B-TEAM INC.

MECE 360 • GROUP 10
12.05.2014

FINAL REPORT

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December 5, 2014

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Dr. Chen,

B-Team Inc. has completed the the final report for our grappling hook lifting system. The attached report describes the brainstorming process behind the conceptualization of our design, descriptions of various components and subsystems of our device, as well as the full design drawing package. A full analysis of this design is also included, with supporting calculations. Based on this analysis, our device safely fulfills the requirements of the project.

The estimated man hour cost is $23,850, while the prototype cost is $4,4047.50. Although deviations were made from the timeline proposed in the original conceptual report, the final design is being delivered on schedule.

Please review the attached report and provide approval to continue on to the manufacturing stage. It has been a pleasure working with you, and we look forward to seeing this product on the market. Please contact us by email should you have any questions regarding this report.

Sincerely,

Shealynn Carpenter, Nicholas Yee, Panveer Dhaliwal, & Alexandre Sauvé
B-Team Inc.

abstract

B-Team Inc. was tasked with the design and engineering of a lift system device to be used alongside a grappling hook device, with consideration of some target market. The following requirements were outlined by our client, Dr. Chen of Chen Inc. (see Appendix A):

- The device must be able to lift a 90kg person
- The device must provide lift at a vertical speed of 4m/s
- The design should strive to be lightweight and compact

Our device design includes a barbed pulley system that consists of a series of pulleys through which the rope is fed. The pulley system is powered by an electric motor and a planetary gear train, allowing for a lightweight mechanism that has sufficient torque and speed for the pulleys to pull a user up the rope. The rope enters through the top of the device and exits through the bottom, passing through an ascender (which safely ensures the mechanism can only climb upwards) while the machine pulls itself along the rope. By winding the rope around a series of pulleys, the system forces the rope to be in contact with a large portion of the main pulley, which grips the rope. To ensure that the rope does not slip through the system, a no-slip condition is guaranteed with a series of barbs embedded in the driving pulley to actively engage the rope. The encasing body of the device has a lid, which can be lifted to allow the user to easily wind the rope around the pulley system prior to ascent. This allows the device to be used in a variety of situations, with some flexibility over the kind of rope, and the grappling hook device used.

The final estimate for the total cost of the project is $27,897.03, including the estimated cost of the prototype, $4,047.03.

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INTRODUCTION

In September 2014, B-Team Inc. was tasked with developing a grappling hook lift system for use in some target market. The group was asked to brainstorm three potential concepts and select a final design through consultation with our client, Dr. Chen. For the final design, the group was to perform detailed engineering calculations and analysis, develop a drawing package and 3D assembly model, and review the manufacturability of the product. Safety was to be ensured while fulfilling the project requirements of lifting a 90kg person upwards at 4m/s (see Appendix A).

existing designs

There are few grappling hook lifting systems currently available on the market. One product, the ATLAS device, was designed by a former MIT student. This grappling hook is used by the military, firemen and police officers. It uses a winch, where the rope is wrapped around the drum several times. The ATLAS Device is quite bulky and heavy, but has a very high speed of ascent up a rope.

legal requirements

Failure or misuse of a rope lifting system could result in injury or loss of life. There are many legal requirements regarding safety equipment and safe use, but no established standards exist for this specific kind of device due to its scarcity on the market.

DESIGN METHODOLOGY

The discussion towards what kind of design B-Team Inc. would pursue began with determining the scenarios where a grappling hook lifting system would be needed. The team also discussed the available options for power sources, and attempted to match these options to appropriate scenarios.

Since electric motors and batteries are relatively light and provide more torque than gasoline powered systems, the team proceeded with incorporation of these components. From there, the following three concepts became the focus of discussion.

tension pulley

The Tension Pulley system (see Appendix B) consists of a series of pulleys through which the rope is fed. The central pulley is powered by an electric motor which, along with a gear train, would allow for sufficient power and speed for the pulleys to pull in the desired direction. The rope would enter through the top of the device and exit through the bottom as the machine pulls itself upwards. By winding the rope around a series of pulleys, the system creates considerable tension in the rope to allow for sufficient force and friction to create the motion desired. The user must wrap the rope through the pulleys device before the lift system can be used. This allows the device to be used in a variety of situations, with flexibility over the kind of rope used.

high-friction winch

Similar to the Tension Pulley system, the High-Friction Winch (see Appendix B) uses an electric motor to create tension in the rope, which then produces the desired upward motion. Rather than a series of pulleys, this system uses a rope that is wrapped around a grooved winch at an angle of 45 degrees, incurring enough friction to prevent slipping. The device is detachable from the rope, which enters and exits the device as it is pulled through.

fishing reel

The Fishing Reel system (see Appendix B) is different from the other two as it permanently attaches the rope to a spool, which a operates in the same way as a fishing reel. This can be disengaged, allowing the rope to unspool, similar to when the line is cast. The motor engages the spool, winding the rope and causing the device to ascend. A large spool is needed in order to be able to store approximately 25m of rope, and a sturdy winding arm would be needed to support the torque caused by the winding motion. This design is suitable for situations where the rope cannot be left hanging following ascent.

DESIGN METHODOLOGY

discussion

The Tension Pulley and High-Friction Winch are very similar as they both rely on friction and redirection of tension forces. They are both detachable devices because they allow the rope to enter and exit the system, thus bypassing the need for storage of the rope. While the Tension Pulley system allows for a lower weight and size, reducing cost, the High-Friction Winch is able to generate higher friction forces. This is achieved because the rope wraps around the winch several times.

Compared to the previous two designs, the Fishing Reel is safer because the rope is permanently attached to the spool. However, this system does not allow for interchangeable ropes because the rope is fixed to the device. Since the entire length of rope is stored within the Fishing Reel, it will be the largest of the three possible designs.

The Tension Pulley system was determined to be the best design concept through the use of a decision matrix (see Appendix B).

DESIGN DESCRIPTION

solution

The lifting device designed by B-Team Inc. is an ideal solution to the design problem (see Appendix A) presented by Chen Incorporated. The device enables the user to ascend at an estimated 4m/s up a tethered rope. The final design is a refined version of the Tension Pulley design concept. Through the addition of barbs, slipping of the rope is eliminated. The device uses a series of pulleys to maximize tension of the rope. A planetary gear train is used to increase input torque of the motor which is powered by batteries. Safety of the user is ensured by an ascender, which acts as a locking mechanism to prevent downward motion.

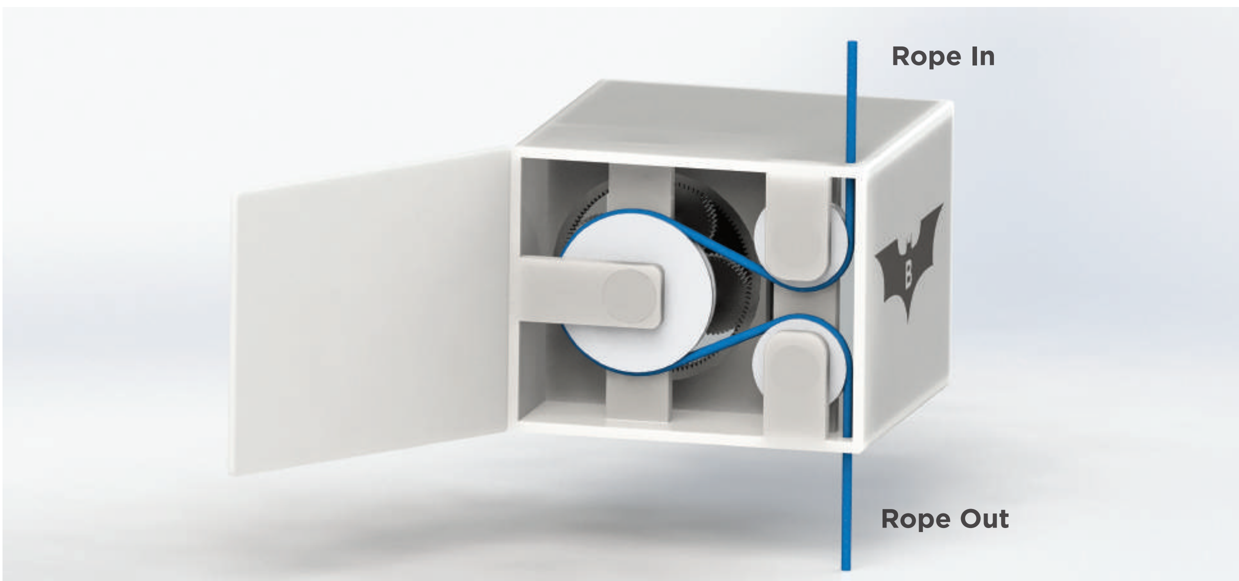


FIGURE 1 Full Device (Open Lid)*

* Third-party components (Ascender, Battery, Carabiner, and Chain) not shown

DESIGN DESCRIPTION

pulley system

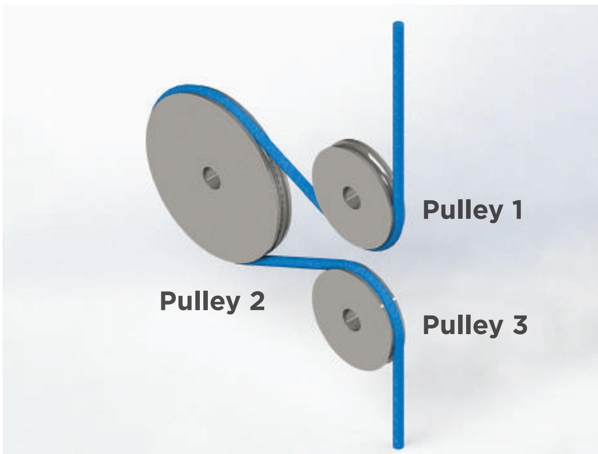


FIGURE 2 Pulley System

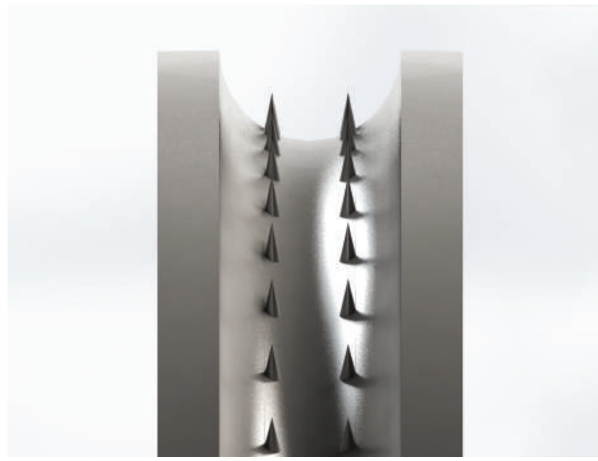


FIGURE 3 Barbed Pulley

A system of three fixed pulleys (one driven, two idlers) is used to transfer the motor speed and torque to the rope and generate upward motion. The pulleys are made of stainless steel, an industry standard.

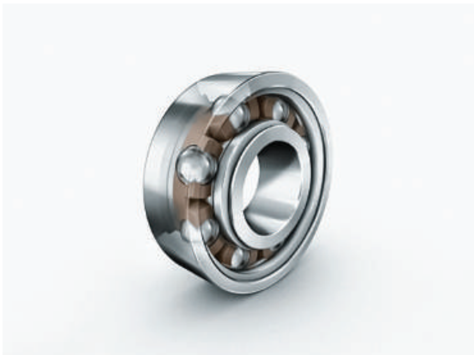
The rope enters through the top of the device and is wrapped around Pulley 1[4], which is an unpowered idler pulley (see Figure 2). The rope then wraps around a larger pulley (Pulley 2 [3]), which is driven by the gear train. To eliminate unwanted slipping between Pulley 2 and the rope, Pulley 2 is outfitted with stainless steel barbs along the contact surface (Figure 3). These barbs are designed to impale the rope ensuring no slipping between the rope and Pulley 2. A smaller barbed pulley (Pulley 3 [4]) disengages the rope from Pulley 2 and directs the rope towards the exit of the mechanism. Pulley 3 is indirectly rotated by a chain and sprocket mechanism, connecting it to Pulley 2. This mechanism provides a speed ratio which ensures that the rope velocity is consistent around both pulleys.

There are 111° of contact between the rope and Pulleys 1 and 3, and 222° of contact between the rope and Pulley 2 (Appendix D). The large contact angle of Pulley 2 allows the lifting force to be distributed along a greater area and therefore a larger number of barbs, increasing safety.

The pulleys rotate on stainless steel shafts, to which they are connected by splines. Splines are used for their strength and resistance to fatigue.

DESIGN DESCRIPTION

bearings



Since all shafts are rotating at the same angular speed as the pulleys that they are supporting, 7004-B-TVP Angular Contact Ball Bearings are used (Figure 4). These bearings provide the radial support necessary to support the shafts, while maintining low friction. They do not require lubrication, making them the ideal bearings for our design.

FIGURE 4 Angular Contact Ball Bearing [8]

gear system

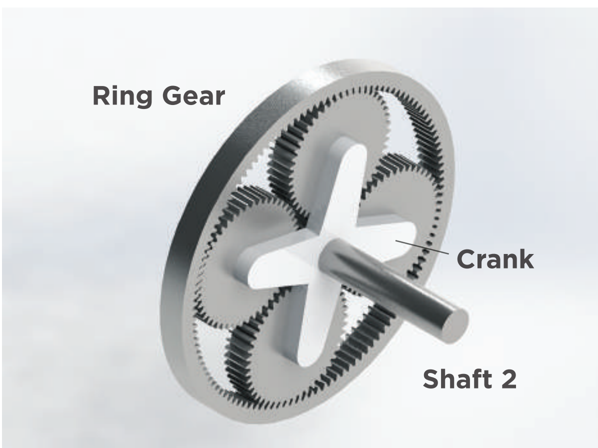


FIGURE 5.1 Gear Train, Front View

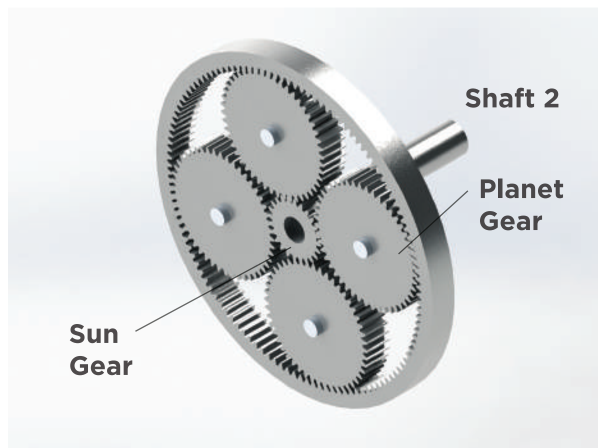


FIGURE 5.2 Gear Train, Back View

A planetary gear train is used to transmit the necessary torque to the pulley system at the necessary speed. Space is conserved since planetary gear trains are more compact than typical ordinary gear trains. The ring gear is fixed to the device casing, and power output is transmitted through a crank connecting the planet gears to Shaft 2.

The gears rotate on stainless steel shafts, to which they are connected by splines, which again are used for their strength and resistance to fatigue.

DESIGN DESCRIPTION

motor

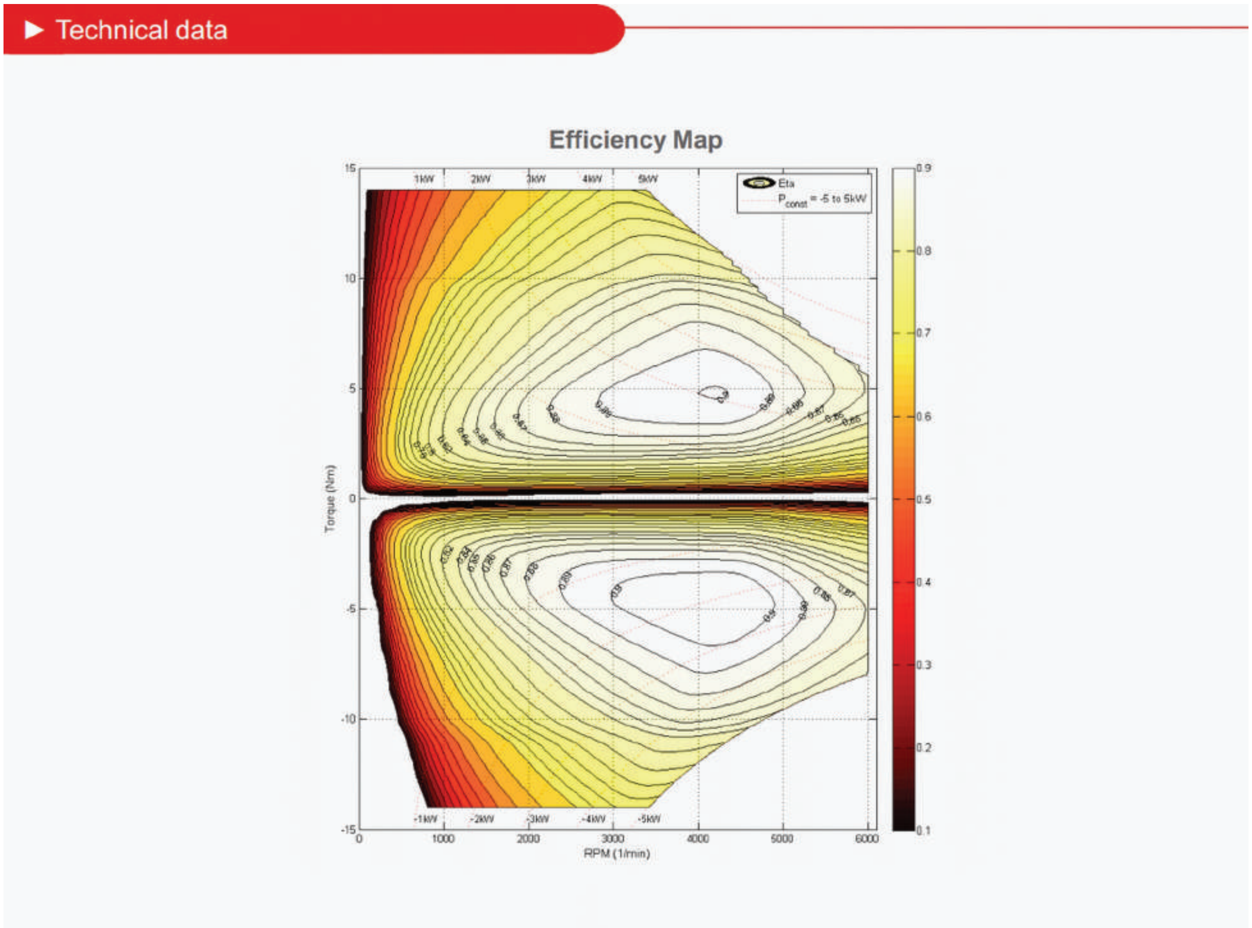


FIGURE 6 Motor Torque-Speed Curve [9]

The Compact Power BLDC CPM90 motor by SONCEBOZ is used to power the device. The CPM90 is a brushless DC motor which can provide torque up to 15 Nm and speeds up to 6000rpm (see Figure 6), with a 5000W peak power [9]. The motor exhibits a maximum efficiency of over 90% for all low voltage applications and operates with voltages between 36-56V [9]. Its compact design and ability to surpass the required specifications make it an ideal choice for the device.

The motor is connected to Shaft 4 by a keyway to protect itself in the case of sudden or irregular applied stress.

DESIGN DESCRIPTION

battery

Three Antigravity 16-Volt VTX12-20 batteries power the motor. The VTX12-20 is an ultra-lightweight, high power lithium nanophosphate battery [1]. Each battery is compact, weighing only 2 kg with dimensions of 150 x 87 x 130mm.

casing

FIGURE 7 Casing (Open Lid)

All the components of the device are housed within a high density polyethylene (HDPE) casing that supports each component and maintains its functional position. The casing also protects the components from any external damage. A lid attached to the casing is removable, allowing the user to thread the rope through the pulley system.

DESIGN DESCRIPTION

ascender + carabiner

After exiting the device, the rope moves through an ascender fixed onto the bottom of the casing. The ascender employs a cam that allows the rope to slide freely in the desired direction, but clamps on the rope firmly when moved in the opposite direction. This adds a measure of safety to the device, preventing the system from falling in case of failure and allows the user to stop in the middle of the lift.

A carabiner is attached to the ascender, and is the point of attachment between the user and the device.

FIGURE 8 Ascender + Carabiner [6]

rope

Kernmantle rope consists of core fibers protected by a woven exterior sheath. The core (kern) fibers provide tensile strength, while the sheath (mantle) prevents the rope from being affected by abrasion. Nylon Kernmantle rope is used in the device for its safety and versatility.

DESIGN ANALYSIS

mass

A conservatively large approximation of 14kg was used for the mass of the device, resulting in a system (user and device) mass of 104kg. This value was used to safely design the device before the actual mass was known. The resultant weight of the system, based on this assumption, is 1.022 kN (Appendix D).

pulley system

Assuming a constant upward velocity of the system (no acceleration), the tension in the rope around each pulley was calculated. This was then used to calculate the reactionary forces of the supporting shafts of each pulley (Appendix D).

The normal force on Pulley 1 and its shaft (Shaft 1) is 1.685kN.
The normal force on Pulley 2 and its shaft (Shaft 2) is 1.022kN.
The normal force on Pulley 3 and its shaft (Shaft 3) is 0.

The sprocket and chain mechanism connecting Pulleys 2 and 3 was ignored in these calculations. An assumption was made that the effect of the chain on the shaft is negligible because Pulley 3 acts as an idler and does not contribute to the upward motion of the device. Rather, its sole purpose is to disengage the rope from Pulley 2 and direct the rope through the exit of the device.

Barbs

Pulleys 2 and 3 are outfitted with a series of barbs on the surface that ensure no slippage occurs, allowing full transmission of the torque to the rope. Based on the required strength, durability, cost, ductility, and manufacturability, the ideal material for the barbs is stainless steel (see Appendix C).

A detailed stress analysis was performed on the barbs to ensure safe operation. The barbs must transfer force to the rope while withstanding a large shear stress at their base. Based on the angle of rope contact around Pulley 2, the number of Pulley 2 barbs in the rope at any given time is approximately 88 (see Appendix D). The force in each barb is 11.5N, assuming even distribution of tension among the barbs (Appendix D). A large minimum safety factor of 4 was used, since failure of the barbs will lead to failure of the device. The calculated safety factors of shear stress and fatigue are 8.5 and 10.1, respectively (Appendix D). Both values are larger than the required safety factor, justifying the choice of stainless steel barbs.

DESIGN ANALYSIS

gear system

For Pulley 2 to move the system mass upward at 4m/s, the required torque is 76.67Nm, with an angular velocity of 53.33rad/s. With the motor rotating at 320rad/s this angular velocity can be achieved through a gear ratio of 6. This results in 90Nm of torque, which satisfies the torque requirement.

Using the tabular method, a ratio of 5 was obtained for the ring-to-sun gear teeth relationship. Assuming 24 teeth on the sun gear, the planet and ring gears will have 48 and 120 teeth, respectively.

Based on these values, a maximum of 4 planet gears can be used, among which the torque loading will be shared (Appendix D). With a module of 1.5 and a face width of 16mm, assuming a lifetime of one million cycles, a minimum safety factor of 4 was satisfied by each gear:

Sun Gear

Surface stress safety factor of 4.2
Bending stress safety factor of 14.1

Ring Gear

Surface stress safety factor of 6.5
Bending stress safety factor of 4.3

Planet Gears

Surface stress safety factor of 4.2
Bending stress safety factor of 11.3

This illustrates that none of the gears will fail within one million cycles of stress.

shafts

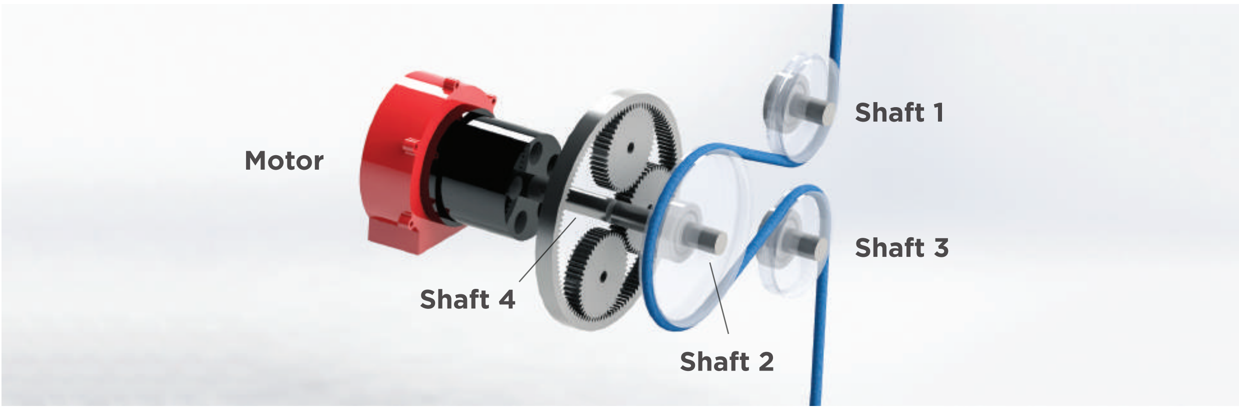


FIGURE 9 Shafts

All shafts are made of stainless steel with an ultimate tensile strength of 505 MPa and a yield strength of 215 MPa [2]. To minimize the mass of the device, the minimum diameter of each shaft was calculated based on the strength of stainless steel (Appendix D). Based on an analysis of Shaft 1, which experiences the largest applied load and therefore greatest deflection, shaft deflection was determined to be relatively small and not limiting (Appendix D). Analysis of Shafts 2 and 4, which undergo applied torque, indicated that torsional deflection was relatively small and not limiting (Appendix D). A factor of safety of 2 was assumed for each shaft. For all shaft calculations, both the mass of the pulleys and shafts were considered negligible due to their low relative magnitude.

Shaft 1

Shaft 1 supports and rotates with Pulley 1, and has the largest normal forces acting on it. It is supported at both ends by ball bearings. The minimum diameter of this shaft is 13mm. The length of this shaft is 70mm.

Shaft 2

The rotational energy of the gear system is transferred to the pulley system through Shaft 2, which is supported at both ends by ball bearings. The minimum diameter of this shaft is 17mm. The length of this shaft is 100mm.

Shaft 3

Shaft 3 supports and rotates with Pulley 3 and is supported at both ends by ball bearings. Shaft 3 only transmits torque: there are no applied forces because Pulley 3 does not experience a normal force. The minimum diameter of Shaft 3 was conservatively assumed to be equal to that of Shaft 1 (13mm). The length of this shaft is 70mm.

Shaft 4

The rotational power from the motor is transmitted to the planetary gear train through Shaft 4, which is attached to the gear train's sun gear. There are no applied forces acting on Shaft 4. The minimum diameter of this shaft is 9.32mm.

DESIGN ANALYSIS

bearings

Angular contact ball bearings are consistently used throughout the device for manufacturing simplicity. Shaft 2 is used to select the bearings as it withstands the greatest load. The lightest angular contact ball bearing manufactured by Schaeffler Germany is the 7004-B-TVP bearing (0.061kg) [7]. Conservatively assuming a bearing use of 20 million revolutions, the dynamic loading factor is 4.24kN (Appendix D). This is considerably smaller than the maximum dynamic loading factor of 13.4kN, ensuring the bearings will not fail [7].

casing

High density polyethylene was selected as the material used for the casing, based on criteria of strength, durability, weight, cost, and ductility (see Appendix C).

motor

The maximum power output of electric motors occurs at half of the maximum angular speed [5]. According to the manufacturer's specifications, the motor will provide maximum power at approximately 3000rpm (~314rad/s) . An angular speed of 320rad/s was approximated to meet the speed requirements at a whole-number gear ratio, and the corresponding motor torque was used in the gear system calculations to conduct a safe analysis.

DESIGN ANALYSIS

rope

Nylon was selected as the material used for the rope, based on the criteria of strength, stretch, cost, and resistance to abrasion, water, UV, and rotting (see Appendix C).

A rope of 9mm diameter is used in the design. As the smallest available size of Kernmantle rope, it is the lightest of the standard sizes and therefore ideal. A minimum factor of safety of the rope was assumed to be 4 due to the potential loss of life if the rope fails. A stress concentration on the rope is produced by the barbs embedded within the rope. Due to the non-standard shape of the barbs, SolidWorks was used to determine the stress concentration factor:

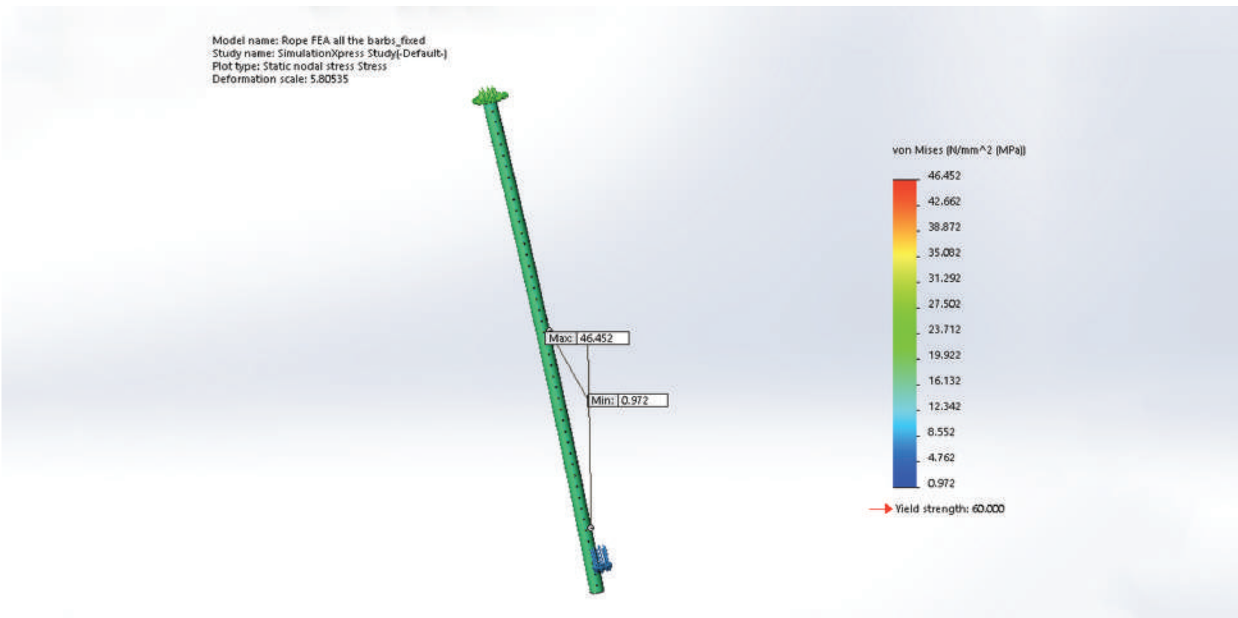


FIGURE 10 Barb Stress Concentration Factor (SolidWorks)

The factor of safety based on the ultimate stress of the rope and the stress applied due to the system weight, with the included stress concentration, is 7.5 (Appendix D). This is larger than the minimum factor of safety, indicating rope selected is safe for use.

COST ESTIMATION

Certain components in the device are existing industry standard parts with known costs, including the rope, gears, motor, batteries, bearings, ascender, and carabiner. Other components in the device, such as the pulleys and chain, are customized existing components. The device also includes components manufactured from raw materials, including the shafts, crank, and casing. The total cost estimation is broken down in Table 1:

Q = Quantity L = Labour M = Materials E = Equipment T = Total

TABLE 1 Cost Estimation Break-Down

Item	Unit	Q	Rate ($/unit)				Cost ($)			
			L*	M	E	T	L*	M	E	T
Pulley 2	Item	1	60	38.53		98.53	60	38.53		98.53
Pulley 1/3	Item	2	60	11.10		71.10	120	22.20		142.20
Motor	Item	1			1170.50	1170.50			1170.50	1170.50
Battery	Item	3			400	400			1200	1200
Shaft Stainless Steel	Kg	0.35633		2.8586		2.8586		1.0186		1.02
Shaft 1/3	Item	2	15			15	30			30
Shaft 2	Item	1	15			15	15			15
Shaft 4	Item	1	15			15	15			15
Rope	Item	1			96.93	96.93			96.83	96.93
Ascender	Item	1			27.95	27.95			27.95	27.95
Bearing	Item	6			1.08	1.08			6.48	6.48
Sun Gear	Item	1			59.57	59.57			59.57	59.57
Planetary Gear	Item	4			107.50	107.50			790	790
Ring Gear	Item	1	20		151.45	171.45	20		151.45	171.45
Crank Stainless Steel	Kg	0.44282		2.8586		2.8586		1.2659		1.27
Crank	Item	1	20			20	20			20
Carabiner	Item	1			49.50	49.50			49.50	49.50
Casing Titanium	Kg	1.6988		6.85		6.85		11.637		11.64
Casing	Item	1	30			30	30			30
Chain	Item	1	15	34.99		49.99	15	34.99		49.99
Assembly	Item	1	60			60	60			60
Total										**4047.03**

* Shop cost is estimated as $60/hr

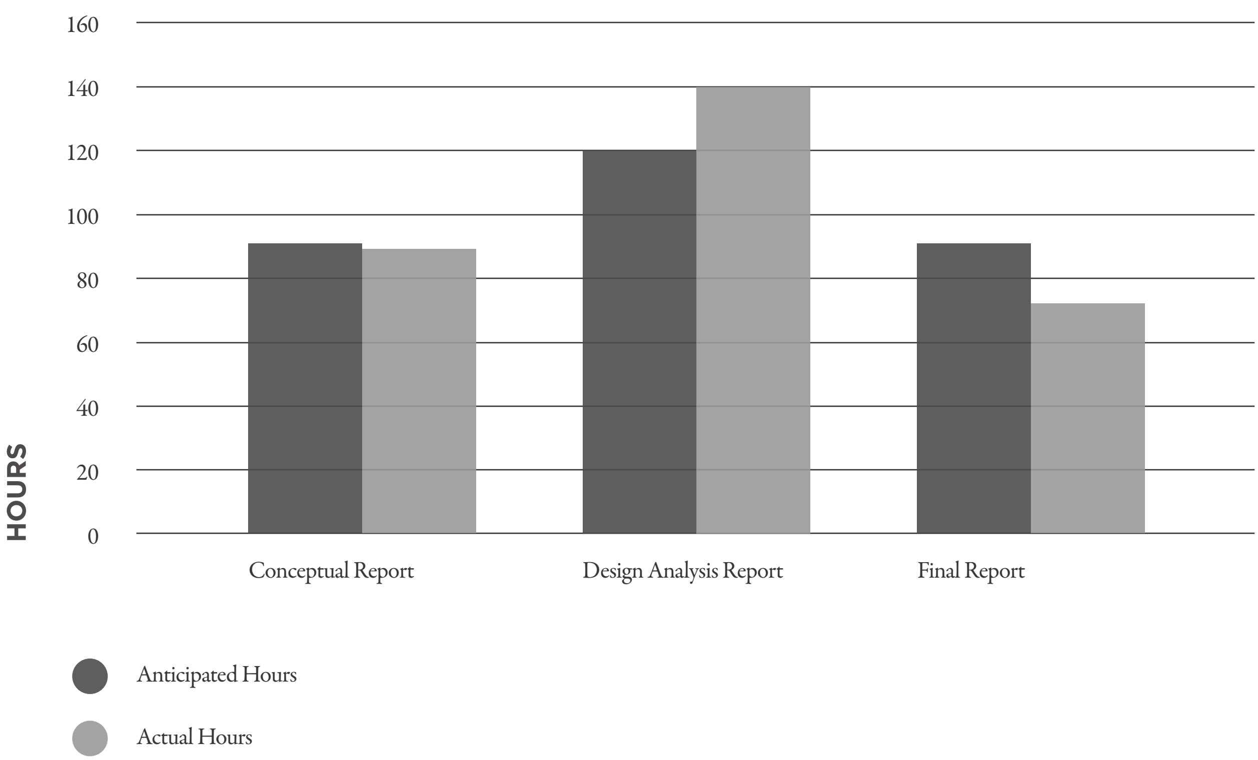
The estimated total cost of the device is $4047.03.

TIME MANAGEMENT

Time spent on the project was used to design the components and device assembly, and present the design and findings through a series of reports. Much more time was spent on the design and analysis than was originally allocated; largely to ensure thoroughness and precision. An emphasis was placed on material selection, with care and consideration taken to choose appropriate and feasible materials for all components. As well, a decision was made to distribute the analysis of the design among all group members rather than isolating these tasks to a single member. This inevitably led to inefficiency when it became necessary to learn new software (MathCad) and gear analysis concepts. However, this also led to a quicker and more cohesive report writing process as it built a broader understanding of the project within each team member. The extra time taken in the design analysis stage was accommodated by an extended deadline for the design analysis report.

See Appendix F for both the anticipated and actual time management Gantt charts. The team's time allocation between the different stages of the project is shown in Figure 11:

FIGURE 11 Project Time Allocation



CONCLUSION

B-Team Inc. has developed a rope lifting device that safely fulfills the requirements outlined by Chen Inc. The device is able to provide lift at a vertical speed of 4m/s for a 90kg user, and is fit within a compact and lightweight casing. Based on a full design and analysis of this design, the team is proposing that the project be moved forward to manufacturing stage.

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APPENDIX A
project description

MecE 360
Engineering Design II

Request for bids package

Fall 2014

Professor
Zengtao Chen

MecE 360: Engineering Design II

Design Bids Fall 2014

Chen Inc, a production firm, was recently awarded contracts to produce new equipment for various applications. The following contracts were awarded:

1. Design of a single speed, 120 volt, meat grinder/pasta maker.
2. Design a proof of concept wind powered oil derrick for the **personal use** in Alberta.
3. Battles Manufacturing Inc, known on the stock market as Batman, requires a grappling hook lifting system capable of lifting a 90Kg person. The device must be small, light and the person must be lifted at an average speed of 4m/s.

To meet the contract requirements, Chen Inc must tender out bids to design the various appliances as they are not a Mechanical Engineering firm. As such, competing bids are requested. **An equal number of bids per project will be accepted (±1).** Bids will be accepted on a first come first served basis. EC must provide a second and third preferred choice of bids in case their first choice is not accepted. EC will be notified which bid they are assigned**. Detailed project specifications will be provided to EC during their first meeting with Chen inc in the form a request for bid document.**

Bidding letter must be submitted **by uploading to course website** to Chen Inc by noon **of September 11 2014** and contain the following:

- Contain EC company name
- Organizational structure (manager, etc). Manager takes responsibility for human resources and project management – this can be a rotating positions
- it must be explicitly written and acknowledged that group members accept the rules and regulations of EC companies as stipulated in this bidding documentation
- First choice of bid
- Second and third choice of bid
- List **10 possible meeting times in order of preference** that the entire group is available as per Chen Inc's schedule below. **Provide the timeslot number**
- Group member contact information (email of one contact speaking for the group)
- Signatures of all members of EC groups accepting the projects/bids

FIRST COME FIRST SERVE FOR BIDS AND TIMESLOTS

Affiliates: It is imperative that EC use components from Chen Inc affiliates QTC gears (www.qtcgears.com), SDP/SI (http://www.sdp-si.com/) and FAG bearings (http://www.fag.com/content.fag.de/en/index.jsp), if required, as cost will be minimized.

EC must spec or investigate their power sources for their bid applications.

Throughout the bidding process it is mandatory that each engineering contractor discuss project requirements and progress with Carey Inc on a weekly basis. Chen Inc will set aside 15 minutes per 5 groups per week for meetings. **Block off weekly meetings** for the term based on following schedule; first come first serve. If a group intends to miss a meeting they **must provide Chen Inc 24hrs notice**. Chen Inc may have administrative meetings on a monthly basis and may need to reschedule meetings; ECs will be advised by email if that is the case.

EC can always email Chen Inc for questions, clarifications or schedule additional meetings. Emails will be answered as soon as possible. Important notices will be posted on the website ASAP.

	Monday	Wednesday	Friday
10:00am-10:50am	Lecture	Lecture	Lecture
2:00pm-2:15pm	Group 1-5	Group 11-15	Group 21-25
2:15pm-2:30pm	Group 5-10	Group 16-20	Group 26-30

MecE 360: Engineering Design II

Bid requirements:

- All reports must be **<u>uploaded electronically as a single pdf document</u> <u>on eclass</u>**
- All calculations submitted using MathCad files converted to pdf or scanned **clean hand** calculations converted to pdf.
- File names: file submission must have the following name structure
  - **Concept report, group 8: mece360_CR_08.pdf**
  - **Analysis report, group 15: mece360_AR_15.pdf**
  - **Final bid, group 1: mece360_FR_01.pdf**
- Submissions are done via **eclass website. Emailed reports are not acceptable**

Deadlines (<u>submitted before 5PM</u>)

- September 30: report 1
- November 7: report 2
- December 10: Final bid

Critical elements for design and analysis (IMPORTANT NOTE you must only do what is critical, but figuring that out is the key exercise!!!)

- Power selection/description
- Power transmission elements
- Other structural elements
- Bearings
- Shafts
- Connections
- General casing (no calculations but must be part of drawings)
- Lubrication
- Heat transfer considerations must be discussed not calculated.
- Fits
- Full set of correct engineering drawings of designed components, full assembly
- EC should base their submission on existing applications and adhere to any and all safety standards.

Group rules, regulations and expectations

If a team member refuses to cooperate on an assignment, his/her name should not be included on the completed work. If the non-cooperation continues, the team should meet with the instructor so that the problem can be resolved, if possible. If no resolution is achieved, the cooperating team members may notify the uncooperative member in writing that he/she is in danger of being fired, sending a copy of the memo to the instructor. If there is no subsequent improvement, they should notify the individual in writing (copy to the instructor) that he/she is no longer with the team. **Students who do not participate actively in group work and are fired will receive a zero for all group work and are at risk of failing the course**.

As you will find out, group work isn't always easy-team members sometimes cannot prepare for or attend group sessions because of other responsibilities, and conflicts often result from differing skill levels and work ethics. When teams work and communicate well, however, the benefits more than compensate for the difficulties. One way to improve the chances that a team will work well is to agree beforehand on what everyone on the team expects from everyone else.

Request for Bids Proposal 1

Project Title: *meat grinder/pasta maker*

Objective

Design a meat grinder/pasta maker. The clear need for humans to start making their food from scratch is critical as exemplified by sanitation problems at XL Meats and other distributors.

Scope of Work

- Define design specifications through conversation with customer.
- Prepare three design concepts by creative brainstorming.
- Select design concept through consultation with customer.
- Prepare detailed design calculations and analysis.
- Prepare assembly and parts drawings suitable for manufacture.
- Review detailed design drawings for manufacturability.

Preliminary Design Specifications

- Single speed
- 120 volt
- Easy to clean
- Adaptable to meat and/or pasta
- Elegant

Deliverables

- Dimensioned and toleranced assembly and parts manufacturing drawings.
- Engineering report

Project Sponsor

Name: Chen Inc
all communications through Carey Inc president/CEO Jason Carey
Company: University of Alberta
Office: 4-31G MecE building
Email: zengtao.chen@ualberta.ca

Meeting Time

Engineering companies and client must meet once a week for 15 minutes at a predetermined meeting time.

Intellectual Property Ownership

Design IP will remain the property of the Company

Request for Bids Proposal 2

Project Title: *Wind powered oil derrick*

Objective

Design a proof of concept wind powered oil derrick for the personal use in Alberta. With an impending energy crisis the government of Alberta will want to maximize on two of its resources, oil and wind.

Scope of Work

- Define design specifications through conversation with customer.
- Prepare three design concepts by creative brainstorming.
- Select design concept through consultation with customer.
- Prepare detailed design calculations and analysis.
- Prepare assembly and parts drawings suitable for manufacture.
- Review detailed design drawings for manufacturability.

Preliminary Design Specifications

- Capable of converting average wind forces found in the Alberta plains into a pumping action
- Personal sized Derrick

Deliverables

- Dimensioned and toleranced assembly and parts manufacturing drawings.
- Engineering report

Project Sponsor

Name: Chen Inc
all communications through Chen Inc president/CEO Zengtao
Chen Company: University of Alberta
Office: 4-31G MecE building
Email: zengtao.chen@ualberta.ca

Meeting Time

Engineering companies and client must meet once a week for 15 minutes at a predetermined meeting time.

Intellectual Property Ownership

Design IP will remain the property of the Company

Request for Bids Proposal 3

Project Title: *Grappling hook lifting system*

Objective

Battles Manufacturing Inc, known on the stock market as Batman, requires a grappling hook lifting system capable of lifting a person.

Scope of Work

- Define design specifications through conversation with customer.
- Prepare three design concepts by creative brainstorming.
- Select design concept through consultation with customer.
- Prepare detailed design calculations and analysis.
- Prepare assembly and parts drawings suitable for manufacture.
- Review detailed design drawings for manufacturability.

Preliminary Design Specifications

- Lift a 90Kg person
- small, light
- Lift at an average speed of 4m/s.

Project Sponsor

Name: Chen Inc
All communications through Chen Inc president/CEO Zengtao Chen
Company: University of Alberta
Office: 4-31G MecE building
Email: zengtao.chen@ualberta.ca

Meeting Time

Engineering companies and client must meet once a week for 15 minutes at a predetermined meeting time.

Intellectual Property Ownership

Design IP will remain the property of the Company

MecE 360: Mechanical Design II

What is expected in a project and report?[1]

Conceptual Report: No more than 1500 words of text for the main body

Cover letter (applies to all reports)
- Standard letter of transmittal format addressed to the project client
- Description of the material being submitted
- Estimated time and cost to complete the project
- Professional, concise, informative . . . should reflect professionalism

Title page (applies to all reports)
- bid number,
- your company name/logo,
- presented to,
- forget the MecE 360 context, think real world
- date

Abstract (applies to all reports)
- Short no more than 100 words
- Main goals/findings of the report
- Exciting

Table of content- include word count for main body at the bottom of TOC
List of figures
List of tables
Main body:

1. Introduction
   - Discuss what is expected of you
   - Discuss current existing designs
   - Discuss business side of the work
   - Legal/standard requirements
   - Before writing something down ask: would a CEO of a company care (i.e. this is NOT a lab report – applies to entire report)
2. Design methodology
   - Results of brainstorming, decision matrices
   - Specifications
   - Assumptions
   - Design selection process
     - Pick three top designs
     - Proof of concept calculations (i.e. will either of the three designs work conceptually!)
     - Pick the top design and justify your choice
3. Time management – Consider liquid planner or MS Project. It can generate great reports. Time

[1] Note, the items listed are NOT necessarily in the logical order

estimate per group member for this report 15-25hr, average must be within 10% of 18hrs. **Manage your time!!** (Same for all projects)

- Practice collaboration & project management
- Use a sufficiently detailed activity scheme for schedule
- Describe a baseline schedule for tracking subsequent time expenditures
- Keep detailed design/engineering hours logged against the project
- Updated task descriptions and time spent
- Engineering/design costs throughout the project
    - Junior Engineer: $90 per hour
    - Intermediate Engineer: $150 per hour
- Perform a “gut” check. Is it reasonable? Achievable?
- I am looking for engagement with the entire project management process and experience, not the “right” answers. Add your comments to the schedule. I expect the project schedules to change

4. Conclusion

References

Appendix

- Gantt Chart
- Proof of concept calculations
- Any additional info

MecE 360: Mechanical Design II

Analysis Report: No more than 1500 words of text for the main body
Cover letter
Abstract
Table of content - include word count for main body at the bottom of TOC
List of figures
List of tables
Main body:

1. Introduction
2. Full critical analysis of selected design
    - Critical elements to analyse
    - Results of analysis
3. Time management: Time estimate per group member for this report 30-40hr, average must be within 10% of 35hrs. **Manage your time!!**
4. Conclusion

References
Appendix

- All analysis used up to here (MathCAD if you can).
- Gantt chart
- Any additional info

MecE 360: Mechanical Design II

Bid Proposal report:

- No more than 3500 words of text for the main body
- This report is basically putting completed (if they weren't to start with) reports 1 and 2 together and making sure it all flows correctly and add the engineering drawings.

Cover letter
Abstract
Table of content- include word count for main body at the bottom of TOC
List of figures and tables
Main body:

1. Introduction
   - Discuss what is expected of you
   - Discuss current existing designs
   - Legal/standard requirements
2. Design methodology
   - brainstorming, decision matrices, Gantt chart
   - Specifications
   - Assumptions
   - Design selection process
3. Full critical analysis of selected design
   - Critical elements to analyse
   - Results of analysis
4. Discuss design and compare to specifications
5. Time management: Time estimate per group member for this report 15-20hr, average must be within 10% of 18hrs. **Manage your time!!**
6. Estimated costs
   - costs for your time as per time management
   - gear reducer parts using information you can find online and discussions with industrial partner
7. Conclusion

References
Appendix

- All analysis (MathCAD if you can), with FBD etc. See example of analysis format
- Engineering drawings
- Gantt Chart
- Any additional info

Introduction

An engineering report is a direct reflection of its writer's ability and knowledge of the subject. A well written report cannot cover up a poorly executed effort; however, a poorly written report can seriously damage an excellent piece of engineering work. In a bid process, the writer must remember that one aim is to impress the reader and some level of proper salesmanship is required.

All engineers must master the art of effective communication, of which report writing is an important part. Fortunately it is a skill that can be learned and practice is the best learning tool. The personal effort you make at improving your report writing is well worth it and the writing skills you master will greatly help you in your professional career. The points that follow provide some guidelines for effective technical report writing.

1. Always have in mind a specific reader, real or imagined, when you are writing a report. Assume that the reader is intelligent and write considering the level of knowledge they should have.

2. Although it seems like extra work, an outline will always save time by helping to organize your thoughts. The result will be a report that not only effectively presents the material but also takes less time to write.

3. Before starting to write, decide the exact purpose of your report, and make sure that every word, every sentence and every paragraph makes a contribution to this purpose and makes it at the appropriate time. The report should reflect a sound understanding of the presented material and should be as objective as possible. Subjective materials such as personal opinions, suggestions or complaints are best stated in the covering letter or letter of transmittal.

4. Use language that is simple, accurate and familiar. Highly technical terms or jargon make it very difficult for the reader to understand what you are trying to say. If highly technical terms cannot be avoided then they must be defined in simple terms so that the reader can understand them. Keep sentences short, especially when dealing with complex material. Point form can be effective, but should be used sparingly.

5. At the beginning of every section of your report check your writing according to the following principle: "First tell the reader what you are going to say, then actually tell the reader what you want to say, and finally tell the reader what you have just said." A report should not be a mystery story. Don't keep the reader in suspense by leaving essential details until the end. Phrases such as “it will be shown later that ... " and" this manufacturing error will be shown to be the key to the poor performance..." are useful in tying the various parts of the report together.

6. State all the important points explicitly; don't leave it to the reader to try and guess what it is you are trying to say. It may be obvious to you what the results or conclusions of your analysis or experiment are but it is probably not so obvious to the reader.

7. Use the third person, passive voice. The personal pronouns, (I, me, you, us, and we), should not appear. Passive voice is used because reports usually deal with something done in the past. For example, use "The voltmeter was calibrated." instead of “We calibrated the voltmeter.". The choice of using past or present tense is often a matter of personal choice. Rules that work well are:

a. If the event relates specifically to something which has been completed, use the past tense. For example, "The meter reading fluctuated between ... ".

b. If you are stating something that should be true today as well during your experiment, then use the present tense. For example, "Jets spread rapidly as they mix with their surroundings" .

8. Never force the reader to search back and forth through the report in order to find information. It never hurts to repeat information to help the reader understand the point you are trying to make. For example,

Use: "The 40% increase in CO with the vents closed indicates that there was insufficient air for complete combustion. This is the predominant reason for the drop in efficiency from 82% to 68%."

Not: "The value of 12% CO with the vents closed can be compared to the data with the vents open to show how the position affects efficiency."

9. If you are having trouble with a sentence it is likely that you are trying to tie together two unrelated ideas. Stop for a moment and think about what it is you are trying to say. You may find that a number of shorter sentences conveys the information more clearly and makes the passage more readable.

Formatting

Figures - figures should be numbered in order and called in the text. Figure captions appear at the bottom of the figure (see Figure 1), usually centered, in bold and in 10 point font. Always include units in graphs (see Figure 2).

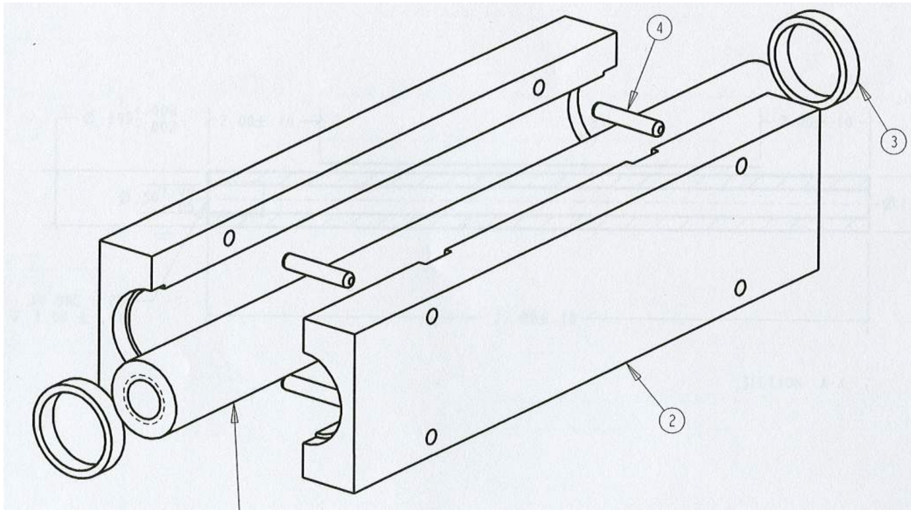


Figure 1: Mould and mandrel assembly

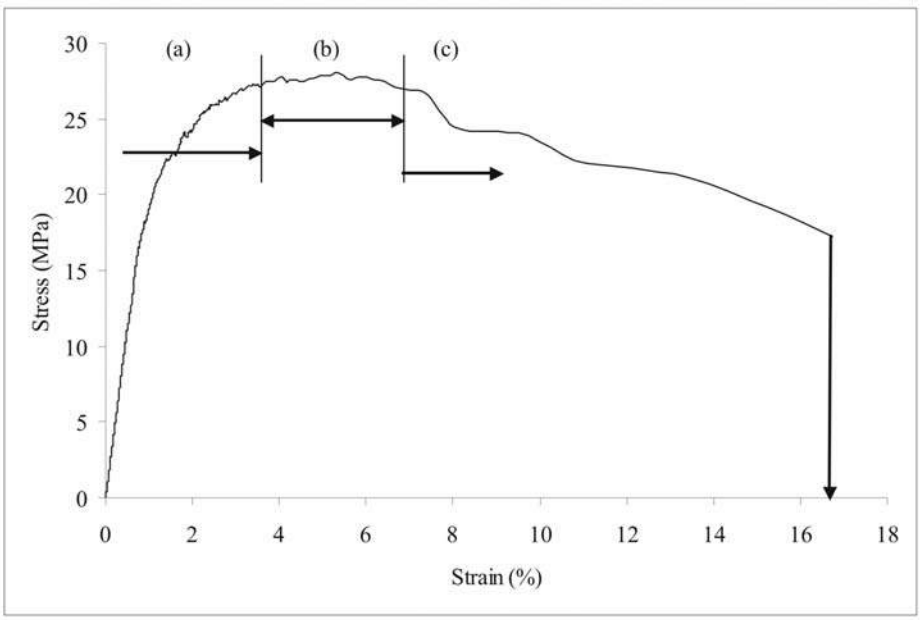


Figure 2: Stress-strain diagram. (a) elastic and preliminary yielding, (b) plastic deformation without necking, and (c) plastic deformation with necking and specimen failure. Note that all units MUST be given

Tables - table captions are bold, centered, in 100 point font and found above the table. Tables can have formatting or no formatting as seen in Table 1 and Table 2, respectively. Again, all units should be present.

Table 1: Experimental materials' properties

Material				
	E_m (GPa)		**G_m (GPa)**	**ν_m**
Epon 825/Ancamine 1482	3.5	-	1.3	0.35
	E_{11} (GPa)	**E_{22} (GPa)**	**G_{12} (GPa)**	**ν_{12}**
Kevlar/epoxy composite	79.7	5.9	2.3	0.33

Table 2: Range of target rigidity required for proposed braided catheter

Characteristic	Target range
Flexural	12.6 - 94.5 (10^{-3} Nm^2)
Torsional	17-1400 (10^{-3} Nm^2)
Axial	39.4 - 147 (kN)

References – sources used in a report (book, journal article, report, website, your friend/neighbour/dog, etc) must be cited in the reference section and in the text where the information was used. References should always be at the end of the paper in the order they appeared in the paper.

In the report, reference numbers should appear in brackets [] or in ^{superscript}. If text was taken word for word, or paraphrased, quotation marks should be used before and after; often the text is also placed as an individual paragraph in the text with 0.5” indent on both sides as shown in the following text. APA citation format guide is one format widely used. http://www.michener.ca/lrc/lrcapa.php

APPENDIX B
design methodology

table of contents

Figures

Figure B1 : Tension Pulley Concept Sketch
Figure B2 : High-Friction Winch Concept Sketch
Figure B3 : Fishing Reed Concept Sketch

Tables

Table B1 : Design Concept Selection

TABLE B1 Design Concept Selection

Design	Speed	Weight*	Size*	Simplicity	Cost*	Total	Ranking
Weighting	100	90	90	80	60		
Rating	10	10	10	10	10		
Tension Pulley	8	8	8	6	7	3140	1
High-Friction Winch	9	6	7	6	5	2850	2
Fishing Reel	5	3	6	9	8	2510	3

* A low weight, size and cost results in a high rating value

The team selected the Tension Pulley concept to continue with through the design and analysis stages.

Speed

The design needs to allow for quick rope ascent.

- The Tension Pulley and High-Friction Winch designs both allow for quick and flexible rope attachment, but the High-Friction Winch incurs greater friction, allowing for a slightly quicker ascent.
- The Fishing Reel design does not allow for quick or flexible rope attachment.

Weight

As outlined in the project requirements (see Appendix A), the design should strive to be lightweight.

- The Tension Pulley design contains three light pulleys, while the High-Friction Winch design contains a heavier winch component. The Fishing Reel design stores the entire length of rope around a very large and heavy spool.

Size

As outlined in the project requirements (see Appendix A), the design should strive to be compact.

- The Tension Pulley design contains three small pulleys and a quick entrance and exit from the device, while the High-Friction Winch design contains a winch component large enough to wrap the rope around several times within the device. The Fishing Reel design stores the entire length of rope around a very large spool.

Simplicity

A relatively simple design reduces the opportunity for failure and improves manufacturability.

- The Fishing Reel design is the most straightforward concept.
- The Tension Pulley and High-Friction Winch designs involve a greater number of components and interactions.

Cost

Although there are no specified budget constraints in this project, a lower cost is preferable from an economic standpoint.

- The Tension Pulley involes more components than the other two designs, and therefore requires a considerable amount of manufacturing time. However, the High-Friction Winch design involves a very precise winch component that would require the longest amount of manufacturing time.
- The Fishing Reel, as the simplest design, requires the least manufacturing time.

FIGURE B1 Tension Pulley Concept Sketch

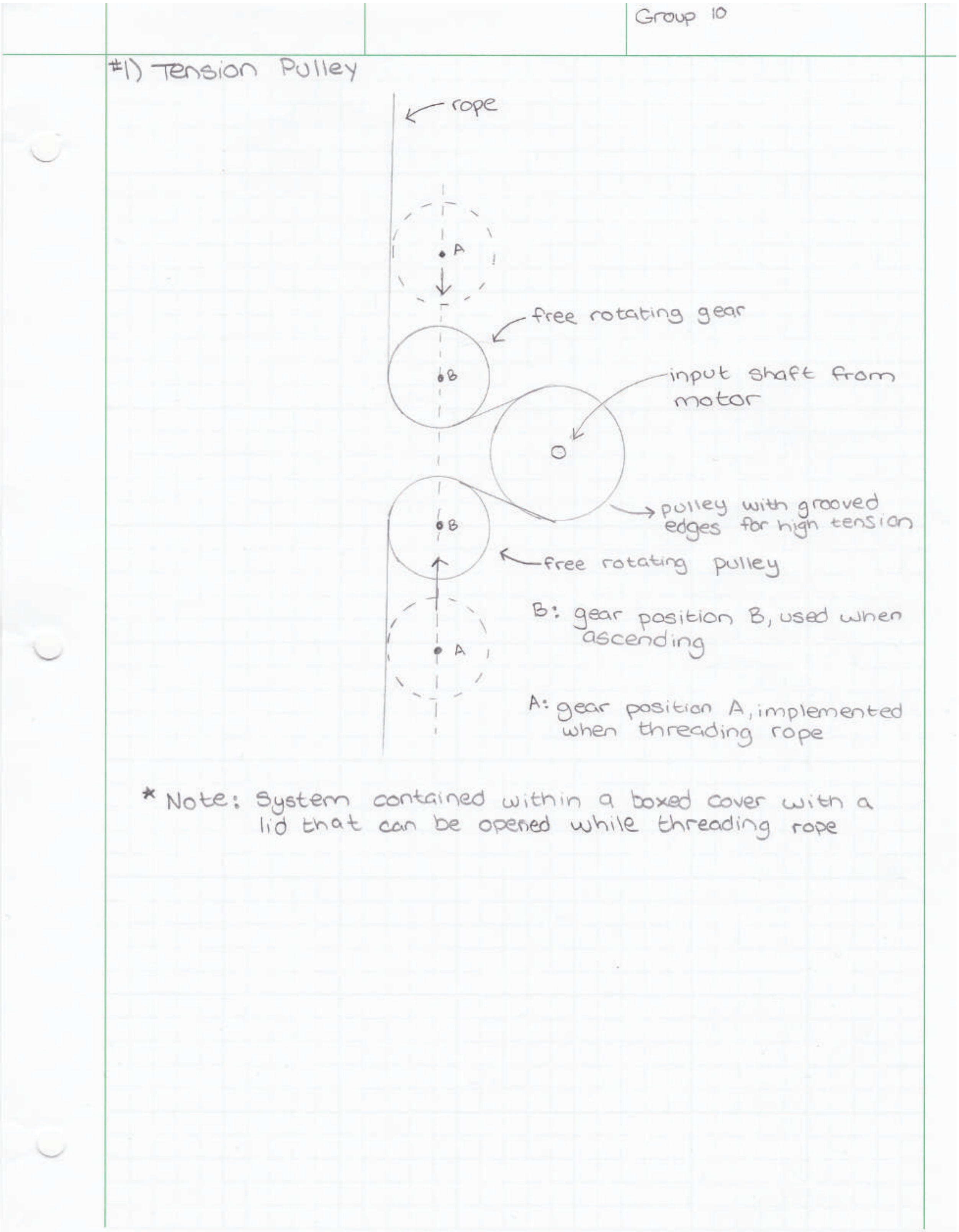


FIGURE B2 High-Friction Winch Concept Sketch

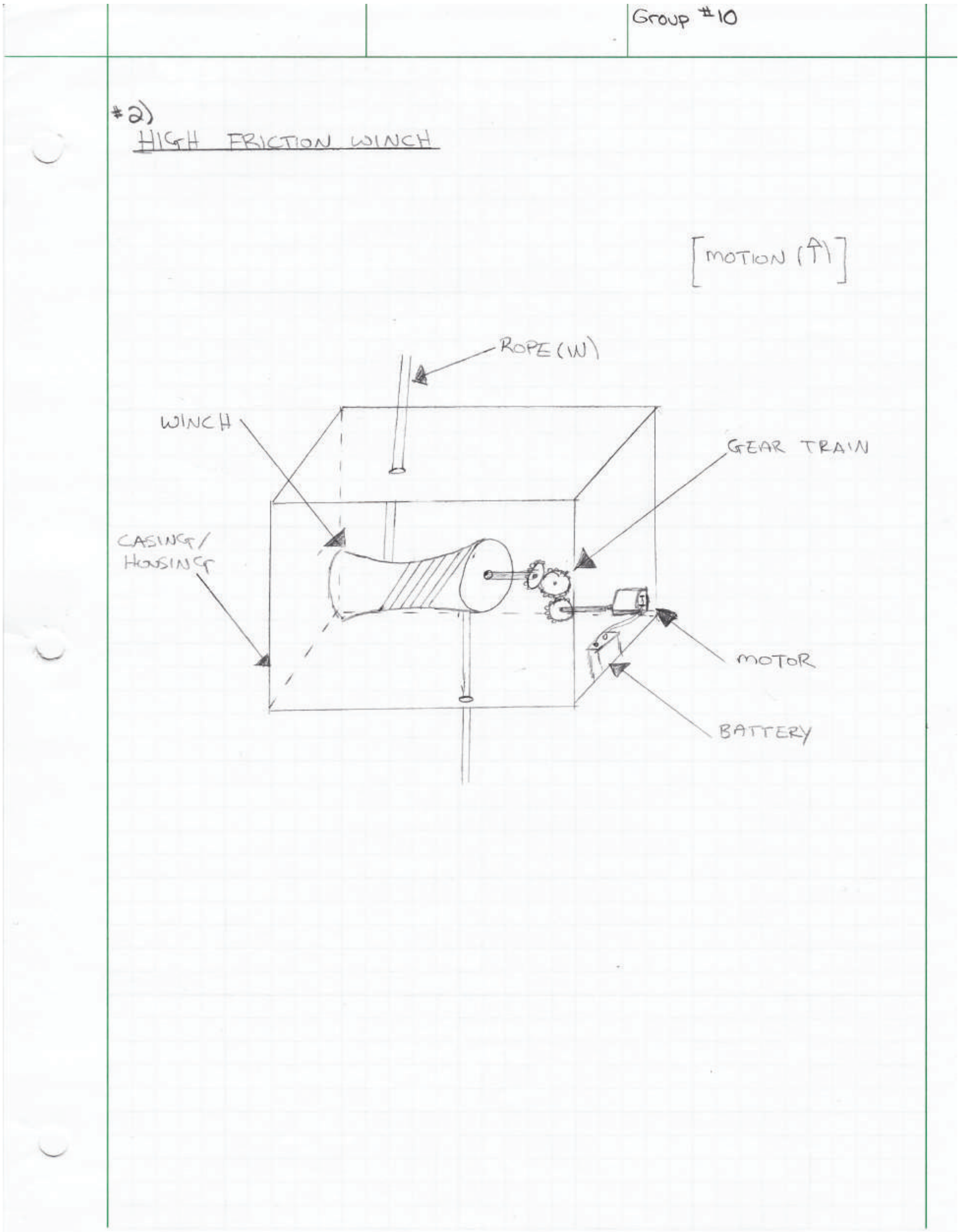
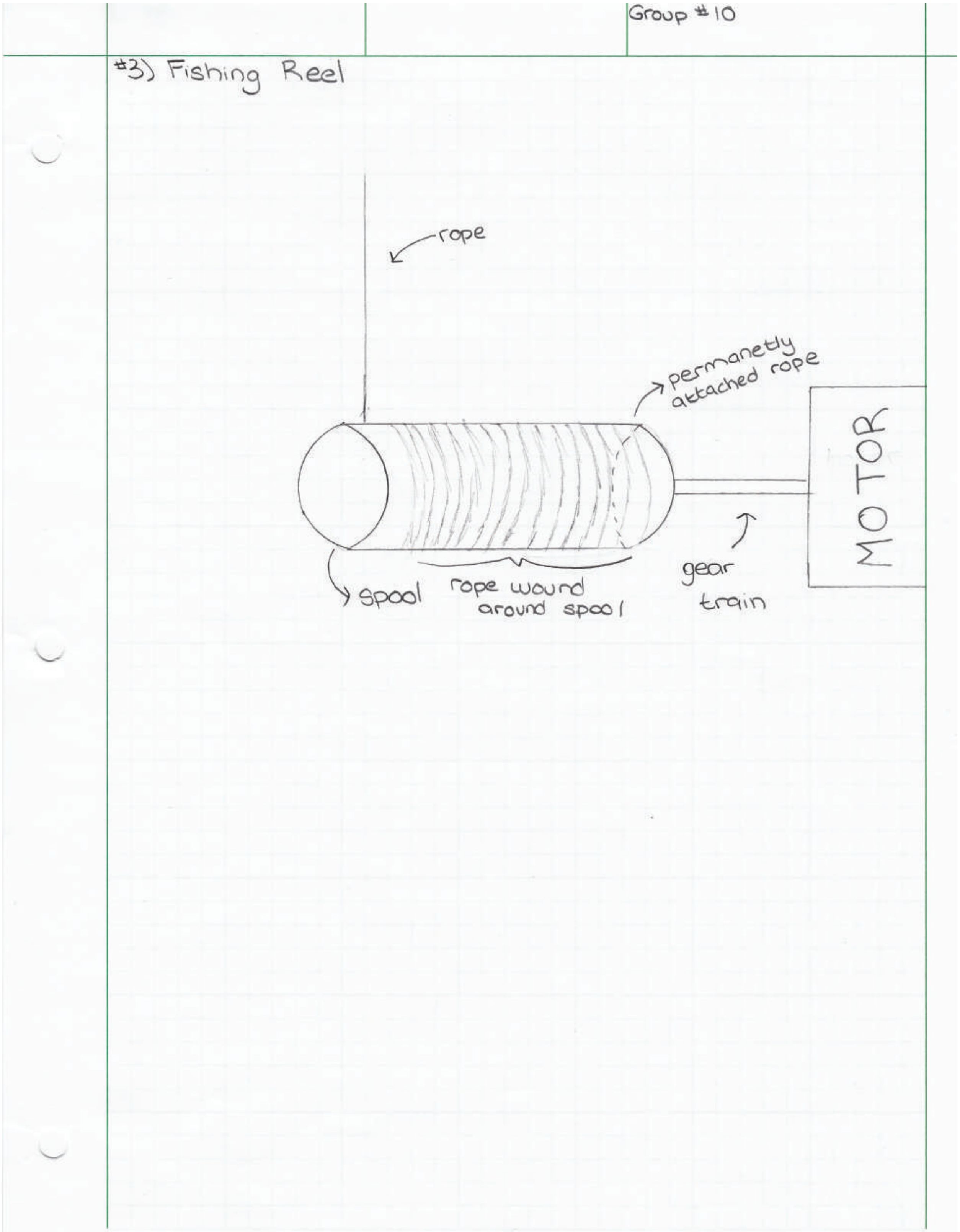


FIGURE B3 Fishing Reel Concept Sketch



APPENDIX C
decision matrices

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Rope Material	C1
Barb Material	C3
Casing Material	C5
References	C7

Tables

Table C1 : Rope Material Selection
Table C2 : Barb Material Selection
Table C3 : Casing Material Selection

rope material

TABLE C1 Rope Material Selection

Material	Strength	Stretch	Cost	Abrasion Resistance	Water Resistance	UV Resistance	Rotting Resistance	Total	Ranking
Weighting	100	50	20	20	10	10	10		
Rating	10	10	10	10	10	10	10		
Polypropylene	6.5	7	5	5	10	4	8	1410	3
Nylon	9	10	5	8	7	8	8	1890	1
Polyester	8	5	5	9	9	10	8	1610	2
Manila	5.5	4	10	5	2	8	0	1080	4

Nylon was selected as the rope material.

Strength

The strength of the rope is paramount in our device's ability to provide lift to a user.

- Polypropylene's breaking tenacity is approximately 6.5 grams/denier [10]
- Nylon's breaking tenacity is up to 9 grams/denier
- Polyester's breaking tenacity is 7-9 grams/denier
- Manila's breaking tenacity is 6-5 grams/denier

Stretch

Stretch is an important factor in ensuring the safety of our lift system, and the reliability of the pulley system, which stretches the rope. However, the rope does not need to be designed for very long periods of sustained use due to the relatively high speed of our lift system.

- Polypropylene typically breaks at 15-25% elongation, with high creep. [10]
- Nylon typically breaks at 18-25% elongation, with moderate creep.
- Polyester typically breaks at 12-15% elongation, with low creep.
- Manila typically breaks at 10-12% elongation, with very low creep.

Cost

The cost of the rope will be significantly smaller than the overall cost of our device, but it is an important factor to consider since our device is designed to allow for the substitution and replacement of rope.

- Polypropylene, nylon, and polyester ropes all have a similar consumer cost. [11] [1]
- Manila and other natural fiber ropes are much cheaper than synthetic ropes

rope material (continued)

Abrasion Resistance

As the rope passed through the pulley system, it will undergo high tension making it more susceptible to abrasion from friction, as well as direct abrasion due to the barbs on the exit pulley.

- Polypropylene has good surface and internal abrasion resistance. [9] [10]
- Nylon has very good surface abrasion resistance and excellent internal abrasion resistance.
- Polyester has excellent surface abrasion resistance and excellent internal abrasion resistance.
- Manila has good surface abrasion resistance and fair internal abrasion resistance

Water Resistance

Due to the likelihood of our device being used outdoors, water resistance (and ability to perform in wet conditions) can be an important factor in deciding rope material. As well, too much water absorption could lead to water build-up inside the device, which will affect performance.

- Polypropylene is completely water resistant. [10]
- Nylon is fairly water resistant (2-8% absorption of individual fibers) but can lose up to 25% of its strength when wet. [9]
- Polyester is nearly completely water resistant (less than 1% absorption of individual fibers)
- Manila and natural fibers absorb water very easily, and can shrink when wet.

UV Resistance

Due to the likelihood of our device being used outdoors, UV resistance (and ability to perform in wet conditions) can be an important factor in deciding rope material.

- Polyester has excellent resistance to UV radiation, with a UV Ranking of 5 [10]
- Nylon and Manila both have good UV resistance, with a UV Ranking of 4
- Polypropylene has generally poor resistance to UV radiation

Rotting Resistance

Due to the likelihood of our device being used outdoors, rotting and mildew resistance (and ability to perform in wet conditions) can be an important factor in deciding rope material.

- Polypropylene, Nylon, and Polyester, like other synthetic rope materials, have very good resistance to rotting and mildew [9]
- Manila, as a natural fiber, is extremely susceptible to rotting and mildew and can easily become brittle as a result.

barb material

TABLE C2 Barb Material Selection

Material	Strength	Durability	Cost	Ductility	Manufacturability	Total	Ranking
Weighting	100	30	30	20	20		
Rating	10	10	10	10	10		
Polyvinyl Chloride (PVC)	4	6	10	5	10	1180	4
Acrylonitrile butadiene styrene (ABS)	6	6	7	6	10	1290	2
High Density Polyethylene (HDPE)	5	7	8	7	8	1230	3
Stainless Steel	10	9	4	10	5	1690	1

*Each material meets the minimum strength requirement, therefore strength is not a factor in choosing the material

Stainless Steel was selected as the barb material.

Strength
The main function of the barbs is to hold the rope in place relative to Pulleys 2 and 3 as they rotate. The strength of the barb material is important in allowing the barbs to maintain their position.
- PVC typically has an ultimate tensile strength of less than 16 MPa. [14]
- ABS has an average ultimate tensile strength of 40 MPa.
- HDPE typically has an ultimate tensile strength of over 29 MPa.
- Stainless Steel has an ultimate tensile strength of 505 MPa.

Durability
The barbs will undergo much repeated stress during device operation.
- HDPE is the hardest material of the plastics. [4]
- Stainless Steel is harder than the plastics and therefore more durable.

Cost
Considering the number of barbs to be made (216 in total), cost is an important factor in determining their material.
- The average market price of PVC is 0.6063 Canadian dollars per kg. [12]
- The average market price of ABS is 1.1562 Canadian dollars per kg.
- The average market price of HDPE is 0.9729 Canadian dollars per kg.
- The average market price of Stainless Steel is 2.8586 Canadian dollars per kg. [8]

barb material (continued)

Ductility

Due to the intended versatility of the device, the barbs should be designed to undergo various stress concentrations. This could cause more brittle materials to fail.

- PVC has a glassy temperature of 10°C. [13]
- ABS has a glassy temperature of -20ºC.
- HDPE has a glassy temperature of -40°C, allowing it to withstand greater temperatures before becoming very brittle.
- Stainless Steel, as a metal, is much more ductile than plastics

Manufacturability

The barbs are very small and must be precisely produced in large numbers.

- ABS and PVC are generally very manufacturable as they can be formed by injection molding. [2]
- HDPE cannot be injection molded and cannot be turned. [4]
- Stainless Steel is much more costly to manufacture than synthetic products.

casing material

TABLE C3 Casing Material Selection

Material	Strength	Durability	Weight	Cost	Ductility	Total	Ranking
Weighting	100	100	80	50	20		
Rating	10	10	10	10	10		
Stainless Steel	10	7	6	6	6	2600	2
Titanium	8	10	7	4	6	2680	1
High Density Polyethylene (HDPE)	3	3	10	10	8	2060	3

Titanium was selected as the barb material.

Strength
The main function of the casing it to contain the components of our device and hold them in their functional positions. The strength of the material will be important in determining the casing's ability to maintain position under fluctuating stresses.
- Stainless Steel has an ultimate tensile strength of 505 MPa. [6]
- Titanium has an ultimate tensile strength of up to 370 MPa. [14]
- HDPE typically has an ultimate tensile strength of over 29 MPa. [7]

Durability
The casing will likely undergo repeated stresses throughout multiple uses of the device and must remain reliable. Since the casing also serves to protect the inner components from the effects of varying external conditions, it must be able to withstand these conditions.
- Titanium can easily withstand high and low temperatures, while stainless steel is more likely to shatter. Titanium is also nonmagnetic and anti-corrosive, whereas steel is corrosive and may be affected by magnetism. [3]
- Stainless Steel and Titanium, as metals, are harder and more durable than plastics.

Weight
The casing will be relatively large compared to the inner components of the device, and so will likely comprise a major portion of the device's weight. The material for the casing should strive to be lightweight, with regards to the original project requirements.
- Stainless Steel has an average density of 8,000 kg/m^3 [6]
- Titanium has an average density of 4,428.8 kg/m^3
- HDPE has an average density of 950 kg/m^3

casing material (continued)

Cost

Considering the large amount of material used to make the casing, cost is an important factor in determining its material.

- The average market price of Stainless Steel is 2.8586 Canadian dollars per kg. [8]
- The average market price of Titanium is 6.85 Canadian dollars per kg. [5]
- The average market price of HDPE is 0.9729 Canadian dollars per kg. [12]

Ductility

Due to the intended versatility of the device, the case should be designed to undergo various stress concentrations. This could cause more brittle materials to fail.

- HDPE has a glassy temperature of -40°C, allowing it to withstand considerable temperatures before becoming very brittle. [13]
- Stainless Steel and Titanium, as metals, are much more ductile than plastics.

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APPENDIX D
calculations

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Pulley Normal Force Calculations

Objective:
To determine the normal forces applied to each pulley.

Solution Method:
The normal force applied to the pulleys can be determined though static force analysis of each pulley individually.

Known:
- weight of system
- radii of pulleys

Assumptions:
- each pulley is rotating at constant angular velocity
- the weight of each pulley is negligible

Pulley 1 Calculations:

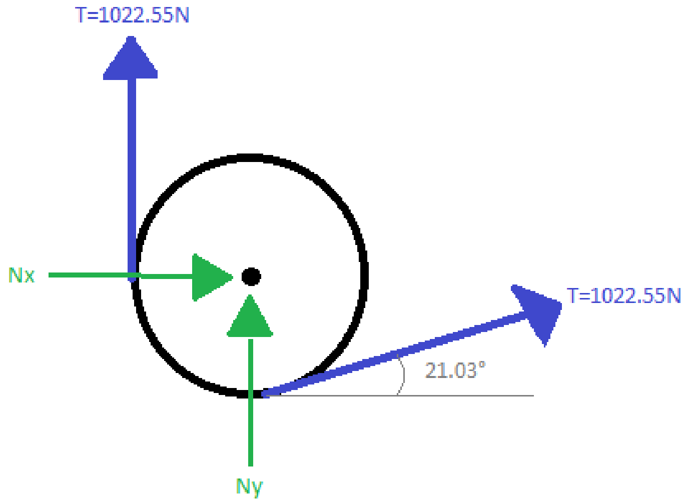
$g = 9.807\frac{m}{s^2}$ — Gravitational acceleration

$m_t := 104.235\,kg$ — Total mass of man and device

$T_r := m_t \cdot g = 1.022 \times 10^3\,N$ — Tension of rope

$\theta := 21.03\,deg$ — Angle of rope

Free Body Diagram:



Force Analysis:

$\Sigma F_x := 0$

$N_{x1} := -T_r \cos(\theta) = -954.11\,N$ — Reactionary force in horizontal direction

$\Sigma F_y := 0$

$N_{y1} := -T_r \sin(\theta) - T_r = -1.389 \times 10^3 \, N$ Reactionary force in vertical direction

$N_{T1} := \left(N_{y1}{}^2 + N_{x1}{}^2\right)^{0.5}$ Magnitude of reactionary force on Pulley 1

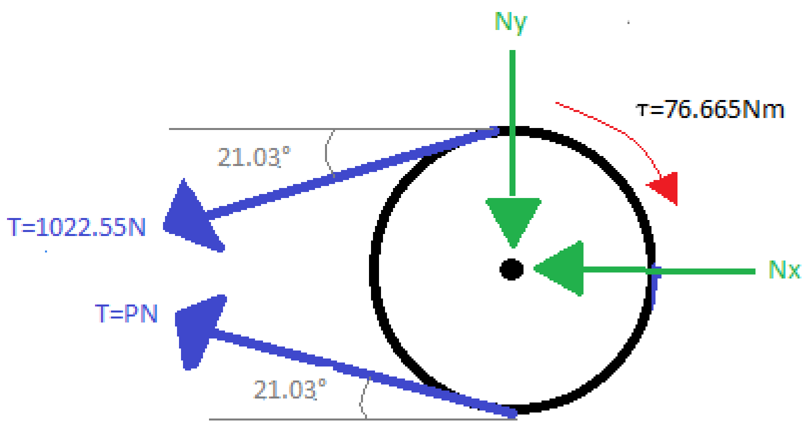
$N_{T1} = 1.685 \times 10^3 \, N$

Pulley 2 Calculations:

$r_2 := 0.075m$ Radius of Pulley 2

$\tau := T_r \cdot r_2 = 76.665 \, N \cdot m$ Torque applied to Pulley 2

Free Body Diagram:



Force Analysis:

$\Sigma M_o := 0$

$P := \frac{(r_2 \cdot T_r - \tau)}{r_2} = 0 \, N$ Tension of rope after Pulley 2

$\Sigma F_{x2} := 0$

$N_{x2} := -T_r \cos(\theta) = -954.11 \, N$ Reactionary force in horizontal direction

$\Sigma F_{y2} := 0$

$N_{y2} := -T_r \sin(\theta) = -366.822 \, N$ Reactionary force in vertical direction

$N_{T2} := \left(N_{y2}{}^2 + N_{x2}{}^2\right)^{0.5}$ Magnitude of reactionary force on Pulley 2

$N_{T2} = 1.022 \times 10^3 \, N$

Conclusion:

The magnitude of the normal force applied to Pulley 1 is 1.69kN and the magnitude of the normal force applied to Pulley 2 is 1.02kN. The tension of the rope after Pulley 2 is 0N therefore there is normal force applied Pulley 3.

Barb Calculations

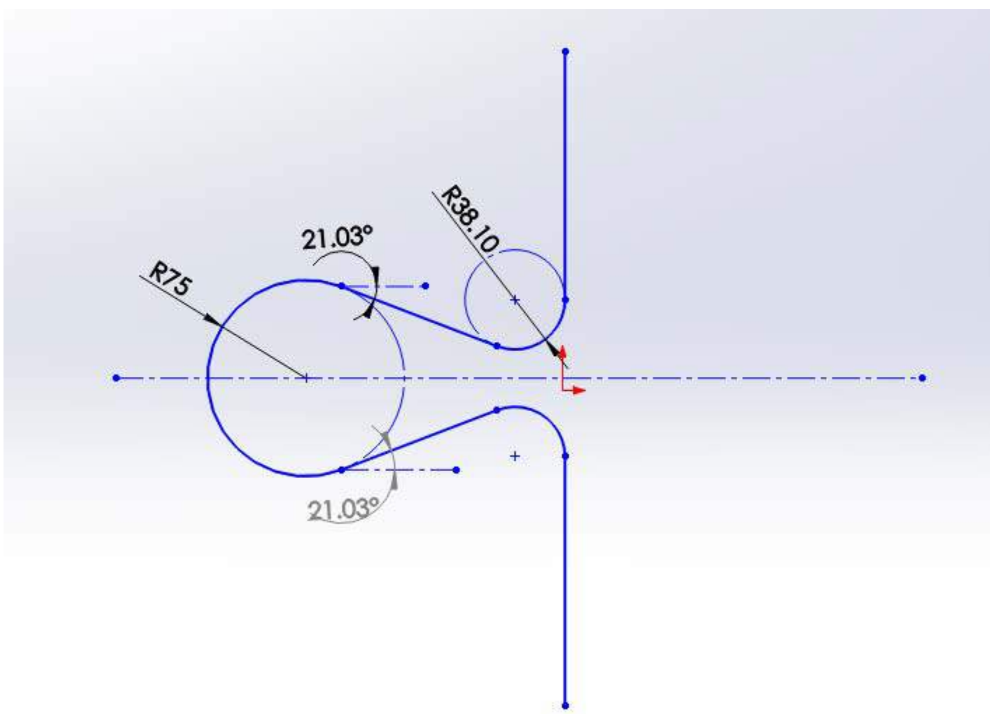
Objective:
To determine if stainless steel barbs will fail.

Solution Method:
A stress analysis on the barbs can be completed to calculate the safety factor of the applied load on the barbs. The resultant factor of safety must be larger than 4, the minimum factor of safety or a different material must be used. The barbs must be tested for shear and fatigue failure.

Known:
- barb dimensions
- angle of rope
- minimum safety factor (n=4)

Assumptions:
- the barbs are loaded in pure shear
- the ideal material of the barbs is stainless steel (Appendix C)
- force is evenly distributed between barbs

Geometry of Pulley 2 diagram:



From geometry, the the contact angle between the rope and pulley can be calculated:

$$contact\_angle := (180 + 2 \cdot 21.03) \cdot \frac{\pi}{180}$$

Angle of the rope in contact with Pulley 2

$$contact\_angle = 3.876 \cdot rad$$

General barb information:

- Each barb is a cone shape
- Barbs occur in pairs every 5° along Pulley 2
- The height of the barbs is 2.5mm
- The base diameter of the barbs is 0.5mm

$h := 2.5\text{mm}$ Heigh of barbs

$r := 0.5\text{mm}$ Base radius of barbs

$\text{deg\_occurance} := 5\cdot\frac{\pi\,\text{rad}}{180}$ Degree of occurance of barbs

$n_b := \frac{(\text{contact\_angle})}{(\text{deg\_occurance})}\cdot 2$ Total number of barbs in contact with the rope

$n_b = 88.824$

General stainless steel properties[1]:

$S_y := 215\text{MPa}$

$S_{ut} := 505\text{MPa}$

Shear Failure Calculations:

The force on each barb is calculated via the decrease in tension of the rope across Pulley 2 (Appendix D - Normal Force Pulley Calculations)

$F_t := 1022.55\text{N}$ Tension of the rope

$F_b := \frac{F_t}{n_b}$ Applied force to each barb assuming even distribution of force beween all barbs

$F_b = 11.512\,\text{N}$

Shear failure of barbs is checked at the bottom of the cone where shear stress is at a maximum

$\text{area} := \pi\cdot r^2$ Area at base of barb

$\text{area} = 7.854\times 10^{-7}\,\text{m}^2$

$$\tau := \frac{F_b}{area}$$ Shear stress applied to each barb

$$\tau = 1.466 \times 10^{7}\,\text{Pa}$$

Stainless steel is a ductile material therefore Distortion Theorem is used to determine shear failure:

$$\tau_{max} := \frac{S_y}{(\sqrt{3})}$$ Assuming pure shear

$$n := \frac{\tau_{max}}{\tau}$$ n is the safety factor

$$n = 8.469$$

The calculated safety factor is 8.5 which is greater than 4, the minimum factor of safety, therefore the barbs will not fail from shear.

Fatigue Failure Calculations:

Each barb is loaded and unloaded with every revolution of Pulley 2. This will cause eventual fatigue failure.

$$\sigma_{xx} := 0$$

$$\sigma_{yy} := 0$$ Only shear stress is applied to the barbs

$$\sigma_{zz} := 0$$

$$\tau_{xy} := \tau$$ Pure shear stress as calculated above

$$\tau_{xy} = 1.466 \times 10^{7}\,\text{Pa}$$

$$\tau_{yz} := 0$$

$$\tau_{zx} := 0$$

$$\sigma_{vm} := \frac{1\cdot\left[\left(\sigma_{xx}-\sigma_{yy}\right)^2+\left(\sigma_{yy}-\sigma_{zz}\right)^2+\left(\sigma_{zz}-\sigma_{xx}\right)^2+6\left(\tau_{xy}{}^2+\tau_{yz}{}^2+\tau_{zx}{}^2\right)\right]^{0.5}}{(\sqrt{2})}$$

$\sigma_{vm} = 2.539 \times 10^{7}\,\mathrm{Pa}$ Von Mises stress on each barb

$\sigma_{max} := \sigma_{vm}$ Max stress is when the barb is loaded in shear

$\sigma_{max} = 2.539 \times 10^{7}\,\mathrm{Pa}$

$\sigma_{min} := 0\mathrm{MPa}$ The minimum stress is when the barb is unloaded

$\sigma_a := \dfrac{(\sigma_{max} - \sigma_{min})}{2}$ Alternating stress

$\sigma_a = 1.269 \times 10^{7}\,\mathrm{Pa}$

$\sigma_m := \dfrac{(\sigma_{max} + \sigma_{min})}{2}$ Mean Stress

$\sigma_m = 1.269 \times 10^{7}\,\mathrm{Pa}$

$S_{eu} := 0.5045 \cdot S_{ut}$ Uncorrected endurance strength

$a := 1.58$

$b := -0.085$

$k_a := a \cdot \left(\dfrac{S_{ut}}{\mathrm{MPa}}\right)^{b}$ Assuming ground surface conditions

$d := 2 \cdot r$ Base diameter of barbs

$d = 1 \times 10^{-3}\,\mathrm{m}$

$k_b := 0.879 \cdot \left(\dfrac{d}{m}\right)^{-0.107}$

$k_b = 1.841$ Assuming a circular shaft for simplicity

$k_c := 0.59$ Pure shear loading

$k_d := 1$ Assuming no temperature correction

$k_e := 0.62$ Assuming 99.9999% reliability

$k_f := 1$ No miscellaneous modification factors

$S_e := k_a \cdot k_b \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S_{eu}$ Corrected endurance strength

$S_e = 1.597 \times 10^8\,\text{Pa}$

$$n_f := \sqrt{\frac{1}{\left(\frac{\sigma_a}{S_e}\right)^2 + \left(\frac{\sigma_m}{S_y}\right)^2}}$$ Fatigue factor of safety

$n_f = 10.099$

The minimum factor of safety for fatigue is 4. The calculated factor of safety for fatigue is 10.1. Therefore stainless steel will not fail in fatigue.

<u>Conclusion:</u>
The ideal material for the barbs was concluded to be stainless steel (Appendix C). To ensure that stainless steel is strong enough under both shear and fatigue stress the factor of safety was calculated for both forms of stress. A minimum factor of safety of 4 was assumed. The calculated factor of safety for shear was 8.5, therefore stainless steel is proven to be safe in shear stress. The fatigue factor of safety was calculated to be 10.1, therefore stainless steel is also safe from fatigue failure. These results show that stainless steel is a safe material and is ideal for the production of the barbs.

Rope Calculations

Objective:
To determine if the 9mm diameter Kernmantle nylon rope will fail.

Solution Method:
To determine if the rope will fail, the stress applied to the rope must be calculated. This value can then be compared to the ultimate stress of the rope through the calculation of the resultant factor of safety. If the calculated factor of safety is less than the minimum factor of safety then a rope with a larger diameter must then be tested.

Known:
- properties of Kernmantle nylon rope
- diameter of rope
- force applied to rope
- stress concentration factor

Assumptions:
- minimum factor of safety of 4
- total mass of device and user

Properties of Kernmantle nylon rope[2]:

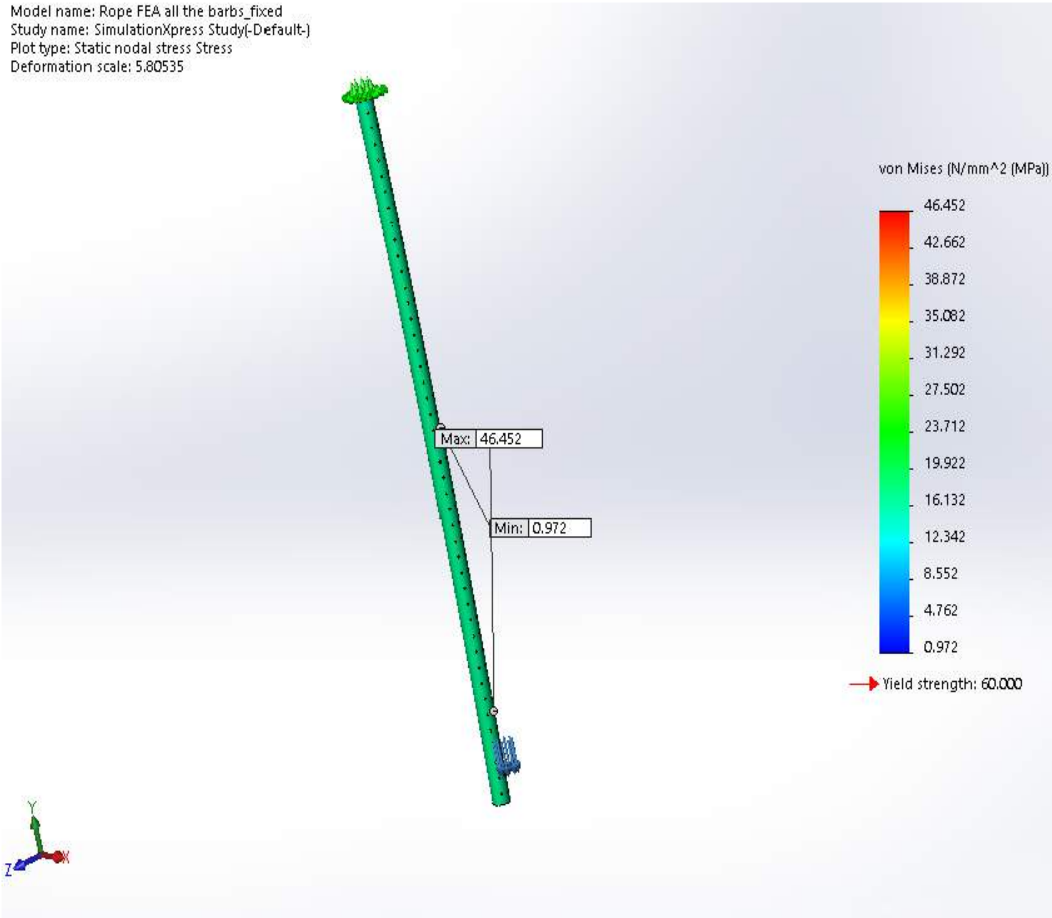
$F_{ut} := 21\text{kN}$ Ultimate Force

$r := 4.5\text{mm}$ Radius of rope

$a := \pi r^2 = 6.362 \times 10^{-5}\,\text{m}^2$ Cross sectional area of rope

$S_{ut} := \dfrac{F_{ut}}{a} = 3.301 \times 10^{8}\,\text{Pa}$ Ultimate stress of rope

Stress concentration factor calulations using SolidWorks finite element analysis:



$\sigma_{vm} := 46.452\text{MPa}$	Von Mises stress as calculated by Solidworks
$F_{in} := 1079.1\text{N}$	Input finite analysis force
$\sigma_{nsw} := \dfrac{F_{in}}{a}$	Nominal SolidWorks stress
$K_t := \dfrac{\sigma_{vm}}{\sigma_{nsw}}$	Stress concentration factor determined by Solidworks
$K_t = 2.739$	

Stress applied to rope calculations:

$m_t := 104.235\text{kg}$	Total mass of device and user
$g = 9.807\,\dfrac{\text{m}}{\text{s}^2}$	Gravitational acceleration

$P := m_t \cdot g$ Force applied to rope

$P = 1.022 \times 10^3\,N$

$\sigma_n := \frac{P}{a} = 1.607 \times 10^7\,Pa$ Nominal stress applied to rope

$\sigma_{max} := K_t \cdot \sigma_n = 4.4 \times 10^7\,Pa$ Maximum stress applied to rope

$n := \frac{S_{ut}}{\sigma_{max}}$ Factor of safety

$n = 7.502$

Conclusion:

The calculated factor of safety is 7.5 which is greater than 4, the minimum factor of safety. Therefore the 9mm diameter Kernmantle rope can be used safely.

Gear Train Calculations

Objective:
To determine the necessary planetary gear reduction required to meet the design requirements, as well as the maximum number of planet gears and finally the pitch line velocity of each gear.

Solution Method:
The gear reduction necessary from the motor to Pulley 2 can be calculated by determining the minimum angular velocity of Pulley 2 and the minimum torque. When the necessary gear reduction has been found the number of teeth of each gear can then be determined. Finally the maximum number of planet gears and the pitch velocity of each gear can also be determined via the known gear reduction.

Known:
- minimum speed of upward ascent
- ring gear is fixed and there has no angular velocity
- radius of Pulley 2
- motor specifications

Assumptions:
- no interference between gears
- device travels at a constant velocity
- approximate total mass of device
- motor provides maximum power at half the maximum angular velocity
- sun gear has 24 teeth

Design Requirements:

$$v_{min} := 4\frac{m}{s}$$

Minimum velocity that the device travels up the rope

$$r_2 := 0.075m$$

Radius of Pulley 2

$$\omega_{required} := \frac{v_{min}}{r_2}$$

Angular velocity of Pulley 2 required to maintain the minimum velocity of the device

$$\omega_{required} = 53.333\cdot\frac{rad}{s}$$

$$g = 9.807\frac{m}{s^2}$$

Acceleration due to gravity

$$m_t := 104.235kg$$

Approximated mass of the person and the device

$$F_{weight} := g\cdot m_t = 1.022\times10^3 N$$

Applied force to the device due to the total mass of person and device

$$\tau_{required} := r_2 \cdot F_{weight}$$

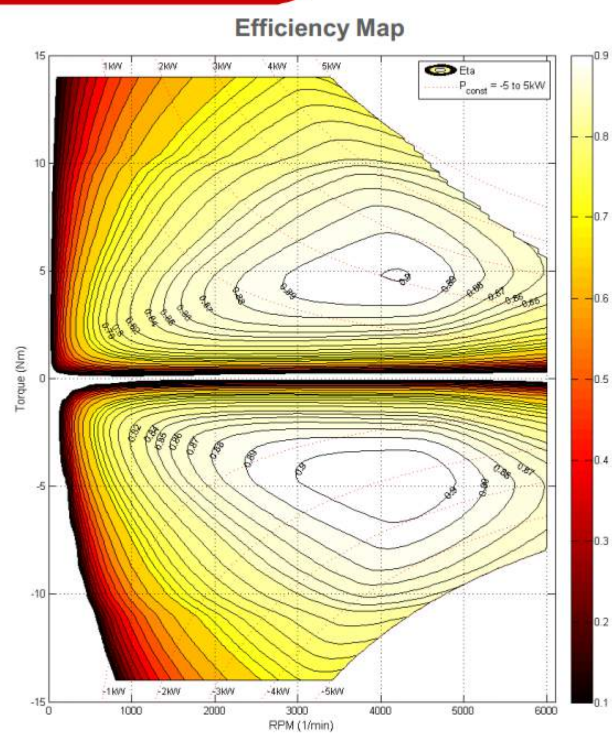
Minimum torque of Pulley 2 required to lift the total mass up the rope at the minimum specified velocity

$$\tau_{required} = 76.665 \cdot N \cdot m$$

Motor Specifications:

Torque Speed Curve provided by manufacturer[3]:

▶ Technical data



$$\omega_{max} := 6000rpm$$

Maximum angular velocity input from the motor - determined via the Torque Speed Curve

$$\tau_{max} := 15N \cdot m$$

Maximum power occurs at approximately half the angular velocity - value found using the Torque Speed Curve

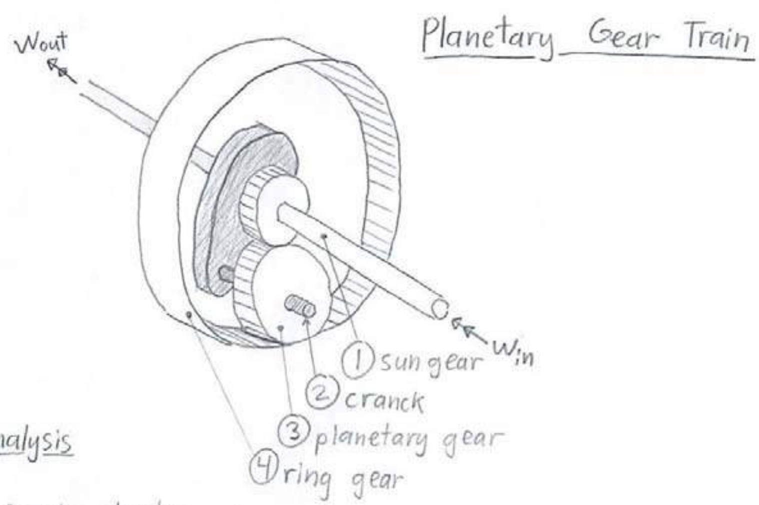
$$\omega := \frac{\omega_{max}}{2}$$

Angular velocity input of the motor at which torque input is max

$$\omega_{in} := 320\,\frac{rad}{s}$$

Angular velocity input from the motor to the gear train (approximately half the maximum angular velocity of the motor)

Gear Reduction Calculations:



Tabular analysis

sun ω_1	Cranck ω_2	planetary ω_3	ring ω_4	
x	x	x	x	RBM
y	0	$-\frac{N_1}{N_3}y$	$-\frac{N_1}{N_4}y$	relative motion
$x+y$	x	$x-\frac{N_1}{N_3}y$	$x-\frac{N_1}{N_4}y$	superposition

Boundary Conditions

- the sun gear is the input to the gear system
- the crank is the output of the gear system
- the ring gear is fixed to the casing and cannot rotate

$$\omega_1 := \omega_{in}$$

$$\omega_1 = 320\cdot\frac{rad}{s}$$

Angular velocity of the sun gear- the sun gear shaft is attached to the motor therefore the angular velocity of the sun gear is equal to that of the input by the motor

$$\omega_{out} := \omega_{required}$$

$$\omega_{out} = 53.333\cdot\frac{rad}{s}$$

The angular output speed of the gear system is the required angular velocity based on the design conditions

$\omega_2 := \omega_{out}$ Angular velocity of the crank

$\omega_4 := 0\,\frac{rad}{s}$ Angular velocity of the ring gear

$x := \omega_2 = 53.333 \cdot \frac{rad}{s}$ Rigid body angular velocity of gear system

$y := \omega_1 - x = 266.667 \cdot \frac{rad}{s}$ Relative angular velocity of gear system

Gear Ratio Calculations:

$e_{ratio} := \frac{\omega_{in}}{\omega_{out}} = 6$ Speed ratio - gear reduction

$E := \frac{y}{x} = 5$ Ring gear to sun gear teeth ratio

Number of teeth calculations:

$N_1 := 24$ Assuming initial number of teeth of the sun gear is 24

$N_4 := E \cdot N_1 = 120$ Number of teeth of the ring gear

$N_3 := \frac{(N_4 - N_1)}{2}$ Number of teeth of the planet gear - calculated using centre to centre distance of sun to ring gear mesh

$N_3 = 48$

Output torque calculations:

$\tau_{output} := \tau_{max} \cdot e_{ratio}$ Torque applied to Pulley 2 after the application of the gear system

$\tau_{output} = 90 \cdot N \cdot m$

$\tau_{required} = 76.665 \cdot N \cdot m$ Required torque of Pulley 2

The torque output is greater than the torque required therefore the minimum design requirements are met.

Maximum number of planet gear calculations:

$\alpha := 0.8$ Standard value for stub tooth gears

$$n_{max} := \frac{180deg}{asin\left[\frac{(N_3 + 2\alpha)}{N_1 + N_3}\right]}$$

Maximum number of planet gears

$$n_{max} = 4.134$$

$$n < 4.134$$

$$n := 4$$

Assume n=4 and check whether the integer variable is a whole number

$$integer := \frac{(N_1 + N_4)}{n}$$

Number of teeth of the sun gear plus the ring gear must be a whole number when divided by the number of planet gears

$$integer = 36$$

A whole number is calculated when there are 4 planet gears

The maximum number of planet gears for the planetary gear system is 4.

Pitch line velocity calculations:

The pitch line velocity is produced by relative motion between the gears, therefore rigid body motion is neglected and pitch line velocity is calculated based on the equivalent ordinary gear train system.

$$module := 1.5mm$$

Module of all the gears

$$d_{p1} := module \cdot N_1$$

Pitch diameter of the sun gear

$$r_1 := \frac{d_{p1}}{2}$$

Radius of the sun gear

$$r_1 = 0.018\,m$$

$$v_{13} := y \cdot r_1$$

Pitch line velocity of gear mesh 1-3

$$v_{13} = 4.8\frac{m}{s}$$

$$v_{34} := v_{13} = 4.8\frac{m}{s}$$

Pitch line velocity of gear mesh 3-4

Conclusion:

The necessary speed ratio to produce the minimum angular velocity and torque of Pulley 2 is 6. Therefore the number of teeth of the sun, planet and ring gears are 24, 48 and 120 respectively. This results in Pulley 2 rotating at an angular velocity of 53.3rad/s. Pulley 2 will also have a resultant torque of 90Nm. The minimum angular velocity of pulley 2 is 53.3rad/s and the minimum torque requirement is 76.7Nm. Both of these minimum values are achieved therefore the calculated gear system will meet the design requirements. The maximum number of planet gears was also determined to be 4 based on the number of teeth of the sun and ring gears. Finally the pitch line velocity of of each gear was calculated to be 4.8m/s.

Gear Stresses - Sun Gear Calculations

Objective:
To determine if the sun gear will fail.

Solution Method:
Use of a stress analysis method can determine the contact stress factor of safety as well as the bending stress factor of safety. These values can then be compared to 4, the minimum factor of safety to determine if the gear fails. If the gear fails the gear material and/or face width must be adjusted.

Known:
- number of gear teeth
- input torque
- gear face width
- gear material (stainless steel)
- minimum safety factor (n=4)

Assumptions:
- no gear interference
- negligible frictional forces
- nonzero backlash
- root fillets are standard
- no teeth are pointed

General gear information:
- all gears are spur gears

$N_1 := 24$ Number of teeth on gear 1 (pinion)

$N_3 := 48$ Number of teeth on gear 3

$N_4 := 120$ Number of teeth on gear 4

$module := 1.5mm$ Module of all gears

$d_{p.1} := module \cdot N_1$ Pitch diameter

$d_{p.1} = 0.036\,m$

$d_{p.3} := module \cdot N_3$

$d_{p.3} = 0.072\,m$

$d_{p.4} := module \cdot N_4$

$d_{p.4} = 0.18\,m$

$\omega_{in} := 266.6667 \frac{rad}{s}$ Angular velocity input (Appendix D - Gear Train Calculations)

$Torque := 15N \cdot m$ Applied pinion torque

$\phi := 20 \frac{\pi}{180}$ Pressure angle

Contact Stress Calculations:

$V_{t1} := 0.5 \cdot d_{p.1} \cdot \omega_{in}$ Pitch line velocity

$V_{t1} = 4.8 \frac{m}{s}$

Transverse Power Load:

$W_t := \frac{Torque}{0.5 \cdot d_{p.1}}$ Force tangential - torque on the pinion (sun gear) divided by the pinion radius

$W_t = 833.333\,N$

AGMA Equation:

$Q_v := 8$ Assuming medium quality gears

$$B := \frac{\left(12 - Q_v\right)^{\frac{2}{3}}}{4}$$

$B = 0.63$

$A_1 := 50 + 56(1 - B)$

$A_1 = 70.722$

$$C_v := \left(\frac{A_1}{A_1 + \sqrt{200 \cdot \frac{V_{t1}}{1 \frac{m}{s}}}} \right)^B$$ Dynamic factor

$C_v = 0.795$

$F_w := 16mm$ Face width

$C_a := 1.25$ Application factor assuming moderate shock

$C_s := 1$ Standard size factor

$C_p := 191MPa^{0.5}$ Elastic coefficient for a steel pinion gear

$C_m := 1.6$ Load distribution factor when face width is 16mm

$C_f := 1$ Standard surface finish

$I := \frac{1}{2}(\sin(\phi)\cdot\cos(\phi))\frac{N_3}{N_3 + N_1}$ Geometric factor for external gear mesh

$I = 0.107$

$\sigma_c := C_p\cdot\sqrt{\frac{W_t\cdot C_a\cdot C_m\cdot C_s\cdot C_f}{F_w\cdot I\cdot d_{p.3}\cdot C_v}}$ Contact stress on sun gear

$\sigma_c = 7.87 \times 10^8 Pa$

Contact Stress Factor of Safety Calculations:

$C_T := 1$ Temperature factor assuming a temperature less than 250°F

$C_R := 0.85$ Reliability factor assuming 90% reliability

$C_H := 1$ Hardness ratio factor - same gear material

$N_C := 10^6$ One million cycles of use

$C_L := 2.466 N_C^{-0.056}$ Life factor at one million cycles

$C_L = 1.138$

$S_{fcp} := 1200\text{MPa}$ Uncorrected surface fatigue strength assuming steel at 400 HB Grade 2

$S_{fc} := \dfrac{(C_L \cdot C_H) S_{fcp}}{(C_T \cdot C_R)}$ Corrected surface fatigue strength of sun gear

$S_{fc} = 1.606 \times 10^9\,\text{Pa}$

$N_c := \left(\dfrac{S_{fc}}{\sigma_c}\right)^2$ Contact stress factor of safety

$N_c = 4.164$

The contact stress factor of safety is 4.2 for the sun gear. This is greater than the minimum value of 4, therefore the sun gear does not fail from contact stress.

Bending Stress Calculations:

$K_a := C_a$ Application factor assuming moderate shock

$K_a = 1.25$

$K_m := C_m$ Load distribution factor when face width is 16mm

$K_m = 1.6$

$K_s := 1$ Standard size factor

$K_B := 1$ Rim thickness factor for solid disc gears

$K_I := 1$ Idler factor - sun gear is not an idler

$K_v := C_v$ Dynamic factor

$C_v = 0.795$

$J_1 := 0.35$ Geometry factor[4]

$$\sigma_b := \frac{\left(W_t \cdot K_a \cdot K_m \cdot K_s \cdot K_B \cdot K_I\right)}{F_w \cdot \text{module} \cdot J_1 \cdot K_v}$$ Bending stress on sun gear

$\sigma_b = 2.494 \times 10^8\, \text{Pa}$

Bending Stress Factor Of Safety:

$S_{fbp} := 390\text{MPa}$ Uncorrected bending strength assuming steel at 400 HB Grade 2

$K_L := 9.4518 N_c^{-0.148}$ Life factor at a million cycles

$K_L = 7.653$

$K_T := 1$ Temperature factor assuming a temperature less than 250°F

$K_R := 0.85$ Reliability factor assuming 90% reliability

$$S_{fb} := \frac{K_L \cdot S_{fbp}}{K_T \cdot K_R}$$ Corrected bending strength assuming steel at 400 HB Grade 2

$S_{fb} = 3.511 \times 10^9\, \text{Pa}$

$$N_b := \frac{S_{fb}}{\sigma_b}$$ Bending stress factor of safety

$N_b = 14.076$

The bending stress factor of safety is 14.1. This calculated value is greater than the minimum factor of safety of 4. Therefore the sun gear will not fail from bending.

Conclusion:

The factor of safety for contact stress is 4.2, and the factor of safety for bending stress 14.1. Both of these values are above the minimum factor of safety of 4. Therefore, the sun gear will not fail from bending or contact stresses. This indicates that the material and face width of the gear are appropriate for the device and will not lead to gear failure.

Gear Stresses - Planet Gear Calculations

Objective:
To determine if the planet gear will fail.

Solution Method:
A stress analysis method can be used to determine the contact stress factor of safety as well as the bending stress factor of safety. These values can then be compared to 4, the minimum factor of safety, to determine if the gear fails. If the gear fails, the gear material and/or face width must be adjusted.

Known:
- number of gear teeth
- input torque
- gear face width
- gear material (stainless steel)
- minimum safety factor (n=4)

Assumptions:
- no gear interference
- negligible frictional forces
- nonzero backlash
- root fillets are standard
- no teeth are pointed

General gear information:
- all gears are spur gears

$N_1 := 24$ Number of teeth on gear 1 (pinion)

$N_3 := 48$ Number of teeth on gear 3

$N_4 := 120$ Number of teeth on gear 4

$\text{module} := 1.5\text{mm}$ Module for all gears

$d_{p.1} := \text{module} \cdot N_1$ Pitch diameter

$d_{p.1} = 0.036\,\text{m}$

$d_{p.3} := \text{module} \cdot N_3$

$d_{p.3} = 0.072\,\text{m}$

$d_{p.4} := \text{module} \cdot N_4$

$d_{p.4} = 0.18\,\text{m}$

$$\omega_{in} := 266.6667 \frac{rad}{s}$$

Angular velocity input (Appendix D - Gear Train Calculations)

$$Torque := 15N \cdot m$$

Applied pinion torque

$$\phi := 20 \frac{\pi}{180}$$

Pressure angle

Contact Stress Calculations:

$$V_{t3} := d_{p.1} \cdot 0.5 \cdot \omega_{in}$$

Pitch line velocity

$$V_{t3} = 4.8 \frac{m}{s}$$

Transverse Power Load:

$$W_t := \frac{Torque}{0.5 \cdot d_{p.1}}$$

Force tangential - torque on the pinion (sun gear) divided by the pinion radius

$$W_t = 833.333\,N$$

AGMA Equation:

$$Q_v := 8$$

Assuming medium quality gears

$$B := \frac{\left(12 - Q_v\right)^{\frac{2}{3}}}{4}$$

$$B = 0.63$$

$$A_1 := 50 + 56(1 - B)$$

$$A_1 = 70.722$$

$$C_v := \left(\frac{A_1}{A_1 + \sqrt{200 \cdot \frac{V_{t3}}{1 \frac{m}{s}}}} \right)^B$$

Dynamic factor

$C_v = 0.795$

$F_w := 16mm$ Face width

$C_a := 1.25$ Application factor assuming moderate shock

$C_s := 1$ Standard size factor

$C_p := 191MPa^{0.5}$ Elastic coefficient for a steel pinion gear

$C_m := 1.6$ Load distribution factor when face width is 16mm

$C_f := 1$ Standard surface finish

$I := \frac{1}{2}(\sin(\phi)\cdot\cos(\phi))\frac{N_3}{N_3 + N_1}$ Geometric factor for external gear mesh

$I = 0.107$

$\sigma_c := C_p\cdot\sqrt{\frac{W_t\cdot C_a\cdot C_m\cdot C_s\cdot C_f}{F_w\cdot I\cdot d_{p.3}\cdot C_v}}$ Contact stress on planet gear

$\sigma_c = 7.87 \times 10^8 Pa$

Contact Stress Factor of Safety Calculations:

$C_T := 1$ Temperature factor assuming a temperature less than 250°F

$C_R := 0.85$ Reliability factor assuming 90% reliability

$C_H := 1$ Hardness ratio factor - same gear material

$N_C := 10^6$ One million cycles of use

$C_L := 2.466 N_C^{-0.056}$ Life factor at one million cycles

$C_L = 1.138$

$S_{fcp} := 1200\text{MPa}$ Uncorrected surface fatigue strength assuming steel at 400 HB Grade 2

$$S_{fc} := \frac{(C_L \cdot C_H) S_{fcp}}{(C_T \cdot C_R)}$$

Corrected surface fatigue strength of planet gear

$S_{fc} = 1.606 \times 10^9 \text{Pa}$

$$N_c := \left(\frac{S_{fc}}{\sigma_c}\right)^2$$

Contact stress factor of safety

$N_c = 4.164$

The contact stress factor of safety is 4.2 for the planet gear. This is greater than the minimum value of 4, therefore the planet gear does not fail from contact stress.

Bending Stress Calculations:

$K_a := C_a$ Application factor assuming moderate shock

$K_a = 1.25$

$K_m := C_m$ Load distribution factor when face width is 16mm

$K_m = 1.6$

$K_s := 1$ Standard size factor

$K_B := 1$ Rim thickness factor for solid disc gears

$K_I := 1.42$ Idler factor - planet gear is an idler

$K_v := C_v$ Dynamic factor

$C_v = 0.795$

$J_1 := 0.4$ Geometry factor[4]

$$\sigma_b := \frac{\left(W_t \cdot K_a \cdot K_m \cdot K_s \cdot K_B \cdot K_I\right)}{F_w \cdot module \cdot J_1 \cdot K_v}$$ Bending stress on planet gear

$\sigma_b = 3.099 \times 10^8\,Pa$

Bending Stress Factor Of Safety:

$S_{fbp} := 390MPa$ Uncorrected bending strength assuming steel at 400 HB Grade 2

$K_L := 9.4518 N_c^{-0.148}$ Life factor at a million cycles

$K_L = 7.653$

$K_T := 1$ Temperature factor assuming a temperature less than 250°F

$K_R := 0.85$ Reliability factor assuming 90% reliability

$$S_{fb} := \frac{K_L \cdot S_{fbp}}{K_T \cdot K_R}$$ Corrected bending strength assuming steel at 400 HB Grade 2

$S_{fb} = 3.511 \times 10^9\,Pa$

$$N_b := \frac{S_{fb}}{\sigma_b}$$ Bending stress factor of safety

$N_b = 11.329$

The bending stress factor of safety is 11.3. This calculated value is greater than the minimum factor of safety of 4. Therefore, the planet gear will not fail from bending stress.

Conclusion:

The factor of safety for contact stress is 4.2, and the factor of safety for bending stress is 11.3. Both of these values are above the minimum factor of safety of 4. Therefore, the planet gear will not fail from bending or contact stresses. This indicates that the material and face width of the gear are appropriate for the device and will not lead to gear failure.

Gear Stresses - Ring Gear Calculations

Objective:
To determine if the ring gear will fail.

Solution Method:
A stress analysis method can be used to determine the contact stress factor of safety as well as the bending stress factor of safety. These values can then be compared to 4, the minimum factor of safety, to determine if the gear fails. If the gear fails the gear material and/or face width must be adjusted.

Known:
- number of gear teeth
- input torque
- gear face width
- gear material (stainless steel)
- minimum safety factor (n=4)

Assumptions:
- no gear interference
- negligible frictional forces
- nonzero backlash
- root fillets are standard
- no teeth are pointed

General gear information:
- all gears are spur gears

$N_1 := 24$ Number of teeth on gear 1

$N_3 := 48$ Number of teeth on gear 3 (pinion)

$N_4 := 120$ Number of teeth on gear 4

$\text{module} := 1.5\text{mm}$ Module of all gears

$d_{p.1} := \text{module} \cdot N_1$ Pitch diameter

$d_{p.1} = 0.036\,\text{m}$

$d_{p.3} := \text{module} \cdot N_3$

$d_{p.3} = 0.072\,\text{m}$

$d_{p.4} := \text{module} \cdot N_4$

$d_{p.4} = 0.18\,\text{m}$

$$\omega_{in} := 53.333\,\frac{rad}{s}$$

Angular velocity input (Appendix D - Gear Train Calculations)

$$Torque := 120N\cdot m$$

Applied pinion torque

$$\phi := 20\frac{\pi}{180}$$

Pressure angle

Contact Stress Calculations:

$$V_{t3} := \omega_{in}\cdot d_{p.4}\cdot 0.5$$

Pitch line velocity

$$V_{t3} = 4.8\,\frac{m}{s}$$

Transverse Power Load:

$$W_t := \frac{Torque}{0.5\cdot d_{p.3}}$$

Force tangential - torque on the pinion (planet gear) divided by the pinion radius

$$W_t = 3.333\times 10^3\,N$$

AGMA Equation:

$$Q_v := 8$$

Assuming medium quality gears

$$B := \frac{\left(12 - Q_v\right)^{\frac{2}{3}}}{4}$$

$$B = 0.63$$

$$A_1 := 50 + 56(1 - B)$$

$$A_1 = 70.722$$

$$C_v := \left(\frac{A_1}{A_1 + \sqrt{200\cdot\frac{V_{t3}}{1\frac{m}{s}}}}\right)^B$$

Dynamic factor

$C_v = 0.795$

$F_w := 16mm$ Face width

$C_a := 1.25$ Application factor assuming moderate shock

$C_s := 1$ Standard size factor

$C_p := 191MPa^{0.5}$ Elastic coefficient for a steel pinion gear

$C_m := 1.6$ Load distribution factor when face width is 16mm

$C_f := 1$ Standard surface finish

$I := \frac{1}{2}(\sin(\phi)\cdot\cos(\phi))\frac{N_4}{N_4 - N_3}$ Geometric factor for an internal gear mesh

$I = 0.268$

$\sigma_c := C_p\cdot\sqrt{\frac{W_t\cdot C_a\cdot C_m\cdot C_s\cdot C_f}{F_w\cdot I\cdot d_{p.4}\cdot C_v}}$ Contact stress on ring gear

$\sigma_c = 6.296 \times 10^8\,Pa$

Contact Stress Factor of Safety Calculations:

$C_T := 1$ Temperature factor assuming a temperature less than 250°F

$C_R := 0.85$ Reliability factor assuming 90% reliability

$C_H := 1$ Hardness ratio factor - same gear material

$N_C := 10^6$ One million cycles of use

$C_L := 2.466 N_C^{-0.056}$ Life factor at one million cycles

$C_L = 1.138$

$S_{fcp} := 1200\text{MPa}$ Uncorrected surface fatigue strength assuming steel at 400 HB Grade 2

$$S_{fc} := \frac{(C_L \cdot C_H) S_{fcp}}{(C_T \cdot C_R)}$$

Corrected surface fatigue strength of ring gear

$S_{fc} = 1.606 \times 10^9\,\text{Pa}$

$$N_c := \left(\frac{S_{fc}}{\sigma_c}\right)^2$$

Contact stress factor of safety

$N_c = 6.507$

The contact stress factor of safety is 6.5 for the ring gear. This is greater than the minimum value of 4, therefore the ring gear does not fail from contact stress.

Bending Stress Calculations:

$K_a := C_a$ Application factor assuming moderate shock

$K_a = 1.25$

$K_m := C_m$ Load distribution factor when face width is 16mm

$K_m = 1.6$

$K_s := 1$ Standard size factor

$K_B := 1$ Rim thickness factor for solid disc gears

$K_I := 1$ Idler factor - ring gear is not an idler

$K_v := C_v$ Dynamic factor

$C_v = 0.795$

$J_1 := 0.46$ Geometry factor[4]

$$\sigma_b := \frac{\left(W_t \cdot K_a \cdot K_m \cdot K_s \cdot K_B \cdot K_I\right)}{F_w \cdot \text{module} \cdot J_1 \cdot K_v}$$ Bending stress on ring gear

$\sigma_b = 7.592 \times 10^8 \, \text{Pa}$

Bending Stress Factor Of Safety:

$S_{fbp} := 390\text{MPa}$ Uncorrected bending strength assuming steel at 400 HB Grade 2

$K_L := 9.4518 N_c^{-0.148}$ Life factor at a million cycles

$K_L = 7.164$

$K_T := 1$ Temperature factor assuming a temperature less than 250°F

$K_R := 0.85$ Reliability factor assuming 90% reliability

$$S_{fb} := \frac{K_L \cdot S_{fbp}}{K_T \cdot K_R}$$ Corrected bending strength assuming steel at 400 HB Grade 2

$S_{fb} = 3.287 \times 10^9 \, \text{Pa}$

$$N_b := \frac{S_{fb}}{\sigma_b}$$ Bending stress factor of safety

$N_b = 4.33$

The bending stress factor of safety is 4.3. This calculated value is greater than the minimum factor of safety of 4. Therefore, the ring gear will not fail from bending.

Conclusion:

The factor of safety for contact stress is 6.5, and the factor of safety for bending stress is 4.3. Both of these values are above the minimum factor of safety of 4. Therefore, the ring gear will not fail from bending or contact stresses. This indicates that the material and face width of the gear are appropriate for the device and will not lead to gear failure.

Shaft Design Calculations - Shaft 1 & 3

Objective:

To determine the minimum diameter of Shaft 1.

Solution Method:

The minimum diameter of the shaft can be calculated based on the strength of the material of which the shaft is made. An iterative technique will be applied to determine the minimum diameter.

Known:

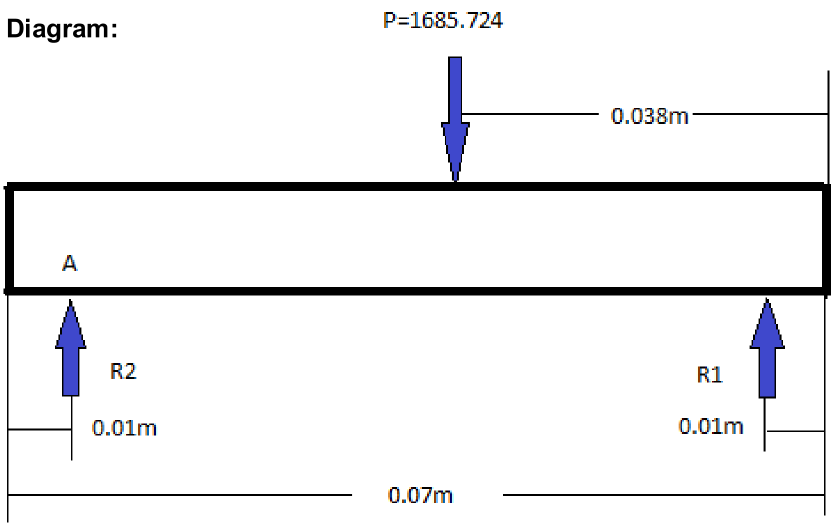
- applied force
- applied torque
- material of shaft - stainless steel
- material properties
- length of shaft

Assumptions:

- no stress concentrations
- factor of safety of 2
- mass of shaft is negligible

Note: Shaft 1 experience a much larger applied load than Shaft 3 and as a result the minimum diameter of Shaft 3 would be smaller than that of Shaft 1. Therefore the minimum diameter of Shaft 1 was deemed safe for Shaft 3 as well and no Shaft 3 minimum diameter calculations were preformed.

Free Body Diagram:



Solving Reaction Forces for Shaft 1:

$P := 1685.724N$ — Applied force due to pulley (Appendix D - Normal Pulley Force Calculations)

$\Sigma M_A := 0$ — Sum of moments about point A

$$R_1 := P \cdot \frac{0.022}{0.05}$$

$\Sigma F_y := 0$ — Sum of vertical forces

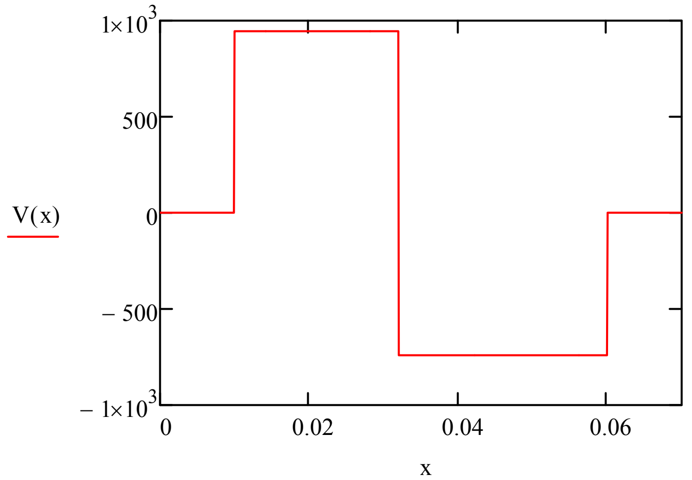
$R_2 := P - R_1$

$R_1 = 741.719\,N$

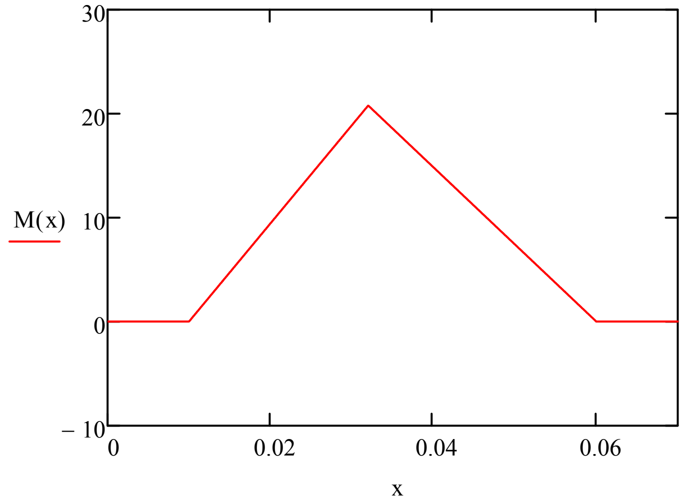
$R_2 = 944.005\,N$

Reactionary forces applied by the bearings

Shear Force Diagram:



Bending Moment Diagram:



Stainless Steel Material Properties[1]**:**

$S_{ut} := 505MPa$ Ultimate stress

$S_y := 215MPa$ Yield stress

Minimum Diameter Calculations:

$d := 20mm$	Initial diameter guess
$n := 2$	Factor of safety
$k_a := 1.1012$	Assuming shaft is ground
$k_b := 1.24\cdot\left(\frac{d}{1mm}\right)^{-0.107}$	Assuming d is greater than 2.79mm and less than 51 mm
$k_b = 0.9$	
$k_c := 1$	Bending dominates
$k_d := 1$	No temperature effects
$k_e := 1$	Assuming 50% reliability
$k_f := 1$	No miscellaneous effects
$S_{e.uncorrected} := 0.504\cdot S_{ut}$	Uncorrected endurance strength
$S_{e.uncorrected} = 2.545 \times 10^{8}\,Pa$	
$S_e := k_a\cdot k_b\cdot k_c\cdot k_d\cdot k_e\cdot k_f\cdot S_{e.uncorrected}$	Corrected endurance strength
$S_e = 2.522 \times 10^{8}\,Pa$	
$K_f := 1$ $K_{fs} := 1$	Assuming no stress concentrations
$T_{max} := 0N\cdot m$ $T_{min} := 0N\cdot m$	No torque applied to the shaft
$M_{max} := 20.768N\cdot m$	Maximum bending moment due to tension of the shaft
$M_{min} := -20.768N\cdot m$	Minimum bending moment due to compression of the shaft

$$T_a := \frac{T_{max} - T_{min}}{2}$$ Alternating torque

$$T_m := \frac{T_{max} + T_{min}}{2}$$ Mean torque

$$M_a := \frac{M_{max} - M_{min}}{2}$$ Alternating moment

$$M_m := \frac{M_{max} + M_{min}}{2}$$ Mean moment

$$d_1 := \left[16\frac{n}{\pi}\cdot\sqrt{\frac{4\left(K_f\cdot M_a\right)^2 + 3\left(K_{fs}\cdot T_a\right)^2}{S_e^{\,2}} + \frac{4\left(K_f\cdot M_m\right)^2 + 3\left(K_{fs}\cdot T_m\right)^2}{S_y^{\,2}}}\right]^{\frac{1}{3}}$$

$$d_1 = 0.012\,\mathrm{m}$$ Minimum shaft diameter

Conclusion:
The minimum shaft diameter of Shaft 1 is 12mm. Therefore any diameter above 13mm will not fail due to the stresses applied to the shaft, with a factor of safety of 2.

Shaft Design Calculations - Shaft 2

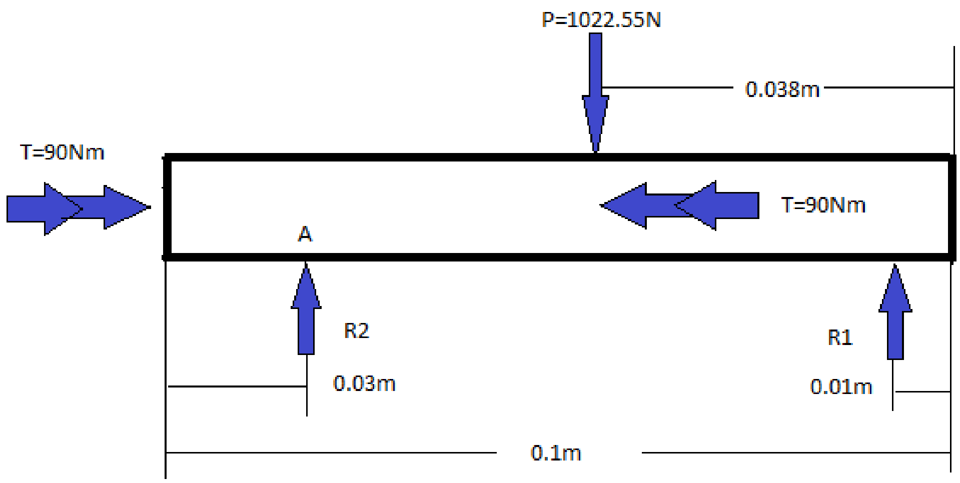
Objective:
To determine the minimum diameter of Shaft 2.

Solution Method:
The minimum diameter of the shaft can be calculated based on the strength of the material of the shaft. An iterative technique will be applied to determine the minimum diameter.

Known:
- applied force
- applied torque
- material of shaft - stainless steel
- material properties
- length of shaft

Assumptions:
- no stress concentrations
- factor of safety of 2
- mass of shaft is negligible

Free Body Diagram:



Solving Reaction Forces for Shaft 2:

$P := 1022.55N$ — Applied force due to pulley (Appendix D-Normal Pulley Force Calculations)

$\Sigma M_A := 0$ — Sum of moments about point A

$$R_1 := P \cdot \frac{0.032}{0.06}$$

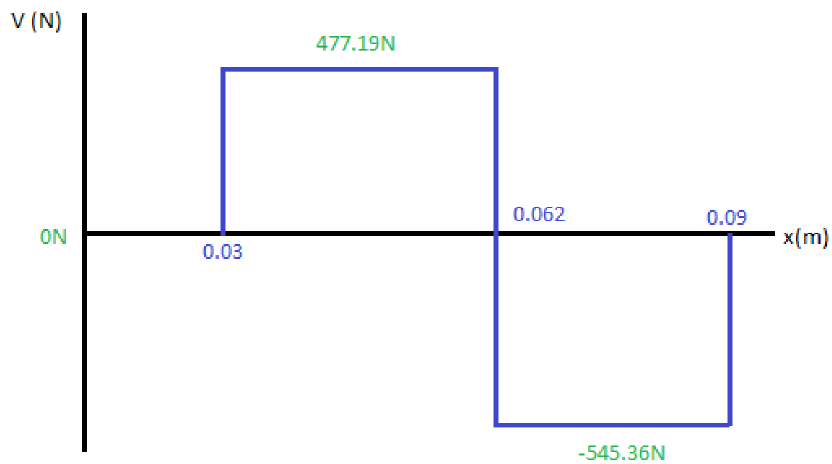
$\Sigma F_y := 0$ Sum of vertical forces

$R_2 := P - R_1$

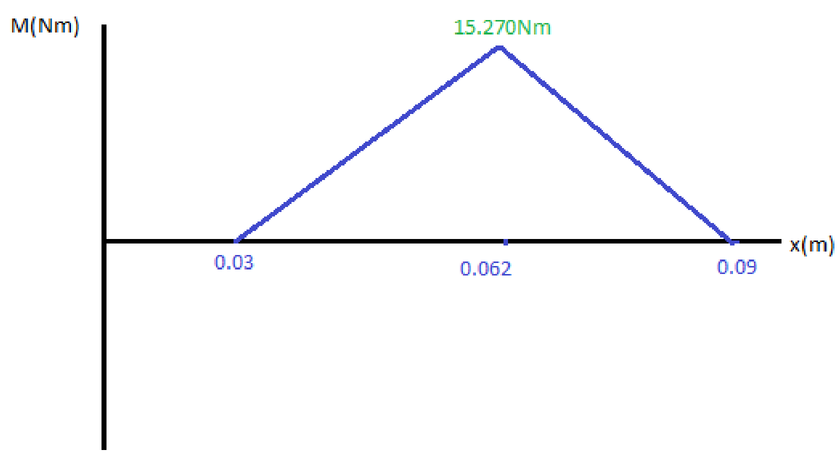
$R_1 = 545.36\,N$

$R_2 = 477.19\,N$ Reactionary forces applied by the bearings

Shear Force Diagram:



Bending Moment Diagram:



Stainless Steel Material Properties[1]:

$S_{ut} := 505MPa$ Ultimate stress

$S_y := 215MPa$ Yield stress

Minimum Diameter Calculations:

$d := 20mm$ Initial diameter guess

$n := 2$ Factor of safety

$k_a := 1.1012$ Assuming ground

$k_b := 1.24 \cdot \left(\frac{d}{1mm}\right)^{-0.107}$ Assuming d is greater than 2.79 and less than 51 mm

$k_b = 0.9$

$k_c := 1$ Bending dominates

$k_d := 1$ No temperature effects

$k_e := 1$ Assuming 50% reliability

$k_f := 1$ No miscellaneous effects

$S_{e.uncorrected} := 0.504 \cdot S_{ut}$ Uncorrected endurance strength

$S_{e.uncorrected} = 2.545 \times 10^8 \, Pa$

$S_e := k_a \cdot k_b \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S_{e.uncorrected}$ Corrected endurance strength

$S_e = 2.522 \times 10^8 \, Pa$

$K_f := 1$

Assuming no stress concentrations

$K_{fs} := 1$

$T_{max} := 90N \cdot m$ Torque applied to the shaft

$T_{min} := 0N \cdot m$

$M_{max} := 15.270N \cdot m$ Maximum bending moment due to tension of the shaft

$M_{min} := -15.270N \cdot m$ Minimum bending moment due to compression of the shaft

$$T_a := \frac{T_{max} - T_{min}}{2}$$ Alternating torque

$$T_m := \frac{T_{max} + T_{min}}{2}$$ Mean torque

$$M_a := \frac{M_{max} - M_{min}}{2}$$ Alternating moment

$$M_m := \frac{M_{max} + M_{min}}{2}$$ Mean moment

$$d_2 := \left[16 \frac{n}{\pi} \cdot \sqrt{\frac{4\left(K_f \cdot M_a\right)^2 + 3\left(K_{fs} \cdot T_a\right)^2}{S_e^{\,2}} + \frac{4\left(K_f \cdot M_m\right)^2 + 3\left(K_{fs} \cdot T_m\right)^2}{S_y^{\,2}}}\right]^{\frac{1}{3}}$$

$d_2 = 0.017\,m$ Minimum shaft diameter

<u>Conclusion:</u>

The minimum shaft diameter of Shaft 2 is 17mm, with a factor of safety of 2. Therefore, any diameter equal to or above 17mm will certainly not result in failure due to the stresses applied to the shaft.

Shaft Design Calculations - Shaft 4

Objective:
To determine the minimum diameter of Shaft 4.

Solution Method:
The minimum diameter of the shaft can be calculated based on the strength of the material of the shaft. An iterative technique will be applied to determine the minimum diameter.

Known:
- applied force
- applied torque
- material of shaft - stainless steel
- material properties
- length of shaft

Assumptions:
- no stress concentrations
- factor of safety of 2
- the mass of the shaft is negligible

Free Body Diagram:

- No reactionary forces on Shaft 4 - only the transmission of power

Stainless Steel Material Properties[1]:

$S_{ut} := 505\text{MPa}$ Ultimate stress

$S_y := 215\text{MPa}$ Yield stress

Minimum Diameter Calculations:

$d := 20\text{mm}$ Initial diameter guess

$n := 2$ Factor of safety

$k_a := 1.1012$ Assuming ground

$$k_b := 1.24 \cdot \left(\frac{d}{1mm}\right)^{-0.107}$$

Assuming d is greater than 2.79mm and less than 51 mm

$$k_b = 0.9$$

$$k_c := 1$$

Bending dominates

$$k_d := 1$$

No temperature effects

$$k_e := 1$$

Assuming 50% reliability

$$k_f := 1$$

No miscellaneous effects

$$S_{e.uncorrected} := 0.504 \cdot S_{ut}$$

Uncorrected endurance strength

$$S_{e.uncorrected} = 2.545 \times 10^{8} \, Pa$$

$$S_e := k_a \cdot k_b \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S_{e.uncorrected}$$

Corrected endurance strength

$$S_e = 2.522 \times 10^{8} \, Pa$$

$$K_f := 1$$

$$K_{fs} := 1$$

Assuming no stress concentrations

$$T_{max} := 15N \cdot m$$

Max torque applied to shaft

$$T_{min} := 0N \cdot m$$

Minimum torque applied

$$M_{max} := 0N \cdot m$$

Maximum bending moment - no force apllied

$$M_{min} := 0N \cdot m$$

Minimum bending moment - no force applied

$$T_a := \frac{T_{max} - T_{min}}{2}$$

Alternating torque

$$T_m := \frac{T_{max} + T_{min}}{2}$$

Mean torque

$$M_a := \frac{M_{max} - M_{min}}{2}$$ Alternating moment

$$M_m := \frac{M_{max} + M_{min}}{2}$$ Mean moment

$$d_4 := \left[16\frac{n}{\pi}\cdot\sqrt{\frac{4\left(K_f\cdot M_a\right)^2 + 3\left(K_{fs}\cdot T_a\right)^2}{S_e^{\,2}} + \frac{4\left(K_f\cdot M_m\right)^2 + 3\left(K_{fs}\cdot T_m\right)^2}{S_y^{\,2}}}\right]^{\frac{1}{3}}$$

$$d_4 = 9.317 \times 10^{-3}\,\mathrm{m}$$ Minimum shaft diameter

Conclusion:

The minimum shaft diameter of Shaft 4 is 9.3mm, with a factor of safety of 2. Therefore, any diameter equal to or above 9.3mm will certainly not result in failure due to the stresses applied to the shaft.

Bearing Calculations

Objective:

To determine the lightest deep groove ball bearing that can be used with a 20mm bore diameter.

Solution Method:

An iterative approach will be used to determine the lightest angular contact ball bearing that can be used. The lightest bearing produced by Schaeffler Germany will first be test. Should the lightest bearing fail due to the stress applied the next will be tested until a strong enough bearing is found. Failure of the bearing will be determined by calculation of the basic load rating applied to the bearing.

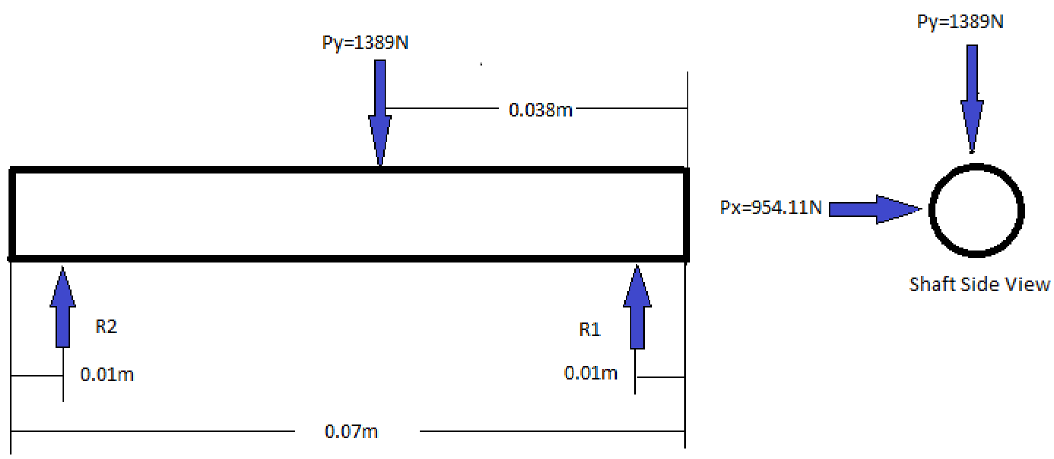
Known:

- bearing bore
- maximum dynamic loading rating
- axial and radial forces
- lightest angular contact ball bearing is 7004-B-TVP produced by Schaeffler

Assumptions:

- 20 million revolutions bearing lifetime
- radial load is applied entirely to one bearing

Free Body Diagram:



Bearing 7004-B-TVP properties[5]:

$C_0 := 7000N$ Static load rating

$C_{rmax} := 13400N$ Maximum dynamic load rating

Dynamic load rating calculations:

$F_a := \frac{954.11}{2}N$ Axial load applied to bearing

$F_r := 1389N$

Radial load assuming entire value is applied to one bearing

$$U := \frac{F_a}{C_0}$$

Axial force to static load rating ratio

$U = 0.068$

$V_1 := 1$

Rotating factor for a rotating shaft

$$e_1 := 0.26 + \frac{(U - 0.056)\cdot(0.28 - 0.26)}{(0.84 - 0.056)}$$

e value found through interpolation[5]

$e_1 = 0.26$

$$U_2 := \frac{F_a}{\left(V_1\cdot F_r\right)} = 0.343$$

Ratio between radial and axial force

$U_2 - e_1 = 0.083$

The ratio between the radial and axial force is greater than the e value therefore axial force cannot be ignored

$X := 0.56$

Radial factor[4]

$$Y := 1.71 + \frac{(U - 0.056)\cdot(1.55 - 1.71)}{(0.084 - 0.056)}$$

Thrust factor[4]

$Y = 1.641$

$F_{eff} := X\cdot V_1\cdot F_r + Y\cdot F_a = 1.56 \times 10^3\, N$

Effective force

$LD := 20$

Assuming 20 million revolutions

$$C_r := F_{eff}\cdot LD^{\left(\frac{1}{3}\right)}$$

Dynamic loading factor applied to bearing

$$C_r = 4.236 \times 10^3\,\mathrm{N}$$

$$C_{rmax} = 1.34 \times 10^4\,\mathrm{N} \qquad C_r = 4.236 \times 10^3\,\mathrm{N}$$

Conclusion:
The calculated dynamic load rating is less than the maximum load rating. Therefore, the bearing will not fail. 7004-B-TVP Angular Contact Ball Bearing is sufficient for the designed device.

Maximum Shaft Slope Analysis

Objective :
To determine if shaft deflections result in unacceptable bearing misalignment.

Solution Method:
Using the singularity method, determine shear flow, shear force, and bending moment in target shafts. Use these forces to determine the slope of the shaft along it's length, and evaluate at locations where the shaft is bound to a transmission element. Determine if slope at the bearings is less than the maximum value.

Known:
- diameter of shaft
- force applied to shaft
- material of shaft - stainless steel
- material properties
- length of shaft
- maximum slope at bearing

Assumptions:
- no stress concentrations
- mass of shaft is negligible

Note:
Shafts 1, 2, and 3 have identical diameters and support bearings. However, Shaft 1 has the greatest force applied and as a result Shaft 1 will undergo the largest deflection. Therefore calculations are completed solely on Shaft 1 because safety of Shaft 1 ensures the deflection safety of Shaft 2 and 3.

General Bearing Properties[5]:

$\theta_{max} := 0.004\text{rad}$ Maximum slope at each bearing

General Stainless Steel Properties[1]:

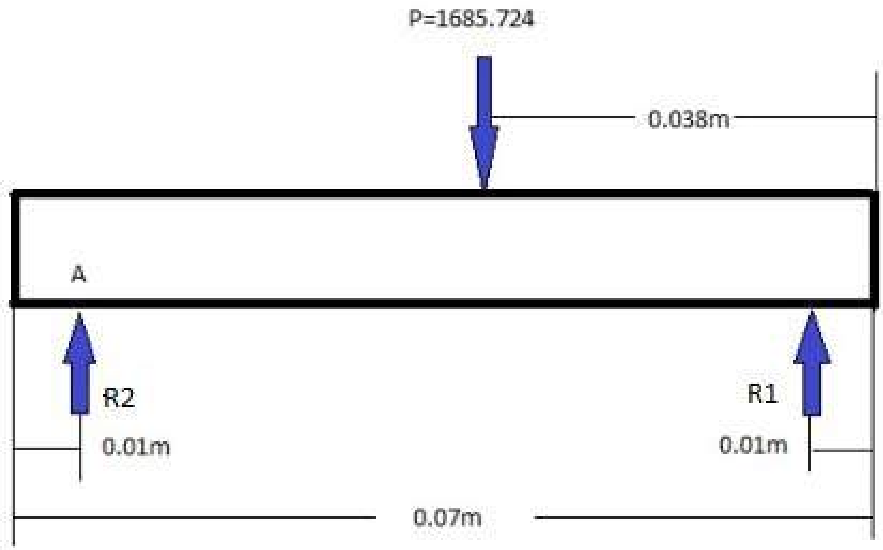
$E := 200\text{GPa}$ Modulus of elasticity

Second Moment of Inertia Calculations:

$d := 0.02\text{m}$ Diameter of Shaft 1

$I := \pi \cdot \frac{d^4}{64} = 7.854 \times 10^{-9}\text{m}^4$ Second moment of inertia

Free Body Diagram of Shaft 1:



Reactionary Force Calculations:

$P := 1685.724N$ — Applied force due to pulley (Appendix D - Normal Pulley Force Calculations)

$\Sigma M_A := 0$ — Sum of moments about point A

$$R_1 := P \cdot \frac{0.022}{0.05}$$

$\Sigma F_y := 0$ — Sum of vertical forces

$$R_2 := P - R_1$$

$R_1 = 741.719\,N$ — Bearing reaction force 1

$R_2 = 944.005\,N$ — Bearing creation force 2

Singularity Function:

$$S(x,a,n) := \begin{cases} (x-a)^n & \text{if } (x-a) > 0 \wedge n \geq 0 \\ 0 & \text{if } (x-a) \leq 0 \vee n < 0 \end{cases}$$

Input singularity code[7]

Shear Flow:

$$q(x) := R_2 \cdot S(x, 0.01, -1) - P \cdot S(x, 0.032, -1) + R_1 \cdot S(x, 0.06, -1)$$

Shear Force:

$$V(x) := R_2 \cdot S(x, 0.01, 0) - P \cdot S(x, 0.032, 0) + R_1 \cdot S(x, 0.06, 0)$$

Bending Moment:

$$M(x) := R_2 \cdot S(x, 0.01, 1) - P \cdot S(x, 0.032, 1) + R_1 \cdot S(x, 0.06, 1)$$

Slope Singularity Equation:

$$\theta(x) := \left(\frac{R_2}{2} \cdot S(x, 0.01, 2) \cdot m^2 - \frac{P}{2} \cdot S(x, 0.032, 2) \cdot m^2 + \frac{R_1}{2} \cdot S(x, 0.06, 2)\, m^2\right) \cdot \frac{1}{E \cdot I} + C_1$$

Deflection Singularity Equation:

$$y(x) := \left(\frac{R_2}{6} \cdot S(x, 0.01, 3) \cdot m^2 - \frac{P}{6} \cdot S(x, 0.032, 3) \cdot m^2 + \frac{R_1}{6} \cdot S(x, 0.06, 3) \cdot m^2\right) \cdot \frac{1}{E \cdot I} + C_1 \cdot x + C_2$$

Calculation of C1, C2 Values:

Initial guesses

$C_1 := 0$ Constant of integration 1

$C_2 := 0$ Constant of integration 2

Given

y(0.01)=0 Deflection at bearing 1 must be 0

y(0.06)=0 Deflection at bearing 2 must be 0

$$\left(\frac{R_2}{6} \cdot S(0.01, 0.01, 3) \cdot m^2 - \frac{P}{6} \cdot S(0.01, 0.032, 3) \cdot m^2 + \frac{R_1}{6} \cdot S(0.01, 0.06, 3)\, m^2\right) \cdot \frac{1}{E \cdot I} + C_1 \cdot 0.01 + C_2 =$$

$$\left(\frac{R_2}{6} \cdot S(0.06, 0.01, 3) \cdot m^2 - \frac{P}{6} \cdot S(0.06, 0.032, 3) \cdot m^2 + \frac{R_1}{6} \cdot S(0.06, 0.06, 3) \cdot m^2\right) \cdot \frac{1}{E \cdot I} + C_1 \cdot 0.06 + C_2 =$$

$$C_{1new} := \mathrm{Find}(C_1) = -1.394 \times 10^{-4}$$

$$C_{2new} := -0.01 \cdot C_{1new} = 1.394 \times 10^{-6}$$

Slope (in radians):

$$\theta(x) := \left(\frac{R_2}{2}\cdot S(x,0.01,2)\cdot m^2 - \frac{P}{2}\cdot S(x,0.032,2)\cdot m^2 + \frac{R_1}{2}\cdot S(x,0.06,2)\cdot m^2\right)\cdot\frac{1}{E\cdot I} + C_{1new}$$

Deflection (in meters):

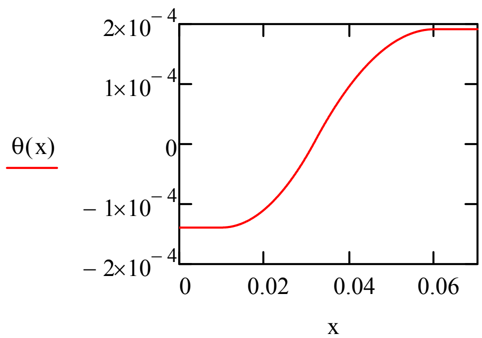
$$y(x) := \left(\frac{R_2}{6}\cdot S(x,0.01,3)\cdot m^2 - \frac{P}{6}\cdot S(x,0.032,3)\cdot m^2 + \frac{R_1}{6}\cdot S(x,0.06,3)\cdot m^2\right)\cdot\frac{1}{E\cdot I} \;... \\ + C_{1new}\cdot x + C_{2new}$$

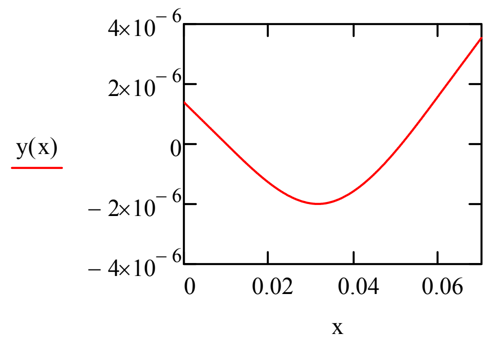
Verify that calculated C_{1new}, C_{2new} satisfy boundary conditions:

$y(0.01) = 0$

$y(0.06) = 1.626 \times 10^{-6}$

Deflection at the bearing supports should be equal to zero; Mathcad solve block has found the solution to C1 and C2 within reasonable accuracy





$\theta(0.01) = -1.394 \times 10^{-4}\cdot rad$ Slope at bearing 1

$\theta(0.06) = 1.912 \times 10^{-4}\cdot rad$ Slope at bearing 2

$\theta_{max} = 4 \times 10^{-3}\cdot rad$

Conclusion:

For Shaft 1, the largest angle between the shaft and a bearing is 1.912×10^{-4} rad. Given the acceptable shaft misalignment of 4×10^{-4} rad for angular contact ball bearings supplied by Schaeffer Germany, this deflection is acceptable. This conclusion proves that Shafts 1,2 and 3 all have acceptable deflections, since all shafts have identical diameters and support bearings and Shaft 1 has the most extreme loading.

Free Body Diagram of Shaft 4:

<u>**Note**:</u> Shaft 4 transmits power between the motor and the sun gear. Gear reactions do generate normal forces, however, since the sun gear meshes with 4 equally spaced planet gears, the sun gear will exert no radial normal force on Shaft 4. Since there were no forces acting on Shaft 4 that could cause axial deflection, B team inc did not support Shaft 4 with bearings, as the shaft ends are held in place by the planetary gear train and the motor.

<u>**Conclusion:**</u>
Shaft 4 undergoes no force which can generate slope deflection.

Angle of Twist Calculations

Objective :
To determine if Shafts 2 and 4 will have acceptable torsional deflections, as caused by the applied torque.

Solution Method:
By comparing the torsional deflection per meter of Shafts 1 and 4 to the maximum acceptable torsional deflection per meter, the safety of the shafts is determined.

Known:
- diameter of shaft
- torque applied to shaft
- material of shaft - stainless steel
- material properties
- length of shaft
- maximum torsional deflection

Assumptions:
- no stress concentrations
- mass of shaft is negligible

Note:
Shafts 2 and 4 are the only shafts which have an applied torque. Therefore the torsional deflection is calculated only for these shafts.

Maximum Torsional Deflection:

$$\phi_{max} := 3\,\frac{deg}{m}$$

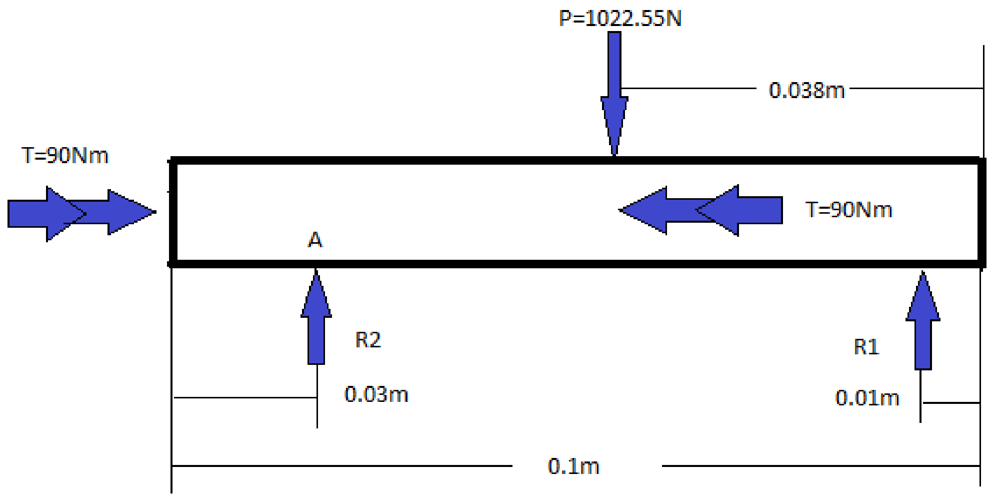
Maximum angle of twist per meter[6]

General Stainless Steel Properties[1]:

$$G_s := 86GPa$$

Shear module

Shaft 2 Calculations:



$T_2 := 90N\cdot m$ — Internal torsion acting over length L_2

$L_2 := 0.062m$ — Length of Shaft 2 undergoing internal torque

$d_2 := 20mm$ — Diameter of Shaft 2

$J_2 := \left(\frac{d_2}{2}\right)^4 \cdot \frac{\pi}{2}$ — Polar moment of inertia of Shaft 2

$\phi_2 := L_2 \cdot \frac{T_2}{G_s \cdot J_2} = 3.165 \times 10^{-3}$ — Angle of twist of Shaft 2

$\phi_2 = 0.181\ deg$

$\frac{\phi_2}{L_2} = 2.925 \frac{1}{m} deg$ — Torsional Deflection per Meter

Shaft 4 Calculations:

$T_{max4} := 15N\cdot m$ — Internal torque acting over length L_4

$L_4 := 0.1m$ — Length of Shaft 4 undergoing internal torque

$d_4 := 20mm$ — Diameter of Shaft 4

$J_4 := \frac{\pi}{2}\left(\frac{d_4}{2}\right)^4$ — Polar moment of inertia of Shaft 4

$\phi_4 := L_4 \cdot \frac{T_{max4}}{G_s \cdot J_4} = 1.11 \times 10^{-3}$ — Angle of twist of Shaft 4

$\phi_4 = 0.064 \cdot deg$

$\frac{\phi_4}{L_4} = 0.636 \frac{1}{m} \cdot deg$ — Torsional Deflection per Meter

Conclusion:
The rate of twist of Shaft 2 is 2.925°/meter and the rate of twist of Shaft 4 is 0.636°/meter. The maximum value of rate of twist is 3°/meter. Both rates of twist were below the maximum value therefore no shafts fail due to torsional deflection.

References

1. ASM Material Data Sheet. (n.d.). Retrieved November 24, 2014, from http://asm.matweb.com/search/SpecificMaterial.asp?bassnum=MQ304A
2. Kernmantle rope. (2014, October 24). Retrieved November 24, 2014, from http://en.wikipedia.org/wiki/Kernmantle_rope
3. Compact Power Motors. (n.d.). Retrieved November 24, 2014, from http://www.cpmotors.eu/fileadmin/media/downloads/datasheets/CPM90-45-3000-L_EN.pdf
4. Carey, J. (2014). Gears. In *MecE 360 Engineering Design II*. Edmonton: University of Alberta. (Figure 7-11)
5. Angular Contact Ball Bearings. (2014). In *Rolling Bearings*. Schaeffler Technologies.
6. Carey, J. (2014). Gears. In *MecE 360 Engineering Design II*. Edmonton: University of Alberta. (Table 8-3)
7. Cheung, B. (Teaching Assistant) (2014, October 1). MecE 360 Seminar. Lecture conducted from University of Alberta, Edmoton.

APPENDIX E
drawing package

design tree

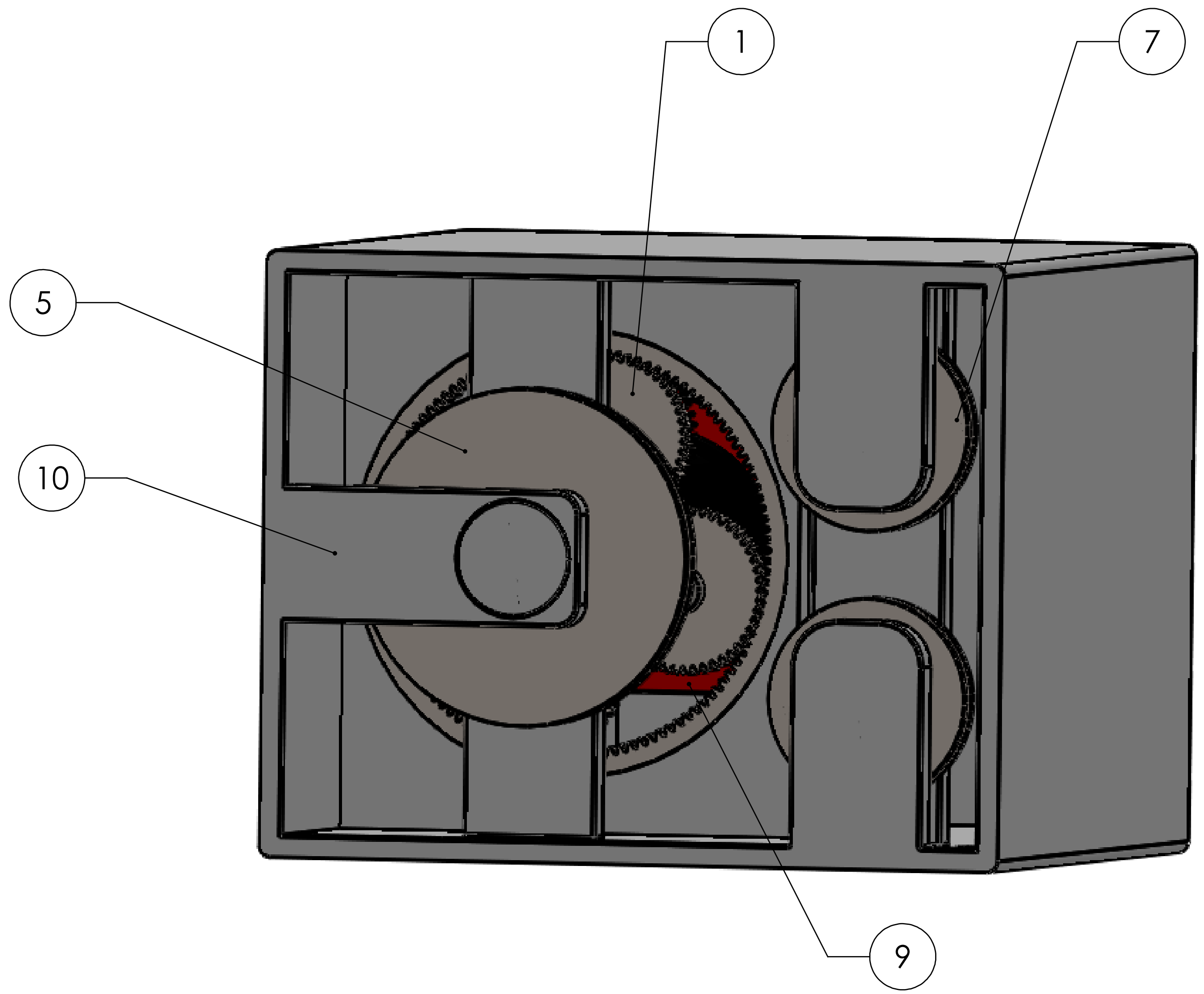
Name
Filename
Model
Drawing
Sheet #

- Main Assembly
- Sub-Assembly
- Part
- Commercial Part

Name	Filename	Model	Drawing	Sheet #
Full Assembly	Grappling Hook Assembly	Panveer Dhaliwal	Panveer Dhaliwal	1 - 3
Casing	casingquicksketch	Panveer Dhaliwal	Panveer Dhaliwal	4 - 5
Gear Train	UpdatedGearAssembly	Panveer Dhaliwal	Alexandre Sauvé	6 - 8
Sun Gear	SunGear_20mmBore	Panveer Dhaliwal	Panveer Dhaliwal	9
Planet Gear	PlanetaryGear	Panveer Dhaliwal	Panveer Dhaliwal	10
Ring Gear	RingGear	Panveer Dhaliwal	Panveer Dhaliwal	11
Crank	crank	Alexandre Sauvé	Alexandre Sauvé	12
Shaft 4	Shaft4	Panveer Dhaliwal	Panveer Dhaliwal	13
Motor	CPM90_Motor	SONCEBOZ	SONCEBOZ	14
Pulley System	PulleyAssembly	Panveer Dhaliwal	Alexandre Sauvé	15
Pulley 1	idlerpulley	Alexandre Sauvé	Alexandre Sauvé	16
Pulley 2	poweredpulley	Alexandre Sauvé	Alexandre Sauvé	17
Pulley 3	Exit pulley	Alexandre Sauvé	Alexandre Sauvé	18
Shaft 1/3	Shaft_1_3	Panveer Dhaliwal	Panveer Dhaliwal	19
Shaft 2	Shafr2	Panveer Dhaliwal	Panveer Dhaliwal	20
Bearing	20mm_Ball-Bearing	McMaster-Carr	McMaster-Carr	21

All parts and assemblies were reiewed by all team members.

ITEM NO.	Description	Material	Quantity
1	Gear Assembly	Stainless Steel	1
2	Shafts 1&3	Stainless Steel (ferritic)	1
3	Shaft 2	Stainless Steel (ferritic)	1
4	Shaft 4	Stainless Steel (ferritic)	2
5	Pulley 2	Stainless Steel (ferritic)	1
6	Pulley 3	Stainless Steel (ferritic)	1
7	Pulley 1	Stainless Steel (ferritic)	1
8	Angular Contact Bearing	Steel	6
9	CPM90_Motor	Various	1
10	Casing	HDPE	1



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LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025

SURFACE FINISH µm 0.6

DO NOT SCALE DRAWING

MATERIAL: Various

FILE NAME: Assembly_With_Kindofa_Case

DRAWN BY: Group 10

Student # 1370542

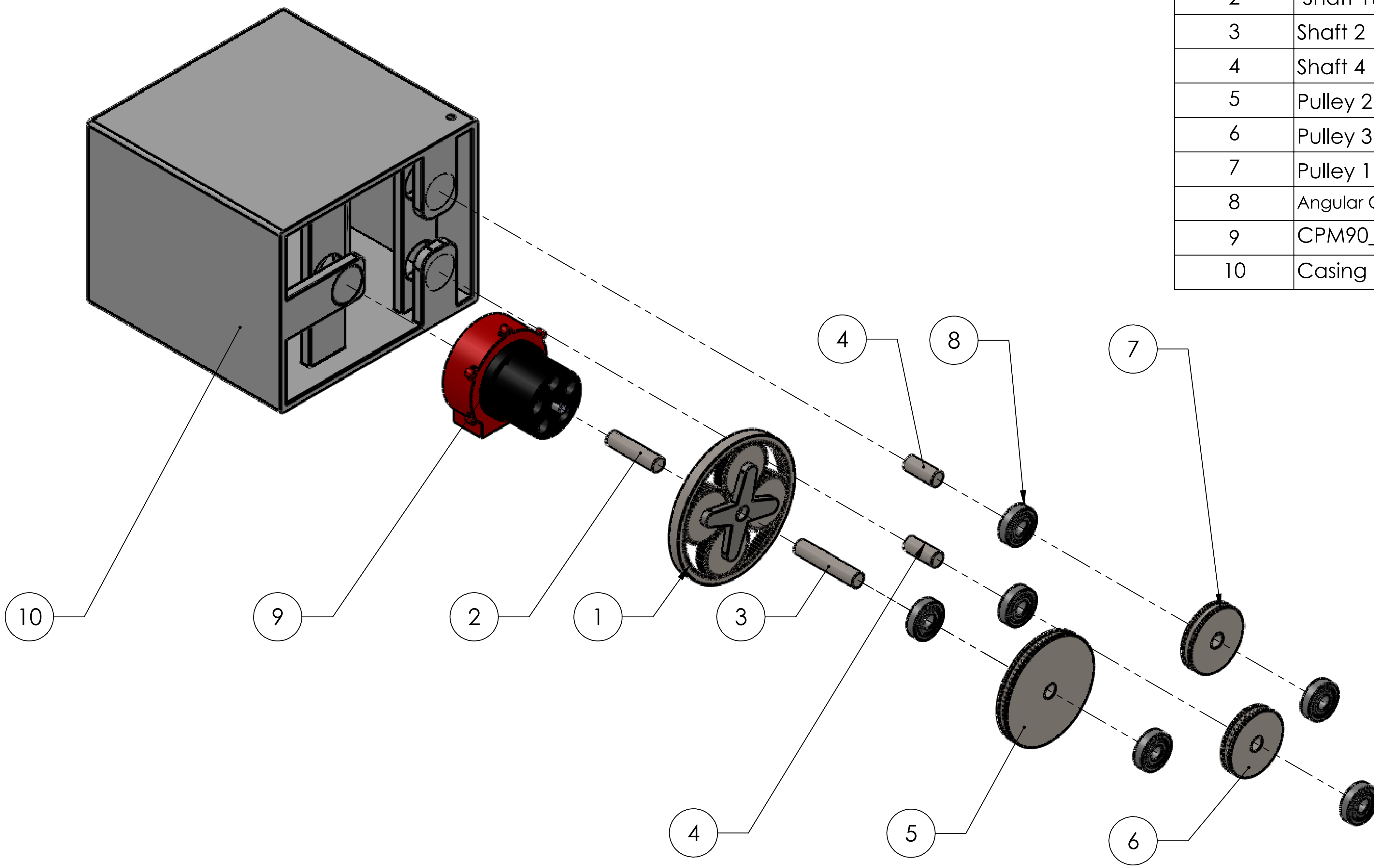
SM By Group 10

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE: Grappling Hook Assembly w/o Door

SIZE B | Mec E 360 | REV A

SCALE: 1:3 | Mass: 12624.91 | SHEET 1 OF 21



ITEM NO.	Description	Material	Quantity
1	Gear Assembly	Stainless Steel	1
2	Shaft 1&3	Stainless Steel (ferritic)	1
3	Shaft 2	Stainless Steel (ferritic)	1
4	Shaft 4	Stainless Steel (ferritic)	2
5	Pulley 2	Stainless Steel (ferritic)	1
6	Pulley 3	Stainless Steel (ferritic)	1
7	Pulley 1	Stainless Steel (ferritic)	1
8	Angular Contact Bearing	Steel	6
9	CPM90_Motor	Various	1
10	Casing	HDPE	1

Some third party components not shown:
-Battery
-Ascender & Carabiner
-Rope

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X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH µm 0.6
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MATERIAL: Various
FILE NAME: Assembly_With_Kindofa_Case

DRAWN BY: Group 10
Student # 1370542
SM By Group 10
December-03-14 7:51:56 PM
November-12-14 2:03:15 PM

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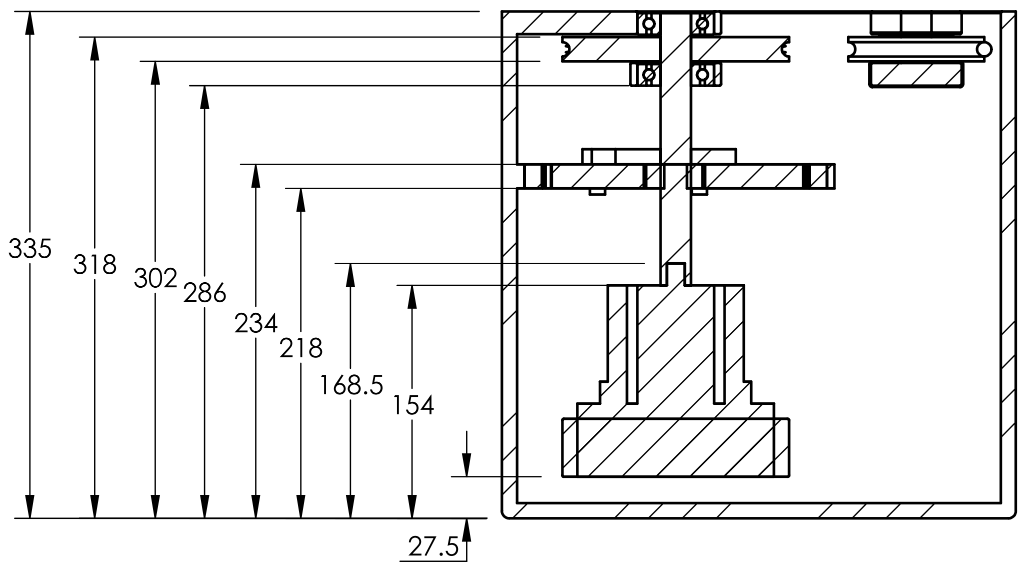
UNIVERSITY OF ALBERTA

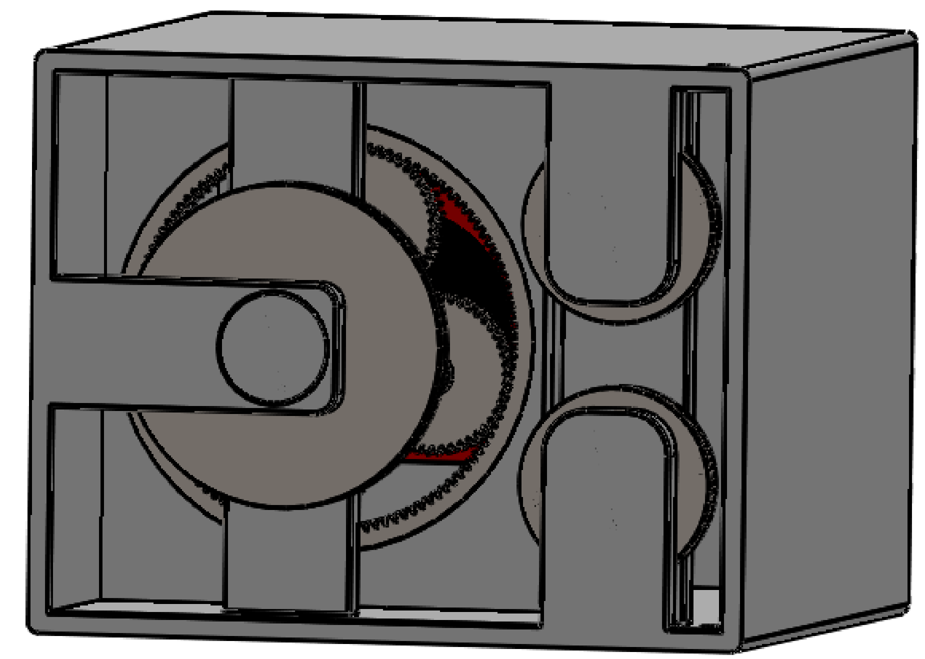
TITLE: Grappling Hook Assembly w/o Door

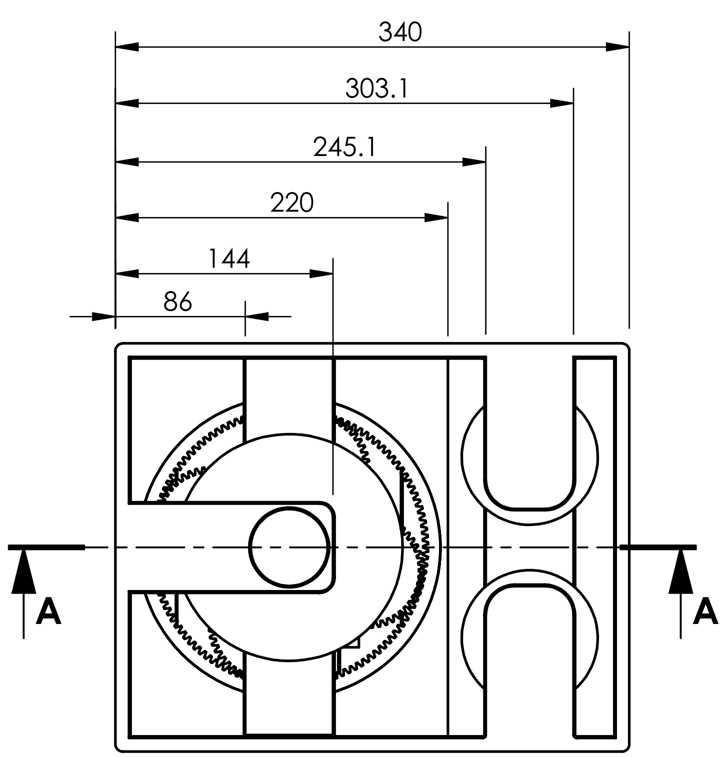
SIZE B | Mec E 360 | REV A

SCALE: 1:6 | Mass: 12624.91 | SHEET 2 OF 21



SECTION A-A





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X.X = ± 0.1
X.XX = ± 0.025

SURFACE FINISH µm 0.6

DO NOT SCALE DRAWING

DRAWN BY: Group 10

Student # 1370542

SM By Group 10

MATERIAL: Various

FILE NAME: Assembly_With_Kindofa_Case

December-03-14 7:51:56 PM
November-12-14 2:03:15 PM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

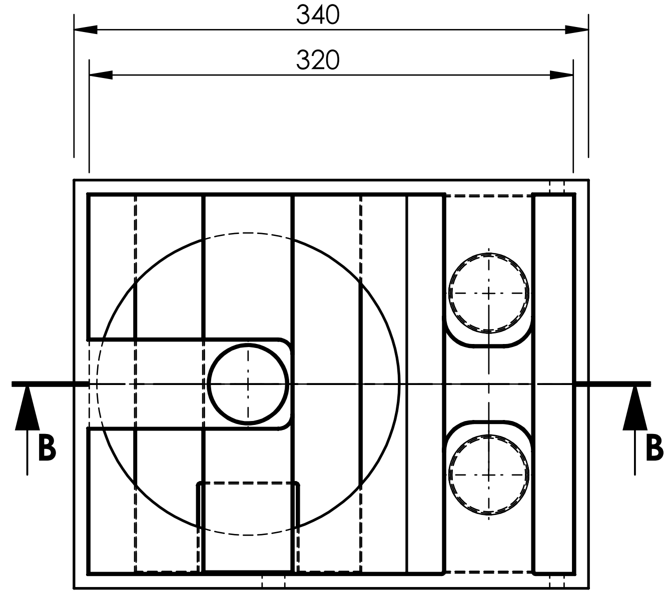
TITLE: Grappling Hook Assembly w/o Door

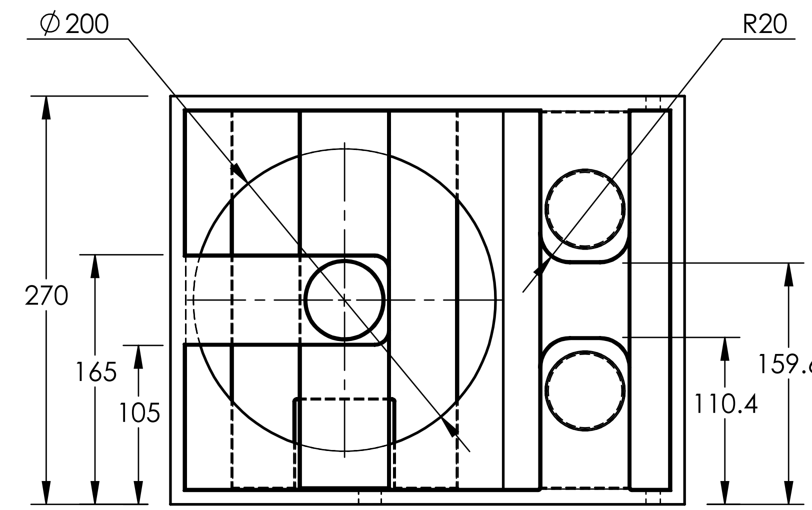
SIZE B

Mec E 360

REV A

SCALE: 1:5 | Mass: 12624.91 | SHEET 3 OF 21





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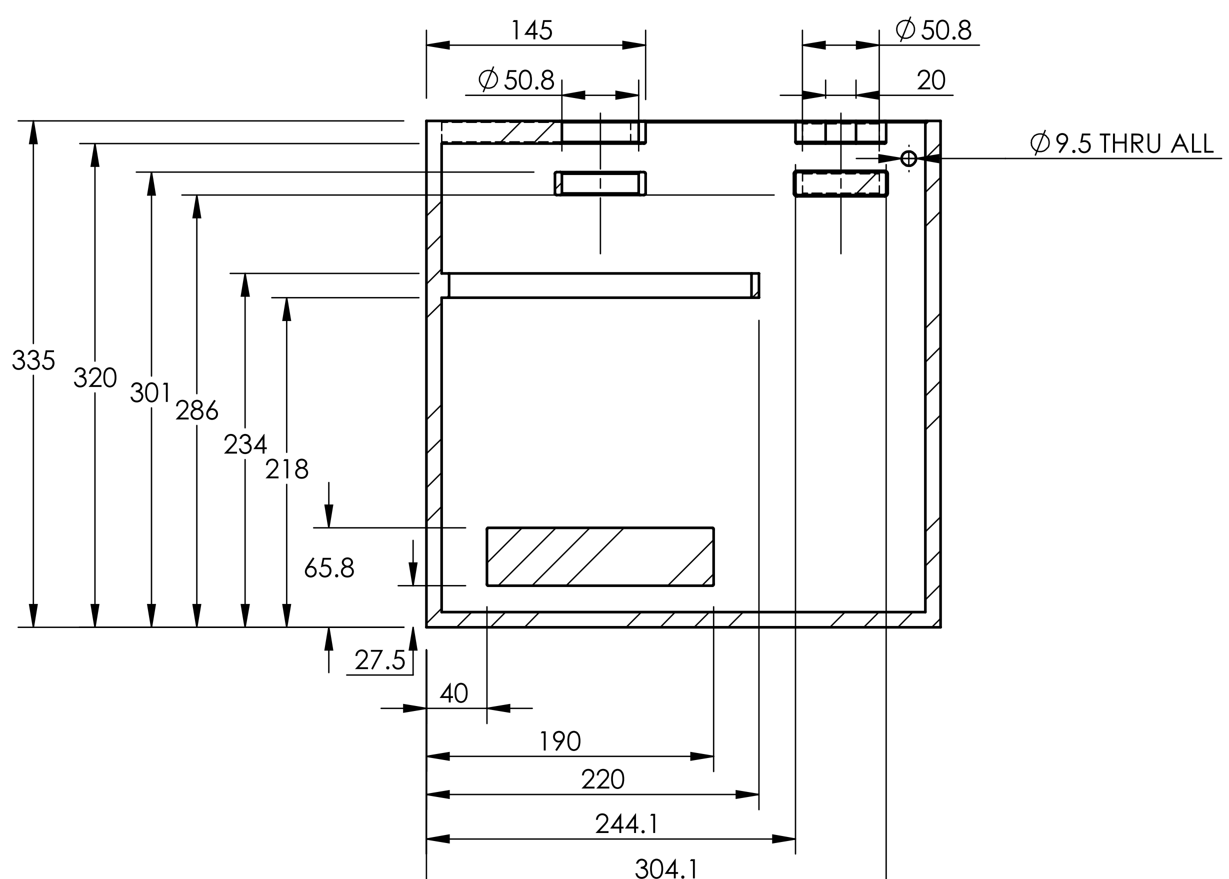

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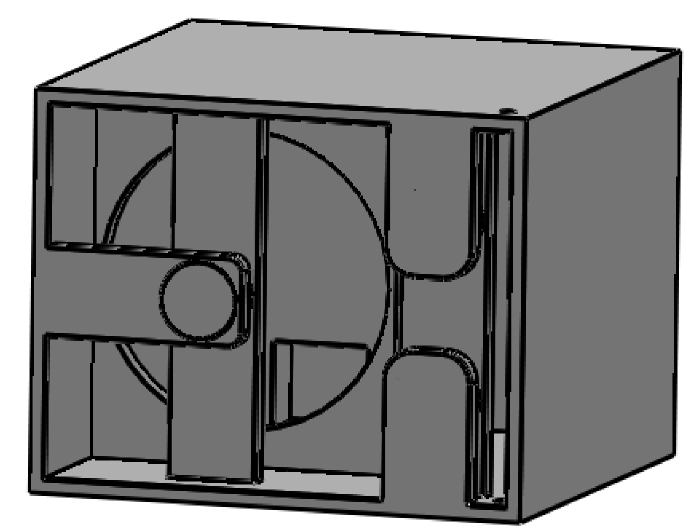

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LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH µm 0.6
DO NOT SCALE DRAWING

MATERIAL:
PE High Density Film
FILE NAME:
casingquicksketch

DRAWN BY:
Group 10
Student # 1370542
SM By Panveer Dhaliwal
December-03-14 6:29:09 PM
November-19-14 5:12:41 PM

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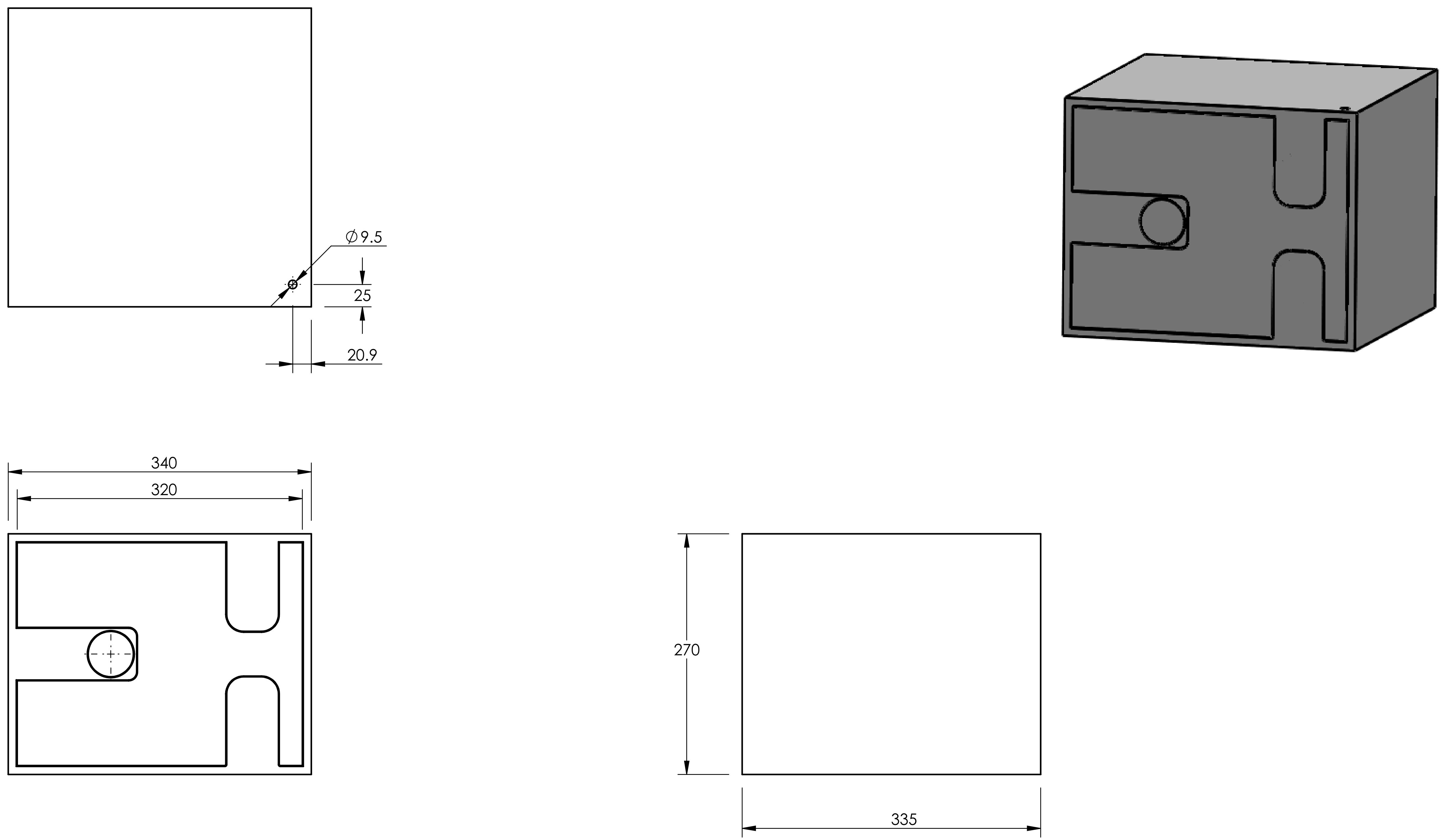
TITLE:
Grappling Hook Case

SIZE B | Grappling Hook | REV A

SCALE: 1:5 | Mass: 6598.95 | SHEET 4 OF 21



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MATERIAL: PE High Density FILE NAME: casingquicksketch_withdoor		December-03-14 7:21:08 PM November-19-14 5:12:41 PM	SIZE B · Grappling Hook · REV A SCALE: 1:5 · Mass: 7152.60 · SHEET 5 OF 21

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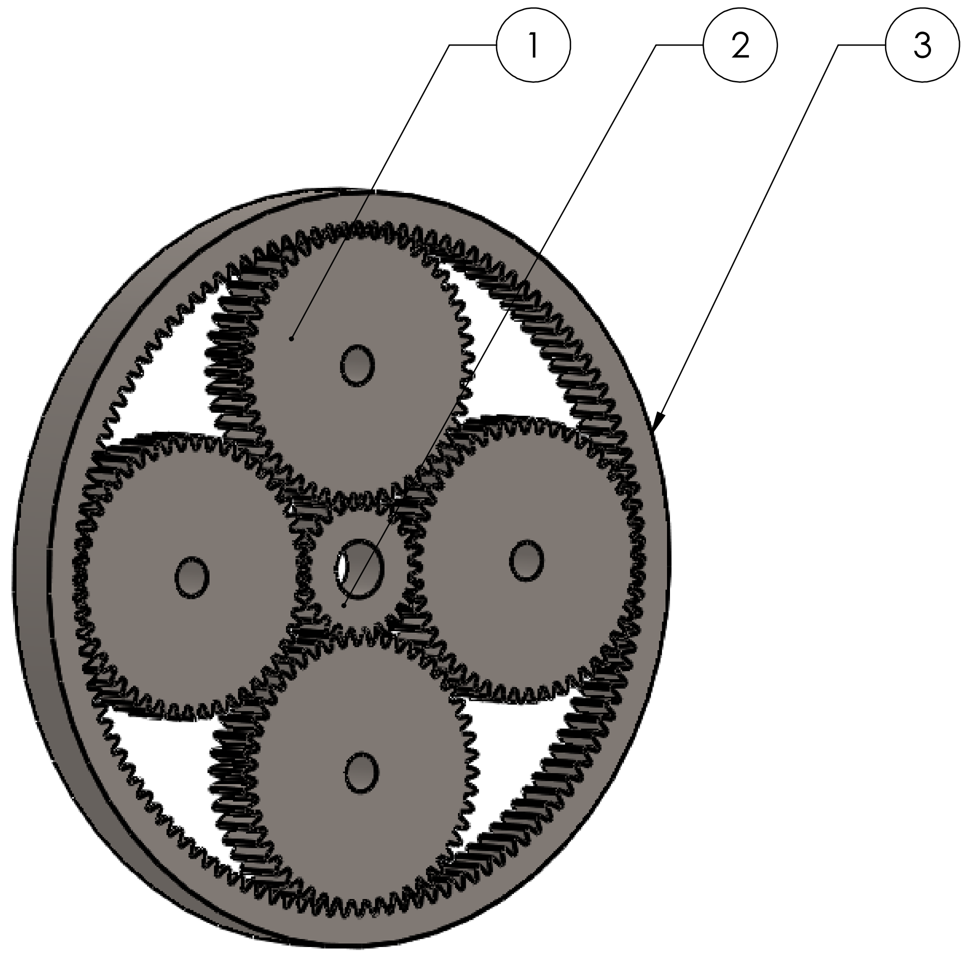
2	Sun Gear	Stainless Steel	1
3	Ring Gear	Stainless Steel	1



Crank not shown for clarity

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LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH µm 0.6
DO NOT SCALE DRAWING

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Student # 1370542
SM By Panveer Dhaliwal
December-03-14 7:25:32 PM
November-16-14 1:25:21 PM

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TITLE: Planetary Gear Train Assembly

MATERIAL: Stainless Steel
FILE NAME: UpdatedGearAssembly (1)

SIZE		REV
B	Grappling Hook	A
SCALE: 1:2	Mass: 789.85	SHEET 6 OF 21

ITEM NO.	Description	Material	QTY.
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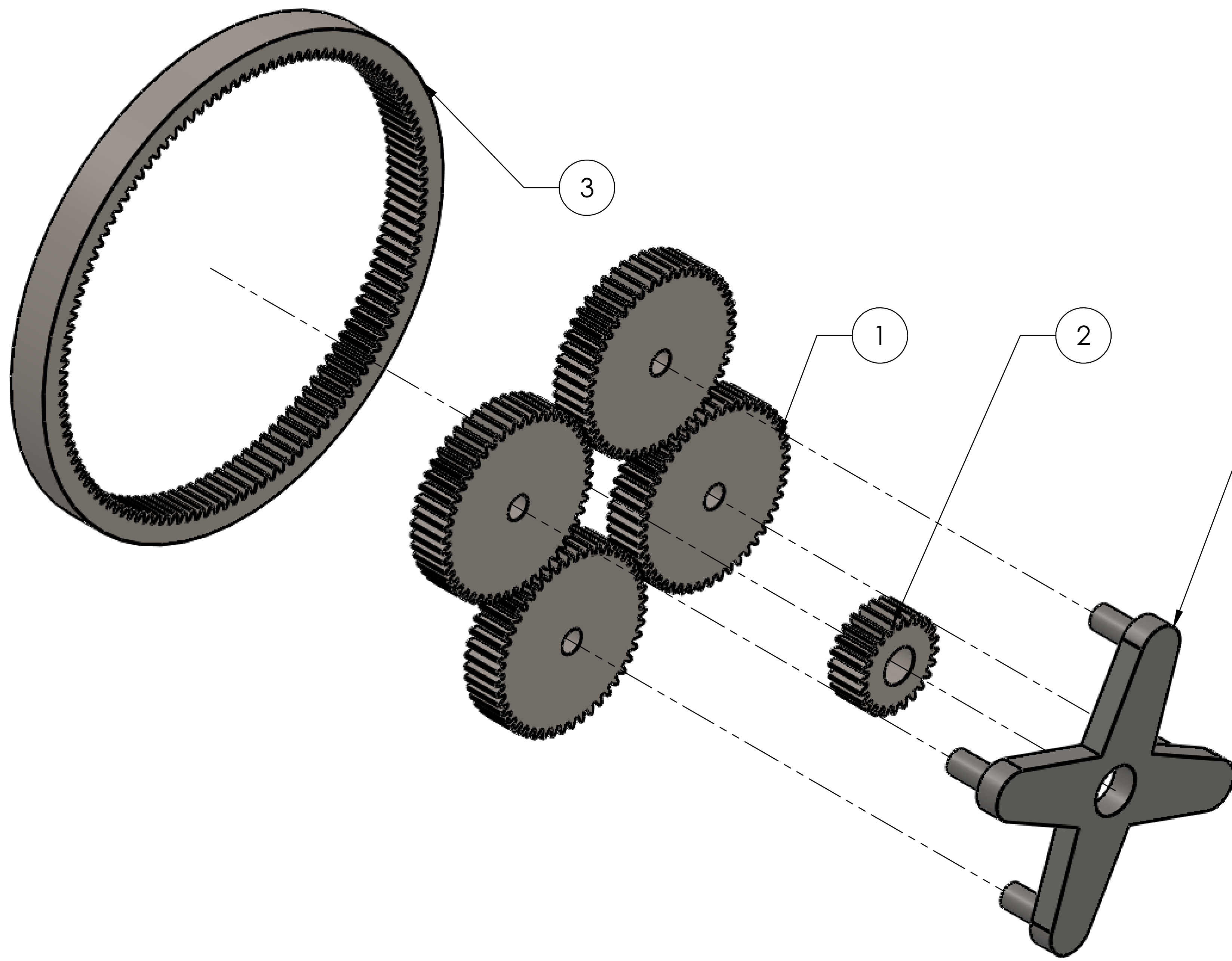
2	Sun Gear	Stainless Steel	1
3	Ring Gear	Stainless Steel	1
4	Crank	Stainless Steel	1



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LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH µm 0.6
DO NOT SCALE DRAWING

MATERIAL:
Stainless Steel
FILE NAME:
UpdatedGearAssembly (1)

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Student # 1370542
SM By Panveer Dhaliwal
December-03-14 7:25:32 PM
November-16-14 1:25:21 PM

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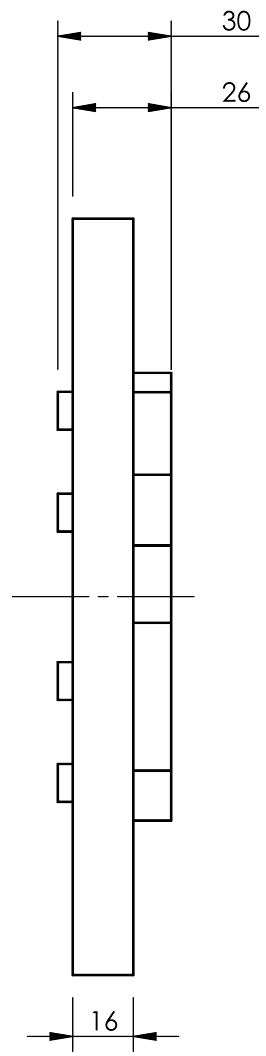
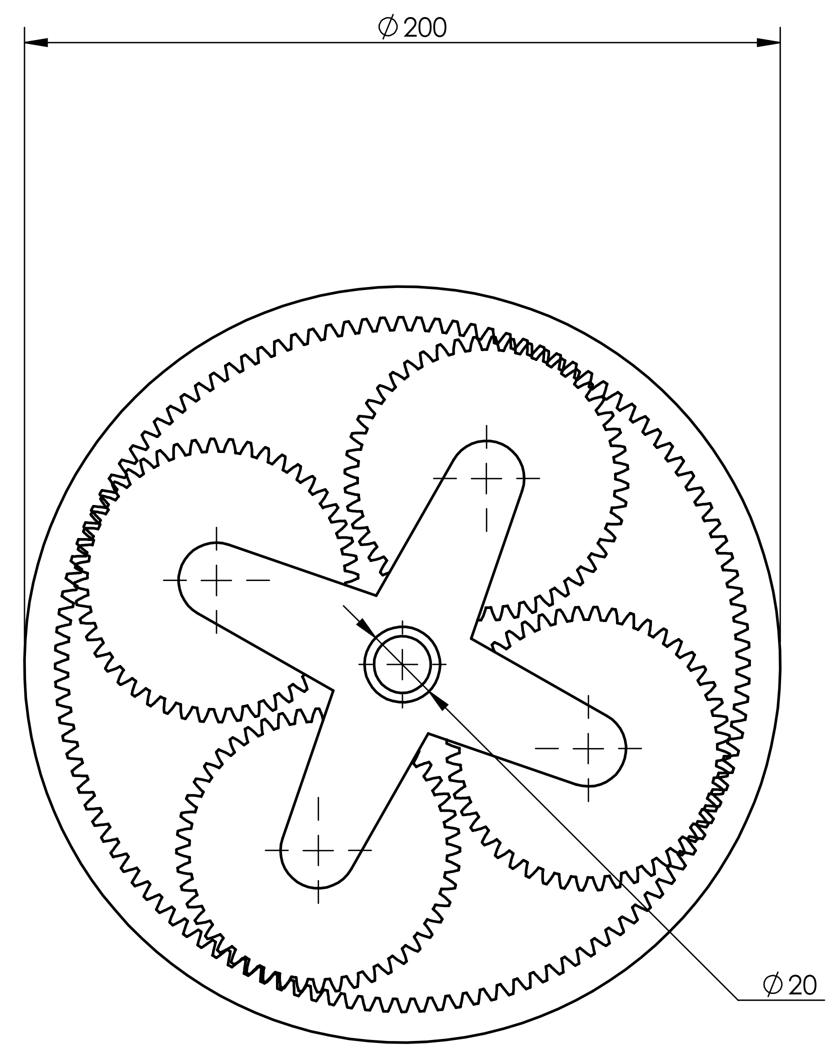
TITLE:
Planetary Gear Train Assembly

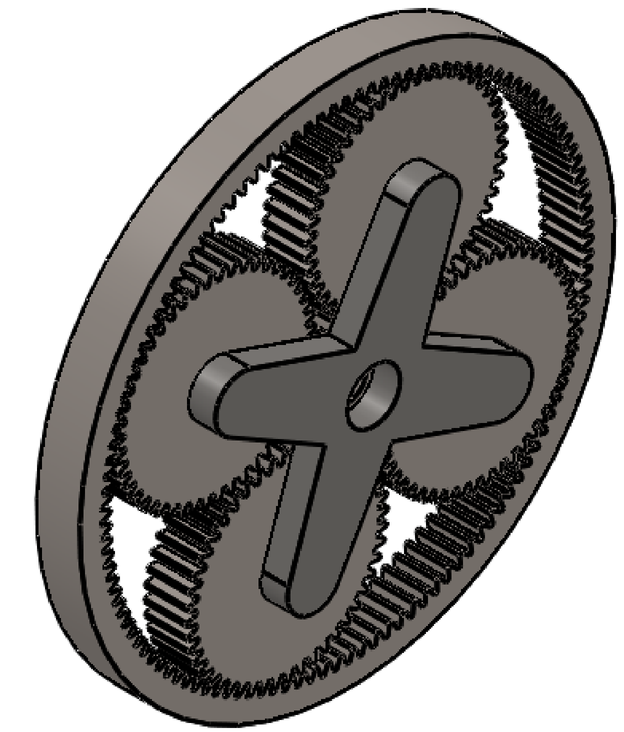
SIZE **B** Grappling Hook REV A

SCALE: 1:2 Mass: 789.85 SHEET 7 OF 21

D C B A







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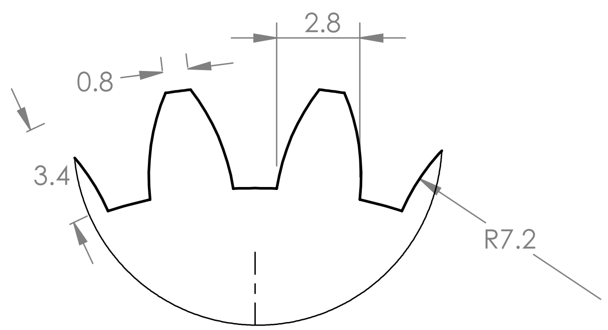
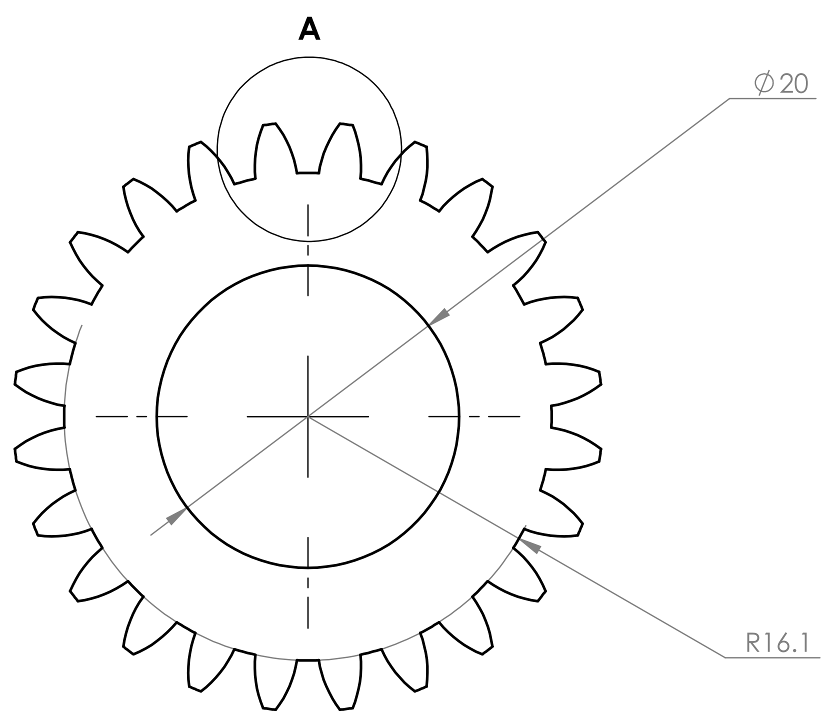
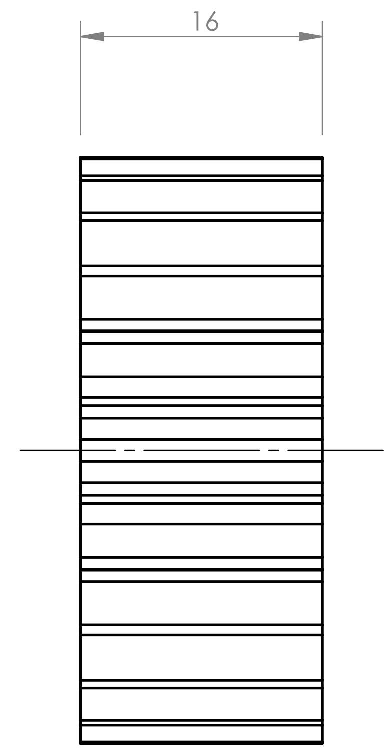
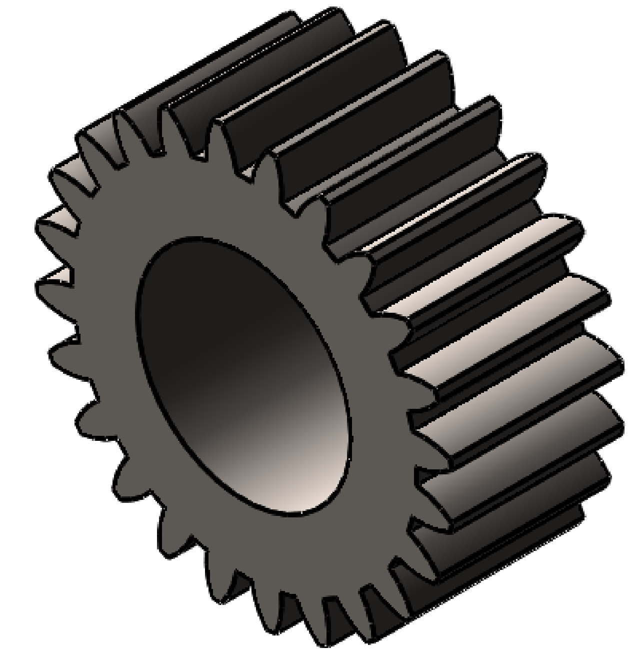
Mec E 360 Instructor: Dr. Zengtao Chen Fall 2014 Comments:	UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025 SURFACE FINISH μm 0.6 DO NOT SCALE DRAWING	DRAWN BY: Group 10 Student # 1370542 SM By Panveer Dhaliwal	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: Planetary Gear Train Assembly
MATERIAL: Stainless Steel FILE NAME: UpdatedGearAssembly (1)		December-03-14 7:25:32 PM November-16-14 1:25:21 PM	SIZE **B** Grappling Hook REV A SCALE: 2:1 Mass: 789.85 SHEET 8 OF 21



DETAIL A
SCALE 4 : 1







-Module: 1.5 mm
-Stub Tooth Gears
-Pressure Angle: 20 degrees

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Instructor:
Dr. Zengtao Chen
Comments:

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH µm 0.6
DO NOT SCALE DRAWING

MATERIAL:
Stainless Steel
FILE NAME:
SunGear_200mmBore

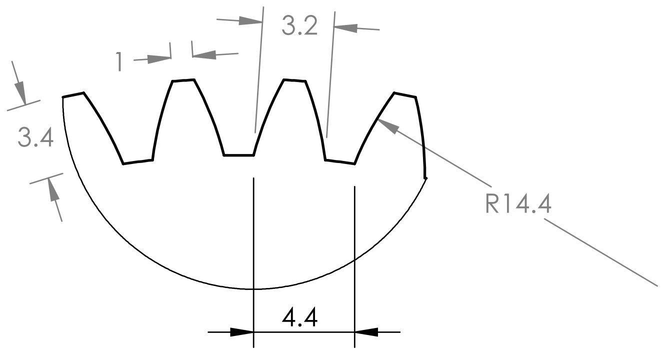
DRAWN BY:
Group 10
Student #
SM By Panveer Dhaliwal
December-03-14 6:45:36 PM
February-08-00 8:41:59 AM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

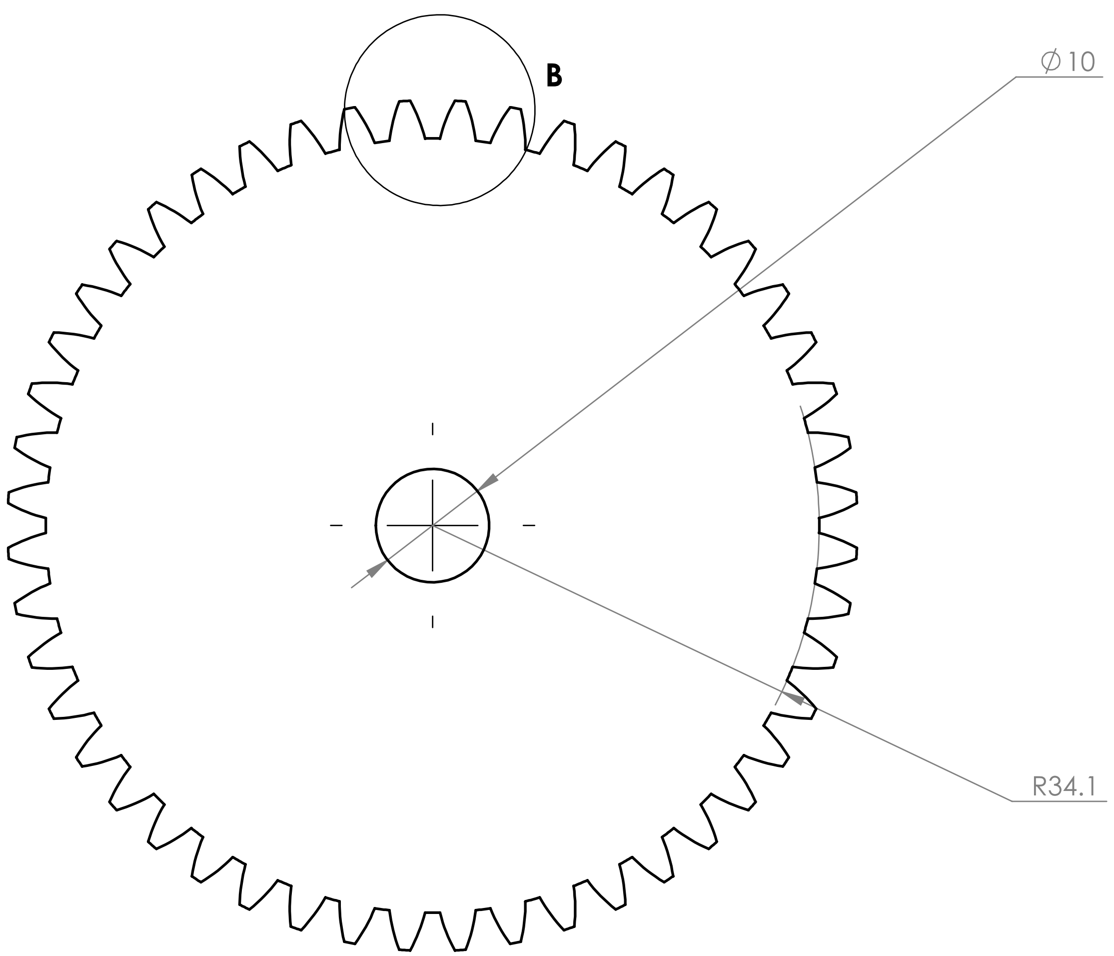
TITLE:
Sun Gear

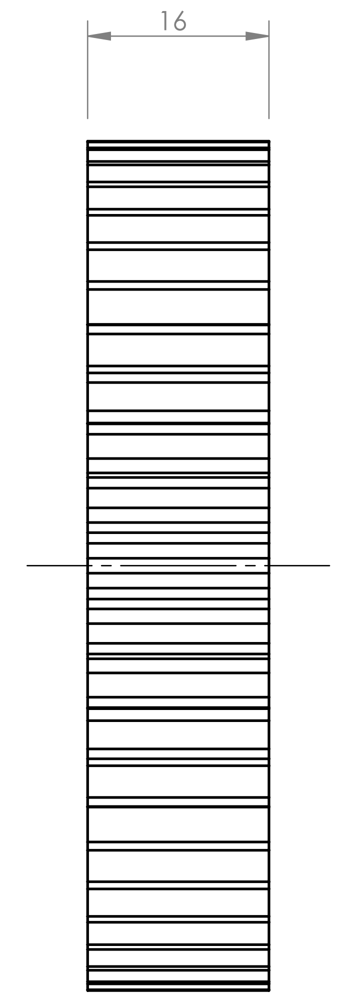
SIZE B
Grappling Hook
REV

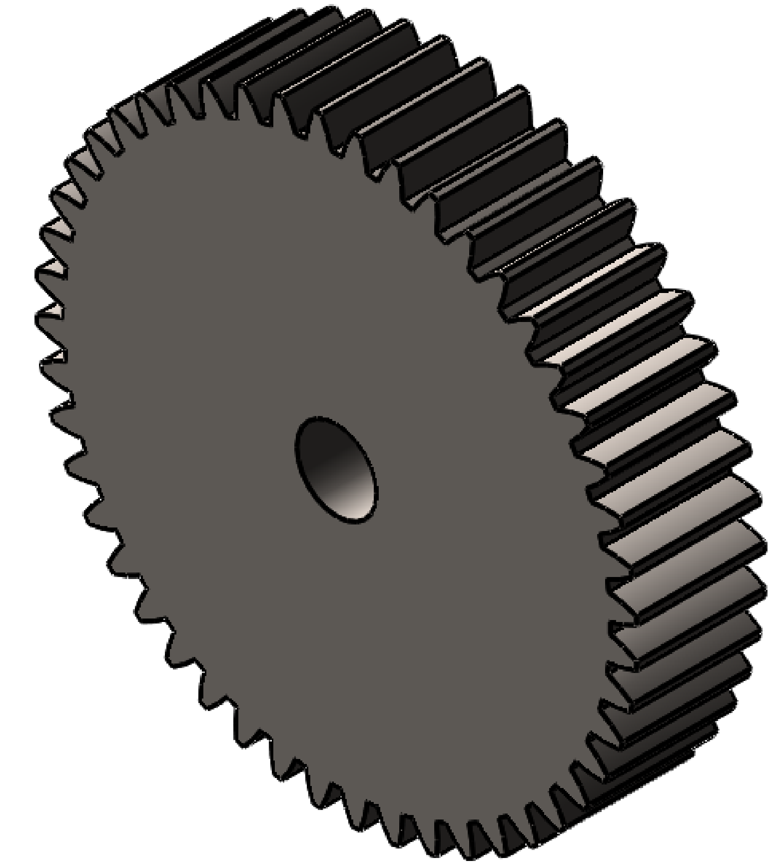
SCALE: 2:1
Mass:
SHEET 9 OF 21



DETAIL B
SCALE 3 : 1







-Module: 1.5 mm
-Stub Tooth Gears
-Pressure Angle: 20 degrees

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Instructor:
Dr. Zengtao Chen

Comments:

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DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025

SURFACE FINISH µm 0.6

DO NOT SCALE DRAWING

DRAWN BY:
Group 10

Student #

SM By Panveer Dhaliwal

December-03-14 6:45:55 PM
February-08-00 8:41:59 AM

MATERIAL:
Stainless Steel

FILE NAME:
PlanetaryGear

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

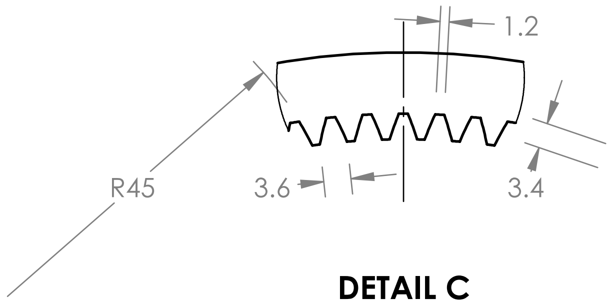
TITLE:
Planetary Gear

SIZE
B

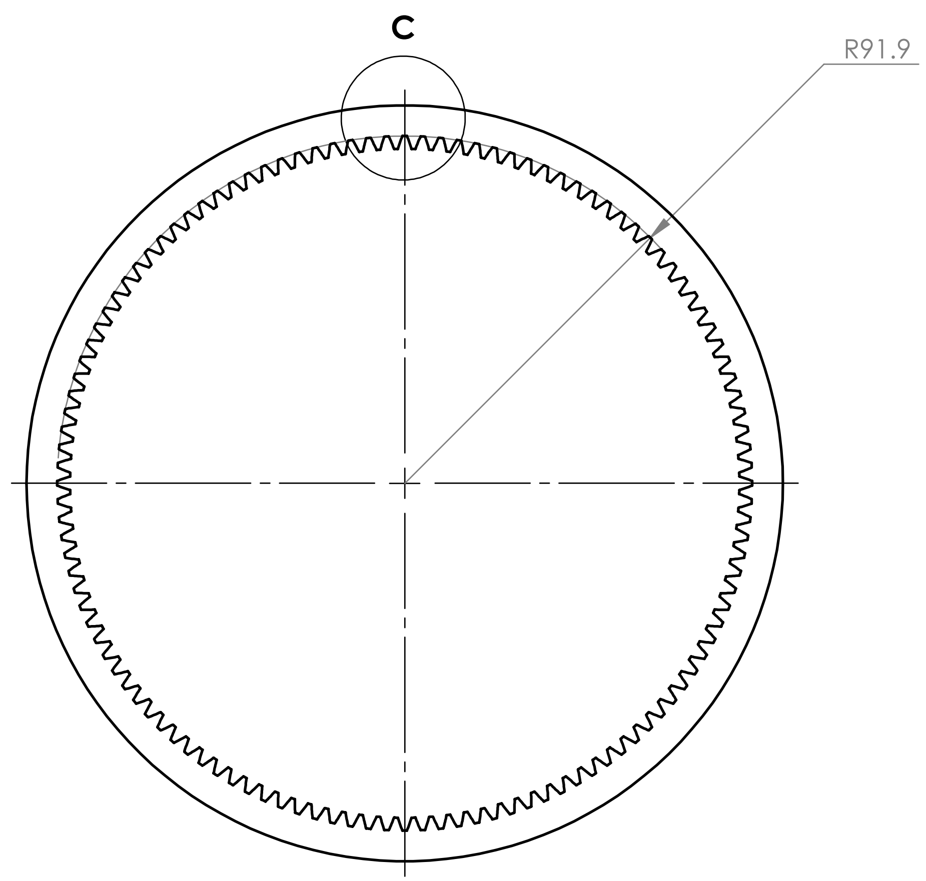
Grappling Hook

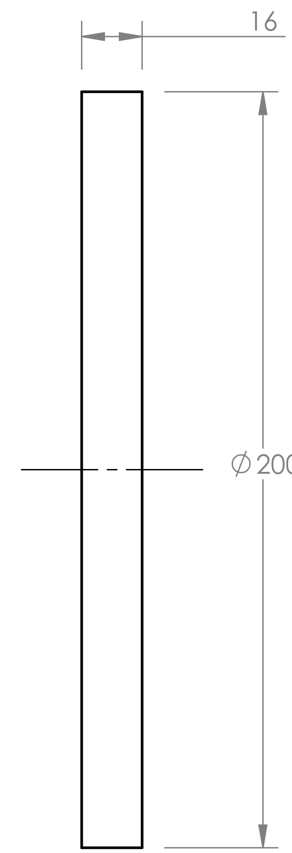
REV

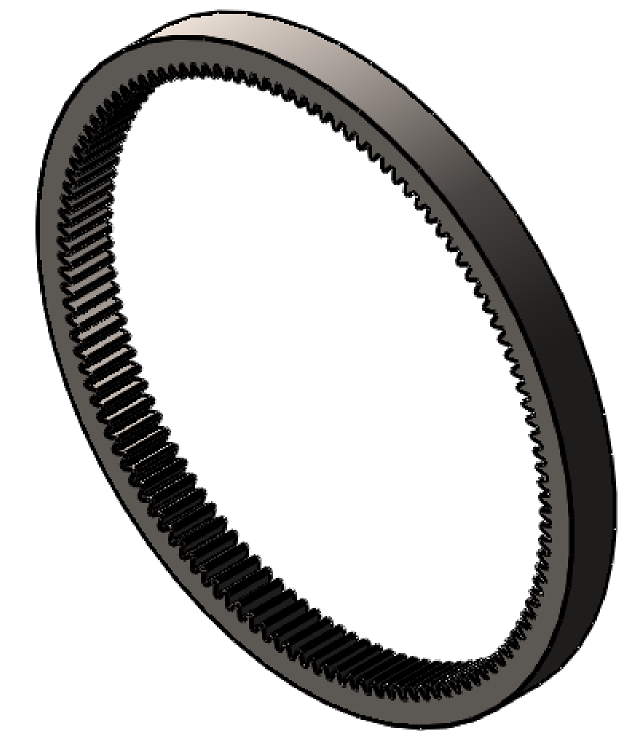
SCALE: 2:1 | Mass: | SHEET 10 OF 21



DETAIL C
SCALE 1 : 1







-Module: 1.5 mm
-Stub Tooth Gears
-Pressure Angle: 20 degrees

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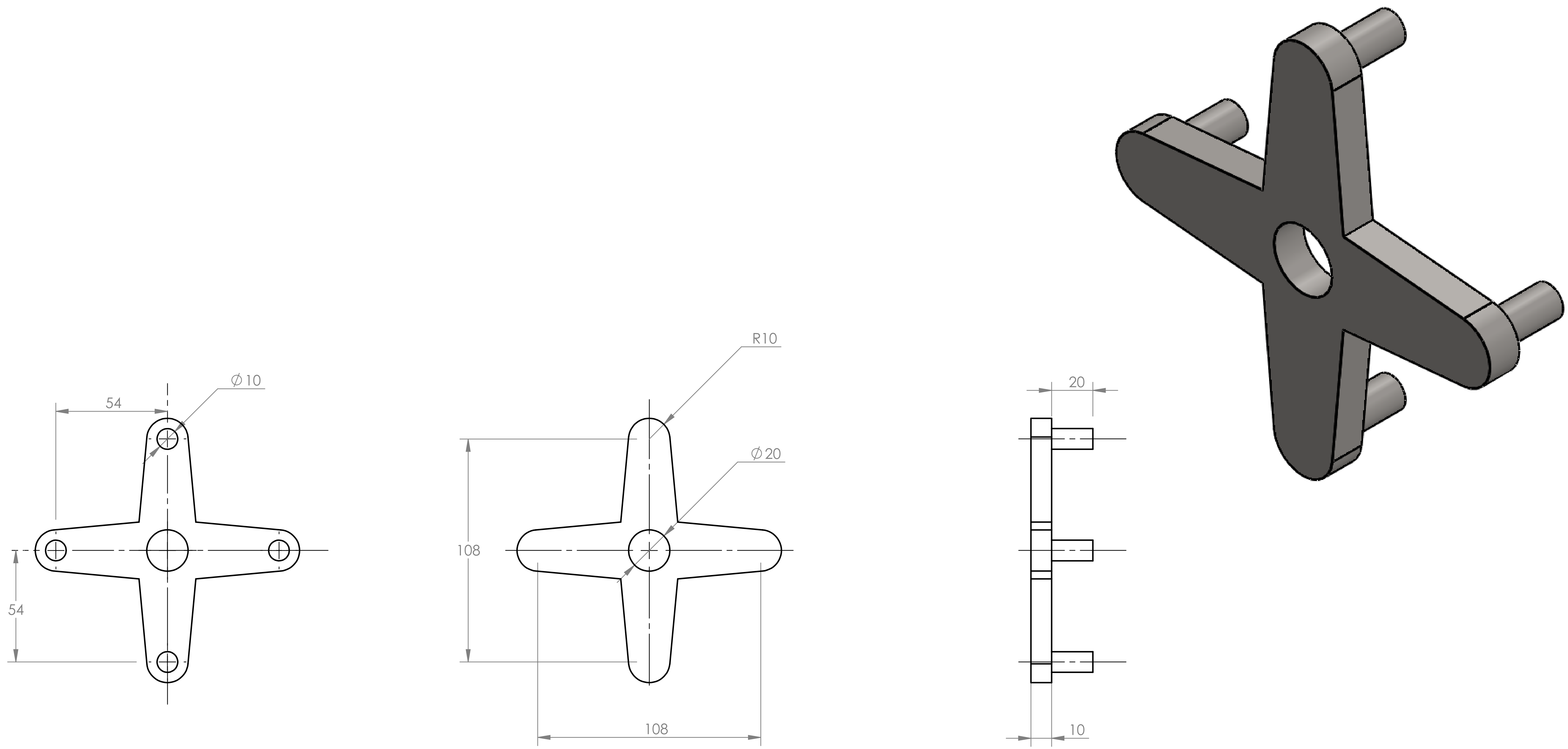
Instructor:
Dr. Zengtao Chen
Comments:

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH µm 0.6
DO NOT SCALE DRAWING

MATERIAL:
Stainless Steel
FILE NAME:
RingGear

DRAWN BY:
Group 10
Student #
SM By Panveer Dhaliwal
December-03-14 6:46:10 PM
February-08-00 8:41:59 AM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA
TITLE:
Ring Gear
SIZE B
Grappling Hook
REV
SCALE: 2:1
Mass:
SHEET 11 OF 21



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Instructor: Dr. Zengtao Chen

Comments:

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025

SURFACE FINISH µm 0.6

DO NOT SCALE DRAWING

MATERIAL: Stainless Steel

FILE NAME: crank

DRAWN BY: Group 10

Student #

SM By Alexandre Suave

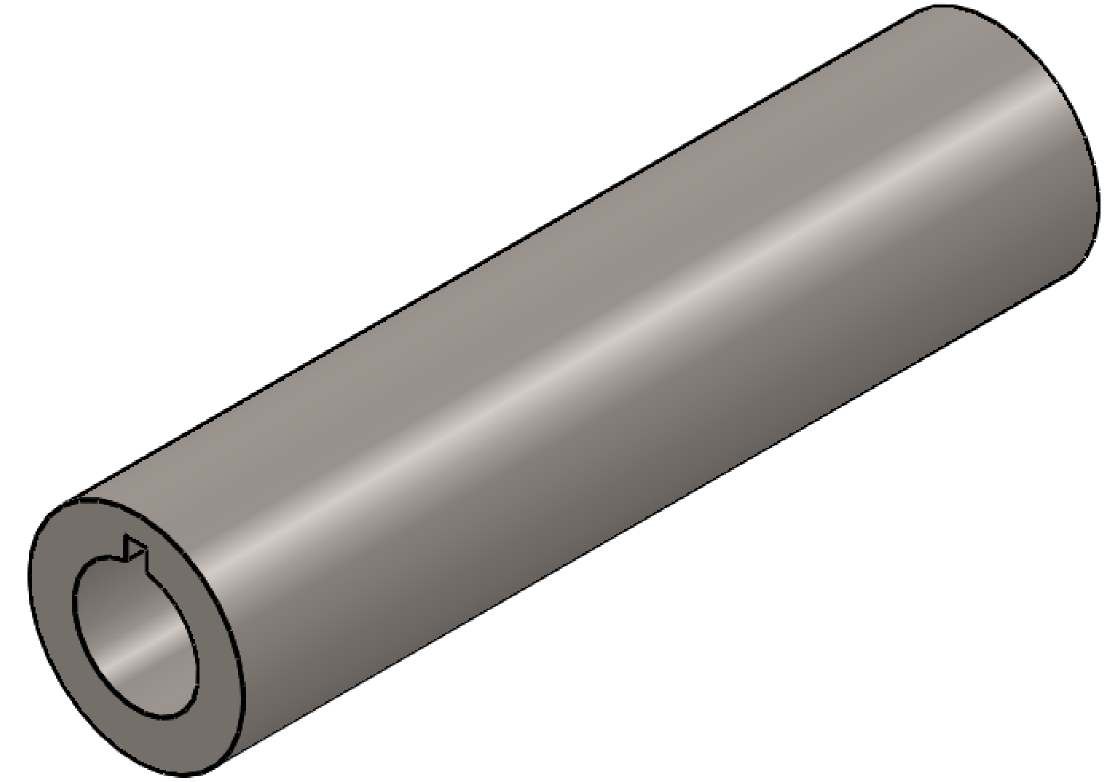
December-03-14 6:42:29 PM
November-16-14 4:10:54 PM

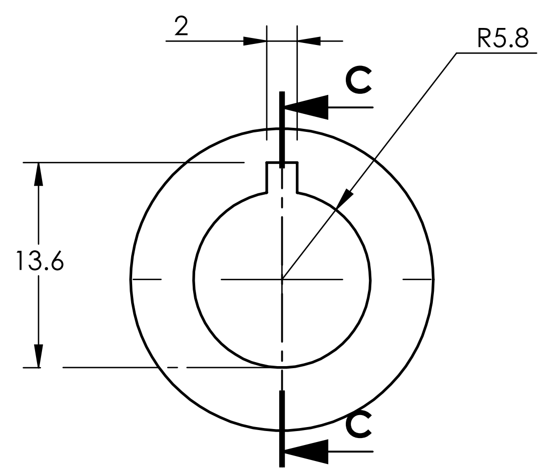
The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

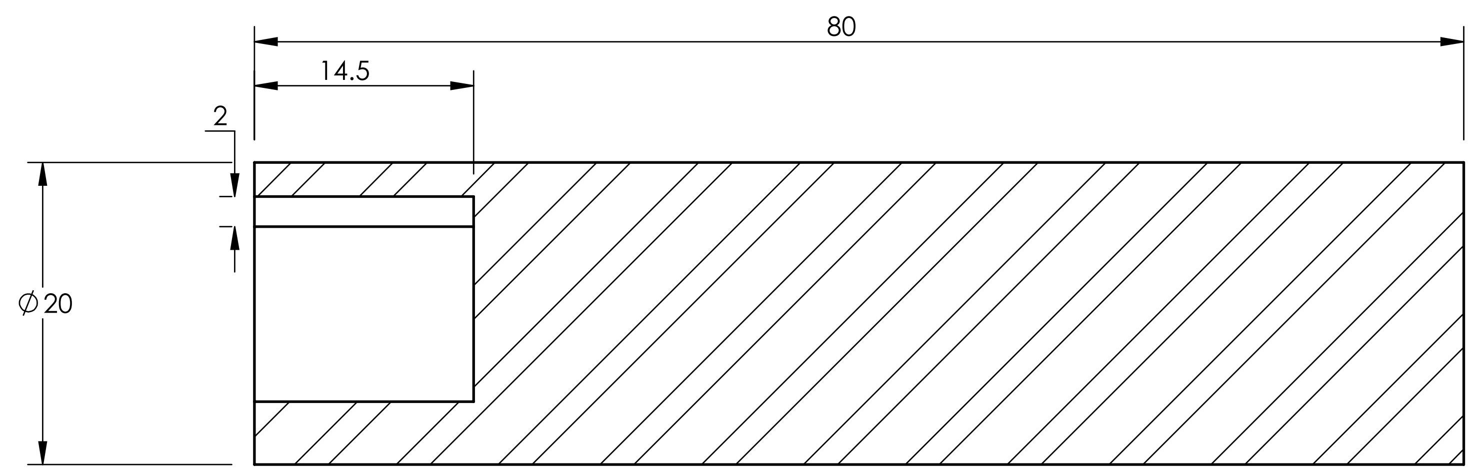
TITLE: Crank

SIZE B | Grappling Hook | REV

SCALE: 2:1 | Mass: | SHEET 12 OF 21

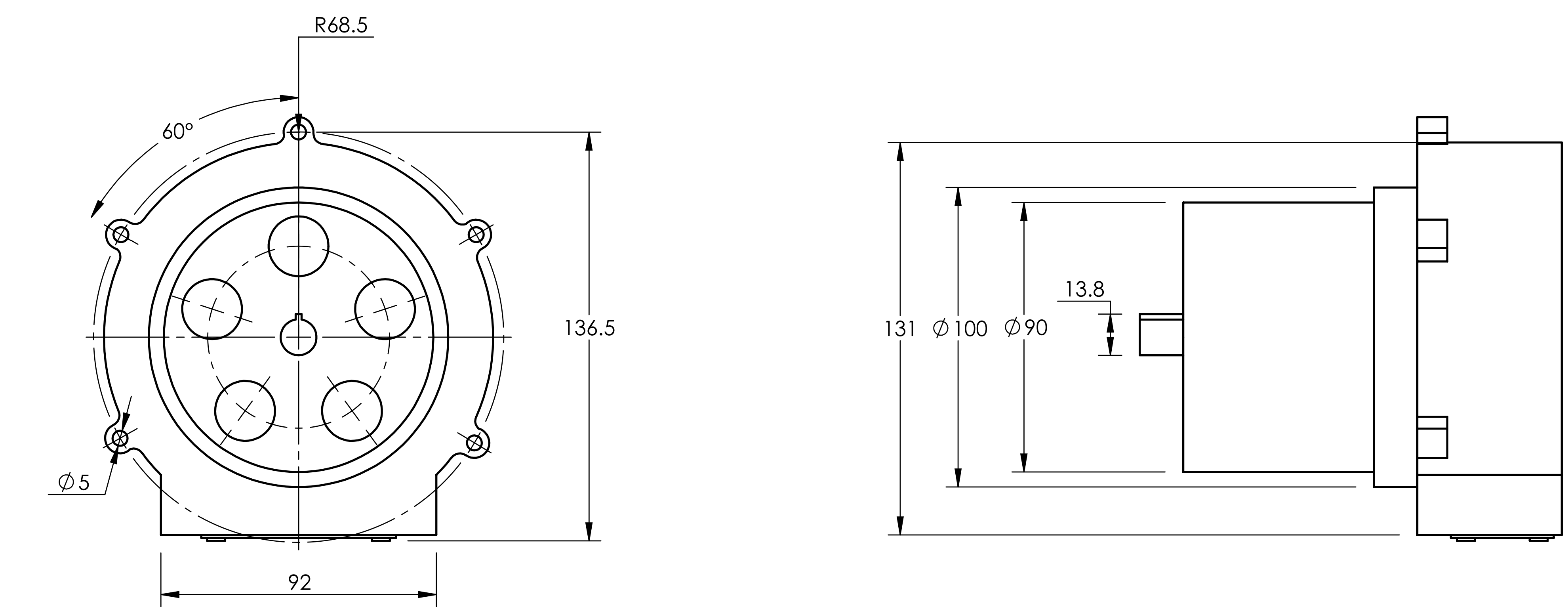


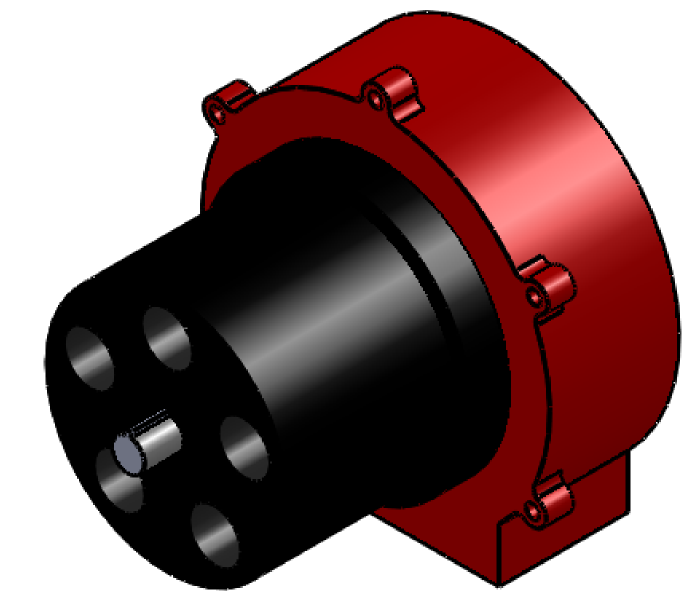


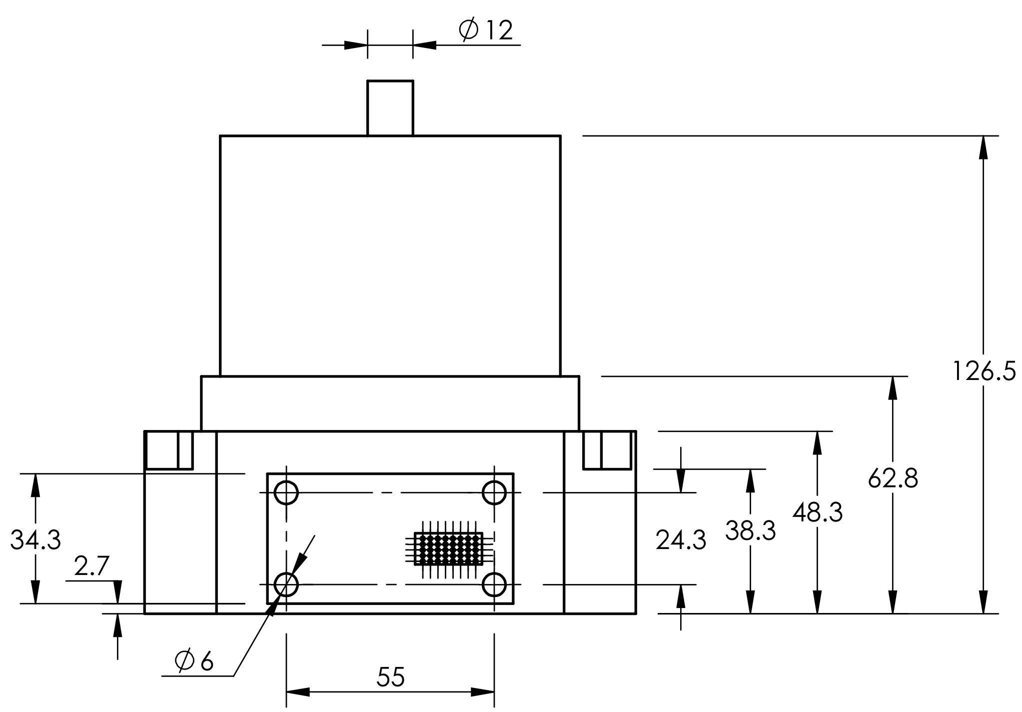


SECTION C-C

Mec E 360 Instructor: Dr. Zengtao Chen Fall 2014 Comments:	UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025 SURFACE FINISH µm 0.6 DO NOT SCALE DRAWING	DRAWN BY: Group 10 Student # SM By: Panveer Dhaliwal	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: Shaft 4
MATERIAL: Stainless Steel (ferritic) FILE NAME: Shaft_1_3		December-03-14 8:33:57 PM November-16-14 12:54:25 PM	SIZE **B** Grappling Hook REV A SCALE: 2:1 Mass: 183.49 SHEET 13 OF 21







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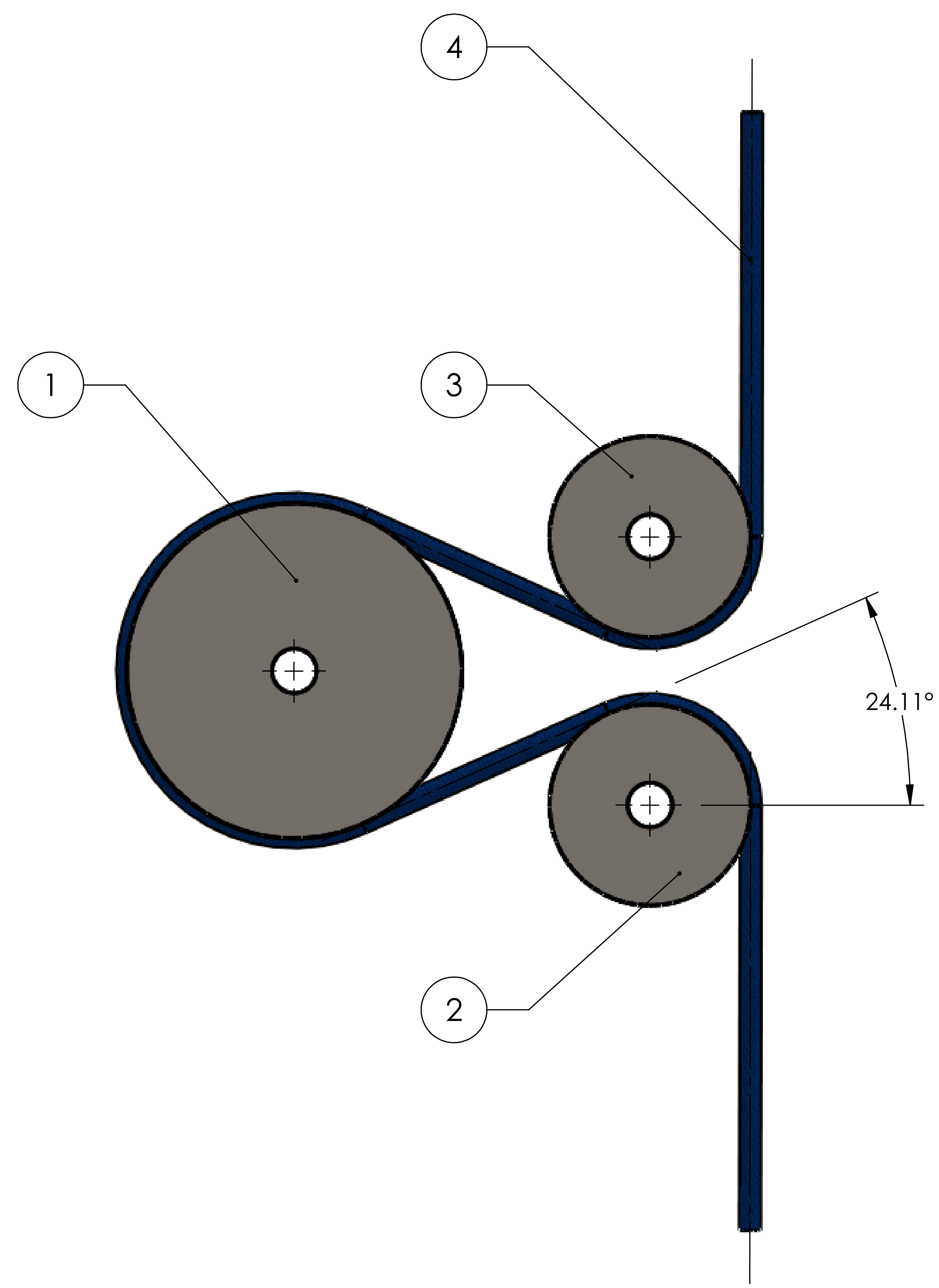
Mec E 360	UNLESS OTHERWISE SPECIFIED:	DRAWN BY: Group 10	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA
Instructor: Dr. Zengtao Chen Fall 2014	DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025	Student # 1370542	TITLE: CPM90 Motor
Comments: Purchased	SURFACE FINISH µm 0.6	SM By SONCEBOZ	
	DO NOT SCALE DRAWING		SIZE B Grappling Hook REV A
MATERIAL: Various		December-03-14 6:48:52 PM November-17-14 12:27:49 PM	SCALE: 2:1 Mass: 1072.81 SHEET 14 OF 21
FILE NAME: CPM90_Motor			

ITEM NO.	Description	Material	QTY.
1	Pulley 2	Stainless Steel (ferritic)	1
2	Pulley 3	Stainless Steel (ferritic)	1
3	Pulley 1	Stainless Steel (ferritic)	1
4	Rope	Nylon	1



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Instructor:
Dr. Zengtao Chen
Comments:

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH µm 0.6
DO NOT SCALE DRAWING

DRAWN BY:
Group 10
Student #
SM By Alexander Suave
December-03-14 7:45:08 PM
November-12-14 2:03:15 PM

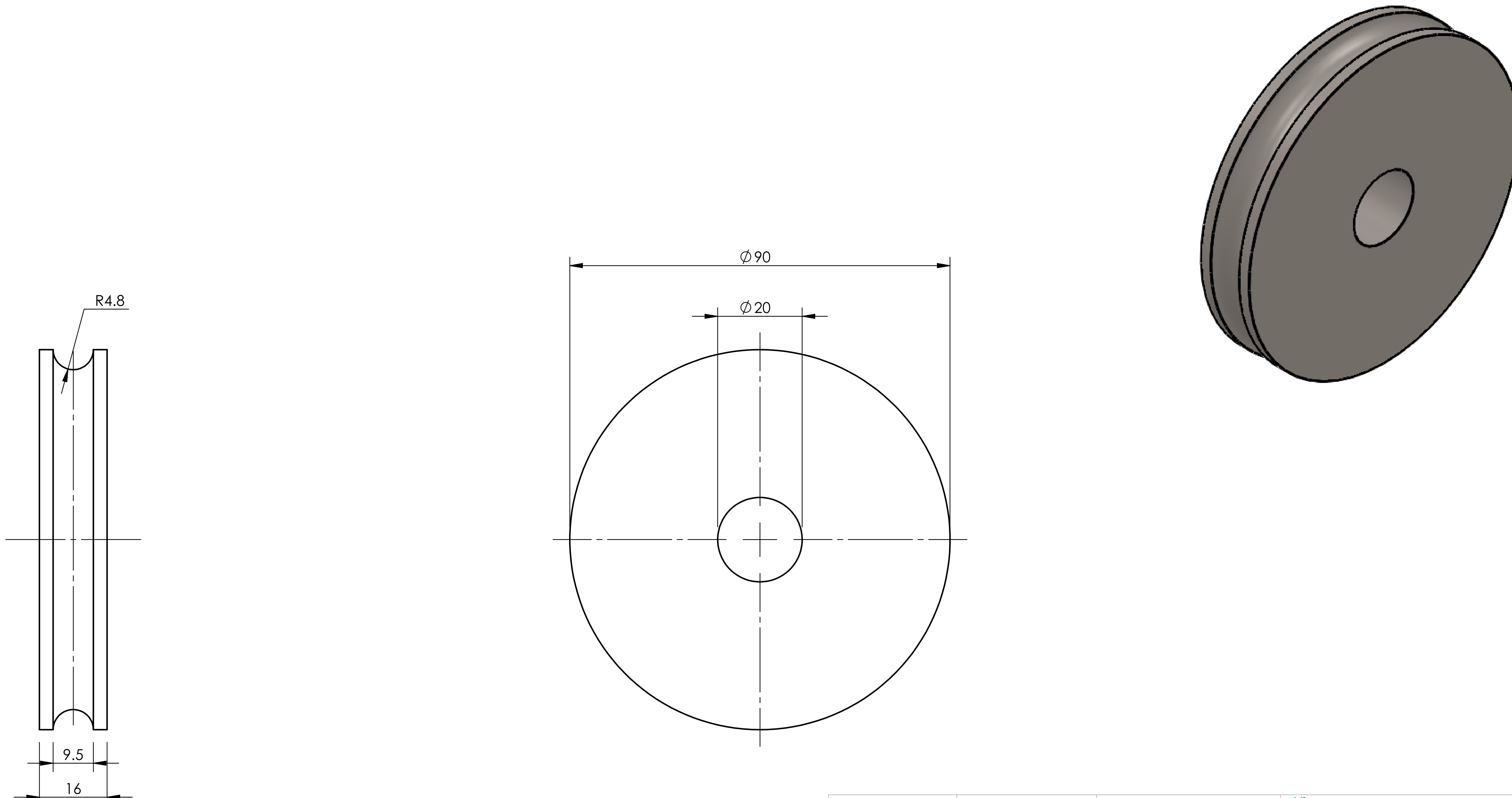
The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Pulley Assembly

MATERIAL:
Various
FILE NAME:
PulleyAssembly

SIZE B | Grappling Hook | REV A

SCALE: 1:3 | Mass: 3479.20 | SHEET 15 OF 21



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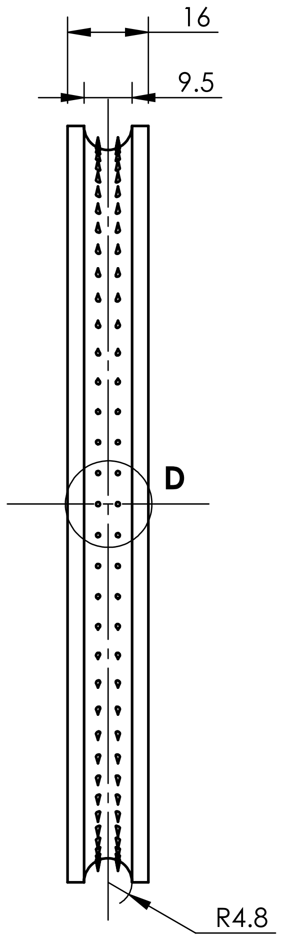
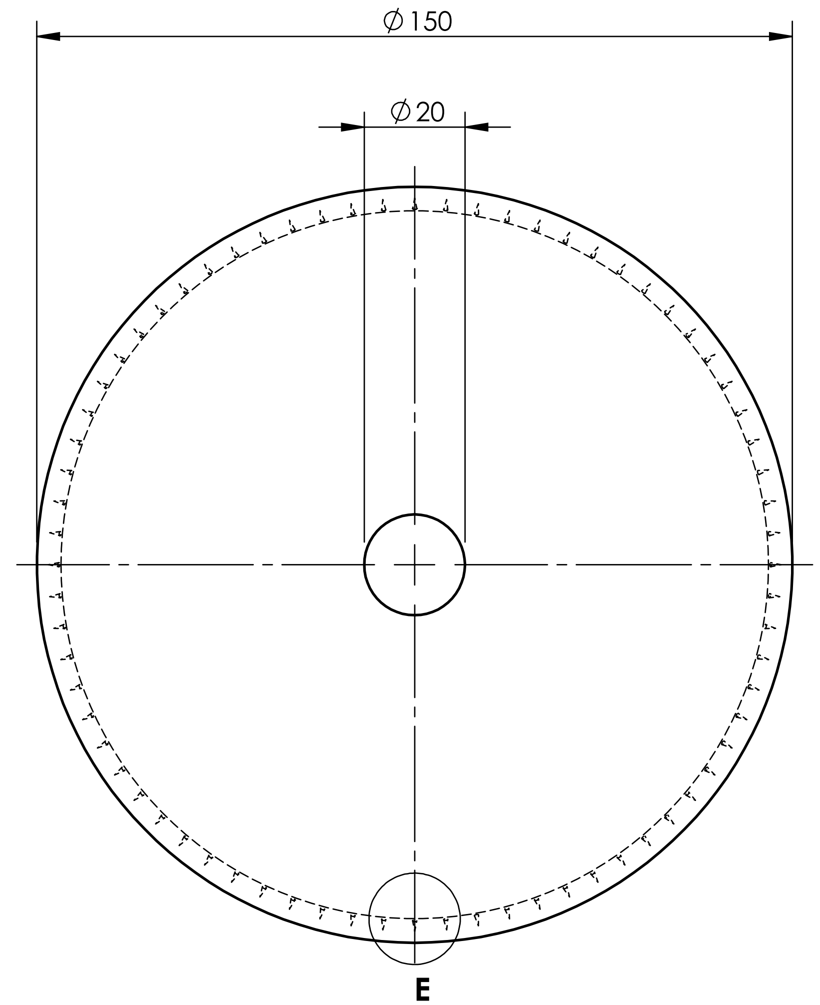
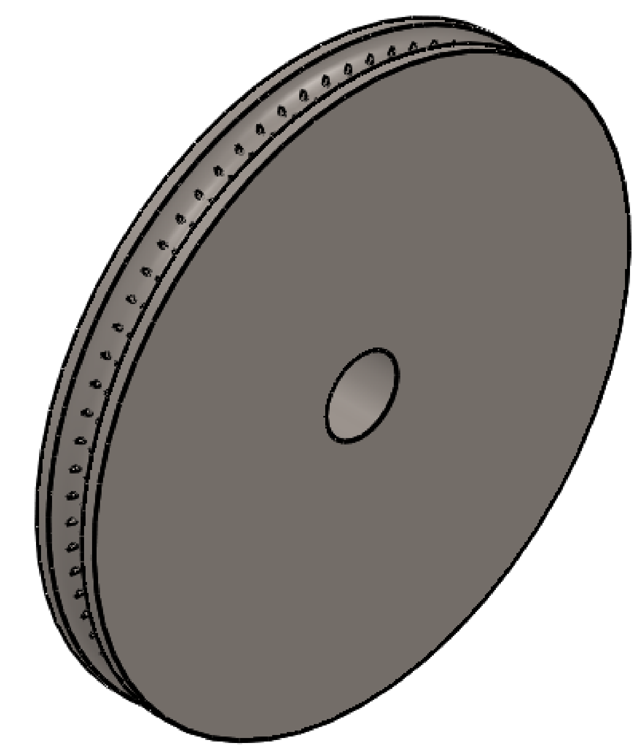
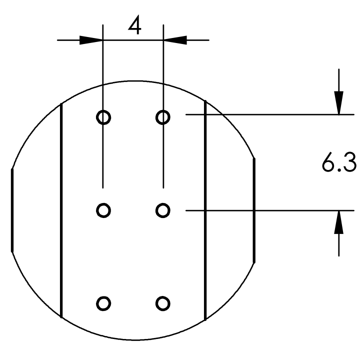
MecE 265 Instructor: Dr KK Duke Fall 2014 Comments: Purchased	UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025 SURFACE FINISH µm 0.6 DO NOT SCALE DRAWING	DRAWN BY: Group 10 Student # SM By Alexandre Suave December-03-14 7:55:07 PM November-17-14 1:02:48 PM	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: Pulley 1
MATERIAL: Stainless Steel (ferritic) FILE NAME: idler pulley			SIZE B Grappling Hook REV A SCALE: 2:1 Mass: 679.69 SHEET 16 OF 21



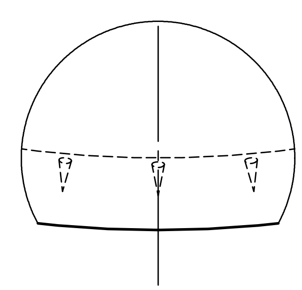




All barbs are conical in shape with dimensions:
Height: 2.5 mm
Base Diameter: 1 mm



DETAIL D
SCALE 2 : 1



DETAIL E
SCALE 2 : 1

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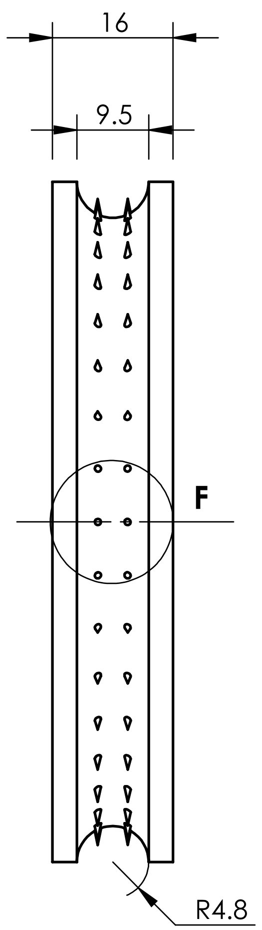
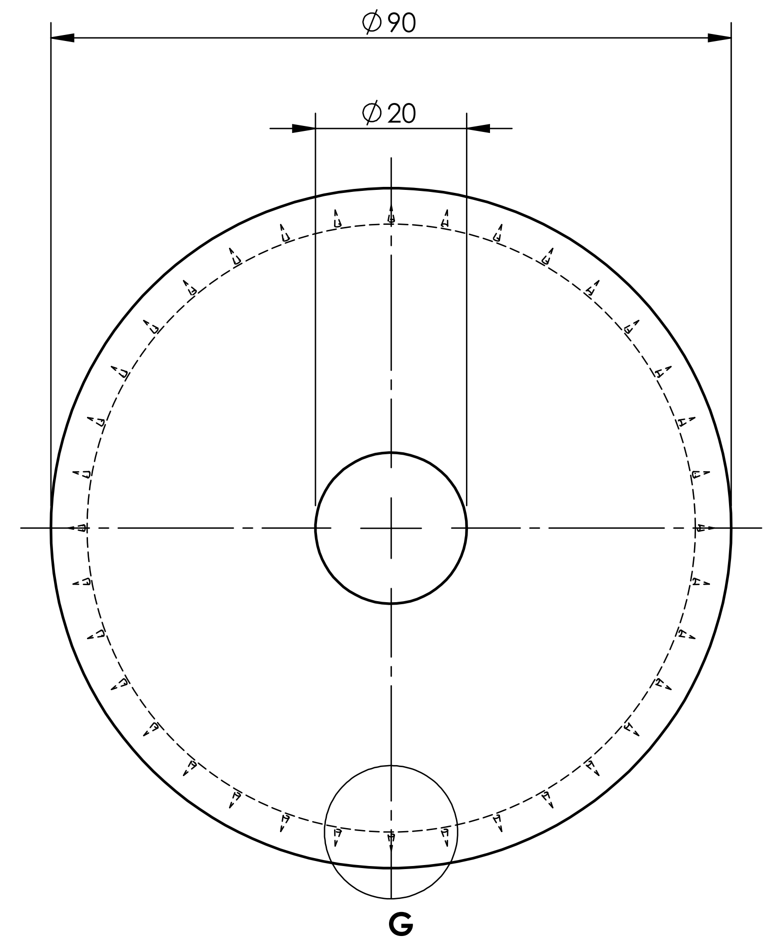
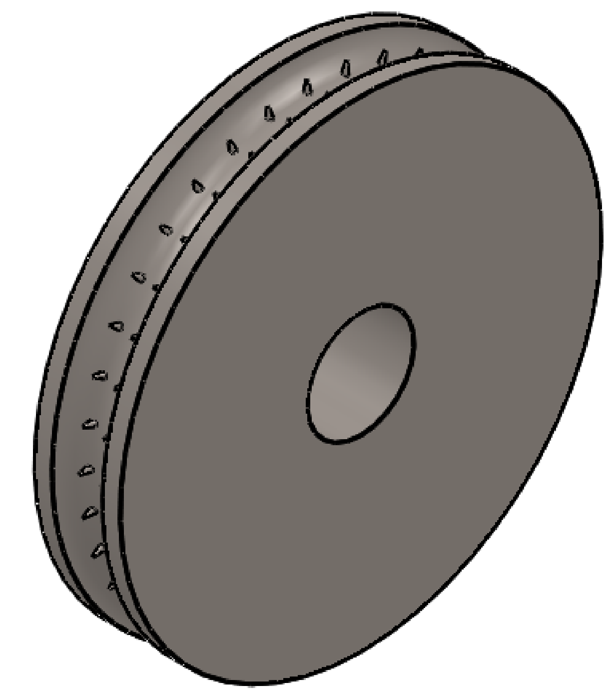
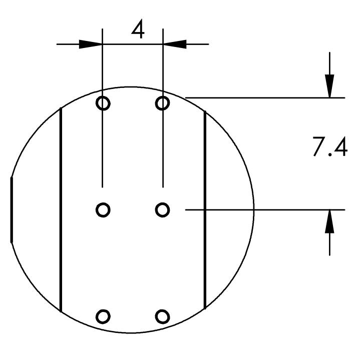
Mec E 360 Instructor: Dr. Zengtao Chen Fall 2014 Comments: Custom Pulley	UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025 SURFACE FINISH µm 0.6 DO NOT SCALE DRAWING	DRAWN BY: Group 10 Student # SM By Alexander Suave	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: Pulley 2 (Powered)
MATERIAL: Stainless Steel (ferritic)		December-03-14 7:55:29 PM November-17-14 12:25:33 PM	SIZE **B** Grappling Hook REV A
FILE NAME: powered pulley			SCALE: 1:1.5 Mass: 2039.18 SHEET 17 OF 21



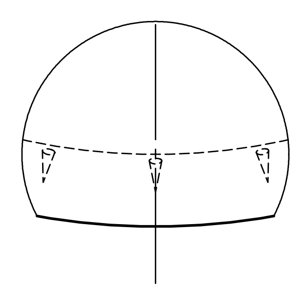




All barbs are conical in shape with dimensions:
Height: 2.5 mm
Base Diameter: 1 mm



DETAIL F



DETAIL G

Mec E 360
Instructor:
Dr. Zengtao Chen
Fall 2014
Comments:
Custom Pulley

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH µm 0.6
DO NOT SCALE DRAWING

DRAWN BY:
Group 10
Student #
SM By Alexandre Suave

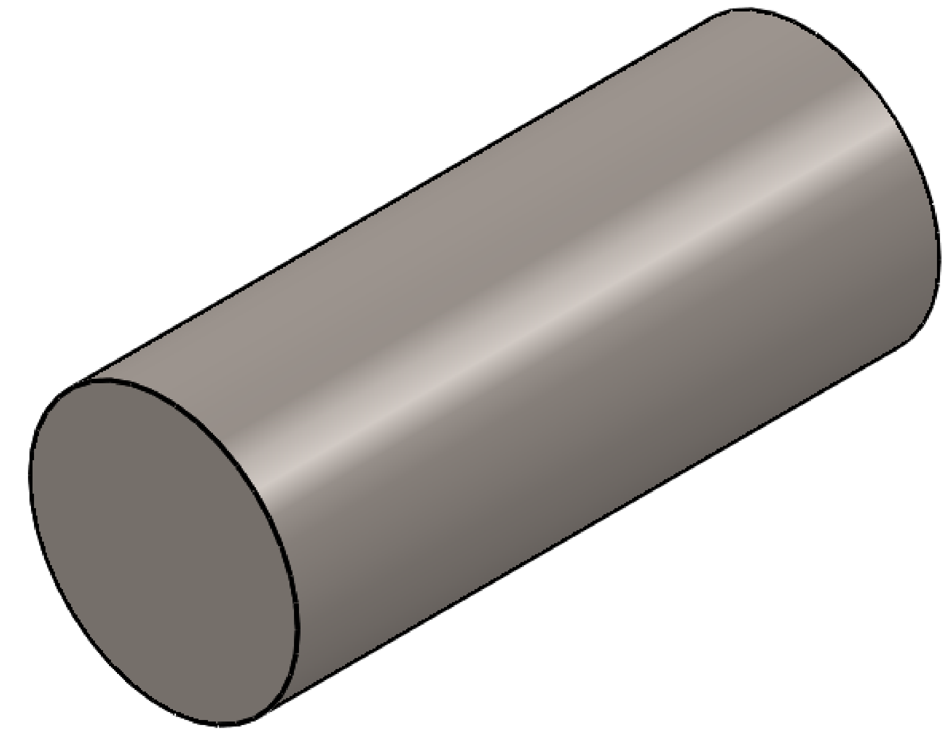
The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA
TITLE:
Pulley 3 (Exit)

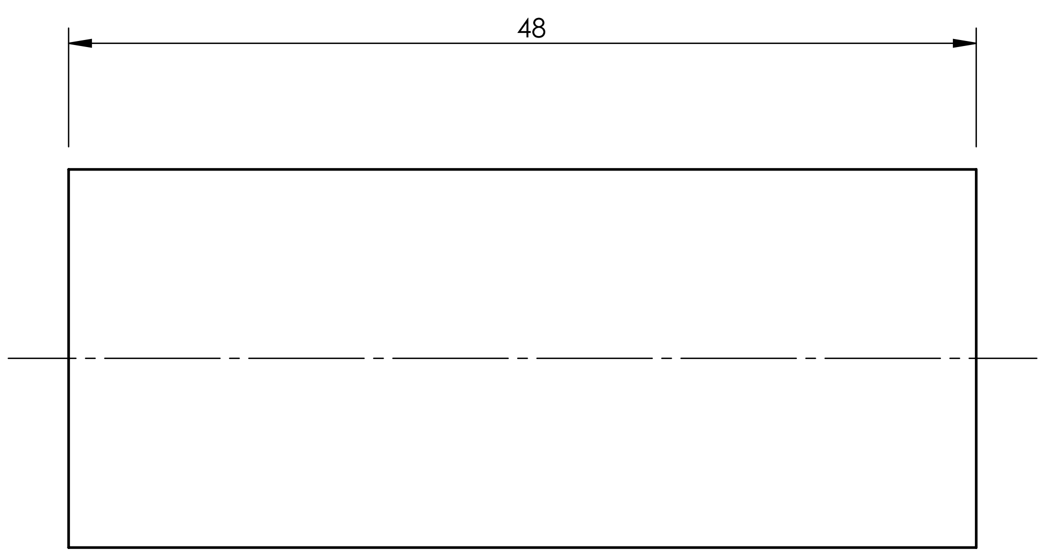
MATERIAL:
Stainless Steel (ferritic)
FILE NAME:
exit pulley

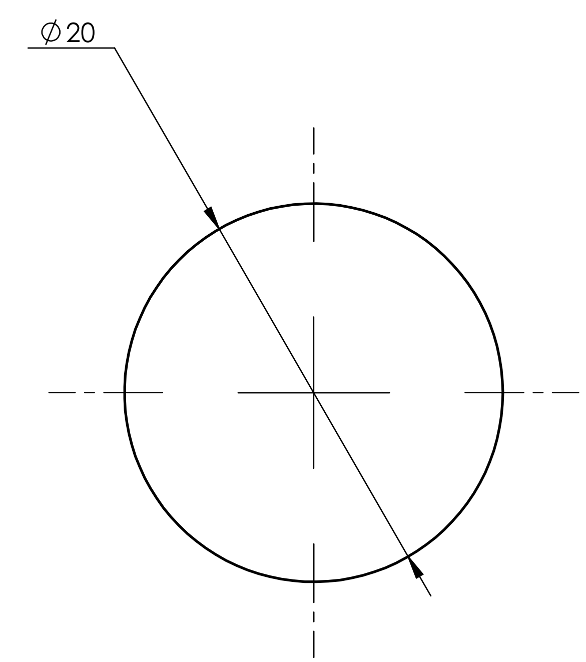
November-26-14 4:01:31 PM
November-17-14 1:02:48 PM

SIZE B Grappling Hook REV A

SCALE: 2:1 Mass: 679.90 SHEET 18 OF 21







Mec E 360

Instructor:
Dr. Zengtao Chen
Fall 2014
Comments:

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH µm 0.6
DO NOT SCALE DRAWING

DRAWN BY:
Group 10

Student #	
SM By	Panveer Dhaliwal

December-03-14 8:33:40 PM
November-16-14 12:54:25 PM

MATERIAL:
Stainless Steel (ferritic)
FILE NAME:
Shaft4

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Shaft 1 & 3

SIZE		REV
B	Grappling Hook	A

SCALE: 2.5:1 | Mass: 117.62 | SHEET 19 OF 21

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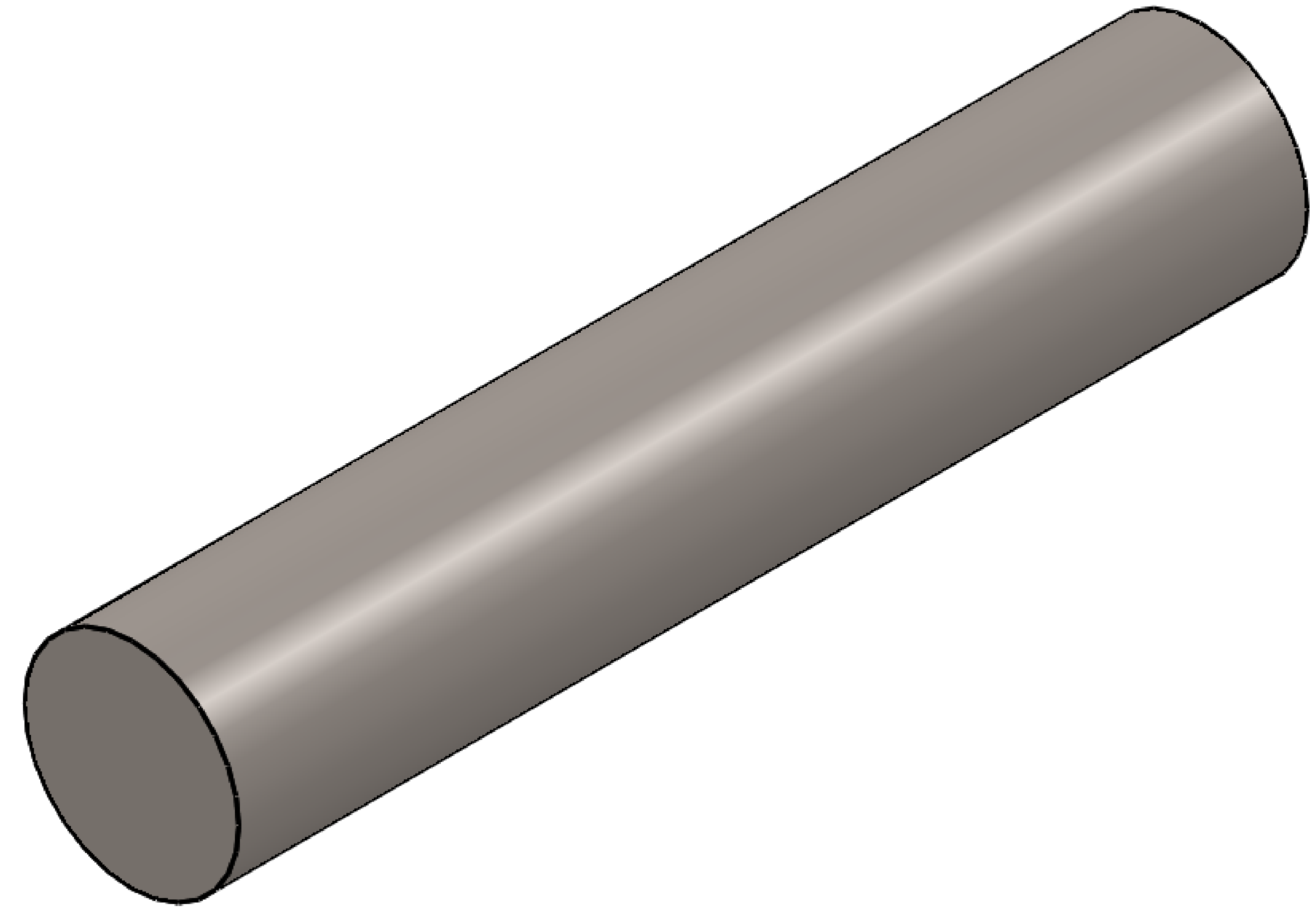

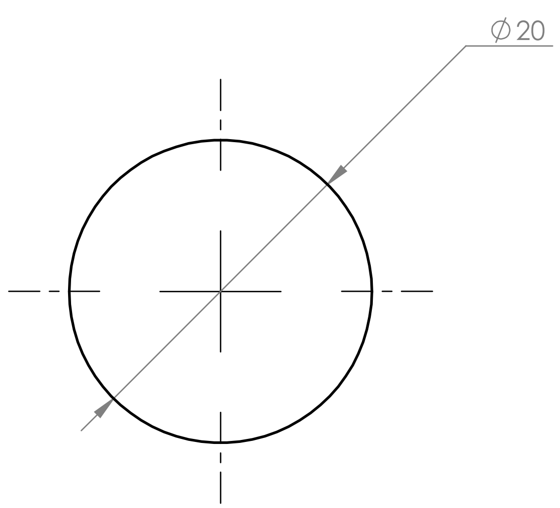

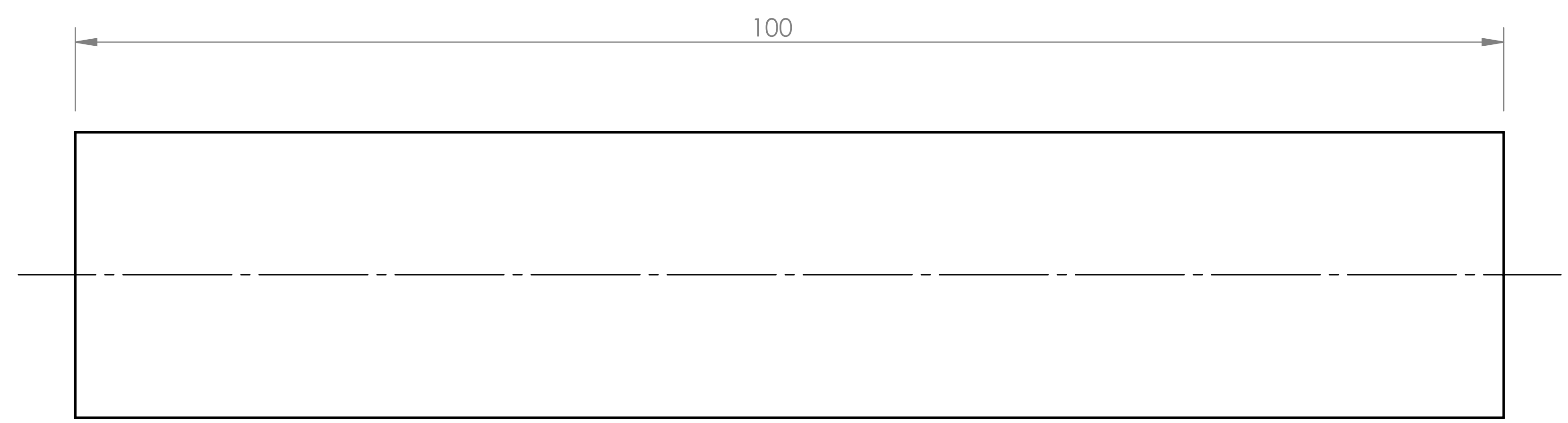

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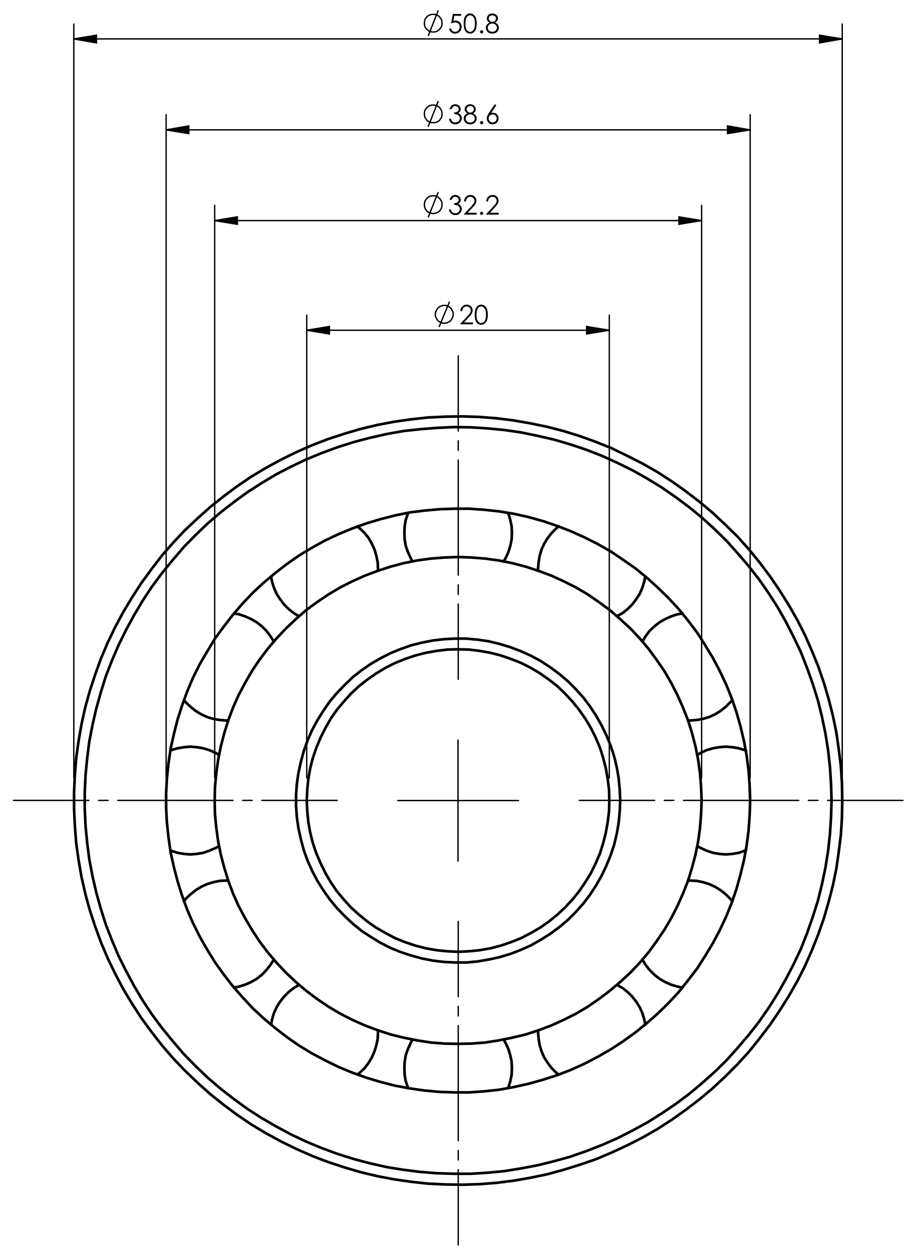
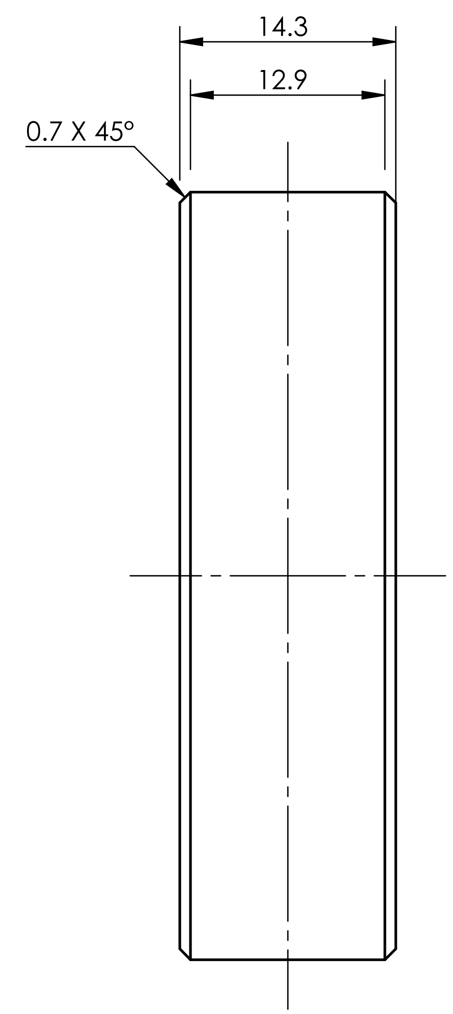
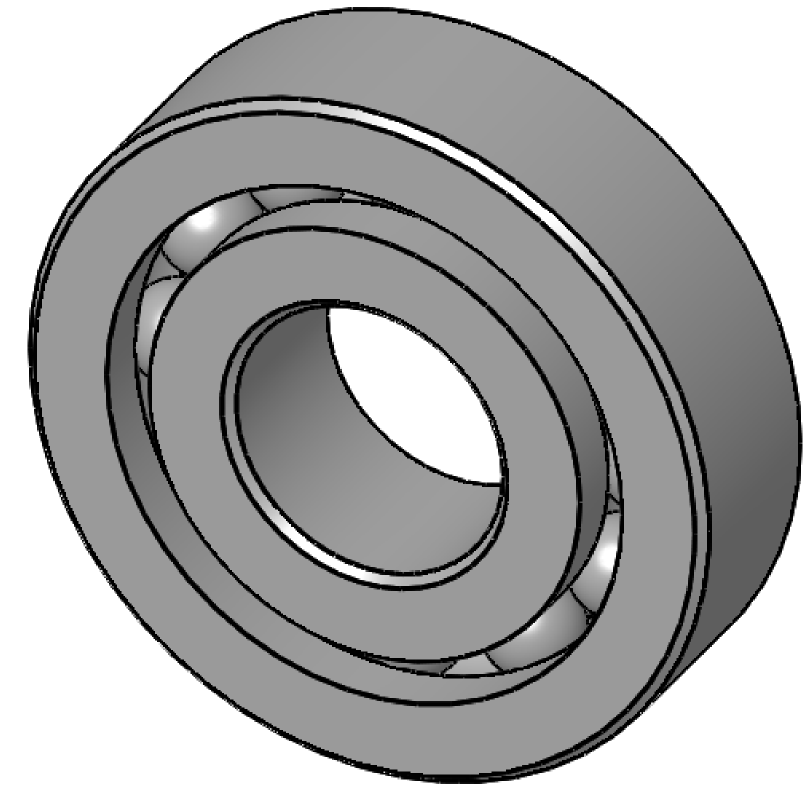
Mec E 360 Instructor: Dr. Zengtao Chen Fall 2014 Comments:	UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025 SURFACE FINISH µm 0.6 DO NOT SCALE DRAWING	DRAWN BY: Group 10 Student # SM By Panveer Dhaliwal November-23-14 6:14:12 PM November-16-14 12:54:25 PM	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: Shaft 2
MATERIAL: Stainless Steel (ferritic) FILE NAME: Shaft2			SIZE **B** Grappling Hook REV A SCALE: 2:1 Mass: 245.04 SHEET 20 OF 21







7004-B-TVP FAG Bearing

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Instructor:
Dr. Zengtao Chen
Comments:

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH µm 0.6
DO NOT SCALE DRAWING

MATERIAL:
Steel
FILE NAME:
20mm_Ball-Bearing

DRAWN BY:
Group 10
Student #
SM By McMaster
December-03-14 6:36:31 PM
January-31-07 7:25:07 AM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Ball Bearing - 20 mm

SIZE B
Grappling Hook
REV
SCALE: 2:1
Mass:
SHEET 21 OF 21

APPENDIX F
Gantt charts

table of contents

Anticipated Time Management	F1
Actual Time Management	F2

Tables

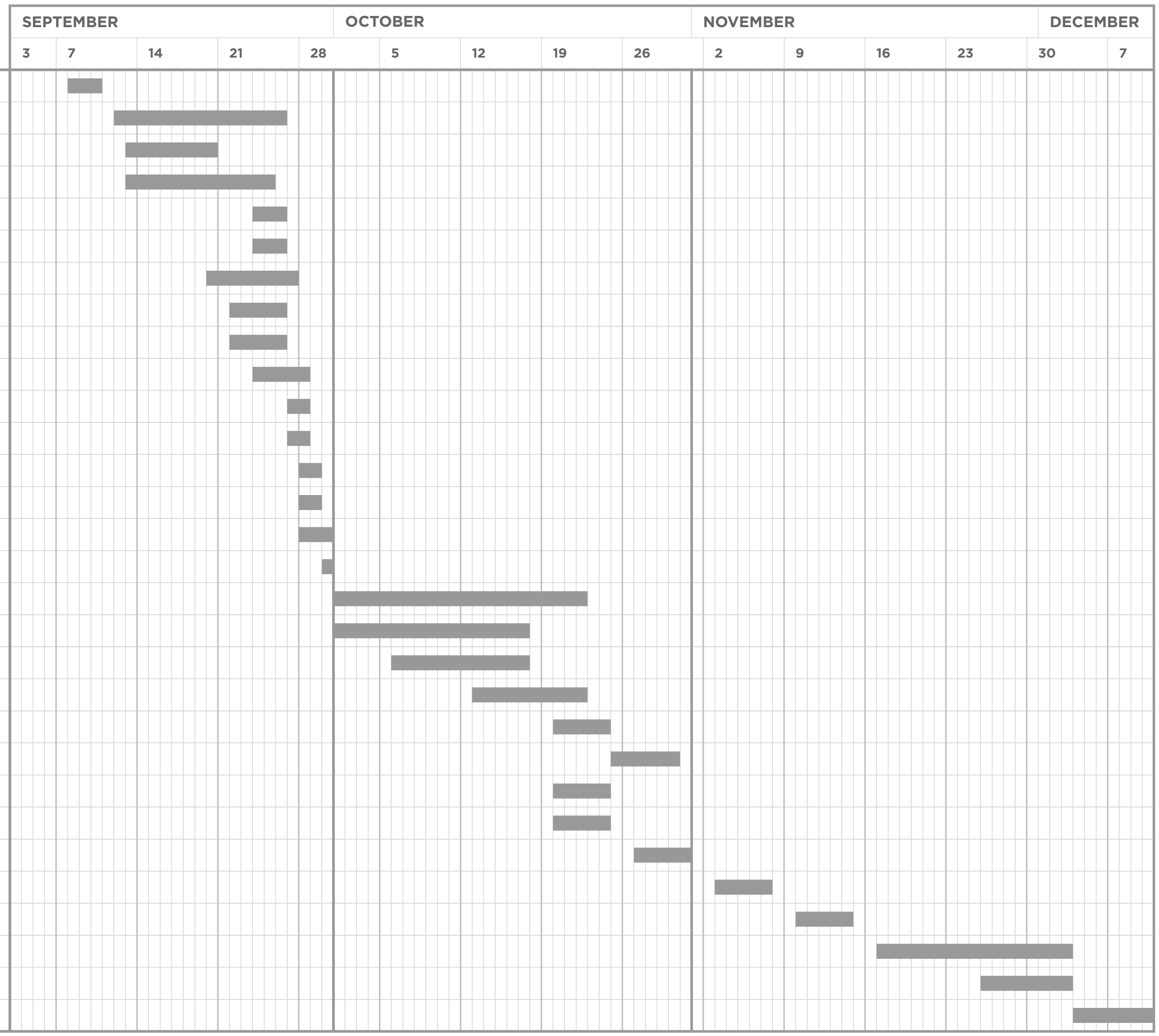
Table F1 : Anticipated Time Management Chart

Table F2 : Actual Time Management Chart

anticipated time management

TABLE F1 Anticipated Time Management Chart

bid letter	all
brainstorming	all
preliminary online research	all
professor consultation	all
gantt chart	ny
decision matrix	sc
initial calculations	all
final design • firing	as, ny
final design • winch	sc, pd
engineering analysis	all
finalize components	all
design/source components	all
material list	as
cost calculations	pd
report 1 • body	all
report 1 • assemble/submit	all
report 1 • edit	all
model parts	all
model assemblies	all
drawing pkg • assemble	all
drawing pkg • edit	ny, pd
drawing pkg • review	all
report 2 • introduction	sc, as
report 2 • conclusion	ny, pd
report 2 • body	all
report 2 • assemble/submit	ny, sc
report 2 • edit	all
final report • assemble	all
appendices	pd, as
final bid	all



as Alexandre Sauve
ny Nicholas Yee
pd Panveer Dhaliwal
sc Shealynn Carpenter

actual time management

TABLE F2 Actual Time Management Chart

bid letter	all
brainstorming	all
preliminary online research	all
professor consultation	all
gantt chart	ny
decision matrix	sc
initial calculations	all
final design • firing	as, ny
final design • winch	sc, pd
engineering analysis	all
decide components	all
design/source components	all
material list	as
cost calculations	pd
report 1 • body	all
report 1 • assemble/submit	all
report 1 • edit	all
model parts	all
model assemblies	pd, as
drawing pkg • assemble	pd, as
drawing pkg • edit	pd, sc
drawing pkg • review	all
report 2 • introduction	sc, as
report 2 • conclusion	ny, pd
report 2 • body	all
report 2 • assemble/submit	ny, sc
report 2 • edit	all
final report • assemble	ny
appendices	ny, sc
final bid	all

as Alexandre Sauve
ny Nicholas Yee
pd Panveer Dhaliwal
sc Shealynn Carpenter