to 1440 rpm, so considering this; we can state that speed is medium

8.1.6 Quiet running

Noise of running is partially important considering that machine will work in public area

8.1.7 Stiffness

Stiffness of material should be high to avoid unnecessary moves of axis

8.1.8 Axial displacement

To avoid failures of gears and noise shafts should not move axially. So bearings should hold axis of spinning tight.

8.1.9 Mounting and dismounting integral

Considering that mounting is one of the critical stages of the bearing's life cycle, bearing should be mounted and dismounted with special tools.

8.1.10 Seals

Bearings should have contact seal from one side.

According to the table below, which is acquired from https://www.skf.com/ the deep groove ball bearing meet the requirements.

Suitability of rolling bearings for industrial applications

Symbols

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

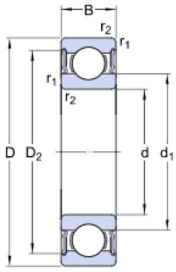
↔ double direction
← single direction
□ non-locating displacement on the seat
■ non-locating displacement within the bearing
✓ yes
✗ no

	Load carrying capability			Misalignment		Arrangement				Suitable for					Design features			
Bearing type	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 ✓	✗	✗	✗

According to loads and initial shaft diameters bearings are selected as below.

Gear Shaft 1

SKF 61905-2RS1



DIMENSIONS

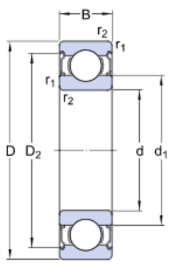
d	25 mm
D	42 mm
B	9 mm
d_1	≈ 30.25 mm
D_2	≈ 37.7 mm
$r_{1,2}$	min. 0.6 mm

CALCULATION DATA

Basic dynamic load rating	C	7.02 kN
Basic static load rating	C_0	4.3 kN
Fatigue load limit	P_u	0.193 kN
Limiting speed		10 000 r/min
Calculation factor	k_r	0.02
Calculation factor	f_0	14.7

Gear Shaft 2

SKF W 6206-2Z



DIMENSIONS

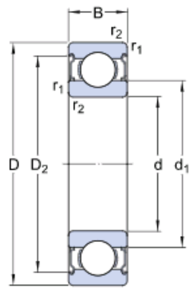
d	30 mm
D	62 mm
B	16 mm
d_1	≈ 40.7 mm
d_2	≈ 40.7 mm
D_2	≈ 55.13 mm
$r_{1,2}$	min. 1 mm

CALCULATION DATA

Basic dynamic load rating	C	16.5 kN
Basic static load rating	C_0	11.2 kN
Fatigue load limit	P_u	0.48 kN
Reference speed		26 000 r/min
Limiting speed		13 000 r/min
Calculation factor	k_r	0.03
Calculation factor	f_0	13.9

Gear Shaft 3

SKF 6010-2Z



DIMENSIONS

d	50 mm
D	80 mm
B	16 mm
d_1	≈ 59.75 mm
D_2	≈ 72.8 mm
$r_{1,2}$	min. 1 mm

CALCULATION DATA

Basic dynamic load rating	C	22.9 kN
Basic static load rating	C_0	15.6 kN
Fatigue load limit	P_u	0.71 kN
Reference speed		18 000 r/min
Limiting speed		9 000 r/min
Calculation factor	k_r	0.025
Calculation factor	f_0	14.7

9 Calculation of shafts

Shaft design and calculation | **Shaft calculation** | **Clear table of results**

i **Calculation without errors.**

ii **Project information**

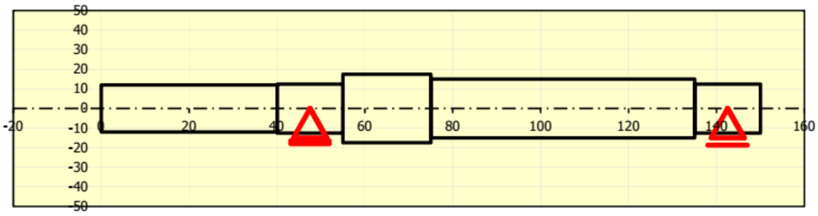
? **Input sectinon**

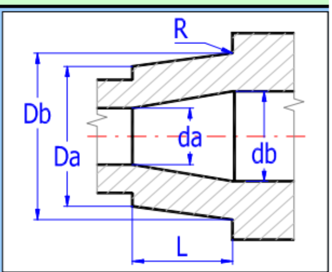
1.0 Preliminary shaft diameter design

1.1 Calculation units	SI Units (N, mm, kW...)			
1.2 Transmitted power	4.00	[kW]	1.6 Type of shaft load	C...Repeated torsion + bending
1.3 Shaft speed	1440	[/min]		
1.4 Torsion moment	26.53	[Nm]	1.7 Material of the shaft	A...Common structural steel (500)
1.5 Preliminary min. diameter	24.47	[mm]		

2.0 Shaft shape and dimensions

2.1 The scale of the displayed shaft diameter. ☑ Calculation units SI Units (N, mm, kW...)



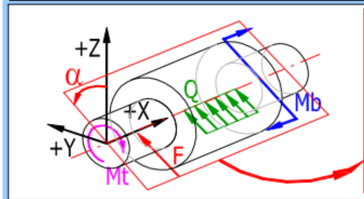


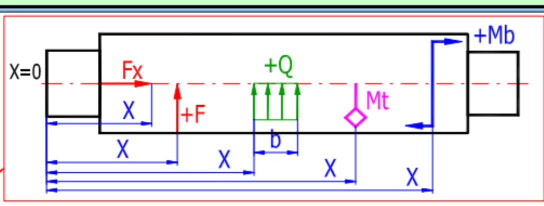
2.2 Table	1	2	3	4	5	6	7	8	9	10
Origin	0.00	40.00	55.00	75.00	135.00	150.00	150.00	150.00	150.00	150.00
L	40.000	15.000	20.000	60.000	15.000					
ø Da	24.000	25.000	35.000	30.000	25.000					
ø Db	24.000	25.000	35.000	30.000	25.000					
ø da	0.000	0.000	0.000	0.000	0.000					
ø db	0.000	0.000	0.000	0.000	0.000					
R	0.500	0.500	0.500	0.500	0.500					

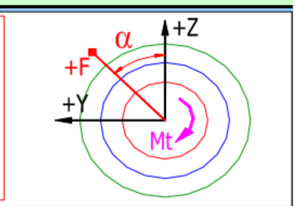
2.3 Total length of the shaft		150.00	[mm]	2.6 The shaft surface (Roughness Ra)	
2.4 X-coordinate of the left support (bearing)	Fixed	47.50 ◉	[mm]	C...Ground (0.8)	
2.5 X-coordinate of the right support (bearing)	Free	142.50 ○	[mm]		

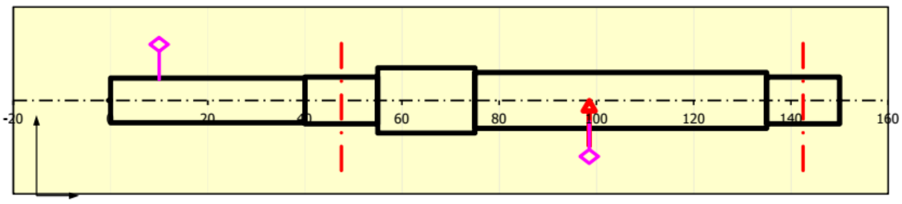
3.0 Notches and necking-down on the shaft

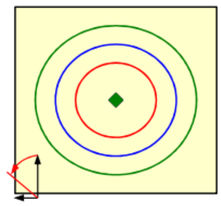
4.0 Loading of the shaft











4.1 Loading	X	Fx	F	alfa	Mt	Mb	alfa	Q	b	alfa
	[mm]	[N]		[°]	[Nm]		[°]	[N/mm]	[mm]	[°]
1	98.50	0.0	1054.0	0	26.00	0.00	0			
2	10.00				-26.00					
3										
4										
5										
6										

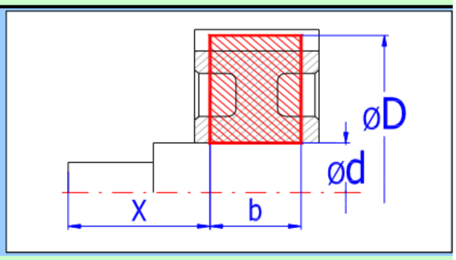
7										
8										
9										
10										

5.0 Rotating masses

5.1 Additional rotating masses (resonance speed)

5.2 Use loading from the weight of disks in the calculation? Yes

5.3	X	D	d	b	Ro	m	
	[mm]				[kg/m^3]	[kg]	
M1	75.00	52.50	30.00	47.00	7800.0	0.53	☑
M2	0.00	0.00	0.00	0.00	7800.0	0.00	☑
M3	0.00	0.00	0.00	0.00	7800.0	0.00	☑
M4	0.00	0.00	0.00	0.00	7800.0	0.00	☑
M5	0.00	0.00	0.00	0.00	7800.0	0.00	☑



6.0 Material and the type of loading

? Results section

7.0 Results - summary

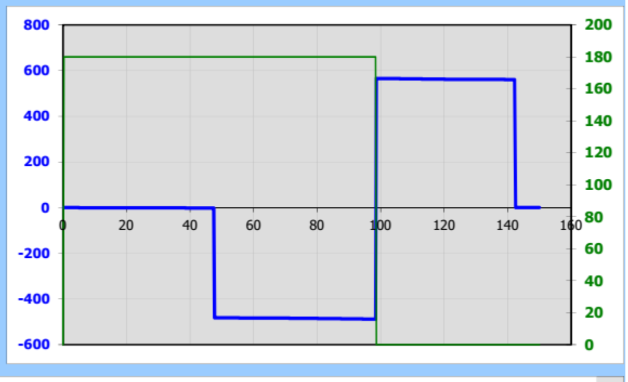
		x	y	z	Σy+z	
7.1	Reaction in the support R1	0	-1.431E-09	-480.87936	480.879364	[N]
7.2	Reaction in the support R2	0	-1.662E-09	-560.58863	560.588632	[N]

7.3	Total shaft weight	m	0.74	[kg]
7.4	Maximum deflection	y	0.0030	[mm]
7.5	Maximum torsional deflectio	φ	0.0326	[°]
7.6	Angular deflection in R1	ϑ	0.0036	[°]
7.7	Angular deflection in R2	ϑ	0.0041	[°]
7.8	Max. bending stress	σ_e	9.3	[MPa]
7.9	Max. stress in shear	τ_s	1.1	[MPa]
7.10	Max. stress in torsion	τ_t	-9.6	[MPa]
7.11	Max. stress in tension/pressu	σ_g	0.0	[MPa]
7.12	Max. equivalent stress	σ_r	14.1	[MPa]
7.13	Min. static safety	SF_{St}	16.62	
7.14	Min. dynamic safety	SF_D	21.67	
7.15	Critical speed (A)	n_c	253536.3	[/min]
	Critical speed (B)	n_c	270447.4	[/min]
	Critical speed (C)	n_c	253350.6	[/min]

7.17 Graph

01...Z - Shear force [N]

24...Stress in shear - Direction of max. value [°]



Shaft freely rotating in bearings, rotating disc between the bearings (K=1)

7.16 Results for X co-ordinate	40.00	55.00	75.00	135.00	150.00	150.00	150.00	150.00
07...Y - Shear force [N]	-1.131E-26	-1.431E-09	-1.421E-09	1.6617E-09	0	0	0	0
42...Safety coefficient (dynamic)	20	20	20	20	20	20	20	20
31...Total coefficient - bending	1.81265805	2.28039196	2.4201247	2.19111611	1.26390293	1.26390293	1.26390293	1.26390293
42...Safety coefficient (dynamic)	20	20	20	20	20	20	20	20
43...Empty graph	0	0	0	0	0	0	0	0

8.0 Graph - Deflection, Bending angle

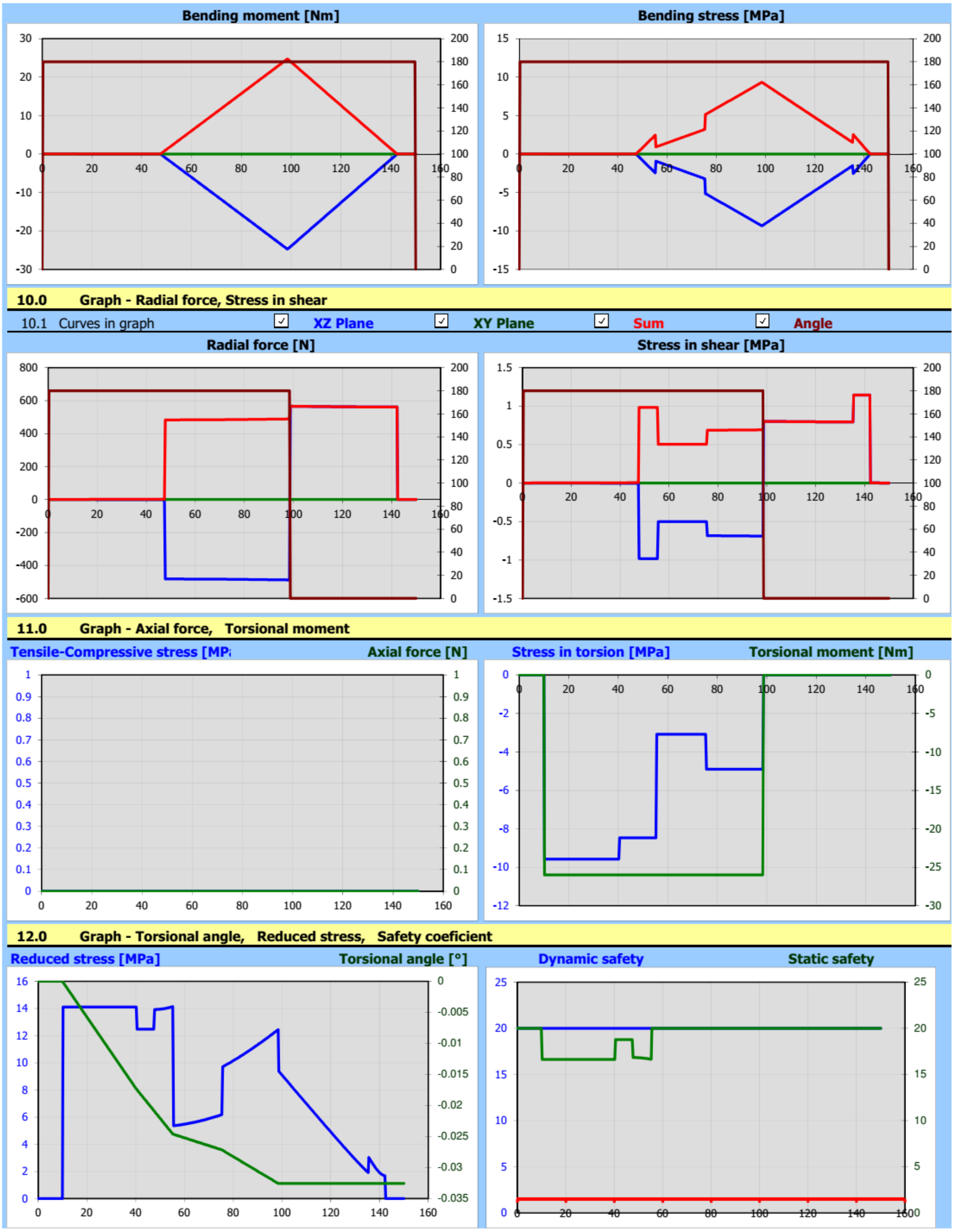
8.1 Curves in graph ☑ XZ Plane ☑ XY Plane ☑ Sum ☑ Angle

Deflection [mm]

Bending angle [°]

9.0 Graph - Bending moment, Bending stress

9.1 Curves in graph ☑ XZ Plane ☑ XY Plane ☑ Sum ☑ Angle



Bending moment [Nm]
Bending stress [MPa]
10.0 Graph - Radial force, Stress in shear
10.1 Curves in graph
XZ Plane
XY Plane
Sum
Angle
Radial force [N]
Stress in shear [MPa]
11.0 Graph - Axial force, Torsional moment
Tensile-Compressive stress [MPa
Axial force [N]
Stress in torsion [MPa]
Torsional moment [Nm]
12.0 Graph - Torsional angle, Reduced stress, Safety coeficient
Reduced stress [MPa]
Torsional angle [°]
Dynamic safety
Static safety

Gear Shaft 2

Shaft design and calculation | **Shaft calculation** | **Clear table of results**

i Calculation without errors.

ii Project information

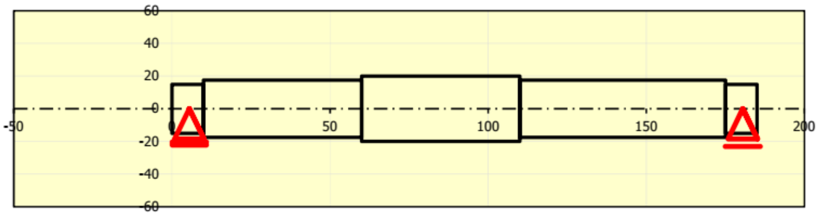
? **Input sectinon**

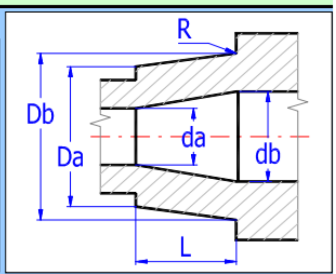
1.0 Preliminary shaft diameter design

1.1	Calculation units	SI Units (N, mm, kW...)		
1.2	Transmitted power	3.76	[kW]	1.6 Type of shaft load
1.3	Shaft speed	332	[/min]	C...Repeated torsion + bending
1.4	Torsion moment	108.16	[Nm]	1.7 Material of the shaft
1.5	Preliminary min. diameter	38.92	[mm]	A...Common structural steel (500)

2.0 Shaft shape and dimensions

2.1 The scale of the displayed shaft diameter. ☑ Calculation units SI Units (N, mm, kW...)





2.2 Table

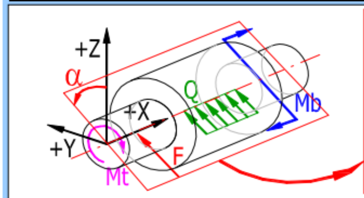
	1	2	3	4	5	6	7	8	9	10
Origin	0.00	10.00	60.00	110.00	175.00	185.00	185.00	185.00	185.00	185.00
L	10.000	50.000	50.000	65.000	10.000					
ø Da	30.000	35.000	40.000	35.000	30.000					
ø Db	30.000	35.000	40.000	35.000	30.000					
ø da	0.000	0.000	0.000	0.000	0.000					
ø db	0.000	0.000	0.000	0.000	0.000					
R	0.500	0.500	0.500	0.500	0.500					

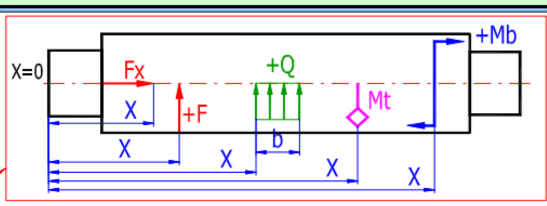
2.3	Total length of the shaft		185.00		[mm]
2.4	X-coordinate of the left support (bearing)	Fixed	5.50	◉	[mm]
2.5	X-coordinate of the right support (bearing)	Free	180.50	○	[mm]

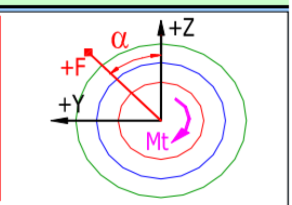
2.6 The shaft surface (Roughness Ra): C...Ground (0.8)

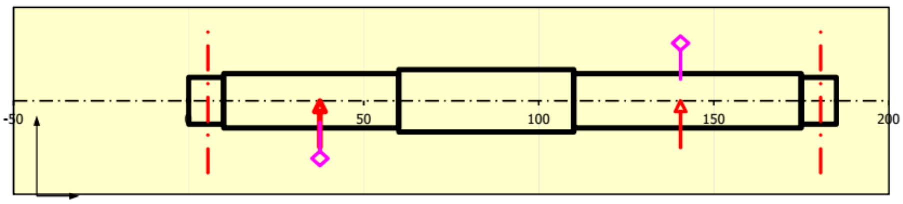
3.0 Notches and necking-down on the shaft

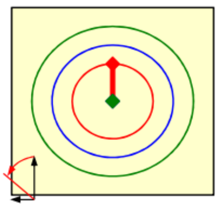
4.0 Loading of the shaft











4.1 Loading

	X	Fx	F	alfa	Mt	Mb	alfa	Q	b	alfa
	[mm]	[N]		[°]	[Nm]		[°]	[N/mm]	[mm]	[°]
1	37.50	0.0	1054.0	0	112.61	0.00	0			
2	140.50	0.0	3413.0	0	-108.24	0.00	0			
3										
4										
5										
6										

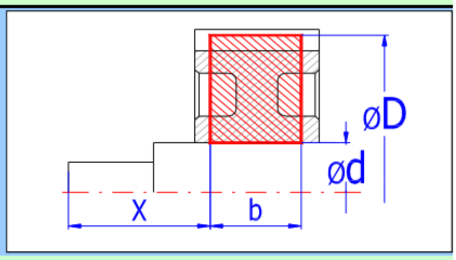
7										
8										
9										
10										

5.0 Rotating masses

5.1 Additional rotating masses (resonance speed)

5.2 Use loading from the weight of disks in the calculation? Yes

5.3	X	D	d	b	Ro	m	
	[mm]				[kg/m^3]	[kg]	
M1	15.00	227.50	35.00	45.00	7800.0	13.93	☑
M2	110.00	67.50	35.00	61.00	7800.0	1.24	☑
M3	0.00	0.00	0.00	0.00	7800.0	0.00	☐
M4	0.00	0.00	0.00	0.00	7800.0	0.00	☐
M5	0.00	0.00	0.00	0.00	7800.0	0.00	☐



6.0 Material and the type of loading

? Results section

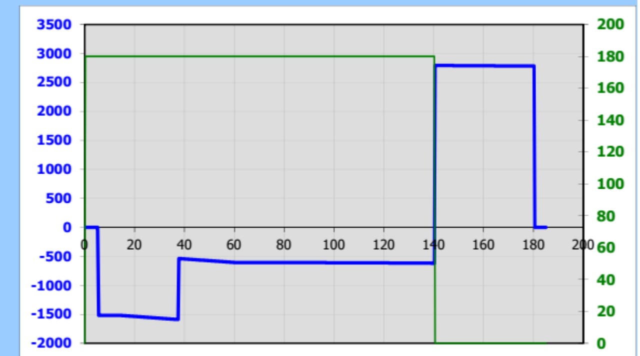
7.0 Results - summary

			x	y	z	Σy+z	
7.1	Reaction in the support R1		0	-6.748E-08	-1519.6648	1519.66475	[N]
7.2	Reaction in the support R2		0	-2.032E-08	-2784.0081	2784.00814	[N]
7.3	Total shaft weight	m		1.47	[kg]		
7.4	Maximum deflection	y		0.0155	[mm]		
7.5	Maximum torsional deflectio	φ		0.0461	[°]		
7.6	Angular deflection in R1	ϑ		0.0155	[°]		
7.7	Angular deflection in R2	ϑ		0.0204	[°]		
7.8	Max. bending stress	σ_e		26.5	[MPa]		
7.9	Max. stress in shear	τ_s		3.9	[MPa]		
7.10	Max. stress in torsion	τ_t		13.4	[MPa]		
7.11	Max. stress in tension/pressu	σ_g		0.0	[MPa]		
7.12	Max. equivalent stress	σ_r		33.6	[MPa]		
7.13	Min. static safety	SF_{St}		7.64			
7.14	Min. dynamic safety	SF_D		5.03			
7.15	Critical speed (A)	n_c		52114.6	[/min]		
	Critical speed (B)	n_c		51617.6	[/min]		
	Critical speed (C)	n_c		42866.8	[/min]		

7.17 Graph

01...Z - Shear force [N]

24...Stress in shear - Direction of max. value [°]



Shaft freely rotating in bearings, rotating disc between the bearings (K=1)

7.16 Results for X co-ordinate	40.00	55.00	75.00	135.00	150.00	150.00	150.00	150.00
07...Y - Shear force [N]	-2.255E-08	3.8721E-09	1.3121E-08	1.6046E-08	1.7836E-08	1.7836E-08	1.7836E-08	1.7836E-08
42...Safety coefficient (dynamic)	14.5831807	13.0162205	16.7245982	7.82748342	10.1639362	10.1639362	10.1639362	10.1639362
31...Total coefficient - bending	1.35869565	1.35869565	1.35869565	1.35869565	1.35869565	1.35869565	1.35869565	1.35869565
42...Safety coefficient (dynamic)	14.5831807	13.0162205	16.7245982	7.82748342	10.1639362	10.1639362	10.1639362	10.1639362
43...Empty graph	0	0	0	0	0	0	0	0

8.0 Graph - Deflection, Bending angle

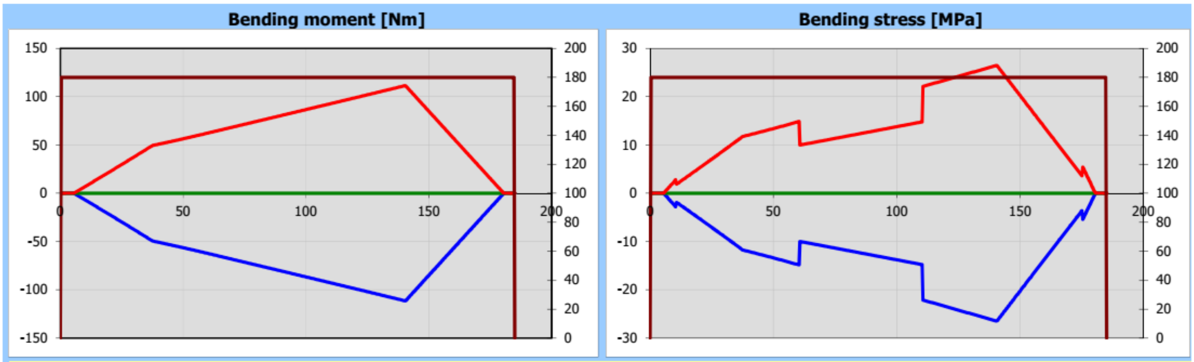
8.1 Curves in graph ☑ XZ Plane ☑ XY Plane ☑ Sum ☑ Angle

Deflection [mm]

Bending angle [°]

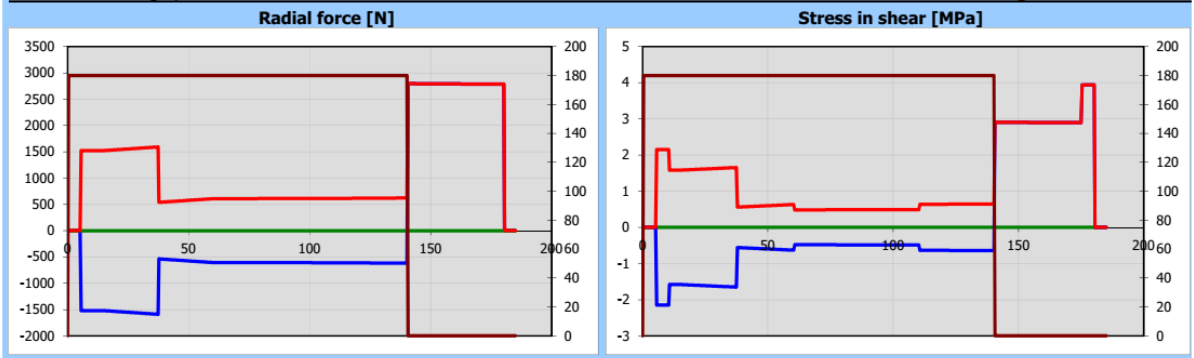
9.0 Graph - Bending moment, Bending stress

9.1 Curves in graph ☑ XZ Plane ☑ XY Plane ☑ Sum ☑ Angle

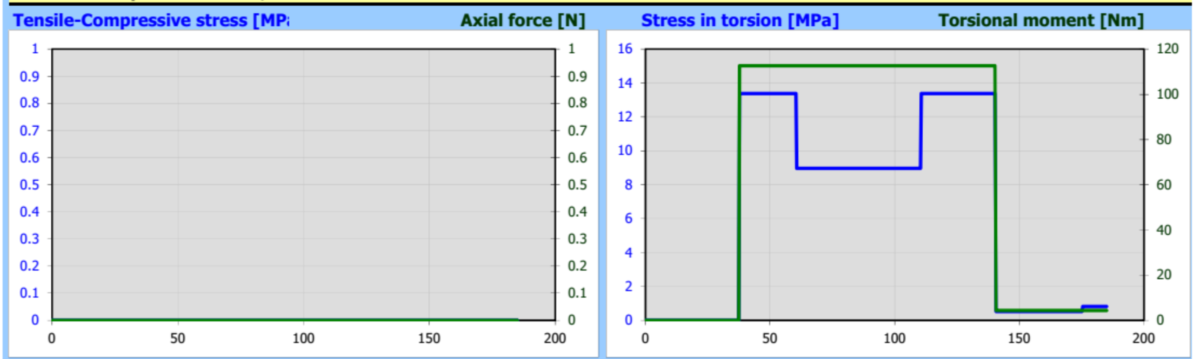


10.0 Graph - Radial force, Stress in shear

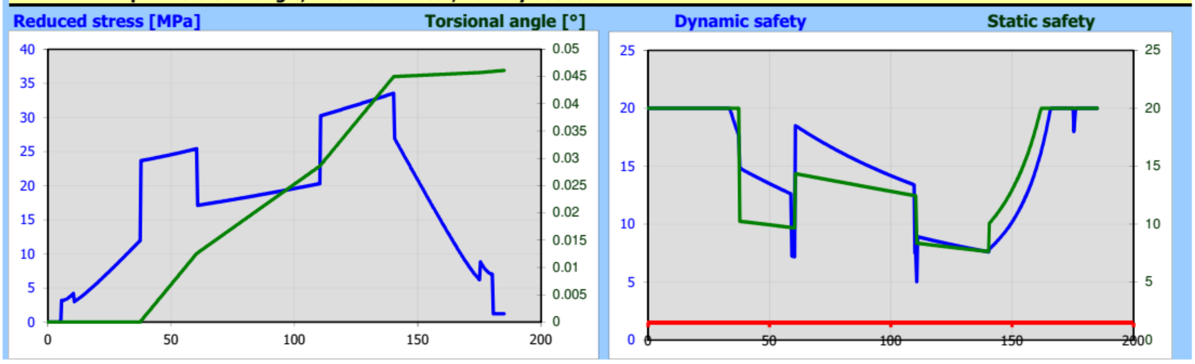
10.1 Curves in graph ☑ **XZ Plane** ☑ **XY Plane** ☑ **Sum** ☑ **Angle**



11.0 Graph - Axial force, Torsional moment



12.0 Graph - Torsional angle, Reduced stress, Safety coeficient



Gear Shaft 3

Shaft design and calculation | **Shaft calculation** | **Clear table of results**

i **Check lines:7.5;**

ii **Project information**

? **Input sectinon**

1.0 Preliminary shaft diameter design

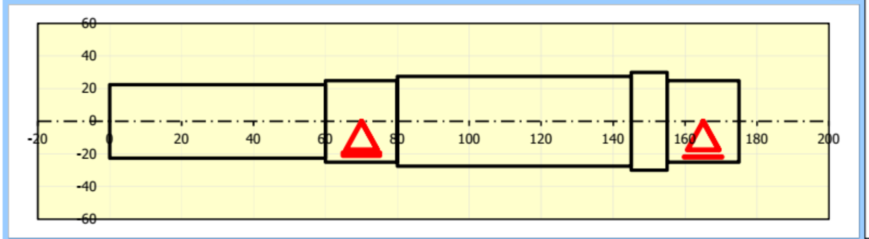
1.1	Calculation units	SI Units (N, mm, kW...)	
1.2	Transmitted power	3.60	[kW]
1.3	Shaft speed	76	[/min]
1.4	Torsion moment	452.37	[Nm]
1.5	Preliminary min. diameter	62.40	[mm]

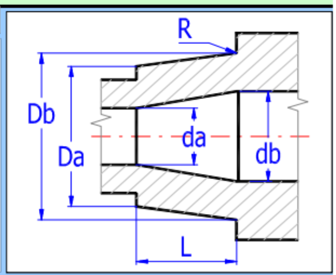
1.6 Type of shaft load: C...Repeated torsion + bending

1.7 Material of the shaft: A...Common structural steel (500)

2.0 Shaft shape and dimensions

2.1 The scale of the displayed shaft diameter. ☑ Calculation units: SI Units (N, mm, kW...)





2.2 Table

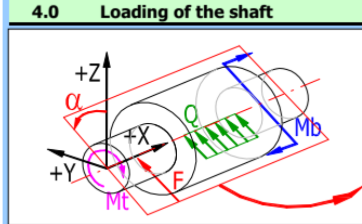
	1	2	3	4	5	6	7	8	9	10
Origin	0.00	60.00	80.00	145.00	155.00	175.00	175.00	175.00	175.00	175.00
L	60.000	20.000	65.000	10.000	20.000					
ø Da	45.000	50.000	55.000	60.000	50.000					
ø Db	45.000	50.000	55.000	60.000	50.000					
ø da	0.000	0.000	0.000	0.000	0.000					
ø db	0.000	0.000	0.000	0.000	0.000					
R	0.500	0.500	0.500	0.500	0.500					

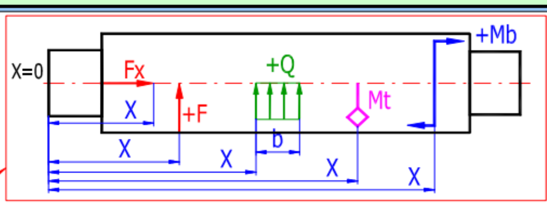
2.3	Total length of the shaft		175.00	[mm]
2.4	X-coordinate of the left support (bearing)	Fixed	70.00	[mm]
2.5	X-coordinate of the right support (bearing)	Free	165.00	[mm]

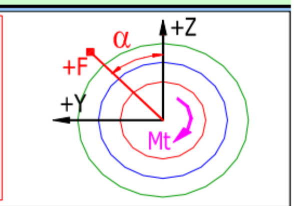
2.6 The shaft surface (Roughness Ra): C...Ground (0.8)

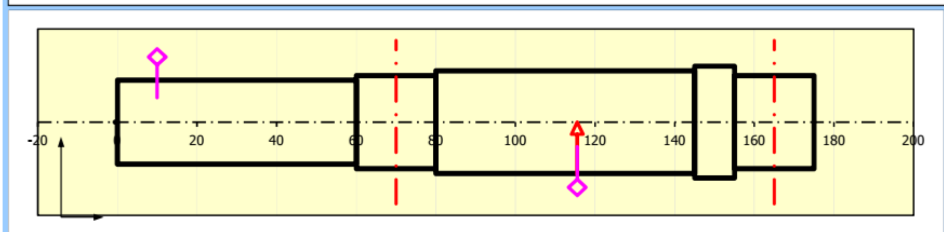
3.0 Notches and necking-down on the shaft

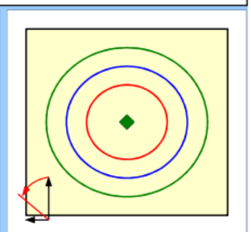
4.0 Loading of the shaft











4.1 Loading

	X	Fx	F	alfa	Mt	Mb	alfa	Q	b	alfa
	[mm]	[N]		[°]	[Nm]		[°]	[N/mm]	[mm]	[°]
1	10.00	0.0	0.0	0	-469.00	0.00	0			
2	115.50	0.0	3413.0	0	469.00	0.00	0			
3										
4										
5										
6										

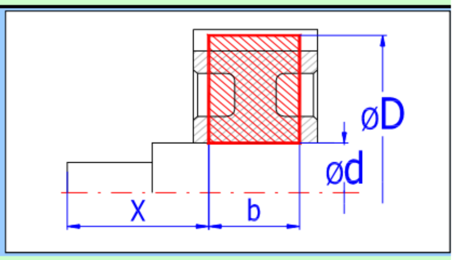
	7									
	8									
	9									
	10									

5.0 Rotating masses

5.1 Additional rotating masses (resonance speed)

5.2 Use loading from the weight of disks in the calculation? Yes

5.3	X	D	d	b	Ro	m	
		[mm]			[kg/m^3]	[kg]	
M1	86.00	292.50	55.00	59.00	7800.0	29.83	☑
M2	0.00	0.00	35.00	61.00	7800.0	1.24	☐
M3	0.00	0.00	0.00	0.00	7800.0	0.00	☐
M4	0.00	0.00	0.00	0.00	7800.0	0.00	☐
M5	0.00	0.00	0.00	0.00	7800.0	0.00	☐



6.0 Material and the type of loading

? Results section

7.0 Results - summary

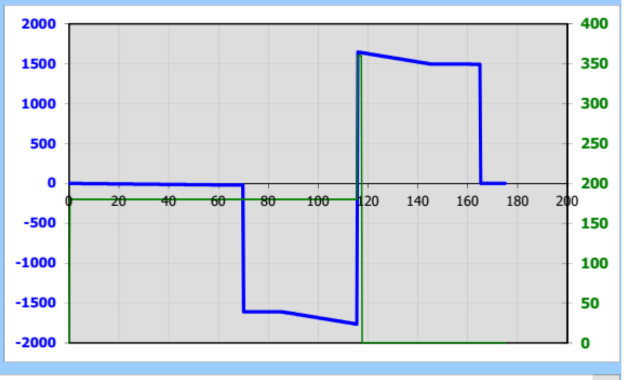
			x	y	z	Σ y+z	
7.1	Reaction in the support R1		0	-1.001E-07	-1588.207	1588.20696	[N]
7.2	Reaction in the support R2		0	-7.968E-08	-1492.454	1492.45399	[N]

7.17 Graph

01...Z - Shear force [N]

24...Stress in shear - Direction of max. value [°]

7.3	Total shaft weight	m	2.80	[kg]
7.4	Maximum deflection	y	0.0014	[mm]
7.5	Maximum torsional deflectio	φ	0.0662	[°]
7.6	Angular deflection in R1	ϑ	0.0011	[°]
7.7	Angular deflection in R2	ϑ	0.0011	[°]
7.8	Max. bending stress	σ_e	4.7	[MPa]
7.9	Max. stress in shear	τ_s	0.8	[MPa]
7.10	Max. stress in torsion	τ_t	-26.2	[MPa]
7.11	Max. stress in tension/pressu	σ_g	0.0	[MPa]
7.12	Max. equivalent stress	σ_r	38.6	[MPa]
7.13	Min. static safety	SF_{St}	6.07	
7.14	Min. dynamic safety	SF_D	9.92	
7.15	Critical speed (A)	n_c	143854.1	[/min]
	Critical speed (B)	n_c	144146.5	[/min]
	Critical speed (C)	n_c	109780.5	[/min]



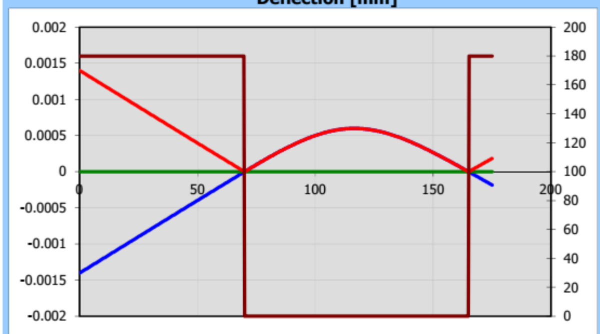
Shaft freely rotating in bearings, rotating disc between the bearings (K=1)

7.16 Results for X co-ordinate	40.00	55.00	75.00	135.00	150.00	150.00	150.00	150.00
07...Y - Shear force [N]	4.7126E-09	6.4824E-09	-9.292E-08	5.0061E-08	7.9678E-08	7.9678E-08	7.9678E-08	7.9678E-08
42...Safety coefficient (dynamic)	14.0940404	14.0902802	17.6073731	20	20	20	20	20
31...Total coefficient - bending	1.44927536	1.44927536	1.44927536	1.53092468	1.53092468	1.53092468	1.53092468	1.53092468
42...Safety coefficient (dynamic)	14.0940404	14.0902802	17.6073731	20	20	20	20	20
43...Empty graph	0	0	0	0	0	0	0	0

8.0 Graph - Deflection, Bending angle

8.1 Curves in graph ☑ **XZ Plane** ☑ **XY Plane** ☑ **Sum** ☑ **Angle**

Deflection [mm]



Bending angle [°]

9.0 Graph - Bending moment, Bending stress

9.1 Curves in graph ☑ **XZ Plane** ☑ **XY Plane** ☑ **Sum** ☑ **Angle**

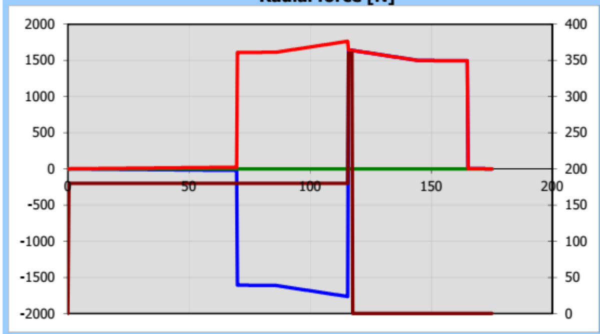
Bending moment [Nm]

Bending stress [MPa]

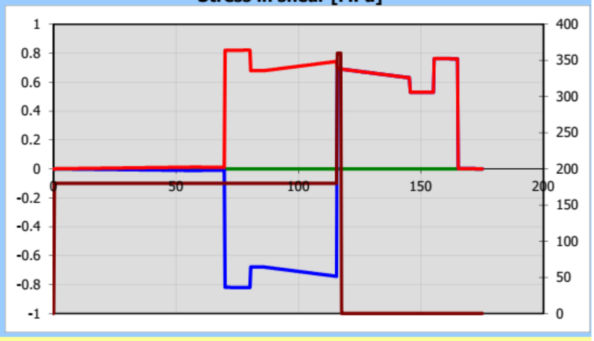
10.0 Graph - Radial force, Stress in shear

10.1 Curves in graph ☑ **XZ Plane** ☑ **XY Plane** ☑ **Sum** ☑ **Angle**

Radial force [N]

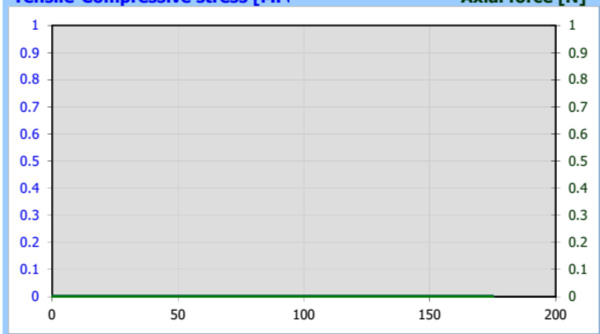


Stress in shear [MPa]

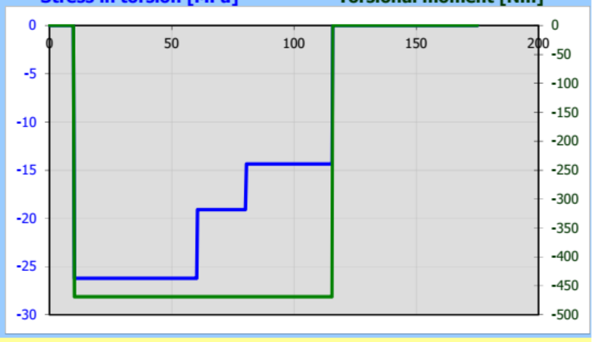


11.0 Graph - Axial force, Torsional moment

Tensile-Compressive stress [MPa] / Axial force [N]

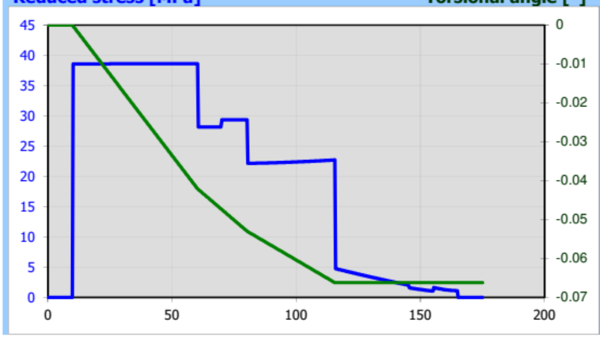


Stress in torsion [MPa] / Torsional moment [Nm]

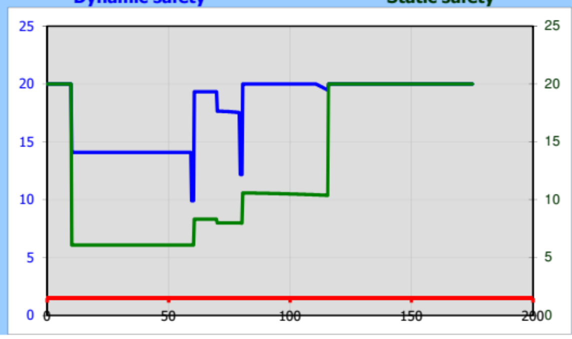


12.0 Graph - Torsional angle, Reduced stress, Safety coeficient

Reduced stress [MPa] / Torsional angle [°]



Dynamic safety / Static safety



10 Selection of bearings by dynamical loads

According to calculations reactions in bearings are acquired as below.

First Shaft:

	x	y	z	Σ y+z
Reaction in the support R1	0	-1.431E-09	-480.87936	480.879364
Reaction in the support R2	0	-1.662E-09	-560.58863	560.588632

Theatrically calculated max value $K_{c1} = 560.6$

Rotational speed of shaft is $n_1 = 1440$

Second Shaft:

	x	y	z	Σ y+z
Reaction in the support R1	0	-6.748E-08	-1519.6648	1519.66475
Reaction in the support R2	0	-2.032E-08	-2784.0081	2784.00814

Theatrically calculated max value $K_{c2} = 2784.0$

Rotational speed of shaft is $n_2 = 332$

Third Shaft:

	x	y	z	Σ y+z
Reaction in the support R1	0	-1.001E-07	-1588.207	1588.20696
Reaction in the support R2	0	-7.968E-08	-1492.454	1492.45399

Theatrically calculated max value $K_{c3} = 1588.2$

Rotational speed of shaft is $n_3 = 76$

10.1 Actual Shaft loads

Shock value for the project is medium so 1.3 is chosen for load factor $f_w = 1.3$ $(Table\ 4.1(Jurkonis\ E., n.d.))$ using formula.

$$K = f_w . K_c \qquad 10\text{-}1$$

We will have

$$K_1 = f_w . K_{c1} = 1.3 * 560.6 = 728.8\ N$$

$$K_2 = f_w . K_{c2} = 1.3 * 2784.0 = 3619.2\ N$$

$$K_3 = f_w . K_{c3} = 1.3 * 1588.2 = 2064.7\ N$$

10.2 Dynamic Equivalent Load

By using following formula,

$$P_r = V\,X\,F_r + YF_a \qquad \text{10-2}$$

In all cases $F_a = 0$, because all used gearings is spur gearing.
$V = 1$

From the table: $X = 1$

$$P_1 = K_1 * 1 * 1 = 728.8\ N$$
$$P_2 = K_2 * 1 * 1 = 3619.2\ N$$
$$P_3 = K_3 * 1 * 1 = 2064.7\ N$$

10.3 Bearing Life

From (Jurkonis E. , n.d.) we also have formula for Bearing Life:

$$L_{10} = \left(\frac{C}{P}\right)^3 \qquad \text{10-3}$$

For selected bearings we will have,

$$L_{10_1} = \left(\frac{C_1}{P_1}\right)^3 = \left(\frac{7000}{728.8}\right) = 889$$
$$L_{10_2} = \left(\frac{C_2}{P_2}\right)^3 = \left(\frac{16500}{3619.2}\right) = 94.75$$
$$L_{10_3} = \left(\frac{C_3}{P_3}\right)^3 = \left(\frac{22900}{2064.7}\right) = 1364$$

From here, we see that for all cases $L10$ is enough Some of Life hours big because dynamic load of bearings is more than sufficient. This type and size of bearings was chosen to fit the bore of the shaft, which was calculated to be at certain diameter due to occurring stresses.

10.4 Safety Factor

From (Jurkonis E. , n.d.) we have formula for safety factor:

$$S_0 = \frac{C_0}{P_0} \qquad \text{10-4}$$

We will find that for selected bearings,

$$S_1 = \frac{C_1}{P_1} = \frac{4300}{728.8} = 5.9$$

$$S_2 = \frac{C_2}{P_2} = \frac{11200}{3619.2} = 3.09$$

$$S_3 = \frac{C_3}{P_3} = \frac{15600}{2064.7} = 7.55$$

Minimum safety factor value $S \geq 2$
Calculations satisfies safety factor requirement.

11 Selection of bearing tolerances

The load rotates with respect to inner ring of bearing, thus it should fit in transition fit **(H7/j6)**. Bearings are subjected primarily to a radial load.

Warming – up during the operation can occur, especially in the first shaft that rotates at 292 radians per second speed, therefore differences in distance should be compensated. In our case, we have deep groove ball bearings that can function as floating bearings when one bearing ring is provided with loose fit. (Jurkonis E., n.d.) . The ring under point load is therefore given a loose fit. Therefore, outer ring will be given a loose fit **(H7/h6).**

12 Selection and control calculations of key and spline joints, other fasteners

Square key joint

From (Jurkonis E. , n.d.) we will have:

$$\tau = \frac{F}{A} = \frac{2T}{DWL} \quad or \; L \geq \frac{2T}{\tau WD} \qquad \textit{12-1}$$

By following this formula and tables indicated in course slides following keys can be found.

First Shaft

Key b×h: 10×8,

Shaft t1: 5 ,

Hub t2: 3.3 ,

D=30mm,

T=26 N.m,

$$L \geq \frac{2T}{\tau WD} = \frac{2\ \times 26}{20\ \times 10\ \times 30} = 8.7\ mm$$

Chosen **L=36 mm**

Second Shaft

Key b×h: 10×8,

Shaft t1: 5 ,

Hub t2: 3.3 ,

D=35mm,

T=112.6 N.m,

$$L \geq \frac{2T}{\tau WD} = \frac{2\ \times 112.6}{20\ \times 10\ \times 35} = 32.1\ mm$$

Chosen **L=36 mm**

Key b×h: 10×8,
Shaft t1: 5 ,
Hub t2: 3.3 ,
D=35mm,
T=108.2 N.m,

$$L \geq \frac{2T}{\tau WD} = \frac{2 \times 108.2}{20 \times 10 \times 35} = 31.0\, mm$$

Chosen **L=50 mm**

Third Shaft
Key b×h: 16×10,
Shaft t1: 6 ,
Hub t2: 4.3 ,
D=55mm,
T=469 N.m,

$$L \geq \frac{2T}{\tau WD} = \frac{2 \times 469}{20 \times 16 \times 55} = 53.3\, mm$$

Chosen **two** sides each has **L=45 mm**

13 Selection of tolerances, roughness, shape variations for surfaces of elements

Material will be hot rolled. Surface roughness of hot rolled material varies from 7 to 100 micrometers. Shaft will be not spinning more than 15000 rpm, therefore there is no need to use interference fit - we will use transition fit for shafts and inner ring:

For first shaft (d25): H7/j6
Therefore hole's diameter tolerance +0.021mm 0mm
Shaft's diameter tolerance: 0.009mm -0.004mm

For second shaft (d30): H7/j6
Therefore hole's diameter tolerance +0.021mm 0mm
Shaft's diameter tolerance: 0.009mm -0.004mm

For third shaft (d30): H7/j6

Therefore hole's diameter tolerance +0.025mm 0mm

Shaft's diameter tolerance: 0.011mm -0.005mm

14 Selection of lubrication types for drives and bearings

Spur gears will be spinning from 76 rpm to 1440 rpm, with load from 1 kN up to 3.4 kN. According to the table found in (Ludwig, n.d.) article, spur gear requires Low slide, low speed lubricant.

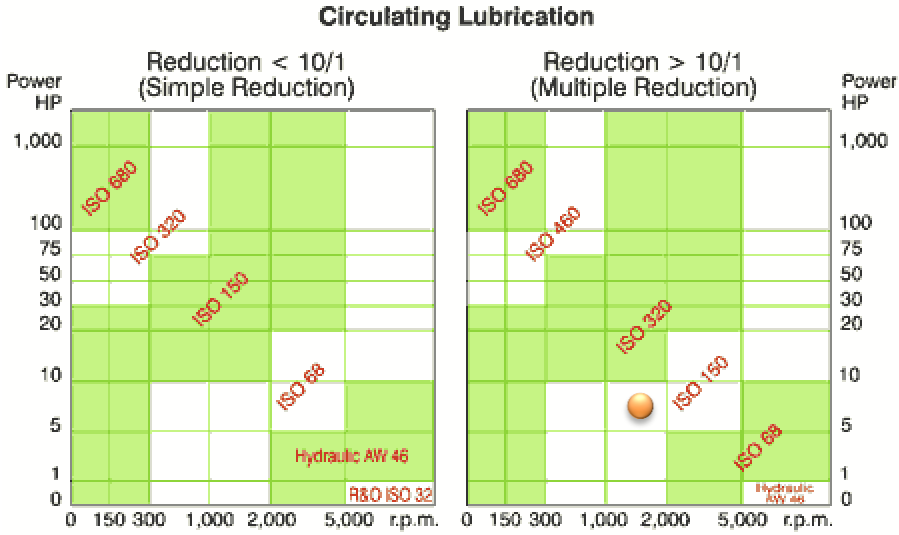
Loading is not as big as other highly loaded industrial gear drives, so they will not require of extreme pressure gear lubricants.

Surface is smoother in these gearings, so it will require lower viscosity oil. Transmitted power is 4kW (5.3 HP).

Gear speed is only high in first shaft, in others it is lower so according to speed oil viscosity also should be lower.

Material used in production is common structural steels, so it will also require light viscosity oil. According to:

https://www.machinerylubrication.com/Read/707/enclosed-gear-drives



The working point is shown on figure so lubricant should fall into **ISO 150** Lubricant will be mineral for environmental reasons.

https://www.grainger.com/product/MOBIL-Mineral-Circulating-Oil-6Y782

15 Selection of box dimensions

200511350

Material for box production: Iron alloy, without heat treatment. Length of the box is 511mm, Width 200mm and height 350mm. According to formula found in (Jurkonis E. , n.d.) slides.

$$Dim = \frac{2L + W + H}{3} \qquad 15\text{-}1$$

Dim will be 0.5m according to the following table, Thickness of walls will be 7 mm, except in places of bigger shafts bearings it will be wider. Box will consist of two parts: upper and lower

Dim, m	0.05	0.15	0.3	0.7	1	1.5	2	3
Thikness,mm	4	5	6	8	10	12	15	20

16 Specifications

The drive housing is waterproof and will be manufactured from cast iron and mounted on a base plate. Access into the housing will be through a screwed points.

The drive system will be through a three phase 400V, 50Hz.

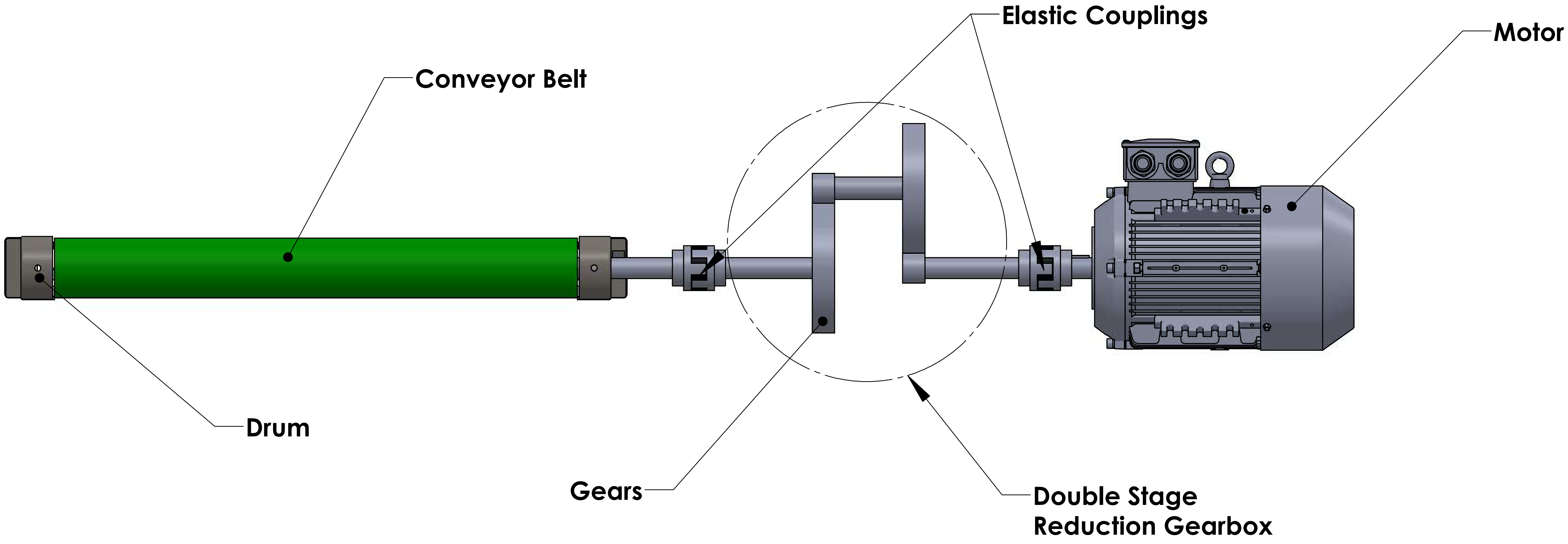
The oil shall be mineral R&O ISO 150.

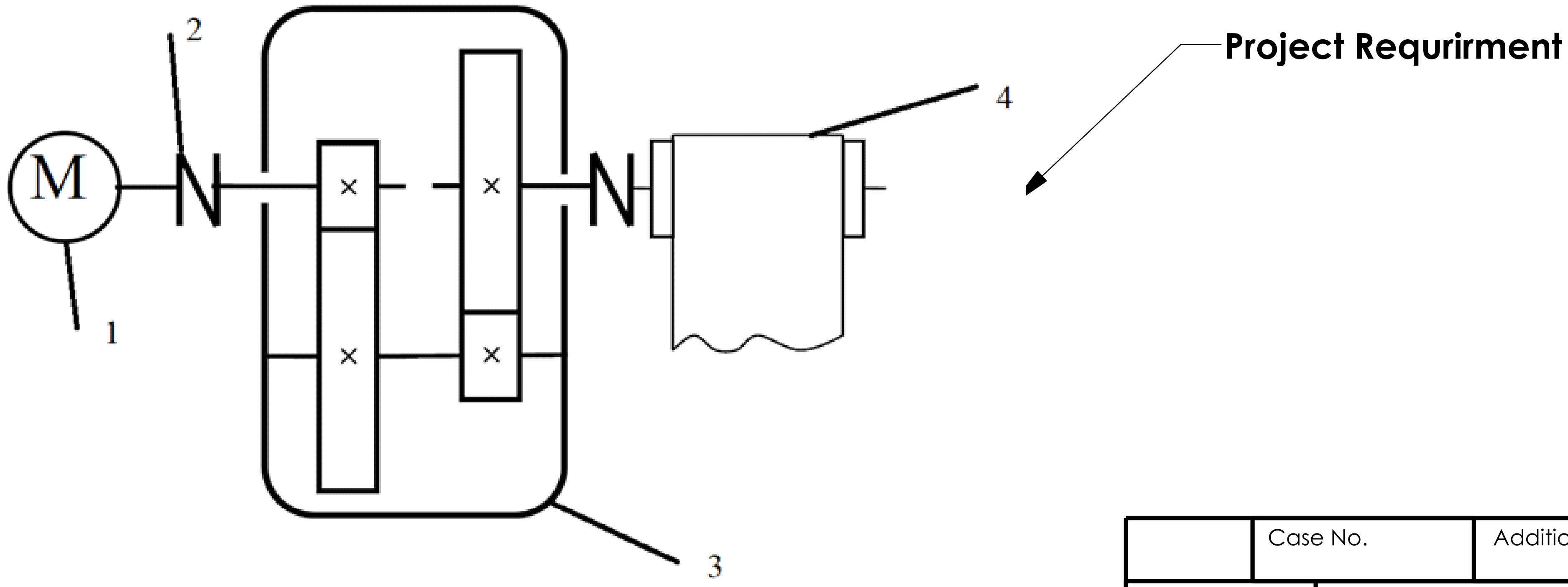
Cost to purchase: 1200Eur.

Cost to use: 2.73 ¢/hour using 4kW motor, when electricity price is 0,683 ¢/kWh (ESO, n.d.) in Lithuania.

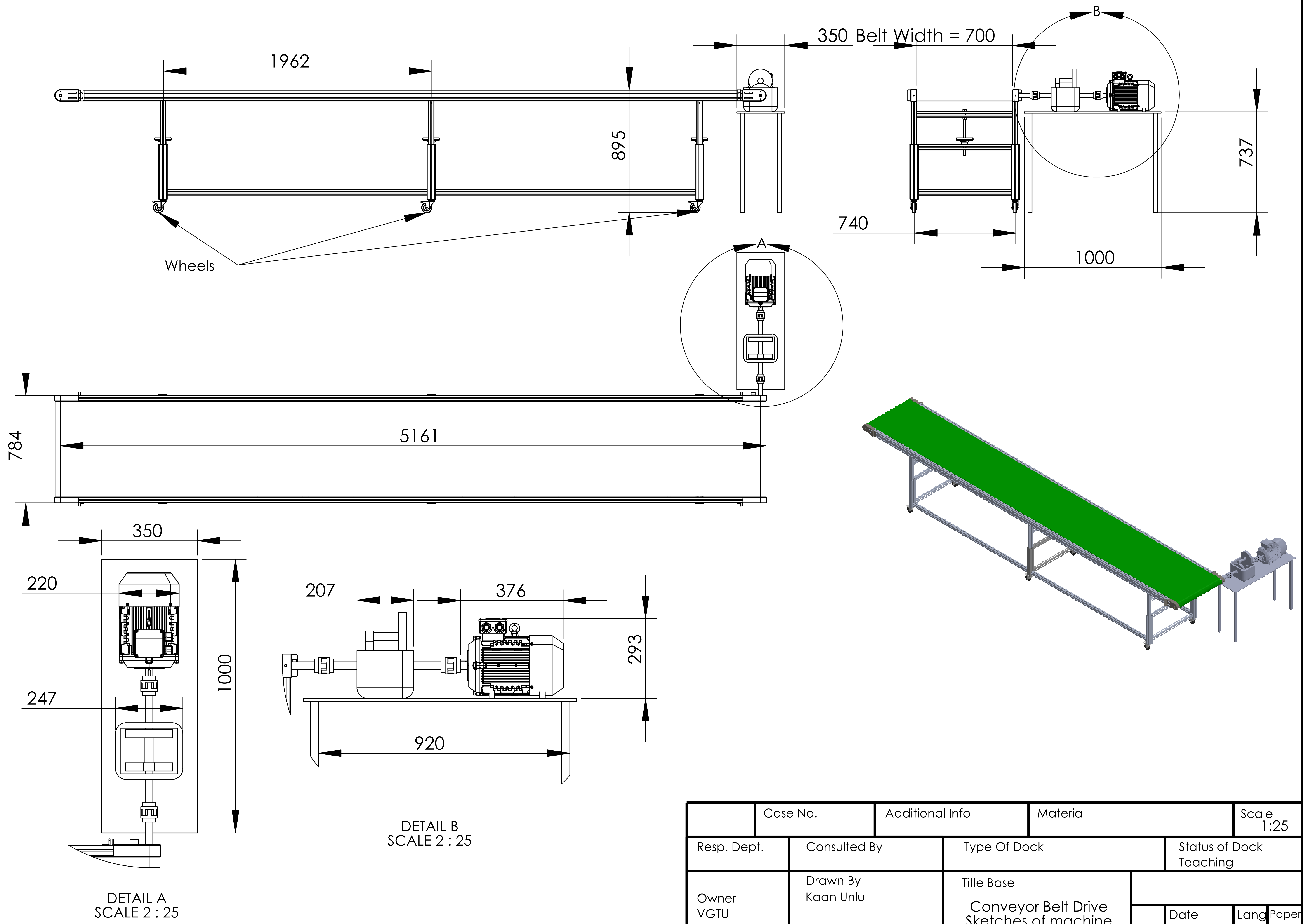
Cost to repair: Motor 200euros, bearings 15 euros.

Color: Blue Cast Iron

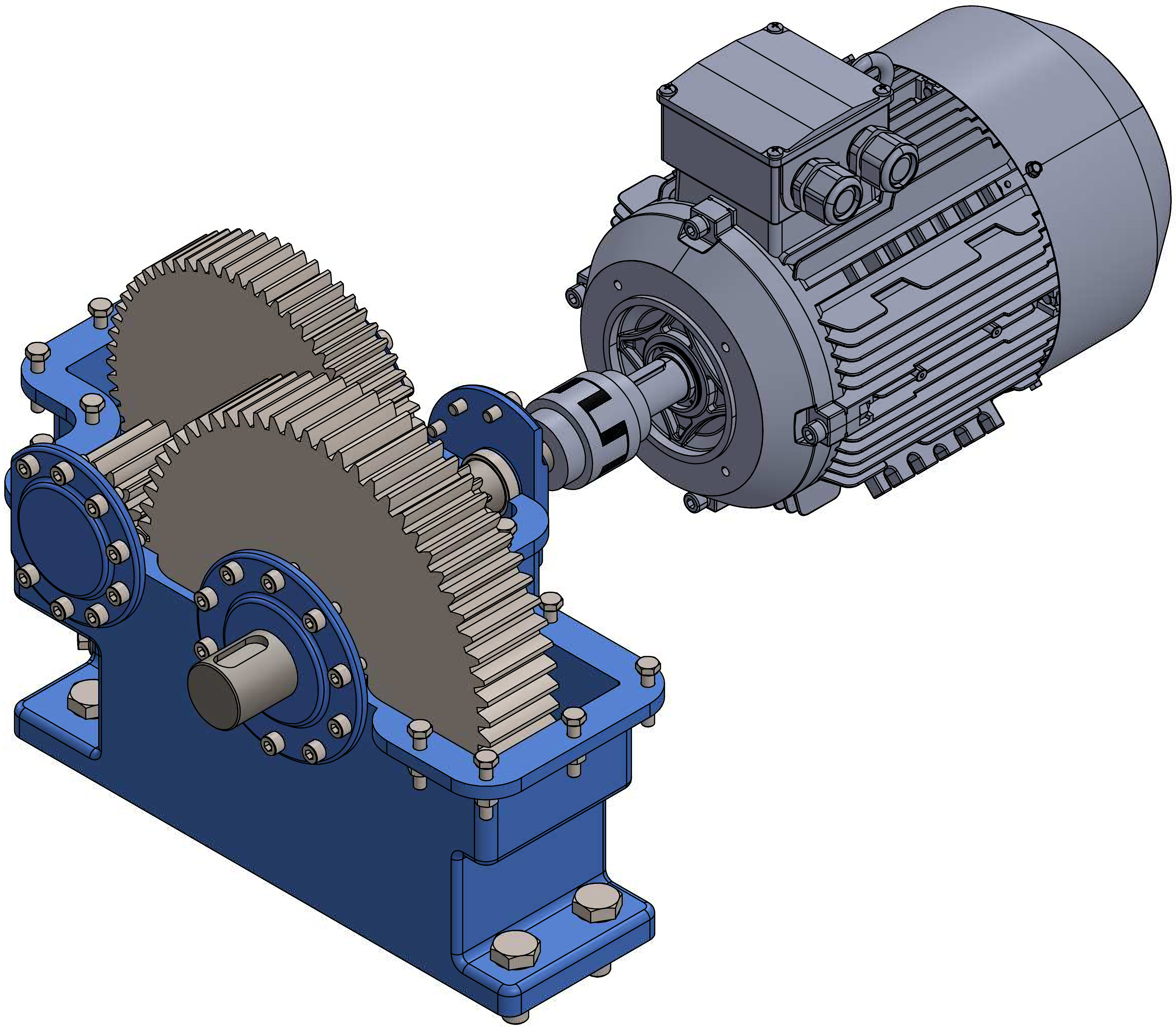




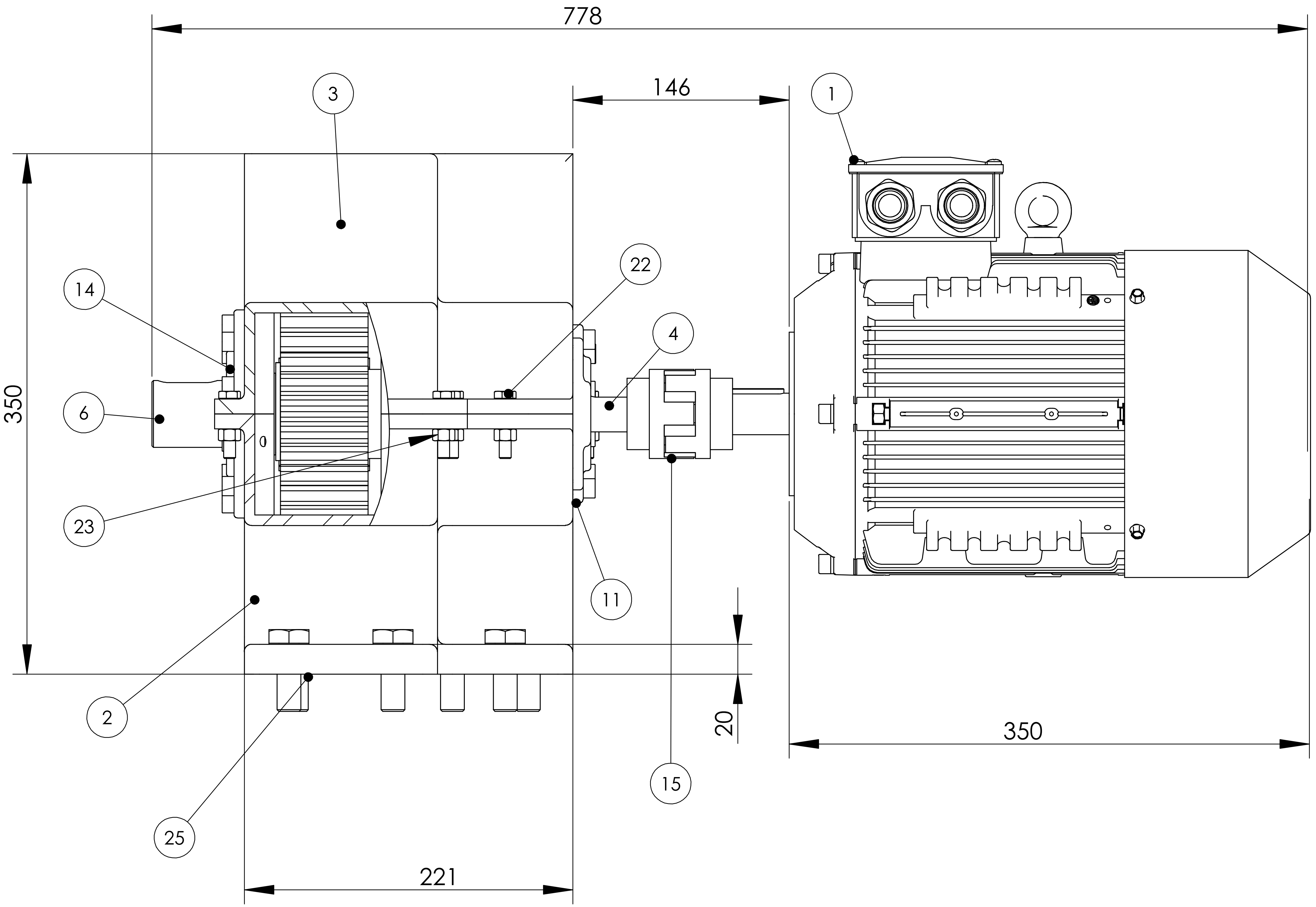
	Case No.	Additional Info	Material	Scale 1:6
Resp. Dept.	Consulted By	Type Of Dock	Status of Dock Teaching	
Owner VGTU	Drawn By Kaan Unlu	Title Base Conveyor Belt Drive Kinematical scheme	Date 2021-19-03	Lang Eng / Paper 1/2



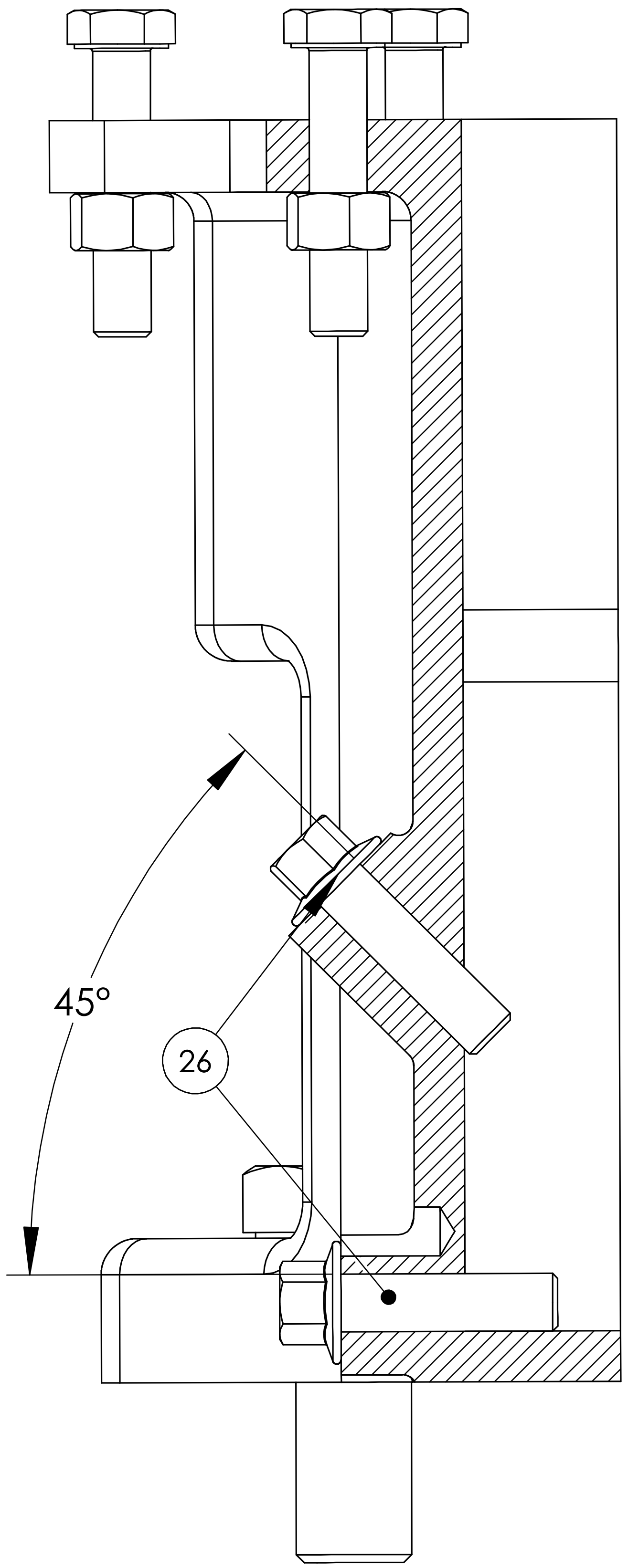
	Case No.	Additional Info	Material	Scale 1:25
Resp. Dept.	Consulted By	Type Of Dock	Status of Dock Teaching	
Owner VGTU	Drawn By Kaan Unlu	Title Base Conveyor Belt Drive Sketches of machine	Date 2021-19-03	Lang Eng / Paper 2/2



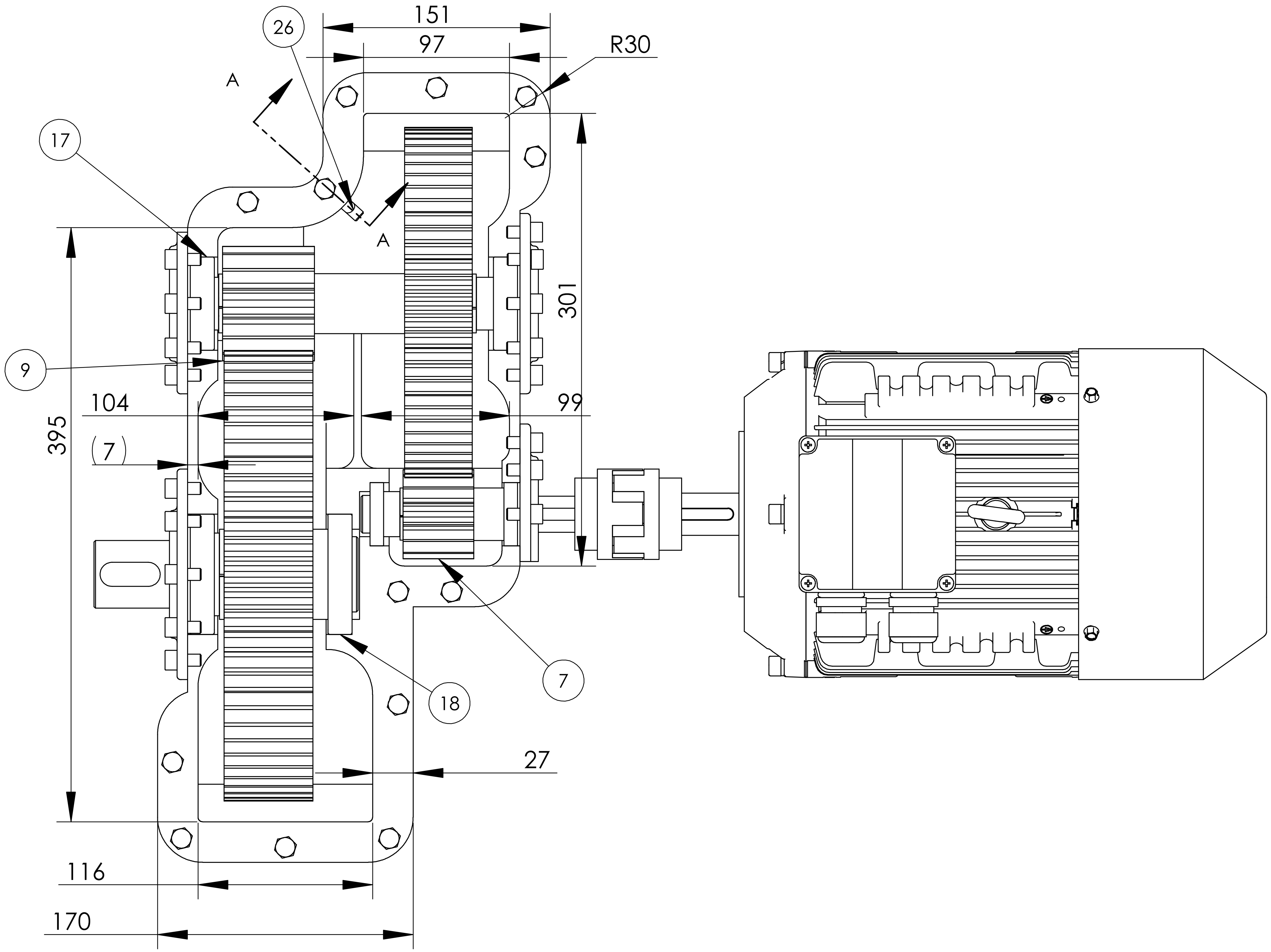
	Case No.	Additional Info	Material	Scale 1:3
Resp. Dept.	Consulted By	Type Of Dock		Status of Dock Teaching
Owner VGTU	Drawn By Kaan Unlu	Title Base Conveyor Belt Drive	Date 2021-19-03	Lang Eng / Paper 1/11



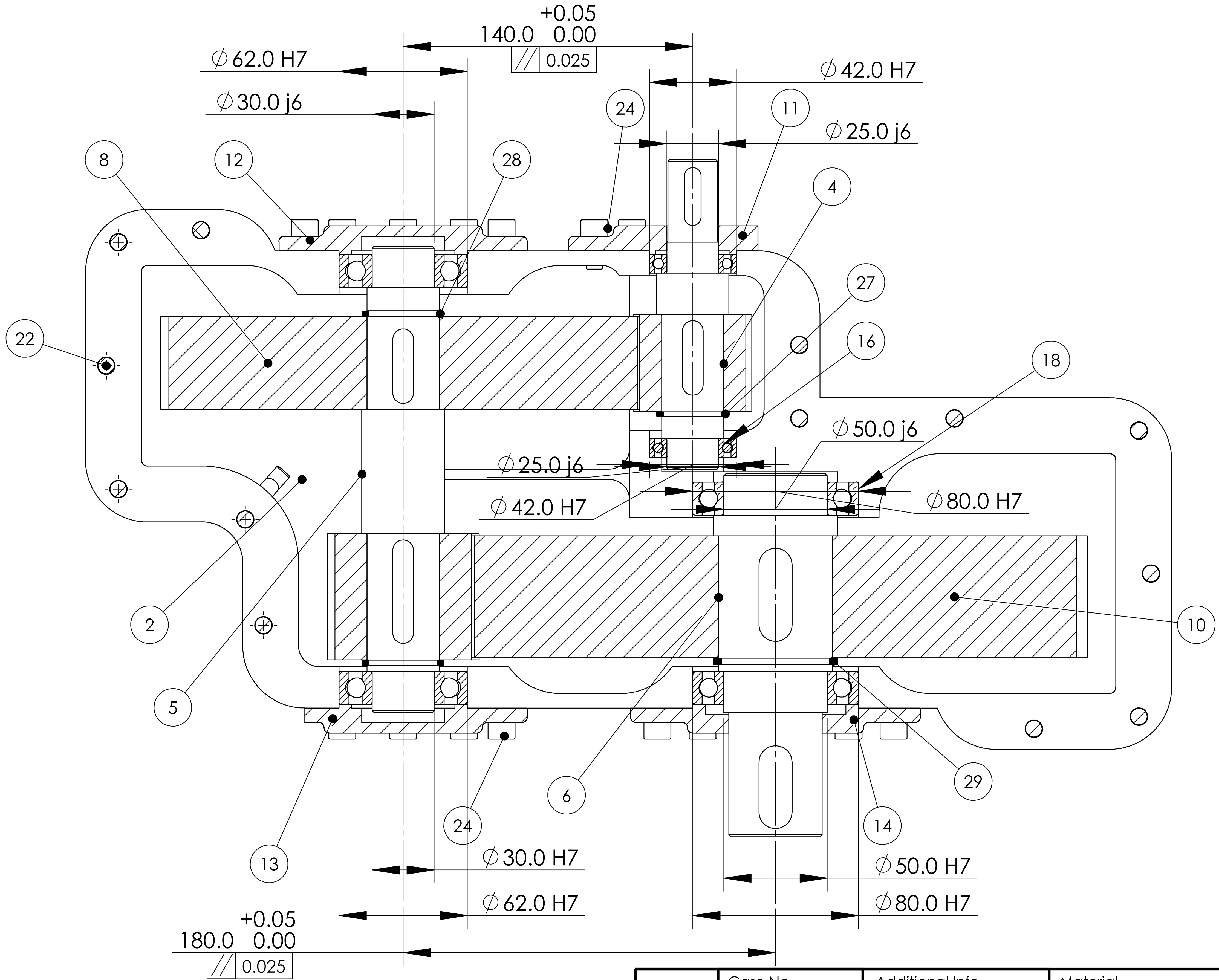
	Case No.	Additional Info	Material	Scale 1:3
Resp. Dept.	Consulted By	Type Of Dock		Status of Dock Teaching
Owner VGTU	Drawn By Kaan Unlu	Title Base Motor and Housing	Date 2021-19-03	Lang Eng	Paper 2/11



SECTION A-A
SCALE 1 : 1



	Case No.	Additional Info	Material	Scale 1:3
Resp. Dept.	Consulted By	Type Of Dock		Status of Dock Teaching
Owner VGTU	Drawn By Kaan Unlu	Title Base Motor and Reducer Oil leveler		Date 2021-19-03 / Lang Eng / Paper 3/11

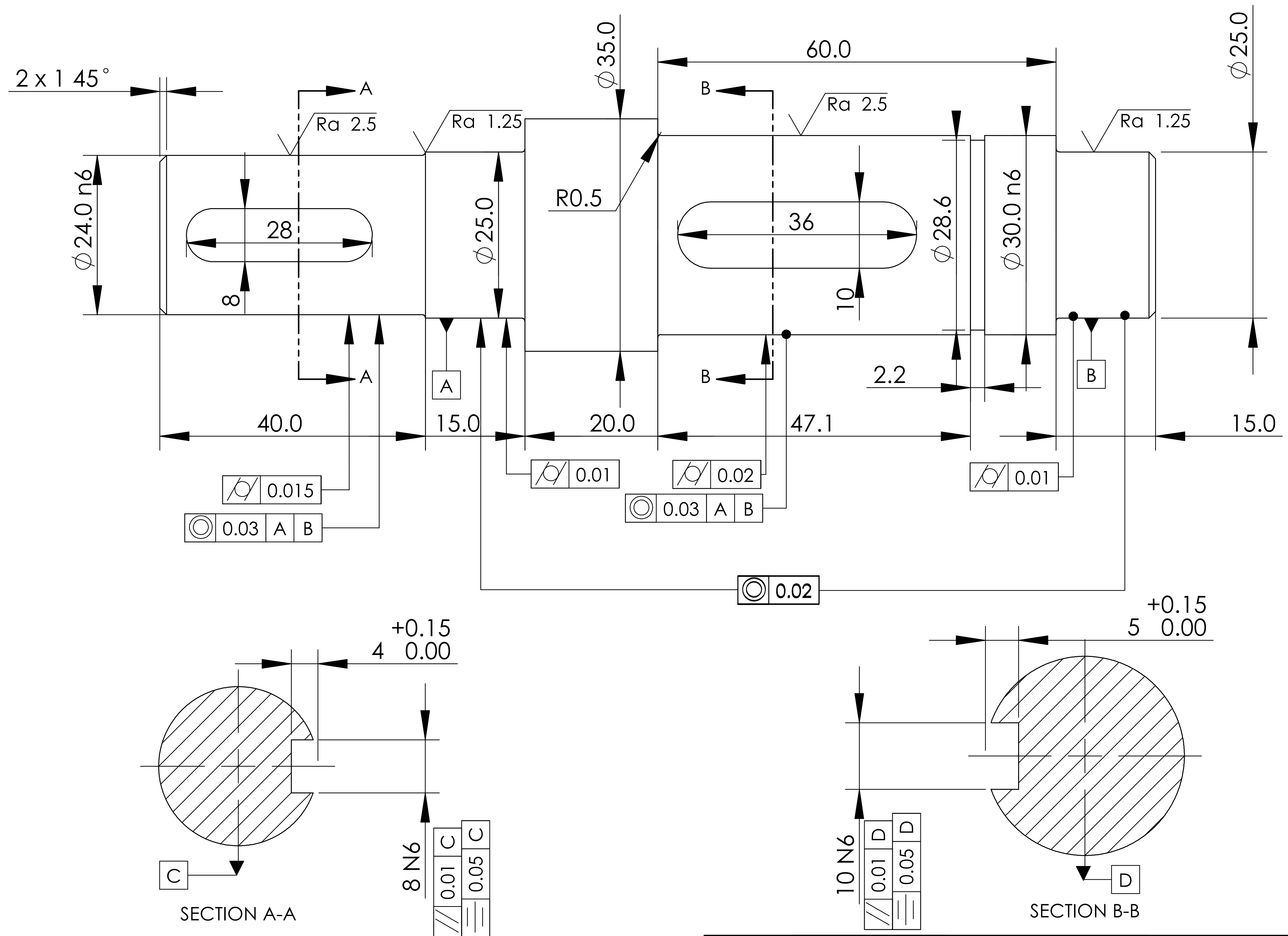


	Case No.	Additional Info	Material	Scale 1:2
Resp. Dept.	Consulted By	Type Of Dock	Status of Dock Teaching	
Owner VGTU	Drawn By Kaan Unlu	Title Base Reducer Schematic	Date 2021-19-03	Lang Eng	Paper 4/11

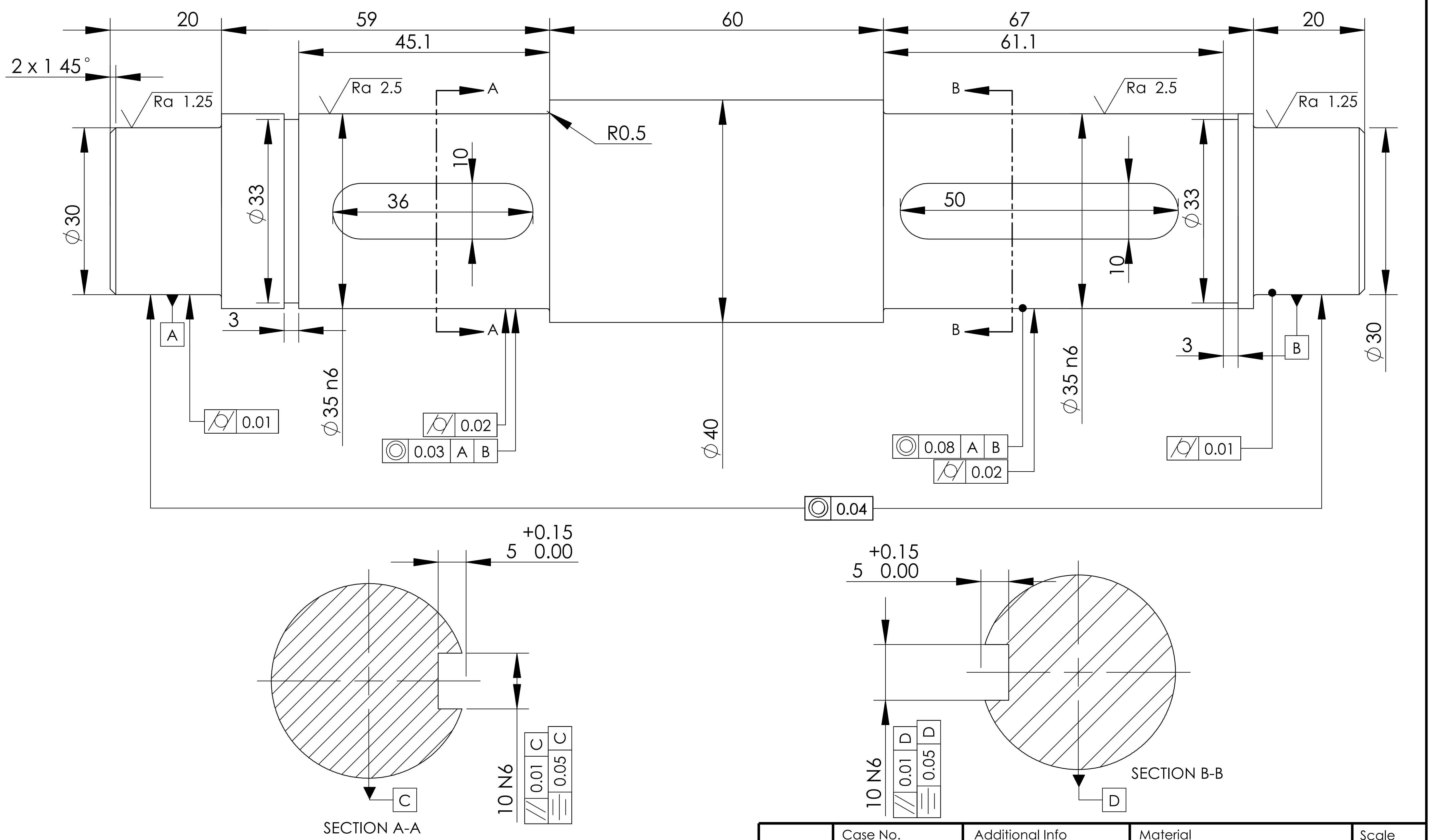
ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	MX3-112_B14	Electro Motor (4kW)	1
2	Housing - Bottom	7mm Cast Housing	1
3	Housing - Top	7mm Cast Housing	1
4	GearShaft1		1
5	GearShaft2		1
6	GearShaft3		1
7	Pinion 1	Spur gear 3.5M 15T 47FW	1
8	Gear 1	Spur gear 3.5M 65T 45FW	1
9	Pinion 2	Spur gear 4.5M 15T 61FW	1
10	Gear 2	Spur gear 4.5M 65T 59FW	1
11	Seal1		1
12	Seal2		1
13	Seal3		1
14	Seal4		1
15	Coupling	Coupling between EM Shaft and first Shaft	2
16	SKF - 61905 - 16,DE,NC,16_68	Bearing Shaft 1	2
17	SKF - 6206 - 12,DE,NC,12_68	Bearing Shaft 2	2
18	SKF - 6010 - 18,DE,NC,18_68	Bearing Shaft 3	2

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
19	Parallel key A10 x 8 x 36 DIN 6885	Key 10 x 8 L=36	2
20	Parallel key A10 x 8 x 50 DIN 6885	Key 10 x 8 L=50	1
21	Parallel key A16 x 10 x 45 DIN 6885	Key 16 x 10 L=45	2
22	ISO 4014 - M8 x 40 x 22-N	Housing Fastners Screw	14
23	ISO - 4034 - M8 - N	Housing Fastners Nut	14
24	ISO 4762 M8 x 16 - 16N	Sealing Fastners	34
25	ISO 7412 - M16 x 45 --- 26-WN	Base Screws	6
26	ISO 4162 - M8 x 30 x 30-N	Oil Control Screws	2
27	Circlip DIN 471 - 30 x 2	Pinion 1 Circlip	1
28	Circlip DIN 471 - 35 x 2.5	Pinion 2 & Gear 1 Circlip	2
29	Circlip DIN 471 - 55 x 3	Gear 2 Circlip	1

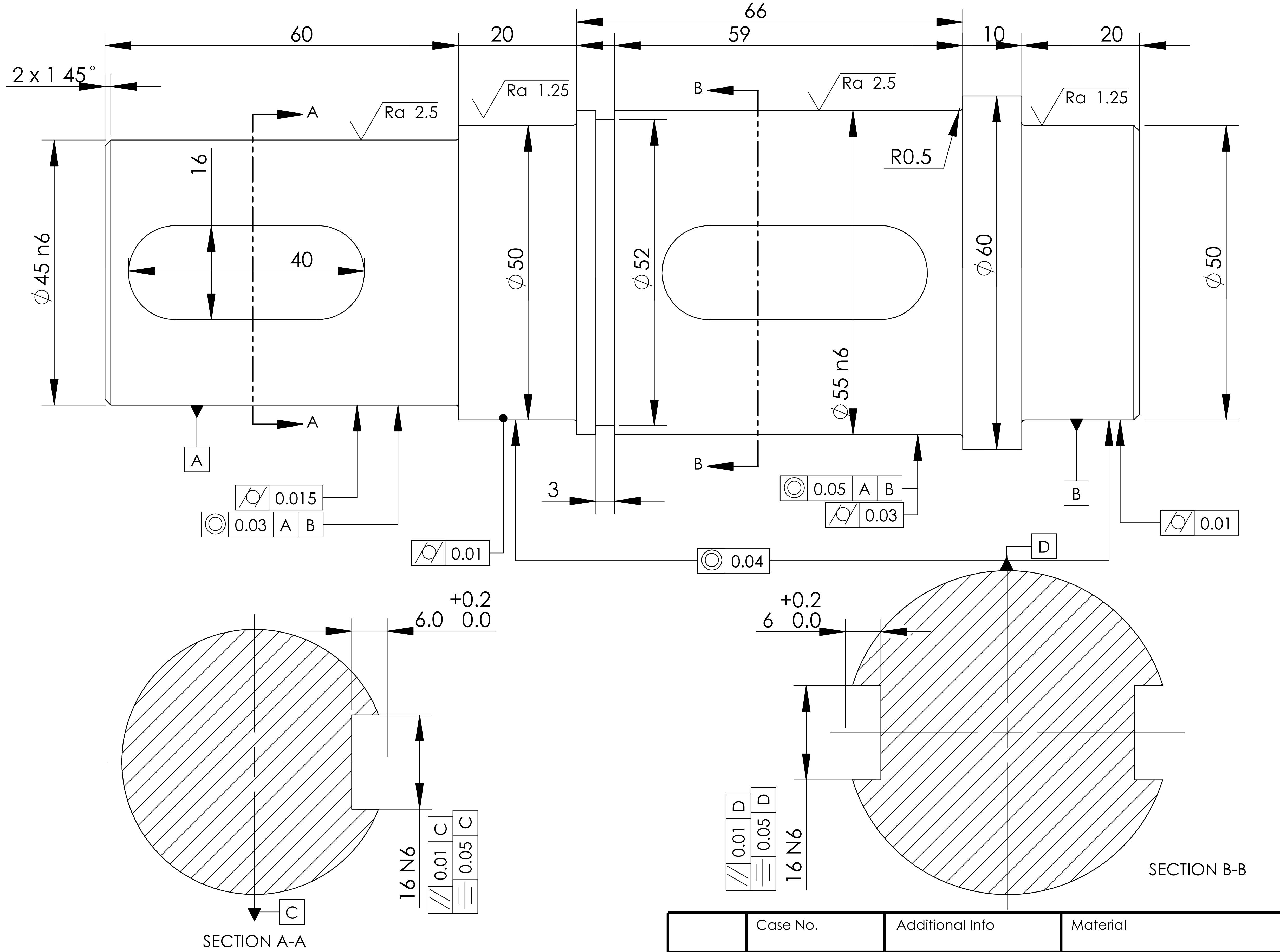
	Case No.	Additional Info	Material	Scale 1:2
Resp. Dept.	Consulted By	Type Of Dock	Status of Dock Teaching	
Owner VGTU	Drawn By Kaan Unlu	Title Base Table of components	Date 2021-19-03	Lang Eng / Paper 5/11



	Case No.	Additional Info	Material	Scale 3:2
Resp. Dept.	Consulted By	Type Of Dock	Status of Dock Teaching	
Owner VGTU	Drawn By Kaan Unlu	Title Base Gear Shaft 1	Date 2021-19-03	Lang Eng / Paper 6/11



	Case No.	Additional Info	Material	Scale 3:2
Resp. Dept.	Consulted By	Type Of Dock	Status of Dock Teaching	
Owner VGTU	Drawn By Kaan Unlu	Title Base Gear Shaft 2	Date 2021-19-03	Lang Eng / Paper 7/11



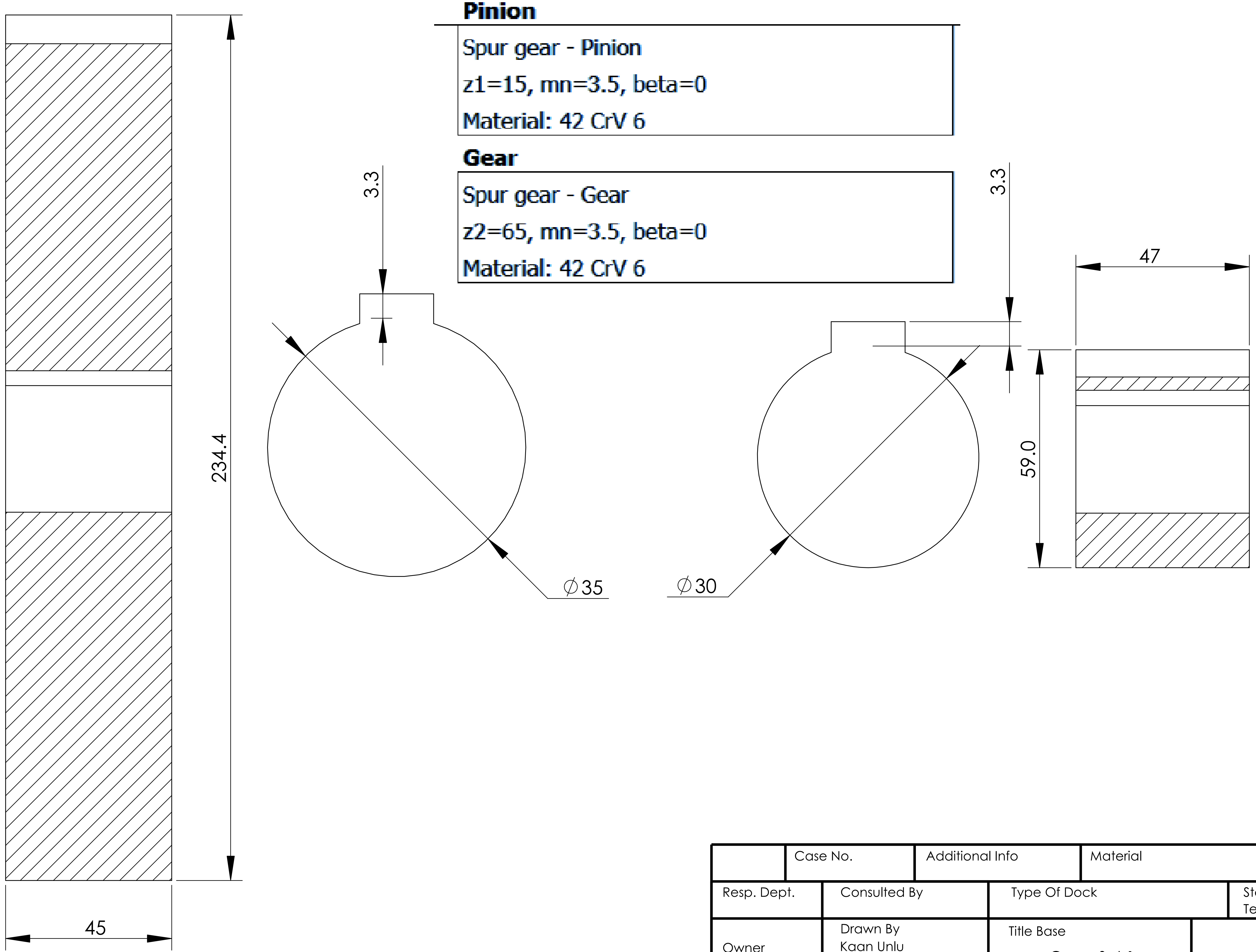
	Case No.	Additional Info	Material	Scale 3:2
Resp. Dept.	Consulted By	Type Of Dock		Status of Dock Teaching
Owner VGTU	Drawn By Kaan Unlu	Title Base Gear Shaft 3	Date 2021-19-03	Lang Eng / Paper 8/11

Pinion

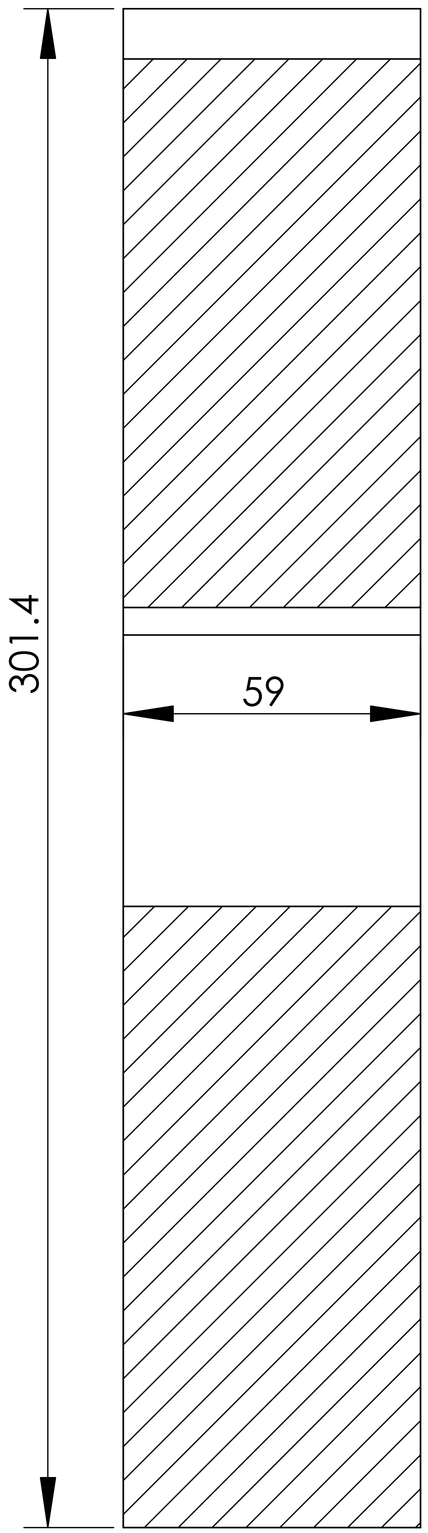
Spur gear - Pinion
z1=15, mn=3.5, beta=0
Material: 42 CrV 6

Gear

Spur gear - Gear
z2=65, mn=3.5, beta=0
Material: 42 CrV 6



	Case No.	Additional Info	Material	Scale 1:1	
Resp. Dept.	Consulted By	Type Of Dock	Status of Dock Teaching		
Owner VGTU	Drawn By Kaan Unlu	Title Base Gears Set 1	Date 2021-19-03	Lang Eng	Paper 9/11



Pinion

Spur gear - Pinion

z1=15, mn=4.5, beta=0

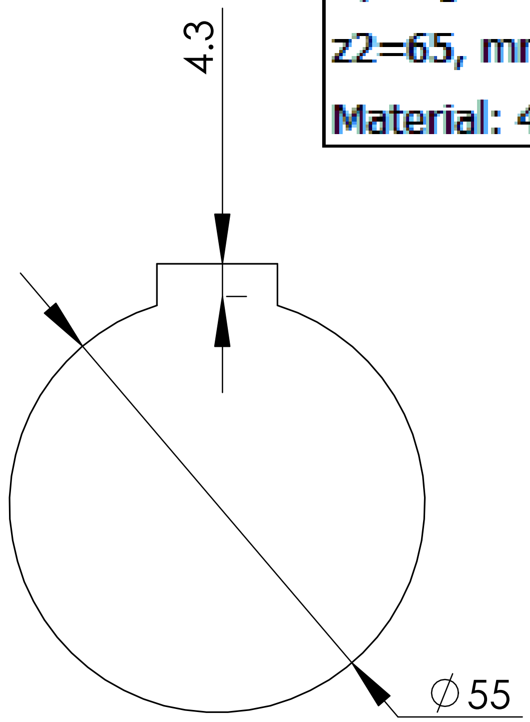
Material: 42 CrV 6

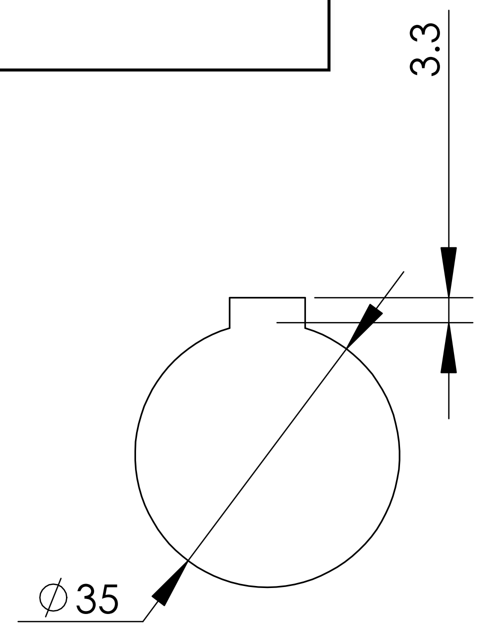
Gear

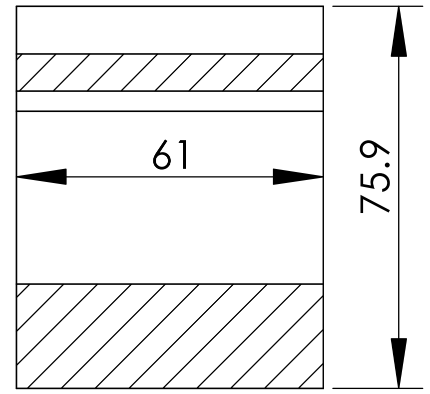
Spur gear - Gear

z2=65, mn=4.5, beta=0

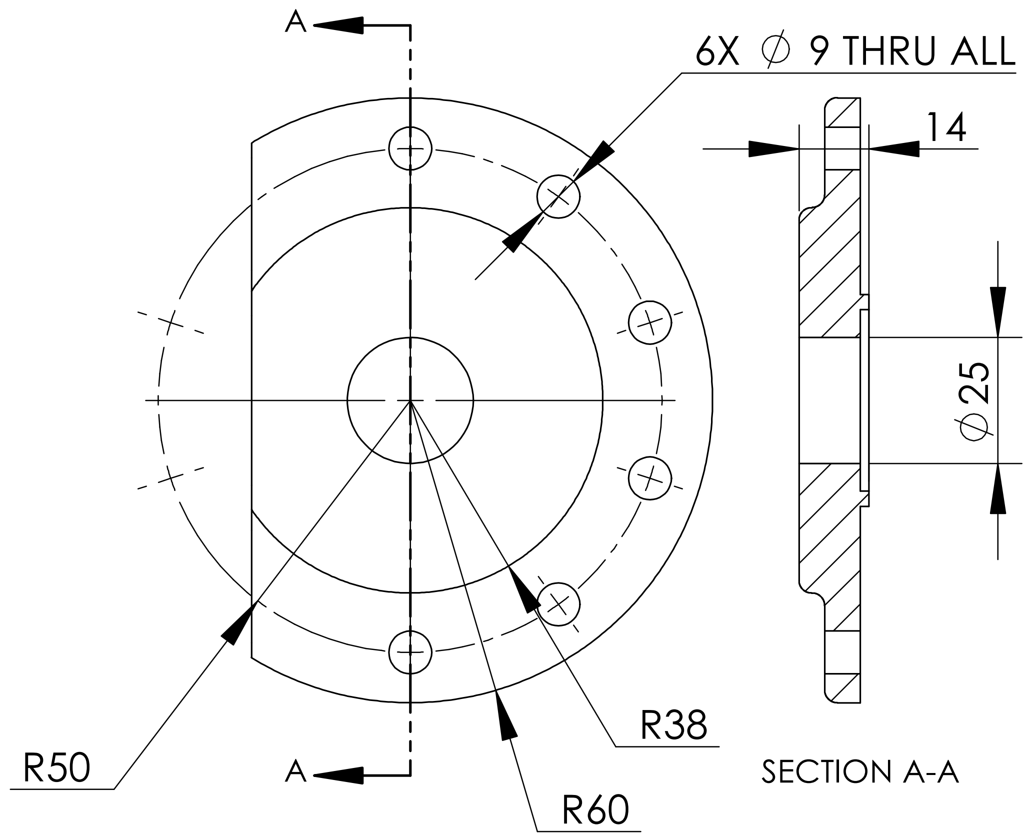
Material: 42 CrV 6

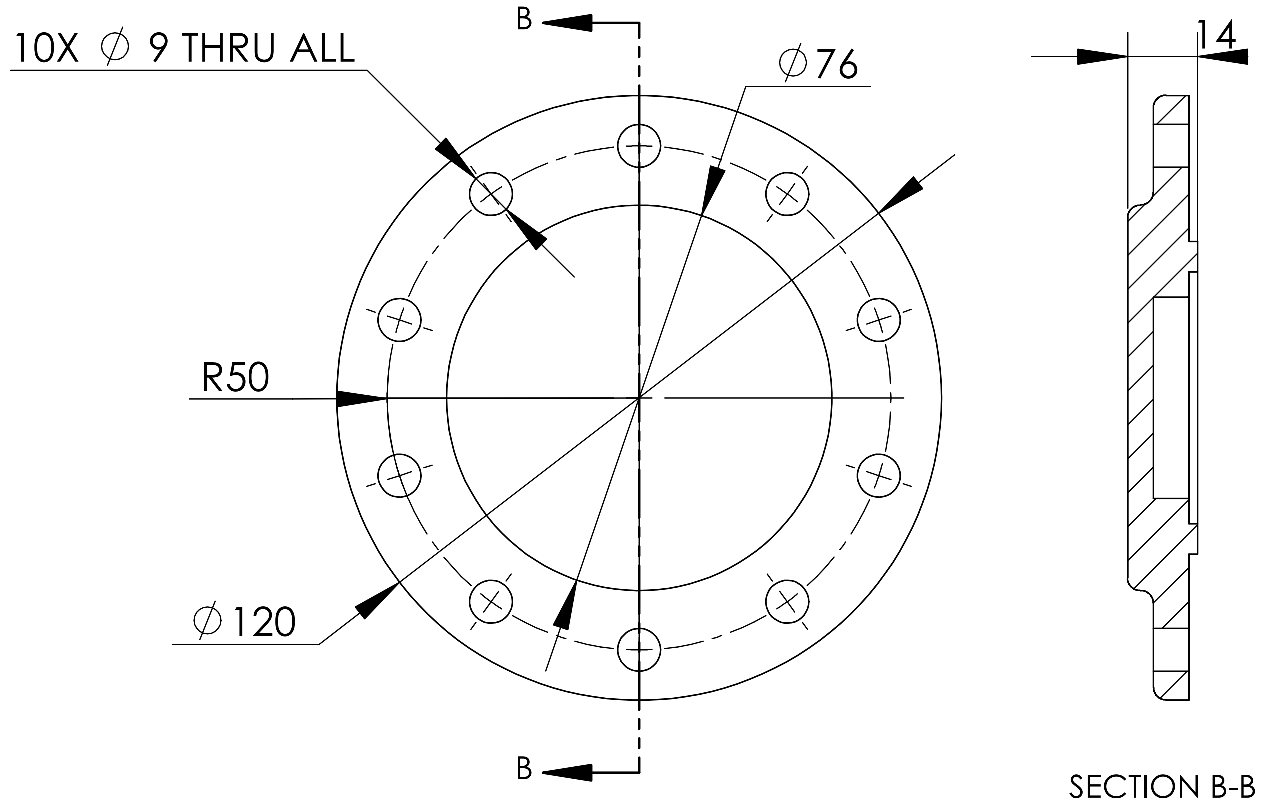


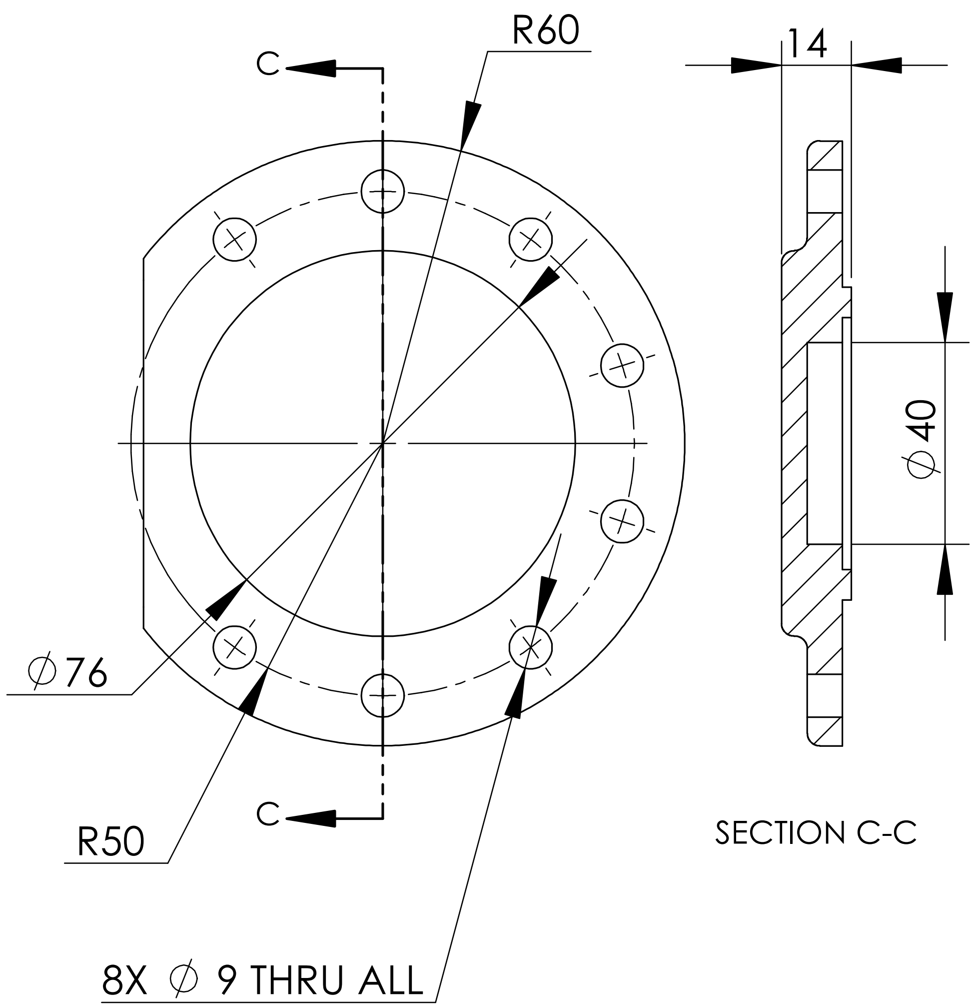


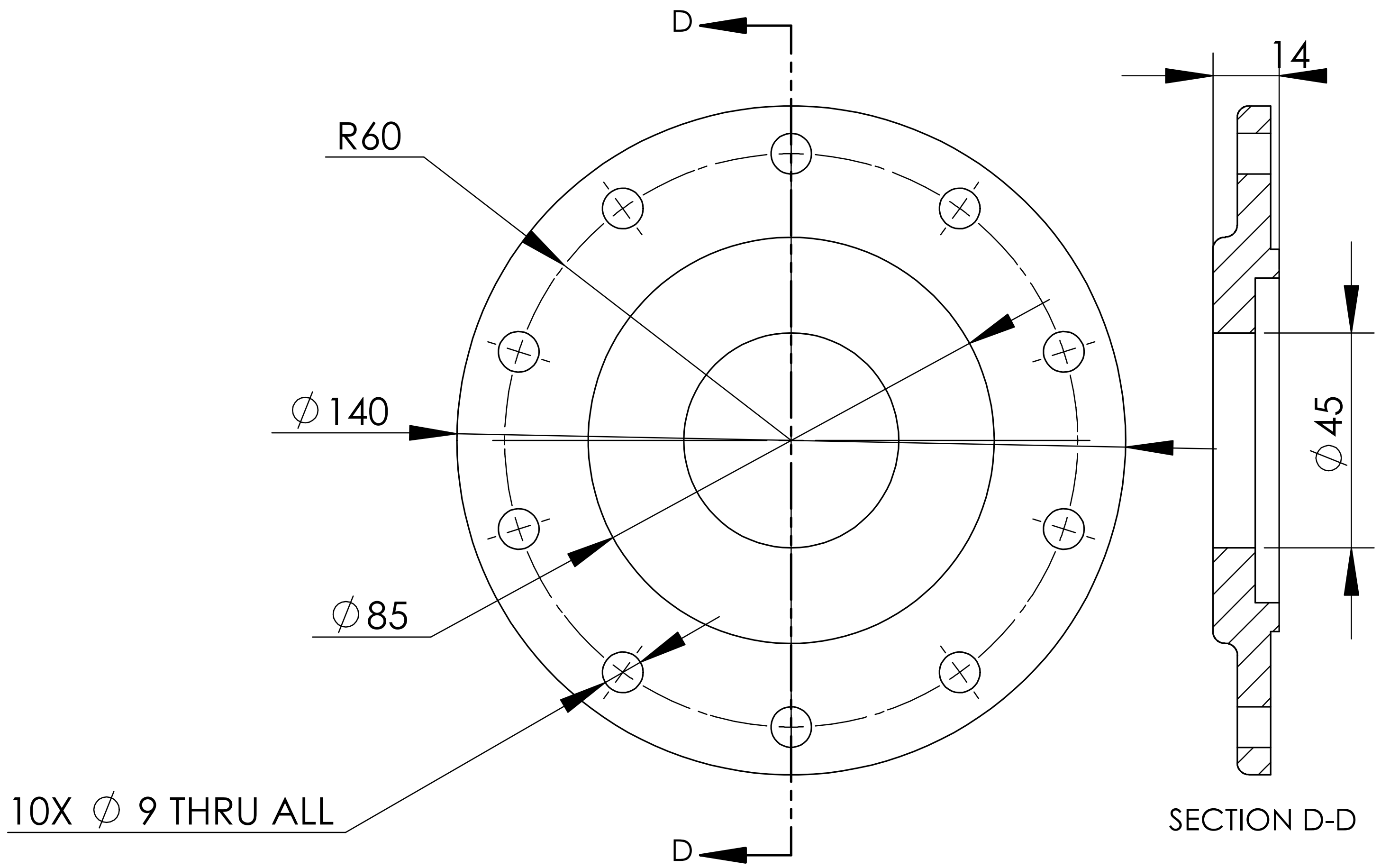


	Case No.	Additional Info	Material	Scale 2:3
Resp. Dept.	Consulted By	Type Of Dock	Status of Dock Teaching	
Owner VGTU	Drawn By Kaan Unlu	Title Base Gears Set 2	Date 2021-19-03	Lang Eng / Paper 1∅1 1









	Case No.	Additional Info	Material	Scale 2:3
Resp. Dept.	Consulted By	Type Of Dock		Status of Dock Teaching
Owner VGTU	Drawn By Kaan Unlu	Title Base Seals		Date 2021-19-03 · Lang Eng · Paper 1/11