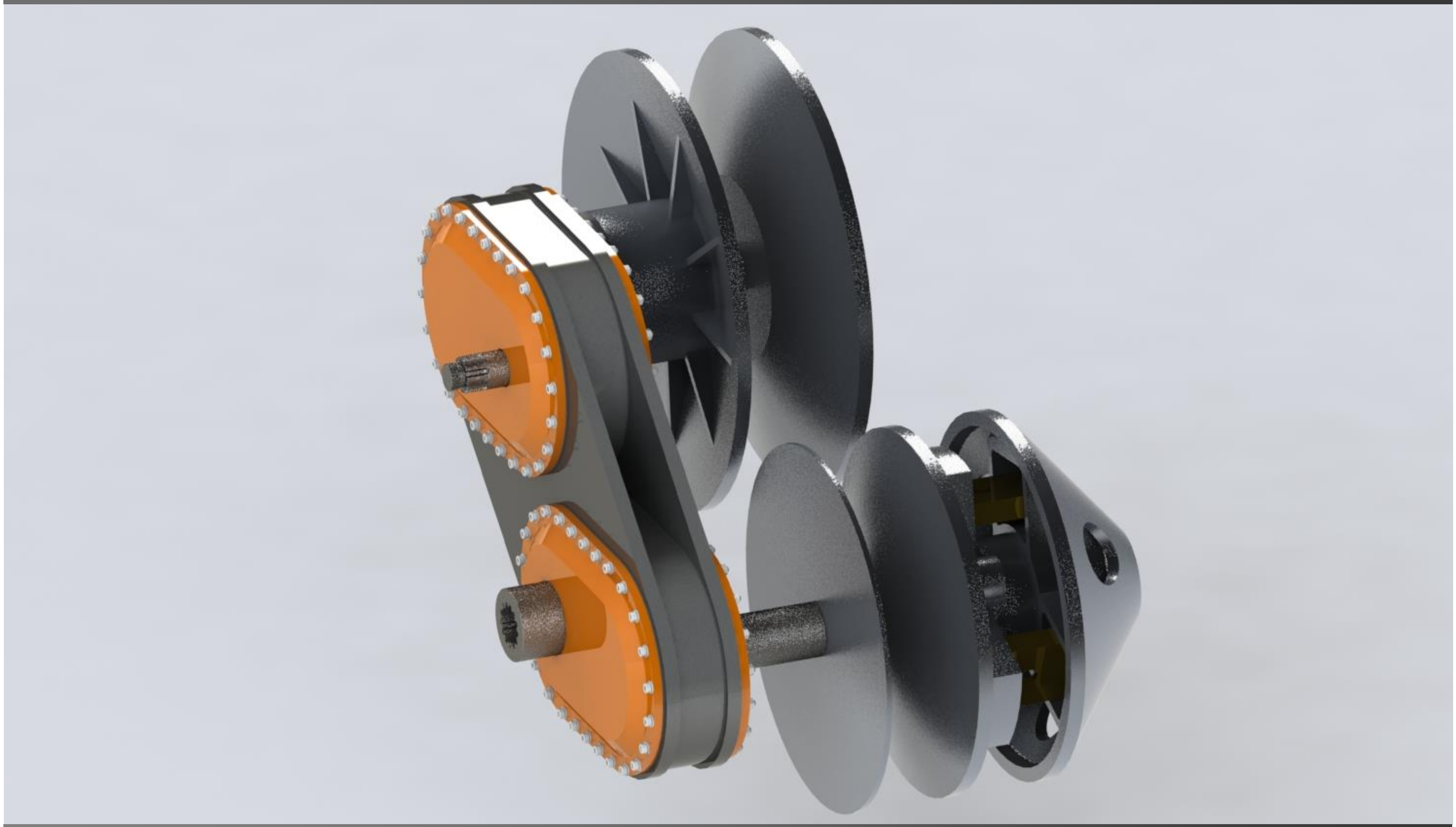

FSAE Continuously Variable Transmission

PHASE 3

Team 13

2-1 Mechanical Engineering Building
Edmonton AB, T6G 2G8

December 8, 2017

Trysten den Hartog
FSAE University of Alberta
9211-116 Street NW
Edmonton, AB T6G 1H9

Re: FSAE Continuously Variable Transmission – Phase 3 Deliverables

Dear Mr. den Hartog:

Team 13 has been contracted to design a continuously variable transmission (CVT) for the Formula SAE (FSAE) team at the University of Alberta. Upon completion of this project, Team 13 would like to present to you the Phase 3 report, which contains:

- An incremental description of the final design
- Detailed design analysis of critical regions in the CVT
- Engineering design cost, project management and scheduling
- Design sustainability and environmental impact
- Future works and research

After reviewing the project schedule, the total time invested in Phase 3 was 340 hours totaling a cost of $30,720. Given the hours invested in Phases 1 & 2, the total time and cost for the completion of this project is 698 hours and $66,503. The estimated cost of the transmission design is $2,570 USD ($3,310 CAD).

On behalf of Team 13, I would like to thank you for the opportunity to work on this project and hope to see our product used in future FSAE vehicles.

Sincerely,

Colin Reimer, Project Lead

enclosed.

cc.

Dr. Mohtada Sadrzadeh P.Eng
Evan Woynorowski, FSAE
Joshua Mulder, Team 13
Zi Cheng Cao, Team 13
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Table of Contents

Word Count (2549)

List of Figures

List of Tables

1 Executive Summary

Team 13 was contracted to design a continuous variable transmission (CVT) for the University of Alberta's Formula SAE (FSAE) team. The design must be lightweight and maintain peak acceleration during their vehicle's operation.

The Phase 2 design recommendation of Concept 2 was revised to Concept 1 during phase 3. Early belt force calculations proved that Concept 2 was not physically feasible for this design application. Therefore, Team 13 pivoted to Concept 1 for Phase 3 design.

The final design proposal is a dual clutch CVT with a torque sensing element. The design uses a primary clutch with sliding weights to create a centrifugal upshifting force on the V-belt. The secondary clutch uses a torque sensing cam, or helix, to generate a downshifting force from torque applied by the engine. The design has a total mass of 23.9 kg. The cost of all custom off the shelf (COTS) and machined components leads to the design costing 2600 USD (3300 CAD) USD. The design maintains an engine speed of 7750 rpm while it shifts from low to high gear. After completing the third phase of this project, the total engineering design cost is $62,303 CAD with Phase 3 comprising of $30,420. Additionally, 338 h was spent on completing Phase 3, resulting in 692 h being spent on the completion of this project.

After conducting further research for future developments of the design, Team 13 recommends the client considers mounting the CVT directly to the engine. This will reduce the weight and eliminate additional mechanical losses. Also, a 2007 Yamaha Phazer 500 engine should be considered since it offers greater compatibility with a CV Transmission. It has a similar engine displacement, produces slightly more horsepower, and is slightly heavier since it is a 2-cylinder engine. It is also recommended that the CVT and engine be prepared prior to designing the chassis.

2 Description of CVT Design

2.1 CVT vs. Sequential Gearbox

Where the sequential gearbox can only achieve peak power at different gears for specific engine speeds, the CVT is able to achieve peak power throughout its operation by varying the transmission gear ratio. This also eliminates the error that comes from the driver manually shifting gears and the shift lag associated with changing gears. The shift curves in Figure 1 show the difference between the two transmissions.

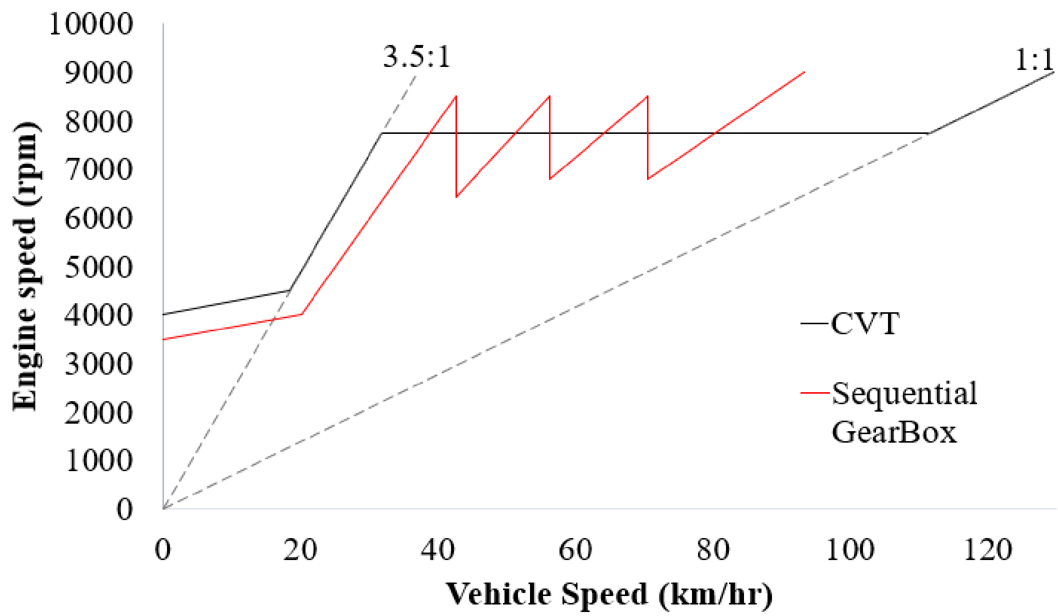

Figure 1 Shift Curves of the Sequential Gearbox and the CVT

2.2 CVT Operation and Physical Parameters

The CVT design transmits power from the engine to the wheels using two sets of angled sheaves with a rubber belt under tension between them called the primary and secondary clutches (also called drive pulley and driven pulley respectively). This design responds to the forces between the primary and secondary clutch to vary the transmission gear ratio.

2.4 Key Features

There are three primary components/sub-assemblies that make up this concept as seen in Figure 2. The drive and driven pulley work together to balance shifting forces and adjust the transmission gear ratio. The gearbox increases the ratio into the pulleys and reduces it after to achieve appropriate torque for each stage and the correct total reduction to the wheels. The tuning parameters for the pulleys were evaluated and the pulleys were selected via consultation with CVTech. The gearbox was designed and analyzed specifically for transmission implementation.

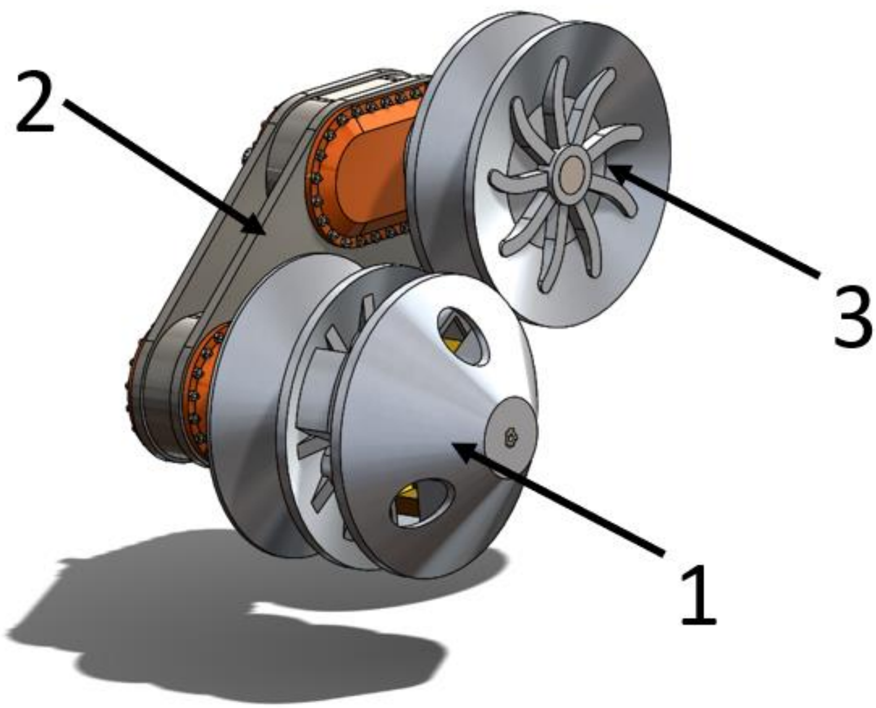

Figure 2 Full Assembly of the CVT

Table 1 Bill of Materials (BOM) for the CVT

Item	Component	Description
1	Primary Clutch	Drive pulley/angular velocity shifting clutch. Mounted to power output shaft of vehicle (CV.1100-0046 PWB 80 Drive Pulley)**
2	CVT Gearbox	CVT transmission housing containing the initial and final gear reductions needed to ensure proper shifting
3	Secondary Clutch	Driven pulley/torque-sensing clutch. Mounted to vehicle differential (CV.6000-0031 PWB 80 Driven Pulley 1” shaft)

NOTE: The rubber V-belt is not shown in this assembly
**See CVTech quote in Appendix B for configuration details (flyweights and springs)

2.4.1 Primary Clutch

The primary clutch in Figure 3 uses the radial acceleration of spinning weights, commonly referred to as centrifugal force, to induce an upshifting force. It also contains a compression spring that is preloaded to induce a downshifting force. This downshifting force increases as the primary clutch shifts and compresses the spring.

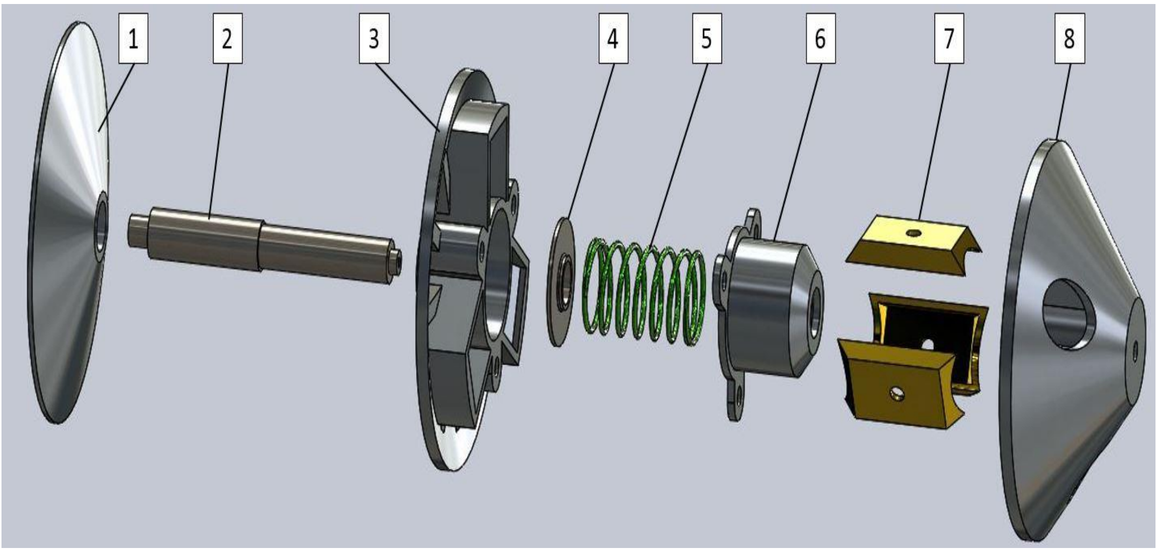

Figure 3 Primary Clutch Exploded View

Table 2 BOM for the Primary Clutch

Item	Component	Description
1	Fixed Sheave	Aluminum, fixed pulley sheave, permanently installed on the clutch shaft
2	Primary Clutch Stepped Shaft	Steel, stepped shaft for primary clutch - integrated into existing engine/transmission output
3	Primary Clutch Moveable Sheave	Aluminum, moveable pulley sheave, position dictated by compression spring clutch weights
4	Spring Shoulder	Steel shoulder contacts stepped shaft and creates dynamic relationship between spring compression and moveable sheave
5*	Primary Spring	Preloaded compression spring
6	Spring Cap	Transmits spring force to moveable sheave and maintains spring preload during assembly
7*	Clutch Weights	Sliding weights that shift radially outward during shifting
8	Spider	Contains sliding weights. Fixed to shaft. As weights move radially forces axial movement of moveable sheave
***These are the tuning parameters that will influence the clamping force on the belt and shifting behavior of the CVT**		

2.4.2 Secondary Clutch

An exploded assembly view of the secondary clutch can be seen in Figure 4.

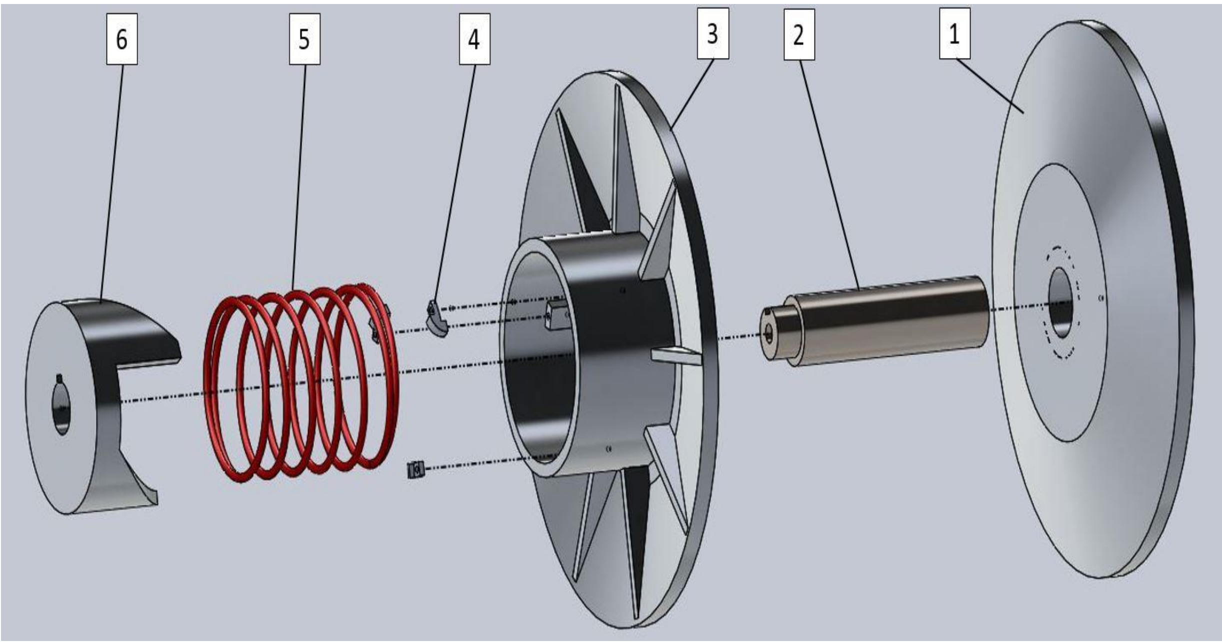

Figure 4 Secondary Clutch Exploded View

Table 3 BOM for the Secondary Clutch

Item	Component	Description
1	Fixed Sheave	Aluminum, pulley sheave that is permanently installed on the clutch shaft
2	Secondary Clutch Stepped Shaft	Steel, stepped shaft for secondary clutch - either additional transmission shaft or integrated into differential
3	Secondary Clutch Moveable Sheave	Aluminum, pulley sheave that slides on shaft based on forces from helix and spring
4	Helix Slider	Sliding contact point between helix and moveable sheave
5*	Secondary Compression Spring	Compression spring
6*	Helix	Torque sensing cam, pushes against moveable helix when under load
*These are the tuning parameters that will influence the clamping force on the belt and shifting behavior of the CVT. These tuning parameters were selected for the CVTech pulleys		

This clutch also contains a compression spring like the primary, however it is designed to cause a downshifting force. It also contains a helix, also known as a torque cam, which is the torque-sensing element of the design. The helix generates a downshifting force that is a function of the amount of torque being transmitted through the system.

2.4.3 CVT Gearbox

The gearbox assembly was added out of necessity. The CVT pulley torque requirements and belt calculations led to the discovery that the torque output from the client's existing transmission in any gear was too high to prevent belt slip. Other options explored included modifying the gears in the existing sequential gearbox to achieve a smaller gear reduction or modifying the engine crankshaft such that the primary clutch could be mounted directly on it. However, in client meetings these options were discussed and rejected due to the complexity and difficulty of making these modifications. Therefore, the gearbox shown in Figure 5 was added to achieve the correct operating conditions for the pulleys.

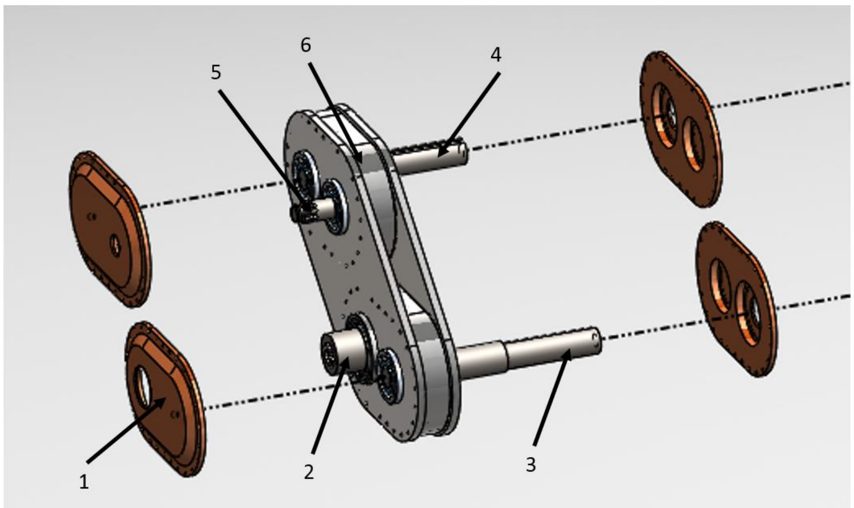

Figure 5 CVT Gearbox Exploded View

Table 4 BOM for the CVT Gearbox

Item	Component	Description
1	Bearing Cap	Seals the gearbox components together and carries bearing load
2	CVT Input Shaft	Transmits power from the engine to the Primary Clutch input shaft
3	Primary Clutch Input Shaft	Transmits power to the primary clutch from the CVT input shaft
4	Secondary Clutch Output Shaft	Transmits power from the secondary clutch to the CVT output shaft
5	CVT Output Shaft	Transmits power from the CVT to the sprocket (sprocket/chain drive to differential)
6	Gear Case	Houses gears, shafts, and ATF fluid for transmission

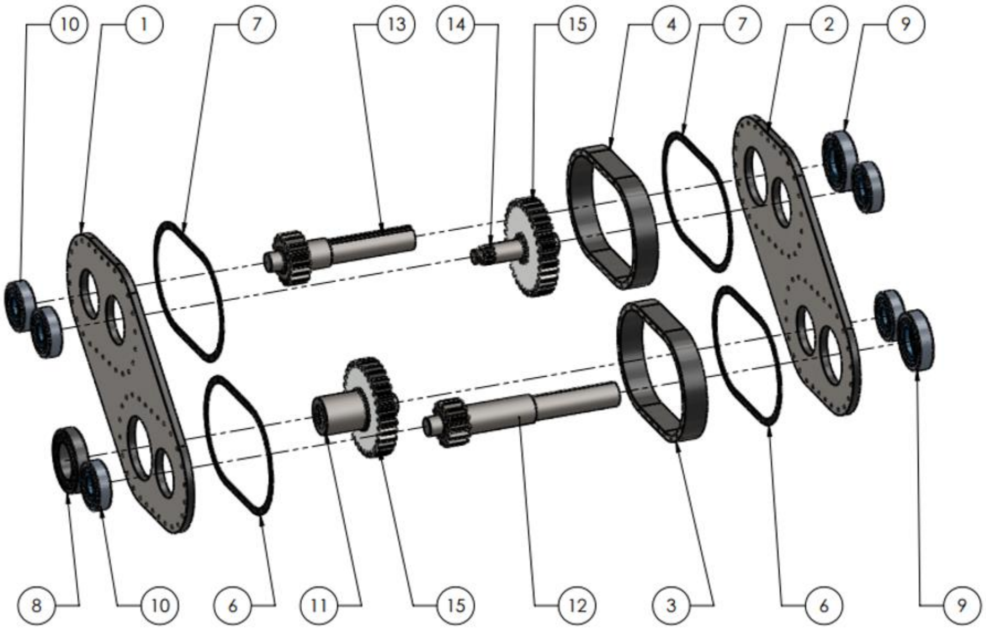

Figure 6 Gear Case Exploded View

Table 5 BOM for the Gear Case

Item(s)	Component	Description
1, 2	Frame Plates	Used to line up shafts and gear for correct meshing as part of gearbox housing
3, 4	Seals	Metal case surrounding gears and shafts as part of gearbox housing
6	69 mm Gasket Seal	Oil paper jointing or nitrile rubber gasket for sealing gearbox
7	72 mm Gasket Seal	Oil paper jointing or nitrile rubber gasket for sealing gearbox
8	40 mm FAG Bearing	40mm ID angular Contact bearing (7008-b-xl-tvp)
9	30 mm FAG Bearing	30mm ID 4-point contact bearing (qj206-xl-mpa)
10	20 mm FAG Bearing	20mm ID 4-point contact bearing (qj304-xl-mpa)
11, 14	Shaft 1, Shaft 4	Custom made CVT input and output shafts
12, 13*	Shaft2, shaft 3	Custom made primary and secondary clutch shafts
15*	32 tooth QTC Gears	32 tooth gear (A 1C 1MYK30032)
*Gears have a standard 20° pressure angle and modulus of 3.		

3 Detailed Design Analysis

3.1 Shifting Forces

3.1.1 Primary Clutch

The free body diagram (FBD) of the primary clutch can be seen in Figure 7.

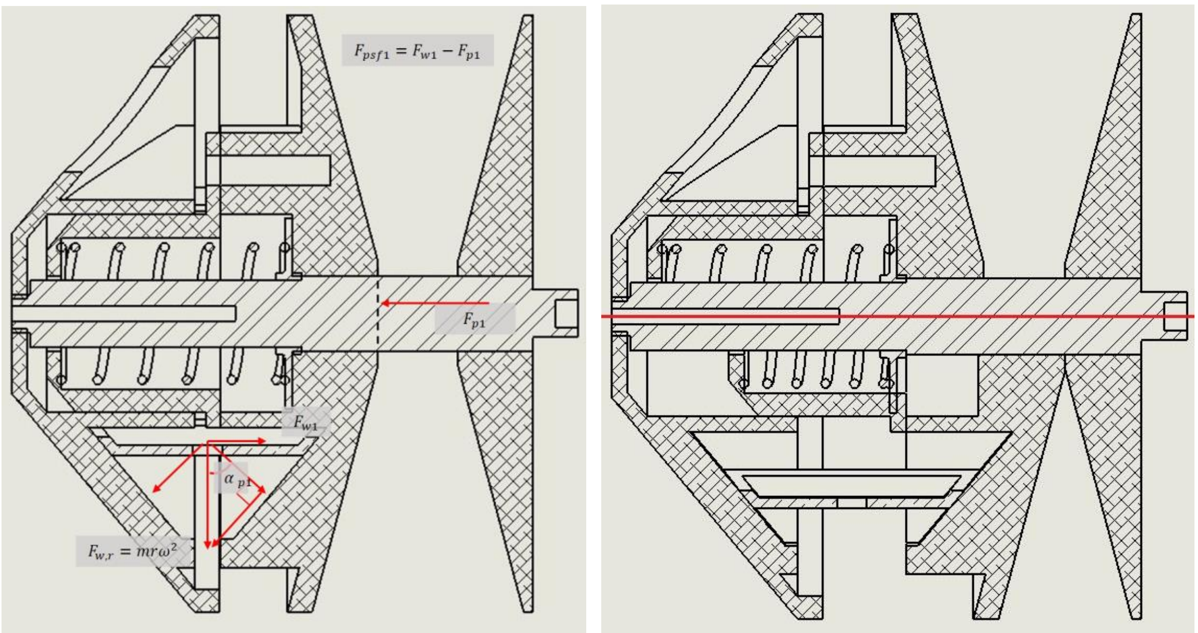

Figure 7 Left: Free body diagram of shifting forces in primary clutch. F.w1, axial force on moveable sheave due to centrifugal force; F.p1, axial force of compression spring. Right: Open (top) and closed (bottom) positions of the primary clutch

This upshifting force is a function of the weights in the primary clutch and spring compression force. The spring force was determined from its stiffness and the preload needed to engage the clutch. The force from the clutch weights depends on their angular acceleration, which can be determined given their radial positions, masses and sliding angles.

3.1.2 Secondary Clutch

The FBD of the secondary is shown in Figure 8.

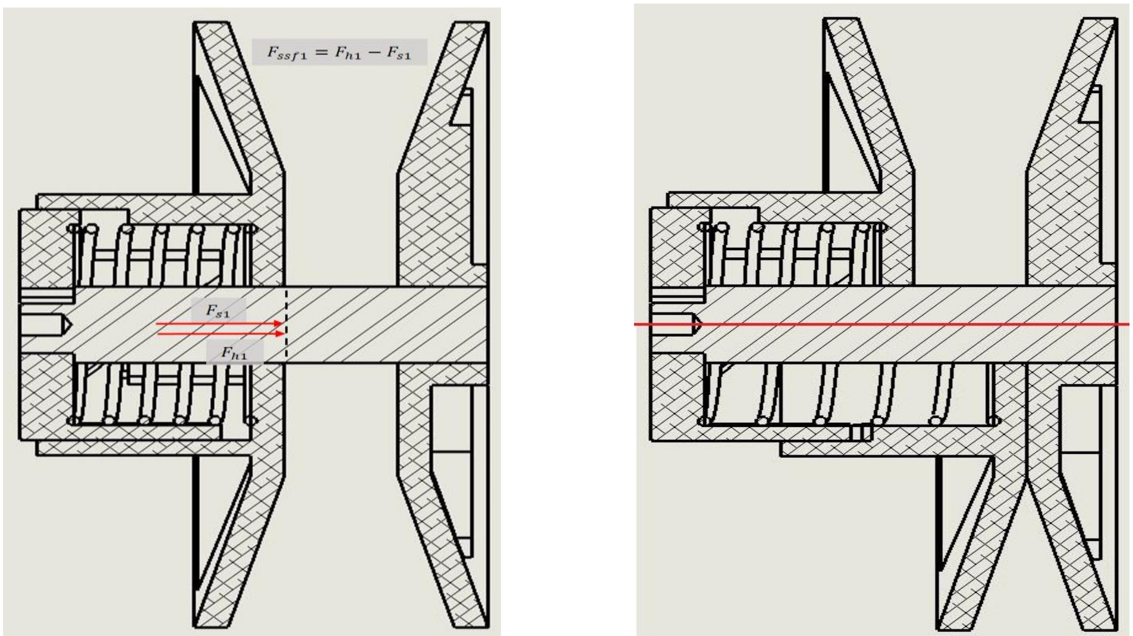

Figure 8 Left: Free body diagram of shifting forces in secondary clutch. F.h1, axial force on moveable sheave due to helix; F.s1, axial force of compression spring. Right: Open (top) and closed (bottom) positions of the secondary clutch

The downshifting force depends on the spring force, like the primary clutch, but also the force from the helix. The spring force can be determined using its stiffness and displacement of the moveable sheave. The helix force is governed by the helix angle, α_{h1}, and radius, r_{h1}, such that:

$$F_{h1} = T_{s1}\frac{\sin\alpha_{s1}}{r_{h1}} \qquad \textbf{Equation 1}$$

where T_{s1} is the torque experienced by the secondary clutch.

3.2 Belt Tension and Spring Preload

The FBD governing the belt force and spring preload can be seen in Figure 9.

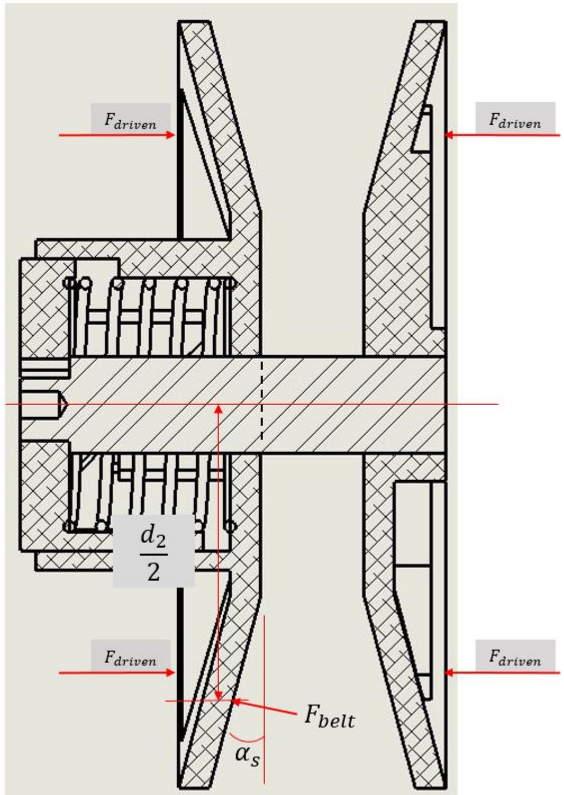

Figure 9 FBD of secondary clutch showing the belt clamping force

For the belt to shift, the following condition must be satisfied:

$$\tan \alpha_s > \mu_{belt} \qquad \textbf{Equation 2}$$

where α_s is the sheave angle and μ_{belt} is friction coefficient of the belt. Using the selected sheaves and belt, the clamping force applied to the belt by the sheaves were found using the following:

$$F_{driven} = \frac{T_{driven}}{\mu_{belt} d_s} \cos \alpha_s \qquad \textbf{Equation 3}$$

The preload applied to the primary clutch spring depends on the clamping force from the sheaves and the force from the helix. Note that frictional force will occur from the helix since it is sliding and rotating on the output shaft of the CVT. The maximum preload for compression springs in CVT is typically 1 kN, so the spring preload was described as follows:

$$F_{preload} = F_{driven} - \left(F_{h1} + F_{hf}\right) \leq 1 \text{ kN} \qquad \textbf{Equation 4}$$

where F_{h1} is the helix force from the torque output and F_{hf} is friction force from the helix. This spring preload condition is a typical industry standard that is used in many CVT designs.

3.3 Force Applied to the Road (Ground Force)

To compare the existing sequential gearbox to the CVT design, the ground force of the client's existing transmission and CVT were calculated. The ground force was determined from engine's power output, HP_{engine}, and vehicle speed, $v_{vehicle}$, such that:

$$F_{ground} = \eta_{transmission} \frac{HP_{engine}}{v_{vehicle}} \qquad \textbf{Equation 5}$$

where $\eta_{transmission}$ is the transmission's efficiency. Using Equation 5, the ground force plot in Figure 10 was generated for both transmissions.

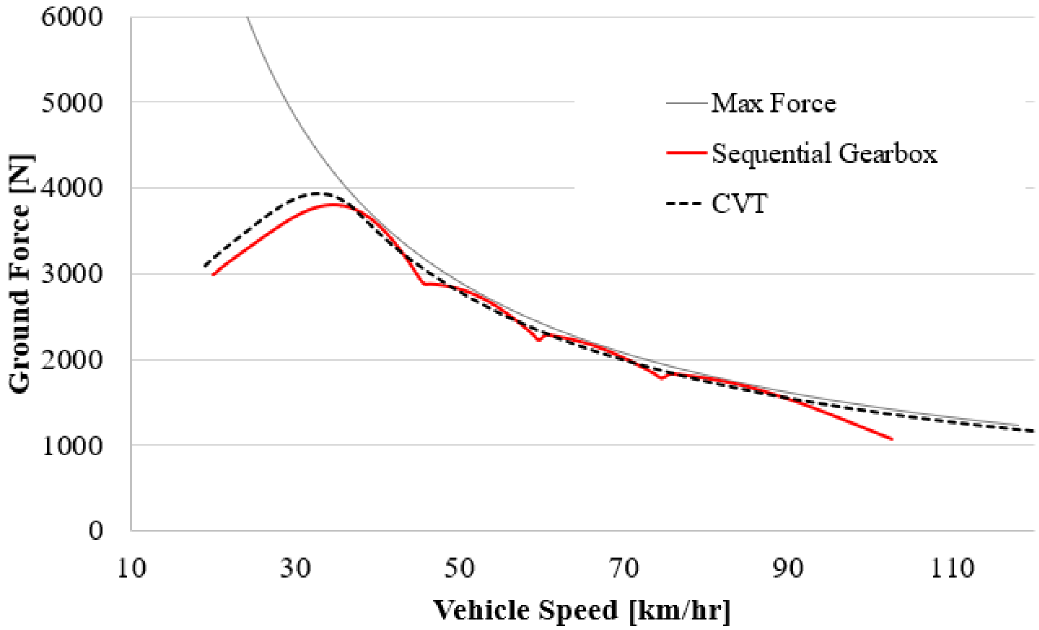

Figure 10 Ground Force of the Sequential Gearbox and the CVT

The CVT has an estimated mechanical efficiency of ~96%, whereas the sequential gearbox has an ~98% efficiency. This is due to the additional losses that come from the belt. However, the ground force plots for the CVT and sequential gearbox show the CVT can smoothly transmit force to the ground at any given speed. At some speeds, the sequential gearbox does transmit a larger ground force, but this plot does not illustrate two important factors:

1) The sequential gearbox does not shift gears instantly and;
2) The driver using a sequential gearbox cannot shift at the ideal engine speed all the time.

Without these limitations, the CVT reaches top speed faster than the sequential gearbox and transmits on average a higher ground force even with the additional mechanical losses.

3.4 Gear Analysis

The CVT gearbox in Figure 11 was added to ensure the CVT went through the correct gear reductions to ensure proper shifting. The diagram in Figure 12 shows the different torque regions the CVT goes through during its operation.

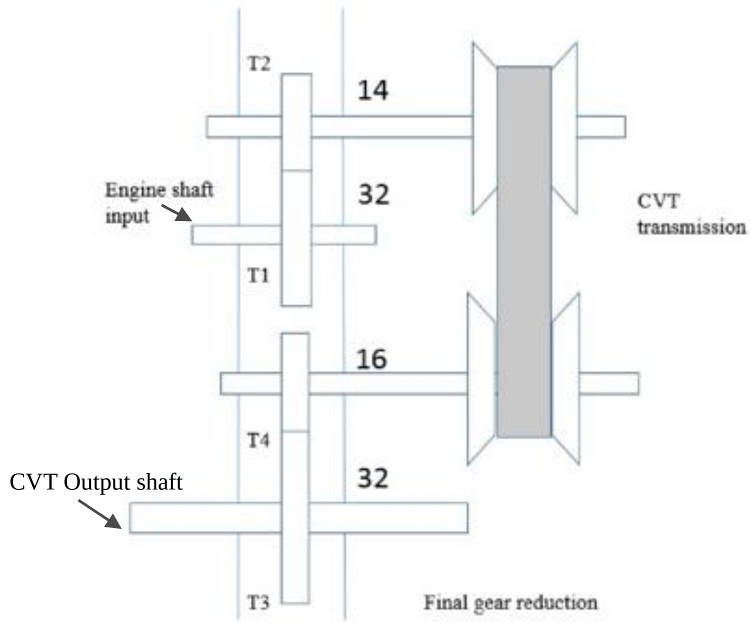

Figure 11 Gear ratio changes that occur in the CVT gearbox

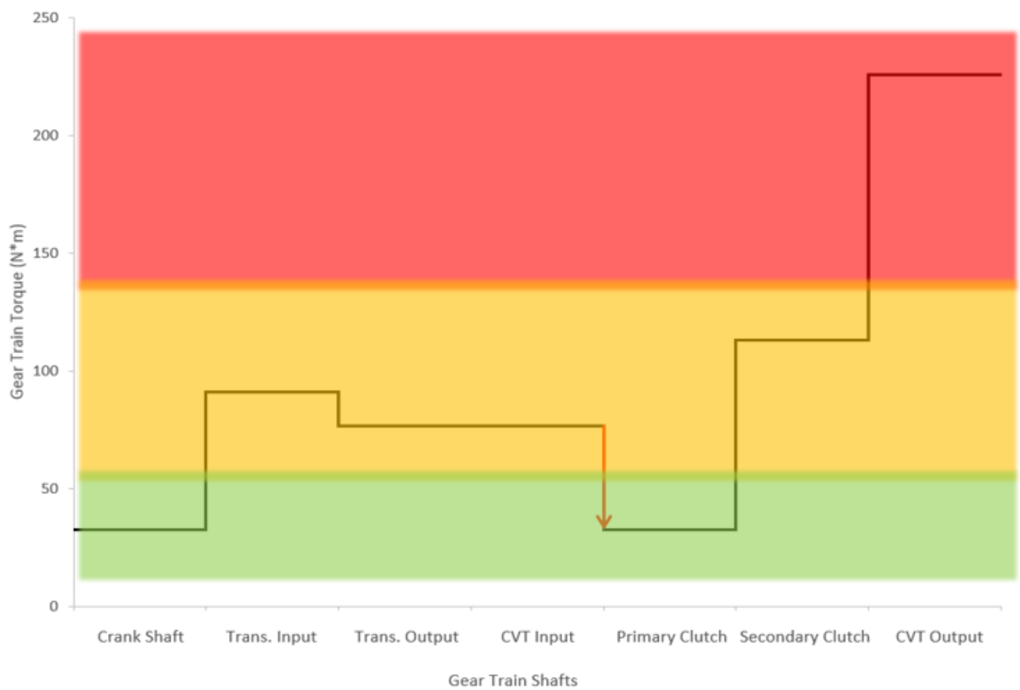

Figure 12 Torque Outputs of the CVT due to Gear Reductions in the CVT Gearbox. Red Region: Output torque is too large for CVT to handle; Yellow Region: Output torque is undesirable but achievable; Green Region: Optimal output torque region.

The three coloured regions in Figure 12 represent approximate torque design regions. The red region is an area where the CVT pulleys are experiencing such severe torque that a rubber belt driven transmission is not suitable. The yellow region represents a torque that is too severe for a compression spring alone to achieve sufficient clamping force.

The red arrow shown between the CVT input and primary clutch is the gear reduction required to have the primary clutch operate in the green region and the secondary clutch in the yellow. The chosen gear ratio is an industry standard ratio that is meant to lower the torque and raise the rotational speed. Although the torque in the secondary clutch is undesirable, it is achievable because of the force applied by the helix.

Since a gear ratio addition was made into the primary clutch, a gear reduction was necessary after the secondary clutch. This was done to ensure the vehicle operation speed of approximately 100 km/hr at an engine speed of 9000 rpm was achieved as shown in Figure 13.

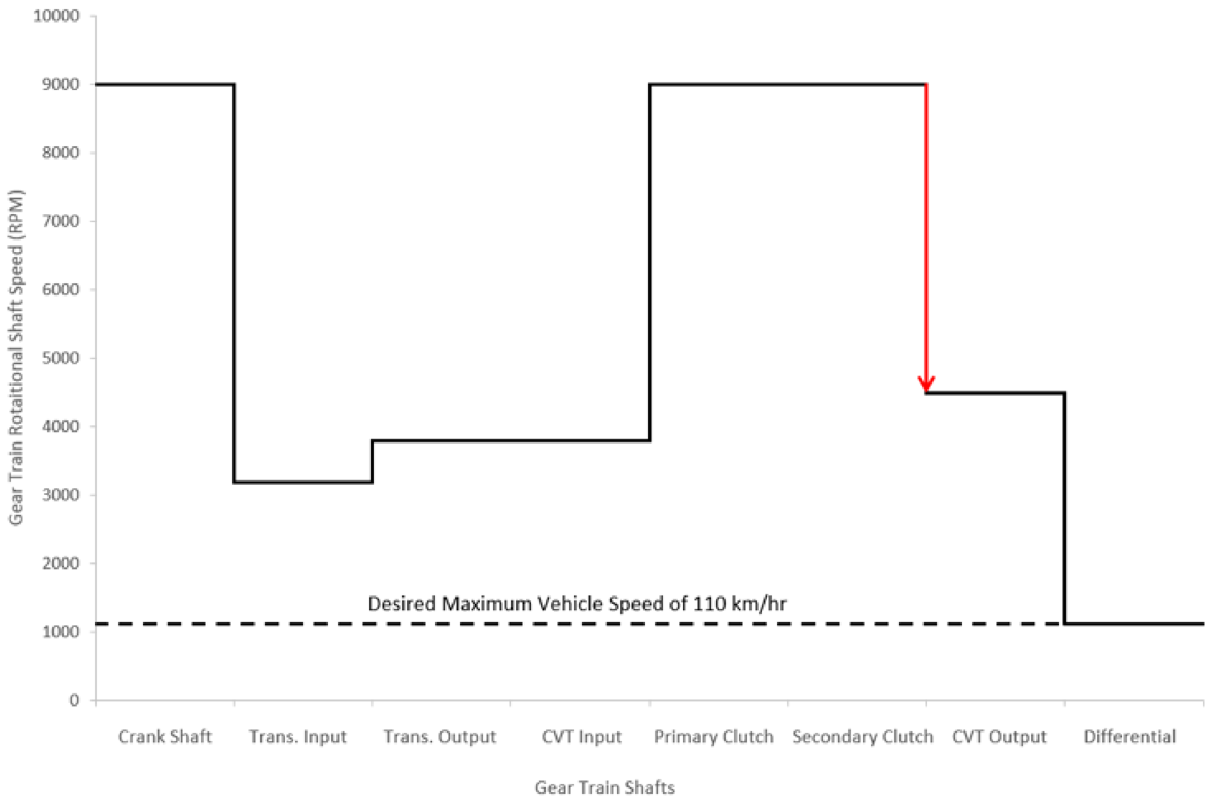

Figure 13 Rotational Speed Changes in the CVT due to Gear Reductions from the CVT Gearbox

To ensure the gear selected were sufficient for the design application a torque and stress analysis was completed and can be seen in Appendix A.2 . The maximum stress found for any of the gears used was 342 MPa. The yield stress for the gears is 410 MPa resulting in a safety factor of 1.2. This safety factor meets the minimum requirements based on the design compliance matrix.

3.5 Shaft Analysis

The shafts in the CVT gearbox were designed to handle the torque output from the client's engine at the required optimal speed. This was done by:

1) Determining the number of gears needed on each shaft for the gear train;
2) Conducting an FBD analysis on each shaft from the torque applied from the engine and bending moment/shear force diagrams from the gears and bearing supports;
3) Generating torque diagrams to locate regions on each shaft that experienced the highest torque;
4) Generating shear force and bending moment diagrams to locate regions with the largest force and moment;
5) Determining the diameters of the shaft in these critical locations and selecting the correct bearings to prevent system failure (based on DE-ASME elliptic criterion).
6) Conducting a stress concentration factor analysis for specific regions such as steps where the torque and moment are severe, splines, and keyways

See Appendix A for the detailed calculations and design process used for sizing the shafts and selecting the appropriate bearings. The shafts were analyzed using a safety factor of 1.5 for the design to account for any unforeseen loading conditions and potential changes to the power output of the engine.

3.7 Manufacturing Cost Analysis

The total cost of the CVT is $2,570 USD ($3,310 CAD). After reviewing the cost each subcomponent, the primary and secondary clutches make up the largest proportion of the total cost. Custom parts for the gearbox cost ~880 USD with approximately 30.5h needed for machining. Table 6 contains detailed description of the cost of the CVT.

Table 6 Detailed Cost Analysis of the CVT

Part	Source	Price (USD)	Quantity	Cost (USD)	Percentage
KSS3-15	QTC gears	$26.99	1	$26.99	1.05%
KSS3-36	QTC gears	$84.93	1	$84.93	3.30%
A 1C 1MYK30016	QTC gears	$35.66	1	$35.66	1.39%
A 1C 1MYK30032	QTC gears	$82.08	1	$82.08	3.19%
Gears Total				**$229.66**	**8.92%**
Shaft 1	Solidworks	$124.77	1	$124.77	4.85%
Shaft 2	Solidworks	$90.31	1	$90.31	3.51%
Shaft 3	Solidworks	$89.72	1	$89.72	3.49%
Shaft 4	Solidworks	$185.19	1	$185.19	7.19%
Shaft Total				**$489.99**	**19.03%**
BELT 9.370 IN / 238MM (Part # CV.JM52-3201-C)	CVTech	$83.45	1	$83.45	3.24%
Clutch PWB 80 (Part # CV.1100-0046)	CVTech	$393.66	1	$393.66	15.29%
Clutch (Driven) ID 1" Shaft (Part # CV.1130-3003)	CVTech	$330.67	1	$330.67	12.84%
BLOC BOMBÉ NOIR PWB 80 (Part # CV.1130-3003)	CVtech	$11.01	3	$33.03	1.28%
CAP (TREADED) PWB 80 (PQT 3) (Part # CV.1150-3001-3)	CVTech	$4.70	1	$4.70	0.18%
WEIGHT 5.5G PWB80 (QT 12) (Part # CV.1135-3001-12)	CVTech	$9.44	2	$18.88	0.73%
SPRING 800N. / 1100N. PWB 80 SERIE 0200 (Part # CV.1151-1132)	CVTech	$27.55	1	$27.55	1.07%

Part	Source	Price (USD)	Quantity	Cost (USD)	Percentage
CVT System Total				**$891.94**	**34.65%**
FAG QJ304-MPA Four Point Contact Bearing (For primary,secondary and output shaft)	Quality Bearings Online	$82.92	3	$248.76	9.66%
7008-B-2RS-TVP (For trans output shaft)	Applied	$138.57	1	$138.57	5.38%
52 mm Bearing Cap	Solidworks	$31.28	3	$93.84	3.65%
68 mm Bearing Cap	Solidworks	$162.02	1	$162.02	6.29%
Bearing Total				**$643.19**	**24.98%**
18-8 Stainless Steel Socket Head Screws (10/pack) (Part # 92196A218)	Mcmaster-Carr	$8.99	6	$53.94	2.10%
18-8 Stainless Steel Hex Nuts (100/pack) (Part # 91841A005)	Mcmaster-Carr	$2.61	1	$2.61	0.10%
General Purpose Washers (100/pack) (Part # 90107A005)	Mcmaster-Carr	$3.23	1	$3.23	0.13%
Frame Plate 1	Solidworks	$97.40	1	$97.40	3.78%
Frame Plate 2	Solidworks	$97.04	1	$97.04	3.77%
Gaskets (500x500mm sheet)	RAM	$14.50	1	$14.50	0.56%
O-ring (19-22 mm) 100/pack (Part # 9262K639)	Mcmaster-Carr	$7.81	1	$7.81	0.30%
O-ring (29-32 mm) 100/pack (Part # 9262K649)	Mcmaster-Carr	$13.40	2	$26.80	1.04%
O-ring (39-42 mm) 100/pack (Part # 9262K661	Mcmaster-Carr	$16.35	1	$16.35	0.64%
Casing Total				**$319.68**	**12.42%**
Total Cost				**$2,574.46**	**100.00%**

4 Design Sustainability and Environmental Impact

All the main components of the design (i.e. sheaves, belt, etc.) can be bought from local supply chains like Mcmaster-Carr and CVtech. Additionally, the CVT was designed for failure to occur at the belt first, since this would be the easiest component to replace.

The components of the CVT are made of aluminum, steel and rubber, all of which are not hazardous to the environment nor to employees manufacturing and machining any custom parts.

The CVT was designed to operate for 100 hours during competitions. Since the client will only operate the CVT for 50 hours during each yearly competition, the CVT only needs to be replaced every two years.

After two years, components like the engine and pulleys can still be reused. The custom shafts and plates for the gearbox will need to be replaced, but any other internal components can be bought from local suppliers.

5 Design Compliance Matrix

The design specification matrix was updated to the compliance matrix shown in Table 8, which replaces the conceptual design evaluation columns with a final design compliance column. This compliance matrix reports the qualitative and quantitative specifications achieved, and addresses the safety and regulatory requirements of the design. Also, the compliance matrix considers the environmental, social and ethical factors of the design and discusses the client expectations. After reviewing these factors with the client, approval was given for the final design. The rating system in Table 7 was created to establish a definitive meaning for each rating.

Table 7 Design Compliance Matrix Rating System

Rating of Importance	Description
5	Vital
4	Important
3	Desirable
2	Nice to have
1	Not important

A rating of 5 means that without meeting the specification the design is unusable; 4 is a high priority, but could potentially be worked around if other benefits are great enough; 3 carries significant weight but is not no longer vital; 2 means it is a nice feature though not technically necessary; and 1 implies that while the specification exists and is still applicable, it is not a significant factor for design success. Scores for the concept were decided upon by the design team via unanimous agreement on the relative score based on the quantitative/qualitative achievement and a comparison with the design specification

Table 8 Design Compliance Matrix

Description	Specification	Design Authority	Importance	Score (end of Phase 2)	Score (end of Phase 3)	Quantitative /Qualitative Achievement	Reference
Dimensions							
Mass	Minimum total mass	Client	4	7	2	24.0kg	CAD files
Size	Fit into chassis/within FSAE body parameters	Client/FSAE rules	5	4	4	431x344x331mm Fits in FSAE chassis	Appendix-D Drawing package
Rotational Inertia	Minimum possible rotational inertia	Client/Design Team	2	5	6	0.01719 kg*m^2	CAD files
Fasteners	Fasteners must be metric	Client	4	10	0	Gearbox fasteners metric. CVT fasteners imperial (CVTech)	CAD files
Safety							
Reliability	Component safety factor of 1.2 under competition operation	Design Team	4	7	10	1.5 safety factor used for shafts. 1.2 for gears	Appendix A

Description	Specification	Design Authority	Importance	Score (end of Phase 2)	Score (end of Phase 3)	Quantitative /Quantitative Achievement	Reference
Durability	50 engine hours operation	Client	4	N/A	N/A	Components selected for 100 hours' operation.	Appendix A
Shielding	Finger guard must cover all moving parts. Guard must prevent passage of 12mm diameter object through guard	Client	4	10	10	Gearbox Shields all gears. CVT shielding incorporated in FSAE body design.	
Cost							
Machinability Hours	Less than 10 total machining hours	Client	4	9	0	Total machinability time is 30.5 hours	Appendix-B
Total Cost	$2500 limit for sourced components	Client	5	8	0	$3,310 CAD	Section 3.7
Function							
Power	Range of vehicle speed operating at max power 7750rpm ~50HP	Design Team	5	10	10	goal achieved	Section 3.3
Stationary Operation	Capable of maintaining neutral for vehicle noise test in competition. Engine speed of 9000 rpm without moving	Client/Design Team	3	4	6	Belt removable via 2 bolts fastening pulleys	CAD File
Max Vehicle Speed	Sustained transmission operation at 110km/hour	Client	4	10	10	110km/h @ 9000rpm	Section 3.4
Transmission Tunability	Number of design variables that can be readily adjusted after transmission assembly	Design Team	3	8	6	3	
Installation/Manufacturing							
Mounting	Number of necessary mounts on chassis	Client	5	10	8	Indeterminate - up to FSAE team to install on chassis	
Material	Use readily available from local suppliers	Client	5	10	10	Yes - steel and aluminum	Appendix D
Maintenance							
Simplicity	Minimum number of electrical parts possible	Client	1	10	10	0	
Part Sourcing	Available for purchase in typical Canadian/American suppliers	Client	4	10	10	Parts source from Canadian/American Suppliers.	Section 3.7
Serviceability	Belt accessible and interchangeable within 30 minutes	Client	4	9	10	2 bolts removed to access/service belt	

The system designed failed to meet some of the design specifications. The design is over budget by $810 CAD and the total machine hours quoted is 30.5 exceeding the limit by 20.5 hours. The fasteners selected for the gearbox are all metric but the fasteners provided by CVTech for assembling the pulleys are imperial – 1/2x20 UNC screws (these come with the CVTech pulleys and are not included in the drawing files). The shielding of the CVTech pulleys was decided to be neglected. Incorporating shielding for the pulleys into the body of the FSAE vehicle will be necessary. The gearbox was designed from weldable steels so that mounting brackets can be added after the chassis design is finalized and the position of mounts can be decided upon. The pulley's specified by CVTech have been specifically selected for the FSAE vehicle. However, because actual performance can vary, the springs and weights can be easily adjusted. The helix can be modified if necessary though is not recommended.

The overall mass and inertia of the system are much larger than initially expected. The addition of the gearbox for the necessary gear additions/reductions resulted in a total mass of 24.0 kg. The expectation for this design is that the first component to fail will be the belt from wear. The shafts and gears were all designed to safety factors equal to or greater than the specified value of 1.2 and the bearings selected for operational lifetime of 100 hours. Overall the expectation is that the design will exceed the 50 hours of operation, including the belt.

The design does not incorporate any electrical components and all parts and materials are available from local suppliers (within North America). ATF fluid for the gear box can be added during installation either through one of the bolt openings or by removing the bearing cap and positioned vertically. To remove the belt, 1 bolt in each of the pulleys must be removed and can easily be serviced in under 30 minutes. Since the existing transmission clutch and gearbox has not been removed, the transmission can be set in neutral without any modifications.

6 Future Works, Research and Development

During the critical analysis of the dual clutch design, the team ran into obstacles that arose from limitations placed on the scope of the design. The addition of the gearbox can be largely if not entirely eliminated by mounting the CVT directly to the engine shaft. This is a complex modification to the engine that would require significant investment and time on the part of the FSAE Club. However, it would enable the transmission weight, rotational inertia, and cost to be reduced significantly and improve the overall performance of the vehicle assuming the engine modifications could be made successfully.

6.1 CVT Mounting/Shielding

The transmission does not come with any mounting brackets. The final dimensions and design of the chassis of the 2017/2018 competition vehicle was not known. Space was left on the gear box, which is made of carbon steel, such that it can be welded to the chassis or outfitted with a mounting system.

The gears are self-contained in the gearbox and are effectively shielded. The CVTech pulley's shielding must be added/incorporated into the vehicle body design by the FSAE team.

6.2 Engine

As previously mentioned, if the CVT were mounted directly to the client's engine the performance of the system could be improved. Team 13 researched into the suitability of another engine for the FSAE vehicle and found the 2007 Yamaha Phazer 500 engine is commonly used. This engine has slightly more horsepower but is slightly heavier since it is a 2-cylinder engine. Since the engine is designed to operate with a CVT it offers much greater compatibility with a custom continuously variable transmission.

7 Engineering Design Cost and Schedule

During Phase 3 several scheduling issues were encountered that were overcome. Initially Concept 2 had been recommended and planned for from Phase 2. Due to calculations that demonstrated the unsuitability of this concept the schedule and plan for Phase 3 had to be adjusted. This resulted in a significant increase in total hours due to the need to pivot and reinvest time in the appropriately selected concept. Additionally, the design conference offered its own challenges in the schedule that were overcome. Overall, the project was completed on time but with a greater amount of time invested to complete.

The total number of engineering hours invested in Phase 3 was 340 hours – 338 junior engineering hours and 2 intermediate hours. At a rate of $90/hour for junior engineers and $150/hour for intermediate the design cost for Phase 3 is $30,720. For the entire project (Phase 1 through 3) the number of hours and cost comes to 698 hours and $66,503. A graphical representation of the design hours and associated cost can be seen in Figure 14 and Figure 15.

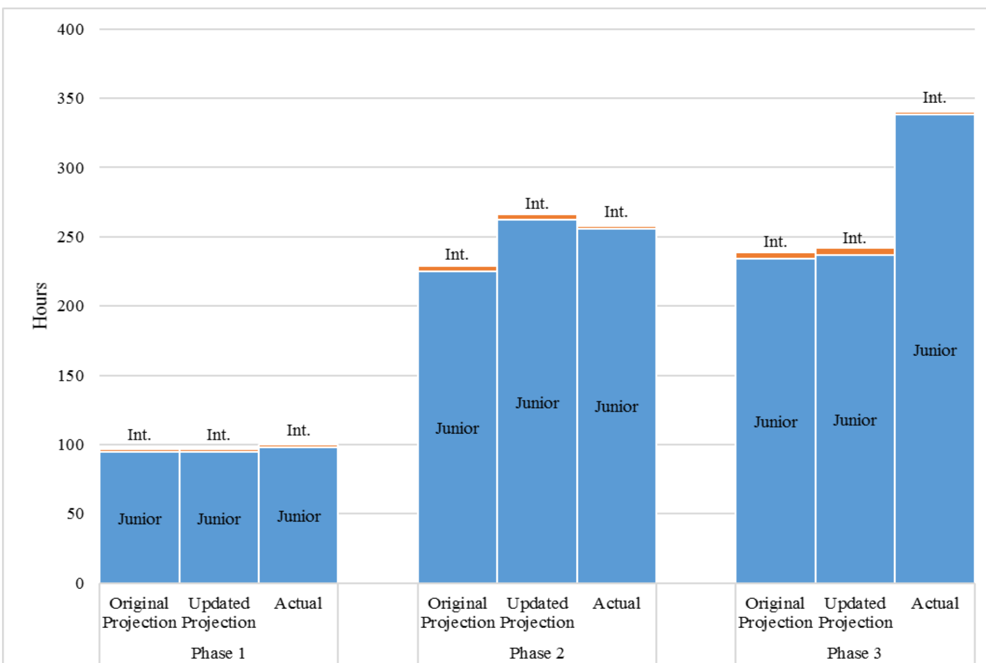

Figure 14 Design engineer hours for each phase including projections and actual hours

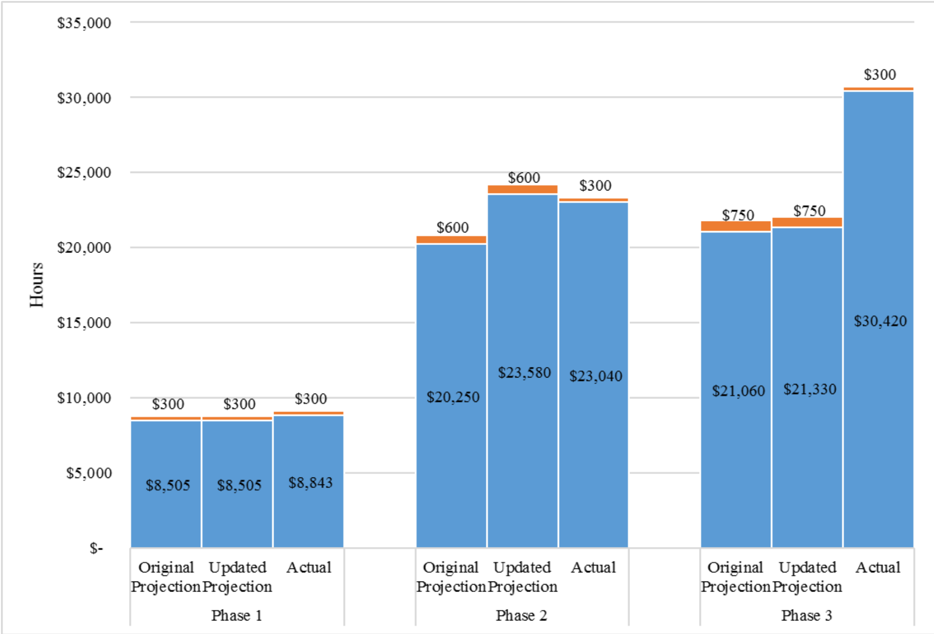

Figure 15 Design engineer cost (CAD) for each phase including projections and actual hours

8 Conclusion

With the Team 13 CVT Transmission design the performance of the FSAE vehicle in terms of the force applied to the ground and thereby the vehicle acceleration can be expected to improve. The design loading conditions were effectively managed in the analysis and design to ensure a reliable and effective design. While not all of the design specifications were met the design it does comply with all required standards. Team 13 recommends that investigations into more suitable engines and alternative engine mounting methods be considered in future CVT design work.

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Appendix A. Detailed Design Calculations

A.1. Belt Tension

Concept 2 Belt Force Analysis

Maximum force applid to road gives maximum torque that must be transmitted through the transmission belt. Therefore, solve for minimum belt tension to prevent belt slip considering this maximum load condition.

The industry standard based on the physical limitations of the system is to have a maximum spring preload force of 1 kN. The following calculations revealed the unsuitability of concept 2.

$F_{max} := \max\left(\mathrm{col}\left(GF_{C2}, 2\right)\right) = 4725.9453$ maximum torque Output of engine based on system gearing

$v_{Fmax} := \mathrm{col}\left(\mathrm{findrows}\left(GF_{C2}, \max\left(\mathrm{col}\left(GF_{C2}, 2\right)\right), 2\right), 1\right)_{1\,1} = 28.4108$

$F_{max.friction} := 0.8 \cdot \mu_{tire} \cdot m_v \cdot g = 3466.4101\ N$

Since the ground force due to friction is less than the max ground force based on the system gearing, the maximum friction force, also known as the traction limit will be used for torque calculations

The following calculations were completed iteratively. The additional gear reduction after the driven pulley was assumed and the process was iterated until a suitable maximum spring preload force was found.

$\boxed{gr_{red} := \frac{1}{28}}$

Additional reduction needed between driven pulley and drive axle
This value was selected to achieve the value that follows (gr_add) equal to 1 which represents the additional gear reduction needed to have a spring preload force within the physical limit previously outline of 1 kN

Torque applied to driven pulley:

$$T_{driven} := F_{max.friction} \cdot \frac{d_T}{2} \cdot gr_{red} = 32.2314\ N\ m$$

Assume a sheave angle measured from the radial axis and a sheave radius

$\alpha_s := 10\ deg$ $\boxed{\mu_{belt} := 0.15}$

$d_{sheave} := (200)\ mm$

Note that the maximum value for the belt coefficient of friction is as follows:

$$\mu_{belt.max} := \tan\left(\alpha_s\right) = 0.1763$$

This condition arises from the self locking behavior of a belt that has a higher coefficient of friction. Put simply, if the coefficient of friction is higher than this value, the belt will not shift under any clamping conditions. The coefficient used is based on typical rubber belt-aluminum coefficients of friction selected from engineering toolbox.

From CVT design equations we know that the clamping force for the driven pulley is:

$$F_{driven} = \frac{Torque_{out}}{\mu_{belt} \cdot d_2} \cdot \cos\left(\alpha_s\right)$$

Solving for the driving force which yields the minimum belt tension created by the driven pulley spring:

$$F_{preload} := \frac{T_{driven}}{\mu_{belt} \cdot d_{sheave}} \cdot \cos\left(\alpha_s\right) = 1058.0586\,N$$

This force represents the sheave clamping force on the belt that must be applied before the system starts to shift. For a variator design this force is stricly due to the preload on the spring. This preload force is much too large for a CVT compression spring.

Assume max preload possible for the compression spring

$$F_{preload.max} := 1\,kN$$

Solve for a gear reduction to achieve a reasonable load for the driven pulley

$$gr_{add} := \frac{F_{preload}}{F_{preload.max}} = 1.0581$$

$$gr_{red} = \frac{1}{28}$$

This reduction is far too large to be added after the transmission. The number of gears necessary to achieve this reduction was deemed unfeasible by the client and the design team. It was also discovered that Concept 2 designs are generally limited to applications where rapid acceleration is not required and the power transmission is far lower than the FSAE vehicle applicatoin.

THEREFORE CONCEPT 2 DESPITE BEING RECOMMENDED WAS FOUND UNVIABLE.
The decision to pivot to Concept 1, which includes a helix to add additional belt clamping force was made and is frequently used in performance applications.

Concept 1 Belt Force Analysis

Key Belt Calculations Must Occur during the maximum force transmission since this is the scenario in which the belt is under the greatest tension. This situation corresponds to the region of the gorund force curve where the vehicle power exceeds the traction limit of the vehicle. Assuming the vehicle is operated effectively, the ground force for this entire region will be approximately equal to the traction limit. During this time the CVT is primarily in "Low Gear"

CV transmission physical parameters estimated from CVTech pulleys.

$\alpha_s := 10\ deg$ Sheave Angle

$d_{sheave} := 200\ mm$ Sheave Diameter

$\mu_{belt} := 0.15$ Belt/Sheave friction coefficient

$\alpha_h := 42\ deg$ Helix Angle

$r_{h1} := 50\ mm$ Helix radius

$\mu_{helix} := 0.5$ Helix/roller friction coefficient

$m_v := \dfrac{573\ lbf}{9.81\ \dfrac{m}{s^2}} = 259.8197\ kg$ Vehicle Mass

$\mu_{tire} := 1.7$ Tire/road friction coefficient

$g := 9.81\ \dfrac{m}{s^2}$ Gravity

Condition for Belt to shift: $\boxed{\tan\left(\alpha_s\right) > \mu_{belt}}$

$\tan\left(\alpha_s\right) = 0.1763$

This geometric condition for the belt to not be locked in place by friction severely limits the torque capacity of a belt driven CVT

To properly analyze the belt, the maximum force applied to the wheels which corresponds to the maximum torque applied to the transmission should be determined.

The maximum force corresponds to the minimum when comparing the traction limit for the vehicle and the peak force from the ground force curve above. This is because regardless of how much force the transmission is capable of transmitting, the force experienced by the transmssion cannot physically exceed the traction limit:

$F_{max.GF} := \max\left(\mathrm{col}\left(GF_{C1.2}, 2\right)\right) = 4312.0176$ Maximum ground force applied by the transmission (regardless of traction limit)

$F_{max.friction} := 0.8 \cdot \mu_{tire} \cdot m_v \cdot g = 3466.4101\ N$ Traction limit due to friction between tires and asphalt

Therefore, $F_{max} := \min\left(\left[F_{max.GF}N \;\; F_{max.friction}\right]\right) = 3466.4101\,N$

The maximum torque experienced by the driven clutch can then be determined along with the axial force, F_driven, that must be applied to prevent the belt from slipping

$$T_{driven} := F_{max} \cdot \frac{d_T}{2} \cdot gr_{add} = 150.4133\,N\,m$$

$$F_{driven} := \frac{T_{driven}}{\mu_{belt} \cdot d_{sheave}} \cdot \cos\left(\alpha_s\right) = 4937.6066\,N$$

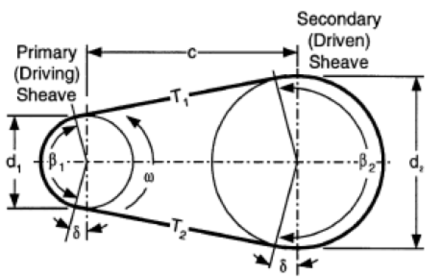

FIGURE 13—BELT LENGTH DIAGRAM

CV ratio for maximum force condition is the low gear ratio, c1

$c_1 = 0.2857$

$d_2 := d_{sheave} = 200\,mm$

$d_1 := d_2 \cdot c_1 = 57.1429\,mm$

Pulley center distance as determined by physical vehicle constraints

$c := 205\,mm$

included angle, delta

$$\delta := \operatorname{asin}\left(\frac{d_1}{2 \cdot c} \cdot \left(\frac{1}{c_1} - 1\right)\right)$$

$$\delta = 20.3914\,deg$$

wrap angles, beta_1 and beta_2

$$\beta_1 := 180\,deg - 2 \cdot \delta$$

$$\beta_1 = 139.2171\,deg$$

$$\beta_2 := 180\,deg + 2 \cdot \delta$$

$$\beta_2 = 220.7829\,deg$$

$$\boxed{OutputTorque := \left(T_1 - T_2\right) \cdot \frac{d_{sheave}}{2}}$$

Using the above torque relation the tensile forces on the belt can be solved for since the output torque expected from the transmission is known - T_driven

$$T_1 := \frac{T_{driven}}{\frac{d_2}{2} \cdot \left(1 - \exp\left(-\frac{\mu_{belt} \cdot \beta_2}{\sin\left(\alpha_h\right)}\right)\right)} = 2600.2766\,N$$

Equations from SAE 730003

$$T_2 := T_1 - \frac{T_{driven}}{\frac{d_2}{2}} = 1096.1434\,N$$

Equations from SAE 730003

From the torque output (also the belt tension difference) using the following equation, the axial force from the helix is determined:

$$F_a := \frac{1}{2 \cdot r_{h1}} \cdot \left(\frac{T_1 - T_2}{2} \cdot d_{sheave}\right) \cdot \left(\frac{\cos\left(\alpha_h\right) + \mu_{helix} \cdot \sin\left(\alpha_h\right)}{\sin\left(\alpha_h\right) - \mu_{helix} \cdot \cos\left(\alpha_h\right)}\right) = 5447.7395\,N$$

The Cantilever force or "Center Pull Force" applied to the shaft bearings is:

$$CP_{driven} := \left(T_1 + T_2\right) \cdot \cos\left(\delta\right)$$

$$CP_{driven} = 3464.7803\,N$$

Completing a force balance on the belt, the center pull force will be approximately the same for driven and drive pulleys. This force represents the cantilever force applied to the CVT shafts and is important for bearing selection and shaft analysis

$$\boxed{CP_{drive} := CP_{driven} = 3464.7803\,N}$$

$$F_{s.preload} := F_{driven} - F_a$$

$F_{s.preload} = -510.1329\,N$ or in pounds $F_{s.preload} = -114.6824\,lbf$

**Note that these forces correspond to a helix angle and diamter as set above. These ought to be adjusted once the helix angle and diameter for the pwb80 CVTech drive clutch are known.

Evaluate the necessary flyweight mass based on a engine shift speed of 7750 rpm

$$\omega_{opt} = 7750\ rpm$$

$$v_{v1}(\omega_{opt})_1 = 34.974\ \frac{km}{hr}$$

$$T_{driven} := \frac{HP_{opt}}{v_{v1}(\omega_{opt})_1} = 3837.8838\ N$$

$$T_1 = 2600.2766\ N$$

$$T_2 = 1096.1434\ N$$

Belt tensions using equation above

$$F_a = 5447.7395\ N$$

Helix axial force

Additionally, the necessary clamping force for this condition can be solved for the drive pulley using:

$$F_r := \frac{T_1 \cdot \beta_1}{2} \cdot \left(\frac{1 - \mu_{belt} \cdot \tan(\alpha_s)}{\mu_{belt} + \tan(\alpha_s)} \right)$$

$$F_r = 9424.6503\ N$$

Select the primary spring weights such that they achieve F_r at 7750 rpm which dictates the CVT's shift rpm.

Given the starting radius of the weights, the weight pressure angle (translating centrifugal force to axial force) and the angular velocity of the drive pulley:

$$r_w := 75\ mm$$

$$\alpha_w := 60\ deg$$

$$\omega_{dp} := \omega_{opt}$$

Therefore, the total mass of the weights to achieve the desired shift rpm is:

$$m_w := \frac{F_r}{\omega_{dp}^2 \cdot r_w \cdot \tan(\alpha_w)} = 0.1101\ kg$$

Given the mass of the flyweights to achieve the approximate engagement rpm, the primary spring preload will be selected to achieve the desired engagement rpm

$$\omega_{eg} := 2400\ rpm$$

$$\omega_{dp} := \omega_{eg}$$

$$F_r := \omega_{dp}^2 \cdot r_w \cdot m_w \cdot \tan(\alpha_w) = 903.8249\ N$$

$$F_{p.preload} := F_r = 903.8249\ N$$

The actual values used were determined through consulatation with CVTech's engineers

The PWB80 Drive and Driven pulleys were selected, the preload on the primary spring selected was 800 N and the total mass for the weights selected was 305.4 g. Note that the weights radius and angle are different for the CVTech pulley.

The specifications for the PWB80 pulley were unable to be acquired. The above process for the belt force calculations and the drive clutch tuning parameters may be repeated if necessary once the physcial parameters of the pulley's are known

Alternatively, if a fully custom set of Pulley's is desired, the above procedure can be used to appropriately design some of the physical parameters of the system

It should be noted that the actual radius of the PWB80 pulley for the flyweights to travel along is not linear. The axial force generated by the weights is a complex function of radial position and the contact angle with the pulleys. Further studies could be made into how these variables influence the behavior of the clutch.

A.2. Gear Analysis

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Gear Ratios

The Primary Reduction, Fifth Gear Reduction, and Sixth Gear Reduction are all on the engine input shaft . The Final Gear Reduction is after the CVT Reduction to lower the rotational speed and achieve the desired optimal vehicle speed. The gear ratios and gear analysis were done for sixth gear and final gear 1 because those two gears were not provided by the client.

Primary Reduction

$pr := \frac{22}{62}$

Fifth Gear Reduction

$g_5 := \frac{25}{21}$

Sixth Gear Reduction

$\boxed{g_6 := \frac{32}{14}}$

CVT Ratios

Low Gear

$c_1 := \frac{1}{3.5}$

High Gear

$c_2 := \frac{1}{1}$

Final Gear Reduction

$g_f := g_{f1} \cdot g_{f2}$ $\boxed{g_{f1} := \frac{16}{32}}$ $g_{f2} := \frac{1}{3}$

Two gear ratios, c1 and c2, were calculated for the CVT transmission c1 and c2 are the minimum and maximum gear ratios respectivly. The two gear ratio calculations below are based on c1 and c2.

Total gear reduction ratio with CVT low gear rato

$Ratio_{low} := pr \cdot g_5 \cdot g_6 \cdot c_1 \cdot g_f$

$Ratio_{low} = 0.046$

Total gear reduction ratio with CVT high gear ratio

$Ratio_{high} := pr \cdot g_5 \cdot g_6 \cdot c_2 \cdot g_f$

$Ratio_{high} = 0.1609$

Veichle Engine Parameters

These are some basic parameters. For rotational speed, maximum speed optimal speed and engagement speed were considered. Optimal horse power is shown in the power section. Tire diameter was found to be 20.5 inches.

Rotational Speed

$\omega_{max} := 9000\ rpm$

$\omega_{opt} := 7750\ rpm$

$\omega_{eg} := 4000\ rpm$

Power

$HP_{opt} := 60\ hp$

Tire Diameter

$d_T := 20.5\ in$

1 / 14

Tire Perimeter

$P := \pi \cdot d_T = 1.6358\ m$

The vehicle wheel angular speeds were calculated using engine angular speeds times the total gear reduction ratio.

Vehicle Wheel Angular Speed

Engagement

$\omega_{Engagement} := Ratio_{low} \cdot \omega_{eg} = 19.2594\ Hz$

Optimal rpm

Low gear

$\omega_{OptimalLOW} := Ratio_{low} \cdot \omega_{opt} = 37.3151\ Hz$

High gear

$\omega_{OptimalHIGH} := Ratio_{high} \cdot \omega_{opt} = 130.6029\ Hz$

Shift Out

Low gear

$\omega_{MaxLOW} := Ratio_{low} \cdot \omega_{max} = 43.3337\ Hz$

High gear

$\omega_{MaxHIGH} := Ratio_{high} \cdot \omega_{max} = 151.6678\ Hz$

The vehicle wheel linear speeds were calculated using the total reduction ratio times the wheel angular speeds and the radius of the wheel. The radius of the wheel was expressed as tire diameter divided by two.

For the vehicle wheel linear speed, 3 conditions were considered: engine engagement, optimal speed, and maximum speed. For both optimal and maximum condition, both low gear ratio and high gear ratio situations were considered.

Vehicle Wheel Linear Speed

Engagement

$$v_{Engagement1} := Ratio_{low} \cdot \omega_{eg} \cdot \frac{d_T}{2} = 18.0511 \frac{km}{hr}$$

Optimal rpm

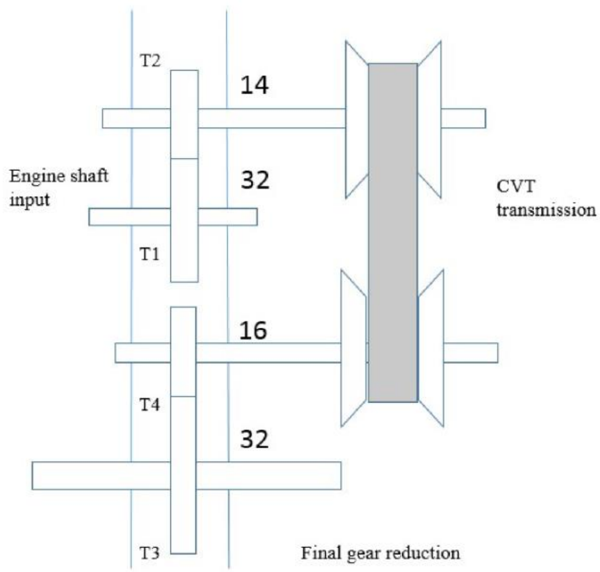
$$v_{OptimalLOW1} := Ratio_{low} \cdot \omega_{opt} \cdot \frac{d_T}{2} = 34.974 \frac{km}{hr}$$

$$v_{OptimalHIGH1} := Ratio_{high} \cdot \omega_{opt} \cdot \frac{d_T}{2} = 122.4088 \frac{km}{hr}$$

Shift Out

$$v_{MaxLOW1} := Ratio_{low} \cdot \omega_{max} \cdot \frac{d_T}{2} = 40.6149 \frac{km}{hr}$$

$$v_{MaxHIGH1} := Ratio_{high} \cdot \omega_{max} \cdot \frac{d_T}{2} = 142.1522 \frac{km}{hr}$$



As shown in the above figure, the four gears are named gear 1, gear 2, gear 3 and gear 4, with tooth number T1, T2, T3 and T4 respectively.

Teeth Number

$T_1 := 32$ $T_2 := 14$

$T_3 := 32$ $T_4 := 16$

The gear used have a modulus of 3. For the sake of calculating the pitch diameter from the number of teeth they were assigned units of meters.

Modulus

$m_1 := 0.003\,m$ $m_2 := 0.003\,m$

$m_3 := 0.003\,m$ $m_4 := 0.003\,m$

Reference Pressure Angle

This information wass taken from the QTC gear webite Product Specifications.

$\alpha_1 := 20\,°$ $\alpha_2 := 20\,°$ $\alpha_3 := 20\,°$ $\alpha_4 := 20\,°$

Profile Shift Coefficient

For standard gears the coefficient was 0.

$x_1 := 0$ $x_2 := 0$ $x_3 := 0$ $x_4 := 0$

Working Pressure Angle

The pressure angles were assumed to be the same as the reference pressure angles for standard gears.

$\alpha_{w1} := 20\,°$ $\alpha_{w2} := 20\,°$ $\alpha_{w3} := 20\,°$ $\alpha_{w4} := 20\,°$

Because the working pressure angles are the same as the reference presure angles, the center to center distance modifition coefficient will be 0. If in the real application the working pressure angles are found to be different. These calculations can easily be adjusted.

Center Distance Modification Coefficient

$$y_{12} := \frac{T_1 + T_2}{2} \cdot \left(\frac{\cos\left(\alpha_1\right)}{\cos\left(\alpha_{w1}\right)} - 1 \right) = 0 \qquad y_{34} := \frac{T_3 + T_4}{2} \cdot \left(\frac{\cos\left(\alpha_3\right)}{\cos\left(\alpha_{w3}\right)} - 1 \right) = 0$$

Center to Center Distance

$$a_{12} := \left(\frac{T_1 + T_2}{2} + y_{12} \right) \cdot m_1 = 0.069\,m \qquad a_{34} := \left(\frac{T_3 + T_4}{2} + y_{34} \right) \cdot m_3 = 0.072\,m$$

Reference diamater is also known as pitch diameter. It is obtained by multiplying the number of teeth by the module number.

Reference Diameter

$D_1 := T_1 \cdot m_1 = 0.096\,m$ $\quad$ $D_2 := T_2 \cdot m_2 = 0.042\,m$

$D_3 := T_3 \cdot m_3 = 0.096\,m$ $\quad$ $D_4 := T_4 \cdot m_4 = 0.048\,m$

Base Diameter

$D_{b1} := D_1 \cdot \cos\left(\alpha_1\right) = 0.0902\,m$ $\quad$ $D_{b2} := D_2 \cdot \cos\left(\alpha_2\right) = 0.0395\,m$

$D_{b3} := D_3 \cdot \cos\left(\alpha_3\right) = 0.0902\,m$ $\quad$ $D_{b4} := D_4 \cdot \cos\left(\alpha_4\right) = 0.0451\,m$

In this case the working pitch diameters were the same as the pitch diameter. This is true because the working pressure angles were same as pressure angles.

Working Pitch Diameter

$$D_{w1} := \frac{D_{b1}}{\cos\left(\alpha_{w1}\right)} = 0.096\,m \qquad D_{w2} := \frac{D_{b2}}{\cos\left(\alpha_{w2}\right)} = 0.042\,m$$

$$D_{w3} := \frac{D_{b3}}{\cos\left(\alpha_{w3}\right)} = 0.096\,m \qquad D_{w4} := \frac{D_{b4}}{\cos\left(\alpha_{w4}\right)} = 0.048\,m$$

For the standard gears, addendum is the same as the module.

Addendum

$h_{a1} := \left(1 + y_{12} - x_2\right) \cdot m_1 = 0.003\,m$ $\quad$ $h_{a2} := \left(1 + y_{12} - x_1\right) \cdot m_2 = 0.003\,m$

$h_{a3} := \left(1 + y_{34} - x_4\right) \cdot m_3 = 0.003\,m$ $\quad$ $h_{a4} := \left(1 + y_{34} - x_3\right) \cdot m_4 = 0.003\,m$

The inner diameter was set to 25mm, the gears will be machined to have 25mm bores.

Inner Diameter

$D_{i1} := 0.025\,m$ $\quad$ $D_{i2} := 0.025\,m$

$D_{i3} := 0.025\,m$ $\quad$ $D_{i4} := 0.025\,m$

Tooth Depth

$h_1 := \left(2.25 + y_{12} - \left(x_1 + x_2\right)\right) \cdot m_1 = 0.0068\,m$ $\quad$ $h_2 := \left(2.25 + y_{12} - \left(x_1 + x_2\right)\right) \cdot m_2 = 0.0068\,m$

$h_3 := \left(2.25 + y_{34} - \left(x_3 + x_4\right)\right) \cdot m_3 = 0.0068\,m$ $\quad$ $h_4 := \left(2.25 + y_{34} - \left(x_3 + x_4\right)\right) \cdot m_4 = 0.0068\,m$

Tip diameters were obtained by adding the reference diameter to two tooth depths

Tip Diameter

$D_{t1} := D_1 + 2 \cdot h_{a1} = 0.102\ m$ $\quad$ $D_{t2} := D_2 + 2 \cdot h_{a2} = 0.048\ m$

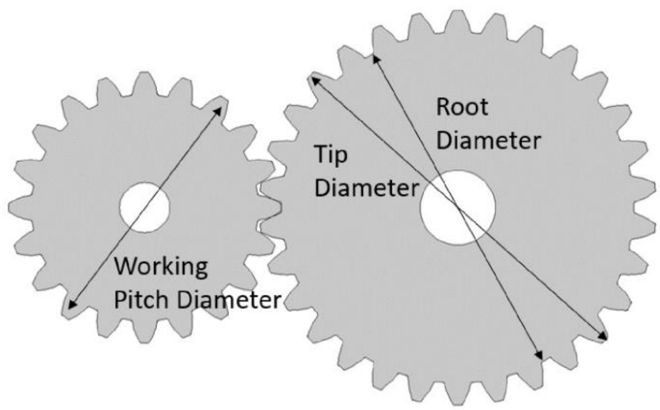
$D_{t3} := D_3 + 2 \cdot h_{a3} = 0.102\ m$ $\quad$ $D_{t4} := D_4 + 2 \cdot h_{a4} = 0.054\ m$

Root diameters were obtained by subtracting two tooth depths from the reference diameter

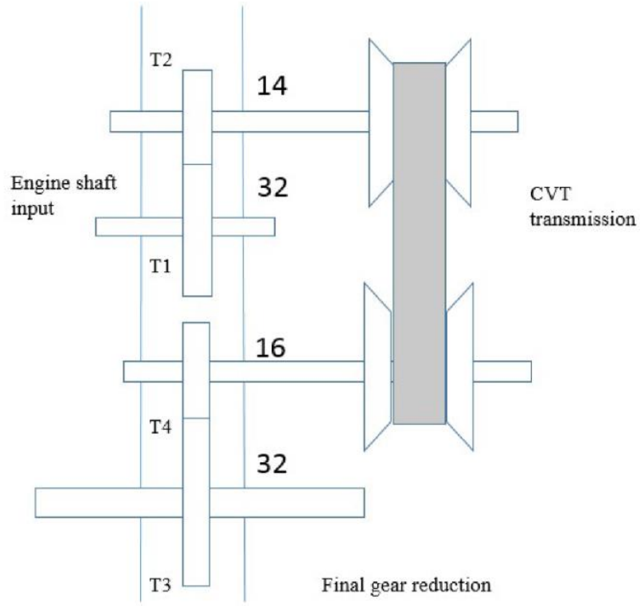
Root Diameter

$D_{r1} := D_{t1} - 2 \cdot h_1 = 0.0885\ m$ $\quad$ $D_{r2} := D_{t2} - 2 \cdot h_2 = 0.0345\ m$

$D_{r3} := D_{t3} - 2 \cdot h_3 = 0.0885\ m$ $\quad$ $D_{r4} := D_{t4} - 2 \cdot h_4 = 0.0405\ m$



This figure illustrats the root diameter, tip diameter, and working pitch diameter



The angular speed of gear 1 is the same speed as the engine output shaft.

Angular Speed of Gears

Gear 1

$\omega_{max\_g1} := 9000\ rpm$

$\omega_{opt\_g1} := 7750\ rpm$

$\omega_{eg\_g1} := 4000\ rpm$

Gear 2

$\omega_{max\_g2} := \omega_{max\_g1} \cdot g_5 = 10714.2857\ rpm$

$\omega_{opt\_g2} := \omega_{opt\_g1} \cdot g_5 = 9226.1905\ rpm$

$\omega_{eg\_g2} := \omega_{eg\_g1} \cdot g_5 = 4761.9048\ rpm$

Gear 3

Low Gear

$\omega_{max\_g3\_low} := \omega_{max\_g2} \cdot c_1 = 3061.2245\ rpm$

$\omega_{opt\_g3\_low} := \omega_{opt\_g2} \cdot c_1 = 2636.0544\ rpm$

$\omega_{eg\_g3\_low} := \omega_{eg\_g2} \cdot c_1 = 1360.5442\ rpm$

High Gear

$\omega_{max\_g3\_high} := \omega_{max\_g2} \cdot c_2 = 10714.2857\ rpm$

$\omega_{opt\_g3\_high} := \omega_{opt\_g2} \cdot c_2 = 9226.1905\ rpm$

$\omega_{eg\_g3\_high} := \omega_{eg\_g2} \cdot c_2 = 4761.9048\ rpm$

Gear 4

Low Gear

$\omega_{max\_g4\_low} := \omega_{max\_g3\_low} \cdot g_{f1} = 1530.6122\ rpm$

$\omega_{opt\_g4\_low} := \omega_{opt\_g3\_low} \cdot g_{f1} = 1318.0272\ rpm$

$\omega_{eg\_g4\_low} := \omega_{eg\_g3\_low} \cdot g_{f1} = 680.2721\ rpm$

High Gear

$\omega_{max\_g4\_high} := \omega_{max\_g3\_high} \cdot g_{f1} = 5357.1429\ rpm$

$\omega_{opt\_g4\_high} := \omega_{opt\_g3\_high} \cdot g_{f1} = 4613.0952\ rpm$

$\omega_{eg\_g4\_high} := \omega_{eg\_g3\_high} \cdot g_{f1} = 2380.9524\ rpm$

These are some other basic dimensions for the gears.

Pitch of the Gears

Gear 1

$P_1 := \frac{T_1}{D_{w1}} = 333.3333 \cdot \frac{1}{m}$

Gear 2

$P_2 := \frac{T_2}{D_{w2}} = 333.3333 \cdot \frac{1}{m}$

Gear 3

$P_3 := \frac{T_3}{D_{w3}} = 333.3333 \cdot \frac{1}{m}$

Gear 4

$P_4 := \frac{T_4}{D_{w4}} = 333.3333 \cdot \frac{1}{m}$

Diameter Pitch

Gear 1

$dP_1 := \frac{1}{P_1} = 0.003\ m$

Gear 2

$dP_2 := \frac{1}{P_2} = 0.003\ m$

Gear 3

$dP_3 := \frac{1}{P_3} = 0.003\ m$

Gear 4

$dP_4 := \frac{1}{P_4} = 0.003\ m$

6 Dec 2017 17:05:43 - gear analysis3.sm

Tooth bending Stress calculation

Torque Calculations

Gear 1

Engagement

$$Tor_{eg\_g1} := \frac{HP_{opt}}{\omega_{eg\_g1}} = 106.8136\ m\ N$$

Optimal

$$Tor_{opt\_g1} := \frac{HP_{opt}}{\omega_{opt\_g1}} = 55.1296\ m\ N$$

Maximum

$$Tor_{max\_g1} := \frac{HP_{opt}}{\omega_{max\_g1}} = 47.4727\ m\ N$$

Gear 2

Engagement

$$Tor_{eg\_g2} := \frac{HP_{opt}}{\omega_{eg\_g2}} = 89.7235\ m\ N$$

Optimal

$$Tor_{opt\_g2} := \frac{HP_{opt}}{\omega_{opt\_g2}} = 46.3089\ m\ N$$

Maximum

$$Tor_{max\_g2} := \frac{HP_{opt}}{\omega_{max\_g2}} = 39.8771\ m\ N$$

Gear 3

Engagement

Low Gear

$$Tor_{eg\_g3\_low} := \frac{HP_{opt}}{\omega_{eg\_g3\_low}} = 314.0321\ m\ N$$

High Gear

$$Tor_{eg\_g3\_high} := \frac{HP_{opt}}{\omega_{eg\_g3\_high}} = 89.7235\ m\ N$$

Optimal

Low Gear

$$Tor_{opt\_g3\_low} := \frac{HP_{opt}}{\omega_{opt\_g3\_low}} = 162.0811\ m\ N$$

High Gear

$$Tor_{opt\_g3\_high} := \frac{HP_{opt}}{\omega_{opt\_g3\_high}} = 46.3089\ m\ N$$

Maximum

Low Gear

$$Tor_{max\_g3\_low} := \frac{HP_{opt}}{\omega_{max\_g3\_low}} = 139.5698\ m\ N$$

High Gear

$$Tor_{max\_g3\_high} := \frac{HP_{opt}}{\omega_{max\_g3\_high}} = 39.8771\ m\ N$$

Gear 4

Engagement

Low Gear

$$Tor_{eg\_g4\_low} := \frac{HP_{opt}}{\omega_{eg\_g4\_low}} = 628.0642\ m\ N$$

High Gear

$$Tor_{eg\_g4\_high} := \frac{HP_{opt}}{\omega_{eg\_g4\_high}} = 179.4469\ m\ N$$

Optimal

Low Gear

$$Tor_{opt\_g4\_low} := \frac{HP_{opt}}{\omega_{opt\_g4\_low}} = 324.1622\ m\ N$$

High Gear

$$Tor_{opt\_g4\_high} := \frac{HP_{opt}}{\omega_{opt\_g4\_high}} = 92.6178\ m\ N$$

Maximum

Low Gear

$$Tor_{max\_g4\_low} := \frac{HP_{opt}}{\omega_{max\_g4\_low}} = 279.1396\ m\ N$$

High Gear

$$Tor_{max\_g4\_high} := \frac{HP_{opt}}{\omega_{max\_g4\_high}} = 79.7542\ m\ N$$

Tangential Load

Gear 1

Engagement

$$W_{t\_eg\_g1} := 2 \cdot \frac{Tor_{eg\_g1}}{D_{w1}} = 2225.2841\ N$$

Optimal

$$W_{t\_opt\_g1} := 2 \cdot \frac{Tor_{opt\_g1}}{D_{w1}} = 1148.5337\ N$$

Maximum

$$W_{t\_max\_g1} := 2 \cdot \frac{Tor_{max\_g1}}{D_{w1}} = 989.0152\ N$$

Gear 2

Engagement

$$W_{t\_eg\_g2} := 2 \cdot \frac{Tor_{eg\_g2}}{D_{w1}} = 1869.2387\ N$$

Optimal

$$W_{t\_opt\_g2} := 2 \cdot \frac{Tor_{opt\_g2}}{D_{w1}} = 964.7683\ N$$

Maximum

$$W_{t\_max\_g2} := 2 \cdot \frac{Tor_{max\_g2}}{D_{w1}} = 830.7727\ N$$

Gear 3

Engagement

Low Gear

$$W_{t\_eg\_g3\_low} := 2 \cdot \frac{Tor_{eg\_g3\_low}}{D_{w1}} = 6542.3354\ N$$

High Gear

$$W_{t\_eg\_g3\_high} := 2 \cdot \frac{Tor_{eg\_g3\_high}}{D_{w1}} = 1869.2387\ N$$

Optimal

Low Gear

$$W_{t\_opt\_g3\_low} := 2 \cdot \frac{Tor_{opt\_g3\_low}}{D_{w1}} = 3376.6892\ N$$

High Gear

$$W_{t\_opt\_g3\_high} := 2 \cdot \frac{Tor_{opt\_g3\_high}}{D_{w1}} = 964.7683\ N$$

Maximum

Low Gear

$$W_{t\_max\_g3\_low} := 2 \cdot \frac{Tor_{max\_g3\_low}}{D_{w1}} = 2907.7046\ N$$

High Gear

$$W_{t\_max\_g3\_high} := 2 \cdot \frac{Tor_{max\_g3\_high}}{D_{w1}} = 830.7727\ N$$

Gear 4

Engagement

Low Gear

$$W_{t\_eg\_g4\_low} := 2 \cdot \frac{Tor_{eg\_g3\_low}}{D_{w1}} = 6542.3354\ N$$

High Gear

$$W_{t\_eg\_g4\_high} := 2 \cdot \frac{Tor_{eg\_g3\_high}}{D_{w1}} = 1869.2387\ N$$

Optimal

Low Gear

$$W_{t\_opt\_g4\_low} := 2 \cdot \frac{Tor_{opt\_g3\_low}}{D_{w1}} = 3376.6892\ N$$

High Gear

$$W_{t\_opt\_g4\_high} := 2 \cdot \frac{Tor_{opt\_g3\_high}}{D_{w1}} = 964.7683\ N$$

Maximum

Low Gear

$$W_{t\_max\_g4\_low} := 2 \cdot \frac{Tor_{max\_g3\_low}}{D_{w1}} = 2907.7046\ N$$

High Gear

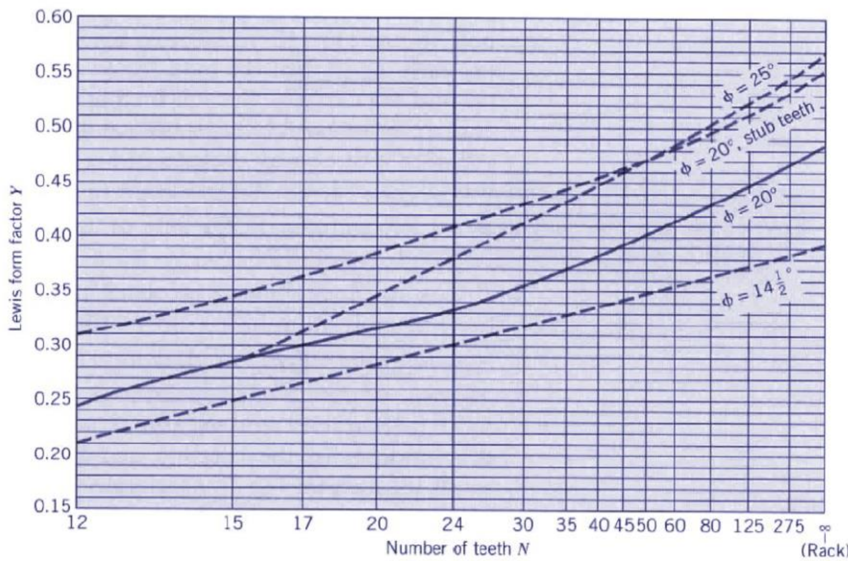
$$W_{t\_max\_g4\_high} := 2 \cdot \frac{Tor_{max\_g3\_high}}{D_{w1}} = 830.7727\ N$$

Face Width of the gears

The gear face width data was obtianed from the QTC gears webite.

$F_1 := 0.02\ m$ $F_2 := 0.022\ m$ $F_3 := 0.02\ m$ $F_4 := 0.022\ m$

Lewis Form Factor



The below data was obtained from the above graph

$Y_1 := 0.35$ $Y_2 := 0.28$ $Y_3 := 0.35$ $Y_4 := 0.29$

Lewis Bending Stress

Gear 1

Engagement

$$\sigma_{t\_eg\_g1} := \frac{W_{t\_eg\_g1} \cdot P_1}{F_1 \cdot Y_1} = 1.0597 \cdot 10^8\ Pa$$

Optimal

$$\sigma_{t\_opt\_g1} := \frac{W_{t\_opt\_g1} \cdot P_1}{F_1 \cdot Y_1} = 5.4692 \cdot 10^7\ Pa$$

Maximum

$$\sigma_{t\_max\_g1} := \frac{W_{t\_max\_g1} \cdot P_1}{F_1 \cdot Y_1} = 4.7096 \cdot 10^7\ Pa$$

Gear 2

Engagement

$$\sigma_{t\_eg\_g2} := \frac{W_{t\_eg\_g2} \cdot P_2}{F_2 \cdot Y_2} = 1.0115 \cdot 10^8 \ Pa$$

Optimal

$$\sigma_{t\_opt\_g2} := \frac{W_{t\_opt\_g2} \cdot P_2}{F_2 \cdot Y_2} = 5.2206 \cdot 10^7 \ Pa$$

Maximum

$$\sigma_{t\_max\_g2} := \frac{W_{t\_max\_g2} \cdot P_2}{F_2 \cdot Y_2} = 4.4955 \cdot 10^7 \ Pa$$

Gear 3

Engagement

Low Gear

$$\sigma_{t\_eg\_g3\_low} := \frac{W_{t\_eg\_g3\_low} \cdot P_3}{F_3 \cdot Y_3} = 3.1154 \cdot 10^8 \ Pa$$

High Gear

$$\sigma_{t\_eg\_g3\_high} := \frac{W_{t\_eg\_g3\_high} \cdot P_3}{F_3 \cdot Y_3} = 8.9011 \cdot 10^7 \ Pa$$

Optimal

Low Gear

$$\sigma_{t\_opt\_g3\_low} := \frac{W_{t\_opt\_g3\_low} \cdot P_3}{F_3 \cdot Y_3} = 1.6079 \cdot 10^8 \ Pa$$

High Gear

$$\sigma_{t\_opt\_g3\_high} := \frac{W_{t\_opt\_g3\_high} \cdot P_3}{F_3 \cdot Y_3} = 4.5941 \cdot 10^7 \ Pa$$

Maximum

Low Gear

$$\sigma_{t\_max\_g3\_low} := \frac{W_{t\_max\_g3\_low} \cdot P_3}{F_3 \cdot Y_3} = 1.3846 \cdot 10^8 \ Pa$$

High Gear

$$\sigma_{t\_max\_g3\_high} := \frac{W_{t\_max\_g3\_high} \cdot P_3}{F_3 \cdot Y_3} = 3.9561 \cdot 10^7 \ Pa$$

Gear 4

Engagement

Low Gear

$$\sigma_{t\_eg\_g4\_low} := \frac{W_{t\_eg\_g4\_low} \cdot P_4}{F_4 \cdot Y_4} = 3.4181 \cdot 10^8 \ Pa$$

High Gear

$$\sigma_{t\_eg\_g4\_high} := \frac{W_{t\_eg\_g4\_high} \cdot P_4}{F_4 \cdot Y_4} = 9.7661 \cdot 10^7 \ Pa$$

Optimal

Low Gear

$$\sigma_{t\_opt\_g4\_low} := \frac{W_{t\_opt\_g4\_low} \cdot P_4}{F_4 \cdot Y_4} = 1.7642 \cdot 10^8 \ Pa$$

High Gear

$$\sigma_{t\_opt\_g4\_high} := \frac{W_{t\_opt\_g4\_high} \cdot P_4}{F_4 \cdot Y_4} = 5.0406 \cdot 10^7 \ Pa$$

Maximum

Low Gear

$$\sigma_{t\_max\_g4\_low} := \frac{W_{t\_max\_g4\_low} \cdot P_4}{F_4 \cdot Y_4} = 1.5192 \cdot 10^8 \ Pa$$

High Gear

$$\sigma_{t\_max\_g4\_high} := \frac{W_{t\_max\_g4\_high} \cdot P_4}{F_4 \cdot Y_4} = 4.3405 \cdot 10^7 \ Pa$$

The maximum stress occurs on gear four at engagement period in low gear ratio.
This maximum stress calculated was 341.81 MPa.

A.3. Shaft Analysis

Shaft 1 Analysis

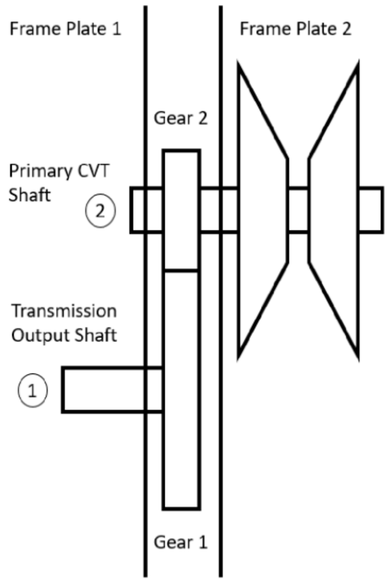
Gear reductions in the transmission to be used for torque calculations

Primary Reduction: $pr := \frac{22}{62}$

Fifth: $g_5 := \frac{25}{21}$

CVT input Reduction: $g_{input} := \frac{1}{pr \cdot g_5}$

transmission Schematic



$CVT_{ratio1} := 3.5$ $CVT_{ratio2} := 1$

CVT output Reduction

$g_{output} := \frac{1}{2}$

$g_{differential} := \frac{1}{4}$

Tire Diameter: $D_t := 20.5\ in$

Vehicle Mass: $m_{total} := 573\ lbm$

Tire Friction Coefficient: $\mu := 1.7$

Force at tire slip:

$f_{friction} := m_{total} \cdot 9.81 \frac{m}{s^2} \cdot \mu \cdot 0.8$

$f_{friction} = 3467.5943\ N$

$g_{input} = 2.3673$

Number of Pinion teeth: $Pinion := 14$

$Spur := round\left(Pinion \cdot g_{input}, 0\right)$

$Spur = 33$

Actual gear selected based on what was available is:

$Spur := 32$

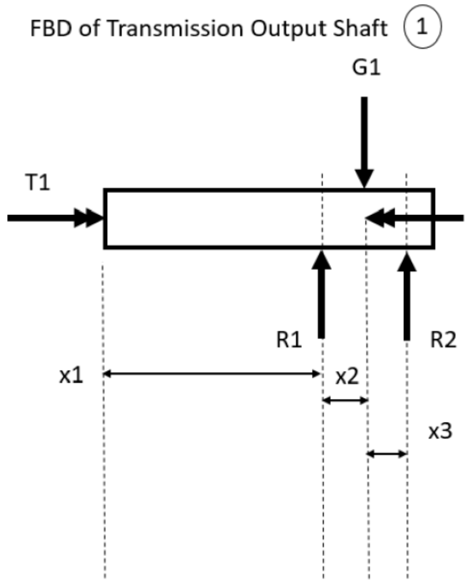
$g_{input} := \frac{Spur}{Pinion} = 2.2857$

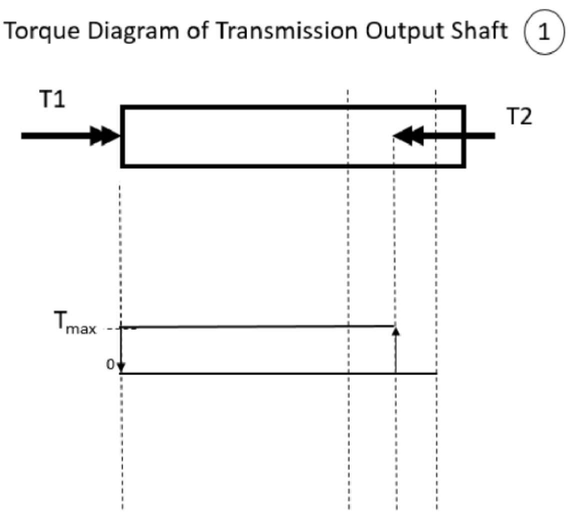
Maximum shaft torque considered based on the traction limit of the vehicle:

$$T_{max} := f_{friction} \cdot \frac{D_t}{2} \cdot g_{output} \cdot g_{differential} \cdot \frac{1}{CVT_{ratio1}} \cdot g_{input}$$

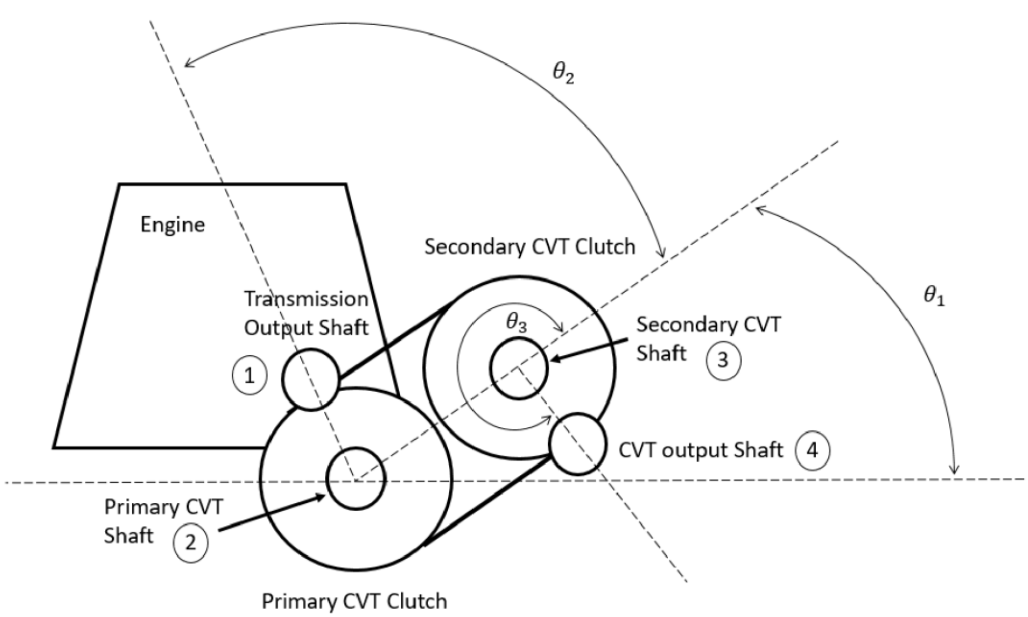
$$T_{max} = 73.697\ N\ m$$

$T1 := T_{max}$ $T2 := T_{max}$



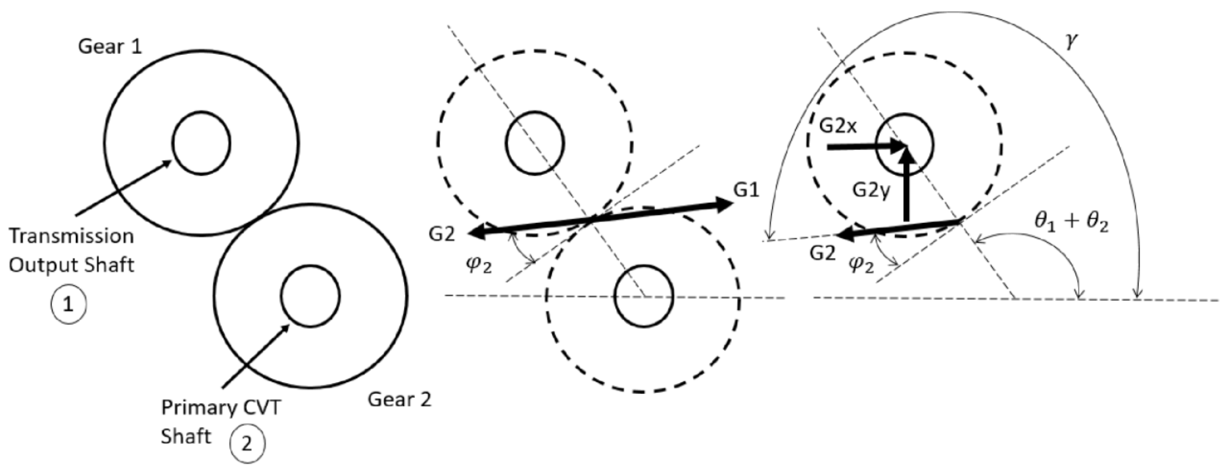


CVT Angle Schematic



4 Dec 2017 20:23:59 - 1 Trans Output Shaft Analysis.sm

Gear Force Schematic



Radial Forces Exerted on Shaft from Gear 2:

$d_p := 96\ mm$ gear diameter

$\varphi_2 := 20\ deg$ gear pressure angle

$\theta_1 := 30\ deg$

$\theta_2 := 330\ deg$

Forces in free body diagram above:

$$G_2 := 2 \cdot \frac{T_{max}}{d_p} \cdot \frac{1}{\cos(\varphi_2)}$$

$$\gamma := \theta_1 + \theta_2 + 90\ deg - \varphi_2$$

$$+\uparrow \sum F_y = 0 \qquad G_{2y} - G_2 \cdot \sin(\gamma) = 0$$

$$G_{2y} := G_2 \cdot \sin(\gamma)$$

$$G_{2y} = 1535.354\ N$$

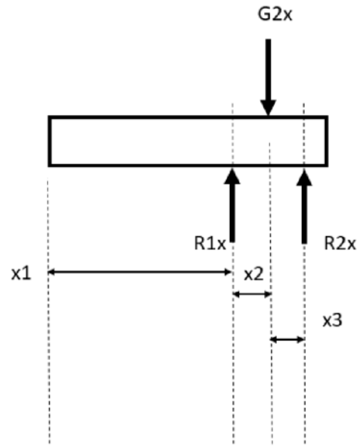
$$+\rightarrow \sum F_x = 0 \qquad -G_{2x} + G_2 \cdot \cos(\gamma) = 0$$

$$G_{2x} := G_2 \cdot \cos(\gamma)$$

$$G_{2x} = 558.8232\ N$$

Bearing Reaction Calculations

FBD of Transmission Output Shaft in X Coordinate (1)



$x_1 := 37.5\ mm$

$x_2 := 20\ mm$

$x_3 := 20\ mm$

Distance values taken as center of gears and bearings

$+\uparrow \sum F_x = 0 \quad R_{1x} + R_{2x} + G_{2x} = 0$

$R_{1x} + R_{2x} = -G_{2x}$

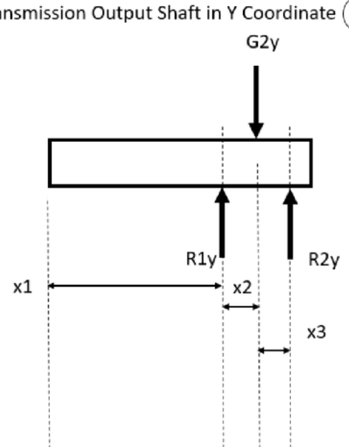
By Symmerty: $R_{1x} := R_{2x}$

$R_{2x} := \frac{-G_{2x}}{2}$

$R_{2x} = -279.4116\ N$

$R_{1x} = -279.4116\ N$

FBD of Transmission Output Shaft in Y Coordinate (1)



$+\uparrow \sum F_y = 0 \quad R_{1y} + R_{2y} + G_{2y} = 0$

$R_{1y} + R_{2y} = -G_{2y}$

By Symmerty: $R_{1y} := R_{2y}$

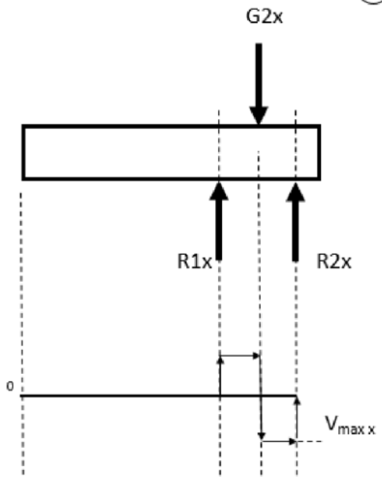
$R_{2y} := \frac{-G_{2y}}{2}$

$R_{2y} = -767.677\ N$

$R_{1y} = -767.677\ N$

Shear Diagrams

Shear Diagram of Trans Output in X Coordinate (1)

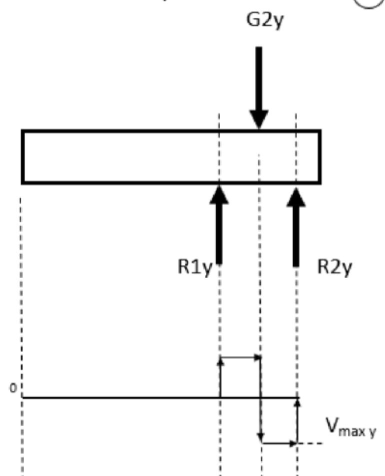


Here maximum shear occurs at V max x.

$V_{max\_x} := R_{1x}$

$V_{max\_x} = -279.4116\ N$

Shear Diagram of Trans Output in Y Coordinate (1)

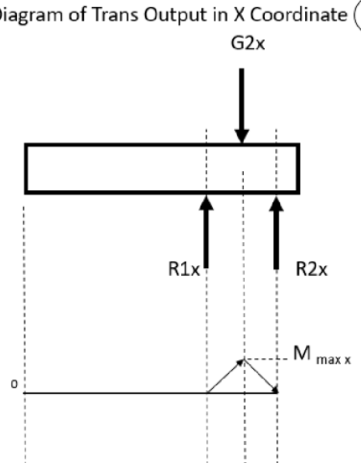


Here maximum shear occurs at V max y.

$V_{max\_y} := R_{1y}$

$V_{max\_y} = -767.677\ N$

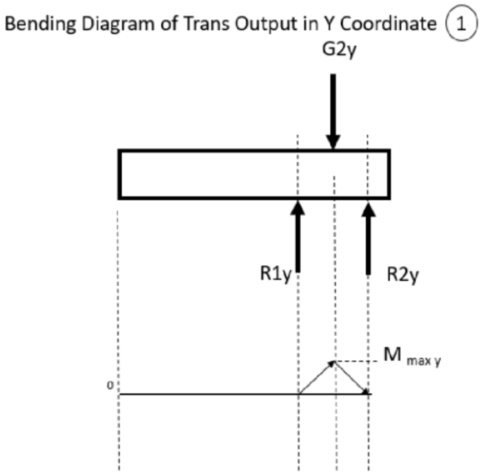
Bending Diagram of Trans Output in X Coordinate (1)



Here maximum bending occurs at M max x.

$M_{max\_x} := R_{1x} \cdot x_2$

$M_{max\_x} = -5.5882\ J$



Here maximum bending occurs at M max y.

$M_{max\_y} := R_{1y} \cdot x_2$

$M_{max\_y} = -15.3535\ J$

Shaft Material:
AISI 4340 Normalized Steel

$S_{yield} := 710\ MPa$ Tensile Strength: Yield

$E := 205\ GPa$ Youngs Modulus

$S_{ut} := 1110\ MPa$ Tensile Strength: Ultimate

Shaft minimum diameter based on DE-ASME elliptic criterion:

$$D_{min} := \left(\left(\frac{16 \cdot n_f}{\pi} \cdot \sqrt{\frac{4 \cdot \left(K_f \cdot M_a\right)^2 + \left(K_{fs} \cdot T_a\right)^2 \cdot 3}{S_e^{\,2}} + \frac{4 \cdot \left(K_f \cdot M_m\right)^2 + 3 \cdot \left(K_{fs} \cdot T_m\right)^2}{S_{yield}^{\,2}}} \right) \right)^{\frac{1}{3}}$$

$n_f := 1.5$ Safety Factor

$K_f := 1$ $K_{fs} := 1$ For shaft calculations use nominal stress concentration factors of 1. Locations of stress concentration will be considered later

$M_{min} := 0$ $M_{max} := \sqrt{M_{max\_x}^{\,2} + M_{max\_y}^{\,2}}$ Min and Max moments based on operating conditions

$M_m := \dfrac{M_{max} + M_{min}}{2}$ $M_a := \dfrac{M_{max} - M_{min}}{2}$ Mean and alternating moments

$T_{min} := 0$ Min torque based on operating condition (max calculated above)

$T_m := \dfrac{T_{max} + T_{min}}{2}$ $T_a := \dfrac{T_{max} - T_{min}}{2}$ Mean and alternating torques

$S_e := 0.504 \cdot S_{ut} \cdot 0.703 \cdot 0.879 \cdot 1 \cdot 1 \cdot 0.868 \cdot 1$ Elliptic stress with factors

$\boxed{D_{min} = 12.2114\ mm}$ Chosen Shaft Diameter = 20 mm

Bearing Selections

	Bearing 1:	Bearing 2:
Reaction Forces:	$R_1 := \sqrt{R_{1x}^2 + R_{1y}^2}$	$R_2 := \sqrt{R_{2x}^2 + R_{2y}^2}$
	$R_1 = 816.9448\ N$	$R_2 = 816.9448\ N$
Number of cycles rated for (million)	$L_{D1} := 20$	$L_{D2} := 20$
Dynamic load factor:	$C_{r1} := R_1 \cdot L_{D1}^{\frac{1}{3}}$	$C_{r2} := R_2 \cdot L_{D2}^{\frac{1}{3}}$
	$C_{r1} = 2217.5294\ N$	$C_{r2} = 2217.5294\ N$

Bearing Selected for R2: FAG QJ304-TVP Four Point Contact Bearing

$d_{inner} := 20\ mm$ Width $:= 15\ mm$

$d_{outer} := 52\ mm$ Weight $:= 0.184\ kg$

Bearing Rating:

$C_{r1.rating} := 36500\ N$ Dynamic force rating

$n_{r1.rating} := 20000\ rpm$ Maximum speed rating

Bearing Selected for R1: FAG 7008-B-TVP Angular Contact Bearing

$d_{inner} := 40\ mm$ Width $:= 15\ mm$

$d_{outer} := 68\ mm$ Weight $:= 0.17\ kg$

Bearing Rating:

$C_{r2.rating} := 26000\ N$ Dynamic force rating

$n_{r2.rating} := 8000 - 8600\ rpm$ Maximum speed rating

Therefore, bearings selected meet the mimimum design criteria based on the operating conditions they have been selected for.

Stress Concentration Factor Analysis

Consider the spline where the gear is mounted since moment and torsion are highest here

Select stress concentration factors based on conservative estimations for splines

$K_{t.spline} := 1.8$

$K_{ts.spline} := 2.5$

$q_{spline} := 0.8$

Therefore the fatigue stress factors can be calculated

$K_{f.spline} := 1 + q_{spline} \cdot \left(K_{t.spline} - 1\right) = 1.64$

$K_{fs.spline} := 1 + q_{spline} \cdot \left(K_{ts.spline} - 1\right) = 2.2$

Fatigue stress concentration factors to use in DE-ASME elliptic criterion

$K_f := K_{f.spline}$

$K_{fs} := K_{fs.spline}$

$$D := \left(\left(\frac{16 \cdot n_f}{\pi} \cdot \sqrt{\frac{4 \cdot \left(K_f \cdot M_a\right)^2 + \left(K_{fs} \cdot T_a\right)^2 \cdot 3}{S_e^{\,2}} + \frac{4 \cdot \left(K_f \cdot M_m\right)^2 + 3 \cdot \left(K_{fs} \cdot T_m\right)^2}{S_{yield}^{\,2}}} \right) \right)^{\frac{1}{3}}$$

$\boxed{D = 15.8088\ mm}$ The minimum diameter of the spline is 27 mm

Consider the step where torsion and moment are severe - the step from the bearing shoulder to the gear spline

$D_1 := 44\ mm$ Diameter of bearing shoulder

$d_1 := 28\ mm$ Average diameterof the spline

$r_1 := 1\ mm$ Fillet Radius after step

$\frac{D_1}{d_1} = 1.5714$ $\frac{r_1}{d_1} = 0.0357$ Ratios for selecting notch sensitivity

From these ratios, the appropriate notch sensitivity factor and stress concentration factors were selected from tables in MECE 360 course notes and design tables.

$q_1 := 0.8$

$K_{t.1} := 1.8$

$K_{ts.1} := K_{t.1}$

Therefore the fatigue stress factors can be calculated

$$K_{f.1} := 1 + q_1 \cdot \left(K_{t.1} - 1\right) = 1.64$$

$$K_{fs.spline} := 1 + q_{spline} \cdot \left(K_{ts.spline} - 1\right) = 2.2$$

Fatigue stress concentration factors to use in DE-ASME elliptic criterion

Finally, the minimum required diameter for this location was evaluated using the most severe stress conditions.

$$K_f := K_{f.1}$$

$$K_{fs} := K_{fs.1}$$

$$\boxed{D = 15.8088\ mm}$$

The minimum diameter in this location is 27 mm

Both major stress concentration locations that were evaluated resulted in minimum diameter requirements less than the shaft diameter. Therefore, the shaft size is sufficient to prevent system failure.

Shaft 2 Analysis

Primary Reduction

$pr := \frac{22}{62}$

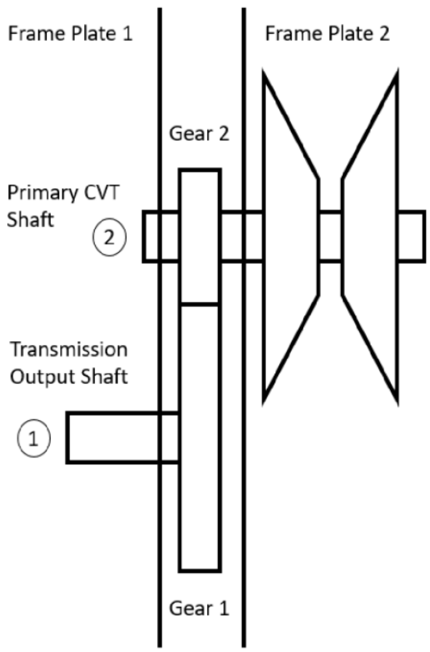
Fifth

$g_5 := \frac{25}{21}$

CVT input Reduction

$g_{input} := \frac{1}{pr \cdot g_5}$

transmission Schematic



$CVT_{ratio1} := 3.5$ $CVT_{ratio2} := 1$

CVT output Reduction

$g_{output} := \frac{1}{2}$

$g_{differential} := \frac{1}{4}$

Tire Diameter: $D_t := 20.5\ in$

Vehicle Mass: $m_{total} := 573\ lbm$

Tire Friction Coefficient: $\mu := 1.7$

Force at tire slip:

$f_{friction} := m_{total} \cdot 9.81 \frac{m}{s^2} \cdot \mu \cdot 0.8$

$f_{friction} = 3467.5943\ N$

$g_{input} = 2.3673$

Number of Pinion teeth: $Pinion := 14$

$Spur := round(Pinion \cdot g_{input}, 0)$

$Spur = 33$

Actual gear selected based on what was available is:

$Spur := 32$

$g_{input} := \frac{Spur}{Pinion} = 2.2857$

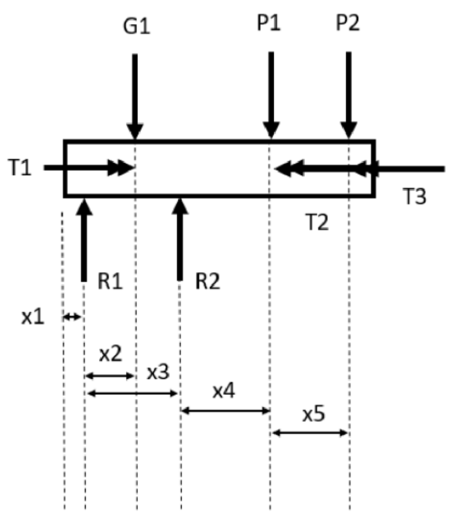
Maximum shaft Torque:

$$T_{max} := f_{friction} \cdot \frac{D_t}{2} \cdot g_{output} \cdot g_{differential} \cdot \frac{1}{CVT_{ratio1}}$$

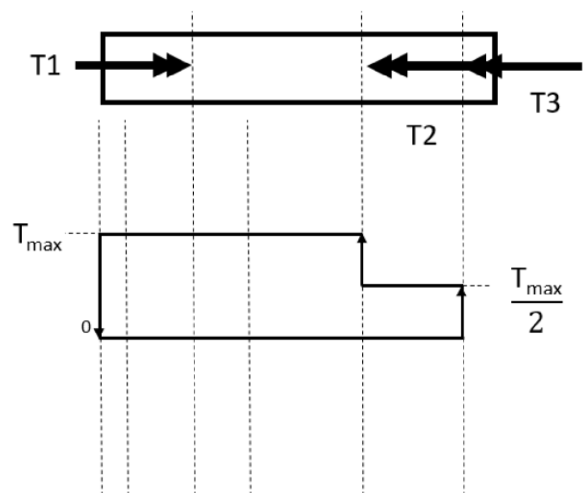
$$T_{max} = 32.2424\ J$$

$T1 := T_{max}$ $\quad T2 := \frac{T_{max}}{2}$ $\quad T3 := T2$

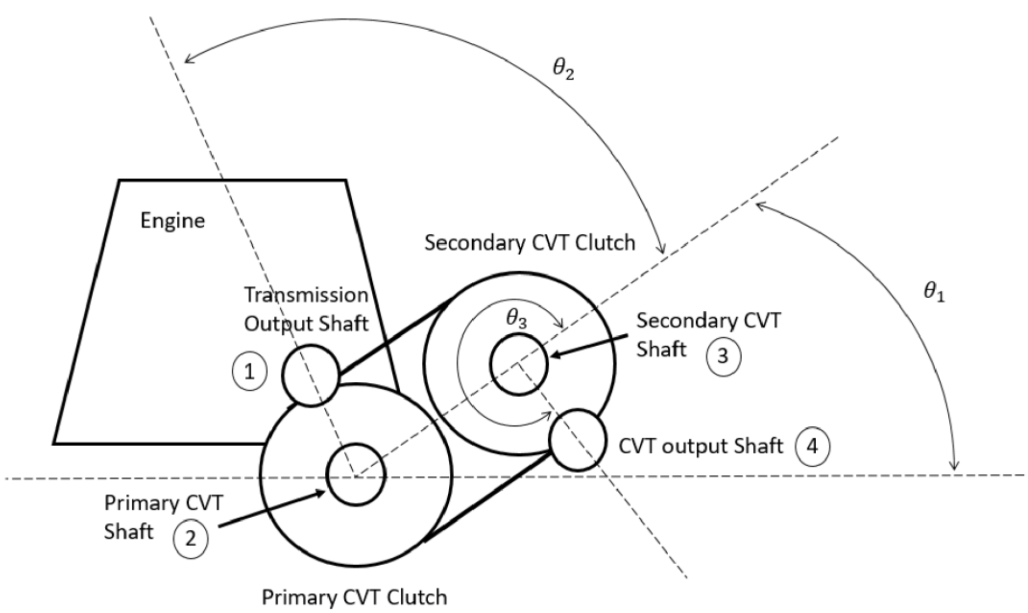
FBD of Primary CVT Shaft (2)



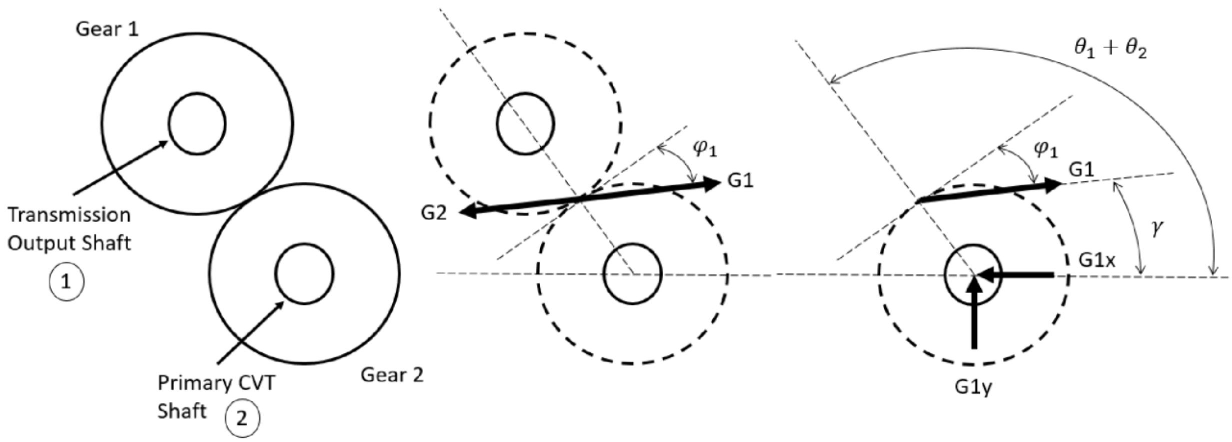
Torque Diagram of Primary CVT Shaft (2)



CVT Angle Schematic



Gear Force Schematic



Radial Forces Exerted on Shaft from Gear 2:

$d_p := 42\ mm$ gear diameter $\qquad \theta_1 := 30\ deg$

$\varphi_1 := 20\ deg$ gear pressure angle $\qquad \theta_2 := 330\ deg$

Forces in free body diagram above:

$G_1 = -G_2$ Due to equal and opposite forces in the gears, the magnitude of G_1 is equal to G_2

$$G_1 := 2 \cdot \frac{T_{max}}{d_p} \cdot \frac{1}{\cos(\varphi_1)}$$

$$\gamma := \theta_1 + \theta_2 - 90\ deg - \varphi_1$$

$$+\uparrow \sum F_y = 0 \qquad G_{1y} + G_1 \cdot \sin(\gamma) = 0$$

$$G_{1y} := G_1 \cdot \sin(\gamma)$$

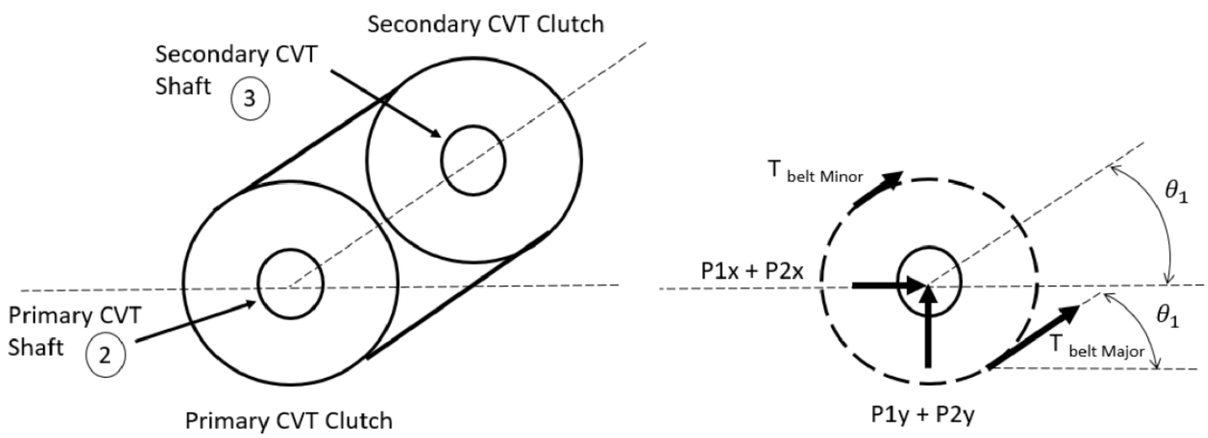
$$G_{1y} = -1535.354\ N$$

$$+\rightarrow \sum F_x = 0 \qquad -G_{1x} + G_1 \cdot \cos(\gamma) = 0$$

$$G_{1x} := G_1 \cdot \cos(\gamma)$$

$$G_{1x} = -558.8232\ N$$

CVT Belt Force Schematic



Radial Forces Exerted on Shaft from CVT Pullies:

$T_{belt\_Major} := 2.6\ kN$

$T_{belt\_Minor} := 1\ kN$

The minor and major belt forces were calculated using the belt tension analysis

By symmetry: $P_{1y} = P_{2y}$ and $P_{1x} = P_{2x}$

$+\uparrow \sum F_y = 0 \quad P_{1y} + P_{2y} + T_{belt\_Major} \cdot \sin(\theta_1) + T_{belt\_Minor} \cdot \sin(\theta_1) = 0$

$P_{1y} := (T_{belt\_Major} + T_{belt\_Minor}) \cdot \sin(\theta_1) \cdot \frac{1}{2}$

$P_{1y} = 900\ N$

$P_{2y} := P_{1y} = 900\ N$

$+\rightarrow \sum F_x = 0 \quad P_{1x} + P_{2x} + T_{belt\_Major} \cdot \cos(\theta_1) + T_{belt\_Minor} \cdot \cos(\theta_1) = 0$

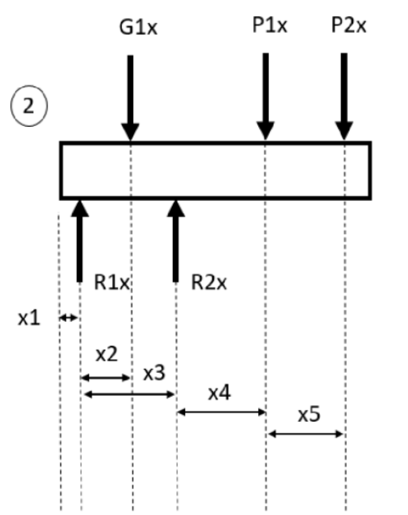
$P_{1x} := (T_{belt\_Major} + T_{belt\_Minor}) \cdot \cos(\theta_1) \cdot \frac{1}{2}$

$P_{1x} = 1558.8457\ N$

$P_{2x} := P_{1x} = 1558.8457\ N$

Bearing Reaction Calculations

FBD of Primary CVT Shaft In X Coordinate



$x_1 := 7.5\ mm$

$x_2 := 25\ mm$

$x_3 := 73\ mm$

$x_4 := 65\ mm$

$x_5 := 27\ mm$

Distance values taken as center of gears and bearings

$$+\uparrow \sum F_x = 0 \qquad R_{1x}+R_{2x}+G_{1x}+P_{1x}+P_{2x}=0$$

$$R_{1x}+R_{2x}=-\left(G_{1x}+P_{1x}+P_{2x}\right)$$

$$+\circlearrowleft \sum M_{R1x} = 0$$

$$G_{1x}\cdot x_2+R_{2x}\cdot x_3-\left(-P_{1x}\right)\cdot\left(x_3+x_4\right)-\left(-P_{2x}\right)\cdot\left(x_3+x_4+x_5\right)=0$$

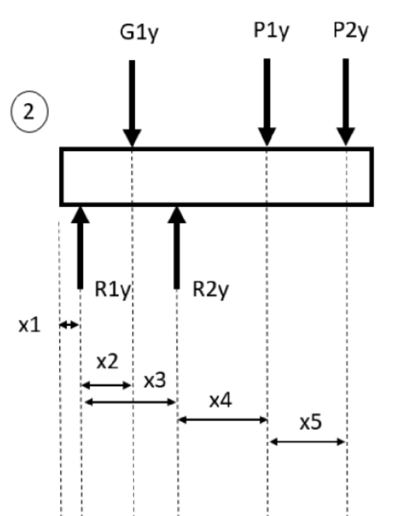
$$R_{2x}:=-\frac{\left(G_{1x}\cdot x_2+\left(-\left(-P_{1x}\right)\right)\cdot\left(x_3+x_4\right)+\left(-\left(-P_{2x}\right)\right)\cdot\left(x_3+x_4+x_5\right)\right)}{x_3}$$

$$R_{2x}=-6278.8997\ N$$

$$R_{1x}:=-\left(G_{1x}+P_{1x}+P_{2x}\right)-R_{2x}$$

$$R_{1x}=3720.0314\ N$$

FBD of Primary CVT Shaft In Y Coordinate



$$+\uparrow \sum F_y = 0 \qquad R_{1y}+R_{2y}+G_{1y}+P_{2y}=0$$

$$R_{1y}+R_{2y}=-\left(G_{1y}+P_{1y}+P_{2y}\right)$$

$$+\circlearrowleft \sum M_{R1y} = 0$$

$$-\left(-G_{1y}\right)\cdot x_2+R_{2y}\cdot x_3-\left(-P_{1y}\right)\cdot\left(x_3+x_4\right)-\left(-P_{2y}\right)\cdot\left(x_3+x_4+x_5\right)=0$$

$$R_{2y}:=-\frac{\left(\left(G_{1y}\right)\cdot x_2+\left(P_{1y}\right)\cdot\left(x_3+x_4\right)+\left(P_{2y}\right)\cdot\left(x_3+x_4+x_5\right)\right)}{x_3}$$

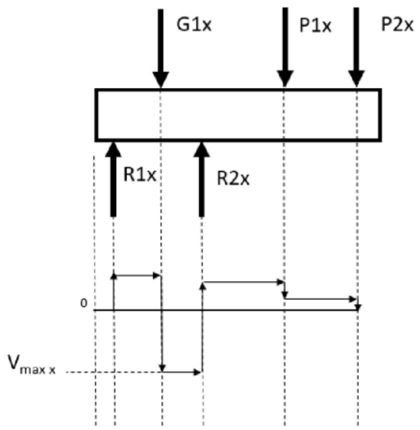
$$R_{2y}=-3209.8103\ N$$

$$R_{1y}:=-\left(G_{1y}+P_{1y}+P_{2y}\right)-R_{2y}$$

$$R_{1y}=2945.1643\ N$$

Shear Diagrams

Shear Diagram of Primary CVT Shaft in X Coordinate 2

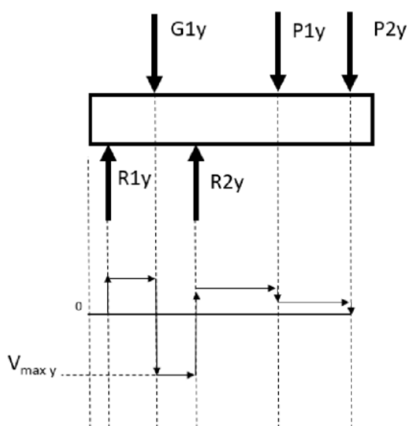


Here maximum shear occurs at V max x.

$$V_{max\_x} := \left(\left|R_{1x}\right| - \left|G_{1x}\right|\right)$$

$$V_{max\_x} = 3161.2082\ N$$

Shear Diagram of Primary CVT Shaft in Y Coordinate 2

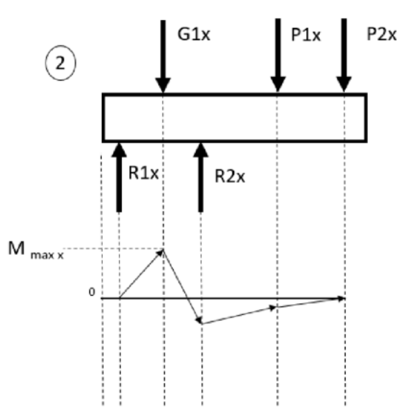


Here maximum shear occurs at V max x.

$$V_{max\_y} := \left(\left|R_{1y}\right| - \left|G_{1y}\right|\right)$$

$$V_{max\_y} = 1409.8103\ N$$

Bending Moment Diagram of Primary CVT Shaft in X Coordinate



Here maximum bending occurs at M max x.

$$M_{max\_x} := R_{1x} \cdot x_2$$

$$M_{max\_x} = 93.0008\ J$$

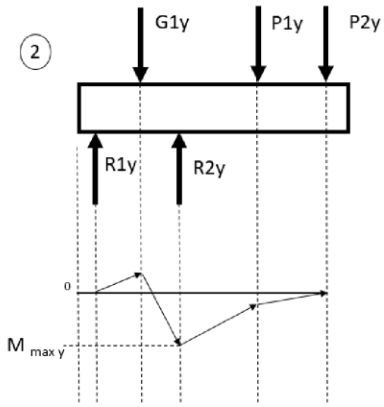
$$R_{1x} \cdot x_2 = 93.0008\ J$$

$$\left(-\left(G_{1x} - R_{1x}\right)\right) \cdot \left(x_3 - x_2\right) = 205.385\ J$$

$$\left(R_{2x} + R_{1x} - G_{1x}\right) \cdot x_4 = -130.0029\ J$$

$$\left(P_{1x} + R_{2x} + R_{1x} - G_{1x}\right) \cdot x_5 = -11.9124\ J$$

Bending Moment Diagram of Primary CVT Shaft in Y Coordinate



Here maximum bending occurs at M max y.

$$M_{max\_y} := \left(R_{1y} \cdot x_2 + \left(G_{1y} + R_{1y}\right) \cdot \left(x_3 - x_2\right)\right)$$

$$M_{max\_y} = 141.3\ J$$

$$R_{1y} \cdot x_2 = 73.6291\ J$$

$$\left(G_{1y} + R_{1y}\right) \cdot \left(x_3 - x_2\right) = 67.6709\ J$$

$$\left(R_{2y} + R_{1y} + G_{1y}\right) \cdot x_4 = -117\ J$$

$$\left(P_{1y} + \left(R_{2y} + R_{1y} + G_{1y}\right)\right) \cdot x_5 = -24.3\ J$$

Shaft Material:
AISI 4340 Normalized Steel

$S_{yield} := 710\ MPa$	Tensile Strength: Yield
$E := 205\ GPa$	Youngs Modulus
$S_{ut} := 1110\ MPa$	Tensile Strength: Ultimate

Shaft minimum diameter based on DE-ASME elliptic criterion:

$$D_{min} := \left(\left(\frac{16 \cdot n_f}{\pi} \cdot \sqrt{\frac{4 \cdot \left(K_f \cdot M_a\right)^2 + \left(K_{fs} \cdot T_a\right)^2 \cdot 3}{S_e^{\,2}} + \frac{4 \cdot \left(K_f \cdot M_m\right)^2 + 3 \cdot \left(K_{fs} \cdot T_m\right)^2}{S_{yield}^{\,2}}}\right)^{\frac{1}{3}}\right)$$

$n_f := 1.5$ — Safety Factor

$K_f := 1$ $\quad K_{fs} := 1$ — For shaft calculations use nominal stress concentration factors of 1. Locations of stress concentration will be considered later.

$M_{min} := 0$ $\quad M_{max} := \sqrt{M_{max\_x}^{\,2} + M_{max\_y}^{\,2}}$ — Min and Max moments based on operating conditions

$M_m := \frac{M_{max} + M_{min}}{2}$ $\quad M_a := \frac{M_{max} - M_{min}}{2}$ — Mean and alternating moments

$T_{min} := 0$ — Min torque based on operating condition (max calculated above)

$T_m := \frac{T_{max} + T_{min}}{2}$ $\quad T_a := \frac{T_{max} - T_{min}}{2}$ — Mean and alternating torques

$$S_e := 0.504 \cdot S_{ut} \cdot 0.703 \cdot 0.879 \cdot 1 \cdot 1 \cdot 0.868 \cdot 1$$

$\boxed{D_{min} = 16.7966\ mm}$ $\boxed{\text{Chosen Shaft Diameter = 20 mm}}$

Bearing Selections

	Bearing 1:		Bearing 2:
Reaction Forces:	$R_1 := \sqrt{R_{1x}^2 + R_{1y}^2}$		$R_2 := \sqrt{R_{2x}^2 + R_{2y}^2}$
	$R_1 = 4744.7472\ N$		$R_2 = 7051.7702\ N$
Number of cycles rated for (million)	$L_{D1} := 20$	Number of cycles rated for (million)	$L_{D2} := 20$
Dynamic load factor:	$C_{r1} := R_1 \cdot L_{D1}^{\frac{1}{3}}$		$C_{r2} := R_2 \cdot L_{D2}^{\frac{1}{3}}$
	$C_{r1} = 12879.2255\ N$		$C_{r2} = 19141.4493\ N$

Bearing 1 Selected: FAG QJ304-MPA Four Point Contact Bearing

$d_{1.inner} := 20\ mm$ $\quad Width_1 := 15\ mm$

$d_{1.outer} := 52\ mm$ $\quad Weight_1 := 0.184\ kg$

Bearing Rating:

$C_{r1.rating} := 30000\ N$ Dynamic force rating

$n_{r1.rating} := 28000\ rpm$ Maximum speed rating

Bearing 2 Selected: FAG QJ206-MPA Four Point Contact Bearing

$d_{2.inner} := 30\ mm$ $\quad Width_2 := 16\ mm$

$d_{2.outer} := 62\ mm$ $\quad Weight_2 := 0.254\ kg$

Bearing Rating:

$C_{r2.rating} := 36500\ N$ Dynamic force rating

$n_{r2.rating} := 20000\ rpm$ Maximum speed rating

Stress Concentration Factor Analysis

Consider the keyway for the primary clutch

Select stress concentration factors for the keys from MEC E 360 shaft tables

$K_{t.key} := 2.2$

$K_{ts.key} := 3.0$

$q_{key} := 0.8$

Therefore the fatigue stress factors can be calculated

$K_{f.key} := 1 + q_{key} \cdot \left(K_{t.key} - 1\right) = 1.96$

$K_{fs.key} := 1 + q_{key} \cdot \left(K_{ts.key} - 1\right) = 2.6$

Fatigue stress concentration factors to use in DE-ASME elliptic criterion

Finally, the minimum required diameter for this location was evaluated using the most severe stress conditions.

$K_f := K_{f.key}$

$K_{fs} := K_{fs.key}$

$\boxed{D_{min} = 21.0903\ mm}$ Diameter in this region is 28 mm

Consider the step where torsion and moment are severe - the step from the machined gear to the bearing support diameter

$D_1 := 42\ mm$ Pitch diameter of machined gear on shaft

$d_1 := 30\ mm$ Shaft diameter of bearing support

$r_1 := 1\ mm$ Fillet Radius after step

$\frac{D_1}{d_1} = 1.4$ $\frac{r_1}{d_1} = 0.0333$ Ratios for selecting notch sensitivity

From these ratios, the appropriate notch sensitivity factor and stress concentration factors were selected from tables in MECE 360 course notes and design tables.

$q_1 := 0.92$

$K_{t.1} := 2.0$

$K_{ts.1} := K_{t.1}$

Therefore the fatigue stress factors can be calculated

$K_{f.1} := 1 + q_1 \cdot \left(K_{t.1} - 1\right) = 1.92$

$K_{fs.1} := 1 + q_1 \cdot \left(K_{ts.1} - 1\right) = 1.92$

Fatigue stress concentration factors to use in DE-ASME elliptic criterion

Finally, the minimum required diameter for this location was evaluated using the most severe stress conditions.

$K_f := K_{f.1}$

$K_{fs} := K_{fs.1}$

$\boxed{D_{min} = 20.8764\ mm}$ Diameter at this location is 25.4 mm

Both major stress concentration locations that were evaluated resulted in minimum diameter requirements less than the shaft diameter. Therefore, the shaft size is sufficient to prevent system failure.

Note that the smallest diameter for this shaft occurs for one of the support bearings. This area was not evaluated for stress concentration since there is no torque applied and the overall stress is relatively low

Shaft 3 Analysis

Primary Reduction

$pr := \frac{22}{62}$

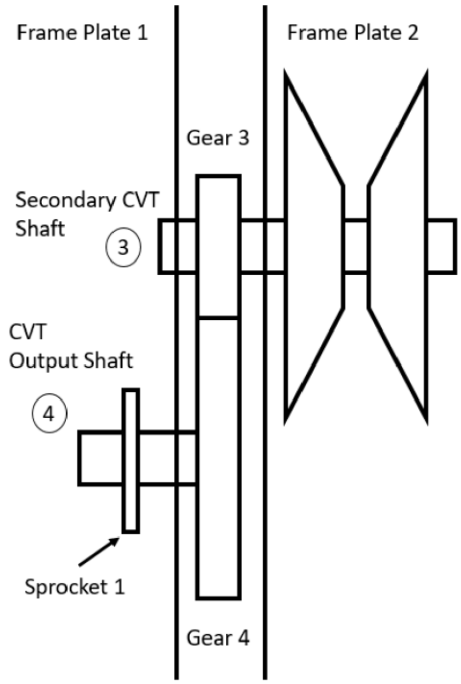
Fifth

$g_5 := \frac{25}{21}$

CVT input Reduction

$g_{input} := \frac{1}{pr \cdot g_5}$

transmission Schematic



$CVT_{ratio1} := 3.5$ $\quad$ $CVT_{ratio2} := 1$

CVT output Reduction

$g_{output} := \frac{1}{2}$

$g_{differential} := \frac{1}{4}$

Tire Diameter: $D_t := 20.5\ in$

Vehicle Mass: $m_{total} := 573\ lbm$

Tire Friction Coefficient: $\mu := 1.7$

Force at tire slip:

$f_{friction} := m_{total} \cdot 9.81 \frac{m}{s^2} \cdot \mu \cdot 0.8$

$f_{friction} = 3467.5943\ N$

$g_{output} = 0.5$

Number of Pinion teeth: $Pinion := 16$

$Spur := round\left(Pinion \cdot \frac{1}{g_{output}}, 0\right)$

$Spur = 32$

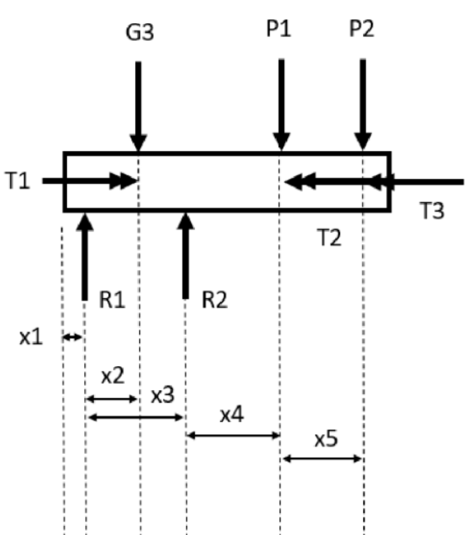
Maximum shaft torque considered based on the traction limit of the vehicle:

$$T_{max} := f_{friction} \cdot \frac{D_t}{2} \cdot g_{output} \cdot g_{differential}$$

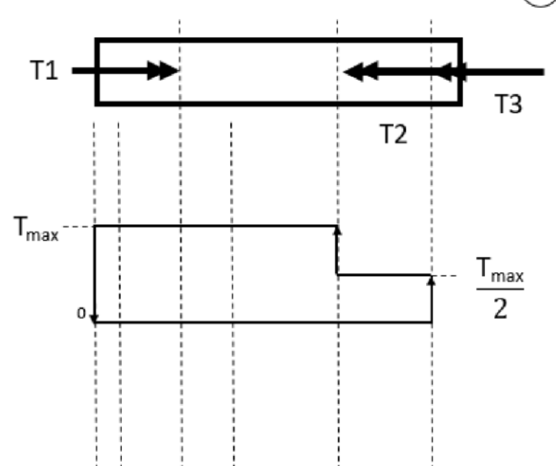
$$T_{max} = 112.8485\ N\ m$$

FBD of Secondary CVT Shaft 3

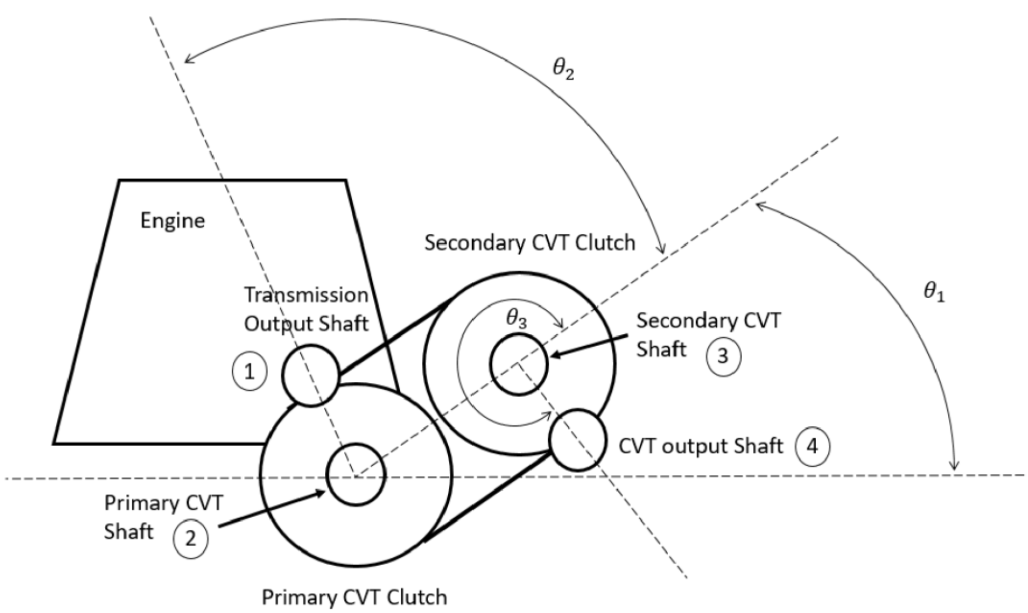
$T1 := T_{max}$ $\quad T2 := \frac{T_{max}}{2}$ $\quad T3 := T2$



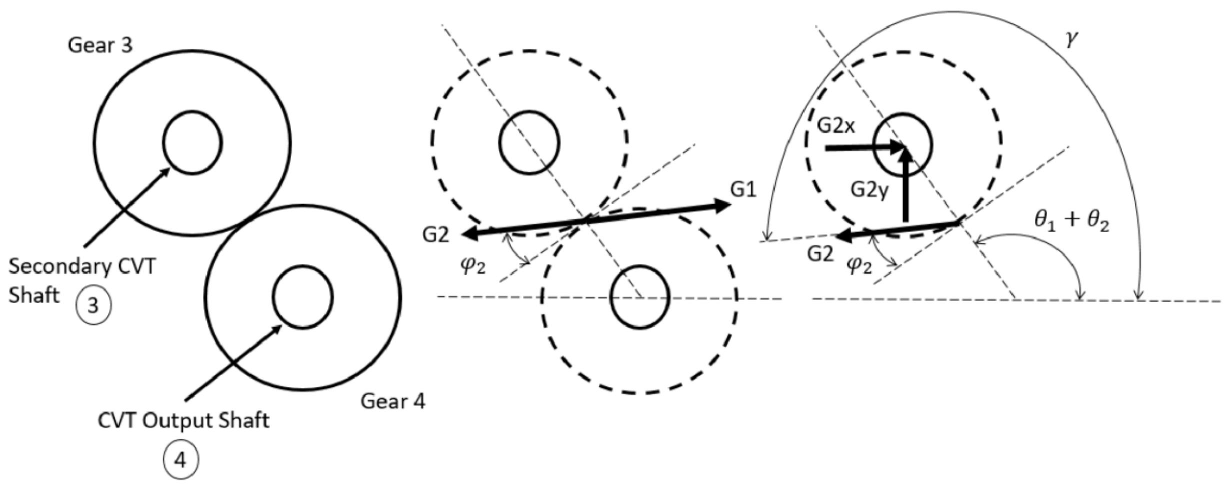
Torque Diagram of Secondary CVT Shaft 3



CVT Angle Schematic



Gear Force Schematic



Radial Forces Exerted on Shaft from Gear 1:

$d_p := 48\ mm$ gear diameter

$\varphi_2 := 20\ deg$ gear pressure angle

$\theta_1 := 30\ deg$ $\theta_3 := 150\ deg$

$\theta_2 := 330\ deg$

Forces in free body diagram above:

$G_3 \equiv G_2$

$G_2 := 2 \cdot \frac{T_{max}}{d_p} \cdot \frac{1}{\cos\left(\varphi_2\right)}$

$\gamma := \theta_1 + \theta_2 + 90\ deg - \varphi_2$

$+\uparrow \sum F_y = 0 \quad G_{1y} + G_1 \cdot \sin\left(\gamma\right) \equiv 0$

$G_{2y} := G_2 \cdot \sin\left(\gamma\right)$

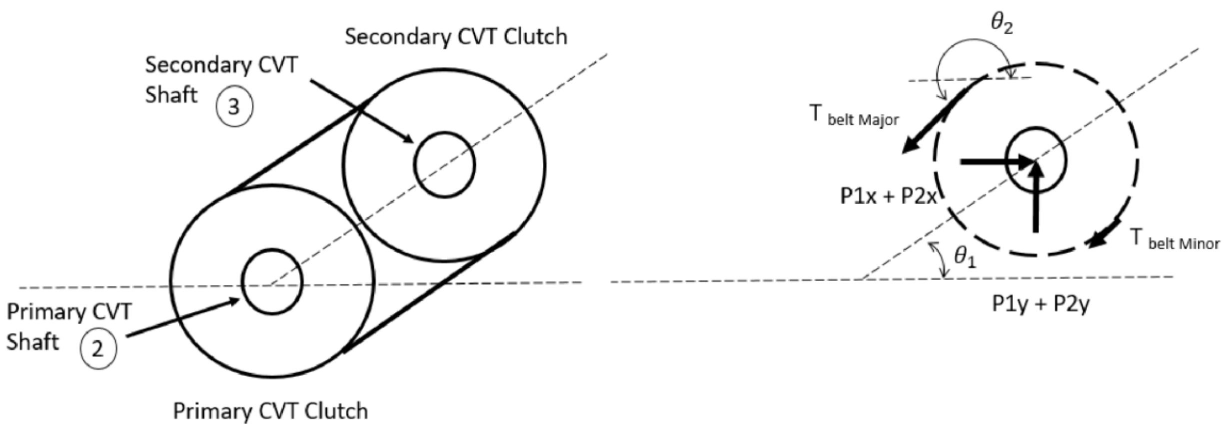
$G_{2y} = 4702.0217\ N$

$+\rightarrow \sum F_x = 0 \quad G_{1x} + G_1 \cdot \cos\left(\gamma\right) \equiv 0$

$G_{2x} := G_2 \cdot \cos\left(\gamma\right)$

$G_{2x} = 1711.3959\ N$

CVT Belt Force Schematic



Radial Forces Exerted on Shaft from CVT Pullies:

$T_{belt\_Major} := 2.6\ kN$

$T_{belt\_Minor} := 1\ kN$

The minor and major belt forces were calculated using the belt tension analysis

By symmetry: $P_{1y} = P_{2y}$ and $P_{1x} = P_{2x}$

$$+\uparrow \sum F_y = 0 \qquad P_{1y} + P_{2y} + T_{belt\_Major} \cdot \sin\left(\theta_1 + 180\right) + T_{belt\_Minor} \cdot \sin\left(\theta_1 + 180\right) = 0$$

$$P_{1y} := \left(T_{belt\_Major} + T_{belt\_Minor}\right) \cdot \sin\left(\theta_1 + 180\ deg\right) \cdot \frac{1}{2}$$

$$P_{1y} = -900\ N$$

$$P_{2y} := P_{1y} = -900\ N$$

$$+\rightarrow \sum F_x = 0 \qquad P_{1x} + P_{2x} + T_{belt\_Major} \cdot \cos\left(\theta_1 + 180\ deg\right) + T_{belt\_Minor} \cdot \cos\left(\theta_1 + 180\ deg\right) = 0$$

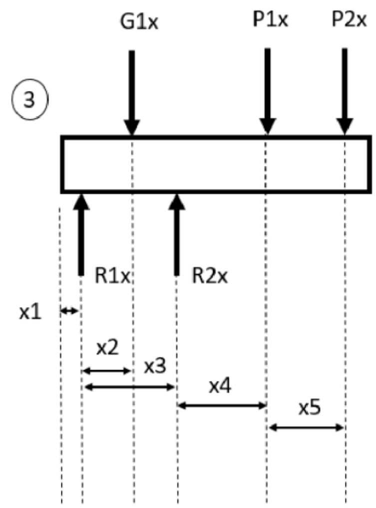
$$P_{1x} := \left(T_{belt\_Major} + T_{belt\_Minor}\right) \cdot \cos\left(\theta_1 + 180\ deg\right) \cdot \frac{1}{2}$$

$$P_{1x} = -1558.8457\ N$$

$$P_{2x} := P_{1x} = -1558.8457\ N$$

Bearing Reaction Calculations

FBD of Secondary CVT Shaft In X Coordinate



$x_1 := 7.5\ mm \quad x_4 := 80\ mm$

$x_2 := 20\ mm \quad x_5 := 27\ mm$

$x_3 := 40.5\ mm$

$$+\uparrow \sum F_x = 0 \qquad R_{1x} + R_{2x} + G_{2x} + P_{1x} + P_{2x} = 0$$

$$R_{1x} + R_{2x} = -\left(G_{2x} + P_{1x} + P_{2x}\right)$$

$$+\circlearrowleft \sum M_{R1x} = 0$$

$$G_{2x} \cdot x_2 + R_{2x} \cdot x_3 + P_{1x} \cdot \left(x_3 + x_4\right) + P_{2x} \cdot \left(x_3 + x_4 + x_5\right) = 0$$

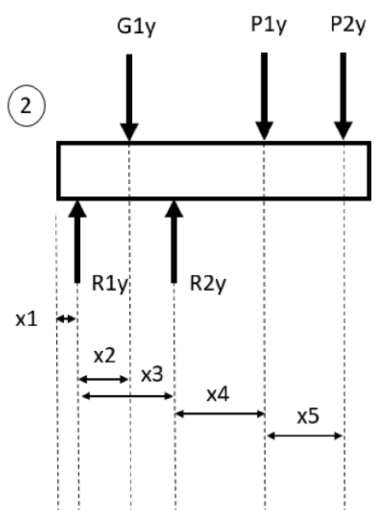
$$R_{2x} := \frac{-\left(G_{2x} \cdot x_2 + P_{1x} \cdot \left(x_3 + x_4\right) + P_{2x} \cdot \left(x_3 + x_4 + x_5\right)\right)}{x_3}$$

$$R_{2x} = 9470.191\ N$$

$$R_{1x} := -\left(G_{2x} + P_{1x} + P_{2x}\right) - R_{2x}$$

$$R_{1x} = -8063.8955\ N$$

FBD of Primary CVT Shaft In Y Coordinate



$$+\uparrow \sum F_y = 0 \qquad R_{1y} + R_{2y} + G_{2y} + P_{1y} + P_{2y} = 0$$

$$R_{1y} + R_{2y} = -\left(G_{2y} + P_{1y} + P_{2y}\right)$$

$$+\circlearrowleft \sum M_{R1y} = 0$$

$$G_{2y} \cdot x_2 + R_{2y} \cdot x_3 + P_{1y} \cdot \left(x_3 + x_4\right) + P_{2y} \cdot \left(x_3 + x_4 + x_5\right) = 0$$

$$R_{2y} := \frac{-\left(G_{2y} \cdot x_2 + P_{1y} \cdot \left(x_3 + x_4\right) + P_{2y} \cdot \left(x_3 + x_4 + x_5\right)\right)}{x_3}$$

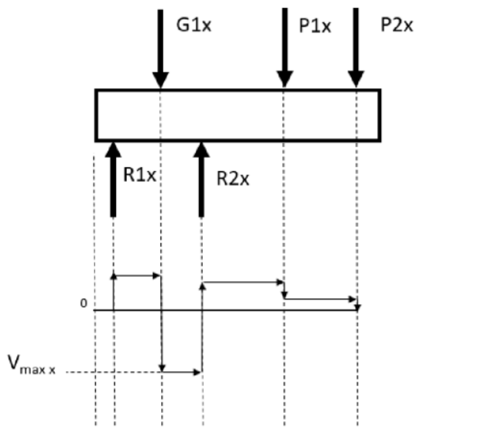
$$R_{2y} = 3633.5695\ N$$

$$R_{1y} := -\left(G_{2y} + P_{1y} + P_{2y}\right) - R_{2y}$$

$$R_{1y} = -6535.5912\ N$$

Shear Diagrams

Shear Diagram of Primary CVT Shaft in X Coordinate (2)

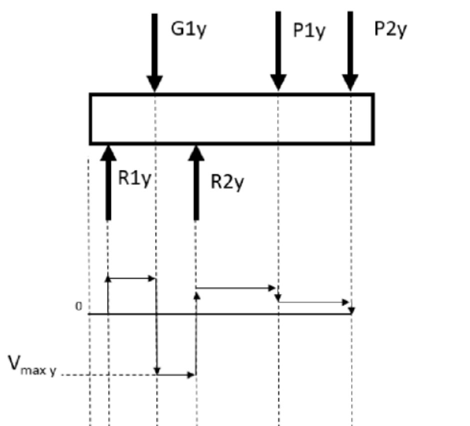


Here maximum shear occurs at V max x.

$V_{max\_x} := \left(\left|R_{1x}\right| - \left|G_{2x}\right|\right)$

$V_{max\_x} = 6352.4996\,N$

Shear Diagram of Primary CVT Shaft in Y Coordinate (2)

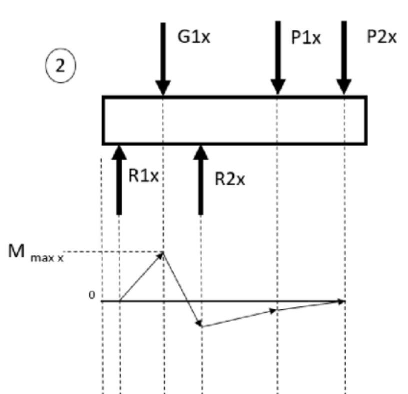


Here maximum shear occurs at V max x.

$V_{max\_y} := \left(\left|R_{1y}\right| - \left|G_{2y}\right|\right)$

$V_{max\_y} = 1833.5695\,N$

Bending Moment Diagram of Primary CVT Shaft in X Coordinate



Here maximum bending occurs at M max x.

$M_{max\_x} := R_{1x} \cdot x_2$

$M_{max\_x} = -161.2779\,J$

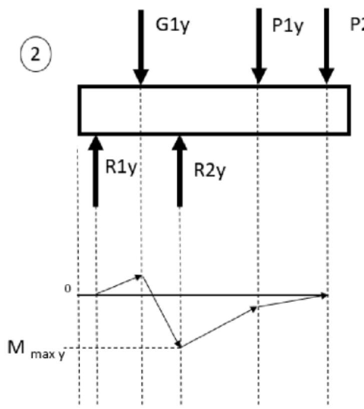
$R_{1x} \cdot x_2 = -161.2779\,J$

$\left(-\left(G_{2x} - R_{1x}\right)\right) \cdot \left(x_3 - x_2\right) = -200.3935\,J$

$\left(R_{2x} + R_{1x} - G_{2x}\right) \cdot x_4 = -24.408\,J$

$$\left(P_{1x}+R_{2x}+R_{1x}-G_{2x}\right)\cdot x_5=-50.3265\ J$$

Bending Moment Diagram of Primary CVT Shaft in Y Coordinate



Here maximum bending occurs at M max y.

$$M_{max\_y}:=\left(R_{1y}\cdot x_2+\left(G_{2y}+R_{1y}\right)\cdot\left(x_3-x_2\right)\right)$$

$$M_{max\_y}=-168.3\ J$$

$$R_{1y}\cdot x_2=-130.7118\ J$$

$$\left(G_{2y}+R_{1y}\right)\cdot\left(x_3-x_2\right)=-37.5882\ J$$

$$\left(R_{2y}+R_{1y}+G_{2y}\right)\cdot x_4=144\ J$$

$$\left(P_{1y}+\left(R_{2y}+R_{1y}+G_{2y}\right)\right)\cdot x_5=24.3\ J$$

Shaft Material:
AISI 4340 Steel

$S_{yield}:=710\ MPa$ Tensile Strength: Yield

$E:=205\ GPa$ Youngs Modulus

$S_{ut}:=1110\ MPa$ Tensile Strength: Ultimate

Shaft minimum diameter based on DE-ASME elliptic criterion:

$$D_{min}:=\left(\left(\frac{16\cdot n_f}{\pi}\cdot\sqrt{\frac{4\cdot\left(K_f\cdot M_a\right)^2+\left(K_{fs}\cdot T_a\right)^2\cdot 3}{S_e^{\ 2}}+\frac{4\cdot\left(K_f\cdot M_m\right)^2+3\cdot\left(K_{fs}\cdot T_m\right)^2}{S_{yield}^{\ 2}}}\right)^{\frac{1}{3}}\right)$$

$n_f:=1.5$ Safety Factor

$K_f:=1$ $K_{fs}:=1$

For shaft calculations use nominal stress concentration factors of 1. Locations of stress concentration will be considered later

$M_{min}:=0$ $M_{max}:=\sqrt{M_{max\_x}^{\ 2}+M_{max\_y}^{\ 2}}$ Min and Max moments based on operating conditions

$M_m:=\dfrac{M_{max}+M_{min}}{2}$ $M_a:=\dfrac{M_{max}-M_{min}}{2}$ Mean and alternating moments

$T_{min}:=0$ Min torque based on operating condition (max calculated above)

$T_m:=\dfrac{T_{max}+T_{min}}{2}$ $T_a:=\dfrac{T_{max}-T_{min}}{2}$ Mean and alternating torques

$S_e := 0.504 \cdot S_{ut} \cdot 0.703 \cdot 0.879 \cdot 1 \cdot 1 \cdot 0.868 \cdot 1$

$\boxed{D_{min} = 19.1166\ mm}$ Chosen Shaft Diameter = 25.4 mm

Bearing Selections

	Bearing 1:	Bearing 2:
Reaction Forces:	$R_1 := \sqrt{R_{1x}^2 + R_{1y}^2}$	$R_2 := \sqrt{R_{2x}^2 + R_{2y}^2}$
	$R_1 = 10379.8056\ N$	$R_2 = 10143.3399\ N$
Number of cycles rated for (million)	$L_{D1} := 20$	$L_{D2} := 20$
Dynamic load factor:	$C_{r1} := R_1 \cdot L_{D1}^{\frac{1}{3}}$	$C_{r2} := R_2 \cdot L_{D2}^{\frac{1}{3}}$
	$C_{r1} = 28175.1271\ N$	$C_{r2} = 27533.2606\ N$

Bearing 1 Selected: FAG QJ304-MPA Four Point Contact Bearing

$d_{1.inner} := 20\ mm$ Width $:= 15\ mm$

$d_{1.outer} := 52\ mm$ Weight $:= 0.184\ kg$

Bearing Rating:

$C_{r1.rating} := 30000\ N$ Dynamic force rating

$n_{r1.rating} := 28000\ rpm$ Maximum speed rating

Bearing 2 Selected: FAG QJ206-MPA Four Point Contact Bearing

$d_{2.inner} := 30\ mm$ $Width_2 := 16\ mm$

$d_{2.outer} := 62\ mm$ $Weight_2 := 0.254\ kg$

Bearing Rating:

$C_{r2.rating} := 36500\ N$ Dynamic force rating

$n_{r2.rating} := 20000\ rpm$ Maximum speed rating

Stress Concentration Factor Analysis

Consider the keyway for the primary clutch

Select stress concentration factors for the keys from MEC E 360 shaft tables

$K_{t.key} := 2.2$

$K_{ts.key} := 3.0$

$q_{key} := 0.8$

Therefore the fatigue stress factors can be calculated as follows:

$K_{f.key} := 1 + q_{key} \cdot \left(K_{t.key} - 1\right) = 1.96$

$K_{fs.key} := 1 + q_{key} \cdot \left(K_{ts.key} - 1\right) = 2.6$

Fatigue stress concentration factors to use in DE-ASME elliptic criterion

$K_f := K_{f.key}$

$K_{fs} := K_{fs.key}$

$\boxed{D_{min} = 24.3566\ mm}$ Diameter in this region is 25.4 mm

Consider the step where torsion and moment are severe - the step from the machined gear to the bearing support diameter

$D_1 := 48\ mm$ Pitch diameter of machined gear on shaft

$d_1 := 30\ mm$ Shaft diameter of bearing support

$r_1 := 1\ mm$ Fillet Radius after step

$\frac{D_1}{d_1} = 1.6$ $\frac{r_1}{d_1} = 0.0333$

From these ratios, the appropriate notch sensitivity factor and stress concentration factors were selected from tables in MECE 360 course notes and design tables.

$q_1 := 0.92$

$K_{t.1} := 2.0$

$K_{ts.1} := K_{t.1}$

Therefore the fatigue stress factors can be calculated

$K_{f.1} := 1 + q_1 \cdot \left(K_{t.1} - 1\right) = 1.92$

$K_{fs.1} := 1 + q_1 \cdot \left(K_{ts.1} - 1\right) = 1.92$

Finally, the minimum required diameter for this location was evaluated using the most severe stress conditions.

$K_f := K_{f.1}$

$K_{fs} := K_{fs.1}$

$\boxed{D_{min} = 23.7599\ mm}$ Diameter at this location is 30 mm

Both major stress concentration locations that were evaluated resulted in minimum diameter requirements less than the shaft diameter. Therefore, the shaft size is sufficient to prevent system failure.

Note that the smallest diameter for this shaft occurs for one of the support bearings. This area was not evaluated for stress concentration since there is no torque applied and the overall stress is relatively low

Shaft 4 Analysis

Primary Reduction

$pr := \frac{22}{62}$

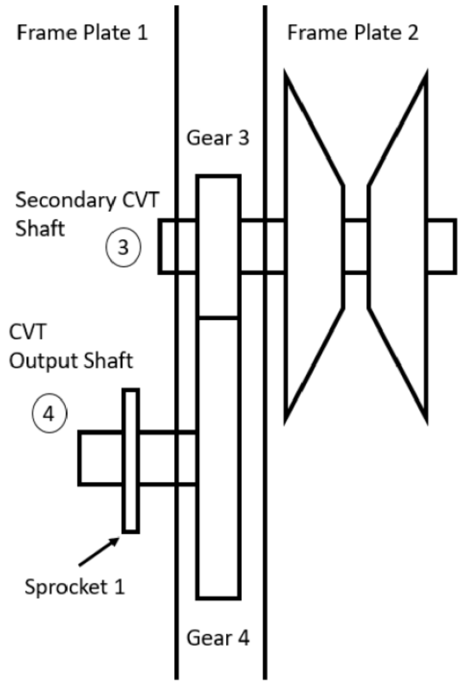
Fifth

$g_5 := \frac{25}{21}$

CVT input Reduction

$g_{input} := \frac{1}{pr \cdot g_5}$

transmission Schematic



$CVT_{ratio1} := 3.5$ $CVT_{ratio2} := 1$

CVT output Reduction

$g_{output} := \frac{1}{2}$

$g_{differential} := \frac{1}{4}$

Tire Diameter: $D_t := 20.5\ in$

Vehicle Mass: $m_{total} := 573\ lbm$

Tire Friction Coefficient: $\mu := 1.7$

Force at tire slip:

$f_{friction} := m_{total} \cdot 9.81 \frac{m}{s^2} \cdot \mu \cdot 0.8$

$f_{friction} = 3467.5943\ N$

$g_{output} = 0.5$

Number of Pinion teeth: $Pinion := 16$

$Spur := round\left(Pinion \cdot \frac{1}{g_{output}}, 0\right)$

$Spur = 32$

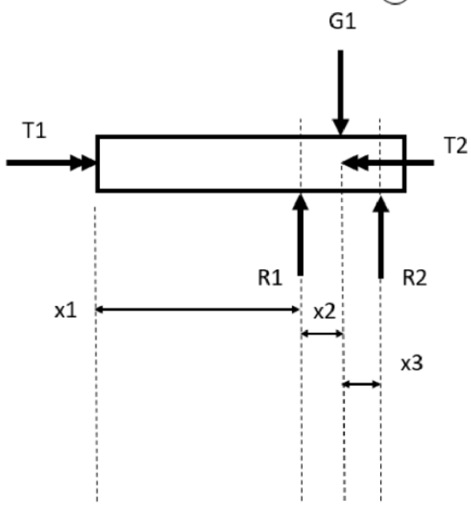
Maximum shaft torque considered based on the traction limit of the vehicle:

$$T_{max} := f_{friction} \cdot \frac{D_t}{2} \cdot g_{differential}$$

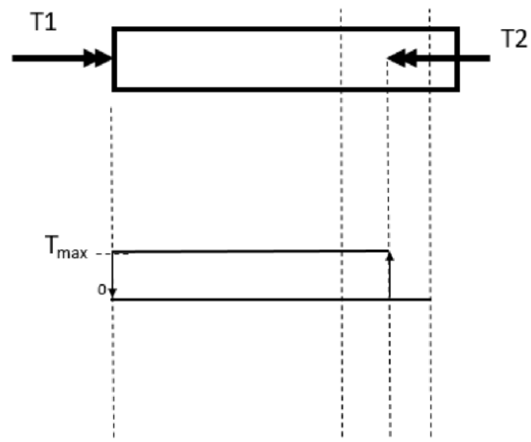
$$T_{max} = 225.697\ J$$

$T1 := T_{max}$ $T2 := \frac{T_{max}}{2}$ $T3 := T2$

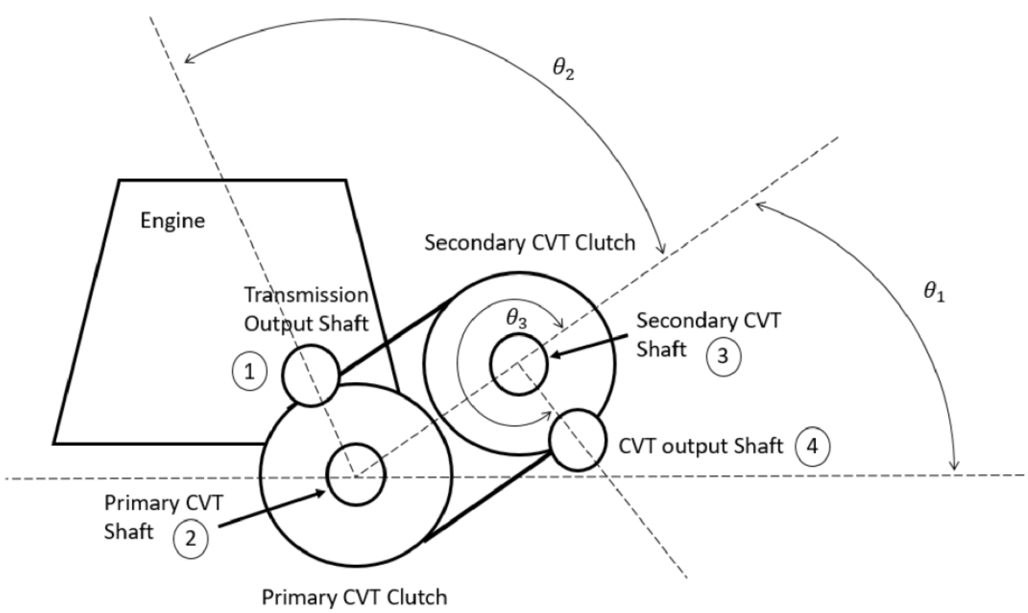
FBD of CVT Output Shaft 4



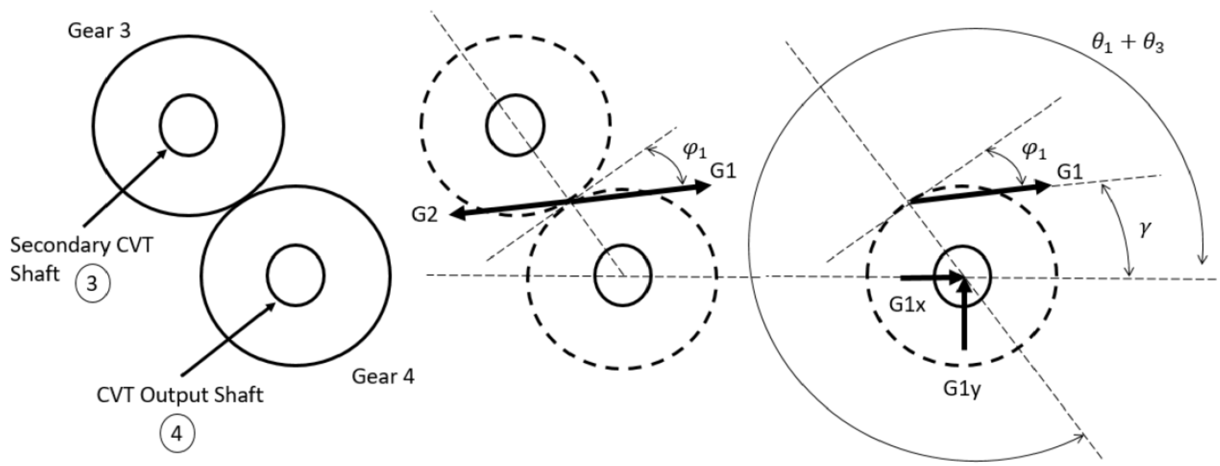
Torque Diagram of CVT Output Shaft 4



CVT Angle Schematic



Gear Force Schematic



Radial Forces Exerted on Shaft from Gear 1:

$d_p := 42\ mm$ gear diameter

$\varphi_1 := 20\ deg$ gear pressure angle

$\theta_1 := 30\ deg$ $\theta_3 := 150\ deg$

$\theta_2 := 330\ deg$

Forces in free body diagram above:

$G_4 = G_1$

$$G_1 := 2 \cdot \frac{T_{max}}{d_p} \cdot \frac{1}{\cos(\varphi_1)}$$

$\gamma := \theta_1 + \theta_3 + 90\ deg - \varphi_1$

$+\uparrow \sum F_y = 0$ $G_{1y} + G_1 \cdot \sin(\gamma) = 0$

$G_{1y} := G_1 \cdot \sin(\gamma)$

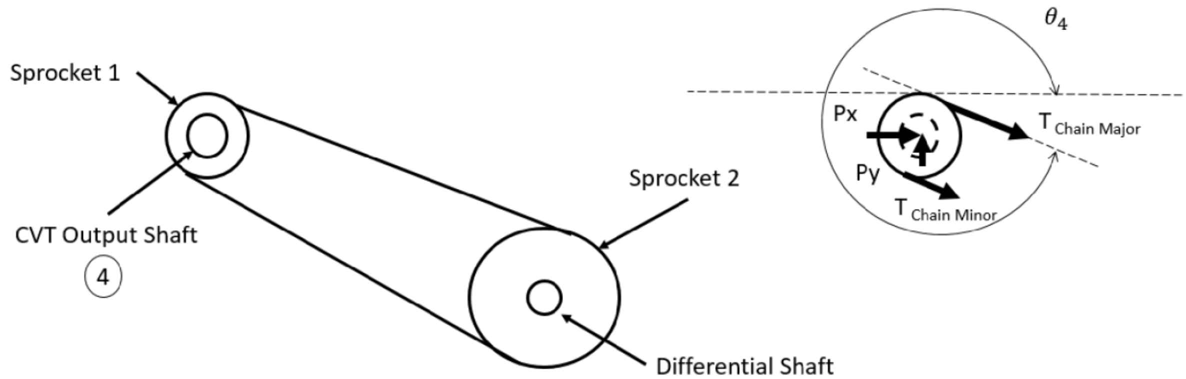
$G_{1y} = -10747.4782\ N$

$+\rightarrow \sum F_x = 0$ $G_{1x} + G_1 \cdot \cos(\gamma) = 0$

$G_{1x} := G_1 \cdot \cos(\gamma)$

$G_{1x} = -3911.7622\ N$

CVT Chain Force Schematic



Radial Forces Exerted on Shaft from Differential Chain:

$T_{chain\_Major} := \frac{T_{max}}{R_{sprocket}}$ $R_{sprocket} := 35\ mm$ Radius of the driving sprocket (pinion)

$T_{chain\_Major} = 6448.4869\ N$ $\theta_4 := 315\ deg$

$T_{chain\_Minor} := 0\ kN$ The minor chain force was assumed negligable

$+\uparrow \sum F_y = 0$ $P_y + T_{chain\_Major} \cdot \sin(\theta_4) = 0$

$P_y := T_{chain\_Major} \cdot \sin(\theta_4) \cdot \frac{1}{2}$

$P_y = -2279.8844\ N$

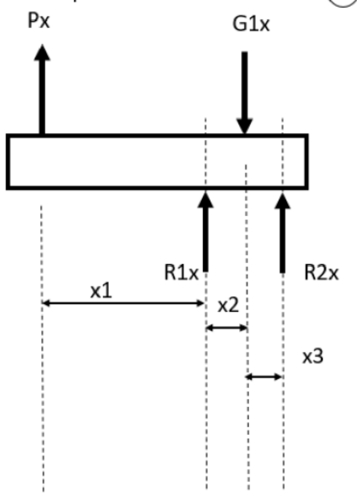
$+\rightarrow \sum F_x = 0$ $P_x + T_{chain\_Major} \cdot \cos(\theta_4) = 0$

$P_x := T_{chain\_Major} \cdot \cos(\theta_4) \cdot \frac{1}{2}$

$P_x = 2279.8844\ N$

Bearing Reaction Calculations

FBD of CVT Output Shaft in X Coordinate (4)



$$x_1 := 32.5\ mm$$

$$x_2 := 20\ mm$$

$$x_3 := 20\ mm$$

$$+\uparrow \sum F_x = 0 \qquad R_{1x} + R_{2x} + G_{1x} + P_x = 0$$

$$R_{1x} + R_{2x} = -\left(G_{1x} + P_x\right)$$

$$+\circlearrowleft \sum M_{R1x} = 0$$

$$G_{1x} \cdot x_2 + R_{2x} \cdot \left(x_2 + x_3\right) - P_x \cdot \left(x_1\right) = 0$$

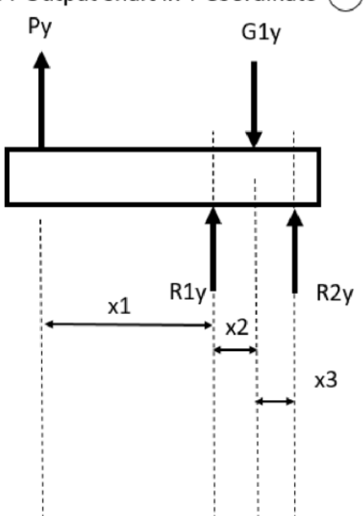
$$R_{2x} := \frac{P_x \cdot \left(x_1\right) - G_{1x} \cdot x_2}{x_2 + x_3}$$

$$R_{2x} = 3808.2872\ N$$

$$R_{1x} := -\left(G_{1x} + P_x\right) - R_{2x}$$

$$R_{1x} = -2176.4094\ N$$

FBD of CVT Output Shaft in Y Coordinate (4)



$$+\uparrow \sum F_y = 0 \qquad R_{1y} + R_{2y} + G_{1y} + P_y = 0$$

$$R_{1y} + R_{2y} = -\left(G_{1y} + P_y\right)$$

$$+\circlearrowleft \sum M_{R1y} = 0$$

$$G_{1y} \cdot x_2 + R_{2y} \cdot \left(x_2 + x_3\right) - P_y \cdot \left(x_1\right) = 0$$

$$R_{2y} := \frac{P_y \cdot \left(x_1\right) - G_{1y} \cdot x_2}{x_2 + x_3}$$

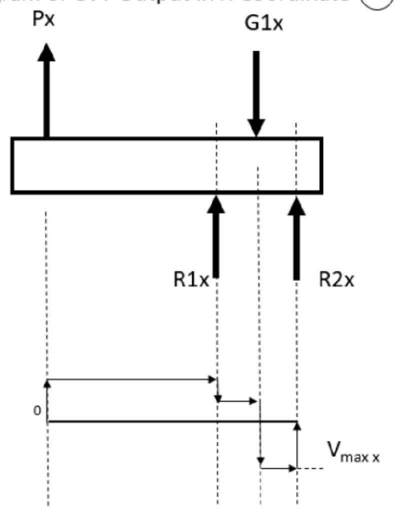
$$R_{2y} = 3521.333\ N$$

$$R_{1y} := -\left(G_{1y} + P_y\right) - R_{2y}$$

$$R_{1y} = 9506.0296\ N$$

Shear Diagrams

Shear Diagram of CVT Output in X Coordinate (4)

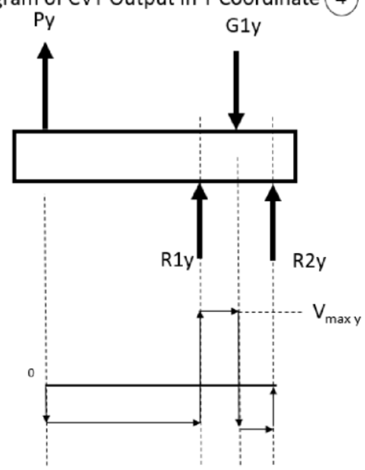


Here maximum shear occurs at V max x.

$$V_{max\_x} := P_x + R_{1x} + G_{1x}$$

$$V_{max\_x} = -3808.2872\,N$$

Shear Diagram of CVT Output in Y Coordinate (4)

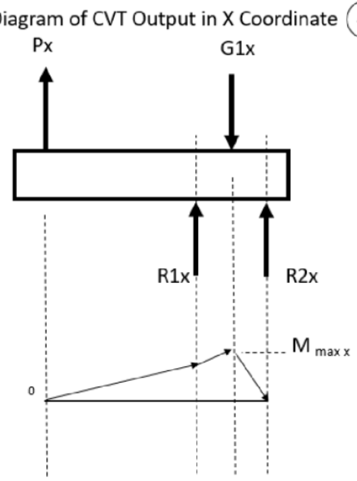


Here maximum shear occurs at V max x.

$$V_{max\_y} := P_y + R_{1y}$$

$$V_{max\_y} = 7226.1452\,N$$

Bending Diagram of CVT Output in X Coordinate (4)

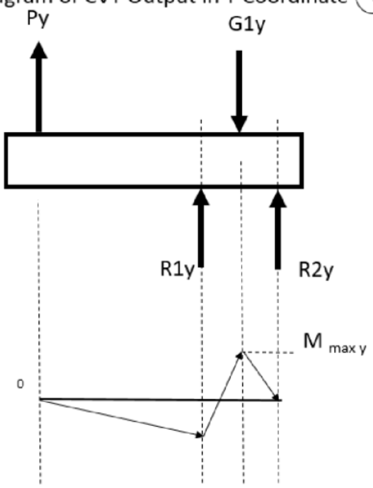


Here maximum bending occurs at M max x.

$$M_{max\_x} := \left(-\left(R_{1x} + P_x + G_{1x}\right)\right) \cdot x_3$$

$$M_{max\_x} = 76.1657\,J$$

Bending Diagram of CVT Output in Y Coordinate 4



Here maximum bending occurs at M max y.

$$M_{max\_y} := \left(-\left(P_y + R_{1y} + G_{1y}\right)\right) \cdot x_3$$

$$M_{max\_y} = 70.4267\ J$$

Shaft Material:
AISI 4340 Normalized Steel

$S_{yield} := 710\ MPa$ Tensile Strength: Yield

$E := 205\ GPa$ Youngs Modulus

$S_{ut} := 1110\ MPa$ Tensile Strength: Ultimate

Shaft minimum diameter based on DE-ASME elliptic criterion:

$$D_{min} := \left(\left(\frac{16 \cdot n_f}{\pi} \cdot \sqrt{\frac{4 \cdot \left(K_f \cdot M_a\right)^2 + \left(K_{fs} \cdot T_a\right)^2 \cdot 3}{S_e^{\ 2}} + \frac{4 \cdot \left(K_f \cdot M_m\right)^2 + 3 \cdot \left(K_{fs} \cdot T_m\right)^2}{S_{yield}^{\ 2}}}\right)^{\frac{1}{3}}\right)$$

$n_f := 1.5$ Safety Factor — Safety Factor

$K_f := 1$ $K_{fs} := 1$ — For shaft calculations use nominal stress concentration factors of 1. Locations of stress concentration will be considered later

$M_{min} := 0$ $M_{max} := \sqrt{M_{max\_x}^{\ 2} + M_{max\_y}^{\ 2}}$ — Min and Max moments based on operating conditions

$M_m := \frac{M_{max} + M_{min}}{2}$ $M_a := \frac{M_{max} - M_{min}}{2}$ — Mean and alternating moments

$T_{min} := 0$ — Min torque based on operating condition (max calculated above)

$T_m := \frac{T_{max} + T_{min}}{2}$ $T_a := \frac{T_{max} - T_{min}}{2}$ — Mean and alternating torques

$S_e := 0.504 \cdot S_{ut} \cdot 0.703 \cdot 0.879 \cdot 1 \cdot 1 \cdot 0.868 \cdot 1$ — Elliptic stress with factors

$\boxed{D_{min} = 18.2877\ mm}$ Chosen Shaft Diameter = 20 mm

Bearing Selections

	Bearing 1:	Bearing 2:
Reaction Forces:	$R_1 := \sqrt{R_{1x}^2 + R_{1y}^2}$	$R_2 := \sqrt{R_{2x}^2 + R_{2y}^2}$
	$R_1 = 9751.9925\,N$	$R_2 = 5186.7945\,N$
Number of cycles rated for (million)	$L_{D1} := 20$	$L_{D2} := 20$
Dynamic load factor:	$C_{r1} := R_1 \cdot L_{D1}^{\frac{1}{3}}$	$C_{r2} := R_2 \cdot L_{D2}^{\frac{1}{3}}$
	$C_{r1} = 26470.9802\,N$	$C_{r2} = 14079.1264\,N$

Bearing Selected (1 and 2): FAG QJ304-MPA Four Point Contact Bearing

$d_{inner} := 20\,mm$ Width $:= 15\,mm$

$d_{outer} := 52\,mm$ Weight $:= 0.184\,kg$

Bearing Rating:

$C_{r.rating} := 30000\,N$ Dynamic force rating

$n_{r.rating} := 28000\,rpm$ Maximum speed rating

Therefore, bearings selected meet the mimimum design criteria based on the operating conditions they have been selected for.

Stress Concentration Factor Analysis

Consider the spline where the gear is mounted since moment and torsion are highest here

Select stress concentration factors based on conservative estimations for splines

$K_{t.spline} := 1.8$

$K_{ts.spline} := 2.5$

$q_{spline} := 0.8$

Therefore the fatigue stress factors can be calculated

$K_{f.spline} := 1 + q_{spline} \cdot \left(K_{t.spline} - 1\right) = 1.64$

$K_{fs.spline} := 1 + q_{spline} \cdot \left(K_{ts.spline} - 1\right) = 2.2$

Fatigue stress concentration factors to use in DE-ASME elliptic criterion

$K_f := K_{f.spline}$

$K_{fs} := K_{fs.spline}$

$$D := \left(\left(\frac{16 \cdot n_f}{\pi} \cdot \sqrt{\frac{4 \cdot \left(K_f \cdot M_a\right)^2 + \left(K_{fs} \cdot T_a\right)^2 \cdot 3}{S_e^{\,2}} + \frac{4 \cdot \left(K_f \cdot M_m\right)^2 + 3 \cdot \left(K_{fs} \cdot T_m\right)^2}{S_{yield}^{\,2}}}\right)^{\frac{1}{3}}\right)$$

$\boxed{D = 23.381\ mm}$ The minimum diameter of the spline is 27 mm

Consider the step where torsion and moment are severe - the step from the bearing shoulder to the gear spline

$D_1 := 27\ mm$ Diameter of bearing shoulder

$d_1 := 20\ mm$ Average diameterof the spline

$r_1 := 1\ mm$ Fillet Radius after step

$\frac{D_1}{d_1} = 1.35$ $\frac{r_1}{d_1} = 0.05$ Ratios for selecting notch sensitivity

From these ratios, the appropriate notch sensitivity factor and stress concentration factors were selected from tables in MECE 360 course notes and design tables.

$q_1 := 0.8$

$K_{t.1} := 1.8$

$K_{ts.1} := K_{t.1}$

Therefore the fatigue stress factors can be calculated

$$K_{f.1} := 1 + q_1 \cdot \left(K_{t.1} - 1\right) = 1.64$$

$$K_{fs.spline} := 1 + q_{spline} \cdot \left(K_{ts.spline} - 1\right) = 2.2$$

Fatigue stress concentration factors to use in DE-ASME elliptic criterion

Finally, the minimum required diameter for this location was evaluated using the most severe stress conditions.

$$K_f := K_{f.1}$$

$$K_{fs} := K_{fs.1}$$

$\boxed{D = 23.381\ mm}$ The minimum diameter in this location is 20 mm

Both major stress concentration locations that were evaluated resulted in minimum diameter requirements less than the shaft diameter. Therefore, the shaft size is sufficient to prevent system failure.

A.4. Finite Element Analysis (FEA)

Objective

The objective of this finite element analysis is to determine the maximum deflection and stress in all custom designed CVT gearbox parts. From these values, a material safety factor was calculated to assure no yielding will occur and that there is insufficient material deflection to risk gasket/O-ring leaks.

Background

While all quoted CVTech parts were picked taking into consideration their specified operating conditions, all custom designed gearbox parts were analyzed using the finite element technique to assure structural soundness with an acceptable minimum allowable factor of safety. For the analysis ANSYS Workbench version 18.1 was used.

Material Properties

To reduce the manufacturing cost of the gearbox, AISI 1020 steel was chosen as a readily available, inexpensive material option. Table A 1 outlines the material properties of this steel taken from the material properties section of SolidWorks 2017 software.

Table A 1 AISI 1020 Steel Material Properties taken from SolidWorks 2017

Property	**Value**
Elastic Modulus	200 GPa
Poisson’s Ratio	0.29
Shear Modulus	77 GPa
Mass Density	7900 kg/m^3
Tensile Strength	421 MPa
Yield Strength	352 MPa

Method

Using the force/torque calculations, bearing forces were applied to each cap/plate under an assumed steady state operating condition in ANSYS Workbench v18.1 finite element software. Selecting the outside perimeter of each cap/plate as a fixed boundary support to avoid singularity (unrealistically high stress concentrations) in each bolt hole, total deformation and Von-Mises stress plots were created for each of the 6 custom made components. This boundary condition is justified as the outside perimeter of each cap/plate was estimated to remain relatively constant and maximum deflections in each cap/plate bolt hole was found to be in the realm of thousandths of a millimeter. Convergence tests were competed for each displacement/stress plot to confirm reliable results. In addition, factory of safety plots were created for each component confirming that none of the designed parts are at any risk of material failure due to excessively high yield stress.

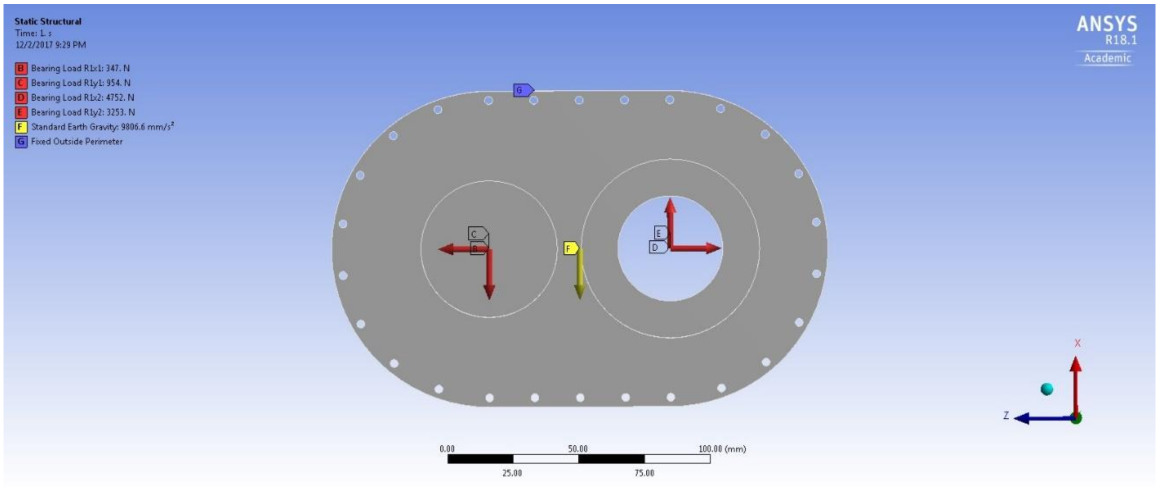
Results

The following Table A 2 summarizes the results of the finite element analysis as calculated by the ANSYS Workbench v18.1 software. Table X.2 shows the meshing information, degrees of freedom, stress type, maximum stress, maximum displacement, and minimum safety factor in each of the 6 gearbox caps/plates designed.

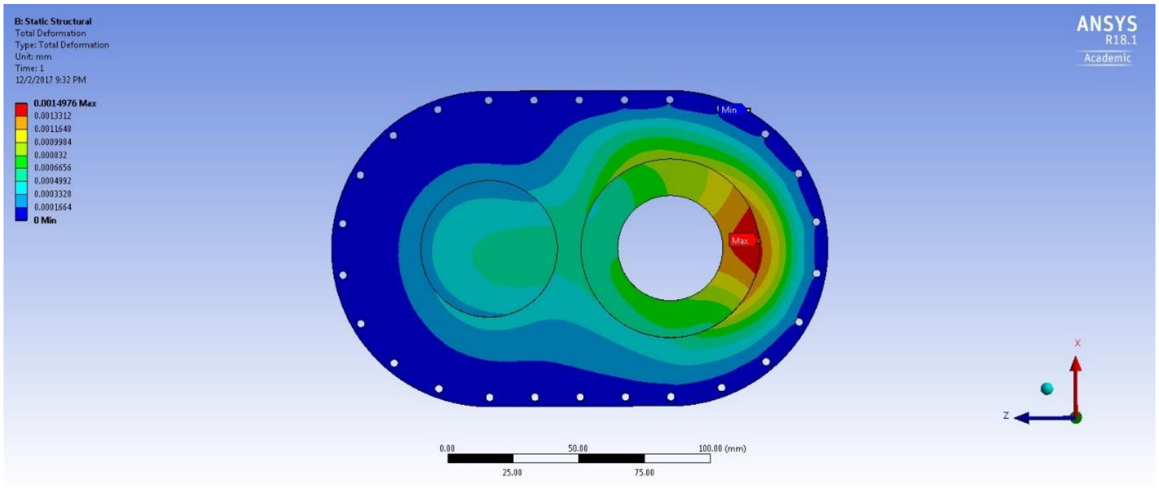
Table A 2 Finite Element Analysis Results Summary Table

	68mm Bearing Cap 1-2	62mm Bearing Cap 1-2	52mm Bearing Cap 3-4	62mm Bearing Cap 3-4	Frame Plate 1	Frame Plate 2
Mesh Type	Tetrahedral	Tetrahedral	Tetrahedral	Tetrahedral	Tetrahedral	Tetrahedral
Mesh Size (mm)	11.175	11.175	11.175	11.175	11.175	11.175
Mesh Tolerance	0.05	0.05	0.05	0.05	0.05	0.05
Degrees of Freedom	85,182	223,263	257,751	266,958	199,779	190,455
Stress Type	Von Mises	Von Mises	Von Mises	Von Mises	Von Mises	Von Mises
Maximum Stress (MPa)	27.317	35.061	51.933	38.008	36.166	44.239
Maximum Displacement (mm)	0.002	0.007	0.003	0.002	0.005	0.010
Minimum Safety Factor	9.152	7.130	4.814	6.578	6.913	5.651
Required Safety Factor	1.50	1.50	1.50	1.50	1.50	1.50

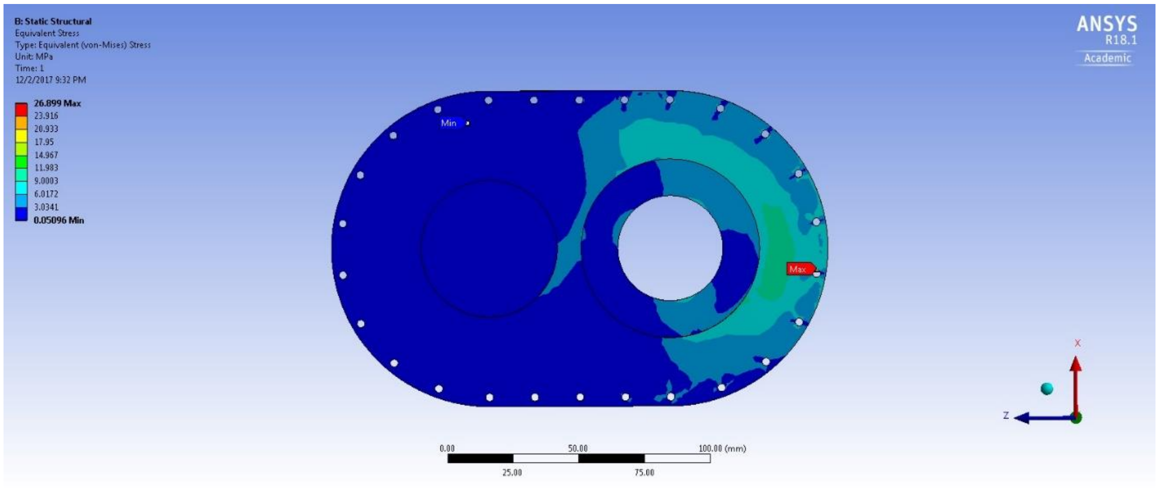
The following Figure A 1 to Figure A 12 below show the boundary and loading conditions applied to each part, along with the total deformation, maximum stress, and factor of safety plots. Additionally, convergence plots for both total deformation and maximum stress were created to assure reliability of the finite element analysis results.



68mm Bearing Cap 1-2 Applied Loads and Boundary Conditions

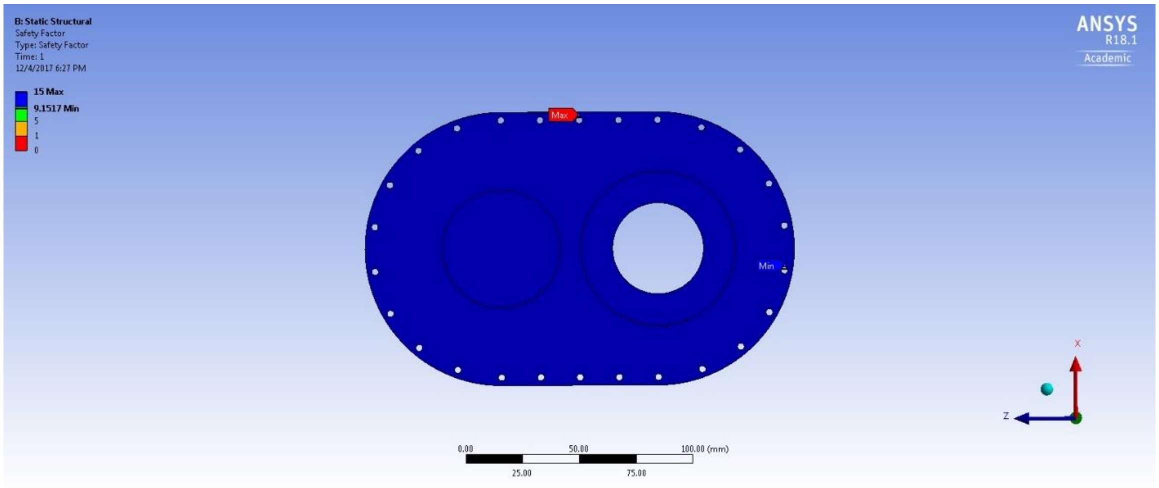


68mm Bearing Cap 1-2 Total Displacement Diagram

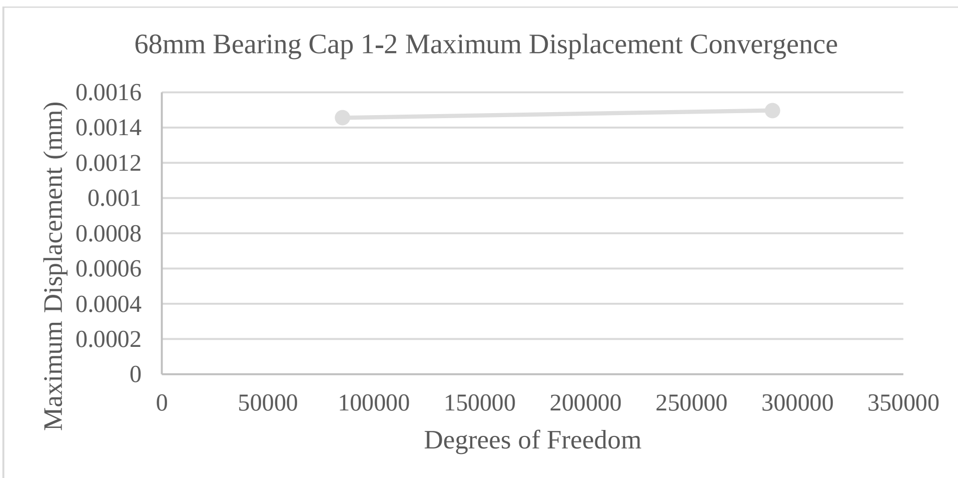


68mm Bearing Cap 1-2 Equivalent Stress Diagram

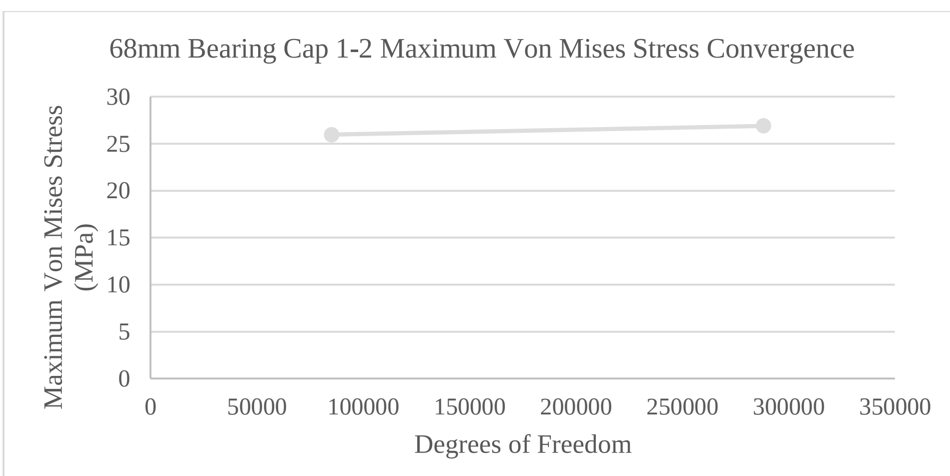
Figure A 1 68mm Bearing Cap 1-2 Finite Element Analysis Results



68mm Bearing Cap 1-2 Factory of Safety Stress Diagram

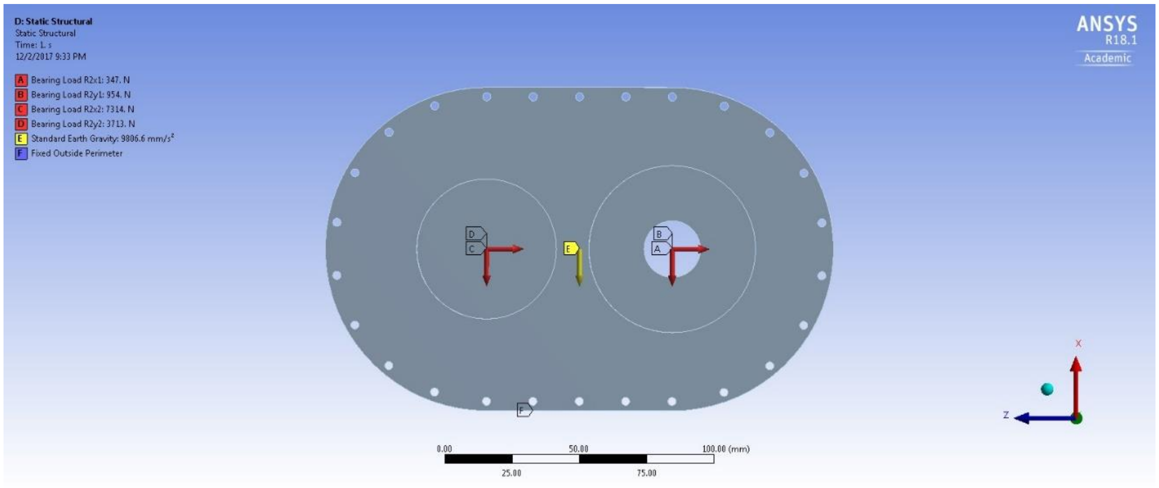


68mm Bearing Cap 1-2 Total Displacement Convergence Plot

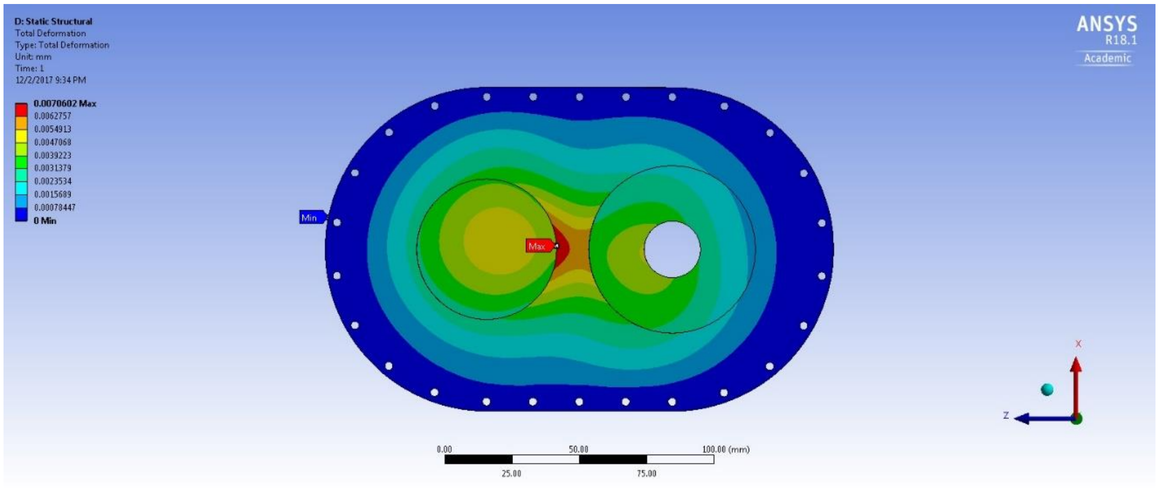


68mm Bearing Cap 1-2 Equivalent Stress Convergence Plot

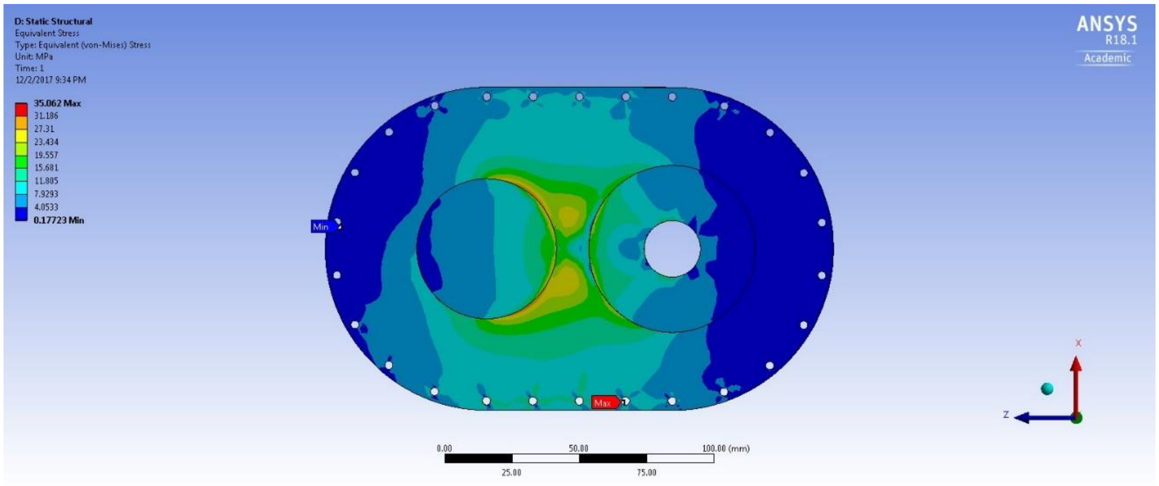
Figure A 2 68mm Bearing Cap 1-2 Finite Element Analysis Results



62mm Bearing Cap 1-2 Applied Loads and Boundary Conditions

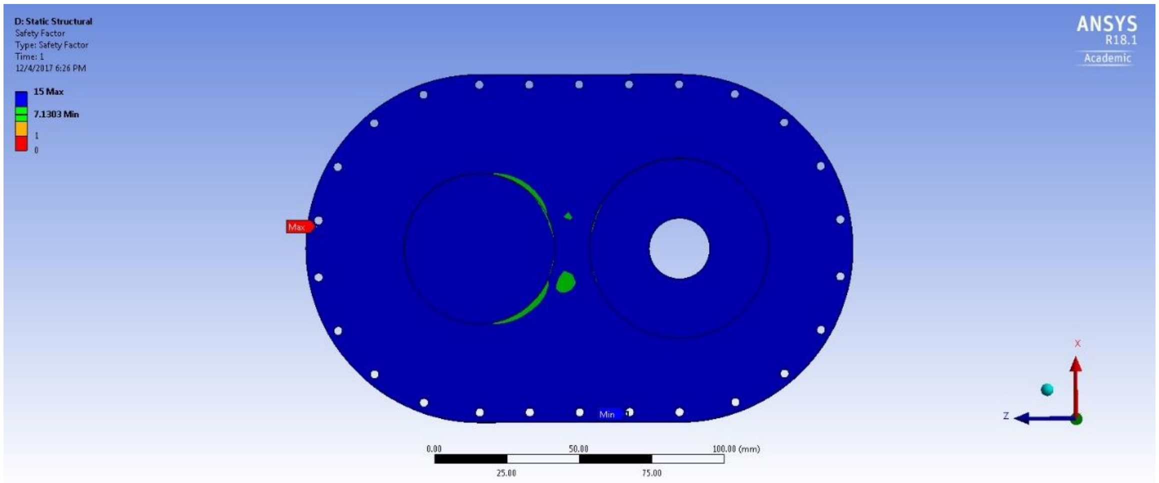


62mm Bearing Cap 1-2 Total Displacement Diagram

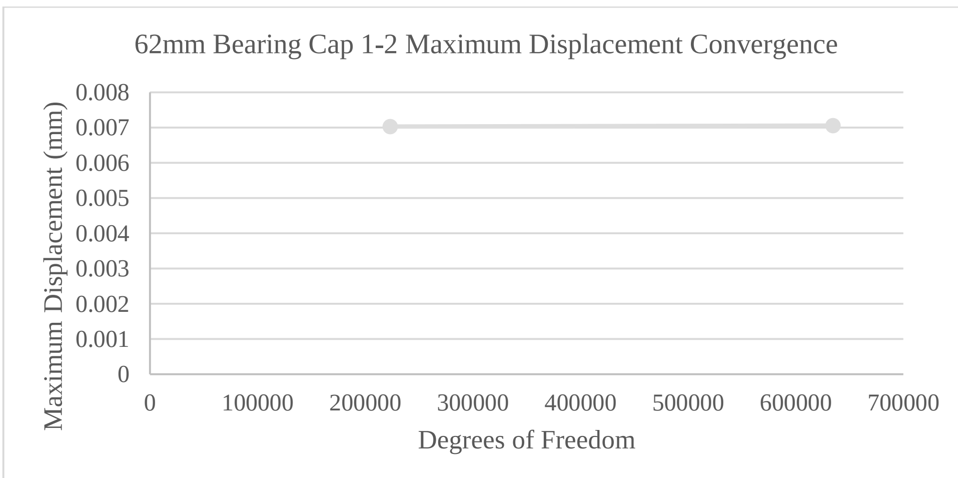


62mm Bearing Cap 1-2 Equivalent Stress Diagram

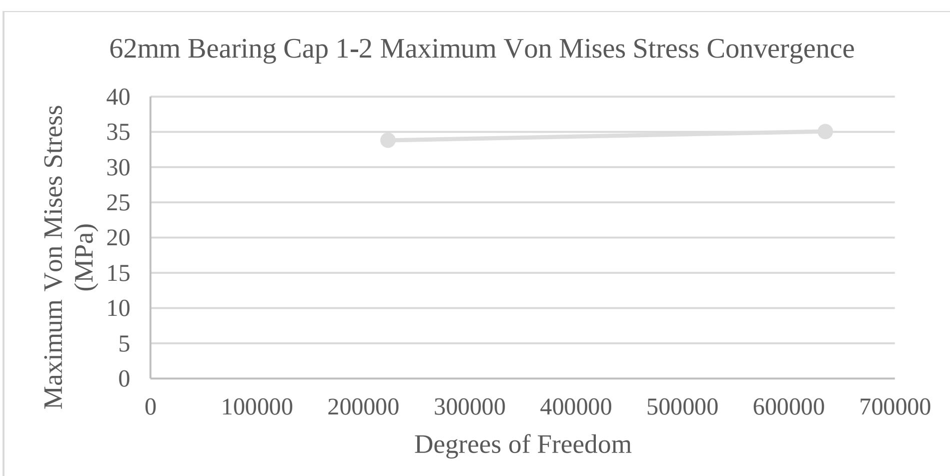
Figure A 3 62mm Bearing Cap 1-2 Finite Element Analysis Results



62mm Bearing Cap 1-2 Factory of Safety Stress Diagram

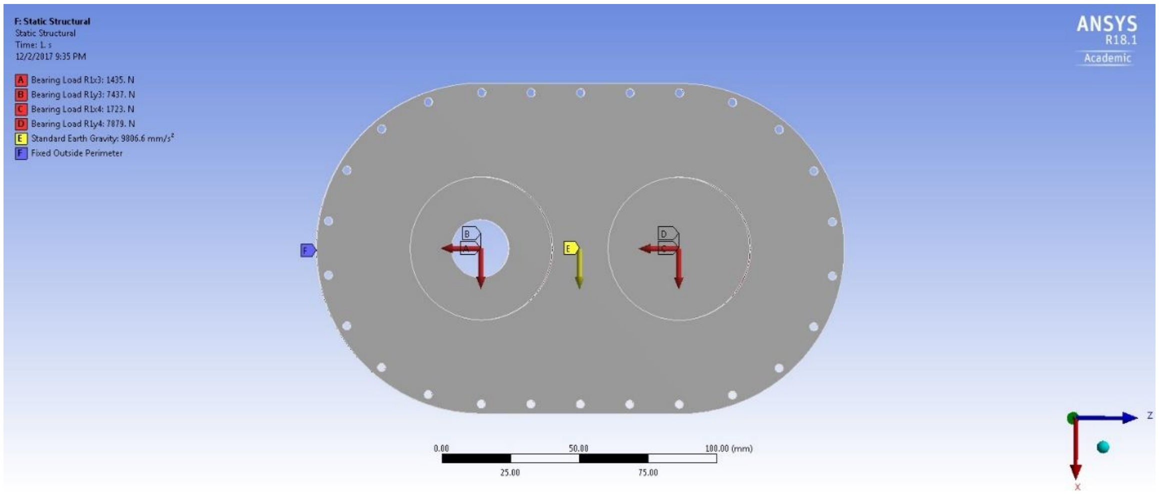


62mm Bearing Cap 1-2 Total Displacement Convergence Plot

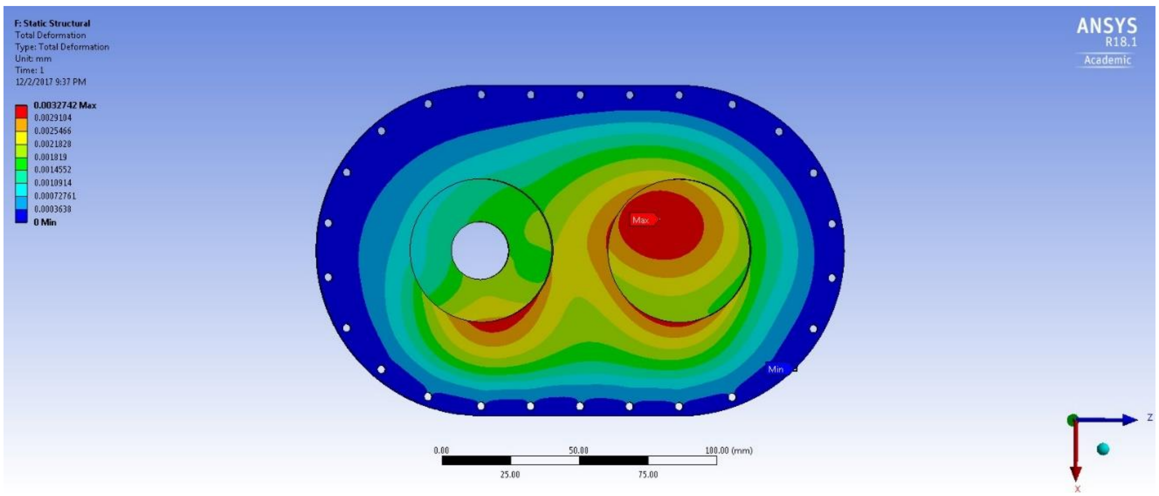


62mm Bearing Cap 1-2 Equivalent Stress Convergence Plot

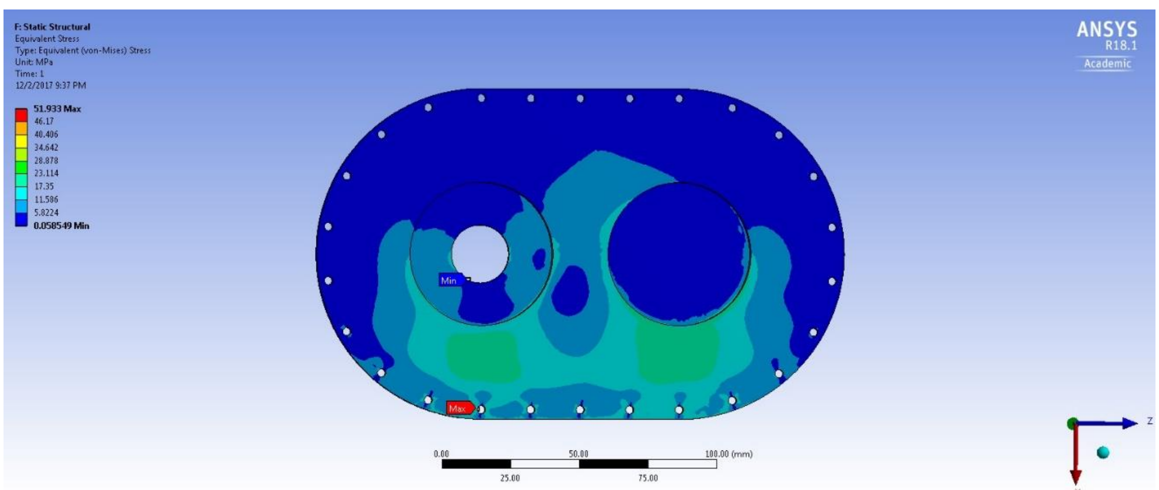
Figure A 4 68mm Bearing Cap 1-2 Finite Element Analysis Results



52mm Bearing Cap 3-4 Applied Loads and Boundary Conditions

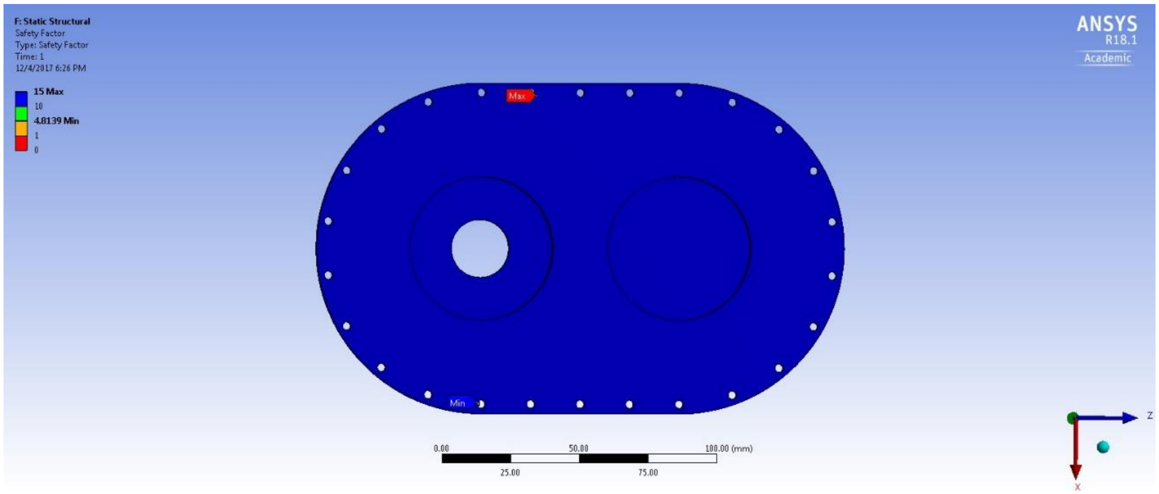


52mm Bearing Cap 3-4 Total Displacement Diagram

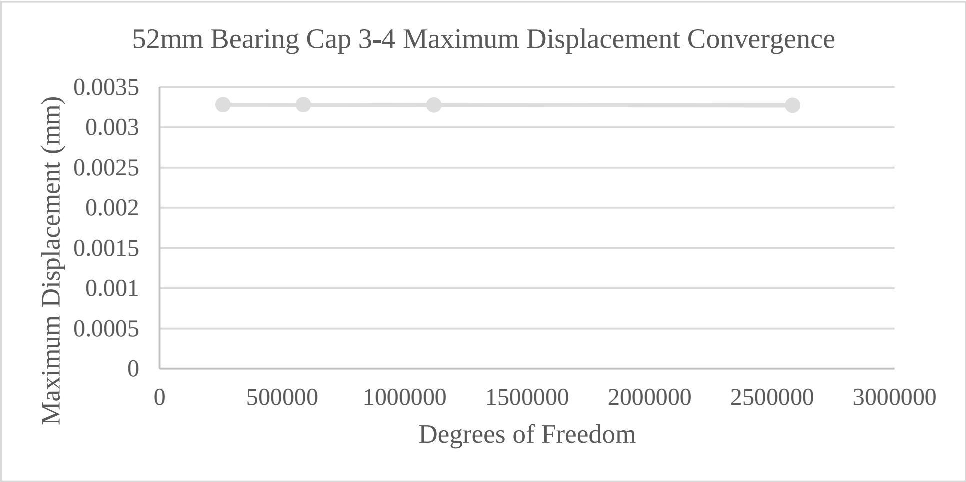


52mm Bearing Cap 3-4 Equivalent Stress Diagram

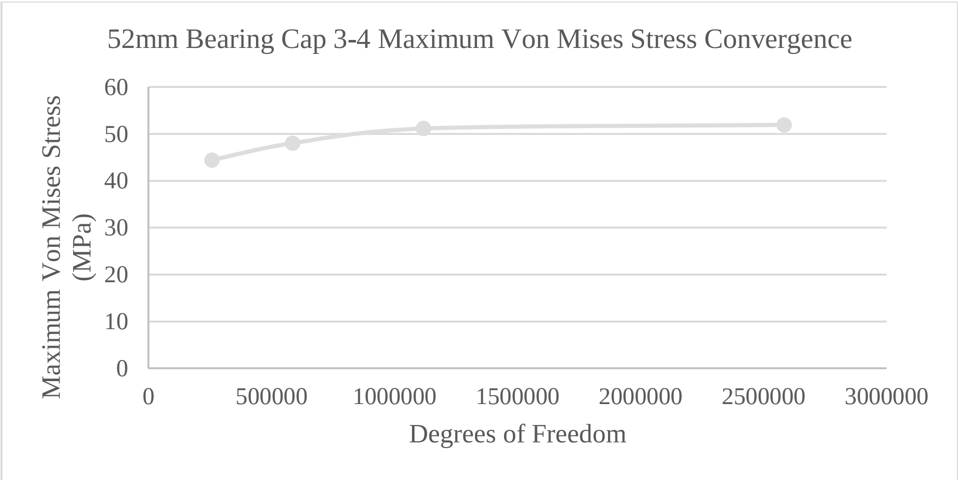
Figure A 5 52mm Bearing Cap 3-4 Finite Element Analysis Results



52mm Bearing Cap 3-4 Factory of Safety Stress Diagram

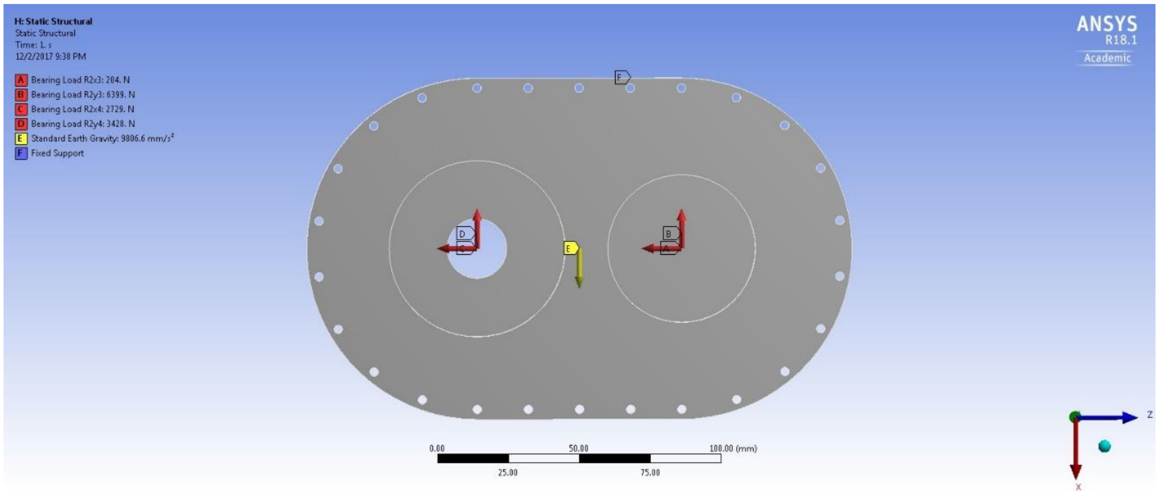


52mm Bearing Cap 3-4 Total Displacement Convergence Plot

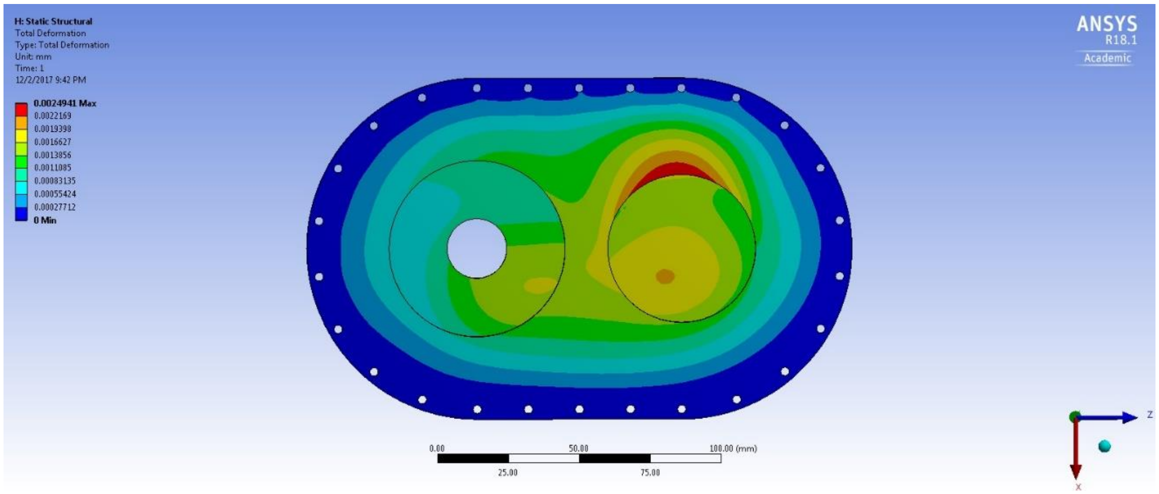


52mm Bearing Cap 3-4 Equivalent Stress Convergence Plot

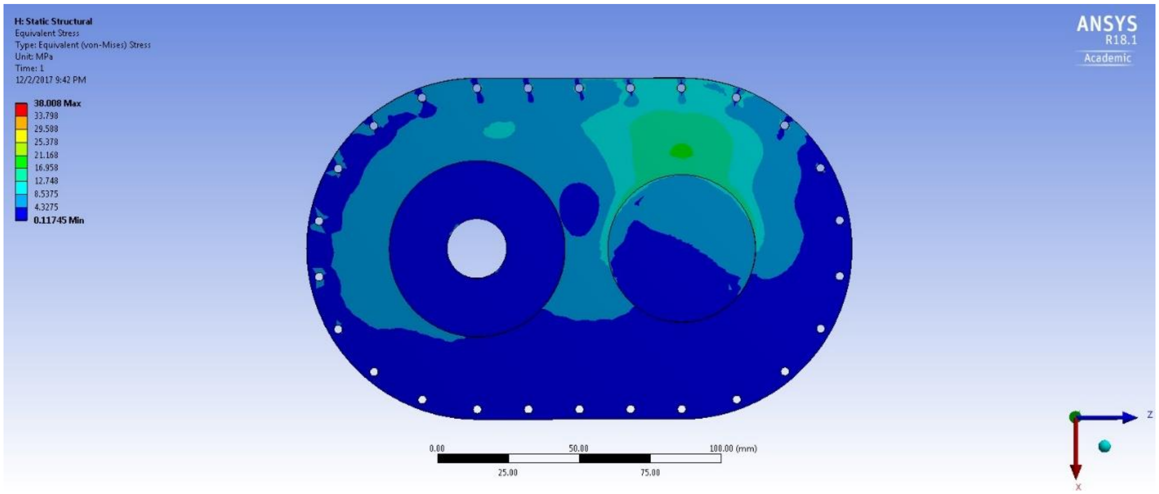
Figure A 6 68mm Bearing Cap 3-4 Finite Element Analysis Results



62mm Bearing Cap 3-4 Applied Loads and Boundary Conditions

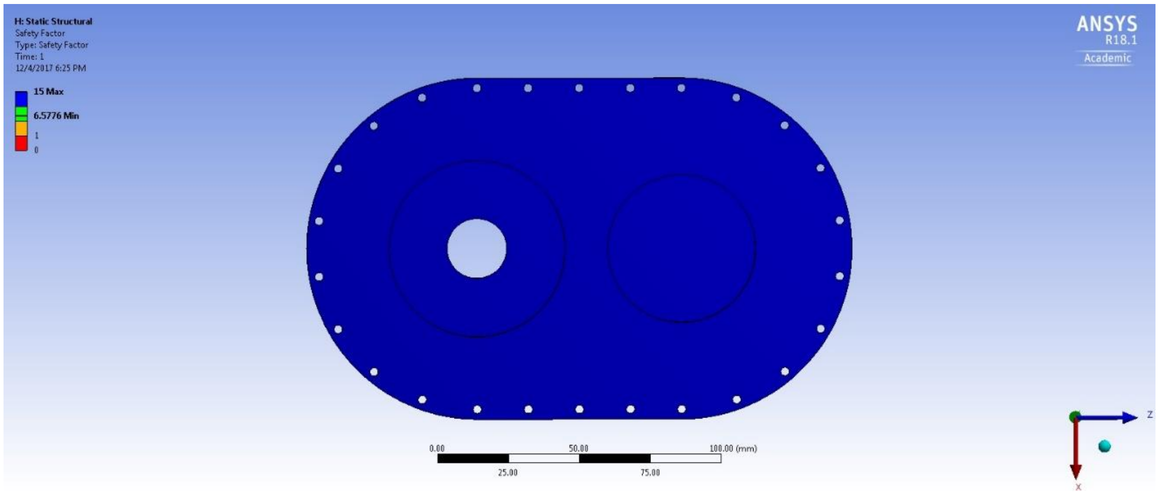


62mm Bearing Cap 3-4 Total Displacement Diagram

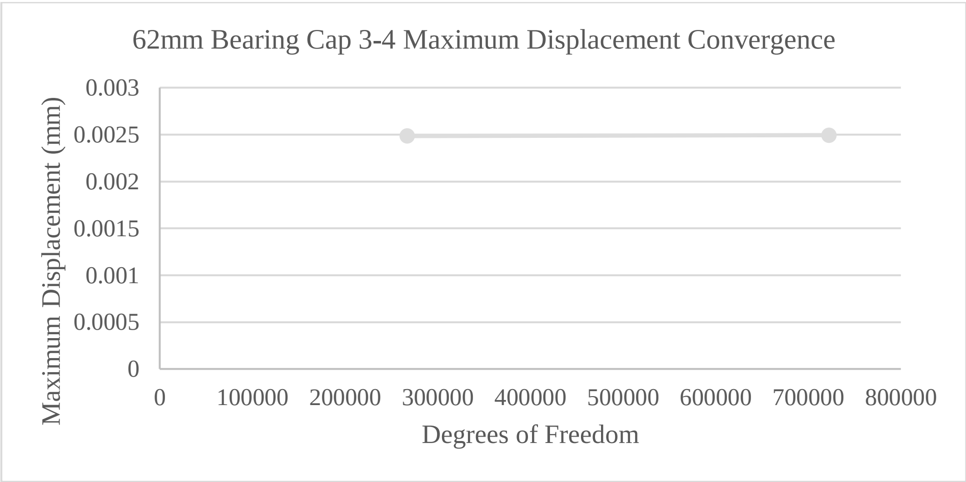


62mm Bearing Cap 3-4 Equivalent Stress Diagram

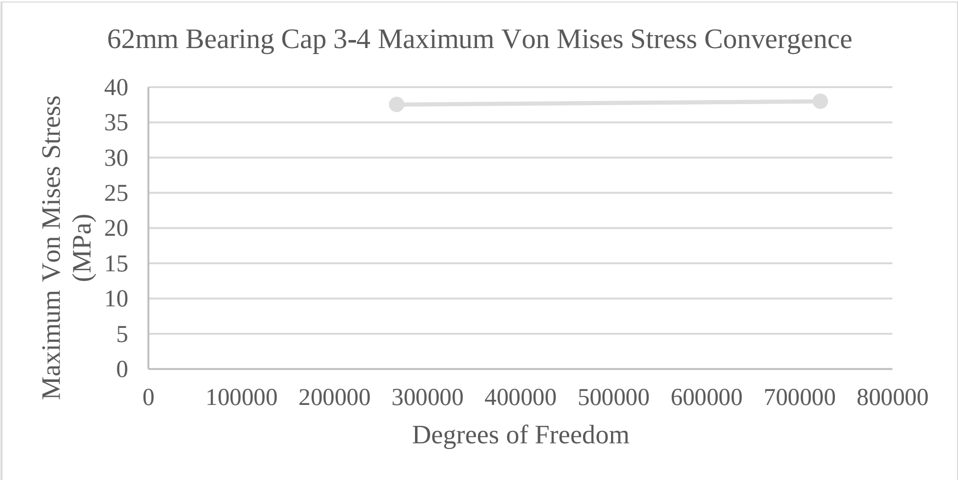
Figure A 7 62mm Bearing Cap 3-4 Finite Element Analysis Results



62mm Bearing Cap 3-4 Factory of Safety Stress Diagram

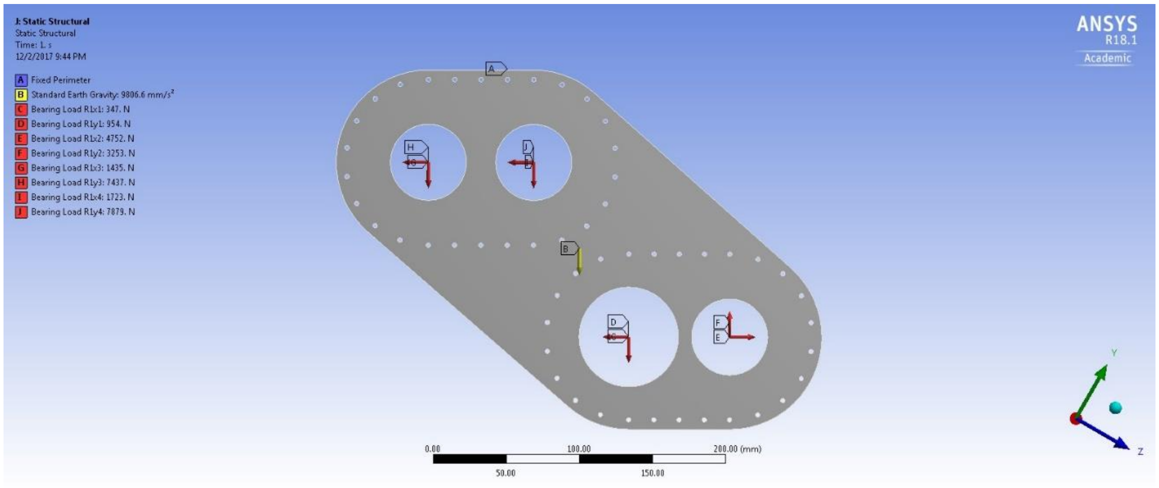


62mm Bearing Cap 3-4 Total Displacement Convergence Plot

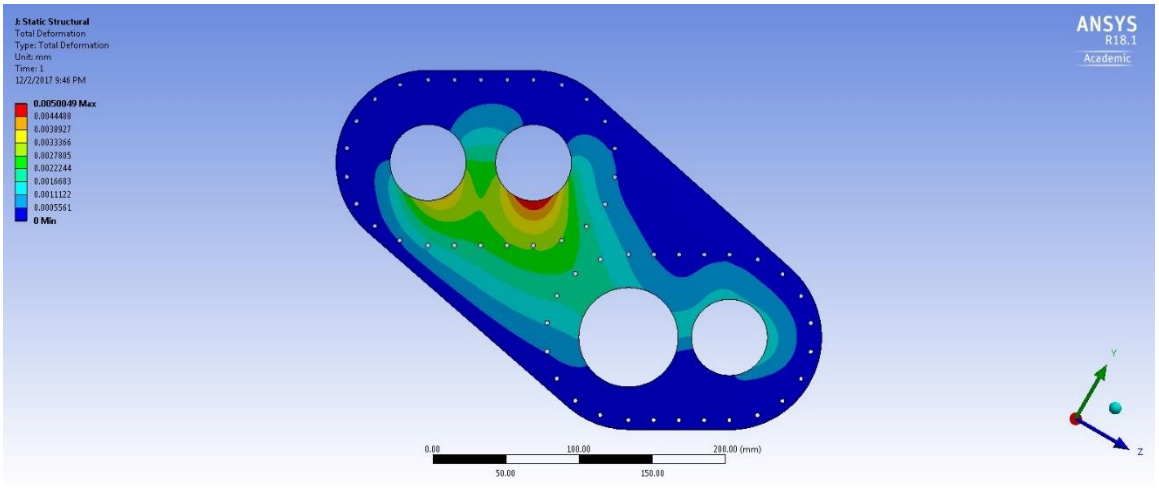


62mm Bearing Cap 3-4 Equivalent Stress Convergence Plot

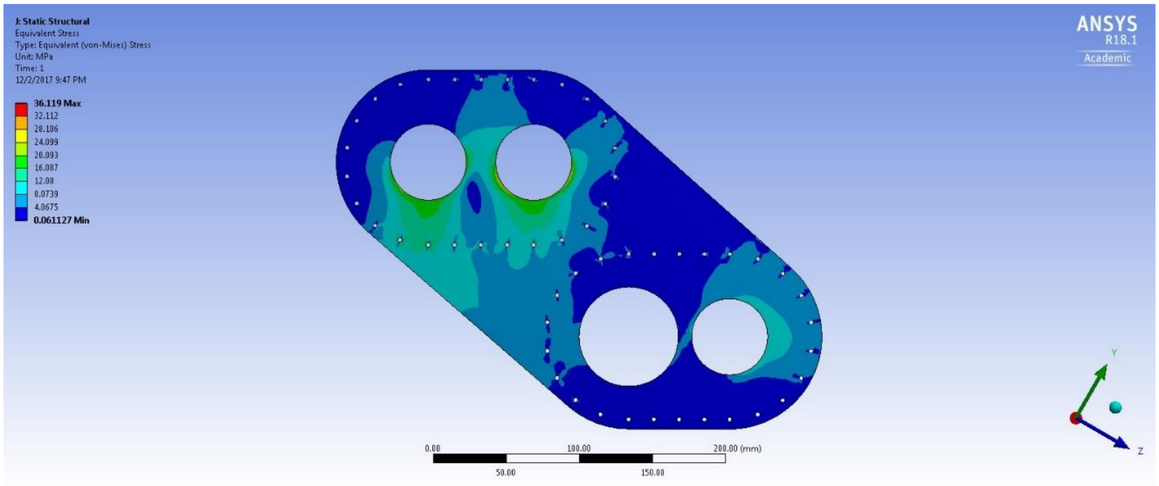
Figure A 8 62mm Bearing Cap 3-4 Finite Element Analysis Results



Frame Plate 1 Applied Loads and Boundary Conditions

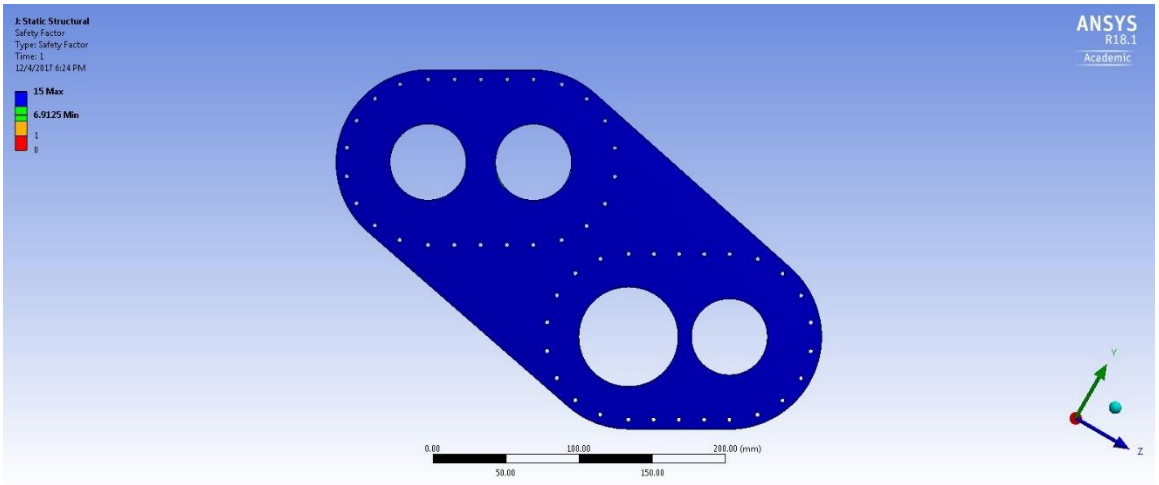


Frame Plate 1 Total Displacement Diagram

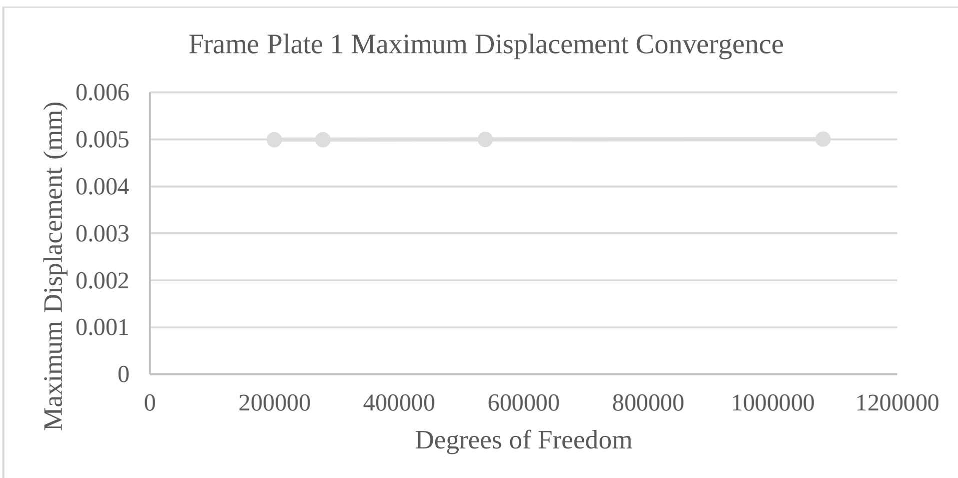


Frame Plate 1 Equivalent Stress Diagram

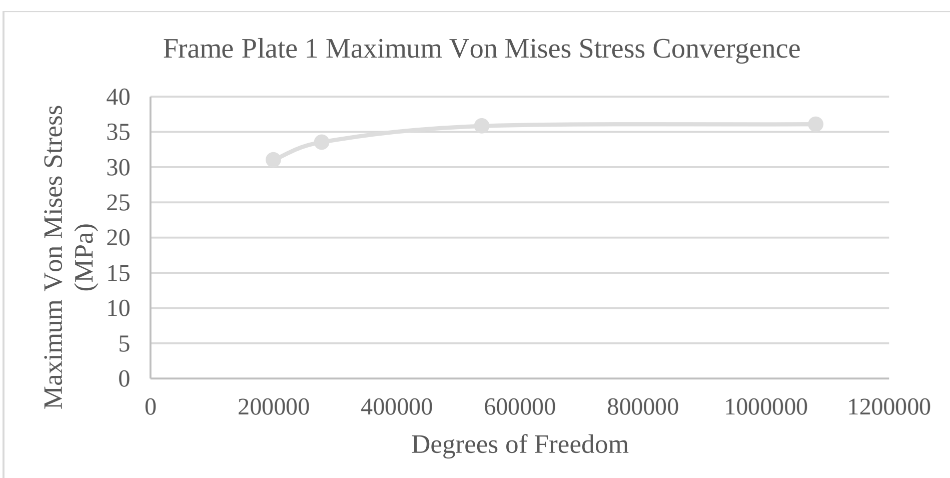
Figure A 9 Frame Plate 1 Finite Element Analysis Results



Frame Plate 1 Factory of Safety Stress Diagram

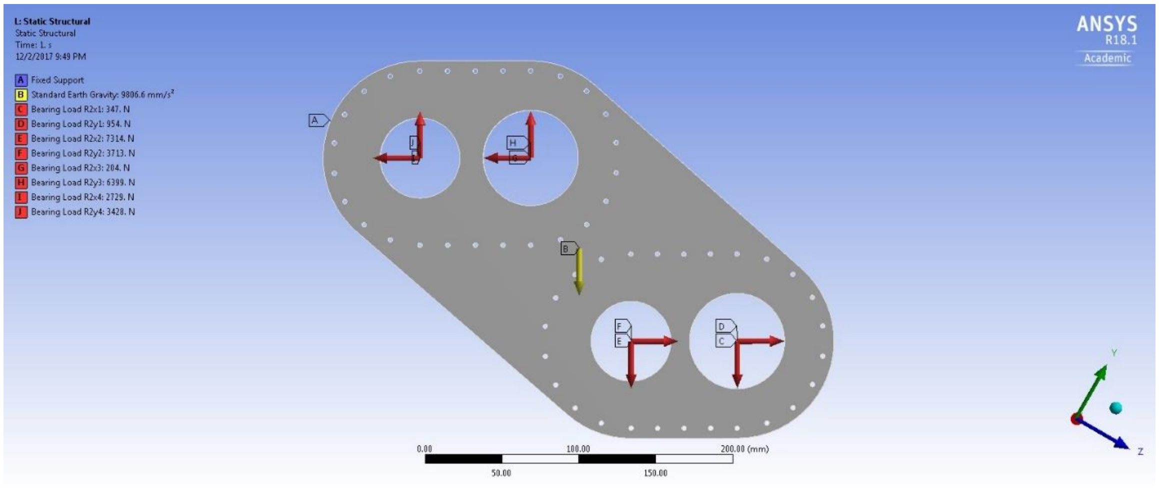


Frame Plate 1 Total Displacement Convergence Plot

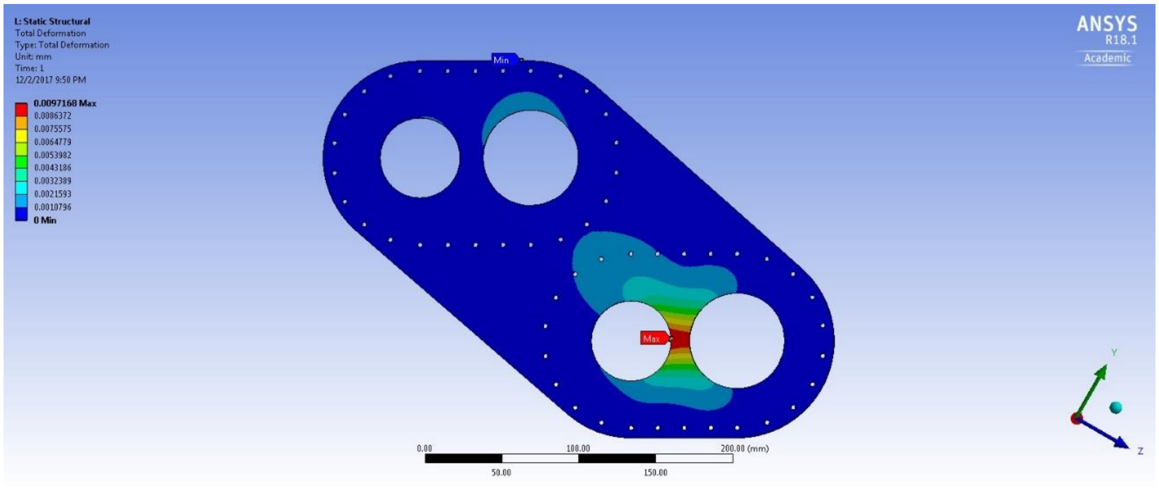


Frame Plate 1 Equivalent Stress Convergence Plot

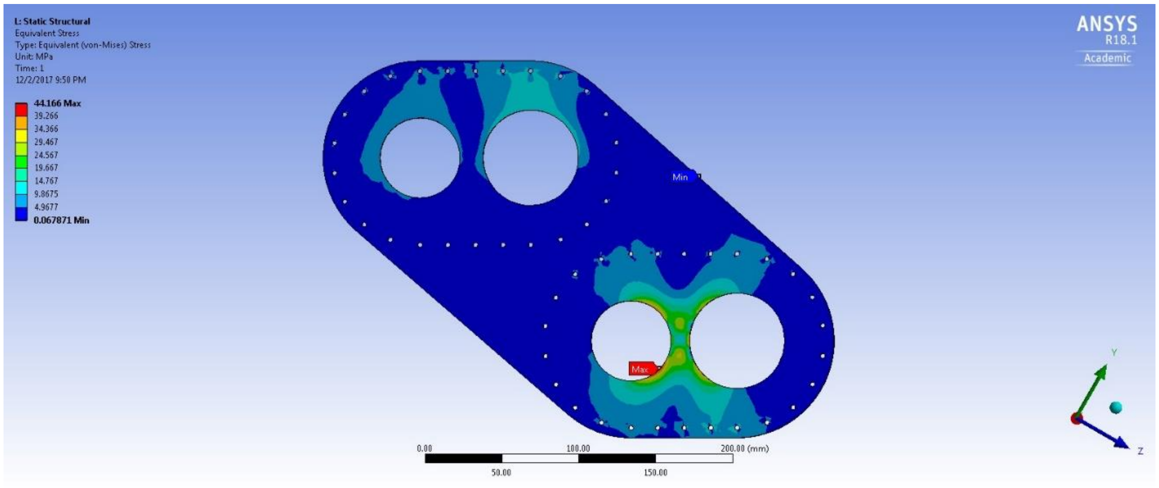
Figure A 10 Frame Plate 1 Finite Element Analysis Results



Frame Plate 2 Applied Loads and Boundary Conditions

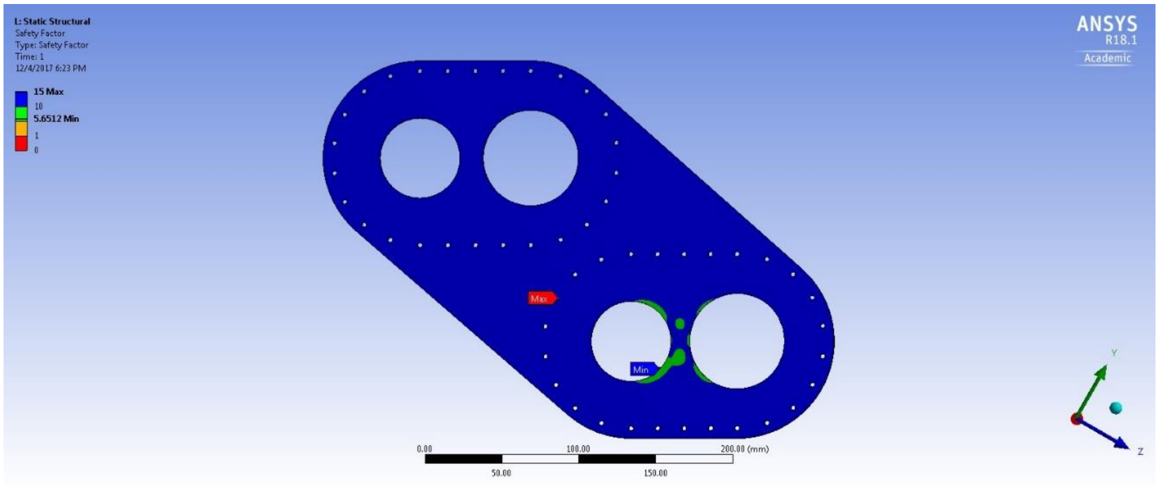


Frame Plate 2 Total Displacement Diagram

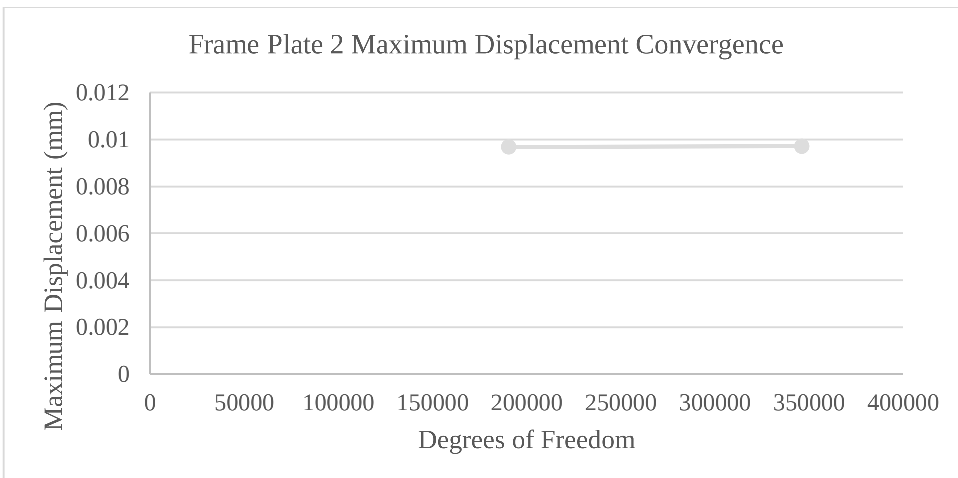


Frame Plate 2 Equivalent Stress Diagram

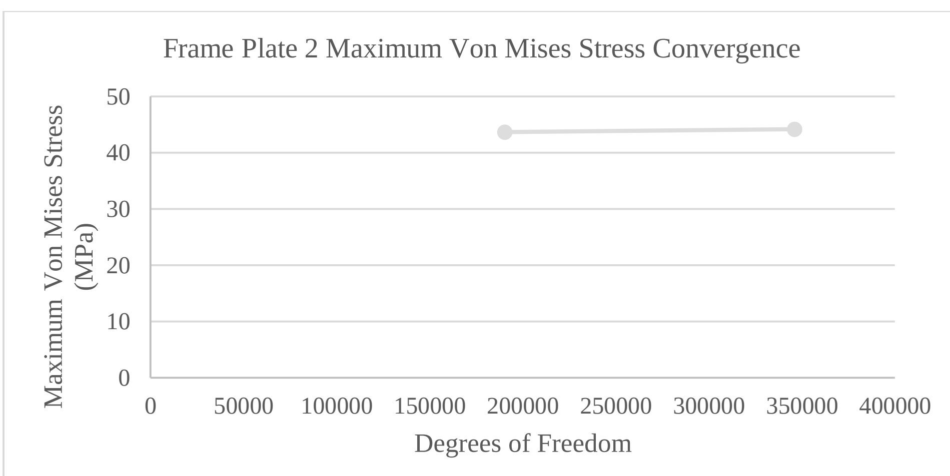
Figure A 11 Frame Plate 2 Finite Element Analysis Results



Frame Plate 2 Factory of Safety Stress Diagram



Frame Plate 2 Total Displacement Convergence Plot



Frame Plate 2 Equivalent Stress Convergence Plot

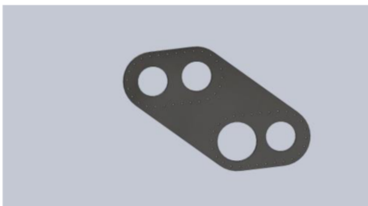
Figure A 12 Frame Plate 2 Finite Element Analysis Results

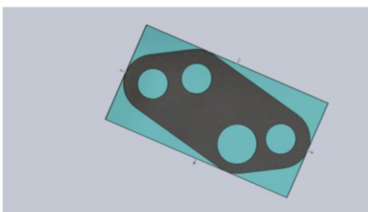
Table A 3 CVT Gearbox Components FEA Convergence Data

Part	Maximum Displacement (mm)	Maximum Von Mises Stress (MPa)	Number of Nodes	Degrees of Freedom	Number of Elements
68mm Bearing Cap 1-2	0.0014564	25.959	28394	85182	16510
	0.0014976	26.899	96054	288162	61321
62mm Bearing Cap 1-2	0.0070334	33.824	74421	223263	47326
	0.0070602	35.062	211530	634590	142684
52mm Bearing Cap 3-4	0.0032816	44.41	85917	257751	54663
	0.0032804	48.012	195470	586410	130436
	0.0032786	51.136	372869	1118607	254582
	0.0032742	51.933	861240	2583720	601455
62mm Bearing Cap 3-4	0.0024859	37.555	88986	266958	56818
	0.0024941	38.008	240775	722325	162246
Frame Plate 1	0.0049942	31.05	66593	199779	40879
	0.0049953	33.572	92642	277926	58501
	0.0050003	35.859	179448	538344	117407
	0.0050049	36.119	360264	1080792	241495
Frame Plate 2	0.009684	43.678	63485	190455	38474
	0.0097168	44.166	115362	346086	73426

Appendix B. Cost Analysis

SOLIDWORKS Costing Report





Model Name:	Frame Plate 1
Date and time of report:	11/30/2017 5:32:47 PM

Manufacturing Method:	Machining
Material:	Plain Carbon Steel
Stock weight:	3.00 kg
Stock Type	Plate
Plate Thickness:	6.50 mm
Material cost/weight:	0.50 USD/kg
Shop Rate:	N/A

Quantity to Produce

Total number of parts:	1
Lot size:	1

Estimated cost per part:	97.40 USD
Costing template used:	machiningtemplate_default(metric).sldctm
Costing mode used:	Manufacturing Process Recognition
Comparison:	100% Current 97.40 USD

Cost Breakdown

Material:	1.50 USD	2%
Manufacturing:	95.90 USD	98%
Markup:	0.00 USD	0%
Mold:	0.00 USD	0%

Estimated time per part:	02:29:33
Setups:	02:10:00
Operations:	00:19:33

Cost Report

1

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Model Name:	Frame Plate 1	Material:	Plain Carbon Steel	Material cost:	1.50 USD	Total cost /part:	97.40 USD
				Manufacturing cost:	95.90 USD	Total time /part:	02:29:33
				Markup:	0.00 USD		

Manufacturing Cost Breakdown

Operation Setups	Time (hh:mm:ss)	Cost (USD / Part)
Setup Operation 1	01:00:00	30.00
Setup Operation 2	01:00:00	45.00
Total	**02:00:00**	**75.00**

Load and Unload Setups	Time (hh:mm:ss)	Cost (USD / Part)
Setup Operation 1	00:05:00	2.50
Setup Operation 2	00:05:00	3.75
Total	**00:10:00**	**6.25**

Cut Operations	Cutting Method	Quantity	Cut Length (mm)	Time (hh:mm:ss)	Cost (USD / Part)
Cut Path 1	Water Jet	1	818.28	00:07:46	5.83
Cut Path 2	Water Jet	1	9.43	00:00:05	0.07
Cut Path 3	Water Jet	1	9.43	00:00:05	0.07
Cut Path 4	Water Jet	1	9.43	00:00:05	0.07
Cut Path 5	Water Jet	1	9.43	00:00:05	0.07
Cut Path 6	Water Jet	1	9.43	00:00:05	0.07
Cut Path 7	Water Jet	1	9.43	00:00:05	0.07
Cut Path 8	Water Jet	1	9.43	00:00:05	0.07
Cut Path 9	Water Jet	1	9.43	00:00:05	0.07
Cut Path 10	Water Jet	1	9.43	00:00:05	0.07
Cut Path 11	Water Jet	1	9.43	00:00:05	0.07
Cut Path 12	Water Jet	1	9.43	00:00:05	0.07
Cut Path 13	Water Jet	1	9.43	00:00:05	0.07
Cut Path 14	Water Jet	1	9.43	00:00:05	0.07
Cut Path 15	Water Jet	1	9.43	00:00:05	0.07

2

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Cut Path 16	Water Jet	1	9.43	00:00:05	0.07
Cut Path 17	Water Jet	1	9.43	00:00:05	0.07
Cut Path 18	Water Jet	1	163.36	00:01:33	1.16
Cut Path 19	Water Jet	1	9.43	00:00:05	0.07
Cut Path 20	Water Jet	1	9.43	00:00:05	0.07
Cut Path 21	Water Jet	1	9.43	00:00:05	0.07
Cut Path 22	Water Jet	1	9.43	00:00:05	0.07
Cut Path 23	Water Jet	1	9.43	00:00:05	0.07
Cut Path 24	Water Jet	1	9.43	00:00:05	0.07
Cut Path 25	Water Jet	1	9.43	00:00:05	0.07
Cut Path 26	Water Jet	1	9.43	00:00:05	0.07
Cut Path 27	Water Jet	1	9.43	00:00:05	0.07
Cut Path 28	Water Jet	1	9.43	00:00:05	0.07
Cut Path 29	Water Jet	1	9.43	00:00:05	0.07
Cut Path 30	Water Jet	1	9.43	00:00:05	0.07
Cut Path 31	Water Jet	1	9.43	00:00:05	0.07
Cut Path 32	Water Jet	1	9.43	00:00:05	0.07
Cut Path 33	Water Jet	1	9.43	00:00:05	0.07
Cut Path 34	Water Jet	1	163.36	00:01:33	1.16
Cut Path 35	Water Jet	1	9.43	00:00:05	0.07
Cut Path 36	Water Jet	1	9.43	00:00:05	0.07
Cut Path 37	Water Jet	1	9.43	00:00:05	0.07
Cut Path 38	Water Jet	1	9.43	00:00:05	0.07
Cut Path 39	Water Jet	1	9.43	00:00:05	0.07
Cut Path 40	Water Jet	1	9.43	00:00:05	0.07
Cut Path 41	Water Jet	1	9.43	00:00:05	0.07
Cut Path 42	Water Jet	1	9.43	00:00:05	0.07
Cut Path 43	Water Jet	1	9.43	00:00:05	0.07
Cut Path 44	Water Jet	1	9.43	00:00:05	0.07

3

Cut Path 45	Water Jet	1	9.43	00:00:05	0.07
Cut Path 46	Water Jet	1	9.43	00:00:05	0.07
Cut Path 47	Water Jet	1	9.43	00:00:05	0.07
Cut Path 48	Water Jet	1	9.43	00:00:05	0.07
Cut Path 49	Water Jet	1	9.43	00:00:05	0.07
Cut Path 50	Water Jet	1	9.43	00:00:05	0.07
Cut Path 51	Water Jet	1	9.43	00:00:05	0.07
Cut Path 52	Water Jet	1	9.43	00:00:05	0.07
Cut Path 53	Water Jet	1	9.43	00:00:05	0.07
Cut Path 54	Water Jet	1	9.43	00:00:05	0.07
Cut Path 55	Water Jet	1	213.62	00:02:01	1.52
Cut Path 56	Water Jet	1	9.43	00:00:05	0.07
Cut Path 57	Water Jet	1	163.36	00:01:33	1.16
Cut Path 58	Water Jet	1	9.43	00:00:05	0.07
Cut Path 59	Water Jet	1	9.43	00:00:05	0.07
Cut Path 60	Water Jet	1	9.43	00:00:05	0.07
Cut Path 61	Water Jet	1	9.43	00:00:05	0.07
Total		**61**	**2049.83**	**00:19:28**	**14.61**

Operation	Surface Finish	Volume Removed (mm^3)	Time (hh:mm:ss)	Cost (USD / Part)	Tooling	Cost-per-Volume (USD/mm^3)
Left Face	Roughing	88652.06	00:00:05	0.04	Face Mill	N/A
Total		**88652.06**	**00:00:05**	**0.04**		

No Cost Features
Fillet 1
Fillet 2

Setup Operations

1. Setup Operation 1
   a. Left Face
2. Setup Operation 2

4

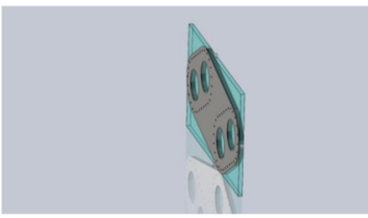
SOLIDWORKS

a. Cut Path 4
b. Cut Path 24
c. Cut Path 8
d. Cut Path 22
e. Cut Path 6
f. Cut Path 9
g. Cut Path 19
h. Cut Path 12
i. Cut Path 26
j. Cut Path 7
k. Cut Path 11
l. Cut Path 21
m. Cut Path 17
n. Cut Path 28
o. Cut Path 18
p. Cut Path 23
q. Cut Path 25
r. Cut Path 13
s. Cut Path 30
t. Cut Path 16
u. Cut Path 10
v. Cut Path 27
w. Cut Path 20
x. Cut Path 5
y. Cut Path 29
z. Cut Path 15
aa. Cut Path 14
bb. Cut Path 3
cc. Cut Path 2
dd. Cut Path 1
ee. Cut Path 31
ff. Cut Path 33
gg. Cut Path 38
hh. Cut Path 39
ii. Cut Path 45
jj. Cut Path 36
kk. Cut Path 37
ll. Cut Path 42
mm. Cut Path 34
nn. Cut Path 41
oo. Cut Path 32
pp. Cut Path 40
qq. Cut Path 35
rr. Cut Path 44

5

ss. Cut Path 43
tt. Cut Path 61
uu. Cut Path 57
vv. Cut Path 59
ww. Cut Path 60
xx. Cut Path 58
yy. Cut Path 54
zz. Cut Path 53
aaa. Cut Path 47
bbb. Cut Path 46
ccc. Cut Path 55
ddd. Cut Path 52
eee. Cut Path 48
fff. Cut Path 49
ggg. Cut Path 50
hhh. Cut Path 51
iii. Cut Path 56

SOLIDWORKS Costing Report



Model Name:	Frame Plate 2
Date and time of report:	11/30/2017 5:35:50 PM
Manufacturing Method:	Machining
Material:	Plain Carbon Steel
Stock weight:	3.00 kg
Stock Type	Plate
Plate Thickness:	6.50 mm
Material cost/weight:	0.50 USD/kg
Shop Rate:	N/A

Quantity to Produce

Total number of parts:	1
Lot size:	1

Estimated cost per part: 97.04 USD

Costing template used:	machiningtemplate_default(metric).sldctm
Costing mode used:	Manufacturing Process Recognition
Comparison:	100% Current 97.04 USD

Cost Breakdown

Material:	1.50 USD	2%
Manufacturing:	95.54 USD	98%
Markup:	0.00 USD	0%
Mold:	0.00 USD	0%

Estimated time per part: 02:29:04

Setups:	02:10:00
Operations:	00:19:04

Cost Report

1

SOLIDWORKS

Model Name:	Frame Plate 2	Material:	Plain Carbon Steel	Material cost:	1.50 USD	Total cost /part:	97.04 USD
				Manufacturing cost:	95.54 USD	Total time /part:	02:29:04
				Markup:	0.00 USD		

Manufacturing Cost Breakdown

Operation Setups	Time (hh:mm:ss)	Cost (USD / Part)
Setup Operation 1	01:00:00	30.00
Setup Operation 2	01:00:00	45.00
Total	**02:00:00**	**75.00**

Load and Unload Setups	Time (hh:mm:ss)	Cost (USD / Part)
Setup Operation 1	00:05:00	2.50
Setup Operation 2	00:05:00	3.75
Total	**00:10:00**	**6.25**

Cut Operations	Cutting Method	Quantity	Cut Length (mm)	Time (hh:mm:ss)	Cost (USD / Part)
Cut Path 1	Water Jet	1	818.28	00:07:46	5.83
Cut Path 2	Water Jet	1	9.43	00:00:05	0.07
Cut Path 3	Water Jet	1	9.43	00:00:05	0.07
Cut Path 4	Water Jet	1	9.43	00:00:05	0.07
Cut Path 5	Water Jet	1	9.43	00:00:05	0.07
Cut Path 6	Water Jet	1	9.43	00:00:05	0.07
Cut Path 7	Water Jet	1	9.43	00:00:05	0.07
Cut Path 8	Water Jet	1	9.43	00:00:05	0.07
Cut Path 9	Water Jet	1	9.43	00:00:05	0.07
Cut Path 10	Water Jet	1	9.43	00:00:05	0.07
Cut Path 11	Water Jet	1	9.43	00:00:05	0.07
Cut Path 12	Water Jet	1	9.43	00:00:05	0.07
Cut Path 13	Water Jet	1	9.43	00:00:05	0.07
Cut Path 14	Water Jet	1	9.43	00:00:05	0.07
Cut Path 15	Water Jet	1	9.43	00:00:05	0.07

2

SOLIDWORKS

Cut Path 16	Water Jet	1	9.43	00:00:05	0.07
Cut Path 17	Water Jet	1	9.43	00:00:05	0.07
Cut Path 18	Water Jet	1	163.36	00:01:33	1.16
Cut Path 19	Water Jet	1	9.43	00:00:05	0.07
Cut Path 20	Water Jet	1	9.43	00:00:05	0.07
Cut Path 21	Water Jet	1	9.43	00:00:05	0.07
Cut Path 22	Water Jet	1	9.43	00:00:05	0.07
Cut Path 23	Water Jet	1	9.43	00:00:05	0.07
Cut Path 24	Water Jet	1	9.43	00:00:05	0.07
Cut Path 25	Water Jet	1	9.43	00:00:05	0.07
Cut Path 26	Water Jet	1	9.43	00:00:05	0.07
Cut Path 27	Water Jet	1	9.43	00:00:05	0.07
Cut Path 28	Water Jet	1	9.43	00:00:05	0.07
Cut Path 29	Water Jet	1	9.43	00:00:05	0.07
Cut Path 30	Water Jet	1	9.43	00:00:05	0.07
Cut Path 31	Water Jet	1	9.43	00:00:05	0.07
Cut Path 32	Water Jet	1	9.43	00:00:05	0.07
Cut Path 33	Water Jet	1	9.43	00:00:05	0.07
Cut Path 34	Water Jet	1	9.43	00:00:05	0.07
Cut Path 35	Water Jet	1	163.36	00:01:33	1.16
Cut Path 36	Water Jet	1	9.43	00:00:05	0.07
Cut Path 37	Water Jet	1	9.43	00:00:05	0.07
Cut Path 38	Water Jet	1	9.43	00:00:05	0.07
Cut Path 39	Water Jet	1	9.43	00:00:05	0.07
Cut Path 40	Water Jet	1	9.43	00:00:05	0.07
Cut Path 41	Water Jet	1	9.43	00:00:05	0.07
Cut Path 42	Water Jet	1	9.43	00:00:05	0.07
Cut Path 43	Water Jet	1	9.43	00:00:05	0.07
Cut Path 44	Water Jet	1	9.43	00:00:05	0.07

3

SOLIDWORKS

Cut Path 45	Water Jet	1	9.43	00:00:05	0.07
Cut Path 46	Water Jet	1	9.43	00:00:05	0.07
Cut Path 47	Water Jet	1	9.43	00:00:05	0.07
Cut Path 48	Water Jet	1	9.43	00:00:05	0.07
Cut Path 49	Water Jet	1	9.43	00:00:05	0.07
Cut Path 50	Water Jet	1	9.43	00:00:05	0.07
Cut Path 51	Water Jet	1	163.36	00:01:33	1.16
Cut Path 52	Water Jet	1	9.43	00:00:05	0.07
Cut Path 53	Water Jet	1	9.43	00:00:05	0.07
Cut Path 54	Water Jet	1	9.43	00:00:05	0.07
Cut Path 55	Water Jet	1	9.43	00:00:05	0.07
Cut Path 56	Water Jet	1	9.43	00:00:05	0.07
Cut Path 57	Water Jet	1	9.43	00:00:05	0.07
Cut Path 58	Water Jet	1	9.43	00:00:05	0.07
Cut Path 59	Water Jet	1	163.36	00:01:33	1.16
Cut Path 60	Water Jet	1	9.43	00:00:05	0.07
Cut Path 61	Water Jet	1	9.43	00:00:05	0.07
Total		**61**	**1999.57**	**00:18:59**	**14.25**

Operation	Surface Finish	Volume Removed (mm^3)	Time (hh:mm:ss)	Cost (USD / Part)	Tooling	Cost-per-Volume (USD/mm^3)
Left Face	Roughing	88652.06	00:00:05	0.04	Face Mill	N/A
Total		**88652.06**	**00:00:05**	**0.04**		

No Cost Features
Fillet 1
Fillet 2

Setup Operations

1. Setup Operation 1
   a. Left Face
2. Setup Operation 2

4

SOLIDWORKS

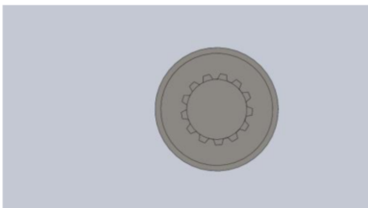
a. Cut Path 44
b. Cut Path 46
c. Cut Path 12
d. Cut Path 13
e. Cut Path 14
f. Cut Path 15
g. Cut Path 16
h. Cut Path 17
i. Cut Path 18
j. Cut Path 19
k. Cut Path 20
l. Cut Path 21
m. Cut Path 22
n. Cut Path 23
o. Cut Path 24
p. Cut Path 25
q. Cut Path 26
r. Cut Path 27
s. Cut Path 28
t. Cut Path 29
u. Cut Path 30
v. Cut Path 31
w. Cut Path 32
x. Cut Path 33
y. Cut Path 34
z. Cut Path 35
aa. Cut Path 36
bb. Cut Path 37
cc. Cut Path 38
dd. Cut Path 39
ee. Cut Path 40
ff. Cut Path 41
gg. Cut Path 42
hh. Cut Path 47
ii. Cut Path 43
jj. Cut Path 11
kk. Cut Path 10
ll. Cut Path 9
mm. Cut Path 8
nn. Cut Path 7
oo. Cut Path 6
pp. Cut Path 5
qq. Cut Path 4
rr. Cut Path 2

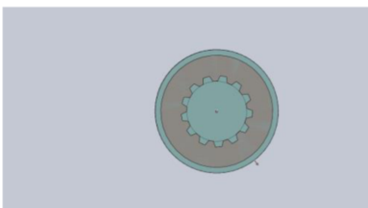
5

ss. Cut Path 3
tt. Cut Path 45
uu. Cut Path 1
vv. Cut Path 57
ww. Cut Path 56
xx. Cut Path 50
yy. Cut Path 61
zz. Cut Path 55
aaa. Cut Path 51
bbb. Cut Path 54
ccc. Cut Path 48
ddd. Cut Path 59
eee. Cut Path 52
fff. Cut Path 49
ggg. Cut Path 53
hhh. Cut Path 58
iii. Cut Path 60

6

SOLIDWORKS Costing Report





Model Name:	Shaft number 1
Date and time of report:	11/30/2017 5:15:17 PM
Manufacturing Method:	Machining
Material:	Plain Carbon Steel
Stock weight:	1.01 kg
Stock Type	Cylinder
Cylinder Size:	44.00x85.00 mm
Material cost/weight:	0.25 USD/kg
Shop Rate:	N/A

Quantity to Produce

Total number of parts:	1
Lot size:	1

Estimated cost per part: 124.77 USD

Costing template used:	machiningtemplate_default(metric).sldctm
Costing mode used:	Manufacturing Process Recognition
Comparison:	

Cost Breakdown

Material:	0.25 USD	0%
Manufacturing:	124.52 USD	100%
Markup:	0.00 USD	0%
Mold:	0.00 USD	0%

Estimated time per part: 03:23:25

Setups:	03:15:00
Operations:	00:08:25

Model Name:	Shaft number 1	Material:	Plain Carbon Steel	Material cost: Manufacturing cost: Markup:	0.25 USD 124.52 USD 0.00 USD	Total cost /part: Total time /part:	124.77 USD 03:23:25

Manufacturing Cost Breakdown

Operation Setups	Time (hh:mm:ss)	Cost (USD / Part)
Setup Operation 1	01:00:00	30.00
Setup Operation 2	01:00:00	40.00
Setup Operation 3	01:00:00	40.00
Total	**03:00:00**	**110.00**

Load and Unload Setups	Time (hh:mm:ss)	Cost (USD / Part)
Setup Operation 1	00:05:00	2.50
Setup Operation 2	00:05:00	3.33
Setup Operation 3	00:05:00	3.33
Total	**00:15:00**	**9.17**

Operation	Surface Finish	Volume Removed (mm^3)	Time (hh:mm:ss)	Cost (USD / Part)	Tooling	Cost-per-Volume (USD/mm^3)
Pocket 1	Roughing	6589.27	00:00:47	0.40	Flat End Mill	N/A
Total		**6589.27**	**00:00:47**	**0.40**		

Turn Operation	Surface Finish	Volume Removed (mm^3)	Time (hh:mm:ss)	Cost (USD / Part)	Tooling	Cost-per-Volume (USD/mm^3)
OD Turn 1	Roughing	42131.35	00:05:21	3.57	OD Turning	N/A
OD Turn 2	Roughing	11878.34	00:01:30	1.01	OD Turning	N/A
Total		**54009.69**	**00:06:51**	**4.57**		

Hole Operation	Surface Finish	Volume Removed (mm^3)	Time (hh:mm:ss)	Cost (USD / Part)	Tooling	Cost-per-Volume (USD/mm^3)
Hole 1	Drill	7261.01	00:00:46	0.39	HSS Drill	N/A

2

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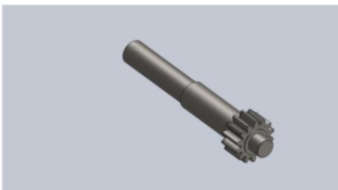
SOLIDWORKS Costing Report



Assembly Name: Shaft 2_with gear

Date and time of report:	12/5/2017 5:49:10 PM
Total weight:	2.19 kg
Total stock weight:	5.70 kg

Quantity to Produce

Total number of assemblies:	1
Lot size:	1

Estimated cost per assembly: 90.31 USD

Costing main template:	multibodytemplate_default(metric).sldctc
Comparison:	0% Current 90.31 USD Previous 90.31 USD

Cost Breakdown

Calculated Parts:	56.31 USD	62%
Purchased Parts:	0.00 USD	0%
Toolbox Parts:	0.00 USD	0%
Operations:	34.00 USD	38%
Markup:	0.00 USD	0%

1

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Component Cost Impact

Top Ten Components Contributing Most to Assembly Cost

Component	Configuration	Material Cost (USD/Assembly)	Manufacturing Cost (USD/Assembly)	Total Cost (USD/Assembly)
Shaft 2_with gear	Default	11.78	24.79	36.57
Shaft number 2	Default	4.66	6.53	11.19
SDP_a_1c_1myk30014	Default	1.29	7.26	8.55
Total		17.73	38.58	56.31

2

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Cost Breakdown for Each Part

Calculated Parts	Method	Quantity	Part Cost (USD/Assembly)	Total Cost (USD / Assembly)	Costing Template
SDP_a_1c_1myk30014 [Default]	Machining	1	8.55	8.55	machiningtemplate_default(metric).sldctm
Shaft 2_with gear [Default]	Machining	1	36.57	36.57	machiningtemplate_default(metric).sldctm
Shaft number 2 [Default]	Machining	1	11.19	11.19	machiningtemplate_default(metric).sldctm
Total			56.31	56.31	

Cost Breakdown at Assembly Level

Setups	Cost (USD / Assembly)
Painting 1	34.00
Total	34.00

Custom Operations Part	Quantity	Cost (USD / Assembly)
Painting 1	1	0.00
Total	1	0.00

No Cost Parts

4

SOLIDWORKS

SOLIDWORKS Costing Report

Assembly Name:	Shaft 3_with gear
Date and time of report:	12/5/2017 5:53:38 PM
Total weight:	1.77 kg
Total stock weight:	5.35 kg

Quantity to Produce

Total number of assemblies:	1
Lot size:	1

Estimated cost per assembly: 89.72 USD

Costing main template:	multibodytemplate_default(metric).sldctc
Comparison:	100% Current 89.72 USD

Cost Breakdown

Calculated Parts:	55.72 USD	62%
Purchased Parts:	0.00 USD	0%
Toolbox Parts:	0.00 USD	0%
Operations:	34.00 USD	38%
Markup:	0.00 USD	0%

SOLIDWORKS

Component Cost Impact

Top Ten Components Contributing Most to Assembly Cost

Component	Configuration	Material Cost (USD/Assembly)	Manufacturing Cost (USD/Assembly)	Total Cost (USD/Assembly)
Shaft 3_with gear	Default	11.45	25.53	36.98
Shaft number 3	Default	3.52	5.95	9.48
SDP_a_1c_1myk30016	Default	1.66	7.60	9.26
Total		16.64	39.08	55.72

2

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Cost Breakdown for Each Part

Calculated Parts	Method	Quantity	Part Cost (USD/Assembly)	Total Cost (USD / Assembly)	Costing Template
Shaft 3_with gear [Default]	Machining	1	36.98	36.98	machiningtemplate_default(metric).sldctm
SDP_a_1c_1myk30016 [Default]	Machining	1	9.26	9.26	machiningtemplate_default(metric).sldctm
Shaft number 3 [Default]	Machining	1	9.48	9.48	machiningtemplate_default(metric).sldctm
Total			55.72	55.72	

3

Cost Breakdown at Assembly Level

Setups	Cost (USD / Assembly)
Painting <1>	34.00
Total	34.00

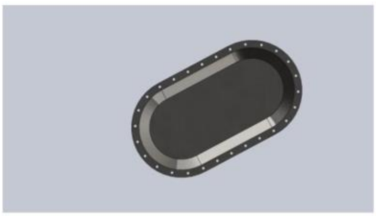
Custom Operations Part	Quantity	Cost (USD / Assembly)
Painting <1>	1	0.00
Total	1	0.00

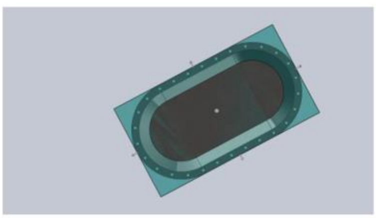
No Cost Parts

4

SOLIDWORKS

SOLIDWORKS Costing Report





Model Name:	52mm Bearing Cap
Date and time of report:	11/30/2017 5:09:44 PM
Manufacturing Method:	Machining
Material:	Plain Carbon Steel
Stock weight:	1.49 kg
Stock Type	Block
Block Size:	94.00x12.00x169.00 mm
Material cost/weight:	0.50 USD/kg
Shop Rate:	N/A

Quantity to Produce

Total number of parts:	3
Lot size:	3

Estimated cost per part: 31.28 USD

Costing template used:	machiningtemplate_default(metric).sldctm
Costing mode used:	Manufacturing Process Recognition
Comparison:	0% Current 31.28 USD Previous 31.28 USD

Cost Breakdown

Material:	0.74 USD	2%
Manufacturing:	30.53 USD	98%
Markup:	0.00 USD	0%
Mold:	0.00 USD	0%

Estimated time per part: 01:01:03

Setups:	00:50:00
Operations:	00:11:03

Cost Report

1

SOLIDWORKS

Model Name:	52mm Bearing Cap	Material:	Plain Carbon Steel	Material cost:	0.74 USD	Total cost /part:	31.28 USD
				Manufacturing cost:	30.53 USD	Total time /part:	01:01:03
				Markup:	0.00 USD		

Manufacturing Cost Breakdown

Operation Setups	Time (hh:mm:ss)	Cost (USD / Part)
Setup Operation 1	00:20:00	10.00
Setup Operation 2	00:20:00	10.00
Total	**00:40:00**	**20.00**

Load and Unload Setups	Time (hh:mm:ss)	Cost (USD / Part)
Setup Operation 1	00:05:00	2.50
Setup Operation 2	00:05:00	2.50
Total	**00:10:00**	**5.00**

Operation	Surface Finish	Volume Removed (mm^3)	Time (hh:mm:ss)	Cost (USD / Part)	Tooling	Cost-per-Volume (USD/mm^3)
Circular Pocket 1	Roughing	16989.73	00:01:51	0.93	Flat End Mill	N/A
Circular Pocket 2	Roughing	16989.73	00:01:51	0.93	Flat End Mill	N/A
Volume 1	Roughing	964.11	00:00:06	0.05	Flat End Mill	N/A
Volume 2	Roughing	56815.26	00:06:11	3.10	Flat End Mill	N/A
Total		**91758.83**	**00:10:00**	**5.00**		

Hole Operation	Surface Finish	Volume Removed (mm^3)	Time (hh:mm:ss)	Cost (USD / Part)	Tooling	Cost-per-Volume (USD/mm^3)
Hole 3	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 4	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 5	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 6	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 7	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 8	Drill	35.34	00:00:02	0.02	HSS Drill	N/A

2

SOLIDWORKS

Hole 9	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 10	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 11	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 12	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 13	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 14	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole Pattern 1	Drill	424.12	00:00:27	0.23	HSS Drill	N/A
Hole Pattern 2	Drill	141.37	00:00:09	0.08	HSS Drill	N/A
Total		**989.60**	**00:01:03**	**0.53**		

Setup Operations

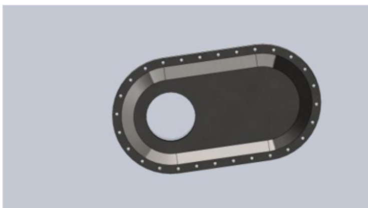
1. Setup Operation 1
   a. Hole 4
   b. Circular Pocket 2
   c. Circular Pocket 1
   d. Hole 6
   e. Hole 8
   f. Hole 10
   g. Hole 11
   h. Hole 9
   i. Hole 7
   j. Hole 5
   k. Hole 3
   l. Hole Pattern 1 - 4
   m. Hole Pattern 1 - 5
   n. Hole 14
   o. Hole 13
   p. Hole 12
   q. Hole Pattern 1 - 6
   r. Hole Pattern 1 - 8
   s. Hole Pattern 1 - 10
   t. Hole Pattern 1 - 12
   u. Hole Pattern 2 - 3
   v. Hole Pattern 2 - 4
   w. Hole Pattern 1 - 1
   x. Hole Pattern 1 - 11
   y. Hole Pattern 1 - 9
   z. Hole Pattern 1 - 7
   aa. Hole Pattern 1 - 3
   bb. Hole Pattern 1 - 2
   cc. Hole Pattern 2 - 1

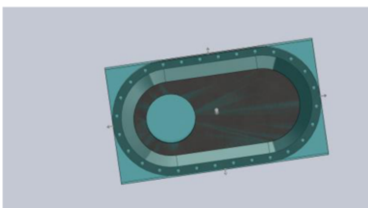
3

dd. Hole Pattern 2 - 2
2. Setup Operation 2
    a. Volume 2
    b. Volume 1

4

SOLIDWORKS Costing Report





Model Name: 68mm Bearing Cap

Date and time of report:	11/30/2017 5:08:32 PM

Manufacturing Method: Machining

Material:	Plain Carbon Steel
Stock weight:	1.49 kg
Stock Type	Block
Block Size:	94.00x12.00x169.00 mm
Material cost/weight:	0.50 USD/kg
Shop Rate:	N/A

Quantity to Produce

Total number of parts:	1
Lot size:	1

Estimated cost per part: 162.02 USD

Costing template used:	machiningtemplate_default(metric).sldctm
Costing mode used:	Manufacturing Process Recognition
Comparison:	100% Current 162.02 USD

Cost Breakdown

Material:	0.74 USD	0%
Manufacturing:	161.27 USD	100%
Markup:	0.00 USD	0%
Mold:	0.00 USD	0%

Estimated time per part: 05:22:32

Setups:	02:10:00
Operations:	03:12:32

Model Name:	68mm Bearing Cap	Material:	Plain Carbon Steel	Material cost:	0.74 USD	Total cost /part:	162.02 USD
				Manufacturing cost:	161.27 USD	Total time /part:	05:22:32
				Markup:	0.00 USD		

Manufacturing Cost Breakdown

Operation Setups	Time (hh:mm:ss)	Cost (USD / Part)
Setup Operation 1	01:00:00	30.00
Setup Operation 2	01:00:00	30.00
Total	**02:00:00**	**60.00**

Load and Unload Setups	Time (hh:mm:ss)	Cost (USD / Part)
Setup Operation 1	00:05:00	2.50
Setup Operation 2	00:05:00	2.50
Total	**00:10:00**	**5.00**

Operation	Surface Finish	Volume Removed (mm^3)	Time (hh:mm:ss)	Cost (USD / Part)	Tooling	Cost-per-Volume (USD/mm^3)
Circular Pocket 1	Roughing	16989.73	00:01:51	0.93	Flat End Mill	N/A
Volume 1	Roughing	964.11	00:00:06	0.05	Flat End Mill	N/A
Volume 2	Roughing	56815.26	00:06:11	3.10	Flat End Mill	N/A
Total		**74769.10**	**00:08:09**	**4.08**		

Hole Operation	Surface Finish	Volume Removed (mm^3)	Time (hh:mm:ss)	Cost (USD / Part)	Tooling	Cost-per-Volume (USD/mm^3)
Hole 2	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 3	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 4	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 5	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 6	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 7	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 8	Drill	35.34	00:00:02	0.02	HSS Drill	N/A

2

SOLIDWORKS

Hole 9	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 10	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 11	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 12	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 13	Drill	35.34	00:00:02	0.02	HSS Drill	N/A
Hole 14	Drill	0.00	03:03:19	91.67	HSS Drill	N/A
Hole Pattern 1	Drill	424.12	00:00:27	0.23	HSS Drill	N/A
Hole Pattern 2	Drill	141.37	00:00:09	0.08	HSS Drill	N/A
Total		**989.60**	**03:04:23**	**92.19**		

Setup Operations

1. Setup Operation 1
    a. Hole 9
    b. Circular Pocket 1
    c. Hole 7
    d. Hole 5
    e. Hole 8
    f. Hole 3
    g. Hole 6
    h. Hole 4
    i. Hole 2
    j. Hole Pattern 1 - 8
    k. Hole Pattern 2 - 1
    l. Hole Pattern 2 - 2
    m. Hole Pattern 2 - 3
    n. Hole Pattern 2 - 4
    o. Hole 10
    p. Hole 11
    q. Hole 13
    r. Hole 14
    s. Hole Pattern 1 - 10
    t. Hole Pattern 1 - 12
    u. Hole Pattern 1 - 2
    v. Hole Pattern 1 - 4
    w. Hole Pattern 1 - 7
    x. Hole Pattern 1 - 5
    y. Hole Pattern 1 - 3
    z. Hole Pattern 1 - 1
    aa. Hole Pattern 1 - 11
    bb. Hole Pattern 1 - 6
    cc. Hole Pattern 1 - 9

3

dd. Hole 12

2. Setup Operation 2
   a. Volume 2
   b. Volume 1

4

SOUMISSION / QUOTE

S 300-PO-F-001 (Révision 2.0)

3037 Frontenac Est
Thetford Mines, Qc
G6G 6P6
Tél: (418) 335-7220
Fax: (418) 335-2206
www.cvtech-aab.com

PIÈCES MOTEURS/PARTS

Motoneiges/Snowmobiles
Motomarines/Watercrafts
V.T.T./A.T.V.
Motocross / Dirtbike
Automobiles/Cars
Industriels/Industrials
Camions/Trucks

RÉUSINAGE MOTEURS / ENGINE REBUILDING

Vilebrequins/Crankshafts
Sleeves/Sleeves
Cylindres/Cylinders
Embrayages / clutches
Culasse/Cyl.head
Replacage cylindres/Replating cylinders

VENDU À / SOLD TO

CASH (ALBERTA)
ALBERTA
CANADA

CASHAB	No QT009355
VOTRE No COMMANDE/ YOUR ORDER #	DATE 04/12/17
NOTRE No BON DE TRAVAIL/ OUR # WORK ORDER: QT009355	CONDITIONS/TERMS: CASH
VENDEUR SALESMAN: ANDRE	TRANSPORT VIA/SHIPPED VIA: PUROLATOR

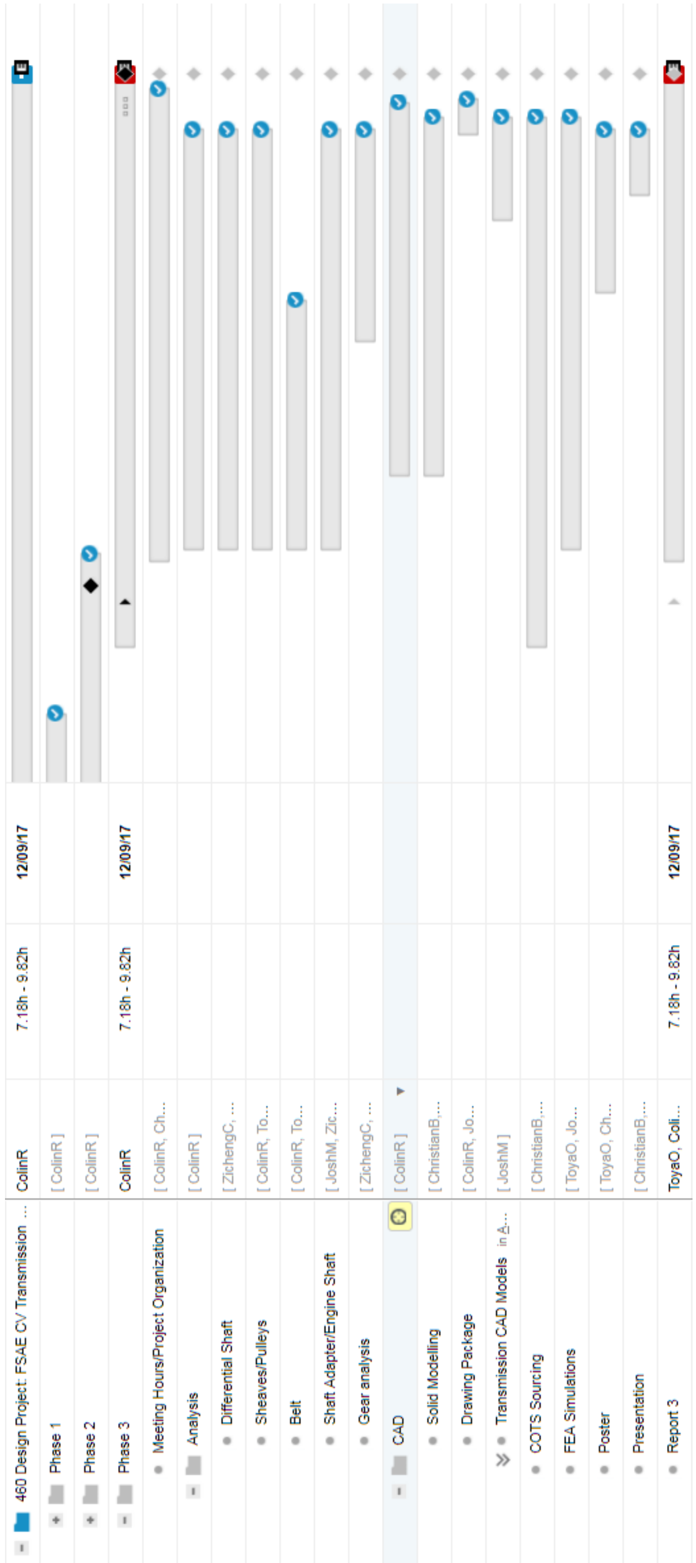
COMMANDÉE ORDERED	LIVRÉE SHIPPED	No PIÈCE PART #	DESCRIPTION	PRIX UNIT. UNIT PRICE	NET NET PRICE	EXTENSION AMOUNT
1.00	0.00	CV.1100-0046	CLUTCH PWB 80	499.99	499.99	499.99
1.00	0.00	CV.6000-0031	CLUTCH (DRIVEN) ID 1" SHAFT	419.99	419.99	419.99
3.00	0.00	CV.1130-3003	BLOC BOMBÉ NOIR PWB 80	13.99	13.99	41.97
1.00	0.00	CV.1150-3001-3	CAP (TREADED) PWB 80 (PQT 3)	5.97	5.97	5.97
2.00	0.00	CV.1135-3001-12	WEIGHT 5.5G PWB80 (QT 12)	11.99	11.99	23.98
1.00	0.00	CV.1151-1132	SPRING 800N. / 1100N. PWB 80 SERIE 0200	34.99	34.99	34.99
1.00	0.00	CV.JM52-3201-C	BELT 9.370 IN / 238MM	105.99	105.99	105.99

*** CECI N'EST PAS UNE FACTURE / THIS IS NOT AN INVOICE ***

Ces prix sont valides pour 30 jours / Those prices are valid for 30 days

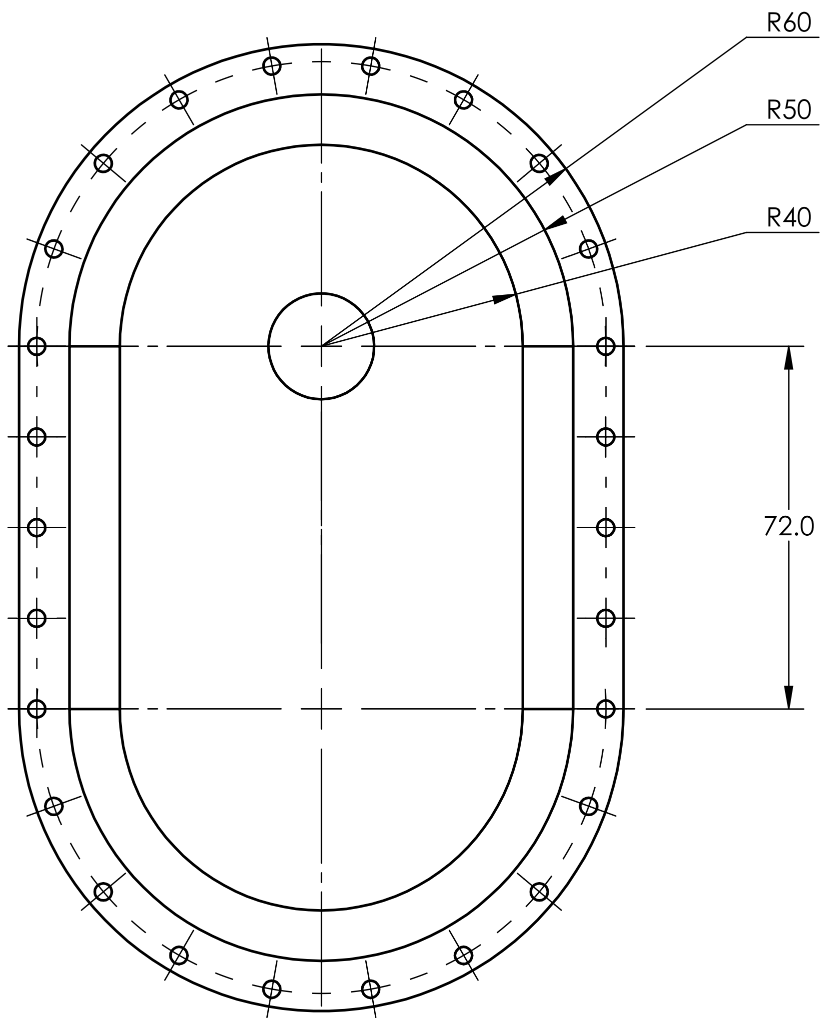
No QT009355	TAXES EN SUS / TAX NOT INCLUDED	TOTAL CAD$ 1,132.88

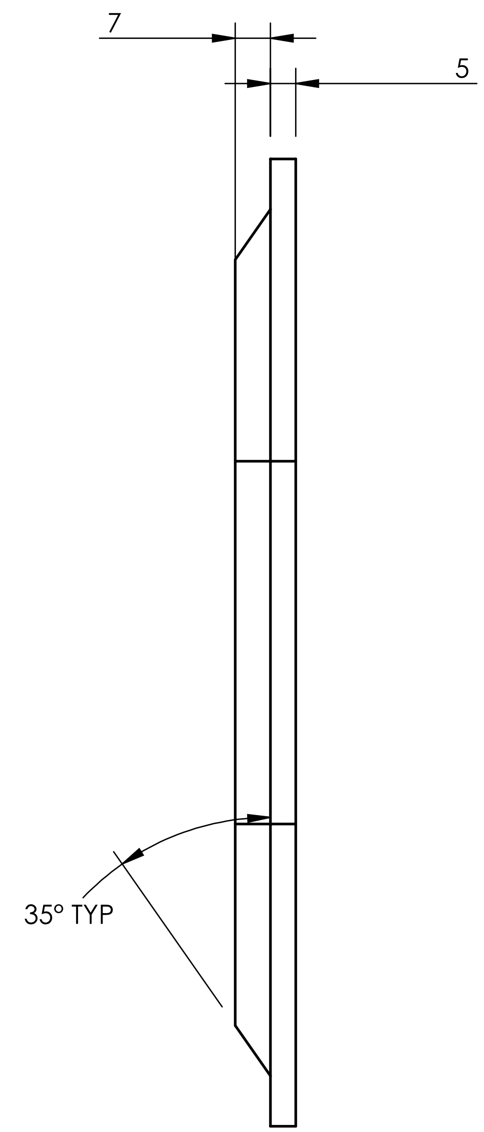
Appendix C. Project Schedule/Timesheets

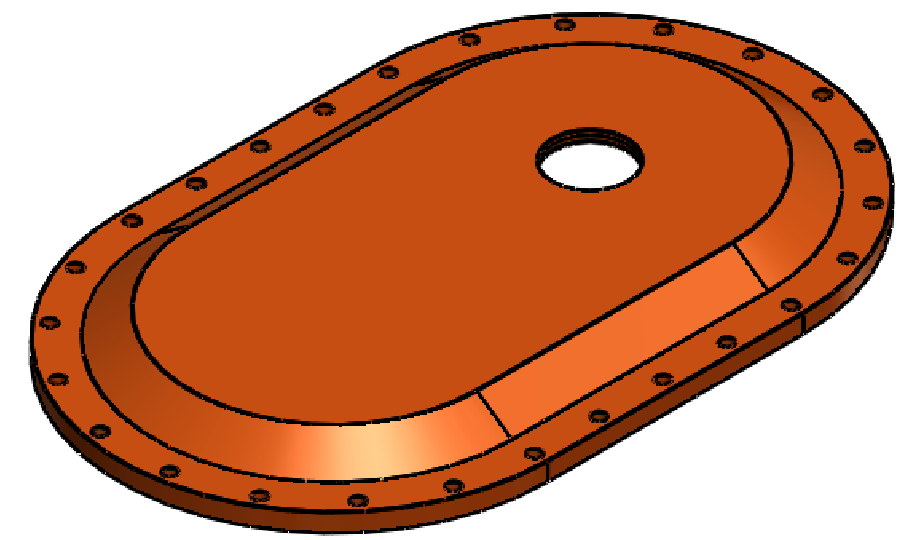


Appendix D. Drawing Package

Please see the attached drawing package at the end of this document







Mec E 460

Instructors:
Lipsett & Sadrzadeh
Fall 2017

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:
AISI 1020

FILE NAME:
52mm Bearing Cap 3-4

DRAWN BY:	**Zicheng Cao**
Group name	Mec E 460 Team 13
Group number	13
SM By	Josh Mulder
Reviewed by	Colin Reimer

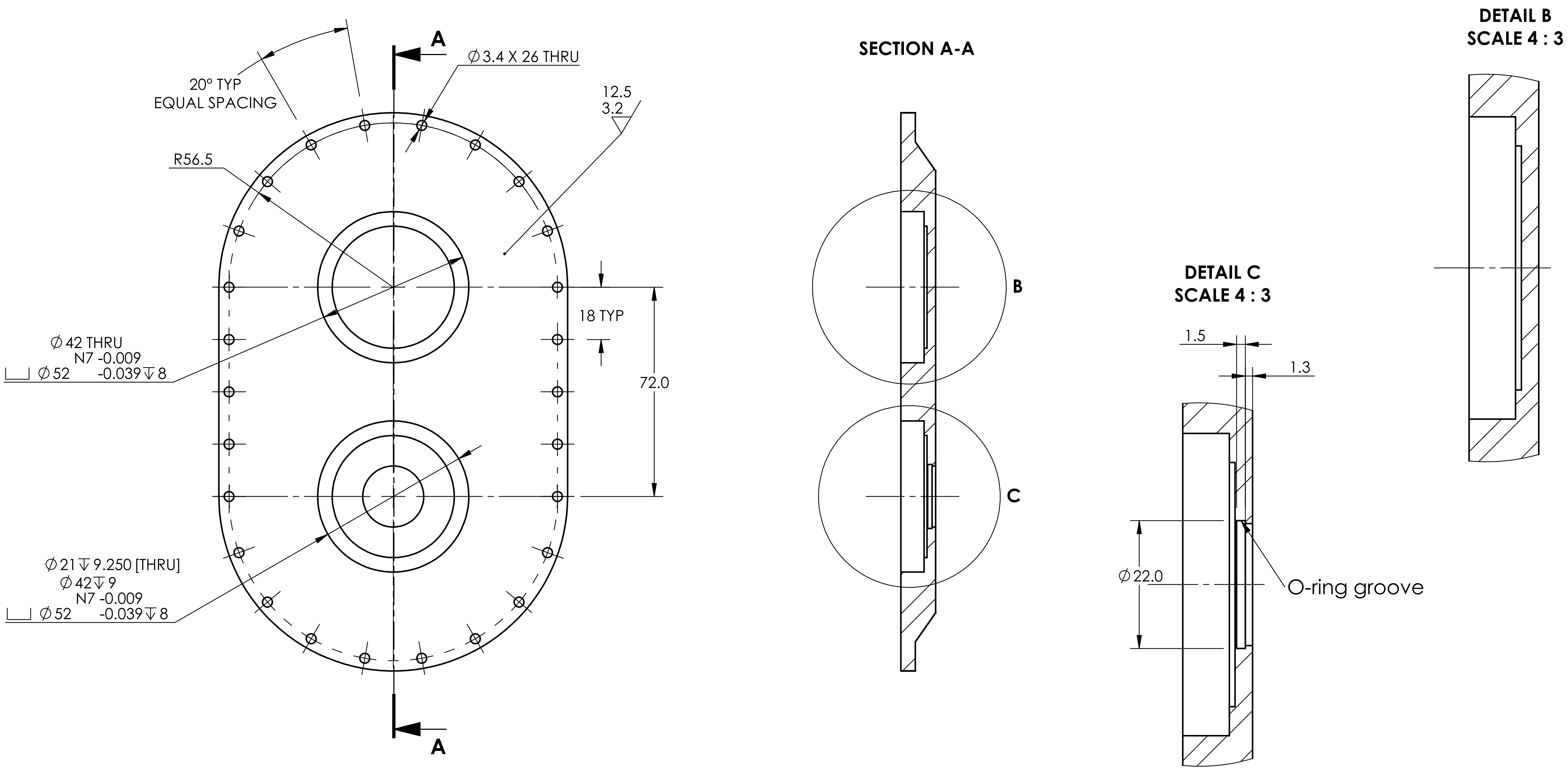
Wednesday, December 06, 2017 2:20:08 PM
Monday, November 27, 2017 8:44:34 PM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

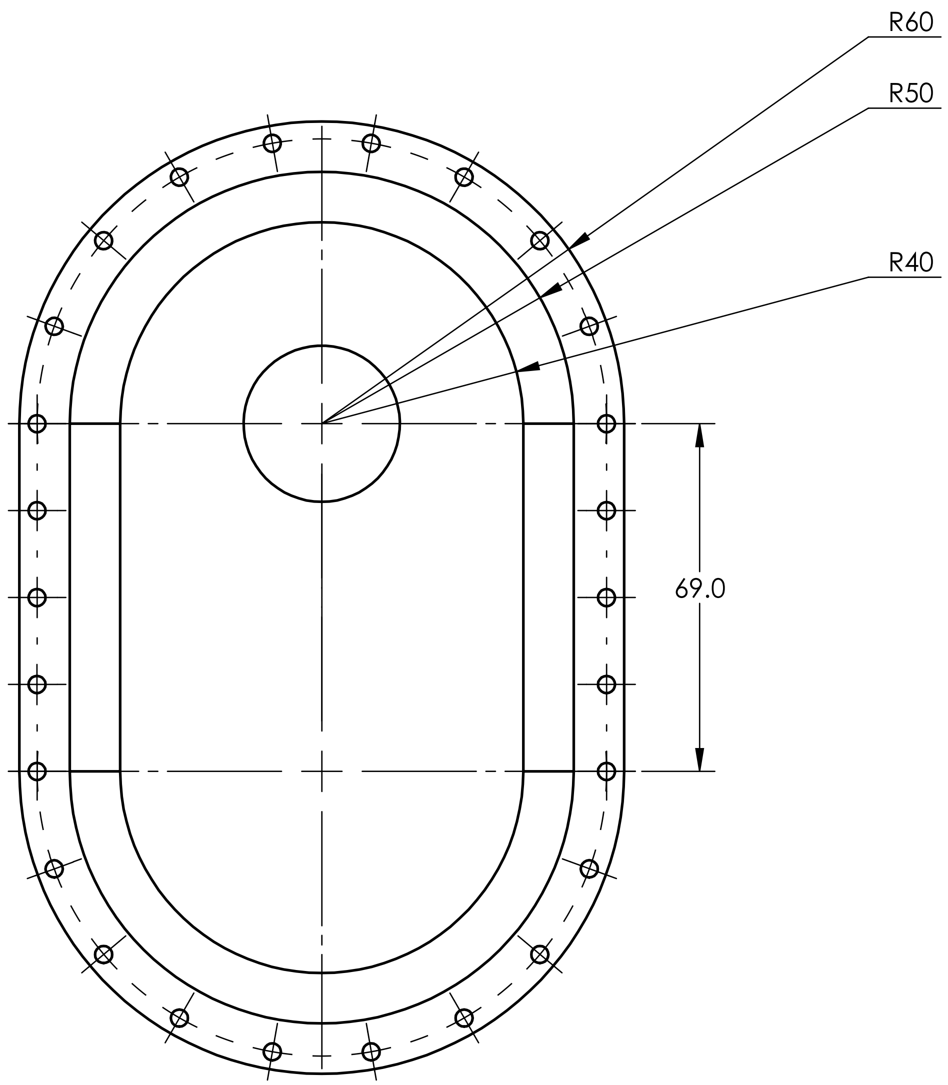
TITLE:
52mm Bearing Cap 3-4

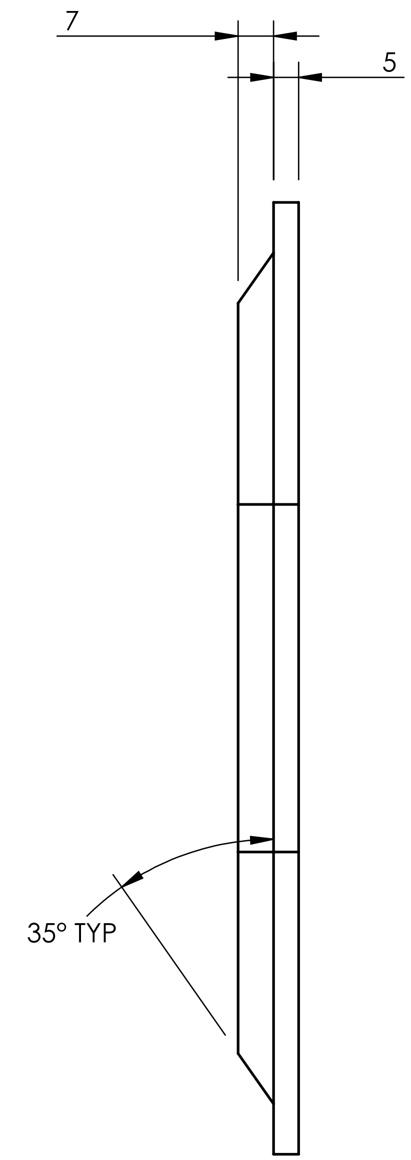
SIZE	Part supplier/manufacturer	REV
B		1

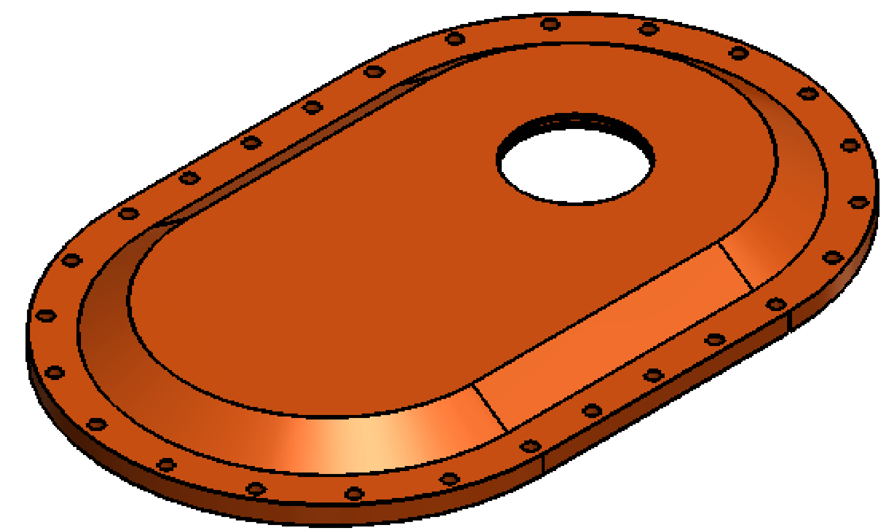
SCALE: 2:3 | Mass: 1191.35 | SHEET 1 OF 2



Mec E 460 Instructors: Lipsett & Sadrzadeh **Fall 2017** 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: **Zicheng Cao**	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA
		Group name: Mec E 460 Team 13	TITLE: 52mm Bearing Cap 3-4
		Group number: 13	
		SM By: Josh Mulder	
MATERIAL: AISI 1020		Reviewed by: Colin Reimer	SIZE **B** / Part supplier/manufacturer / REV 1
FILE NAME: 52mm Bearing Cap 3-4		Wednesday, December 06, 2017 2:20:08 PM Monday, November 27, 2017 8:44:34 PM	Scale: 2:3 / Mass: 1191.35 / SHEET 2 OF 2







Mec E 460

Instructors: Lipsett & Sadrzadeh
Fall 2017

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: AISI 1020

FILE NAME: 62mm Bearing Cap 1-2

DRAWN BY: **Zicheng Cao**

Group name	Mec E 460 Team 13
Group number	13
SM By	Josh Mulder
Reviewed by	Colin Reimer

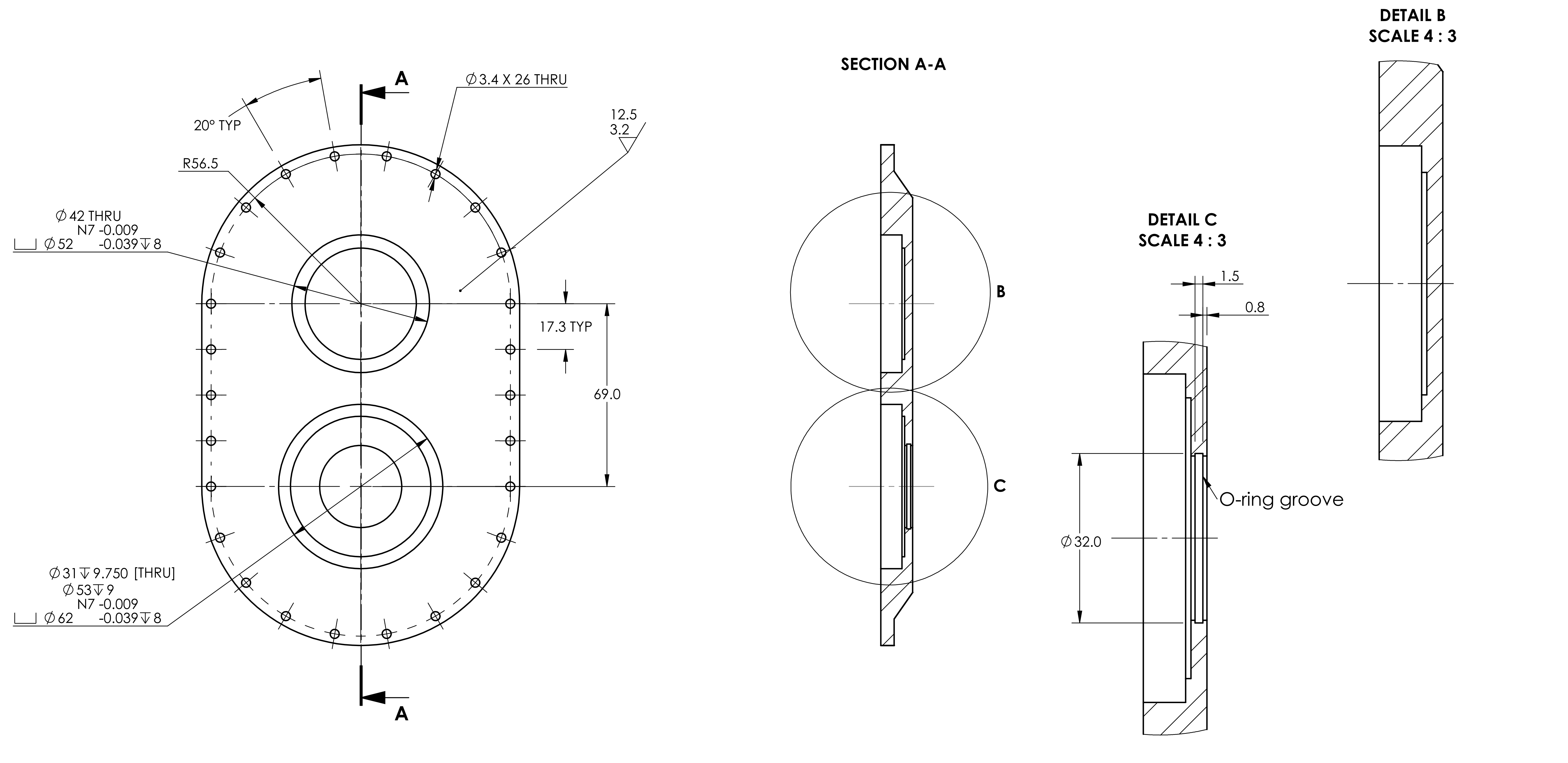
Wednesday, December 06, 2017 2:52:44 PM
Monday, November 27, 2017 8:44:34 PM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

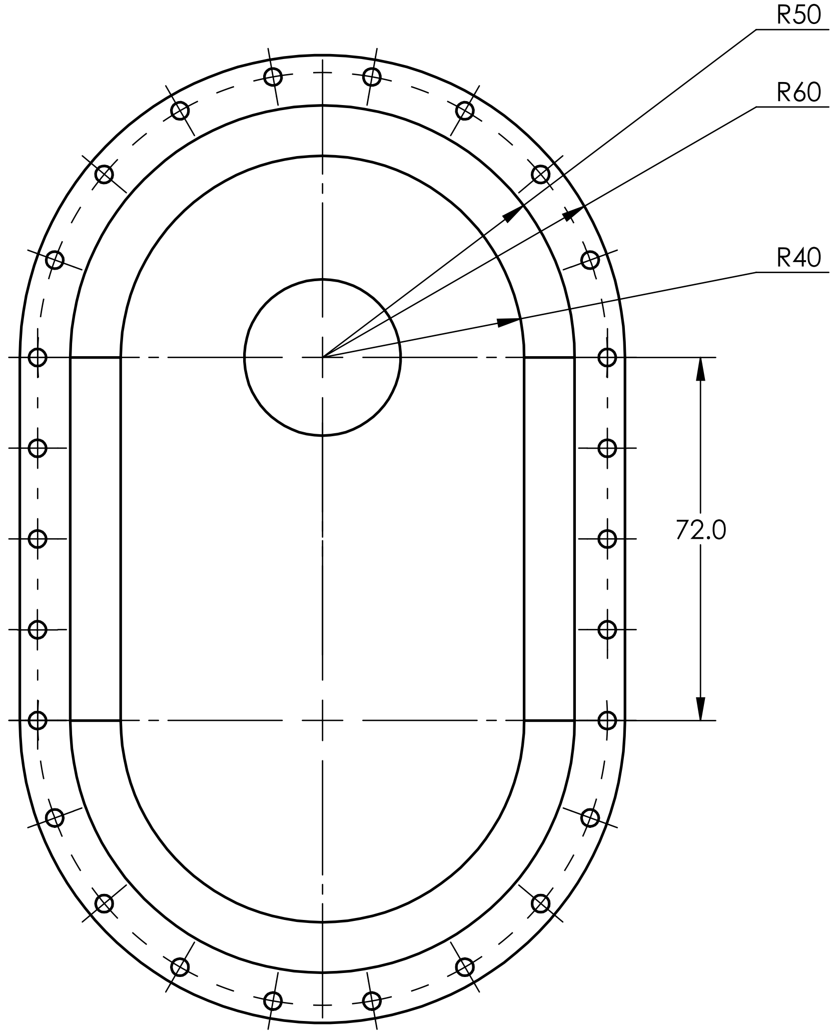
TITLE: 62mm Bearing Cap 1-2

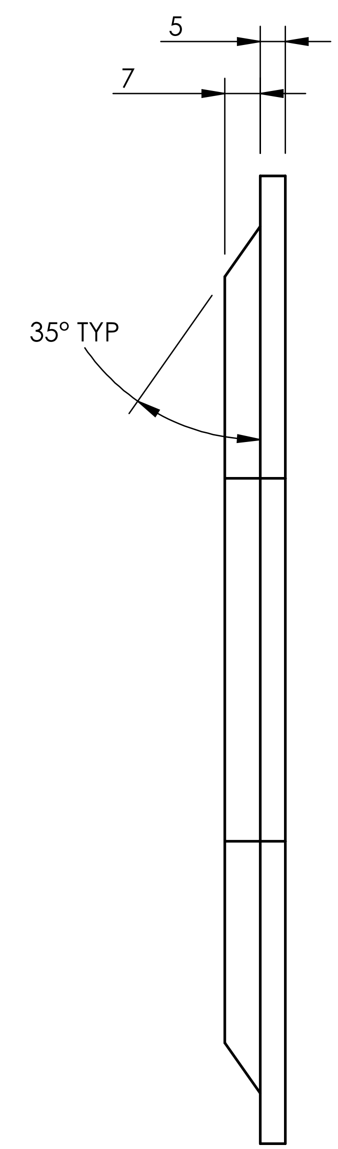
SIZE **B** | Part supplier/manufacturer | REV 1

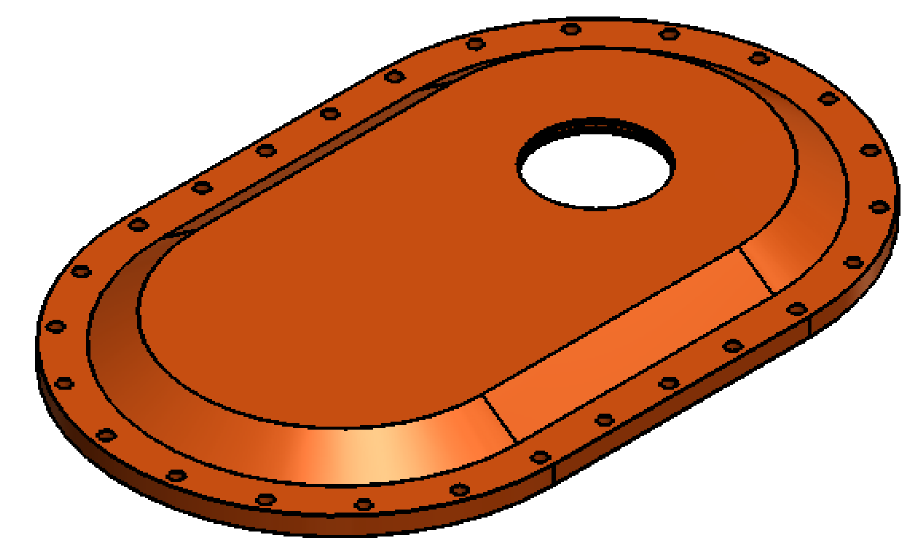
SCALE: 2:3 | Mass: 1089.26 | SHEET 1 OF 2



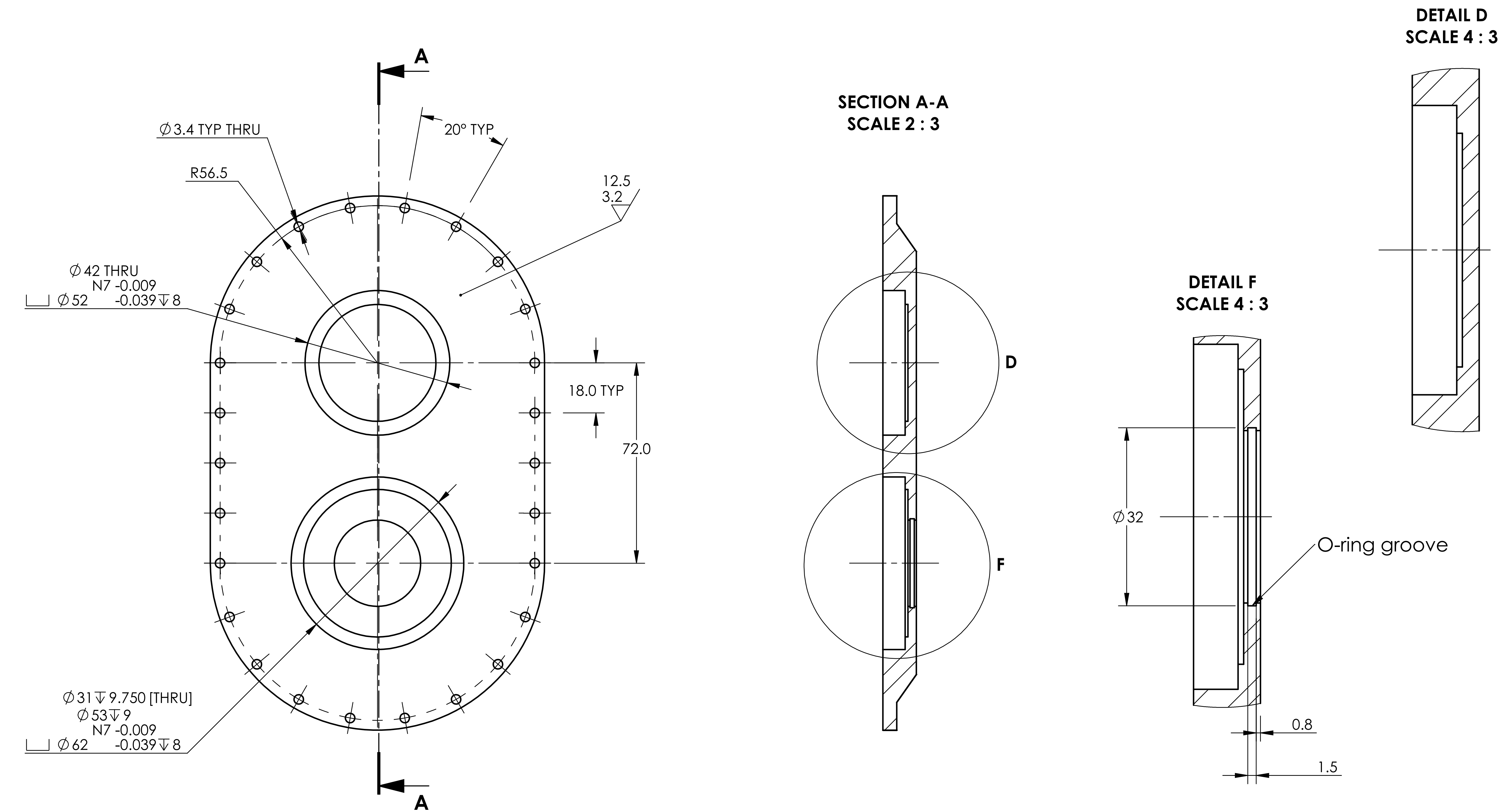
Mec E 460 Instructors: Lipsett & Sadrzadeh **Fall 2017** Comments:	UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025 SURFACE FINISH µm 0.6 DO NOT SCALE DRAWING	DRAWN BY: **Zicheng Cao** Group name: Mec E 460 Team 13 Group number: 13 SM By: Josh Mulder Reviewed by: Colin Reimer Wednesday, December 06, 2017 2:52:44 PM Monday, November 27, 2017 8:44:34 PM	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: 62mm Bearing Cap 1-2 SIZE: **B** — Part supplier/manufacturer — REV: 1 SCALE: 2:3 — Mass: 1089.26 — SHEET 2 OF 2
MATERIAL: AISI 1020 FILE NAME: 62mm Bearing Cap 1-2			



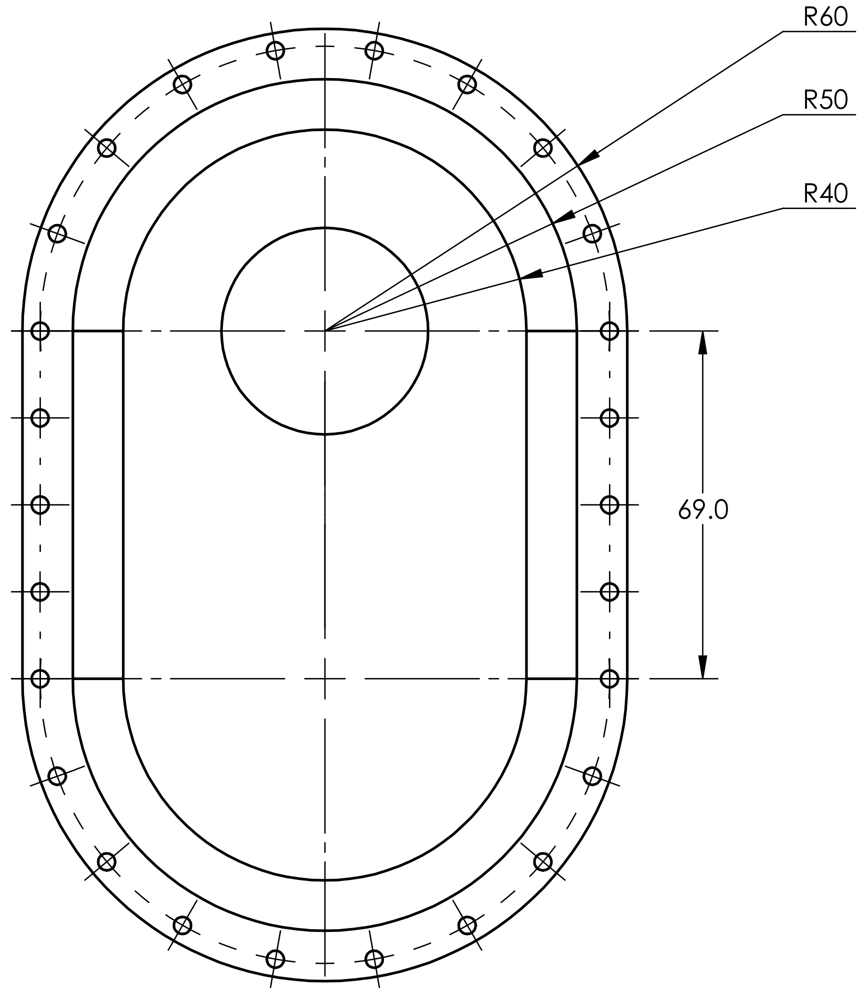


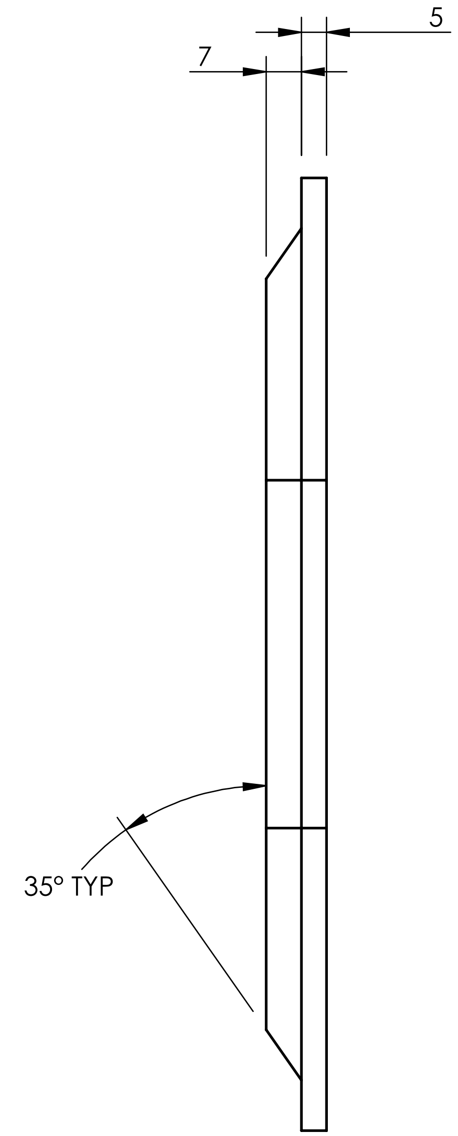


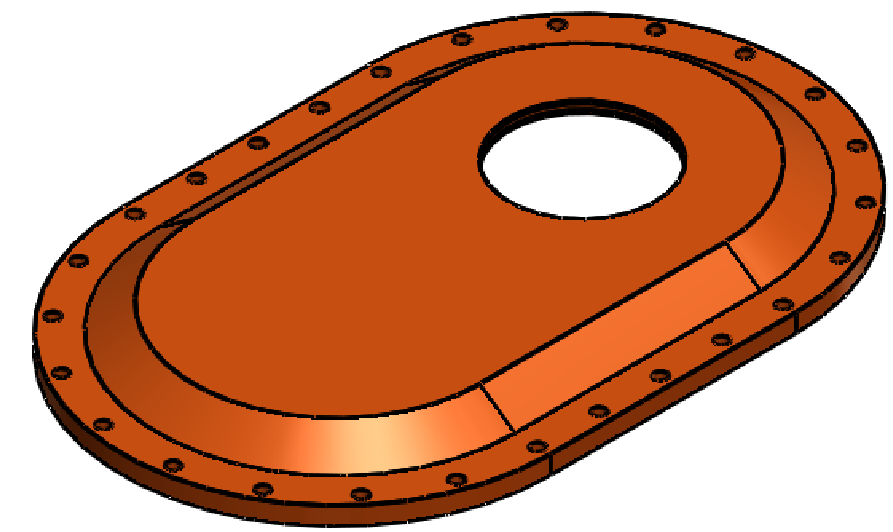
Mec E 460 Instructors: Lipsett & Sadrzadeh **Fall 2017** Comments:	UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025 SURFACE FINISH µm 0.6 DO NOT SCALE DRAWING	DRAWN BY: **Zicheng Cao** Group name: Mec E 460 Team 13 Group number: 13 SM By: Josh Mulder Reviewed by: Colin Reimer	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: 62mm Bearing Cap 3-4
MATERIAL: AISI 1020			SIZE **B** / Part supplier/manufacturer / REV 1
FILE NAME: 62mm Bearing Cap 3-4		Wednesday, December 06, 2017 3:04:13 PM Monday, November 27, 2017 8:44:34 PM	SCALE: 2:3 / Mass: 1118.41 / SHEET 1 OF 2



Mec E 460 Instructors: Lipsett & Sadrzadeh **Fall 2017** 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: **Zicheng Cao** Group name: Mec E 460 Team 13 Group number: 13 SM By: Josh Mulder Reviewed by: Colin Reimer	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: 62mm Bearing Cap 3-4
MATERIAL: AISI 1020 FILE NAME: 62mm Bearing Cap 3-4		Wednesday, December 06, 2017 3:04:13 PM Monday, November 27, 2017 8:44:34 PM	SIZE **B** | Part supplier/manufacturer | REV 1 SCALE: 2:3 | Mass: 1118.41 | SHEET 2 OF 2







Mec E 460

Instructors: Lipsett & Sadrzadeh

Fall 2017

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: **Zicheng Cao**

Group name	Mec E 460 Team 13
Group number	13
SM By	Josh Mulder
Reviewed by	Colin Reimer

Wednesday, December 06, 2017 3:29:36 PM
Monday, November 27, 2017 8:44:34 PM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:

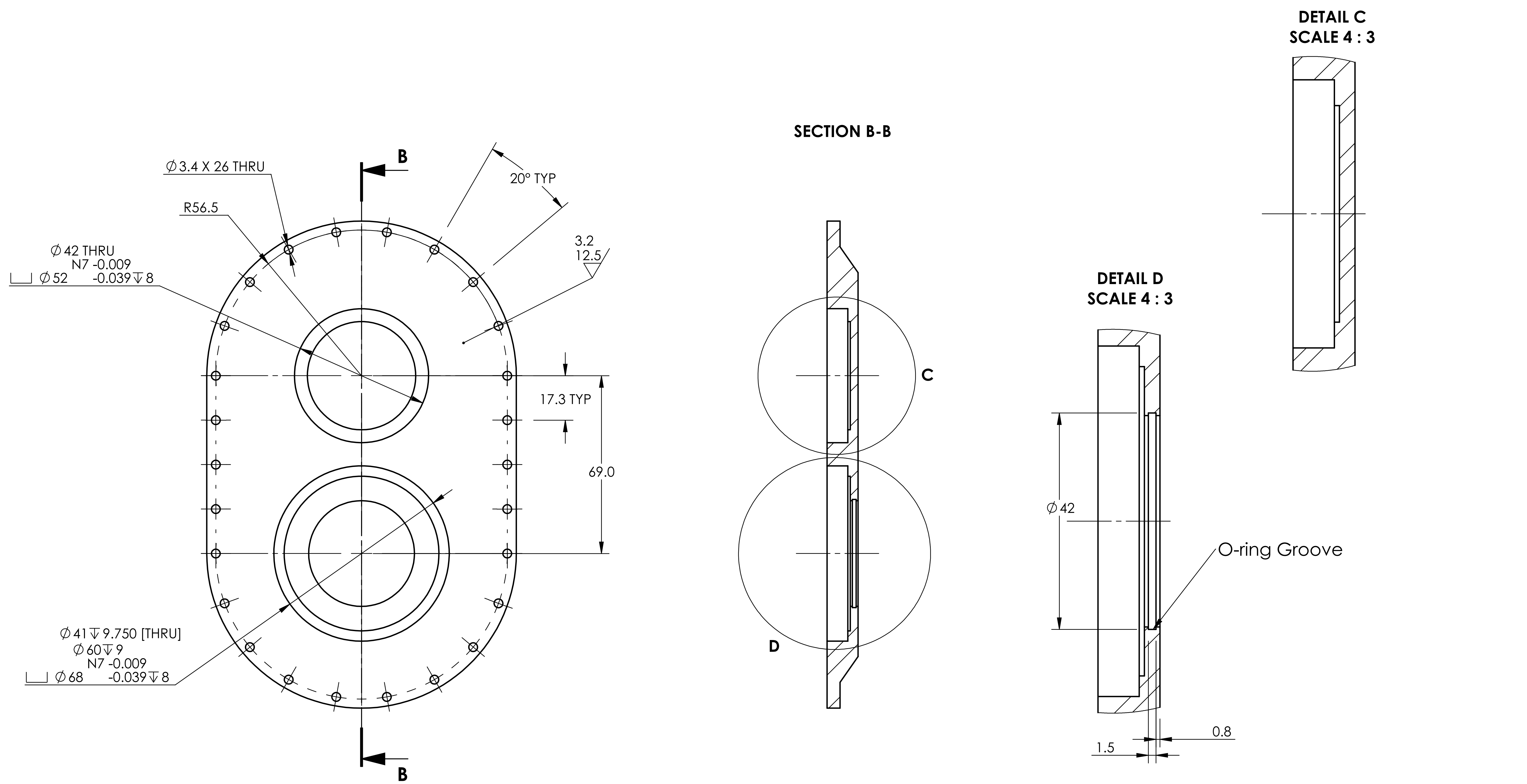
68mm Bearing Cap 1-2

MATERIAL: AISI 1020

FILE NAME: 68mm Bearing Cap 1-2

SIZE	Part supplier/manufacturer	REV
B		1

SCALE: 2:3 | Mass: 1032.05 | SHEET 1 OF 2



Mec E 460 Instructors: Lipsett & Sadrzadeh **Fall 2017** 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: **Zicheng Cao** Group name: Mec E 460 Team 13 Group number: 13 SM By: Josh Mulder Reviewed by: Colin Reimer Wednesday, December 06, 2017 3:29:36 PM Monday, November 27, 2017 8:44:34 PM	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: 68mm Bearing Cap 1-2
MATERIAL: AISI 1020 FILE NAME: 68mm Bearing Cap 1-2			SIZE **B** · Part supplier/manufacturer · REV 1 SCALE: 2:3 · Mass: 1032.05 · SHEET 2 OF 2

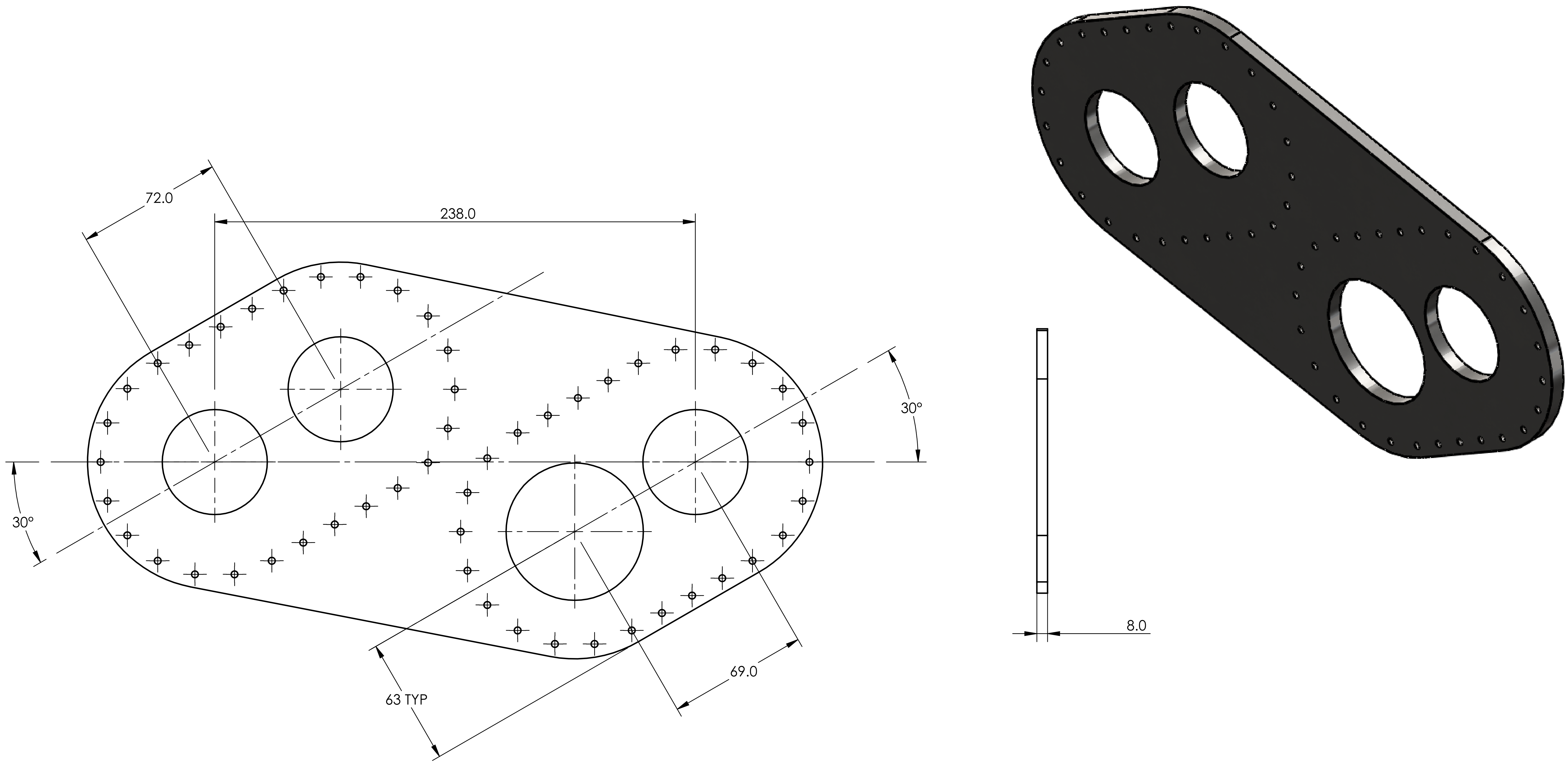


*Plate edges are constant distance from center of holes: 63 mm

Mec E 460

Instructors:
Lipsett & Mohtada Sadrzadeh
Fall 2017

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 0.6 µm

DO NOT SCALE DRAWING

MATERIAL:
AISI 1020

FILE NAME:
Frame Plate 1

DRAWN BY:
Joshua Mulder

Group name	Mec E 460 Team 13
Group number	13
SM By	Joshua Mulder
Reviewed by	Colin Reimer/Zicheng Cao

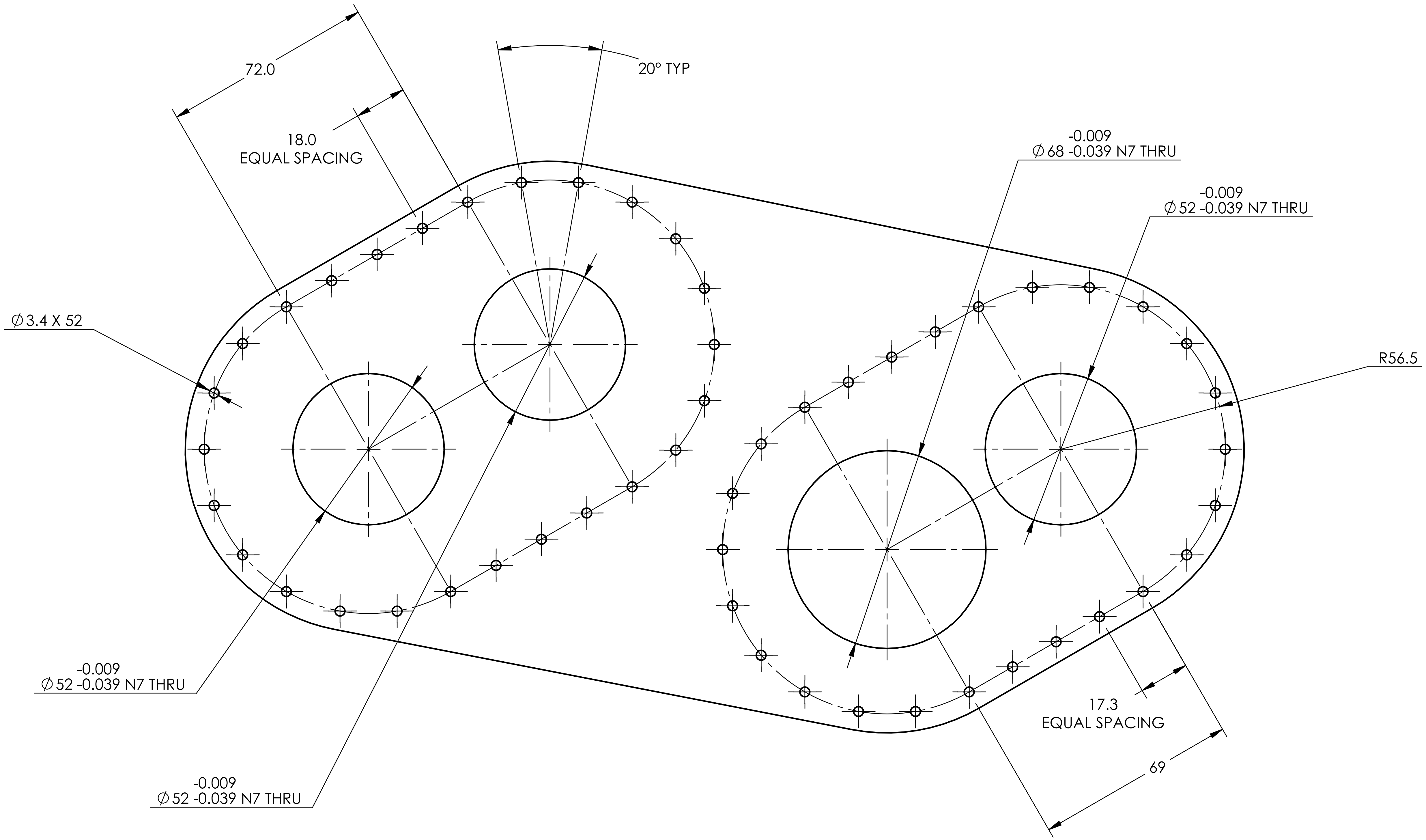
Wednesday, December 06, 2017 2:38:28 PM
Friday, November 24, 2017 3:28:25 PM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Frame Plate 1

SIZE	Part supplier/manufacturer	REV
B		1

SCALE: 1:2 | Mass: 2654.40 | SHEET 1 OF 2



Bolt hole patterns (slot pattern)contain the same number of hole each with the same slot radius. The slot center-center distance is different

Mec E 460

Instructors: Lipsett & Mohtada Sadrzadeh
Fall 2017

Comments:

MATERIAL: AISI 1020

FILE NAME: Frame Plate 1

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: **Joshua Mulder**

Group name	Mec E 460 Team 13
Group number	13
SM By	Joshua Mulder
Reviewed by	Colin Reimer/Zicheng Cao

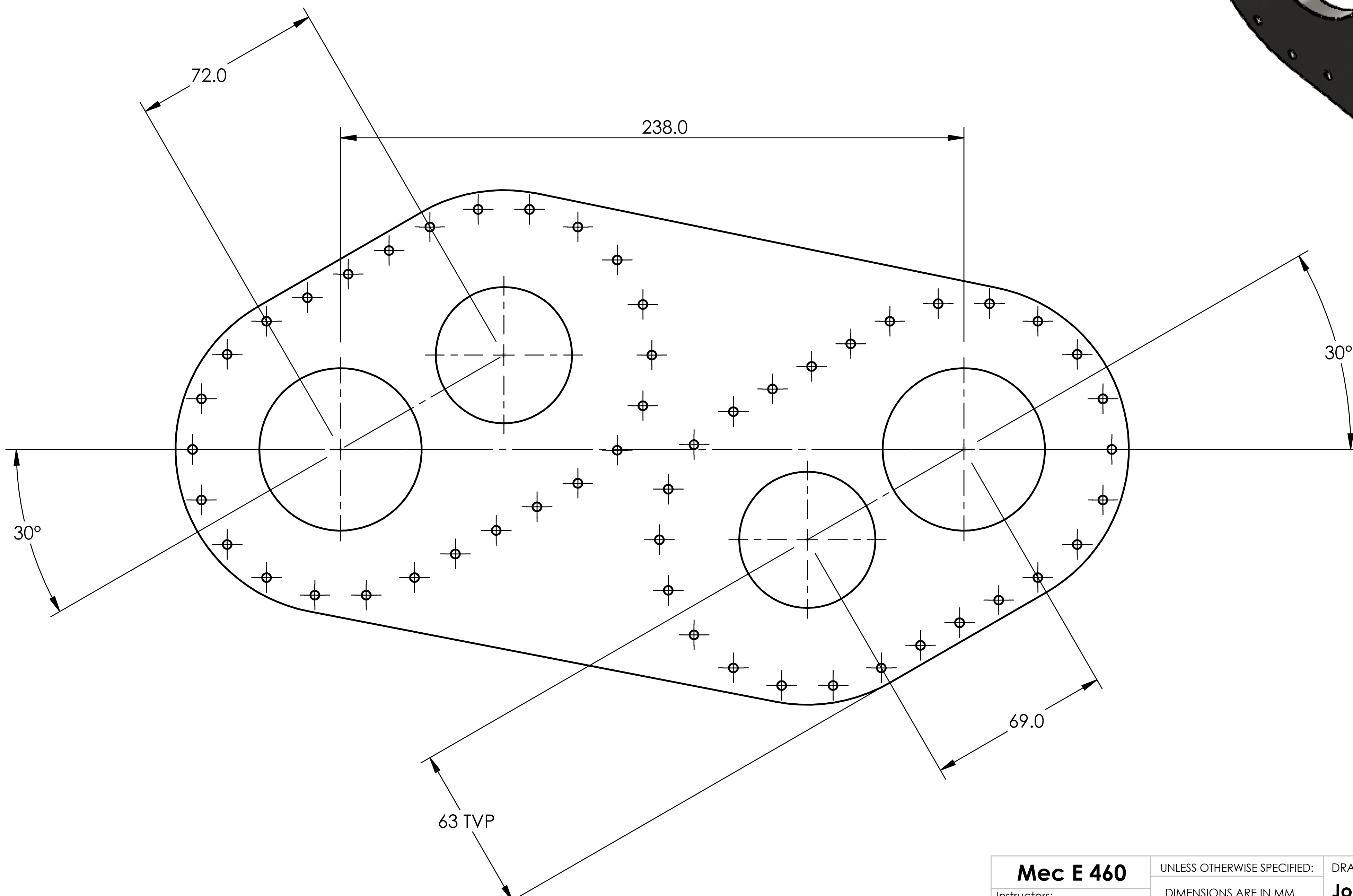
Wednesday, December 06, 2017 2:38:28 PM
Friday, November 24, 2017 3:28:25 PM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE: Frame Plate 1

SIZE	Part supplier/manufacturer	REV
B		1

SCALE: 2:3 | Mass: 2654.40 | SHEET 2 OF 2



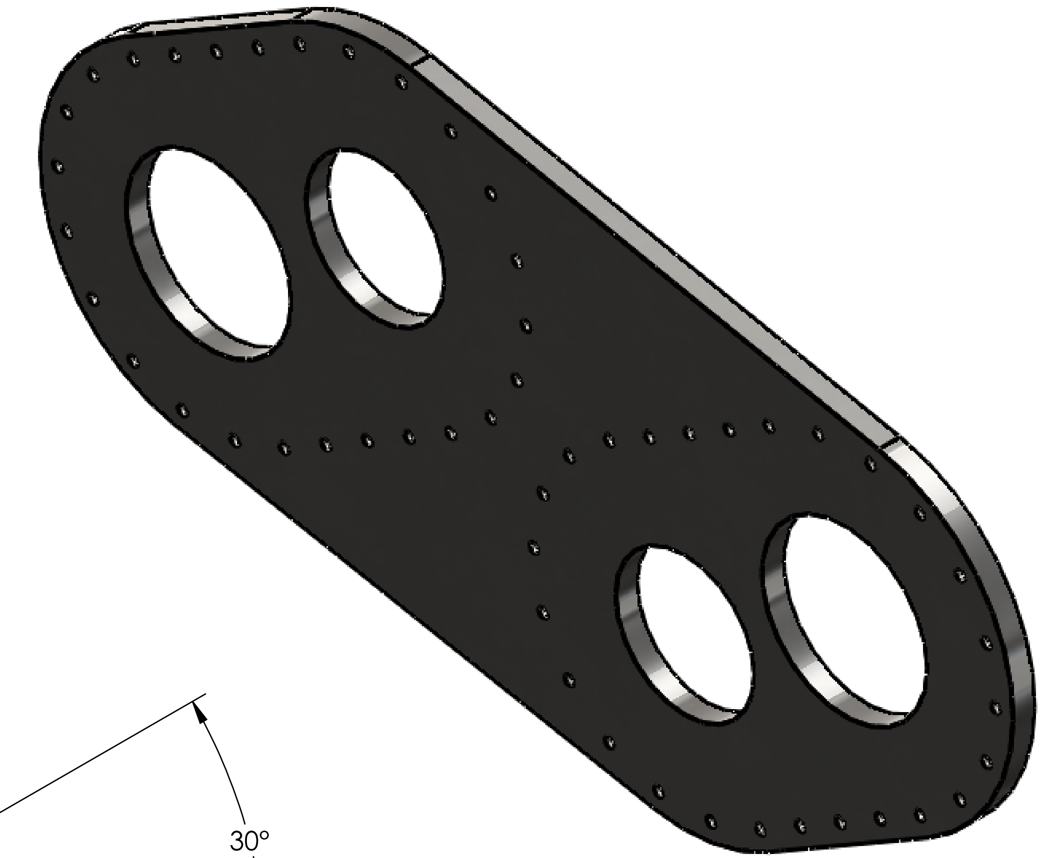
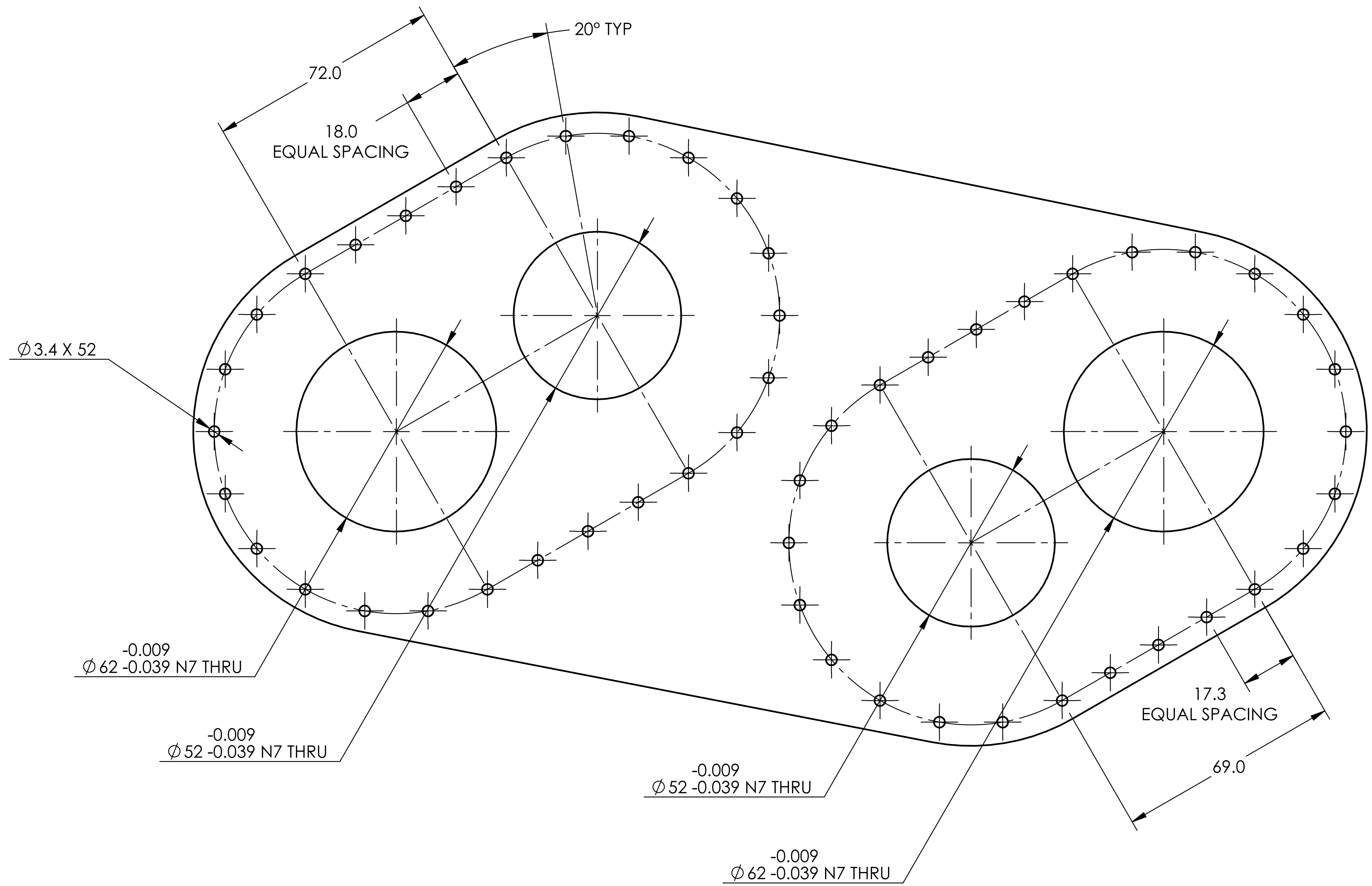


*Plate edges are constant distance from center of holes: 63 mm

Mec E 460 Instructors: Lipsett & Sadrzadeh Fall 2017 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: **Joshua Mulder** Group name: Mec E 460 Team 13 Group number: 13 SM By: Joshua Mulder Reviewed by: Colin Reimer/Zicheng Cao Wednesday, December 06, 2017 2:37:13 PM Friday, November 24, 2017 3:28:25 PM	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: Frame Plate 2 SIZE: B Part supplier/manufacturer REV: 1
MATERIAL: AISI 1020 FILE NAME: Frame Plate 2			SCALE: 1:2 / Mass: 2636.54 / SHEET 1 OF 2



Bolt hole patterns (slot pattern)contain the same number of hole each with the same slot radius. The slot center-center distance is different

Mec E 460
Instructors:
Lipsett & Sadrzadeh
Fall 2017
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:
AISI 1020
FILE NAME:
Frame Plate 2

DRAWN BY:
Joshua Mulder

Group name	Mec E 460 Team 13
Group number	13
SM By	Joshua Mulder
Reviewed by	Colin Reimer/Zicheng Cao

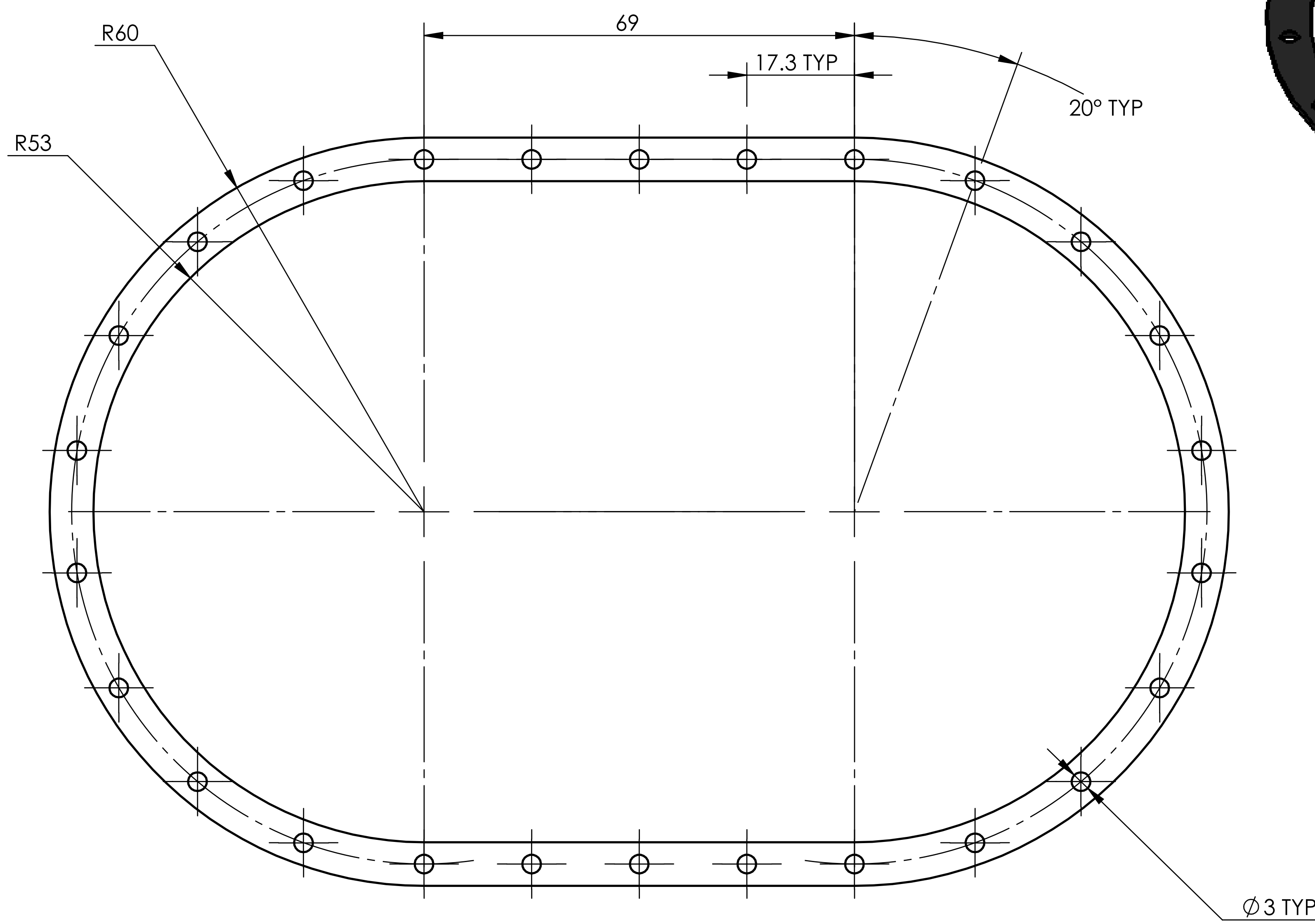
Wednesday, December 06, 2017 2:37:13 PM
Friday, November 24, 2017 3:28:25 PM

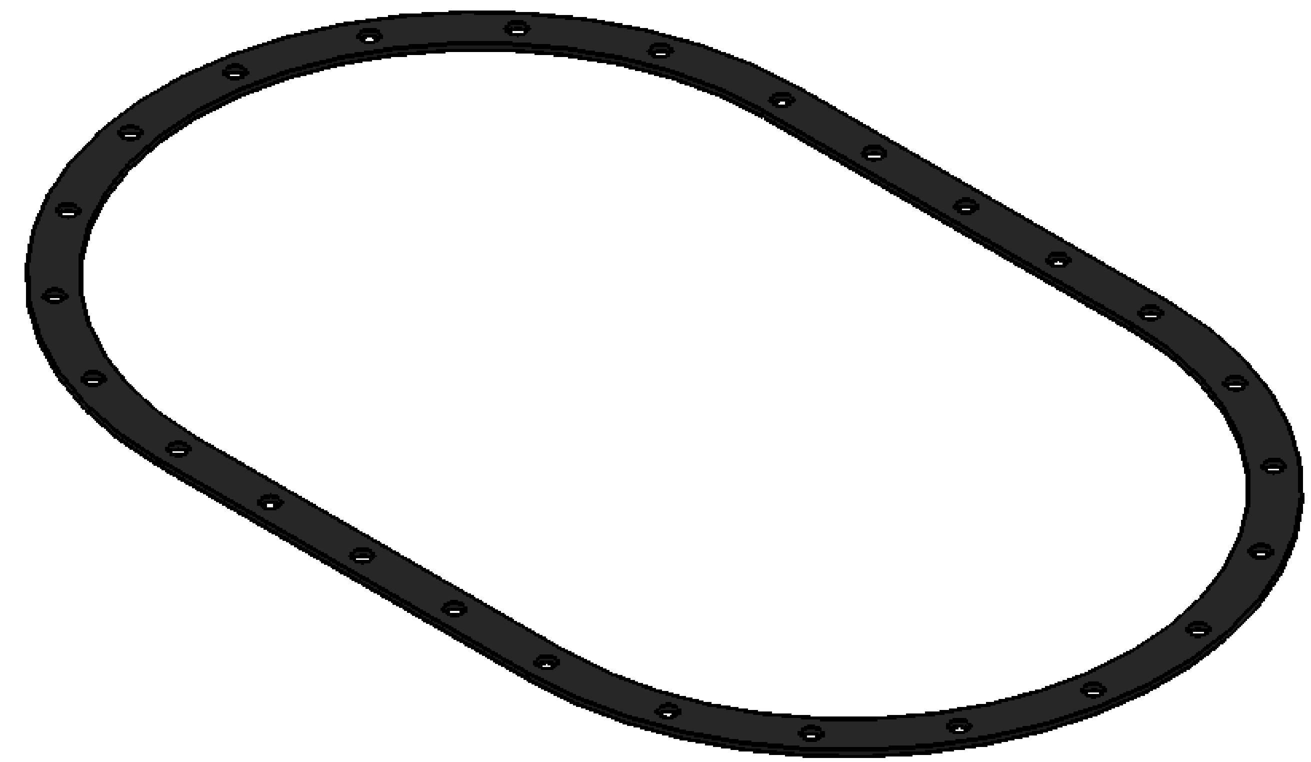
The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Frame Plate 2

SIZE	Part supplier/manufacturer	REV
B		1

SCALE: 2:3 | Mass: 2636.54 | SHEET 2 OF 2





Cut Gasket from 1mm Nitrile Rubber Sheet
or 0.5mm Oil Paper Jointing

Material Supplier: RAM Gasket Solutions Ltd.

Mec E 460
Instructors:
Lipsett & Sadrzadeh
Fall 2017
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:
Colin Reimer

Group name	Mec E 460 Team 13
Group number	13
SM By	Colin Reimer
Reviewed by	Christian Benade

Tuesday, December 05, 2017 3:30:42 PM
Saturday, December 02, 2017 3:26:10 PM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

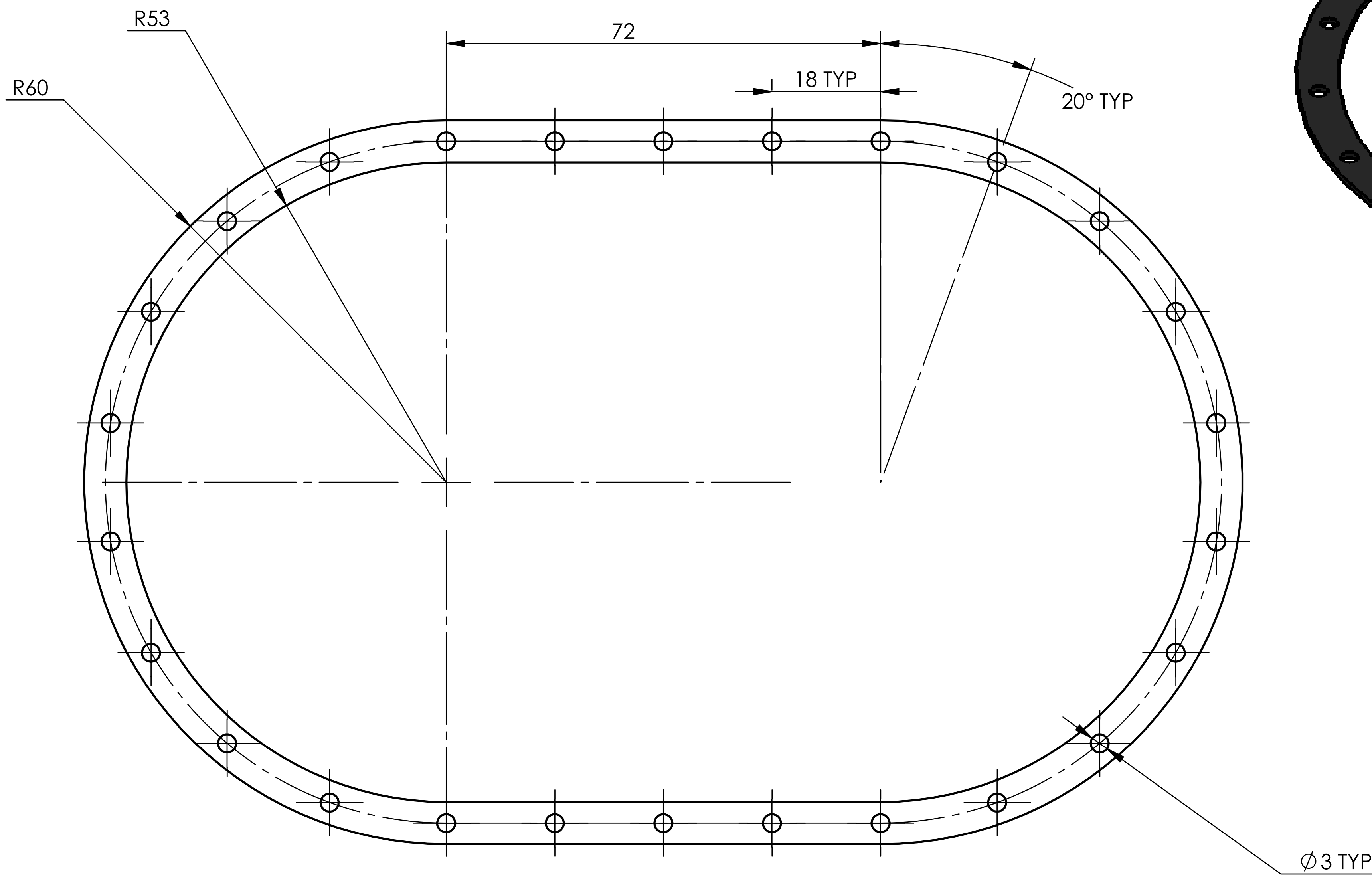
TITLE:
Gasket Seal 69mm

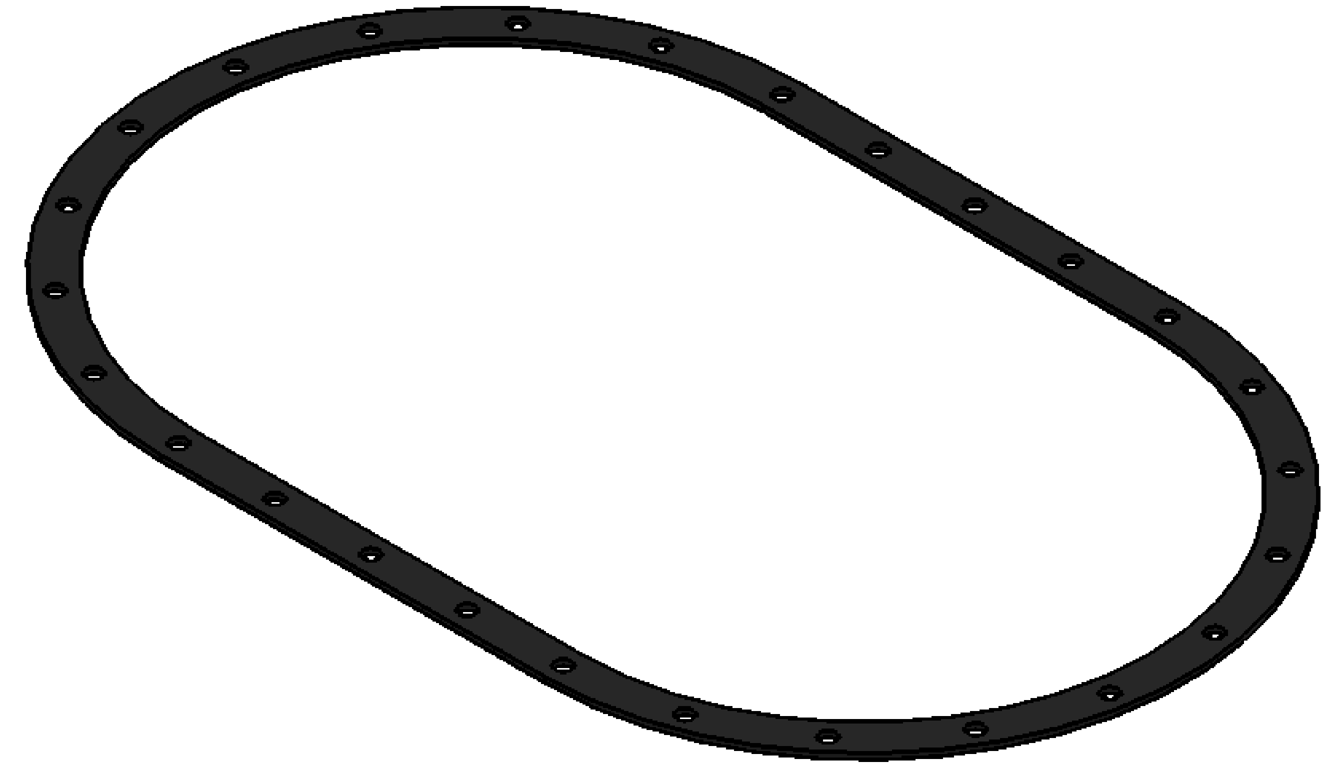
MATERIAL:
BA60 Nitrile Rubber or Oil Paper Jointing

FILE NAME:
Gasket Seal 69mm

SIZE	Part supplier/manufacturer	REV
B		

SCALE: 1:1 | Mass: 4.22 | SHEET 1 OF 1





Cut Gasket from 1mm Nitrile Rubber Sheet
or 0.5mm Oil Paper Jointing

Material Supplier: RAM Gasket Solutions Ltd.

Mec E 460

Instructors: Lipsett & Sadrzadeh
Fall 2017

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: **Colin Reimer**

Group name	Mec E 460 Team 13
Group number	13
SM By	Colin Reimer
Reviewed by	Christian Benade

Tuesday, December 05, 2017 3:30:42 PM
Saturday, December 02, 2017 3:26:10 PM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

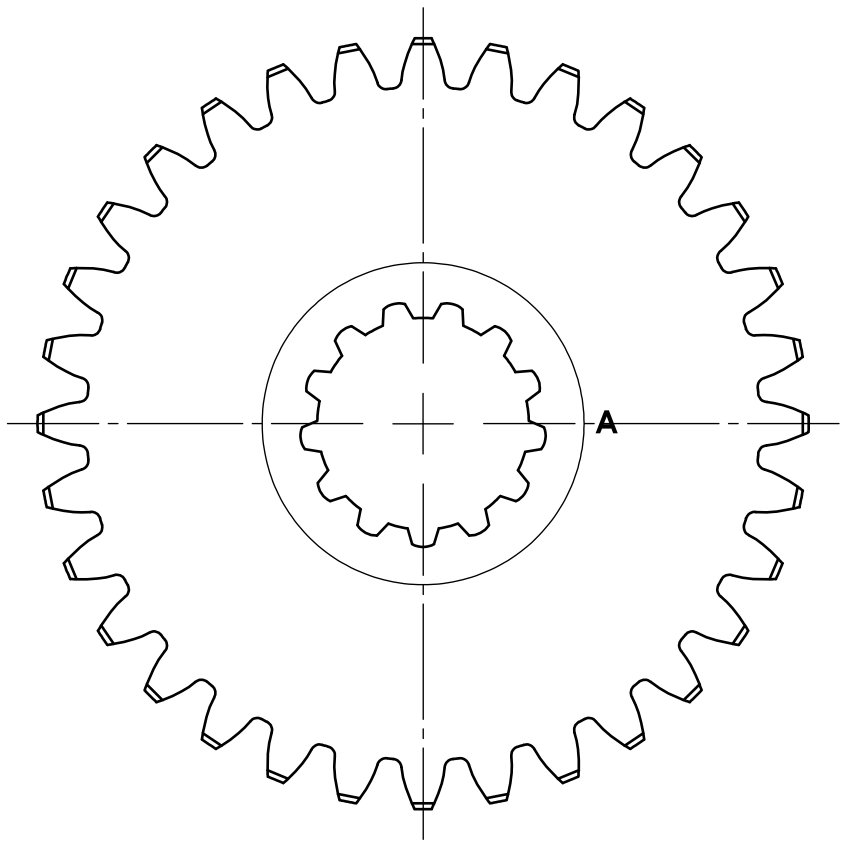
TITLE:
Gasket Seal 72mm

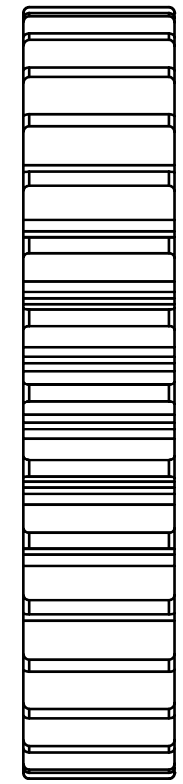
MATERIAL:
BA60 Nitrile Rubber or Oil Paper Jointing

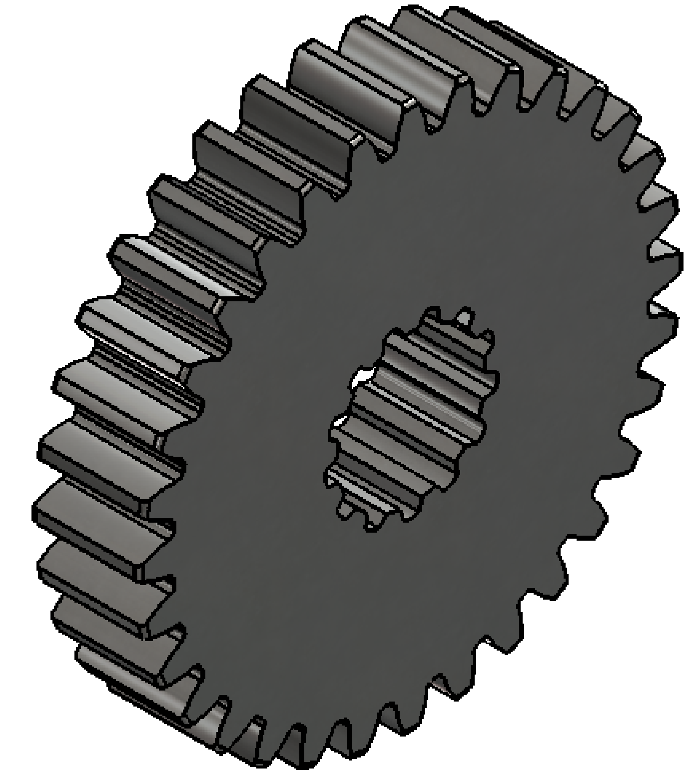
FILE NAME:
Gasket Seal 72mm

SIZE B | Part supplier/manufacturer | REV

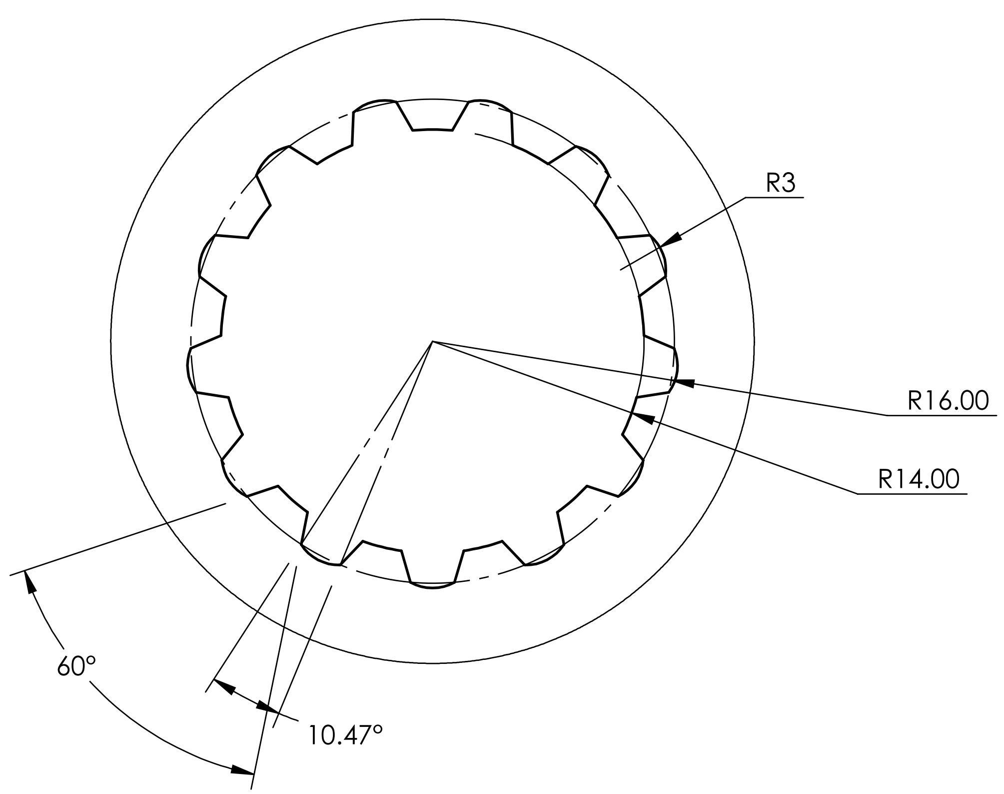
SCALE: 1:1 | Mass: 4.28 | SHEET 1 OF 1







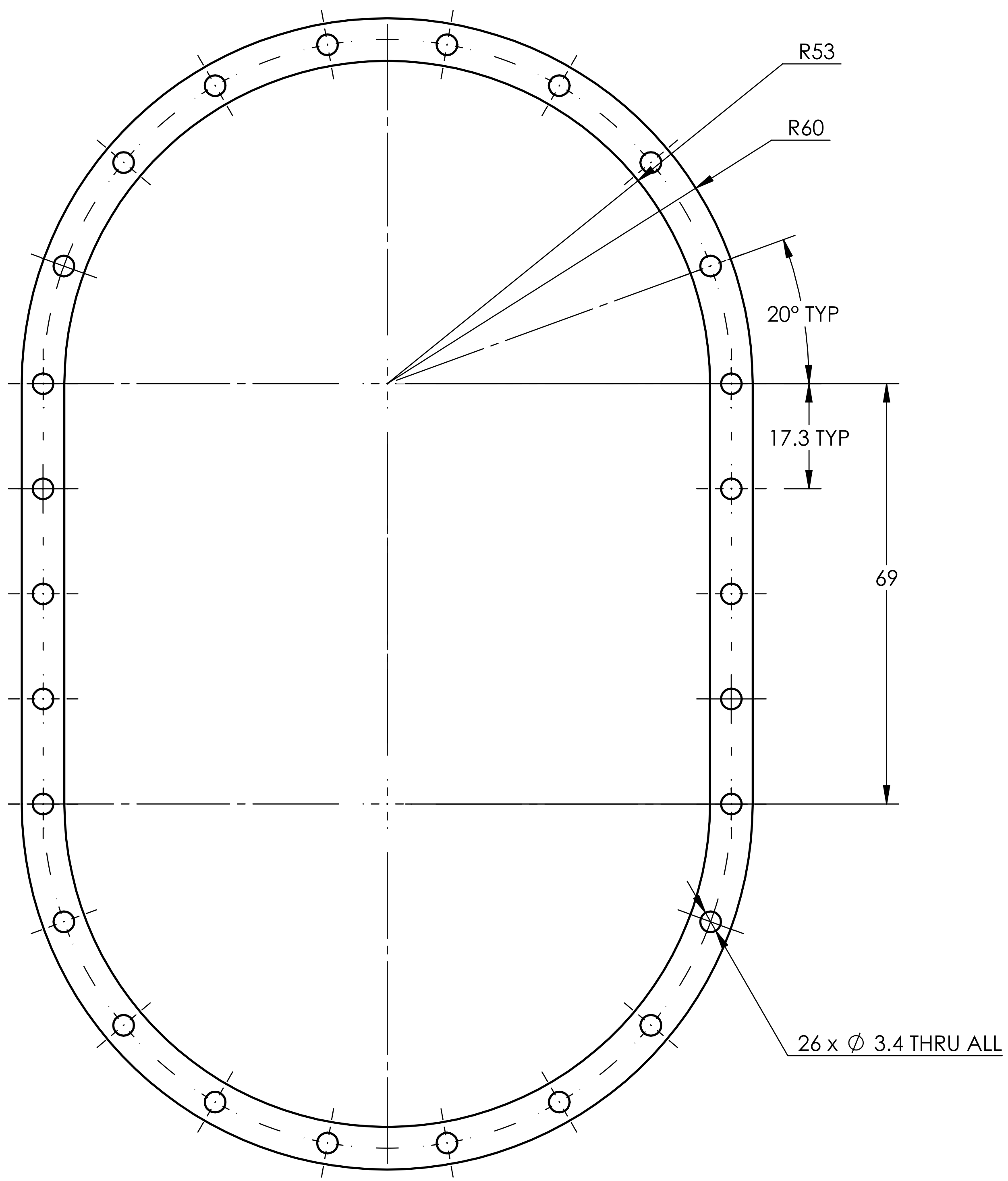
Spline:
13 teeth
Spline depth: 2 mm
Tooth Angle: 60 deg
Undercut Radius: 3 mm

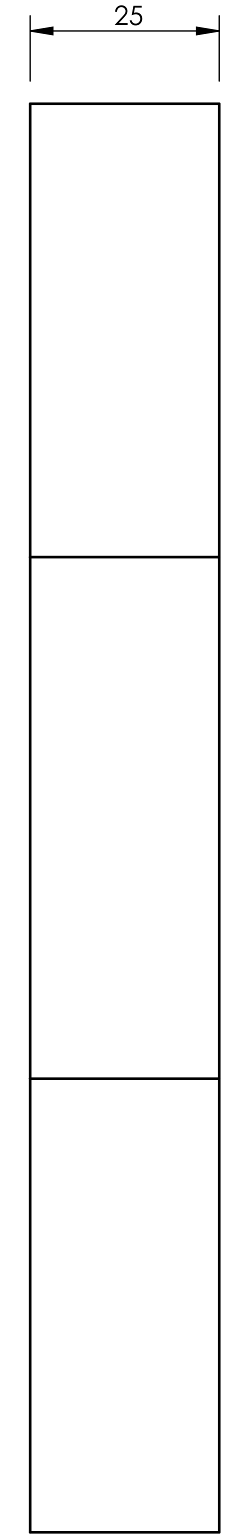


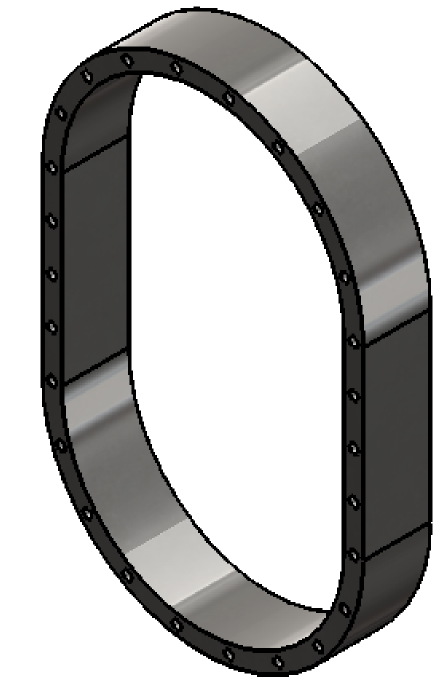
DETAIL A
SCALE 2 : 1

QTC Gear:
Part Number: A 1C 1MYK30032
Modification - Add spline to center for mounting on shaft

Mec E 460 Instructors: Lipsett & Sadrzadeh **Fall 2017** 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: **Colin Reimer** Group name: MECE 460 Team 13 Group number: 13 SM By: Colin Reimer Reviewed by: Christian Benade Tuesday, December 05, 2017 3:30:41 PM Saturday, December 02, 2017 12:19:52 PM	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: 32 Tooth Gear_QTC Mod SIZE B | Part supplier/manufacturer | REV 1 SCALE: 1:1 | Mass: 1010.30 | SHEET 1 OF 1
MATERIAL: Plain Carbon Steel FILE NAME: SDP_a_1c_1myk30032			

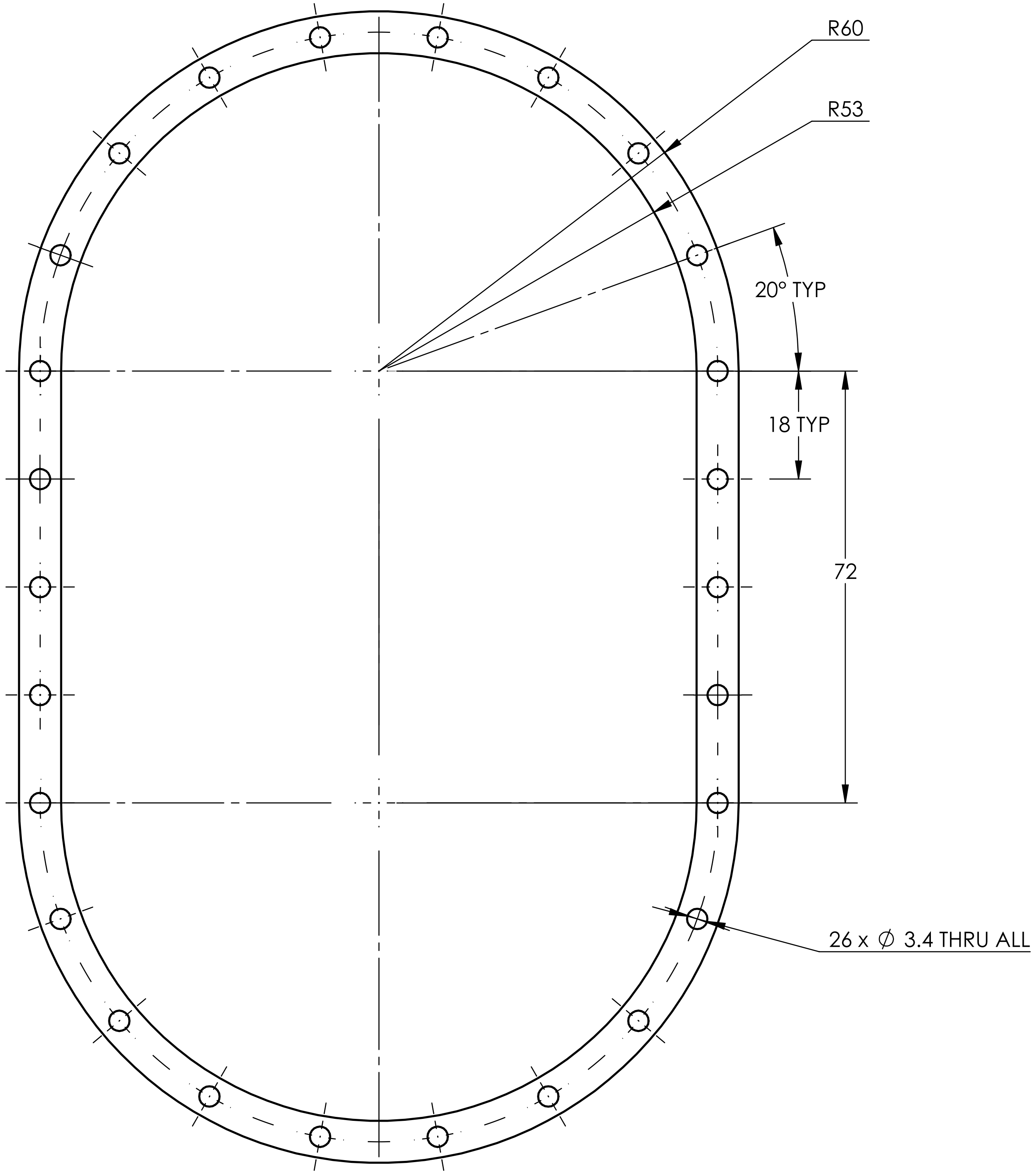


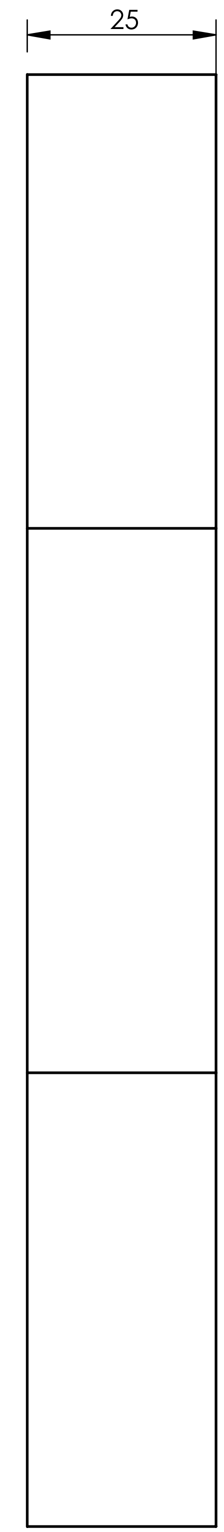


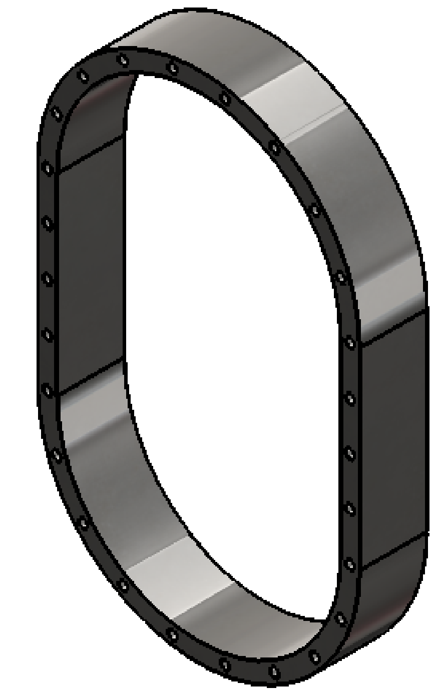


Nominal surface finish is sufficient
for gasket sealing surface

Mec E 460 Instructors: Lipsett & Sadrzadeh **Fall 2017** 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: **Colin Reimer** Group name: Mec E 460 Team 13 Group number: 13 SM By: Josh Mulder Reviewed by: Christian Benade	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: Seal 1-2
MATERIAL: AISI 1020 FILE NAME: Seal 1-2		Tuesday, December 05, 2017 3:41:18 PM Monday, November 27, 2017 10:03:28 PM	SIZE: B · Part supplier/manufacturer · REV: 1 SCALE: 1:1 · Mass: 634.95 · SHEET 1 OF 1







Nominal surface finish is sufficient
for gasket sealing surface

Mec E 460

Instructors: Lipsett & Sadrzadeh
Fall 2017

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:
AISI 1020

FILE NAME:
Seal 3-4

DRAWN BY:
Colin Reimer

Group name	Mec E 460 Team 13
Group number	13
SM By	Josh Mulder
Reviewed by	Christian Benade

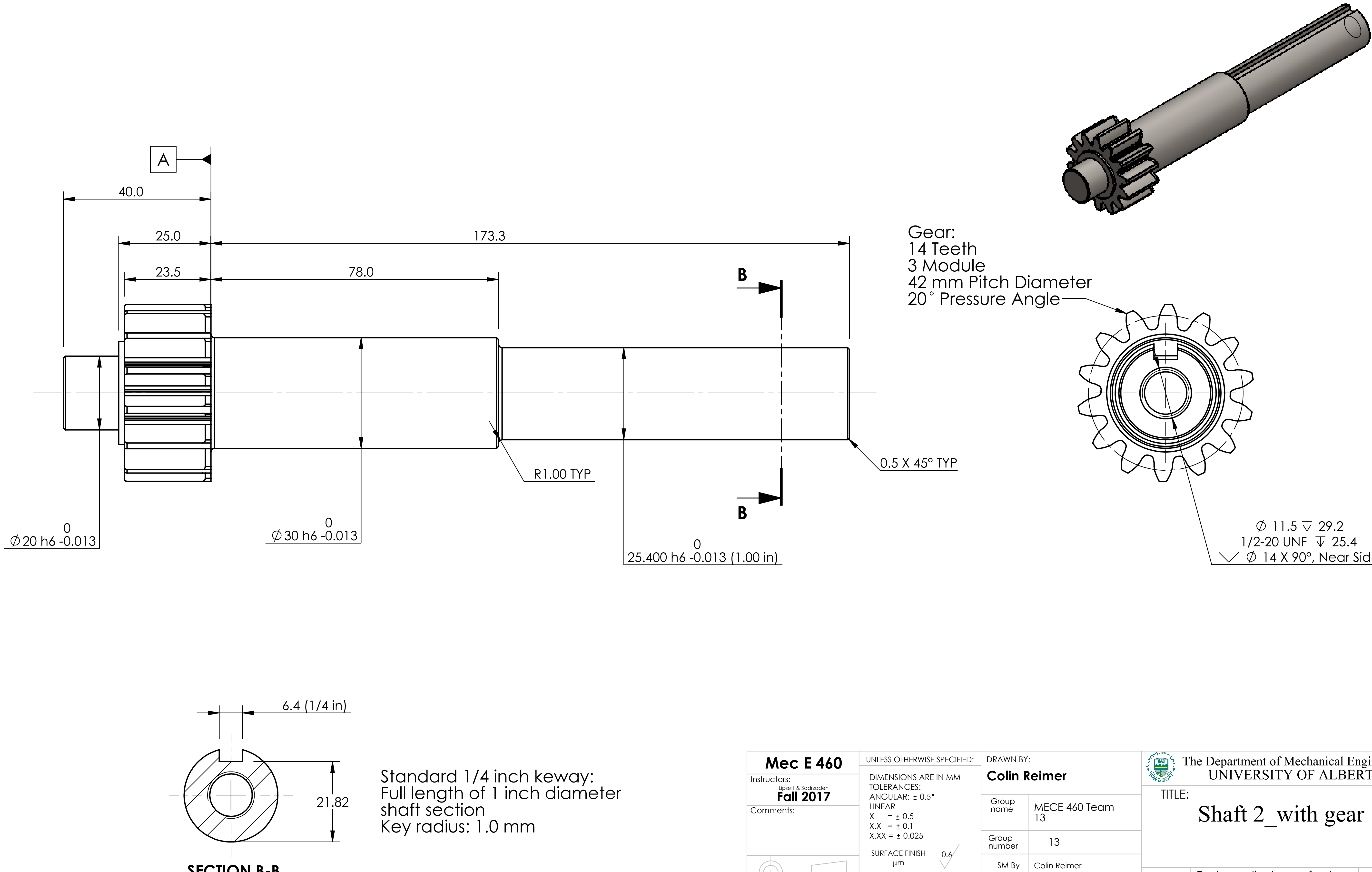
Tuesday, December 05, 2017 3:41:39 PM
Monday, November 27, 2017 10:03:28 PM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Seal 3-4

SIZE	Part supplier/manufacturer	REV
B		1

SCALE: 1:1 | Mass: 643.24 | SHEET 1 OF 1



Mec E 460
Instructors: Lipsett & Sadrzadeh
Fall 2017
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:
AISI 4340 Steel, normalized
FILE NAME:
Shaft 2_with gear

DRAWN BY:
Colin Reimer

Group name	MECE 460 Team 13
Group number	13
SM By	Colin Reimer
Reviewed by	Christian Benade

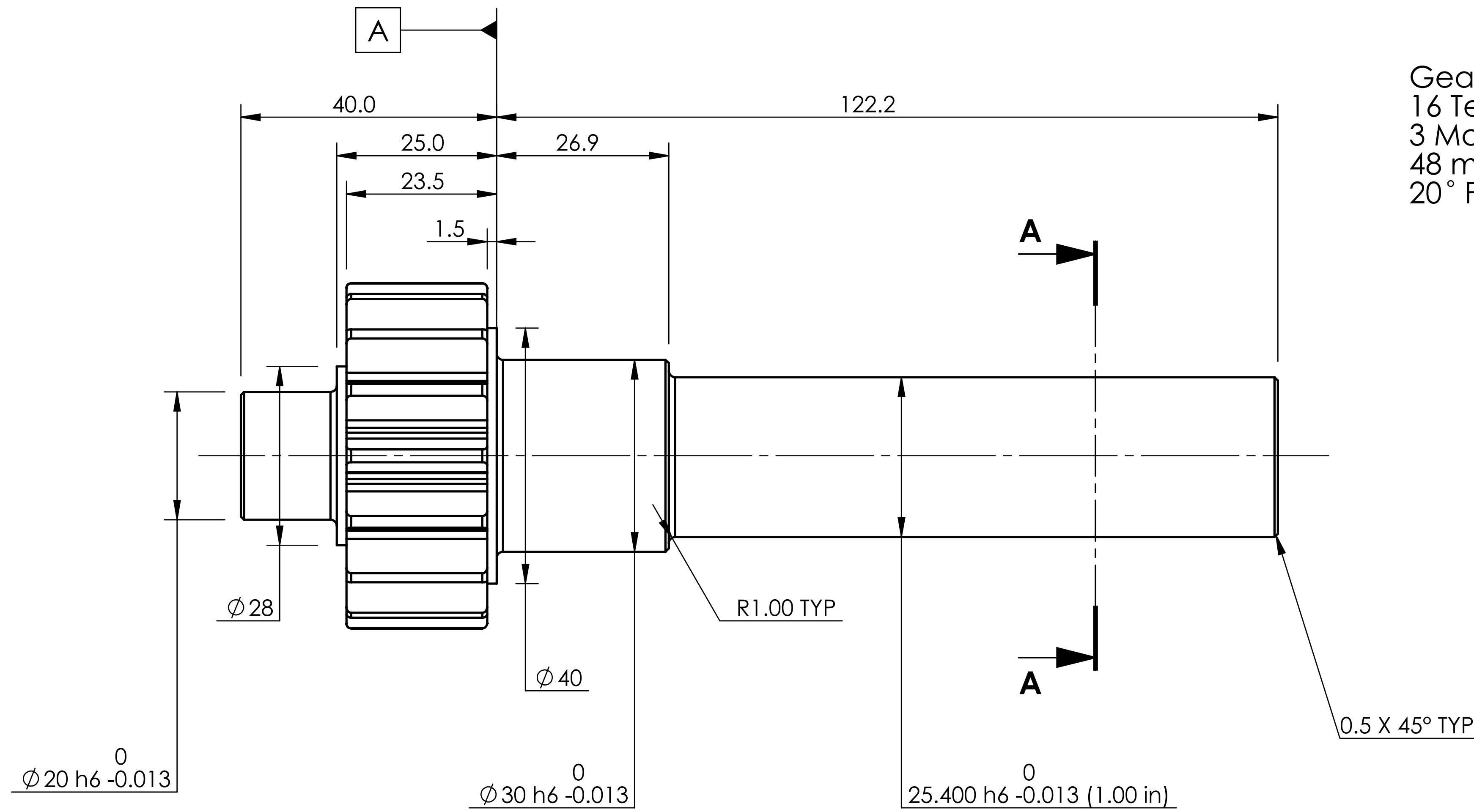
Tuesday, December 05, 2017 3:30:41 PM
Saturday, December 02, 2017 2:18:04 PM

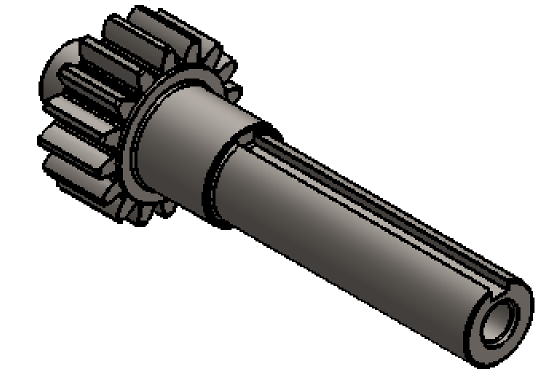
The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Shaft 2_with gear

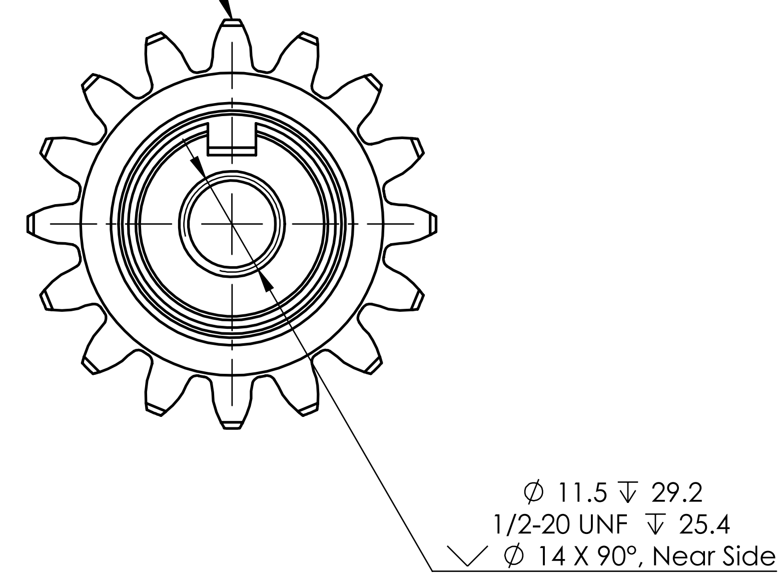
SIZE B | Part supplier/manufacturer | REV 1

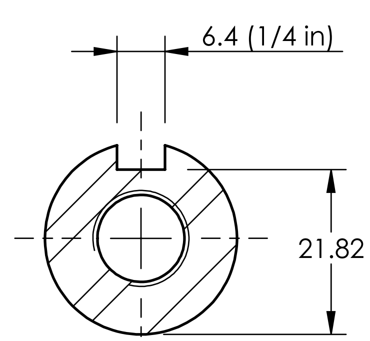
SCALE: 1:1 | Mass: 1066.77 | SHEET 1 OF 1





Gear:
16 Teeth
3 Module
48 mm Pitch Diameter
20° Pressure Angle





SECTION A-A

Standard 1/4 inch keway:
Full length of 1 inch diameter shaft section
Key radius: 1.0 mm

Mec E 460
Instructors: Lipsett & Sadrzadeh
Fall 2017
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: AISI 4340 Steel, normalized
FILE NAME: Shaft 3_with gear

DRAWN BY:	Colin Reimer
Group name	MECE 460 Team 13
Group number	13
SM By	Colin Reimer/Josh Mulder
Reviewed by	Christian Benade

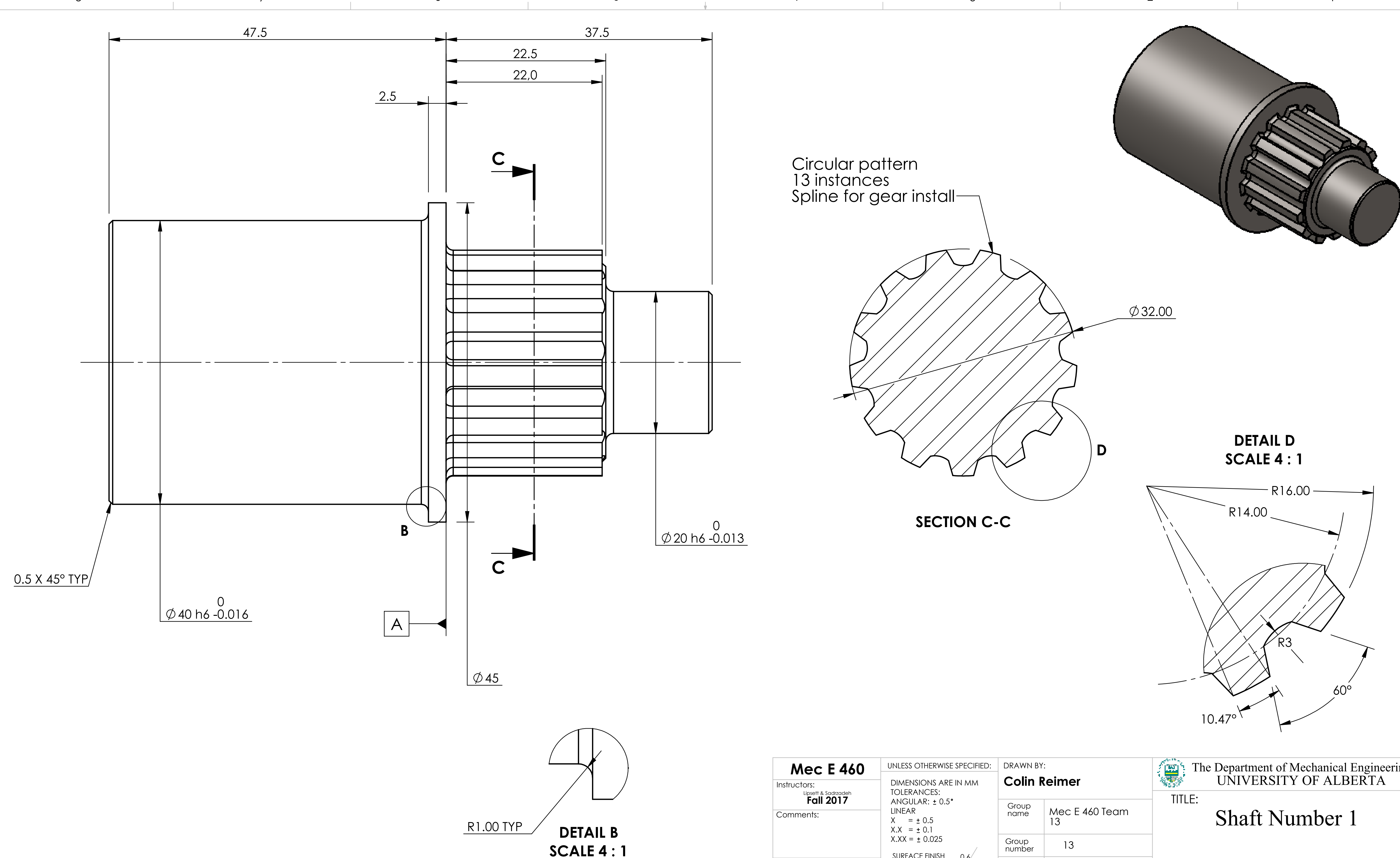
Tuesday, December 05, 2017 3:30:41 PM
Saturday, December 02, 2017 2:31:43 PM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

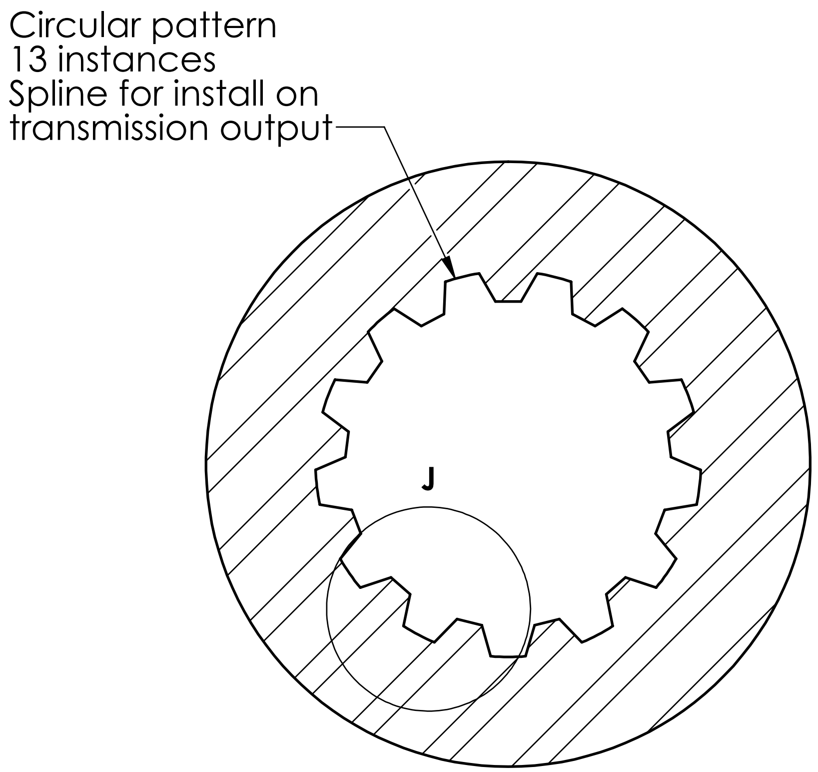
TITLE: Shaft 3_with gear

SIZE	Part supplier/manufacturer	REV
B		1

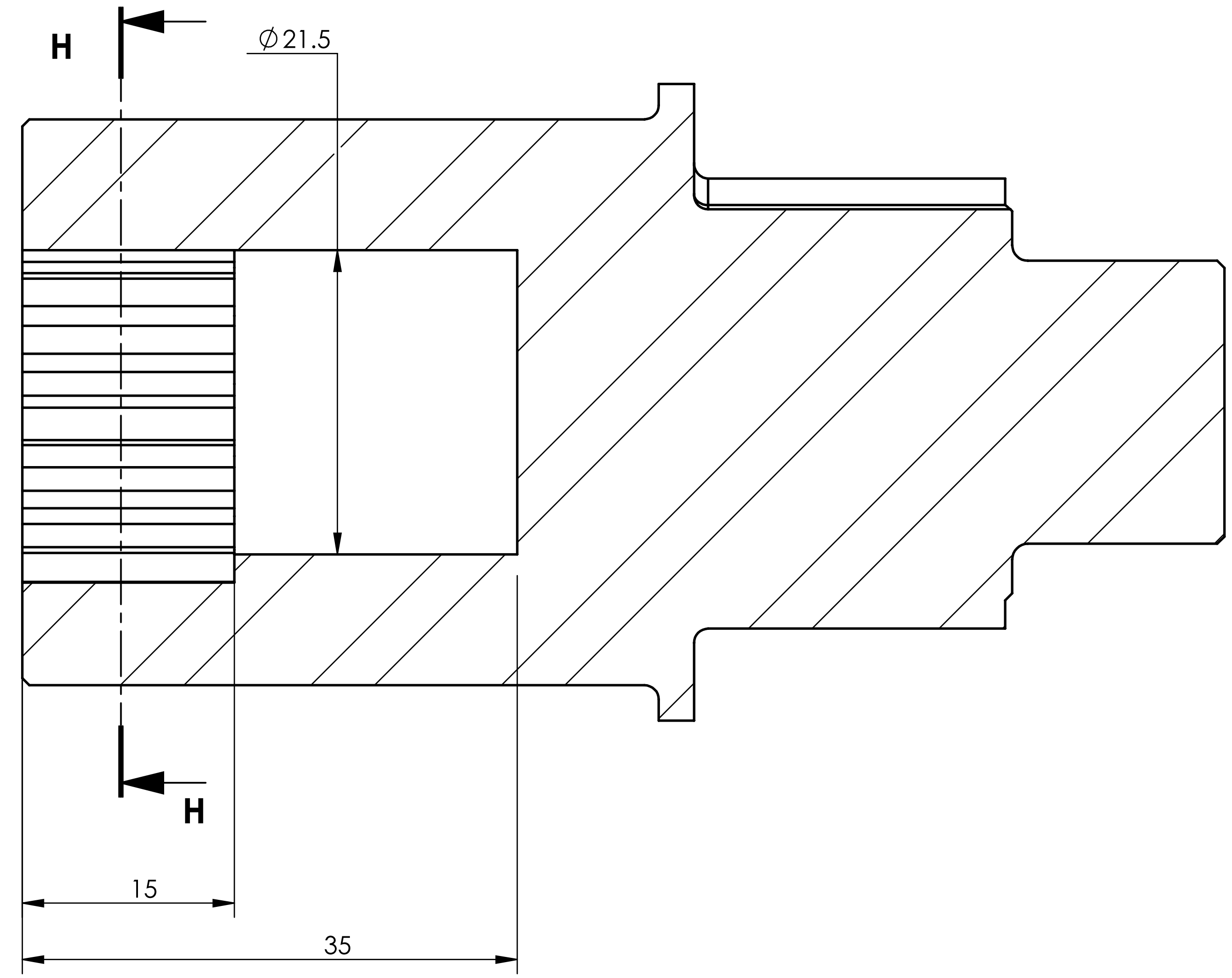
SCALE: 1:1 | Mass: 854.13 | SHEET 1 OF 1

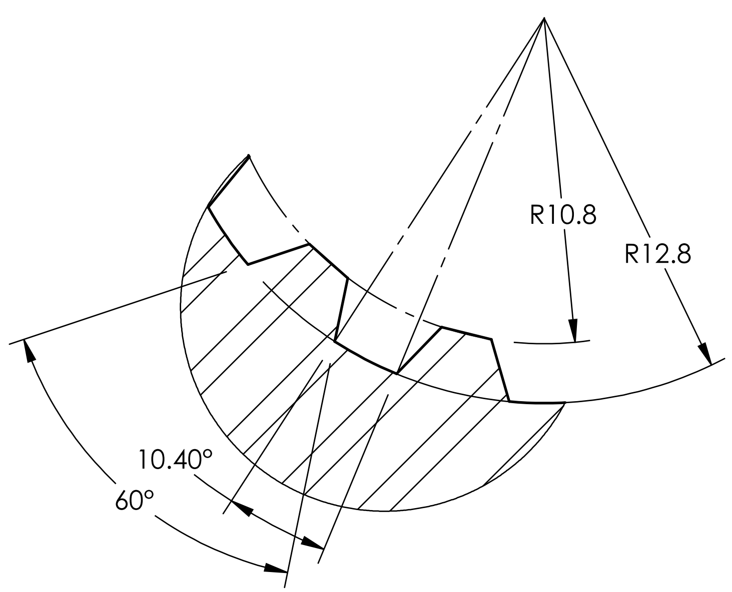


Mec E 460 Instructors: Lipsett & Sadrzadeh Fall 2017 Comments:	UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025 SURFACE FINISH µm 0.6 DO NOT SCALE DRAWING	DRAWN BY: **Colin Reimer** Group name: Mec E 460 Team 13 Group number: 13 SM By: Colin Reimer/Josh Mulder Reviewed by: Christian Benade	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: Shaft Number 1
MATERIAL: AISI 4340 Steel, normalized FILE NAME: Shaft number 1		Tuesday, December 05, 2017 3:30:41 PM Friday, November 24, 2017 2:55:24 PM	SIZE **B** · Part supplier/manufacturer · REV 1 SCALE: 2:1 · Mass: 526.85 · SHEET 1 OF 2



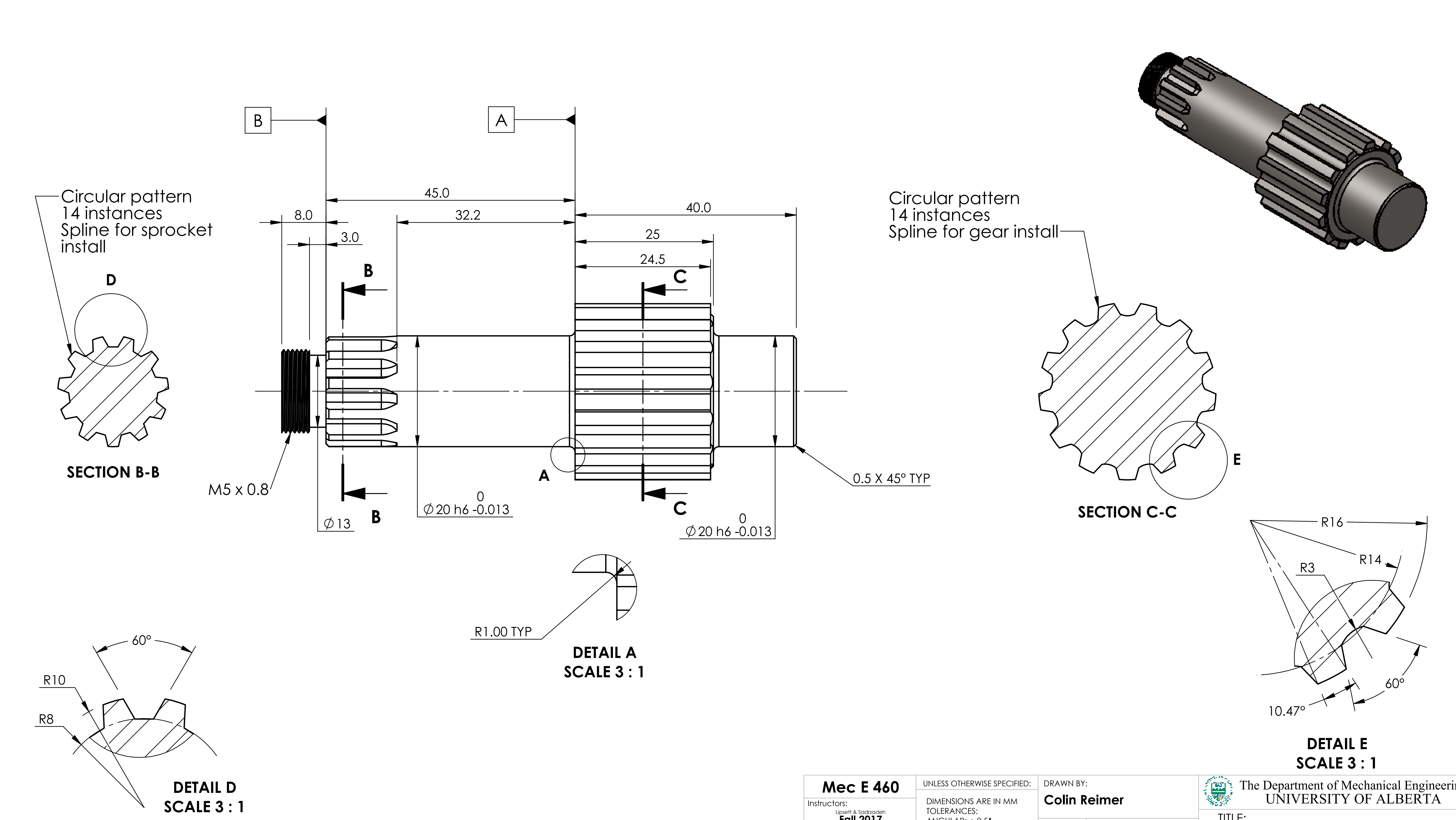
SECTION H-H



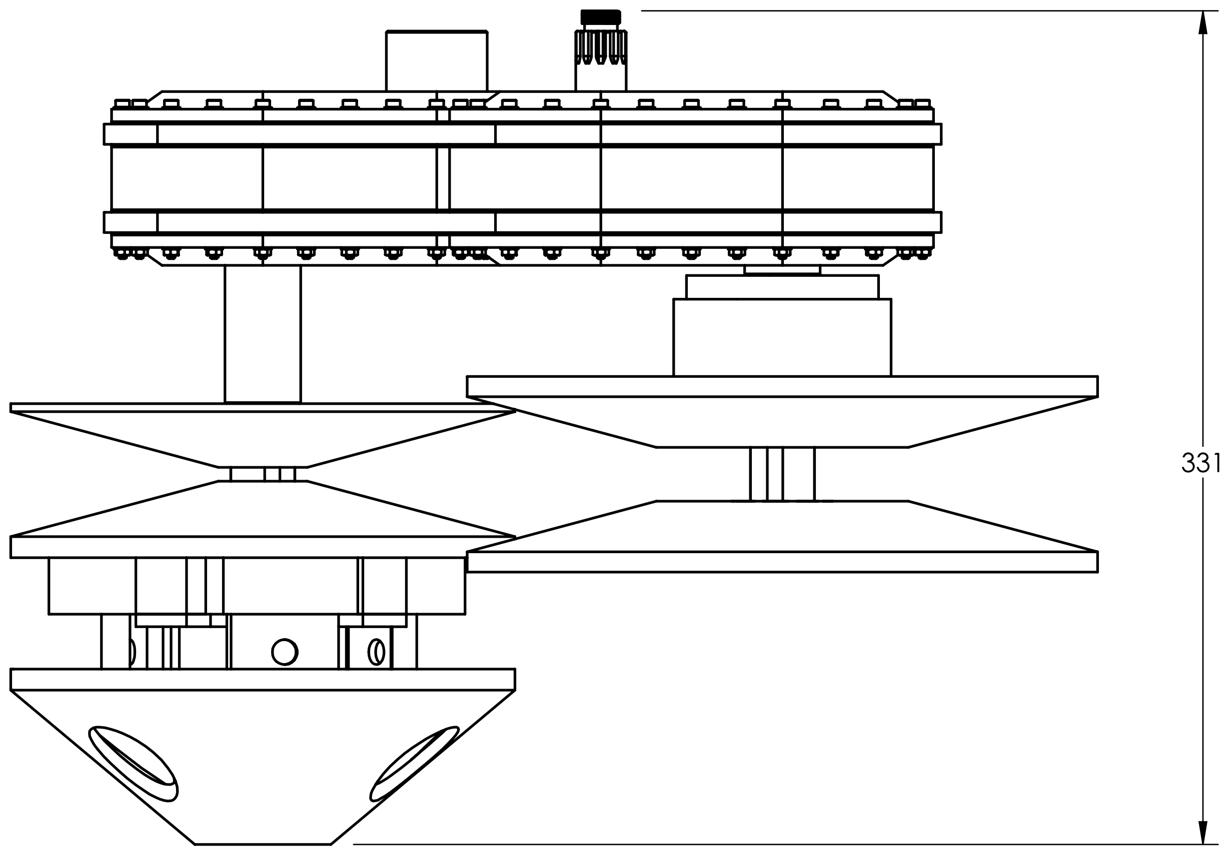


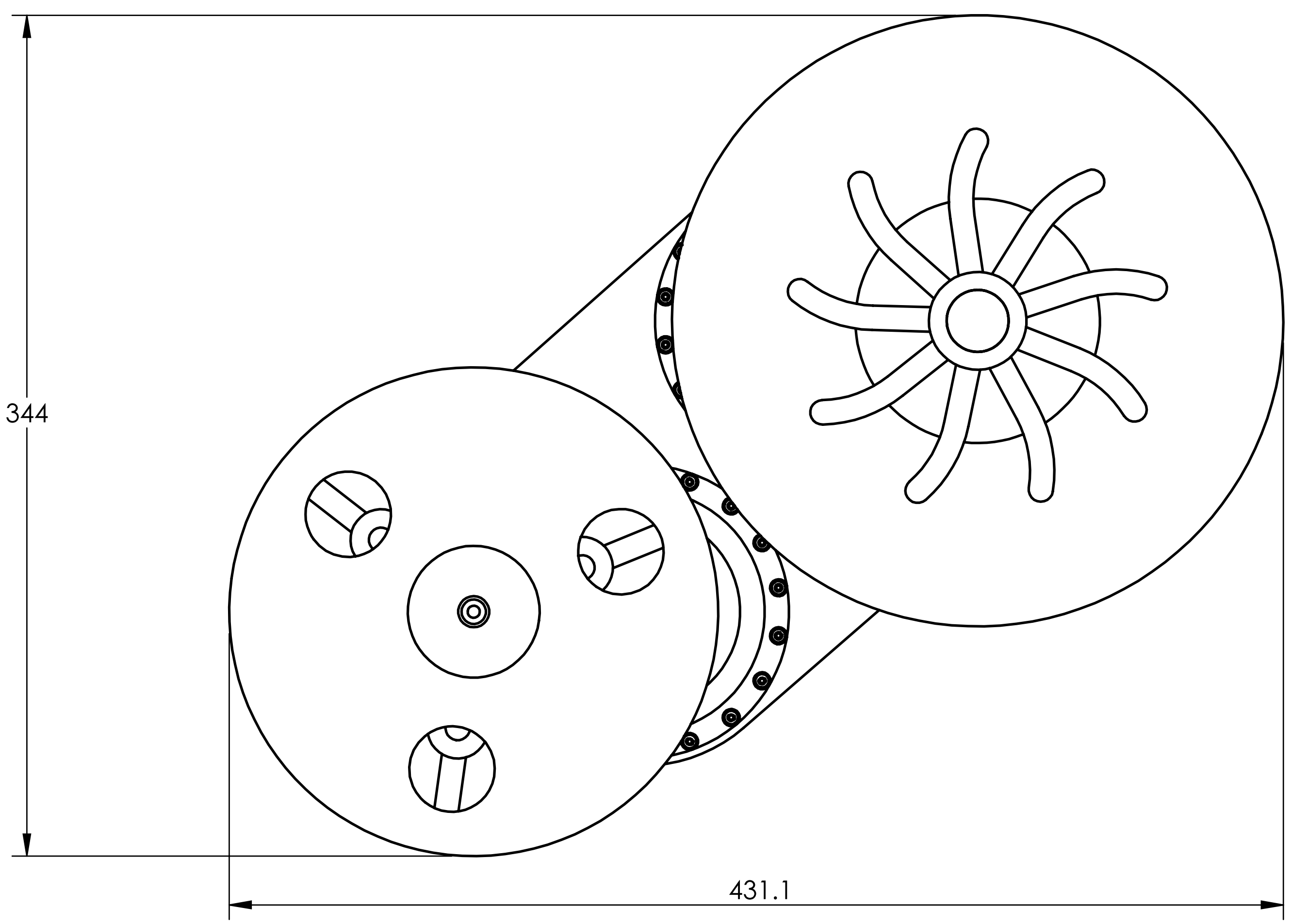
DETAIL J
SCALE 4 : 1

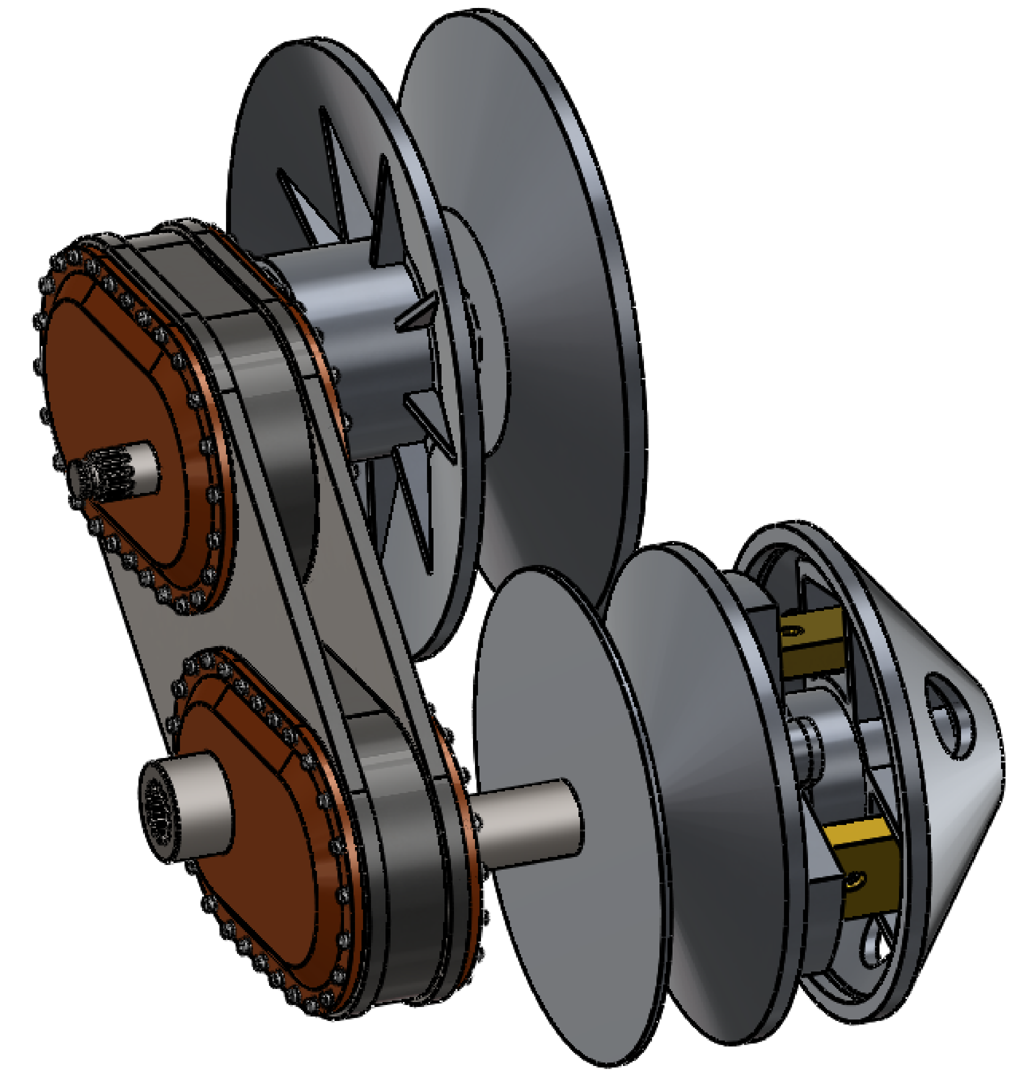
Mec E 460 Instructors: Lipsett & Mohtada **Fall 2017** Comments:	UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025 SURFACE FINISH µm 0.6 DO NOT SCALE DRAWING	DRAWN BY: **Colin Reimer** Group name: Mec E 460 Team 13 Group number: 13 SM By: Colin Reimer/Josh Mulder Reviewed by: Christian Benade Tuesday, December 05, 2017 3:30:41 PM Friday, November 24, 2017 2:55:24 PM	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: Shaft Number 1
MATERIAL: AISI 4340 Steel, normalized FILE NAME: Shaft number 1			SIZE **B** Part supplier/manufacturer REV 1 SCALE: 2:1 Mass: 526.85 SHEET 2 OF 2



Mec E 460 Instructors: Lipsett & Sadrzadeh **Fall 2017** 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: **Colin Reimer** Group name: Mec E 460 Team 13 Group number: 13 SM By: Colin Reimer/Josh Mulder Reviewed by: Christian Benade	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: Shaft Number 4
MATERIAL: AISI 4340 Steel, normalized FILE NAME: Shaft number 4		Tuesday, December 05, 2017 3:30:41 PM Monday, November 27, 2017 8:39:47 PM	SIZE **B** — Part supplier/manufacturer — REV 1 SCALE: 3:2 — Mass: 288.96 — SHEET 1 OF 1

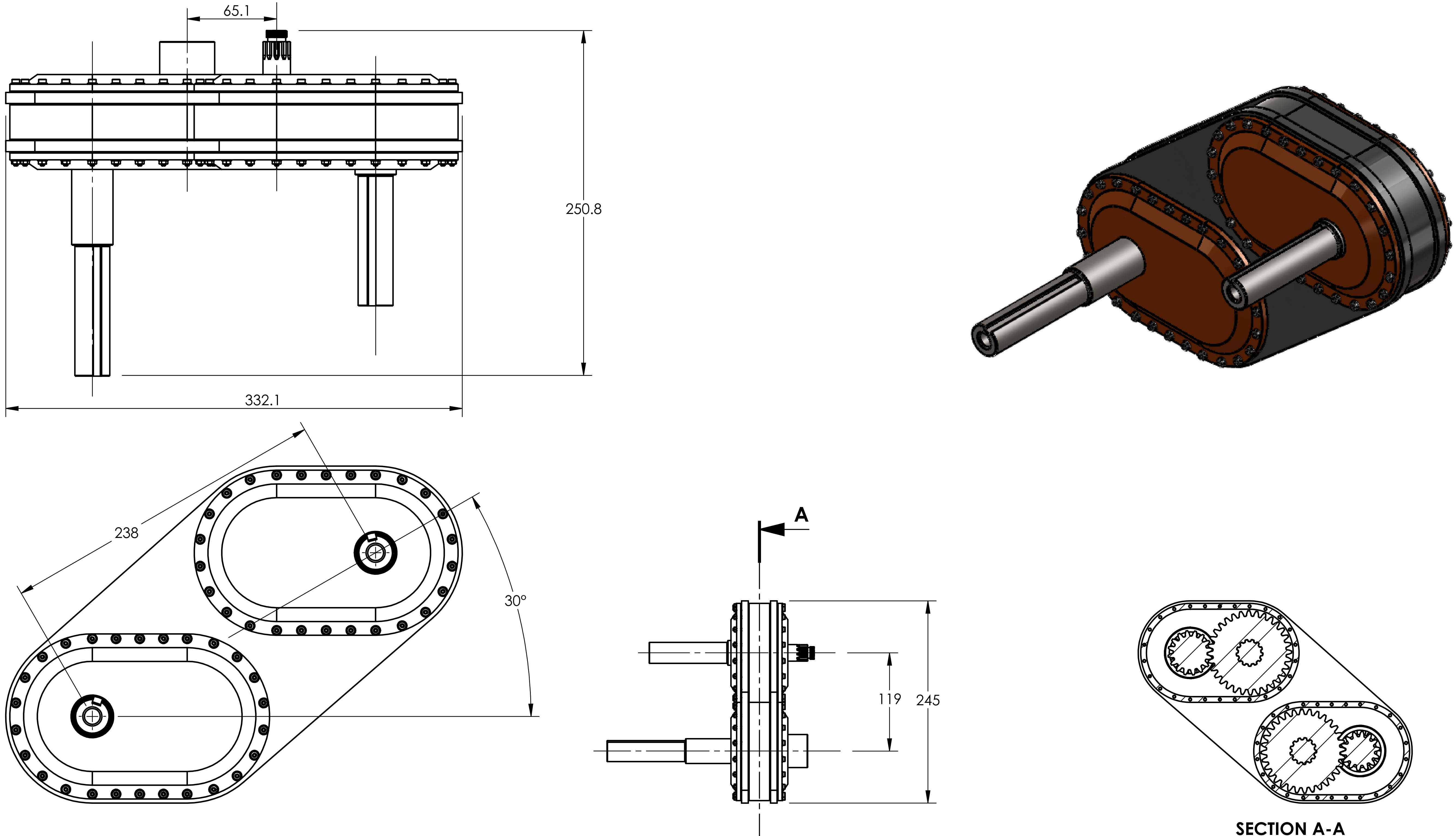






*CVT Pulleys supplied from CVTech - Models here are approximate representations of sourced components

Mec E 460 Instructors: Lipsett & Mohtada Sadrzadeh **Fall 2017** 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: **Joshua Mulder** Group name: Mec E 460 Team 13 Group number: 13 SM By: Joshua Mulder Reviewed by: Set in this file Thursday, December 07, 2017 7:27:02 PM Friday, November 24, 2017 4:11:47 PM	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: Transmission Assembly SIZE **B** — Part supplier/manufacturer — REV SCALE: 1:3 — Mass: 23981.60 — SHEET 1 OF 4
MATERIAL:			
FILE NAME: Full Transmission Assembly			



Mec E 460
Instructors:
Lipsett & Mohtada Sadrzadeh
Fall 2017
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:

FILE NAME:
Full Transmission Assembly

DRAWN BY:
Joshua Mulder

Group name	Mec E 460 Team 13
Group number	13
SM By	Joshua Mulder
Reviewed by	Set in this file

Thursday, December 07, 2017 7:27:02 PM
Friday, November 24, 2017 4:11:47 PM

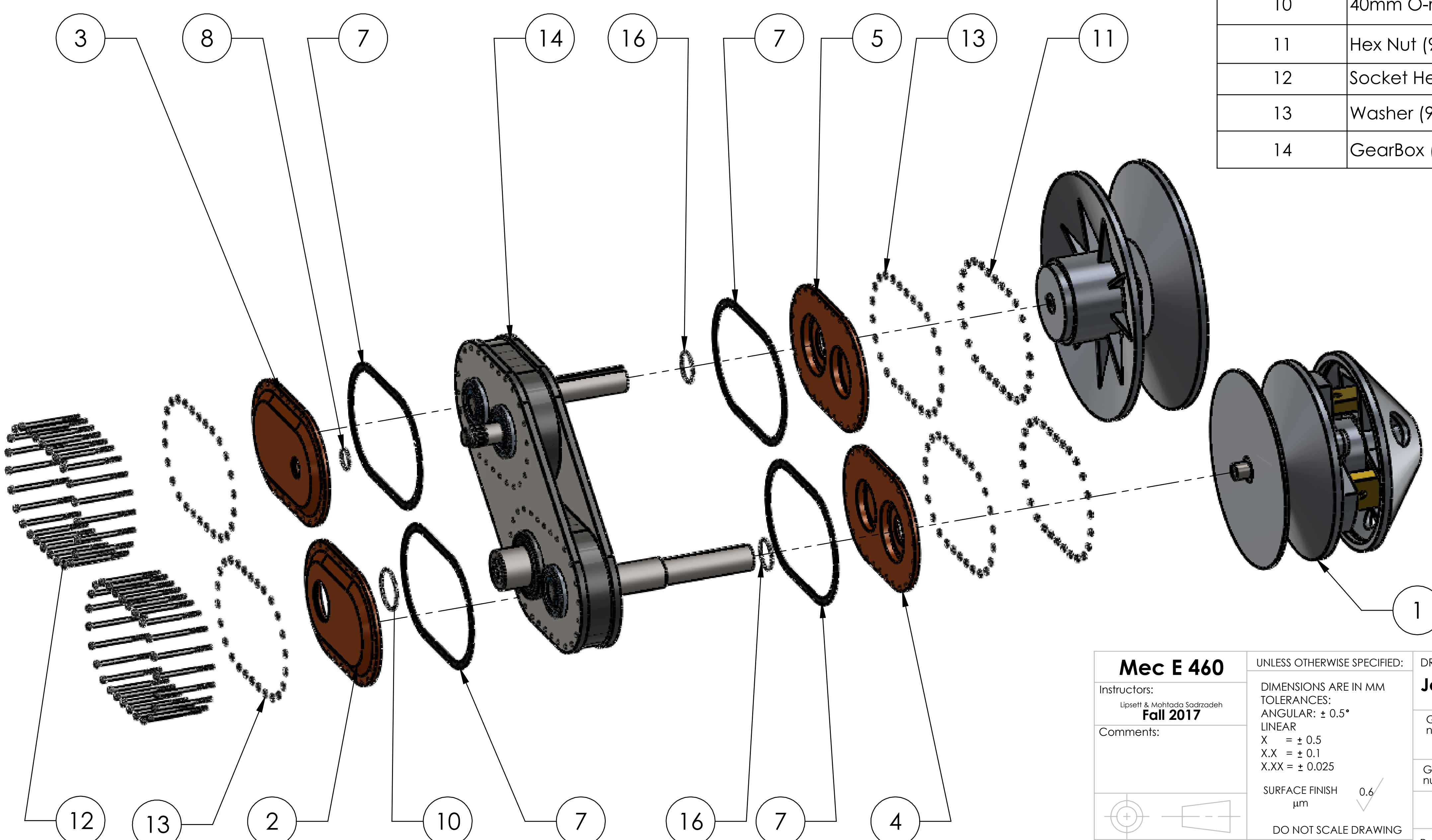
The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Transmission Assembly

SIZE B | Part supplier/manufacturer | REV

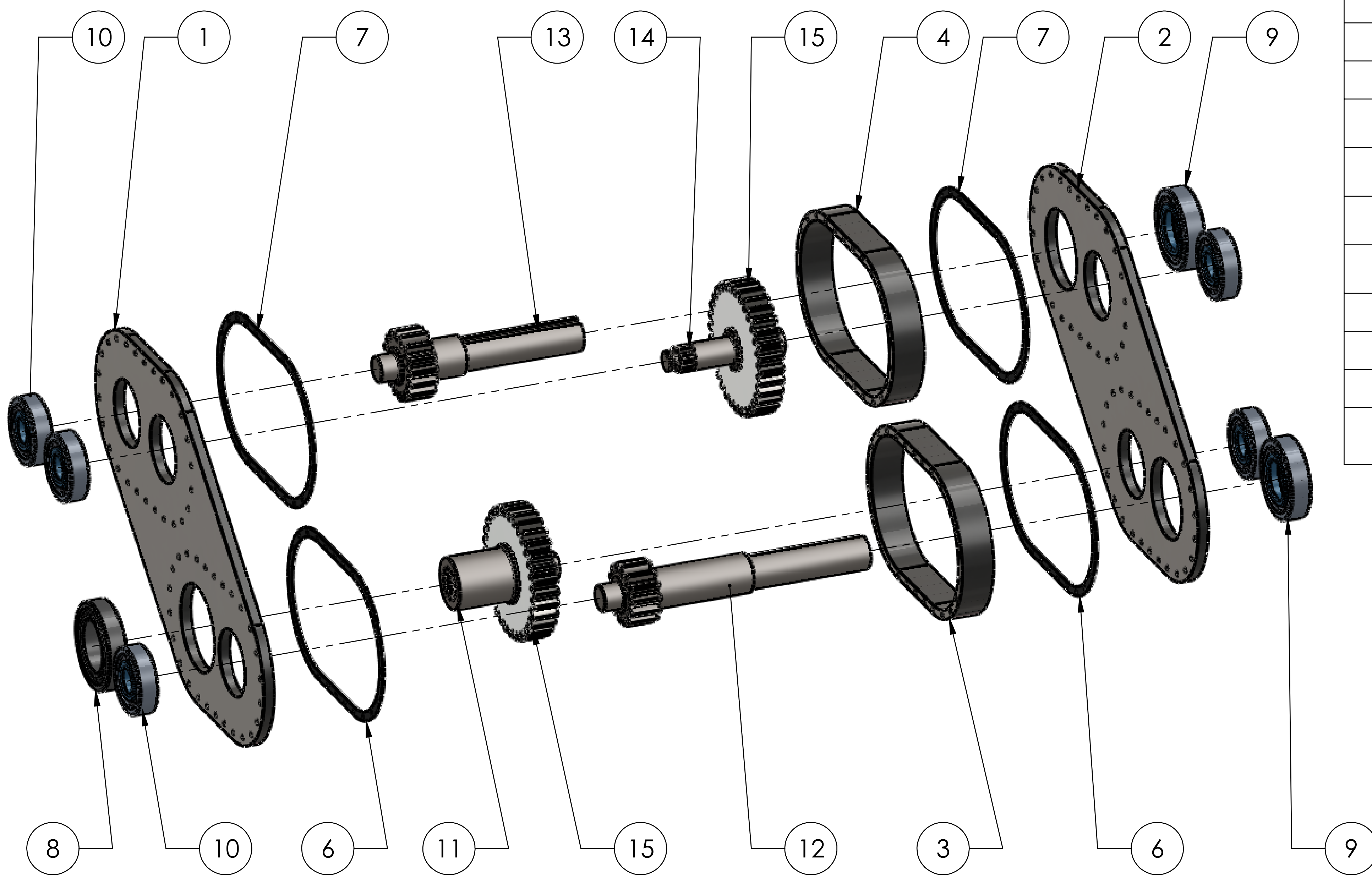
SCALE: 1:3 | Mass: 23981.60 | SHEET 2 OF 4

ITEM NO.	PART NUMBER	Manufacturer	With CVT Pulleys/QTY.
1	CVTech Drive/Driven Pulleys	CVTech	1
2	68mm Bearing Cap 1-2		1
3	52mm Bearing Cap 3-4		1
4	62mm Bearing Cap 1-2		1
5	62mm Bearing Cap 3-4		1
6	Gasket Seal 72mm		4
7	Gasket Seal 69mm		4
8	20mm O-ring (9262K639)	McMaster Carr	1
9	30mm O-ring (9262K649)	McMaster Carr	2
10	40mm O-ring (9262K661)	McMaster Carr	1
11	Hex Nut (91828A211)	McMaster Carr	52
12	Socket Head Cap Screw (91292A267)	McMaster Carr	52
13	Washer (90965A130)	McMaster Carr	104
14	GearBox (Sheet4)	McMaster Carr	1



-Item No. 8-9: O-rings
-Items without manufacturer must be custom made

Mec E 460	UNLESS OTHERWISE SPECIFIED:	DRAWN BY: **Joshua Mulder**	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA
Instructors: Lipsett & Mohtada Sadrzadeh **Fall 2017**	DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025	Group name: Mec E 460 Team 13	TITLE: Transmission Assembly
Comments:	SURFACE FINISH µm 0.6	Group number: 13	
	DO NOT SCALE DRAWING	SM By: Joshua Mulder	
MATERIAL:		Reviewed by: Set in this file	SIZE **B** / Part supplier/manufacturer / REV
FILE NAME: Full Transmission Assembly		Thursday, December 07, 2017 7:27:02 PM Friday, November 24, 2017 4:11:47 PM	SCALE: 1:5 / Mass: 23981.60 / SHEET 3 OF 4



ITEM NO.	PART NUMBER	Manufacturer	Explode 2/QTY.
1	Frame Plate 1		1
2	Frame Plate 2		1
3	Seal 1-2		1
4	Seal 3-4		1
6	Gasket Seal 69mm		4
7	Gasket Seal 72mm		4
8	40mm ID angular Contact bearing (7008-b-xl-tvp)	FAG	1
9	30mm ID 4-point contact bearing (qj206-xl-mpa)	FAG	2
10	20mm ID 4-point contact bearing (qj304-xl-mpa)	FAG	5
11	Shaft number 1		1
12	Shaft 2_with gear		1
13	Shaft 3_with gear		1
14	Shaft number 4		1
15	*32 tooth gear (A 1C 1MYK30032)	QTC Gears	2

*Gear to be modified according SDP_a_1c_1myk30032 in drawing package

-Items without manufacturer must be custom made

-Gears (Item No. 12) installed on shafts (item No 8/12) prior to this assembly - spline install
-Shafts 2 and 3 (Item No9/10) are machined with gears

Mec E 460

Instructors:
Lipsett & Mohtada Sadrzadeh
Fall 2017

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:

FILE NAME:
Full Transmission Assembly

DRAWN BY:
Joshua Mulder

Group name	Mec E 460 Team 13
Group number	13
SM By	Joshua Mulder
Reviewed by	Set in this file

Thursday, December 07, 2017 7:27:02 PM
Friday, November 24, 2017 4:11:47 PM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Transmission Assembly

SIZE	Part supplier/manufacturer	REV
B		

SCALE: 1:4 | Mass: 23981.60 | SHEET 4 OF 4