Wind Powered Water Pump

by

Andrés GalánEscalada

Santiago Pastor Pérez

PROJECT REPORT

Mechanical design

June 2011

Mechanical Design

Dalgas Avenue, 2. 8000 Aarhus C (Denmark)

Telephone: +45 4789 3000

Fax: +45 4189 3001

E-mail: iha@iha.dk

Title:

Wind powered water pump

Project period:

Spring 2011

Project group:

MIAFP1/group 05

Members of the group:

IY10227 Andrés Galán Escalada

IY10226 Santiago Pastor Pérez

Supervisor:

Flemming Hammer

Copies: 2

Pages: 111

Abstract:

The wind is a removable source of energy. Usually it is used to generate electricity, but it can be used for different goals.

In Israel there are different social classes. Some of them are very poor. During the summer, some regions are very very dry. The access to water is not that easy, but there are some wells.

Joining these two ideas the wind powered water pump was thought. With a low maintenance and cheap construction materials this is a good idea and solution.

The wind turbine provides movement to raise the water with a rope water pump able to move bigger flow with less energy than a membrane pump.

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1 Introduction

Background

This Project was presented by Engineers Without Borders. One student of the IHA told us about the situation and also provided us about information and examples of what the project was about. They sent us some information about what they wanted and they needed. The well was ten meters deep but the tank was two meters high. They also had a condition: the well has to stop raising water when the water level is lower than 0.5 m.

To help us about how would be a wind powered water pump they sent us a project designed in Odense with a savonius vertical axis wind turbine. That wind turbine was able to raise the water three meters with a flow of 175 liters per hour working with a wind speed between 6 and 9 meters per second. It had problems to start pumping. Also another horizontal axis wind turbine design was sent, being this one a commercial one that was able to raise water from 4 meters. What we wanted for this project had to be very different, good efficiency and power with slower wind speed. The height goal was more than eleven meters so the machine had to be very different. After a while the rope pump was found. In Nicaragua they are very popular because of the high efficiency and as we are Spanish it was easier for us finding information about that kind of pumps.

Motivation

This project gives us a chance to learn how to organize by ourselves. First time in our lives we have one deadline to do the whole work, dividing it in a complicated schedule and trying to meeting it. At the same time we have to know about our skills to divide different tasks to be able to develop them as good as possible.

Another motivation is the fact that our project is for Engineers Without Borders. This means that this project is designed in order to help people. Because of the project is about a pump that get water from a well the main goal of the project is helping people, even if it is never built. We hope that some parts of the project will inspire other engineers in similar projects.

Project aims

The objective of this Project is designing a wind powered water pump. This pump has to bedesigned for a region of Israel where the access to the water is not enough. This design has to take water from a 10-meter-deep well to a height of two meters to get it into the tank while the flow has to be at least 1 liter per second. It has to be also cheap and simple, trying to be as effective as possible. The maintenance has to be infrequent and simple in order to be able to be done by a person with no advanced knowledge in this field. To get these objectives passive mechanisms and catalogue pieces will be used, but also some pieces from our own design will be necessary.

The pump used for this case is a rope pump. It has a lot of advantages; very high efficiency, it does not have problems starting pumping when is empty at the same time that its maintenance is needed less than once a year and with low cost.

The turbine to get the power from the wind is a horizontal axis wind turbine. The main reasons to choose a horizontal axis wind turbine are that they are more efficient, simpler, and more reliable; also there are more data recorded. This kind has no problems with starting. But it has a big problem with the orientation, especially when it is connected to a rope pump. Usually they cannot rotate 360 degrees so one of the aims of this project is designing one without this problem.

At the same time all this aims are met, it was thought to be a good idea if as many calculations as possible are done in a way that they can be easily understood, so the whole machine could be modified in order to improve it for the final conditions.

The expectances for this project were a 6-meter-height wind powered water pump able to pump more than one liter per second with a low price and maintenance.

Location

The wind turbine is designed to be used in any well in Israel, a country with scarce water resources and where electricity is not available in most of the homes in the countryside. That is why a water pump wind turbine would be very helpful.

Making a wind study in some different places along Israel the next data have been obtained:

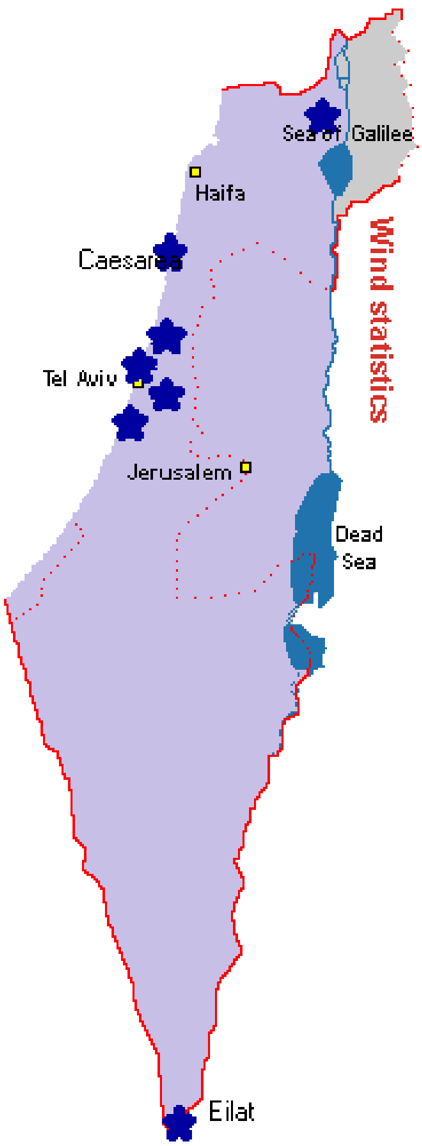


Figure 1.Location of places of wind speed measurements

Lake Kinneret/Sea Galilee (KINNERET)

Statistics based on observations taken between 1/2008 - 3/2011 daily from 7am to 7pm local time.

Month of year	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	SUM
Dominant Wind dir.													
Average Wind speed (m/s)	3	3	4	4	4	4	4	4	4	3	3	3	**3**

Sdot Yam/Caesarea (SDOTYAM)

Statistics based on observations taken between 1/2009 - 3/2011 daily from 7am to 7pm local time.

Month of year	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	SUM
Dominant Wind dir.													
Average Wind speed (m/s)	5	4	5	5	5	5	5	5	5	5	4	4	**5**

Ra'anana (RAANANA)

Statistics based on observations taken between 10/2009 - 3/2011 daily from 7am to 7pm local time.

Month of year	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	SUM
Dominant Wind dir.													
Average Wind speed (m/s)	3	3	4	4	4	4	4	4	4	3	3	3	**3**

Sde-Dov (SDE-DOV)

Statistics based on observations taken between 7/2007 - 3/2011 daily from 7am to 7pm local time.

Month of year	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	SUM
Dominant Wind dir.													
Average Wind speed (m/s)	4	5	5	5	5	5	5	4	5	5	4	4	**4**

Tel Aviv/Ben Gurion (TELAVIV)

Statistics based on observations taken between 2/2003 - 3/2011 daily from 7am to 7pm local time.

Month of year	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	SUM
Dominant Wind dir.													
Average Wind speed (m/s)	4	5	4	5	5	5	5	4	5	4	4	4	**4**

Ness Tsiyona (NSTZIONA)

Statistics based on observations taken between 10/2009 - 3/2011 daily from 7am to 7pm local time.

Month of year	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	SUM
Dominant Wind dir.													
Average Wind speed (m/s)	2	2	2	2	2	2	2	2	2	2	2	3	**2**

Eilat (EILAT)

Statistics based on observations taken between 10/2001 - 3/2011 daily from 7am to 7pm local time.

Month of year	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	SUM
Dominant Wind dir.													
Average Wind speed (m/s)	5	4	5	5	6	6	5	6	7	6	6	5	**5**

Table 1 Wind speeds measurements in Israel

The wind is not very strong, but the direction and the speed are more or less constant.

Average wind speeds are between 3 and 5 m/s except in one of the places (Nstziona) where the wind blows at an average wind speed of 2 m/s. This variation of speed will affect to the water flow, the slower speed the smaller flow. But these values are smaller than the ones in which the wind turbine is going to work because the rotor will be at 4.4 meters high, and the wind speed increases with the height.

2 Design

2.1 Rotor

The rotor is made by six main pieces: three blades and three pieces that join each blade with the shaft. The reason why three blades were chosen instead of a different number is that three is an odd number, what means that the rotor can be considered to be similar to a disc when calculating the dynamic properties of the machine, while the efficiency is only 2% smaller that if it would have four blades.

2.1.1 Blades

The rotor is formed by three blades, which are attached to the hub with one piece that gives the correct angle and bolts. The diameter of the rotor is 3.4 m, which means it can sweep an area of 8.95m^2.

The blades are made of pine wood, which is very common and cheap in Israel. The blades have a NACA profile; it is a NACA 23012, which has to be handmade with the information attached in the drawings.

Each blade has a length of 1.5 meters. The cross section of the blade is changing along its length, as the chord goes from 416 mm to 100 mm the twist is also changing from 0 to 30.4 degrees in the length of the blade.

The next table represents the mean parts in which the blade is divided:

			Sect1	Sect2	Sect3	Sect4	Sect5	Sect6	Sect7
Chord	c	m	0.39	0.30	0.23	0.19	0.16	0.14	0.12
Pitch angle	theta	°	21.7	13.0	8.2	5.2	3.1	1.6	0.5
Solid ratio	sigma	-	0.59	0.27	0.15	0.10	0.07	0.05	0.04
Speed of blade	r*omega	m/s	4.7	7.8	11	14.1	17.2	20.3	23.5

Table 2 Main sections of the blade

The NACA five-digit series describes complex airfoil shapes:

1. The first digit, when multiplied by 0.15, gives the designed coefficient of lift (C_L).
2. Second and third digits, when divided by 2, give *p*, the distance of maximum camber from the leading edge (as per cent of chord).
3. Fourth and fifth digits give the maximum thickness of the airfoil (as per cent of the chord).

The NACA 23012 would give an airfoil with maximum thickness of 12% chord, maximum camber located at 15% chord, with a lift coefficient of 0.30.

<u>Rotor specification sheet</u>

Blades material: Pine wood

Length: 1.5 m

Mass per blade: 2.38 kg

Efficiency of the rotor: 55.5%

Power on the rotor: 268 W

Angular speed: 140 rpm

Av. axial force: 96 N

$C_L(\alpha=7º)$ = 0'877

C_F $(\alpha=7º)$ = 0'0077

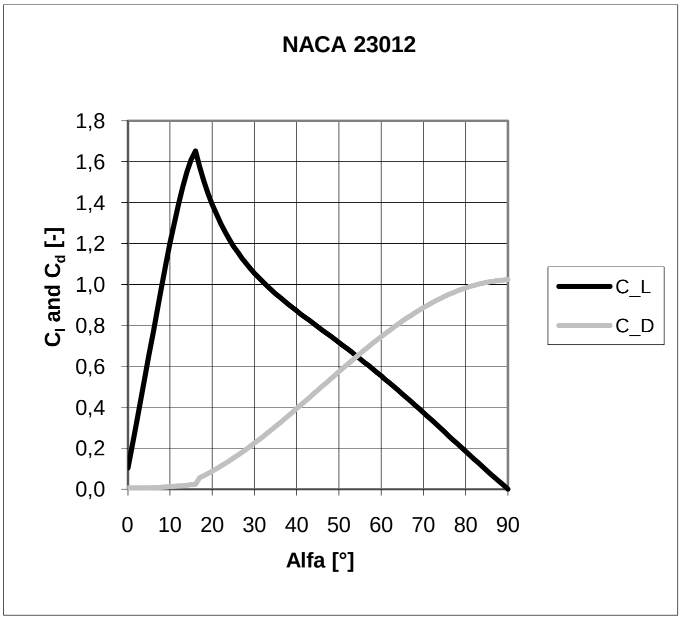


Figure 2Profilecoefficients of drag and lift

The maximum value of coefficient of lift is 1.653 at 16º but the reason why is not the chosen one is because an angle of 7º degrees provides better glide ratio. The glide ratio is the result of the lift coefficient divided by the drag coefficient. The next graphic shows the glide ratio for different angles:

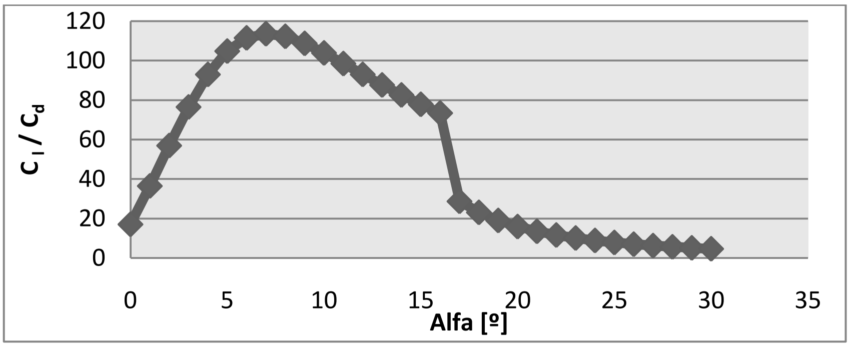


Figure 3 NACA 23012 coefficients ratio

The rotor has been designed to work with an average speed of 4.5 m/s. During the design of the blades the size of them, the tip speed ratio and the efficiency were considered.

Power calculation		
Pitch control	dteta	0 °
Wind speed	V_0	4,5 m/s
Rotational speed	n	140,625 rpm
Rotational speed	ω	14.73 rad/s
Power	P	268.1 W
Efficiency	eta	90,7 %
Torque	T	18.20 N·m
Axial force	F_a	96 N
Tip speed ratio	X	5,56
Mean angle of attack	alpha	7.0 °

Table3Powercalculation in the rotor

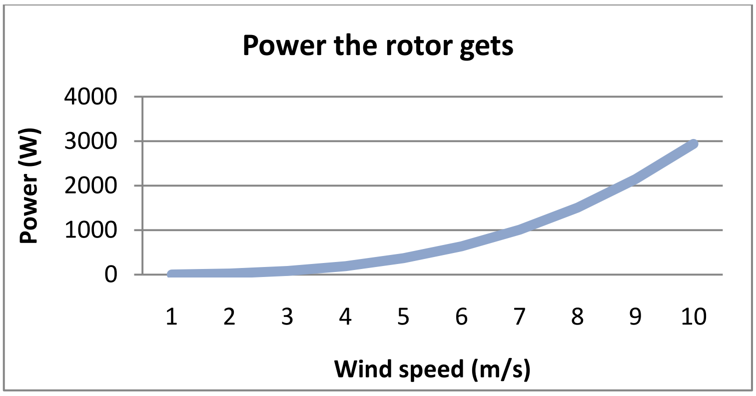


Figure 4Power of the rotor dependingonthewindspeed

Radius	m	0,32	0,53	0,74	0,96	1,17	1,38	1,59
Ring nº	-	1	2	3	4	5	6	7
Power	W	12	23	32	41	50	57	53

Table4Power in thesections of the rotor

The conclusion of this table is that the smaller rings get less power that the bigger rings, so if during the fabrication work there were less material than needed the part that should be modified is the wider end.

Pine wood as material: Wood was chosen as material to make the blades because of his lightness and reduced price. Also, pine wood is a very common material in Israel. It is a strong wood and it holds its shape for long time.

2.1.2 Join parts

In order to join the blades with the correct angle to the shaft one piece has been designed.

These pieces have to support the bending moments, the axial loads because of the centripetal force and the shear loads because of the wind.

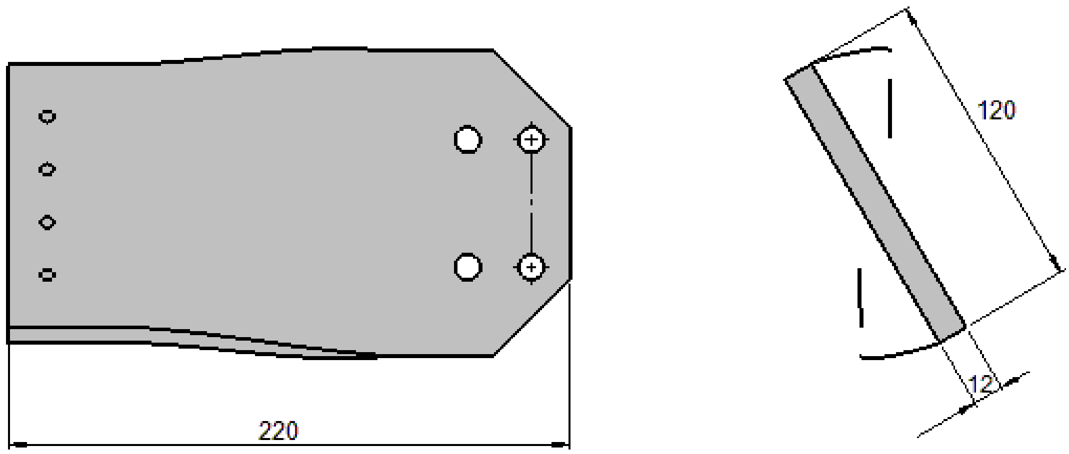
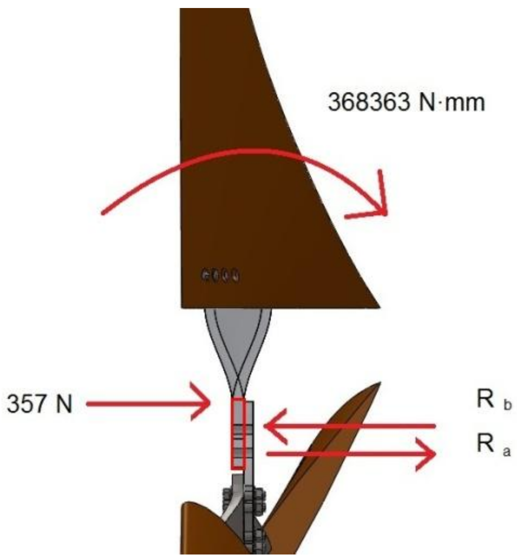


Figure 5 Join part

The calculations are detailed in the annex I but the values that have to support are:



- 357 N in the wind direction

- 1845 N in the radial direction

- 368.363 N·mm

Figure 6Forcesonthejoinpart

The Von Mises stress is 117 MPa which means a safety coefficient of more than 2 using a carbon steel with a yield resistance of 282 MPa.

The bolts needed to support all the loads have been also calculated. In the annex II there is a detailed study about them.

The bolts that join the blade to the join part are M5 and the ones that are screwed to the rotor are M10. The safety coefficient is higher than other parts because bolts can be lost between every maintenance and if the blade will come lose it would be very dangerous.

2.2 Nacelle

It is a device placed on the top of the tower. It connects the rotor with the truss and is able to orientate the rotor into the wind using the yaw mechanism. Its main function is to keep all the pieces necessary to transmit the mechanical energy from the rotor to the vertical shaft. It is designed to be mounted and maintain easily.

2.2.1 Box

The nacelle box has the main and important function of covering the transmission. It is smaller than the internal diameter of the rotor to avoid the blades crashing, and it has two different parts:

The bottom part is made of steel and has been designed to fix all the pieces in the right position. It has a hole where the vertical shaft passes through. Concentric to this hole, there is a recess to keep the bearing, and a cylindrical steel plate to avoid dirt damage the bearing. There are also three holes to fix the yaw mechanism tail.

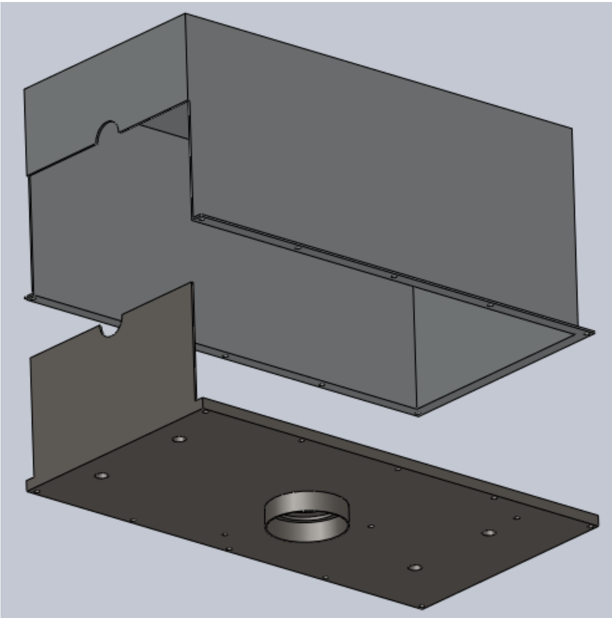


Figure 7Nacelle box

The top part is much lighter because its only function is covering the devices. It is made of aluminum and designed to mount and dismount easily. It can be fixed with the bottom part in the last moment to avoid the disturb to the operator when the assembly is done.

To fix both parts, there are ten holes, five in each side of the nacelle.

2.2.2 Tail

The tail is a device to face the rotor into the wind. This case is different to small American turbines because this tail will be turned 45 degrees to compensate de torque transmitted to the vertical shaft. This torque is 18.21 N·m, so the torque that has to be generated is that one.

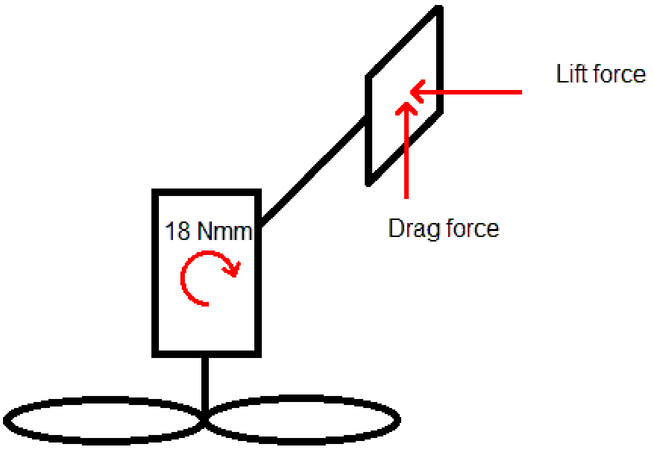


Figure 8 Forces on the tail

The calculations have been done with a wind speed of 4.5 m/s. The tail is 1300x800x1 mm. The thickness is not important because the drag and lift forces do not depend on it but on the surface against the wind.

In order to know the drag and lift coefficient an experiment was done with the wind tunnel and a piece done to scale.Detailed calculations can be found in annex III.The drag and lift forces can be calculated as:

$$F_d = \frac{1}{2} \cdot \rho \cdot A \cdot C_d \cdot v^2 = 9.2\ N$$

$$F_l = \frac{1}{2} \cdot \rho \cdot A \cdot C_l \cdot v^2 = 8.6\ N$$

Being:

- ρ : air density
- A : area of the tail
- C_d : drag coefficient
- C_l: lift coefficient
- v : wind speed

The distance from the center of the nacelle to the center of the tail is 1.7 m but it is turned 45º so to calculate the torque the distance is 1.7 x sin 45.
So the moment created is:

$$M = F_d \cdot 1.7 \cdot \sin 45 + F_l \cdot 1.7 \cdot \cos 45 = 18.21\ N \cdot m$$

As it can be seen in the formulas to calculate the drag and lift forces, those force are directly proportional to the squared wind speed. The same situation happens with the torque generated by the rotor. The power of the rotor is proportional to the cubed wind speed, but the power is the torque times the speed so both torques are proportional to the squared wind speed. The next formulas will demonstrate this statement.

V_{tip}: blade tip speed

V: wind speed

X: tip speed ratio

r: rotor radius

µ: rotor efficiency

D: rotor diameter

$$X = \frac{V_{tip}}{v} = \frac{r \cdot \ \omega}{v} \rightarrow \omega = \frac{X \cdot v}{r}$$

$$M = \frac{1}{2} \cdot \rho \cdot A \cdot C_d \cdot v^2 \cdot 1.7 \cos 45 + \frac{1}{2} \cdot \rho \cdot A \cdot C_l \cdot v^2 \cdot 1.7 \cdot \sin 45 =$$

$$M = (\frac{1}{2} \cdot \rho \cdot A \cdot C_d \cdot 1.7 \cos 45 + \frac{1}{2} \cdot \rho \cdot A \cdot C_l \cdot 1.7 \cdot \sin 45) \cdot v^2$$

$$M = \frac{P}{\omega} = \frac{\mu \cdot {}^{\pi}/_{4} \cdot D^2 \cdot {}^{1}/_{2} \cdot \rho \cdot v^3}{\frac{X \cdot v}{r}} = \left(\frac{\mu \cdot \pi \cdot r^3 \cdot \rho \cdot {}^{1}/_{2}}{X}\right) \cdot v^2$$

As it can be seen those momentums are proportional to the squared wind speed.

2.2.3 Shaft

This shaft is one of the most important parts of the wind powered water pump. This shaft connects the rotor to the rest of the transmission elements, supporting at the same time the weight of the rotor. It has three bearings, one gear, and a rotor on it. All detailed calculations can be seen in annex IV.

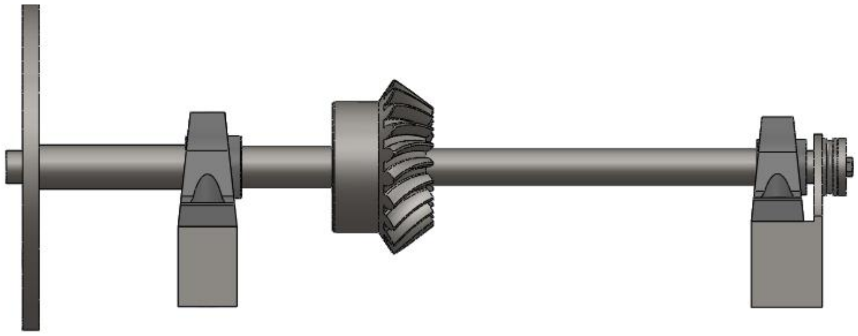


Figure 9 Rotor shaft

The rotor weighs 163.8 N and applies a horizontal force to the shaft. This force is due to the wind, the wind when generates a torque on the shaft, also push the rotor against the shaft. Every force used to calculate the proper dimensions for the pieces in this project are calculated for a wind speed of 15 m/s. For this wind speed the force that the wind will apply on the shaft is 868 N. This has been calculated with the blade element momentum (BEM) theory. There are forces in every bearing as a response of the forces generated by the rotor and the gear that will be calculated below. But gear forces have to be calculated before. If the wind speed is 15 m/s the torque generated by it is 164000 N·mm. The pitch diameter of this gear is 100 mm so the force to transmit is 3277.4 N. The direction of this force is perpendicular to the drawing. The angle of pressure (α) of the gear is 20º what means that there is a separating force between the two gears. The value of this force can be calculated as $F \cdot \tan(\alpha)$, what is 1192 N.

Knowing the forces acting on the shaft, the forces that the bearings have to support can be calculated. These values are 2603 N and 802 N. In the axial bearing the force to support is 263 N.

Following this the load diagrams can be drawn.

The next diagram shows the bending loads in Z-axis, units are in N·mm:

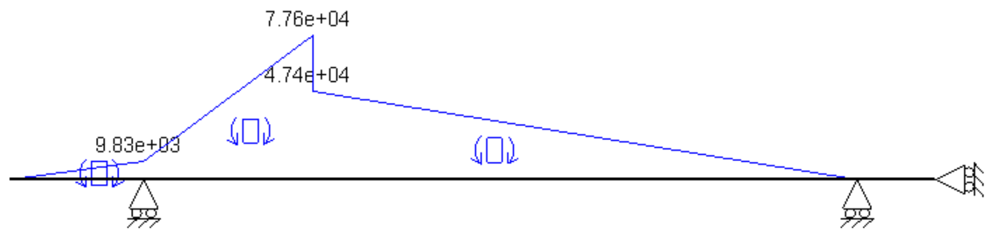


Figure 10 Bending moments in Z-axis in the rotor shaft

The maximum bending moment in the figure is where the gear is.

The next picture shows the bending loads in Y-axis, units are in N·mm:

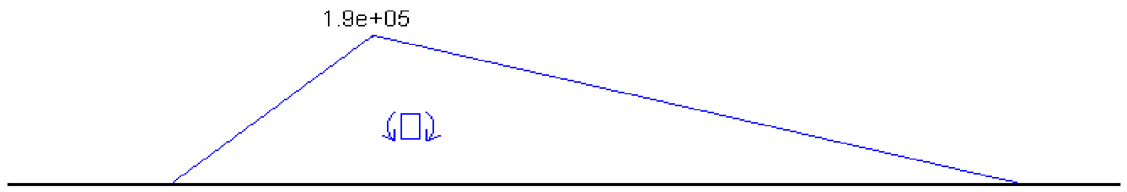


Figure 11 Bending moments in Y axis on the rotor shaft

The maximum bending moment in this figure is also where the gear is.

The total bending moment in this point can be calculated as thesquarerootofthe sumof thesquaredmoments. This value is 20500 N·mm.

This diagram shows the torque in the shaft, units are N·mm:

Figure 12 Torque on the rotor shaft

This means that in the point where the gear is, the torque is 164000 N·mm and the bending moment is 261000N·mm.

The needed diameter can be calculated with this two moments as:

$$d_{min} = \sqrt[3]{\frac{4 \cdot n_s}{\pi \cdot \sigma_y} \sqrt{64M_{max}^2 + 48T_{max}^2}} = 20.4 \approx 22\ mm$$

Being the safety coefficient 2, and the yield stress 710 MPa.

The diameter is 22 because is the hub diameter of the gear and meets the requirements.

2.2.3.1 Bearings

After calculating the loads and forces on the shaft, the values of the force to support by the bearings are known. The radial bearings have to support 2603 and 802 N respectively while the maximum value to support by the axial bearing is 263 N.

They have been chosen from the SKF catalog.

Radial bearings.

Figure 13 Y bearingplummer block

The chosen radial bearings are known as Y-bearing plummer block units. These units comprise:

A “Y-bearing” (insert bearing) which is a single row deep groove ball bearing with convex sphered outside diameter.

A “Y-bearing housing”, which has a correspondingly sphered but concave bore.

Figure 14 Y bearing

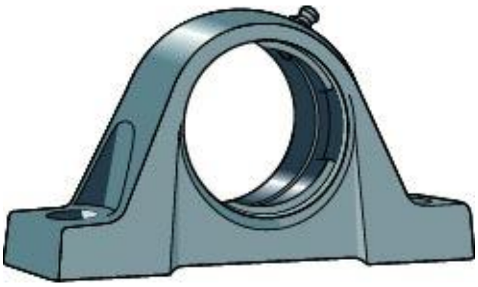


Figure 15 Y bearinghousing

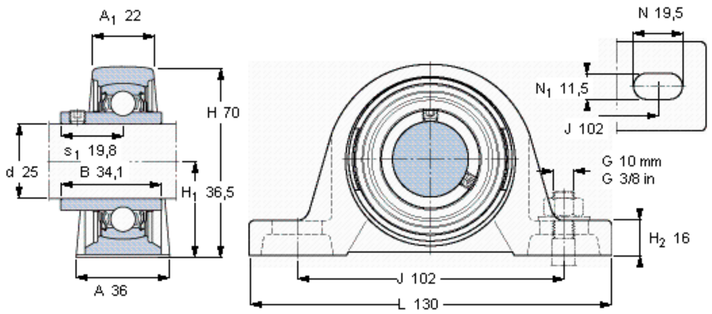
Y-bearing units are ready to mount, greased and sealed units which enable initial errors of alignment to be compensated for.

A support will be added to the bearings so they are easy to join to the nacelle at the proper height.

The needed diameter for the shaft was also known. The bearing that had to support bigger load is where the shaft is 25 mm width. This bearing unit designation is SY 25 TF/VA201. Information about the bearing unit, housing, and Y-bearing can be read in the next figure:

Product information

Dimensions					Basic load rating static	Mass	Designations Bearing unit	Housing	Bearing
d	A	H	H_1	L	C_0				
mm					kN	kg	-		
25	36	70	36,5	130	7,8	0,72	**SY 25 TF/VA201**	SY 505 U/VA228	YAR 205-2FW/VA201



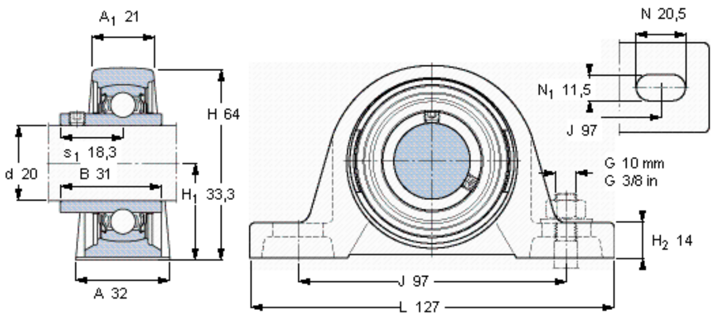
Grub screw	M 6×0.75
Recommended tightening torque [Nm]	4
Hexagonal key size [mm]	3

Figure 16 Y bearing information

The other radial bearing must support 802 N and is installed where the shaft is 20 mm width. Its designation is SY 20 TF/VA201.

Product information

Dimensions					Basic load rating static	Mass	Designations Bearing unit	Housing	Bearing
d	A	H	H_1	L	C_0				
mm					kN	kg	-		
20	32	64	33,3	127	6,55	0,57	**SY 20 TF/VA201**	SY 504 U/VA228	YAR 204-2FW/VA201



Grub screw	M 6×0.75
Recommended tightening torque [Nm]	4
Hexagonal key size [mm]	3

Figure 17 Second Y bearing information

Axial bearing.

The main function of this bearing is to keep the gear and the pinion together. Single direction thrust ball bearings, as their name suggests, can accommodate axial loads in one direction and thus locate the shaft axially in one direction. They must not be subjected to any radial load.

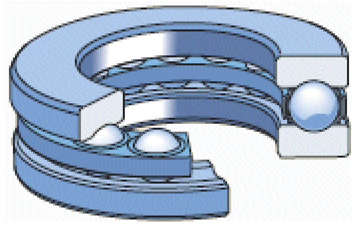


Figure 18 Axial bearing

A single direction thrust ball bearing has been chosen because the load to support is not too big. Its designation is 51104. Dimensions and product information are given in the next figure.

Thrust ball bearings, single direction

Product information | Tolerances , see also text | Recommended fits | Shaft and housing tolerances

Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Minimum load factor	Speed ratings Reference speed	Limiting speed	Mass	Designation
d	D	H	C	C_0	P_u	A				
mm			kN		kN	-	r/min		kg	-
20	35	10	15,1	29	1,08	0,0044	7500	10000	0,039	**51104**

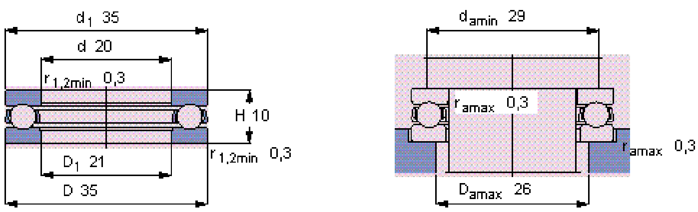


Figure 19Thrust bearing information

2.2.4 Bearing

To support the nacelle over the tower there is one axial bearing. This bearing allows the nacelle facing to the wind.

As the horizontal force of the wind on the blades is transmitted to the shaft, and from this on to the vertical shaft, there are not going to be big forces on it. It will always have to stand the force of the weight of the rotor, and also the nacelle which is 500 N but this force is very small compared to the force it can stand.

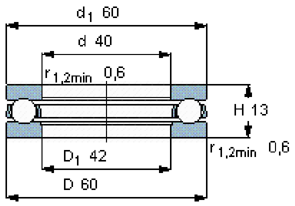
The diameter of this bearing has to be big enough for the shaft to go through it,

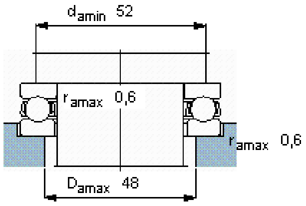
The bearing chosen for this, is a skf 51108, that can be found on the skf website.

Thrust ball bearings, single direction
Product information

Tolerances , see also text
Recommended fits
Shaft and housing tolerances

Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Minimum load factor	Speed ratings Reference speed	Limiting speed	Mass	Designation
d	D	H	C	C_0	P_u	A				
mm			kN		kN	-	r/min		kg	-
40	60	13	26	63	2,32	0,02	5000	7000	0,12	**51108**





2.3 Tower

Instead of a pole a steel truss was used due to a greater rigidity that reduces vibrations and because it is easier to place the different parts needed for the design. But in order to make pieces smaller than 3 m long to be easier to transport and dismount, the tower is made by two trusses than can be joined with screws.

The function of the tower is to get the turbine at a certain height. In the ground level there are some obstacles like houses, hills, trees…which slow the wind and change the wind direction. Due to this friction forces the wind speed increases with the height, the higher the faster the wind is. Knowing the wind velocity at a certain level, this variation can be known with the “Wind shear formula”:

$$v = v_{ref} \frac{\ln(z / z_0)}{\ln(z_{ref} / z_0)}$$

where:

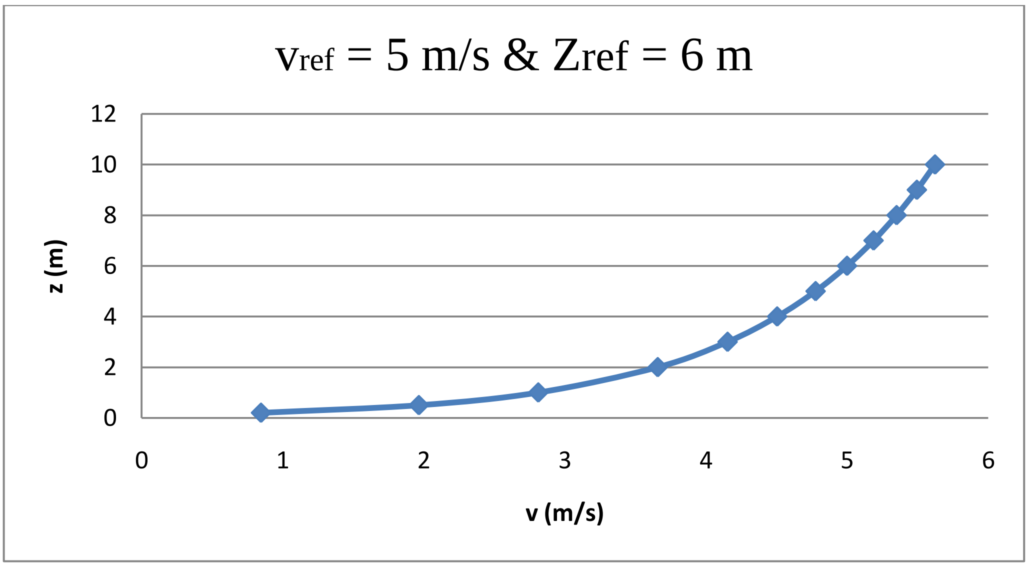
v = wind speed [m/s] at height z above ground level

v_{ref} = reference speed [m/s], i.e. a wind speed known at height z_{ref} .

z = height [m] above ground level for the desired velocity, v.

z_0 = roughness length [m] in the current wind direction, see below

Note: ln(...) is the natural logarithm function.



Class		Roughness Length: m	Landscape features
No.	Name		
1	sea	0.0002	open water, tidal flat, snow with fetch above 3 km
2	smooth	0.005	featureless land, ice
3	open	0.03	flat terrain with grass or very low vegetation, airport runway
4	roughly open	0.10	cultivated area, low crops, obstacles of height H separated by at least 20 H
5	rough	0.25	open landscape, scattered shelter belts, obstacles separated by 15 H or so
6	very rough	0.5	landscape with bushes, young dense forest etc separated by 10 H or so
7	closed	1.0	open spaces comparable with H, eg mature forest, low-rise built-up area
8	chaotic	over 2.0	irregular distribution of large elements, eg city centre, large forest with clearings

Table5Roughnessdependingonthelandscape

The truss is a structure comprising triangular units constructed with straight pipes whose ends are connected at joints referred to as nodes. It is composed of triangles because that shape has a great structural stability. External forces and reactions to those forces are considered to act only at the nodes and result in forces in the members which are either tensile or compressive forces. Moments (torques) are explicitly excluded because, and only because, all the joints in a truss are treated as revolutes. A triangle is the simplest geometric figure that will not change shape when the lengths of the sides are fixed.

Truss structures are easy to make. It is possible to use materials like stainless steel tubes, which are very cheap. Of course these tubes will have to be deformed and welded together.

Calculations have been made supposing a maximum wind speed of 13.5 m/s because statistics say that wind speed in Israel doesn´t reach that magnitude very often. In addition, the tower can be dunk easily to avoid a collapse.

Using the BEM method with the Excel sheet, forces acting in the blades can be known. Adding these forces to other forces acting on the rest of the rotor, an axial force is obtained. The tower must support also the weight of all the components. So the final forces distribution is:

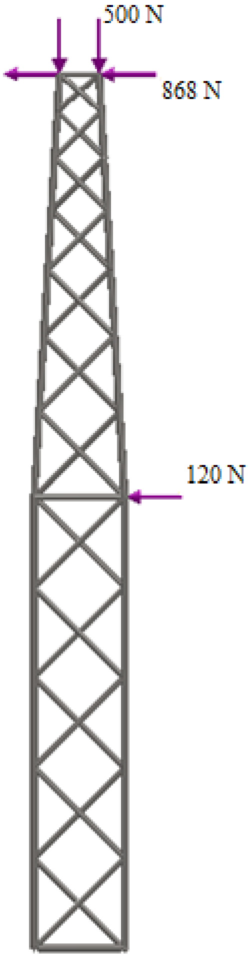


Figure 20Forcesonthetower

The horizontal force has a value of 868 N and the tower has to support around 500 N weight.

Drag force on the tower will act as a distributed force and can be calculated:

$$F_D = \frac{1}{2} \cdot C_D \cdot A \cdot \rho \cdot v^2 [\mathrm{N}]$$

with:

$C_D = 1,2$

$A = 4 \cdot 4,4 \cdot 0,0337 + 2 \cdot 6,5 \cdot 0,0269 = 0,943\mathrm{m}^2$

$\rho = 1,225$ kg/m³

v = 13.5 m/s

$$F_D = \frac{1}{2} \cdot C_D \cdot A \cdot \rho \cdot v^2 = \frac{1}{2} \cdot 1,2 \cdot 0,862 \cdot 1,225 \cdot 13.5^2 = 107\ N$$

The tubes used are:

Tube A: ISO pipe 26,9 x 3,2 mm

Tube B: ISO pipe 33,7 x 4,0 mm

Height = 5 m

Width x length = 0,5 x 0,5 m

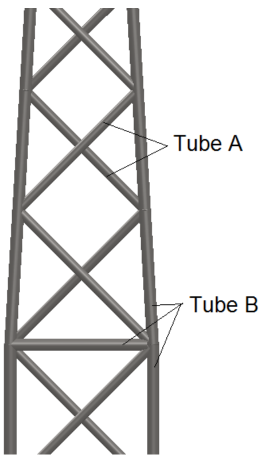


Figure 21Detail of thetower

Study

The structure can be studied using SolidWorks and it is possible to determinate that the model is strong enough.

Displacement

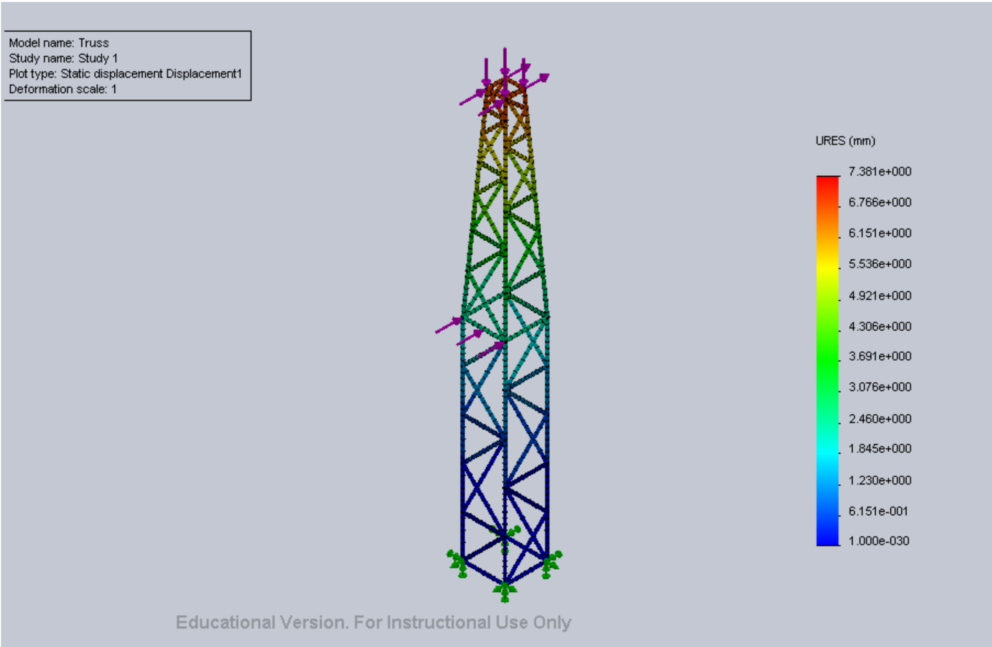


Figure 22 Displacements on the tower

From the simulations it is concluded that the tower will bend with a maximum displacement of 7,4 mm at the top. This is a worst case scenario at a wind speed of 13 [m/s] and is an acceptable value.

Having a look at the stress in the construction:

Stress

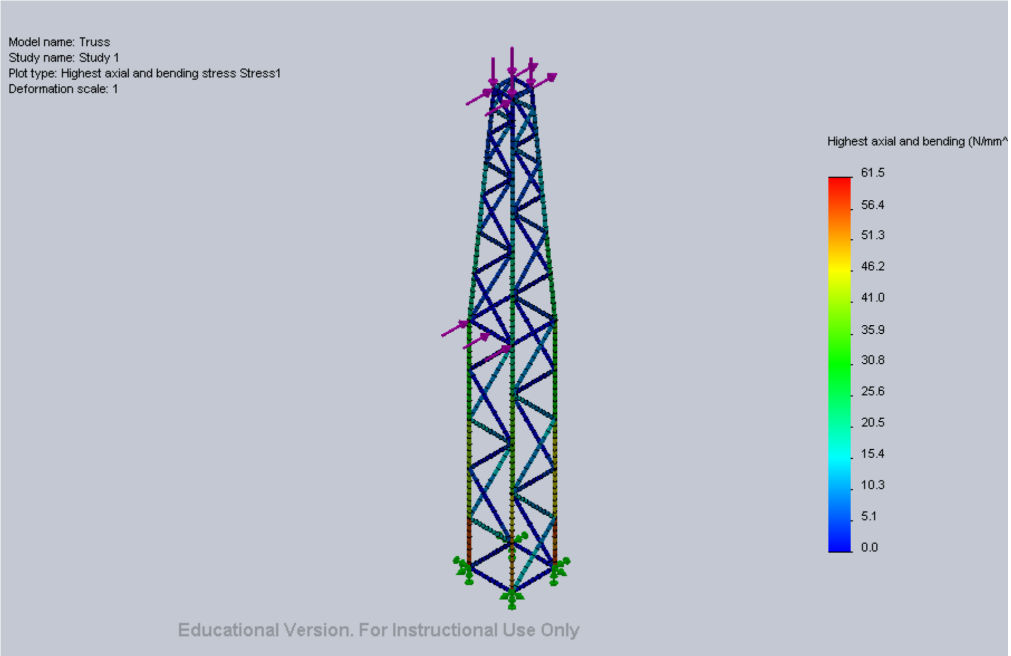


Figure 23 Stresses on the tower

The study shows that there will be a maximum stress of 61,5 [MPa] in the lower part of the tower. To assure this is an acceptable value the tubes will be made of stainless steel ferritic, which has a Yield Strength of 172 MPa, so it's strong enough for the construction. It is resistant to corrosion and has a good weldability.

To fix the truss to the ground the next piece of metal is used.

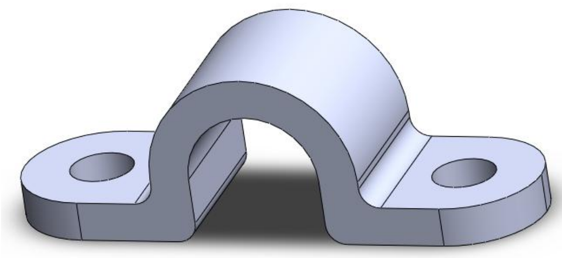


Figure 24 Tower fixingpiece

The bolts used are calculated:

The force on the top of the tower is 1000 N and the drag force of the truss is 100 N (and will be simplified as a force hitting the tower in the middle).

So the moment these forces will generate on the bottom of the tower will be:

$$M = 1000 \cdot 4.4 + 100 \cdot 2 = \ 4200\ N \cdot m$$

Using four connectors, one on each side of the truss, axial forces due to this moment can be calculated:

Axial

$$F_{side} = \frac{M}{distance} = \frac{4600}{0,2} = 23000\ N$$

So if one connection has two screws, and a safety coefficient of 2 is used, the force in each screw will be:

$F_{screw} = \frac{F_{side}}{2} \cdot n_s = \frac{23000}{2} \cdot 2 = 23000\ N$ per screw.

Radial

The radial force on the screws will be the sum of the horizontal forces, which is equal to 1100 N.

The bolts will be M8 with grade 8.8(Max. axial force: 24 kN) and they will be screwed in a concrete block, so an excavation in the ground filled with concrete has to be done.

2.4 Transmission

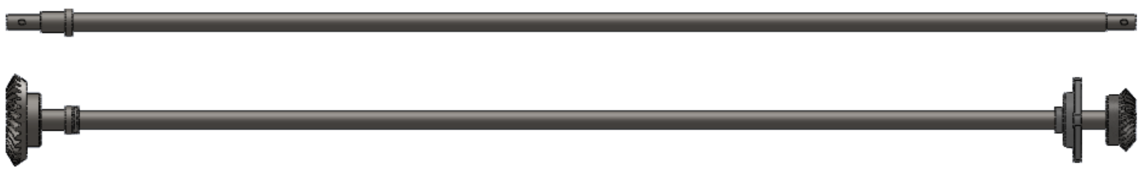
The transmission had to be one of the improvements of the project. There was a problem with it, because the best way to transmit the movement seemed to be two pulleys with one belt. Each pulley would be in one of the horizontal shafts there are in this project. The problem with that kind of transmission is that the nacelle cannot twist 360 degrees so the orientation is not as good as it was wanted for this project. The solution was a vertical shaft inside the tower that would transmit the torque and the speed. So the transmission has five parts, three shafts and two couples of gears.

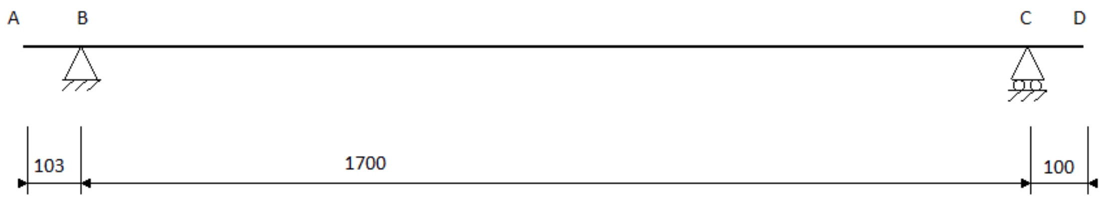
The first designs about the nacelle were working with friction wheels instead of gears. The main reason was that the efficiency of friction wheels is higher than the gears. But one problem was found with that system, the force needed to transmit the torque from one wheel to the next one was too big. The solution to make them work would be more complicated and expensive, so the friction wheels were discarded.

The rotor shaft has been explained in the design of the nacelle design part and calculations can be checked in annex V.

2.4.1 Vertical shaft

The vertical is going to transmit the torque and the angular velocity from the rotor shaft to the pump shaft. In this case the forces can change their direction because the rotor is going to be faced to the wind so the analysis has to be more detailed and thorough. All calculations can be seen in the annex V.





It has to support the separating forces of both gears, the forces of the bearings, the bending moments created by those forces and the torque. Detailed calculations can be found in annex V.

The case were the loads are bigger on it is when the plane, where the rotor shaft and the vertical shaft are, is perpendicular to the plane where the vertical shaft and the pump shaft are.

The next diagram shows the bending moments that the shaft has to support in Y axis:

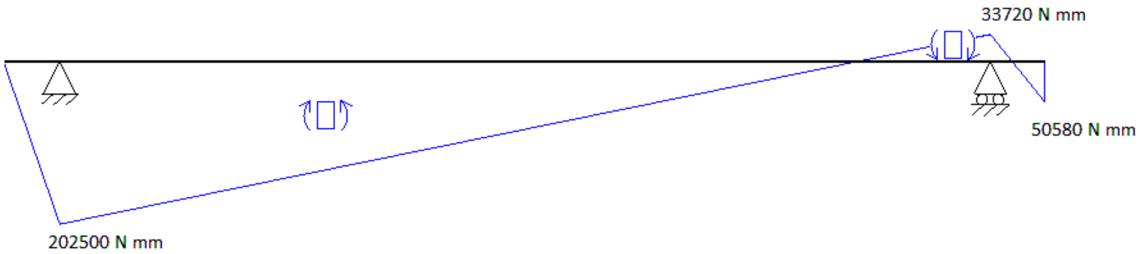


Figure 25Diagram of Y bendingmomentsonthe vertical shaft

The next diagram shows the bending moments that the shaft supports in Z axis:

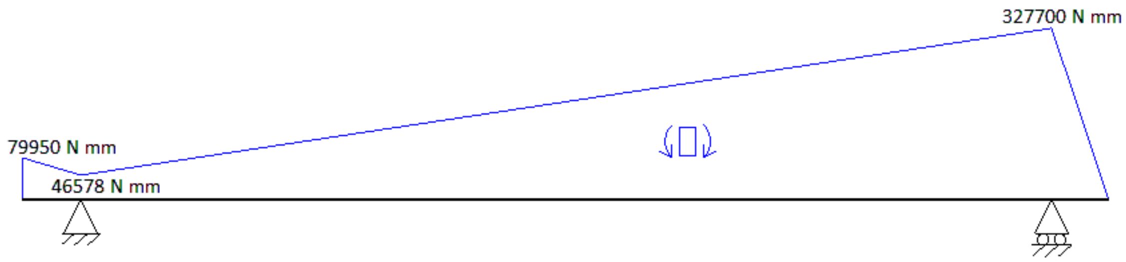


Figure 26Diagram of Z bendingmomentsonthe vertical shaft

In this case the biggest bending moment is where the lower bearing is, and its value is can be calculated as:

$$M_{D3} = \sqrt{{M_{DY3}}^2 + {M_{DZ3}}^2} = 329430\ N \cdot mm$$

The torque transmitted by the shaft is 196600N mm, so the minimum diameter for the shaft is

$$d_{min} = \sqrt[3]{\frac{4 \cdot n_s}{\pi \cdot \sigma_y} \sqrt{64M_{max}^2 + 48T_{max}^2}} = 21.99 \approx 22\ mm$$

Being 2 the safety coefficient (n_s) and 710 MPa the yield strength of the material (σ_y).

The bearings in that part of the shaft have to have a diameter of 30 mm so the diameter has also to be 30 mm. In the widest part is 35 mm diameter.

The maximum stress in the shaft is:

$$\sigma_{vm} = \frac{4 \cdot n_s}{\pi \cdot d^3} \sqrt{64M_{max}^2 + 48T_{max}^2} = 140\ MPa$$

The safety coefficient is:

$$n_s = \frac{\sigma_y}{\sigma_{vm}} = 5$$

After the calculations the shaft had to be modified because of the requirements of the bearings, that is why next to the top of it, its diameter is 46 mm.

2.4.1.1 Bearings

After calculating the loads and forces on the shaft, the values of the force to support by the bearings are known. The radial forces to support by the bearings are 2295 and 3698 N respectively while the maximum value to support by the axial bearing is 205 N.

They have been chosen from the SKF catalog.

<u>Radial bearing</u>

The chosen radial bearing is known as Y-bearing units with square flange. These units comprise:

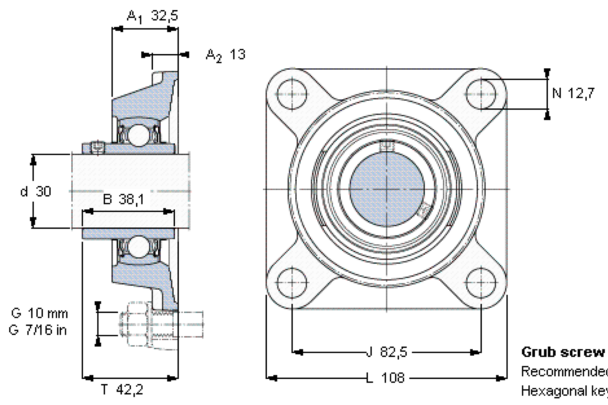
A "Y-bearing" (insert bearing) which is a single row deep groove ball bearing with convex sphered outside diameter.

A "Y-bearing housing", which has a correspondingly sphered but concave bore.

. Information about the bearing unit, housing, and Y-bearing can be read in the next figure:

Y-bearing units for high temperatures, flanged units, square flange, metric bearings
Product information

Dimensions					Basic load rating static	Mass	Designations Bearing unit	Housing	Bearing
d	A_1	J	L	T	C_0				
mm					kN	kg	-		
30	32,5	82,5	108	42,2	11,2	1,10	**FY 30 TF/VA201**	FY 506 U/VA228	YAR 206-2FW/VA201



Grub screw	M 6×0.75
Recommended tightening torque [Nm]	4
Hexagonal key size [mm]	3

Figure 27 Y-bearing FY 30 TF

Axial bearing.

The main function of this bearing is to keep the gear and the pinion together. Single direction thrust ball bearings, as their name suggests, can accommodate axial loads in one direction and thus locate the shaft axially in one direction. They must not be subjected to any radial load.

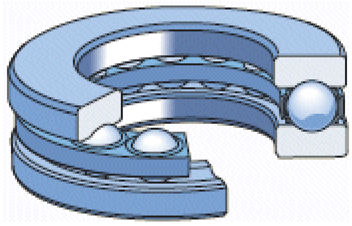


Figure 28 Axial bearing

A single direction thrust ball bearing has been chosen because the load to support is not too big. Its designation is 51104. Dimensions and product information are given in the next figure.

Thrust ball bearings, single direction
Product information

Tolerances , see also text
Recommended fits
Shaft and housing tolerances

Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Minimum load factor	Speed ratings Reference speed	Limiting speed	Mass	Designation
d	D	H	C	C_0	P_u	A				
mm			kN		kN	-	r/min		kg	-
20	35	10	15,1	29	1,08	0,0044	7500	10000	0,039	**51104**

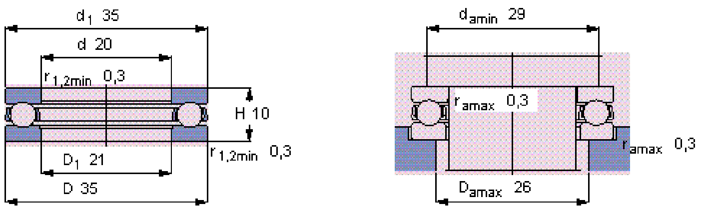


Figure 29 Thrust bearing information

2.4.2Pump shaft

The pump shaft is the one that transmits the angular velocity and the torque from the vertical shaft to the pump. At the same time it has to support the weight of the pump, which stands also the weight of the water.

Detailed calculations can be found in annex VI.

The weight of the pulley is 70 N while the tension of the rope is 230 N. This tension is supporting the weight of the water and the heavy piece at the bottom of the well. This makes possible the rope to have tension enough not to slide over the pulley.

The shaft is fixed to the tower with two bearings, one of the able to fix in X and Y axis at the same time while the other one supports only radial forces.

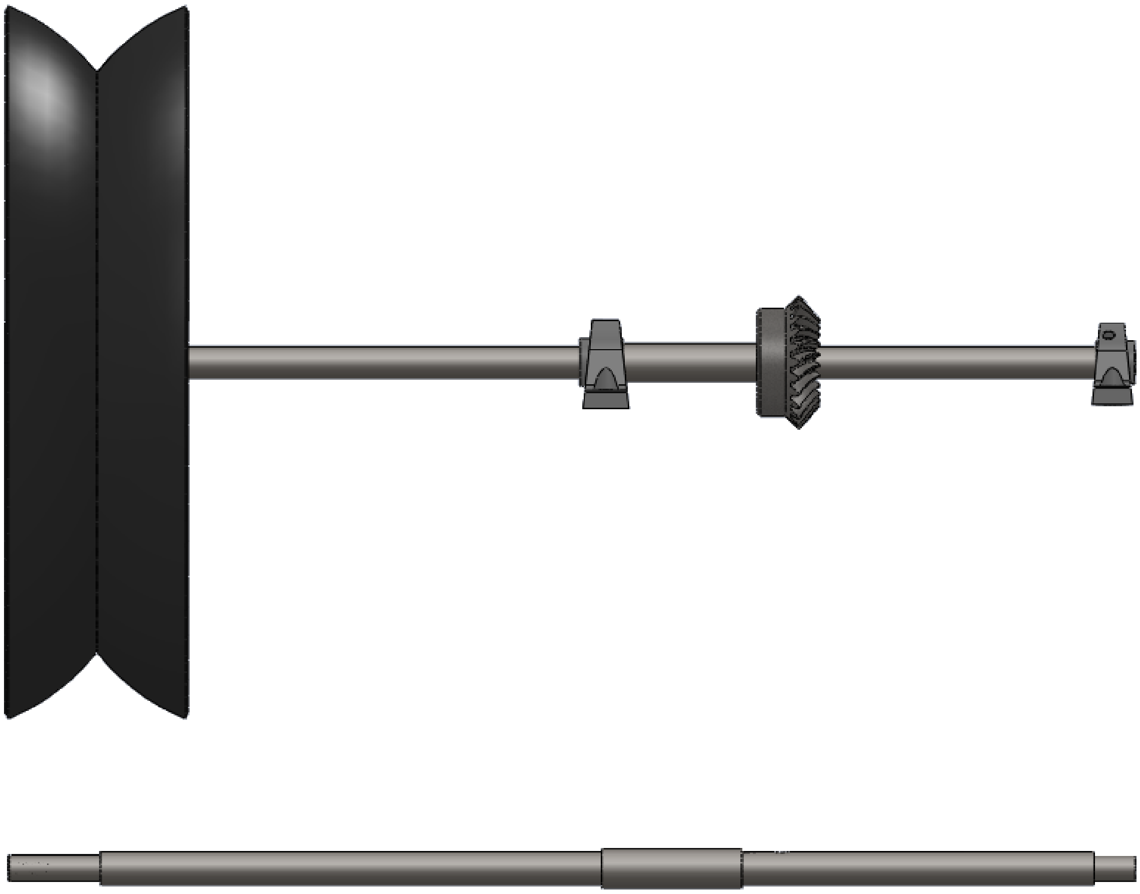


Figure 30: Pump shaft

The next diagram shows the bending moment in Z axis in the shaft:

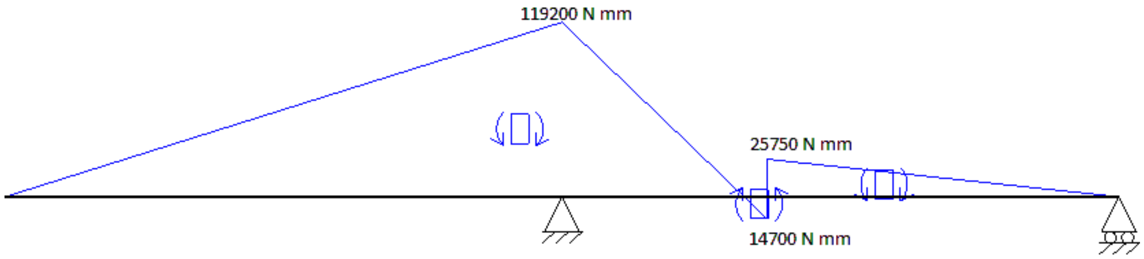


Figure 31Bendingmoments in Z axis onthepumpshaft

The next diagram shows the torque supported by the shaft:

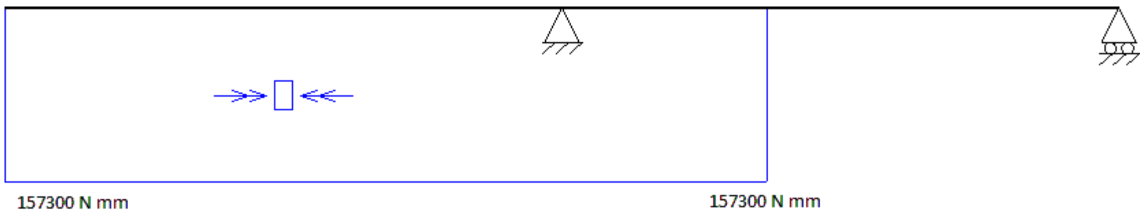


Figure 32 torque in thepumpshaft

As it can be seen in the diagram the maximum bending moment is 293200 N mm while the torque is 157300 N mm from the gear to the pulley. Diameter can be calculated as:

$$d_{min} = \sqrt[3]{\frac{4 \cdot n_s}{\pi \cdot \sigma_y} \sqrt{64M^2 + 48T^2}} = 21{,}01 \approx 22\ mm$$

The final diameter has to be 25 mm because of the hub diameter of the gear.

2.4.2.1 Bearings

After calculating the loads and forces on the shaft, the values of the force to support by the bearings are known. The bearings have to support 2325 and 1539N.

They have been chosen from the SKF catalog.

http://www.skf.com/portal/skf/home/products

Radial bearings.

Figure 33 Y bearingplummer block

The chosen radial bearings are known as Y-bearing plummer block units. These units comprise:

A “Y-bearing” (insert bearing) which is a single row deep groove ball bearing with convex sphered outside diameter.

A “Y-bearing housing”, which has a correspondingly sphered but concave bore.

Figure 35 Y bearing

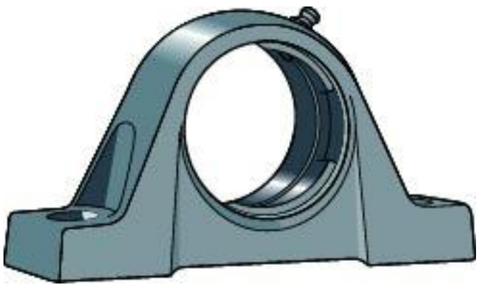


Figure 34 Y bearinghousing

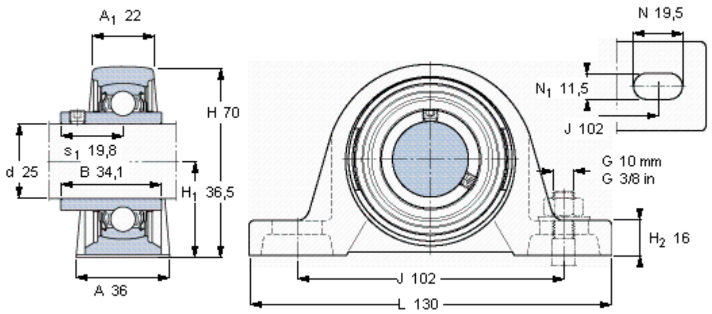
Y-bearing units are ready to mount, greased and sealed units which enable initial errors of alignment to be compensated for.

A support will be added to the bearings so they are easy to join to the nacelle at the proper height.

The needed diameter for the shaft was also known. The bearing that had to support bigger load is where the shaft is 25 mm wide. This bearing unit designation is SY 25 TF/VA201. Information about the bearing unit, housing, and Y-bearing can be read in the next figure:

Product information

Dimensions					Basic load rating static	Mass	Designations Bearing unit	Housing	Bearing
d	A	H	H_1	L	C_0				
mm					kN	kg	-		
25	36	70	36,5	130	7,8	0,72	**SY 25 TF/VA201**	SY 505 U/VA228	YAR 205-2FW/VA201



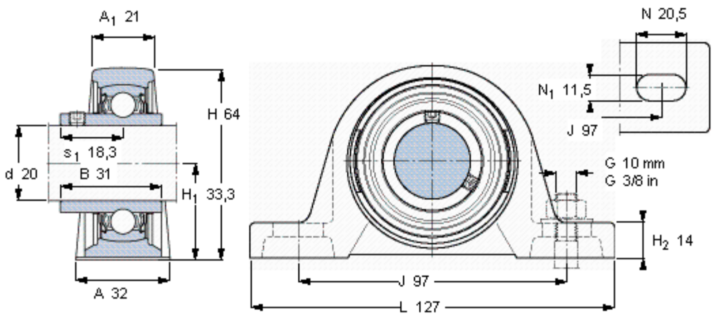
Grub screw	M 6×0.75
Recommended tightening torque [Nm]	4
Hexagonal key size [mm]	3

Figure 36 Y bearing information

The other radial bearing must support 1539 N and is installed where the shaft is 20 mm width. Its designation is SY 20 TF/VA201.

Product information

Dimensions					Basic load rating static	Mass	Designations Bearing unit	Housing	Bearing
d	A	H	H_1	L	C_0				
mm					kN	kg	-		
20	32	64	33,3	127	6,55	0,57	**SY 20 TF/VA201**	SY 504 U/VA228	YAR 204-2FW/VA201



Grub screw	M 6×0.75
Recommended tightening torque [Nm]	4
Hexagonal key size [mm]	3

Figure 37 Second Y bearing information

2.4.3 Gears

First couple of gears istransmitting the power from the rotor shaft to the next one. As the ratio is not 1:1 this kind of gears are called bevel gears. The bigger gear is called gear while the smaller one is called pinion. The pinion is on the horizontal shaft of the rotor and the gear is in the vertical shaft that goes to the pump. The pinion had to support a torque of 163N·m. The ratio is 1:1.5and the efficiency 80%.

Figure 38Bevelgears

The gears chosen are made by Quality Transmission Components. The website where can be found is: http://www.qtcgears.com/KHK/newgears/KHK218.html

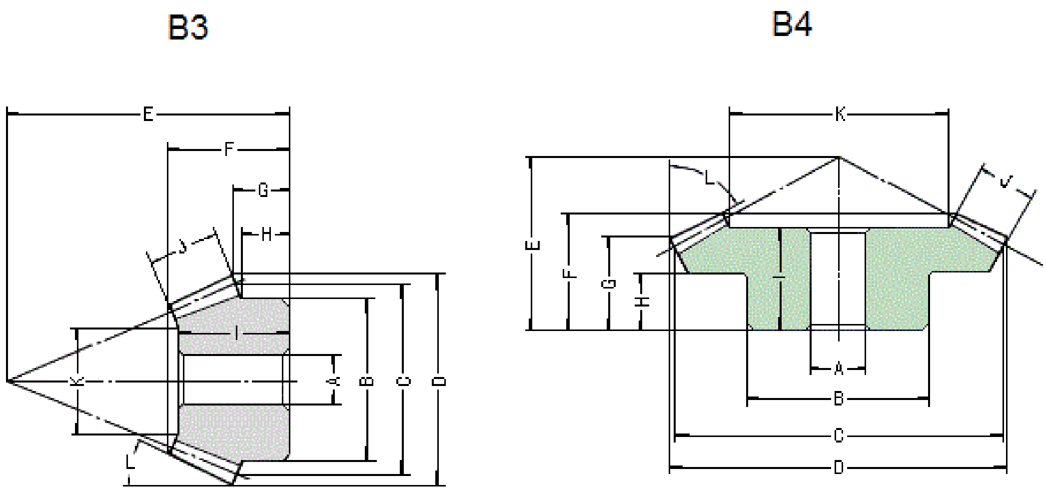


Figure 39 Bevel gears dimensions

Catalog No.	Module	Nº of teeth	A (mm)	C (mm)	J (mm)	Face angle L (°)	Shape	Allowable torque (N·m)
SBS5-3020R	5	30	25	150	30	59° 19’	B5	253.4
SBS5-2030L		20	22	100		38° 31’	B3	172.9

Table 6Bevel gears dimensions and characteristics

Specifications			
Precision grade	JIS B 1704 grade 4	Core hardness	HB165~194
Gear teeth	Gleason	Surface hardness	HRC48~53
Pressure angle	20°	Surface treatment	Black oxide
Helix angle	35°	Surface finish	Hobbed
Material	S45C (Carbon steel)	Datum reference surface for gear curring	Bore
Heat treatment	Induction hardened teeth		

Table7Bevelbearingsspecifications

The next pair is used to transmit the torque from the vertical shaft to the pump shaft. This time the ratio needed is 1:1. These gears are called miter gears. They have to support a torque of 197 N m. The gears chosen are made by Quality Transmission Components. The website where can be found is:

http://www.qtcgears.com/RFQ/default.asp?Page=../KHK/newgears/KHK180.html

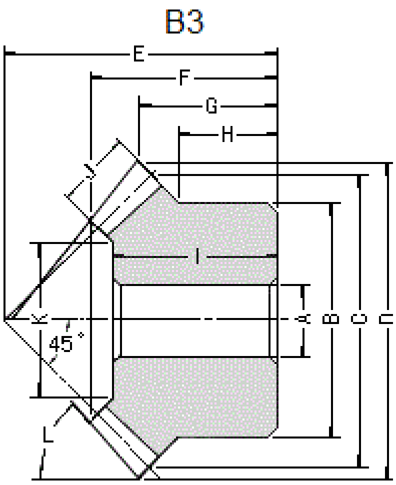


Figure 41 Miter gear dimensions

Figure 40Mitergears

Catalog No.	Module	Nº of teeth	A (mm)	C (mm)	J (mm)	Face angle L (°)	Shape	Allowable torque (N·m)
MMS4-25R	4	25	25	120	25	47° 48’	B3	238
MMS4-25L								

Table8Mitergearsdimensions

Specifications			
Precision grade	JIS B 1704 grade 4	Core hardness	HB 250-300
Gear teeth	Gleason	Surface hardness	HRC55-60
Pressure angle	20°	Surface treatment	Black oxide
Helix angle	35°	Surface finish	Cut
Material	SCM415 (Alloy steel)	Datum reference surface for gear curring	Bore
Heat treatment	Teeth induction hardened after carburizing		

Table9Mitergearsspecifications

2.5 Rope pump

The pump designed in this project is a rope pump. When the different kinds of pumps were studied to choose the best of them, most of them were discarded because of their complexity to be manufactured and be installed in a well. After a shallow study this table was written:

Type	Advantages	Disadvantages
Membrane	-Easier transportation -Easier to face the turbine into the wind -Longer lifetime	-Less deep -Difficult starting -Low efficiency
Rope	-Very deep -Less power needed -Can be connected to a second energy source easily -Simple construction -Cheaper materials	-More difficult transport -More complex orientation mechanism -Higher friction -Lower durability

Table10Advantages of ropepumps

A rope pump is a circuit between the water source and the desired level, using an endless rope with pistons. The rotation of a wheel moves the rope. This rope with pistons pushes the water column up at the top, and sucks another column of water below through a pipe made of PVC.

Some of its advantages are high efficiency, high reliability, it is able to pump a big flow and from deep wells, and above all, low cost and very easy to construct, install and maintain.

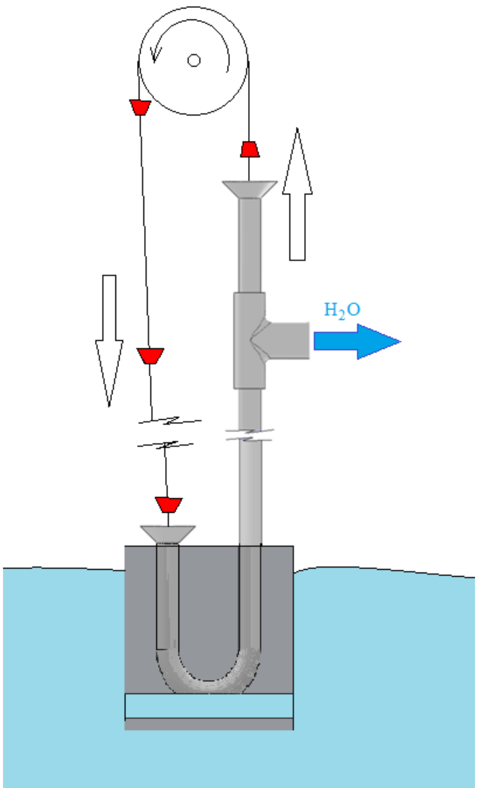
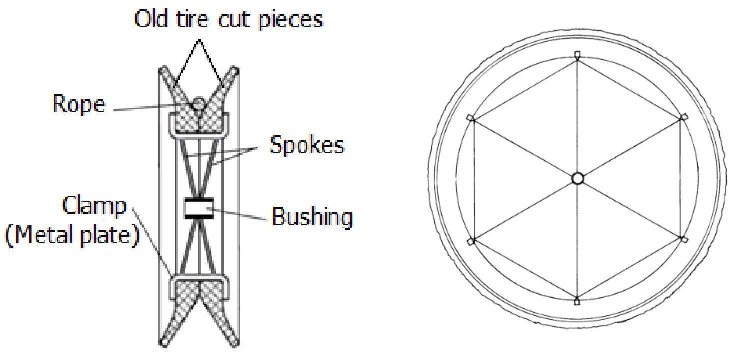


Figure 42Ropepump

A rope pump is composed of the following components:

Wheel (Pulley)

It transmits the input power to all the pump mechanism. The diameter of the wheel is 16 inches so old parts from cars can be used. A 16" tire is used to make the parts in contact with the rope. The parts which section can be seen in the picture below are the lateral parts of the tire, the ones closest to the rim. They have to be cut and joined in "V". For this, six clamps (metal plates) are used. It greatly improves the adhesion of the rope. Each metal plate is joined with two spokes to the central bushing by welding.



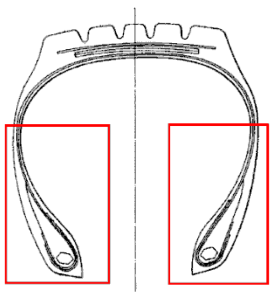


Figure 43 Part of the tyre needed

Figure 44Ropepumpwheel

Rope

A 5 mm diameter polypropylene fibre rope is used. It will support the weight of the water. It must be at least 30 meters long. Polypropylene is water resistant, this rope can support a load of 1600 N, but it is recommended to work with a load of 10% of the maximum capacity.Another good property is that it is not very smooth, so it will not slide on the wheel. The rope, together with the pistons, functions as an endless band transporting water. The pistons are located the entire length of the rope, and are attached with two knots, one in front of the piston and one directly behind it. When installing, the ends are attached by braiding. Knots are not used for this because they are difficult to untie when tautening the rope or for repairs. Once all the knots are installed and the pipes are also ready the pretension of the rope has to be between 20 and 70 N, approximately between 2 and 7 kgf.

Pipes

Figure 46 Pipes

Pumping pipes are a fundamental part of the rope pump. They vary according to the depth of the well. The deeper the well is, the smaller the diameter of the pipe. The chosen pipes have a nominal diameter of 32 mm, and an external of 40 mm. They are made of PVC, 3 meters long and threaded in both ends, so it is easy to join them. Four 3 meters pipes are needed to reach 11,5 m. The lower end should be bell-shaped to allow the movement of the rope and the pistons without damage them. This can be made heating the end of the pipe and pushing with a bottle. At the higher end a "Tee" connection has to be coupled to allow the diversion of water to the tank and avoid losses from the top.

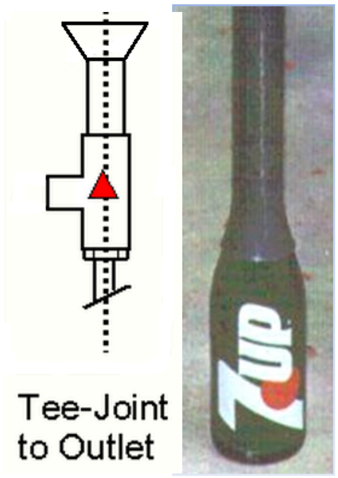


Figure 45Tee-jointtooultlet

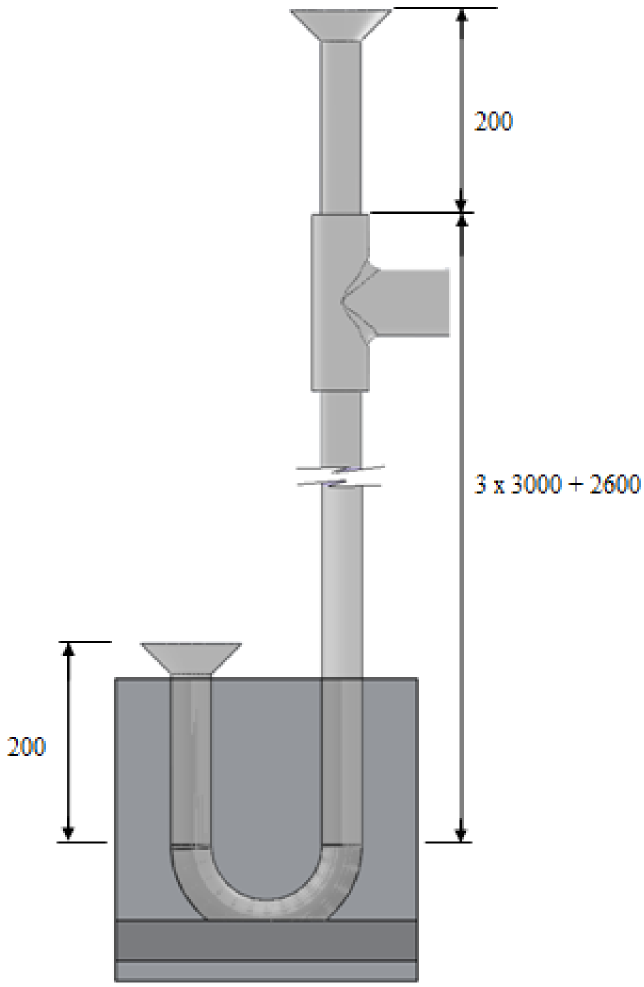


Figure 47Ropepump pipes dimensions

Guide

The guide is installed at the bottom of the well and is where the pumping process is initiated. Its function consists of guiding the rope with pistons so that it enters into the pumping pipe from below. It serves as well as a counterweight to tauten the rope in order to avoid sliding on the wheel. Therefore, the guide has various functions integrated into one piece.

The guide is a concrete box with a base piece, an entry pipe, a pumping pipe, a PVC elbow 180 degree and a transversal cavity. These parts of the guide must be made in such a way that the rope never touches the concrete, which would cause wear to it as well as to the pistons. The entry and pumping pipes on the guide have a wide mouth to facilitate the entry of the rope and pistons. The water enters the guide through the transversal cavity. The guide is placed at 50 cm from the bottom of the well. This allows taking water up to 50 cm deep.

Pistons

The pistons are one of the most sensitive parts of the pump. Together with the rope they form an endless chain. When the rope rotates it leads the piston through the pumping pipe, pushing the water inside upwards.

The piston is a cone shaped part to reduce friction, with a hole for the rope.

They are made of polyethylene; which is very easy to get in the market. A perfect fit is required between the pistons and the pumping pipe. The space between piston and inner wall of the pipe is around 0.50 mm, which is large enough to avoid friction and small enough to avoid loss of water. So the diameter is 31 mm and the distance between two pistons is around two meters.

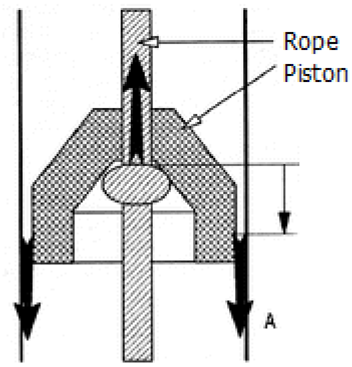


Figure 48Pistonsmountingsystem

3 Manufacture

3.1 Blades

The blades are handmade. There are nine sections defined. The profile is a NACA 23012. The nine sections define the angle and the chord of each section. It is also defined the distance where they are from the end of the blade.

Section number	1	2	3	4	5	6	7	8	9
Distance	0	0,12	0,33	0,54	0,76	0,97	1,18	1,39	1,5
Chord	0,23	0,39	0,30	0,23	0,19	0,16	0,14	0,12	0,117
Angle	30,4	21,66	12,973	8,19	5,18	3,12	1,3	0,5	0

Table11Dimensions of theblade

The join parts need a CNC work, starting with a 230x130x15 or bigger part. It has to be mechanized to get the next dimensions:

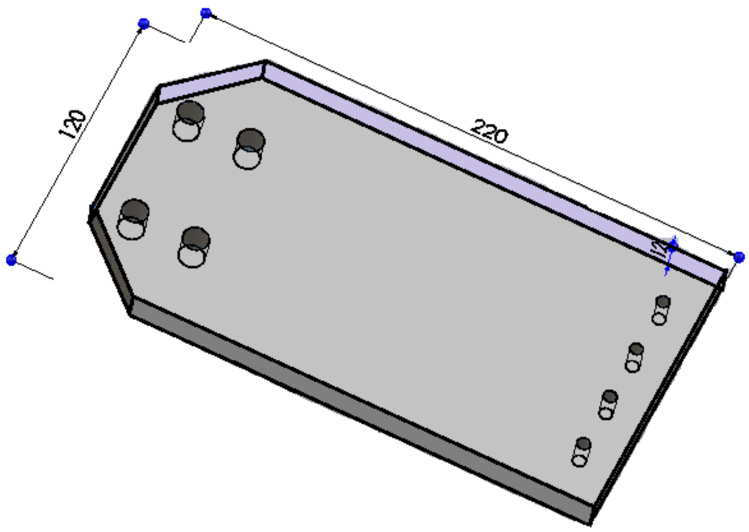


Figure 49Join part before being twisted

Then it has to be twisted 30° the middle area.

3.2 Nacelle

The nacelle is made by two pieces: the base of the nacelle and the cover.

The base of the nacelle is made by three steel parts welded. First of it is a 10 mm wide part. The front sheet is a 2 mm wide sheet that has to be cut and welded. To protect the bearings there is also a cylinder that has to be welded under the tower. Some holes have to be made by drilling process.

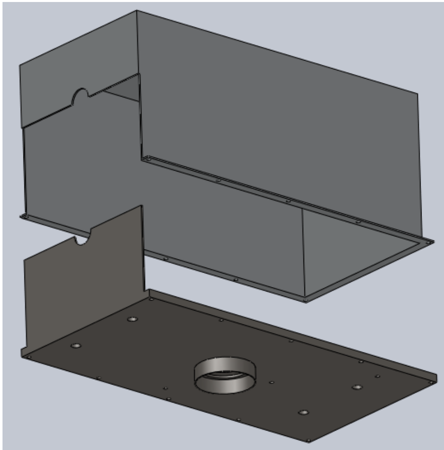


Figure 50Nacelle box

The cover is made by a 2 mm wide aluminum sheet that has to be bended and welded. T has also some holes that have to be made with a drill.

3.3 Transmission

The shafts of the transmission have to be made with CNC. The round shapes with lathe and the keyway with milling machine. In the rotor shaft, the round plate where the blades are screwing has to be welded to the shaft. This welding has only to support the torque because the force of the wind is pushing it against the shaft.

3.4 Tower

For the tower, the ISO tubes have to be bought and bend and weld them together. The welding will take place in Israel itself because transporting an already welded tower from another country is more difficult, time demanding and thus more expensive.

Also other parts have to be welded. The parts that support the pipes have to be welded at 0.5 m high and 1976 mm high from the floor. There is one part to support the radial bearing that fix the shaft to the center of the tower.

After this work the second part consists in installing some screws that fix the two parts of the tower.

3.5 Water pump

3.5.1 Pistons

Piston production requires a small plastic injection machine, and moulds. Polyethylene is poured in the injection machine hopper. As the plastic passes through the heated hopper bottom, it becomes fluid and is injected into the mould. As it cools, the plastic adopts the mould's form.

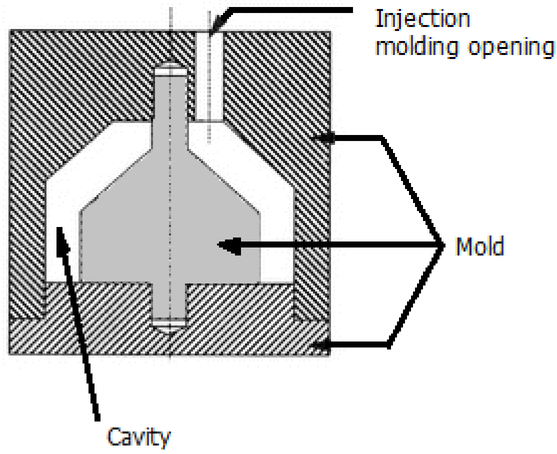


Figure 51Pistonsmoulds

3.5.2 Wheel

The wheel is made from the two 16” tires usually used by cars, buses and trucks. The lateral parts of the tire (the ones closest to the rim) are cut and joined in “V”. For this, six clamps are used. These clamps squeeze and join the two rims together. There are two spokes on each clamp. They must be welded in one side to the clamps and in the other to the central bushing. Finally, this bushing can be fixed to the shaft with a key.

3.5.3Guide

The production process includes the following steps:

-Cut the final extreme of the elbow (The water will flow through this cut).

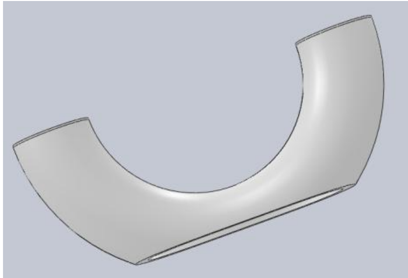


Figure 52 Guide construction 1

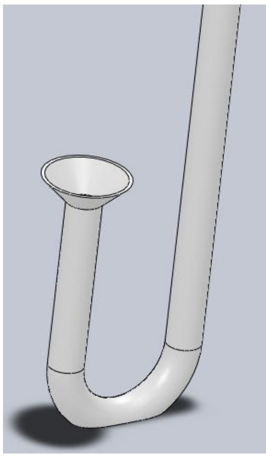


Figure 53 Guide construction 2

- Join the entry pipe with one of the ends of the elbow and the pumping pipe with the other.

Place a piece of polystyrene (or a similar material easy to break) covering the cut made before in the elbow.

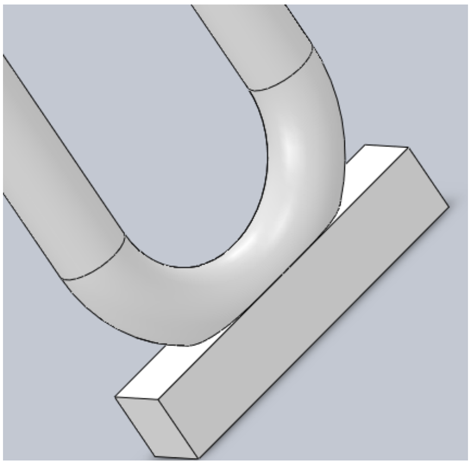


Figure 54 Guide construction 3

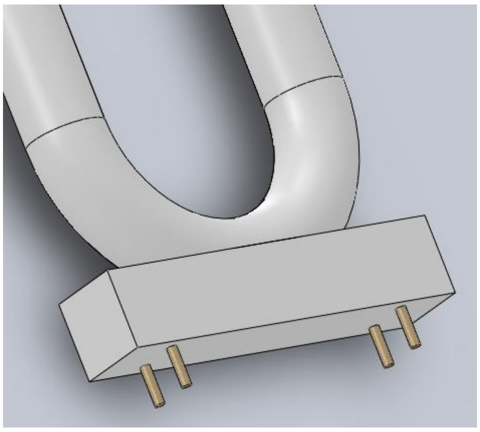


Figure 55 Guide construction 4

- Nail four sticks on the other side of the polystyrene like legs to support the guide.

- Place the guide in a wooden box and fill it with concrete.

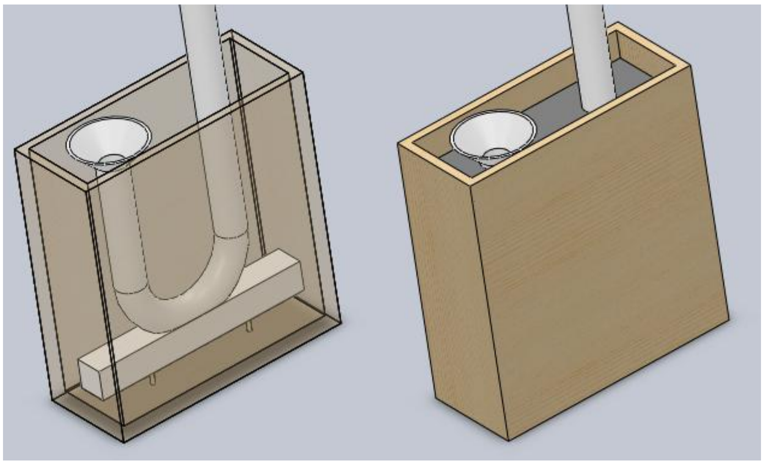


Figure 56 Guide construction 5 Figure 57 Guide construction 6

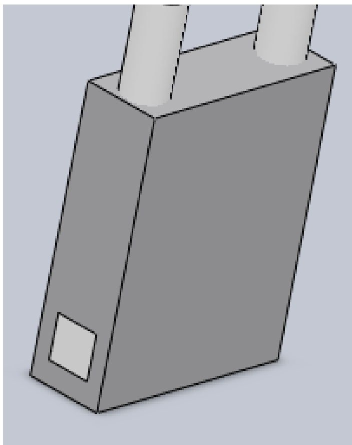


Figure 58 Guide construction 7

- When the concrete dries, remove the guide from the box.

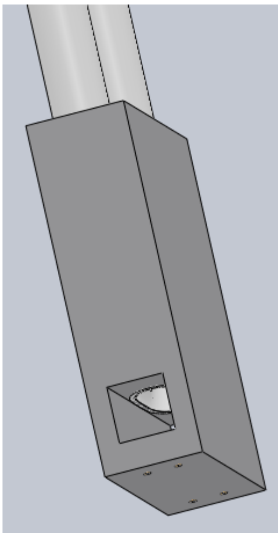


Figure 59 Guide construction 8

- Remove all the polystyrene from the cavity
- Test the quality of the guide with the help of a water container, a rope and piston.

4 Maintenance

Maintenance
The maintenance operations are very simple, and can be divided into:

Proper rope tension
It might be necessary to correct the tension in the rope during the first weeks of use, as the knots tend to lengthen the rope. Lack of tension in the rope will cause the rope to slip over the pulley wheel. To alter the tension in the rope, remove the rope and untie it. Put the rope back on the pulley wheel with the desired tension to measure where the new knot must be made, and repeat the process of braiding.

Greasing
Oil or grease the bushings, the bearings and the gears when considered necessary. They work at very low speed, but it will extend their useful life. Any type of oil or grease can be used to do this.

Pipes
Join parts of pipes should be checked to avoid water losses, and make sure there isn't any obstructing object inside.

Cleaning and painting
To prevent corrosion, clean and paint the metal parts occasionally. Blades should be painted with an outdoors varnish to protect the wood from the wheatear conditions.

The most frequent repairs, with a range of 18 to 24 months, are changing the pistons and the rope.

Pistons	Approx. 18 months
Rope	Approx. 18 months
Pipes	More than 48 months
Guide	More than 48 months
Wheels/Pulleys	More than 48 months

Table 12 Table of rope pump maintenance frequency

Change the rope when it is in very bad conditions or damaged.

5 Economic balance

To estimate the price of each component, the average price of the material which they are made of is looked. Then, knowing how much material these components will need, the final price can be determinate.

The average wage for a worker in Israel was searched in the Internet. The workers will receive 0.10 €/hour for manufacturing and paint all the components.

The following table shows all the manufactured pieces:

	Concept	Price	Per unit	Qty.	Total (€)
Blade	Scotch Pine	98.78 €/m^3	0.12 m^3	x3	32
	Manufacture operation	0.10 €/hour	10 hours	x3	3
	Paint operation	0.10 €/hour	1 hour	x3	0.30
Join Blade	1023 Carbon Steel Sheet (SS)	0.593 €/kg	3 kg	x3	5.35
	Manufacture operation	0.10 €/hour	4 hours	x3	1.2
	Paint operation	0.10 €/hour	1 hour	x3	0.30
Plate	1023 Carbon Steel Sheet (SS)	0.593 €/kg	3 kg	x1	1.8
	Manufacture operation	0.10 €/hour	3 hours	x1	0.30
	Paint operation	0.10 €/hour	1 hour	x1	0.10
Rotor Shaft	AISI 4340 Steel ø32 mm, normalized	69 €/m	0.61	x1	42
	Manufacture operation	0.10 €/hour	1 hour	x1	0.10
	Paint operation	0.10 €/hour	1 hour	x1	0.10
Ø20 Bearing support	1023 Carbon Steel Sheet (SS)	0.593 €/kg	3.5 kg	x1	2.1
	Manufacture operation	0.10 €/hour	2 hours	x1	0.20
	Paint operation	0.10 €/hour	1 hour	x1	0.10
Ø25 Bearing support	1023 Carbon Steel Sheet (SS)	0.593 €/kg	2.6 kg	x1	1.55
	Manufacture operation	0.10 €/hour	1 hour	x1	0.10
	Paint operation	0.10 €/hour	1hour	x1	0.10
Plate ø35	1023 Carbon Steel Sheet (SS)	0.593 €/kg	0.04 kg	x1	0.05
	Manufacture operation	0.10 €/hour	1 hour	x1	0.10
	Paint operation	0.10 €/hour	1 hour	x1	0.10

	Concept		Price	Per unit	Qty.	Total (€)
Nacelle Up	Aluminum (1060 Alloy)		1.8 €/kg	2.8 kg	x1	5.1
	Manufacture operation		0.10 €/hour	2 hours	x1	0.20
	Paint operation		0.10 €/hour	1 hour	x1	0.10
Nacelle Bottom	1023 Carbon Steel Sheet (SS)		0.593 €/kg	8.3 kg	x1	5
	Manufacture operation		0.10 €/hour	3 hours	x1	0.30
	Paint operation		0.10 €/hour	1 hour	x1	0.10
Tail tube	1023 Carbon Steel Sheet (SS)		0.593 €/kg	2.8 kg	x1	1.70
	Manufacture operation		0.10 €/hour	1 hour	x1	0.10
	Paint operation		0.10 €/hour	1 hour	x1	0.10
Tail	Aluminum (1060 Alloy)		1.8 €/kg	5.7 kg	x1	10.30
	Manufacture operation		0.10 €/hour	1 hour	x1	0.10
	Paint operation		0.10 €/hour	1 hour	x1	0.10
Truss	Stainless Steel (ferritic)		0.85 €/kg	117 kg	x1	99.45
	Manufacture operation		0.10 €/hour	50 hours	x1	5
	Paint operation		0.10 €/hour	2 hours	x1	0.20
Vertical Shaft	AISI 4340 Steel ø50 mm, normalized		109 € /m	2 m	x1	218
	Manufacture operation		0.10 €/hour	1 hour	x1	0.10
	Paint operation		0.10 €/hour	1 hour	x1	0.10
Ground Fixations	1023 Carbon Steel Sheet (SS)		0.593 €/kg	0.54 kg	x4	1.30
	Manufacture operation		0.10 €/hour	1 hour	x4	0.40
	Paint operation		0.10 €/hour	1 hour	x4	0.40
Wheel	Cylinder + Clamps + Spokes	1023 Carbon Steel Sheet (SS)	0.593 €/kg	1 kg	x1	0.593
		Manufacture operation	0.10 €/hour	4 hours	x1	0.40
		Paint operation	0.10 €/hour	1 hour	x1	0.10
	Tire	Used tire	-	-	x1	0
		Manufacture operation	0.10 €/hour	1 hour	x1	0.10
Pistons	Polyethylene		0.70 €/kg	0.010 kg	x15	0.15
	Manufacture operation		0.10 €/hour	2 hour	x1	0.20
	Manufacture operation		0.10 €/hour	1 hour	x1	0.10
	Paint operation		0.10 €/hour	1 hour	x1	0.10

	Concept	Price	Per unit	Qty.	Total (€)
Pump Shaft	AISI 4340 Steel ø35 mm, normalized	48 € /m	0.9 m	x1	44
Plate ø40	1023 Carbon Steel Sheet (SS)	0.593 €/kg	0.05 kg	x1	0.03
	Manufacture operation	0.10 €/hour	1 hour	x1	0.10
	Paint operation	0.10 €/hour	1 hour	x1	0.10
Guide	PVC Elbow	4.20	4.20	x1	4.20
	Wood box	-	-	x1	0
	Polystyrene	-	-	x1	0
	Stick	-	-	x4	0
	Manufacture operation	0.10 €/hour	1 hour	x1	0.10
			TOTAL		489.373 €

To complete the fabrication of the wind watered water pump the next commercial components are needed:

Description	Code	Quantity	Cost/ unit (€/u)	Cost (€)
Hexagonal Bolt	M5X50 DIN 933	44	0.12	5.28
Hexagonal Bolt	M6X25 DIN 933	2	0.05	0.1
Hexagonal Bolt	M8X80 DIN 933	14	0.60	8.4
Hexagonal Bolt	M10X110 DIN 933	4	0.84	3.36
Hexagonal Bolt	M10X50DIN 933	12	0.22	2.64
Hexagonal Nut	M5 DIN 934	44	0.04	1.76
Hexagonal Nut	M8 DIN 934	14	0.06	0.84
Hexagonal Nut	M10 DIN 934	24	0.09	2.16
Washer	M5 DIN 125	88	0.03	2.64
Washer	M8 DIN 125	28	0.04	1.12
Washer	M10 DIN 125	40	0.05	2
Threaded rod	M10 L: 1M	1	1.37	1.37
Bearing unit	SKF SY 20 TF	2	27.20	54.4
Bearing unit	SKF SY 25 TF	2	32.30	64.60
Bearing	SKF 51104	1	10	10
Bearing	SKF 51108	1	18.80	18.80
Bearing	SKF 81106 TN	1	39.90	39.90
Bearing	SKF FY 30 TF	1	38.90	38.90

Description	Code	Quantity	Cost/ unit (€/u)	Cost (€)
Bevel Gear (QTC Gears)	SBS5-3020R	1	215	215
	SBS5-2030L	1	141	141
Miter Gear (QTC Gears)	MMS4-25R	1	105	105
	MMS4-25L	1	105	105
Key	DIN 6885 6x6x12	3	0.15	0.45
Key	DIN 6885 8x7x12	2	0.15	0.30
Rope	5 mm polypropylene	30	0.50	15
Pipe	3m Rigid threaded pipe (40 mm)	5	4.25	21.25
PVC Tee	32x32x32 Threaded	1	2.60	2.60
Concrete	Concrete Portland bag (25 kg)	1	2.46	2.46
			TOTAL	866.33 €

PART	COST
Manufactured pieces	489.373 €
Comercial components	866.33€
Total	**1355,70€**

6 Technical considerations

This part of the report is about different problems that have happened during the process and how have they been solved. Here are shown the main problems:

1. The transmission was supposed to be two pulleys and one belt joining them; this kind of transmission did not allow the rotor change its orientation 360°. The idea was designing a vertical shaft that could transmit the torque and the power to the pump. This means that instead of one step (from the rotor shaft to the pump shaft) there are two steps between the rotor and the pump (from the rotor shaft to a vertical shaft and from the vertical shaft to the pump shaft). This decreases the efficiency of the transmission.
2. This kind of transmission has one problem. When a torque is transmitted to a shaft, that shaft transmits a torque in opposite direction with the same value to the shaft that transmitted first. This is the same effect that happens in the helicopters. In the helicopters there is another rotor that controls the direction of the helicopter but in this case something cheaper and simpler was needed. The solution of this problem was changing the orientation of the tail creating a moment with the same value but opposite direction.
3. In order not to decrease the efficiency that much friction wheels were thought as a way to do what gears do. The problem with friction wheels is that they need a big force not to slide. If they work with the force of the wind, they should be 400 mm diameter not to slide.This meant a very big nacelle that would slow down the wind. Finally gears were decided to be the best way to transmit the power and the torque, although the efficiency is not as good as the friction wheels neither the price.
4. The forth problem was the braking system. All the ideas about this were discarded. The one that looked better was a bicycle brake acting on the pulley of the pump but it could be also in the middle of the vertical shaft. Before starting the calculations the tail was supposed to break it, turning it from the wind at higher speeds but as it has been demonstrated it will not. This problem has not been solved because we did not have ideas or time enough.

5. The truss was made later than the nacelle and the rotor and when they were mounted there were problems because the blades were hitting the truss. To solve this problem, three actions were done. The shaft was made longer and all the calculations were made again, changing the diameter as well as the material. Also the distribution inside the nacelle was changed in order to have the bearing between the rotor and the gear as near the wall as possible. Last change was the dimensions of the tower, they used to be 500x500 instead of 400x400 in the base of it.
6. Once the upper pulley of the pump was designed, the shaft to connect it to the transmission was the following step. The problem was that the height was the water outlet pipe was 2.1 m, and the pulley is 402 mm diameter. That means that at least the height of the shaft should be 2302 mm from the floor. The center of the rotor was 4050 mm height and the blades are 1700 mm long from the center of the rotor. There are less than 50 mm were the shaft is, and that distance is too small to be sure that if something is not correctly mounted, it could break itself.
7. Once the tower was joined to the nacelle another problem cropped up. There was no space enough to fit the bar that supports the tail. This problem was solved designing a new join between the nacelle and the tower, the one that is working in the final design.
8. The last problem found is that the separating between the rotor shaft and the vertical shaft is bigger than the force of the weight of the whole nacelle with rotor and tail. This problem appears only when the wind speed is very faster than usual, the wind speed that has been used for the resistance calculations: 13.5 m/s. To solve this problem an additional weight is needed over the nacelle. Also if the bevel gears were installed were the miter gears are and vice versa that force would have been smaller, but we did not have time to change them are recalculate the shafts dimensions.
9. The gears used are very expensive. The reason why these gears are needed is that the torque is very high in comparison to the power. If the angular velocity had been faster, with the same power the torque would have been much smaller. One possible way to improve this characteristic could be designing a rotor with a higher tip speed ratio.

7 Conclusion

Before starting making this project our meaning was creating something new and helpful. The idea was designing a new kind of water pump with a very low cost and a high efficiency.

The flow that this wind powered water pump can raise is bigger than other pumps that were shown to us before, and also the distance that it raises is also longer. We started watching an example of a membrane pump able to raise about3% of the flow that this pump can raise, and only 3 meters high, needing a stronger wind at the same time. This demonstrates that the idea of connecting a rope pump to a wind turbine is good, but some developing ideas are needed in order to make this project real.

After doing all the calculations and make them fix, a lot of expensive parts have been getting more and more needed; especially in the transmission. The problem with the transmission is that although the power is not very high, the rotor has a slow angular velocity which means that the torque is very high.But even with these expensive pieces this pump is 126 times more powerful than the one we had as example. We do not know the price of that pump but we think that this kind of turbine is much cheaper if you compare the final output of water.

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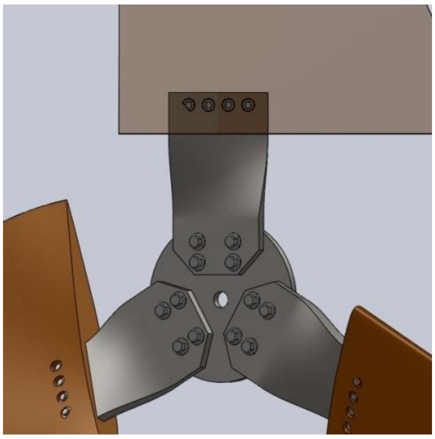
http://www.metalprices.com/index.asp

Annex I: Join parts

To calculate the joins between the blades and the shaft some previous equations and calculations are needed. For a critical wind speed of 15 m/s there are the next values on the blades.



Axial force = 1071.89 N

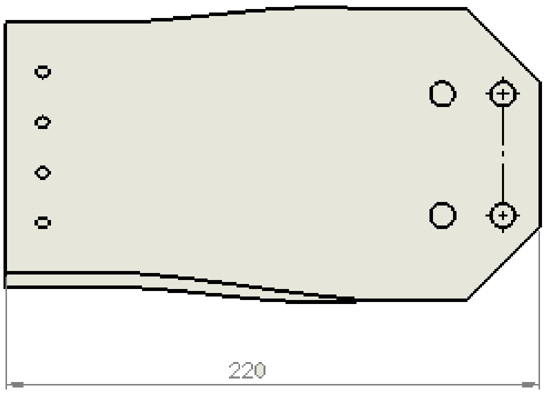
Rotational speed (ω) = 49.125 rad/s

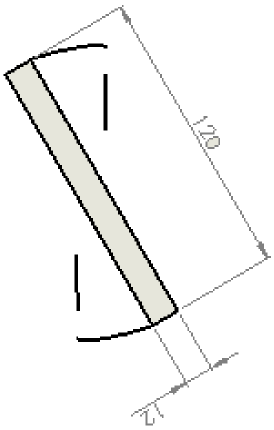
Power (P)= 9931 W

Those values have been calculated using the Blade Element Momentum (BEM) theory.

$$Torque = \frac{P}{\omega} = \frac{9931}{49.125} = 202.3\ N \cdot m$$

We have three blades, so the torque generated by each one (T_b) is 67.44 N·m





There are three forces acting in this piece: centripetal force of the blade, axial force and force created by the torque.

Centripetal force (F_c):

Blade mass = 2.4 kg

Distance between the blade centre of gravity to the rotor centre (r)= 0.316 m

$$F_c = m \cdot r \cdot \omega^2 = 1845\ N$$

Axial force in one blade (F_a):

$$F_a = \frac{Axial\ force}{3} = \frac{1071.89}{3} = 357.3\ N$$

Force generated by torque (F_t):

$$F_t = \frac{T_b}{a} = \frac{67.44}{0.2} = 337.18\ N$$

Once these three forces are known stresses can be also calculated.

Yield stress (σ_y) = 282 Mpa

Cross-sectional area (A) = w · t = 1440 mm^2

Inertia (I) = $\frac{1}{12} \cdot w \cdot t^3$ = 17280 mm^4

Moment of resistance (W_y)=$\frac{I}{y_{max}} = \frac{I}{t/2} = 2880$ mm^3

Shear force (V) =$\sqrt{F_t^2 + F_a^2} = 491.3\ N$

Axial force in this piece (N) = F_c = 1845 N

Momentum (M) = $F_a \cdot b$ = 357.3 · 0.93 = 332600 N · mm

Shear stress (τ)=$\frac{V}{A}$ = 0.341 MPa

Axial tension (σ) = $\frac{N}{A} + \frac{M}{W_y}$ = 117 MPa

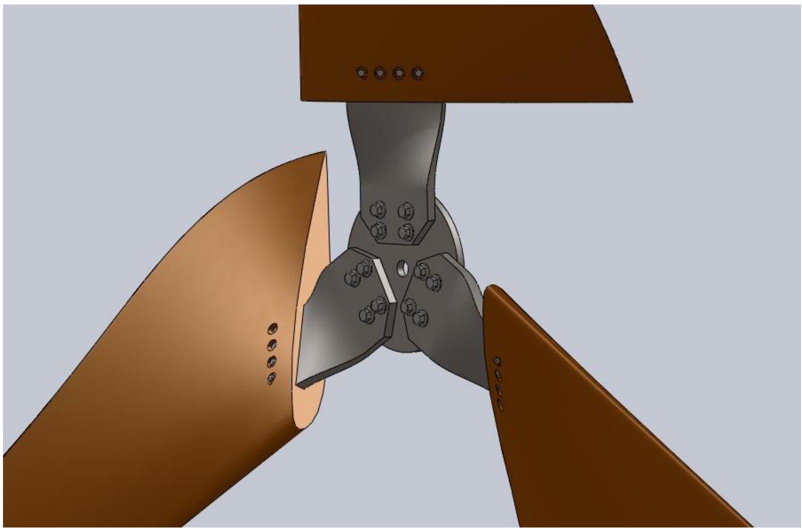
Von Mises stress (σ_{VM}) = $\sqrt{\sigma^2 + 3 \cdot \tau^2}$ = 117 MPa

Safety coefficient (n_s) = $\frac{\sigma_y}{\sigma_{VM}}$ = 2.42

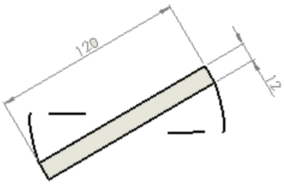
Annex II: Rotor bolts calculations

In this annex the calculation of the bolts that appear in the rotor are going to be calculated.

The pieces studied are those that join the blades to the shaft. There is one piece for each blade that joins it to a disc welded to the shaft.



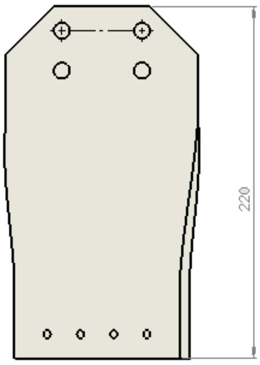
First of all the blots that join the blade to the join piece are going to be calculated. In this case there are four bolts in a row and they do not have to support any bending moment, only the centrifugal force and the weight of the blades. The mass of the blades are 2.38 kg each and the maximum angular velocity expected is 49.125 rad/s. The radio of the turn is 0.316 m. The bolts have a yield stress of 260 MPa.



The centrifugal force can be calculated as:

$$F_c = m_{blade} \cdot r \cdot \omega^2 = 1821.45\ N$$

But the critical moment is when the blade is under the rotor because the centrifugal force has to be added to the gravity force that can be calculated as:



$$F_g = m_{blade} \cdot g = 23.43\ N$$

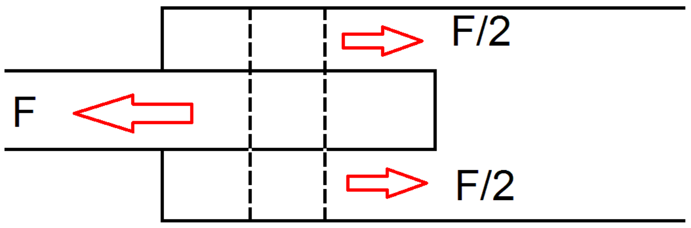
So the force to consider in the following studies is the sum of both:

$$F_{max1} = F_g + F_c = 1844.88\ N$$

There are four bolts with a diameter of 5 mm so the section of all of them can be calculated as:

$$A_{4,5} = n \cdot (\frac{d_1}{2})^2 \cdot \pi = 78.53\ mm^2$$

Being n the number of bolts.



There are three studies that have to be done to calculate the bolts size.

- <u>Shear failure:</u>

$$\tau = \frac{F_{max1}/_2}{A_{4,5}} = 11.74\ MPa$$

Safety coefficient

$$c_s = \frac{\sigma_y}{2 \cdot \tau} = 38$$

- <u>Join piece failure</u>

Stress area: $A_1 = w \cdot t - n \cdot d \cdot t = 1200\ mm^2$

Stress: $\sigma = \frac{F_{max1}}{A_1} = 1.54 MPa$

Safety coefficient: $c_s = \frac{\sigma_y}{\sigma} = 183$

- <u>Bolt flattening failure</u>

Stress area: $A_2 = n \cdot d_1 \cdot t = 240\ mm^2$

Stress: $\sigma = \frac{F_{max1}}{A_2} = 7.7\ MPa$

Safety coefficient: $c_s = \frac{\sigma_y}{\sigma} = 33$

- <u>Blade failure</u>

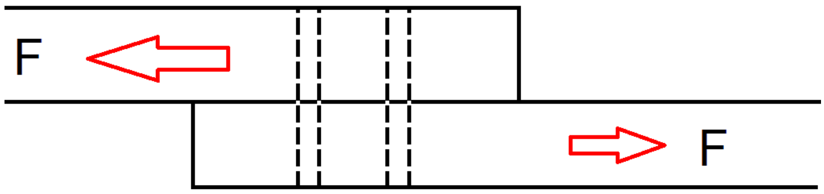
Stress area: $A_3 = A_{wood} - A_{bolts} = 1425\ mm^2$

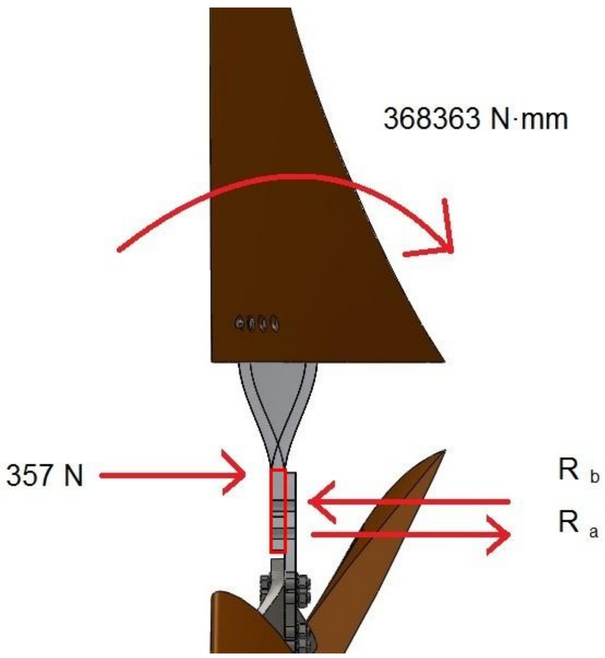
Stress: $\sigma = \frac{F_{max1}/2}{A_3} = 0.64\ MPa$

Safety coefficient: $c_s = \frac{\sigma_y}{\sigma} = 92$

The safety coefficients are so high because the maintenance of the turbine is not going to be very frequent so some bolts may lose between the checking.

The second part of this annex is the study of the belts that support the forces between the disc welded to the shaft and the join piece bolted to the blades.





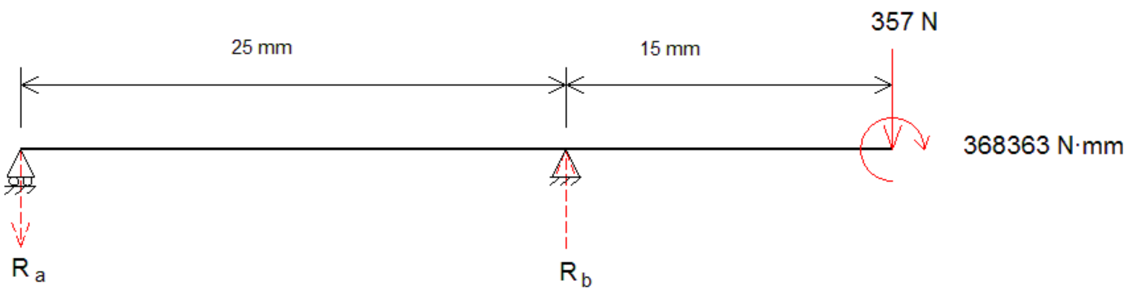
This case has four bolts distributed in two rows and the diameter of the bolts is 10 mm. Also the mass changes because the forces that the bolts have to support include the weight and the centrifugal force of the pieces where they are bolted. Each piece has a mass of 2.4 kg.

$$F_{max2} = m \cdot r \cdot \omega^2 + m \cdot g = 2672.3\ N$$

$$A_{2\ bolts} = 2 \cdot (\frac{d_2}{2})^2 \cdot \pi = 157\ mm^2$$

$$A_{4\ bolts} = 4 \cdot (\frac{d_2}{2})^2 \cdot \pi = 314\ mm^2$$

Before starting the calculation a forces analysis has to be done. This analysis consists in a beam that represents the part of the piece that is next to the disc bolted to the rotor.



$$\sum F_v = 0 \rightarrow R_b - R_a - 357 = 0$$

$$\sum M_z^a = 0 \rightarrow 25 \cdot R_b - 357 \cdot 40 - 368363$$

$$R_b = 15906\ N$$

$$R_a = 15549\ N$$

- <u>Shear failure:</u>

$$\tau = \frac{F_{max2}}{A_{4\ bolts}} = 17.02\ MPa$$

Safety coefficient

$$c_s = \frac{\sigma_y}{2 \cdot \tau} = 15$$

- <u>Bolt flattening failure</u>

Stress area: $A_4 = n \cdot d_2 \cdot t = 240\ mm^2$

Stress: $\sigma = \frac{F_{max}}{A_4} = 11.13\ MPa$

Safety coefficient: $c_s = \frac{\sigma_y}{\sigma} = 23$

- <u>Bolt tensile failure</u>

  Stress area: $A_{2\ bolt} = 157\ mm^2$

  Stress: $\sigma = \frac{R_a}{A_{2\ bolts}} = 99\ MPa$

  Safety coefficient: $c_s = \frac{\sigma_y}{\sigma} = 2.84$

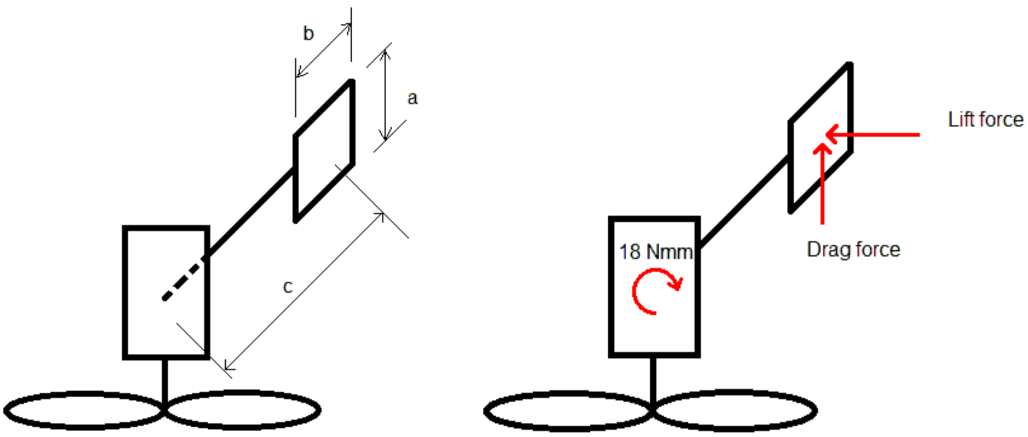
- <u>Von Mises</u>

  VM stress: $\sigma_{vm} = \sqrt{\sigma^2 + 3 \cdot \tau^2} = 103\ MPa$

  Safety coefficient: $c_s = \frac{\sigma_y}{\sigma_{vm}} = 2.51$

Annex III: Tail

This annex is about the tail calculations. This tail was supposed to create a momentum to face the rotor into the wind.



Those drawings show the forces that are going to act and also the dimensions.

First of all, the formulas about drag and lift forces have to be explained. The next values are defined as:

- F_d: Drag force
- F_l: Lift force
- ρ: air density
- A: area of the tail facing the wind
- C_d: coefficient of drag
- C_l: coefficient of lift

$$F_d = \frac{1}{2} \cdot \rho \cdot A \cdot C_d \cdot v^2$$

$$F_l = \frac{1}{2} \cdot \rho \cdot A \cdot C_l \cdot v^2$$

The coefficients of drag and lift have been calculated in an experiment with the wind tunnel, their values are:

C_d=1.6

C_l=1.5

$$b = a \cdot 1.625$$

$$A = a \cdot b \cdot \cos 45 = 1.15 \cdot a^2$$

The area is multiplied by cos45 because the tail is 45º from the wind direction. The ratio between a and b was decided arbitrarily because it is only important that the scale and the final piece have the same proportions.

There is a problem with the wind speed because after going through the rotor the wind is slower. To calculate this speed energy equations are going to be used.

As v is going to be the wind speed where the tail is, v_b is the wind speed before getting to the rotor.

E_1 is the energy that has the wind that goes through the rotor during one second:

$$E_1 = \frac{1}{2} m_{wind} \cdot v_b^2 = \frac{1}{2} \cdot (r^2 \cdot \pi \cdot v_b \cdot \rho) \cdot v_b^2 = 506\,J$$

The energy the rotor gets from the wind is:

$$E_2 = P \cdot t = 268 \cdot 1 = 268\,J$$

The energy the wind has after the rotor is:

$$E_3 = 506 - 268 = 238\,J$$

The wind speed can be cleared:

$$E_3 = \frac{1}{2} m_{wind} \cdot {v_f}^2 \rightarrow v_f = \sqrt{\frac{2 \cdot E_3}{m_{wind}}} = 3.1\,m/s$$

The momentum created by the tail is the sum of every force times the distance perpendicular to the force between the force and the axis that the nacelle is turning around.

$$F_d = \frac{1}{2} \cdot \rho \cdot A \cdot C_d \cdot v^2 = 8.82 \cdot A$$

$$F_l = \frac{1}{2} \cdot \rho \cdot A \cdot C_l \cdot v^2 = 8.27 \cdot A$$

$$M = F_d \cdot c \cdot \cos 45 + F_l \cdot c \cdot \sin 45 = 13.89 \cdot a^2 \cdot c$$

$$13.89 \cdot a^2 \cdot c = 18.21$$

In this case a depends on c or vice versa, so choosing one value the other is calculated.

The designed values are:

a = 0.8 m

b = 1.3 m

c = 1.7 m

The last calculation that has to be done is about the forces in the tail with the maximum wind speed 13.5 m/s.

E_i is the energy that has the wind that goes through the rotor during one second:

$$E_i = \frac{1}{2} m_{wind} \cdot v_i^2 = \frac{1}{2} \cdot (r^2 \cdot \pi \cdot v_i \cdot \rho) \cdot v_i^2 = 13682\, J$$

The energy the rotor gets from the wind is:

$$E_r = P \cdot t = 7239 \cdot 1 = 7239\, J$$

The energy the wind has after the rotor is:

$$E_f = 13682 - 7239 = 6443\, J$$

The wind speed can be cleared:

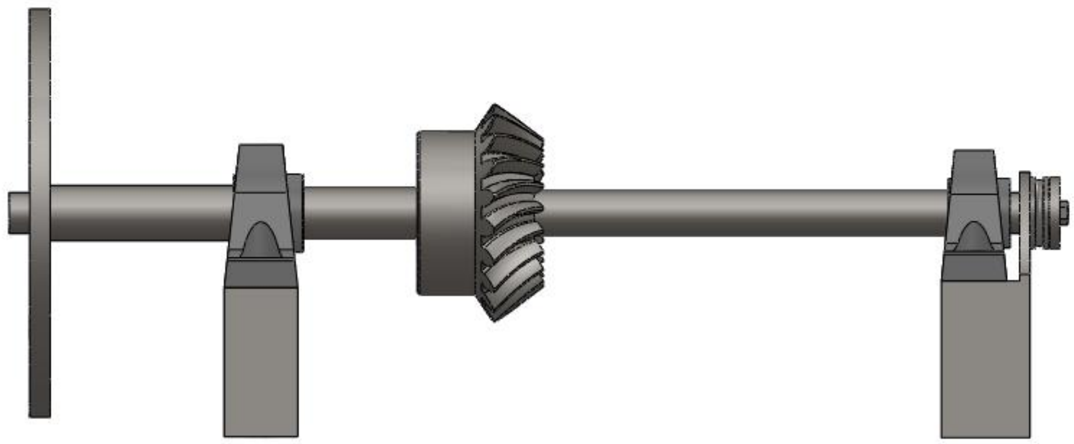
$$E_f = \frac{1}{2} m_{wind} \cdot {v_f}^2 \rightarrow v_f = \sqrt{\frac{2 \cdot E_f}{m_{wind}}} = 9.26\, m/s$$

This wind speed has been calculated so as to calculate the drag forces for the study of the tower.

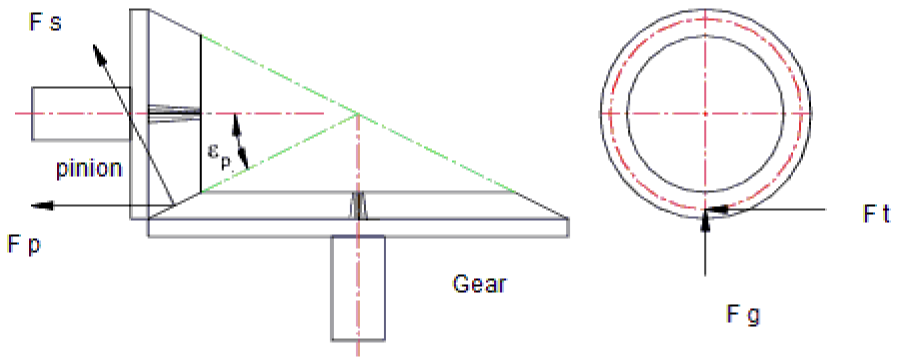
$$F_d = \frac{1}{2} \cdot \rho \cdot A \cdot C_d \cdot v^2 = 62\ N$$

Annex IV: Shaft calculations

The picture below shows the shaft with the main parts that is going to have: one welded piece (part of the rotor) in the front, three bearings and one gear.



First of all, the forces acting on the shaft must be calculated. On the leftmost the rotor is supported. There are eight forces acting here: the weight of the rotor and the axial force generated by the wind on the left; the radial forces in the radial bearings, the separating forces and the transmitting force on the gear and the axial force in the axial bearing. The horizontal one is 868 N due to the wind while the weight is 163.8 N. The forces of the gear and the bearings can be also calculated with the following procedure:



Values to consider:

- M_p≡ torque on the pinion shaft (N·mm)

- P_p ≡ power at the Pinion shaft (W) = 7329 W

- n_p ≡ rotational speed of the pinion shaft (rpm) = 44,18

- F_t ≡ tangential force on the pinion (N)

- d_p≡ pinion pitch circle diameter (mm) = 100 mm

- α ≡ angle of pressure = 20°

- F_s ≡ separating force

- ε_p ≡ pinion pitch angle =63° 6'

- ε_g ≡ gear pitch angle = 26° 54'

- F_p ≡ pinion thrust

- F_g≡ gear thrust

$$M_p = P_p \frac{60 \cdot 1000}{2 \cdot \pi \cdot n_p} = 163870\ N \cdot mm$$

$$F_t = \frac{2 \cdot M_p}{d_p} = 3277.4\ N$$

$$F_s = F_t \cdot \tan\alpha = 1192.87\ N$$

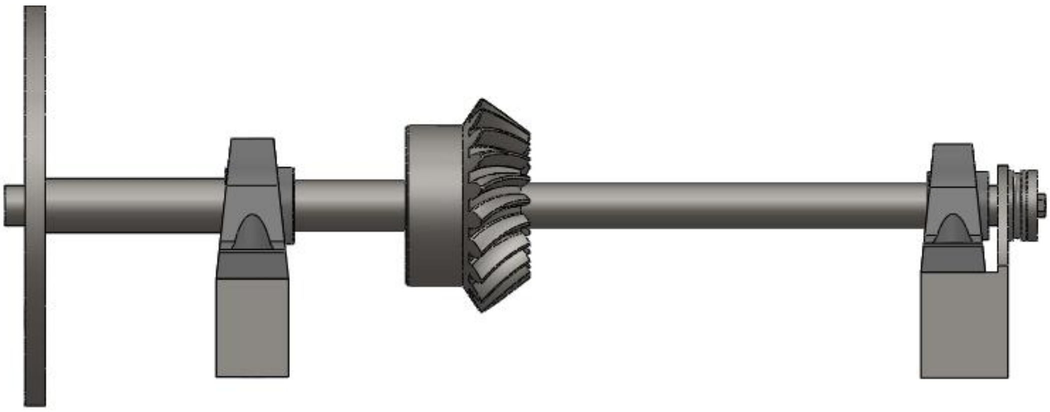
$$F_p = F_s \cdot \sin\varepsilon_p = 553.08\ N$$

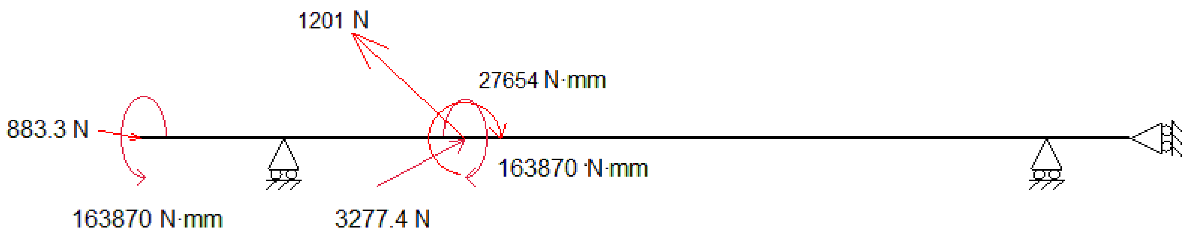
$$F_g = F_s \cdot \sin\varepsilon_g = 1066.16\ N$$

The momentum created by the F_p is this value times the distance to the center of the shaft:

$$M_{pp} = F_p \cdot \frac{d_p}{2} = 27654\ N \cdot mm$$

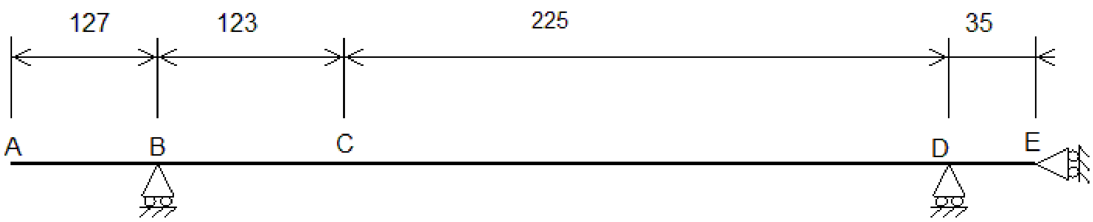
The next diagram shows the forces acting on the shaft. The forces are calculated for a wind speed of 13.5m/s, which is an extreme situation. The forces represented below are the forces of the weight of the rotor, the torque and the forces generated by the gear in opposition to the other gear. The two ovals represent torque and the force with the value of 3277 N is a force perpendicular to the paper, while the rest of them are in the plane of the paper.





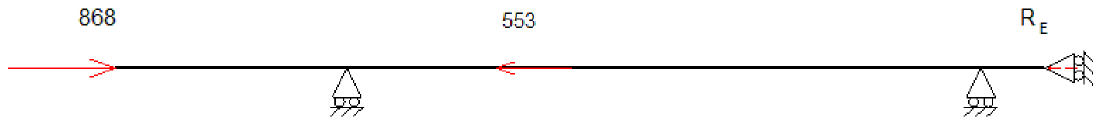
To calculate the reactions forces and the efforts in the shaft the forces are going to be separated in different axis: X, Y and Z. The X axis is parallel to the shaft, the Y axis is vertical and the Z one is perpendicular to both of them.

During the calculations the different points of the shaft are going to be named like in the next picture, being the units mm:



X-axis:

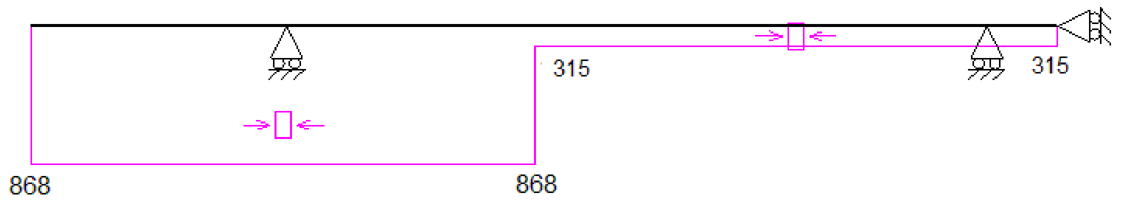
The only forces in the X-axis are the force of the wind in the rotor, the separating force of the gear and the reaction in the bearing.



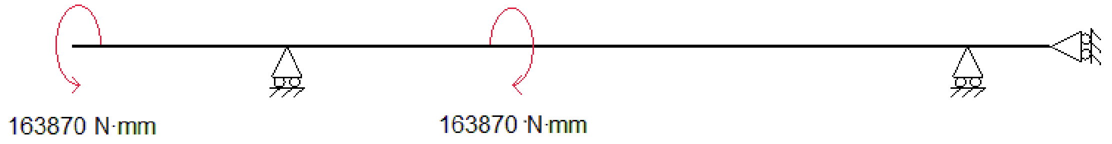
$$\sum F_x = 0 \rightarrow 868 + R_h - 553 = 0 \rightarrow R = -315\ N$$

The negative value means that the force is to the left

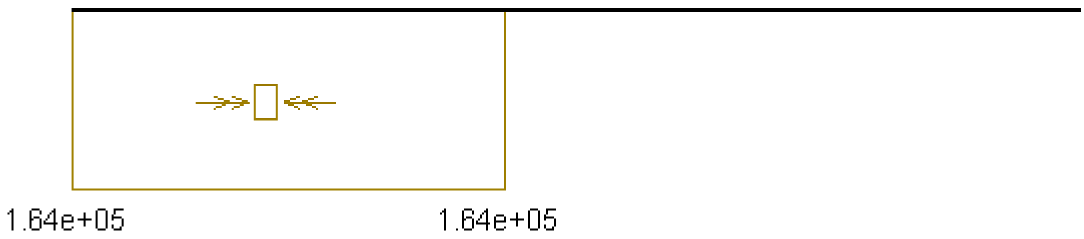
So the axial loads diagram, being the units Newton, is:



In the X-axis there are also torques both with the same value, first one coming from the rotor and the second one to transmit this one to the next part of the transmission:

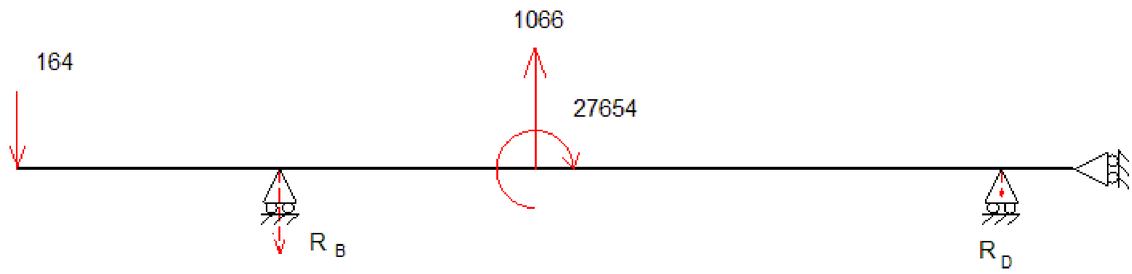


So the torque diagram is, being the units N·mm:



Y-axis

The forces on the Y-axis are the weight of the rotor, vertical separating force of the gears and the two reaction forces, one in each bearing. The forces in the shaft create a bending moment on Z-axis, that is the reason why the moments in Z-axis are studied in this part. Next diagram shows the forces and bending moments in N and mm:



To calculate the reaction forces in the bearings B and D, two equations are going to be used:

$$\sum F_y = 0 \rightarrow -164 + 1066 + R_B^y + R_D^y = 0$$

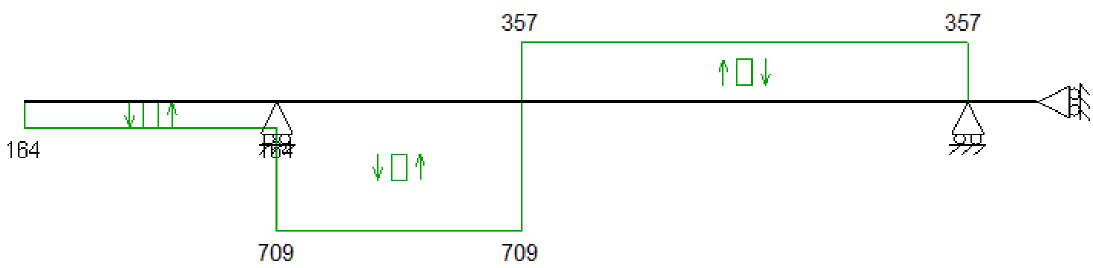
$$\sum M_z^A = 0 \rightarrow 27654 + R_B^y \cdot 127 - 1066 \cdot 250 + R_D^y \cdot 475 = 0$$

$$R_B^y = -545\ N$$

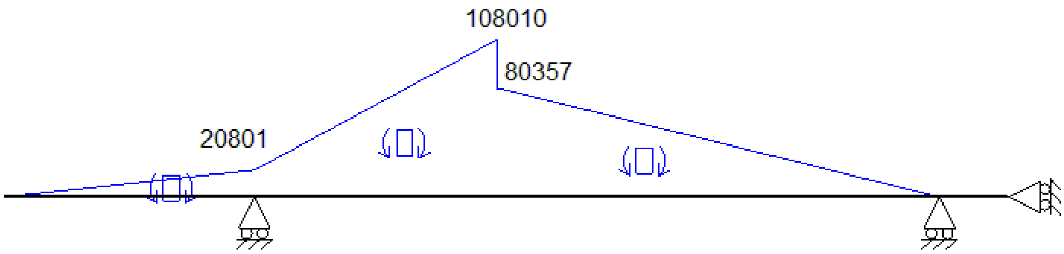
$$R_D^y = -357\ N$$

The negative signal means that the force is downwards.

The next picture is a diagram of the shear loads in the Y-axis, units are in N:



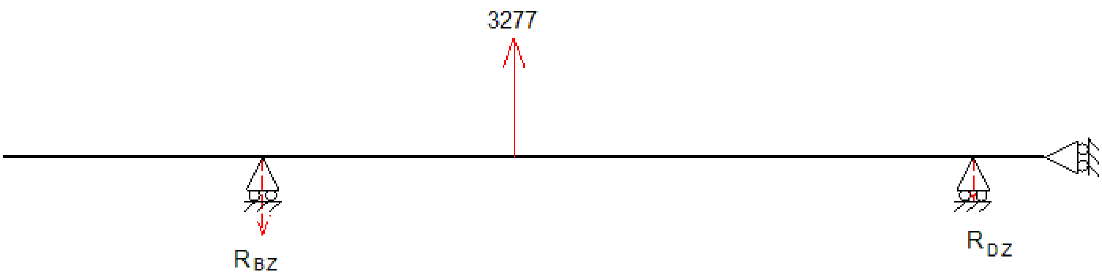
The next picture shows the bending loads in Z-axis, units are in N·mm:



What is most important in these diagrams is that there is a bending moment of 1.08·10^{5}N·mm where the gear is. To be calculated: $M = R_D^z \cdot 225 + M_{pp}$

Z-axis

The forces on Z-axis are the force creating by transmitting the torque from the gear in this shaft to the next one and the reaction forces of the bearings.



To calculate these reactions, the equations needed are the same as the last time but applying them to different forces.

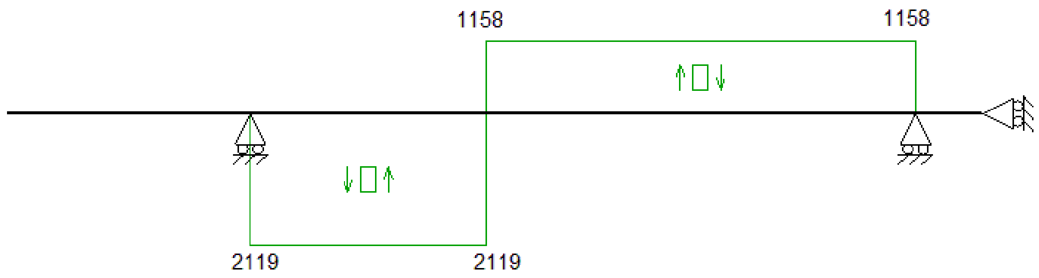
$$\sum F_z = 0 \rightarrow 3277.4 + R_B^z + R_D^z = 0$$

$$\sum M_y^A = 0 \rightarrow R_B^z \cdot 127 + 3277.4 \cdot 250 + R_D^z \cdot 475 = 0$$

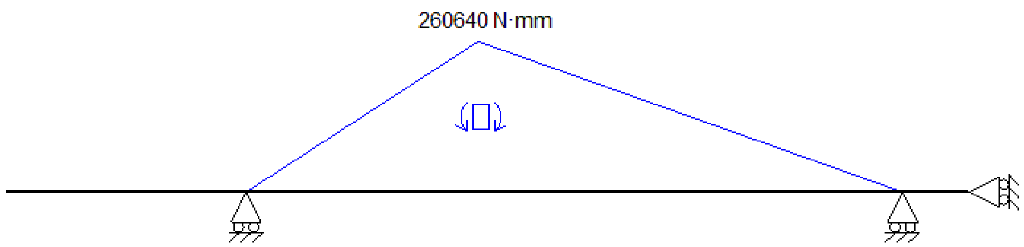
$$R_B^z = -2119\ N$$

$$R_D^z = -1158\ N$$

The next picture is a diagram of the shear loads in the Z-axis, units are in N:



The next picture shows the bending loads in Y-axis, units are in N·mm:



The most important datum in these diagrams is that there is a bending moment of $2.6{\cdot}10^5$ N·mm where the gear is. To be calculated: $M = R_B^z \cdot 123$

Analyzing the preview figures, the most affected part of the shaft is where the gear is.

In that point the bending moment in the Y-axis is $1.9{\cdot}10^5$ and in the Z-axis is $7.76{\cdot}10^4$. To calculate the total bending moment next equation will be used:

$$M = \sqrt{M_z^2 + M_y^2} = 282500\ N \cdot mm$$

In that point the bending moment is 282500 N·mm and the torque is 164000 N·mm.

For this torque and this bending moment the diameter can be calculated:

$$d_{min} = \sqrt[3]{\frac{4 \cdot n_s}{\pi \cdot \sigma_y}\sqrt{64M_{max}^2 + 48T_{max}^2}} = 20.9 \approx 22\ mm$$

Being:

-n_s= 2

-σ_y= 710 MPa

The steel needed is one with a high strength, good for shafts, AISI 4340 steel normalized.

The reason why the chosen diameter is 22 is because the gear has an hub diameter of 22 mm and is bigger than needed so it will not break.

The shame way to calculate the total moment the reactions can be calculated:

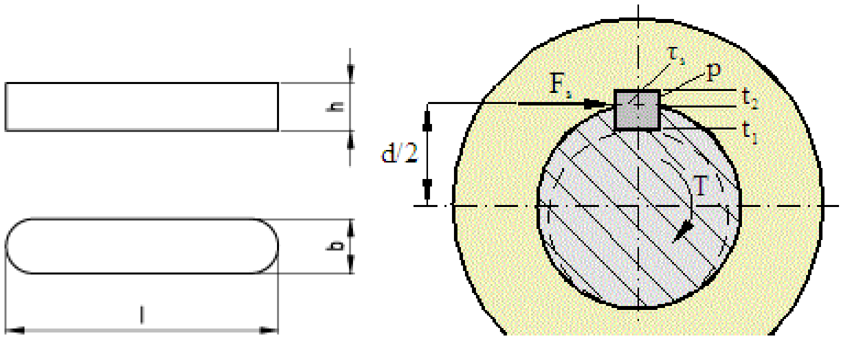
$$R_B = \sqrt{{R_B^y}^2 + {R_B^z}^2} = 2187\ N$$

$$R_D = \sqrt{{R_D^y}^2 + {R_D^z}^2} = 1212\ N$$

As this result the reaction forces are 2187 N in the left bearing, 1212 N in the middle bearing and 315 N in the right one.

Key

In order to transmit the torque from the shaft to the pinion a key is going to be designed.



Because of the shaft diameter is 22 mm the main dimension are 6x6. The yield strength of the key is 60kg/mm^2 = 588 N/mm^2

The keyway depth shaft (t_1) = 3.5 mm

The keyway depth hub (t_2) = 2.8 mm

The force we want to transmit is:

$$F_s = \frac{T}{d} = \frac{164000\ N\cdot mm}{22\ mm} = 7454.5\ N$$

Nominal torsional stress

$$d_k = d - t_1 = 18.5\ mm$$

$$\tau = \frac{T}{\frac{\pi}{16} \cdot d_k^3} = 131\ MPa$$

$$n_s = \frac{\sigma_y}{\sqrt{3} \cdot \tau} = 3.2$$

Shearing stress:

$$\tau = \frac{F_s}{l \cdot b} = \frac{1242.42}{l}$$

Bearing stress:

$$\sigma = \frac{F_s}{\frac{h}{2} \cdot l} = \frac{2484.84}{l}$$

Von Mises stress:

$$\sigma_{vm} = \sqrt{\sigma^2 + 3 \cdot \tau^2} = \frac{2485.6}{l}$$

If the design safety coefficient is 2,

$$n_s = \frac{\sigma_y}{\sigma_{vm}} \rightarrow \sigma_{vm} = \frac{\sigma_y}{n_s} = 355MPa$$

$$\frac{3615.86}{l} = 294 \rightarrow l > 12{,}28 \approx 14\ mm$$

12 is chosen because is the minimum length

$$\sigma_{vm} = \frac{3615.86}{14} = 258\ MPa$$

$$n_s = \frac{\sigma_y}{\sigma_{vm}} = 2.28$$

Annex V: Vertical shaft

This annex explains the procedure to calculate the needed diameter for the shaft. It is not going to be as detailed as the rotor shaft calculations but everything needed will be explained. This shaft has some differences from the rotor one. In this case instead of one pinion there are two gears, one in every end of the shaft. The one that is in the upper end (in the diagrams on the left) is going to be called bevel gar and the one at the bottom (in the diagrams on the right) is called miter gear. This is because of the rotor, as the rotor can turn 360° the gear on it can also be in different positions. The calculations must be done with different cases depending on the position.

The picture below shows the vertical shaft, but drawn in horizontal:

First of all we have to calculate all the forces that are acting on the shaft.

Values to consider:

- P_s≡ power on the shaft (W)

- n≡ rotational speed (rpm)

- M_s≡ torque on the shaft (N·mm)

- F_{bg} ≡ tangential force on the bevel gear (N)

- F_{mg}≡ tangential force on the miter gear (N)

- d_{bg}≡ bevel gear pitch circle diameter (mm)

- d_{mg} ≡ miter gear pitch circle diameter (mm)

- α ≡ angle of pressure = 20°

- F_{sb} ≡ separating force on the bevel gear (N)

- F_{sm} ≡ separating force on the miter gear (N)

- ε_{mg} ≡ miter gear pitch angle = 45°

- ε_{bg} ≡ bevel gear pitch angle = 26° 54’

- F_{mgt}≡ miter gear thrust (N)

- F_{pt} ≡ pinion thrust (N)

- F_{bgt}≡ bevel gear thrust (N)

- M_{bg} ≡ bending moment created by the vertical separating force in the bevel gear

- M_{mg} ≡ bending moment created by the vertical separating force in the miter gear

$$M_s = P_s \frac{60 \cdot 1000}{2 \cdot \pi \cdot n} = 196629\ N \cdot mm$$

$$F_{bg} = \frac{2 \cdot M_s}{d_{bg}} = 1966\ N$$

$$F_{mg} = \frac{2 \cdot M_s}{d_{mg}} = 3277\ N$$

$$F_{sb} = F_{bg} \cdot \tan(\alpha) = 715\ N$$

$$F_{sm} = F_{mg} \cdot \tan(\alpha) = 1192\ N$$

$$F_{pt} = F_{sb} \cdot \cos(\varepsilon_{bg}) = 324\ N$$

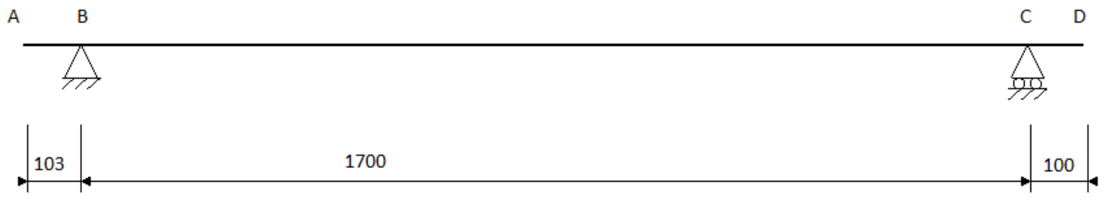
$$F_{bgt} = F_{sb} \cdot \sin(\varepsilon_{bg}) = 628\ N$$

$$F_{sm} = F_{mg} \cdot \sin(\varepsilon_{mg}) = 843\ N$$

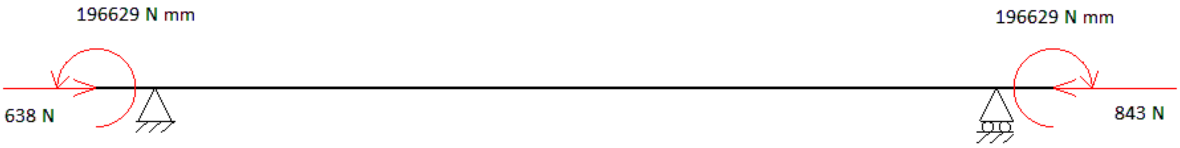
$$M_{bg} = F_{bgt} \cdot \frac{d_{bg}}{2} = 79950\ N \cdot mm$$

$$M_{mg} = F_{mt} \cdot \frac{d_{mg}}{2} = 50580\ N \cdot mm$$

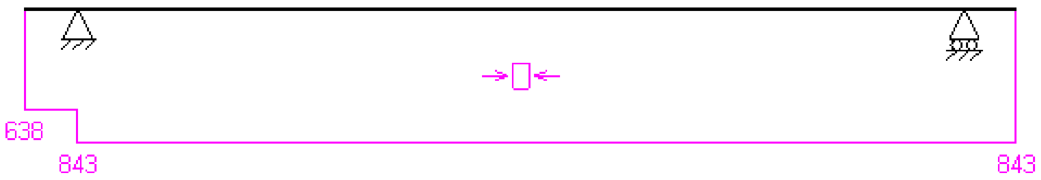
Before studying each case separately the X axis is going to be analyzed. In the Z axis there are only axial forces and torque momentums and they are constant in all the cases.

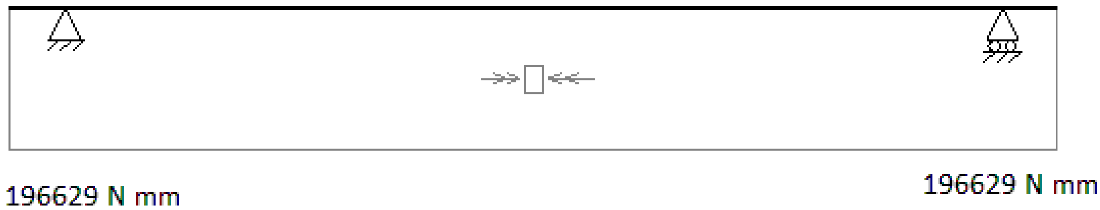


X axis



$$\sum F_H = 0 \rightarrow 638 + R_H - 843 = 0 \rightarrow R_H = 205\ N$$





The torque is constant during the length of the shaft, being its value 196 629 N·mm.

First case

In the first case the gears (the pinion and the miter gear) are in the same side of the shaft.

Y axis

The diagram shows the forces in the Y axis and also the bending moments in the Z axis.

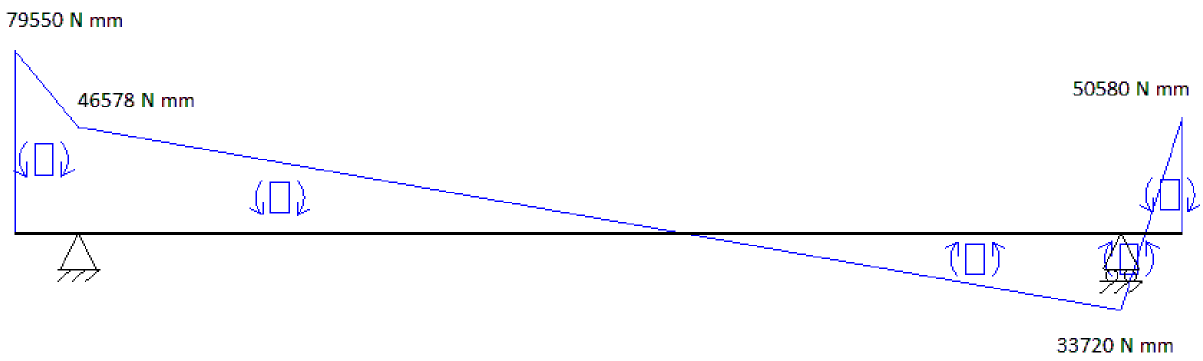
$$\sum F_v = 0 \rightarrow 324 + R_{BY1} + R_{DY1} + 843 = 0$$

$$\sum M_A^z = 0 \rightarrow R_{BY1} \cdot 103 + R_{DY1} \cdot 1803 - 50580 + 843 \cdot 1903 + 79950 = 0$$

$$R_{BY1} = -277\ N$$

$$R_{DY1} = 890\ N$$

The next diagram shows the bending moments in Z axis:



<u>Z axis</u>

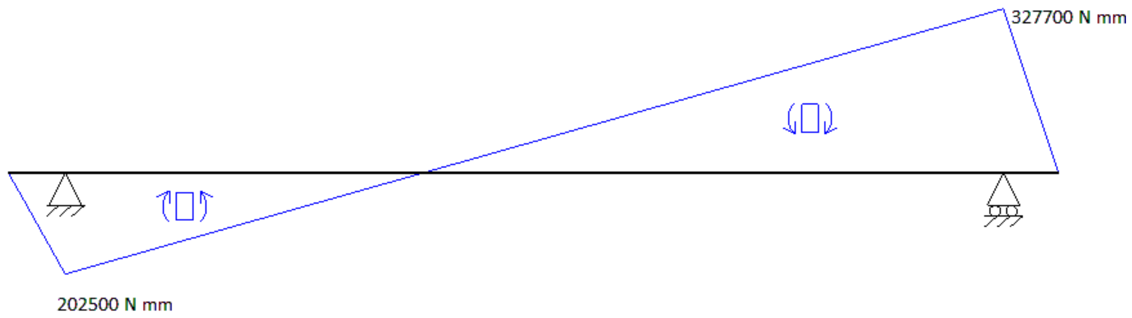
The next diagram shows the forces in Z axis:

$$\sum F_v = 0 \rightarrow 1966 + R_{BZ1} + R_{DZ1} - 3277 = 0$$

$$\sum M_A^z = 0 \rightarrow R_{BZ1} \cdot 103 + R_{DZ1} \cdot 1803 - 3277 \cdot 1903 = 0$$

$$R_{BZ1} = -2278N$$

$$R_{DZ1} = 3589\ N$$



The point of the shaft were the bending moment is maximum is where the second bearing is. The momentum in this point is:

$$M_{D1} = \sqrt{{M_{DY1}}^2 + {M_{DZ1}}^2} = 329430\ N \cdot mm$$

The bearing B has to support:

$$F_{B1} = \sqrt{{F_{BY1}}^2 + {F_{BZ1}}^2} = 2295\ N$$

The bearing D has to support:

$$F_{D1} = \sqrt{{F_{DY1}}^2 + {F_{DZ1}}^2} = 3698\ N$$

Second case

In the second case the gears (the pinion and the miter gear) are in the opposite side of the shaft.

Y axis

The diagram shows the forces in the Y axis and also the bending moments in the Z axis.

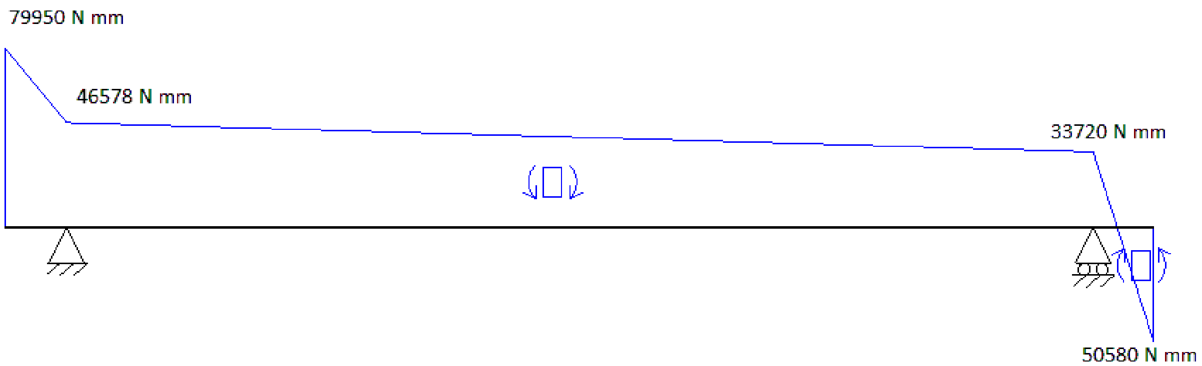
$$\sum F_v = 0 \rightarrow 324 + R_{BY2} + R_{DY2} - 843 = 0$$

$$\sum M_A^z = 0 \rightarrow R_{BY2} \cdot 103 + R_{DY2} \cdot 1803 + 50580 - 843 \cdot 1903 + 79950 = 0$$

$$R_{BY2} = -316\ N$$

$$R_{DY2} = 835\ N$$

The next diagram shows the bending moments in Z axis:



Z axis

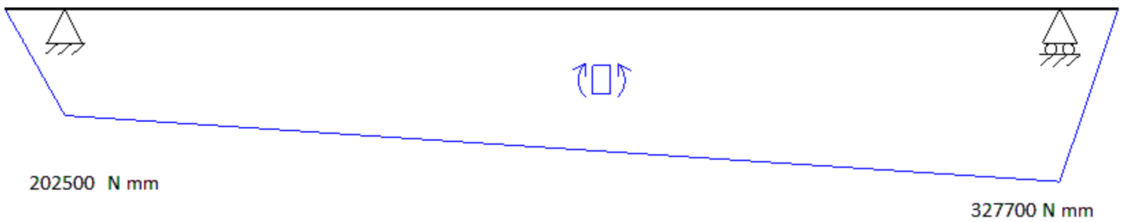
The next diagram shows the forces in Z axis:

$$\sum F_v = 0 \rightarrow 1966 + R_{BZ2} + R_{DZ2} - 3277 = 0$$

$$\sum M_A^z = 0 \rightarrow R_{BZ2} \cdot 103 + R_{DZ2} \cdot 1803 + 3277 \cdot 1903 = 0$$

$$R_{BZ2} = -1892N$$

$$R_{DZ2} = -3350 \ N$$



The point of the shaft were the bending moment is maximum is where the second bearing is. The momentum in this point is:

$$M_{D2} = \sqrt{M_{DY2}{}^2 + M_{DZ2}{}^2} = 329430 \ N \cdot mm$$

The bearing B has to support:

$$F_{B2} = \sqrt{F_{BY2}{}^2 + F_{BZ2}{}^2} = 1892 \ N$$

The bearing D has to support:

$$F_{D2} = \sqrt{F_{DY2}{}^2 + F_{DZ2}{}^2} = 3452 \ N$$

Thrid case

In the third case the plane where the rotor shaft and the vertical shaft is 90° from the plane of the vertical shaft with the pump one.

Y axis

The diagram shows the forces in the Y axis and also the bending moments in the Z axis.

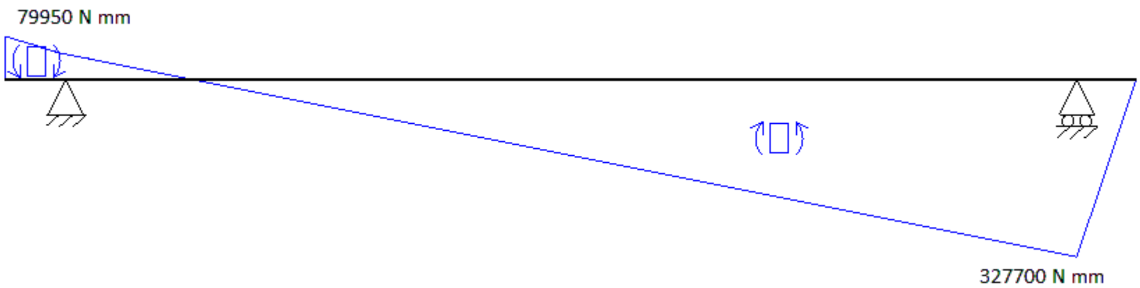
$$\sum F_v = 0 \rightarrow 324 + R_{BY3} + R_{DY3} + 3277 = 0$$

$$\sum M_A^z = 0 \rightarrow R_{BY3} \cdot 103 + R_{DY3} \cdot 1803 - 3277 \cdot 1903 + 79950 = 0$$

$$R_{BY3} = -104\ N$$

$$R_{DY3} = -3497\ N$$

The next diagram shows the bending moments in Z axis:



Z axis

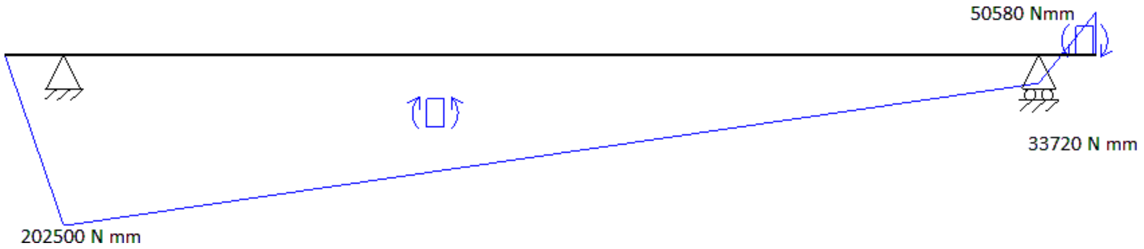
The next diagram shows the forces in Z axis:

$$\sum F_v = 0 \rightarrow 1966 + R_{BZ3} + R_{DZ3} + 843 = 0$$

$$\sum M_A^z = 0 \rightarrow R_{BZ3} \cdot 103 + \ R_{DZ3} \cdot 1803 + 3277 \cdot 1903 = 0$$

$$R_{BZ3} = -2065N$$

$$R_{DZ3} = -744\ \ N$$



The momentum in the point where the bearing B is:

$$M_{B3} = \sqrt{{M_{BY3}}^2 + {M_{BZ3}}^2} = 206408\ N \cdot mm$$

The momentum in the point where the bearing D is:

$$M_{D3} = \sqrt{{M_{DY3}}^2 + {M_{DZ3}}^2} = 329430\ N \cdot mm$$

The maximum momentum is where the bearing D is.

The bearing B has to support:

$$F_{B3} = \sqrt{{F_{BY3}}^2 + {F_{BZ3}}^2} = 2067\ N$$

The bearing D has to support:

$$F_{D3} = \sqrt{{F_{DY3}}^2 + {F_{DZ3}}^2} = 3575\ N$$

Forth case

In the fourth case the plane where the rotor shaft and the vertical shaft is 90° from the plane of the vertical shaft with the pump one.

Y axis

The diagram shows the forces in the Y axis and also the bending moments in the Z axis.

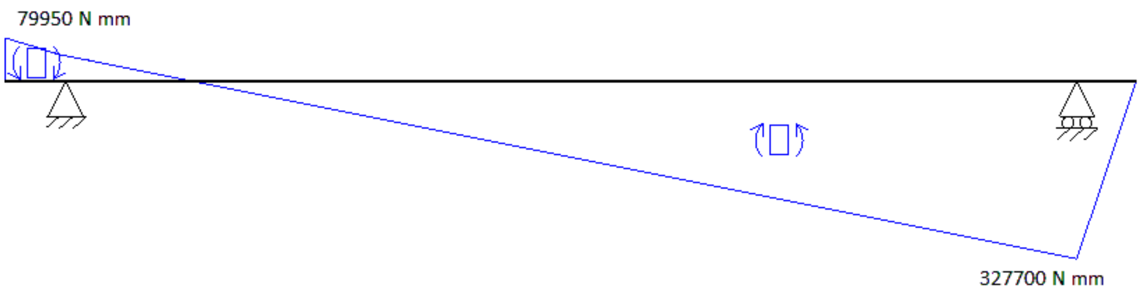
$$\sum F_v = 0 \rightarrow 324 + R_{BY4} + R_{DY4} - 3277 = 0$$

$$\sum M_A^z = 0 \rightarrow R_{BY4} \cdot 103 + \ R_{DY4} \cdot 1803 - 3277 \cdot 1903 + 79950 = 0$$

$$R_{BY3} = -489\ N$$

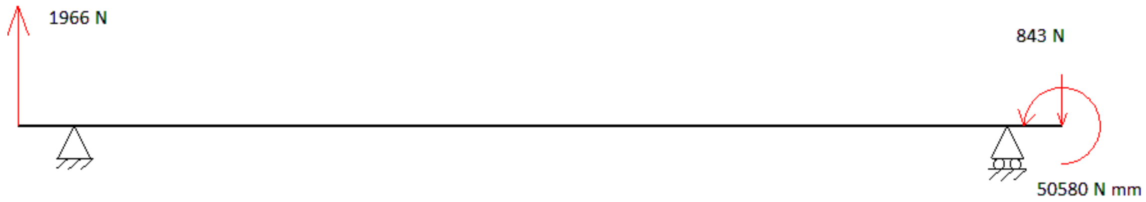
$$R_{DY3} = -3442\ N$$

The next diagram shows the bending moments in Z axis:



Z axis

The next diagram shows the forces in Z axis:

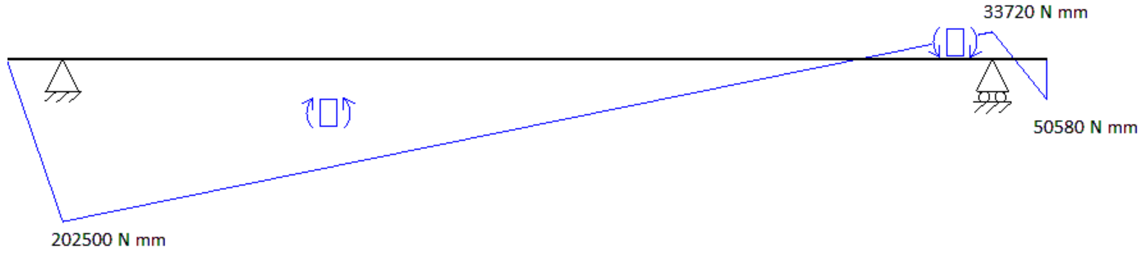


$$\sum F_v = 0 \rightarrow 1966 + R_{BZ4} + R_{DZ4} - 843 = 0$$

$$\sum M_A^z = 0 \rightarrow R_{BZ4} \cdot 103 + \ R_{DZ4} \cdot 1803 + 3277 \cdot 1903 = 0$$

$$R_{BZ4} = -2105N$$

$$R_{DZ4} = 982\ N$$



The momentum in the point where the bearing B is:

$$M_{B3} = \sqrt{{M_{BY3}}^2 + {M_{BZ3}}^2} = 206412\ N \cdot mm$$

The momentum in the point where the bearing D is:

$$M_{D3} = \sqrt{{M_{DY3}}^2 + {M_{DZ3}}^2} = 329430\ N \cdot mm$$

The maximum momentum is where the bearing D is.

The bearing B has to support:

$$F_{B4} = \sqrt{{F_{BY4}}^2 + {F_{BZ4}}^2} = 2161\ N$$

The bearing D has to support:

$$F_{D4} = \sqrt{{F_{DY4}}^2 + {F_{DZ4}}^2} = 3579\ N$$

In that point the bending moment is 329430 N·mm and the torque is 196629 N·mm.

For this torque and this bending moment the diameter can be calculated:

$$d_{min} = \sqrt[3]{\frac{4 \cdot n_s}{\pi \cdot \sigma_y}\sqrt{64M_{max}^2 + 48T_{max}^2}} = 21.99 \approx 22\ mm$$

Being:

-n_s= 2

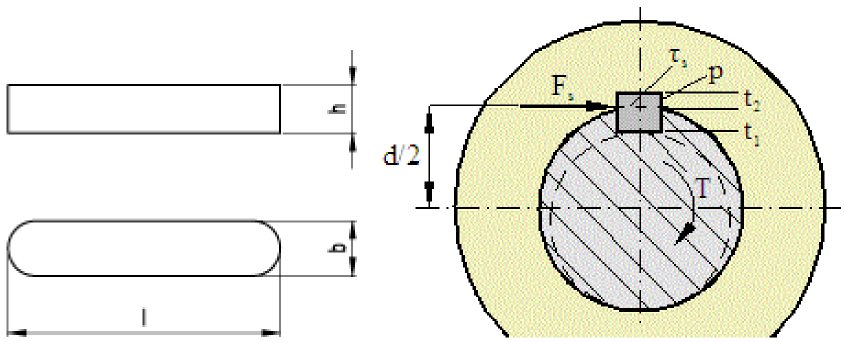
-σ_y= 710 MPa

Because of the hub diameter of the gear, the diameter of the shaft has to be bigger so the safety coefficient will be more than 2.

The biggest force than the bearing B has to support is 2295 N. The biggest force than the bearing B has to support is 3698 N. As the diameter is quite big, the bearings used will support easily that loads.

Top key

In order to transmit the torque from the shaft to the pinion a key is going to be designed.



Because of the shaft diameter is 25 mm the main dimension are 8x7.

The keyway depth shaft (t_1) = 4 mm

The keyway depth hub (t_2) = 3.3 mm

The force we want to transmit is:

$$F_s = \frac{T}{d} = \frac{196629\ N \cdot mm}{25\ mm} = 7865\ N$$

Nominal torsional stress

$$d_k = d - t_1 = 21\ mm$$

$$\tau = \frac{T}{\frac{\pi}{16} \cdot d_k^3} = 108\ MPa$$

$$n_s = \frac{\sigma_y}{\sqrt{3} \cdot \tau} = 3.8$$

Shearing stress:

$$\tau = \frac{F_s}{l \cdot b} = \frac{983}{l}$$

Bearing stress:

$$\sigma = \frac{F_s}{\frac{h}{2} \cdot l} = \frac{2247}{l}$$

Von Mises stress:

$$\sigma_{vm} = \sqrt{\sigma^2 + 3 \cdot \tau^2} = \frac{2819}{l}$$

If the design safety coefficient is 2,

$$n_s = \frac{\sigma_y}{\sigma_{vm}} \rightarrow \sigma_{vm} = \frac{\sigma_y}{n_s} = 294\ MPa$$

$$\frac{2819}{l} = 294 \rightarrow l \geq 9.58 \approx 12\ mm$$

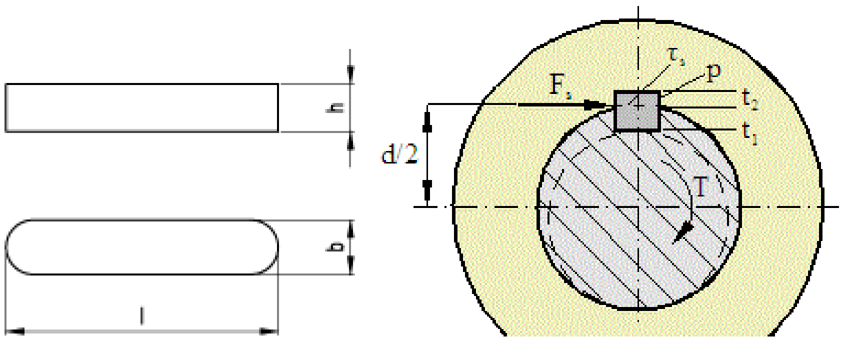
12 is chosen because is the minimum length

$$\sigma_{vm} = \frac{2819}{12} = 234\ MPa$$

$$n_s = \frac{\sigma_y}{\sigma_{vm}} = 2.5$$

Bottom key

In order to transmit the torque from the shaft to the pinion a key is going to be designed.



Because of the shaft diameter is 25 mm the main dimension are 8x7.

The keyway depth shaft (t_1) = 4 mm

The keyway depth hub (t_2) = 3.3 mm

The force we want to transmit is:

$$F_s = \frac{T}{d} = \frac{196629\ N \cdot mm}{25\ mm} = 7865\ N$$

Nominal torsional stress

$$d_k = d - t_1 = 21\ mm$$

$$\tau = \frac{T}{\frac{\pi}{16} \cdot d_k^3} = 108\ MPa$$

$$n_s = \frac{\sigma_y}{\sqrt{3} \cdot \tau} = 3.8$$

Shearing stress:

$$\tau = \frac{F_s}{l \cdot b} = \frac{983}{l}$$

Bearing stress:

$$\sigma = \frac{F_s}{\frac{h}{2} \cdot l} = \frac{2247}{l}$$

Von Mises stress:

$$\sigma_{vm} = \sqrt{\sigma^2 + 3 \cdot \tau^2} = \frac{2819}{l}$$

If the design safety coefficient is 2,

$$n_s = \frac{\sigma_y}{\sigma_{vm}} \rightarrow \sigma_{vm} = \frac{\sigma_y}{n_s} = 355MPa$$

$$\frac{2819}{l} = 355 \rightarrow l \geq 7.95 \approx 12\ mm$$

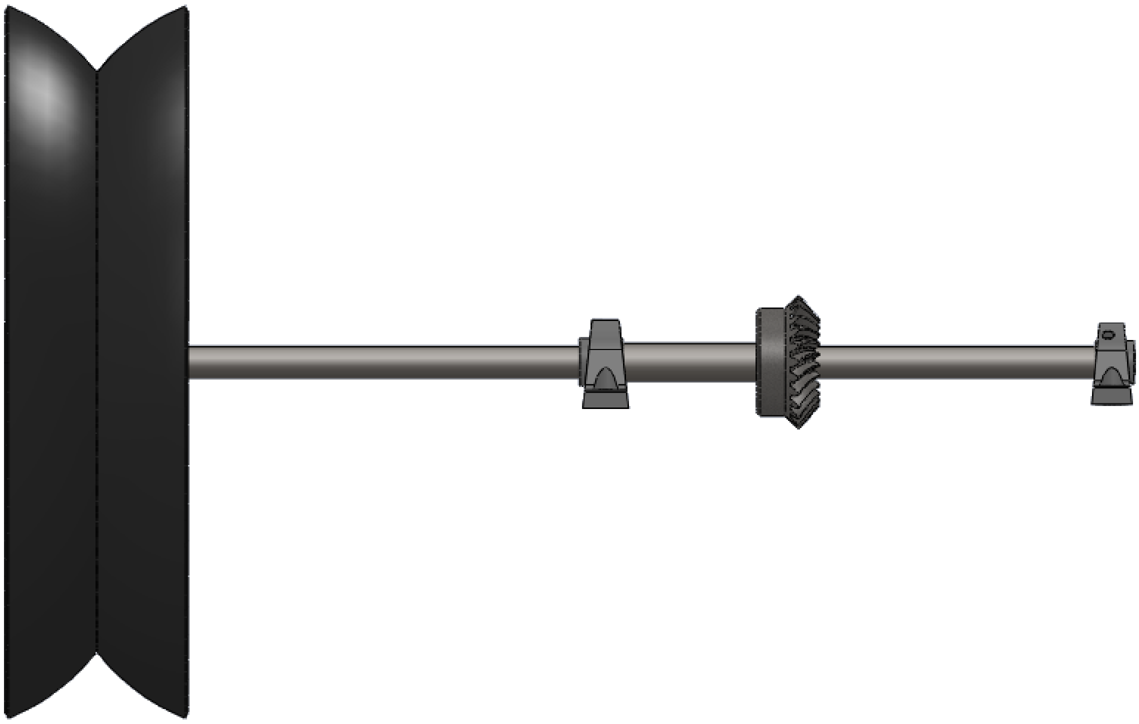
12 is chosen because is the minimum length

$$\sigma_{vm} = \frac{3615.86}{12} = 207\ MPa$$

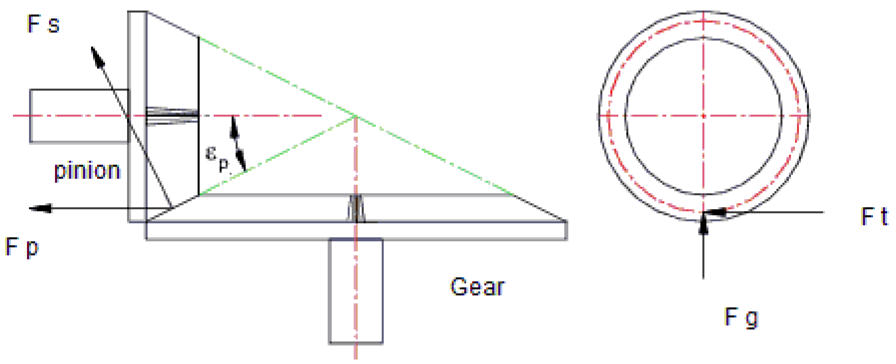
$$n_s = \frac{\sigma_y}{\sigma_{vm}} = 3.4$$

Annex VI: The pump shaft

The picture below shows the shaft with the main parts that is going to have: one welded piece (part of the rotor) in the front, three bearings and one gear.



First of all, the forces acting on the shaft must be calculated. On the leftmost the rotor is pump is supported. There are four forces acting here: the weight and the tension on the pump, the force that transmits the torque and the separating forces. The weight of the pump is 68 N and the tension is 230 N. The forces of the gear can be also calculated with the following procedure:



Values to consider:

- M_s≡ torque on the pump shaft (N·mm)

- P_p ≡ power at the pump shaft (W) = 4633 W

- n ≡ rotational speed of the pinion shaft (rpm) = 281,25 rev/min

- F_t ≡ tangential force on the pinion (N)

- d_g≡ pinion pitch circle diameter (mm) = 100

- α ≡ angle of pressure = 20°

- F_s ≡ separating force

- ε_g ≡ gear pitch angle = 45°

- F_g≡ gear thrust

$$M_s = P_s \frac{60 \cdot 1000}{2 \cdot \pi \cdot \mathrm{n}} = 157100\ N \cdot mm$$

$$F_t = \frac{2 \cdot M_p}{d_p} = 3141\ N$$

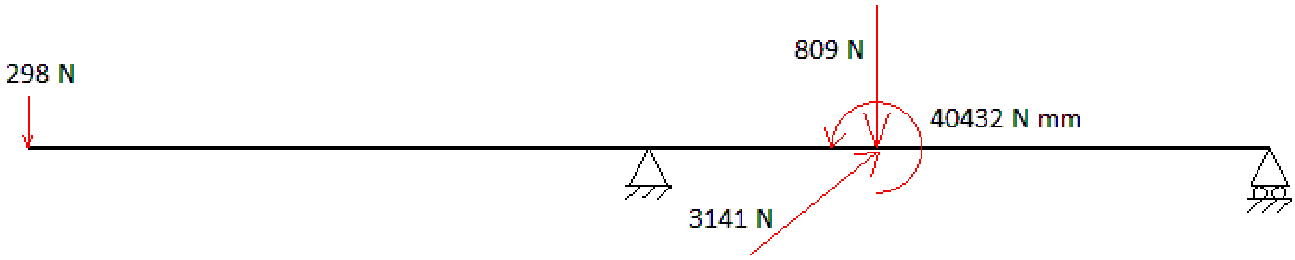
$$F_s = F_t \cdot \tan\alpha = 1144\ N$$

$$F_g = F_t \cdot \sin\varepsilon_g = 809\ N$$

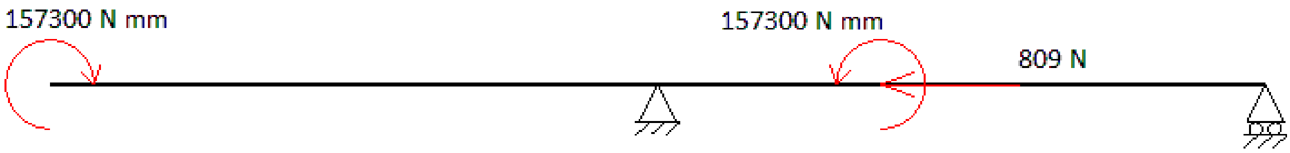
The momentum created by the F_g is this value times the distance to the center of the shaft:

$$M_g = F_g \cdot \frac{d_g}{2} = 40432\ N \cdot mm$$

The next diagram shows the forces acting on the shaft including only forces in Y axis and Z axis; bending moments in Z axis.

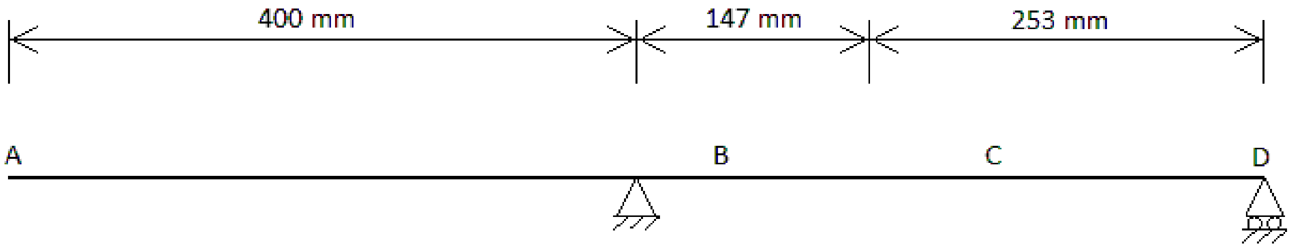


The next diagram shows the forces in X axis and torques in the shaft:



To calculate the reactions forces and the efforts in the shaft the forces are going to be separated in different axis: X, Y and Z. The X axis is parallel to the shaft, the Y axis is vertical and the Z one is perpendicular to both of them.

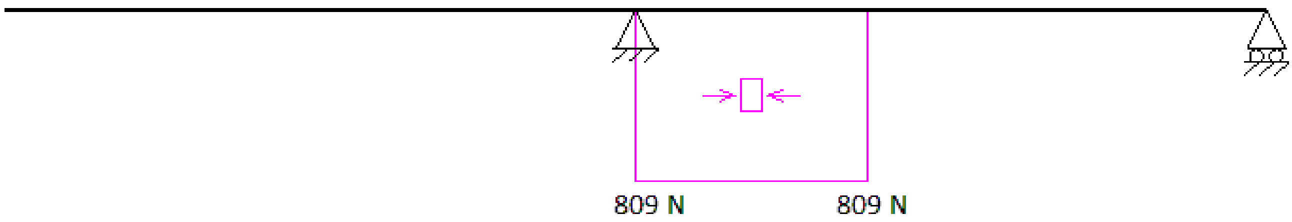
During the calculations the different points of the shaft are going to be named like in the next picture:



X-axis:

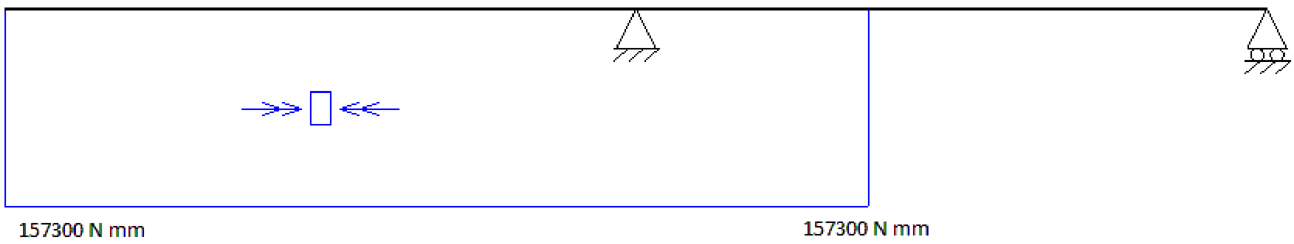
$$\sum F_x = 0 \rightarrow R_H - 809 = 0 \rightarrow R = 809\ N$$

As the value is positive, the force is to the right. So the axial loads diagram, being the units Newton, is:



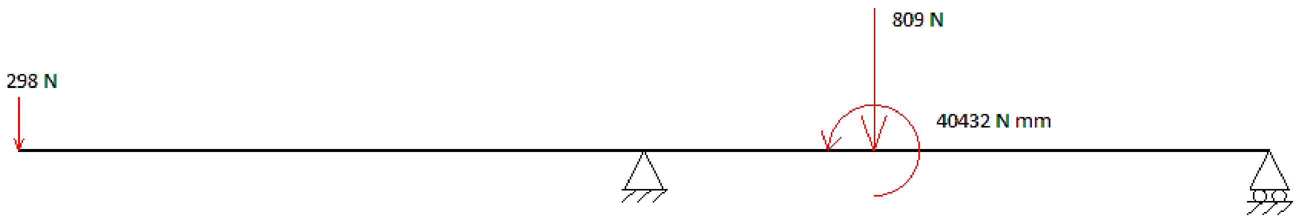
In the X-axis there are also torques both with the same value, first one coming from the vertical shaft and the second one transmitting to the angular speed to the pump.

So the torque diagram is, being the units N·mm:



Y-axis

The forces on the Y-axis are the weight of the pump the tension of the rope where the wheel is, the vertical separating force of the gears and the two reaction forces, one in each bearing.. Next diagram shows the forces and bending moments in N and mm:



To calculate the reaction forces in the bearings B and D, two equations are going to be used:

$$\sum F_y = 0 \rightarrow -298 - 809 + R_{By} + R_{Dy} = 0$$

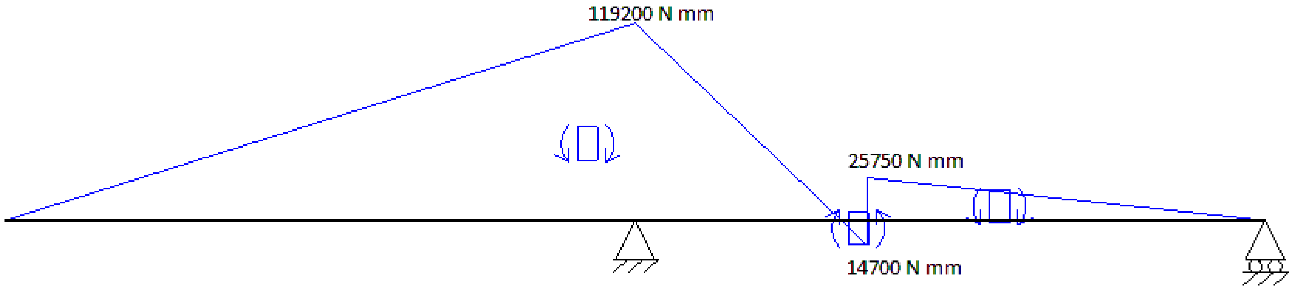
$$\sum M_z^A = 0 \rightarrow 40432 + R_{By} \cdot 400 - 809 \cdot 547 + R_{Dy} \cdot 800 = 0$$

$$R_B^y = 1208\ N$$

$$R_D^y = -1018\ N$$

The negative signal means that the force is downwards.

The next picture shows the bending loads in Z-axis, units are in N·mm:



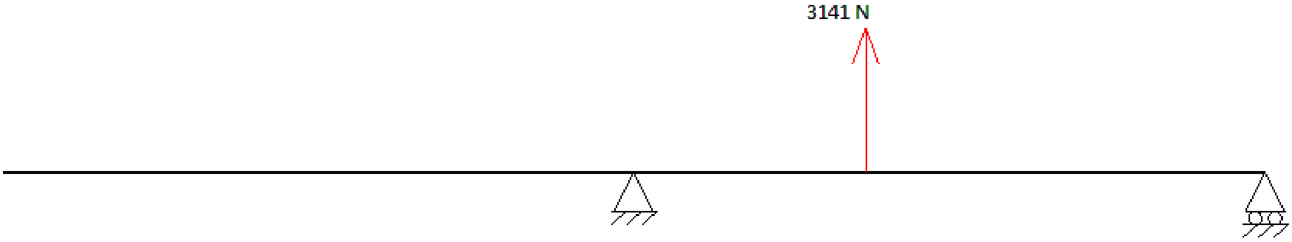
What is most important in these diagrams is that there is a bending moment of $1.19 \cdot 10^5$ N·mm where the bearing B is. To be calculated :

$$M_{Bz} = F_{Ay} \cdot 400$$

Being F_{Ay} the weight of the pump and the tensions, 400 is the distance to the first bearing.

Z-axis

The forces on Z-axis are the force creating by transmitting the torque from the gear in this shaft to the next one and the reaction forces of the bearings.



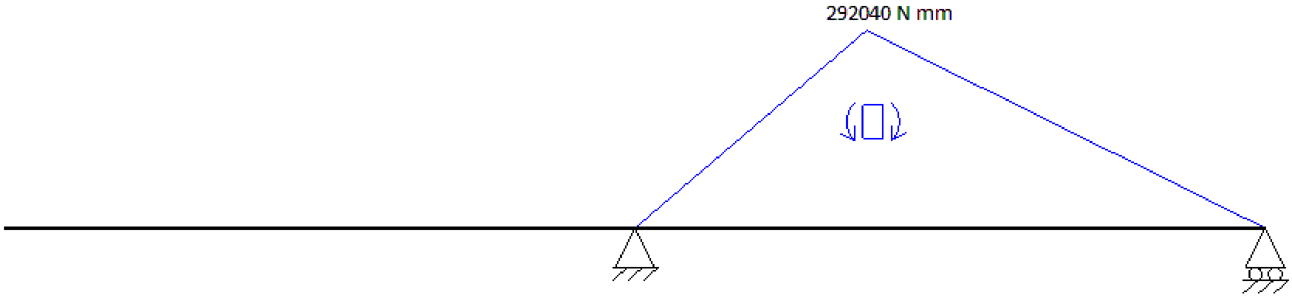
$$\sum F_z = 0 \rightarrow 3141 + R_{Bz} + R_{Dz} = 0$$

$$\sum M_y^A = 0 \rightarrow R_{Bz} \cdot 400 + 3141 \cdot 253 = 0$$

$$R_{Bz} = -1987\ N$$

$$R_{Dz} = -1154\ N$$

The next picture shows the bending loads in Y-axis:



The most important datum in these diagrams is that there is a bending moment of $2.92 \cdot 10^5$ N·mm where the gear is. To be calculated:

$$M = R_{Dz} \cdot 253$$

It is also important that where in the Z-axis the bending moment is maximum, in this axis is zero.

Where the gear is the bending moment is:

$$M = \sqrt{{M_{Cz}}^2 + {M_{Cy}}^2} = 293200\ N \cdot mm$$

In that point the bending moment is 293200N·mm and the torque is 157300 N·mm.

For this torque and this bending moment the diameter can be calculated:

$$d_{min} = \sqrt[3]{\frac{4 \cdot n_s}{\pi \cdot \sigma_y} \sqrt{64M_C^2 + 48T_C^2}} = 21{,}01 \approx 22\ mm$$

Being:

-n_s= 2

-σ_y= 710 MPa

The steel needed is one with a high strength, good for shafts, AISI 4340 steel normalized.

The diameter in that part of the shaft is going to be 25 mm because the hub diameter for the gear is 25 so the safety coefficient will be bigger.

Where the bearing B is, the diameter has to be also calculated, in that point the torque is 157300 N mm and the bending moment is 119200 N mm:

$$d_{min} = \sqrt[3]{\frac{4 \cdot n_s}{\pi \cdot \sigma_y} \sqrt{64M_B^2 + 48T_B^2}} = 17.31 \approx 20\ mm$$

The inner diameter of the bearing is 17 or 20, so 20 is the value.

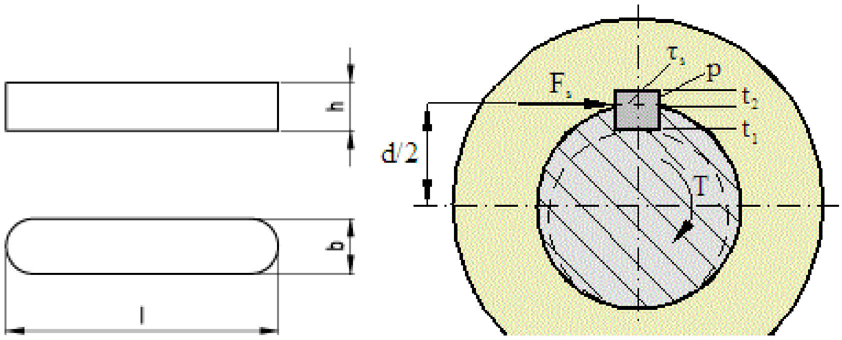
The shame way to calculate the total moment the reactions can be calculated:

$$R_B = \sqrt{{R_{By}}^2 + {R_{Bz}}^2} = 2325\ N$$

$$R_D = \sqrt{{R_{Dy}}^2 + {R_{Dz}}^2} = 1539\ N$$

Key

In order to transmit the torque from the shaft to the pinion a key is going to be designed.



Because of the shaft diameter is 20 mm the main dimension are 6x6.

The keyway depth shaft (t_1) = 3.5 mm

The keyway depth hub (t_2) = 2.8 mm

The force we want to transmit is:

$$F_s = \frac{T}{d} = \frac{157300\ N \cdot mm}{25\ mm} = 6292\ N$$

Nominal torsional stress

$$d_k = d - t_1 = 21\ mm$$

$$\tau = \frac{T}{\frac{\pi}{16} \cdot d_k^3} = 86.5\ MPa$$

$$n_s = \frac{\sigma_y}{\sqrt{3} \cdot \tau} = 3.92$$

Shearing stress:

$$\tau = \frac{F_s}{l \cdot b} = \frac{1573}{l}$$

Bearing stress:

$$\sigma = \frac{F_s}{\frac{h}{2} \cdot l} = \frac{1798}{l}$$

Von Mises stress:

$$\sigma_{vm} = \sqrt{\sigma^2 + 3 \cdot \tau^2} = \frac{3264}{l}$$

If the design safety coefficient is 2,

$$n_s = \frac{\sigma_y}{\sigma_{vm}} \rightarrow \sigma_{vm} = \frac{\sigma_y}{n_s} = 288MPa$$

$$\frac{3264}{l} = 288 \rightarrow l \geq 11.33 \approx 12\ mm$$

12 is chosen because there are not 12 mm long keys

$$\sigma_{vm} = \frac{3264}{12} = 272\ MPa$$

$$n_s = \frac{\sigma_y}{\sigma_{vm}} = 2.16$$

Annex VII: The pump

The rotor has a power of 286 Watts working with a wind speed of 4.6 m/s. Because of the transmission efficiencies, the power that gets to the pump is 229 W. The pump efficiency is not 100% but 80%. So the power for the calculations is 183 W. This power is going to be used to raise the water, and is directly proportional to the flow of water. The power is the product of force times the speed:

$$P = F \cdot v$$

As the speed is constant there is no acceleration so the system is balanced. Then, the force is only the weight of the water that is being raised. This force can be calculated as:

$$F = l \cdot r^2 \cdot \pi \cdot \rho \cdot g$$

Being:

l : the length of the pipe

r : the radius of the pipe

ρ : the water density

g : gravity acceleration

The length is 11.5 meter, from the bottom of the well to the pipe that goes to the tank. The values of the density and the gravity are always the same so the only values that can be changed are the radius and the speed.

The flow can be calculated as:

$$Q = r^2 \cdot \pi \cdot v$$

Joining all the equations the power is:

$$P = l \cdot \rho \cdot g \cdot Q$$

It has been demonstrated that the power is directly proportional to the flow and the length. Clearing the flow:

$$Q = \frac{P}{l \cdot \rho \cdot g} = 1.6x10^{-3}\, m^3/s = 1.6\, l/s$$

What has to be designed is the radius of the pipe and the speed that depends on the radius. But the force also depends on the diameter and the rope has to support it. The final diameter is 32 mm so the speed is:

$$v = \frac{Q}{r^2 \cdot \pi} = 2\, m/s$$

The wheel that will support the rope has a diameter of 16 inches which is 406 mm. With the diameter and the speed of the rope the angular velocity can be also calculated:

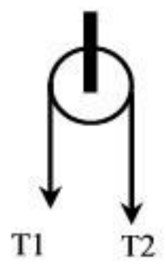
$$\omega = \frac{v_{rope}}{\frac{d}{2}} = 9.84\ {}^{rad}/_{s}$$

$$n = \frac{\omega \cdot 60}{2 \cdot \pi} = 94\ {}^{rev}/_{minute}$$

Having all this data, the force that the rope has to support because of the water is:

$$F = l \cdot r^2 \cdot \pi \cdot \rho \cdot g = 90.7\, N$$

This force is just the weight of the water but for the rope not to slide on the pulley bigger tension is needed.



This drawing represents the pulley, the bigger tension (T2) is the one that supports the water. To measure the maximun difference the next equations have to be used:

$$T2_{max} = T1 \cdot e^{\mu \cdot \Delta\theta}$$

Where:

μ is the friction coefficient

$\Delta\theta$ is the angle that the rope is touching the wheel

To solve the equation another one is needed in order to have system of equations.

$$T2 = T1 + W_{water}$$

Being $W_{water}$the weight of the water.

The rope is made of polypropylene and the surface of the wheel that is in contact is made of rubber. The friction coefficient between those materials is 0.55 and the angle is 180°.

$$T1 + W_{water} = T1 \cdot e^{\mu \cdot \Delta\theta} \rightarrow T1 = 19.582\ N$$

$$T2 = 110.207\ N$$

The value of T1 is the minimum value needed for the rope not to slide over the pulley. This value is a pretension that has to be installed.

The maximum tension the rope maker recommends is 160 N so the maximum values of the tension should be:

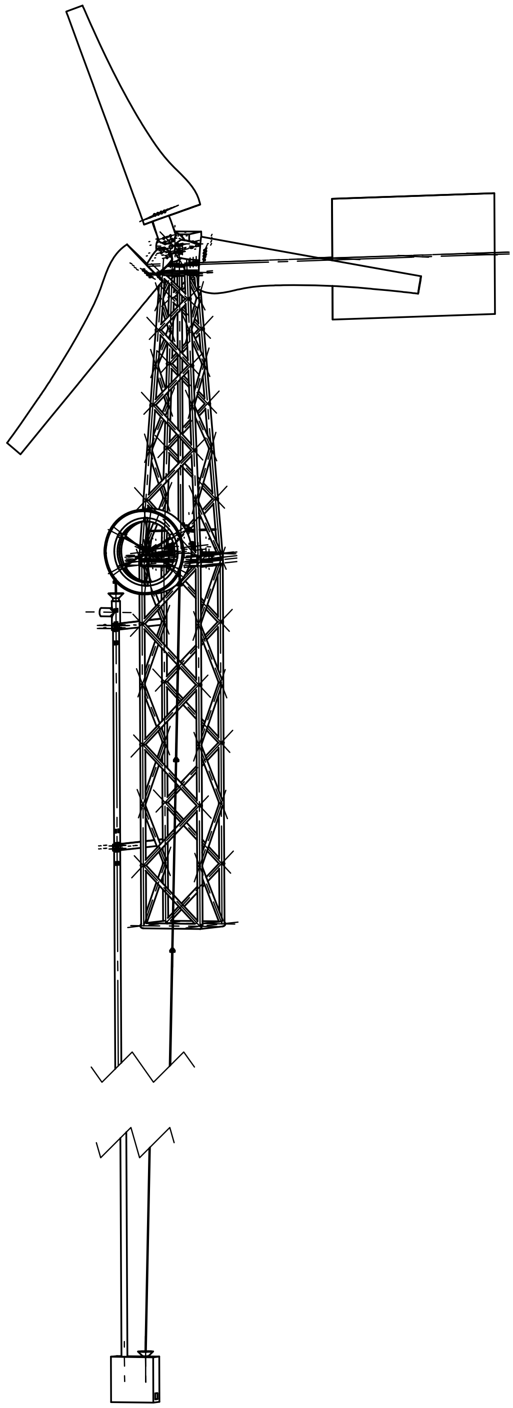
$$T1 = 70\ N$$

$$T2 = 160\ N$$

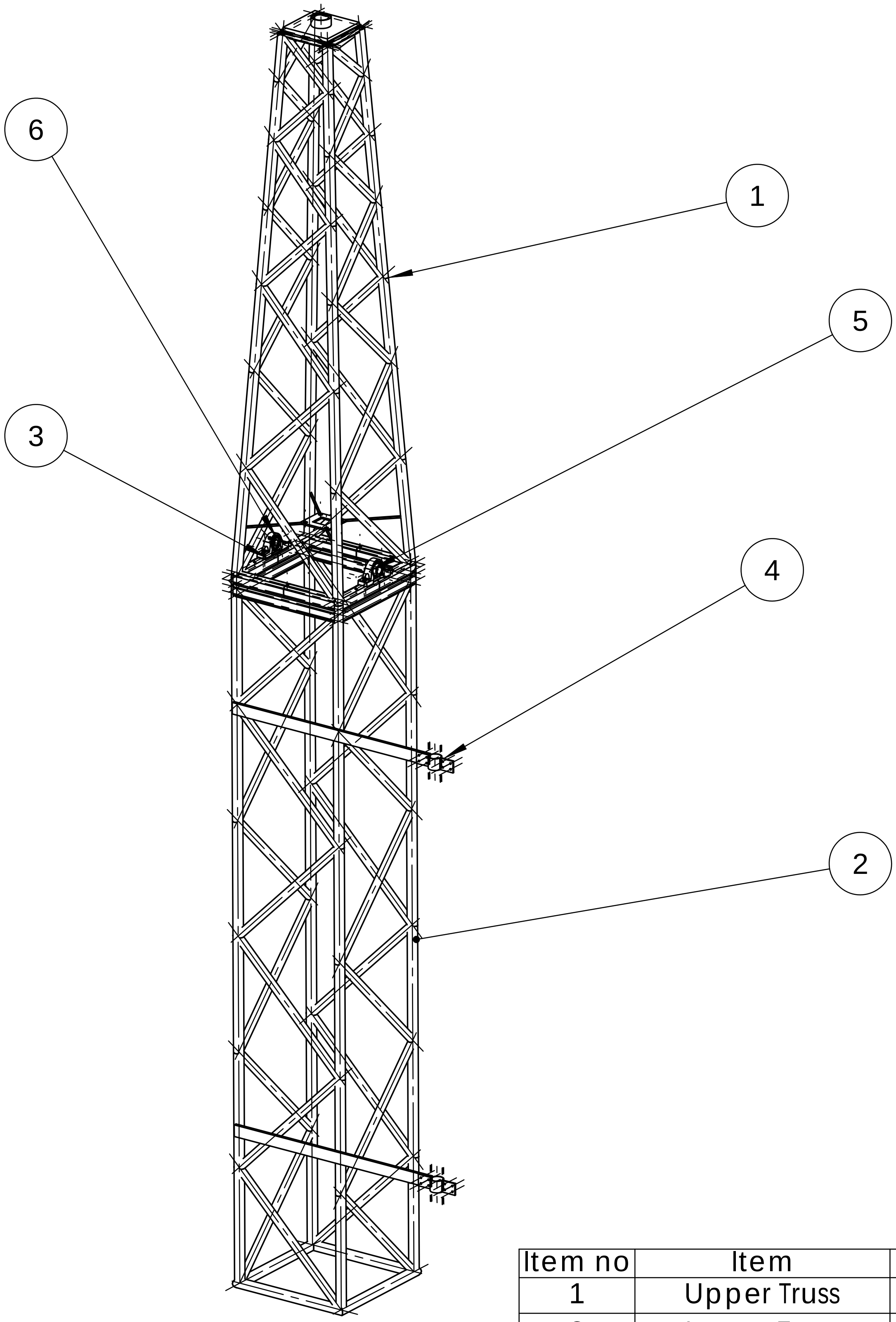
The end of the pipe has to support a force between 130 and 230 N that can make it buckle. In order to avoid that a heavy piece of concrete will be installed, weighting the force needed.

In order to minimize the buckling of the pipes there is a heavy part of concrete.

The weight of the pipe is 6 kg, what is 59 N. So the weight of the guide of the pump has to be at least 70 N of weight and no more than 170 N. As the PVC works better with tension loads than compression loads the weight of the piece is approximately the tensions of the rope.



Ingeniørhøjskolen i Århus Department of Mechanical Engineering Engineering College of Aarhus	Scale: 1:50	Group ID: Student ID: IY10227	Date: 23/5/2011 Initial: A.G.
Description: WIND TURBINE	Drawing no.: 1.0		



Item no	Item	Material	Qty
1	Upper Truss	Stainless Steel	1
2	Lower Truss	Stainless Steel	1
3	Bearing support 3	1023 Carbon Steel	1
4	Pipe support	1023 Carbon Steel	4
5	SKF SY25TF	-	1
6	SKF SY20TF	-	1

Ingeniørhøjskolen i Århus
Department of Mechanical Engineering
Engineering College of Aarhus

Scale: 1:50

Group ID:

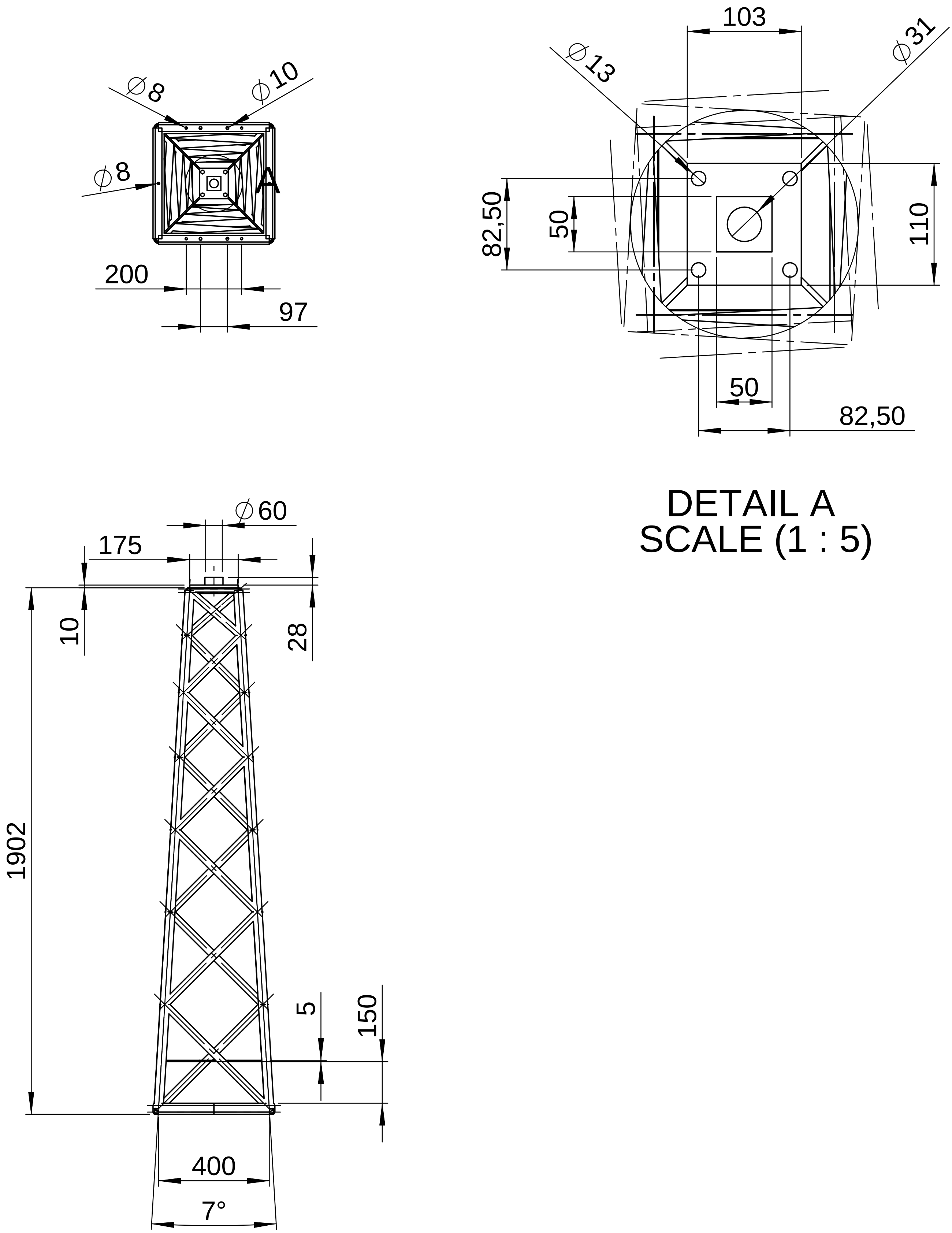
Date: 22/5/2011

Student ID: IY10227

Initial: A.G.

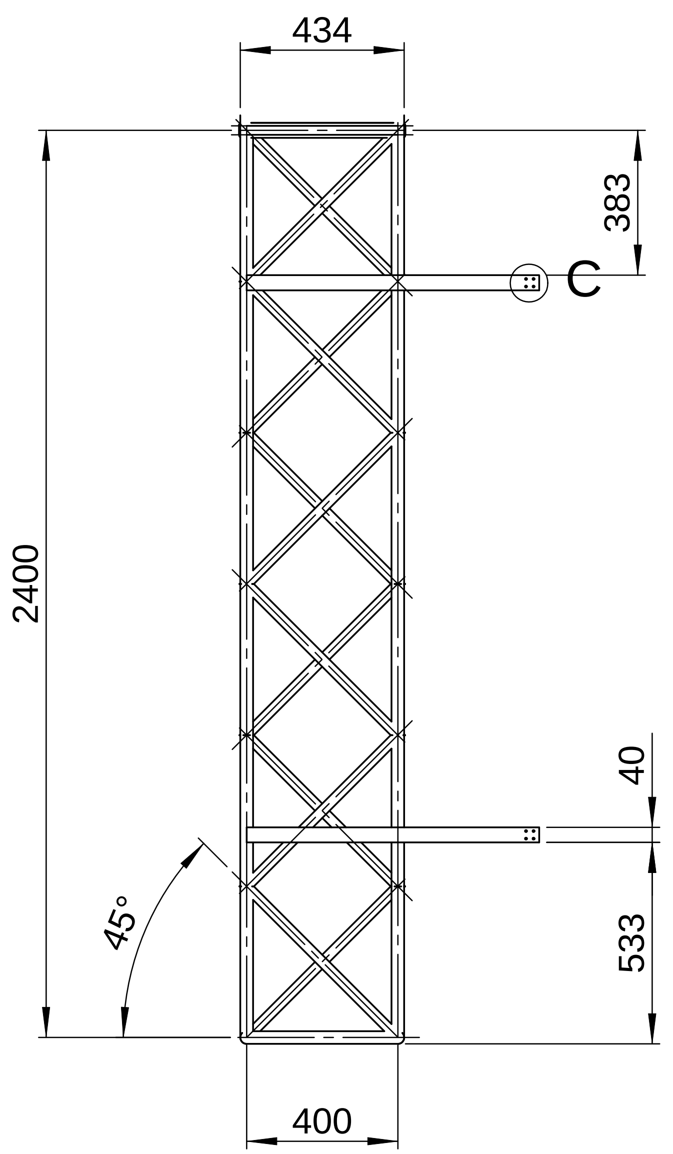
Description: TRUSS

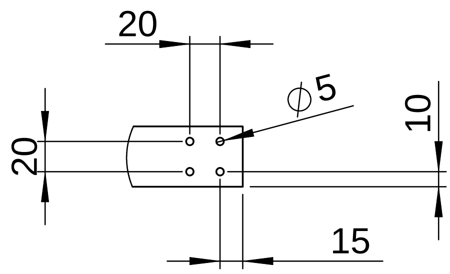
Drawing no.: 2.0



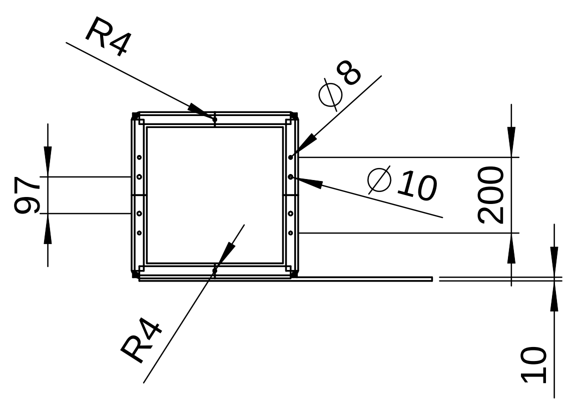
Qty.	Item	Item no.	Drawing no.	Material / Model no.
1	Upper truss			Stainless Steel (ferritic)

Ingeniørhøjskolen i Århus Department of Mechanical Engineering Engineering College of Aarhus	Scale: 1:20	Group ID: Student ID: IY10227	Date: 22/5/2011 Initial: A.G.
Description: UPPER TRUSS	Drawing no.: 2.1		





DETAIL C
SCALE (1 : 5)



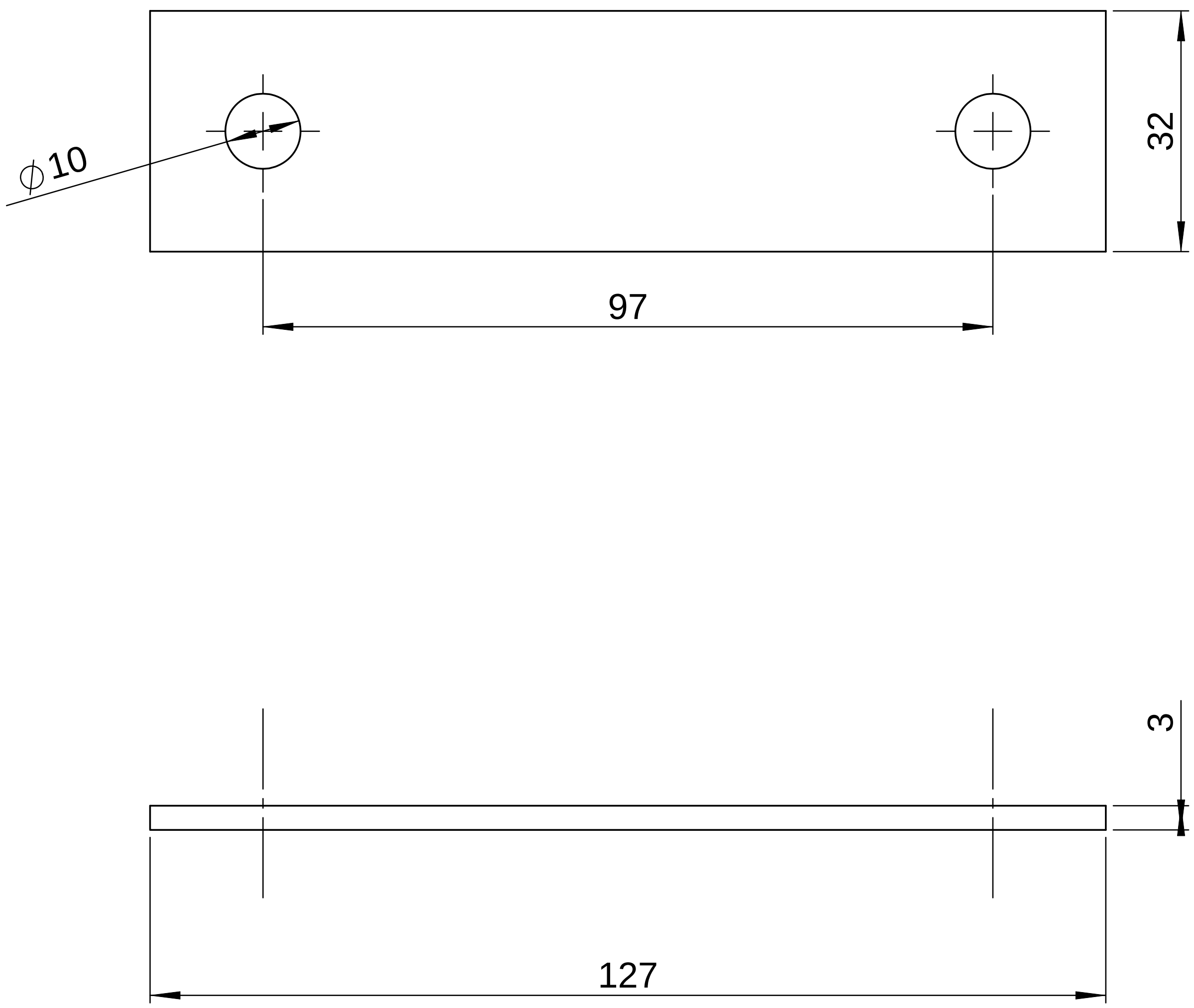
Qty.	Item	Item no.	Drawing no.	Material / Model no.
1	Lower truss			Stainless Steel (ferritic)

Ingeniørhøjskolen i Århus
Department of Mechanical Engineering
Engineering College of Aarhus

Scale: 1:20
Group ID:
Date: 22/5/2011
Student ID: IY10227
Initial: IY10227

Description: LOWER TRUSS

Drawing no.: 2.2



1	Support			1023 Carbon Steel
Qty.	**Item**	**Item no.**	**Drawing no.**	**Material / Model no.**

Ingeniørhøjskolen i Århus
Department of Mechanical Engineering
Engineering College of Aarhus

Scale: 1:1

Group ID:

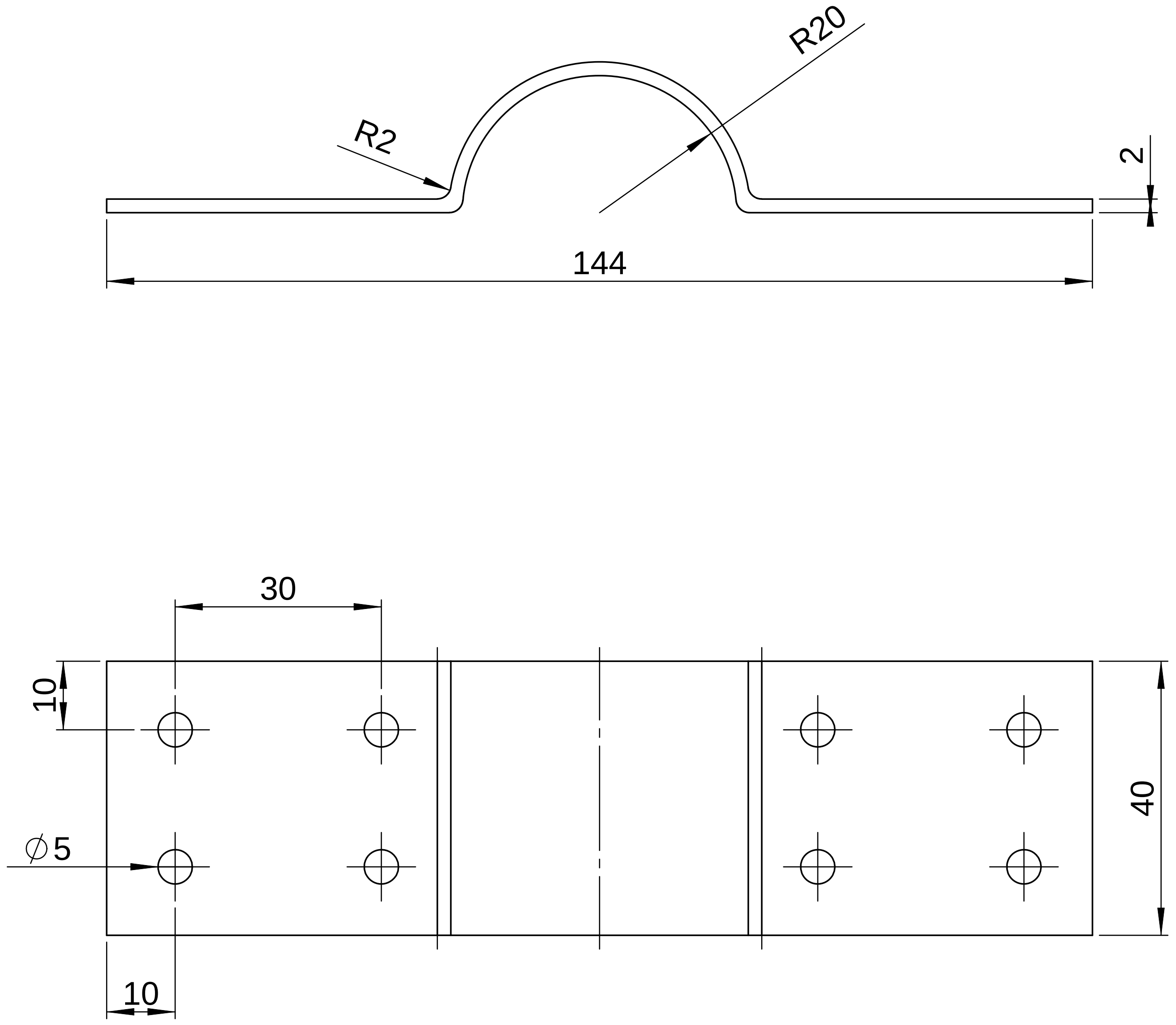
Date: 22/5/2011

Student ID: IY10227

Initial: A.G.

Description: BEARING SUPPORT 3

Drawing no.: 2.3



4	Pipes support			1023 Carbon Steel
Qty.	**Item**	**Item no.**	**Drawing no.**	**Material / Model no.**

Ingeniørhøjskolen i Århus
Department of Mechanical Engineering
Engineering College of Aarhus

Scale: 1:1

Group ID:

Date: 22/5/2011

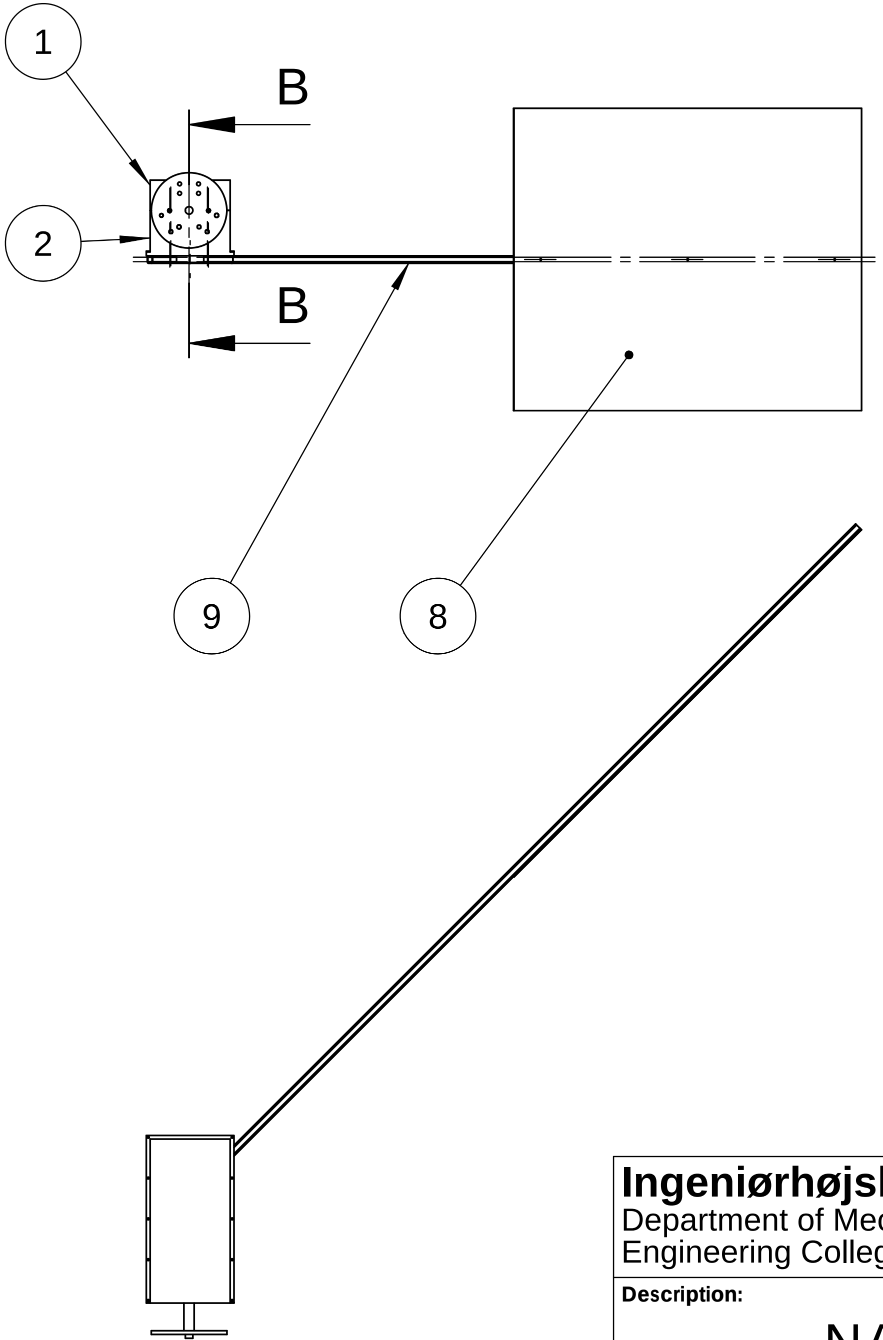
Student ID: IY10227

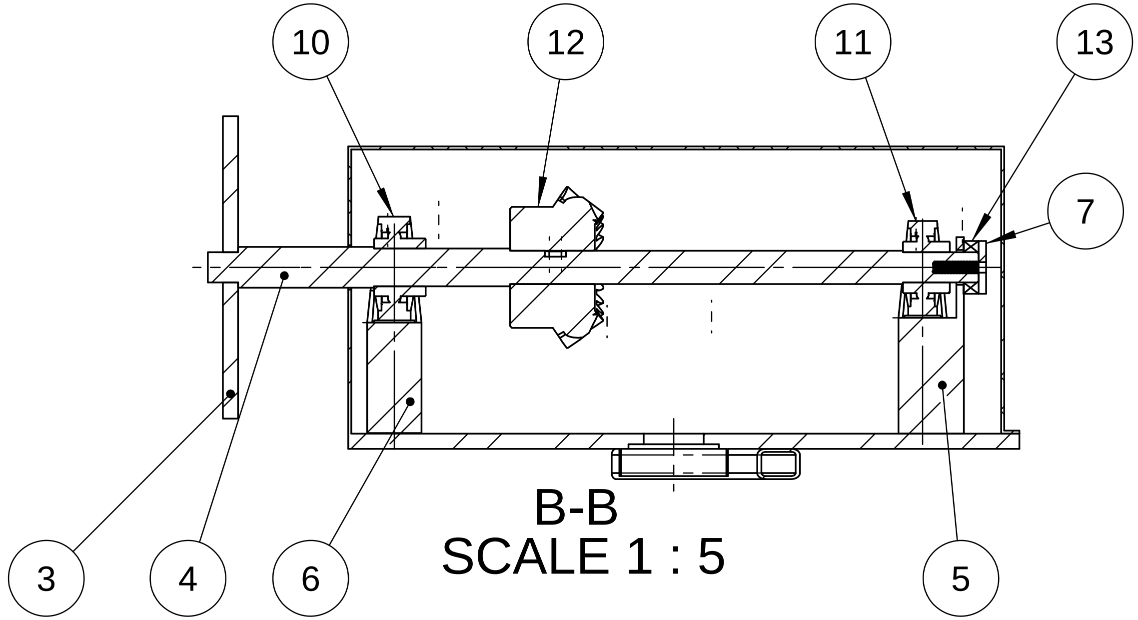
Initial: A.G.

Description:

PIPES SUPPORT

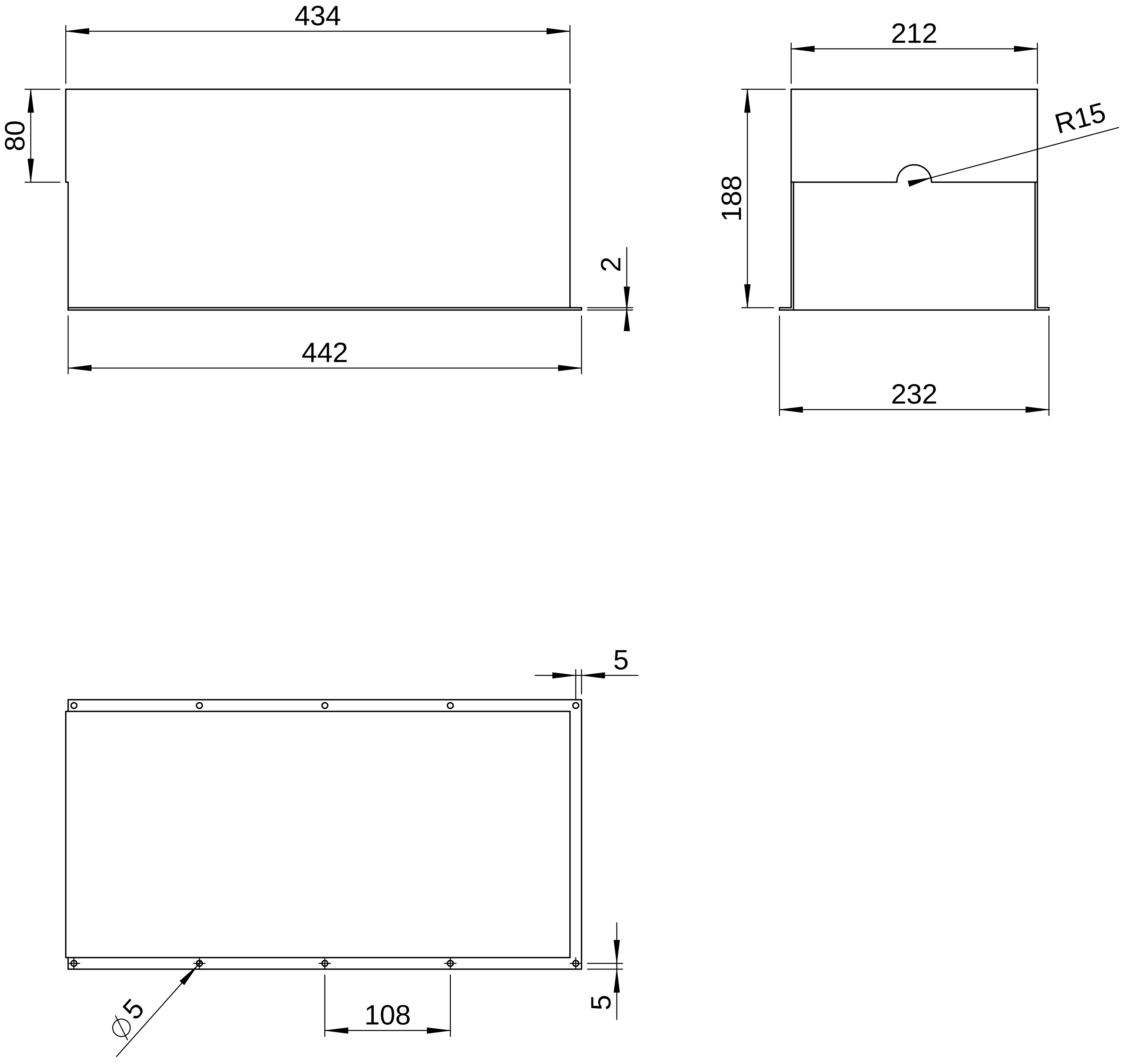
Drawing no.: 2.4





Item no	Item	Material	Qty
1	Nacelle cover	1060 Alloy	1
2	Nacelle base	1023 Carbon steel	1
3	Rotor plate	1023 Carbon steel	1
4	Rotor shaft	AISI 4340 Steel	1
5	Bearing support 1	1023 Carbon steel	1
6	Bearing support 2	1023 Carbon steel	1
7	Rotor shaft plate	1023 Carbon steel	1
8	Tail	1060 Alloy	1
9	Tail support	1023 Carbon steel	1
10	SKF SY 25TF	-	1
11	SKF SY 20TF	-	1
12	QTC SBS5-2030L	-	1
13	SKF 51104	-	1

Ingeniørhøjskolen i Århus Department of Mechanical Engineering Engineering College of Aarhus	**Scale:** 1:50	**Group ID:** **Student ID:** IY10227	**Date:** 23/5/2011 **Initial:** A.G.
Description: NACELLE ASSEMBLY	**Drawing no.:** 3.0		



1	Nacelle cover			1060 Alloy
Qty.	Item	Item no.	Drawing no.	Material / Model no.

Ingeniørhøjskolen i Århus
Department of Mechanical Engineering
Engineering College of Aarhus

Scale: 1:5

Group ID:

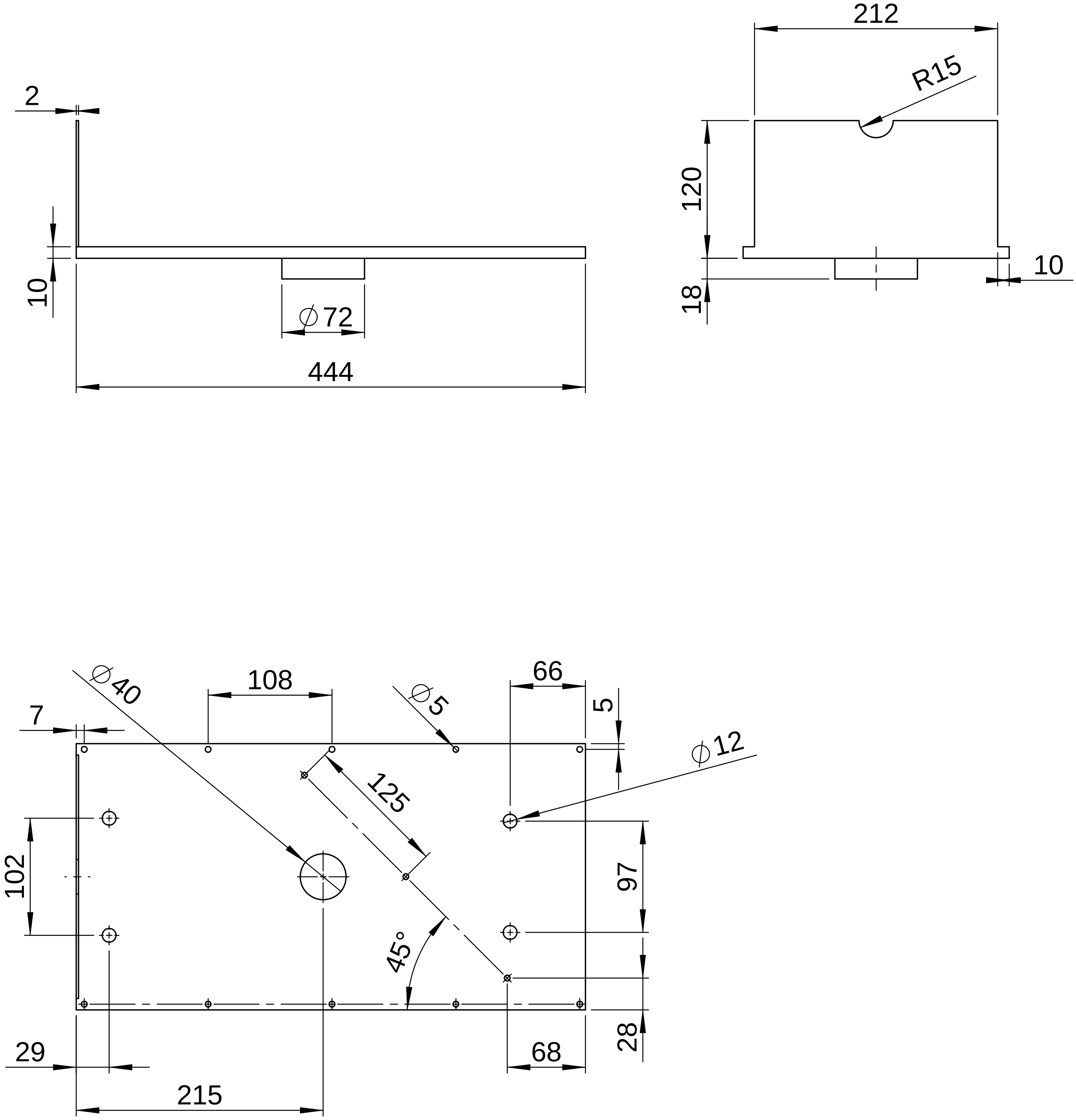
Date: 22/5/2011

Student ID: IY10227

Initial: A.G.

Description: Nacelle cover

Drawing no.: 3.1



1	Nacelle base			1023 Carbon Steel
Qty.	Item	Item no.	Drawing no.	Material / Model no.

Ingeniørhøjskolen i Århus
Department of Mechanical Engineering
Engineering College of Aarhus

Scale: 1:5

Group ID:

Date: 22/5/2011

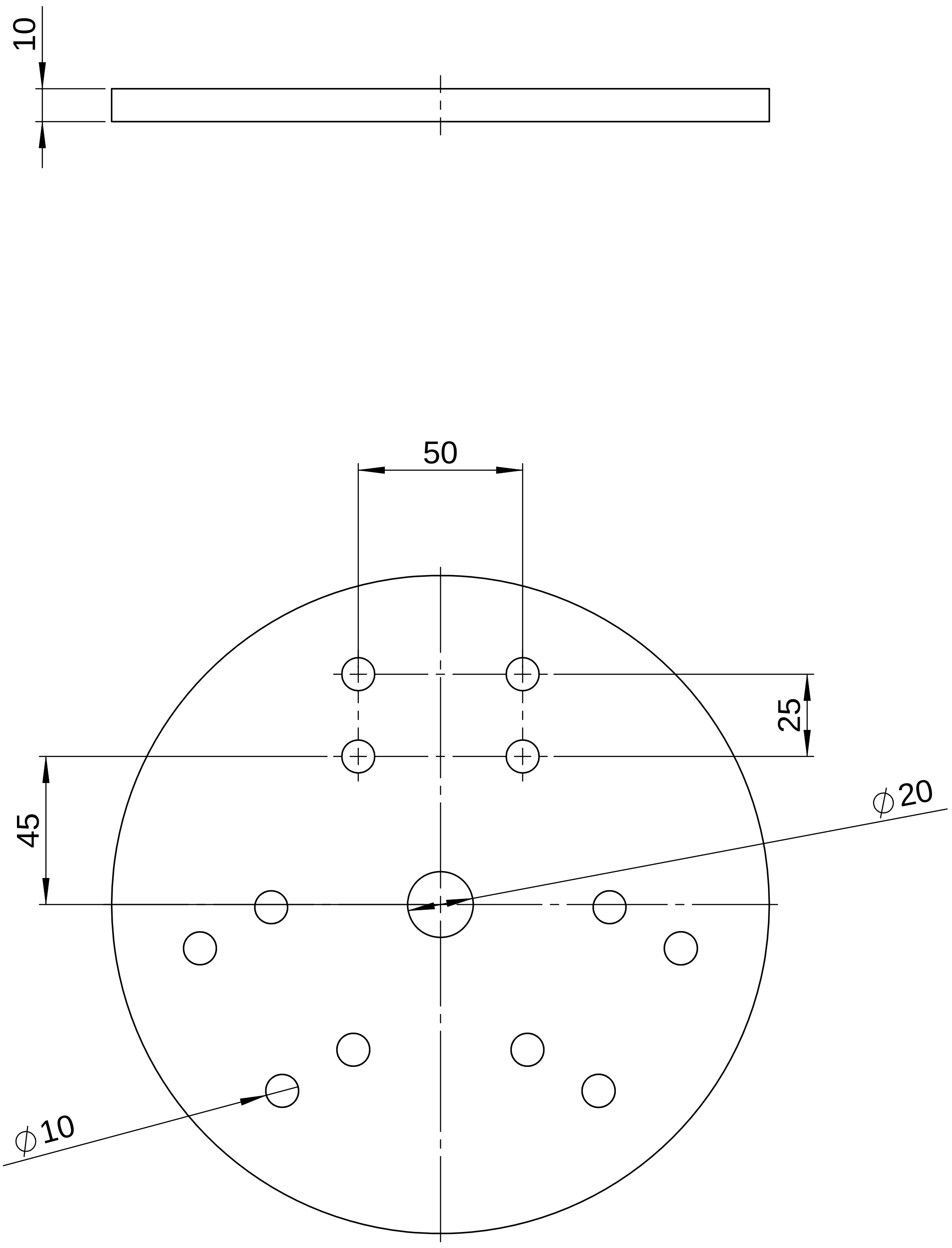
Student ID: IY10227

Initial: A.G.

Description:

NACELLE BASE

Drawing no.: 3.2



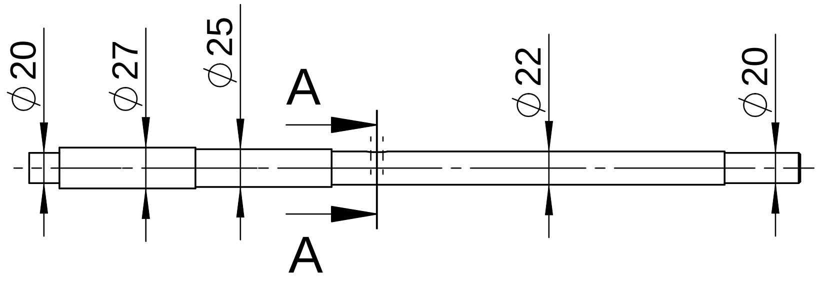
1	Rotor plate			1023 Carbon Steel
Qty.	Item	Item no.	Drawing no.	Material / Model no.

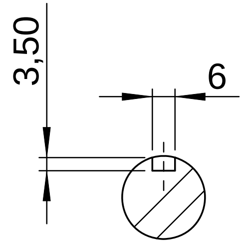
Ingeniørhøjskolen i Århus
Department of Mechanical Engineering
Engineering College of Aarhus

Scale: 1:2	Group ID:	Date: 22/5/2011	
	Student ID: IY10227	Initial: A.G.	

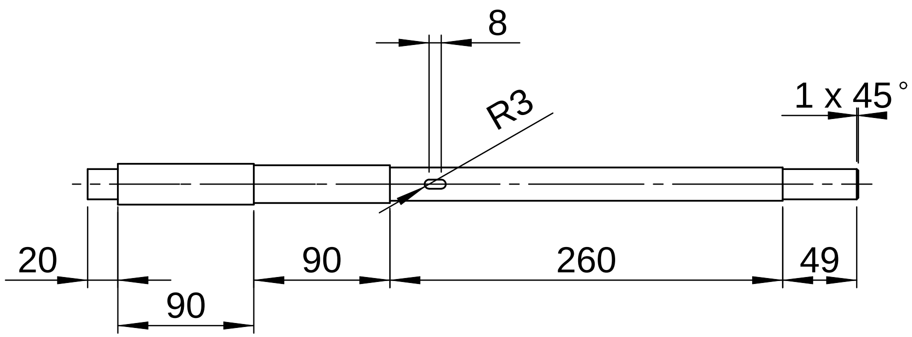
Description: ROTOR PLATE

Drawing no.: 3.3



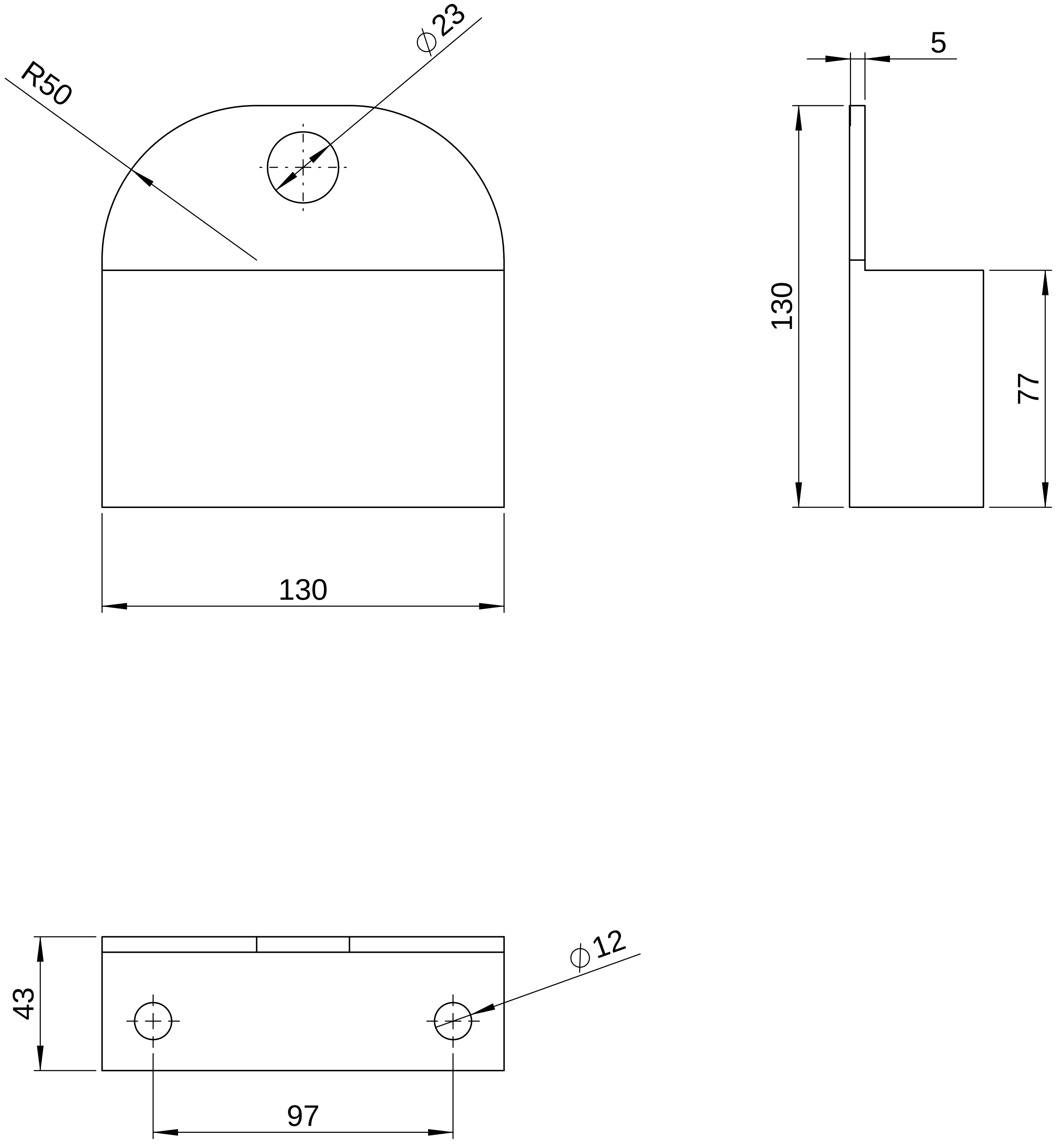


A-A
SCALE (1 : 2)



1	Shaft			AISI 4340 Steel
Qty.	**Item**	**Item no.**	**Drawing no.**	**Material / Model no.**

Ingeniørhøjskolen i Århus Department of Mechanical Engineering Engineering College of Aarhus	**Scale:** 1:5	**Group ID:** **Student ID:** IY10227	**Date:** 22/5/2011 **Initial:** A.G.
Description: ROTOR SHAFT	**Drawing no.:** 3.4		



Qty.	Item	Item no.	Drawing no.	Material / Model no.
1	Support			1023 Carbon Steel

Ingeniørhøjskolen i Århus
Department of Mechanical Engineering
Engineering College of Aarhus

Scale: 1:2

Group ID:

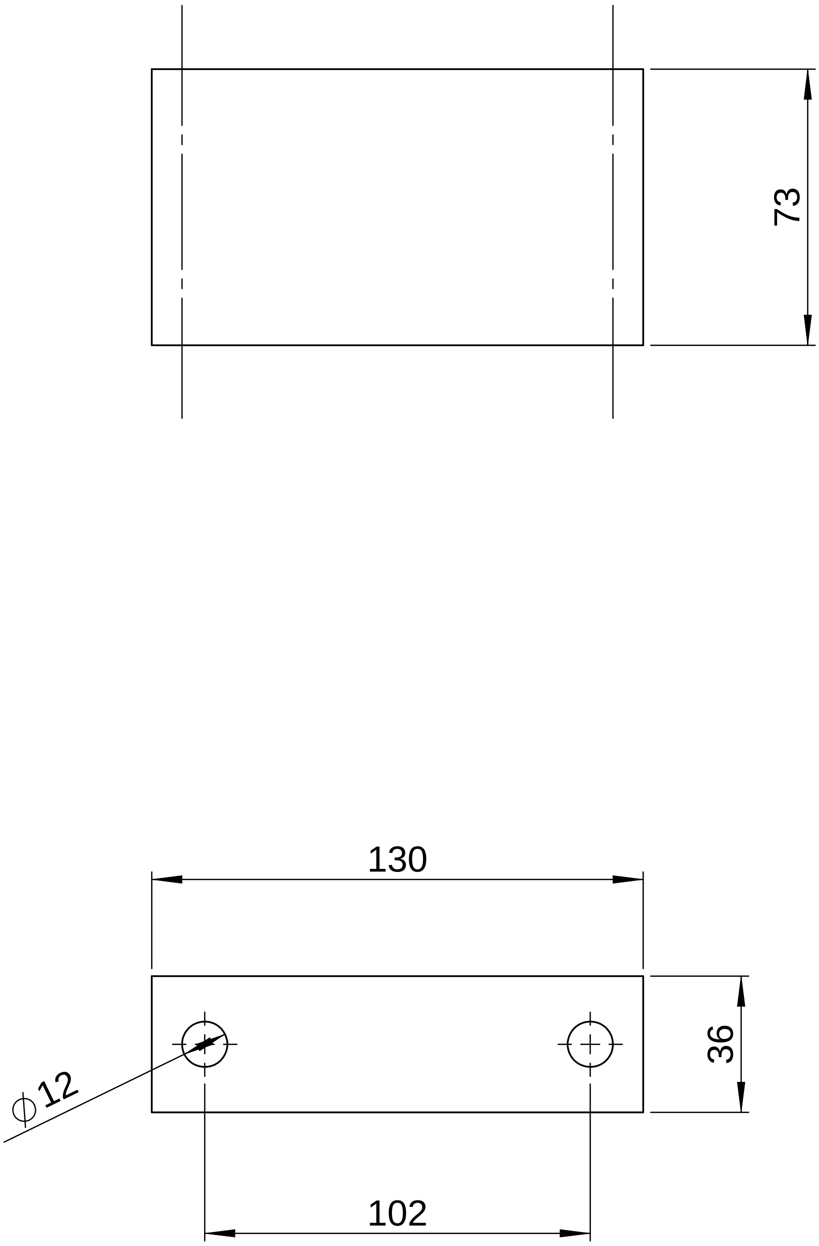
Date: 22/5/2011

Student ID: IY10227

Initial: A.G.

Description: BEARINGS SUPPORT 1

Drawing no.: 3.5



1	Support			1023 Carbon Steel
Qty.	Item	Item no.	Drawing no.	Material / Model no.

Ingeniørhøjskolen i Århus
Department of Mechanical Engineering
Engineering College of Aarhus

Scale: 1:2

Group ID:

Date: 22/5/2011

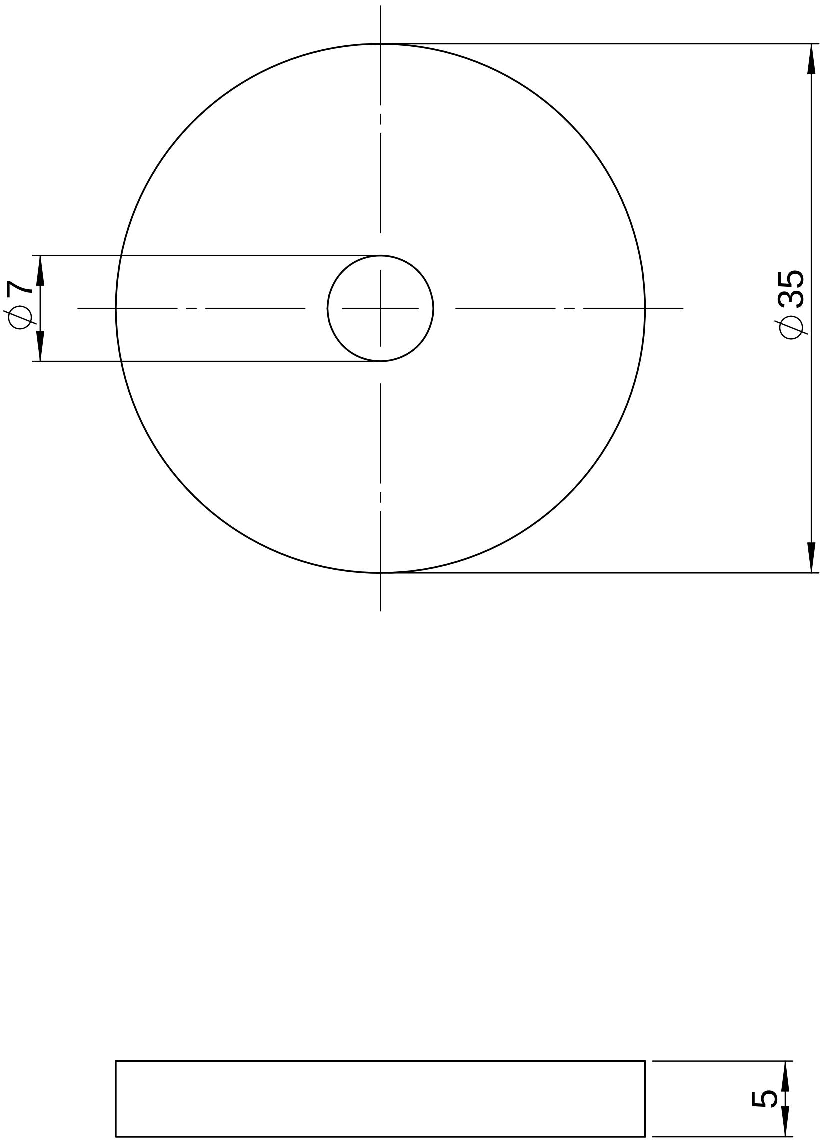
Student ID: IY10227

Initial: A.G.

Description:

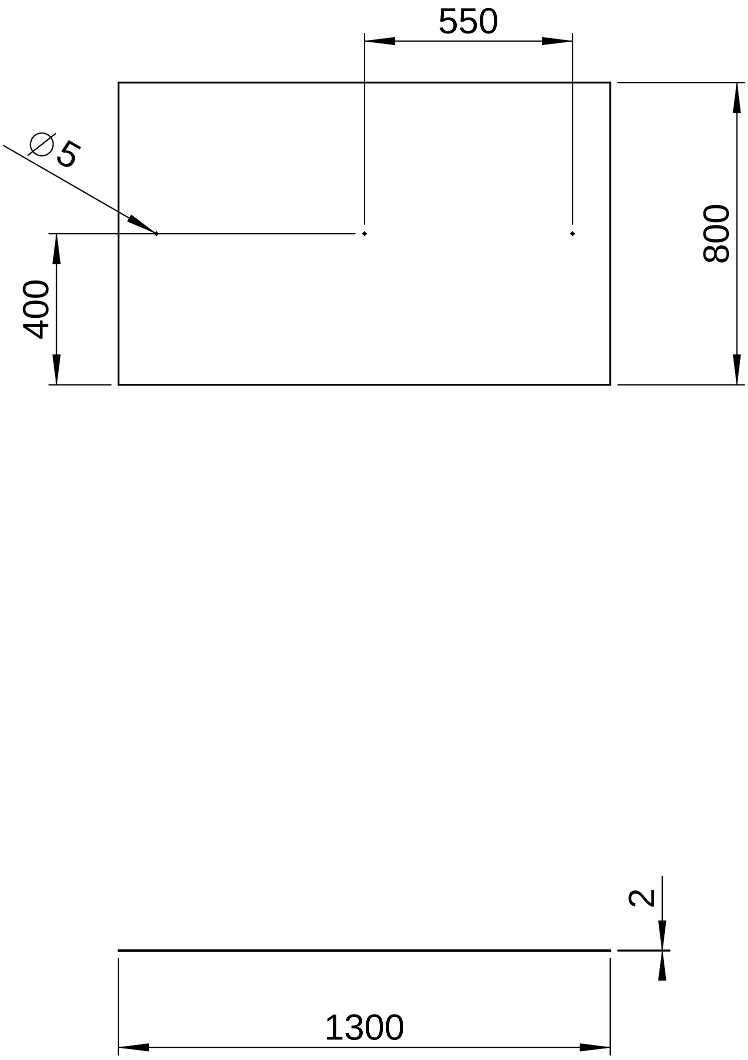
BEARING SUPPORT 2

Drawing no.: 3.6



1	Plate			1023 Carbon steel
Qty.	Item	Item no.	Drawing no.	Material / Model no.

Ingeniørhøjskolen i Århus Department of Mechanical Engineering Engineering College of Aarhus	Scale: 2:1	Group ID:	Date: 22/5/2011
		Student ID: IY10227	Initial: A.G.
Description: ROTOR SHAFT PLATE	Drawing no.: 3.7		



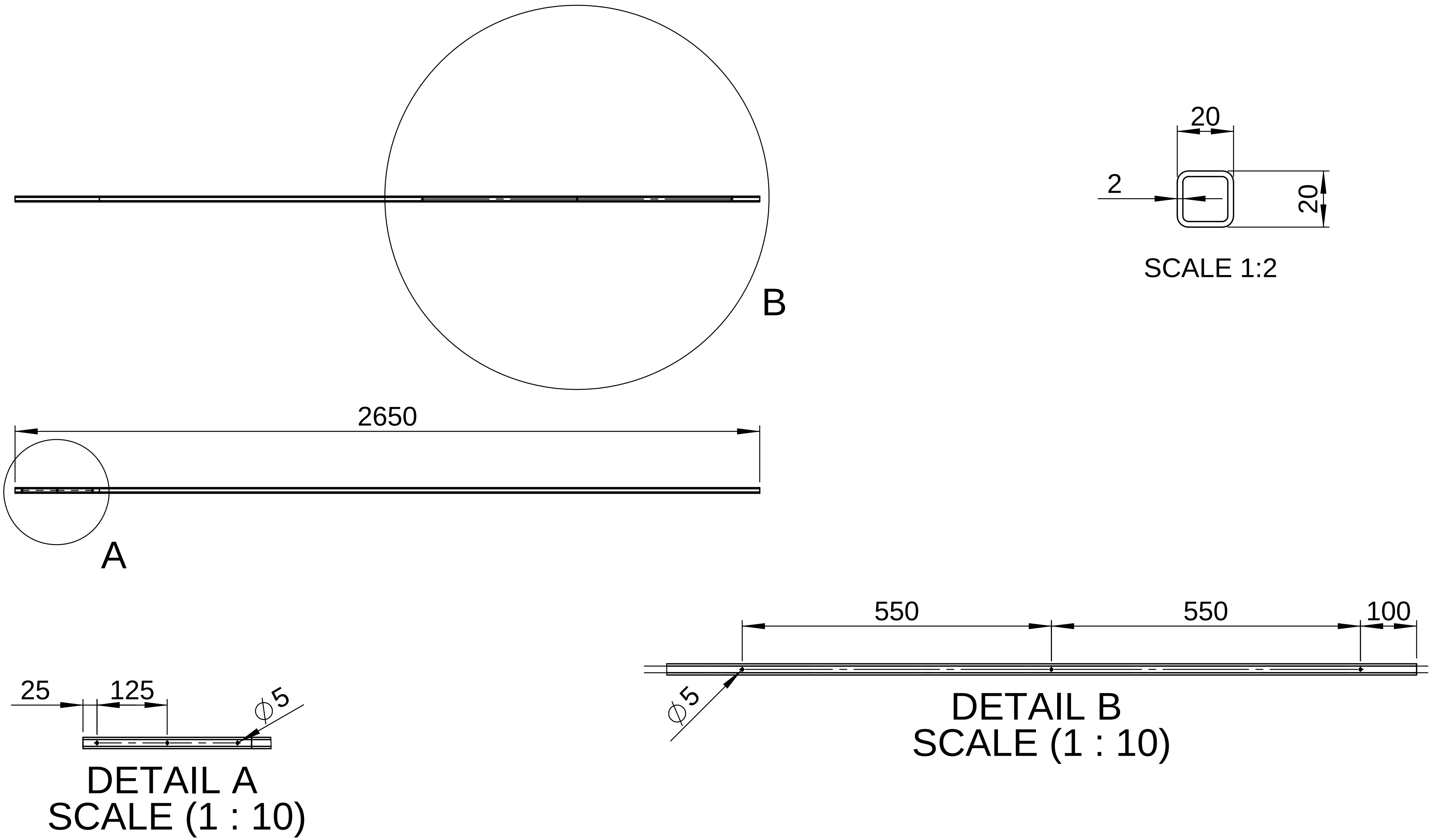
1	Tail			1060 Alloy
Qty.	**Item**	**Item no.**	**Drawing no.**	**Material / Model no.**

Ingeniørhøjskolen i Århus
Department of Mechanical Engineering
Engineering College of Aarhus

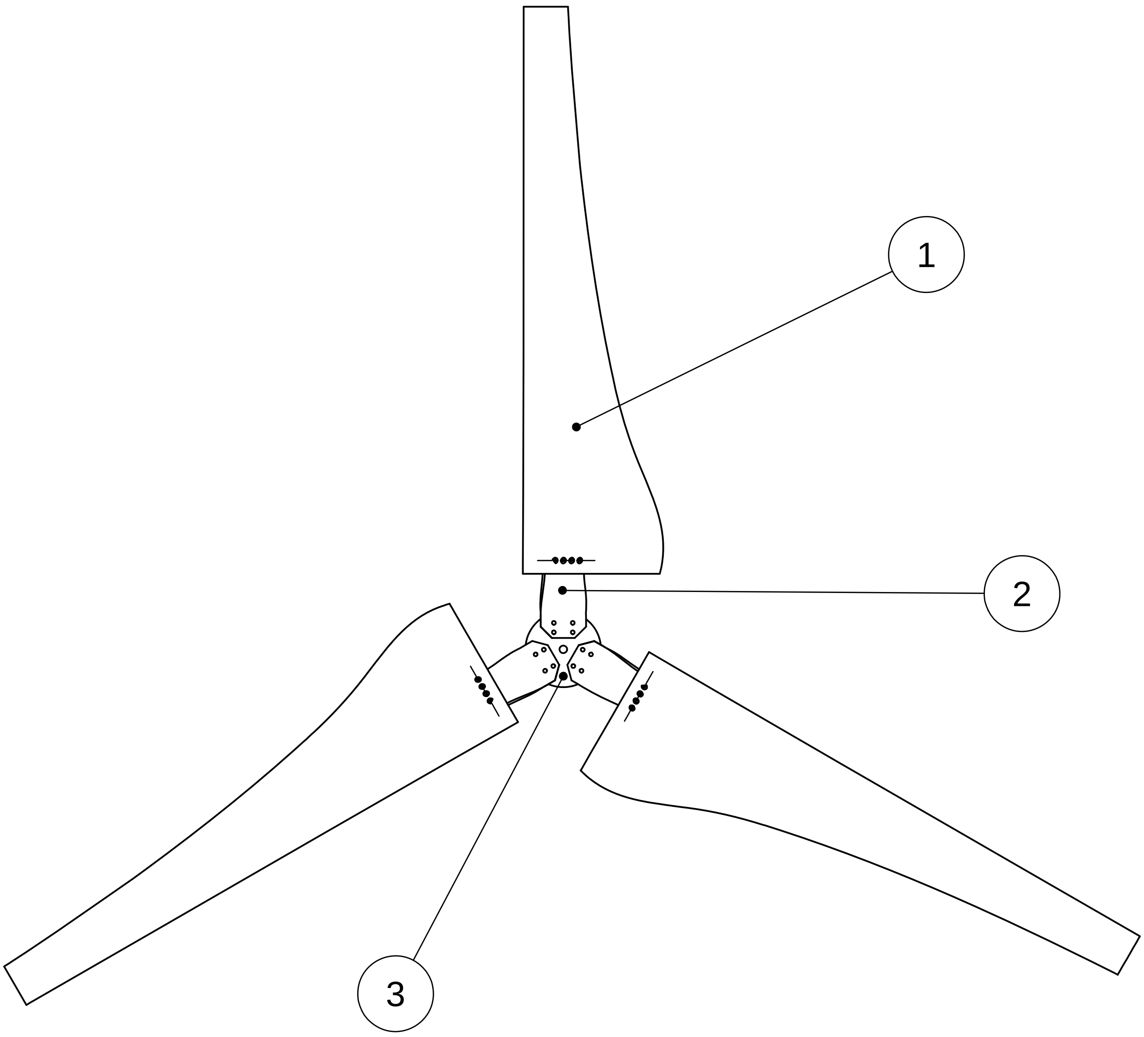
Scale: 1:20	**Group ID:**	**Date:** 22/5/2011
	Student ID: IY10227	**Initial:** A.G.

Description: TAIL

Drawing no.: 3.8



1	Tail support			1023 Carbon Steel	
Qty.	**Item**	**Item no.**	**Drawing no.**	**Material / Model no.**	
Ingeniørhøjskolen i Århus Department of Mechanical Engineering Engineering College of Aarhus			**Scale:** 1:20	**Group ID:**	**Date:** 22/5/2011
				Student ID: IY10227	**Initial:** A.G.
Description: TAIL SUPPORT			**Drawing no.:** 3.9		



Item no	Item	Material	Qty
1	Blade	Pine wood	3
2	Join part	1023 Carbon Steel	3
3	Rotor plate	1023 Carbon Steel	1

Ingeniørhøjskolen i Århus
Department of Mechanical Engineering
Engineering College of Aarhus

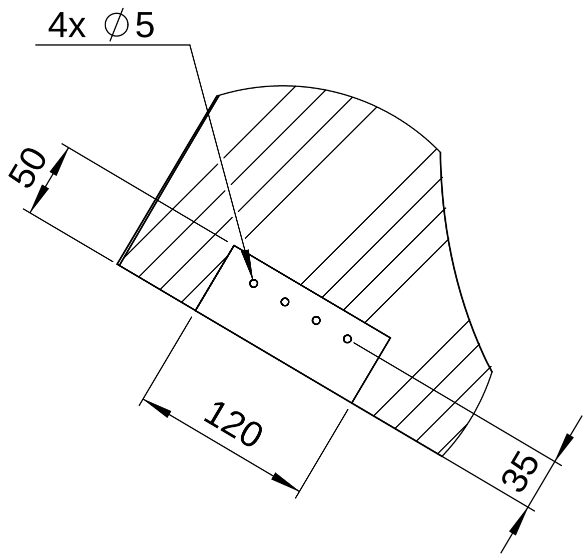
Scale:	Group ID:	Date:
1:50		23/5/2011
	Student ID: IY10227	Initial: A.G.

Description:

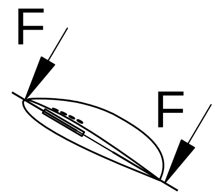
ROTOR ASSEMBLY

Drawing no.:

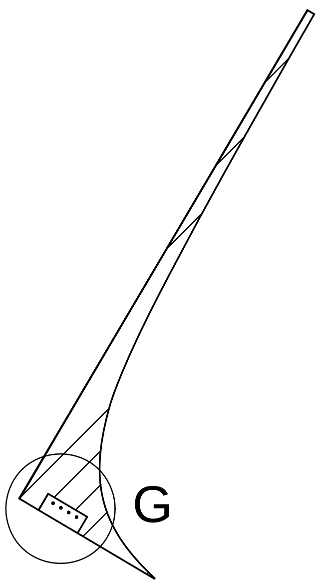
4.0



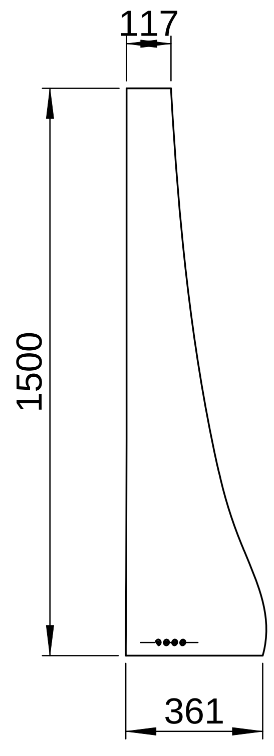
DETAIL G
SCALE (1 : 5)



NACA 23012

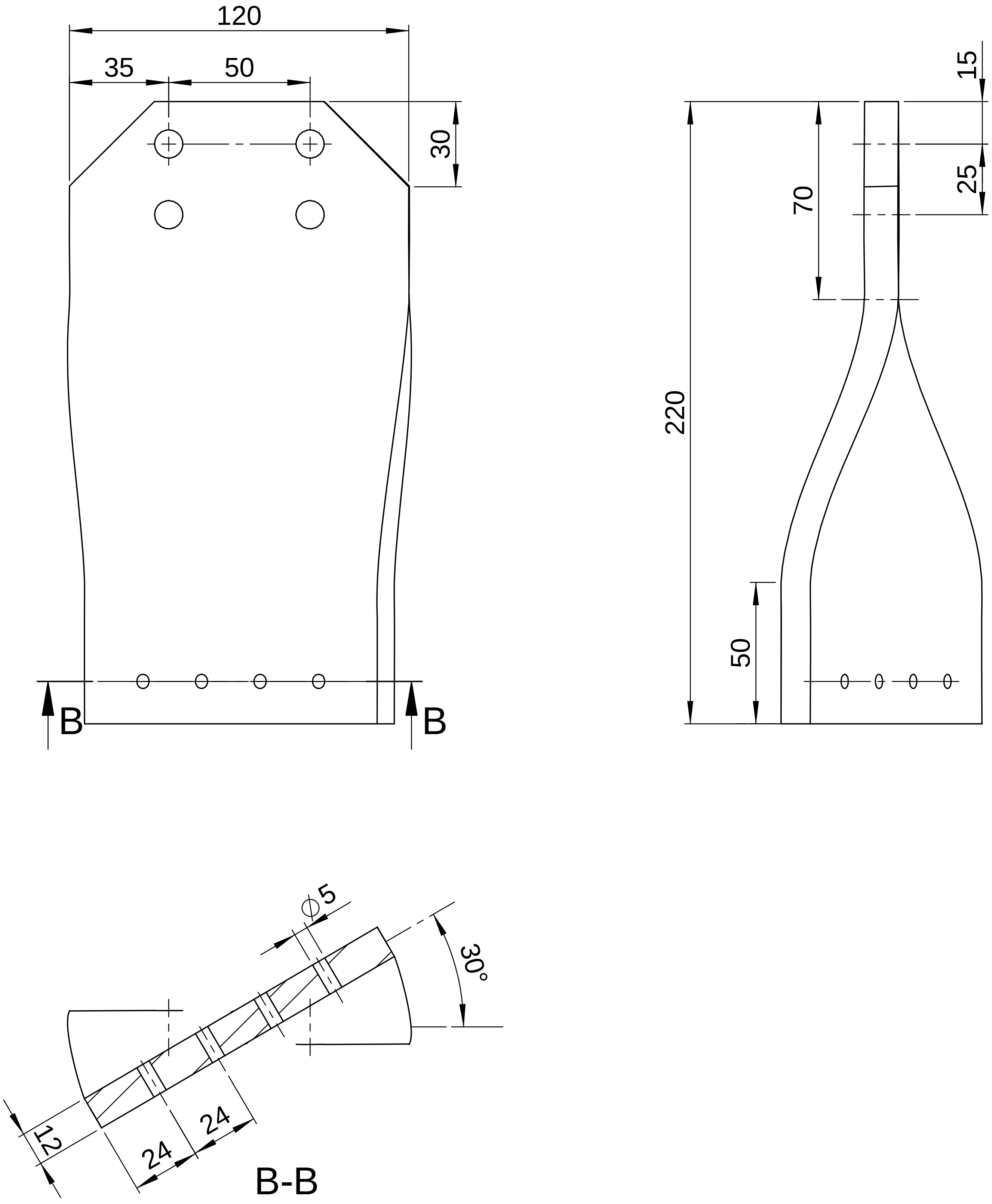


F-F



3	NACA profile 23012			Pine Wood
Qty.	**Item**	**Item no.**	**Drawing no.**	**Material / Model no.**

Ingeniørhøjskolen i Århus Department of Mechanical Engineering Engineering College of Aarhus	**Scale:** 1:20	**Group ID:** **Student ID:** IY10227	**Date:** 22/5/2011 **Initial:** A.G.
Description: BLADE	**Drawing no.:** 4.1		



3	Join Part			1023 Carbon Steel
Qty.	**Item**	**Item no.**	**Drawing no.**	**Material / Model no.**

Ingeniørhøjskolen i Århus
Department of Mechanical Engineering
Engineering College of Aarhus

Scale: 1:2

Group ID:

Date: 22/5/2011

Student ID: IY10227

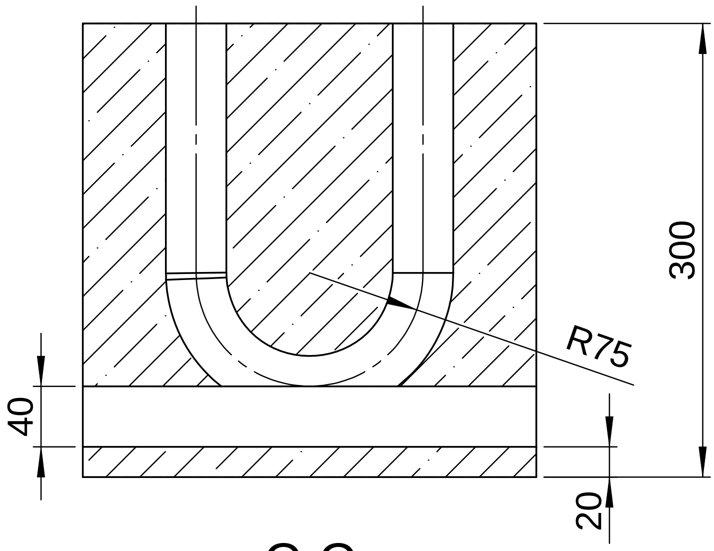
Initial: A.G.

Description:

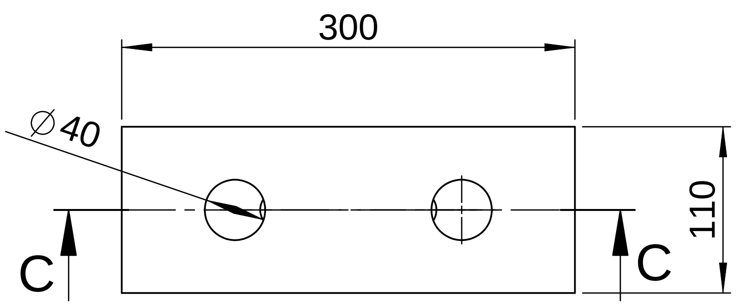
JOIN PART

Drawing no.:

4.2

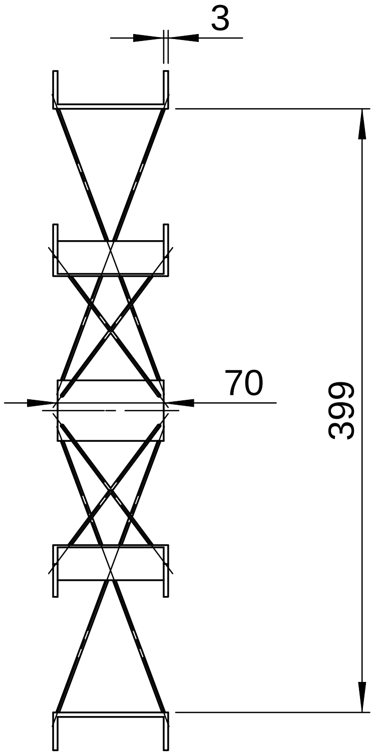


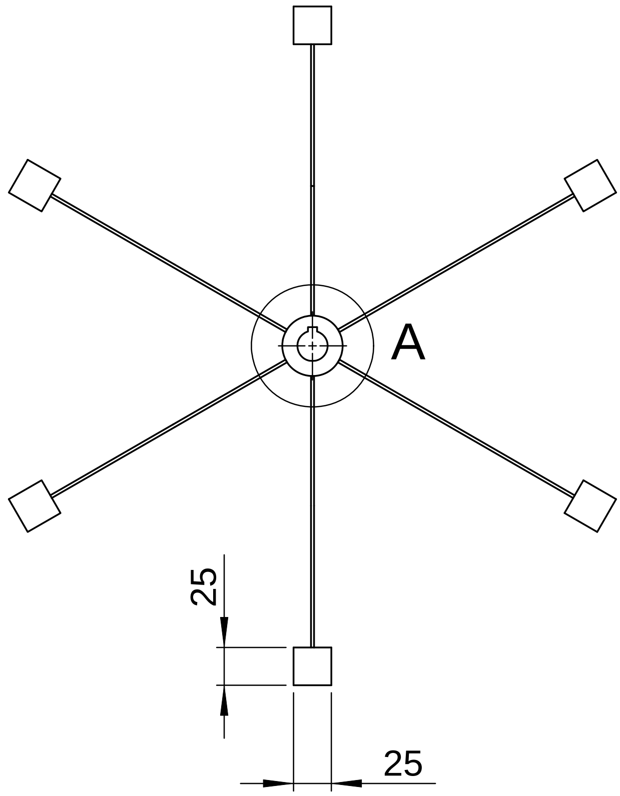
C-C

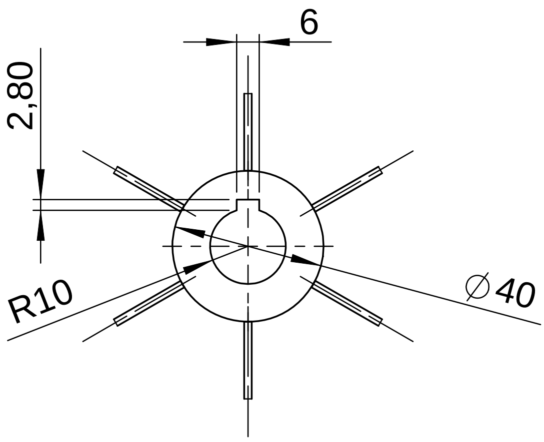


Qty.	Item	Item no.	Drawing no.	Material / Model no.
1	Concrete guide			Concrete

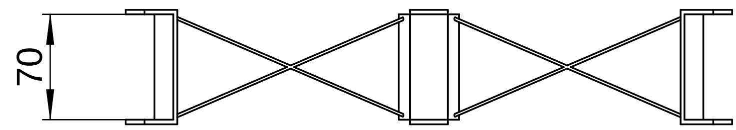
Ingeniørhøjskolen i Århus Department of Mechanical Engineering Engineering College of Aarhus	**Scale:** 1:5	**Group ID:** **Student ID:** IY10227	**Date:** 22/5/2011 **Initial:** A.G.
Description: CONCRETE GUIDE	**Drawing no.:** 5.1		







DETAIL A
SCALE (1 : 2)



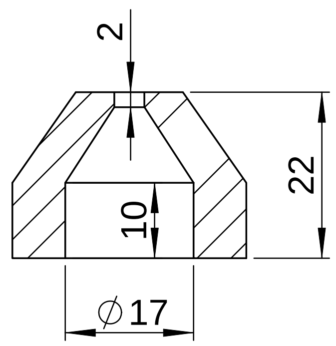
Qty.	Item	Item no.	Drawing no.	Material / Model no.
1	Wheel			1023 Carbon Steel

Ingeniørhøjskolen i Århus
Department of Mechanical Engineering
Engineering College of Aarhus

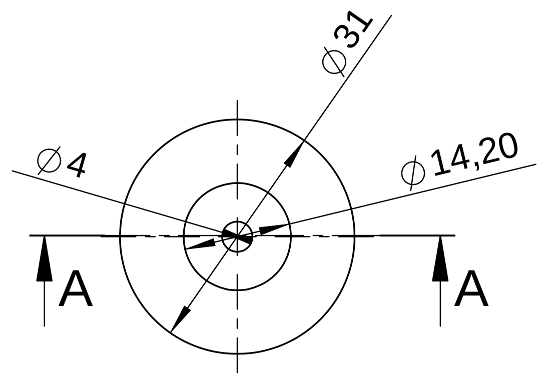
Scale:	Group ID:	Date:
1:5		22/5/2011
	Student ID: IY10227	Initial: A.G.

Description: PUMP WHEEL

Drawing no.: 5.2



A-A



Qty.	Item	Item no.	Drawing no.	Material / Model no.	
15	Piston			Polyethylene	

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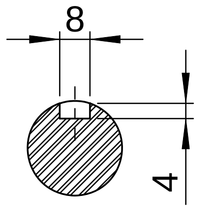
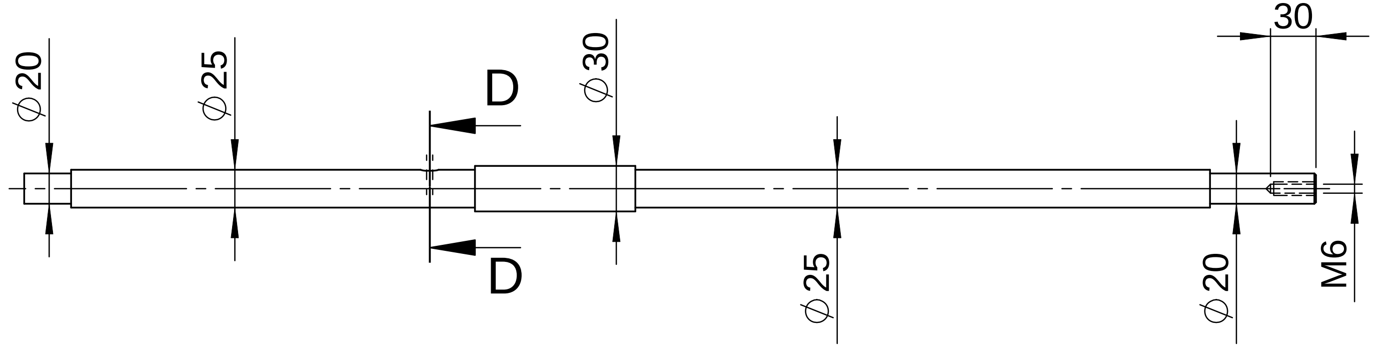
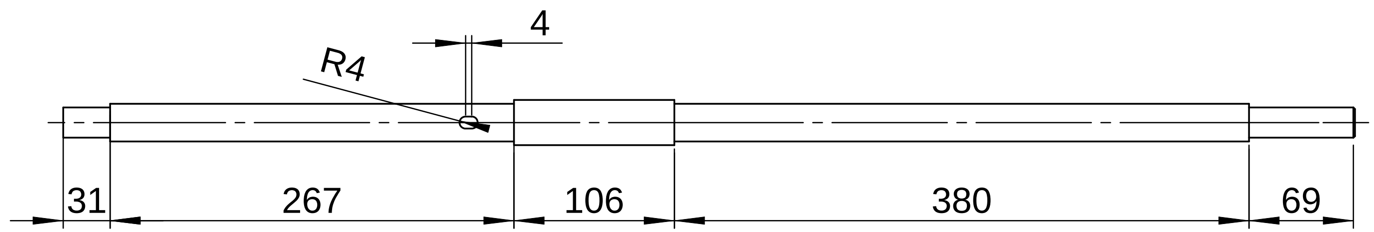
		Student ID: IY10227	**Initial:** A.G.
Description: PISTON	**Drawing no.:** 5.3		

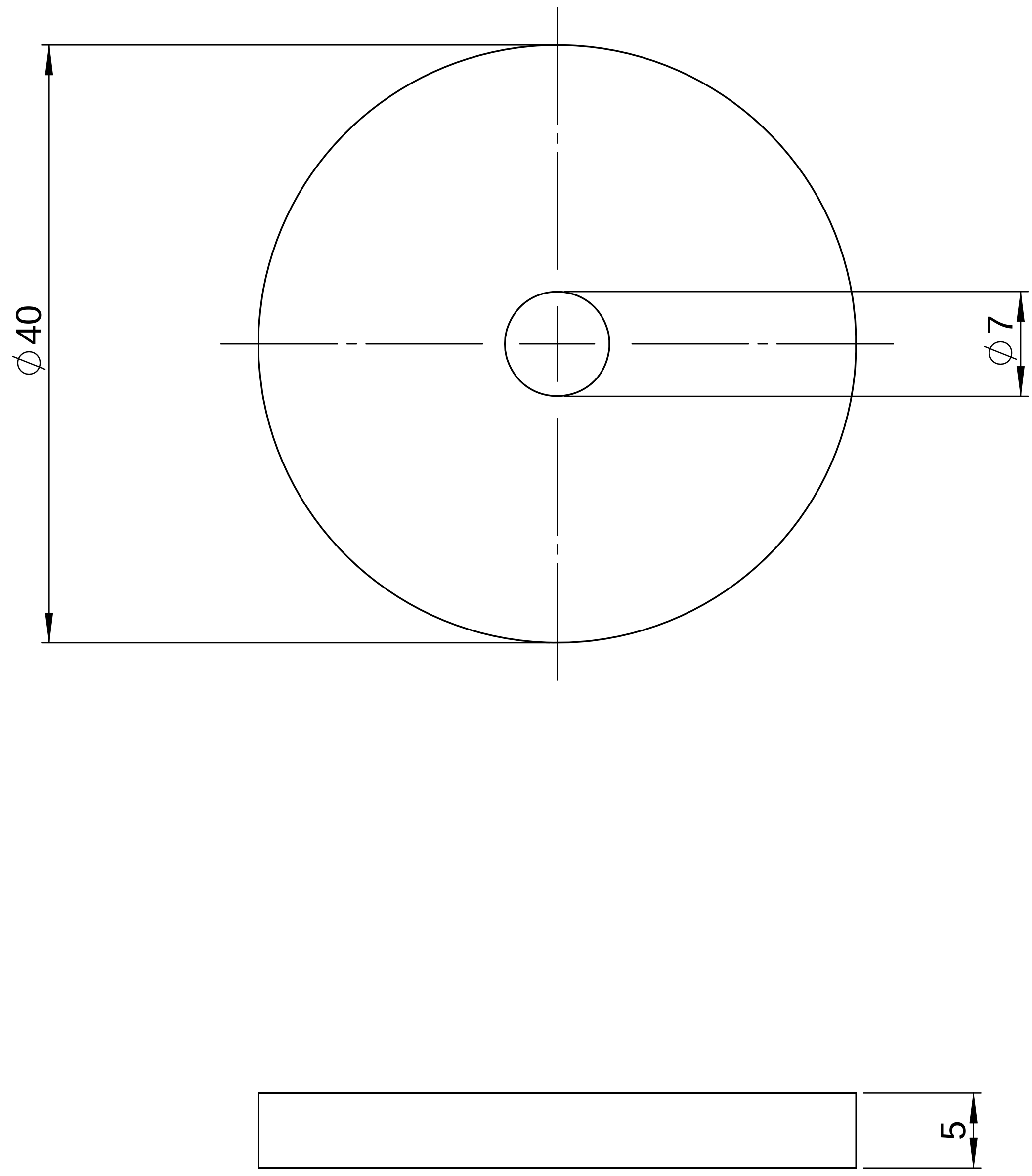


D-D
SCALE (1 : 2)





1	Pump shaft			AISI 4340 Steel	
Qty.	**Item**	**Item no.**	**Drawing no.**	**Material / Model no.**	
Ingeniørhøjskolen i Århus Department of Mechanical Engineering Engineering College of Aarhus			**Scale:** 1:10	**Group ID:** **Student ID:** IY10227	**Date:** 22/5/2011 **Initial:** A.G.
Description: PUMP SHAFT			**Drawing no.:** 5.4		



1	Plate			1023 Carbon Steel
Qty.	**Item**	**Item no.**	**Drawing no.**	**Material / Model no.**

Ingeniørhøjskolen i Århus
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Scale: 2:1

Group ID:

Date: 22/5/2011

Student ID: IY10227

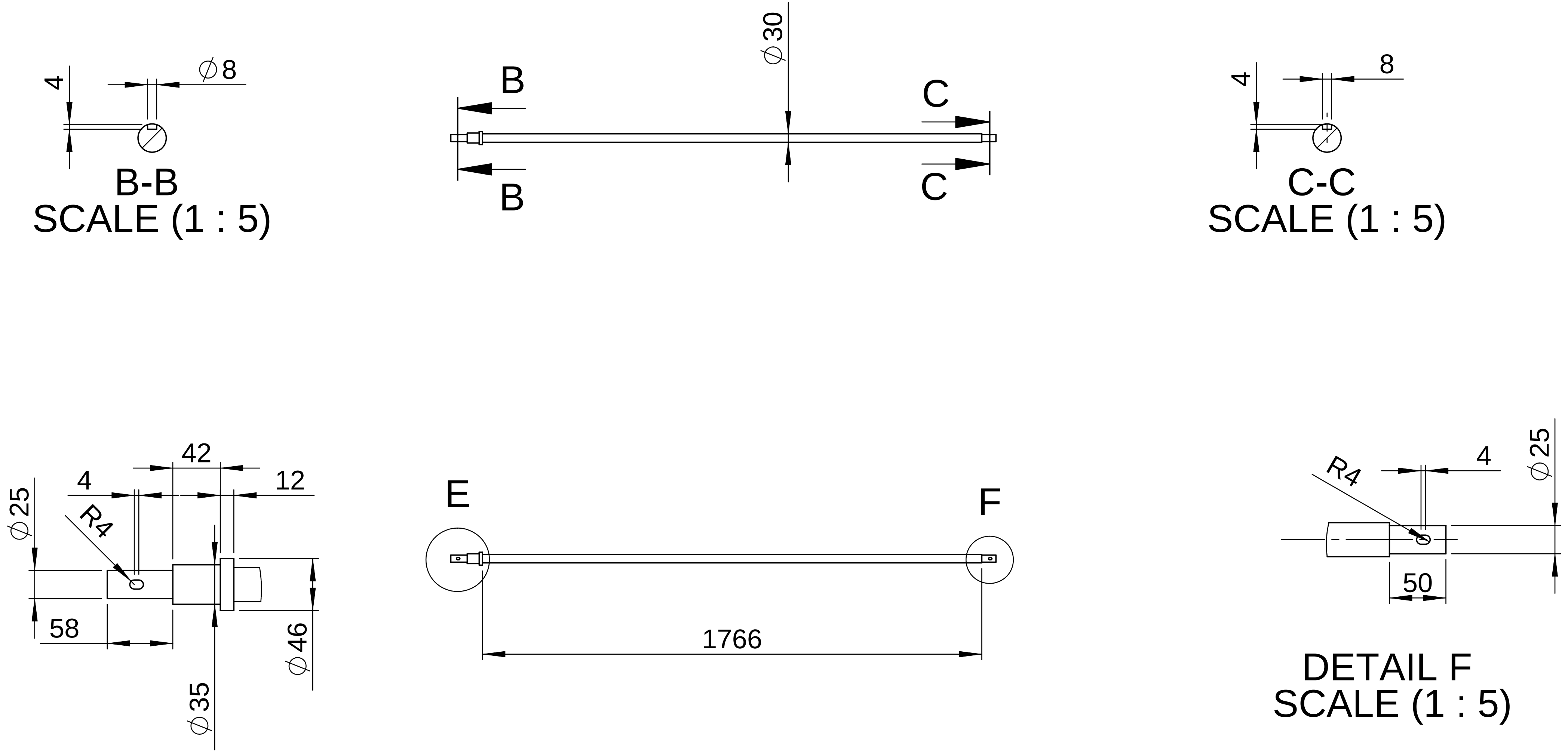
Initial: A.G.

Description:

PUMP SHAFT PLATE

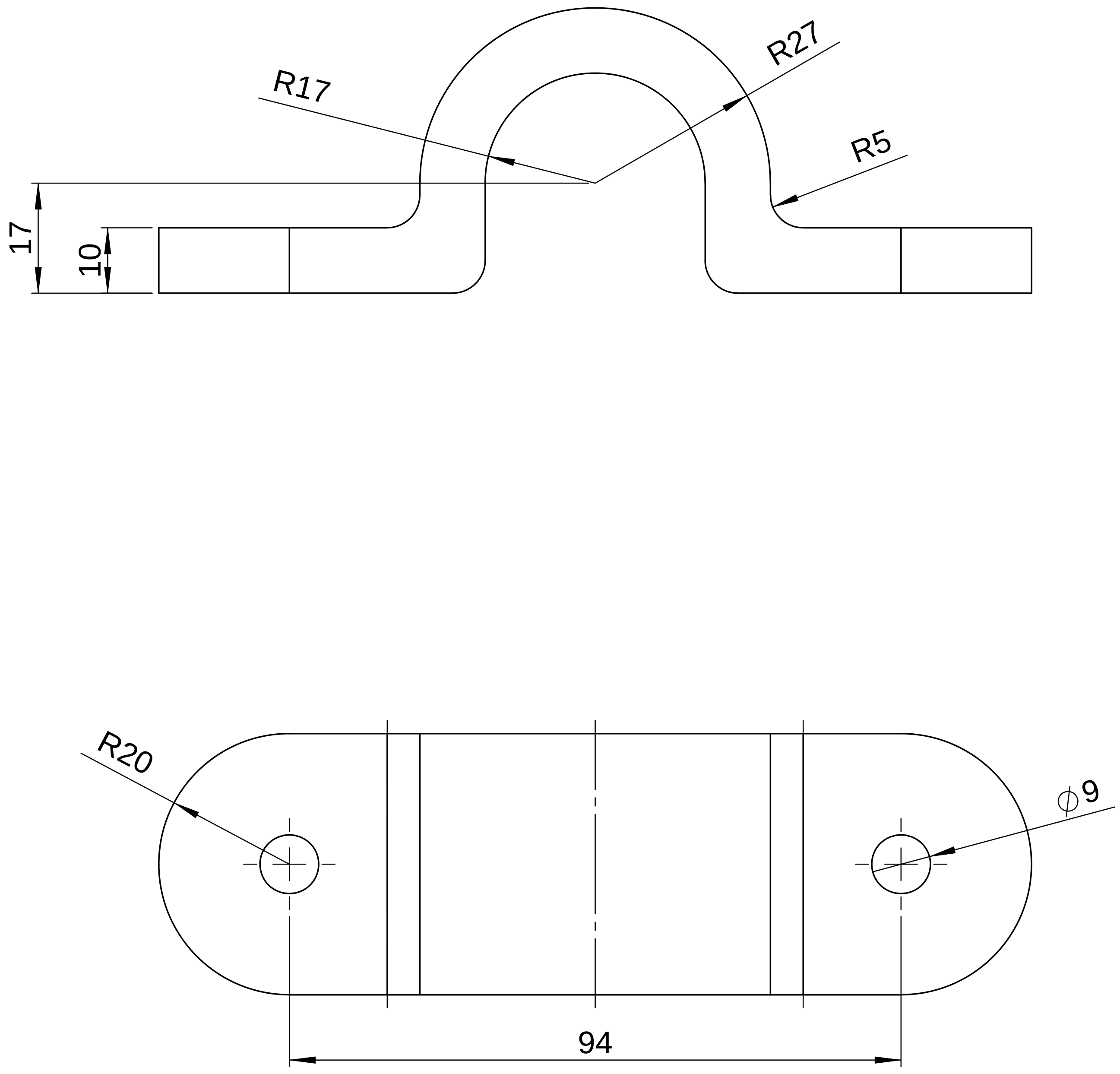
Drawing no.:

5.5



1	Shaft			AISI 4340 Steel
Qty.	**Item**	**Item no.**	**Drawing no.**	**Material / Model no.**

Ingeniørhøjskolen i Århus Department of Mechanical Engineering Engineering College of Aarhus	**Scale:** 1:20	**Group ID:** **Student ID:** IY10227	**Date:** 22/5/2011 **Initial:** A.G.
Description: VERTICAL SHAFT	**Drawing no.:** 6.1		



4	Fixation			1023 Carbon Steel
Qty.	**Item**	**Item no.**	**Drawing no.**	**Material / Model no.**

Ingeniørhøjskolen i Århus Department of Mechanical Engineering Engineering College of Aarhus	**Scale:** 1:1	**Group ID:**	**Date:** 22/5/2011
		Student ID: IY10227	**Initial:** A.G.
Description: GROUND FIXATION	**Drawing no.:** 6.1		