OMIKRON ROBOTICS

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April 15th, 2021

Tonya Wolfe
Chief Executive Officer
Elementiam Materials and Manufacturing
9327-75 Ave NW
Edmonton, AB T6E 1H3

RE: Modular Robotic System for Vessel Inspection and Repair- Phase III Detailed Design Report

Dear Dr. Wolfe,

Omikron Robotics is pleased to present the Phase III Report: Detailed Design for Vessel Inspection and Repair Modular Robotic System, hereby referred to as “OmiBot”.

The following topics are discussed in this report:

1. Overview of design solution
2. Detailed engineering analysis for feasibility
3. Assembly and component drawing package
4. Breakdown of cost and manufacturing estimates

Upon completion of this report, a total of 983.5 hours has been allocated to this project, resulting in an engineering cost of $88,515.

It has been a pleasure working with you on this project, and we appreciate the support and guidance you have provided us throughout this project. We are hopeful this final deliverable meets and exceeds your expectations, and that your experience with us was as excellent as ours. Should you have any questions or concerns, please direct them to our team lead, George Felobes, by email at felobes@ualberta.ca or phone, (780) 710-7095.

Sincerely,

Omikron Robotics

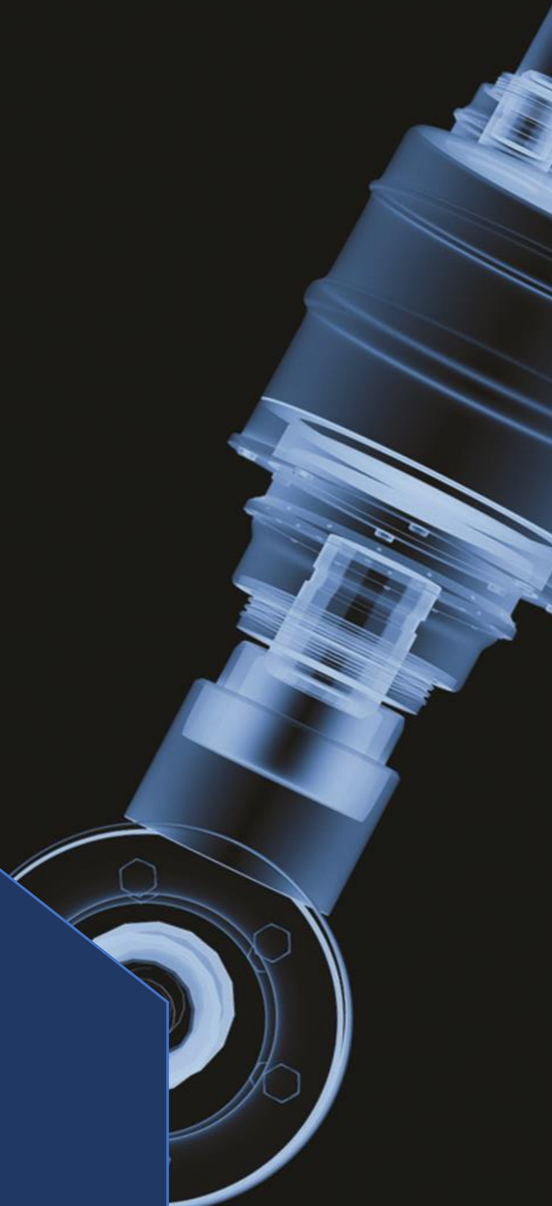
George Felobes	Liam Wolf	Eric Wong
Calvin Chen	Areej Khaddaj	Kenny Okeke

Cc: Dr. Hossein Rouhani University of Alberta
Dr. Tetsu Nakashima, University of Alberta
Dr. Thomas Lehman, Elementiam,
Remy Samson, Elementiam

Phase III

Detailed Design Report
MEC E 460 Capstone Design Project
Vessel Inspection and Repair Robot

Submission Date: April 15th, 2021



Omikron Robotics Design Team:

George Felobes
Liam Wolf
Areej Khaddaj
Kenny Okeke
Eric Wong
Calvin Chen

Executive Summary

Omikron Robotics was assigned to design a modular robot able to conduct inspection and repairs in a vessel. The robot must be able to travel on horizontal and vertical surfaces, traverse a 7.6 cm (3 in) high rib, support a 120 kg manipulator with 5 degrees of freedom and a payload of 20 kg, fit within a 45 cm diameter vessel entry, and remain rigid while conducting inspections and repairs.

The solution proposed by Omikron Robotics is the OmiBot. The OmiBot utilizes a Yaskawa SIA20F manipulator and a passive rocking wheel suspension base derived from the rocker-bogie suspension. The Omibot utilizes a total of eight neodymium magnetic wheels to traverse vertical walls, overcome vessel ribs, and provides sufficient magnetic adhesion force to remain rigid during inspections and repairs. Two linearly actuated permanent magnets are implemented in addition to the eight magnetic wheels, to provide additional adhesion for rigidity during inspections and repairs. The OmiBot consists of four modular subsystems that fit through a 45 cm diameter entryway and can be assembled with bolted connections inside a vessel. These consist of the drive system, modular chassis, manipulator, and linear actuators. The OmiBot satisfies all client specifications, however, it does not comply with all codes and standards set by CSA Z434 or the internal design specification of being waterproof and dustproof. This non-compliance has been acknowledged and accepted by the client as this design is for research and development purposes.

From Phase I, II, and III, a total of 983.5 hours distributed among 6 Junior Engineers and 1 Senior Engineering Advisor have accumulated, resulting in a total cost of $88,515. Due to unforeseen complications and iterations of the drive system design and feasibility analysis in Phases II and III, this final cost is above the original estimate of 640 hours and cost of $57,636. The OmiBot is estimated to have a total production cost of $150,000, which is within the client specified budget of $100,000 - $1,000,000.

Future work is required to develop a commercially viable product. This includes the optimization of the drive system, reduction in size and weight of the robot, the design of controls systems and instrumentation, and compliance to codes and standards set by CSA Z434. These considerations were not a priority as the main objective of the design project was to determine the technical feasibility for a vessel inspection and repair robot with a 20 kg tool capacity.

Word count: (401)

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1 Background and Design Problem

Conducting vessel inspections and repairs poses a safety risk for workers entering confined spaces. Remote robotic systems are being considered by the industry to reduce the need for human presence in these dangerous work environments (Figure 1).

Elementiam Materials and Manufacturing Inc. enlisted the help of Omikron Robotics in designing a modular robotic system capable of performing vessel inspections and repairs. The robot must have a manipulator arm with a 20 kg payload capacity and have at least 5 degrees of freedom (DOF). The robot will utilize magnetic wheels to travel horizontally, vertically, and traverse obstacles modeled as 7.6 cm (3 in) vessel ribs. The robot must remain rigid when using the manipulator during inspections and repairs. Additionally, the entryway of the vessel is limited to 45 cm in diameter; therefore, the robot must consist of modular components that fit within the entryway and can easily be assembled inside the vessel.

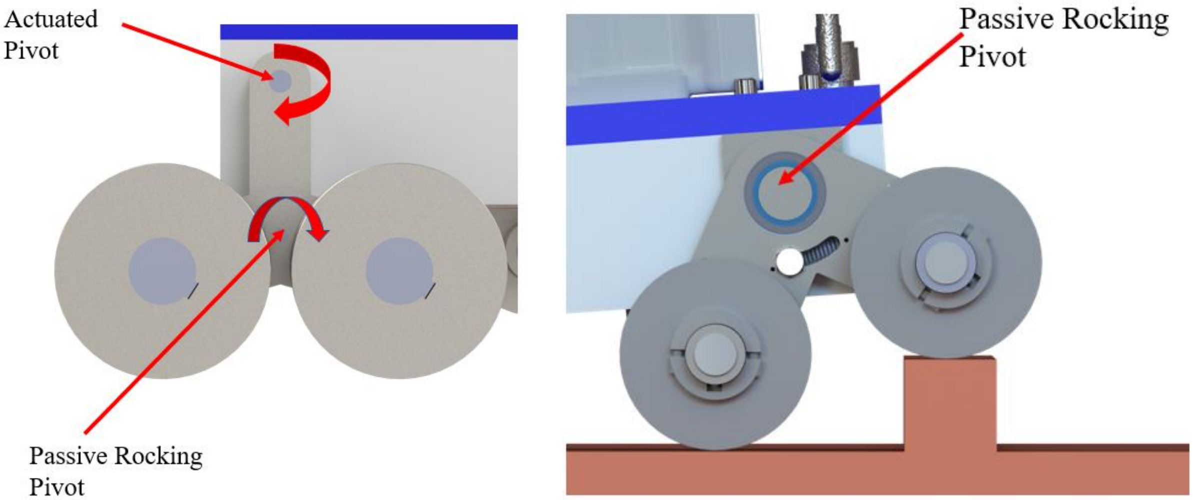
Figure 1 – Magnetic Crawler Conducting Inspections [1]

2 Design Solution

2.1 Concept Refinement

The selected design was revised at the end of Phase II to make the project feasible within the timeline of the project. The following design changes were made:

1. The actuated pivot system from the drive system was removed (Figure 2). The chassis body is no longer able to lower itself onto the vessel surface to provide rigidity during repairs, however sufficient rigidity is achieved from the 8 magnetic wheels and the deployment of permanent magnets using linear actuators.



a) b)

Figure 2 – Concept Refinement Suspension System a) Concept 2 Pivot Rocking System, b) OmiBot Passive Rocking System

2. Hoist rings were added to the platform to ensure the active winch can be connected and engaged in any orientation. The active winch will be utilized as a safety tether and actively bear a portion of the robot weight during operation.

2.2 Overview

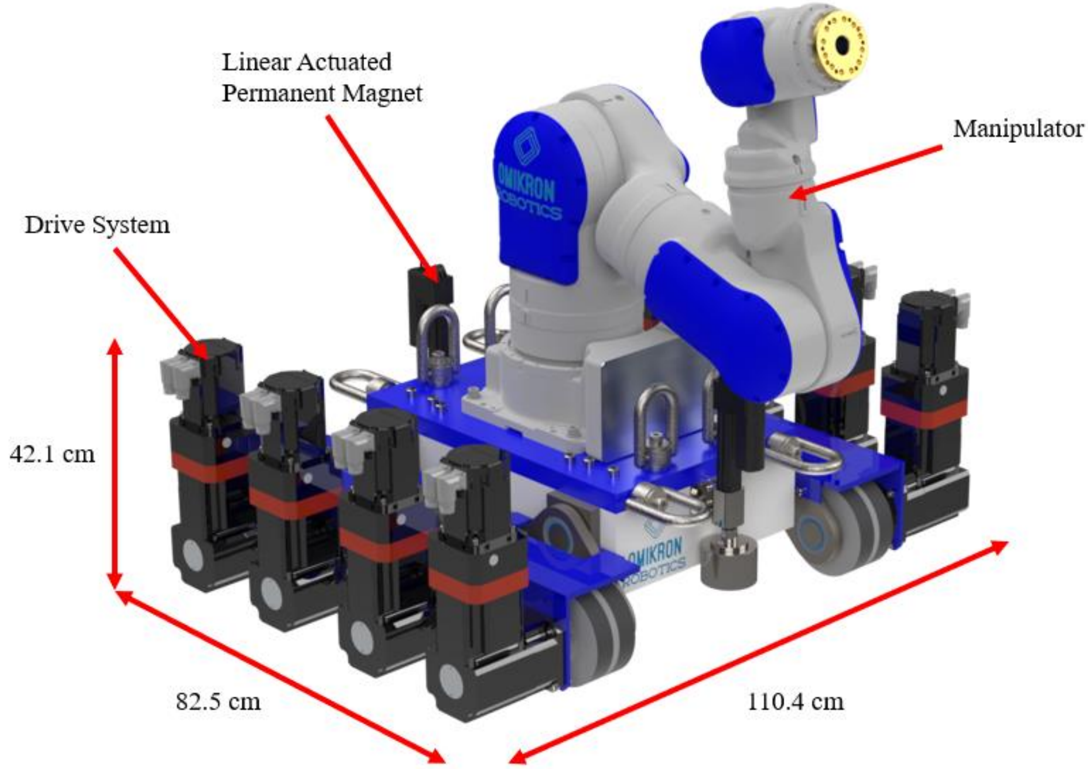


Figure 3 – Omikron Robotics OmiBot Magnetic Crawler for Vessel Inspection and Repairs.

The design solution referred to as the OmiBot, and its overall dimensions are presented in Figure 3. This robot provides a solution to each of the client's specifications. The robot supports 7 DOF manipulator with a payload capacity of 20 kg. Magnetic wheels allow the robot to adhere to the ferromagnetic vessel wall when travelling vertically without sliding or tipping. Hoist rings allow for the attachment of an active winch. Linear actuators with permanent magnets are utilized for additional rigidity. The OmiBot modular components are shown in the exploded view presented in Figure 4. The engineering drawing package developed for the OmiBot is provided in Appendix I.

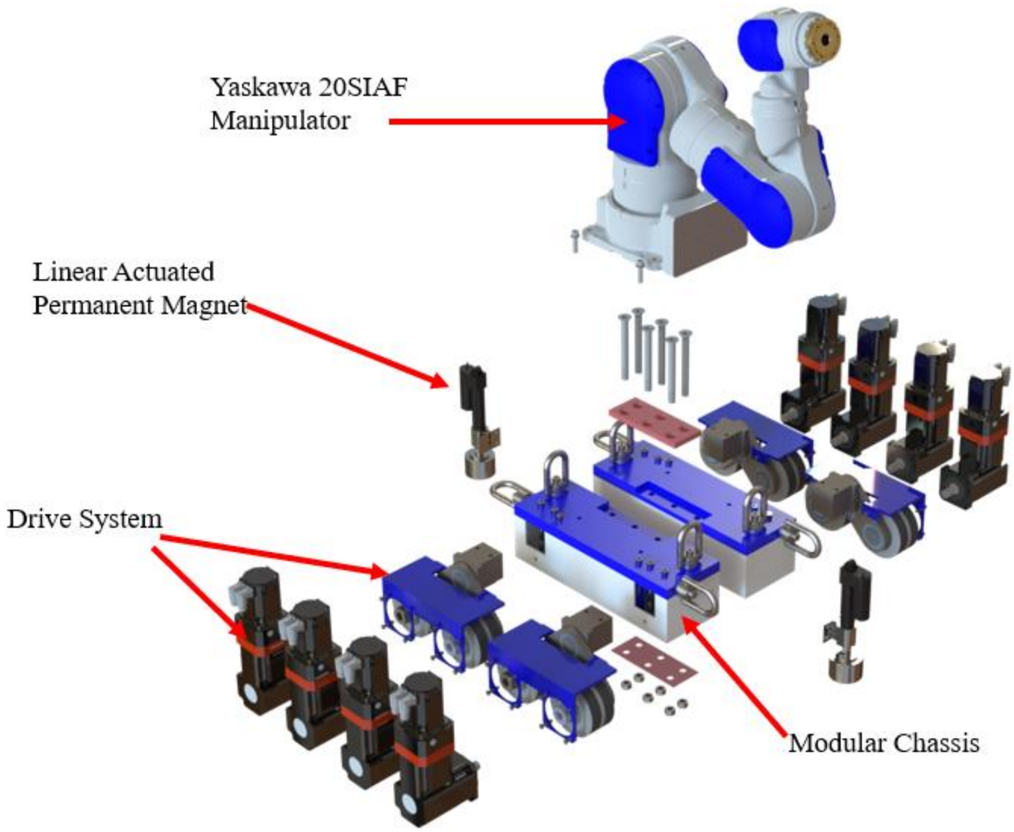


Figure 4 – Omikron Robotics OmiBot Magnetic Crawler Modular Subsystem View

Figure 5 demonstrates the ability of each subassembly to fit into the 45 cm circular vessel opening, represented by a blue circle.

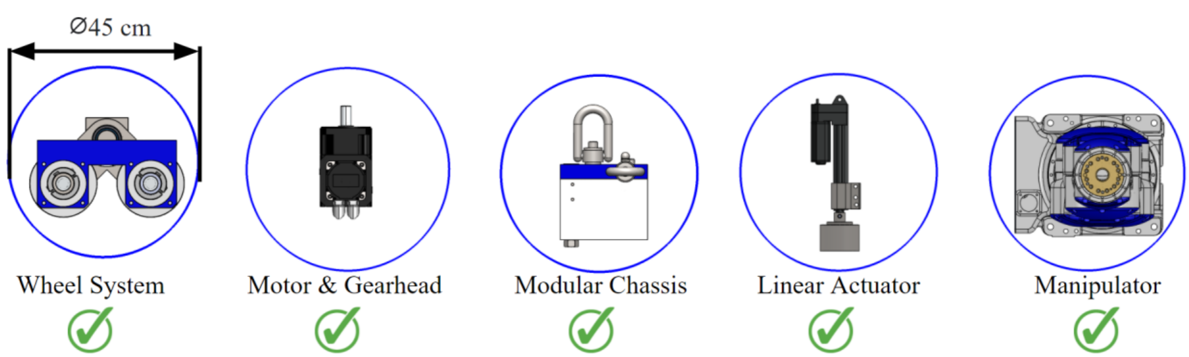


Figure 5 – OmiBot Subassembly View (parts and circles shown to scale, blue circle represents 45 cm vessel opening)

2.3 Manipulator

The chosen manipulator for the OmiBot is the Yaskawa Motoman SIA20F which features 7 DOF alongside a 20 kg payload capacity to meet the design specifications. The manipulator has a mass of 120 kg and is presented in Figure 6 along with its associated dimensions and reach. The manipulator was chosen due to its high payload-to-mass ratio when compared with competitors on the market; this is presented in Appendix B1 with additional details.

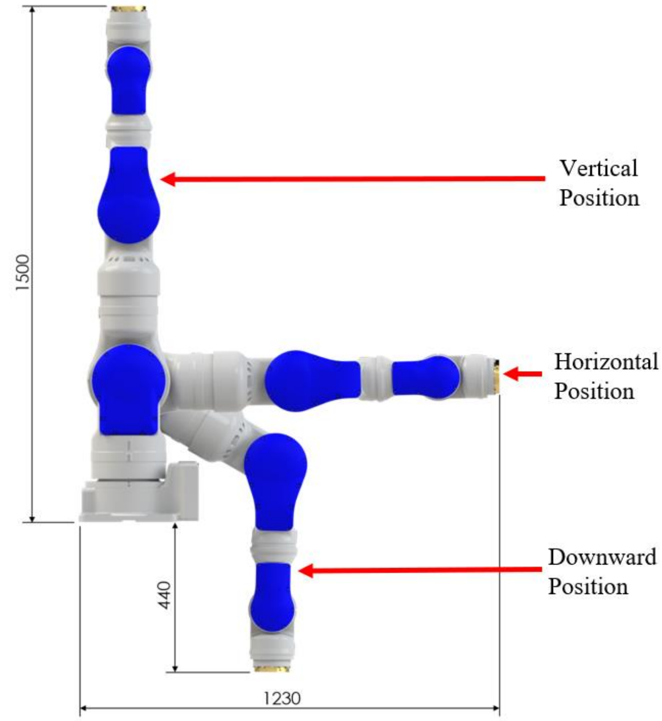


Figure 6 – Yaskawa Motoman SIA20F Manipulator (dimensions are in mm)

2.4 Chassis and Platform

The manipulator platform (top plate) and chassis both consist of two modular halves. The halves of the top plate are joined together with six bolted connections and two modular connection plates. The remaining bolt holes (Figure 7) connect the platform to the chassis and the manipulator to the platform. An operational analysis of how the chassis and platform are connected is provided in Appendix F.

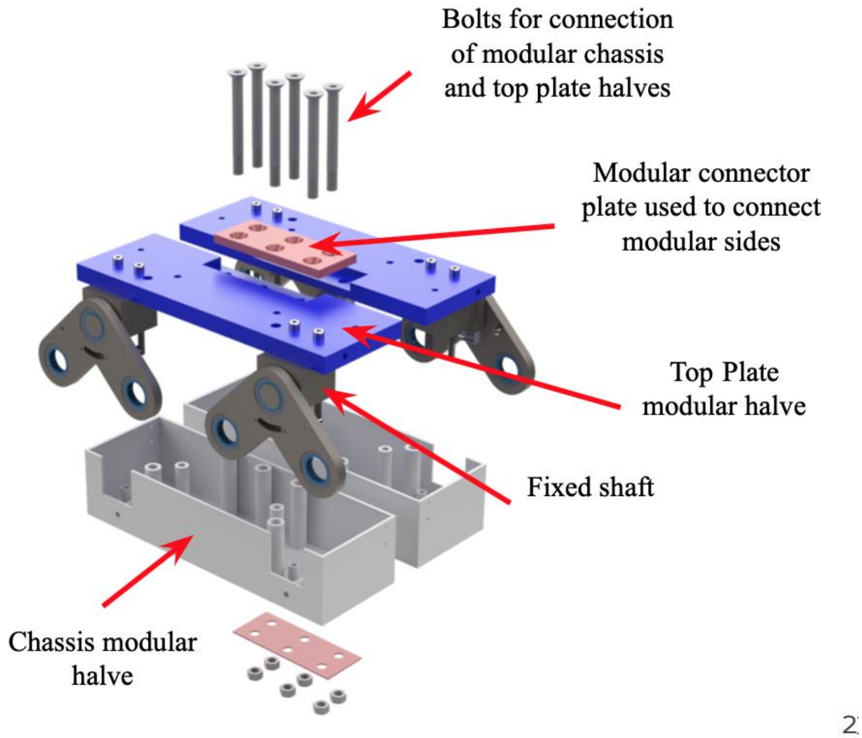


Figure 7 –OmiBot Robot Chassis and Platform

The top plate is designed to withstand the weight of the manipulator and payload, without transferring any loading to the chassis. This is achieved by connecting the fixed shafts from the drive system directly to the top plate. Therefore, the chassis is not in the load path and does not experience significant loading from the manipulator and drive system.

2.5 Drive System

The drive system can be separated into two sub-systems:

1. Pivoting suspension system
2. Power transmission

An overview of the drive system is shown in Figure 8.

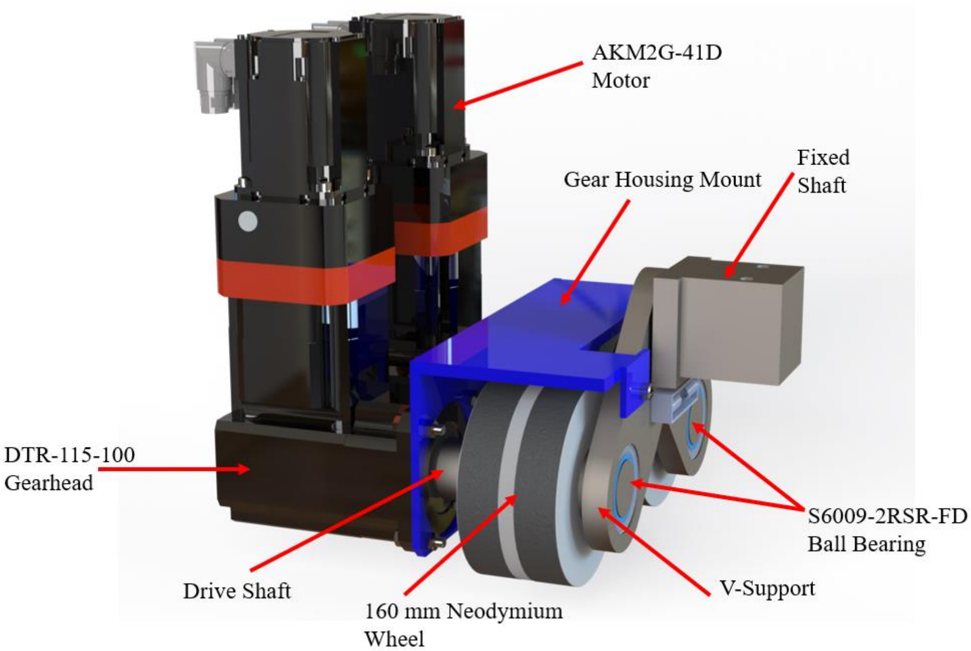


Figure 8 –OmiBot Robot Drive System

2.5.1 Pivoting Mechanism

The magnetic wheels are connected in pairs to the V-support forming a rocker mechanism, enabling each wheel pair to pivot independently. This allows the robot to maintain a minimum of 6 effective points of magnetic contact with the surface, ensuring the robot can maintain traction and avoid tipping when overcoming obstacles.

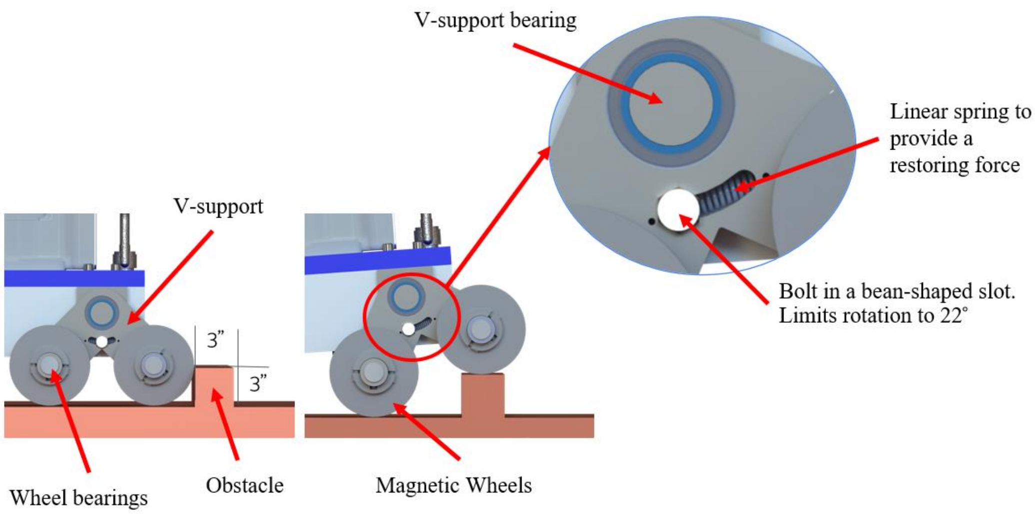


Figure 9 – Drive System Pivot Mechanism

The rocking system, shown in Figure 9, contains a slot and bolt. The bolt serves as a stopper that limits the rotation of the V-support. Additionally, springs are mounted to the V-support providing a restoring force, ensuring that the wheels return to their proper orientation and contact the surface. This is important when traversing an obstacle on a wall since gravity may hamper instead of aiding in restoring the position of the wheel pair.

2.5.2 Power Transmission

The drive system transmission consists of a right-angle planetary gear system (DuraTrue-115-100) and a servo motor (AKM2G-41D). The DuraTRUE gearhead was selected due to its low backlash (8 arc-min) and torque output (185 Nm) with a gear ratio of 100:1, exceeding the required torque of 115 Nm per wheel. The right-angle gear head is selected to minimize the length of the motor and gearhead combination on the exterior of the wheels. The gearhead includes a Redi-mount custom manufactured mount to mount any motor.

The gearhead output shaft is connected to a hollow shaft (drive shaft) and is bolted to the gearhead housing to prevent the motor and gearhead from rotating. The drive shaft transmits power to the neodymium magnetic wheels through a rounded steel key and resists axial motion with the support of a double shaft collar. Each magnetic wheel provides 1800 N of magnetic adhesion, enabling the OmiBot to adhere to vessel surfaces. To increase the friction between a neodymium wheel and a steel surface, a 0.05 mm layer of polyurethane coating will be applied to the wheel by the supplier. It is noted that a 0.05 mm polyurethane layer has a negligible affect on the magnetic force due to the thin layer. Additional information on the magnetic wheels can be found in Appendix B6.

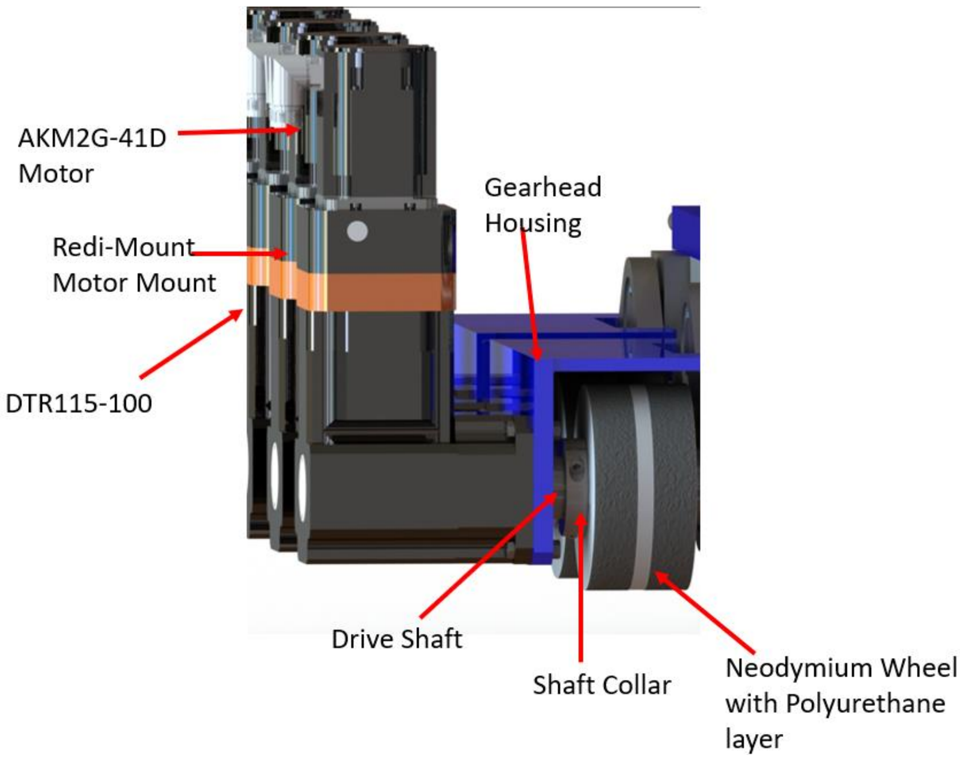


Figure 10 – Power Transmission System

The AKM2G-41D is a servo motor with a continuous output torque of 2.85 Nm with a maximum operating speed of 3000 RPM. The power curve is provided in Appendix B3. This meets the torque and operating speed requirements of the OmiBot; however, a control system will need to be implemented to reduce the total torque output of the motor and satisfy the continuous torque of the gearhead.

2.6 Linear Actuator

The OmiBot accommodates two linear actuators, each connected to a permanent magnet. These actuators are used to deploy the magnets to the vessel surface to provide an additional 1470 N of magnetic axial adhesion for added rigidity. The system can also be used as an emergency precaution as the permanent magnets do not require electrical power to adhere to the vessel walls.

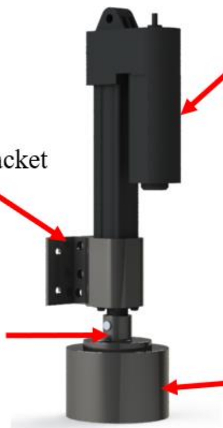


Figure 11 – Linear Actuator with Permanent Magnet for Adhesion Force

3 Engineering Analysis

Technical analysis of OmiBot can be categorized into the following categories:

1. Overall System Analysis
2. Component Analysis

The Overall System Analysis consists of the performance of the assembly while Component Analysis is concerned with the analysis of individual components in the system.

Calculations were performed for the entire assembly to determine if the OmiBot can remain stable and maintain sufficient magnetic adhesion with the vessel surface under a variety of conditions. The main objective of the stability calculations was to ensure tipping, sliding, and involuntary rolling of the wheels does not occur. Figure 12 outlines the key calculations considered for the analysis of the overall assembly.

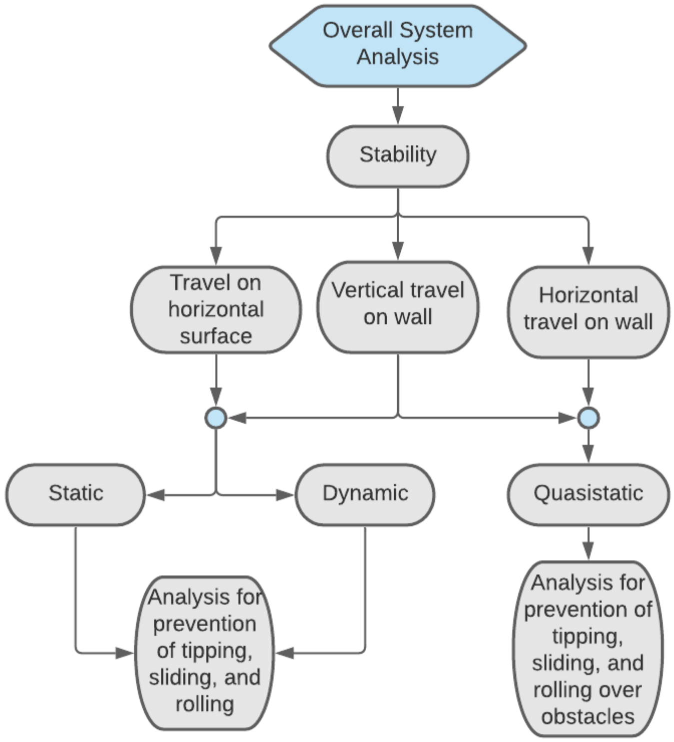
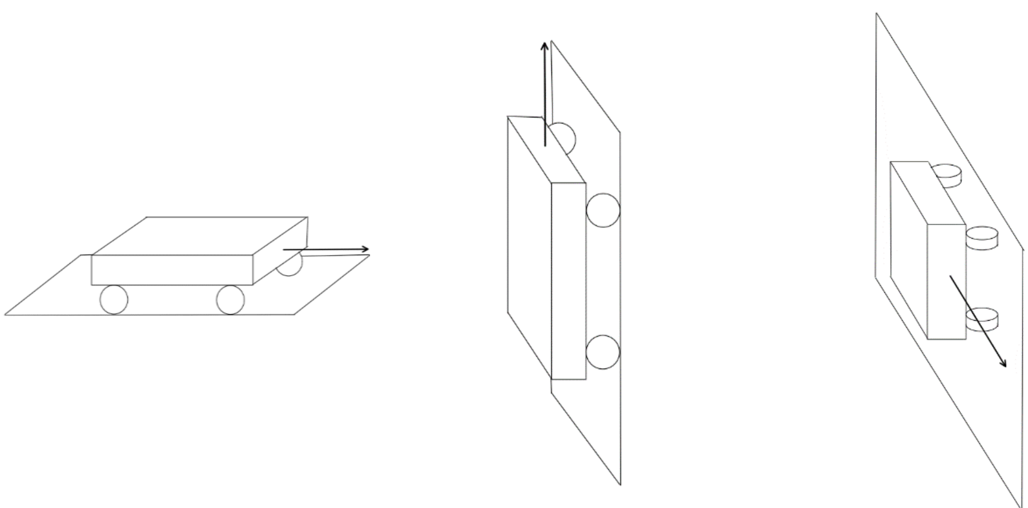


Figure 12 – Overall Analysis Flowchart

The horizontal and vertical motion of the assembly can be visualized from the simplified schematics in Figure 13.



(a) (b) (c)

Figure 13 – Critical Motion Orientations of OmiBot for Consideration in Stability Analysis a) Travel on horizontal surfaces, b) Vertical travel on wall, c) Horizontal travel on wall

Static loading refers to when the entire assembly (including the arm) is stationary, whereas dynamic loading is when the robot body is stationary while the manipulator arm moves to facilitate inspections and repairs. These two loading conditions are considered when the OmiBot is oriented as shown in Figure 13.

The quasistatic condition refers to the entire assembly moving as one rigid body, with no movement from the arm; this condition is used to assess the ability of the OmiBot to maintain stability and traction when overcoming obstacles which were determined to be negligible. The positions outlined in Figure 13 were chosen for analysis as they are the most challenging scenarios for maintaining traction with the vessel surface when the wheels are on the edge of a vessel rib. Skid steering was also analyzed, and the resulting turning rate and curvature charts are available in Appendix D.

3.1 Component Analysis

The primary objective of the component analysis was to determine the maximum induced stress on critical components of the assembly that are subjected to significant loading from the manipulator and payload. A process flow chart of the technical analysis in this section is presented in Figure 14.

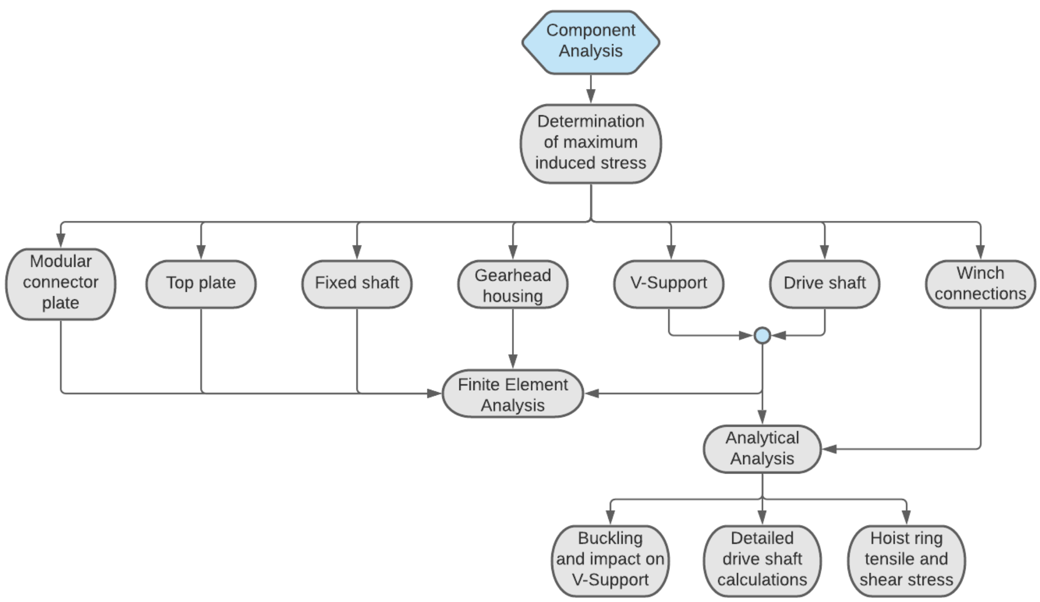


Figure 14 – Component Analysis Flowchart

Finite element analysis (FEA) was performed for complicated geometry, identified in Figure 14 as the top plate, fixed shaft, gearhead housing, and V-support. To achieve the most accurate results, the simulations were run with h-adaptive mesh convergence (Appendix E). Conservative analytic calculations assuming simplified geometry were performed for simulations where the mesh did not converge. These aided in confirming the simulation results are acceptable.

Analytic calculations were performed for the driveshaft, V-support, and winch connection. The shaft calculations determined the stress concentration and minimum diameter size at the bearing, gear shaft, and wheel position. Additionally, buckling and impact calculations were performed for the V-support to account for stresses induced when overcoming obstacles, as a supplement to static FEA. The winch connection is a vendor-supplied component, and therefore a purely analytical analysis was performed to verify its tensile and shear stress is not exceeded at various connection angles to the OmiBot. Conservative calculations were conducted to ensure the bolted connections would not fail. Calculations can be found in Appendix D.

3.2 Technical Analysis Summary

A summary of the engineering analysis results to determine the physical feasibility of the OmiBot is provided in Table 1. The total power consumption is also included, which is the maximum required for operation of the manipulator, motors, and linear actuators. This was calculated using data provided by the vendors.

Table 1 - Summary of Engineering Analysis Results

Stability of Overall System	Driving Vertically on a Wall - Tipping	Driving Horizontally on a Wall - Tipping	Driving Vertically Over an Obstacle on Wall - Slipping	Driving Horizontally Over an Obstacle on Wall - Slipping
Min. Reaction Force at Wheels (N)	612	1116	N/A	N/A
Max. Static Friction (N)	N/A	N/A	5597	5400
Traction Safety Factor	N/A	N/A	1.14	1.10
Magnetic Force Provided by Wheels Normal to Surface (N)	1800	1800	N/A	N/A
Comments	No Tipping	No Tipping	No Sliding	No Sliding

Strength Analysis of Critical Components	Drive Shaft	Fixed Shaft	V-Support	Top Plate	Gearhead housing	Winch Connection	Modular Connection
Maximum Stress (MPa)	125	80.1	98.8	6.89	270	N/A	13.9
Min. Safety Factor	3.6	5.7	4.9	67	1.7	2	33

Torque Analysis	Driving	Braking
Torque Required per Wheel (Nm)	115	96.3

Power Consumption (KW)	6.66

4 Cost Analysis

The total cost of the OmiBot is $150K, with the costs associated with each subsystem shown in Figure 15. The total cost consisting of vendor and component manufacturing processes are detailed in Appendix C. The vendor costs are retrieved from vendor quotes provided in Appendix C, excluding the motor cost which is an analogous estimate based on similar motors in the market. The component manufacturing costs were determined from a Xometry quote (Appendix C9), assuming all machined components are CNC machined from 4130 steel. The client specified budget for the OmiBot is between $100K and $1M, and the design team-specific budget was $150,000. Therefore, the OmiBot is compliant with both the external and internal budget.

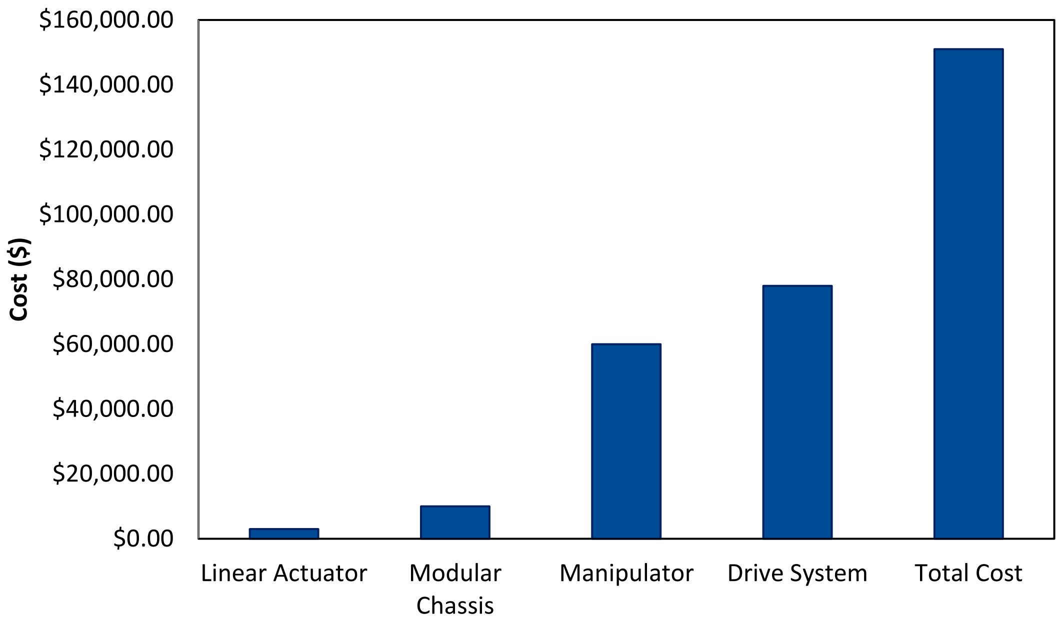


Figure 15 – OmiBot Subsystem and Total Costing

5 Additional Design Considerations

5.1 Sustainability

All machined components are made from AISI steel. The American steel industry is known to be the cleanest and most energy-efficient producer of steel in the world [2]. In contrast to China, steel produced from the U.S uses 50% less energy and is 2.5 times lower in CO2 emissions per ton of steel produced [2]. Steel is 100% recyclable and can continue to be reused without deterioration in product quality. This is evident from the emission intensity chart presented in Figure 16. The modularity of the robot also enables machined components to be easily disassembled for maintenance and reuse, forgoing the need to machine a new part. Commercially sourced parts such as the manipulator and wheels can also be disassembled and used in other applications both for industrial and research purposes.

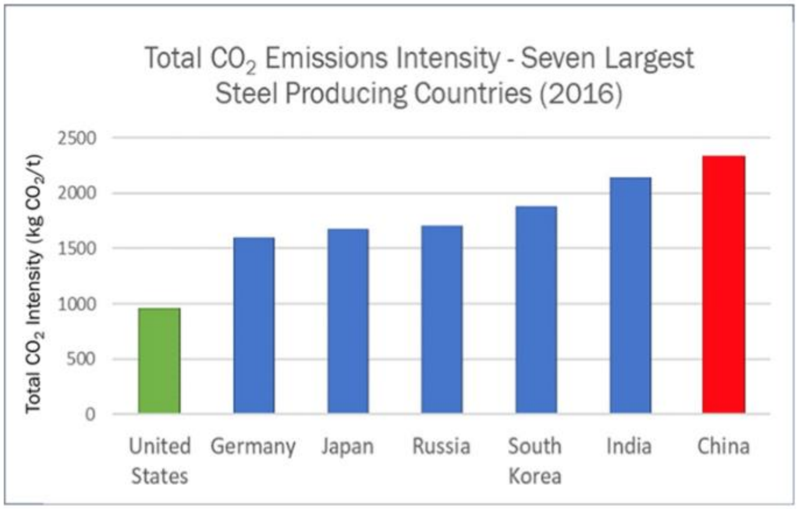


Figure 16 – Steel Production Emissions by Countries [2]

5.2 Legality

The robot is designed for research and development purposes according to the client design specifications. Technical attributes such as limiting the linear velocity to 250 $\frac{mm}{s}$, and coverage of any moving parts in the drive system, are met for CSA Z434. Adherence to additional codes and standards outlined in CSA and ISO regulatory bodies must be considered in future upgrades to the design.

5.3 Manufacturing

Designed components will be manufactured with CNC machining with exception to the shafts, where round portions will need to be lathed and non-circular cross-sections machined. Additionally, the chassis will be comprised of sheet metal which can be folded and welded.

To ensure components fit properly, additional machining will be required after components are delivered. This is because the robot is designed in modular halves, and thus the halves must be connected and machined to reduce variability. Furthermore, the support rods welded to the chassis will require post-machining as they may not perfectly align with their respective holes after welding.

Additional considerations are to trim the bottom edges of the top plate to create a landing for the chassis, and to either create contact pads or shim the interface between the fixed shaft and the top plate to ensure wheel pairs settle at the same height.

6 Design Compliance Matrix

Omikron Robotics designed OmiBot to the design specification matrix presented in Phase I and Phase II reports. Several changes to the design specifications were made in consultations with the client and are outlined in Table 2.

Table 2 - Summary of Design Specification Changes

Specification	Description	Changes
B5	Reduced Vibrations	Removed after consultation with the client
C2	Payload Connection Method	Removed after consultation with the client
E1,3,4	Safety Code, Intrinsic Safety, and Emergency Stop	Weighting was lowered from a 5 (must) to 3 (should) due to conversations with the client; the robot is intended for R&D purposes to determine the feasibility of a magnetic crawler with a 20 kg tool payload

Overall, the OmiBot is compliant with the specifications laid out in the design compliance matrix, except for safety codes, intrinsic safety, emergency stop, and IP rating (waterproof and dustproof rating) of the robot. The specifications of safety codes, intrinsic safety, and emergency stop are outlined by CSA Z434 and are governed by codes and standards. The client has acknowledged the non-compliant specifications and continues to support and endorse the design.

The Design Compliance Matrix is presented on the following page and the client-approved matrix is presented in Appendix A.

Table 3 - Design Compliance Matrix

Requirement Number	Design Function Consideration	Design Specification/Requirement	Design Authority	Weighting (5 for Must, 1 for Low-Priority Preference)	Phase 3 Rating (Compliant, Non-Compliant)	Design Solution
A	Overall Design and Dimensions					
A1	Robot Weight	Point load stress must not exceed the yield stress of the steel vessel walls (300 MPA)	Client	4	C	Compliant: The mass of the robot is 500 kg. The robot weight is equally distributed among 8 wheels; however, each wheel is assumed to support 120 kg as a conservative approach. Assuming a conservative point load of 1mm, this corresponds to a point stress of 11.8 MPa, which is less than the yield strength of steel (300 MPa)
A2	Module Size	Module size must be less than 45 cm in diameter to fit in the vessel entrance. The design should be small enough to be transported in a F150 truck.	Client	5	C	Compliant: The total dimension of the robot is LxWxH: 110.4 x 82.5 x 42.1 cm^3. This is smaller than a bed of a F-150 truck, L x W: 170.4 x 128.5 cm^2 (67.1 x 50.6 in^2). The robot contains total 11 modular components which are less than 45 cm in diameter. The modular subsystems are the following: 1. Two modular chassis halves 2. Two linear actuators 3. Four pivoting drive system 4. Manipulator 5. Two modular connection plates
B	Functionality					
B1	Payload Specifications	Robotic platform must be able to carry a payload of 20 kg. This payload can be approximated as a 30 x 30 x 30 cm^3 cube.	Client	5	C	Compliant: The platform is designed to support the manipulator dynamics with a 20kg payload attached. The most conservative loading condition was used.
B2	Manipulator Degrees of Freedom	The manipulator ~~must~~ should have at least 5 DOF (including producing an incident angle of 15-30 degrees) and be able to work on a surface of 10 cm by 10 cm.	Client	5	C	Compliant: The manipulator selected is the Yaskawa SIA20F, which has 7 degrees of freedom and can provide the required incident angle from the end effector.
B3	Platform Degrees of Freedom	The platform must have 3 DOF (vertical, horizontal, and rotational).	Client	5	C	Compliant: The drive system includes 8 motors for 8 wheels. Vertical and horizontal travel is achieved with the magnetic wheels adhering to the vessel surface. The rotational degree of freedom is achieved with skid steering.
B4	Overcome Obstacles	Platform must be able to overcome a 3" x 3" cube obstacle.	Client	5	C	Compliant: The robotic system utilizes a pivoting rocking system. This provides a physical suspension that will ensure the wheels are always in contact with the surface when traversing obstacles.
~~B5~~	~~Reduced Vibrations~~	~~Vibrations must be damped when the robot is moving such that it will not interfere with modular sensors.~~	~~Client~~	4	~~NC~~	Specification was revised and removed.
B6	Tool Dynamics	The robot must have a system to counteract ~~tool~~ manipulator dynamics.	Client	5	C	Compliant: The robot has a low center of mass and a wide chassis frame, with sufficient magnetic adhesion, preventing tipping of the robot with a varying COM caused by manipulator dynamics.
B7	Flat horizontal and vertical surface travelling	The platform must be able to travel on vertical and horizontal surfaces.	Client	5	C	Compliant: The robot is designed to the most conservative case of vertical travel. The magnetic adhesion force and friction coefficient is sufficient to ensure the robot does not slip or tip on a 90-degree surface.
B8	Adhesion during tool operation	The robotic platform must become stable when performing inspections and repairs and have vibrations damped.	Client	4	C	Compliant: The 8-wheel configuration provides enough magnetic adhesion to allow the robot to remain rigid. The addition of the linear actuated permanent magnets provides additional adhesion forces to provide additional safety factor.
B9	Platform Velocity	Platform velocity must be less than 250 mm/s. The client specified the platform velocity should travel at 25.4 mm/s.	Code/Client	5	C	Compliant: The motor and gear ratio selected can provide a maximum linear velocity of 250 mm/s. Additional control system will be required to regulate the motor output which (out of scope).
B10	Manipulator Velocity	The manipulator must operate at 12.7 – 55.0 mm/s.	Client	4	C	Compliant: The Yaskawa SIA20F includes control systems to operate the manipulator within the specified range.
B11	Terrain	The robot will be designed to the assumption that the vessel curvature is small, and walls can be considered flat.	Client	4	C	Compliant: The robot is designed assuming all 8 wheels are on a flat surface.
C	Assembly					

C1	Attachments Modules	Robot parts and attachments may be modular (if greater than 45 cm). The robot may need to be assembled and disassembled inside the vessel.	Client	4	C	Compliant: The modular subsystems utilize bolted connections to allow assembly and disassembly inside and outside the vessel.
~~C2~~	~~Payload Connection Method~~	~~Reusable quick connect device to install different sensors or tools~~	~~Client~~	~~3~~	~~NC~~	Specification was revised and removed.
D	Operations					
D1	Wireless/Wired Power source	Wired tether voltage source	Client	1	C	Design constraint
D2	Environment Conditions	Vessel will be drained and cleaned before inspection. Atmospheric conditions will be present.	Client	5	C	Design constraint
D3	Operating Condition	Working temperature of the robot will be ~~-40 ºC~~ 0 ºC to +40 ºC.	Client	4	C	Compliant: all components are rated to a working temperature of 0 ºC to 40 ºC.
D4	Remote Operation	The robot may be remote operated with either a joystick control or computer system.	Client	3	C	Design constraint
D5	Waterproof and Dust Resistance	The robot should be design to an IP65 rating to resist low pressure water jet and resistance to dust and particles.	Omikron	3	NC	Non-Compliant: Vendor supplied items (motor, gear housing, linear actuator, and manipulator) are rated to at least IP65. The chassis design with bolted connections is not rated to IP65 and will required additional design to achieve compliance) This was a self-imposed design team specification:
D6	Life Cycle	Expected life cycle is one year with minimal maintenance.	Client	3	C	Compliant: Components are designed for a life cycle of 1 year.
E	Safety					
E1	Safety Code	Must comply with Canadian or Provincial safety code.	Governing Safety Institution	~~5~~ 3	NC	Non-Compliant: The robotic system is designed for R&D purposes; therefore, the system was designed to meet the client specifications. Codes and standards such as limited linear velocity and exposed moving components are met, however, additional codes and standards are not met.
E2	Winch Control	The platform must be designed such that ~~a passive~~ an active winch will secure the entire platform to prevent it from falling and must support the platform weight	Client	3 3	C	Compliant: The winch connection is designed to support 500 kg. The platform includes 8 winch connections. This allows the winch to have multiple configurations.
E3	Intrinsic Safety	Materials must be selected such that it does not retain an electrostatic charge for certain applications where electrostatic charges are a hazard	CSA Z434 / ISO 12100 6.2.11.5	~~5~~ 3	NC	Non-Compliant: The robotic system is designed for R&D purposes; therefore, the system was designed to meet the client specifications. Even though there are no components that retain electrostatic charge, further analysis would be required on such a standard.
E4	Emergency Stop	Emergency stop is required when humans are working around the robot and is independent of the power supply.	CSA Z434	~~5~~ 3	NC	Non-Compliant: The robotic system is designed for R&D purposes; therefore, the priority was given to meeting the client specifications to prove the feasibility of robot of this size, and mass. Note that even though there is not an emergency braking system, controls can be designed to actuate the linear actuators upon power loss to prevent the rolling of the vehicle. Linear actuators have a permanent magnet, and thus, does not need power supply. The magnetic wheels are made from Neodymium permanent magnets, which makes the stability of the robot independent of the power supply.
F	Manufacturing					
F1	Ease of Manufacture	The materials of the robot may be easily sourced to either produce only one unit or batch produce twelve.	Client	3	C	Compliant: The load bearing components (shafts, platform and support) uses 4130 normalized steel, the chassis body is not load bearing and is manufactured with sheet metal.
F2	Production Volume	Maximum production volume of 12 units.	Client	1	C	Compliant: Omikron sourced the cost for a production of 1 unit.
G	Project Management					
G1	Schedule	Schedule of the project will follow the milestones: Phase 1: February 3rd, 2021 Phase 2: March 4h, 2021 Phase 3: April 15th, 2021	Mec E 460	5	C	Compliant
G2	Budget	The client specified a project budget between $100k - $1M. The design team will design a robotic system to meet the market cost of $150,000 for one unit.	Client	3	C	Compliant: The cost of one unit is 150k, this is within the client specified budget of $100k - $1M and is compliant with the design teams stated budget of $150k.

7 Future Work

Future work is required to develop a commercially viable product. This includes optimizing the drive system such that fewer motors/gear pairs are used and introducing a control system to regulate the motor output. The system's geometry and mass can also be optimized, and electrical and instrumentation design will be required for the remote operation of OmiBot. Prototyping must be conducted to ensure the system can operate safely.

8 Project Management

In Phase III, a total of 508 hours were spent, resulting in 216 hours more than the original baseline, and 146 hours more than the revised estimate from Phase II (Table 4). The total project cost is $88,515, which is approximately $31,000 more than the original baseline and is larger than the 10% originally added. This is largely due to the unforeseen complications in the iterative design of the robot and suspension system in Phases II and III and have been approved by the client. Omikron Robotics acknowledges going over budget; however, due to the research nature of the design project, it is taken as a learning opportunity for better future estimations.

A comparison between the baseline, revisited and actual hours, and cost are shown in Figure 17 and Figure 18, respectively. Appendix I contains the individual timesheets, Phase III hours distribution. and the Gantt chart.

Table 4 - Summary of Actual and Baseline Junior Time and Cost for Phase I, II, and III

	Junior time (hr.)		Junior Cost ($)	
	Baseline	Actual	Baseline	Actual
Phase 1	138.0	135	$12,420.00	$12,150.00
Phase 2	211	341	$18,990.00	$30,690.00
Phase 3	291.4	507.5	$26,226.00	$45,675.00
Total	640.4	983.5	$57,636.00	$88,515.00

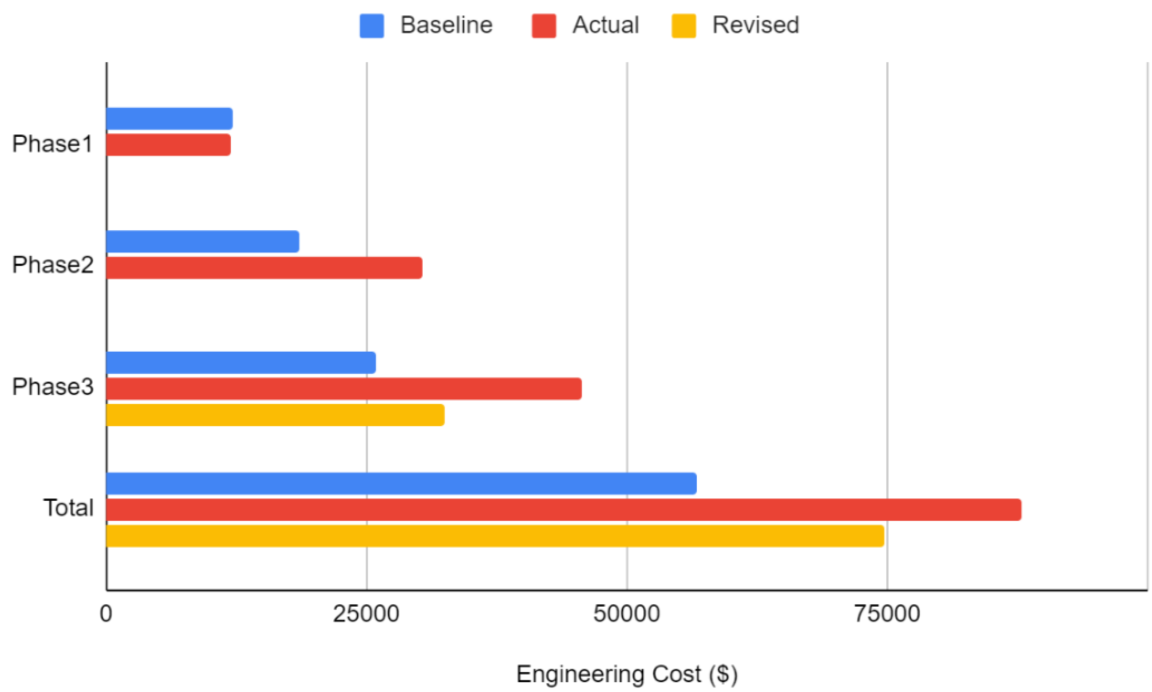


Figure 17 – Omikron Robotics Actual, Baseline and Revised Junior Cost

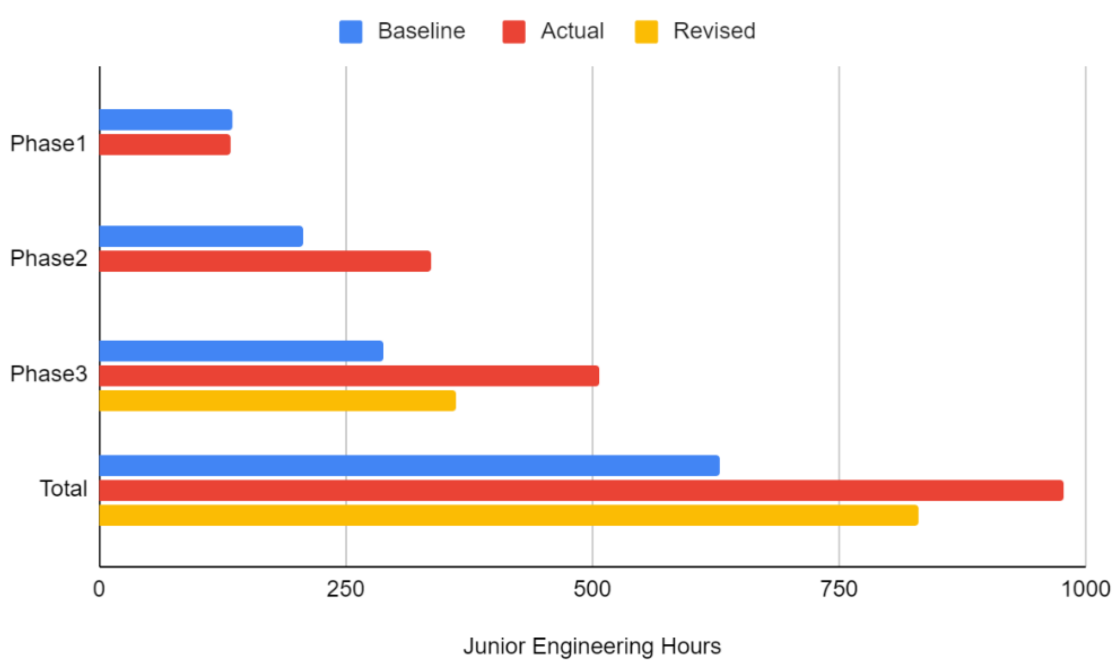


Figure 18 – Omikron Robotics Phase III Actual, Baseline and Revised Junior Time Allocation

9 Conclusion

Omikron Robotics provided an overview of the technical feasibility of a magnetic vessel/inspecting robot, which was the primary goal of the client. The OmiBot can maneuver horizontal and vertical surfaces with 3 DOF, traverse a 7.6 cm (3 in) rib, support a commercial manipulator with a 20 kg payload and at least 5 DOF, incorporate a modular design to fit within a 45 cm diameter opening and for easy assembly within the vessel, and become rigid during inspections and repairs. To produce a commercially viable product, future work must be completed to optimize the drive system, geometry of the chassis, and compliance with the codes and standards of CSA Z434. Non-compliant specifications have been approved by the client. The Junior Engineer time spent on this project was 983.5 hours, resulting in a cost of $88,515.00.

10 References

[1] "Remote Inspection with Robotic Systems Inspections without human entry in Confined spaces | IRISNDT", Irisndt.com, 2021. [Online]. Available: https://www.irisndt.com/us/robotic-remote-inspection/.

[2] Hasanbeigi, A. and Springer, C. 2019. *How Clean is the U.S. Steel Industry? An International Benchmarking of Energy and CO2 Intensities*. San Francisco CA: Global Efficiency Intelligence.

Appendix A: Design Compliance Matrix

The design specifications were updated as more information was provided. Major revision to the specifications from Phase II is the removal of a damping vibrations during motion, reduction of the 25 kg manipulator payload to 20 kg, and the use of an active winch instead of a passive winch. Phase III revisions revised the weighting from a must (5) to a should (3) of the adherence to codes and standards as the main scope of the robot is to the feasibility of a magnetic crawler. The signed Design Compliant Matrix is presented on the following pages.

Requirement Number	Design Function Consideration	Design Specification/Requirement	Design Authority	Weighting (5 for Must, 1 for Low-Priority Preference)	Phase 3 Rating (Compliant, Non-Compliant)	Design Solution
A	**Overall Design and Dimensions**					
A1	Robot Weight	Point load stress must not exceed the yield stress of the steel vessel walls (300 MPA)	Client	4	C	Compliant: The weight of the robot is 500 kg. The robot weight is equally distributed among 8 wheels, however each wheel is assumed to support 120 kg as a conservative approach. Assuming a conservative point load of 1mm, this corresponds to a point stress of 11.8 MPa, which is less than the yield strength of steel (300 MPa)
A2	Module Size	Module size must be less than 45 cm in diameter to fit in the vessel entrance. The design should be small enough to be transported in a F150 truck.	Client	5	C	Compliant: The total dimension of the robot is LxWxH: 110.4 x 82.5 x 42.1 cm^3. This is smaller than a bed of a F-150 truck , L x W: 170.4 x 128.5 cm^2 (67.1 x 50.6 in^2). The robot contains total 11 modular components which are less than 45 cm in diameter. The modular subsystems are the following: 1. Two modular chassis halves 2. Two linear actuator 3. Four pivoting drive system 4. Manipulator 5. Two modular connection plates
B	**Functionality**					
B1	Payload Specifications	Robotic platform must be able to carry a payload of 20 kg. This payload can be approximated as a 30 x 30 x 30 cm^3 cube.	Client	5	C	Compliant: The platform is designed to support the manipulator dynamics with a 20kg payload attached. The most conservative loading condition was used.
B2	Manipulator Degrees of Freedom	The manipulator ~~must~~ should have at least 5 DOF (including producing an incident angle of 15-30 degrees) and be able to work on a surface of 10 cm by 10 cm.	Client	5	C	Compliant: The manipulator selected is the Yaskawa SIA20F, which has 7 degrees of freedom and can provide the required incident angle from the end effector.
B3	Platform Degrees of Freedom	The platform must have 3 DOF (vertical, horizontal, and rotational).	Client	5	C	Compliant: The drive system includes 8 motors for 8 wheels. The rotational degree of freedom is achieved with skid steering.
B4	Overcome Obstacles	Platform must be able to overcome a 3" x 3" cube obstacle.	Client	5	C	Compliant: The robotic system utilizes a pivoting rocking system. This provides a physical suspension that will ensure the wheels are always in contact with the surface when traversing obstacles.
~~B5~~	~~Reduced Vibrations~~	~~Vibrations must be damped when the robot is moving such that it will not interfere with modular sensors.~~	~~Client~~	~~4~~	~~NC~~	Specification was revised and removed.
B6	Tool Dynamics	The robot must have a system to counteract ~~tool~~ manipulator dynamics.	Client	5	C	Compliant: The robot has a low center of mass and a wide chassis frame, with sufficient magnetic adhesion, preventing tipping of the robot with a varying COM caused by manipulator dynamics.
B7	Flat horizontal and vertical surface travelling	The platform must be able to travel on vertical and horizontal surfaces.	Client	5	C	Compliant: The robot is designed to the most conservative case of vertical travel. The magnetic adhesion force and friction coefficient is sufficient to ensure the robot does not slip or tip on a 90-degree surface.
B8	Adhesion during tool operation	The robotic platform must become stable when performing inspections and repairs and have vibrations damped.	Client	4	C	Compliant: The 8 wheel configuration provides enough magnetic adhesion to allow the robot to remain rigid. The addition of the linear actuated permanent magnets provide additional adhesion forces to provide additional safety factor.
B9	Platform Velocity	Platform velocity must be less than 250 mm/s. The client specified the platform velocity should travel at 25.4 mm/s.	Code/Client	5	C	Compliant: The motor and gear ratio selected can provide a maximum linear velocity of 250 mm/s. Additional control system will be required to regulate the motor output which (out of scope).
B10	Manipulator Velocity	The manipulator must operate at 12.7 – 55.0 mm/s.	Client	4	C	Compliant: The Yaskawa SIA20F includes control systems to operate the manipulator within the specified range.
B11	Terrain	The robot will be designed to the assumption that the vessel curvature is small, and walls can be considered flat.	Client	4	C	Compliant: The robot is designed assuming all 8 wheels are on a flat surface.
C	**Assembly**					
C1	Attachments Modules	Robot parts and attachments may be modular (if greater than 45 cm). The robot may need to be assembled and disassembled inside the vessel.	Client	4	C	Compliant: The modular subsystems utilizes bolted connections to allow assembly and disassembly inside and outside the vessel.
~~C2~~	~~Payload Connection Method~~	~~Reusable quick connect device to install different sensors or tools~~	~~Client~~	~~3~~	~~NC~~	Specification was revised and removed.
D	**Operations**					
D1	Wireless/Wired Power source	Wired tether voltage source	Client	1	C	Design constraint
D2	Environment Conditions	Vessel will be drained and cleaned before inspection. Atmospheric conditions will be present.	Client	5	C	Design constraint

D3	Operating Condition	Working temperature of the robot will be ~~-40 ºC~~ 0 ºC to +40 ºC.	Client	4	C	Compliant: all components are rated to a working temperature of 0 ºC to 40 ºC.
D4	Remote Operation	The robot may be remote operated with either a joystick control or computer system.	Client	3	C	Design constraint
D5	Waterproof and Dust Resistance	The robot should be design to an IP65 rating to resist low pressure water jet and resistance to dust and particles.	Omikron	3	NC	Non-Compliant: Vendor supplied items (motor, gear housing, linear actuator and manipulator) are rated to at least IP65. The chassis design with bolted connections is not rated to IP65 and will required additional design to achieve compliance) This was a self-imposed design team specification:
D6	Life Cycle	Expected life cycle is one year with minimal maintenance.	Client	3	C	Compliant: Components are designed for a life cycle of 1 year.
E			**Safety**			
E1	Safety Code	Must comply with Canadian or Provincial safety code.	Governing Safety Institution	~~5~~ 3	NC	Non-Compliant: The robotic system is designed for R&D purposes, therefore the system was designed to meet the client specifications. Codes and standards such as limited linear velocity and exposed moving components are met, however, additional codes and standards are not met.
E2	Winch Control	The platform must be designed such that ~~a passive~~ an active winch will secure the entire platform to prevent it from falling and must support the platform weight	Client	3 3	C	Compliant: The winch connection is designed to support 500 kg. The platform includes 8 winch connections. This allows the winch to have multiple configurations.
E3	Intrinsic Safety	Materials must be selected such that it does not retain an electrostatic charge for certain applications where electrostatic charges are a hazard	CSA Z434 / ISO 12100 6.2.11.5	~~5~~ 3	NC	Non-Compliant: The robotic system is designed for R&D purposes, therefore the system was designed to meet the client specifications. Even though there are no components that retain electrostatic charge, further analysis would be required on such a standard.
E4	Emergency Stop	Emergency stop is required when humans are working around the robot and is independent of the power supply.	CSA Z434	~~5~~ 3	NC	Non-Compliant: The robotic system is designed for R&D purposes, therefore the priority was given to meeting the client specifications to prove the feasibility of robot of this size, and mass. Note that even though there isn't an emergency breaking system, controls can be designed to actuated the linear actuators upon power loss to prevent the rolling of the vehicle. Linear actuators have a permanent magnet, and thus, does not need power supply. The magnetic wheels are made from Neodymium permanent magnets, which makes the stability of the robot independent of the power supply.
F			**Manufacturing**			
F1	Ease of Manufacture	The materials of the robot may be easily sourced to either produce only one unit or batch produce twelve.	Client	3	C	Compliant: The load bearing components (shafts, platform and support) uses 4130 normalized steel, The chassis body is not load bearing and is manufactured with sheet metal.
F2	Production Volume	Maximum production volume of 12 units.	Client	1	C	Compliant: Omikron sourced the cost for a production of 1 unit.
G			**Project Management**			
G1	Schedule	Schedule of the project will follow the milestones: Phase 1: February 3rd, 2021 Phase 2: March 4h, 2021 Phase 3: April 15th, 2021	Mec E 460	5	C	Compliant
G2	Budget	The client specified a project budget between $100k - $1M. The design team will design a robotic system to meet the market cost of $150,000 for one unit.	Client	3	C	Compliant: The cost of one unit is 150k, this is within the client specified budget of $100k - $1MM and is compliant with the design teams stated budget of $150k.

Overall Client Approval:	☑ YES ☐ NO
Client Signature	
Date	April 13, 2021

Appendix B: Vendor Parts

This appendix contains manufacturer specifications for the commercial off-the-shelf components selected for this project.

Appendix B1: Yaskawa Motoman SIA20F

The datasheet for the Yaskawa Motorman SIA20F series manipulator is presented in this section. The datasheet outlines the dimensions, reach and operations of the manipulator.

YASKAWA

MOTOMAN
SIA-series

Flexible Applications with the SIA-series

The SIA-series are small and agile 7-axis robots providing „human-like" flexibility of movement and fast acceleration. They distinguish themselves through slim and lightweight design. They offer high payload and big working ranges. These agile and versatile robots open up a wide range of industrial applications: ideal for assembly, injection moulding, inspection, machine tending and a host of other operations.

This robots are driven by the compact MOTOMAN FS100 controller.

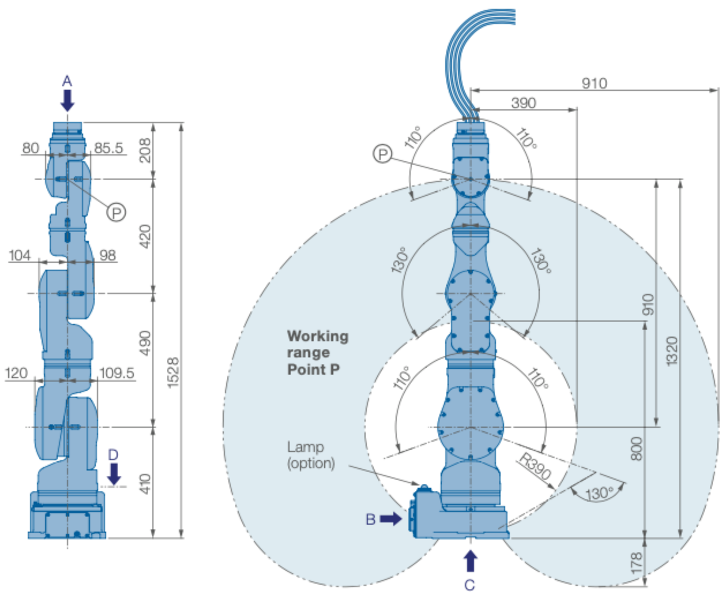
KEY BENEFITS

- 7 axes
- Flexible applications
- Compact design allows maximum performance

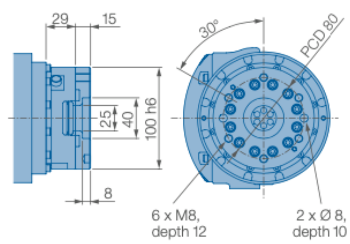
www.yaskawa.eu.com

Controlled by FS100

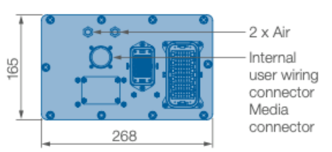
MOTOMAN SIA20F



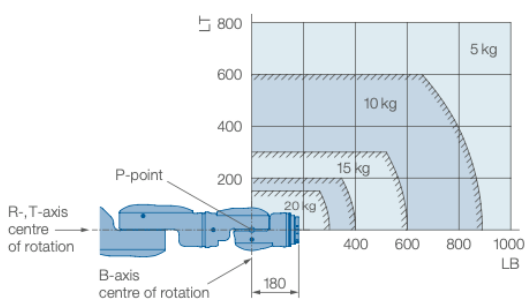
View A



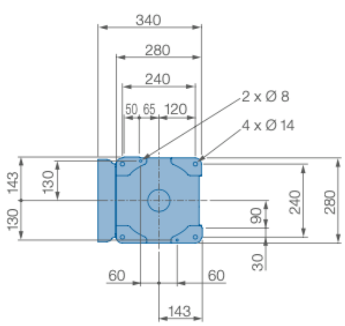
View B



Allowable wrist load



View C



Mounting options: Floor, ceiling, wall

Specifications SIA20F						
Axes	Maximum motion range [°]	Maximum speed [°/sec.]	Allowable moment [Nm]	Allowable moment of inertia [kg · m²]	Controlled axes	7
S	±180	130	–	–	Max. payload [kg]	20
L	±110	130	–	–	Repeatability [mm]	±0,1
Θ	±170	170	–	–	Max. working range R [mm]	910
U	±130	170	–	–	Temperature [°C]	0 bis +40
R	±180	200	58,8	4,0	Humidity [%]	20 – 80
B	±110	200	58,8	4,0	Weight [kg]	120
T	±180	400	29,4	2,0	Power supply, average [KVA]	2,2

Appendix B1.1: Manipulator Payload-to-Mass Ratio Plot

This plot shows the payload-to-mass ratio of commercial manipulators with a payload of 20 kg. The manipulator of the OmiBot should minimize the mass while maximizing the payload, and the Yaskawa Motoman SIA20F (marked in red) accomplishes this, therefore it was chosen.

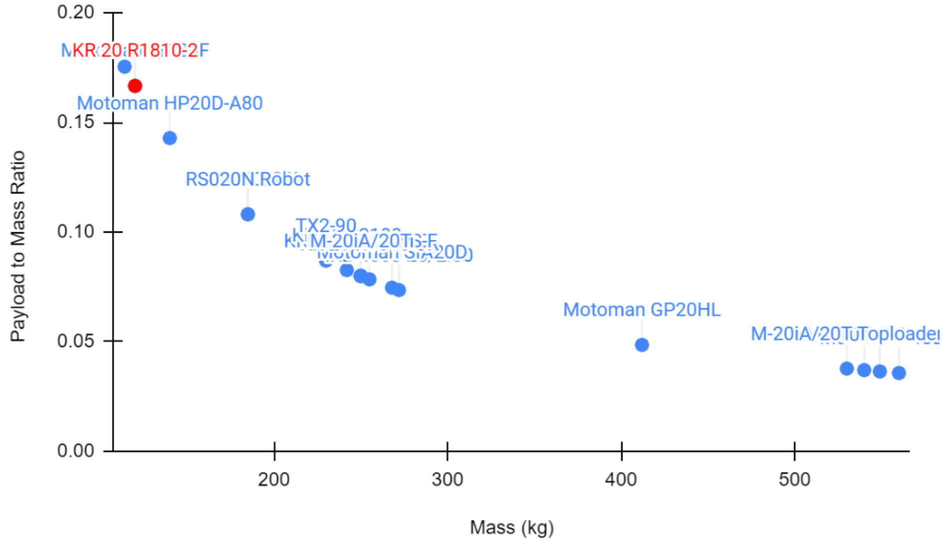


Figure 19 - Manipulator Payload-to-Mass Ratio

Appendix B2: DuraTrue Size 115-100 Gearhead

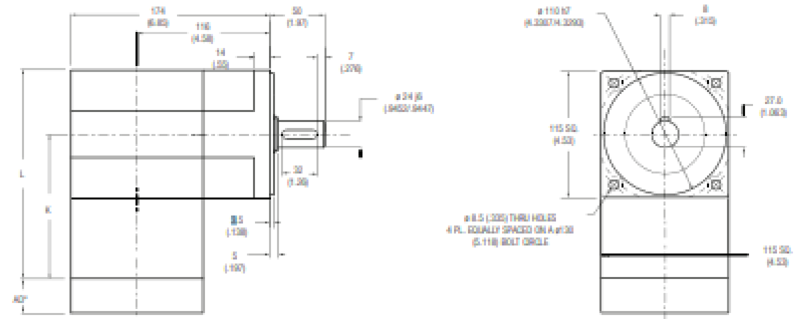
The datasheet for the DuraTRUE gear head is presented in this section. The selected gear head is the DTR115-100, which provides a total gear ratio of 100:1. The maximal radial and axial load is also shown in this section. This provides the maximal load that can be exerted onto the gear head shaft.

Micron TRUE Planetary® Gearheads

DuraTRUE 90 Size 90

Right Angle Gearheads

Metric



Ratio[1]	Dimension 'K' mm [in.]	Dimension 'L' mm [in.]	Backlash [arc-min]	Weight kg [lb.]	Efficiency
5:1 to 50:1	137 [5.40]	195 [7.67]	8 max	11 [24]	93%
60:1 to 500:1	168,4 [6.63]	226 [8.90]	9 max	12 [27]	88%

All dimensions are: mm (in.)
AD** = Adapter length
Adapter length will vary depending on motor. Efficiency is calculated at 100% of the rated torque.

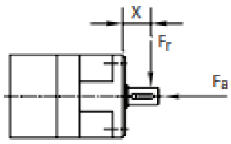
Performance Specifications

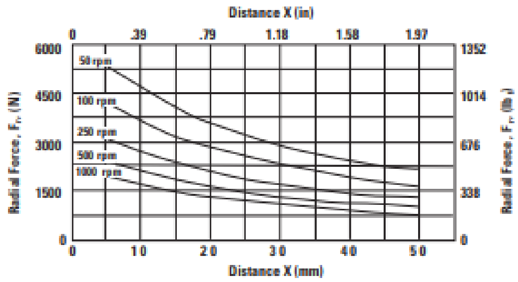
Part Number	Ratio[1]	10000 Hour Life			T_{peak} Nm [in.-lb.]	20000 Hour Life			J kg-cm² [in.-lb.-sec²x10⁻⁴]	Torsional Stiffness Nm/arc-min [in.-lb./arc-min]
		T_r (1000 rpm) Nm [in.-lb.]	T_r (3000 rpm) Nm [in.-lb.]	T_r (5000 rpm) Nm [in.-lb.]		T_r (1000 rpm) Nm [in.-lb.]	T_r (3000 rpm) Nm [in.-lb.]	T_r (5000 rpm) Nm [in.-lb.]		
DTR115-005	5:1	123 [1086]	88 [781]	76 [670]	284 [2511]	100 [882]	72 [634]	61 [544]	2,79 [24.7]	15,3 [135]
DTR115-006	6:1	130 [1147]	93 [825]	80 [708]	284 [2511]	105 [932]	76 [670]	65 [575]	2,77 [24.5]	12,5 [110]
DTR115-009	9:1	146 [1295]	105 [932]	90 [799]	284 [2511]	119 [1052]	86 [757]	73 [649]	2,37 [21.0]	12,6 [112]
DTR115-010	10:1	143 [1262]	103 [908]	88 [779]	284 [2511]	116 [1025]	83 [738]	72 [633]	2,75 [24.3]	13,1 [116]
DTR115-012	12:1	137 [1210]	115 [1016]	98 [871]	284 [2511]	130 [1147]	93 [825]	80 [708]	2,19 [19.4]	12,7 [112]
DTR115-015	15:1	161 [1425]	116 [1025]	99 [880]	284 [2511]	131 [1158]	94 [833]	81 [715]	2,14 [18.9]	12,7 [112]
DTR115-020	20:1	164 [1453]	126 [1118]	108 [959]	284 [2511]	143 [1262]	103 [908]	88 [779]	2,18 [19.3]	13,2 [117]
DTR115-025	25:1	167 [1474]	135 [1195]	116 [1025]	284 [2511]	153 [1350]	110 [971]	94 [833]	2,14 [18.9]	13,2 [117]
DTR115-030	30:1	105 [930]	90 [796]	83 [736]	284 [2511]	97 [861]	83 [737]	77 [681]	2,34 [20.7]	11,5 [102]
DTR115-040	40:1	109 [967]	94 [831]	87 [770]	284 [2511]	101 [894]	87 [769]	80 [712]	2,17 [19.2]	11,5 [102]
DTR115-050	50:1	112 [995]	97 [858]	90 [796]	284 [2511]	104 [921]	90 [794]	83 [737]	2,13 [18.8]	11,5 [102]
DTR115-060	60:1	169 [1495]	121 [1075]	104 [923]	284 [2511]	137 [1214]	99 [873]	85 [749]	2,77 [24.5]	13,2 [117]
DTR115-075	75:1	194 [1715]	139 [1233]	120 [1059]	284 [2511]	139 [1233]	113 [1002]	97 [860]	2,35 [20.8]	13,1 [116]
DTR115-090	90:1	191 [1687]	137 [1214]	118 [1041]	284 [2511]	155 [1371]	111 [986]	96 [846]	2,37 [21.0]	12,5 [110]
DTR115-100	100:1	186 [1644]	134 [1183]	115 [1015]	284 [2511]	151 [1336]	109 [962]	93 [825]	2,75 [24.3]	13,0 [115]
DTR115-120	120:1	178 [1577]	150 [1324]	128 [1135]	284 [2511]	169 [1495]	121 [1075]	104 [923]	2,19 [19.4]	12,6 [111]
DTR115-125	1251	200 [1774]	162 [1438]	139 [1233]	284 [2511]	184 [1624]	132 [1168]	113 [1002]	2,14 [18.9]	13,2 [117]
DTR115-150	150:1	210 [1857]	151 [1336]	130 [1147]	284 [2511]	170 [1509]	123 [1085]	105 [932]	2,35 [20.8]	13,1 [116]
DTR115-200	200:1	214 [1893]	165 [1457]	141 [1250]	284 [2511]	186 [1644]	134 [1183]	115 [1015]	2,73 [24.2]	13,2 [116]
DTR115-250	250:1	217 [1921]	176 [1557]	151 [1336]	284 [2511]	199 [1759]	143 [1265]	123 [1085]	2,14 [18.9]	13,2 [117]
DTR115-300	300:1	137 [1212]	117 [1037]	108 [959]	284 [2511]	127 [1122]	109 [960]	100 [887]	2,34 [20.7]	11,5 [102]
DTR115-400	400:1	142 [1260]	122 [1083]	113 [1003]	284 [2511]	132 [1165]	113 [1002]	105 [928]	2,17 [19.2]	11,5 [102]
DTR115-500	500:1	146 [1296]	126 [1118]	117 [1037]	284 [2511]	136 [1200]	117 [1035]	109 [960]	2,03 [18.0]	11,5 [102]

All ratios are available to ship in 24 hours through the Gearhead Express Program.

[1] Ratios are exact, higher ratios and other custom options are also available, consult factory.
T_r = Rated output torque at rated speed for specific hours of life.
T_{peak} = Allowable momentary peak torque for emergency stop or heavy shock loading.
J = Mass moment of inertia reflected to the input shaft (including pinion assembly).

DT115





Speed rpm	Axial Load, F_a N [lb]
50	8196 [1844]
100	6505 [1464]
250	4793 [1078]
500	3804 [856]
1000	3019 [679]

Appendix B3: AKM2G-41D Servo Motor

The motor selected is the AKM2G-41D 240 Vdc by Kollmorgen. The power output chart is presented in this section.

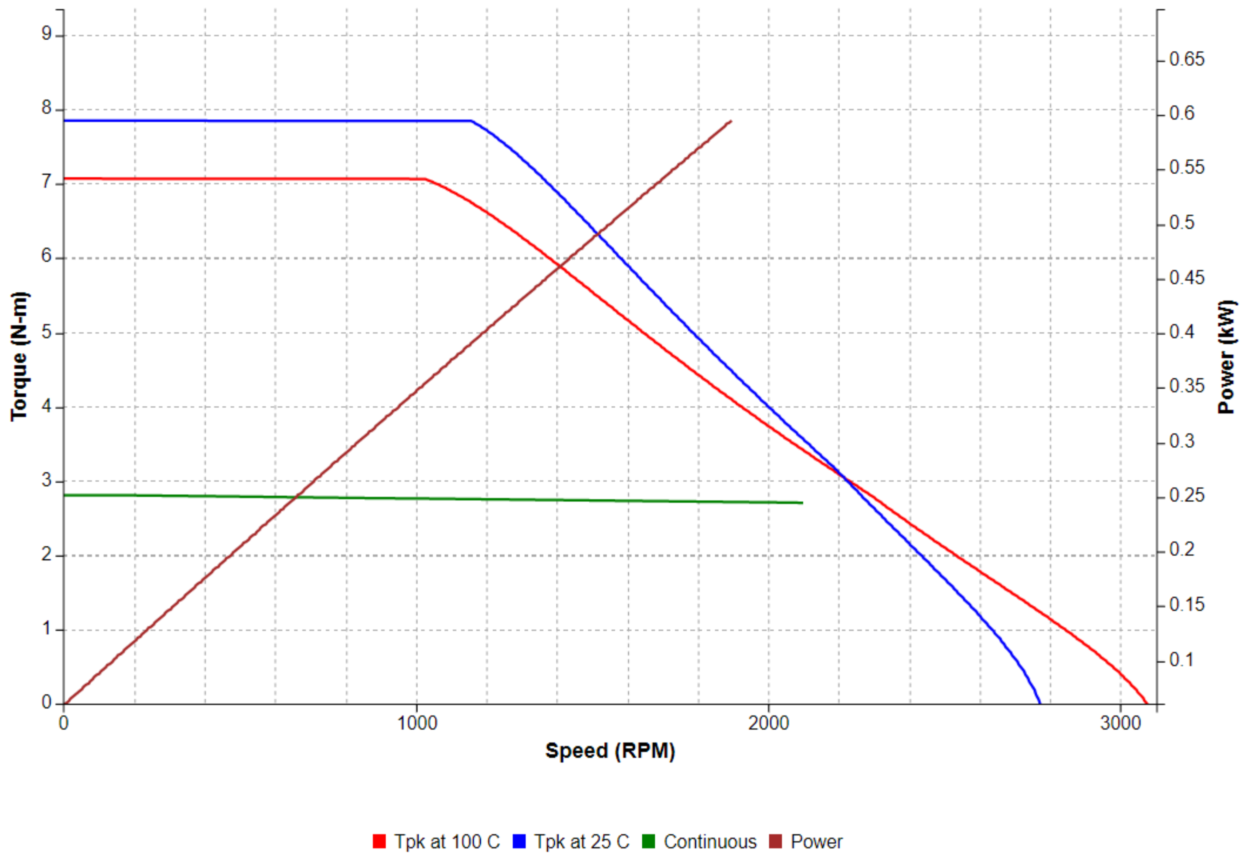
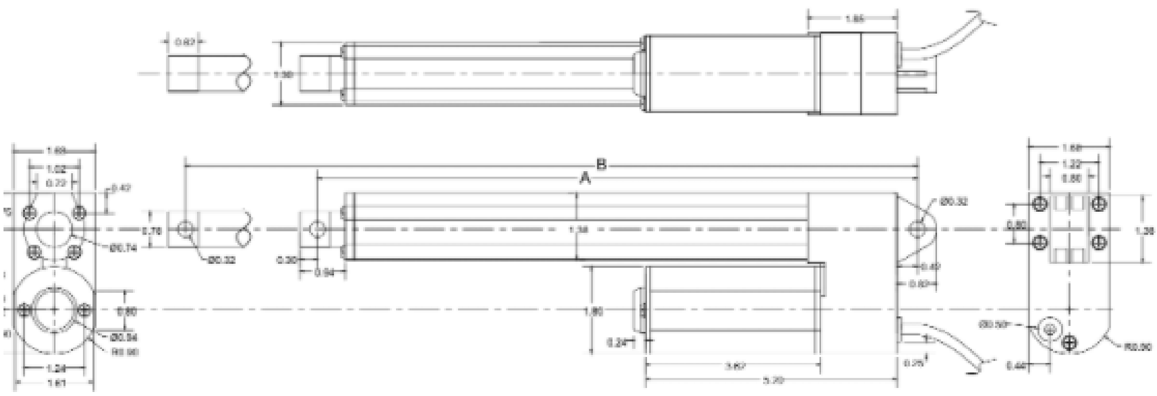


Figure 20 - AKM2G-41D Servo Motor Power Curve

Appendix B4: Linear Actuator Data Sheet

Dimensions

(Dimensions in inches)



Hole to Hole

	Stroke	1	2	3	4	6	8	9	10	12	14	16	18	20	22	24	30	40
PA-09	A	5.53	6.53	7.53	8.53	10.53	12.53	13.53	14.53	17.71	19.71	21.71	23.71	25.71	27.71	30.50	36.50	47.87
	B	6.53	8.53	10.53	12.53	16.53	20.53	22.53	24.53	29.71	33.71	37.71	41.71	45.71	49.71	54.50	66.50	88.87

For Stroke Length less than 12"

A = Stroke Length + 4.53"

B = Stroke Length x 2 + 4.53"

For Stroke Length of 12" to less than 24"

A = Stroke Length + 5.71"

B = Stroke Length x 2 + 5.71"

For Stroke Length 24" to 30"

A = Stroke Length + 6.50"

B = Stroke Length x 2 + 6.50"

For Stroke Length greater than 30"

A = Stroke Length + 7.87"

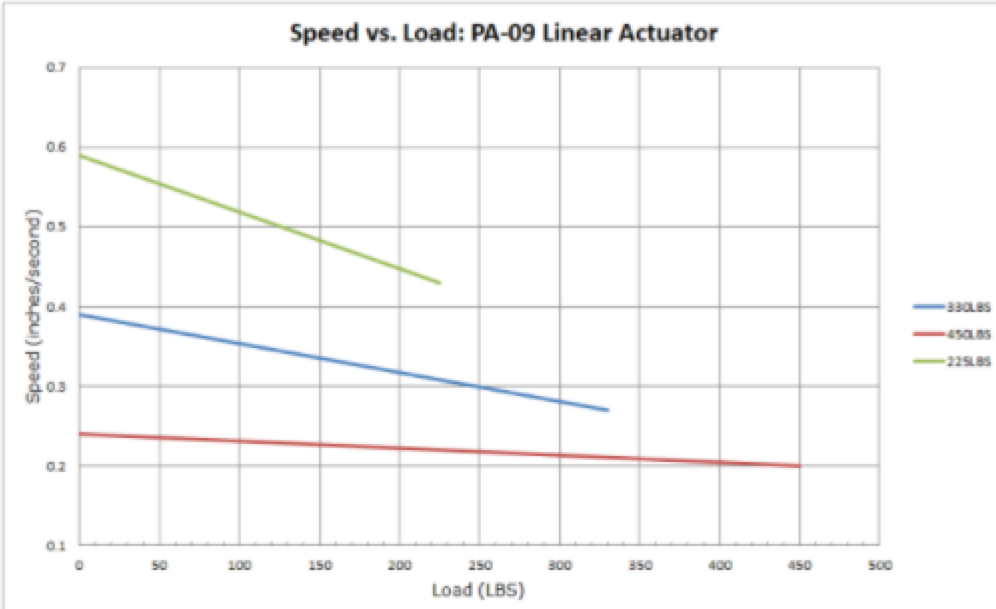
B = Stroke Length x 2 + 7.87"

For Potentiometer Units - Stroke Length up to 8"

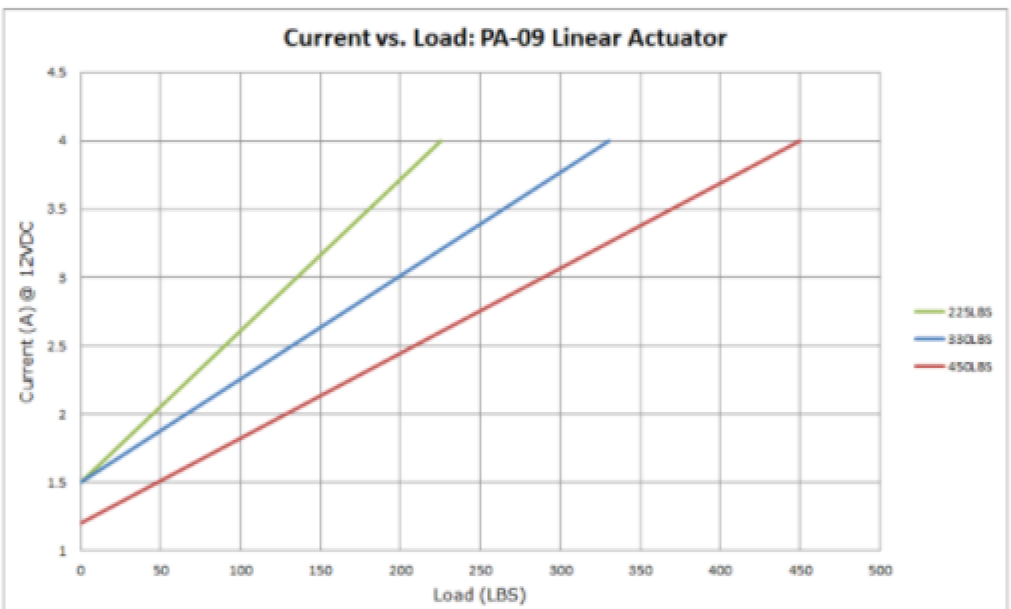
A = Stroke Length + 5.50"

B = Stroke Length x 2 + 5.50"

Speed vs Load



Current vs Load

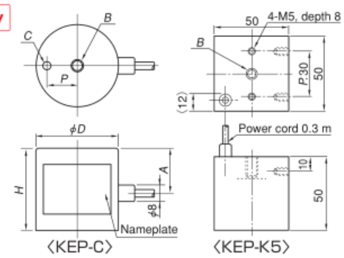


Appendix B5: Permanent Magnet Data Sheet

KE-K / KE-V / KE-M / KEP / KE-H

Model KEP PERMANENT ELECTROMAGNETIC HOLDER

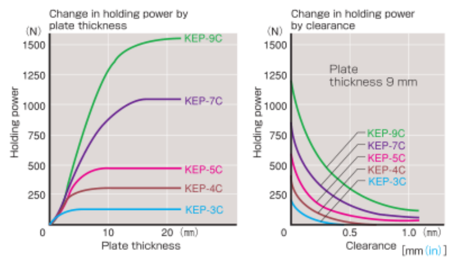
Non-watertight | Electromagnetic release | Rectifier required additionally



Precautions for use
Rust and scratches on the attractive face affect the holding power adversely. Repair it periodically.

[Features]

- No fear of accidents by fallen workpieces due to power failure and no heat generated by continuous power on. These features make these holders suitable for long-hour holding. Workpieces are held by a permanent magnet, but its ON/OFF is controlled electrically.
- The electromagnetic release type that keeps the magnetic force off when power is being supplied. Normally, the magnetic force is kept ON.
- An uninterruptible power supply is not required.
- The square type (KEP-K) is suitable for picking up small parts from corners of containers, etc. and picking up doughnut-shaped workpieces.



How to use

The power source is 24 VDC. When using 4 holders of the same size and same type at the same time, connect their wires in series that are used at a distribution of the voltage 96 VDC (96 V ÷ 4 = 24 V). In this case, a voltage variable rectifier (RH-M) enables adjustment of the demagnetizing voltage (power on amount at OFF) to facilitate operation.

Released only at power on

The power-on time must be 5 seconds or less. The power-off time must be 10 times or longer. (30 seconds or less for KEP-K.)

Residual holding power

As an inevitable nature of permanent electromagnetic holders, 3% to 4% of the holding power will remain as residual holding power after the workpiece has been released. If the weight of the lifted workpiece is smaller than this holding power, it may not be released. In such a case, the workpiece can be released easily by attaching a thin nonmagnetic film on the attractive face. Note, however, that the holding power will drop as the square of clearance.

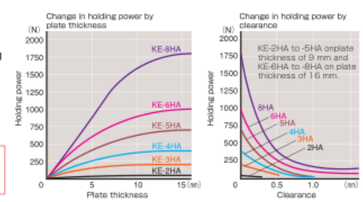
[mm (in)]

Model	Dimensions ϕD	H	P	A	B	C	Max. Holding Power	Voltage	Current	Working Rate	Applicable Rectifier	Mass
KEP-3C	30 (1.18)	40 (1.57)	10 (0.39)	22 (0.86)	M6 (0.23) Depth 10 (0.39)	ϕ4 (0.15) Depth 3 (0.11)	150N (15kgf)	24 VDC	0.45A	10% ED	RH-M303A-6/24 RH-M303A-6/24-C1 RH-M303A-6/24-C2 KR-T101A-6/24 P77, P78	0.17kg/ 0.37 lb
KEP-4C	40 (1.57)		15 (0.59)				250N (25kgf)		0.54A			0.31kg/ 0.68 lb
KEP-5C	50 (1.96)	50 (1.96)	18 (0.70)	25 (0.98)	M8 (0.31) Depth 13 (0.51)	ϕ5 (0.19) Depth 4 (0.15)	340N (35kgf)		0.58A			0.6 kg/ 1.32 lb
KEP-7C	70 (2.75)	60 (2.36)	20 (0.78)	35 (1.37)	M10 (0.39) Depth 16 (0.62)	ϕ6 (0.23) Depth 6 (0.23)	880N (90kgf)		0.50A			1.5 kg/ 3.30 lb
KEP-9C	90 (3.54)						1470N (150kgf)		0.45A			2.4 kg/ 5.29 lb
KEP-K5	50 (1.96) × 50 (1.96) × 50 (1.96)				M8 (0.31) Depth13 (0.51)	—	250N (25kgf)		0.43A	50% ED		0.75kg/ 1.65 lb

※The max. holding power is based on a test piece of SS400, 20 mm thick, ground surface held on the whole area. Therefore, the lifting capacity is normally a third or less of the max. holding power. ※Cord length 0.3 m.

Model KE-H HYBRID HOLDER

Controller required additionally | Non-watertight



Precautions for use
Rust and scratches on the attractive face affect the holding power adversely. Repair it periodically.

[Application]
Suitable for robot hands and such systems that require high-speed operations such as repeated transfer in automated lines.

[Features]

- Very little residual holding power allows workpieces to be released quickly. This enables high-speed operation; for example, light weight workpieces can be attached/detached 5 to 6 times per second.
- Because these holders are of permanent electromagnetic type, the holders consume little power and generate little heat, making these holders suitable for continuous, long-hour operation.
- The holding power is switchable at two stages; High and Low by turning on and off the power supply. The reverse supply of power releases workpieces. This enables a wide variety of usage. (When at "Low," the holding power is about 1/3 of that at "High.")
- The powerful rare earth magnet offers high holding power in spite of its small size.

A type of cord on the top face spec. (KE-HA-U) is also available.

[mm (in)]

Model	Size	Max. Holding Power	Center Tapped Hole on Back	Voltage	Current	Working Rate	Applicable Rectifier	Mass
KE-2HA	ϕ20 (0.78) × 25 (0.98)	50N (5kgf)	M4 (0.15) ×0.7 (0.02) Depth 6 (0.23)	24 VDC	0.07A	100% ED	RH-H303A RH-H303A-C2 P78	60g/ 0.13 lb
KE-3HA	ϕ30 (1.18) × 40 (1.57)	200N (20kgf)	M6 (0.23) ×1.0 (0.03) Depth 6 (0.23)		0.11A			140g/ 0.31 lb
KE-4HA	ϕ40 (1.57) × 40 (1.57)	400N (40kgf)			0.15A			280g/ 0.61 lb
KE-5HA	ϕ50 (1.96) × 50 (1.96)	700N (70kgf)	M8 (0.31) ×1.25 (0.04) Depth 10 (0.39)		0.2 A			530g/ 1.17 lb
KE-6HA	ϕ60 (2.36) × 60 (2.36)	1000N (100kgf)			0.22A			960g/ 2.11 lb
KE-8HA	ϕ80 (3.15) × 60 (2.36)	1800N (180kgf)	M10 (0.39) ×1.5 (0.05) Depth 12 (0.47)		0.28A			1.6kg/ 3.52 lb

※Cord length 0.3 m. (KE-2HA: 0.2 m)
※The max. holding power is based on a test piece of SS400, ground surface held on the whole area. Therefore, the lifting capacity is normally a third or less of the max. holding power.
Test piece thickness: KE-2HA to 4HA … 10 mm, KE-5HA to 8HA … 20 mm

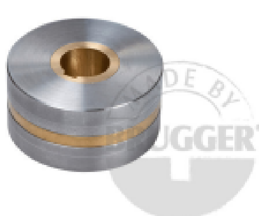
Appendix B6: Neodymium Magnetic Wheel Data Sheet

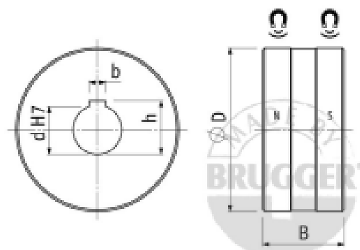
PRODUCT INFORMATION

MAGNETSYSTEME BRUGGER®

Magnetic wheels Neodymium-iron-boron (NdFeB)

Magnetic wheels with two-pole magnetization of NdFeB, bore with fitting tolerance H7 and groove





Article number	D mm	d mm	B mm	b mm	h mm	Force* N	Temperature °C
HRZ25	25 $^{+0.1}/_{-0.1}$	8	16 $^{+0.5}/_{-0.5}$	3	8.6	45	100
HRZ32	32 $^{+0.1}/_{-0.1}$	10	18 $^{+0.5}/_{-0.5}$	4	11.1	65	100
HRZ40	40 $^{+0.1}/_{-0.1}$	12	20 $^{+0.5}/_{-0.5}$	4	13.1	90	100
HRZ50	50 $^{+0.1}/_{-0.1}$	16	25 $^{+0.5}/_{-0.5}$	5	17.3	140	100
HRZ63	63 $^{+0.15}/_{-0.15}$	20	32 $^{+0.5}/_{-0.5}$	6	21.7	270	100
HRZ80	80 $^{+0.15}/_{-0.15}$	25	40 $^{+0.5}/_{-0.5}$	8	26.7	380	100
HRZ100	100 $^{+0.2}/_{-0.2}$	30	50 $^{+0.5}/_{-0.5}$	8	31.7	580	100
HRZ125	125 $^{+0.2}/_{-0.2}$	40	62 $^{+0.5}/_{-0.5}$	12	42.1	1000	100
HRZ160	160 $^{+0.25}/_{-0.25}$	50	80 $^{+0.5}/_{-0.5}$	14	52.6	1800	100

REACH RoHS UV resistance

Appendix C: Cost Analysis

This appendix includes details related to the costing of the commercial off-the-shelf components selected for this project.

Appendix C1: Yaskawa Motoman SIA20F (Manipulator) Quote

sasan alpha-ts.ca <sasan@alpha-ts.ca> Mon, Mar 1, 2021 at 5:00 PM
To: Calvin Chen <calvin1@ualberta.ca>

Hi Calvin,

We can do it for USD48K. If any point, pls let me know.

Appendix C2: Linear Actuator Quote

Home > Mini Industrial Actuator

Mini Industrial Actuator
Model: PA-09

Instant Quote $187.49 CAD

7-9 units	$178.12 CAD	20–49 units	$149.99 CAD
10-19 units	$168.74 CAD	50+	Get a Quote

Stroke * Select Stroke | Force * Select Force | Quantity - 1 +

MAKE A SELECTION

Additional products you may need | Add to Compare

Appendix C3: Linear Actuator Magnets Quote

Electromagnetic/Permanent Electromagnetic Hybrid Holder - KE Series (Kanatec) (KEP-9C)

Kanetec

- Can be used continuously.
- For a wide range of applications, from handles for various types of automated equipment to those for industrial robots.
- Applicable with a wide range of applications, including material feeding for automatic presses, deflection prevention for shearing material and applications ranging from handles for various types of automated equipment to those for industrial robots.

· KEP: With no worry of heat generation or dropping incidents during power failure, it is suitable for long-time adhesion. An electromagnetic release type in which the magnetic force is OFF only when energized. The magnetic force is on during ordinary operation.
· KE-HA: Can quickly release work pieces as there is such little residual adhesion. Therefore, it can operate at high speeds.

Configure | Clear All
Submit Configuration
Completed ✓

Filters
type
☐ KEP

Magnet
☑ Permanent electromagnetic
Clear

External Dimension φ(mm)
☑ 90
Clear

Edit Part Configuration

Part Number **KEP-9C** | Add | Download Product Details | CAD Download

Configured Specifications

Magnet	Permanent electromagnetic	External Dimension φ(mm)	90
External Height(mm)	60	Magnetic force upon energization	OFF
Automatic release function	None	Shape	JPY
JAN code	4544554900080	Ordering Code	406-3457

Price $692.37
Total **$692.37**
Ship Date Sat. Apr 17, 2021
Valid until 8PM, EST
Qty. 1 | Add to Cart

Quantity	Price	Ship Date
1	$692.37	Apr 17, 2021
2-	$692.37	TBD

NOTE : Ship Dates above are subject to change depending upon availability.

Appendix C4: Motor Quote (Analogous Estimate)

iAUTOMATION

Welcome to Our Pop-Up Store!
Limited Listings Available - List Prices Only
Click Here to Call for More Info

- Home
- Banner Engineering
- B&R
- Kollmorgen
- Festo
- Email Our Specialists
- Call One of Our Specialists

Home › AKM2G Servo Motor Series - Kollmorgen

AKM2G Servo Motor Series - Kollmorgen
$1,319.00

Model
AKM2G-42H-ANC2R-00

ADD TO CART
Buy with G Pay
More payment options

Share Tweet Pin it

Appendix C5: Bearing Quote

Hi Dear Calvin,

Good day! This is April,from AOBOTE BEARING CO.,LTD.
So glad to receive your inquiry for our bearings.

Regard to this item,please kindly check the following details:
Item: **Original Fracen SKF S6009-2RSR-FD Deep Groove Ball Bearing, In Stock, Stainless Steel**
Size(d*D*B) : 45*75*16mm
Weight: 0.251kg
EXW Wuxi Unit Price: **USD9.5/pc**

Appendix C6: Gearhead Quote

Micron Motioneering Product Details

Part Number
DTR115-100-0-RM115-40

Part Number Description
DuraTRUE Right Angle (DTR)

Gearhead Model Details

Peak Torque	283.7 Nm
Rated Torque	962.0 Nm
Inertia	2.746 kg cm2
Gearhead Express	No
Torsional Stiffness	12.993 Nm/arc-min
RediMount ID	RM115-40
ADmax Distance	50.038 mm
Bearing Type	Ball Bearing

Total Price List : $ 5228.5 *

Appendix C7: Magnetic Wheels Quote

FAIZEAL 宁波风正磁应用科技有限公司

NINGBO FAIZEAL MAGNETIC TECHNOLOGY CO.,LTD

宁波市江北区通宁路288号
No.288, TongNing Road, Jiangbei District, NingboChina.

To:	Omikron Robotics	Tel:	0574-87520996
Attn:	Liam	Fax:	0574-83091125
Tel:		From:	Ailsa
Fax:		询价日期RFQ Date：	2021/2/10
Add:		报价日期Quotation Date:	2021/2/20

报价单
Quotation Sheet

NO.: SLQ03210220-3

序号 S/N	产品描述 Product Description	图号 Drawing No.	数量 QTY	单价 Unit Price	毛重/运费 G.W./ Delivery cost	数量 QTY	单价 Unit Price	毛重/运费 G.W./ Delivery cost	交期 ETD
1	FZW25		6PCS	US$26.0	4.0KGS/$59	12PCS	US$26.0	4.5kgs/$64	
2	FZW32		6PCS	US$28.0	4.5kgs/$64	12PCS	US$28.0	5.0kgs/$69	
3	FZW40		6PCS	US$32.0	5.0kgs/$69	12PCS	US$31.0	6.0kgs/$79	
4	FZW50		6PCS	US$40.0	5.5kgs/$75	12PCS	US$39.0	8.0kgs/$102	
5	FZW63	/	6PCS	US$88.0	8.0kgs/$102	12PCS	US$86.0	12kgs/$143	25 working days
6	FZW80		6PCS	US$92.0	12.5kgs/$148	12PCS	US$90.0	25kgs/$255	
7	FZW100		6PCS	US$145.0	24kgs/$246	12PCS	US$142.0	41kgs/$398	
8	FZW125		6PCS	US$210.0	43kgs/$418	12PCS	US$207.0	75kgs/$706	
9	FZW160		6PCS	US$295.0	89.5kgs/$775	12PCS	US$290.0	179kgs	

其他条款(other terms):

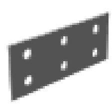
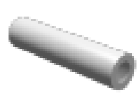
1、有效期：以上价格有效期为7天，自报价之日起开始计
Validity: quotation is valid for 7 days

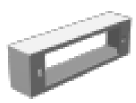
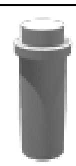
2、付款方式：银行汇款100%
Payment terms: 100%T/T for sample order, for mass production, we accept 30% in advance, balance against to copy of B/L.

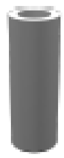
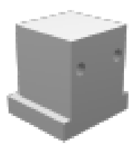
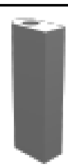
Appendix C8: OmiBot Xometry Quote

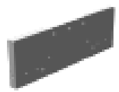
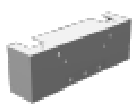
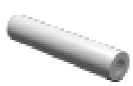
Xometry	Quote ID: 301BE-15005
7951 Cessna Avenue | Gaithersburg, MD | 20879 | 240-252-1138	Date: 04/04/2021 1:04 AM EDT

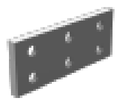
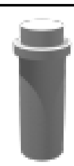
Contact Info:	Estimated ship date:	Requirements:
Calvin Chen	**Friday, May 07** (if you order by 11:59PM EDT Monday, April 5) Lead time: 20 business days	

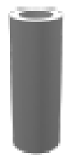
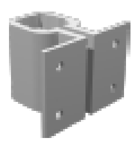
Item	Part ID	Description	Qty.	Unit Price	Extended Price
1	 01F6FC4 Made in China	Bottom Connector Plate.SLDPRT **Bounding Box:** 254.0mm x 101.6mm x 3.2mm 10.00in x 4.00in x 0.12in **Process:** CNC Machining **Material:** Steel 4130 **Finish:** Standard **Inspection:** Standard Inspection	1	$133.60	$133.60
2	 01F6FC9 Made in China	Threaded Support Rod.SLDPRT **Bounding Box:** 133.4mm x 31.8mm x 31.8mm 5.25in x 1.25in x 1.25in **Process:** CNC Machining **Material:** Steel 4130 **Finish:** Standard **Inspection:** Standard Inspection	4	$66.02	$264.08

Item	Part ID	Description	Qty.	Unit Price	Extended Price
3	 020B378 Made in China	drive housing.SLDPRT **Bounding Box:** 330.0mm x 160.0mm x 160.0mm 12.99in x 6.30in x 6.30in **Process:** CNC Machining **Material:** Steel 4130 **Finish:** Standard **Inspection:** Standard Inspection	4	$799.26	$3,197.04
4	 020B37A Made in China	compression spring mount.SLDPRT **Bounding Box:** 79.0mm x 24.0mm x 17.0mm 3.11in x 0.94in x 0.67in **Process:** CNC Machining **Material:** Steel 4130 **Finish:** Standard **Inspection:** Standard Inspection	4	$62.74	$250.96
5	 020B37D Made in China	Driveshaft.SLDPRT **Bounding Box:** 149.0mm x 60.0mm x 60.0mm 5.87in x 2.36in x 2.36in **Process:** CNC Machining **Material:** Steel 4130 **Finish:** Standard **Inspection:** Standard Inspection	4	$183.02	$732.08

Item	Part ID	Description	Qty.	Unit Price	Extended Price
6	 020B37E Made in China	Fixed Shaft Threaded Support.SLDPRT **Bounding Box:** 53.2mm x 19.0mm x 19.0mm 2.09in x 0.75in x 0.75in **Process:** CNC Machining **Material:** Steel 4130 **Finish:** Standard **Inspection:** Standard Inspection	4	$36.54	$146.16
7	 020B37F Made in China	Fixed Shaft.SLDPRT **Bounding Box:** 122.0mm x 100.0mm x 87.7mm 4.80in x 3.94in x 3.45in **Process:** CNC Machining **Material:** Steel 4130 **Finish:** Standard **Inspection:** Standard Inspection	4	$234.02	$936.08
8	 020B384 Made in China	Pivot Shaft Housing.SLDPRT **Bounding Box:** 100.0mm x 38.8mm x 19.3mm 3.94in x 1.53in x 0.76in **Process:** CNC Machining **Material:** Steel 4130 **Finish:** Standard **Inspection:** Standard Inspection	4	$62.75	$251.00

Item	Part ID	Description	Qty.	Unit Price	Extended Price
9	 0208387 Made in China	Manipulator Platform.SLDPRT **Bounding Box:** 600.0mm x 200.0mm x 33.0mm 23.62in x 7.87in x 1.30in **Process:** CNC Machining **Material:** Steel 4130 **Finish:** Standard **Inspection:** Standard Inspection	2	$703.38	$1,406.76
10	 0208388 Made in China	Modular Chassis (Half).SLDPRT **Bounding Box:** 600.0mm x 200.0mm x 139.7mm 23.62in x 7.87in x 5.50in **Process:** CNC Machining **Material:** Steel 4130 **Finish:** Standard **Inspection:** Standard Inspection	2	$2,090.49	$4,180.98
11	 0208393 Made in China	Threaded Manipulator Support Rod.SLDPRT **Bounding Box:** 133.4mm x 26.7mm x 26.7mm 5.25in x 1.05in x 1.05in **Process:** CNC Machining **Material:** Steel 4130 **Finish:** Standard **Inspection:** Standard Inspection	4	$56.49	$225.96

Item	Part ID	Description	Qty.	Unit Price	Extended Price
12	 020B395 Made in China	Top Connector Plate.SLDPRT **Bounding Box:** 254.0mm x 101.6mm x 16.5mm 10.00in x 4.00in x 0.65in **Process:** CNC Machining **Material:** Steel 4130 **Finish:** Standard **Inspection:** Standard Inspection	1	$226.85	$226.85
13	 020B398 Made in China	V Bracket.SLDPRT **Bounding Box:** 306.5mm x 203.3mm x 20.0mm 12.07in x 8.00in x 0.79in **Process:** CNC Machining **Material:** Steel 4130 **Finish:** Standard **Inspection:** Standard Inspection	4	$304.81	$1,219.24
14	 020B37D Made in China	Driveshaft.SLDPRT **Bounding Box:** 149.0mm x 60.0mm x 60.0mm 5.87in x 2.36in x 2.36in **Process:** CNC Machining **Material:** Steel 4130 **Finish:** Standard **Inspection:** Standard Inspection	4	$183.02	$732.08

Item	Part ID	Description	Qty.	Unit Price	Extended Price
15	 020B37E Made in China	Fixed Shaft Threaded Support.SLDPRT **Bounding Box:** 53.2mm x 19.0mm x 19.0mm 2.09in x 0.75in x 0.75in **Process:** CNC Machining **Material:** Steel 4130 **Finish:** Standard **Inspection:** Standard Inspection	4	$36.54	$146.16
16	 020B3B7 Made in China	BRK-10.SLDPRT **Bounding Box:** 82.5mm x 66.0mm x 59.7mm 3.25in x 2.60in x 2.35in **Process:** CNC Machining **Material:** Steel A36 **Finish:** Standard **Inspection:** Standard Inspection	2	$256.51	$513.02

SUBTOTAL	$14,562.05
TOTAL	$14,562.05

This Quote does not include sales tax or shipping. If applicable, sales tax and shipping will be added to your order at checkout or upon invoice.

Checkout

Appendix C9: OmiBot Component Cost

Table 5 - OmiBot Vendor Component Cost

Item	Quantity	Cost per Unit (USD)	Cost per Unit (CAD)	Packs Of	Total Cost	McMaster Item #
Manipulator	1	$48,000.00	$60,232.80	-	$60,232.80	-
Linear Actuator	2	-	$187.49	-	$374.98	-
Linear Actuator Magnets	2	$692.37	$868.82	-	$1,737.64	-
Motor	8	-	$1,319.00	-	$10,552.00	-
Bearing	12	9.50	$11.93	-	$143.16	-
Gearhead	8	$5,228.50	$6,560.98	-	$52,487.87	-
Magnetic Wheels	8	$290.00	$363.91	12	$4,366.88	-
Winch Hoist Rings	8	$71.52	$89.75	1	$717.97	2994T91
Linear Springs	4	$6.40	$8.03	1	$32.12	9663K69
Gear Housing bolt	32	$10.04	$12.60	50	$12.60	91280A086
Gear Housing Nut	32	$7.77	$9.75	100	$9.75	95462A030
Gear Housing to Pivot Bolt	8	$8.80	$11.04	50	$11.04	91290A432
Gear Housing to Pivot Nut	8	$7.77	$9.75	100	$9.75	95462A030
Motor Mount Screw Bottom	32	$14.34	$17.99	50	$17.99	90044A123
Motor Mount Screw	32	$14.34	$17.99	50	$17.99	90044A123
Modular Plate Bolt	6	24.73	$31.03	1	$186.19	91253A867
Modular Plate Nut	6	$14.09	$17.68	20	$17.68	94895A426
Manipulator Screw	4	$12.00	$15.06	10	$15.06	96144A302
Manipulator Washer	4	$8.07	$10.13	5	$10.13	93413A180
Keys	8	$5.19	$6.51	1	$52.10	96717A631
Shaft Collars	8	$98.72	$123.88	1	$991.03	3329K18
Fixed Shaft	8	12.59	$15.80	1	$126.39	90044A193
Linear Actuator Bolt	8	$10.04	$12.60	50	$12.60	91280A086
Linear Actuator Nut	8	$7.77	$9.75	100	$9.75	95462A030
Linear Actuator Mount Bolt	6	$13.25	$16.66	50	$16.66	91251A442
Fixed Shaft Bolt	8	$3.21	$4.04	1	$32.32	91251A031
Fixed Shaft Nut	8	$17.42	$21.90	100	$21.90	95462A525
Total					**$132,216.36**	

Table 6 - OmiBot Manufactured Component Cost

Item	Quantity	Material	Process	Cost per Unit (USD)	Cost per Unit (CAD)	Total Cost
Bottom Connector Plate	1	Steel 4130	CNC Machining	$133.60	$167.65	$167.65
Threaded Support Rod	4	Steel 4130	CNC Machining	$66.02	$82.85	$331.38
Drive Housing	4	Steel 4130	CNC Machining	$799.26	$1,002.95	$4,011.81
Compression Spring Mount	4	Steel 4130	CNC Machining	$62.74	$78.73	$314.92
Driveshaft	8	Steel 4130	CNC Machining	$183.02	$229.66	$1,837.30
Fixed Shaft Threaded Support	8	Steel 4130	CNC Machining	$36.54	$45.85	$366.82
Fixed Shaft	4	Steel 4130	CNC Machining	$234.02	$293.66	$1,174.64
Pivot Shaft Housing	4	Steel 4130	CNC Machining	$62.75	$78.74	$314.97
Manipulator Platform	2	Steel 4130	CNC Machining	$703.38	$882.64	$1,765.27
Modular Chassis Half	2	Steel 4130	CNC Machining	$2,090.49	$2,623.25	$5,246.50
Threaded Manipulator Support Rod	4	Steel 4130	CNC Machining	$56.49	$70.89	$283.55
Top Connector Plate	1	Steel 4130	CNC Machining	$226.85	$284.66	$284.66
V Bracket	4	Steel 4130	CNC Machining	$304.81	$382.49	$1,529.96
Linear Actuator Mounting Bracket	2	Steel A36	CNC Machining	$256.51	$321.88	$643.76
Total						**$18,273.19**

Appendix D: Analytic Solutions

The analytic solutions (hand calculations) are presented in this section.

Appendix D1: Derivation of Dynamic Loads

Dynamic Force Calculations: Determination of inertial forces from rotating manipulator arm

Author : Areej Khaddaj   Date : March 24, 2021

Objective

To determine the forces induced on the platform when the manipulator arm is undergoing maximum angular acceleration while the chassis is stationary. The resulting dynamic forces will be used in a static FEA analysis to ensure stresses and displacements on the chassis do not exceed a safety factor of XX, and for the stability analysis for the dynamic case of the arm moving.

Assumptions

(1) The manipulator arm is modelled as a rod, rotating about the manipulator base identified by the orange circle in the FBD/MAD. The robot without the arm is reffered to as the "robot body", and is treated as a separate rigid body from the rotating rod.

(2) The center of mass for the rotating arm is at the center of the arm, which is half of its length.

(3) The acceleration of the arm is constant..

(4) The robot body and manipulator base are treated as a rigid body.

(5) The manipulator arm accelerates from rest.

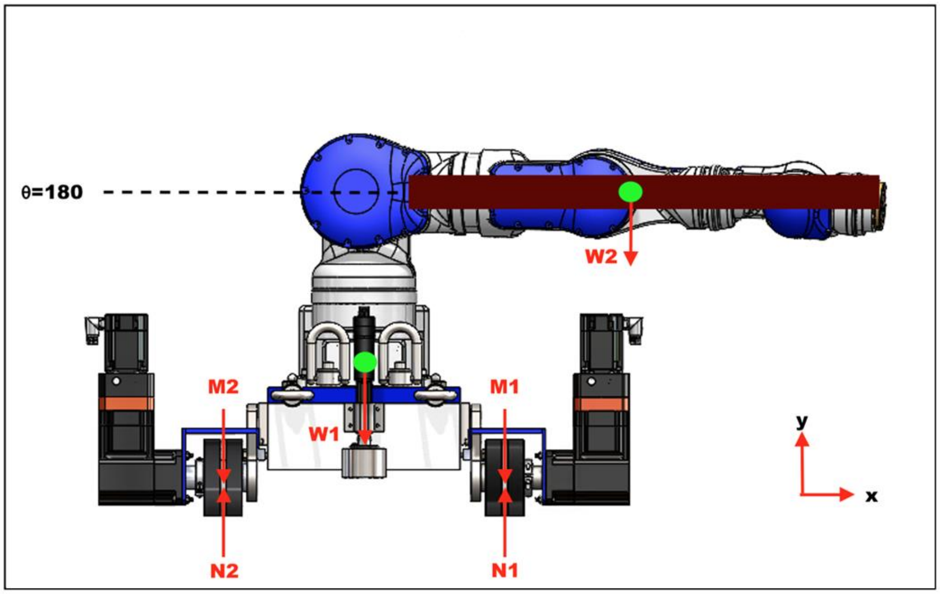
(6) The rod begins at $\theta = 0$ rad, and reaches maximum angular velocity at $\theta = \pi$ rad.

Analysis

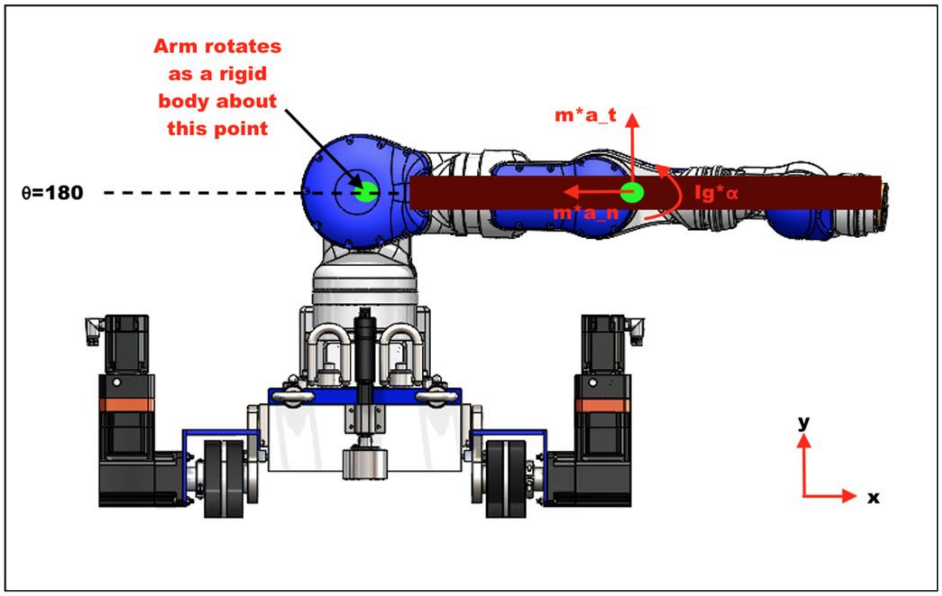
HORIZONATAL CASE:

Assembly free body diagram and mass acceleration diagram, showing the arm beginning to move from rest:

FBD



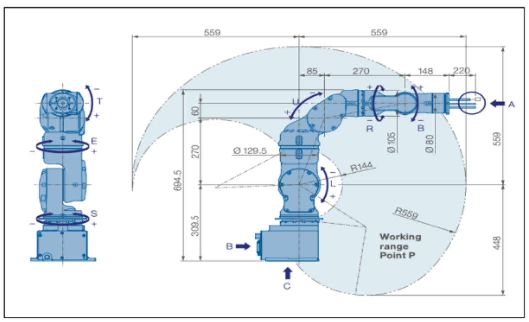
MAD



Define variables :

a_n	normal component acceleration of arm
a_t	tangential component acceleration of arm
α	angular acceleration of arm
$w_0 := 0$	initial angular velocity of arm
$w_{max} := 2.27 \ \frac{rad}{s}$	maximum allowable acceleration of arm, based on manufacturer specifications
$w_f := w_{max} = 2.27$	final angular acceleration of arm
$\theta_0 := 0 \ rad$	initial angular position of arm
$\theta := \pi = 3.1416$	final angular position of arm
$m_{rod} := 120 \ kg$	mass of the rod, separate from the robot body

For reference, the maximum allowable angular velocity is retrieved from the manufacturer specifications for rotational axes L:



Axes	Maximum motion range [°]	Maximum speed [°/sec.]	Allowable moment [Nm]	Allowable moment of inertia [kg · m²]
				Specifications SIA20F
S	±180	130	–	–
L	±110	130	–	–
Θ	±170	170	–	–
U	±130	170	–	–
R	±180	200	58,8	4,0
B	±110	200	58,8	4,0
T	±180	400	29,4	2,0

For constant angular acceleration, the following relationships between θ, w, and α exist (retrieved from Engineering Mechanics Dynamics, R.C Hibbeler):

Constant Angular Acceleration. If the angular acceleration of the body is *constant*, $\alpha = \alpha_c$, then Eqs. 16–1, 16–2, and 16–4, when integrated, yield a set of formulas which relate the body's angular velocity, angular position, and time. These equations are similar to Eqs. 12–4 to 12–6 used for rectilinear motion. The results are

$$\omega = \omega_0 + \alpha_c t \qquad (16\text{–}5)$$
$$\theta = \theta_0 + \omega_0 t + \tfrac{1}{2}\alpha_c t^2 \qquad (16\text{–}6)$$
$$\omega^2 = \omega_0^2 + 2\alpha_c(\theta - \theta_0) \qquad (16\text{–}7)$$

Constant Angular Acceleration

Here θ_0 and ω_0 are the initial values of the body's angular position and angular velocity, respectively.

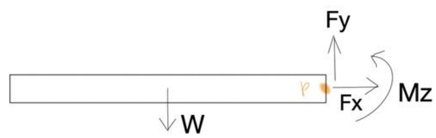
Knowing initial and final angular velocity, and assuming rotation from 0 to π rad, the angular acceleration can be calculated using equation (16-7):

$$\alpha := \frac{\left(w_f^{\,2} - w_0^{\,2}\right)}{2 \cdot \left(\theta - \theta_0\right)} = 0.8201 \qquad \frac{rad}{s^2}$$

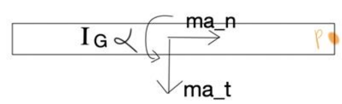
The orange pivot point acts as a common point between two rigid bodies: the arm and the robot body. A free body diagram and mass acceleration diagram of the arm when it is at it's maximum angular velocity where θ = π rad.

Manipulator arm FBDs:

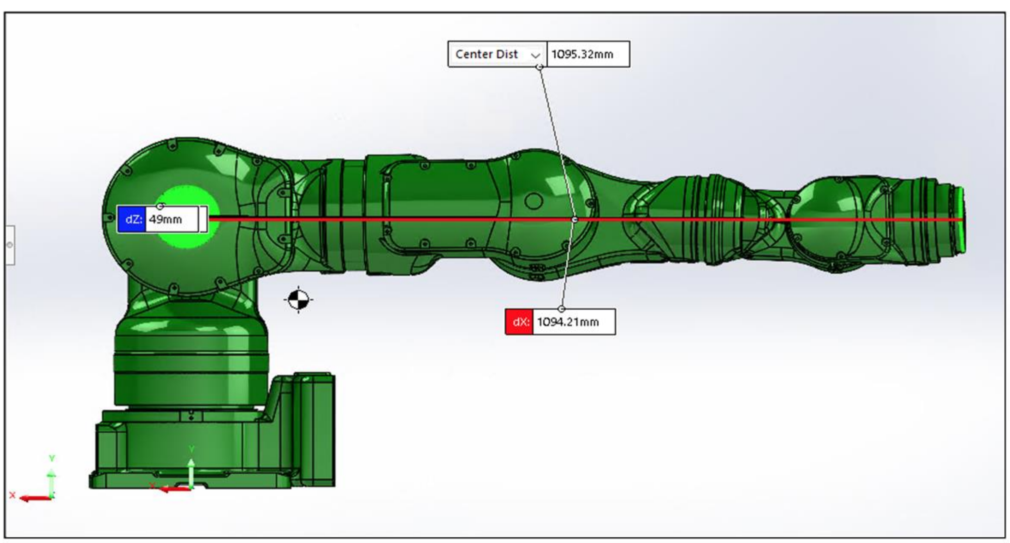
FBD



MAD



The length of the rod is retrieved from solidworks: $l_{rod} := 1.094 \quad m$



Mass moment of inertia, normal and tangential acceleration components:

$$I_G := m_{rod} \cdot \frac{l_{rod}^{\ 2}}{12} = 11.9684 \quad kg \cdot m^2 \qquad a_n := w_f^{\ 2} \cdot \frac{l_{rod}}{2} = 2.8186 \quad \frac{m}{s^2}$$

$$a_t := \alpha \cdot \frac{l_{rod}}{2} = 0.4486 \quad \frac{m}{s^2}$$

Equations of motion for general plane motion:

In some problems it may be convenient to sum moments about a point P other than G in order to eliminate as many unknown forces as possible from the moment summation. When used in this more general case, the three equations of motion are

$$\begin{aligned} \Sigma F_x &= m(a_G)_x \\ \Sigma F_y &= m(a_G)_y \\ \Sigma M_P &= \Sigma(\mathcal{M}_k)_P \end{aligned} \qquad (17\text{–}18)$$

Here $\Sigma(\mathcal{M}_k)_P$ represents the moment sum of $I_G\boldsymbol{\alpha}$ and $m\mathbf{a}_G$ (or its components) about P as determined by the data on the kinetic diagram.

$$\Sigma F_x := m \cdot \overrightarrow{a_x}$$

$$F_{x.horiz} := m_{rod} \cdot a_n = 338.236\ N$$

$$\Sigma F_y := m \cdot \overrightarrow{a_y}$$

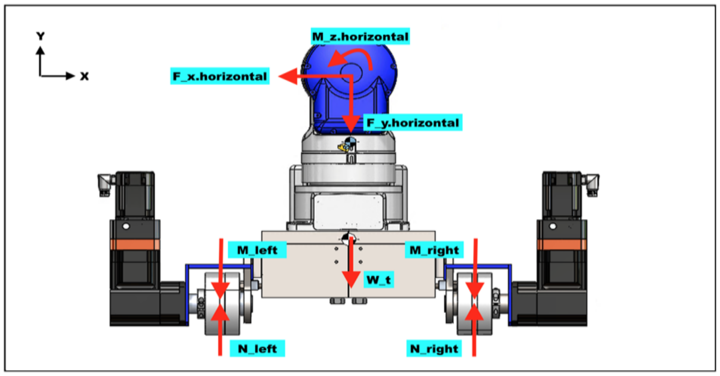
$$W := 120 \cdot 9.81 = 1177.2 \quad N$$

$$F_{y.horiz} := m_{rod} \cdot a_t + W = 1231.032 \quad N$$

$$\Sigma M_p := r_{g.p} \ X \ m \cdot \overrightarrow{a_G} + I_G \cdot \vec{\alpha}$$

$$M_z := m_{rod} \cdot a_t \cdot \frac{l_{rod}}{2} + I_G \cdot \alpha - W \cdot \frac{l_{rod}}{2} = -604.6669 \quad Nm$$

Fy, Fx, and Mp are internal forces between the rod and the rest of the robot body. The forces exerted on the robot body are therefore equal and opposite.

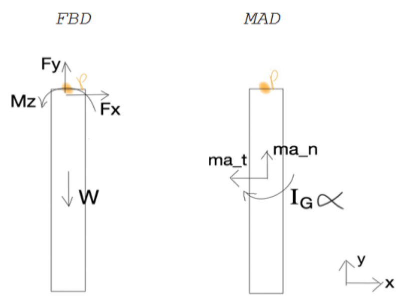


The calculated dynamic forces are used to determine stability for tipping and slipping in the horizontal direction. They are also used in a static FEA analysis to determine the effect of the motion of the arm on the stresses induced throughout the chassis body.

VERTICAL CASE

The approach used to determine the angular acceleration is the same as before. The rod begins at θ=0 and rotates to its maximum angular velocity at θ= π rad, defined in the drawings above. Therefore, the resulting angular acceleration is the same asthe horizontal case, resulting in the same tangential and normal acceleration components.

Manipulator arm FBD and MAD:



Equations of motion:

$$\Sigma F_x := m \cdot \vec{a}_x$$

$$F_{x.vertical} := m_{rod} \cdot a_t = 53.832 \quad N$$

$$\Sigma F_y := m \cdot \vec{a}_y$$

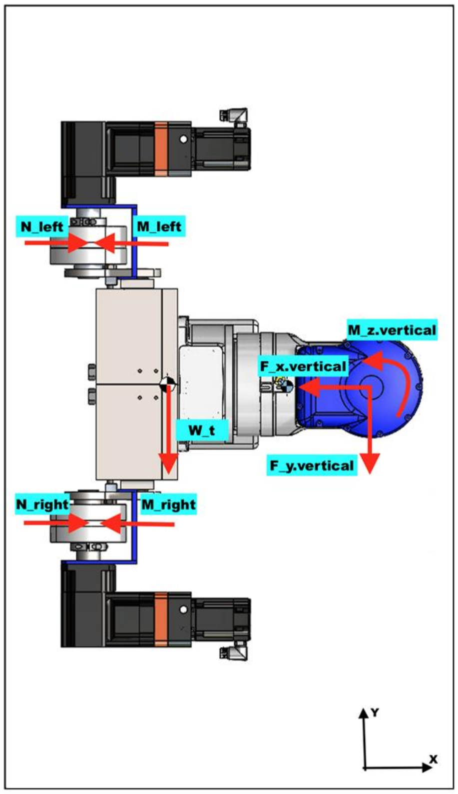
$$W := 120 \cdot 9.81 = 1177.2 \quad N$$

$$F_{y.vertical} := m_{rod} \cdot a_n + W = 1515.4364 \quad N$$

$$\Sigma M_p := r_{g.p} \; X \; m \cdot \vec{a}_G + I_G \cdot \vec{\alpha}$$

$$M_{p.vertical} := \left(\left(-m_{rod} \right) \cdot a_t \cdot \frac{l_{rod}}{2} - I_G \cdot \alpha \right) = -39.2615 \quad Nm$$

Forces from rod are equal and opposite on robot body:



Conclusion

The FBDs with known forces and moments when the assembly is undergoing maximum loading under dynamic movement of the manipulator arm have been provided for the vertical and horizontal robot motion. These are used for both stability and finite element analysis.

Appendix D2: Dynamic Model Appendix C3: Static Model and Braking Torque

This section of the appendix highlights the calculations used to determine the stability of the overall assembly when the robot is completely stationary. Three cases are considered: horizontal driving on wall, vertical driving on wall, and driving on a flat surface.

General

Static Stability Calculations

Author: Eric Wong   Date: April 12, 2021

Objective

To determine reaction forces on the assembly for different orientations. Cases considered:
1. Horizontal driving on wall
2. Vertical driving on wall
3. Driving on flat ground.

The reaction forces will tell us whether or not the robot tips or slides.
Some secondary calculations were done to determine loading on individual components but are not the focus of this code.

Assumptions

1. For driving cases 1 and 3, there is symmetry along the left and right half of robot.
2. Magnetic force of each wheel is treated as a concentrated force.
3. Reaction forces are concentrated loads.

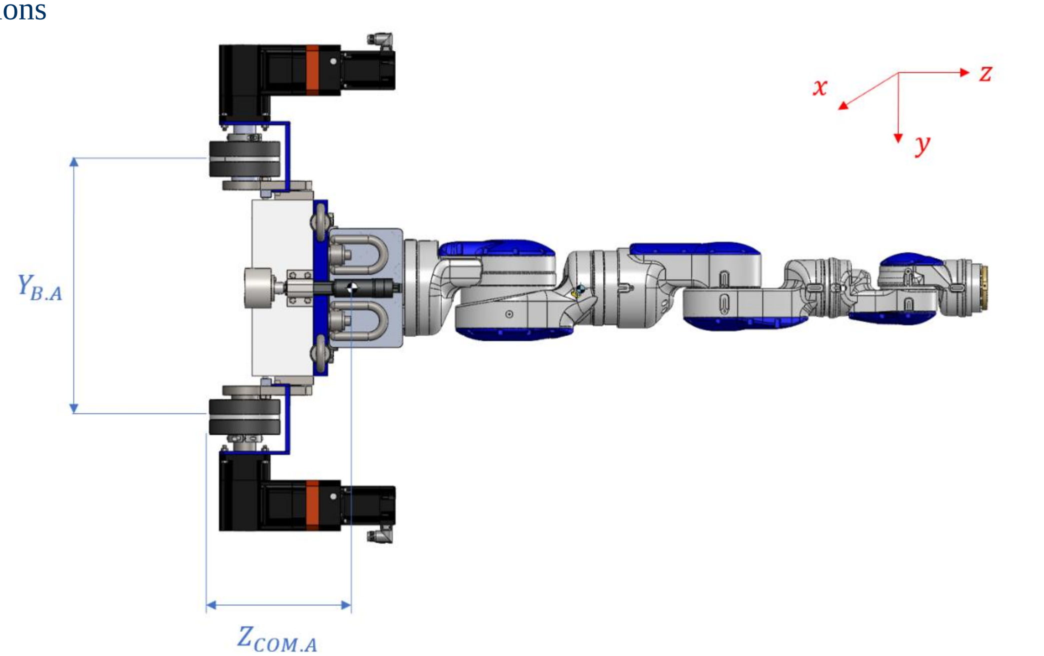
Shared Known Parameters

Some parameters are common across all cases of analysis which are the mass of the system, magnetic force of each wheel, and the number of wheels.

$m_t := 500\ \text{kg} \quad g := g_e \quad F_{mag} := 1800\ \text{N} \cdot 100\ \% \quad n_{wheel} := 8 \quad W := m_t \cdot g = 4903.325\ \text{N}$

Horizontal Driving on Wall

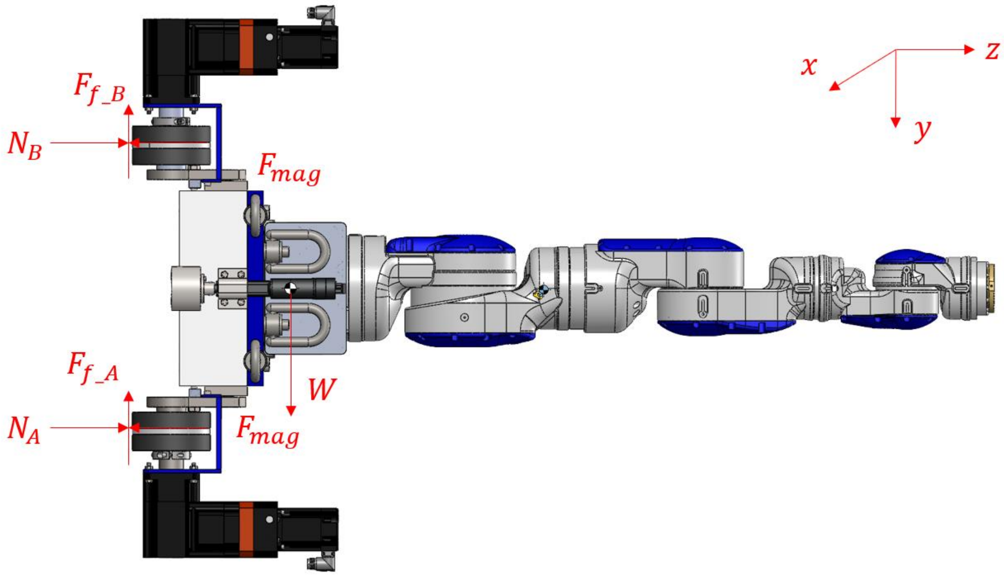
Dimensions



The relevant dimensions are:

$y_{b.a} := 0.563\ \text{m} \quad z_{com.a} := 0.275\ \text{m}$

Calculations



Tipping

Consider the equations of motion:
Taking the moment at A in the x-direction we find:

$$0 = y_{B.A} \frac{n_{wheel}}{2} \left(F_{mag} - N_B\right) - z_{com.A} W$$

Consider the sum of forces in the z-direction:

$$0 = \frac{n_{wheel}}{2} (N_A + N_B) - n_{wheel} F_{mag}$$

Rearranging and solving the two equations:

$$N_B := F_{mag} - \frac{2}{n_{wheel}} \cdot \frac{W \cdot z_{com.a}}{y_{b.a}} = 1201.237\ \text{N}$$

$$N_A := 2 \cdot F_{mag} - N_B = 2398.763\ \text{N}$$

The positive normal forces indicate there is no tipping.

Quasi-Static Analysis for Traversing Obstacle - Slipping

As a conservative estimate of traversing the obstacle, we assumed the loss of the front 2 wheels. The total traction of the robot is then estimated using maximum static friction of each remaining wheel. The static friction between polyurethane and steel is $\mu = 0.5$. There are 3 pairs of wheels. The traction was computed:

$$F_{f\_maxtotal} := \left(\sum \begin{bmatrix} N_A \\ N_B \end{bmatrix}\right) \cdot 3 \cdot 0.5 = 5400\ \text{N}$$

The total traction is greater than the weight of the robot (4903 N) thus no sliding occurs.

Vertical Driving on Wall

The calculations for this case are similar to the previous one except the system is statically indeterminate considering the friction. So as an approximation, we have solved for the average normal force for each wheel pair. Then we assumed that the true normal forces must average to the average normal force. Thus, we were able to approximate the true normal forces by considering the moment about passive pivot.

Note that the calculations are involved and cannot be simplified like in the previous section. Instead the SMATH code is pasted in the following pages for further reading.

Summary of Tipping and Sliding

The summary results were:

$$N_{A2} := F_{mag} - F_{z\_shaft\_A2} = 3536.2717 \text{ N}$$

$$N_{A1} := F_{mag} - F_{z\_shaft\_A1} = 1417.558 \text{ N}$$

$$N_{B2} := F_{mag} - F_{z\_shaft\_B2} = 1603.4198 \text{ N}$$

$$N_{B1} := F_{mag} - F_{z\_shaft\_B1} = 642.7505 \text{ N}$$

The positive normal forces indicate there is no tipping.

$$F_{f\_maxtotal} := \left(\sum \begin{bmatrix} N_{A1} & N_{A2} \\ N_{B1} & 0 \end{bmatrix} \right) \cdot 2 \cdot 0.5 = 5596.5802 \text{ N}$$

The total traction is greater than the weight of the robot (4903 N) thus no sliding occurs.

Braking Torque

The braking torque was determined by considering the wheel with the highest static friction. Thus, the calculated braking torque is a conservative estimate.

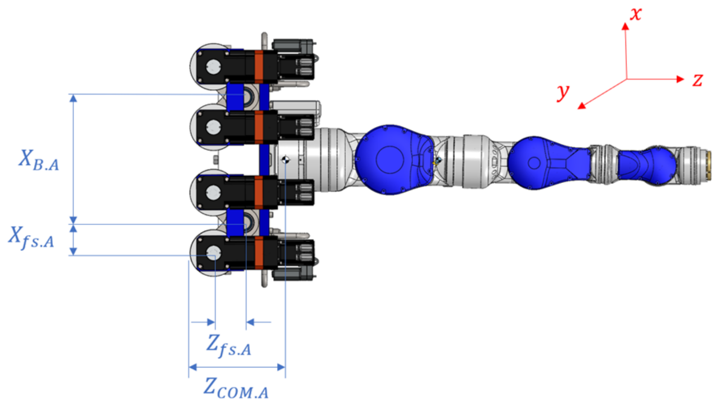
$$T_{brake} := \max \left(\begin{bmatrix} M_{y\_shaft\_A1} \\ M_{y\_shaft\_A2} \end{bmatrix} \right) = 96.3305 \text{ J}$$

The braking torque was determined to be 96.3 $N \cdot m$. The driving torque is computed as the sum of braking torque and rolling resistance.

⊟— Vertical on Wall

⊟— Normal Force

Normal forces are solved the same way as shown in stability calculations. The system is statically indeterminate. As an approximation, each wheel pair will be treated as a singular point of contact. The results are the average normal force for each wheel.

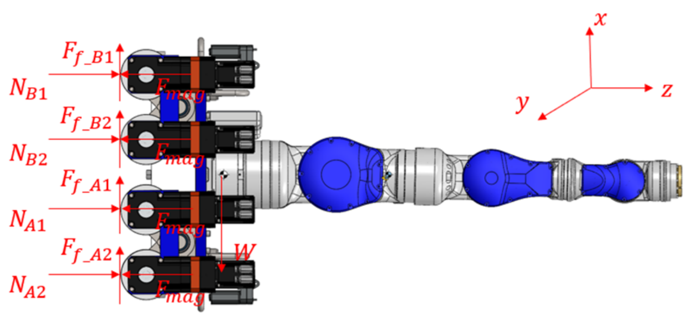


$x_{b.a} := 0.498\ m$

$x_{fs.A} := 103.5\ mm$

$z_{com.a} := 0.26777\ m$

$z_{fs.A} := 130\ mm$



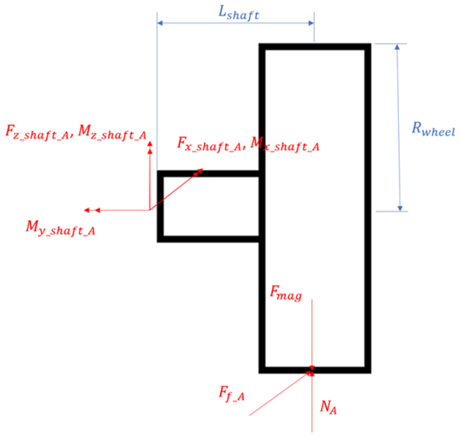
Calculations:

$$N_{B\_AVG} := F_{mag} - \frac{2}{n_{wheel}} \cdot \frac{W \cdot z_{com.a}}{x_{b.a}} = 1140.8819 \text{ N}$$

$$N_{A\_AVG} := 2 \cdot F_{mag} - N_{B\_AVG} = 2459.1181 \text{ N}$$

⊟— Wheel Pin/Shaft Loading

Evidently, the loading conditions on A side are more critical than on B side. Loading analysis will focus on A side. Will still check B side for stability. The loading conditions on the shaft directly connected to the wheel. Here we solve for averaged loading ...



Calculations:

Total friction:

$$F_{f\_total} = \frac{n_{wheel}}{2}(F_{f\_A} + F_{f\_B})$$

Since friction is proportional to normal force:

$$\frac{F_{f\_A}}{N_A} = \frac{F_{f\_A} + F_{f\_B}}{N_A + N_B} \qquad F_{f\_A} = \frac{N_A}{N_A + N_B}(F_{f\_A} + F_{f\_B})$$

Rearranging gives:

$$F_{f\_A} = \frac{2}{n_{wheel}} \frac{N_A}{N_A + N_B} F_{f\_total}$$

$$F_{f\_A\_AVG} := \frac{2}{n_{wheel}} \cdot \frac{N_{A\_AVG}}{N_{A\_AVG} + N_{B\_AVG}} \cdot F_{f\_total} = 837.3511 \text{ N}$$

$$F_{f\_B\_AVG} := \frac{F_{f\_A\_AVG}}{N_{A\_AVG}} \cdot N_{B\_AVG} = 388.4802 \text{ N}$$

Averaged shaft loading conditions (moments ignored, calculated later with more accurate values):

$$F_{x\_shaft\_A\_AVG} := -F_{f\_A\_AVG} = -837.3511 \text{ N}$$

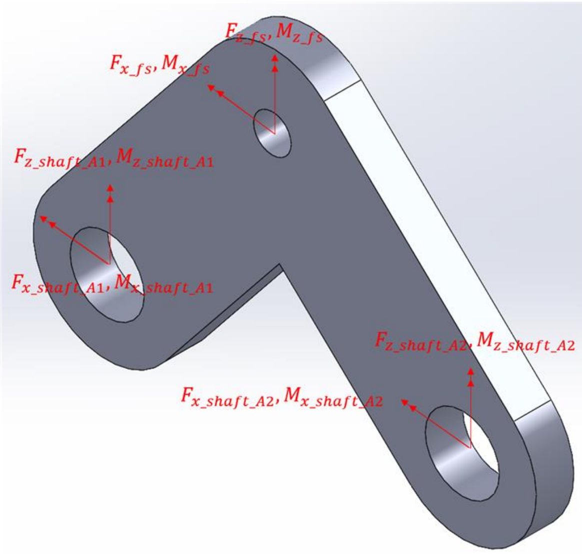
$$F_{z\_shaft\_A\_AVG} := F_{mag} - N_{A\_AVG} = -659.1181 \text{ N}$$

$F_{x\_shaft\_B\_AVG} := -F_{f\_B\_AVG} = -388.4802 \text{ N}$

$F_{z\_shaft\_B\_AVG} := F_{mag} - N_{B\_AVG} = 659.1181 \text{ N}$

V-Support Loading

This section solves for the loading conditions on the V-support.



The forces must sum to twice the average value in general:

$$F_{x\_shaft\_A1} + F_{x\_shaft\_A2} = 2F_{x\_shaft\_A\_AVG}$$

$$F_{z\_shaft\_A1} + F_{z\_shaft\_A2} = 2F_{z\_shaft\_A\_AVG}$$

For static equilibrium, the sum of moments must be zero. In y-direction:

$$z_{fs.A}(F_{x\_shaft\_A1} + F_{x\_shaft\_A2}) = x_{fs.A}(F_{z\_shaft\_A2} - F_{z\_shaft\_A1})$$

Combining the results:

$$F_{z\_shaft\_A2} = \frac{z_{fs.A}}{x_{fs.A}}(F_{x\_shaft\_A\_AVG}) + F_{z\_shaft\_A\_AVG}$$

$$F_{z\_shaft\_A1} = 2F_{z\_shaft\_A\_AVG} - F_{z\_shaft\_A2}$$

$F_{z\_shaft\_A2} := \frac{z_{fs.A}}{x_{fs.A}} \cdot F_{x\_shaft\_A\_AVG} + F_{z\_shaft\_A\_AVG} = -1710.8634 \text{ N}$

$F_{z\_shaft\_A1} := 2 \cdot F_{z\_shaft\_A\_AVG} - F_{z\_shaft\_A2} = 392.6272 \text{ N}$

$F_{z\_shaft\_B2} := \frac{z_{fs.A}}{x_{fs.A}} \cdot F_{x\_shaft\_B\_AVG} + F_{z\_shaft\_B\_AVG} = 171.172 \text{ N}$

$F_{z\_shaft\_B1} := 2 \cdot F_{z\_shaft\_B\_AVG} - F_{z\_shaft\_B2} = 1147.0643\ N$

$N_{A2} := F_{mag} - F_{z\_shaft\_A2} = 3510.8634\ N$

$N_{A1} := F_{mag} - F_{z\_shaft\_A1} = 1407.3728\ N$

$N_{B2} := F_{mag} - F_{z\_shaft\_B2} = 1628.828\ N$

$N_{B1} := F_{mag} - F_{z\_shaft\_B1} = 652.9357\ N$

$\min\left(\begin{bmatrix} N_{A1} & N_{A2} \\ N_{B1} & N_{B2} \end{bmatrix}\right) > 0 = 1$

Friction is proportional to normal force:

$$\frac{F_{f\_A1}}{N_{A1}} = \frac{F_{f\_A2}}{N_{A2}} = \frac{F_{f\_A\_AVG}}{N_{A\_AVG}}$$

$F_{f\_A2} := \frac{N_{A2}}{N_{A\_AVG}} \cdot F_{f\_A\_AVG} = 1195.4795\ N$

$F_{f\_A1} := \frac{N_{A1}}{N_{A\_AVG}} \cdot F_{f\_A\_AVG} = 479.2227\ N$

Shaft 2:

$F_{x\_shaft\_A2} := -F_{f\_A2} = -1195.4795\ N$

$F_{z\_shaft\_A2} := F_{mag} - N_{A2} = -1710.8634\ N$

$M_{x\_shaft\_A2} := L_{shaft} \cdot \left(N_{A2} - F_{mag}\right) = 102.6518\ J$

$M_{y\_shaft\_A2} := R_{wheel} \cdot F_{f\_A2} = 95.6384\ J$

$M_{z\_shaft\_A2} := -L_{shaft} \cdot F_{f\_A2} = -71.7288\ J$

Note that the signs for shaft loading are based off of the drawing of the wheel rather than the V-support. For the V-support, use the opposite sign.

Shaft 1:

$F_{x\_shaft\_A1} := -F_{f\_A1} = -479.2227\ N$

$F_{z\_shaft\_A1} := F_{mag} - N_{A1} = 392.6272\ N$

$M_{x\_shaft\_A1} := L_{shaft} \cdot \left(N_{A1} - F_{mag}\right) = -23.5576\ J$

$M_{y\_shaft\_A1} := R_{wheel} \cdot F_{f\_A1} = 38.3378\ J$

$M_{z\_shaft\_A1} := -L_{shaft} \cdot F_{f\_A1} = -28.7534\ J$

V-Support Calculations:

$F_{x\_fs} := F_{x\_shaft\_A1} + F_{x\_shaft\_A2} = -1674.7021\ N$

Signs for V-support are based off the drawing of the V-support.

$F_{z\_fs} := F_{z\_shaft\_A1} + F_{z\_shaft\_A2} = -1318.2363\ N$

$M_{x\_fs} := M_{x\_shaft\_A1} + M_{x\_shaft\_A2} = 79.0942\ J$

$M_{z\_fs} := M_{z\_shaft\_A1} + M_{z\_shaft\_A2} = -100.4821\ J$

$$T_{brake} := \max\left(\begin{bmatrix} M_{y\_shaft\_A1} \\ M_{y\_shaft\_A2} \end{bmatrix}\right) = 95.6384\ J$$

Sliding Calcs:

$$F_{f\_maxtotal} := \left(\sum \begin{bmatrix} N_{A1} & N_{A2} \\ N_{B1} & N_{B2} \end{bmatrix}\right) \cdot 2 \cdot 0.5 = 7200\ N$$

$$W = 4903.325\ N$$

```
if W < F_f_maxtotal = "No sliding"
  "No sliding"
else
  "Sliding"
```

Conservative estimate of max friction force when going over obstacle. Assuming loss of front 2 wheels:

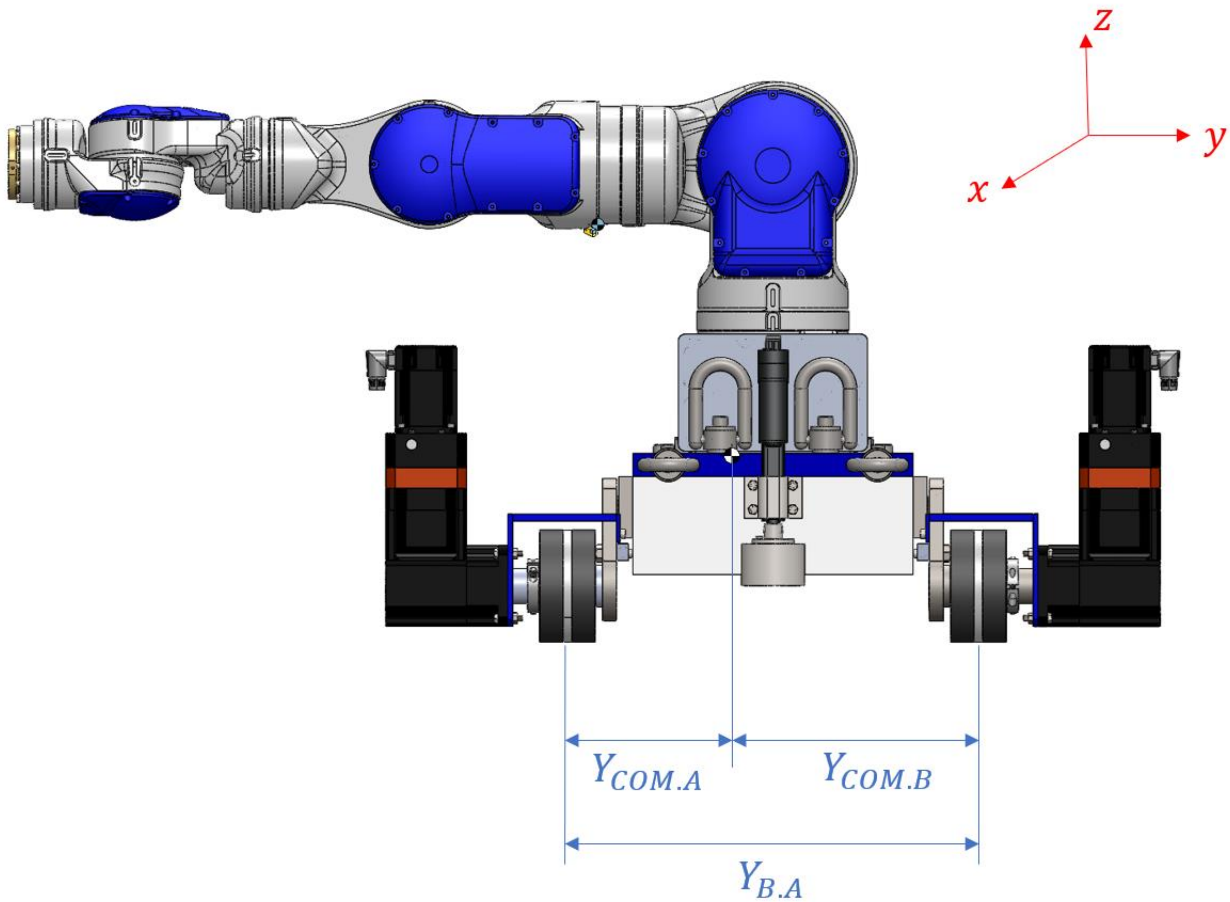
$$F_{f\_maxtotal} := \left(\sum \begin{bmatrix} N_{A1} & N_{A2} \\ N_{B1} & 0 \end{bmatrix}\right) \cdot 2 \cdot 0.5 = 5571.172\ N$$

$$W = 4903.325\ N$$

$$\sigma_{traction} := \frac{F_{f\_maxtotal}}{W} = 1.1362$$

```
if W < F_f_maxtotal = "No sliding"
  "No sliding"
else
  "Sliding"
```

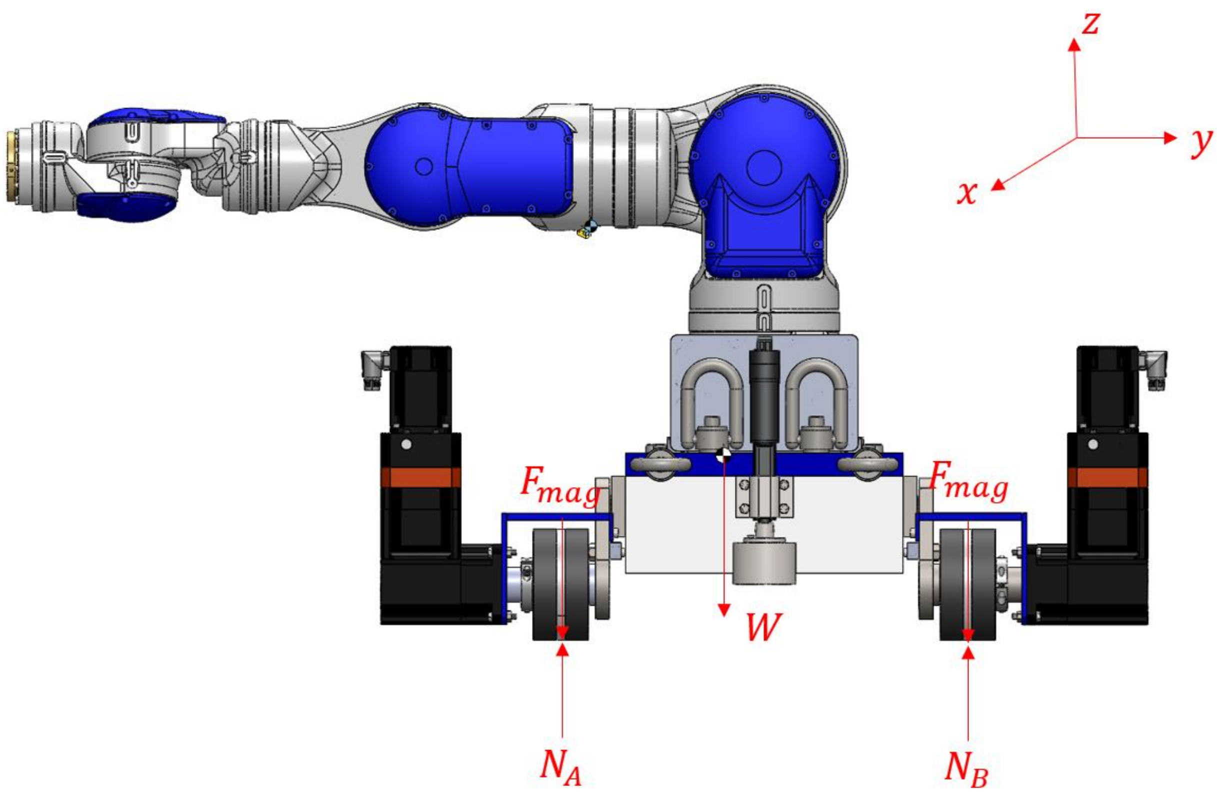
Driving on Flat Surface
Dimensions



Relevant dimensions:

$y_{b.a} := 0.563\ m$ $y_{com.b} := 0.344\ m$ $y_{com.a} := (0.563 - 0.344)\ m = 0.219\ m$

Calculations



This case is similar to the first case, driving horizontally on a wall. The only difference is that the arm is extended to side of the robot. Sliding is not a concern while on flat ground, so calculations focused only on tipping.

$$N_B := F_{mag} + \frac{2}{n_{wheel}} \cdot \frac{W \cdot y_{com.a}}{y_{b.a}} = 2276.8331 \text{ N}$$

$$N_A := 2 \cdot F_{mag} - N_B + \frac{2}{n_{wheel}} \cdot W = 2548.9981 \text{ N}$$

The positive normal forces indicate there is no tipping.

Appendix D3: Dynamic Model

The forces derived from the dynamic loads in Appendix D1 were used as forces in the static model to account for the inertial forces applied by the manipulator. The calculations were the same as in Append D2, but with the additional forces and moments imposed by the manipulator. As the calculations are the same, the explanation will not be repeated. Instead the results will be summarized and the full SMATH code will be available for reading on the following pages.

Also note that the robot should never be driving while the manipulator is moving, so quasistatic analysis was not performed for the dynamic cases.

Horizontal Driving on Wall

$$N_B := F_{mag} - \frac{2}{n_{wheel}} \cdot \frac{z_{com.a} \cdot W - M_{p\_inertial} + z_{p.A} \cdot F_{y\_inertial} - y_{p.A} \cdot F_{z\_inertial}}{y_{b.a}} = 1116.0171 \text{ N}$$

$$N_A := \frac{2}{n_{wheel}} \cdot \left(F_{z\_inertial} + n_{wheel} \cdot F_{mag} \right) - N_B = 2497.4409 \text{ N}$$

The positive normal forces indicate there is no tipping.

Vertical Driving on Wall

$$N_{A2} := F_{mag} - F_{z\_shaft\_A2} = 3688.1463 \text{ N}$$

$$N_{A1} := F_{mag} - F_{z\_shaft\_A1} = 1484.2097 \text{ N}$$

$$N_{B2} := F_{mag} - F_{z\_shaft\_B2} = 1465.0031 \text{ N}$$

$$N_{B1} := F_{mag} - F_{z\_shaft\_B1} = 589.5568 \text{ N}$$

The positive normal forces indicate there is no tipping.

Driving on Flat Surface

$$N_B := F_{mag} + \frac{2}{n_{wheel}} \cdot \frac{W \cdot y_{com.a} + M_{p\_inertial} - z_{p.A} \cdot F_{y\_inertial} + y_{p.A} \cdot F_{z\_inertial}}{y_{b.a}} = 2461.3473 \text{ N}$$

$$N_A := \frac{2}{n_{wheel}} \cdot \left(W + F_{z\_inertial} + n_{wheel} \cdot F_{mag} \right) - N_B = 2377.9844 \text{ N}$$

The positive normal forces indicate there is no tipping.

SMATH Code

Dynamic Stability Calculations

Author: Eric Wong Date: April 12, 2021

Objective

To determine reaction forces on the assembly for different orientations. Cases considered:
1. Horizontal driving on wall
2. Vertical driving on wall
3. Driving on flat ground.

The reaction forces will tell us whether or not the robot tips or slides.
Some secondary calculations were done to determine loading on individual components but are not the focus of this code.

Assumptions

1. For driving cases 1 and 3, there is symmetry along the left and right half of robot.
2. Magnetic force of each wheel is treated as a concentrated force.
3. Reaction forces are concentrated loads.

Notes:

1. Dots indicate relative coordinates. "B.A" mean B relative to A.

Common Parameters

$m_t := 500\ \text{kg}$ $g := g_e$ $F_{mag} := 1800\ \text{N}$ $n_{wheel} := 8$

$W := m_t \cdot g = 4903.325\ \text{N}$ $F_{f\_total} := W$

Common Dimensions:

$D_{wheel} := 160\ \text{mm}$ $R_{wheel} := \frac{D_{wheel}}{2}$

$L_{shaft} := 60\ \text{mm}$

$z_{fs.A} := 130\ \text{mm}$

$x_{fs.A} := 103.5\ \text{mm}$

These dimensions locate the manipulators first joint, where the inertial forces act.

$x_{p.A} := 300\ \text{mm}$

$y_{p.A} := 280.10\ \text{mm}$

$z_{p.A} := 600\ \text{mm}$

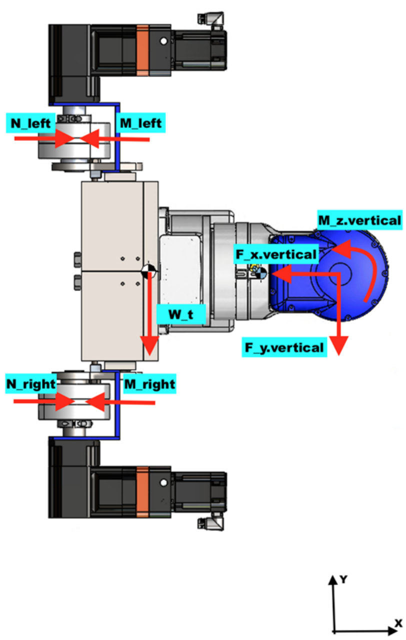
Horizontal on Wall

Inertial Forces:

$F_{y\_inertial} := 1515.4364\ \text{N} - 120\ \text{kg}\ g_e = 338.6384\ \text{N}$

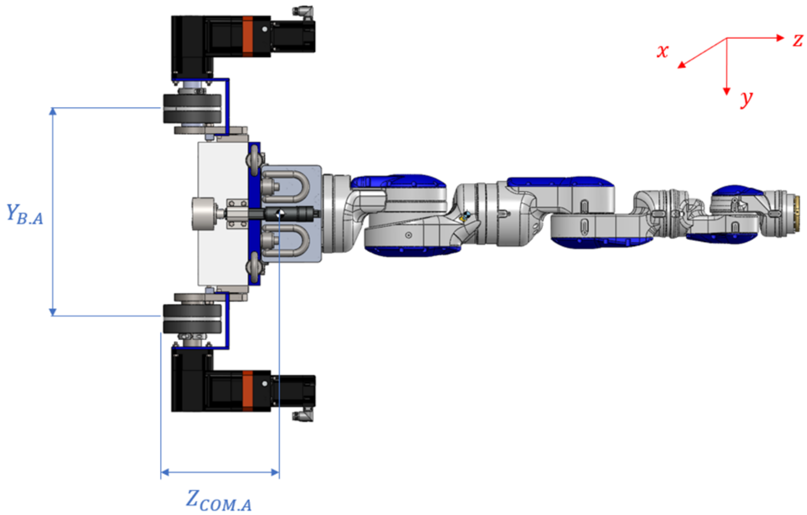
$F_{z\_inertial} := 53.832\ \text{N}$

$M_{p\_inertial} := -39.2615\ \text{N m}$



⊟— Normal Force

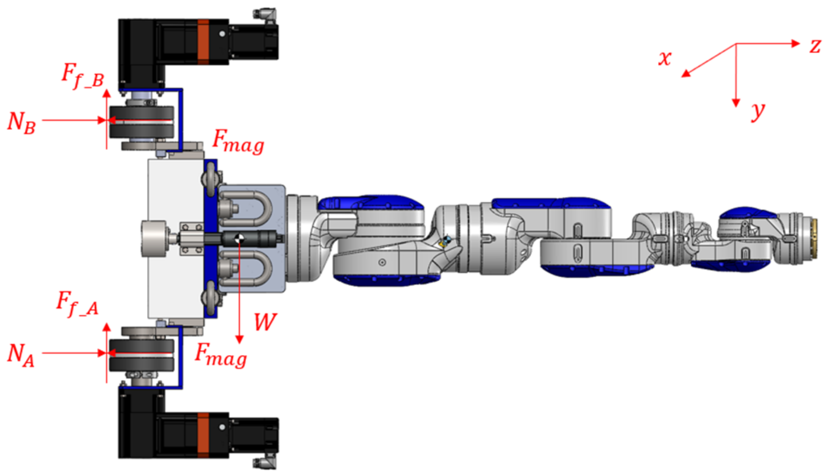
Section for solving normal forces.



Dimensions:

$y_{b.a} := 0.563\ \text{m}$

$z_{com.a} := 0.26777\ \text{m}$



Calculations:

The summation of moments about A must be zero to avoid tipping:

$$0 = y_{B.A}\frac{n_{wheel}}{2}\left(F_{mag} - N_B\right) - z_{com.A}W + M_{p\_inertial} - z_{p.A}F_{y\_inertial} + y_{p.A}F_{z\_inertial}$$

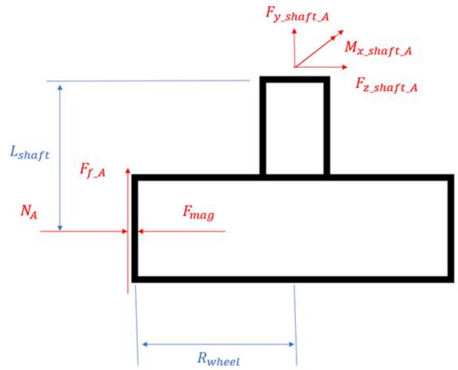
$$N_B = F_{mag} - \frac{2}{y_{B.A}n_{wheel}}\left(z_{com.A}W - M_{p\_inertial} + z_{p.A}F_{y\_inertial} - y_{p.A}F_{z\_inertial}\right)$$

$$N_B := F_{mag} - \frac{2}{n_{wheel}} \cdot \frac{z_{com.a} \cdot W - M_{p\_inertial} + z_{p.A} \cdot F_{y\_inertial} - y_{p.A} \cdot F_{z\_inertial}}{y_{b.a}} = 1116.0171\ \text{N}$$

$$N_A := \frac{2}{n_{wheel}} \cdot \left(F_{z\_inertial} + n_{wheel} \cdot F_{mag}\right) - N_B = 2497.4409\ \text{N}$$

Wheel Pin/Shaft Loading

Evidently, the loading conditions on A side are more critical than on B side. Analysis will focus on A side. The loading conditions on the shaft directly connected to the wheel:



Calculations:

Total friction:

$$F_{f\_total} = \frac{n_{wheel}}{2}(F_{f\_A} + F_{f\_B})$$

Since friction is proportional to normal force:

$$\frac{F_{f\_A}}{N_A} = \frac{F_{f\_A} + F_{f\_B}}{N_A + N_B} \qquad F_{f\_A} = \frac{N_A}{N_A + N_B}(F_{f\_A} + F_{f\_B})$$

Rearranging gives:

$$F_{f\_A} = \frac{2}{n_{wheel}} \frac{N_A}{N_A + N_B} F_{f\_total}$$

$$F_{f\_A} := \frac{2}{n_{wheel}} \cdot \frac{N_A}{N_A + N_B} \cdot F_{f\_total} = 847.2331 \text{ N}$$

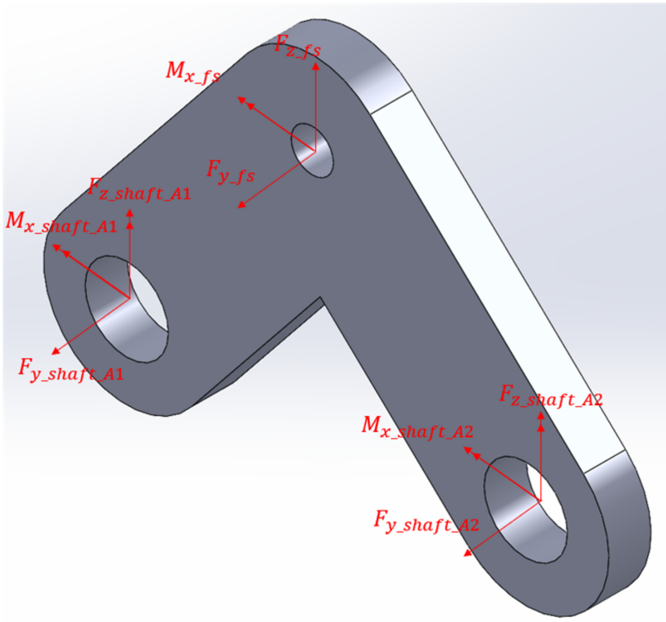
Shaft loading conditions:

$$F_{y\_shaft\_A} := -F_{f\_A} = -847.2331 \text{ N}$$

$$F_{z\_shaft\_A} := F_{mag} - N_A = -697.4409 \text{ N}$$

$$M_{x\_shaft\_A} := L_{shaft} \cdot (N_A - F_{mag}) - R_{wheel} \cdot F_{f\_A} = -25.9322 \text{ J}$$

V-Support Loading



Known:

Loading on V-support is equal and oppposite to loading on the wheel/shaft.

$F_{y\_shaft\_A1} := -F_{y\_shaft\_A}$ $\quad F_{y\_shaft\_A2} := -F_{y\_shaft\_A} = 847.2331\ \text{N}$

$F_{z\_shaft\_A1} := -F_{z\_shaft\_A}$ $\quad F_{z\_shaft\_A2} := -F_{z\_shaft\_A} = 697.4409\ \text{N}$

$M_{x\_shaft\_A1} := -M_{x\_shaft\_A}$ $\quad M_{x\_shaft\_A2} := -M_{x\_shaft\_A} = 25.9322\ \text{J}$

Calculations:

$$F_{y\_fs} := -\left(F_{y\_shaft\_A1} + F_{y\_shaft\_A2}\right) = -1694.4661\ \text{N}$$

$$F_{z\_fs} := -\left(F_{z\_shaft\_A1} + F_{z\_shaft\_A2}\right) = -1394.8818\ \text{N}$$

$$M_{x\_fs} := -\left(M_{x\_shaft\_A1} + M_{x\_shaft\_A2}\right) - z_{fs.A} \cdot \left(F_{y\_shaft\_A1} + F_{y\_shaft\_A2}\right) = -272.145\ \text{J}$$

Sliding Calculations:

$$F_{f\_maxtotal} := \left(\sum \begin{bmatrix} N_A \\ N_B \end{bmatrix}\right) \cdot 4 \cdot 0.5 = 7226.916\ \text{N}$$

$W = 4903.325\ \text{N}$

```
if  W < F_f_maxtotal  = "No sliding"
  "No sliding"
else
  "Sliding"
```

Conservative estimate of max friction force when going over obstacle. Assuming loss of front 2 wheels:

$$F_{f\_maxtotal} := \left(\sum \begin{bmatrix} N_A \\ N_B \end{bmatrix}\right) \cdot 3 \cdot 0.5 = 5420.187\ \text{N} \qquad W = 4903.325\ \text{N}$$

```
if  W < F_f_maxtotal = "No sliding"
  "No sliding"
else
  "Sliding"
```

Vertical on Wall

Inertial Forces:

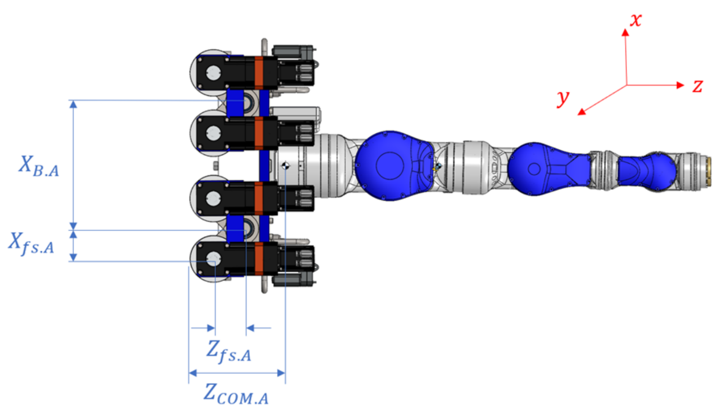
$F_{y\_inertial} := 1515.4364\ \text{N} - 120\ \text{kg}\ g_e = 338.6384\ \text{N}$

$F_{z\_inertial} := 53.832\ \text{N}$

$M_{p\_inertial} := -39.2615\ \text{N m}$

Normal Force

Normal forces are solved the same way as shown in stability calculations. The system is statically indeterminate. As an approximation, each wheel pair will be treated as a singular point of contact. The results are the average normal force for each wheel.



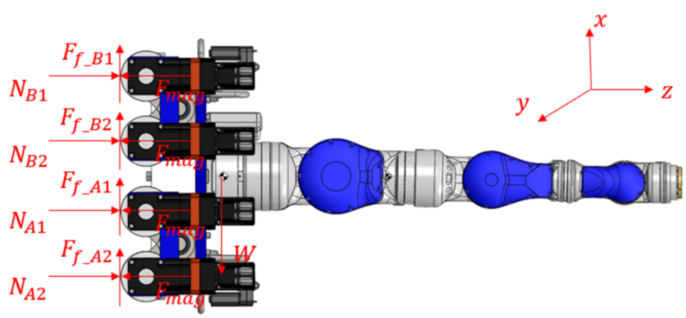
Dimensions:

$x_{b.a} := 0.498\ \text{m}$

$x_{fs.A} := 103.5\ \text{mm}$

$z_{com.a} := 0.26777\ \text{m}$

$z_{fs.A} := 130\ \text{mm}$



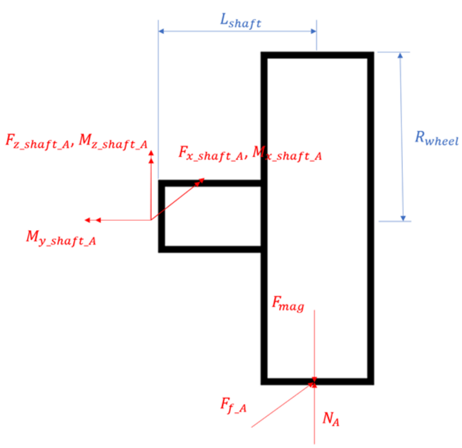
Calculations:

$$N_{B\_AVG} := F_{mag} - \frac{2}{n_{wheel}} \cdot \frac{z_{com.a} \cdot W - M_{p\_inertial} + z_{p.A} \cdot F_{y\_inertial} - x_{p.A} \cdot F_{z\_inertial}}{x_{b.a}} = 1027.28 \text{ N}$$

$$N_{A\_AVG} := \frac{2}{n_{wheel}} \cdot \left(F_{z\_inertial} + n_{wheel} \cdot F_{mag}\right) - N_{B\_AVG} = 2586.178 \text{ N}$$

Wheel Pin/Shaft Loading

Evidently, the loading conditions on A side are more critical than on B side. Loading analysis will focus on A side. Will still check B side for stability. The loading conditions on the shaft directly connected to the wheel. Here we solve for averaged loading conditions.



Calculations:

Total friction:

$$F_{f\_total} = \frac{n_{wheel}}{2}(F_{f\_A} + F_{f\_B})$$

Since friction is proportional to normal force:

$$\frac{F_{f\_A}}{N_A} = \frac{F_{f\_A} + F_{f\_B}}{N_A + N_B} \qquad F_{f\_A} = \frac{N_A}{N_A + N_B}(F_{f\_A} + F_{f\_B})$$

Rearranging gives:

$$F_{f\_A} = \frac{2}{n_{wheel}} \frac{N_A}{N_A + N_B} F_{f\_total}$$

$$F_{f\_A\_AVG} := \frac{2}{n_{wheel}} \cdot \frac{N_{A\_AVG}}{N_{A\_AVG} + N_{B\_AVG}} \cdot F_{f\_total} = 877.3363 \text{ N}$$

$$F_{f\_B\_AVG} := \frac{F_{f\_A\_AVG}}{N_{A\_AVG}} \cdot N_{B\_AVG} = 348.495 \text{ N}$$

Averaged shaft loading conditions (moments ignored, calculated later with more accurate values):

$F_{x\_shaft\_A\_AVG} := -F_{f\_A\_AVG} = -877.3363\ N$

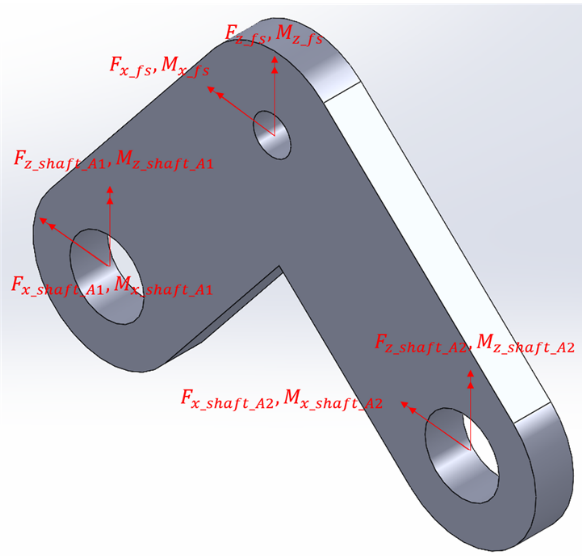
$F_{z\_shaft\_A\_AVG} := F_{mag} - N_{A\_AVG} = -786.178\ N$

$F_{x\_shaft\_B\_AVG} := -F_{f\_B\_AVG} = -348.495\ N$

$F_{z\_shaft\_B\_AVG} := F_{mag} - N_{B\_AVG} = 772.72\ N$

V-Support Loading

This section solves for the loading conditions on the V-support.



The forces must sum to twice the average value in general:

$$F_{x\_shaft\_A1} + F_{x\_shaft\_A2} = 2F_{x\_shaft\_A\_AVG}$$

$$F_{z\_shaft\_A1} + F_{z\_shaft\_A2} = 2F_{z\_shaft\_A\_AVG}$$

For static equilibrium, the sum of moments must be zero. In y-direction:

$$z_{fs.A}(F_{x\_shaft\_A1} + F_{x\_shaft\_A2}) = x_{fs.A}(F_{z\_shaft\_A2} - F_{z\_shaft\_A1})$$

Combining the results:

$$F_{z\_shaft\_A2} = \frac{z_{fs.A}}{x_{fs.A}}(F_{x\_shaft\_A\_AVG}) + F_{z\_shaft\_A\_AVG}$$

$$F_{z\_shaft\_A1} = 2F_{z\_shaft\_A\_AVG} - F_{z\_shaft\_A2}$$

$F_{z\_shaft\_A2} := \frac{z_{fs.A}}{x_{fs.A}} \cdot F_{x\_shaft\_A\_AVG} + F_{z\_shaft\_A\_AVG} = -1888.1463\ N$

$F_{z\_shaft\_A1} := 2 \cdot F_{z\_shaft\_A\_AVG} - F_{z\_shaft\_A2} = 315.7903 \text{ N}$

$F_{z\_shaft\_B2} := \frac{z_{fs.A}}{x_{fs.A}} \cdot F_{x\_shaft\_B\_AVG} + F_{z\_shaft\_B\_AVG} = 334.9969 \text{ N}$

$F_{z\_shaft\_B1} := 2 \cdot F_{z\_shaft\_B\_AVG} - F_{z\_shaft\_B2} = 1210.4432 \text{ N}$

$N_{A2} := F_{mag} - F_{z\_shaft\_A2} = 3688.1463 \text{ N}$

$N_{A1} := F_{mag} - F_{z\_shaft\_A1} = 1484.2097 \text{ N}$

$N_{B2} := F_{mag} - F_{z\_shaft\_B2} = 1465.0031 \text{ N}$

$N_{B1} := F_{mag} - F_{z\_shaft\_B1} = 589.5568 \text{ N}$

$\frac{W}{8} = 612.9156 \text{ N}$

$\min\left(\begin{bmatrix} N_{A1} & N_{A2} \\ N_{B1} & N_{B2} \end{bmatrix}\right) > 0 = 1$

Friction is proportional to normal force:

$$\frac{F_{f\_A1}}{N_{A1}} = \frac{F_{f\_A2}}{N_{A2}} = \frac{F_{f\_A\_AVG}}{N_{A\_AVG}}$$

$F_{f\_A2} := \frac{N_{A2}}{N_{A\_AVG}} \cdot F_{f\_A\_AVG} = 1251.1685 \text{ N}$

$F_{f\_A1} := \frac{N_{A1}}{N_{A\_AVG}} \cdot F_{f\_A\_AVG} = 503.504 \text{ N}$

Shaft 2:

$F_{x\_shaft\_A2} := -F_{f\_A2} = -1251.1685 \text{ N}$

$F_{z\_shaft\_A2} := F_{mag} - N_{A2} = -1888.1463 \text{ N}$

$M_{x\_shaft\_A2} := L_{shaft} \cdot \left(N_{A2} - F_{mag}\right) = 113.2888 \text{ J}$

$M_{y\_shaft\_A2} := R_{wheel} \cdot F_{f\_A2} = 100.0935 \text{ J}$

$M_{z\_shaft\_A2} := -L_{shaft} \cdot F_{f\_A2} = -75.0701 \text{ J}$

Note that the signs for shaft loading are based off of the drawing of the wheel rather than the V-support. For the V-support, use the opposite sign.

Shaft 1:

$F_{x\_shaft\_A1} := -F_{f\_A1} = -503.504 \text{ N}$

$F_{z\_shaft\_A1} := F_{mag} - N_{A1} = 315.7903 \text{ N}$

$M_{x\_shaft\_A1} := L_{shaft} \cdot \left(N_{A1} - F_{mag}\right) = -18.9474 \text{ J}$

$M_{y\_shaft\_A1} := R_{wheel} \cdot F_{f\_A1} = 40.2803 \text{ J}$

$M_{z\_shaft\_A1} := -L_{shaft} \cdot F_{f\_A1} = -30.2102 \text{ J}$

V-Support Calculations:

$F_{x\_fs} := F_{x\_shaft\_A1} + F_{x\_shaft\_A2} = -1754.6726 \text{ N}$

Signs for V-support are based off the drawing of the V-support.

$F_{z\_fs} := F_{z\_shaft\_A1} + F_{z\_shaft\_A2} = -1572.356 \text{ N}$

$M_{x\_fs} := M_{x\_shaft\_A1} + M_{x\_shaft\_A2} = 94.3414 \text{ J}$

$M_{z\_fs} := M_{z\_shaft\_A1} + M_{z\_shaft\_A2} = -105.2804\ J$

$T_{brake} := \max\left(\begin{bmatrix} M_{y\_shaft\_A1} \\ M_{y\_shaft\_A2} \end{bmatrix}\right) = 100.0935\ J$

Sliding Calcs:

$F_{f\_maxtotal} := \left(\sum \begin{bmatrix} N_{A1} & N_{A2} \\ N_{B1} & N_{B2} \end{bmatrix}\right) \cdot 2 \cdot 0.5 = 7226.916\ N$

$W = 4903.325\ N$

```
if  W < F_f_maxtotal = "No sliding"
  "No sliding"
else
  "Sliding"
```

Conservative estimate of max friction force when going over obstacle. Assuming loss of front 2 wheels:

$F_{f\_maxtotal} := \left(\sum \begin{bmatrix} N_{A1} & N_{A2} \\ N_{B1} & 0 \end{bmatrix}\right) \cdot 2 \cdot 0.5 = 5761.9129\ N$

$W = 4903.325\ N$

```
if  W < F_f_maxtotal = "No sliding"
  "No sliding"
else
  "Sliding"
```

Horizontal - Lateral Arm

Inertial Forces:

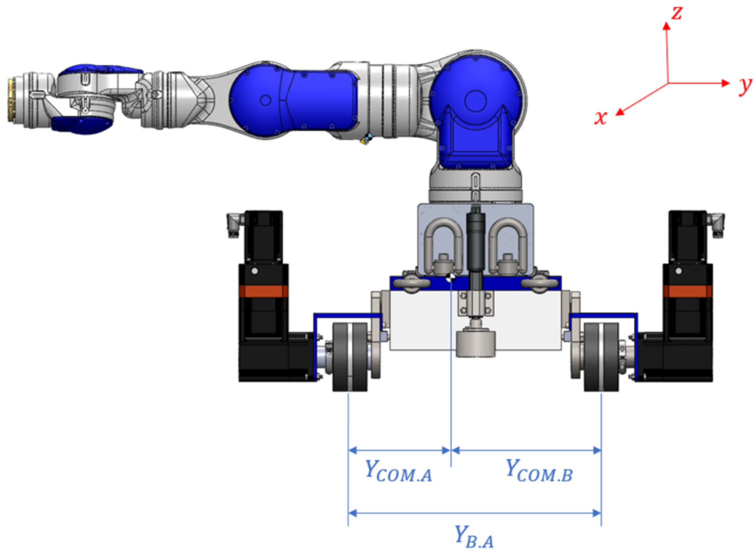
$F_{y\_inertial} := 337\ N$

$F_{z\_inertial} := 1230.8\ N - 120\ kg\ g_e = 54.002\ N$

$M_{p\_inertial} := 602.6\ N\ m$

Normal Force

Normal forces are solved the same way as shown in stability calculations. Assume the arm is rotating CW which is the worse case scenario.

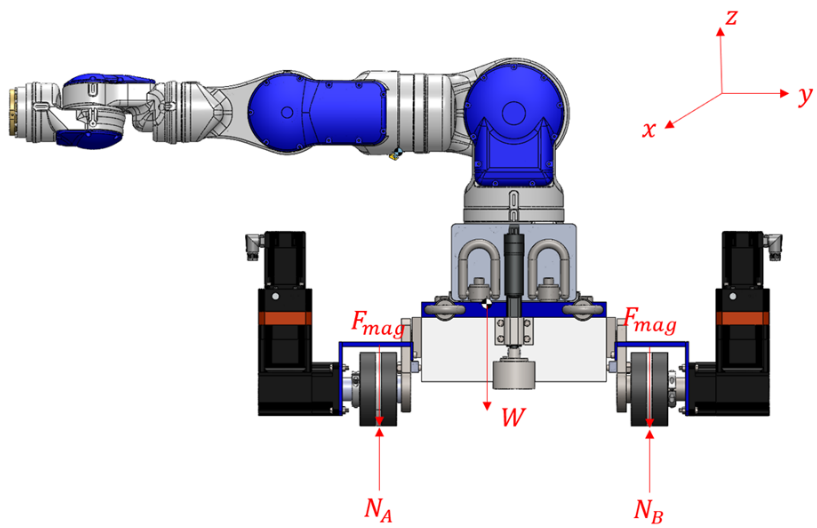


Dimensions:

$y_{b.a} := 0.563\ \text{m}$

$y_{com.b} := 0.344\ \text{m}$

$y_{com.a} := y_{b.a} - y_{com.b} = 0.219\ \text{m}$



Calculations:

The summation of moments about A must be zero to avoid tipping:

$$0 = \frac{y_{B.A}\left(N_B - F_{mag}\right)n_{wheel}}{2} - y_{com.A}W - M_p + z_{p.A}F_y - y_{p.A}F_z$$

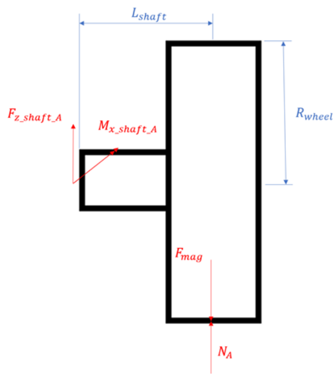
$$N_B = \frac{2}{n_{wheel}y_{B.A}}\left(y_{com.A}W + M_p - z_{p.A}F_y + y_{p.A}F_z\right) + F_{mag}$$

$$N_B := F_{mag} + \frac{2}{n_{wheel}} \cdot \frac{W \cdot y_{com.a} + M_{p\_inertial} - z_{p.A} \cdot F_{y\_inertial} + y_{p.A} \cdot F_{z\_inertial}}{y_{b.a}} = 2461.3473 \text{ N}$$

$$N_A := \frac{2}{n_{wheel}} \cdot \left(W + F_{z\_inertial} + n_{wheel} \cdot F_{mag}\right) - N_B = 2377.9844 \text{ N}$$

Wheel Pin/Shaft Loading

Evidently, the loading conditions on A side are more critical than on B side. Analysis will focus on A side. The loading conditions on the shaft directly connected to the wheel:



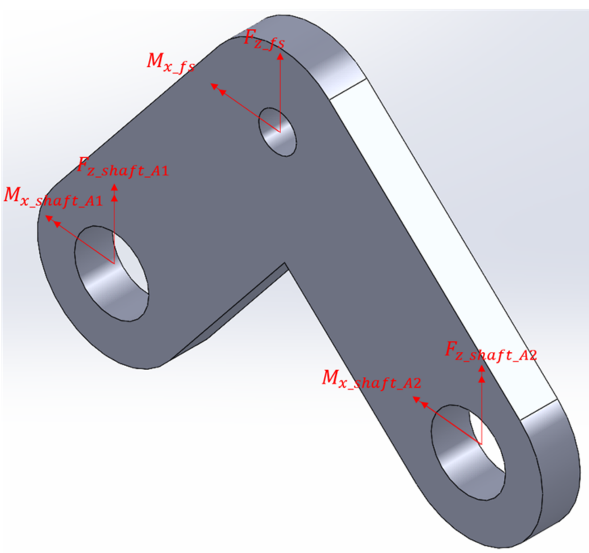
Calculations:

Shaft loading conditions:

$$F_{z\_shaft\_A} := F_{mag} - N_A = -577.9844 \text{ N}$$

$$M_{x\_shaft\_A} := L_{shaft} \cdot \left(N_A - F_{mag}\right) = 34.6791 \text{ J}$$

V-Support Loading



Known:

Loading on V-support is equal and oppposite to loading on the wheel/shaft.

$F_{z\_shaft\_A1} := -F_{z\_shaft\_A}$ $F_{z\_shaft\_A2} := -F_{z\_shaft\_A} = 577.9844\ N$

$M_{x\_shaft\_A1} := -M_{x\_shaft\_A}$ $M_{x\_shaft\_A2} := -M_{x\_shaft\_A} = -34.6791\ J$

Calculations:

$F_{z\_fs} := -\left(F_{z\_shaft\_A1} + F_{z\_shaft\_A2}\right) = -1155.9689\ N$

$M_{x\_fs} := -\left(M_{x\_shaft\_A1} + M_{x\_shaft\_A2}\right) = 69.3581\ J$

Appendix D4: Testing for Sliding at Any Angle

Author: George Felobes
Date: February 25, 2021
Revised: April 1, 2021

Objective:

The aim of this analysis is to determine if OmiBot would experience sliding due to the large weight of the robot. The study was done at an arbitrary angle that range from a horizontal to a vertical surface. The results at all angles are studied.

Solution Method:

The static equilibrium was determined for the model to calculate the frictional forces at each point. This was then compared to the maximum frictional values. MATLAB was used to calculate the sliding conditions at all angles B, where B range from 0 degrees (horizontal surface) to 90 degrees (vertical surface).
The variables, and methodology are described in the MATLAB code comments. The code can be found at the end of this analysis after the conclusion section.

Methodology summary:

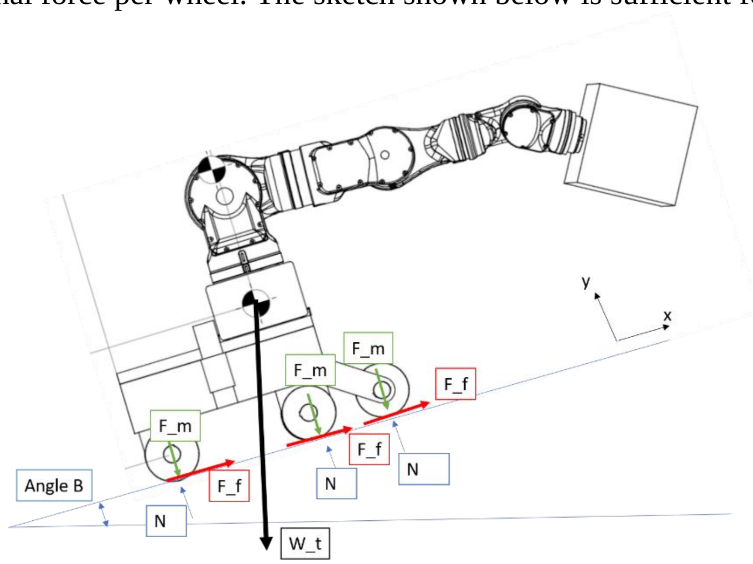
- The forces along the x-direction were used to determine the frictional forces required to prevent sliding.
- The sum of the forces along the y-directions allows us to determine the reaction forces on each wheel. It was initially assumed that the weight would be equally distributed along each wheel, leading to equal reaction forces. This assumption would be analyzed further in the analysis.
- By considering the most conservative design with the lowest number of wheels, and thus, the lowest number of contact points, and therefore, frictional forces. If the case of 6 wheels is sufficient, then the cases of 8 wheels (concept design 3) is sufficient.
- For static analysis, the inertial forces, inertial moments, accelerations, and driving torques are set to zero.

Assumptions:

1. System is a rigid body. With no deformation or relative motion between the parts.
2. Wheels have negligible deformation with the wall. Point of contact rather than an area.
3. Position of the platform is perfectly changing with angle, with no deformation. Centre of mass relative to a platform coordinate does not change location.
4. Material for the platform is 4130 steel
5. The magnetic force provided by each of the wheel is constant and equal between all the wheels.
6. The system platform is assumed to be symmetric along the axis shown. This allows the assumption that the normal forces are equivalent on each side of the axis.
7. The gravitational acceleration is constant.
8. Dimensions are based off the drawings

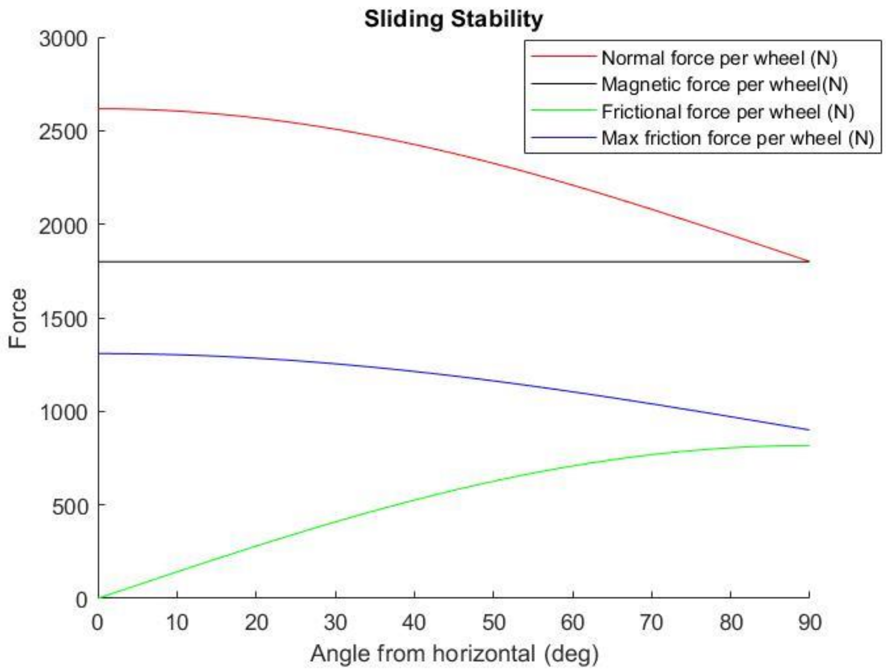
Sketch:

The purpose of this sketch is to show the force locations for a robot climbing a surface at an angle B from the horizontal. The main parameter in the analysis is the number of wheels to determine the average frictional force per wheel. The sketch shown below is sufficient for this purpose.



Analysis:

The code was run, and the following plot was produced. The plot shows how the reaction forces required change with angle. As expected, the reaction forces are the highest in the horizontal position (B=0) since it would oppose both the weight and the magnetic adhesive forces. In the vertical position (B=90), the normal force is equal to the magnetic adhesive force. Note that the forces shown are per wheel. Moreover, the frictional forces required at all angles studied and compared to the maximum frictional force available at that point. Since the required frictional force is always less than the maximum frictional force. Sliding does not occur.



Conclusion:

As the frictional force required per wheel is always less than the maximum frictional force per wheel. The robot will not slip for the case of 6 wheels, and thus, will not slip for the 8 wheel design.

Code:

```
%Calculate stability (static and dynamic) with respect to angle
%% Input Variables

n=6; % # of wheels

mu_s = 0.5;           % static friction factor (unitless)
mu_d = 0.2;           % dynamic friction coefficient (unitless)

T = 0;                  % Torque per wheel in N.m. Ignored for static
analysis
r = 0.08;             % radius of the wheel in m
a_x = 0;               % required acceleration along the wall. Zero for
static analysis
e = 1;                  % accelerate components like motors and gears
(unitless)

F_m_defined = 1800;           % Magnetic adhesion force per wheel in N

g = 9.81;             % gravitational acceleration in m/s^2
%Total weight of the system:
W = 500*g; %in kg
```

```
Blow=0;                %Define angle B lower limit in degree
Bhigh=90;              %Define angle B upper limit in degree

%% This section for sliding stability
%We can also get relationship of reaction force with angle
%We can determine acceleration achieved with a certain torque
%Modify code to obtain torque required


%Determine external force on the wheel
F_ext = T/r;

%Discretize angle B to calculate forces
B =  (Blow*pi/180):0.01:(Bhigh*pi/180);


for i=1:length(B)
    %Sum of the forces in y leads to equation for reaction force "N"
    %define magnetic force
    F_m(i)= F_m_defined; %in N

    %Calculate reaction force for every angle "B"
    N(i) = (W*cos(B(i)) + n*F_m(i))/n; %Equation for reaction force "N"
    F_f_max(i) = mu_s*N(i); %Calculate the maximum friction for rolling
(no sliding)

    %Sum of the forces in x
    F_f(i) = ((W/g)*a_x + W*sin(B(i)) - F_ext)/n; %used friction force

    %Check if maximum friction occurs
    if F_f(i)>= F_f_max(i)
    fprintf('The robot will slide, max friction is %f, at an angle %f
degrees from the horizontal \n', F_f(i), (180*B(i))/pi)
    end
end


%Clear Plot
clf
%Plotting overall
hold on
title('Sliding Stability')
plot((180*B)/pi,N(1:length(B)),'r')
plot((180*B)/pi,F_m(1:length(B)),'k')
plot((180*B)/pi,F_f(1:length(B)),'g')
plot((180*B)/pi,F_f_max(1:length(B)),'b')
%Adjust plot parameters
xlim([Blow,Bhigh])
legend('Normal force per wheel (N)','Magnetic force per
wheel(N)','Frictional force per wheel (N)','Max friction force per wheel
(N)')
```

```
xlabel('Angle from horizontal (deg)')
ylabel('Force')
hold off

%Plotting frictional force
title('Friction')
hold on
plot((180*B)/pi,F_f(1:length(B)),'r')
plot((180*B)/pi,F_f_max(1:length(B)),'b')
xlim([Blow,Bhigh])
xlabel('Angle from horizontal (deg)')
ylabel('Force per wheel (N)')
legend('Friction force per wheel (N)','Max friction force per wheel(N)')
hold off
```

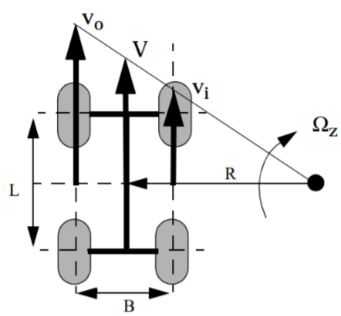
Appendix D5: Skid Steering Vertical Turn Radius and Vehicle Angular Velocity

Author: George Felobes
Date: March 12, 2021
Revised: April 12, 2021

Objective:
The objective of this analysis is to determine the required relative velocity between the inner and outer wheels to achieve a certain turning radius and turning rate. This will aid in the development of the control system to allow skid steering.
Solution Method:
The kinematic analysis of skid steering allows a preliminary determination of wheel velocities given the vehicle dimensions, the desired radius, and the desired turn rate. The kinematic model can provide accurate results if an accurate slippage model is used. Wheel velocities and turn radius can only be assumed to be estimates in the absence of a longitudinal slip model [1].
When a driving torque is applied to a wheel, the wheel travels less distance than a wheel moving in an unloaded and free rolling condition. This is referred to as longitudinal slip. Slippage is accounted for by calculating a longitudinal slip percentage using the relationship:

$$i = \left(1 - \frac{V}{r\omega}\right) \times 100$$

Where V is the linear velocity of the wheel center. r is the rolling radius, ω is the angular velocity of the wheel.
The skid steering model can be described by the following figure [1]:



Variables:

v_o = outside wheel velocity [m/s]
v_i = inside wheel velocity [m/s]
V = vehicle velocity [m/s]
Ω_z = vehicle angular velocity [rad/s]
R = vehicle turn radius [m]
L = vehicle length [m]
B = vehicle width [m]

The turning radius is calculated using similar triangles as shown below [1]:

$$\frac{v_o}{v_i} = \frac{R + \frac{B}{2}}{R - \frac{B}{2}}$$

$$R = \frac{\frac{B}{2}\left(\frac{v_o}{v_i} + 1\right)}{\left(\frac{v_o}{v_i} - 1\right)} = \frac{B}{2}\left(\frac{v_o + v_i}{v_o - v_i}\right)$$

When accounting for slippage, the turning radius becomes:

$$R' = \frac{B}{2}\left(\frac{v_o(1 - i_o) + v_i(1 - i_i)}{v_o(1 - i_o) - v_i(1 - i_i)}\right)$$

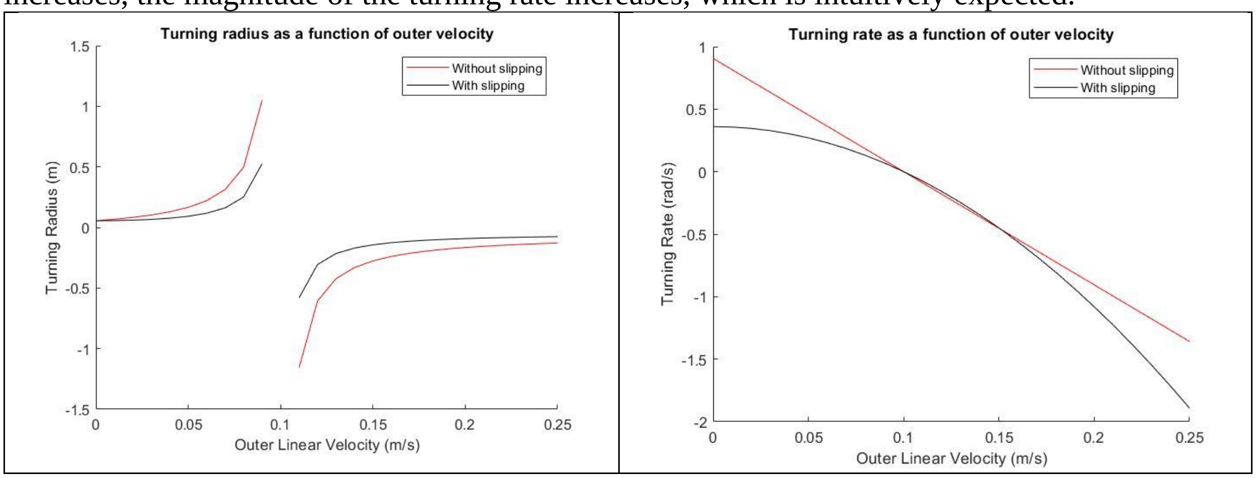
The turn rate (yaw velocity) is calculated using:

$$\Omega_z = \frac{v_o + v_i}{2R} = \frac{v_i\left(\frac{v_o}{v_i} - 1\right)}{B}$$

By quantifying the amount of steering in a path, relative to path and vehicle dimensions, understanding can be gained of the optimal driving mode to be used to traverse a given path.

Analysis:

To simplify the plots, the inner velocity was fixed to a linear velocity of 0.1 m/s. A MATLAB code was developed to try various combinations of the outer velocity compared to a constant inner velocity. The results are described in the following figures. As observed, the turning radius is infinite when the outer and inner velocities are equal (in this case they are both 0.1 m/s). This is indicated by the asymptotic behavior at 0.1m/s outer velocity. This is the case for both cases, with and without slip. The turning rate sign is different depending on the magnitude of the outer velocity compared to the inner velocity. Which is expected, when the outer velocity is less than the inner velocity, the turning radius is positive, and negative when its is greater than the inner velocity.
The turning rate is linear without slip, and non-linear when slip is considered. The turning rate decreases as the outer velocity increases. As expected, the turning rate is zero when the outer velocity is equal to the inner velocity. As the difference between the outer and inner velocities increases, the magnitude of the turning rate increases, which is intuitively expected.



Conclusion:

To conclude, the turning radius, and turning rate can be determined based on the linear velocities of the inner and outer wheels. Given that the linear velocities can be determined directly from the rotational speed of the wheel (based on the motor output), control systems can be developed to adjust the motor output to achieve a certain turning radius and rate to allow the rotation of Omibot on the surface.

References:

[1] Shamah, Benjamin. (2011). Experimental Comparison of Skid Steering vs. Explicit Steering for Wheeled Mobile Robot," M.Sc.

Code:

```
%Note the slippage model provide results that can only be treated as an
%estimate, since it is only a longitudinal slip model
% source:
%
https://www.ri.cmu.edu/pub_files/pub1/shamah_benjamin_1999_1/shamah_benjamin_
1999_1.pdf

%Assumed parameters based on the problem studied
B = (110.4)/1000; %vehicle width (m) based from the drawings
r = (160/2)/1000; %wheel radius (m)
w = 3.1415; %rotational speed of the wheel (rad/s)

%maximum linear velocity, due to the codes and standards
v_max = 0.25; % m/s


%Define an array of value for the outer and inner linear velocities

vo_data = 0:0.01:v_max; %outside wheel velocity (m/s)
vi_data = 0:0.01:v_max; %inside wheel velocity (m/s)
L = length(vo_data);

vo = zeros(2, L);
vo(1,:)= 0.1;
vo(2,:)= vo_data;

vi = zeros(2, L);
vi(1,:)= vi_data;
vi(2,:)= 0.1;

%Loop to find the slippage factor, radius, and yaw rate at various outer
%and inner wheel linear velocities

for j = 1:2
    for i = 1:L
        %Slippage factors
        i_o(j,i) = 1-(vo(j,i)/(r*w)); %slippage on the outer wheel (unitless)
        i_i(j,i) = 1-(vi(j,i)/(r*w)); %slippage on the inner wheel (unitless)


        %Radius of turn assuming zero slippage
        R(j,i)  =  (B/2)*((vo(j,i)+vi(j,i))/(vo(j,i)-vi(j,i)));  %vehicle  turn
radius (m)
        %Radius of turn accounting for slippage
        R_prime(j,i)         =          (B/2)*((vo(j,i)*(1-i_o(j,i))+vi(j,i)*(1-
i_i(j,i)))/(vo(j,i)*(1-i_o(j,i))-vi(j,i)*(1-i_i(j,i)))); % vehicle turn radius
accounting for slippage (m)

        %rate of turning assuming zero slippage
        yaw(j,i)   =  (vo(j,i)+vi(j,i))/(2*R(j,i));  %vehicle  angular  velocity
(rad/s)
        %rate of turning accounting for slippage
```

```
        yaw_prime(j,i)            =            (vo(j,i)*(1-i_o(j,i))+vi(j,i)*(1-
i_i(j,i)))/(2*R_prime(j,i)); %vehicle angular velocity accounting for slippage
(rad/s)

    end

end

%Plotting
f1 = figure;
hold on
%Adjust plot parameters
xlim([0,v_max])
xlabel('Outer Linear Velocity (m/s)')
ylabel('Turning Radius (m)')
str = sprintf('Turning radius as a function of outer velocity');
title(str)

plot(vo(2,:), R(1,:),'r')
plot(vo(2,:), R_prime(1,:),'k')
legend('Without slipping','With slipping')
hold off

%Plotting
f2 = figure;
hold on
%Adjust plot parameters
xlim([0,v_max])
xlabel('Outer Linear Velocity (m/s)')
ylabel('Turning Rate (rad/s)')
str = sprintf('Turning rate as a function of outer velocity');
title(str)

plot(vo(2,:), yaw(1,:),'r')
plot(vo(2,:), yaw_prime(1,:),'k')
legend('Without slipping','With slipping')
hold off
```

Appendix D6: Drive Shaft

Title - Bearing Calculation for Wheel Drive Shaft
Date - March 12, 2021
Author - Calvin Chen

Objective

Determine the minimum bearing size to withstand radial and axial loading on the wheel drive shaft
Parametric equation will be derived and bearing size will be iterated with step shaft iterative design

Variables

Xm : distance of the motor
Fw - weight of the robot
Xw - distance of wheel
Xb1 - distance of bearing 1
Rw(x,y,z) : Force in the (x,y,z) direction of the wheel

Mw(x,y,z): Moment in the (x,y,z) direction of the wheel
Rb(x,y,z) : Force in the (x,y,z) direction of the bearing
Mb(x,y,z): Moment in the (x,y,z) direction of the bearing
Fr : Radial force due to the gearhub and motor

d1 wheel shaft diameter
d2 step shaft diameter
d3 bearing shaft diameter

Additional varibles are defined as the calculation progresses

Solution Method

Shaft Deflection Calculation for Shaft with 1 gear shaft input, 1 wheel and 1 Ball Bearings
Note : Datum is on left end of beam established at x=0

Assume gravity forces are insignificant compared to applied loading.

Assume the material used is 4130 Normalize Steel Cold Drawn

$g := g_e = 9.8066 \frac{m}{s^2}$ Gravity

$theda := \frac{90\ deg}{2} = 0.7854$

$\omega_{motor} := 1750\ rpm$

$T_{in} := 1.85\ N\ m$

$GR := 100$

$\omega_{out} := \frac{\omega_{motor}}{GR} = 17.5\ rpm$

$T_{out} := T_{in} \cdot GR = 185\ N\ m$

Add control system to regualte torque to the rated capacity of the gear head

$T_{out} := 185\ N\ m$

Assumption: robot will work 8 hours a day, 5 days a week for 1 year

$n_{cycle1} := \omega_{out} \cdot 60 \frac{min}{hr} \cdot 8 \frac{hr}{day} \cdot 5 \frac{day}{week} \cdot 52 \frac{week}{yr} \cdot 1\ yr = 2.184 \cdot 10^{6}\ rev$

From Manufacture spec

Motor Weight

$F_m := 2.9\ kg \cdot 9.81 \frac{m}{s^2} = 28.449\ N$

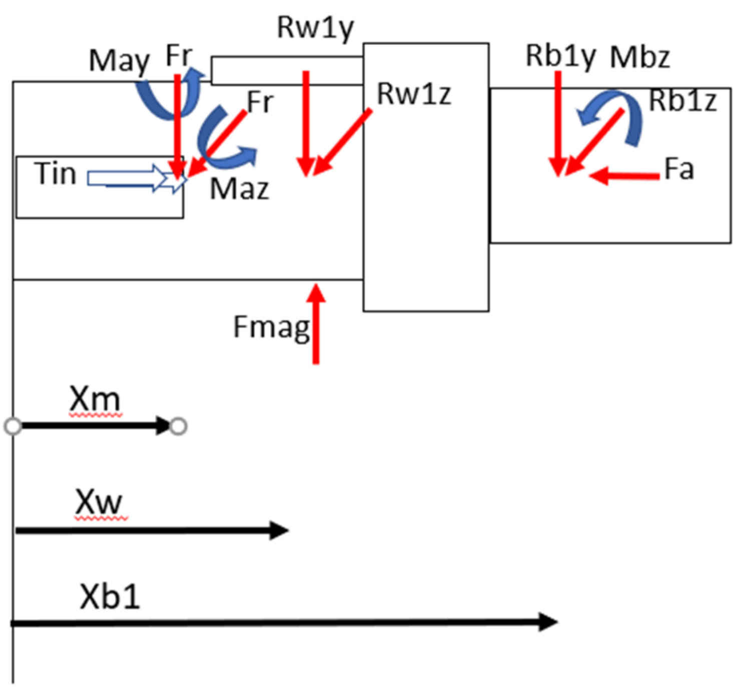
Gearhead Weight

$F_{gh} := 12\ kg \cdot 9.81 \frac{m}{s^2} = 117.72\ N$

Radial Load

$F_r := F_m + F_{gh} = 146.169\ N$

Shaft Deflection Calculation for shaft with 1 motor shaft, 1 magnetic wheel and 1 ball bearings
NOTE : Datum is establies on left end of the beam (x=0)
Assume gravity forces are insignifacnt compared to applied loading.



Define forces on the shaft

$M_{robot} := 500\ kg$ Mass of the robot

$F_w := M_{robot} \cdot g = 4903.325\ N$ force of the robot due to gravity

Horizontal Case

X-Y plane

Shaft contains Gear shaft, magnetic wheel, 1 ball bearing

Distance of motor:

$x_m := 45.01\ \text{mm}$

$F_{mag} := 1800\ \text{N}$

Distance of wheel:

$x_w := 80\ \text{mm}$

Distance of bearing 1:

$x_{b1} := 130\ \text{mm}$

Force in y-direction of bearing:

$R_{b1y} := 577.984\ \text{N} = 577.984\ \text{N}$

Force in y-direction of wheel:

$R_{w1y} := \left(F_r + R_{b1y} + F_{mag}\right) = 2524.153\ \text{N}$

Moment Balance

Moment About y

$M_{by} := 0\ \text{N m}$

$M_{ay} := \left(R_{b1y} \cdot \left(x_w - x_{b1}\right) + F_r \cdot \left(x_w - x_m\right) + M_{by}\right) = -23.7847\ \text{J}$

Material Properties of Shaft

Length of shaft:

$L := 160\ \text{mm}$

Define the singularity function:

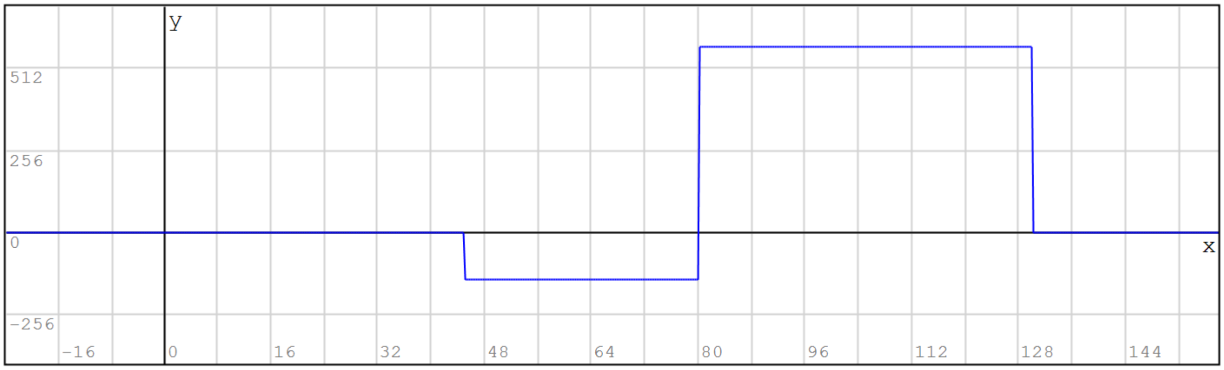
$$S(x, a, n) := \text{if } \left(\left((x - a) > 0\right) \wedge (n \geq 0)\right) \; (x-a)^n \; \text{else} \; 0$$

Shear and bending moment eqns:

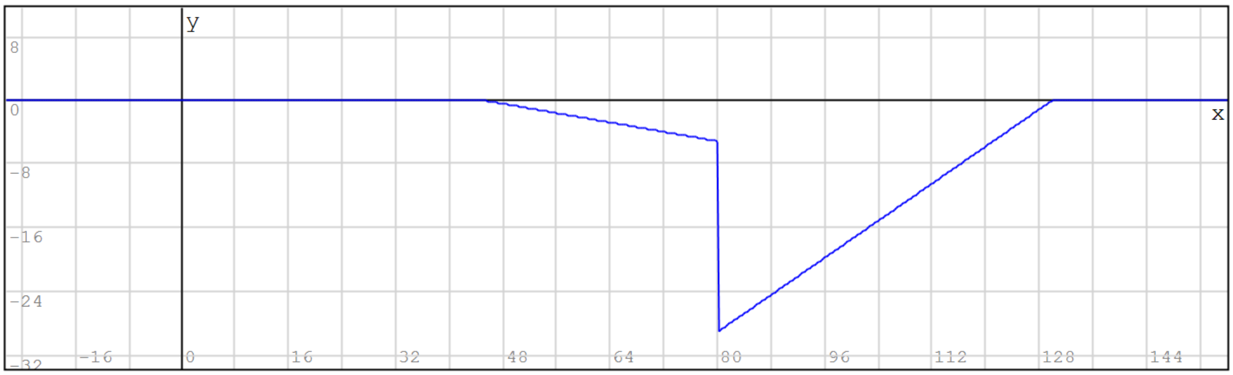
$$q_y(x) := \left(\left(-R_{b1y}\right) \cdot S\left(x, x_{b1}, -1\right) + \left(R_{w1y}\right) \cdot S\left(x, x_w, -1\right) - F_r \cdot S\left(x, x_m, -1\right)\right)$$

$$V_y(x) := \left(\left(\left(-R_{b1y}\right) \cdot S\left(x, x_{b1}, 0\right) + \left(R_{w1y}\right) \cdot S\left(x, x_w, 0\right) - F_r \cdot S\left(x, x_m, 0\right) - F_{mag} \cdot S\left(x, x_w, 0\right)\right)\right)$$

$$M_y(x) := \left(\left(-R_{b1y}\right) \cdot S\left(x, x_{b1}, 1\right) + \left(R_{w1y}\right) \cdot S\left(x, x_w, 1\right) - F_r \cdot S\left(x, x_m, 1\right) + M_{ay} \cdot S\left(x, x_w, 0\right) - F_{mag} \cdot S\left(x, x_w, 1\right) + M_{by} \cdot S\left(x, x_{b1}, 0\right)\right)$$



$V_y\left(x\ \mathrm{mm}\right)$



$M_y\left(x\ \mathrm{mm}\right)$

Moments at critical points in xy plane (total moment):

$M_{b1y} := M_y\left(x_{b1}\right) = 0\ \mathrm{N\ m}$

$M_{m1y} := M_y\left(x_m\right) = 0\ \mathrm{N\ m}$

$M_{wy} := M_y\left(x_w\right) = -5.1145\ \mathrm{N\ m}$

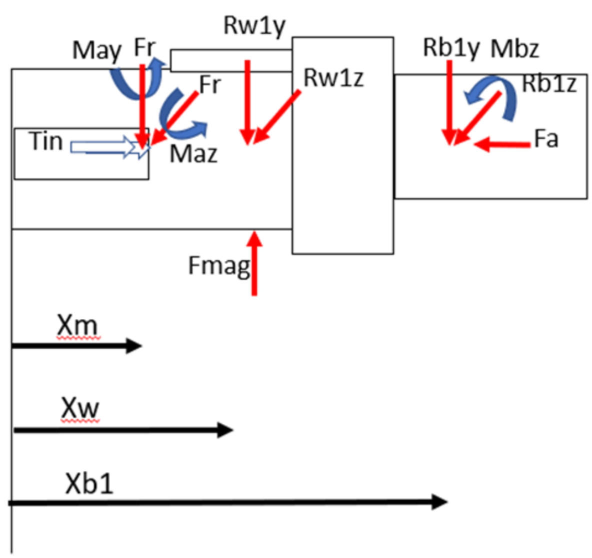
$V_{b1y} := V_y\left(x_{b1}\right) = 577.984\ \mathrm{N}$

$V_{m1y} := V_y\left(x_m\right) = 0\ \mathrm{N}$

$V_{wy} := V_y\left(x_w\right) = -146.169\ \mathrm{N}$

y-z plane (Note that radial forces act on this plane)

Shaft contains Gear shaft, magnetic wheel, 1 ball bearing



Distance of motor:

$x_m := 45.01\ mm$

$F_{mag} := 1800\ N$

Distance of wheel:

$x_w := 80\ mm$

Distance of bearing 1:

$x_{b1} := 130\ mm$

Force in z-direction of bearing conservative assumption of Rb1y=Rb1z:

$R_{b1z} := R_{b1y} = 577.984\ N$

Force in z-direction of wheel:

$R_{w1z} := \left(R_{b1z} + F_r\right) = 724.153\ N$

Moment Balance

$M_{bz} := 28.8992\ N\ m$

$M_{az} := \left(R_{b1z} \cdot \left(x_w - x_{b1}\right) + F_r \cdot \left(x_w - x_m\right) + M_{bz}\right) = 5.1145\ J$

Length of shaft:

$L := 160\ mm$

Define the singularity function:

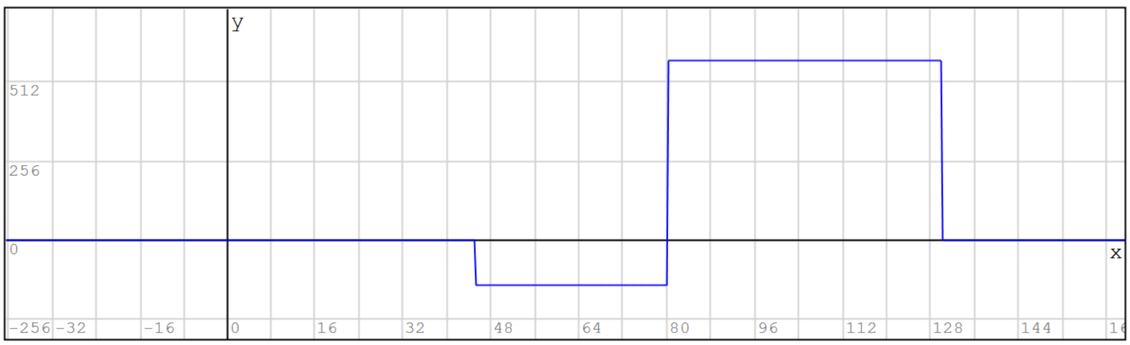
$$S(x, a, n) := \text{if } (((x-a) > 0) \wedge (n \geq 0))\ (x-a)^n \ \text{else}\ 0$$

Shear and bending moment eqns:

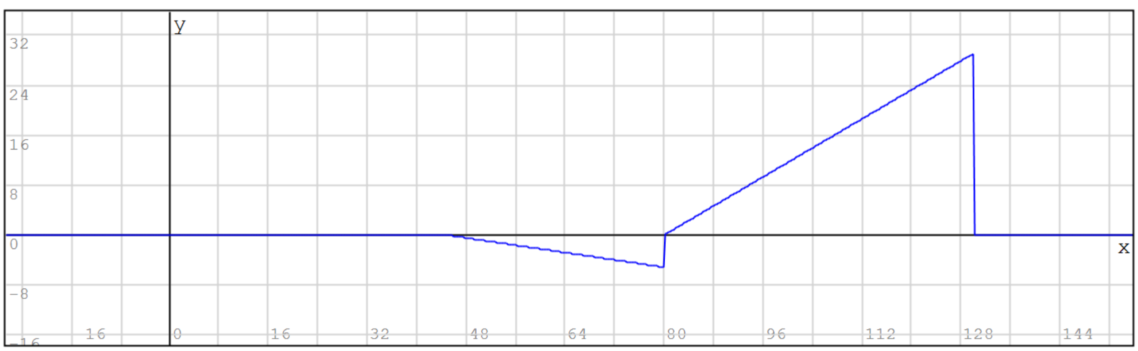
$$q_z(x) := ((-R_{b1z}) \cdot S(x, x_{b1}, -1) + (R_{w1z}) \cdot S(x, x_w, -1) - F_r \cdot S(x, x_m, -1))$$

$$V_z(x) := ((-R_{b1z}) \cdot S(x, x_{b1}, 0) + (R_{w1z}) \cdot S(x, x_w, 0) - F_r \cdot S(x, x_m, 0))$$

$$M_z(x) := ((-R_{b1z}) \cdot S(x, x_{b1}, 1) + (R_{w1z}) \cdot S(x, x_w, 1) - F_r \cdot S(x, x_m, 1) + M_{az} \cdot S(x, x_w, 0) - M_{bz} \cdot S(x, x_{b1}, 0))$$



$V_z(x\ \text{mm})$



$M_z(x\ \text{mm})$

Moments at critical points in xy plane:

$M_{b1z} := M_z(x_{b1}) = 28.8992\ \text{N m}$

$M_{m1z} := M_z(x_m) = 0\ \text{N m}$

$M_{wz} := M_z(x_w) = -5.1145\ \text{N m}$

$V_{b1z} := V_z(x_{b1}) = 577.984\ \text{N}$

$V_{m1z} := V_z(x_m) = 0\ \text{N}$

$V_{wz} := V_z(x_w) = -146.169\ \text{N}$

Total Alternating moments at critical locations on shaft 1:

Motor:

$$M_m := \sqrt{M_{m1z}^2 + M_{m1y}^2} = 0 \text{ N m}$$

Bearing:

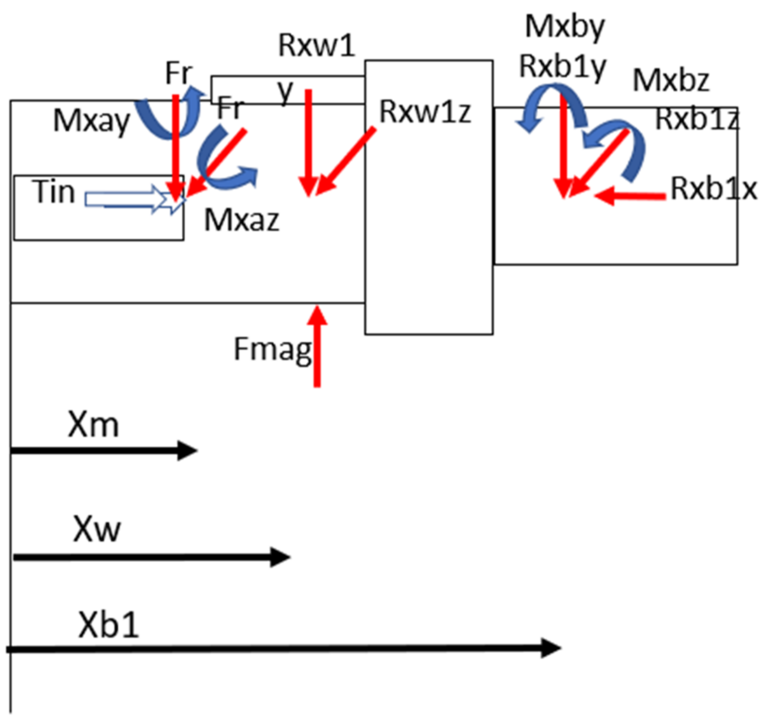
$$M_B := \sqrt{\left(M_{b1y}^2 + M_{b1z}^2\right)} = 28.8992 \text{ N m}$$

Wheel:

$$M_w := \sqrt{M_{wy}^2 + M_{wz}^2} = 7.2329 \text{ N m}$$

X-Y plane for axially loaded shaft (robot is on vertical surface

Shaft contains gear shaft, magnetic wheel, 1 ball bearing,



Distance of motor:

$x_m := 45.01\ mm$

$F_{mag} := 1800\ N$

Distance of wheel:

$x_w := 80\ mm$

Distance of bearing 1:

$x_{b1} := 130\ mm$

Force in y-direction of bearing:

$R_{xb1y} := 1831.93\ N$

Force in y-direction of wheel:

$R_{xw1y} := \left(R_{xb1y} + F_r - F_{mag}\right) = 178.099\ N$

Forces in x-direction of Bearing

$R_{xb1x} := 1694.4661\ N$

$R_{xb1x} = 1694.4661\ N$

Moment Balance

$M_{xby} := 61.6\ N\ m$

$M_{xay} := \left(R_{xb1y} \cdot \left(x_w - x_{b1}\right) + F_r \cdot \left(x_w - x_m\right) - M_{xby}\right) = -148.082\ J$

Material Properties of Shaft

$E := 29000\ \text{ksi}$

Length of shaft:

$L := 160\ \text{mm}$

Define the singularity function:

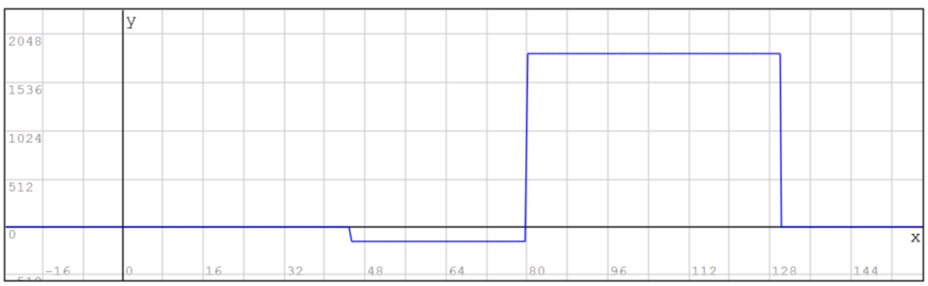
$$S(x, a, n) := \text{if } \left(\left((x-a)>0\right) \wedge (n \geq 0)\right) \quad (x-a)^n \quad \text{else} \quad 0$$

Shear and bending moment eqns:

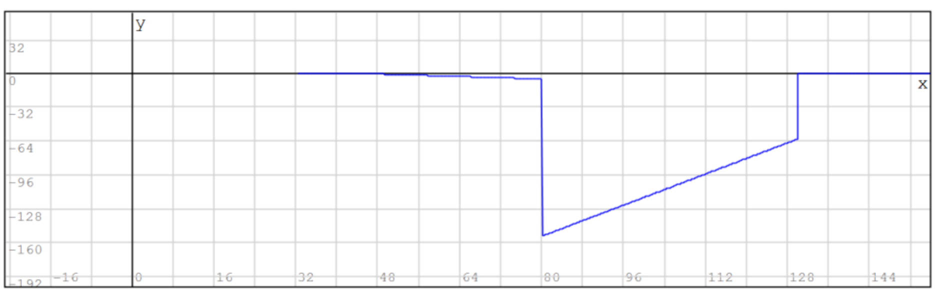
$$q_{xy}(x) := \left(\left(-R_{xb1y}\right) \cdot S\left(x, x_{b1}, -1\right) + \left(R_{xw1y}\right) \cdot S\left(x, x_w, -1\right) - F_r \cdot S\left(x, x_m, -1\right)\right)$$

$$V_{xy}(x) := \left(\left(-R_{xb1y}\right) \cdot S\left(x, x_{b1}, 0\right) + \left(R_{xw1y}\right) \cdot S\left(x, x_w, 0\right) - F_r \cdot S\left(x, x_m, 0\right) + F_{mag} \cdot S\left(x, x_w, 0\right)\right)$$

$$M_{xy}(x) := \left(\left(-R_{xb1y}\right) \cdot S\left(x, x_{b1}, 1\right) + \left(R_{xw1y}\right) \cdot S\left(x, x_w, 1\right) - F_r \cdot S\left(x, x_m, 1\right) + M_{xay} \cdot S\left(x, x_w, 0\right) + F_{mag} \cdot S\left(x, x_w, 1\right) + M_{xby} \cdot S\left(x, x_{b1}, 0\right)\right)$$



$V_{xy}(x\ \text{mm})$



$M_{xy}(x\ \text{mm})$

Moments at critical points in xy plane:

$M_{xb1y} := M_{xy}\left(x_{b1}\right) = -61.6 \text{ N m}$

$M_{xm1y} := M_{xy}\left(x_m\right) = 0 \text{ N m}$

$M_{xwy} := M_{xy}\left(x_w\right) = -5.1145 \text{ N m}$

$V_{xb1zy} := V_{xy}\left(x_{b1}\right) = 1831.93 \text{ N}$

$V_{xm1y} := V_{xy}\left(x_m\right) = 0 \text{ N}$

$V_{xwzy} := V_{xy}\left(x_w\right) = -146.169 \text{ N}$

y-z plane for axially loaded shaft (robot is on vertical surface)

Shaft contains gear shaft, magnetic wheel, 1 ball bearing,

Assuming equal distribution at the COM

$RCOM := 200 \text{ mm}$

Distance of motor:

$x_m := 45.01 \text{ mm}$

$F_{mag} := 1800 \text{ N}$

Distance of wheel:

$x_w := 80 \text{ mm}$

Distance of bearing 1:

$x_{b1} := 130 \text{ mm}$

Force in z-direction of bearing Conservative approach of y forces = z forces:

$R_{xb1z} := R_{xb1y} = 1831.93 \text{ N}$

$R_{xb1z} := 1232.0979 \text{ N}$

Force in z-direction of wheel:

$R_{xw1z} := \left(R_{xb1z} + F_r\right) = 1378.2669 \text{ N}$

Moment Balance

$M_{xzb} := 98.5678 \text{ N m}$

$M_{xza} := \left(R_{xb1z} \cdot \left(x_w - x_{b1}\right) + F_r \cdot \left(x_w - x_m\right) - M_{xzb}\right) = -155.0582 \text{ J}$

Material Properties of Shaft

$E := 29000 \text{ ksi}$

Length of shaft:

$L := 160 \text{ mm}$

Define the singularity function:

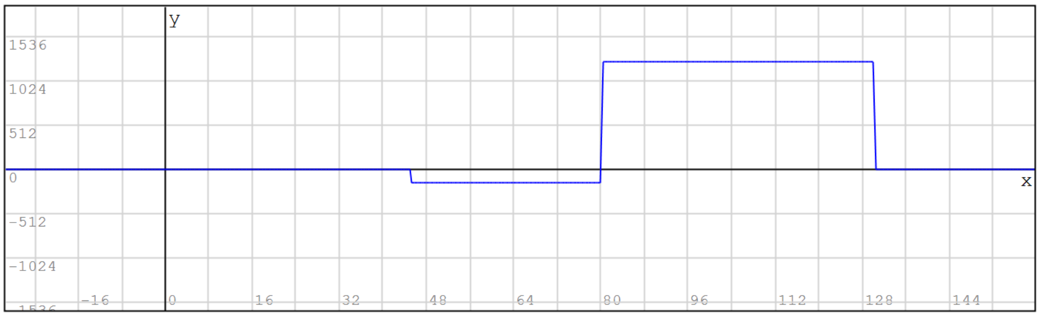
$$S(x, a, n) := \text{if } \left(\left((x-a) > 0\right) \wedge (n \geq 0)\right)$$
$$\quad (x-a)^n$$
$$\text{else}$$
$$\quad 0$$

Shear and bending moment eqns:

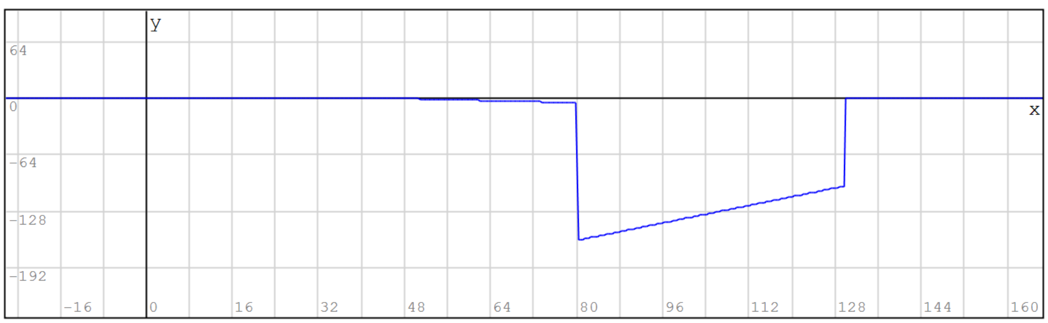
$$q_{xz}(x) := \left(\left(-R_{xb1z}\right) \cdot S\left(x, x_{b1}, -1\right) + \left(R_{xw1z}\right) \cdot S\left(x, x_w, -1\right) - F_r \cdot S\left(x, x_m, -1\right)\right)$$

$$V_{xz}(x) := \left(\left(-R_{xb1z}\right) \cdot S\left(x, x_{b1}, 0\right) + \left(R_{xw1z}\right) \cdot S\left(x, x_w, 0\right) - F_r \cdot S\left(x, x_m, 0\right) + M_{xza} \cdot S\left(x, x_w, -1\right) + M_{xzb} \cdot S\left(x, x_{b1}, -1\right)\right)$$

$$M_{xz}(x) := \left(\left(-R_{xb1z}\right) \cdot S\left(x, x_{b1}, 1\right) + \left(R_{xw1z}\right) \cdot S\left(x, x_w, 1\right) - F_r \cdot S\left(x, x_m, 1\right) + M_{xza} \cdot S\left(x, x_w, 0\right) + M_{xzb} \cdot S\left(x, x_{b1}, 0\right)\right)$$



$V_{xz}(x\ \mathrm{mm})$



$M_{xz}(x\ \mathrm{mm})$

Moments at critical points in xy plane:

$M_{xb1z} := M_{xz}\left(x_{b1}\right) = -98.5678\ \mathrm{N\ m}$

$M_{xwz} := M_{xz}\left(x_w\right) = -5.1145\ \mathrm{N\ m}$

$V_{xb1z} := V_{xz}\left(x_{b1}\right) = 1232.0979\ \mathrm{N}$

$V_{xwz} := V_{xz}\left(x_w\right) = -146.169\ \mathrm{N}$

$M_{xm1z} := M_{xz}\left(x_m\right) = 0\ \mathrm{N\ m}$

$V_{xm1z} := V_{xz}\left(x_m\right) = 0\ \mathrm{N}$

Total Alternating moments at critical locations on shaft 1:

Motor:

$$M_{xm} := \sqrt{M_{xm1z}^2 + M_{xm1y}^2} = 0 \text{ N m}$$

Wheel:

$$M_w := \sqrt{M_{wy}^2 + M_{wz}^2} = 7.2329 \text{ N m}$$

Bearing:

$$M_{xB} := \sqrt{\left(M_{xb1y}^2 + M_{xb1z}^2\right)} = 116.2333 \text{ N m}$$

Endurance Strength (simple)

$S_{ut} := 560 \text{ MPa}$ $S_y := 460 \text{ MPa}$ $S'_e := 0.504 \cdot S_{ut} = 282.24 \text{ MPa}$

$$S_{ut'} := S_{ut} \cdot \frac{1}{\text{MPa}} = 560$$

Step 1 Motor and wheel (hollow shaft with step and keyway)

Assume cold drawn:

$a := 4.51$ $b := -0.265$

$$k_a := \left(a \cdot S_{ut'}^{\;b}\right) = 0.8432$$

assume 2 ≤ d ≤ 10 in

$$k_{b11}\left(d_{11}\right) := 0.91 \cdot d_{11}^{-0.157}$$

$k_c := 1$ (bending dominates)

$k_d := 1$ (assume room temp)

$k_e := 0.897$ (assume 90% reliability)

$k_f := 1$ (no misc. effects)

Corrected Endurance Strength:

initial guess: $d_{11} := 50 \text{ mm}$

$$S_{e11} := k_a \cdot k_{b11}\left(d_{11}\right) \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S'_e = 105.1044 \frac{\text{N}}{\text{mm}^2}$$

Step 2 (no load - wheel support)

Assume cold drawn:

$a := 4.51$ $b := -0.265$

$k_a := \left(a \cdot S_{ut}{}^{b}\right) = 0.8432$

assume 2 ≤ d ≤ 10 in

$k_{b12}\left(d_{12}\right) := 0.91 \cdot d_{12}{}^{-0.157}$

(bending dominates)

(assume room temp)

(assume 90% reliability)

(no misc. effects)

Corrected Endurance Strength:

initial guess: $d_{12} := 60\ mm$

$S_{e12} := k_a \cdot k_{b12}\left(d_{12}\right) \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S'_e = 102.1385\ \frac{N}{mm^2}$

Step 3 (Bearing)

Assume cold drawn:

$a := 4.51$ $b := -0.265$

$k_a := \left(a \cdot S_{ut}{}^{b}\right) = 0.8432$

assume 0.11 ≤d ≤2 in

$k_{b13}\left(d_{13}\right) := 0.879 \cdot d_{13}{}^{-0.107}$

$k_c := 1$ (bending dominates)

$k_d := 1$ (assume room temp)

$k_e := 0.897$ (assume 90% reliability)

$k_f := 1$ (no misc. effects)

Corrected Endurance Strength:

initial guess: $d_{13} := 29.8478\ mm$

$S_{e13} := k_a \cdot k_{b13}\left(d_{13}\right) \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S'_e = 130.4641\ \frac{N}{mm^2}$

Initial Safety Factor Calculations:

$n_m := 1.1$ Material properties found in handbook or are manufacturer values

$n_s := 1.3$ Average load known although some error of overload may be present

$n_g := 1.0$ Manufacturing tolerances are tight to avoid vibrations

$n_f := 1.1$ Failure analysis based on multiaxial loading

$n_r := 1.2$ Assume a reliability of 90%

$N_f := n_m \cdot n_s \cdot n_g \cdot n_f \cdot n_r = 1.8876$

SCF Information:

Assumptions

$r := 1$

$q_{11} := 0.5$ $q_{12} := 0.5$ $q_{13} := 0.5$ $q_{14} := 0.5$ $q_{15} := 0.5$

Assume keyways gear head and wheel

Stepped Shaft Assumption

D/d=2.381
r/d=0.0635
Kt=2.1
Kts=1.75

Key way

Kt=2.2
Kts=3

Hollow Shaft Assumption

D/d=1.2
r/d=0.02
Kt=2.6
Kts=1.55

di /d=0.66

Key Way SCF is governing

step 1 Motor SCF - Hollow shaft

$q_{11} = 0.5$ $q_{11s} := 0.56$

$K_{ts11} := 3$ $K_{t11} := 2.6$

$K_{fs11} := q_{11s} \cdot (K_{ts11} - 1) + 1 = 2.12$ $K_{f11} := q_{11} \cdot (K_{t11} - 1) + 1 = 1.8$

step 2 Wheel step

$q_{12} = 0.5$ $q_{12s} := 0.56$

$K_{ts12} := 1.75$ $K_{t12} := 2.1$

$K_{fs12} := q_{12s} \cdot (K_{ts12} - 1) + 1 = 1.616$ $K_{f12} := q_{12} \cdot (K_{t12} - 1) + 1 = 1.3$

step 3 Bearing 2 SCF

$q_{13} = 0.5 \quad q_{13s} := 0.56$

$K_{ts13} := 1.75 \qquad K_{t13} := 2.1$

$K_{fs13} := q_{13s} \cdot (K_{ts13} - 1) + 1 = 1.42 \qquad K_{f13} := q_{13} \cdot (K_{t13} - 1) + 1 = 1.55$

Assuming no yield (This assumption is checked below)

$K_{fsm11} := K_{fs11} \quad K_{fsm12} := K_{fs12} \quad K_{fsm15} := K_{fs15} \quad K_{fsm13} := K_{fs13} \quad K_{fsm14} := K_{fs14}$

$K_{fm21} := K_{f21} \quad K_{fm22} := K_{f22} \quad K_{fm25} := K_{f25} \quad K_{fm23} := K_{f23} \quad K_{fm24} := K_{f24}$

Minimum Diameter at Gearhead and wheel on Shaft 1:

$T_b := 0 \text{ N m}$

$$d'_{11} := \left(\frac{16 \cdot N_f}{\pi} \cdot \sqrt{\frac{4 \cdot \left(K_{f11} \cdot (M_m + M_w)\right)^2}{(S_{e11})^2} + \frac{3 \cdot (K_{fsm12} \cdot T_b)^2}{(S_y)^2}} \right)^{\frac{1}{3}} = 0.0134 \text{ m}$$

$d'_{11} = 13.3544 \text{ mm}$

Assume cold drawn:

$a := 4.51 \qquad b := -0.265$

$k_a := \left(a \cdot S_{ut'}{}^{b} \right) = 0.8432$

assume 0.11 ≤d ≤2 in

$k_{b11} := 0.879 \cdot d_{11}^{-0.107}$

$k_c := 1$ (bending dominates)

$k_d := 1$ (assume room temp)

$k_e := 0.897$ (assume 90% reliability)

$k_f := 1$ (no misc. effects)

Corrected Endurance Strength:

initial guess: $d_{11} := d'_{11} = 13.3544 \text{ mm}$

$S_{e11} := k_a \cdot k_{b11} \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S'_e = 1.2346 \cdot 10^8 \text{ Pa}$

$$d'_{11} := \left(\frac{16 \cdot N_f}{\pi} \cdot \sqrt{\frac{4 \cdot \left(K_{f11} \cdot \left(M_m + M_w\right)\right)^2}{\left(S_{e11}\right)^2} + \frac{3 \cdot \left(K_{fsm12} \cdot T_b\right)^2}{\left(S_y\right)^2}} \right)^{\frac{1}{3}} = 0.0127\ \mathrm{m}$$

$d'_{11} := 12.6569$

Assume cold drawn:

$a := 4.51 \qquad b := -0.265$

$k_a := \left(a \cdot S_{ut'}{}^{b}\right) = 0.8432$

assume $0.11 \le d \le 2$ in

$k_{b11}\left(d_{11}\right) := 0.879 \cdot d_{11}{}^{-0.107}$

$k_c := 1$ (bending dominates)

$k_d := 1$ (assume room temp)

$k_e := 0.897$ (assume 90% reliability)

$k_f := 1$ (no misc. effects)

Corrected Endurance Strength:

initial guess: $d_{11} := d'_{11} = 12.6569$

$S_{e11} := k_a \cdot k_{b11} \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S'_e = 123.4573\ \frac{\mathrm{N}}{\mathrm{mm}^2}$

$$d'_{11} := \left(\frac{16 \cdot N_f}{\pi} \cdot \sqrt{\frac{4 \cdot \left(K_{f11} \cdot \left(M_m + M_w\right)\right)^2}{\left(S_{e11}\right)^2} + \frac{3 \cdot \left(K_{fsm12} \cdot T_b\right)^2}{\left(S_y\right)^2}} \right)^{\frac{1}{3}} = 0.0127\ \mathrm{m}$$

$d'_{11} = 12.6569\ \mathrm{mm}$

Convergence matrix

$$\begin{bmatrix} 1 & 50 & 13.3544 \\ 2 & 13.3544 & 12.6569 \\ 3 & 12.6569 & 12.6569 \end{bmatrix}$$

Diameter converges at 12.6569 mm

Minimum Diameter at wheel support on Shaft 1:

$$d'_{12} := \left(\frac{16 \cdot N_f}{\pi} \cdot \sqrt{\frac{4 \cdot (K_{f12} \cdot 0)^2}{(S_{e12})^2} + \frac{3 \cdot (K_{fsm12} \cdot T_b)^2}{(S_y)^2}} \right)^{\frac{1}{3}} = 0$$

$d'_{12} = 0\ mm$ there is no load at section 12

Minimum Diameter at bearing on Shaft 1:

$$d'_{13} := \left(\frac{16 \cdot N_f}{\pi} \cdot \sqrt{\frac{4 \cdot (K_{f13} \cdot M_{xB})^2}{(S_{e13})^2} + \frac{3 \cdot (K_{fsm12} \cdot T_b)^2}{(S_y)^2}} \right)^{\frac{1}{3}} = 0.0298\ m$$

$d'_{13} = 29.8328\ mm$

Testing for convergence

$$\begin{bmatrix} 1 & 45 & 30.2728 \\ 2 & 32.36890 & 29.8478 \\ 3 & 29.8328 & 29.83 \end{bmatrix}$$

the bearing diameter converges at 29.8328 mm

Bearing diameter is also dependent on bearing

$R_{bearingaxial} := R_{xb1x} = 1694.4661\ N$

Twist Calculations for shaft 1, sections 1:

$G := 78\ GPa$

Section 1:

$$J_{11} := \frac{\pi \cdot d'_{11}{}^4}{32} = 2.5195 \cdot 10^{-9}\ m^4$$

$$\theta_{11} := \frac{T_{in}}{G \cdot J_{11}} = 0.5394\ \frac{deg}{m}$$

Both angles of twist are less than 3 deg/m, thus criteria is met!

Von Mises Stress Calculations:

shaft 1 section 1 :

$$I_{11} := \frac{J_{11}}{2} = 1259.7273\ mm^4$$

$$\sigma_{an11} := \frac{(M_w + M_m) \cdot \frac{d'_{11}}{2}}{I_{11}} = 36.3358 \frac{N}{mm^2}$$

$$\tau_{mn11} := \frac{T_{out} \cdot \frac{d'_{11}}{2}}{J_{11}} = 464.6884 \frac{N}{mm^2}$$

$$\sigma_{a11} := K_{f11} \cdot \sigma_{an11} = 65.4044 \frac{N}{mm^2}$$

$$\tau_{m11} := K_{fs11} \cdot \tau_{mn11} = 985.1394 \frac{N}{mm^2}$$

Assuming fully reversed bending:

$$\sigma'_{a11} := \sigma_{a11} = 65.4044 \frac{N}{mm^2}$$

$$\sigma'_{m11} := \frac{1}{\sqrt{2}} \cdot \left(6 \cdot \tau_{m11}^{\ 2}\right)^{\frac{1}{2}} = 1706.3116 \frac{N}{mm^2}$$

Calculated Safety Factor:

$$n_{f11} := \sqrt{\frac{1}{\left(\frac{\sigma'_{a11}}{S_{e11}}\right)^2 + \left(\frac{\sigma'_{m11}}{S_y}\right)^2}} = 0.2669$$

Shaft 1, will reiterate with a new shaft diameter

$$d'_{11} := 50\ mm$$

Section 1:

$$J_{11} := \frac{\pi \cdot d'^{\,4}_{11}}{32} = 6.1359 \cdot 10^{-7}\ m^4$$

$$\theta_{11} := \frac{T_{in}}{G \cdot J_{11}} = 0.0022 \frac{deg}{m}$$

Both angles of twist are less than 3 deg/m, thus criteria is met!

Von Mises Stress Calculations:

shaft 1 section 1 :

$$I_{11} := \frac{J_{11}}{2} = 3.068 \cdot 10^5\ mm^4$$

$$M_m = 0$$

$$M_w = 7.2329\ J$$

$$\sigma_{an11} := \frac{(M_w + M_m) \cdot \frac{d'_{11}}{2}}{I_{11}} = 0.5894 \frac{N}{mm^2}$$

$$\tau_{mn11} := \frac{T_{out} \cdot \frac{d'_{11}}{2}}{J_{11}} = 7.5376 \frac{N}{mm^2}$$

$$\sigma_{a11} := K_{f11} \cdot \sigma_{an11} = 1.0609 \frac{N}{mm^2}$$

$$\tau_{m11} := K_{fs11} \cdot \tau_{mn11} = 15.9797 \frac{N}{mm^2}$$

Assuming fully reversed bending:

$$\sigma'_{a11} := \sigma_{a11} = 1.0609 \frac{N}{mm^2}$$

$$\sigma'_{m11} := \frac{1}{\sqrt{2}} \cdot \left(6 \cdot \tau_{m11}{}^2\right)^{\frac{1}{2}} = 27.6776 \frac{N}{mm^2}$$

Calculated Safety Factor:

$$n_{f11} := \sqrt{\frac{1}{\left(\frac{\sigma'_{a11}}{S_{e11}}\right)^2 + \left(\frac{\sigma'_{m11}}{S_y}\right)^2}} = 16.453$$

Twist Calculations for shaft 1, sections 3:

$$G := 78\ GPa$$

Section 3:

$$J_{13} := \frac{\pi \cdot d'_{13}{}^4}{32} = 77763.3311\ mm^4$$

$$\theta_{13} := \frac{T_{in}}{G \cdot J_{13}} = 0.0175 \frac{deg}{m}$$

Von Mises Stress Calculations:

shaft 1 section 3 :

$$I_{13} := \frac{J_{13}}{2} = 38881.6656\ mm^4$$

$$\sigma_{an13} := \frac{\left(M_{xB}\right) \cdot \frac{d'_{13}}{2}}{I_{13}} = 44.5912 \frac{N}{mm^2}$$

$$\tau_{mn13} := \frac{T_b \cdot \frac{d'_{13}}{2}}{J_{13}} = 0 \frac{N}{mm^2}$$

$$\sigma_{a13} := K_{f13} \cdot \sigma_{an13} = 69.1164 \frac{N}{mm^2}$$

$$\tau_{m13} := K_{fs13} \cdot \tau_{mn13} = 0 \frac{N}{mm^2}$$

Assuming fully reversed bending:

$$\sigma'_{a13} := \sigma_{a13} = 69.1164 \frac{N}{mm^2}$$

$$\sigma'_{m13} := \frac{1}{\sqrt{2}} \cdot \left(6 \cdot \tau_{m13}{}^2\right)^{\frac{1}{2}} = 0 \frac{N}{mm^2}$$

Calculated Safety Factor:

$$n_{f13} := \sqrt{\frac{1}{\left(\frac{\sigma'_{a13}}{S_{e13}}\right)^2 + \left(\frac{\sigma'_{m13}}{S_y}\right)^2}} = 1.8876$$

The calculated minimum diameter based on stress will fail, the bearing is dependent on bearing size,

Minimum bearing diameter:

Bearing 1:

$$C_{r1} := \left(n_{cycle1} \cdot 10^{-6}\right)^{\frac{1}{3}} \cdot \sqrt{R_{b1y}^{\ 2} + R_{b1z}^{\ 2}} = 1956.9286 \text{ N}$$

$$F_{br} := \sqrt{R_{b1y}^{\ 2} + R_{b1z}^{\ 2}} = 817.3928 \text{ N}$$

$$F_{eff} := X \cdot V \cdot F_{br} + Y \cdot R_{bearingaxial}$$

V : Rotation factor rotating shaft
X: Radial Factor $V := 1$
Y: Thrust Factor

Using these dynamic load ratings and the provided bearing diameter chart the minimum diameter and mximum fillet of the bearings are:

Use bearing PWTR30-2RS-RR-XL:

$d_{b1} := 45 \text{ mm}$

$d_{b1o} := 62 \text{ mm}$

$d_{b1w} := 29 \text{ mm}$

$C_r := 35000 \text{ N}$

$C_0 := 46000 \text{ N}$

Calculating the new safety factor based on new bearing diameter

$d'_{13} := 45 \text{ mm}$

shaft 1 section 3 :

$$J_{13} := \frac{\pi \cdot d'_{13}{}^{4}}{32} = 4.0258 \cdot 10^{5} \text{ mm}^{4}$$

$$\theta_{13} := \frac{T_{in}}{G \cdot J_{13}} = 0.0034 \frac{\text{deg}}{\text{m}}$$

Both angles of twist are less than 3 deg/m, thus criteria is met!

Von Mises Stress Calculations:

$$I_{13} := \frac{J_{13}}{2} = 2.0129 \cdot 10^{5}\ mm^{4}$$

$$\sigma_{an13} := \frac{\left(M_{xB}\right) \cdot \frac{d'_{13}}{2}}{I_{13}} = 12.9925\ \frac{N}{mm^{2}}$$

$$\tau_{mn13} := \frac{T_b \cdot \frac{d'_{13}}{2}}{J_{13}} = 0\ \frac{N}{mm^{2}}$$

$$\sigma_{a13} := K_{f13} \cdot \sigma_{an13} = 20.1384\ \frac{N}{mm^{2}}$$

$$\tau_{m13} := K_{fs13} \cdot \tau_{mn13} = 0\ \frac{N}{mm^{2}}$$

Assuming fully reversed bending:

$$\sigma'_{a13} := \sigma_{a13} = 20.1384\ \frac{N}{mm^{2}}$$

$$\sigma'_{m13} := \frac{1}{\sqrt{2}} \cdot \left(6 \cdot \tau_{m13}^{2}\right)^{\frac{1}{2}} = 0\ \frac{N}{mm^{2}}$$

Calculated Safety Factor:

$$n_{f13} := \sqrt{\frac{1}{\left(\frac{\sigma'_{a13}}{S_{e13}}\right)^{2} + \left(\frac{\sigma'_{m13}}{S_y}\right)^{2}}} = 6.4784$$

Slope and Deflection Equations:

Building off of the singularity equations for shear and bending moment, the slope and deflection formulas are defined as follows:

$d_b := d_{b1} = 45\ mm$ Bearing diameter

$$I := \frac{\pi \cdot d_b^{4}}{64} = 2.0129 \cdot 10^{5}\ mm^{4}$$

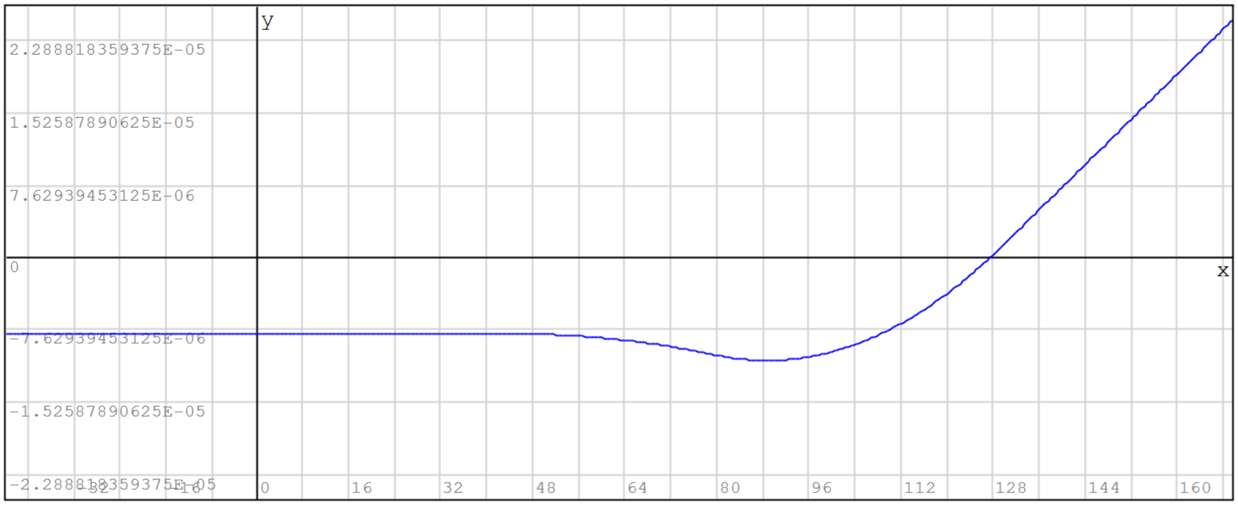
x-y plane

$$\theta_y(x) := \frac{1}{E \cdot I} \cdot \left(\frac{-R_{b1y}}{2} \cdot S\left(x, x_{b1}, 2\right) + \frac{\left(R_{w1y}\right)}{2} \cdot S\left(x, x_w, 2\right) - \frac{\left(F_r\right)}{2} \cdot S\left(x, x_m, 2\right) - \frac{F_{mag}}{2} \cdot S\left(x, x_w, 2\right)\right) + C_{1xy}$$

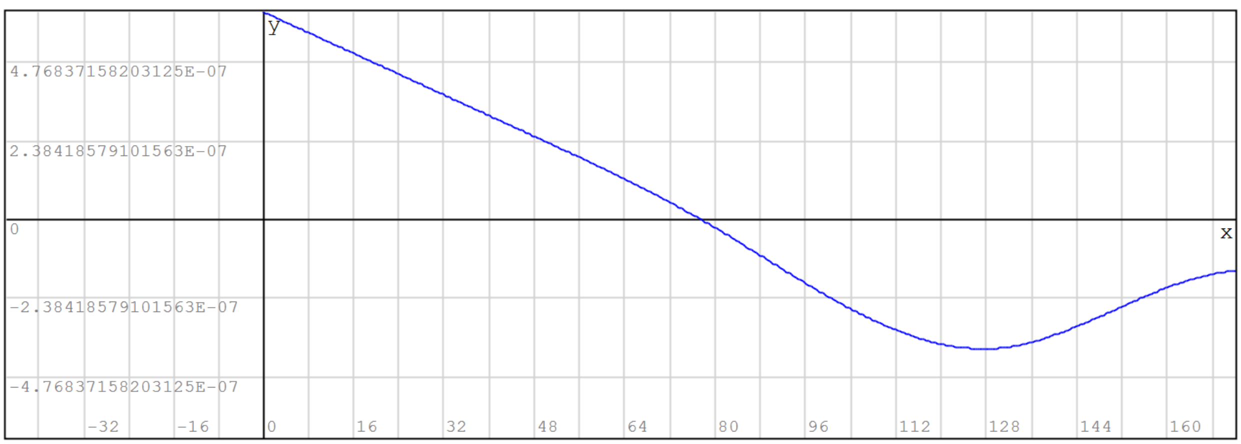
$$\delta_y(x) := \frac{1}{E \cdot I} \cdot \left(\frac{-R_{b1y}}{2} \cdot S\left(x, x_{b1}, 3\right) + \frac{\left(R_{w1y}\right)}{6} \cdot S\left(x, x_w, 3\right) - \frac{\left(F_r\right)}{6} \cdot S\left(x, x_m, 3\right) - \frac{F_{mag}}{6} \cdot S\left(x, x_w, 3\right)\right) + C_{1xy} \cdot x + C_{2xy}$$

$$C_{1xy} := \frac{R_{b1y} \cdot \left(x_w - x_{b1}\right)^3 - R_{w1y} \cdot \left(x_w - x_w\right)^3 + F_r \cdot \left(x_w - x_m\right)^3 + F_{mag} \cdot \left(x_w - x_w\right)^3}{6 \cdot E \cdot I \cdot \left(x_w - x_m\right)} = -7.810 \cdot 10^{-6}$$

$$C_{2xy} := -C_{1xy} \cdot x_w = 0.0006248 \text{ mm}$$



$\theta_y\ (x \text{ mm})$



$\delta_y\ (x \text{ mm})$

Slopes at critical points in xy plane:

$\theta_{yb1} := \theta_y\left(x_{b1}\right) = 1.5646 \cdot 10^{-6} \text{ rad}$

$\theta_{ym} := \theta_y\left(x_m\right) = -7.8095 \cdot 10^{-6} \text{ rad}$

$\theta_{yw} := \theta_y\left(x_w\right) = -1.0033 \cdot 10^{-5} \text{ rad}$

Deflections at critical points in xy plane:

$\delta_{yb1} := \delta_y\left(x_{b1}\right) = -0.0004\ mm$

$\delta_{ym} := \delta_y\left(x_m\right) = 0.0003\ mm$

$\delta_{yw} := \delta_y\left(x_w\right) = -2.593 \cdot 10^{-5}\ mm$

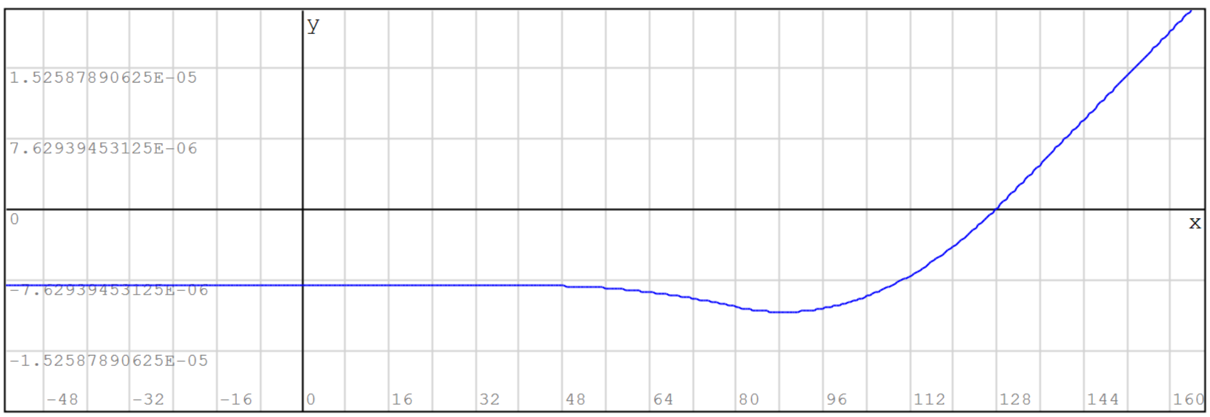
y-z plane

$$\theta_z(x) := \frac{1}{E \cdot I} \cdot \left(\frac{-R_{b1z}}{2} \cdot S\left(x, x_{b1}, 2\right) + \frac{\left(R_{w1z}\right)}{2} \cdot S\left(x, x_w, 2\right) - \frac{\left(F_r\right)}{2} \cdot S\left(x, x_w, 2\right)\right) + C_{1yz}$$

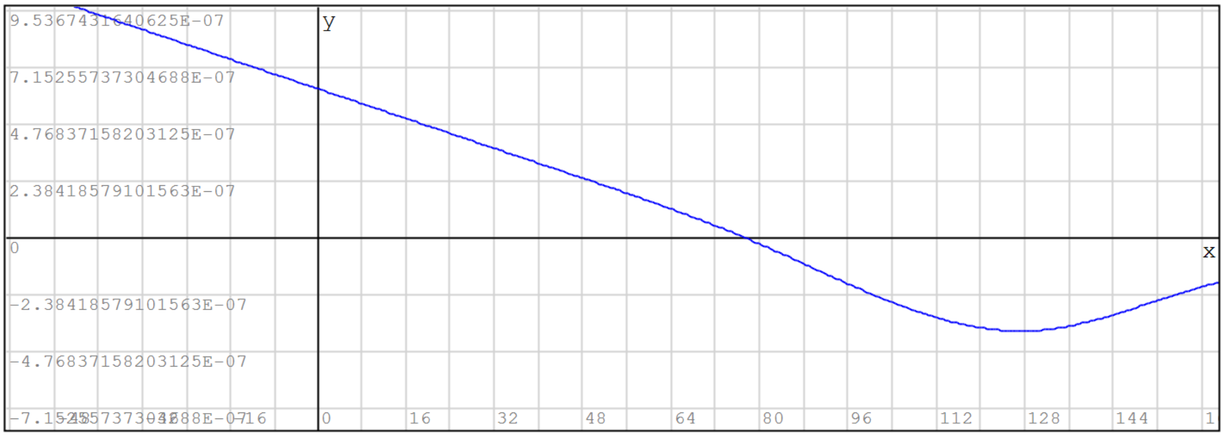
$$\delta_z(x) := \frac{1}{E \cdot I} \cdot \left(\frac{-R_{b1z}}{6} \cdot S\left(x, x_{b1}, 3\right) + \frac{\left(R_{w1z}\right)}{6} \cdot S\left(x, x_w, 3\right) - \frac{\left(F_r\right)}{6} \cdot S\left(x, x_w, 3\right)\right) + C_{1yz} \cdot x + C_{2yz}$$

$$C_{1yz} := \frac{-R_{b1z} \cdot \left(x_w - x_{b1}\right)^3 + R_{w1z} \cdot \left(x_w - x_m\right)^3 - F_r \cdot \left(x_w - x_m\right)^3}{6 \cdot E \cdot I \cdot \left(x_w - x_m\right)} = 1.148 \cdot 10^{-5}$$

$$C_{2yz} := -C_{1yz} \cdot x_{b1} = -0.001493\ mm$$



$\theta_y(x\ mm)$



$\delta_y\left(x\ \mathrm{mm}\right)$

Slopes at critical points in yz plane:

$\theta_{zb1} := \theta_z\left(x_{b1}\right) = 2.9432 \cdot 10^{-5}\ \mathrm{rad}$

$\theta_{zm} := \theta_z\left(x_m\right) = 1.1481 \cdot 10^{-5}\ \mathrm{rad}$

$\theta_{zw} := \theta_z\left(x_w\right) = 1.1481 \cdot 10^{-5}\ \mathrm{rad}$

Deflections at critical points in yz plane:

$\delta_{zb1} := \delta_z\left(x_{b1}\right) = 0.0003\ \mathrm{mm}$

$\delta_{zm} := \delta_z\left(x_m\right) = -0.001\ \mathrm{mm}$

$\delta_{zw} := \delta_z\left(x_w\right) = -0.0006\ \mathrm{mm}$

Total deflections at critical locations on shaft 1:

Bearing 1:

$\delta_{Tb1} := \sqrt{\delta_{yb1}^{\ 2} + \delta_{zb1}^{\ 2}} = 0.0005\ \mathrm{mm}$

Motor:

$\delta_{Tm} := \sqrt{\delta_{ym}^{\ 2} + \delta_{zm}^{\ 2}} = 0.001\ \mathrm{mm}$

Wheel:

$\delta_{Tw} := \sqrt{\left(\delta_{yw}^{\ 2} + \delta_{zw}^{\ 2}\right)} = 0.0006\ \mathrm{mm}$

Total slopes at critical locations on shaft 1:

Bearing 1:

$$\theta_{Tb1} := \sqrt{\theta_{yb1}^{2} + \theta_{zb1}^{2}} = 2.9473 \cdot 10^{-5}\ \text{rad}$$

Motor:

$$\theta_{Tm} := \sqrt{\theta_{ym}^{2} + \theta_{zm}^{2}} = 1.3885 \cdot 10^{-5}\ \text{rad}$$

Wheel:

$$\theta_{Tw} := \sqrt{\left(\theta_{yw}^{2} + \theta_{zw}^{2}\right)} = 1.5247 \cdot 10^{-5}\ \text{rad}$$

Shaft rounding to fit selected gears and bearings

$d1r := 50\ \text{mm}$
$d2r := 55\ \text{mm}$
$d3r := 45\ \text{mm}$

Check maximum stepped shaft dimensions

$r12 := 0.2 \cdot d1r = 10\ \text{mm}$
$r_{23} := 0.2 \cdot d2r = 11\ \text{mm}$

therefor max diameter for d12 is
$d_{2max} := d1r + r12 = 60\ \text{mm}$

The max diamerter for d3 is
$d_{3max} := d2r + r_{23} = 66\ \text{mm}$

$d_{3min} := d2r - r_{23} = 44\ \text{mm}$

Bearing size needs to be reiterted

using bearing S6009-2RSR-FD

$C_r := 17800\ \text{N}$

$C_0 := 12100\ \text{N}$

$$\frac{R_{bearingaxial}}{C_0} = 0.14$$

$$\frac{R_{bearingaxial}}{V \cdot F_r} = 11.5925 \qquad e := 0.42$$

Value is greater then e, therefore

$Y := 1.04$

$X := 0$

$F_{eff} = 1762.2447\ \text{N}$

$$L_c := \left(\frac{C_r}{F_{br}}\right)^{\frac{10}{3}} = 28837.9219$$

$$\left(L_{c'} := \frac{L_c}{10^6}\right) = 0.0288$$

$$C := F_{eff} \cdot L_{c'}^{\frac{1}{3}} = 540.4069\ N$$

this is less than the basic load rating of the bearing, therefore this bearing will work

final Shaft diameters:

$d1r := 50\ mm$
$d2r := 55\ mm$
$d3r := 45\ mm$

Check assumptions

1. Check kb diameter

2 in < d1r<10 in

2 in < d2r < 10 in

0.11 ≤d ≤2 in

2. Check SCF

1. snap ring

$\frac{d2r}{d1r} = 1.1$

$k_{t1} := 2.15$

$k_{ts} := 1.8$

$r_1 := 0.05 \cdot d1r = 2.5\ mm$

2. snap ring

$\frac{d3r}{d2r} = 0.8182$

$k_{t1} := 2.15$

$k_{ts} := 1.8$

$r_3 := 0.05 \cdot d2r = 2.75\ mm$

3. Check Step Size

$\frac{d2r}{d1r} = 1.1$

$r_2 := 0.2 \cdot d1r = 10\ mm$

$\frac{r_2}{d1r} = 0.2$

$k_t := 1.35$

$K_{ts} := 1.25$

$\frac{d3r}{d2r} = 0.8182$

$r_3 := 0.2 \cdot d2r = 11\ mm$

$\frac{r_2}{d2r} = 0.1818$

$k_t := 1.35$

$K_{ts} := 1.1$

Kt≤ Ktkeyway=2.2

Kts ≤Kts Keyway = 3

Assumed Kfs=Kfsm

$K_{fs11} \cdot \tau_{m11} = 4.9134 \text{ ksi}$

$K_{fs13} \cdot \tau_{m13} = 0 \text{ ksi}$

$\frac{S_y}{2} = 33.3587 \text{ ksi}$

Thus $K_{fs11} = K_{fsm11}$ & $K_{fs13} = K_{fsm13}$

Assumed Kfm=Kf

$K_{f11} \cdot \left|\left(\sigma'_{a11} + \sigma'_{m11}\right)\right| = 7.5027 \text{ kpsi}$

$K_{f13} \cdot \left|\left(\sigma'_{a13} + \sigma'_{m13}\right)\right| = 4.5273 \text{ kpsi}$

< $S_y = 66.7174 \text{ ksi}$

Thus $K_{fm11} = K_{f11}$ & $K_{fm13} = K_{f13}$

Assumption is valid!

Keyways and Keys dimensions

Using the diameters at each gear as well as the face widths:

$d1r = 50 \text{ mm}$

$d3r = 45 \text{ mm}$

The dimensions of the keyways and keys were provided by an online catalogue as:

$W_{12} := 2.5 \text{ mm}$

$T_{12} := 2.5 \text{ mm}$

$L_{12} := 4 \text{ mm}$

Note: W=Width, T=Depth, L=Length

Shaft Recalculations Using Actual Diameters

Endurance Strength (simple)

$S_{ut} := 560\ MPa$ $S_y := 460\ MPa$ $S'_e := 0.504 \cdot S_{ut} = 282.24\ MPa$

Assume cold drawn:

$a := 4.51$ $b := -0.265$

$k_a := \left(a \cdot S_{ut}{}'^{\,b}\right) = 0.8432$

assume 2 ≤ d ≤ 10 in

$k_{b11}\left(d_{11}\right) := 0.91 \cdot d_{11}^{\;-0.157}$

$k_c := 1$ (bending dominates)

$k_d := 1$ (assume room temp)

$k_e := 0.897$ (assume 90% reliability)

$k_f := 1$ (no misc. effects)

Corrected Endurance Strength:

initial guess: $d_{11} := 50\ mm$

$S_{e11} := k_a \cdot k_{b11}\left(d_{11}\right) \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S'_e = 105.1044\ \frac{N}{mm^2}$

$d_{11} := 50\ mm$

Step 2 (no load - wheel support)

Assume cold drawn:

$a := 4.51$ $b := -0.265$

$k_a := \left(a \cdot S_{ut}{}'^{\,b}\right) = 0.8432$

assume 2 ≤ d ≤ 10 in

$k_{b12}\left(d_{12}\right) := 0.91 \cdot d_{12}^{\;-0.157}$

(bending dominates)

(assume room temp)

(assume 90% reliability)

(no misc. effects)

Corrected Endurance Strength:

Actual Diameter: $d_{12} := 55\ mm$

$$S_{e12} := k_a \cdot k_{b12}\left(d_{12}\right) \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S'_e = 103.5433 \frac{N}{mm^2}$$

$d_{12} := 55\ mm$

Step 3 (Bearing)

Assume cold drawn:
$a := 4.51$ $b := -0.265$

$$k_a := \left(a \cdot S_{ut'}{}^{b}\right) = 0.8432$$

assume 0.11 ≤d ≤2 in

$$k_{b13}\left(d_{13}\right) := 0.879 \cdot d_{13}{}^{-0.107}$$

$k_c := 1$ (bending dominates)

$k_d := 1$ (assume room temp)

$k_e := 0.897$ (assume 90% reliability)

$k_f := 1$ (no misc. effects)

Corrected Endurance Strength:

Actual Diameter: $d_{13} := 45\ mm$

$$S_{e13} := k_a \cdot k_{b13}\left(d_{13}\right) \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S'_e = 124.857 \frac{N}{mm^2}$$

$d_{13} := 45\ mm = 0.045\ m$

SCF Information:

Assumptions

$r := 0.05$ $q_{11} := 0.68$ $q_{12} := 0.68$ $q_{13} := 0.68$

$q_{11s} := 0.73$ $q_{12s} := 0.73$ $q_{13s} := 0.73$

Assume snapring at bearings and keyways at wheel

Snapring Assumption Stepped Shaft Assumption

$\frac{d_{13}}{d_{12}} = 0.8182$ $\frac{d_{12}}{d_{11}} = 1.1$ $\frac{d_{13}}{d_{12}} = 0.8182$

Kt=2.2
Kts=3

$\frac{r}{d_{12}} = 0.9091 \cdot \frac{1}{m}$ $\frac{r}{d_{12}} = 0.9091 \cdot \frac{1}{m}$ $\frac{r}{d_{13}} = 1.1111 \cdot \frac{1}{m}$

Kt = 2.25 Kt = 2.45 Kt = 1.65 Kt =1.4 Kt = 1.7 Kt 1.75
Kts = 1.55 Kts = 1.6 Kts =1.45 Kts =1.2 Kts =1.5 Kts =1.55

Key Way SCF is governing

Step 1 wheel

$K_{ts11} := 3$ $K_{t11} := 2.2$

$K_{fs11} := q_{11s} \cdot (K_{ts11} - 1) + 1 = 2.46$ $K_{f11} := q_{11} \cdot (K_{t11} - 1) + 1 = 1.816$

Step 2

$K_{ts12} := 1.45$ $K_{t32} := 1.65$

$K_{fs12} := q_{12s} \cdot (K_{ts12} - 1) + 1 = 1.3285$ $K_{f12} := q_{12} \cdot (K_{t12} - 1) + 1 = 1.408$

step 3 Bearing

$K_{ts13} := 1.2$ $K_{t13} := 1.65$

$K_{fs13} := q_{13s} \cdot (K_{ts13} - 1) + 1 = 1.146$ $K_{f13} := q_{13} \cdot (K_{t13} - 1) + 1 = 1.442$

$d_{11in} = 0.032$ m

Assuming no yield

$K_{fsm11} := K_{fs11}$ $K_{fm11} := K_{f11}$

$K_{fsm12} := K_{fs12}$ $K_{fm12} := K_{f12}$

$K_{fsm13} := K_{fs13}$ $K_{fm13} := K_{f13}$

Twist Calculations for shaft 1, sections 1:

$G := 78$ GPa

Section 1: Now accounting for hollow shaft (ID = 32 mm)

$$J_{11} := \frac{\pi \cdot \left(d_{11}^{\ 4} - d_{11in}^{\ 4}\right)}{32} = 5.1065 \cdot 10^{5}\ \mathrm{mm}^{4}$$

$$\theta_{11} := \frac{T_{in}}{G \cdot J_{11}} = 0.0027\ \frac{\mathrm{deg}}{\mathrm{m}}$$

Both angles of twist are less than 3 deg/m, thus criteria is met!

Von Mises Stress Calculations:

shaft 1 section 1 :

$$I_{11} := \frac{J_{11}}{2} = 2.5532 \cdot 10^{5}\ \mathrm{mm}^{4}$$

$$\sigma_{an11} := \frac{\left(M_w + M_m\right) \cdot \frac{d'_{11}}{2}}{I_{11}} = 0.7082 \; \frac{N}{mm^2}$$

$$\tau_{mn11} := \frac{T_{out} \cdot \frac{d'_{11}}{2}}{J_{11}} = 9.0571 \; \frac{N}{mm^2}$$

$$\sigma_{a11} := K_{f11} \cdot \sigma_{an11} = 1.2861 \; \frac{N}{mm^2}$$

$$\tau_{m11} := K_{fs11} \cdot \tau_{mn11} = 22.2805 \; \frac{N}{mm^2}$$

Assuming fully reversed bending:

$$\sigma'_{a11} := \sigma_{a11} = 1.2861 \; \frac{N}{mm^2}$$

$$\sigma'_{m11} := \frac{1}{\sqrt{2}} \cdot \left(6 \cdot \tau_{m11}^{2}\right)^{\frac{1}{2}} = 38.5909 \; \frac{N}{mm^2}$$

Calculated Safety Factor:

$$n_{f11} := \sqrt{\frac{1}{\left(\frac{\sigma'_{a11}}{S_{e11}}\right)^2 + \left(\frac{\sigma'_{m11}}{S_y}\right)^2}} = 11.7951$$

Twist Calculations for shaft 1, sections 3:

$G := 78 \; GPa$

Section 3:

$$J_{13} := \frac{\pi \cdot d_{13}^{4}}{32} = 4.0258 \cdot 10^{5} \; mm^{4}$$

$$\theta_{13} := \frac{T_{in}}{G \cdot J_{13}} = 0.0034 \; \frac{deg}{m}$$

Both angles of twist are less than 3 deg/m, thus criteria is met!

Von Mises Stress Calculations:

shaft 1 section 3 :

$$I_{13} := \frac{J_{13}}{2} = 2.0129 \cdot 10^{5} \; mm^{4}$$

$$\sigma_{an13} := \frac{\left(M_{xB}\right) \cdot \frac{d_{13}}{2}}{I_{13}} = 12.9925 \; \frac{N}{mm^2}$$

$$\tau_{mn13} := \frac{T_b \cdot \frac{d_{13}}{2}}{J_{13}} = 0 \; \frac{N}{mm^2}$$

$\sigma_{a13} := K_{f13} \cdot \sigma_{an13} = 18.7352 \frac{N}{mm^2}$

$\tau_{m13} := K_{fs13} \cdot \tau_{mn13} = 0 \frac{N}{mm^2}$

Assuming fully reversed bending:

$\sigma'_{a13} := \sigma_{a13} = 18.7352 \frac{N}{mm^2}$

$\sigma'_{m13} := \frac{1}{\sqrt{2}} \cdot \left(6 \cdot \tau_{m13}^{2}\right)^{\frac{1}{2}} = 0 \frac{N}{mm^2}$

Calculated Safety Factor:

$$n_{f13} := \sqrt{\frac{1}{\left(\frac{\sigma'_{a13}}{S_{e13}}\right)^2 + \left(\frac{\sigma'_{m13}}{S_y}\right)^2}} = 6.6643$$

Slope and Deflection Equations:

Building off of the singularity equations for shear and bending moment, the slope and deflection formulas are defined as follows:

$$I := \frac{\pi \cdot d_{13}^{4}}{64} = 2.0129 \cdot 10^{5} \text{ mm}^4$$

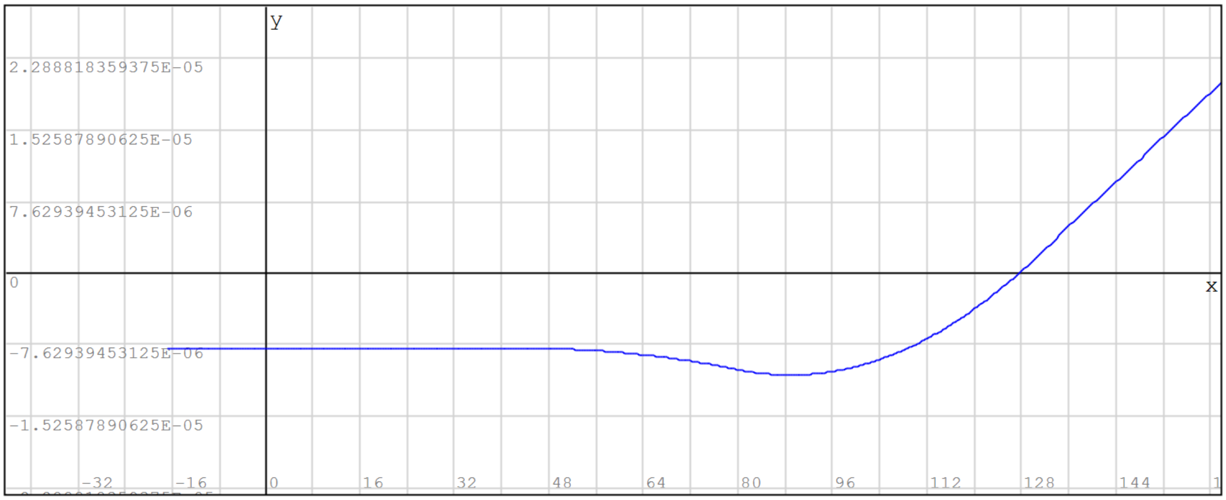
x-y plane

$$\theta_y(x) := \frac{1}{E \cdot I} \cdot \left(\frac{-R_{b1y}}{2} \cdot S(x, x_{b1}, 2) + \frac{(R_{w1y})}{2} \cdot S(x, x_w, 2) - \frac{(F_r)}{2} \cdot S(x, x_m, 2) - \frac{F_{mag}}{2} \cdot S(x, x_w, 2)\right) + C_{1xy}$$

$$\delta_y(x) := \frac{1}{E \cdot I} \cdot \left(\frac{-R_{b1y}}{2} \cdot S(x, x_{b1}, 3) + \frac{(R_{w1y})}{6} \cdot S(x, x_w, 3) - \frac{(F_r)}{6} \cdot S(x, x_m, 3) - \frac{F_{mag}}{6} \cdot S(x, x_w, 3)\right) + C_{1xy} \cdot x + C_{2xy}$$

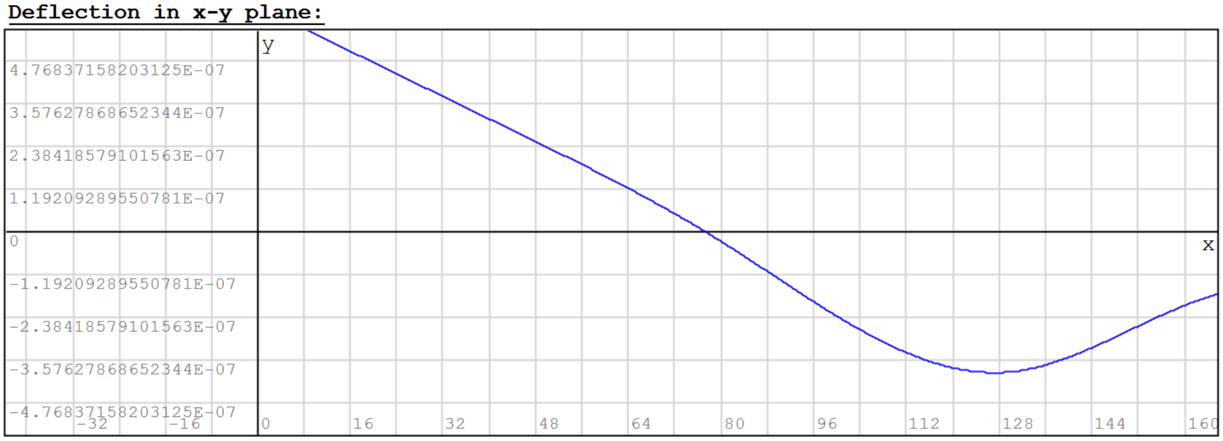
$$C_{1xy} := \frac{R_{b1y} \cdot (x_w - x_{b1})^3 - R_{w1y} \cdot (x_w - x_w)^3 + F_r \cdot (x_w - x_m)^3 + F_{mag} \cdot (x_w - x_w)^3}{6 \cdot E \cdot I \cdot (x_w - x_m)} = -7.810 \cdot 10^{-6}$$

$$C_{2xy} := -C_{1xy} \cdot x_w = 0.0006248 \text{ mm}$$



$\theta_y\,(x\ \mathrm{mm})$

Deflection in x-y plane:



$\delta_y\,(x\ \mathrm{mm})$

Slopes at critical points in xy plane:

$\theta_{yb1} := \theta_y\left(x_{b1}\right) = 1.5646 \cdot 10^{-6}\ \mathrm{rad}$

$\theta_{ym} := \theta_y\left(x_m\right) = -7.8095 \cdot 10^{-6}\ \mathrm{rad}$

$\theta_{yw} := \theta_y\left(x_w\right) = -1.0033 \cdot 10^{-5}\ \mathrm{rad}$

Deflections at critical points in xy plane:

$\delta_{yb1} := \delta_y\left(x_{b1}\right) = -0.0004\ \mathrm{mm}$

$\delta_{ym} := \delta_y\left(x_m\right) = 0.0003\ \mathrm{mm}$

$\delta_{yw} := \delta_y\left(x_w\right) = -2.593 \cdot 10^{-5}\ \mathrm{mm}$

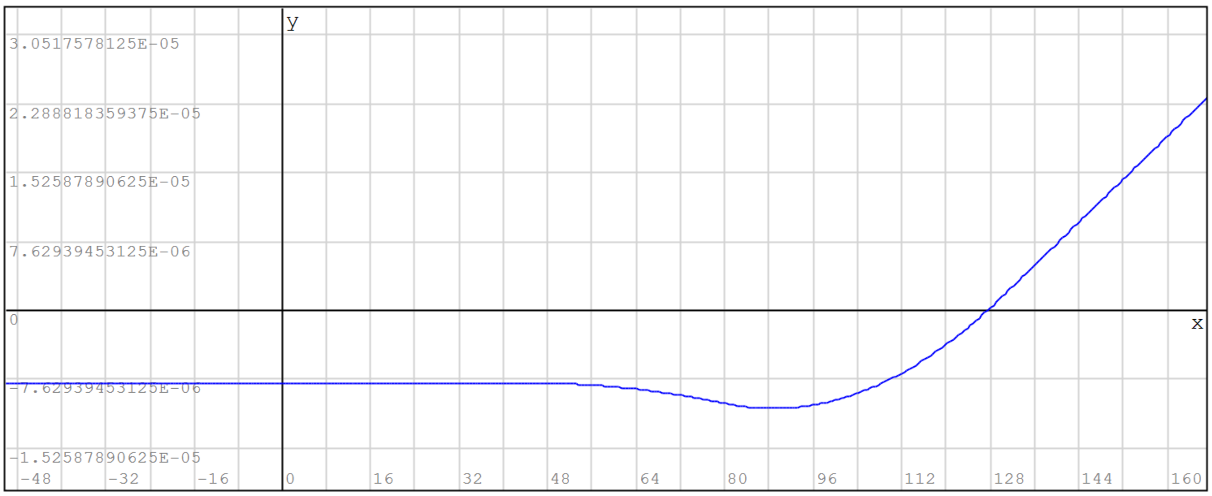
y-z plane

$$\theta_z(x) := \frac{1}{E \cdot I} \cdot \left(\frac{-R_{b1z}}{2} \cdot S(x, x_{b1}, 2) + \frac{(R_{w1z})}{2} \cdot S(x, x_w, 2) - \frac{(F_r)}{2} \cdot S(x, x_w, 2) \right) + C_{1yz}$$

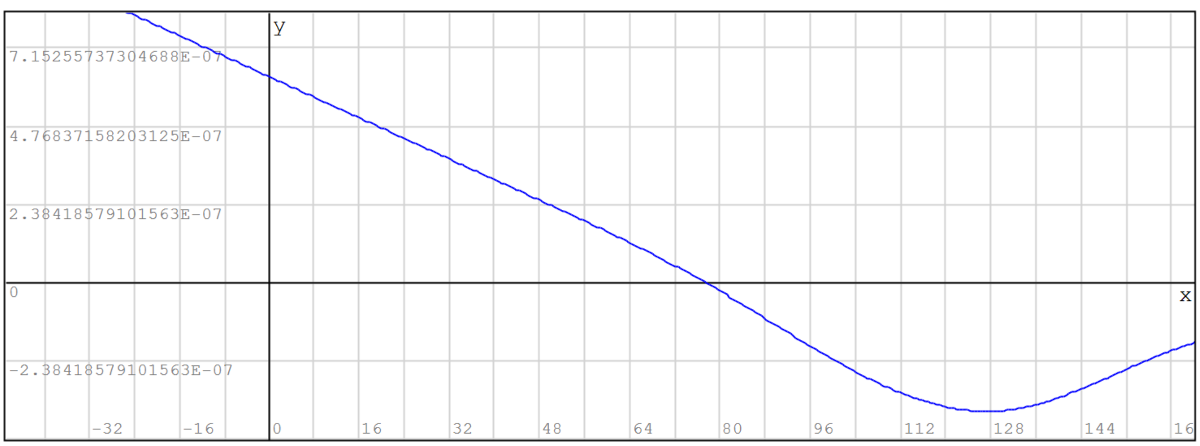
$$\delta_z(x) := \frac{1}{E \cdot I} \cdot \left(\frac{-R_{b1z}}{6} \cdot S(x, x_{b1}, 3) + \frac{(R_{w1z})}{6} \cdot S(x, x_w, 3) - \frac{(F_r)}{6} \cdot S(x, x_w, 3) \right) + C_{1yz} \cdot x + C_{2yz}$$

$$C_{1yz} := \frac{-R_{b1z} \cdot (x_w - x_{b1})^3 + R_{w1z} \cdot (x_w - x_m)^3 - F_r \cdot (x_w - x_m)^3}{6 \cdot E \cdot I \cdot (x_w - x_m)} = 1.148 \cdot 10^{-5}$$

$$C_{2yz} := -C_{1yz} \cdot x_{b1} = -0.001493 \text{ mm}$$



$\theta_y(x \text{ mm})$



$\delta_y(x \text{ mm})$

Slopes at critical points in yz plane:

$\theta_{zb1} := \theta_z(x_{b1}) = 2.9432 \cdot 10^{-5}$ rad

$\theta_{zm} := \theta_z(x_m) = 1.1481 \cdot 10^{-5}$ rad

$\theta_{zw} := \theta_z(x_w) = 1.1481 \cdot 10^{-5}$ rad

Deflections at critical points in yz plane:

$\delta_{zb1} := \delta_z(x_{b1}) = 0.0003$ mm

$\delta_{zm} := \delta_z(x_m) = -0.001$ mm

$\delta_{zw} := \delta_z(x_w) = -0.0006$ mm

Total deflections at critical locations on shaft 1:

Bearing 1:

$\delta_{Tb1} := \sqrt{\delta_{yb1}^2 + \delta_{zb1}^2} = 0.0005$ mm

Motor:

$\delta_{Tm} := \sqrt{\delta_{ym}^2 + \delta_{zm}^2} = 0.001$ mm

Wheel:

$\delta_{Tw} := \sqrt{\left(\delta_{yw}^2 + \delta_{zw}^2\right)} = 0.0006$ mm

Total slopes at critical locations on shaft 1:

Bearing 1:

$\theta_{Tb1} := \sqrt{\theta_{yb1}^2 + \theta_{zb1}^2} = 2.9473 \cdot 10^{-5}$ rad

Motor:

$\theta_{Tm} := \sqrt{\theta_{ym}^2 + \theta_{zm}^2} = 1.3885 \cdot 10^{-5}$ rad

Wheel:

$\theta_{Tw} := \sqrt{\left(\theta_{yw}^2 + \theta_{zw}^2\right)} = 1.5247 \cdot 10^{-5}$ rad

total deflection < 0.127 mm Deflection criteria met!

Assumption and Criterion Check:

Assumed Kfs=Kfsm

$K_{fs11} \cdot \tau_{m11} = 7.9495\ \text{ksi}$

$K_{fs13} \cdot \tau_{m13} = 0\ \text{ksi}$

$\frac{S_y}{2} = 33.3587\ \text{ksi}$

Thus $K_{fs11} = K_{fsm11}$ & $K_{fs13} = K_{fsm13}$

Assumption is valid!

Assumption is valid!

Assumed Kfm=Kf

$K_{f11} \cdot \left|\left(\sigma'_{a11} + \sigma'_{m11}\right)\right| = 10.5032\ \text{kpsi}$

$K_{f13} \cdot \left|\left(\sigma'_{a13} + \sigma'_{m13}\right)\right| = 3.9184\ \text{kpsi}$

$< \quad S_y = 66.7174\ \text{ksi}$

Thus $K_{fm11} = K_{f11}$ & $K_{fm13} = K_{f13}$

All twists must be less than 3 deg/m:

$\theta_{11} = 0.0027\ \frac{\text{deg}}{\text{m}} \qquad \theta_{13} = 0.0034\ \frac{\text{deg}}{\text{m}}$

Criterion is met!

All deflections must be less than 0.127 mm:

Criterion is met!

$\delta_{Tb1} = 0.0005\ \text{mm} \qquad \delta_{Tm} = 0.001\ \text{mm} \qquad \delta_{Tw} = 0.0006\ \text{mm}$

All assumptions/criterion is met, Drive Shaft Complete

Conclusion

The minimum shaft diameter was calculated for the loaded sections, (section d1 and section d3) and factor of safety was determined at the rounded shaft diameters.

The minimum shaft diameter at section 1 is 12.6 mm and the minimum diameter at section 3 is 30mm

The rounder diameters are as follows:
Section 1: 50mm
Section 2: 55mm
Section 3: 45mm

at section 1 and 3 the factory of safety are as follows:
Section 1: 11.79
Section 2: 6.66

Therefore the shaft does not fail at the most conservative loading conditions

Appendix D7: Fixed Shaft

Title - Bearing Calculation and Shaft sizing for fixed shaft
Date - March 12, 2021
Author - Calvin Chen

Objective

Determine the minimum bearing size to withstand radial and axial loading on the pivot fixed shaft
Parametric equation will be derived and bearing size will be iterated with step shaft iterative design

Variables

Xb - Distance of the bearing
Xwall - Distance of the shaft to the chassis
Xs - Distance to the fixed shaft connection point

Rb(x,y,z) - Bearing forces in the (x,y,z) plane
Rs(x,y,z) - Support forces in the (x,y,z) plane
Mb - bearing moment in the

Ms - Support Moment

All additional varibles will be defined along the calculation

Solution Method

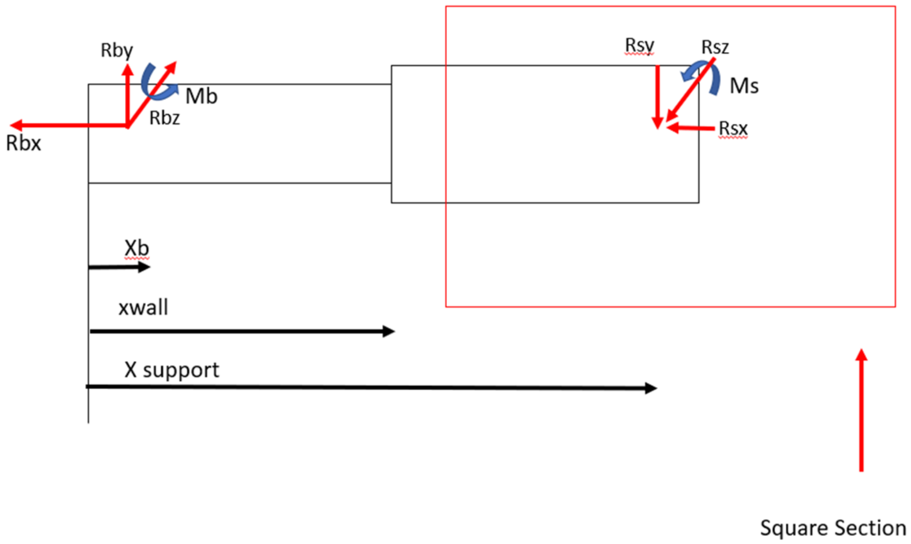
Forces were obtained from the stability calculations for the V-Support

```
Shaft Deflection Calculation for Shaft with 1 bearing and 1 support fixture

Note : Datum is on left end of beam established at x=0

Assume gravity forces are insignificant compared to applied loading.
```

Shaft Deflection Calculation for shaft wiht bearing and a supporting connection, will assume transverse hole
NOTE : Datum is establies on left end of the beam (x=0)
Assume gravity forces are insignifacnt compared to applied loading.



Define forces on the shaft

$M_{robot} := 500\ kg$

$F_w := M_{robot}\ g_e = 4903.325\ N$

X-Y plane (Note that radial forces act on this plane)

Shaft contains motor shaft, magnetic wheel, 1 ball bearing, 2 couplings

Distance of bearing:

$x_b := 8\ mm$

Distance of support:

$x_s := 92\ mm$

Distance of wall suppot:

$x_1 := 43\ mm$

bearing forces:

$R_{by} := 1497.9963\ N$ Force obtained from stability calculation

Force in y-direction of support:

$R_{sy} := \left(R_{by}\right)$

Moment Balance

$M_{by} := 0\ m$

$M_s := \left(R_{by} \cdot \left(x_s - x_b\right) - M_{by}\right) = 125.8317\ J$

Material Properties of Shaft

Length of shaft:

$L := 122\ mm$

Define the singularity function:

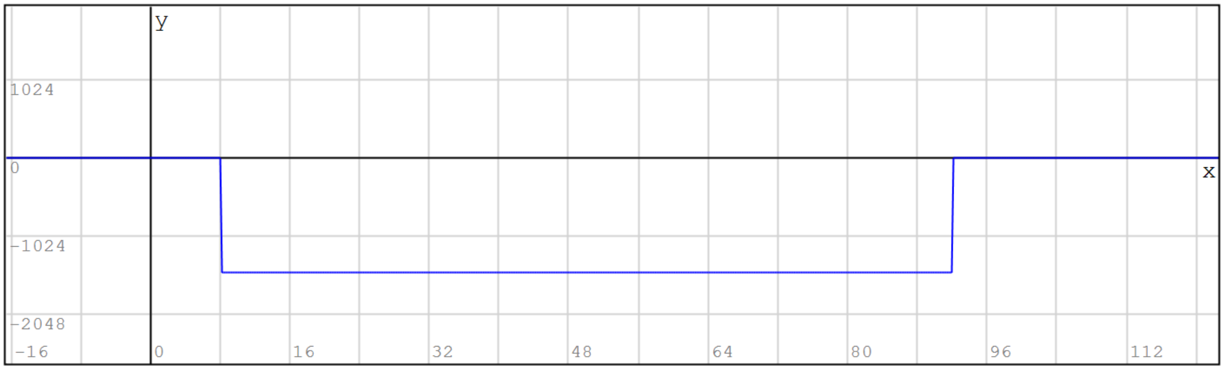
$$S(x, a, n) := \text{if } \left(\left((x - a) > 0\right) \wedge (n \geq 0)\right)$$
$$\quad (x - a)^n$$
$$\text{else}$$
$$\quad 0$$

Shear and bending moment eqns:

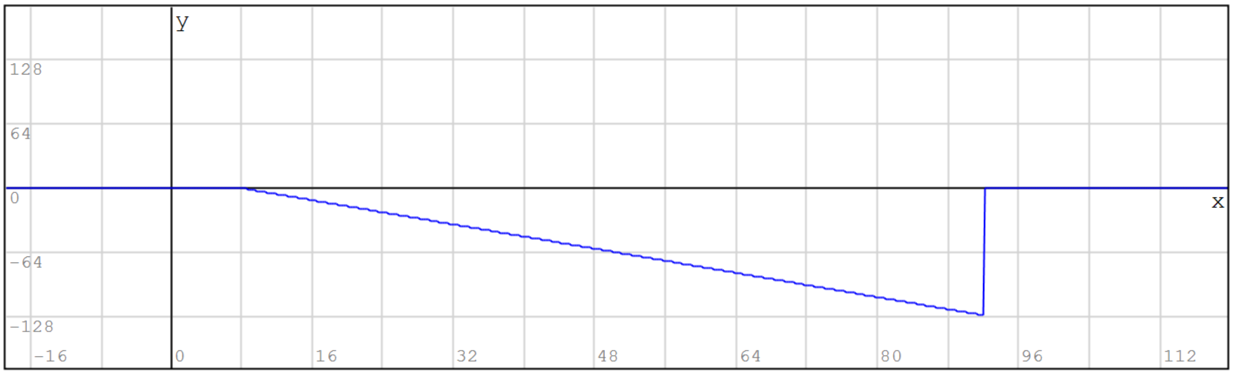
$$q_y(x) := \left(\left(-R_{by}\right) \cdot S\left(x, x_b, -1\right) + \left(R_{sy}\right) \cdot S\left(x, x_s, -1\right)\right)$$

$$V_y(x) := \left(\left(-R_{by}\right) \cdot S\left(x, x_b, 0\right) + \left(R_{sy}\right) \cdot S\left(x, x_s, 0\right)\right)$$

$$M_y(x) := \left(\left(-R_{by}\right) \cdot S\left(x, x_b, 1\right) + \left(R_{sy}\right) \cdot S\left(x, x_s, 1\right) + M_s \cdot S\left(x, x_s, 0\right) + M_{by} \cdot S\left(x, x_b, 0\right)\right)$$



$V_y\,(x\ \mathrm{mm})$



$M_y\,(x\ \mathrm{mm})$

Moments at critical points in xy plane:

$M_{By} := M_y\left(x_b\right) = 0\ \mathrm{N\ m}$

$M_{sy} := M_y\left(x_s\right) = -125.8317\ \mathrm{N\ m}$

$V_{by} := V_y\left(x_b\right) = 0\ \mathrm{N}$

$V_{sy} := V_y\left(x_s\right) = -1497.9963\ \mathrm{N}$

y-z plane (Note that radial forces act on this plane)
Shaft contains motor shaft, magnetic wheel, 1 ball bearing, 2 couplings

Distance of bearing:
$x_b := 8\ \mathrm{mm}$

Distance of support:
$x_s := 92\ \mathrm{mm}$

Distance of wall suppot:
$x_1 := 43\ \mathrm{mm}$

bearing forces:
$R_{bz} := 1497.9963\ \mathrm{N}$ Obtained from V-Support Stability Calculation

Force in z-direction of support:

$R_{sz} := \left(R_{bz}\right) = 1497.9963\ \mathrm{N}$

Moment Balance

$M_{bz} := 299.5993\ N\ m$

$M_s := \left(R_{bz} \cdot \left(x_s - x_b\right) - M_{bz}\right) = -173.7676\ J$

Material Properties of Shaft

Length of shaft:
$L := 122\ mm$

Define the singularity function:

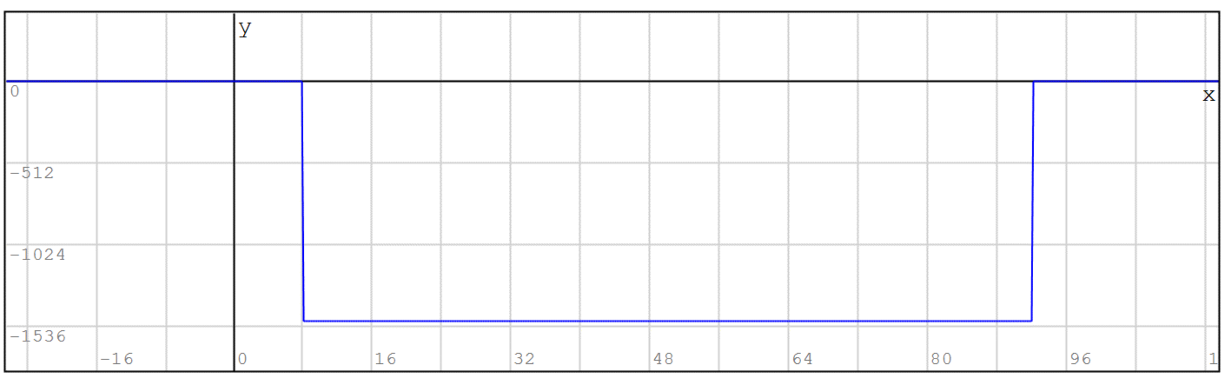
$S(x, a, n) := \text{if } \left(\left((x-a) > 0\right) \wedge (n \ge 0)\right)$
$\quad (x-a)^n$
else
$\quad 0$

Shear and bending moment eqns:

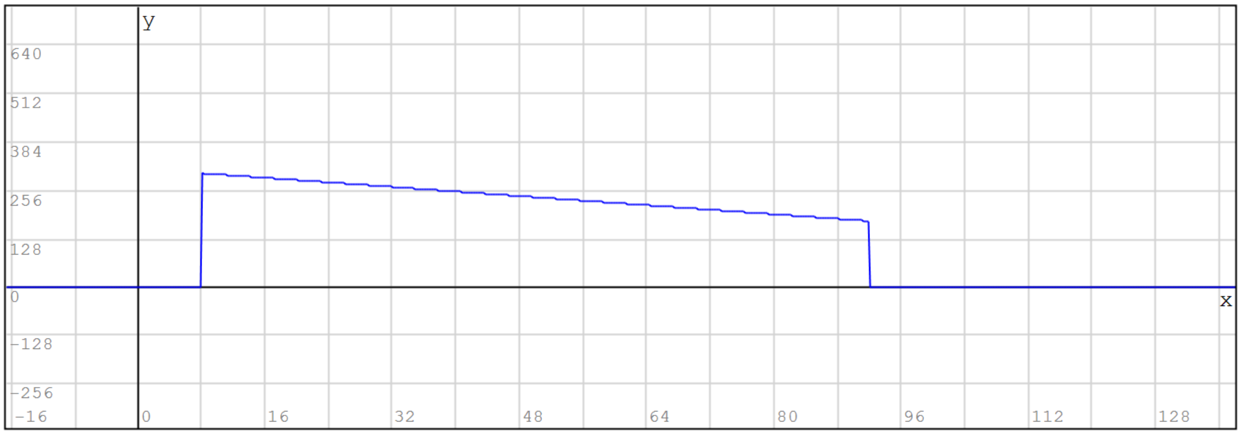
$q_z(x) := \left(\left(-R_{bz}\right) \cdot S\left(x, x_b, -1\right) + \left(R_{sy}\right) \cdot S\left(x, x_s, -1\right)\right)$

$V_z(x) := \left(\left(-R_{bz}\right) \cdot S\left(x, x_b, 0\right) + \left(R_{sy}\right) \cdot S\left(x, x_s, 0\right)\right)$

$M_z(x) := \left(\left(-R_{bz}\right) \cdot S\left(x, x_b, 1\right) + \left(R_{sy}\right) \cdot S\left(x, x_s, 1\right) + M_s \cdot S\left(x, x_s, 0\right) + M_{bz} \cdot S\left(x, x_b, 0\right)\right)$



$V_z(x\ mm)$



$M_z\left(x\ \mathrm{mm}\right)$

Moments at critical points in xy plane:

$M_{Bz} := M_z\left(x_b\right)$

$M_{sz} := M_z\left(x_s\right) = 173.7676\ \mathrm{N\ m}$

$V_{bz} := V_z\left(x_b\right) = 0\ \mathrm{N}$

$V_{sz} := V_z\left(x_s\right) = -1497.9963\ \mathrm{N}$

Total Alternating moments at critical locations on shaft 1:

Motor:

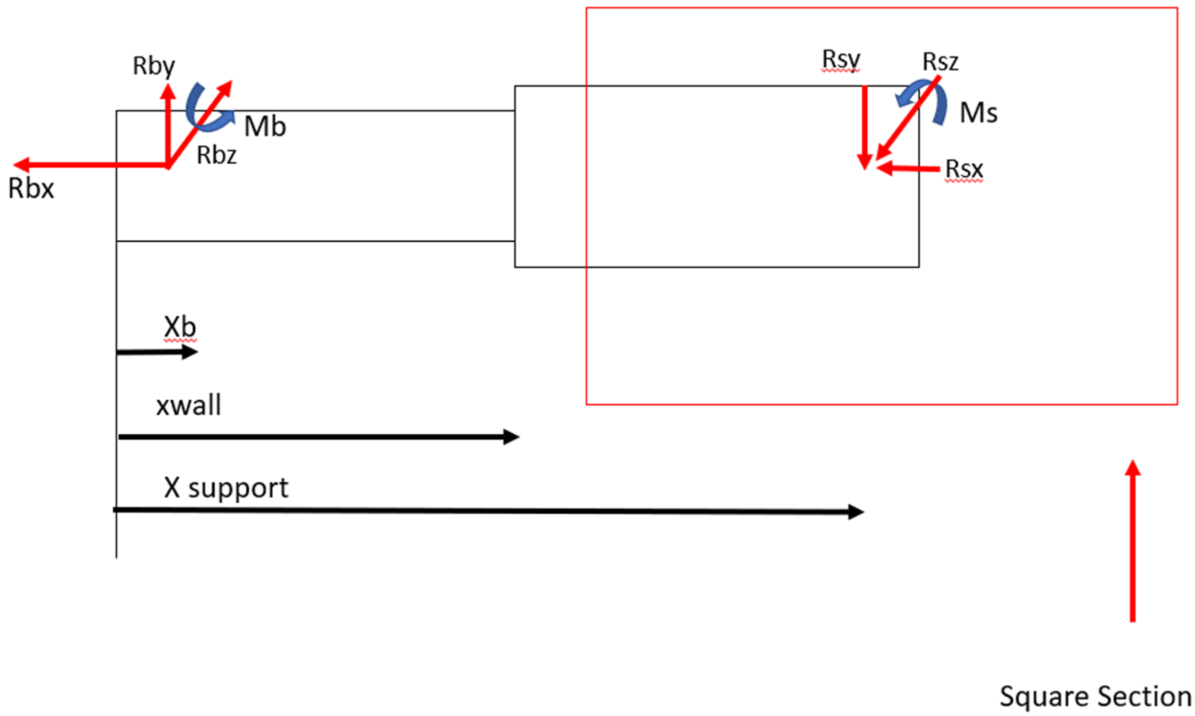
$M_{ST} := \sqrt{{M_{sz}}^2 + {M_{sy}}^2} = 214.5432\ \mathrm{N\ m}$

Bearing:

$M_{BT} := \sqrt{\left({M_{By}}^2 + {M_{Bz}}^2\right)} = 0\ \mathrm{N\ m}$

$M_{BT} := M_{bz} = 299.5993\ \mathrm{J}$

X-Y plane for axially loaded shaft (robot is on vertical surface
Shaft contains motor shaft, magnetic wheel, 1 ball bearing, 2 couplings



Distance of bearing:
$x_b := 8\ mm$

Distance of support:
$x_s := 92\ mm$

Distance of wall suppot:
$x_1 := 43\ mm$

bearing forces:
$R_{by} := 1197.5261\ N = 1197.5261\ N$

$R_{bx} := 1633.5993\ N$

Force in x-direction of support:

$R_{sx} := \left(R_{bx}\right) = 1633.5993\ N$

Moment Balance

$M_{xby} := 0\ N\ m$

$M_{xsy} := \left(R_{bx} \cdot \left(x_s - x_b\right) - M_{xby}\right) = 137.2223\ J$

Material Properties of Shaft

Length of shaft:

$L := 122\ \text{mm}$

Define the singularity function:

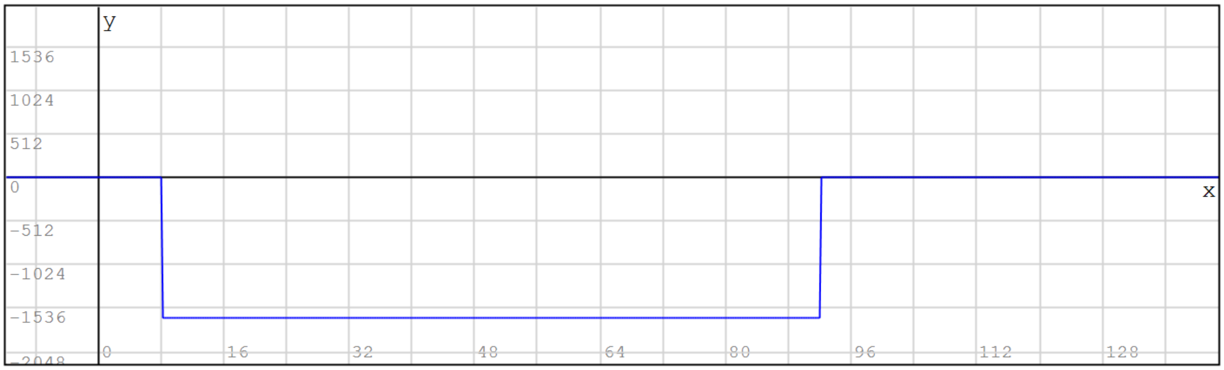
$$S(x, a, n) := \text{if } \left(\left((x - a) > 0\right) \wedge (n \geq 0)\right) \quad (x - a)^n \quad \text{else} \quad 0$$

Shear and bending moment eqns:

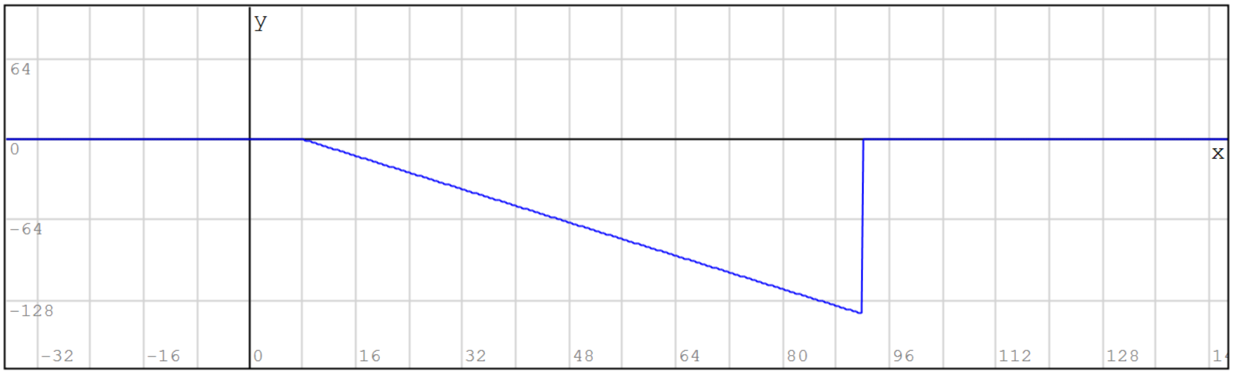
$$q_{xy}(x) := \left(\left(-R_{bx}\right) \cdot S\left(x, x_b, -1\right) + \left(R_{sx}\right) \cdot S\left(x, x_s, -1\right)\right)$$

$$V_{xy}(x) := \left(\left(-R_{bx}\right) \cdot S\left(x, x_b, 0\right) + \left(R_{sx}\right) \cdot S\left(x, x_s, 0\right)\right)$$

$$M_{xy}(x) := \left(\left(-R_{bx}\right) \cdot S\left(x, x_b, 1\right) + \left(R_{sx}\right) \cdot S\left(x, x_s, 1\right) + M_{xsy} \cdot S\left(x, x_s, 0\right)\right)$$



$V_{xy}(x\ \text{mm})$



$M_{xy}(x\ \text{mm})$

Moments at critical points in xy plane:

$$M_{xby} := M_{xy}\left(x_b\right) = 0\ \text{N m}$$

$$M_{xsy} := M_{xy}\left(x_s\right) = -137.2223\ \text{N m}$$

$V_{xbzy} := V_{xy}\left(x_b\right) = 0\ \text{N}$

$V_{xszy} := V_{xy}\left(x_s\right) = -1633.5993\ \text{N}$

y-z plane for axially loaded shaft (robot is on vertical surface)

Shaft contains motor shaft, magnetic wheel, 1 ball bearing, 2 couplings

Assuming equal distribution at the COM

$RCOM := 200\ \text{mm}$

Distance of bearing:

$x_b := 8\ \text{mm}$

Distance of support:

$x_s := 92\ \text{mm}$

Distance of wall suppot:

$x_1 := 43\ \text{mm}$

Force in z-direction of bearing:

$R_{xbz} := 1633.5993\ \text{N}$

Force in z-direction of wheel:

$R_{xsz} := \left(R_{xbz}\right) = 1633.5993\ \text{N}$

Forces in x-direction of wheel

$R_{xb1x} := 1633.5993\ \text{N} = 1633.5993\ \text{N}$

Moment Balance

$M_{xbz} := 103.5506\ \text{N m}$

$M_{xsz} := \left(R_{xbz} \cdot \left(x_s - x_b\right) - M_{xbz}\right) = 33.6717\ \text{J}$

Material Properties of Shaft

$E := 29000\ \text{ksi}$

Length of shaft:

$L := 122\ \text{mm}$

Define the singularity function:

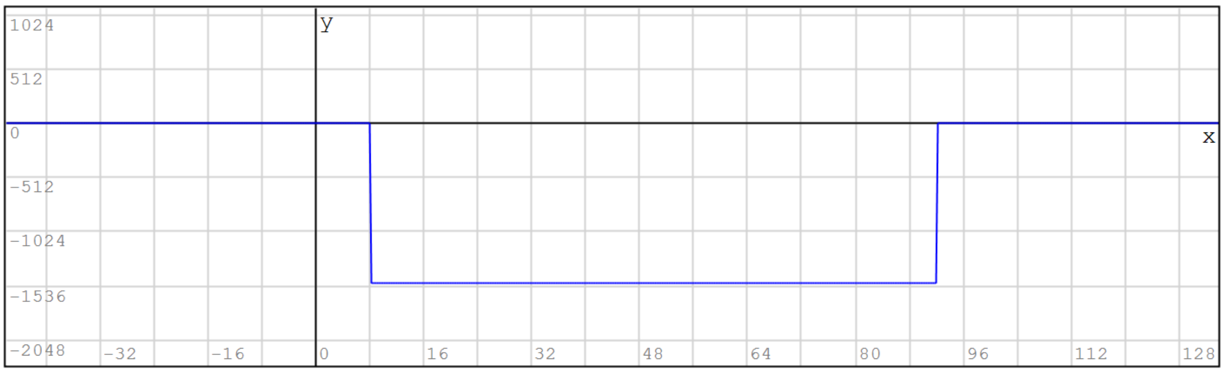
$$S(x, a, n) := \text{if } \left(\left((x-a) > 0\right) \wedge (n \geq 0)\right) \; (x-a)^n \; \text{else } 0$$

Shear and bending moment eqns:

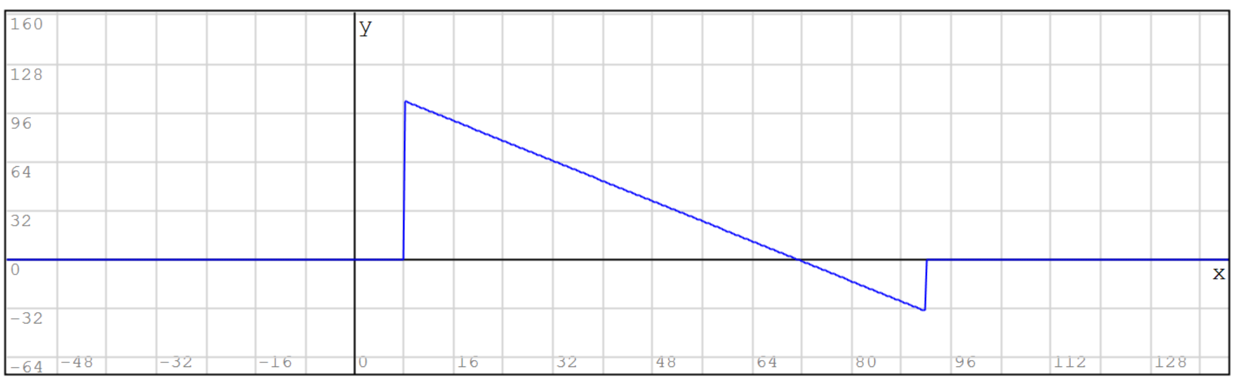
$$q_{xz}(x) := \left(\left(-R_{xb1z}\right) \cdot S\left(x, x_{b1}, -1\right) + \left(R_{xw1z}\right) \cdot S\left(x, x_w, -1\right) - F_r \cdot S\left(x, x_m, -1\right)\right)$$

$$V_{xz}(x) := \left(\left(-R_{bz} \right) \cdot S\left(x, x_b, 0\right) + \left(R_{sz} \right) \cdot S\left(x, x_s, 0\right) \right)$$

$$M_{xz}(x) := \left(\left(-R_{xbz} \right) \cdot S\left(x, x_b, 1\right) + \left(R_{xsz} \right) \cdot S\left(x, x_s, 1\right) + M_{xbz} \cdot S\left(x, x_b, 0\right) + M_{xsz} \cdot S\left(x, x_s, 0\right) \right)$$



$V_{xz}(x\ \mathrm{mm})$



$M_{xz}(x\ \mathrm{mm})$

Moments at critical points in xy plane:

$M_{xBz} := M_{xz}\left(x_b\right) = 0\ \mathrm{N\ m}$

$M_{xSz} := M_{xz}\left(x_s\right) = -33.6717\ \mathrm{N\ m}$

$V_{xBz} := V_{xz}\left(x_b\right) = 0\ \mathrm{N}$

$V_{xDz} := V_{xz}\left(x_s\right) = -1497.9963\ \mathrm{N}$

Total Alternating moments at critical locations on shaft 1:

Bearing:

$M_{xB} := \sqrt{\left(M_{xby}{}^2 + M_{xbz}{}^2 \right)} = 103.5506\ \mathrm{N\ m}$

$M_{xB} := M_{xbz} = 103.5506\ \mathrm{J}$

Wheel:

$M_{xs} := \sqrt{M_{xsy}{}^2 + M_{xsz}{}^2} = 141.2932\ \mathrm{N\ m}$

Endurance Strength (simple)

$S_{ut} := 560\ MPa$ $S_y := 460\ MPa$ $S'_e := 0.504 \cdot S_{ut} = 282.24\ MPa$

4340 cold drawn steel

$S_{ut'} := S_{ut} \cdot \frac{1}{MPa} = 560$

Step 1 Bearing)

Assume cold drawn:

$a := 4.51$ $b := -0.265$

$k_a := \left(a \cdot S_{ut'}{}^{b}\right) = 0.8432$

assume 0.11 ≤d ≤2 in

$k_{b11}\left(d_{11}\right) := 0.879 \cdot d_{11}{}^{-0.107}$

$k_c := 1$ (bending dominates)

$k_d := 1$ (assume room temp)

$k_e := 0.897$ (assume 90% reliability)

$k_f := 1$ (no misc. effects)

Corrected Endurance Strength:

initial guess: $d_{11} := 45\ mm$

$S_{e11} := k_a \cdot k_{b11}\left(d_{11}\right) \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S'_e = 124.857\ \frac{N}{mm^2}$

Step 2 (support)

Assume cold drawn:

$a := 4.51$ $b := -0.265$

$k_a := \left(a \cdot S_{ut'}{}^{b}\right) = 0.8432$

assume 2 ≤ d ≤ 10 in

$k_{b12}\left(d_{12}\right) := 0.91 \cdot d_{12}{}^{-0.157}$

(bending dominates)

(assume room temp)

(assume 90% reliability)

(no misc. effects)

Corrected Endurance Strength:

initial guess: $d_{12} := 60\ mm$

$S_{e12} := k_a \cdot k_{b12}\left(d_{12}\right) \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S'_e = 102.1385\ \frac{N}{mm^2}$

Initial Safety Factor Calculations:

$n_m := 1.1$ Material properties found in handbook or are manufacturer values

$n_s := 1.3$ Average load known although some error of overload may be present

$n_g := 1.0$ Manufacturing tolerances are tight to avoid vibrations

$n_f := 1.1$ Failure analysis based on multiaxial loading

$n_r := 1.2$ Assume a reliability of 90%

$N_f := n_m \cdot n_s \cdot n_g \cdot n_f \cdot n_r = 1.8876$

SCF Information:

Assumptions

$r := 1$

$q_{11} := 0.5$ $q_{12} := 0.5$ $q_{13} := 0.5$ $q_{14} := 0.5$ $q_{15} := 0.5$

Assume snapring at bearings and keyways motor and wheel

Snapring Assumption	Stepped Shaft Assumption	Tranverse Hole
D/d=1.1455	D/d=2.381	d/D = 0.1
r/d=0.07274	r/d=0.0635	
Kt=2.1	Kt=2.1	
Kts=1.65	Kts=1.75	

Key Way SCF is governing

Snap Ring SCF is governing

step 1 Motor SCF - bearing

$q_{11} = 0.5$ $q_{11s} := 0.56$

$K_{ts11} := 1.75$ $K_{t11} := 2.1$

$K_{fs11} := q_{11s} \cdot \left(K_{ts11} - 1\right) + 1 = 1.42$ $K_{f11} := q_{11} \cdot \left(K_{t11} - 1\right) + 1 = 1.55$

step 2 pin connection

$q_{12} = 0.5$ $q_{12s} := 0.56$

$K_{ts12} := 2.8$ $K_{t12} := 2.2$

$K_{fs12} := q_{12s} \cdot \left(K_{ts12} - 1\right) + 1 = 2.008$ $K_{f12} := q_{12} \cdot \left(K_{t12} - 1\right) + 1 = 1.6$

Assuming no yield (This assumption is checked below)

$K_{fsm11} := K_{fs11}$ $K_{fsm12} := K_{fs12}$

Minimum Diameter at Bearing on Shaft 1:

$T_b := 0 \text{ N m}$

$$d'_{11} := \left(\frac{16 \cdot N_f}{\pi} \cdot \sqrt{\frac{4 \cdot \left(K_{f11} \cdot \left(M_{BT}\right)\right)^2}{\left(S_{e11}\right)^2} + \frac{3 \cdot \left(K_{fsm12} \cdot T_b\right)^2}{\left(S_y\right)^2}} \right)^{\frac{1}{3}} = 0.0415 \text{ m}$$

$d'_{11} = 41.5072 \text{ mm}$

Testing for convergence

$$\begin{bmatrix} 1 & 45 & 41.5072 \\ 2 & 41.5072 & 41.3877 \\ 3 & 41.3835 & 41.3835 \end{bmatrix}$$

$d'_{11} := 41.3835 \text{ mm}$

Bearing diameter converges at 41.38 mm

d11 is dependent on bearing size

assume

$d_{11} := 45 \text{ mm}$

$d'_{11} := d_{11} = 0.045 \text{ m}$

Minimum Diameter at support:

$$d'_{12} := \left(\frac{16 \cdot N_f}{\pi} \cdot \sqrt{\frac{4 \cdot \left(K_{f12} \cdot M_{xs}\right)^2}{\left(S_{e12}\right)^2} + \frac{3 \cdot \left(K_{fsm12} \cdot T_b\right)^2}{\left(S_y\right)^2}} \right)^{\frac{1}{3}} = 0.0349 \text{ m}$$

$d'_{12} = 34.913 \text{ mm}$

$d_{12} := d'_{12} = 0.0349 \text{ m}$

Testing for convergence

$$\begin{bmatrix} 1 & 60 & 34.913 \\ 2 & 34.913 & 32.358 \\ 3 & 32.358 & 32.2673 \\ 4 & 32.2673 & 32.2673 \end{bmatrix}$$

The diameter converges to 32.26 mm

$d'_{12} := 32.2673 \text{ mm}$

$d'_{12} := 32.2673 \text{ mm}$

Twist Calculations for shaft 1, sections 1:

$G := 78\ \text{GPa}$

Section 1: $T_{in} := 0$ $T_{out} := 0$

$$J_{11} := \frac{\pi \cdot {d'_{11}}^{4}}{32} = 4.0258 \cdot 10^{-7}\ \text{m}^{4}$$

$$\theta_{11} := \frac{T_{in}}{G \cdot J_{11}} = 0\ \frac{\text{deg}}{\text{m}}$$

Both angles of twist are less than 3 deg/m, thus criteria is met!

Von Mises Stress Calculations:

shaft 1 section 1 :

$$I_{11} := \frac{J_{11}}{2} = 2.0129 \cdot 10^{5}\ \text{mm}^{4}$$

$$\sigma_{an11} := \frac{\left(M_{BT}\right) \cdot \frac{d'_{11}}{2}}{I_{11}} = 33.4891\ \frac{\text{N}}{\text{mm}^{2}}$$

$$\tau_{mn11} := \frac{T_{out} \cdot \frac{d'_{11}}{2}}{J_{11}} = 0\ \frac{\text{N}}{\text{mm}^{2}}$$

$$\sigma_{a11} := K_{f11} \cdot \sigma_{an11} = 51.9081\ \frac{\text{N}}{\text{mm}^{2}}$$

$$\tau_{m11} := K_{fs11} \cdot \tau_{mn11} = 0\ \frac{\text{N}}{\text{mm}^{2}}$$

Assuming fully reversed bending:

$$\sigma'_{a11} := \sigma_{a11} = 51.9081\ \frac{\text{N}}{\text{mm}^{2}}$$

$$\sigma'_{m11} := \frac{1}{\sqrt{2}} \cdot \left(6 \cdot {\tau_{m11}}^{2}\right)^{\frac{1}{2}} = 0\ \frac{\text{N}}{\text{mm}^{2}}$$

Calculated Safety Factor:

$$n_{f11} := \sqrt{\frac{1}{\left(\frac{\sigma'_{a11}}{S_{e11}}\right)^{2} + \left(\frac{\sigma'_{m11}}{S_{y}}\right)^{2}}} = 2.4053$$

Twist Calculations for shaft 1, sections 3:

$G := 78\ \text{GPa}$

Section 3:

$$J_{12} := \frac{\pi \cdot {d'_{12}}^{4}}{32}$$

$$J_{12} := \frac{\pi \cdot {d'_{12}}^{4}}{32} = 1.0643 \cdot 10^{5}\ \text{mm}^{4}$$

$$\theta_{12} := \frac{T_{in}}{G \cdot J_{12}} = 0 \ \frac{deg}{m}$$

Von Mises Stress Calculations:

shaft 1 section 2 :

$$I_{12} := \frac{J_{12}}{2} = 53213.3262 \ mm^4$$

$$\sigma_{an12} := \frac{\left(M_{ST}\right) \cdot \frac{d'_{12}}{2}}{I_{12}} = 65.047 \ \frac{N}{mm^2}$$

$$\tau_{mn12} := \frac{T_b \cdot \frac{d'_{12}}{2}}{J_{12}} = 0 \ \frac{N}{mm^2}$$

$$\sigma_{a12} := K_{f12} \cdot \sigma_{an12} = 104.0751 \ \frac{N}{mm^2}$$

$$\tau_{m12} := K_{fs12} \cdot \tau_{mn12} = 0 \ \frac{N}{mm^2}$$

Assuming fully reversed bending:

$$\sigma'_{a12} := \sigma_{a12} = 104.0751 \ \frac{N}{mm^2}$$

$$\sigma'_{m12} := \frac{1}{\sqrt{2}} \cdot \left(6 \cdot \tau_{m12}^{\ 2}\right)^{\frac{1}{2}} = 0 \ \frac{N}{mm^2}$$

Calculated Safety Factor:

$$n_{f12} := \sqrt{\frac{1}{\left(\frac{\sigma'_{a12}}{S_{e12}}\right)^2 + \left(\frac{\sigma'_{m12}}{S_y}\right)^2}} = 0.9814$$

Ignore section 2 diameter as this is a square shaft that will be computed via FEA

Minimum bearing diameter:

$n_{cycle1} := 10^6$

Bearing 1:

$C_{r1} := \left(n_{cycle1} \cdot 10^{-6}\right)^{\frac{1}{3}} \cdot \sqrt{R_{by}^2 + R_{bz}^2} = 1917.8273\ N$

$F_{br} := \sqrt{R_{by}^2 + R_{bz}^2} = 1917.8273\ N$

$F_{eff} := X \cdot V \cdot F_{br} + Y \cdot R_{bearingaxial}$

V : Rotation factor   rotating shaft
X: Radial Factor $V := 1$
Y: Thrust Factor
$R_{bearingaxial} := R_{bz}$

Using these dynamic load ratings and the provided bearing diameter chart the minimum diameter and mximum fillet of the bearings are:

$Cr := F_{br} \cdot \left(n_{cycle1} \cdot 10^{-6}\right)^{-\frac{1}{3}} = 1917.8273\ N$

use bearingPWTR30

using bearing S6009-2RSR-FD

$C_r := 17800\ N$

$C_0 := 12100\ N$

$\frac{R_{bearingaxial}}{C_0} = 0.1238$

$\frac{R_{bearingaxial}}{V \cdot R_{bx}} = 0.917$ $e := 0.42$

Value is greater then e, therefore

$Y := 1.04$
$X := 0$

$F_{eff} = 1557.9162\ N$

$L_c := \left(\frac{C_r}{F_{br}}\right)^{\frac{10}{3}} = 1680.2284$

$\left(L_c' := \frac{L_c}{10^6}\right) = 0.0017$

$$C := F_{eff} \cdot L_{c'}^{\frac{1}{3}} = 185.211\ \mathrm{N}$$

this is lower than the basic load rating of the bearing, therefore this bearing will work

Calculating the new safety factor based on new bearing diameter

$$d'_{11} := 45\ \mathrm{mm}$$

$$J_{11} := \frac{\pi \cdot d'^{4}_{11}}{32} = 4.0258 \cdot 10^{5}\ \mathrm{mm}^{4}$$

$$\theta_{11} := \frac{T_{in}}{G \cdot J_{11}} = 0\ \frac{\mathrm{deg}}{\mathrm{m}}$$

Both angles of twist are less than 3 deg/m, thus criteria is met!

Von Mises Stress Calculations:

$$I_{11} := \frac{J_{11}}{2} = 2.0129 \cdot 10^{5}\ \mathrm{mm}^{4}$$

$$\sigma_{an11} := \frac{\left(M_{BT}\right) \cdot \frac{d'_{11}}{2}}{I_{11}} = 33.4891\ \frac{\mathrm{N}}{\mathrm{mm}^{2}}$$

$$\tau_{mn11} := \frac{T_b \cdot \frac{d'_{11}}{2}}{J_{11}} = 0\ \frac{\mathrm{N}}{\mathrm{mm}^{2}}$$

$$\sigma_{a11} := K_{f11} \cdot \sigma_{an11} = 51.9081\ \frac{\mathrm{N}}{\mathrm{mm}^{2}}$$

$$\tau_{m11} := K_{fs11} \cdot \tau_{mn11} = 0\ \frac{\mathrm{N}}{\mathrm{mm}^{2}}$$

Assuming fully reversed bending:

$$\sigma'_{a11} := \sigma_{a11} = 51.9081\ \frac{\mathrm{N}}{\mathrm{mm}^{2}}$$

$$\sigma'_{m11} := \frac{1}{\sqrt{2}} \cdot \left(6 \cdot \tau_{m11}^{2}\right)^{\frac{1}{2}} = 0\ \frac{\mathrm{N}}{\mathrm{mm}^{2}}$$

Calculated Safety Factor:

$$n_{f11} := \sqrt{\frac{1}{\left(\frac{\sigma'_{a11}}{S_{e11}}\right)^{2} + \left(\frac{\sigma'_{m11}}{S_y}\right)^{2}}} = 2.4053$$

Slope and Deflection Equations:

Building off of the singularity equations for shear and bending moment, the slope and deflection formulas are defined as follows:

$$d_b := d_{11} = 45\ \mathrm{mm}$$ Bearing diameter

$$I := \frac{\pi \cdot d_b^{4}}{64} = 2.0129 \cdot 10^{5}\ \mathrm{mm}^{4}$$

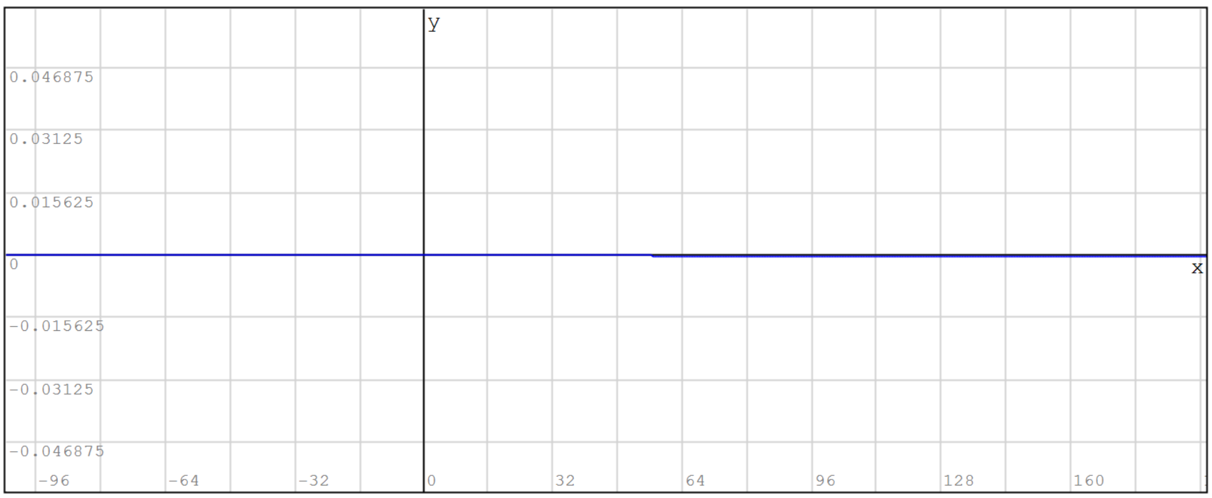
x-y plane

$$\theta_y(x) := \frac{1}{E \cdot I} \cdot \left(\frac{-R_{by}}{2} \cdot S(x, x_b, 2) + \frac{(R_{sy})}{2} \cdot S(x, x_s, 2) \right) + C_{1xy}$$

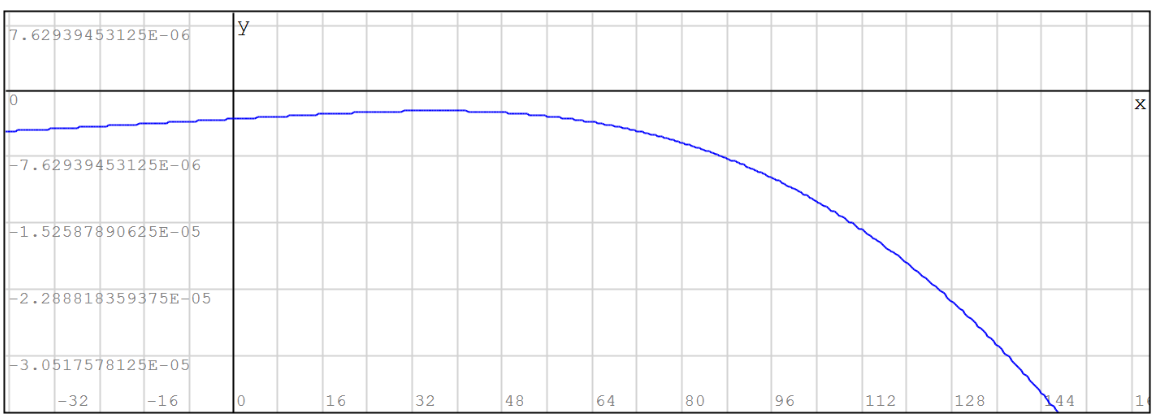
$$\delta_y(x) := \frac{1}{E \cdot I} \cdot \left(\frac{-R_{by}}{2} \cdot S(x, x_b, 3) + \frac{(R_{sy})}{6} \cdot S(x, x_s, 3) \right) + C_{1xy} \cdot x + C_{2xy}$$

$$C_{1xy} := \frac{R_{by} \cdot (x_s - x_b)^3 - R_{sy} \cdot (x_s - x_s)^3}{6 \cdot E \cdot I \cdot (x_s - x_b)} = 3.499 \cdot 10^{-5}$$

$$C_{2xy} := -C_{1xy} \cdot x_s = -0.003219 \text{ mm}$$



$\theta_y(x \text{ mm})$



$\delta_y(x \text{ mm})$

Slopes at critical points in xy plane:

$$\theta_{yb} := \theta_y\left(x_b\right) = 3.4991 \cdot 10^{-5}\ \text{rad}$$

$$\theta_{ys} := \theta_y\left(x_s\right) = -6.9982 \cdot 10^{-5}\ \text{rad}$$

Deflections at critical points in xy plane:

$$\delta_{yb} := \delta_y\left(x_b\right) = -0.0029\ \text{mm}$$

$$\delta_{ys} := \delta_y\left(x_s\right) = -0.0088\ \text{mm}$$

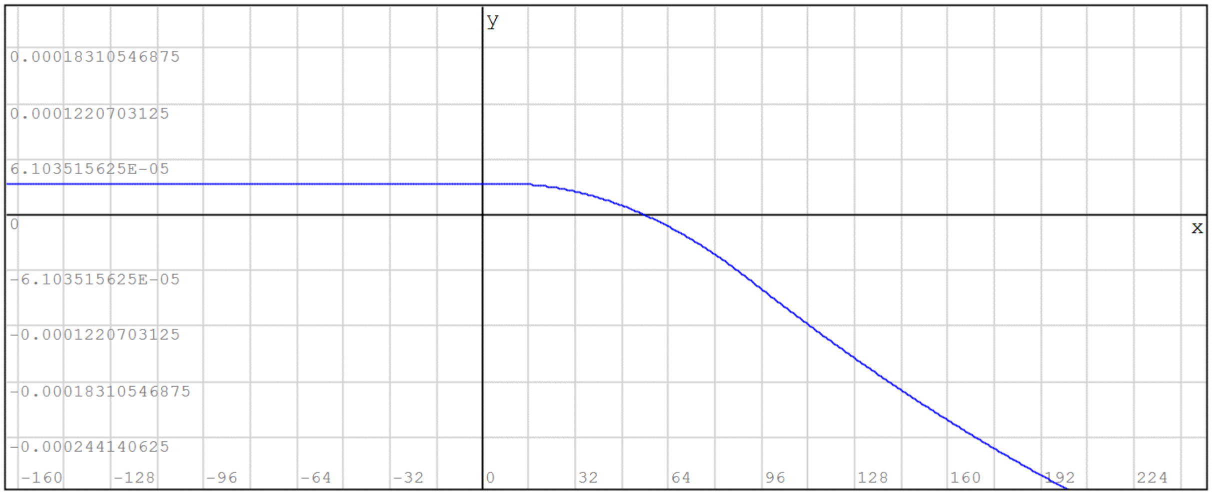
y-z plane

$$\theta_z(x) := \frac{1}{E \cdot I} \cdot \left(\frac{-R_{bz}}{2} \cdot S\left(x, x_b, 2\right) + \frac{\left(R_{sz}\right)}{2} \cdot S\left(x, x_s, 2\right)\right) + C_{1yz}$$

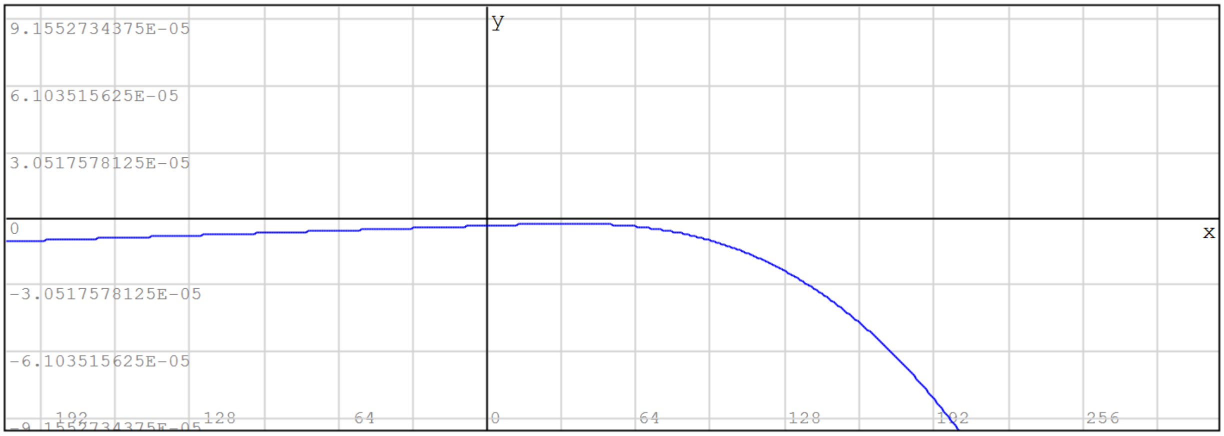
$$\delta_z(x) := \frac{1}{E \cdot I} \cdot \left(\frac{-R_{bz}}{6} \cdot S\left(x, x_b, 3\right) + \frac{\left(R_{sz}\right)}{6} \cdot S\left(x, x_s, 3\right)\right) + C_{1yz} \cdot x + C_{2yz}$$

$$C_{1yz} := \frac{-R_{bz} \cdot \left(x_s - x_b\right)^3 + R_{sz} \cdot \left(x_s - x_s\right)^3}{6 \cdot E \cdot I \cdot \left(x_s - x_b\right)} = -4.377 \cdot 10^{-5}$$

$$C_{2yz} := -C_{1yz} \cdot x_b = 0.0003502\ \text{mm}$$



$\theta_y\left(x\ \text{mm}\right)$



$\delta_y(x\ \mathrm{mm})$

Slopes at critical points in yz plane:

$\theta_{zb} := \theta_z(x_b) = -4.377 \cdot 10^{-5}\ \mathrm{rad}$

$\theta_{zs} := \theta_z(x_s) = -0.0002\ \mathrm{rad}$

Deflections at critical points in yz plane:

$\delta_{zb} := \delta_z(x_b) = 0\ \mathrm{mm}$

$\delta_{zs} := \delta_z(x_s) = -0.0074\ \mathrm{mm}$

Total deflections at critical locations on shaft 1:

Bearing 1:

$\delta_{Tb} := \sqrt{\delta_{yb}^{\ 2} + \delta_{zb}^{\ 2}} = 0.0029\ \mathrm{mm}$

Wheel:

$\delta_{Ts} := \sqrt{\left(\delta_{ys}^{\ 2} + \delta_{zs}^{\ 2}\right)} = 0.0115\ \mathrm{mm}$

Total slopes at critical locations on shaft 1:

Bearing 1:

$\theta_{Tb} := \sqrt{\theta_{yb}^{\ 2} + \theta_{zb}^{\ 2}} = 5.6038 \cdot 10^{-5}\ \mathrm{rad}$

Wheel:

$$\theta_{Ts} := \sqrt{\left(\theta_{ys}^{\ 2} + \theta_{zs}^{\ 2}\right)} = 0.0002\ \text{rad}$$

$d1r := 45\ \text{mm}$
$d2r := d'_{12} = 0.0323\ \text{m}$

Check maximum stepped shaft dimensions

$r12 := 0.2 \cdot d1r = 9\ \text{mm}$
$r_{23} := 0.2 \cdot d2r = 6.4535\ \text{mm}$

therefor max diameter for d12 is
$d_{2max} := d1r + r12 = 54\ \text{mm}$

The max diamerter for d3 is
$d_{3max} := d2r + r_{23} = 38.7208\ \text{mm}$

$d_{3min} := d2r - r_{23} = 25.8138\ \text{mm}$

Bearing size needs to be reiterted

using bearing S6009-2RSR-FD

$C_r := 17800\ \text{N}$

$C_0 := 12100\ \text{N}$

$$\frac{R_{bearingaxial}}{C_0} = 0.1238$$

$$\frac{R_{bearingaxial}}{V \cdot F_{br}} = 0.7811 \qquad e := 0.42$$

Value is greater then e, therefore

$Y := 1.04$
$X := 0$

$F_{eff} = 1557.9162\ \text{N}$

$$L_c := \left(\frac{C_r}{F_{br}}\right)^{\frac{10}{3}} = 1680.2284$$

$$\left(L_{c'} := \frac{L_c}{10^6}\right) = 0.0017$$

$$C := F_{eff} \cdot L_{c'}^{\ \frac{1}{3}} = 185.211\ \text{N}$$

this is lower than the basic load rating of the bearing, therefore this bearing will work

final Shaft 1 diameters:

$d1r := 45\ \text{mm}$

$d2r := 50\ mm$

Check assumptions

1. Check kb diameter

2 in < d1r<10 in

2 in < d2r < 10 in

0.11 ≤d ≤2 in

2. Check SCF

$$\frac{d2r}{d1r} = 1.1111$$

$$k_{t1} := 2.15$$

$$k_{ts} := 1.8$$

$$r_1 := 0.05 \cdot d1r = 2.25\ mm$$

3. Check Step Size

$$\frac{d2r}{d1r} = 1.1111$$

$$r_2 := 0.2 \cdot d1r = 9\ mm$$

$$\frac{r_2}{d1r} = 0.2$$

$$k_t := 1.35$$

$$K_{ts} := 1.25$$

Assumed Kfs=Kfsm

$$K_{fs11} \cdot \tau_{m11} = 0\ ksi \qquad \frac{S_y}{2} = 33.3587\ ksi \quad \text{Thus} \quad K_{fs11} = K_{fsm11}$$

$$K_{fs12} \cdot \tau_{m12} = 0\ ksi \qquad \& \qquad K_{fs12} = K_{fsm12}$$

Assumed Kfm=Kf

$$K_{f11} \cdot \left|\left(\sigma'_{a11} + \sigma'_{m11}\right)\right| = 11.6694\ kpsi$$

$$K_{f12} \cdot \left|\left(\sigma'_{a12} + \sigma'_{m12}\right)\right| = 24.1517\ kpsi$$

$$< \quad S_y = 66.7174\ ksi \quad \text{Thus} \quad K_{fm11} = K_{f11} \ \& \ K_{fm12} = K_{f12}$$

Assumption is valid!

Shaft Recalculations Using Actual Diameters

Endurance Strength (simple)

$S_{ut} := 560\ \text{MPa}$ $S_y := 460\ \text{MPa}$ $S'_e := 0.504 \cdot S_{ut} = 282.24\ \text{MPa}$

Assume cold drawn:

$a := 4.51$ $b := -0.265$

$k_a := \left(a \cdot S_{ut'}{}^{b}\right) = 0.8432$

assume 2 ≤ d ≤ 10 in

$k_{b11}(d_{11}) := 0.91 \cdot d_{11}^{-0.157}$

$k_c := 1$ (bending dominates)

$k_d := 1$ (assume room temp)

$k_e := 0.897$ (assume 90% reliability)

$k_f := 1$ (no misc. effects)

Corrected Endurance Strength:

initial guess: $d_{11} := 45\ \text{mm}$

$S_{e11} := k_a \cdot k_{b11}(d_{11}) \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S'_e = 106.8574\ \frac{\text{N}}{\text{mm}^2}$

$d_{11} := 45\ \text{mm}$

Step 2 (no load - wheel support)

Assume cold drawn:

$a := 4.51$ $b := -0.265$

$k_a := \left(a \cdot S_{ut'}{}^{b}\right) = 0.8432$

assume 2 ≤ d ≤ 10 in

$k_{b12}(d_{12}) := 0.91 \cdot d_{12}^{-0.157}$

(bending dominates)

(assume room temp)

(assume 90% reliability)

(no misc. effects)

Corrected Endurance Strength:

Actual Diameter: $d_{12} := 45\ \text{mm}$

$S_{e12} := k_a \cdot k_{b12}\left(d_{12}\right) \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S'_e = 106.8574 \frac{N}{mm^2}$

$d_{12} := 55\ mm$

SCF Information:

Assumptions

$r := 0.05$ $q_{11} := 0.68$ $q_{12} := 0.68$ $q_{13} := 0.68$

$q_{11s} := 0.73$ $q_{12s} := 0.73$ $q_{13s} := 0.73$

Stepped Shaft Assumption

$\frac{d_{12}}{d_{11}} = 1.2222$

$\frac{r}{d_{12}} = 0.9091 \cdot \frac{1}{m}$

Kt = 1.65 Kt =1.4
Kts =1.45 Kts =1.2

Key way

Kt=2.2
Kts=3

Step 1 Bearing

$K_{ts11} := 1.45$ $K_{t11} := 1.2$

$K_{fs11} := q_{11s} \cdot \left(K_{ts11} - 1\right) + 1 = 1.3285$ $K_{f11} := q_{11} \cdot \left(K_{t11} - 1\right) + 1 = 1.136$

Step 2

$K_{ts12} := 1.45$ $K_{t12} := 1.65$

$K_{fs12} := q_{12s} \cdot \left(K_{ts12} - 1\right) + 1 = 1.3285$ $K_{f12} := q_{12} \cdot \left(K_{t12} - 1\right) + 1 = 1.442$

step 3

$K_{ts13} := 1.6$ $K_{t13} := 2.45$

$K_{fs13} := q_{13s} \cdot \left(K_{ts13} - 1\right) + 1 = 1.438$ $K_{f13} := q_{13} \cdot \left(K_{t13} - 1\right) + 1 = 1.986$

Assuming no yield

$K_{fsm11} := K_{fs11}$ $K_{fm11} := K_{f11}$

$K_{fsm12} := K_{fs12}$ $K_{fm12} := K_{f12}$

$K_{fsm13} := K_{fs13}$ $K_{fm13} := K_{f13}$

Twist Calculations for shaft 1, sections 1:

$G := 78\ GPa$

Section 1:

Now accounting for hollow shaft (ID = 32 mm)

$$J_{11} := \frac{\pi \cdot \left(d_{11}{}^{4}\right)}{32} = 4.0258 \cdot 10^{5}\ mm^{4}$$

$$\theta_{11} := \frac{T_{in}}{G \cdot J_{11}} = 0\ \frac{deg}{m}$$

Both angles of twist are less than 3 deg/m, thus criteria is met!

Von Mises Stress Calculations:

shaft 1 section 1 :

$$I_{11} := \frac{J_{11}}{2} = 2.0129 \cdot 10^{5}\ mm^{4}$$

$$\sigma_{an11} := \frac{\left(M_{BT}\right) \cdot \frac{d'_{11}}{2}}{I_{11}} = 33.4891\ \frac{N}{mm^{2}}$$

$$\tau_{mn11} := \frac{T_{out} \cdot \frac{d'_{11}}{2}}{J_{11}} = 0\ \frac{N}{mm^{2}}$$

$$\sigma_{a11} := K_{f11} \cdot \sigma_{an11} = 38.0436\ \frac{N}{mm^{2}}$$

$$\tau_{m11} := K_{fs11} \cdot \tau_{mn11} = 0\ \frac{N}{mm^{2}}$$

Assuming fully reversed bending:

$$\sigma'_{a11} := \sigma_{a11} = 38.0436\ \frac{N}{mm^{2}}$$

$$\sigma'_{m11} := \frac{1}{\sqrt{2}} \cdot \left(6 \cdot \tau_{m11}{}^{2}\right)^{\frac{1}{2}} = 0\ \frac{N}{mm^{2}}$$

Calculated Safety Factor:

$$n_{f11} := \sqrt{\frac{1}{\left(\frac{\sigma'_{a11}}{S_{e11}}\right)^{2} + \left(\frac{\sigma'_{m11}}{S_{y}}\right)^{2}}} = 2.8088$$

Twist Calculations for shaft 1, sections 3:

$$G := 78\ GPa$$

Section 2:

$$J_{12} := \frac{\pi \cdot d_{12}{}^{4}}{32} = 8.9836 \cdot 10^{5}\ mm^{4}$$

$$\theta_{12} := \frac{T_{in}}{G \cdot J_{12}} = 0\ \frac{deg}{m}$$

Both angles of twist are less than 3 deg/m, thus criteria is met!

Von Mises Stress Calculations:

shaft 1 section 2 :

$$I_{12} := \frac{J_{12}}{2} = 4.4918 \cdot 10^{5}\ mm^{4}$$

$$\sigma_{an12} := \frac{(M_{ST}) \cdot \frac{d_{12}}{2}}{I_{12}} = 13.1349\ \frac{N}{mm^2} \qquad \tau_{mn12} := \frac{T_b \cdot \frac{d_{12}}{2}}{J_{12}} = 0\ \frac{N}{mm^2}$$

$$\sigma_{a12} := K_{f12} \cdot \sigma_{an12} = 18.9405\ \frac{N}{mm^2} \qquad \tau_{m12} := K_{fs12} \cdot \tau_{mn12} = 0\ \frac{N}{mm^2}$$

Assuming fully reversed bending:

$$\sigma'_{a12} := \sigma_{a12} = 18.9405\ \frac{N}{mm^2} \qquad \sigma'_{m12} := \frac{1}{\sqrt{2}} \cdot \left(6 \cdot \tau_{m12}^{\ 2}\right)^{\frac{1}{2}} = 0\ \frac{N}{mm^2}$$

Calculated Safety Factor:

$$n_{f12} := \sqrt{\frac{1}{\left(\frac{\sigma'_{a12}}{S_{e12}}\right)^2 + \left(\frac{\sigma'_{m12}}{S_y}\right)^2}} = 5.6417$$

Slope and Deflection Equations:

Building off of the singularity equations for shear and bending moment, the slope and deflection formulas are defined as follows:

$$I := \frac{\pi \cdot d_{11}^{\ 4}}{64} = 2.0129 \cdot 10^{5}\ mm^{4}$$

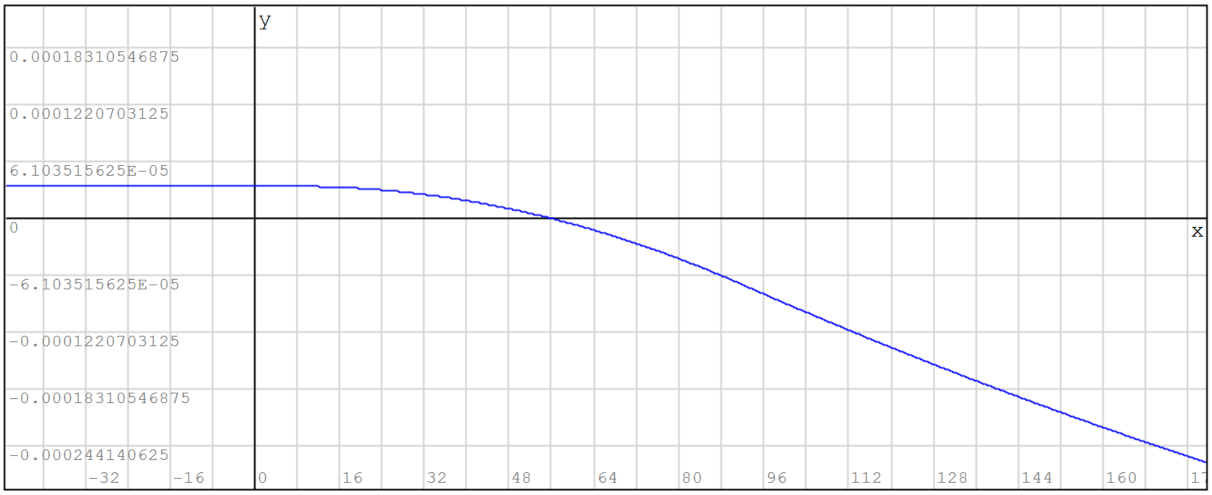
x-y plane

$$\theta_y(x) := \frac{1}{E \cdot I} \cdot \left(\frac{-R_{by}}{2} \cdot S(x, x_b, 2) + \frac{(R_{sy})}{2} \cdot S(x, x_s, 2)\right) + C_{1xy}$$

$$\delta_y(x) := \frac{1}{E \cdot I} \cdot \left(\frac{-R_{by}}{2} \cdot S(x, x_b, 3) + \frac{(R_{sy})}{6} \cdot S(x, x_s, 3)\right) + C_{1xy} \cdot x + C_{2xy}$$

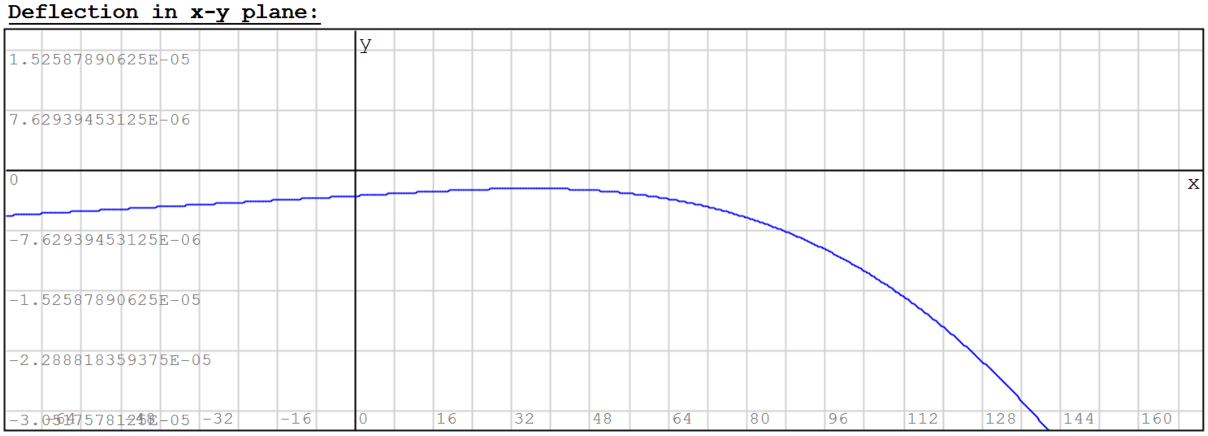
$$C_{1xy} := \frac{R_{by} \cdot \left(x_s - x_b\right)^3 - R_{sy} \cdot \left(x_s - x_s\right)^3}{6 \cdot E \cdot I \cdot \left(x_s - x_b\right)} = 3.499 \cdot 10^{-5}$$

$$C_{2xy} := -C_{1xy} \cdot x_s = -0.003219 \text{ mm}$$



$\theta_y\left(x \text{ mm}\right)$

Deflection in x-y plane:



$\delta_y\left(x \text{ mm}\right)$

Slopes at critical points in xy plane:

$$\theta_{yb1} := \theta_y\left(x_b\right) = 3.4991 \cdot 10^{-5} \text{ rad}$$

$$\theta_{ys} := \theta_y\left(x_s\right) = -6.9982 \cdot 10^{-5} \text{ rad}$$

Deflections at critical points in xy plane:

$$\delta_{yb1} := \delta_y\left(x_b\right) = -0.0029 \text{ mm}$$

$$\delta_{ys} := \delta_y\left(x_s\right) = -0.0088 \text{ mm}$$

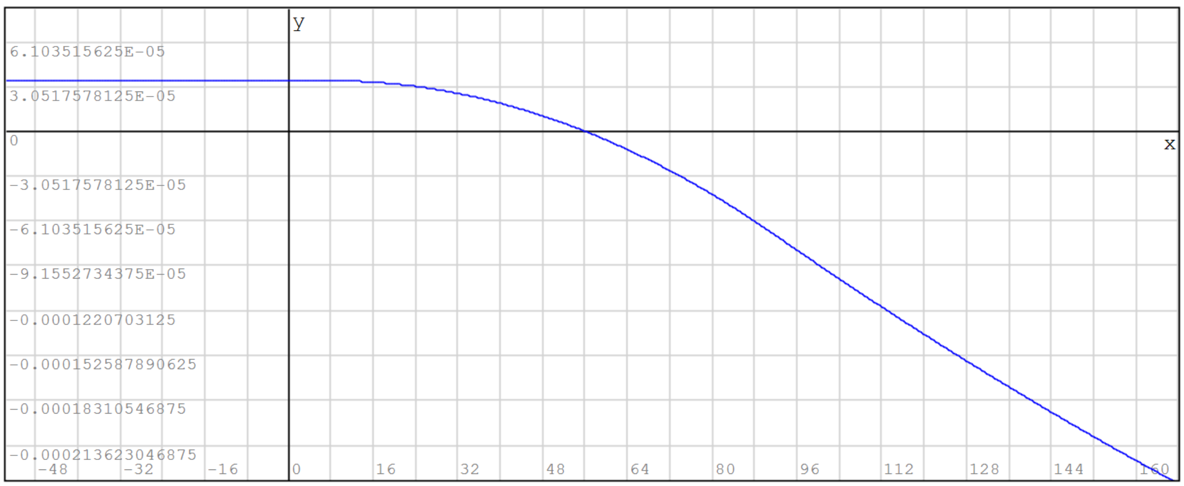
y-z plane

$$\theta_z(x) := \frac{1}{E \cdot I} \cdot \left(\frac{-R_{bz}}{2} \cdot S(x, x_b, 2) + \frac{(R_{sz})}{2} \cdot S(x, x_s, 2) \right) + C_{1yz}$$

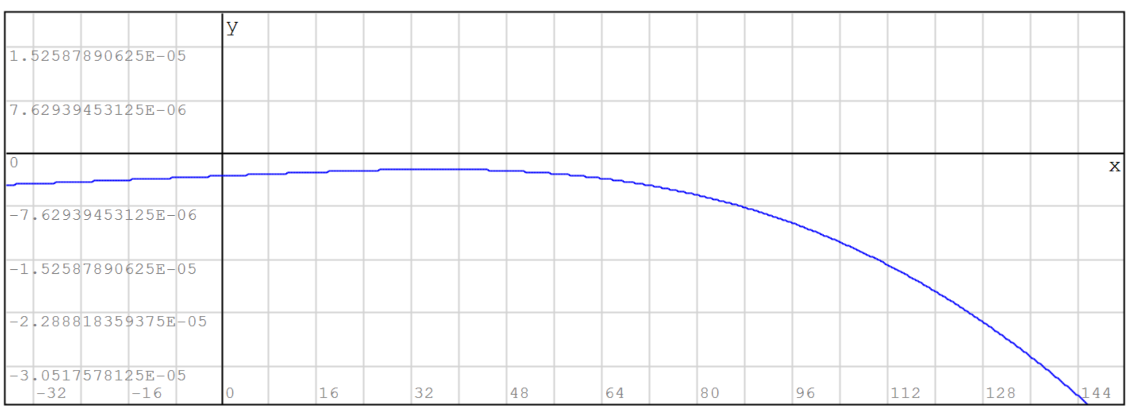
$$\delta_z(x) := \frac{1}{E \cdot I} \cdot \left(\frac{-R_{bz}}{6} \cdot S(x, x_b, 3) + \frac{(R_{sz})}{6} \cdot S(x, x_s, 3) \right) + C_{1yz} \cdot x + C_{2yz}$$

$$C_{1yz} := \frac{-R_{bz} \cdot (x_s - x_b)^3 + R_{sz} \cdot (x_s - x_s)^3}{6 \cdot E \cdot I \cdot (x_s - x_b)} = -4.377 \cdot 10^{-5}$$

$$C_{2yz} := -C_{1yz} \cdot x_b = 0.0003502 \text{ mm}$$



$\theta_y(x \text{ mm})$



$\delta_y(x \text{ mm})$

Slopes at critical points in yz plane:

$$\theta_{zb} := \theta_z\left(x_b\right) = -4.377 \cdot 10^{-5}\ \text{rad}$$

$$\theta_{zs} := \theta_z\left(x_s\right) = -0.0002\ \text{rad}$$

Deflections at critical points in yz plane:

$$\delta_{zb} := \delta_z\left(x_b\right) = 0\ \text{mm}$$

$$\delta_{zs} := \delta_z\left(x_s\right) = -0.0074\ \text{mm}$$

Total deflections at critical locations on shaft 1:

Bearing 1:

$$\delta_{Tb} := \sqrt{\delta_{yb}^{\ 2} + \delta_{zb}^{\ 2}} = 0.0029\ \text{mm}$$

Support:

$$\delta_{Ts} := \sqrt{\left(\delta_{ys}^{\ 2} + \delta_{zs}^{\ 2}\right)} = 0.0115\ \text{mm}$$

Total slopes at critical locations on shaft 1:

Bearing 1:

$$\theta_{Tb} := \sqrt{\theta_{yb}^{\ 2} + \theta_{zb}^{\ 2}} = 5.6038 \cdot 10^{-5}\ \text{rad}$$

Wheel:

$$\theta_{Ts} := \sqrt{\left(\theta_{ys}^{\ 2} + \theta_{zs}^{\ 2}\right)} = 0.0002\ \text{rad}$$

total deflection < 0.127 mm   Deflection criteria met!

Assumption and Criterion Check:

Assumed Kfs=Kfsm

$$K_{fs11} \cdot \tau_{m11} = 0\ \text{ksi}$$

$$K_{fs12} \cdot \tau_{m12} = 0\ \text{ksi}$$

$$\frac{S_y}{2} = 33.3587\ \text{ksi}$$

Thus $K_{fs11} = K_{fsm11}$ & $K_{fs12} = K_{fsm12}$

Assumption is valid!

Assumed Kfm=Kf

Assumption is valid!

$$K_{f11} \cdot \left|\left(\sigma'_{a11} + \sigma'_{m11}\right)\right| = 6.2682\ \text{kpsi}$$

$$K_{f12} \cdot \left|\left(\sigma'_{a12} + \sigma'_{m12}\right)\right| = 3.9613\ \text{kpsi}$$

$< \quad S_y = 66.7174\ \text{ksi}$

Thus $K_{fm11} = K_{f11}$ & $K_{fm13} = K_{f12}$

All twists must be less than 3 deg/m:

$\theta_{11} = 0 \frac{deg}{m}$ $\theta_{12} = 0 \frac{deg}{m}$ Criterion is met!

All deflections must be less than 0.127 mm: Criterion is met!

$\delta_{Tb} = 2.9392 \cdot 10^{-6}\ m$ $\delta_{Ts} = 0.0115\ mm$

All assumptions/criterion is met, Analytic Shaft Complete

FEA will validate the square shaft

Conclusion

The minimum shaft diameter was calculated for the loaded sections, (section d1 and section d3) and factor of safety was determined at the rounded shaft diameters. However, section d3 will be a square shaft that will be determined with FEA. and the analytics solution method shown in this calculation will be ignored.

The minimum shaft diameter at section 1 is 41.38 mm and the minimum diameter at section 2 is 32.27mm

The rounded diameters are as follows:
Section 1: 45 mm
Section 2: 55mm

at section 1 and 2 the factory of safety are as follows:
Section 1: 2.8
Section 2: 5.64

Therefore the shaft does not fail at the most conservative loading conditions

Appendix D8: V Support Buckling

Title: Buckling Calculations
Date: April 10, 2021
Author: Kenny Okeke

Objective:
To determine the maximum mass of the robot before it fails to buckling. Steel 4130 is the material choice.

Variables: P.cr - Critical Load [N]
E - Modulus of Elasticity [GPa]
I - Moment of Inertia of the Cross-Section [m^4]
L - Length of the Column [m]
a - Length of Cross-Section [m]
b - Width of Cross-Section [m]
g - Acceleration due to Gravity [m/s^2]
m - Mass of Robot [kg]
W - Weight of Robot [N]
N - Number of V Brackets

Solution Method: Determine the Moment of Inertia
Determine the critical load
Determine the maximum mass

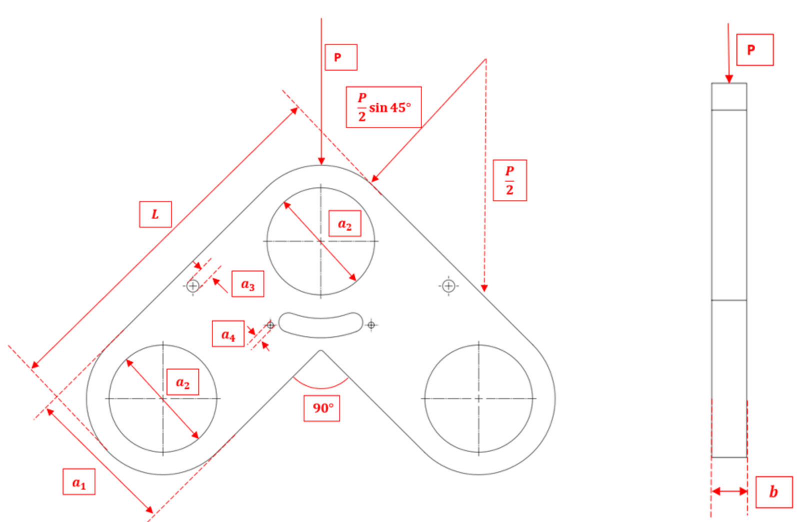
Equation: $P_{cr.equation} := \frac{\pi^2 \cdot E \cdot I}{{L_e}^2}$

Assumptions: The moment of inertia of the cross section considers the smallest cross section of the column
The material considered is Steel 4130: $E := 200 \text{ GPa}$
Acceleration is due to gravity: $g := 9.81 \frac{\text{m}}{\text{s}^2}$
Mass of the robot is assumed: $m := 500 \text{ kg}$
The V-Support is asssumed to be made up of two identical rectangular sections.

Calculations:

Assumptions: This case considers that both ends of the column are pinned: Le = L
The cross section is assumed to be a rectangular section
Slot dimensions are ignored for the moment of inertia calculations
The robot is supported by 4 separate V brackets

Sketch:



Analysis: $a_1 := 100\ \text{mm}$

$a_2 := 70.40\ \text{mm}$

$a_3 := 8\ \text{mm}$

$a_4 := 4\ \text{mm}$

$b := 20\ \text{mm}$

$L := 246.05\ \text{mm}$

$L_e := L = 246.05\ \text{mm}$

$N := 4$

$m = 500\ \text{kg}$

$$I_1 := \frac{a_1 \cdot b^3}{12} - \frac{a_2 \cdot b^3}{12} = 1.9733 \cdot 10^{-8}\ \text{m}^4$$

$$I_2 := \frac{b \cdot a_1^3}{12} - \frac{b \cdot a_2^3}{12} = 1.0851 \cdot 10^{-6}\ \text{m}^4$$

$$P_{cr.1} := \frac{2}{\sin\left(45 \cdot \frac{\pi}{180}\right)} \cdot \frac{\pi^2 \cdot E \cdot I_1}{L_e^2} = 1.8198 \cdot 10^6\ \text{N}$$

$$P_{cr.2} := \frac{2}{\sin\left(45 \cdot \frac{\pi}{180}\right)} \cdot \frac{\pi^2 \cdot E \cdot I_2}{L_e^2} = 1.0007 \cdot 10^8\ \text{N}$$

$P_{cr} :=$ if $P_{cr.1} < P_{cr.2}$
    $P_{cr.1}$
  else
    $P_{cr.2}$

$P_{cr} = 1.8198 \cdot 10^6\ \text{N}$

$W_{max} := P_{cr} \cdot N = 7.2793 \cdot 10^6\ \text{N}$ Each of the 4 V brackets will be supporting the same load

$$m_{max} := \frac{W_{max}}{g} = 7.4203 \cdot 10^5\ \text{kg}$$

$m < m_{max} = 1$ Therefore the mass of the robot is less than the maximum mass

Conclusion:

These calculations determined the maximum mass of the robot is 7.4203E5 kg before buckling occurs. The actual mass of the robot is 500 kg, so it will not fail to buckling. The smallest cross-section was considered, thus it can be assumed that all other cross sections will also resist buckling.

References:

Engineer4Free. “Column buckling example problem #1: both ends pinned,” YouTube, 2 November 2017. [Video file]. Available: https://www.youtube.com/watch?v=SIRY5ZZDEF0&t=452s&ab_channel=Engineer4Free. [Accessed: 1 March 2021]

AZoM, "AZO Materials," 12 November 2012. [Online]. Available: https://www.azom.com/article.aspx?ArticleID=6742. [Accessed 1 March 2021].

Appendix D9 : V Support Impact

Impact Loading Calculations
Date: April 10, 2021
Author: Kenny Okeke

Objective:
To determine if the robot material can resist failure due to impact loading. Steel 4130 is the material considered.

Variables: σ.max - Maximum Stress [MPa]
σ.yield - Yield Stress [MPa]
P - Load [N]
A - Area [m^2]
E - Modulus of Elasticity [GPa]
h - Height [m]
L - Length [m]
g - Acceleration due to Gravity [m/s^2]
m - Mass of Robot [kg]
W - Weight of Robot [N]
N - Number of V brackets

Solution Method: Determine the load
Determine the area
Determine the stress induced by impact loading
Compare this stress with the yield strength of the material

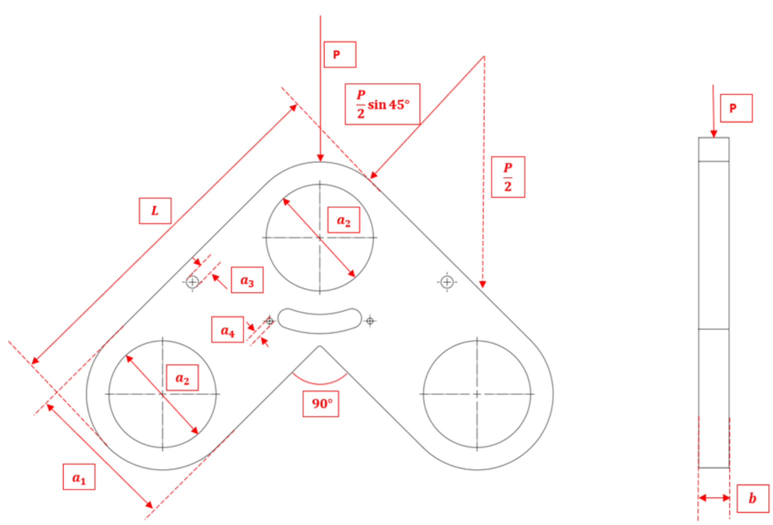
Equation:

$$\sigma_{max.equation} := \frac{P}{A} + \sqrt{\left(\frac{P}{A}\right)^2 + \frac{2 \cdot E \cdot P \cdot h}{A \cdot L}}$$

Assumptions: The load will be defined as the weight of the divided by the number of V brackets
The material considered is Steel 4130: $E := 200\ \text{GPa}$
Acceleration is due to gravity: $g := 9.81\ \frac{\text{m}}{\text{s}^2}$
The height is considered to be the height of the obstacle: $h := 3\ \text{in} = 0.0762\ \text{m}$
Mass of the robot is considered to be: $m := 500\ \text{kg}$
The V-Support is assumed to be made up of two identical rectangular sections
The slot is not considered in the area calulcations

Concept 2:

Sketch:



Analysis: $m = 500\ \text{kg}$

$W := m \cdot g = 4905\ \text{N}$

$N := 4$

$P := \frac{W}{N} = 1226.25\ \text{N}$

$P_{cr} := \frac{P}{2} \cdot \sin\left(45 \cdot \frac{\pi}{180}\right) = 433.5448\ \text{N}$

$a_1 := 100\ \text{mm}$

$a_2 := 70.40\ \text{mm}$

$a_3 := 8\ \text{mm}$

$a_4 := 4\ \text{mm}$

$b := 20\ \text{mm}$

$L := 246.05\ \text{mm}$

$$A_1 := \left(a_1 \cdot L\right) - \left(\pi \cdot \left(\frac{2 \cdot a_2}{2}\right)^2\right) - \left(\pi \cdot \left(\frac{a_3}{2}\right)^2\right) - \left(\pi \cdot \left(\frac{a_4}{2}\right)^2\right) = 0.009\ \text{m}^2$$

$A_2 := b \cdot L = 0.0049\ \text{m}^2$

Calculations

$$\sigma_1 := \frac{P_{cr}}{A_1} + \sqrt{\left(\frac{P_{cr}}{A_1}\right)^2 + \frac{2 \cdot E \cdot P_{cr} \cdot h}{A_1 \cdot L}} = 7.7418 \cdot 10^7\ \text{Pa}$$

$$\sigma_2 := \frac{P_{cr}}{A_2} + \sqrt{\left(\frac{P_{cr}}{A_2}\right)^2 + \frac{2 \cdot E \cdot P_{cr} \cdot h}{A_2 \cdot L}} = 1.0456 \cdot 10^8\ \text{Pa}$$

$$\sigma_{max} := \text{if } \sigma_1 > \sigma_2 \ \ \sigma_1 \ \ \text{else} \ \ \sigma_2 = 1.0456 \cdot 10^8\ \text{Pa}$$

$\sigma_{yield} := 4.6 \cdot 10^8\ \text{Pa}$

$\sigma_{max} < \sigma_{yield} = 1$ Therefore, the robot does not fail to impact

Conclusions:

These calculations determined the maximum stress induced by impact loading is 1.0456E8 Pa. The yield stress of Steel 4130 is 4.6E8Pa, which is greater than the maximum stress, thus the robot does not fail to impact.

References:

Impact Academy Official. “Stress Due to Impact Load | Strain Energy | Strength of Materials |,” YouTube, 15 November 2018. [Video file]. Available: https://www.youtube.com/watch?v=pS2-QAZ57_U&feature=emb_logo&ab_channel =ImpactAcademyOfficial. [Accessed : 1 March 2021]

AZoM, "AZO Materials," 12 November 2012. [Online]. Available: https://www.azom.com/article.aspx?ArticleID=6742. [Accessed 1 March 2021].

Appendix D10 : Bolt Analysis

Title: Bolt Calculations
Date: April 20, 2021
Author: Kenny Okeke

Modular Plate Bolt Torque Calculations

Objective:
To determine the preload torque applied to the bolts to prevent joint separation of the modular plate.

Variables: T - Torque [N*m]
k - Torque Coefficient
d - Diameter of Bolt Shank [m]
Fi = Preload [N]
S.ty = Yield Strength [Pa]
A.t = Tensile Stress Area [m^2]

Solution Method: Determine the Preload
Determine the Torque

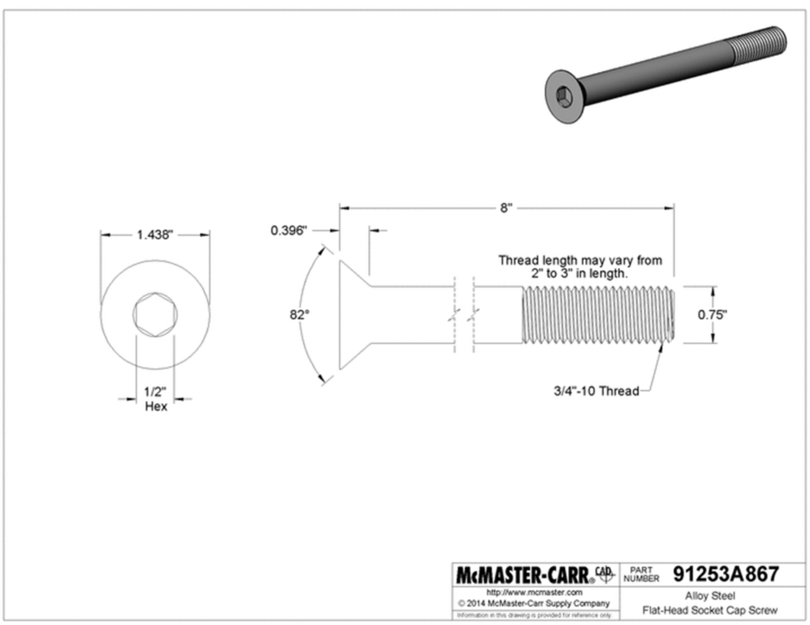
Equations: $F_{i.eq} := 0.64 \cdot S_{ty} \cdot A_t$

$T_{eq} := k \cdot d \cdot F_i$

Assumptions: The torque coefficent is assumed to be: $k := 0.2$
The preload used is for a nonpermanent joint (reused fastener)

The yield strength of bolt (from McMaster Carr): $S_{ty.MP} := 120000\ psi = 8.2737 \cdot 10^{8}\ Pa$

Sketch:



A. Diameters and Areas of Unified Screw Threads

Size designation	Nominal major diameter, in.	Coarse series—UNC			Fine series—UNF		
		Threads per inch, N	Tensile-stress area A_t, in.^{2}	Minor-diameter area A_r, in.^{2}	Threads per inch, N	Tensile-stress area A_t, in.^{2}	Minor-diameter area A_r, in.^{2}
0	0.0600				80	0.00180	0.00151
1	0.0730	64	0.00263	0.00218	72	0.00278	0.00237
2	0.0860	56	0.00370	0.00310	64	0.00394	0.00339
3	0.0990	48	0.00487	0.00406	56	0.00523	0.00451
4	0.1120	40	0.00604	0.00496	48	0.00661	0.00566
5	0.1250	40	0.00796	0.00672	44	0.00830	0.00716
6	0.1380	32	0.00909	0.00745	40	0.01015	0.00874
8	0.1640	32	0.0140	0.01196	36	0.01474	0.01285
10	0.1900	24	0.0175	0.01450	32	0.0200	0.0175
12	0.2160	24	0.0242	0.0206	28	0.0258	0.0226
1/4	0.2500	20	0.0318	0.0269	28	0.0364	0.0326
5/16	0.3125	18	0.0524	0.0454	24	0.0580	0.0524
3/8	0.3750	16	0.0775	0.0678	24	0.0878	0.0809
7/16	0.4375	14	0.1063	0.0933	20	0.1187	0.1090
1/2	0.5000	13	0.1419	0.1257	20	0.1599	0.1486
9/16	0.5625	12	0.182	0.162	18	0.203	0.189
5/8	0.6250	11	0.226	0.202	18	0.256	0.240
3/4	0.7500	10	0.334	0.302	16	0.373	0.351
7/8	0.8750	9	0.462	0.419	14	0.509	0.480
1	1.0000	8	0.606	0.551	12	0.663	0.625
1 1/4	1.2500	7	0.969	0.890	12	1.073	1.024
1 1/2	1.5000	6	1.405	1.294	12	1.315	1.260

Analysis:

The bolt properties are then determined based on the sketch:

$$d_{MP} := \frac{3}{4}\ \text{in} = 19.05\ \text{mm}$$

$$A_{t.MP} := 0.334\ \text{in}^2 = 215.4834\ \text{mm}^2$$

The preload can now be calculated:

$$F_{i.MP} := 0.64 \cdot S_{ty.MP} \cdot A_{t.MP} = 1.141 \cdot 10^5\ \text{N}$$

Finally, the torque preload is calculated:

$$T_{MP} := k \cdot d_{MP} \cdot F_{i.MP} = 434.7295\ \text{N m}$$

Conclusion:

Thus, a preload torque of 434.7295 N*m must be applied to each bolt in order to prevent joint separation of the modular plate.

References:

MechaniCalc. (2021, April 12). Bolted Joint Analysis. Retrieved from MechaniCalc: https://mechanicalc.com/reference/bolted-joint-analysis

McMaster-Carr. (2021, April 12). Black-Oxide Alloy Steel Hex Drive Flat Head Screw. Retrieved from McMaster-Carr: https://www.mcmaster.com/91253A867/

Fixed Shaft Bolt Torque Calculations

Objective:
To determine the preload torque applied to the bolts to prevent joint separation of the fixed shaft.

Variables: T - Torque [N*m]
k - Torque Coefficient
d - Diameter of Bolt Shank [m]
Fi = Preload [N]
S.ty = Yield Strength [Pa]
A.t = Tensile Stress Area [m^2]

Solution Method: Determine the Preload
Determine the Torque

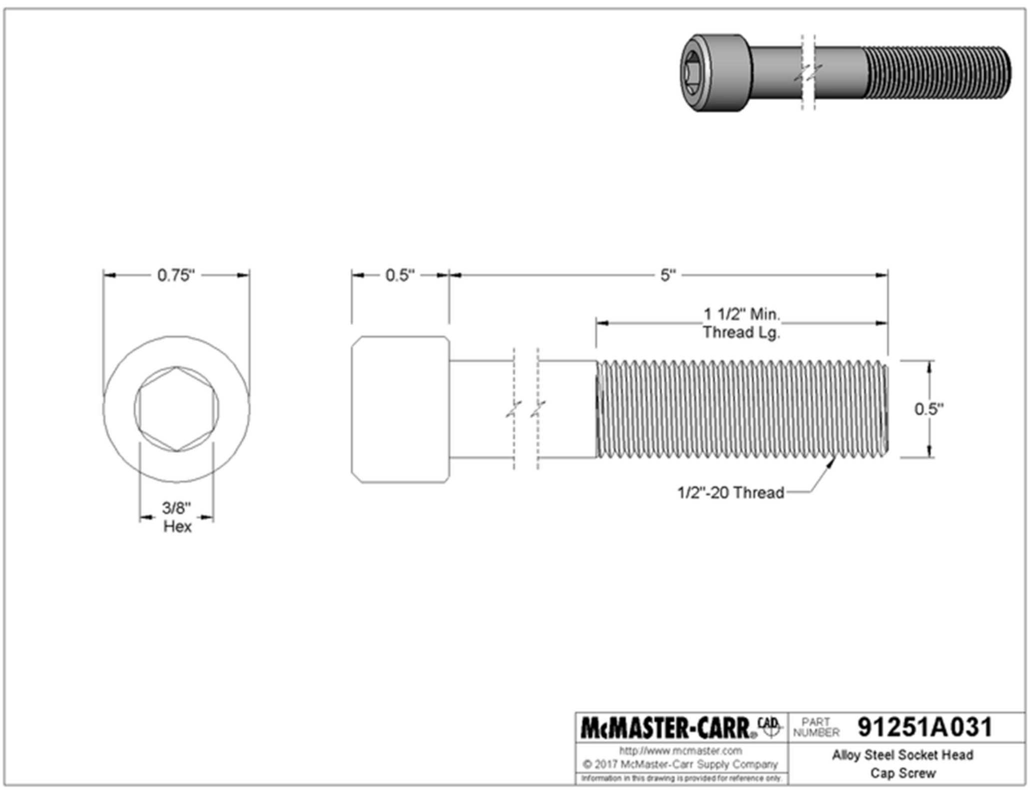
Equations: $F_{i.eq} := 0.64 \cdot S_{ty} \cdot A_t$

$T_{eq} := k \cdot d \cdot F_i$

Assumptions: The torque coefficent is assumed to be: $k := 0.2$
The preload used is for a nonpermanent joint (reused fastener)

The yield strength of bolt (from McMaster Carr): $S_{ty.FS} := 170000 \ \text{psi} = 1.1721 \cdot 10^{9} \ \text{Pa}$

Sketch:



A. Diameters and Areas of Unified Screw Threads

Size designation	Nominal major diameter, in.	Coarse series—UNC			Fine series—UNF		
		Threads per inch, N	Tensile-stress area A_t, in.²	Minor-diameter area A_r, in.²	Threads per inch, N	Tensile-stress area A_t, in.²	Minor-diameter area A_r, in.²
0	0.0600				80	0.00180	0.00151
1	0.0730	64	0.00263	0.00218	72	0.00278	0.00237
2	0.0860	56	0.00370	0.00310	64	0.00394	0.00339
3	0.0990	48	0.00487	0.00406	56	0.00523	0.00451
4	0.1120	40	0.00604	0.00496	48	0.00661	0.00566
5	0.1250	40	0.00796	0.00672	44	0.00830	0.00716
6	0.1380	32	0.00909	0.00745	40	0.01015	0.00874
8	0.1640	32	0.0140	0.01196	36	0.01474	0.01285
10	0.1900	24	0.0175	0.01450	32	0.0200	0.0175
12	0.2160	24	0.0242	0.0206	28	0.0258	0.0226
1/4	0.2500	20	0.0318	0.0269	28	0.0364	0.0326
5/16	0.3125	18	0.0524	0.0454	24	0.0580	0.0524
3/8	0.3750	16	0.0775	0.0678	24	0.0878	0.0809
7/16	0.4375	14	0.1063	0.0933	20	0.1187	0.1090
1/2	0.5000	13	0.1419	0.1257	20	0.1599	0.1486
9/16	0.5625	12	0.182	0.162	18	0.203	0.189
5/8	0.6250	11	0.226	0.202	18	0.256	0.240
3/4	0.7500	10	0.334	0.302	16	0.373	0.351
7/8	0.8750	9	0.462	0.419	14	0.509	0.480
1	1.0000	8	0.606	0.551	12	0.663	0.625
1 1/4	1.2500	7	0.969	0.890	12	1.073	1.024
1 1/2	1.5000	6	1.405	1.294	12	1.315	1.260

Analysis:

The bolt properties are then determined based on the sketch:

$$d_{FS} := \frac{1}{2}\ \text{in} = 12.7\ \text{mm}$$

$$A_{t.FS} := 0.1599\ \text{in}^2 = 103.1611\ \text{mm}^2$$

The preload can now be calculated:

$$F_{i.FS} := 0.64 \cdot S_{ty.FS} \cdot A_{t.FS} = 77386.2452\ \text{N}$$

Finally, the torque preload is calculated:

$$T_{FS} := k \cdot d_{FS} \cdot F_{i.FS} = 196.5611\ \text{N m}$$

Conclusion:

Thus, a preload torque of 196.5611 N*m must be applied to each bolt in order to prevent joint separation of the fixed shaft.

References:

MechaniCalc. (2021, April 12). Bolted Joint Analysis. Retrieved from MechaniCalc: https://mechanicalc.com/reference/bolted-joint-analysis

McMaster-Carr. (2021, April 12). Black-Oxide Alloy Steel Socket Head Screw. Retrieved from McMaster-Carr: https://www.mcmaster.com/91251A031/

Modular Plate Bolt Shear Stress Analysis

Objective:
To conduct a conservative study to determine if the modular plate bolt fails from shear.

Variables: d - Diameter of Bolt Shank [m]
A = Area [m^2]
S.ty = Yield Strength [Pa]
m = Mass of Robot [kg]
g = Gravity [m/s^2]
F = Force [N]
τ = Shear Stress [MPa]
FS = Factor of Safety

Solution Method: Determine the Area of the Bolt Shank
Determine the force
Determine the shear stress
Compare shear stress to yield strength of the bolt

Equations:

$$A_{eq} := \pi \cdot \left(\frac{d}{2}\right)^2$$

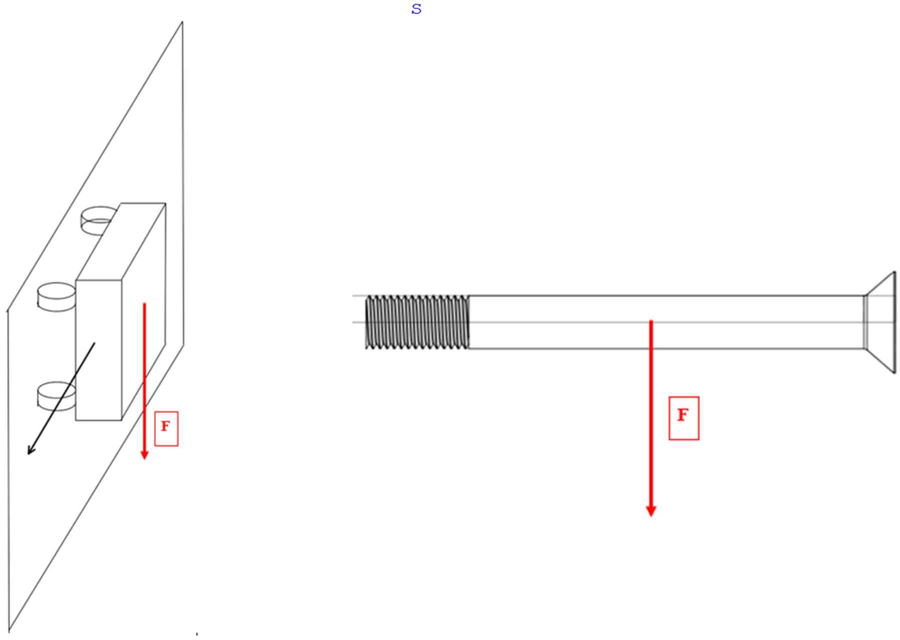
$$F_{eq} := m \cdot g$$

$$\tau_{eq} := \frac{F_{eq}}{A_{eq}}$$

$$FS_{eq} := \frac{S_{ty.eq}}{\tau_{eq}}$$

Assumptions: The worst case scenario experienced by the bolts is when the robot is travelling horizontal along the wall
The shear stress applied to the bolt is a conservative estimate, and therefore the force experienced by one bolt is the weight of the whole robot.
The yield strength of bolt (from McMaster Carr): $S_{ty.MP} = 8.2737 \cdot 10^8\ \mathrm{Pa}$
The mass of the robot is assumed to be: $m := 500\ \mathrm{kg}$
The acceleration due to gravity: $g := 9.81\ \frac{\mathrm{m}}{\mathrm{s}^2}$

Sketch:



Analysis:

The area of the bolt shank can be calculated:

$$A := \pi \cdot \left(\frac{d_{MP}}{2}\right)^2 = 0.0003\ m^2$$

The force experienced by the bolt shank is then calculated:

$$F := m \cdot g = 4905\ N$$

The shear stress experienced by the bolt shank is then calculated:

$$\tau := \frac{F}{A} = 1.7209 \cdot 10^7\ Pa$$

$$S_{ty.MP} = 8.2737 \cdot 10^8\ Pa$$

$$\tau < S_{ty.MP} = 1$$

The shear stress experienced by the bolt is less than

$$FS_{eq} := \frac{S_{ty.MP}}{\tau} = 48.0774$$

Conclusion:

This conservative estimate of the shear stress experienced by the modular plate bolts resulted in a factor of safety of 48. Due to this high factor of safety value achieved, no more further analysis is required to determine if the bolts of the robot fail. In reality, the shear stress experienced by the modular plate bolts will be much less than the conservative estimate used in this analysis.

References:

McMaster-Carr. (2021, April 12). Black-Oxide Alloy Steel Hex Drive Flat Head Screw. Retrieved from McMaster-Carr: https://www.mcmaster.com/91253A867/

Appendix D11 : Winch Analysis

Winch Connection Calculations

Title - Tensile & Shear Stress on the Winch Connection
Date - March 15, 2021
Author - Liam Wolf & George Felobes

Objective

Determine the average tensile, average shear and maximum shear stress in the winch connection to make sure it does not fail under shear or tension. The pull strength of the threads used to connect the connection point to the robot will also be analysed. This will confirm the selection of a hoist ring to connect the vehicle to the winch.

Variables

m_t = Total mass of the system

D = Diameter of load bearing portion of hoist ring. Shown in diagram

b = effective width

I_c= Moment of inertia around the centriod

g = Gravitational acceleration of earth, equal to 9.81 m/s^2

FOS = Factor of safety

V_avg = Average shear force calculated

A = Area that the V_avg acts upon on hoist ring

σ_avg = Maximum shear stress calculated through load bearing portion of hoist ring

σ_T_failure = Yield shear stress found from material properties

τ_max = Maximum shear stress calculated through load bearing portion of hoist ring

τ_failure = Yield shear stress found from material properties

n_t = Number of threads per inch on winch connection

D_t = Major diameter of threaded winch connection

A_t = Effective tensile strength area though threaded connection

σ_avg_t = Maximum shear stress calculated through thread

Solution Method

Determine the estimated tension on the winch connection to be the extreme value when it is carrying the whole weight of the robot.

Determine the effective area that the force is transmitted through

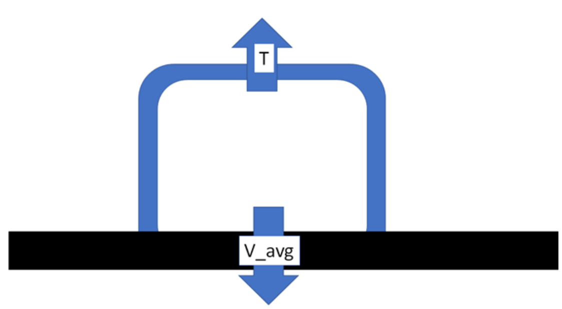
Calculate the maximum shear stress as the force divide by the effective area

The maximum shear stress is calculated using the relations shown in the analysis section

Employ a factor of safety to be used

This will guide the selection of a commercially available connection to avoid failure. If all materials cannot meet the failure criteria requirements, the geometry would be modified to reduce the shear stresses.

Sketches



Knowns

Winch connection dimensions based on preliminary selection of the smallest available hoist ring which capable of supporting the robot mass of 500 kg. A hoist ring able to lift 1134 kg (2500 lbs) vertically or 756 kg (1667 lbs) at a lifting angle of 45 degrees (to account for potential swaying of the robot if detached from vessel) was selected; a cross-sectional diameter and thread properties can be assumed for this calculation as shown below.

Source for preliminary hoist ring selection: https://www.mcmaster.com/2994T91/

$m\_t := 500\ \text{kg}$ $g := 9.81\ \frac{\text{m}}{\text{s}^2}$ $D := 0.75\ \text{in}$ $FOS := 2$ $n\_t := 13\ \frac{\text{rad}}{\text{in}}$ $D\_t := 0.5\ \text{in}$

Assumptions

The system will be analyzed when the robot is placed on a vertical surface such that the weight vector is perpendicular to the ground. The winch is assumed to carry the whole weight of the robot. This is an extreme cases to test the limits of a winch connection

Assume a constant thickness perfectly circular hoist ring cross section

Assume tensile and shear force distribution is uniform along the effective area

Connection is symmetric, with symmetric force distribution

The winch connection is assumed to be made out of stainless steel

Analysis

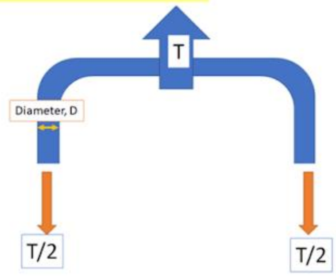
$r := \frac{D}{2} = 0.0095\ \text{m}$

$V_{avg} := m\_t \cdot g = 4905\ \text{N}$ Determine weight of the platform from the mass

$A := \left(r^2\right) \cdot \pi = 0.0003\ \text{m}^2$

Tension of the winch is equal to the weight of the whole platform (assumed conservativly)

Tensile Stress Calculation



Thus the maximum force used for the tension calculation is half the weight of the robot

$$\sigma_{avg} := \frac{\frac{V_{avg}}{2} \cdot FOS}{A} = 17.2091\ \text{MPa}$$

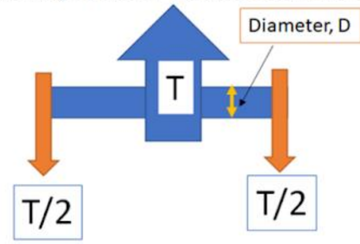
Looking at tensile strength of Stainless Steel:
Source for tensile strength: https://www.thomasnet.com/articles/metals-metal-products/all-about-304-steel-properties-strength-and-uses/

$\sigma_{T\_failure} := 215\ \text{MPa}$

```
if σavg < σT_failure                    = "No failure under tension"
  "No failure under tension"
else
  "Winch connection fails under tension"
```

Shear Stress Calculation

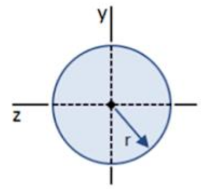
Following the approach from the following source in order to determine the maximum shear force in a hollow shaft:



Thus the maximum force used for the shear calculation is half the weight of the robot

Shear Stresses in Circular Sections

A circular cross section is shown in the figure below:



The equations for shear stress in a beam were derived using the assumption that the shear stress along the width of the beam is constant. This assumption is valid at the centroid of a circular cross section, although it is not valid anywhere else. Therefore, while the distribution of shear stress along the height of the cross section cannot be readily determined, the maximum shear stress in the section (occuring at the centroid) can still be calculated. The maximum value of first moment, Q, occurring at the centroid, is given by:

$$Q_{max} = \frac{2r^3}{3}$$

The maximum shear stress is then calculated by:

$$\tau_{max} = \frac{VQ_{max}}{I_c b} = \frac{4V}{3A}$$

where $b = 2r$ is the diameter (width) of the cross section, $I_c = \pi r^4/4$ is the centroidal moment of inertia, and $A = \pi r^2$ is the area of the cross section.

Source to determine the shear stress in a hollow shaft: https://mechanicalc.com/reference/beam-analysis

$$I_c := \frac{(r^4)\cdot \pi}{4} = 6.4647 \cdot 10^{-9}\ \mathrm{m}^4$$

$$b := (r)\cdot 2 = 0.019\ \mathrm{m}$$

$$Q_{max} := \frac{2\cdot (r^3)}{3} = 5.7611 \cdot 10^{-7}\ \mathrm{m}^3$$

$$\tau_{max} := \frac{\frac{V_{avg}}{2}\cdot Q_{max}}{I_c \cdot b}\cdot FOS = 22.9455\ \mathrm{MPa}$$

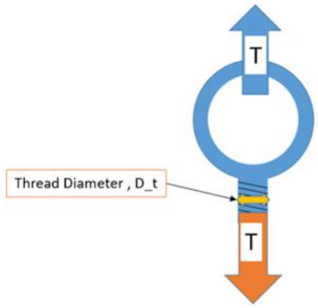
Looking at shear strength of Stainless Steel:

Source for shear strength: http://www.matweb.com/search/datasheet_print.aspx?matguid =71396e57ff5940b791ece120e4d563e0

$$\tau_{failure} := 74.5\ \mathrm{MPa}$$

```
if τmax < τfailure                    = "No failure under shear"
  "No failure under shear"
else
  "Wheel axle fails under shear"
```

Thread Strength Calculation



Thus the maximum force used for the thread strength calculation is the entire weight of the robot

Based on the following source, the pull strength of the threaded connection between the hoist ring and the robot plate can be calculated.

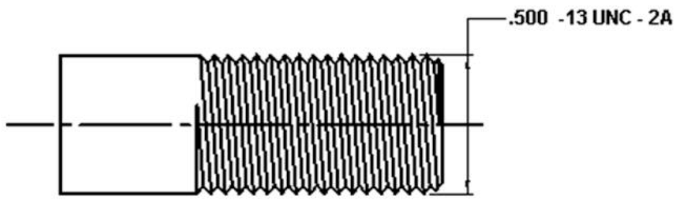
https://www.engineersedge.com/thread_stress_area_a.htm

The following are definitions for the elements:

A_t = tensile strength area of screw thread.

D = *Basic major diameter*, example; .500 for a 1/2 - 13 UNC - 2B Thread Engineering Data Chart

n = Number of threads per inch



Equation:

$$A_t = 0.7854\left(D - \frac{0.9743}{n}\right)^2$$

This equation provides an approximate result by extrapolation on the thread stress area of a fastener.This equation is adequate for design applications on engineering materials or less than 100 ksi ultimate strength.

$$A\_t := 0.7854 \cdot \left(D\_t - \frac{0.9743}{n\_t}\right)^2 = 9.155 \cdot 10^{-5}\ \mathrm{m}^2$$

$$\sigma_{avg\_t} := \frac{V_{avg} \cdot FOS}{A\_t} = 107.1576\ \mathrm{MPa}$$

Looking at tensile strength of Stainless Steel:

Source for tensile strength: https://www.thomasnet.com/articles/metals-metal-products/all-about-304-steel-properties-strength-and-uses/

$$\sigma_{T\_failure} := 215\ \mathrm{MPa}$$

```
if σavg_t < σT_failure                    = "No thread failure under tension"
  "No thread failure under tension"
else
  "Thread fails under tension"
```

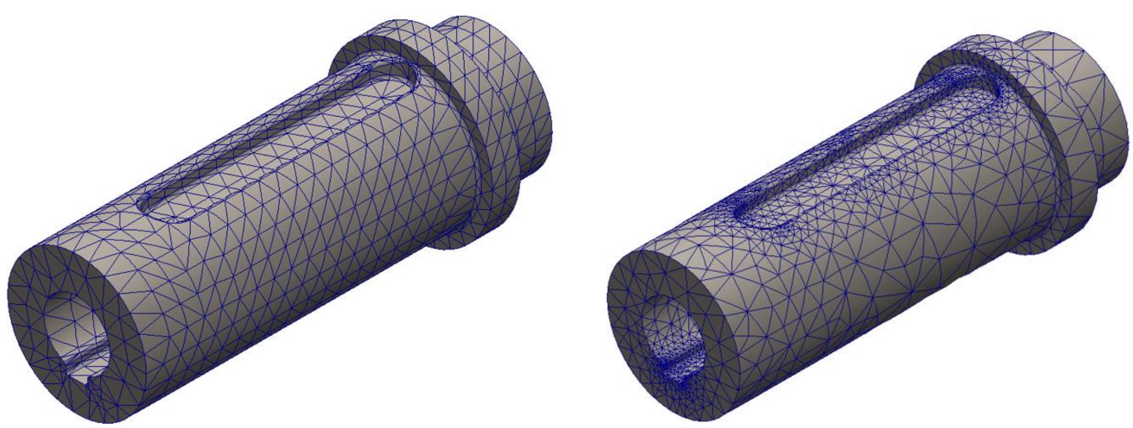
Appendix E: Finite Element Analysis (FEA)

This appendix details the process and results of the FEA analysis carried out on the critical components of this design.

The worst-case loading scenarios for each of the critical components were determined analytically as per the calculations in Appendix D. From these forces, FEA simulations were carried out to determine the maximum stress experienced by each component. The minimum factors of safety in each part were determined to ensure they were greater than 1 and would not fail under their worst-case loading conditions.

Appendix E1: Mesh Convergence

The models in the following FEA studies utilized an h-adaptive mesh to achieve optimal mesh convergence while prioritizing computational power and time. The h-adaptive meshing procedure involves running multiple iterations of a simulation and varying the refinement level or density of the mesh between each iteration. This is done automatically by SolidWorks in areas where stress is the highest or lowest without the need for user input. In areas of high stress, the mesh will be refined to achieve a high level of accuracy and achieve convergence, while in areas of low stress the mesh is loosened/coarsened to reduce the computational power and time required. An example of an h-adaptive mesh after 5 iterations versus a standard mesh for a preliminary model of the drive shaft is shown below:



a) *b)*

Figure 21 - Standard Solidworks Mesh (a) vs. h-adaptive Mesh (b)

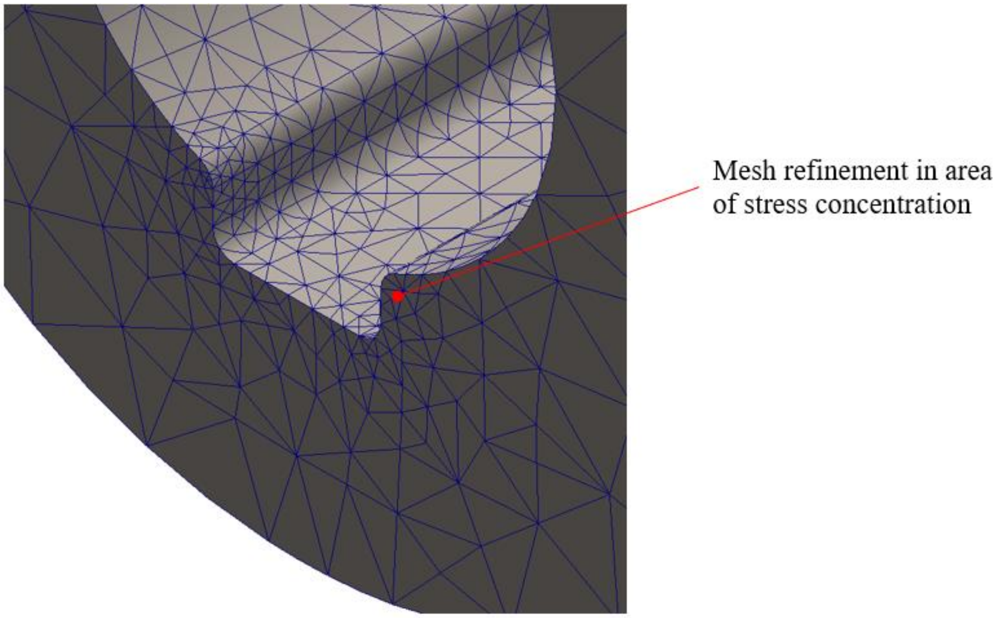


Figure 22 - Mesh Refinement in Area of Stress Concentration After 5 Iterations of h-adaptive Meshing

To continue, mesh convergence can be achieved via the mesh refinement and coarsening of the h-adaptive meshing technique. Convergence of the maximum Von-Mises stress calculated within this example of the driveshaft using the h-adaptive model is shown below:

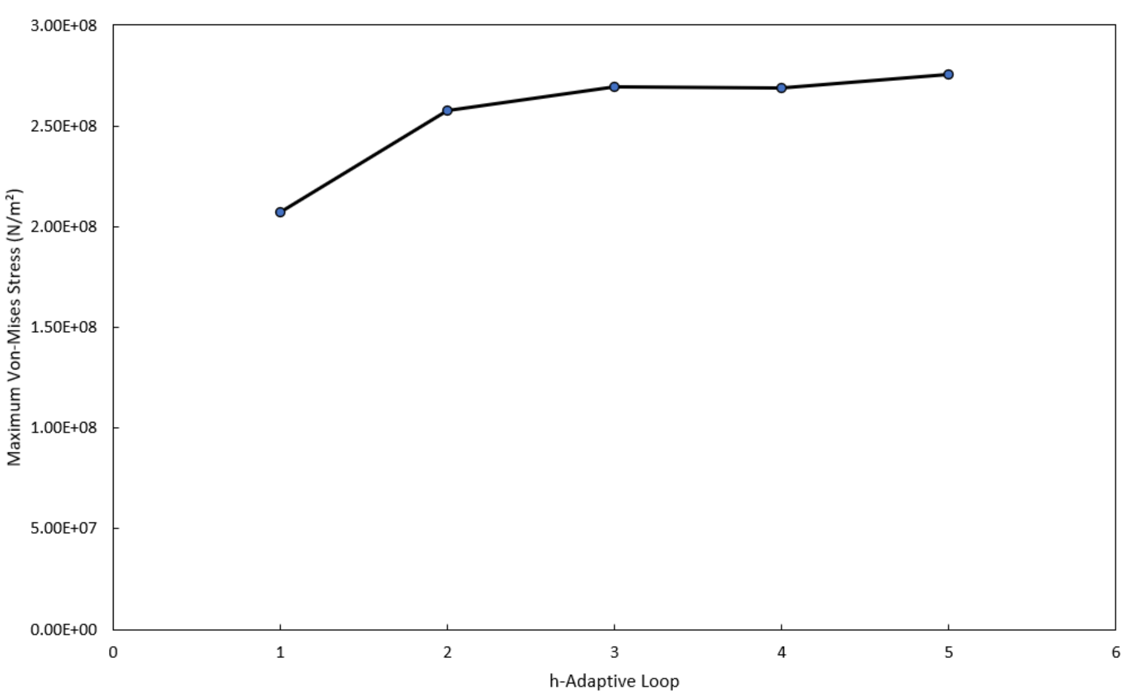


Figure 23 - Maximum Von-Mises Stress for Each h-adaptive Loop

This example demonstrates how the h-adaptive method was proven to be capable of achieving mesh convergence in the driveshaft, which is the individual component with the most complex geometry. Therefore, the h-adaptive meshing technique was used in all the following FEA studies to achieve the most accurate possible meshing possible given the available time and computational power. For any simulations which were still unable to converge due to limitations in computational power, conservative analytical calculations assuming simplified geometry were carried out to confirm the simulation results as acceptable.

Appendix E2: Gearhead Housing

Author: George Felobes
Date: April 6th, 2021

Objective:
To make sure the gearhead housing does not fail under the most conservative loading conditions. The gearhead housing should be able to resist the torque provided by the gearhead, and also resist the axial load due to the weight of the motor and gearhead.

Assumptions and model set-up:
Some of the assumptions made in analysis include the following:

1. Material: AISI 4130 Steel, normalized at 870C
2. Model type: Linear Elastic Isotropic
3. Yield strength:
4. 4.6e+08 N/m^2
5. Tensile strength: 7.31e+08 N/m^2
6. Elastic modulus: 2.05e+11 N/m^2
7. Poisson's ratio: 0.285
8. Mass density: 7,850 kg/m^3
9. Shear modulus: 8e+10 N/m^2
10. The weight of the drive system is equally distributed on the gearhead housing when in a horizontal position on a wall.
11. The worst-case loading scenario on the gearhead housing occurs when the robot is travelling horizontally on a wall, and the entire weight of the drive system is acting on the gearhead housing. This will be represented as a distributed load equivalent to the weight of the drive system which was conservatively used to be 150 N.
12. The gear head housing is connected to the v-support, which will be represented as a fixed support
13. The torque provided by the gearhead has been simulated to be distributed uniformly on the inner edge of the gearhead housing. The torque value is conservatively used to be 180 Nm. The torque acts along a virtual axis set in the centre of where the gearhead is located.

Figure 24 shows the loading and boundary conditions of the gearhead housing. Figure 25 shows the meshing used before solving the model using the Static model in SolidWorks.

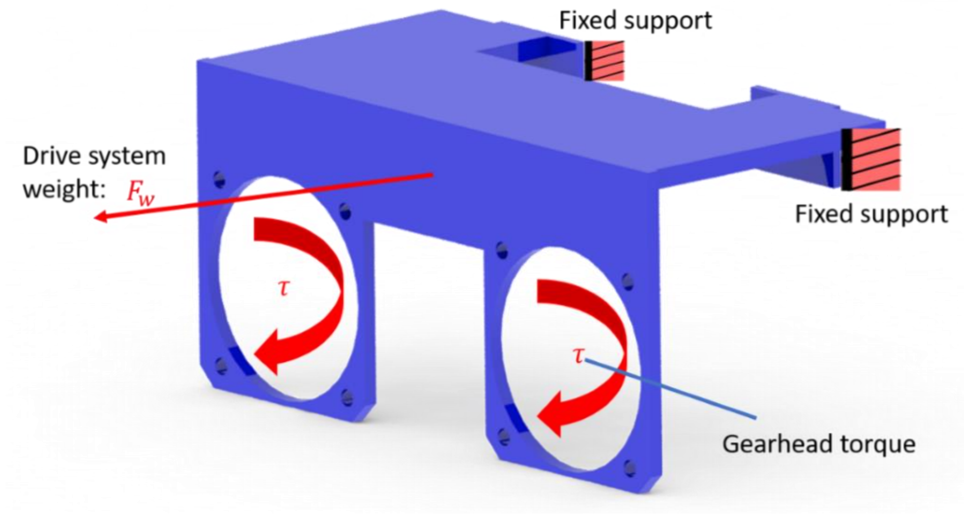


Figure 24 – Loading Conditions of the Gearhead Housing

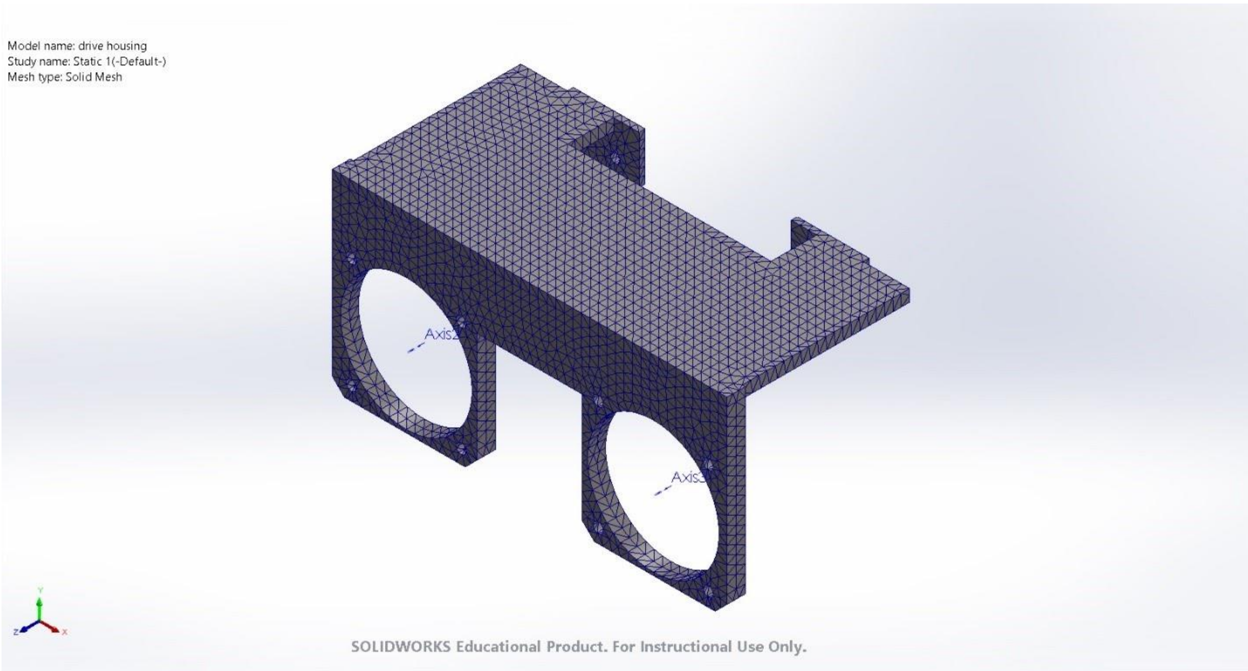


Figure 25 - Meshing used for the Gearhead Housing Model.

Results:
The maximum von misses stress is 270 MPA, which is located at the fixed support, where the v-support would be connected. This will be used as a failure criterion and compared to the yield stress which is 460 MPa. The factor of safety (FOS) of the design is 1.7. This is an acceptable FOS for the conservative loading used on this part. The results are shown in Figure 26 and Figure 27.

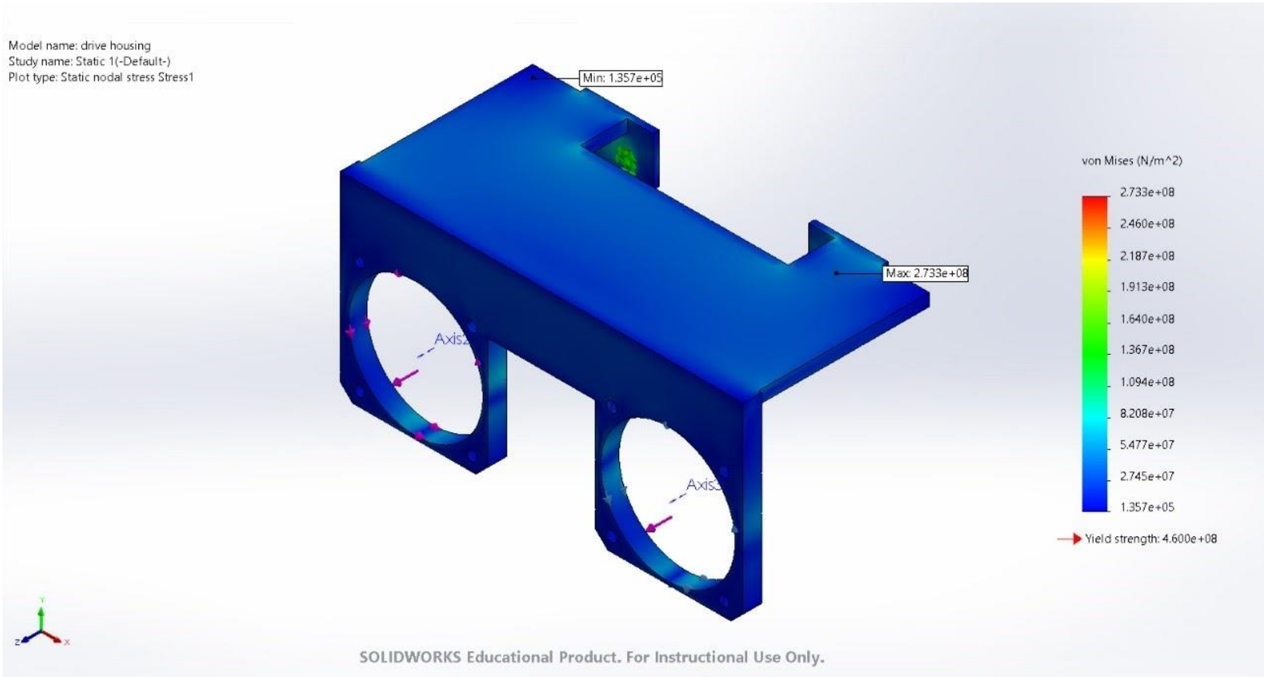


Figure 26 - Stress Concentration Along the Gearhead Housing

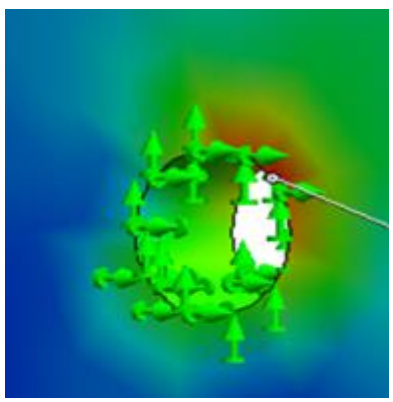


Figure 27 - Closeup of Stress Concentration Around the Gearhead Housing Connection to the V-Support at the Fixture Location

Analytical Validation:

Due to the complex nature of this loading scenario, no analytical validation via hand calculations was carried out for this component. Instead, the loading conditions used in the FEA model were based on the most conservative loading scenarios.

Conclusion:

The study suggest that the gearhead housing will not fail under conservative loading conditions and would satisfy its role in holding the drive system axially and prevent the gearhead from spinning around its axis.

Appendix E3: Top Plate

Author: Areej Khaddaj

Date: April 2, 2021

Objective:

To determine the highest stress induced in the top plate due to the loading of the manipulator and payload. Four simulations were run under the following conditions:

1. Static loading in the horizontal
2. Static loading in the vertical
3. Dynamic loading in the horizontal
4. Dynamic loading in the vertical

It was found that the highest stress occurs when the plate is subjected to dynamic loading conditions while it is oriented in the vertical position (when the robot is traversing a vertical wall). For brevity, only the simulation resulting in highest stress is presented.

Assumptions:

9. The material used is 4130 steel, normalized at 870 C
10. Linear elastic isotropic properties.
11. Yield strength:
12. 4.6e+08 N/m^2
12. Tensile strength: 7.31e+08 N/m^2
13. Elastic modulus: 2.05e+11 N/m^2
14. Poisson's ratio: 0.285
15. Mass density: 7,850 kg/m^3
16. Shear modulus: 8e+10 N/m^2
17. Contact area with the body does not change with time.
18. Stresses induced by torque preloads from bolts in the plate are negligible.
19. Loads from the manipulator are distributed equally between the two halves of the plate.
20. The modular halves of the plate are modelled together as one body.
21. The plate is supported by the fixed shafts of the drive system, with no loading transferred to the chassis.

Simulation Set-up Conditions & Procedure:

Figure 28 below shows the fixtures on the plate where the fixed shafts are connected. These are modelled as areas of fixed geometry on the plate, identified by the green clusters of arrows. Dynamic loads resulting from the motion of the manipulator are applied to the plate as remote forces and moments, acting a distance in the x-coordinate equal to the height of the manipulator base from the plate. They are symmetrically positioned in the middle of the plate. In SolidWorks, these remote loads are rigidly connected to the plate, shown by the lines connecting the loads to the plate. The loads used are the same ones previously calculated in the derivation of dynamic loads in a vertical orientation:

Fx = 54 N
Fy = 1515 N
Mz = 39 Nm

In Figure 29, the blue areas on the plate are the contact faces for the loads. These are the areas where the bottom of the manipulator base is in contact with the plate.

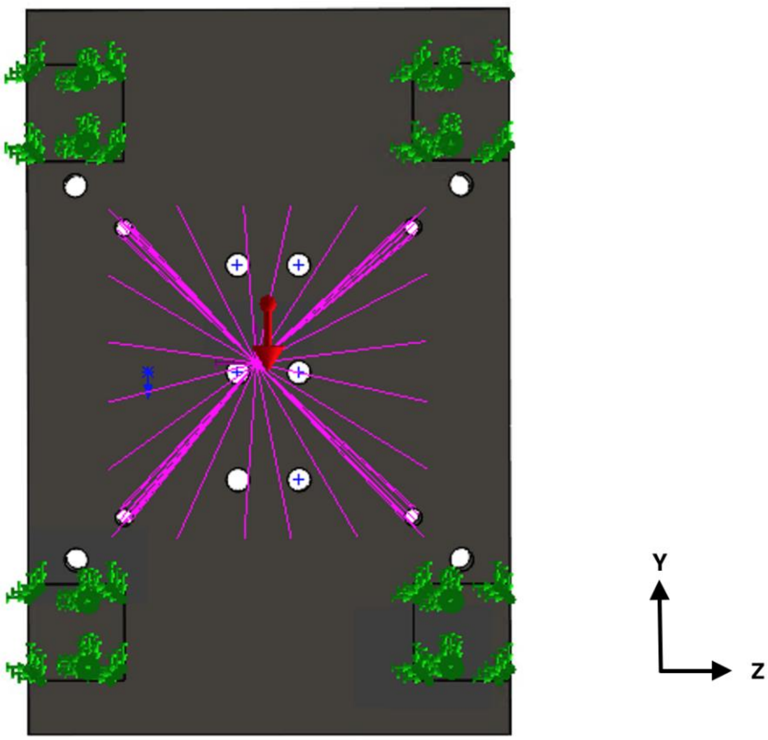


Figure 28 - Fixtures on the Plate, Bottom View

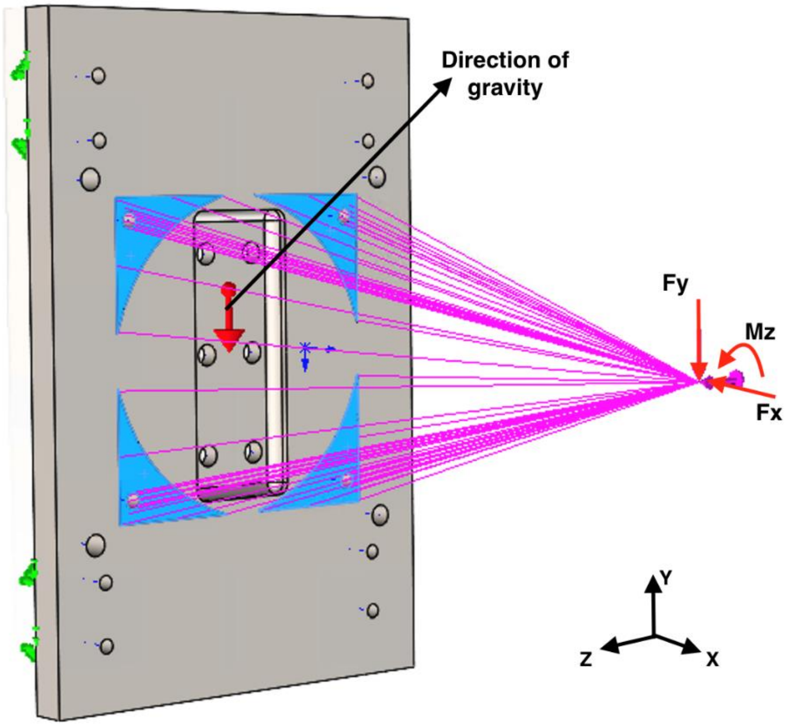


Figure 29 - Dynamic Loads Applied to the Plate from a Remote Distance.

Results:

The simulation was run and the deformed state with the maximum and minimum induced stress is shown in Figure #. Clearly, the yield stress of 460 MPa is not exceeded. The maximum stress is 6.89 MPa resulting in a minimum safety factor of 67.

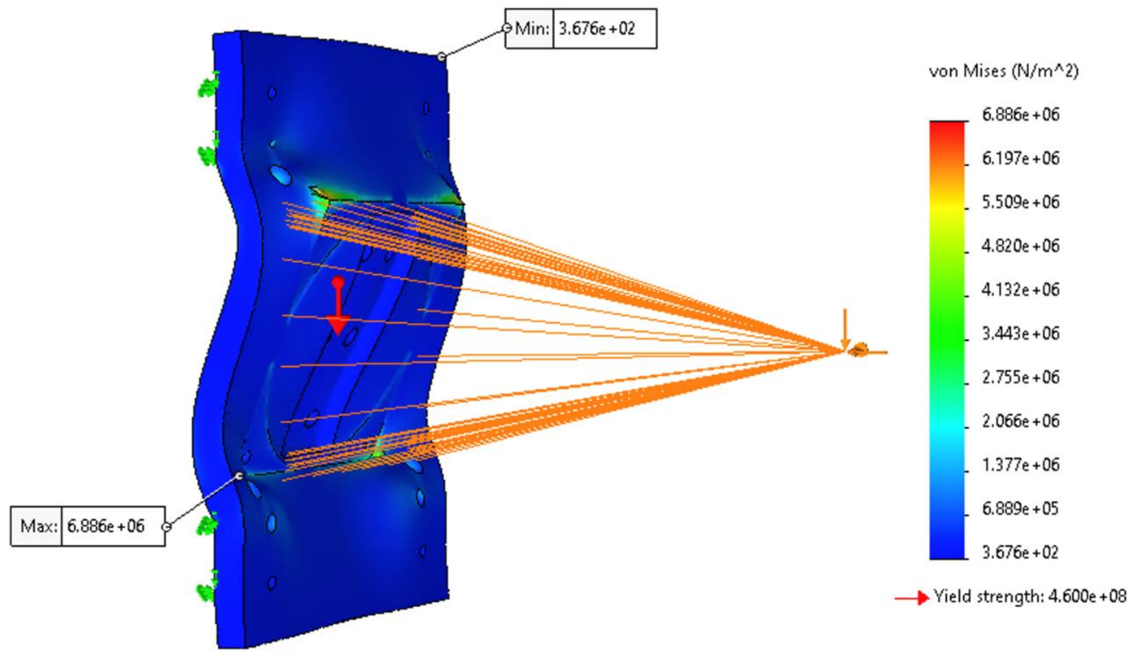


Figure 30 – Top Plate Max Stress Result Plot

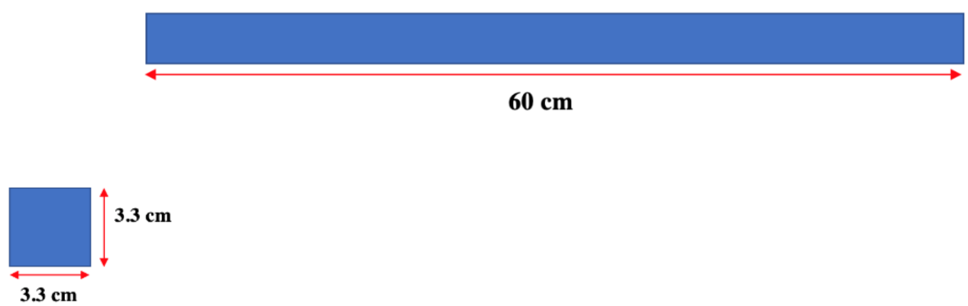
Analytical Validation:

Due to limitations in computing processing memory, convergence using h-adaptive meshing could not be achieved. Therefore, a conservative analytical calculation will be used to ensure the plate does not fail under the load of the manipulator and payload.

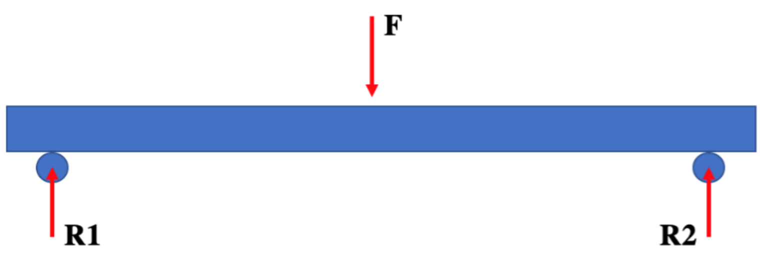
Assumptions:

- As a highly conservative model, the plate will be modelled as a square shaft with the thickness and length of the top plate.
- The shaft will have a thickness of 3.3 cm and length of 60 cm.
- The material is 4130 steel with linear isotropic properties.
- The shaft is simply supported at the ends, and a concentrated load acting in the middle

Model dimensions:



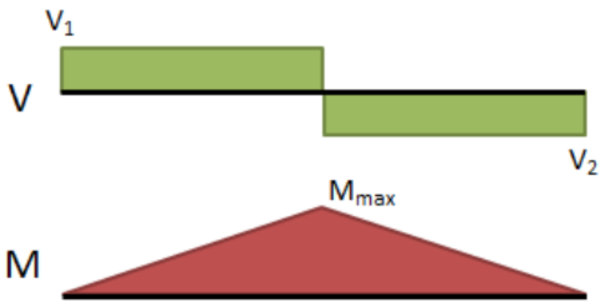
FBD:



F = 140 kg x 9.81 m//s^2 = 1373 N

R1 = R2 = 0.5 x 1373 = 687 N

For this simple loading condition, the shear and bending moment diagrams are given by:



Where V1=R1 and V2=R2

The maximum shear stress occurs at the neutral axis of the cross-sectional area, is calculated as:

$$\tau_{max} = \frac{3V}{2A} = \frac{3 * 1373\ N}{2 * (0.033\ m)^2} = 1.89\ MPa$$

The maximum bending moment is given by:

$$M_{max} = \frac{FL}{4} = \frac{1373\ N * 0.6m}{4} = 205.95\ Nm$$

The maximum bending stress that occurs:

$$\sigma_{max} = \frac{Mc}{I_c} = \frac{205.95\ Nm * 0.0165\ m}{\frac{(0.033m)^4}{12}} = 34.4\ MPa$$

The yield strength of 4130 is 860 MPa, therefore the minimum safety factor achieved in this conservative calculation is 25.

Reference for formulas used: https://mechanicalc.com/reference/beam-analysis

Conclusion:

The FEA results can be used to verify that under a dynamic loading condition in the vertical orientation, the top plate will not fail, with a safety factor of 67. Analytical calculations of a simplified highly conservative model of the top plate also results in a high safety factor of 25.

The high safety factor can be attributed to the distributed loading condition on the top plate – the manipulator with the payload does not contact the plate in a small, concentrated area. Additionally, although the plate thickness of 33 mm can be reduced, it was not optimized for this study. For mounting purposes specified by the manipulator manufacturer, the plate on which the manipulator sits must be at least 33 mm for the bolted connections.

Appendix E4: Drive Shaft

Author: Calvin Chen
Date: 4/1/2021

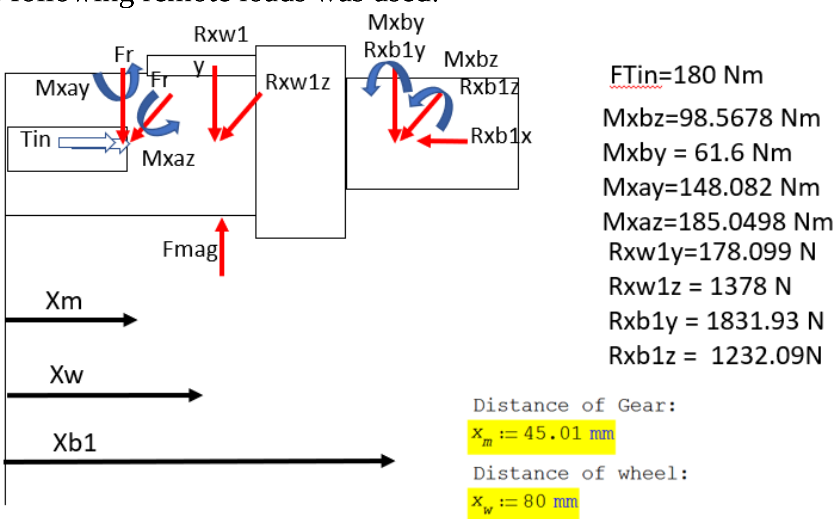
Objective:
Validate analytic drive shaft calculation and determine FEA FOS and maximum stress.

Assumptions:

- The material of the shaft is isotropic and linear elastic
- The material is 4130 Normalized Steel
- The most conservative loading condition will be used to determine it will fail.

Simulation Set-up Conditions & Procedure:
Figure 31 demonstrates the loading conditions applied to the model in the most conservative loading condition, which is axially loaded travelling on a vertical wall. The boundary conditions were determined from the V-Support Calculation and are presented in the Figure. A combined loading of the forces in the X, Y, and Z plane are loaded onto the shaft via remote loading conditions. The wheel shaft keyway is set as the fixture, where the wheel will be torqued. Prior to running the simulation, the H-adaptive method was utilized for a mesh convergence test. The H-adaptive resulted in a 95% accuracy rating, thus the simulation can be conducted.

The following remote loads was used:



Distance of Gear:
$x_m := 45.01$ mm

$F_{mag} := 1800$ N

Distance of wheel:
$x_w := 80$ mm

Distance of bearing 1:
$x_{b1} := 130$ mm

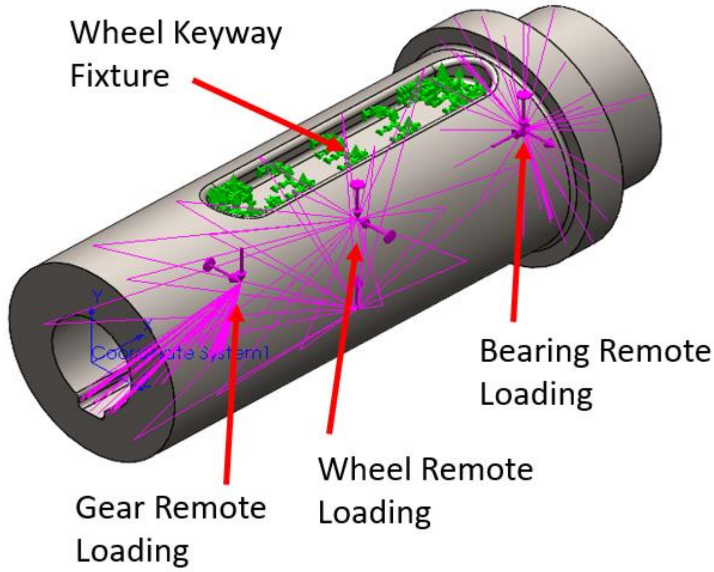


Figure 31 – Simulation Set-Up Conditions for the Fixed Shaft

Results:
From the results of the simulations shown in Figure 32,Figure 33, and Figure 34. It is clear, under the most conservative loading condition, the shaft does not fail, where the maximum stress is 1.26 MPa, located on the gear head keyway, and the lowest safety factor is 3.66.

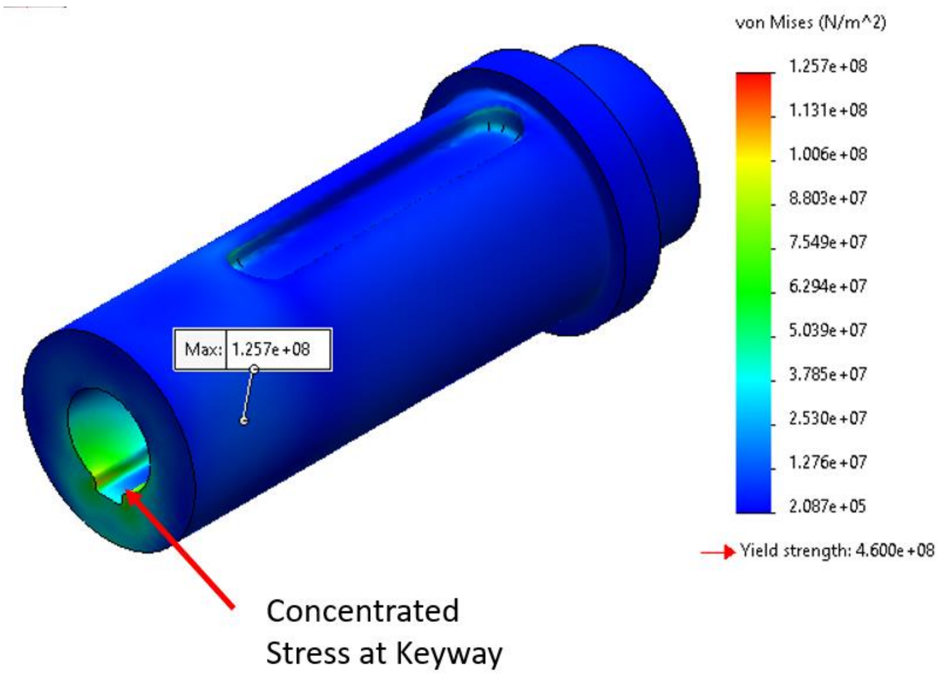


Figure 32 – Drive Shaft FEA Max Stress Result Plot Isometric View

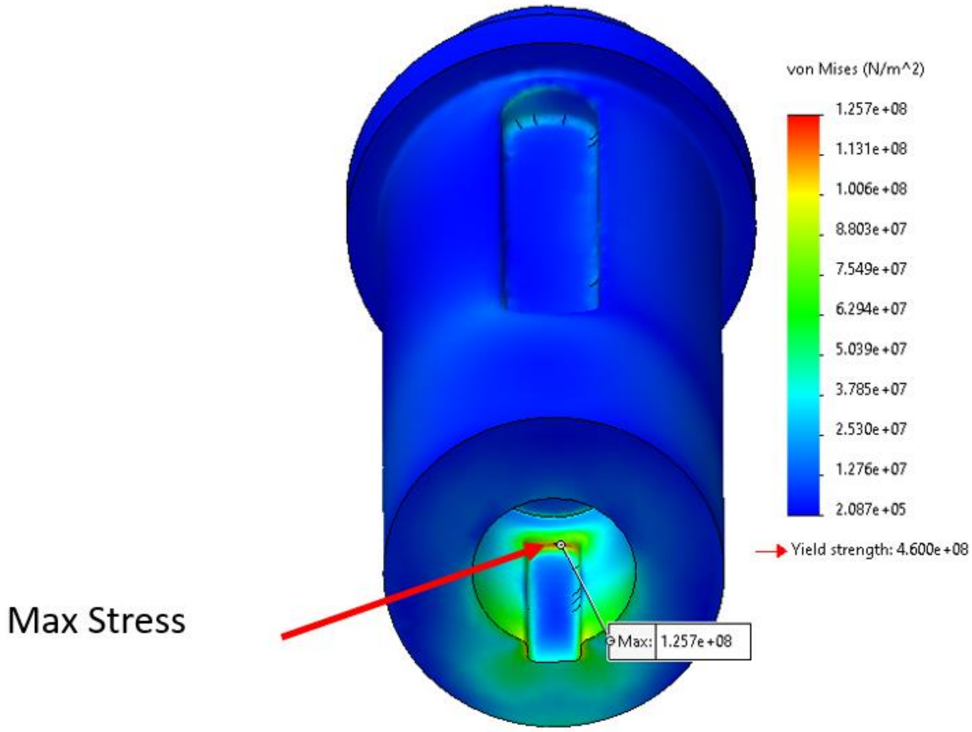


Figure 33 – Drive Shaft FEA Max Stress Result Plot Keyway View

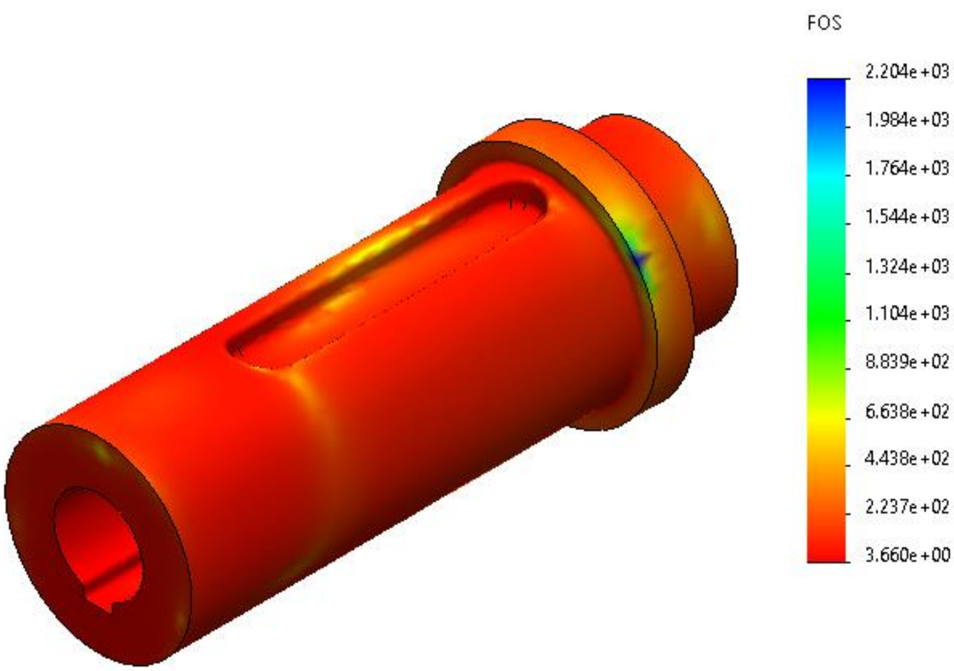


Figure 34 – Drive Shaft FEA FOS Result Plot

Analytical Validation:

The hand calculations conducted in Appendix C: Drive shaft calculated a factor of safety of 6.6. This is over 2x greater than the analytic result. This may be due to the empirical tables utilized for stress concentration values in the hand calculations. As the simulation simulated the face forces acting on the shaft as in the hand calculations, the FEA results will be utilized to be the more conservative choice.

Conclusion:

The FEA results provide to be 2x less than the analytic result when the same forces were utilized in both cases. Since the FEA resulted in a much lower FOS, the FEA results will be utilized. In both cases, the shaft does not fail.

Appendix E5: Fixed Shaft

Author: Calvin Chen
Date: 4/1/2021

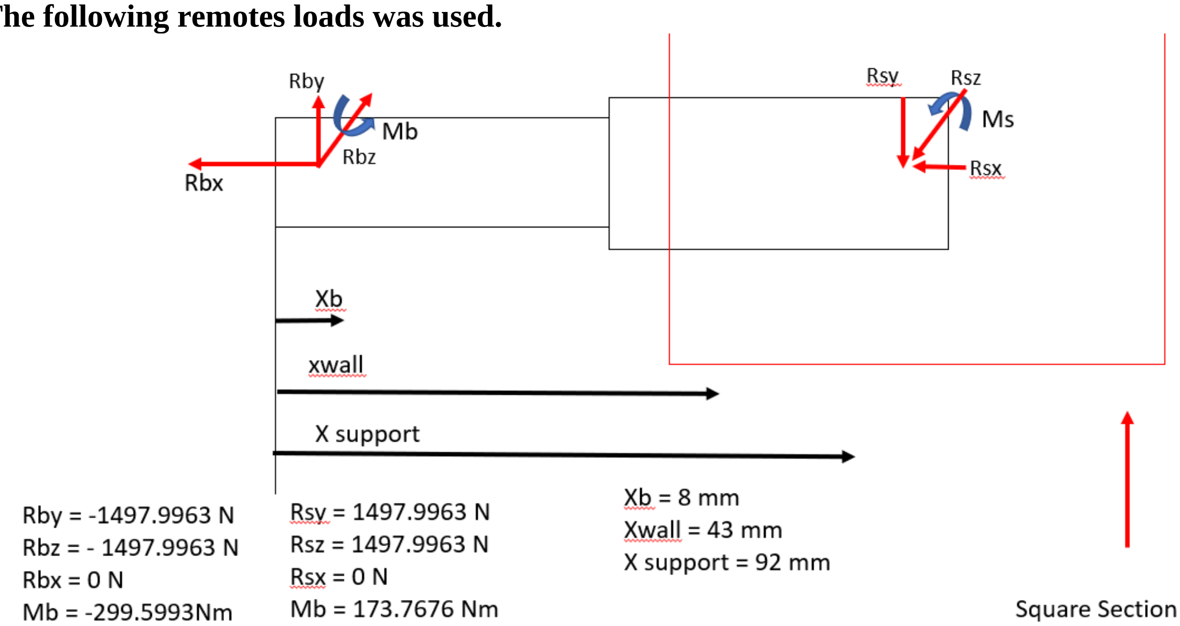
Objective:
Validate analytic fixed shaft calculation and determine FEA FOS and maximum stress. H-adaptive mesh is utilized to determine the proper mesh

Assumptions:
- The material of the shaft is isotropic and linear elastic
- The material is 4130 Normalized Steel
- The most conservative loading condition will be used to determine it will fail. The most conservative loading condition is on a horizontal surface.

Simulation Set-up Conditions & Procedure:

Figure 35demonstrates the loading conditions applied to the model. The loading condition on the bearing was determined from the stability calculations, the shaft is bearing the entire weight of the robot. The remote loads are applied to the round, bearing shaft and a fixture is applied to the top square portion. This is due to the shaft being bolted to the top plate. The most conservative case seen was the horizontal configuration. Like the drive shaft FEA, H-adaptive mesh convergence was used, and the accuracy proved to be the same at 95%.

The following remotes loads was used.



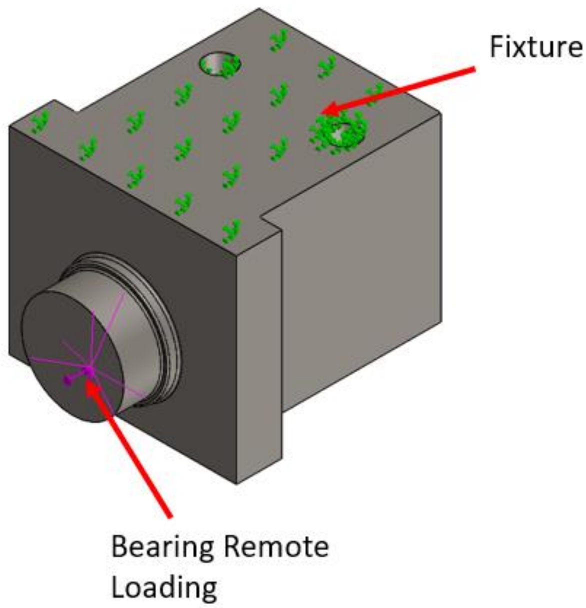


Figure 35 – Simulation Set-Up for Fixed Shaft

Results:

The stress and FOS results of the simulation are presented in Figure 36 and Figure 37. The maximum stress determined is 0.801 MPa and is located on the step shaft of the bearing portion. This corresponds to the lowest factor of safety rating, which is 5.74.

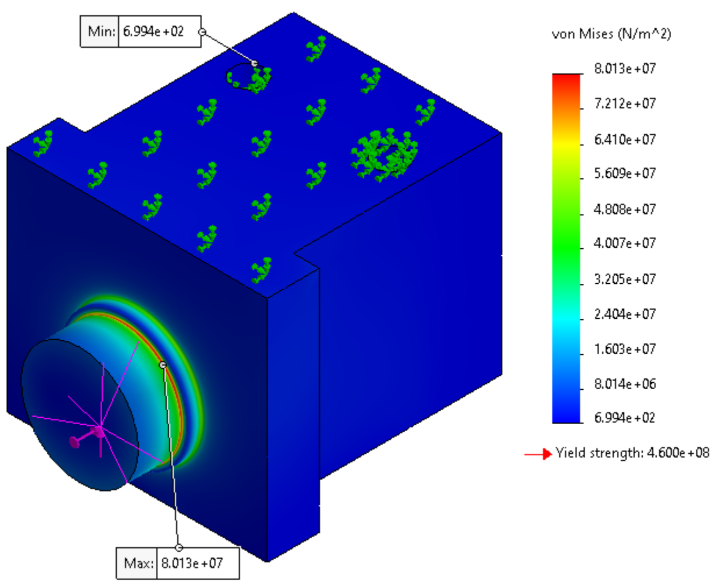


Figure 36 – Fixed Shaft FEA Max Stress Result Plot

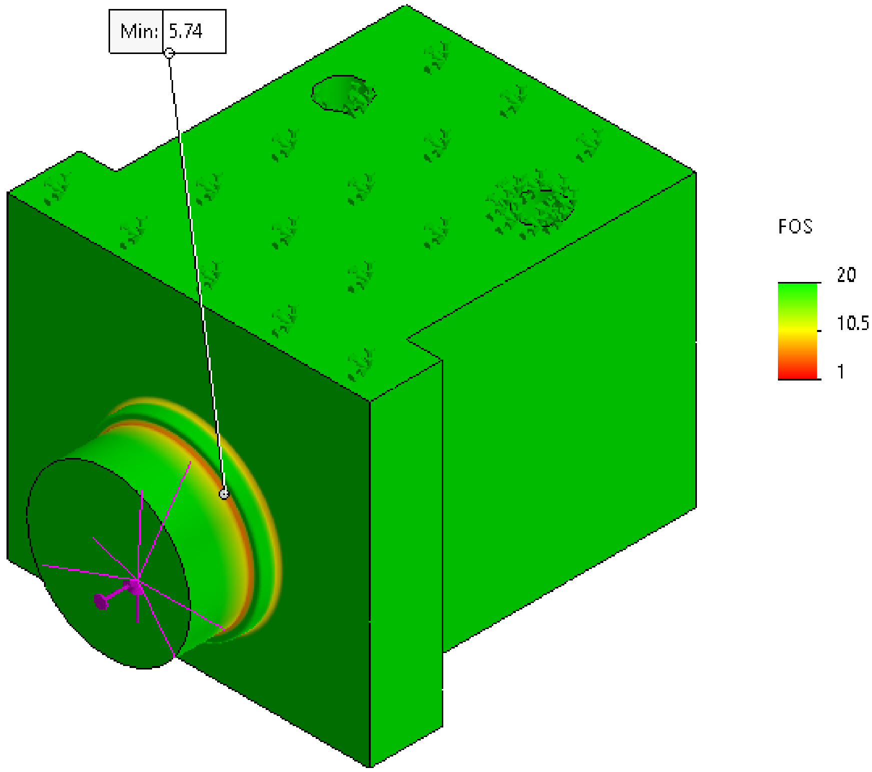


Figure 37 – Fixed Shaft FEA FOS Plot

Analytical Validation:
The analytic calculation for the fixed shaft only calculated the rounded bearing shaft. Due to the geometry of the square shaft, this calculator was left to FEA to determine. Of the round portion, the FOS determined is 2.4, compared to the FEA result, the FOS is 5.74. This difference is due to the analytic solution only calculating the FOS based on a rounded shaft. The square portion utilized in the FEA will add greater strength to the shaft. Therefore, the FEA results will be considered.

Conclusion:
This study shows the FEA factor of safety and maximum stress of the fixed shaft. Both the analytic calculator and the FEA simulation proved the shaft does not fail. Due to the geometry of the shaft, analytic calculation only determined the FOS for the bearing portion, disregarding the square portion entirely.

Appendix E6: V Support

Author: Liam Wolf
Date: April 5, 2021

Objective:
To determine the highest stress induced on the V Support due to the loading of the drive shaft. Three simulations were run under the following conditions:

- Horizontal on Wall
- Vertical on Wall
- Totally Horizontal

It was found that the highest stress occurs when the V Support is vertical on the wall. To avoid redundancy, only this simulation set-up will be presented.

Assumptions:
The following assumptions were made for the purposes of this simulation:

- The loads applied to the V Support are the forces and moments from the drive shaft.
- The worst-case loading scenario on the V Support occurs when the robot is vertical on the wall.
- Assume 4130 steel.
- The V support is supported by the fixed shafts of the drive system, with no loading transferred to the V support.

Simulation Set-up Conditions & Procedure:
Figure 23 demonstrates the loading conditions applied to the model, with the analysis to calculate the loads presented in Appendix D. The following remote loads were used:

$F_{x1}=495$ N $Fx1=495$ N

$F_{x2}=1232$ N $Fx2=1232$ N

$F_{z1}=-338$ N $Fz1=-338$ N

$F_{z2}=1831$ N $Fz2=1831$ N

$M_{x1}=20$ Nm $Mx1=20$ Nm

$M_{x2}=-109$ Nm $Mx2=-109$ Nm

$M_{y1}=-39$ Nm $My1=-39$ Nm

$M_{y2}=-98$ Nm $My2=-98$ Nm

$M_{z1}=29$ Nm $Mz1=29$ Nm

$M_{z2}=73$ Nm $Mz2=73$ Nm

The inner surfaces of the holes connecting to the fixed shaft were considered fixed due to them being the axis of rotation for the V support as seen in Figure 39.

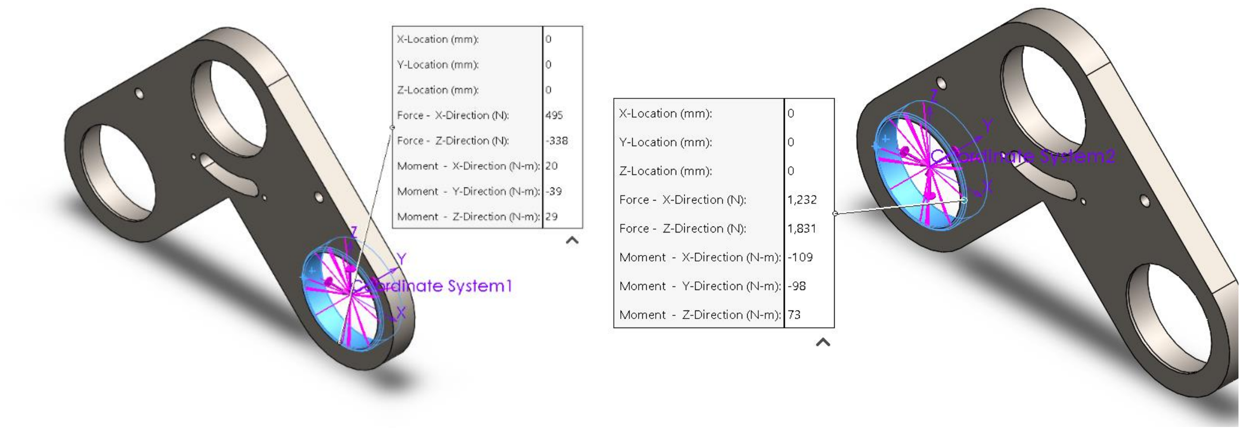


Figure 38 – V Support Loading Conditions

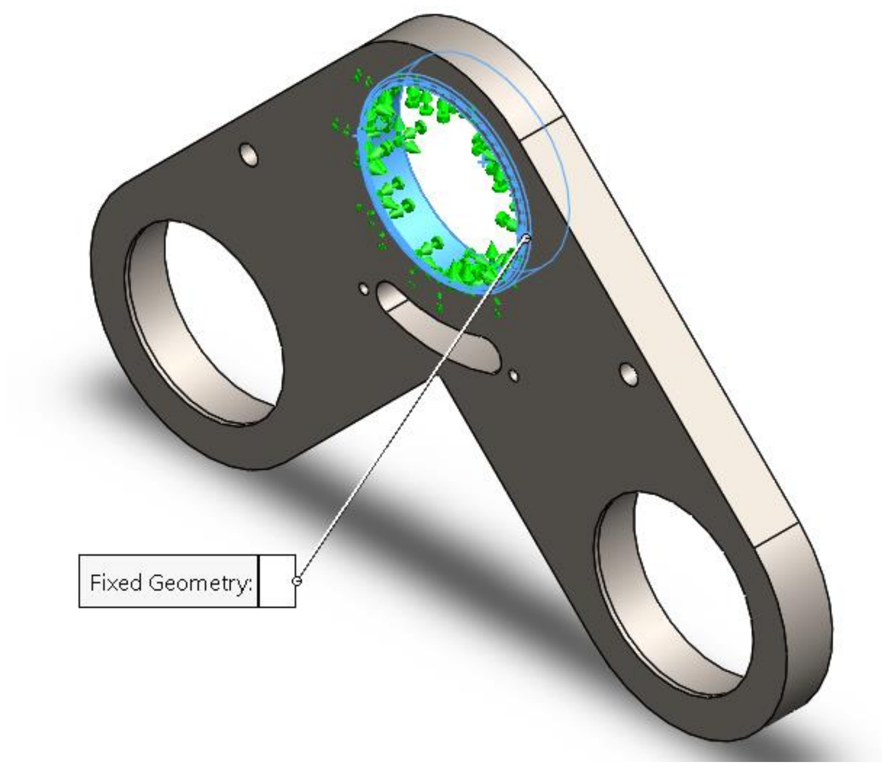


Figure 39 – Simulation set-up conditions for the V Support

Results:
From the results of the simulation shown in Figure 40, the maximum stress occurs around the slot of the V support. This is expected due to the high stress concentration in this area caused by the slot and the hole for the screws to hold the spring box. The minimum factor of safety was determined to be 4.8 which greatly exceeds the factor of safety of 1.2 assumed as standard for this project.

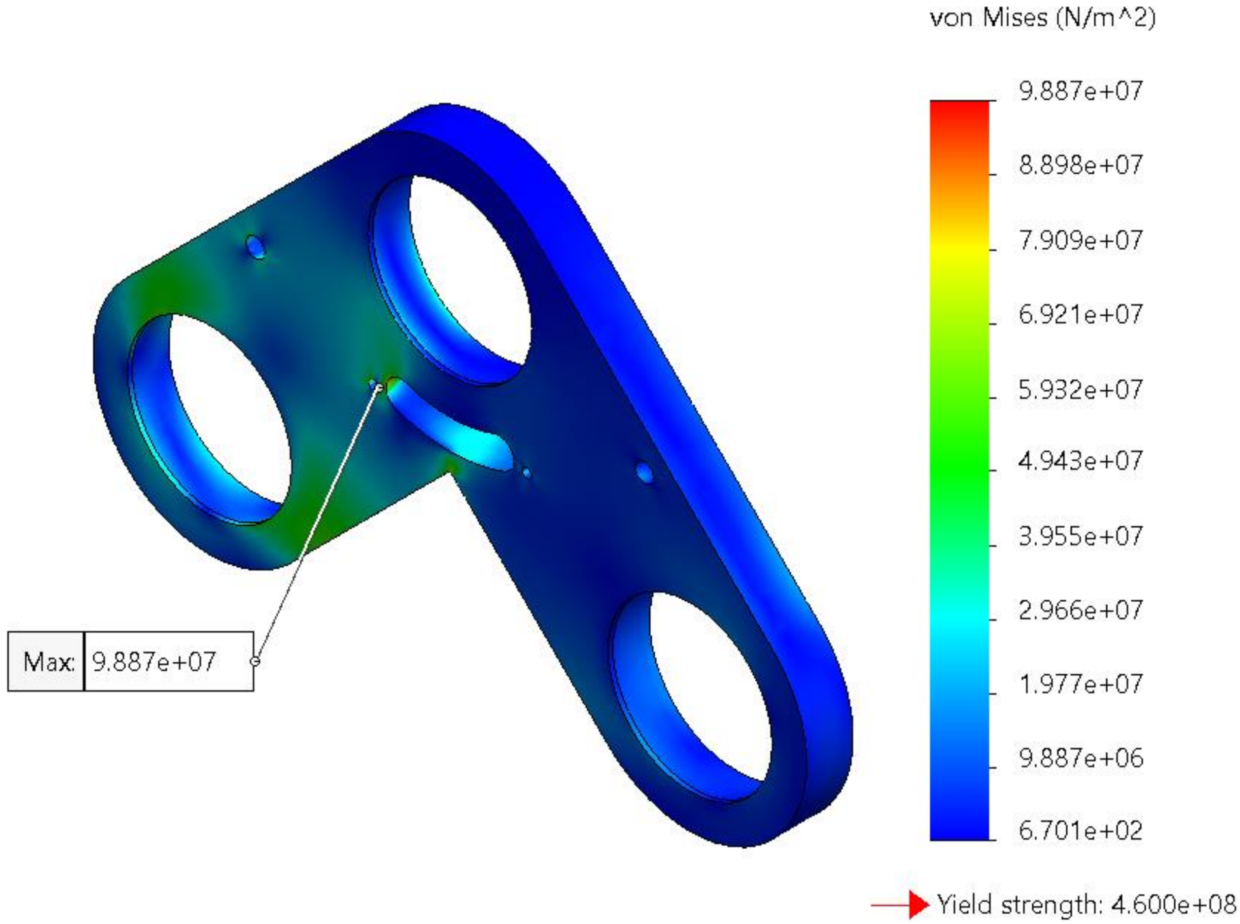


Figure 40 – Maximum Von Mises Stress of the V Support

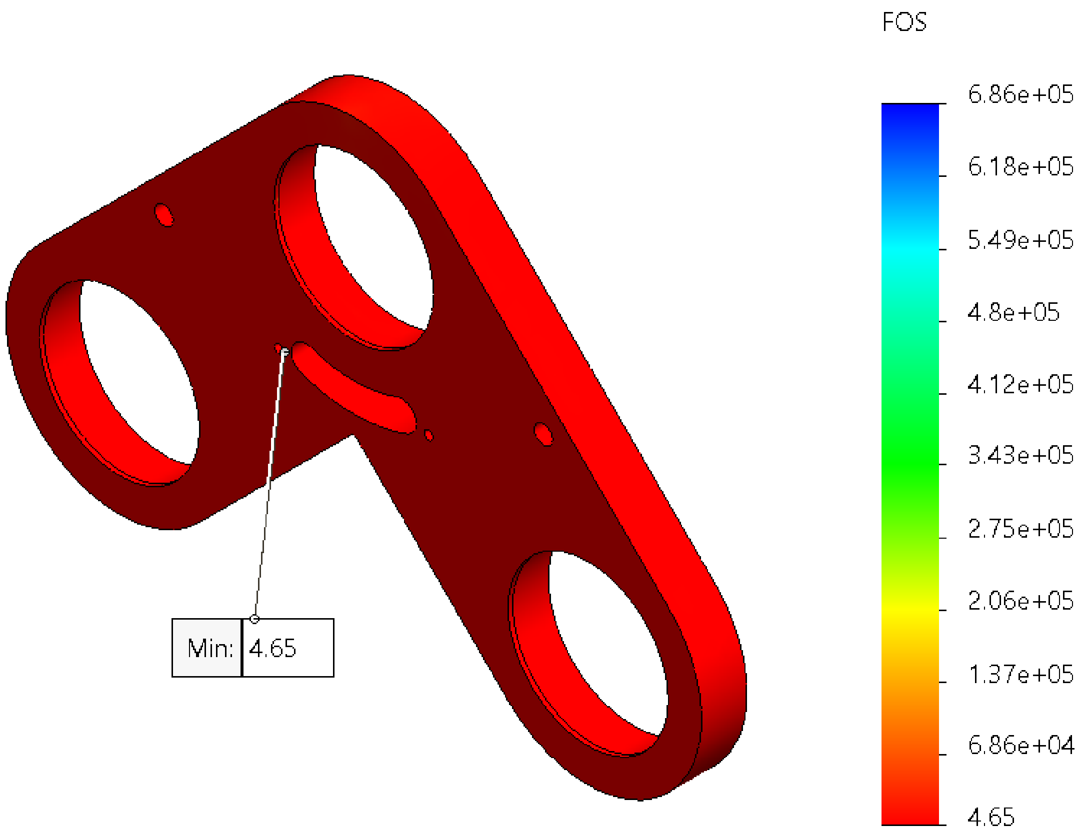


Figure 41 – Minimum Factor of Safety of the V Support

Analytical Validation:
Due to the complex nature of this loading scenario, no analytical validation via hand calculations was carried out for this component.

Conclusion:
This study suggests that due to the strength of the selected material, the V support is strong enough to support the load of the drive shafts.

Appendix E7: Modular Connector Plate

Author: Kenny Okeke
Date: April 13, 2021

Objective:
To determine the highest stress induced on the modular connector plate due to the conservative estimate of the weight of the robot. Two simulations were run under the following conditions:

- Horizontal on Wall
- Vertical on Wall

It was found that the highest stress occurs when the modular plate is vertical on the wall. To avoid redundancy, only this simulation set-up will be presented.

Assumptions:
The following assumptions were made for the purposes of this simulation:

- The load applied to the modular plate is the weight of the entire robot.
- The worst-case loading scenario on the modular plate occurs when the robot is vertical on the wall.
- Assume 4130 steel.
- The robot has a mass of 500 kg.

Simulation Set-up Conditions & Procedure:
Figure 42 demonstrates the loading conditions applied to the model, with the weight of the robot equating 4905 N. The top side surface of the modular plate was considered fixed for the purposes of the conservative estimate.

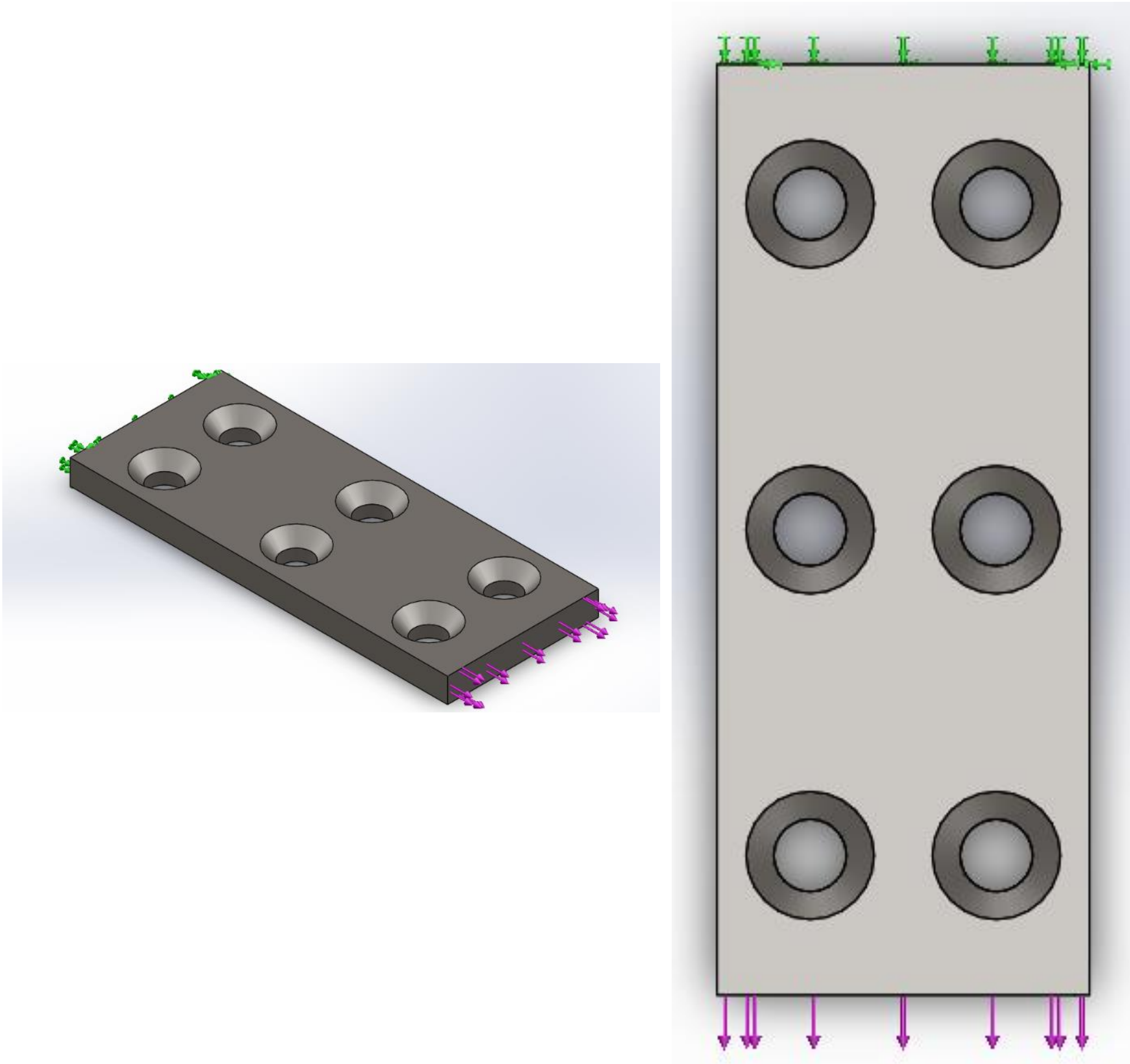


Figure 42 – Simulation set-up conditions for the Modular Plate

Results:
From the results of the simulation shown in Figure 43, the maximum stress occurs around the bolt holes of the modular plate. This is expected due to the high stress concentration in this area caused by the hole for the bolt and nut to connect the modular plate. The minimum factor of safety was determined to be 33 which greatly exceeds the factor of safety of 1.2 assumed as standard for this project as seen in Figure 44.

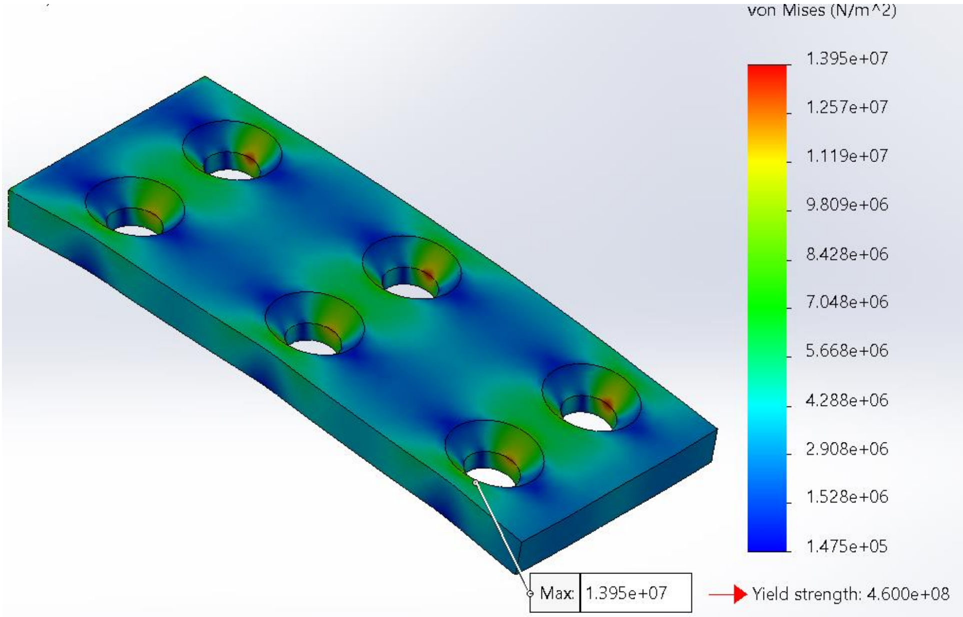


Figure 43 – Maximum Von Mises Stress of the Modular Plate

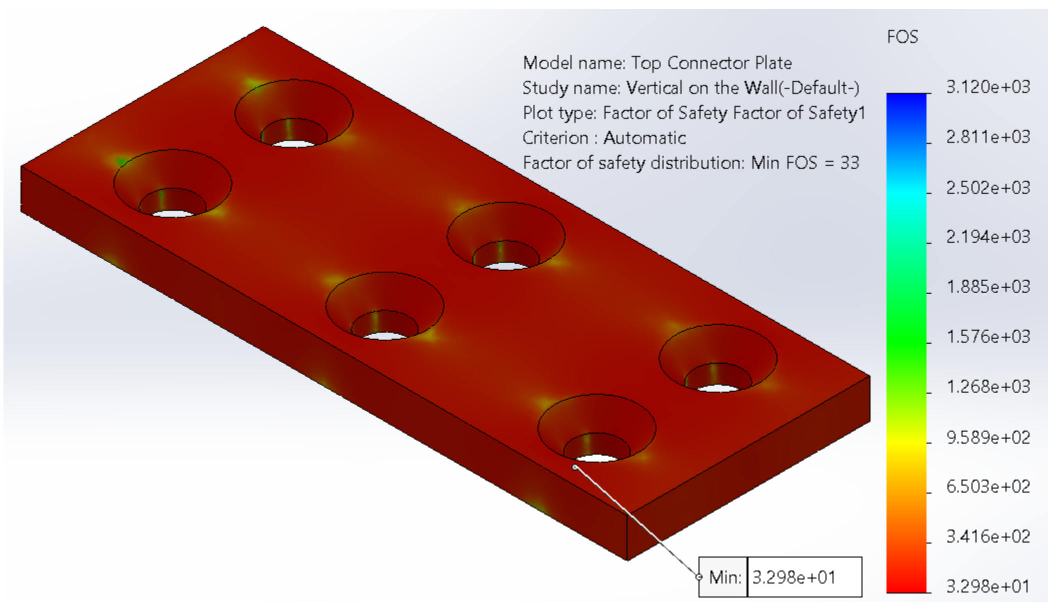


Figure 44 – Minimum Factor of Safety of the Modular Plate

Analytical Validation:

Due to the complex nature of this loading scenario, no analytical validation via hand calculations was carried out for this component.

Conclusion:

This study suggests that due to the strength of the selected material, the modular connector plate is strong enough to support the load of the robot. This was a conservative estimate, the actual load applied to the modular plate will be greatly reduced, and thus the component will not fail.

Appendix F: Operational Analysis: Set-up Procedure and Time Estimation

This appendix details the estimation of the assembly and set-up procedure for the OmiBot once inserted into the vessel opening.

Author: Areej Khaddaj
Date: April 12, 2021

Objective

Approximate the time required to set up the modular robotic system, from a disassembled to an assembled state.
Establish set-up procedure of the modular subsystems.

Assumptions

1. Assumed one minute required to torque a bolt, based on video reference.
2. Transition time between bolted connections is 30 seconds.
3. Additional time of 15 minutes is estimated to include time required to raise the manipulator onto the platform.
4. Set up of electronics, controls, and the winch connections is not included for the time estimation.

Analysis

The modular robotic system is assembled in the following order:

a. Platform modules are connected together with top modular plate.
b. Drive systems are connected to the top plate by bolting the fixed shaft to the plate.
c. Manipulator is loaded onto the top plate and connected.
d. Halves of the chassis are connected together using the bottom modular plate.
e. Linear actuators are mounted onto the front and back end of the chassis.
f. Chassis connected to the top plate with bolted threads.
g. Winch is connected to the hoist rings to provide support while the robot travels vertically.

Set-up time estimation:

Table 7 - OmiBot Estimated Assembly Time

Task	Number of bolt connections	Time estimate (s)
(a) Assemble modular halves of top plate	6	510
(b) Connect drive systems to top plate	8	690
(c) Connect manipulator to top plate	4	330 + 15(60s)
(d) Connect halves of chassis with bottom modular plate	6	510
(e) Mount linear actuators to chassis	8	690
(f) Connect top plate and chassis	8	690
Total	44	4320

Conclusion:

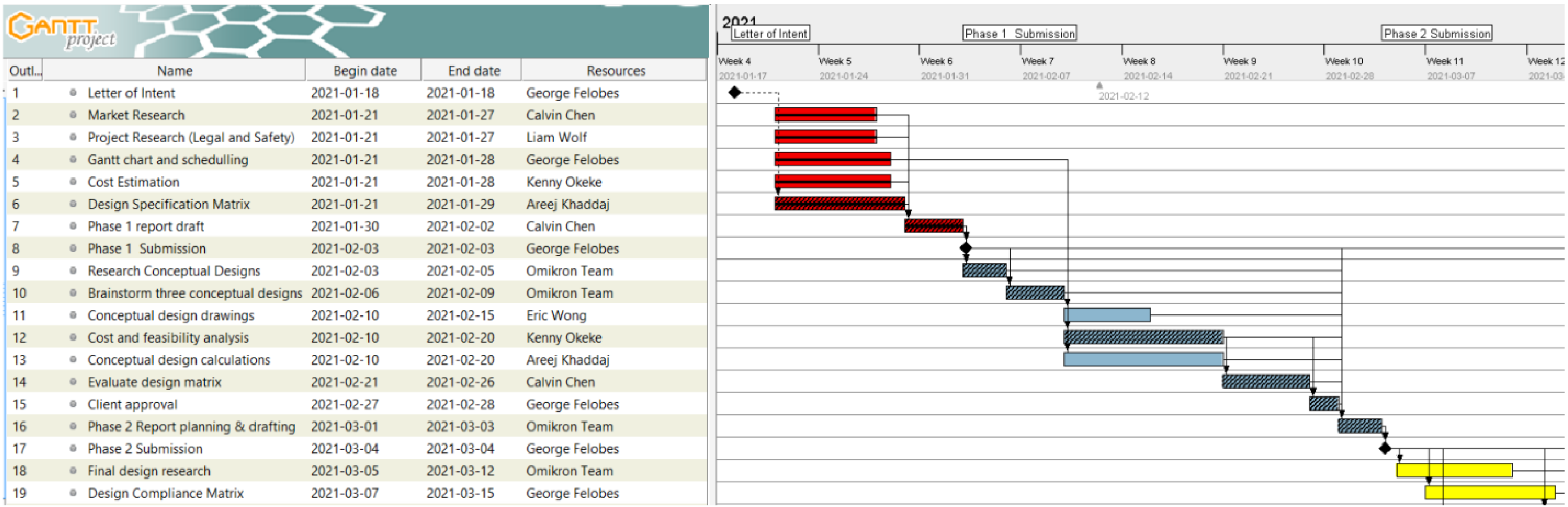
Assembly of the OmiBot modules from a disassembled to and assembled state takes approximately 72 minutes to complete.

Appendix G: Project Management

This appendix contains details relating to the project management of this project.

Appendix G1: Project Schedule Management: Gantt Chart

The Gantt chart is presented in Figure 45 and Figure 46 shows Phase I in red, Phase II in grey, Phase III in yellow and the poster/presentation section in dark blue. The diamond shapes indicate a milestone, such as Phase I, Phase II, Phase III submission, the poster, and presentation submission. The black lines between the bars indicate the dependencies between the tasks. The shaded bars show the critical path where potential bottlenecks would occur in the project. The critical path is based on the dependencies between the tasks. The tasks, along with the start and finish dates are shown in a tabular form within the Gantt chart. The vertical blue lines indicate the start and end of the project, while the vertical red line indicates the current position in time, when this report is submitted. Note that certain tasks in phase III have been revisited, and further refined.



Outl...	Name	Begin date	End date	Resources
1	Letter of Intent	2021-01-18	2021-01-18	George Felobes
2	Market Research	2021-01-21	2021-01-27	Calvin Chen
3	Project Research (Legal and Safety)	2021-01-21	2021-01-27	Liam Wolf
4	Gantt chart and schedulling	2021-01-21	2021-01-28	George Felobes
5	Cost Estimation	2021-01-21	2021-01-28	Kenny Okeke
6	Design Specification Matrix	2021-01-21	2021-01-29	Areej Khaddaj
7	Phase 1 report draft	2021-01-30	2021-02-02	Calvin Chen
8	Phase 1 Submission	2021-02-03	2021-02-03	George Felobes
9	Research Conceptual Designs	2021-02-03	2021-02-05	Omikron Team
10	Brainstorm three conceptual designs	2021-02-06	2021-02-09	Omikron Team
11	Conceptual design drawings	2021-02-10	2021-02-15	Eric Wong
12	Cost and feasibility analysis	2021-02-10	2021-02-20	Kenny Okeke
13	Conceptual design calculations	2021-02-10	2021-02-20	Areej Khaddaj
14	Evaluate design matrix	2021-02-21	2021-02-26	Calvin Chen
15	Client approval	2021-02-27	2021-02-28	George Felobes
16	Phase 2 Report planning & drafting	2021-03-01	2021-03-03	Omikron Team
17	Phase 2 Submission	2021-03-04	2021-03-04	George Felobes
18	Final design research	2021-03-05	2021-03-12	Omikron Team
19	Design Compliance Matrix	2021-03-07	2021-03-15	George Felobes

Figure 45 - Omikron Robotics Gantt Chart (1)

Outl...	Name	Begin date	End date	Resources
13	Conceptual design calculations	2021-02-10	2021-02-20	Areej Khaddaj
14	Evaluate design matrix	2021-02-21	2021-02-26	Calvin Chen
15	Client approval	2021-02-27	2021-02-28	George Felobes
16	Phase 2 Report planning & drafting	2021-03-01	2021-03-03	Omikron Team
17	Phase 2 Submission	2021-03-04	2021-03-04	George Felobes
18	Final design research	2021-03-05	2021-03-12	Omikron Team
19	Design Compliance Matrix	2021-03-07	2021-03-15	George Felobes
20	CAD modelling	2021-03-15	2021-03-25	Eric Wong
20.1	Transmission	2021-03-15	2021-03-25	Eric Wong
20.2	Actuation	2021-03-15	2021-03-18	Liam Wolf
20.3	Connections	2021-03-15	2021-03-18	Calvin Chen
20.4	Chassis	2021-03-15	2021-03-25	Calvin Chen
21	Detailed Design Calculations	2021-03-08	2021-03-20	Areej Khaddaj
21.1	Dynamic and Static Analysis	2021-03-08	2021-03-20	George Felobes
21.2	Stability at different angles	2021-03-10	2021-03-15	George Felobes,Areej Khadd...
21.3	Stability with an obstacle	2021-03-10	2021-03-15	George Felobes,Areej Khadd...
21.4	Motor Specifications	2021-03-08	2021-03-14	Calvin Chen
22	Simulation	2021-03-15	2021-03-28	Liam Wolf
22.1	Optimization	2021-03-15	2021-03-20	Liam Wolf
22.2	FEA	2021-03-15	2021-03-28	Eric Wong
22.3	Motion Analysis	2021-03-20	2021-03-25	Eric Wong,Liam Wolf,Calvin ...
23	Compile Drawing Package	2021-03-26	2021-03-28	Calvin Chen
24	Poster drafting	2021-03-26	2021-03-29	Omikron Team
25	Poster	2021-03-30	2021-03-30	Omikron Team
26	Presentation drafting	2021-03-26	2021-04-04	Omikron Team
27	Presentation	2021-04-05	2021-04-05	Omikron Team
28	Draft Phase 3 report	2021-03-29	2021-04-10	Omikron Team
29	Phase 3 Submission	2021-04-15	2021-04-15	George Felobes

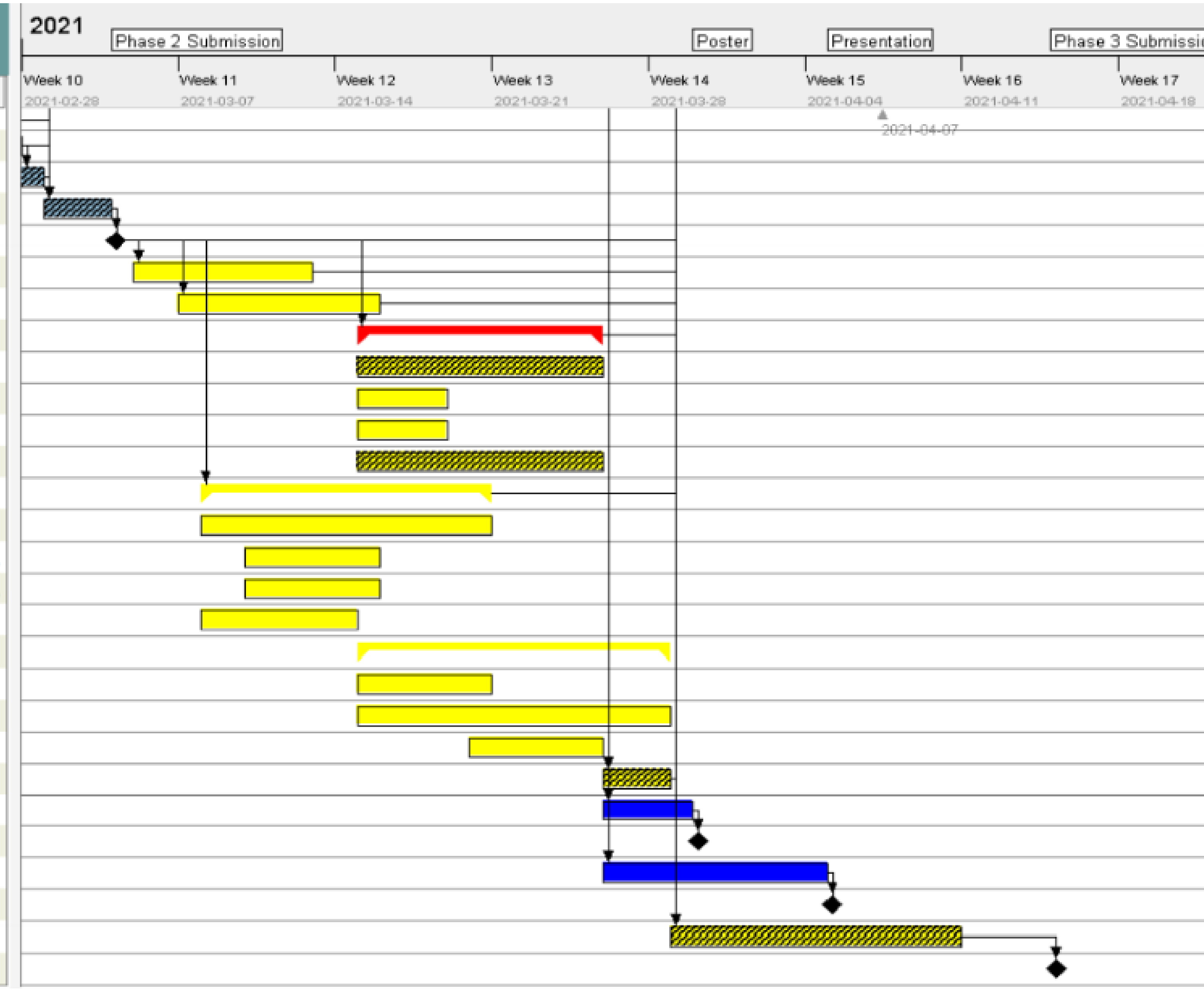


Figure 46 - Omikron Robotics Gantt Chart (2)

Appendix G2: Omikron Robotics Time Sheets

The timesheets used to track individual hours for Phase III. The poster and presentation effort is shown in Table 8.

Table 8 - Omikron Robotics Phase III Time Sheet

Phase 3												
Start date	End date	Responsibility	Task	Baseline total hours	Actual total hours	Liam Wolf	Calvin Chen	George Felobes	Eric Wong	Areej Khaddaj	Kenny Okeke
5-Mar-2021	12-Mar-2021	All	Additional Research	24	35	3	3	10	6	9	4
7-Mar-2021	15-Mar-2021	GF	Design Compliance Matrix	20	18	1	10	3	1	1	2
9-Mar-2021	9-Mar-2021	All	Client Meeting - #7	6	6	1	1	1	1	1	1
15-Mar-2021	25-Mar-2021	EW	CAD Modelling (transmission)	10	16	8	8	0	0	0	0
15-Mar-2021	18-Mar-2021	LW	CAD Modelling (actuation)	8	9	0	0	5	4	0	0
15-Mar-2021	18-Mar-2021	CC	CAD Modelling (connections)	4	10	4	2	0	0	0	4
15-Mar-2021	25-Mar-2021	CC	CAD Modelling (chassis)	8	4.5	3	1	0.5	0	0	0
15-Mar-2021	20-Mar-2021	LW	Simulations (optimization)	12	14	5	0	2	0	5	2
20-Mar-2021	28-Mar-2021	EW	Simulations (FEA)	8	39.5	25	0.5	4	0	8	2
20-Mar-2021	25-Mar-2021	EW, LW, CC	Simulations (motion analysis)	10	10	0	10	0	0	0	0
9-Mar-2021	9-Mar-2021	All	Faculty Advisor Meeting -#7	6	6	1	1	1	1	1	1
8-Mar-2021	20-Mar-2021	GF	Detailed Design Calculations (static and	20	69	2	10	8	25	12	12

			dynamic analysis)								
10-Mar-2021	15-Mar-2021	GF, AK	Detailed Design Calculations (stability with obstacle)	10	11	0	0	6	5	0	0
10-Mar-2021	15-Mar-2021	GF, AK	Detailed Design Calculations (Stability at all angles)	10	9	0	0	9	0	0	0
8-Mar-2021	14-Mar-2021	CC	Detailed Design Calculations (motor specs)	10	3	0	3	0	0	0	0
16-Mar-2021	24-Mar-2021	KO	Costing Analysis	25	14	1	1	0	0	0	12
16-Mar-2021	16-Mar-2021	All	Client Meeting - #8	6	6	1	1	1	1	1	1
16-Mar-2021	16-Mar-2021	All	Faculty Advisor Meeting - #8	6	6	1	1	1	1	1	1
23-Mar-2021	23-Mar-2021	All	Client Meeting - #9	6	6	1	1	1	1	1	1
24-Mar-2021	28-Mar-2021	EW	Compile Drawing Package	10	3	0	0	3	0	0	0
1-Apr-2021	8-Apr-2021	All	Phase 3 Report Planning	15	22	3	3	6	3	4	3
8-Apr-2021	12-Apr-2021	All	Phase 3 Report drafting	36	68	8	8	8	4	20	20
23-Mar-2021	23-Mar-2021	All	Faculty Advisor Meeting - #9	6	6	1	1	1	1	1	1
12-Apr-2021	14-Apr-2021	All	Phase 3 Report Review	6	19	5	5	4	1	3	1
10-Apr-2021	15-Apr-2021	CC	Compile & Submit Phase 3 Report	2	9	4	4	1	0	0	0
Totals				**284**	**419**	**78**	**74.5**	**75.5**	**55**	**68**	**68**

Design Poster/ Presentation												
Start date	End date	Responsibility	Task	Baseline total hours	Actual total hours	Liam Wolf	Calvin Chen	George Felobes	Eric Wong	Areej Khaddaj	Kenny Okeke
26-Mar-2021	30-Mar-2021	All	Initial Poster Draft	18	35	10	1	4	2	8	10
25-Mar-2021	7-Apr-2021	All	Oral Design Presentation Preparation	12	24	4	4	2	2	8	4
30-Mar-2021	30-Mar-2021	All	Faculty Advisor Meeting - #10	6	6	1	1	1	1	1	1
30-Mar-2021	30-Mar-2021	All	Client Meeting - #10	6	6	1	1	1	1	1	1
2-Apr-2021	2-Apr-2021	GF	Design Poster Submission	1	0.5	0	0	0.5	0	0	0
8-Apr-2021	8-Apr-2021	All	Final Poster Presentation	30	13	2	2	4	2	1	2
Total				73	84.5	18	9	12.5	8	19	18

Appendix G3: Project Hours distribution

Omikron Robotics hour distribution between the team members is shown in Figure 47. This is for the actual hours spend on the project. Note: Phase III also includes the hours spend on designing and drafting both the poster and presentation.

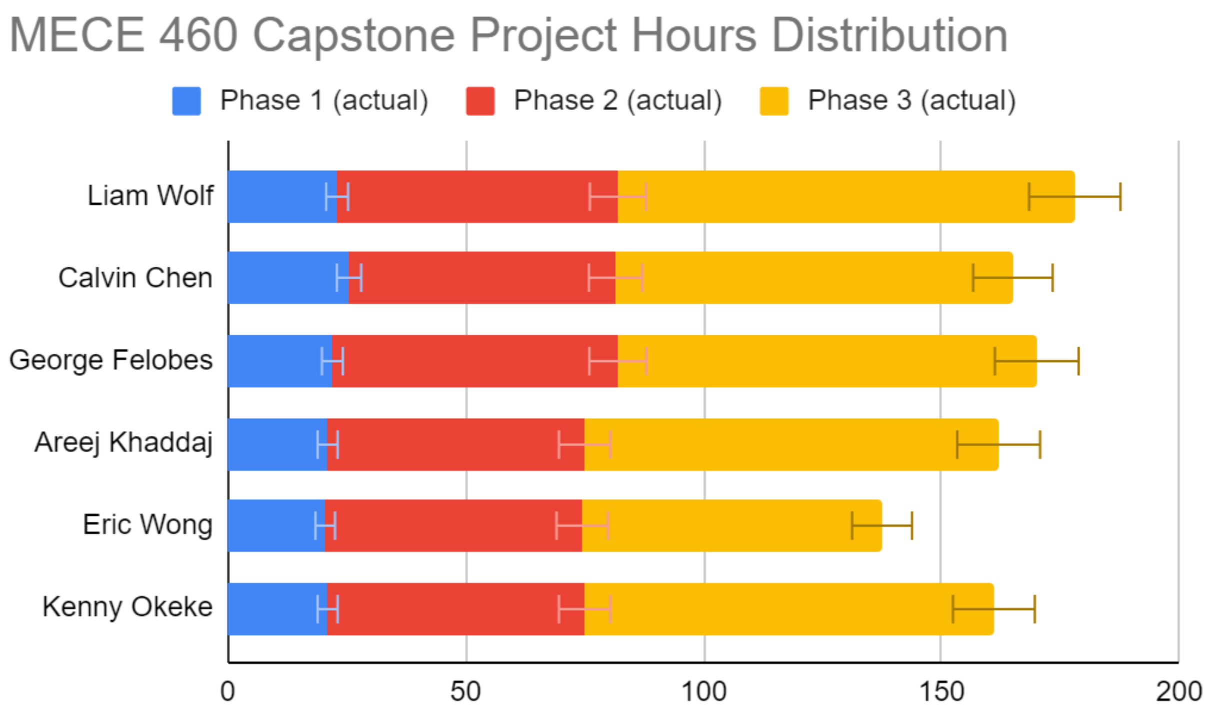


Figure 47 - Omikron Robotics Hour Distribution Per Team Member

Appendix H: Team Charter

Mec E 460 – Senior Capstone Design Project
Group Charter

Mec E 460 – Senior Capstone Design Project

Group Charter

Our Group – Contact Information

Team Name: Omikron Robotics

Name: George Felobes (team lead) E-mail: Felobes@ualberta.ca	Name: Liam Wolf E-mail: lwolf@ualberta.ca
Name: Calvin Chen E-mail: calvin1@ualberta.ca	Name: Eric Wong E-mail: ewwong@ualberta.ca
Name: Areej Khaddaj E-mail: areej@ualberta.ca	Name: Kenny Okeke E-mail: kenechuk@ualberta.ca

About Our Relationship

Group Norms

We consider the following attitudes and behaviors to be important to our group and will strive to uphold these in our work as a group:

- We treat each other with respect.
- We discuss issues and agree by consensus.
- We are honest with each other.
- We admit when we aren't sure about something.
- We get our work done on time.

Rules and responsibilities:

? All group members will actively contribute to the completion of all assignments; all contributing team members will have their names included on the assignment. An uncooperative or non-performing team member may have his/her name(s) excluded from the submitted assignment.

? After each assignment, the team will meet to discuss why marks were lost and how the submission could have been improved. The performance of each group member will be assessed according to assigned task(s).

? Meeting schedule will be determined by team. Members are expected to attend and e-mail/text if unable. Show up on time.

? Team members are expected to complete individual preparation for team meetings.

? Team members, when given a task, should report back to the group in a timely manner as per deadlines discussed in team meetings.

? Team members will support each other in accomplishing individual tasks offering encouragement, resources, and assistance when necessary. Team members will encourage participation of each other during meetings.

? **Team members will respect all members' opinions and offer constructive criticism when necessary, remembering to criticize the idea NOT the person.**

? In an effort to respect each other's time commitments, the team will make every effort possible to adhere to the meeting agenda. The team lead will chair meetings.

? No member will be able to intimidate another member, e.g., shouting or other aggressive behaviors will NOT be tolerated by the group.

- ? Active participation at meetings by all members is required. At times, members may be called upon to verbalize (or make explicit) opinions, support, acceptance, rejection, understanding, elaboration, justification, etc.
- ? Every effort will be made to retain each group member. Underperforming members, for example, will be encouraged or given tasks at which they can better succeed.
- ? For legitimate absences from group meetings, the team will make sure that the absent member gets caught up on missed material or information.
- ? Conflicts will be resolved via consensus-based philosophy, i.e., every team member's contribution/opinion is valued and will be used in the resolution.
- ? If conflicts cannot be resolved within the group, then the MEC E 460 conflict resolution procedure will be followed.
- ? Regularly and equally post to the eclass discussion forum.
- ? Each person gets to chair and take meeting minutes on the faculty advisor meetings at least ones.

Guidelines for Communication

Best way to communicate: Messenger for chats. Weekly meetings (Google Meets) as a regular communication. Eclass for final formal communication

Messaging: Slack

Meetings:

1. Team meetings: Thursday 1 PM + On demand - Google Meets
2. Faculty advisor: Weekly, Tuesdays at 1 PM - Google Meets
3. Client: Weekly, Tuesdays at 4:30 PM (On demand) - Google Meets

Our Task-related Goals

We have identified the following tasks (e.g., related to preliminary & further research; selecting the technology; selecting implementation strategy, key concepts to focus on; conducting analysis; report writing & editing; and work coordination). By signing our initials below, we accept responsibility for completing these tasks by dates indicated.

Name	Title	Role
George Felobes	Project Manager	Ensure all deliverables are met. Stay on schedule. Update schedule as needed. Point of contact.
Calvin Chen	Deliverable's manager	Compiling reports and ensure all components are coherent.
Areej Khaddaj	Technical Analysis	Technical Calculations
Eric Wong	CAD Modelling	SolidWorks
Liam Wolf	Numerical simulation	FEA, dynamics, trajectory
Kenny Okeke	Feasibility/Cost Analysis	Pricing, bill of materials, stay within budget.

Signatures

Our signatures below indicate that we have seen and agreed to the terms of this Charter. Note: it is a good idea for each student to initial each page of the Charter.

Name of Group Member	Signature (Jan 21, 2021)	New Signature (Jan 30, 2021)
George Felobes		
Calvin Chen		
Liam Wolf		
Areej Khaddaj	Areej	Areej
Kenny Okeke		
Eric Wong	Eric Wong	Eric Wong

Appendix I: Engineering Drawings

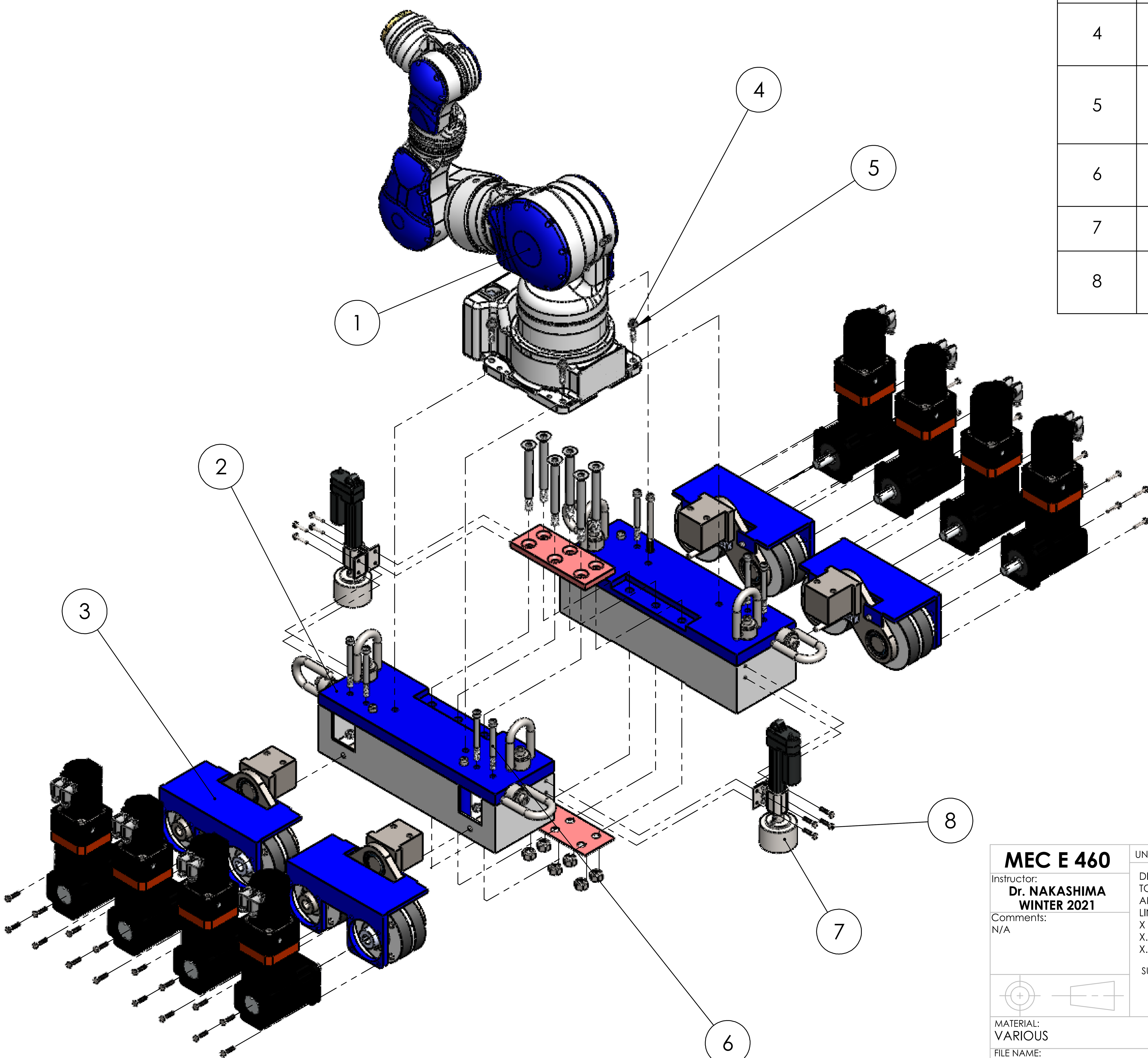
The engineering drawing package for the OmiBot is presented in the following pages of this appendix.

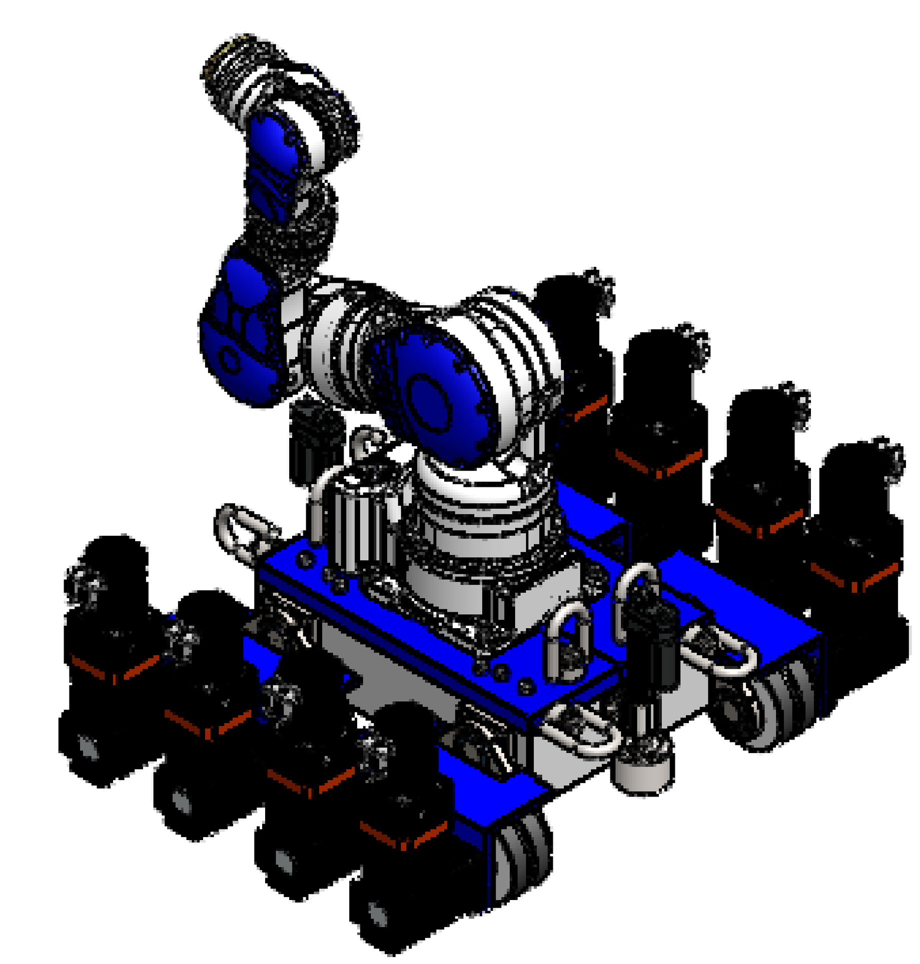
Drawing Tree

Notes:
1. Hardware and fasteners were included in drawings, but omitted from drawing tree for clarity purposes.
2. Manufacturer part numbers of purchased components are included in the BOM of their respective subassembly.

NOTES:
1. FASTENERS NOT INDICATED IN THIS BOM ARE INCLUDED IN THE BOM OF THEIR RESPECTIVE SUBASSEMBLY.

ITEM NO.	DESCRIPTION	VENDOR	VENDOR PART NUMBER	QTY.
1	Manipulator Arm Subassembly	Yaskawa	SIA20D ROBOT	1
2	Modular Chassis Subassembly	-	-	2
3	Modular Drive Subassembly	-	-	4
4	Fine-Thread Alloy Steel Socket Head Screw, M12 x 1.5 mm Thread, 50 mm Long	McMaster-Carr	96144A302	4
5	Black Ultra-Corrosion-Resistant Coated Steel Washer for M12 Screw Size, 13 mm ID, 24 mm OD	McMaster-Carr	93413A180	4
6	Black-Oxide Alloy Steel Socket Head Screw, 1/2"-20 Thread Size, 5" Long	McMaster-Carr	91251A031	8
7	Modular Linear Actuator Subassembly	-	-	2
8	Black-Oxide Alloy Steel Socket Head Screw, M8 x 1.25 mm Thread, 35 mm Long	McMaster-Carr	91290A438	8





MEC E 460
Instructor: **Dr. NAKASHIMA**
WINTER 2021
Comments: N/A

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: VARIOUS
FILE NAME: Model 2 - Pivot Tank Outside Config

OMIKRON ROBOTICS Group 15

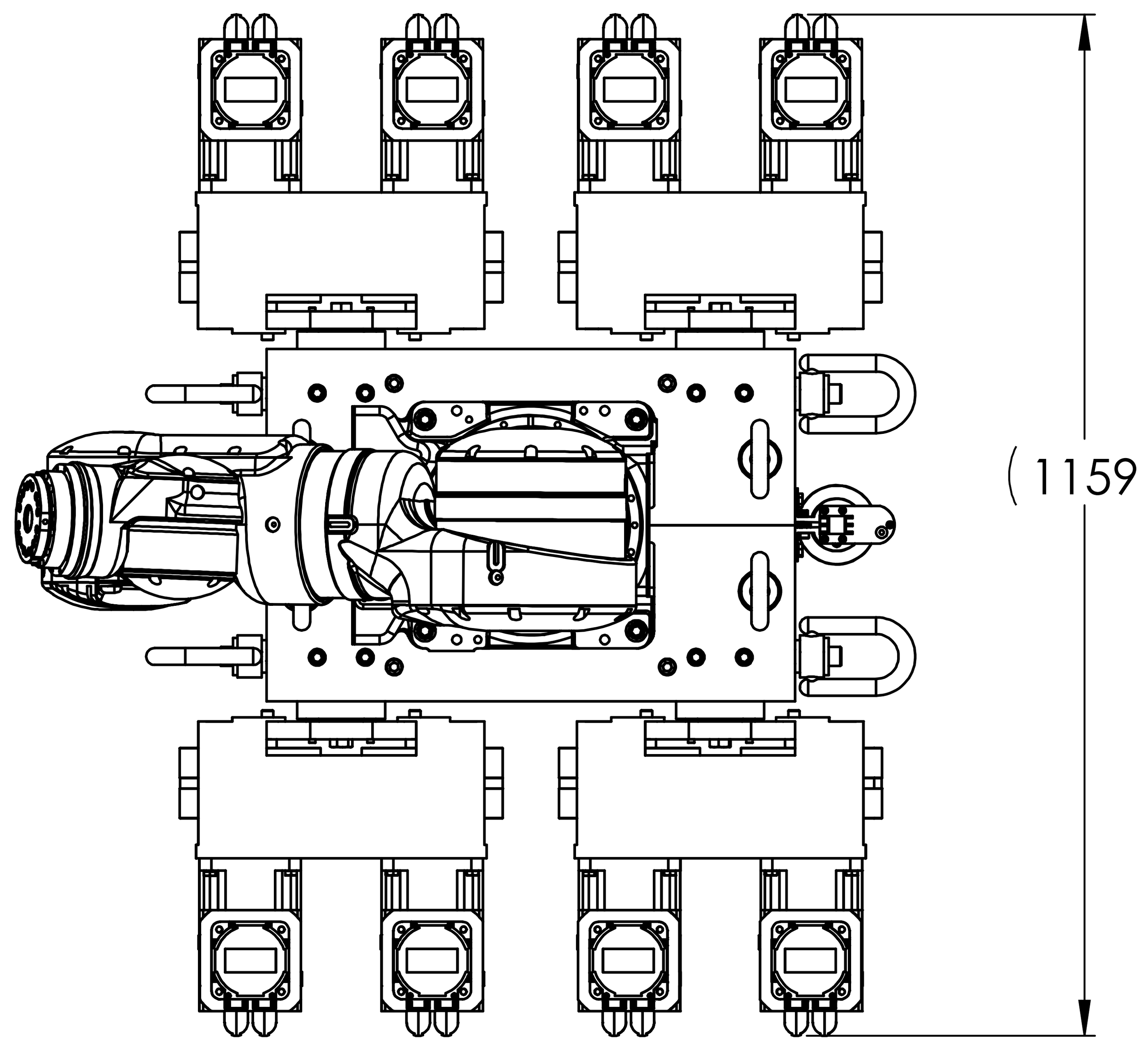
Modeled By **Liam Wolf**
Drawn By **Liam Wolf**
Checked By **Calvin Chen**
Liam
April 13, 2021 12:21:07 AM
February 24, 2021 11:15:24 PM

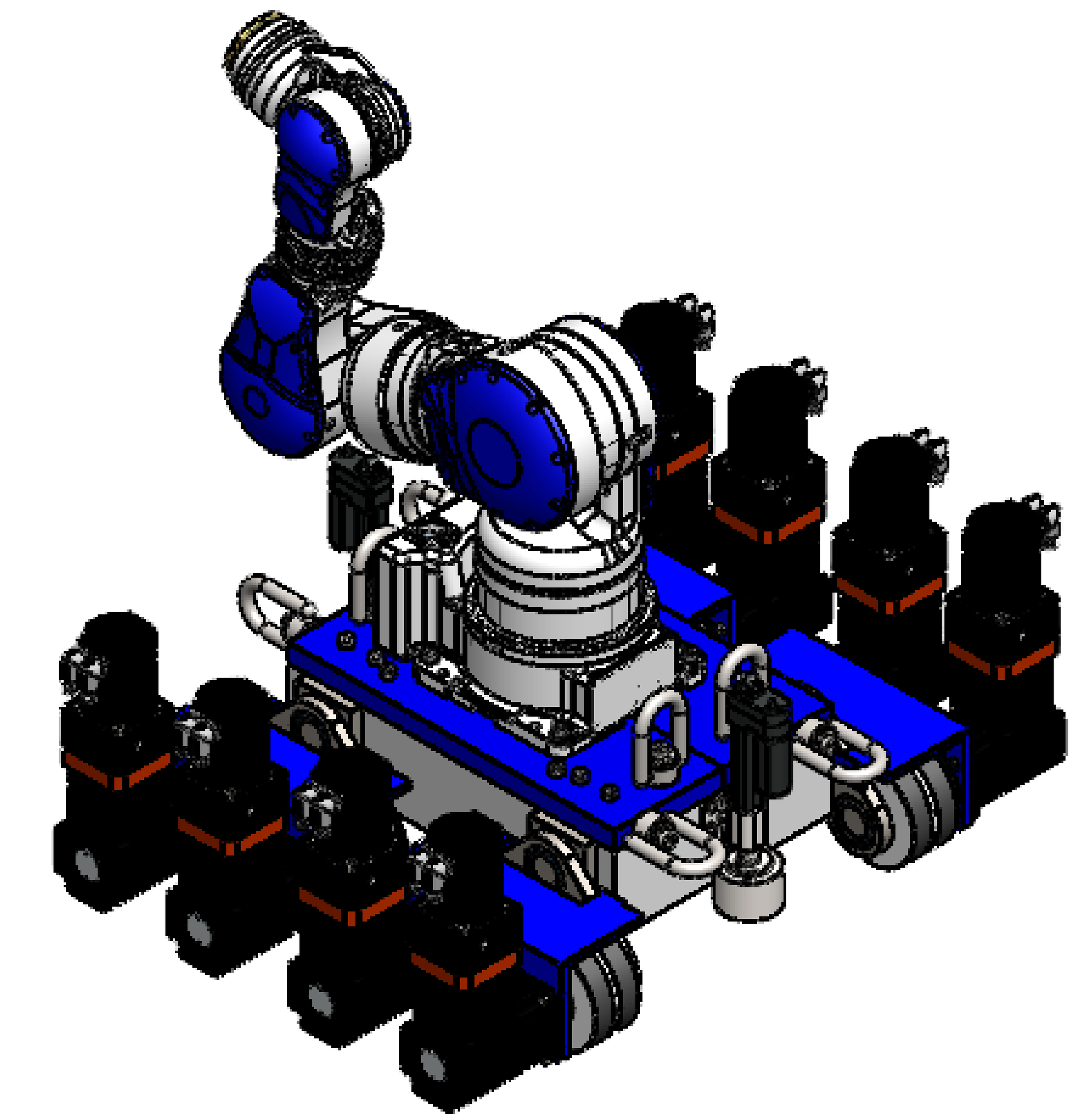
The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

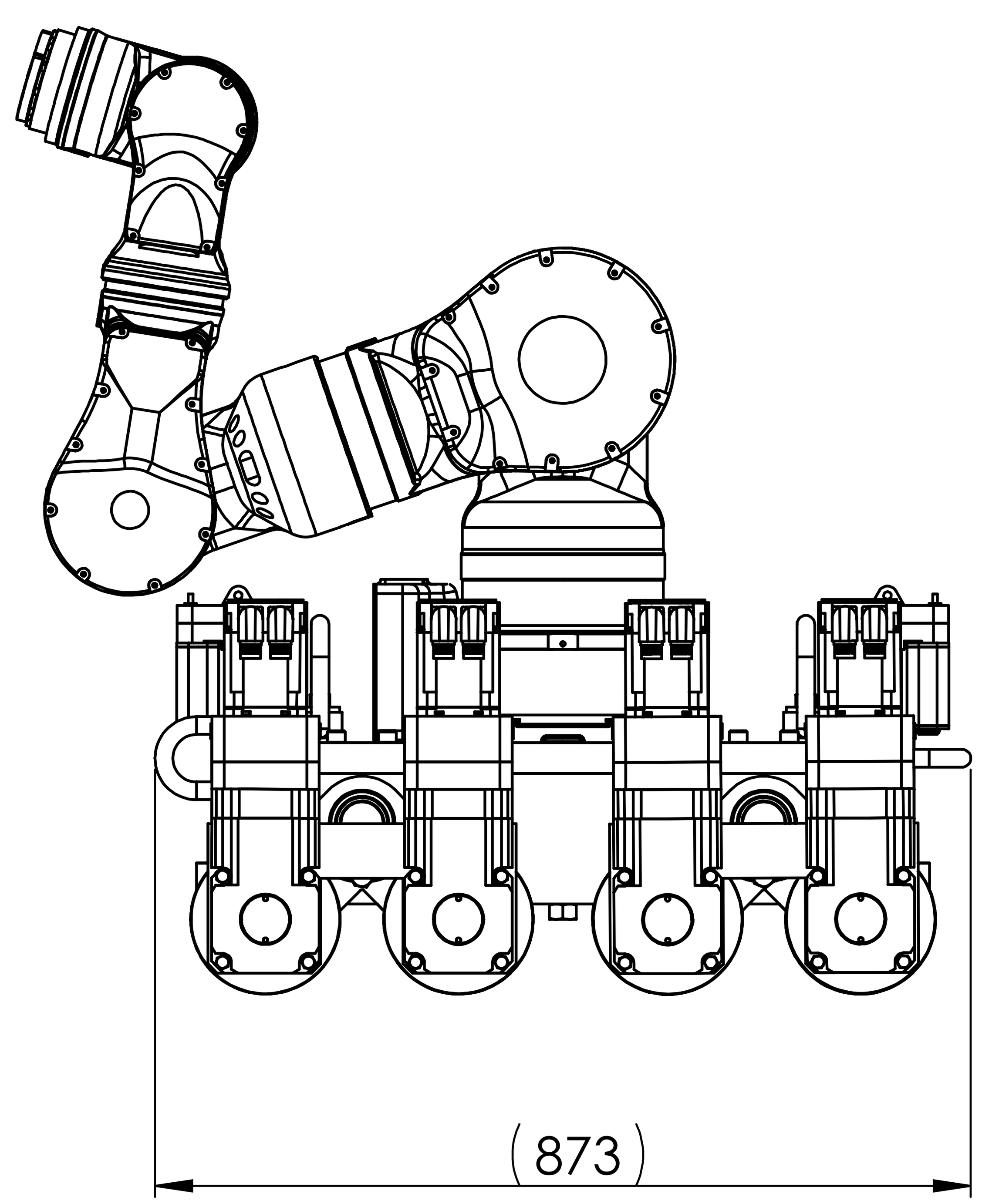
TITLE: OmiBot Assembly

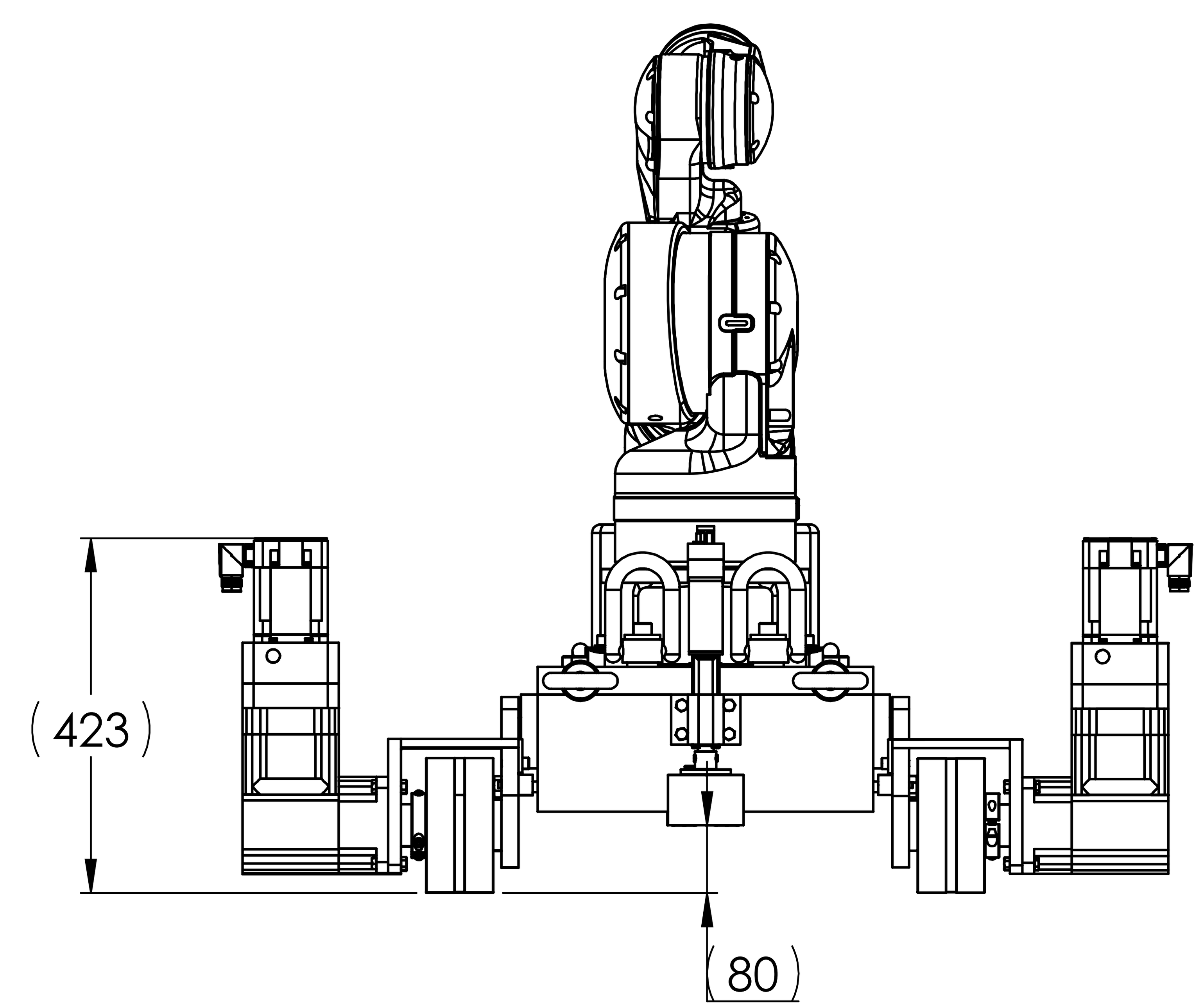
SIZE	Assignment Number	REV
B	N/A	1

SCALE: 1:11 | Mass: 516284.190 | SHEET 1 OF 2



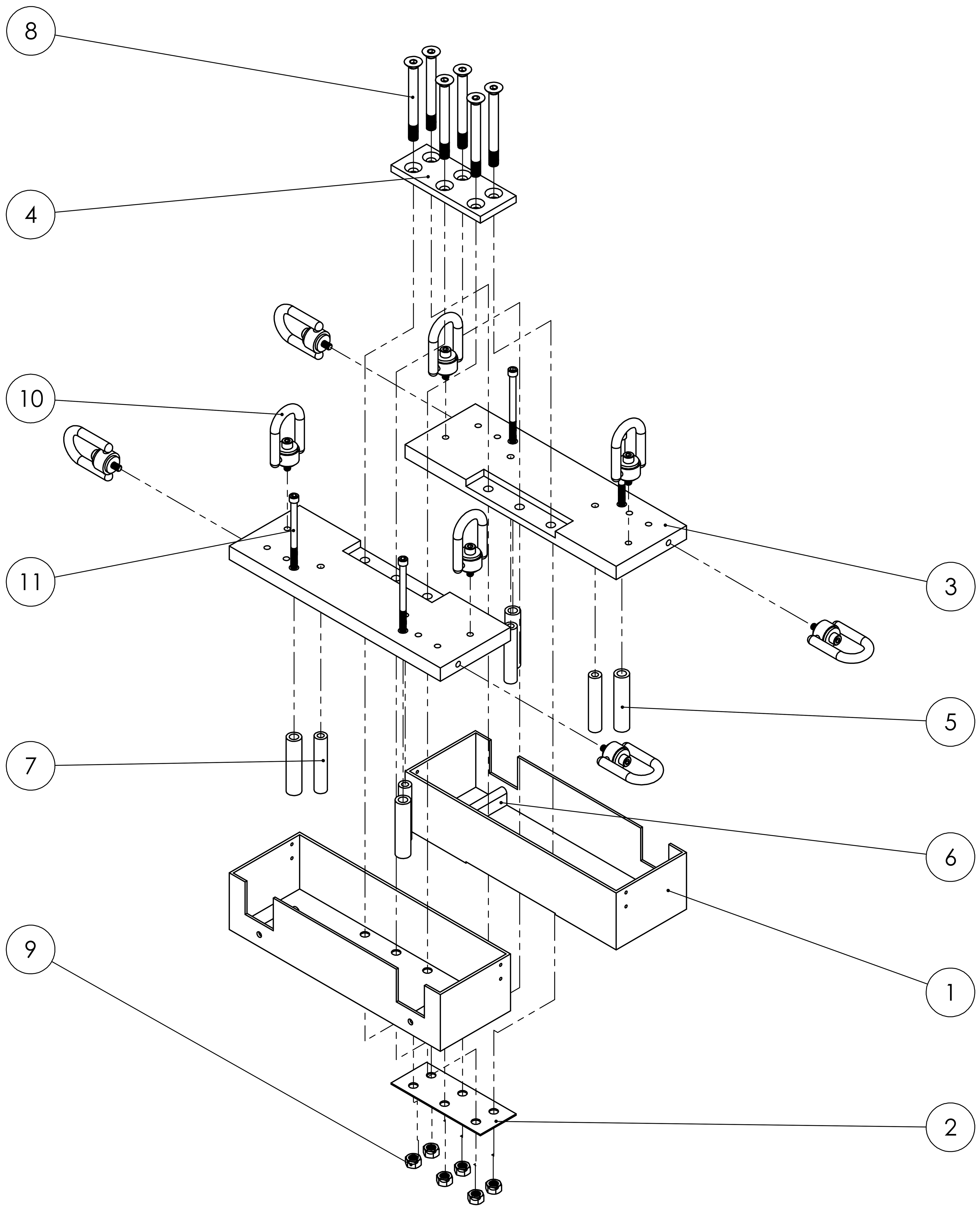


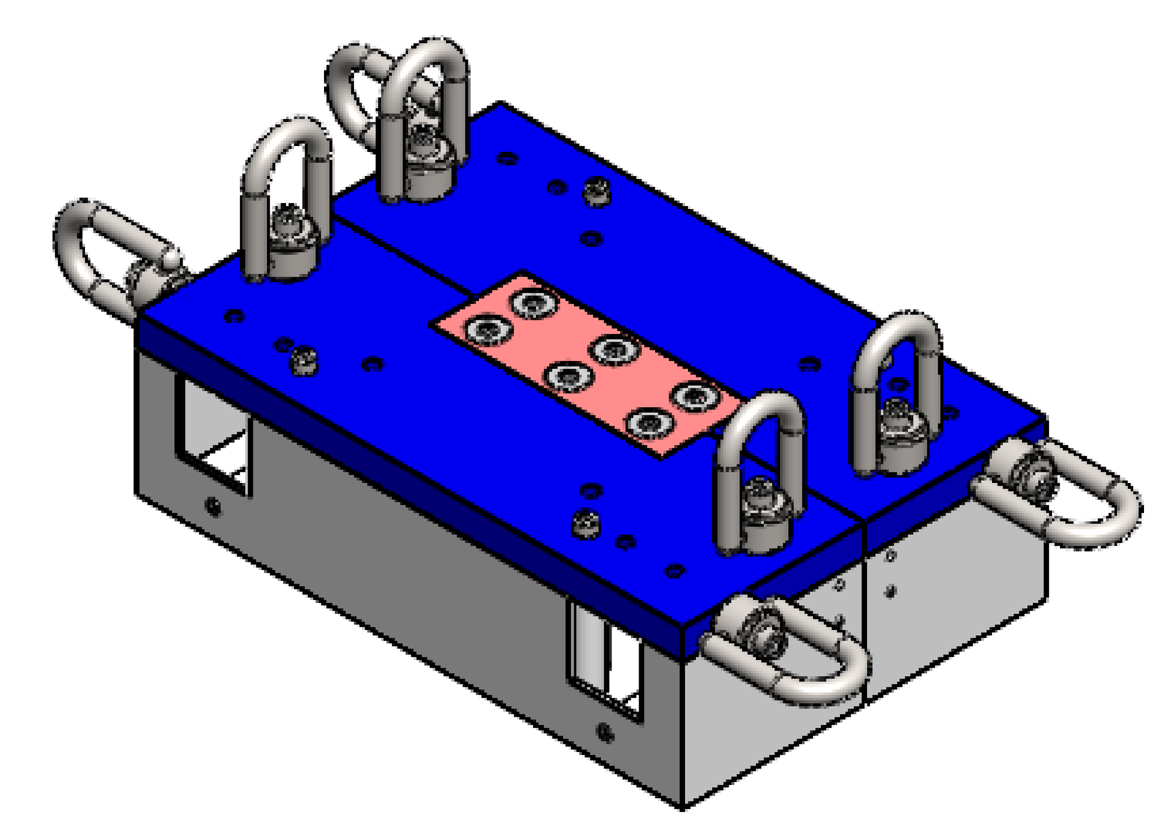




MEC E 460 Instructor: **Dr. NAKASHIMA** **WINTER 2021** Comments: N/A	UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MM TOLERANCES: ANGULAR: ± 0.5° LINEAR X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.025 SURFACE FINISH 0.6 µm DO NOT SCALE DRAWING	OMIKRON ROBOTICS Group 15 Modeled By **Liam Wolf** Drawn By **Liam Wolf** Checked By **Calvin Chen** Liam April 13, 2021 12:21:07 AM February 24, 2021 11:15:24 PM	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: OmiBot Assembly SIZE **B** Assignment Number N/A REV 1
MATERIAL: VARIOUS			
FILE NAME: Model 2 - Pivot Tank Outside Config			SCALE: 1:12 Mass: 516284.190 SHEET 2 OF 2

ITEM NO.	DESCRIPTION	VENDOR	VENDOR PART NUMBER	QTY.
1	Modular Chassis Half	-	-	2
2	Bottom Modular Connector Plate	-	-	1
3	Manipulator Platform	-	-	2
4	Top Modular Connector Plate	-	-	1
5	Threaded Support Rod for Manipulator Platform	-	-	4
6	Pivot Shaft Housing	-	-	4
7	Threaded Support Rod for Manipulator	-	-	4
8	Black-Oxide Alloy Steel Hex Drive Flat Head Screw, 3/4"-10 Thread Size, 8" Long	McMaster-Carr	91253A867	6
9	High-Strength Steel Hex Nut, Grade 8, Black-Oxide, 3/4"-10 Thread Size	McMaster-Carr	94895A426	6
10	Forged Steel Hoist Ring for Lifting, 1/2"-13 Thread Size, 6-1/8" Overall Height	McMaster-Carr	2994T91	8
11	Black-Oxide Alloy Steel Socket Head Screw 10-24 Thread Size, 6-1/2" Long	McMaster-Carr	91251A904	4





MEC E 460
Instructor:
Dr. NAKASHIMA
WINTER 2021
Comments:
N/A

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

OMIKRON ROBOTICS Group 15

Modeled By: Liam Wolf
Drawn By: Liam Wolf
Checked By: George Felobes
Liam
April 12, 2021 12:15:48 AM
February 24, 2021 11:15:24 PM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

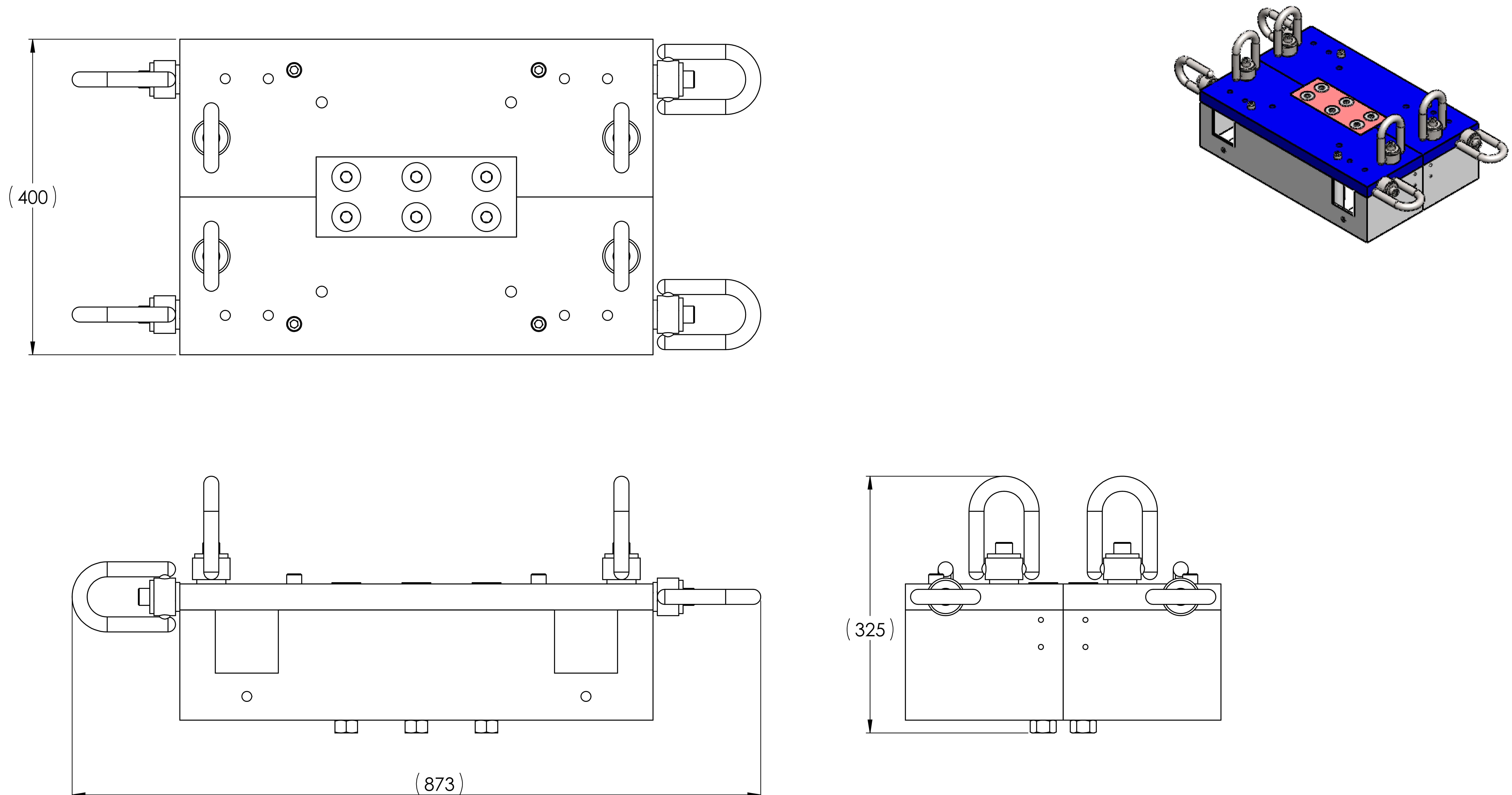
TITLE:
Modular Chassis Subassembly

MATERIAL:
VARIOUS

FILE NAME:
Modular Chassis Subassembly for Drawing

SIZE: B | Assignment Number: N/A | REV: 1

SCALE: 1:10 | Mass: 108053.020 | SHEET 1 OF 2



MEC E 460
Instructor: **Dr. NAKASHIMA**
WINTER 2021
Comments:
N/A

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:
VARIOUS

FILE NAME:
Modular Chassis Subassembly for Drawing

OMIKRON ROBOTICS Group 15

Modeled By	**Liam Wolf**
Drawn By	**Liam Wolf**
Checked By	**George Felobes**

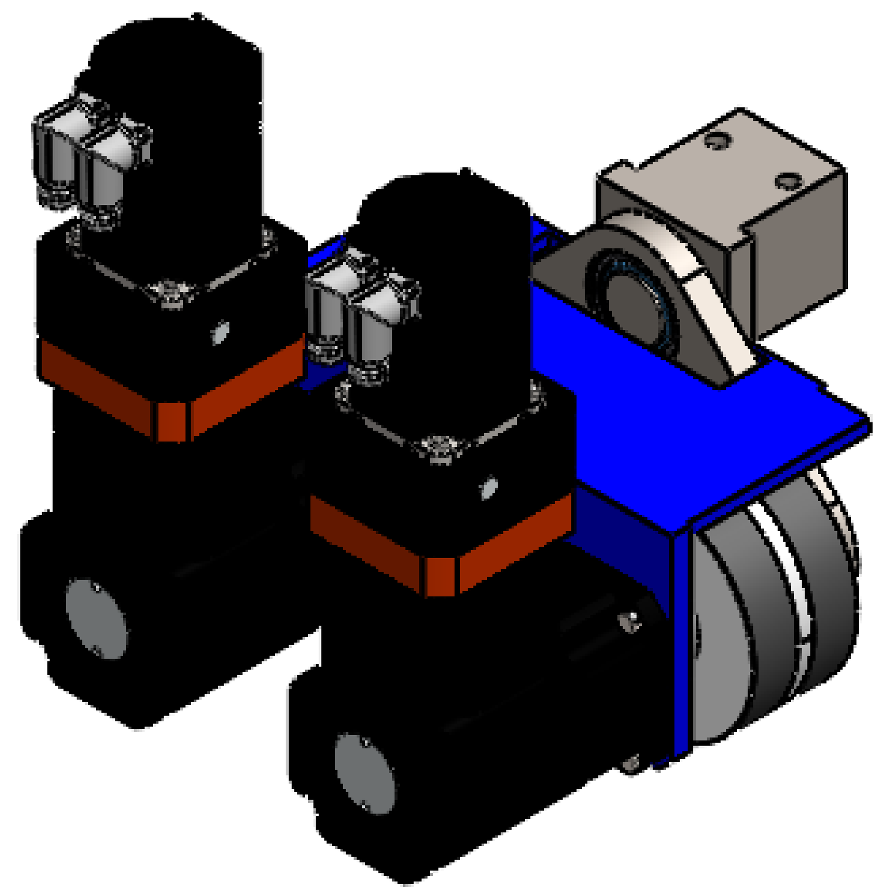
Liam
April 12, 2021 12:15:48 AM
February 24, 2021 11:15:24 PM

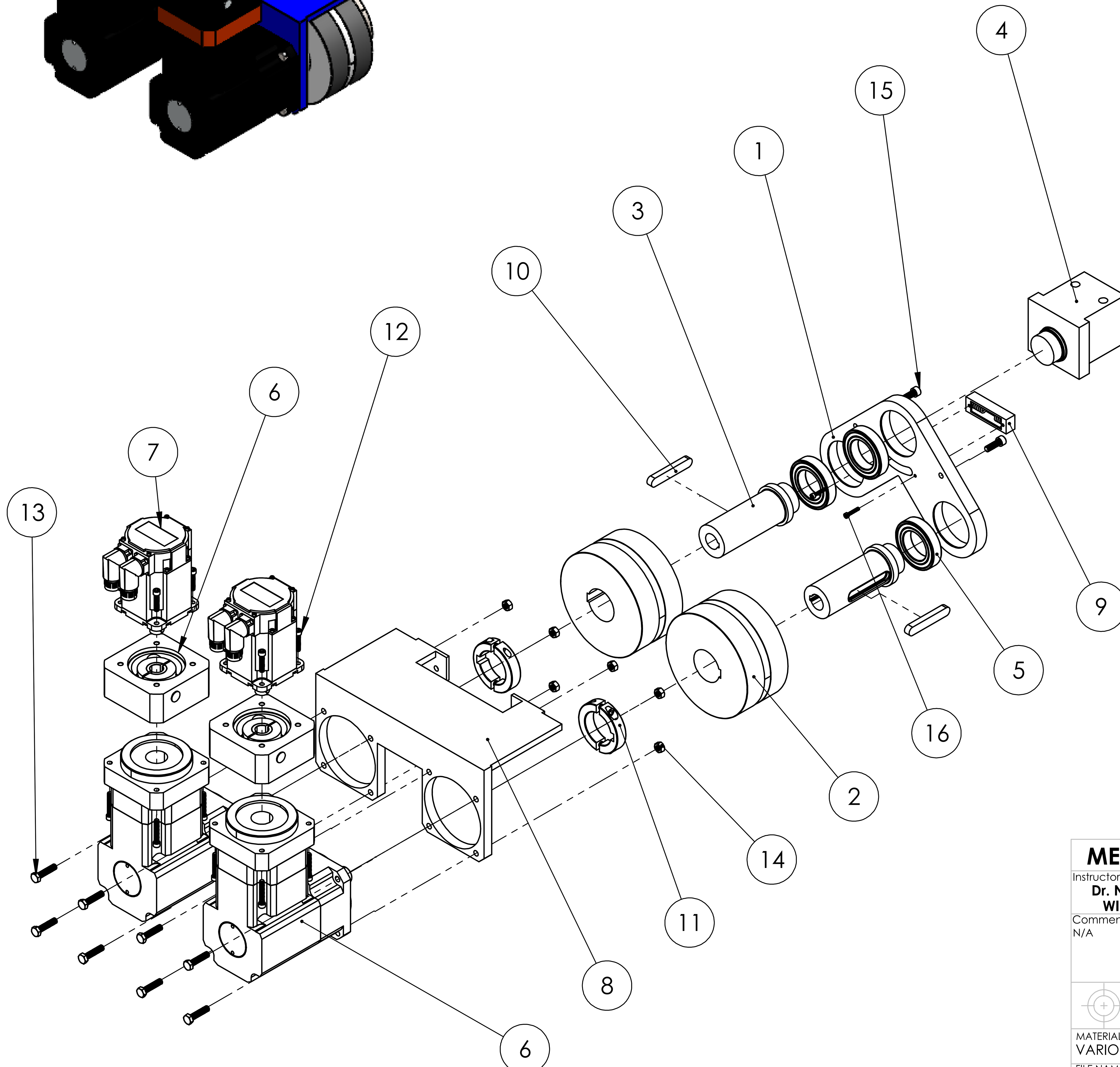
The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Modular Chassis Subassembly

SIZE	Assignment Number	REV
B	N/A	1

SCALE: 1:5 | Mass: 108053.020 | SHEET 2 OF 2





ITEM NO.	DESCRIPTION	VENDOR	VENDOR PART NUMBER	QTY.
1	V-Pivot	-	-	1
2	160 mm Magnetic Wheel	Magnetsysteme Brugger	HRZ160	2
3	Driveshaft	-	-	2
4	Fixed Shaft	-	-	1
5	Deep Groove Ball Bearing	Schaeffler	s6009-2rsr-fd	3
6	Gearhead and Redi-Mount Motor Mount	Kollmorgen	DTR115_120_0_RM115_40	2
7	Servo Motor	Kollmorgen	AKM2G_41X_ACCNR_00	2
8	Gearhead Housing	-	-	1
9	Linear Spring Box	-	-	1
10	Rounded Machine Key, 1018-1045 Carbon Steel, 9 mm x 14 mm, 80 mm Long	McMaster-Carr	96717A631	2
11	Clamping Two-Piece Shaft Collar with Keyway, for 2" Diameter, 303 Stainless Steel	McMaster-Carr	3329K18	2
12	Black-Oxide Alloy Steel Socket Head Screw 1/4"-20 Thread Size, 1-1/8" Long	McMaster-Carr	91251A560	8
13	Black-Oxide Alloy Steel Socket Head Screw M8 x 1.25 mm Thread, 35 mm Long	McMaster-Carr	91290A438	8
14	Steel Hex Nut Medium-Strength, Class 8, M8 x 1.25 mm Thread	McMaster-Carr	90592A022	8
15	Black-Oxide Alloy Steel Socket Head Screw M8 x 1.25 mm Thread, 25 mm Long	McMaster-Carr	91290A432	2
16	Black-Oxide Alloy Steel Socket Head Screw, M4 x 0.7 mm Thread, 22 mm Long	McMaster-Carr	91290A170	2

MEC E 460

Instructor:
Dr. NAKASHIMA
WINTER 2021

Comments:
N/A

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:
VARIOUS

FILE NAME:
Drive Assembly outside Config

OMIKRON ROBOTICS Group 15

Modeled By: **Liam Wolf**
Drawn By: **Liam Wolf**
Checked By: **Kenny Okeke**

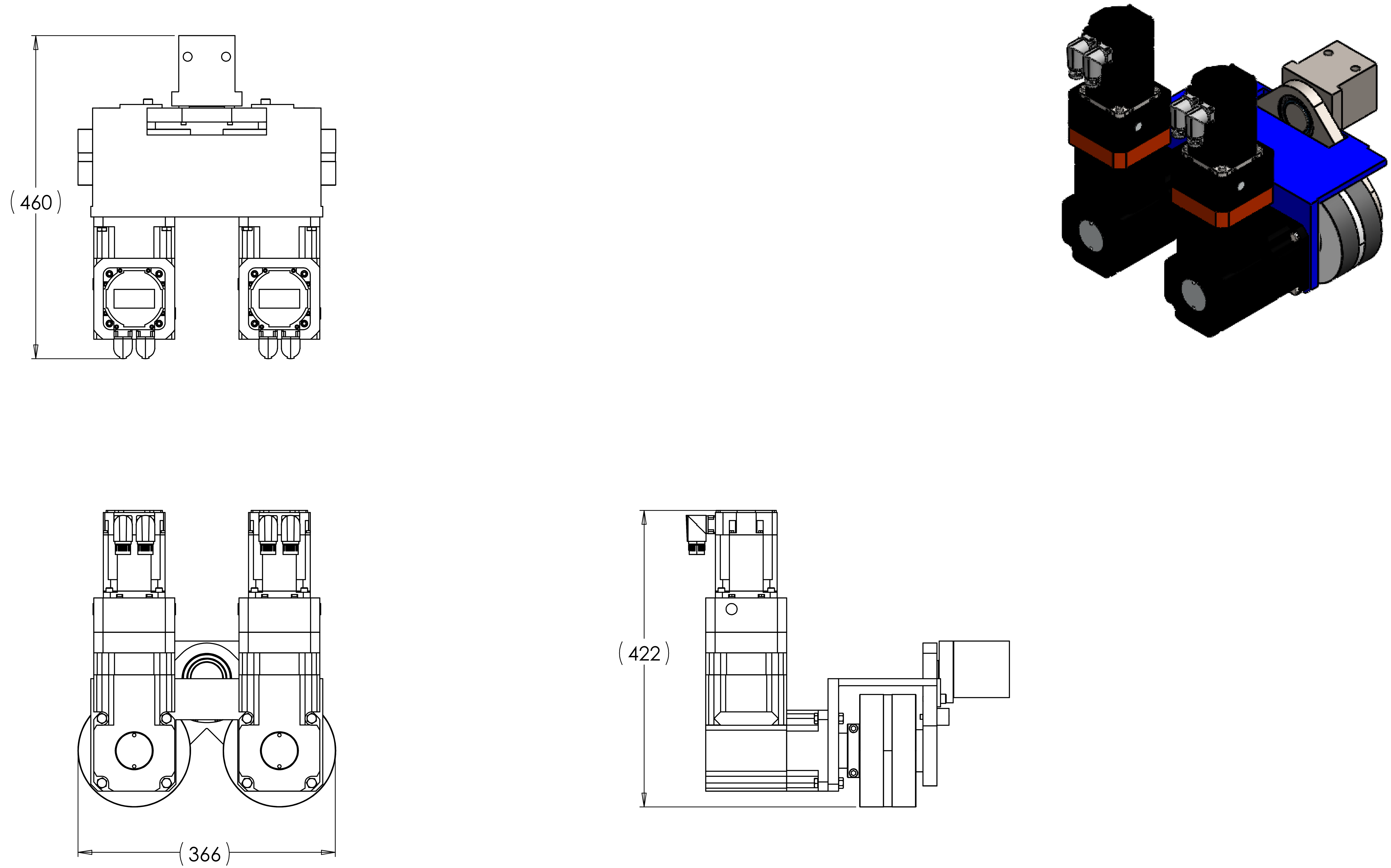
Liam
April 14, 2021 2:13:38 AM
March 9, 2021 12:35:30 AM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Modular Drive Subassembly

SIZE: **B** | Assignment Number: N/A | REV: 1

SCALE: 1:7 | Mass: 69047.460 | SHEET 1 OF 2



MEC E 460
Instructor:
Dr. NAKASHIMA
WINTER 2021
Comments:
N/A

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

OMIKRON ROBOTICS Group 15

Modeled By	Liam Wolf
Drawn By	Liam Wolf
Checked By	Kenny Okeke

Liam
April 14, 2021 2:13:38 AM
March 9, 2021 12:35:30 AM

MATERIAL:
VARIOUS

FILE NAME:
Drive Assembly outside Config

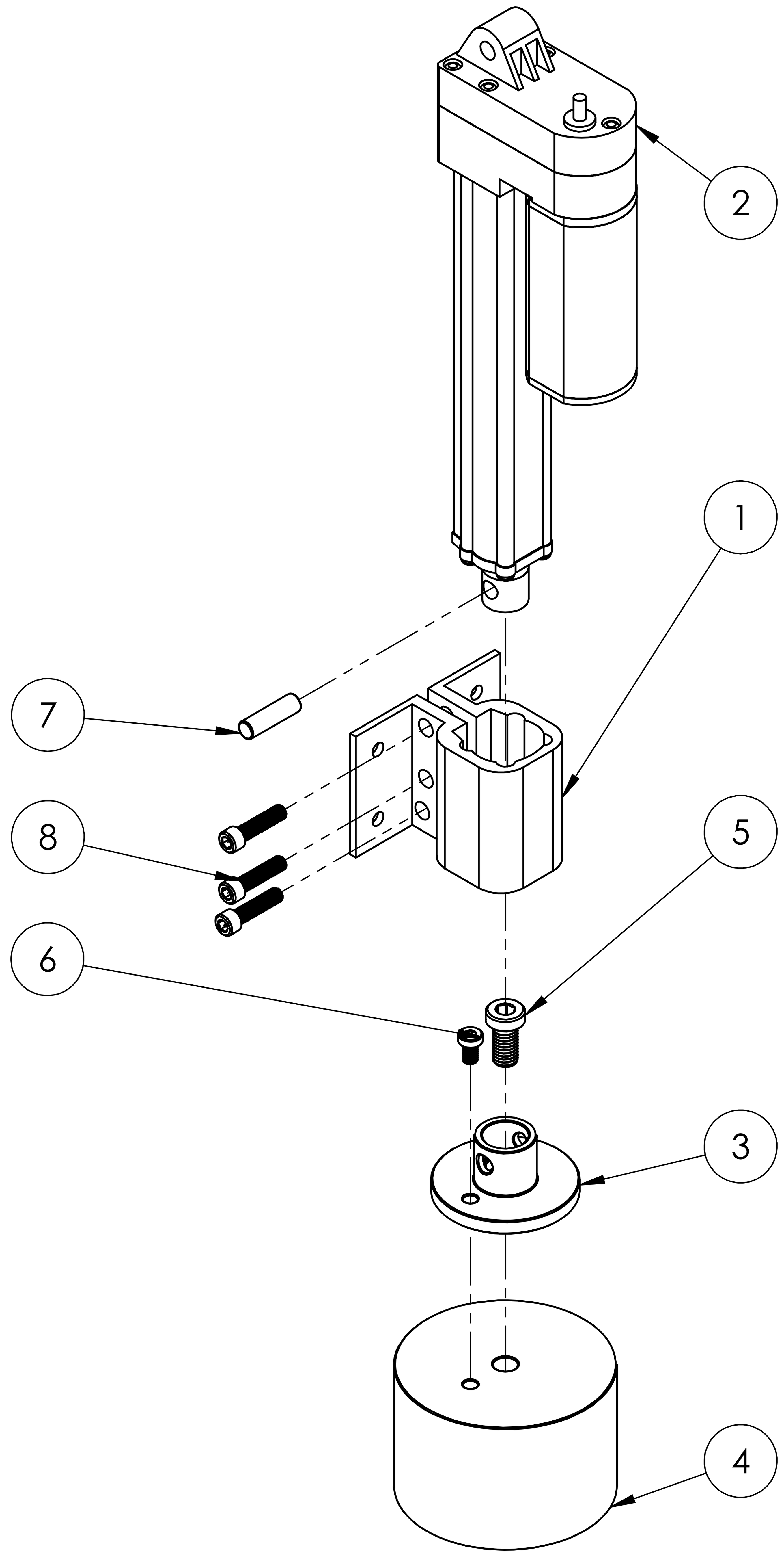
The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

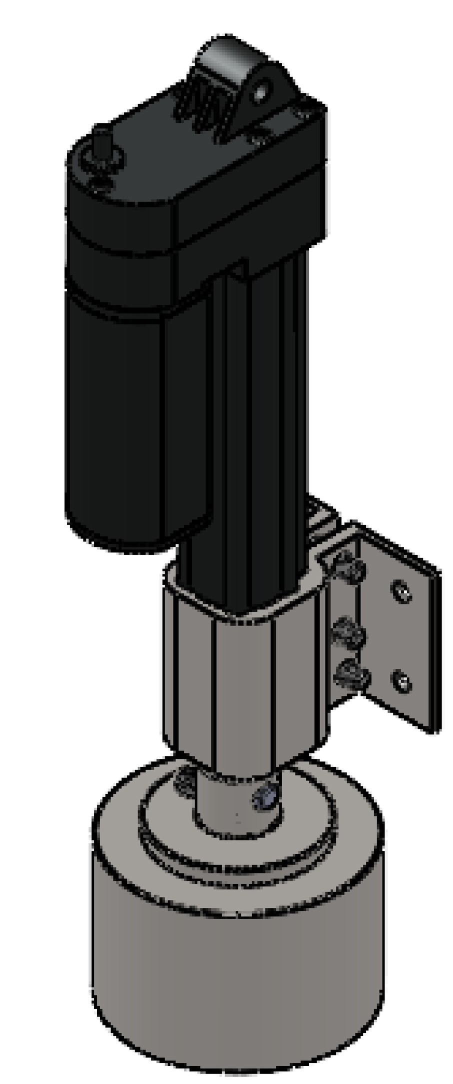
TITLE:
Modular Drive Subassembly

SIZE	Assignment Number	REV
B	N/A	1

SCALE: 1:6 | Mass: 69047.460 | SHEET 2 OF 2

ITEM NO.	DESCRIPTION	VENDOR	VENDOR PART NUMBER	QTY.
1	Shaft Mounting Bracket for PA-09	Progressive Automations	BRK-10	1
2	Mini Industrial Actuator	Progressive Automations	PA-09	1
3	Linear Actuator Connection	-	-	1
4	Kanetec Electro Magnetic Holder	Kanetec	KEP-C9 - Magnet	1
5	Super-Corrosion-Resistant 316 Stainless Steel Socket Head Screw, Low-Profile, M10 x 1.5 mm Thread, 20 mm Long	McMaster-Carr	90666A144	1
6	Super-Corrosion-Resistant 316 Stainless Steel Socket Head Screw, Low-Profile, M6 x 1 mm Thread, 10 mm Long	McMaster-Carr	90666A130	1
7	Linear Actuator Pin Connector	-	-	1
8	Black-Oxide Alloy Steel Socket Head Screw, 1/4"-28 Thread Size, 1" Long	McMaster-Carr	91251A442	3





MEC E 460

Instructor: Dr. NAKASHIMA WINTER 2021

Comments: N/A

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

OMIKRON ROBOTICS Group 15

Modeled By: Liam Wolf
Drawn By: Liam Wolf
Checked By: Eric Wong

Liam
April 13, 2021 12:21:08 AM
March 16, 2021 2:16:30 PM

The Department of Mechanical Engineering UNIVERSITY OF ALBERTA

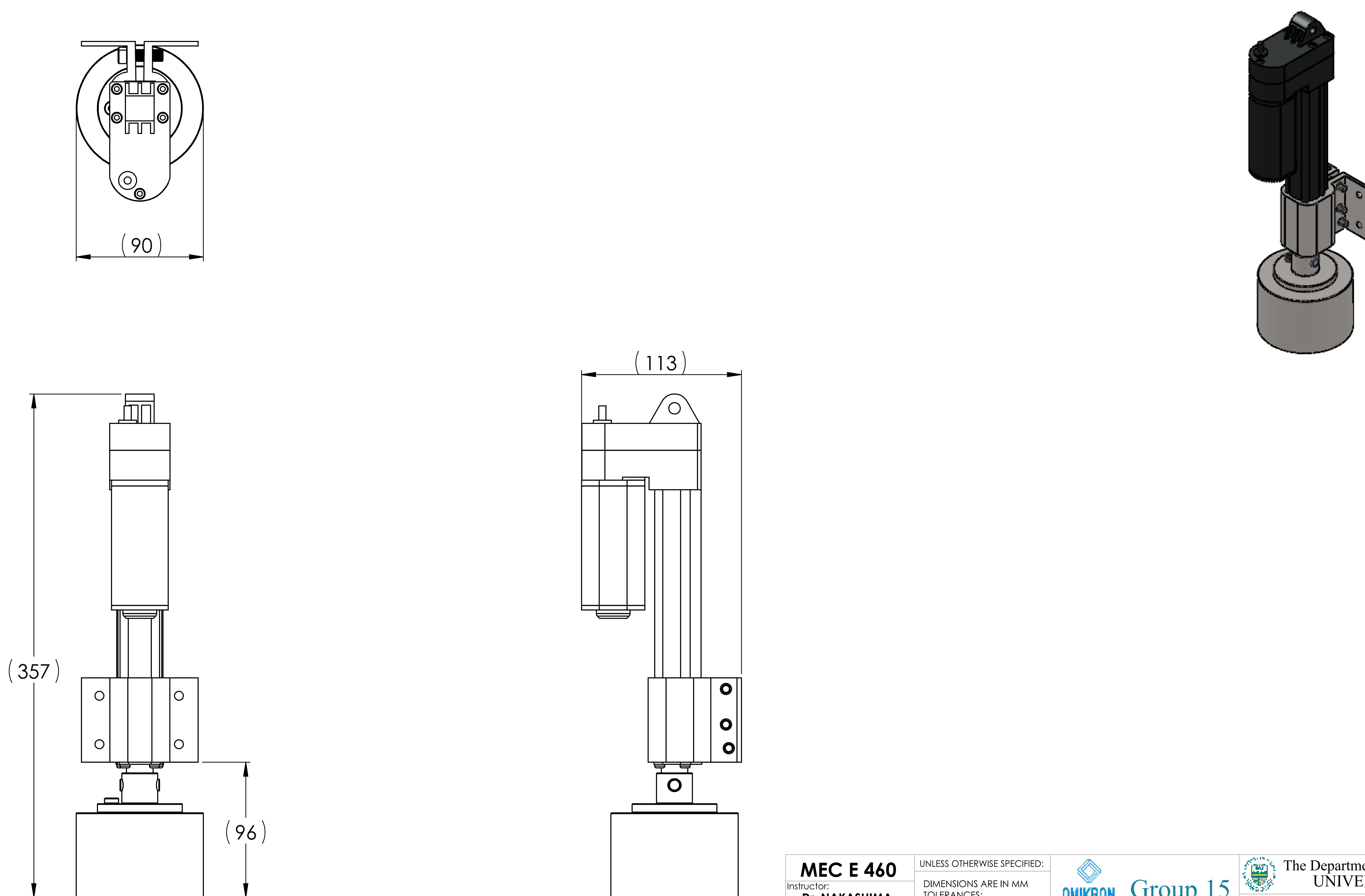
TITLE: Modular Linear Actuator Subassembly

MATERIAL: VARIOUS

FILE NAME: Linear Actuator

SIZE: B | Assignment Number: N/A | REV: 1

SCALE: 1:3 | Mass: 4167.13 | SHEET 1 OF 2



MEC E 460
Instructor:
Dr. NAKASHIMA
WINTER 2021
Comments:
N/A

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:
VARIOUS
FILE NAME:
Linear Actuator

OMIKRON ROBOTICS Group 15

Modeled By	**Liam Wolf**
Drawn By	**Liam Wolf**
Checked By	**Eric Wong**

Liam
April 13, 2021 12:21:08 AM
March 16, 2021 2:16:30 PM

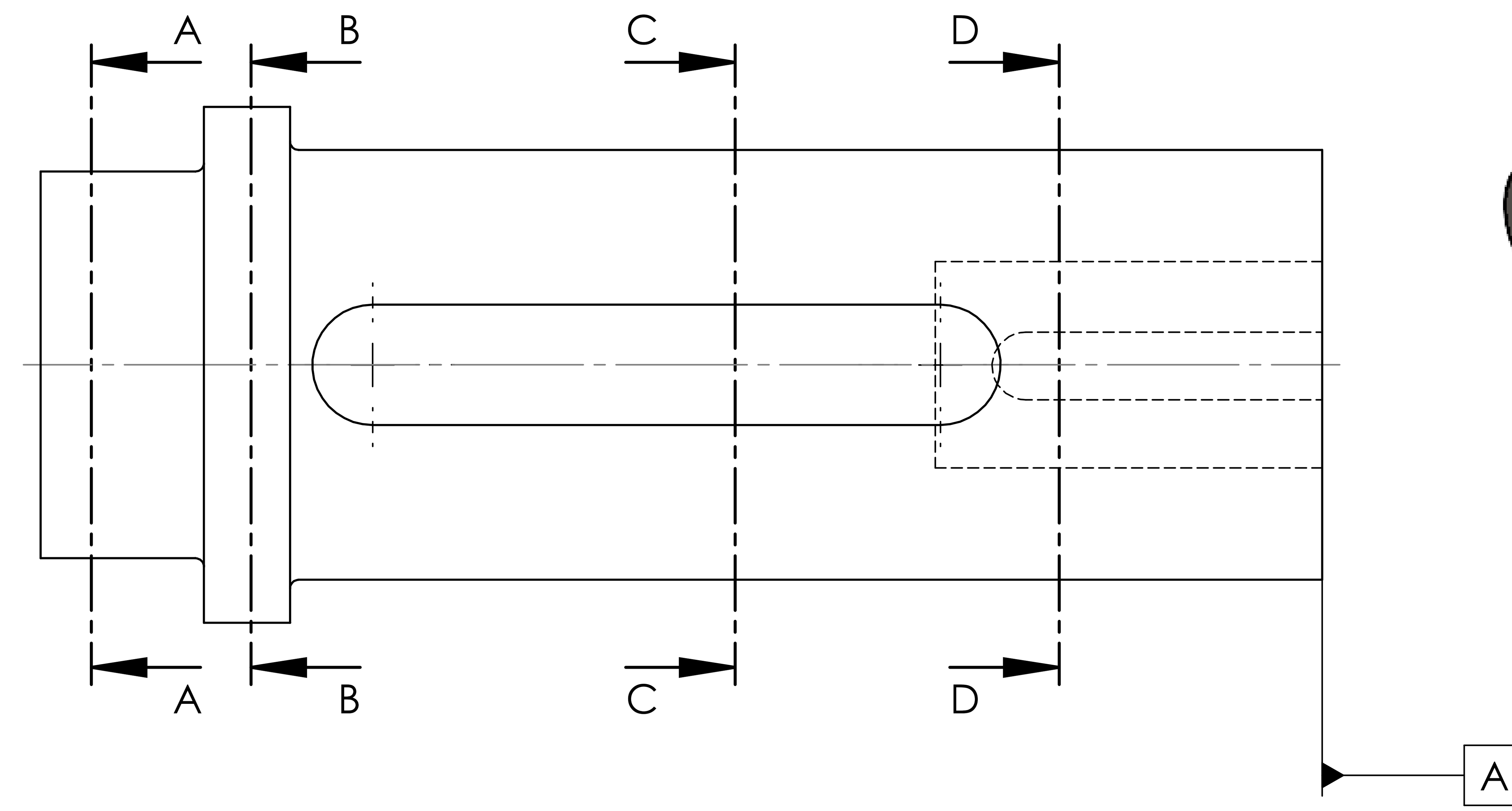
The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

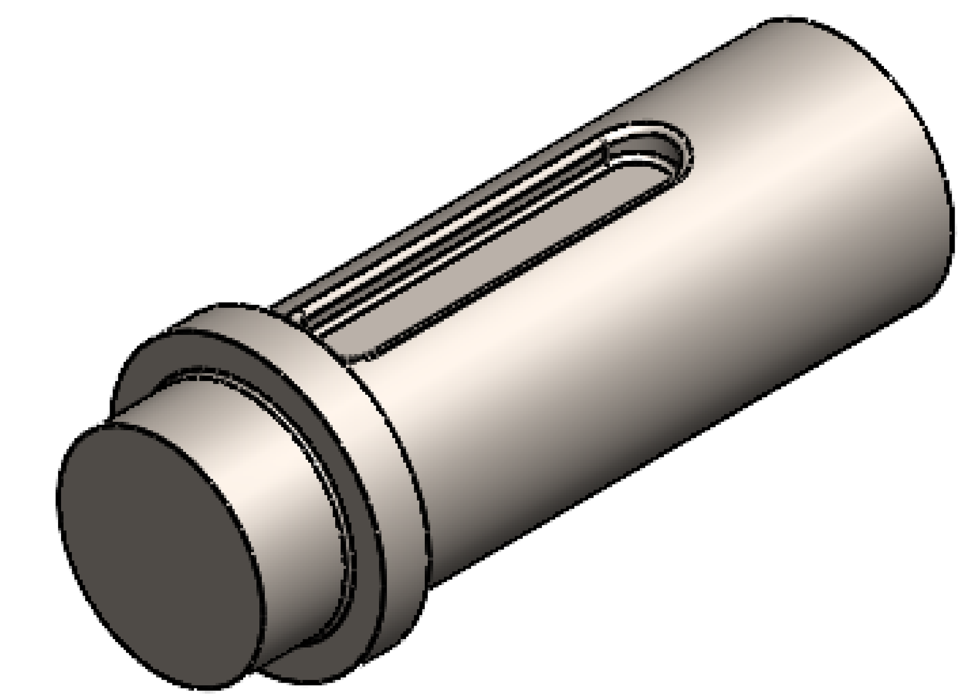
TITLE:
Modular Linear Actuator Subassembly

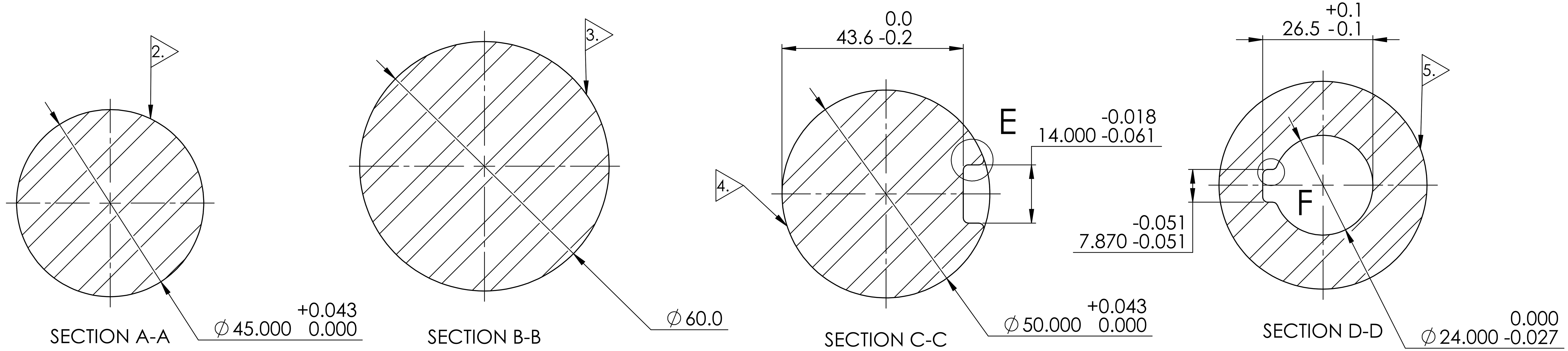
SIZE	Assignment Number	REV
B	**N/A**	**1**

SCALE: 1:3 | Mass: 4167.13 | SHEET 2 OF 2

NOTES:
1. REMOVE ALL BURS AND BREAK SHARP EDGES
2. A-A FITS INTO BEARING ID (INTERFERENCE FIT)
3. B-B IS A LOCATING STEP
4. C-C CONNECTS DRIVESHAFT TO WHEELS (INTERFERENCE FIT)
5. D-D ID CONNECTS DRIVESHAFT TO GEARHEAD (INTERFERENCE FIT)







MEC E 460
Instructor:
Dr. NAKASHIMA
WINTER 2021
Comments:
N/A

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

OMIKRON ROBOTICS Group 15

Modeled By **Calvin Chen**
Drawn By **Liam Wolf**
Checked By **Calvin Chen**
Liam
April 12, 2021 10:14:35 PM
March 14, 2021 7:05:48 PM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Driveshaft

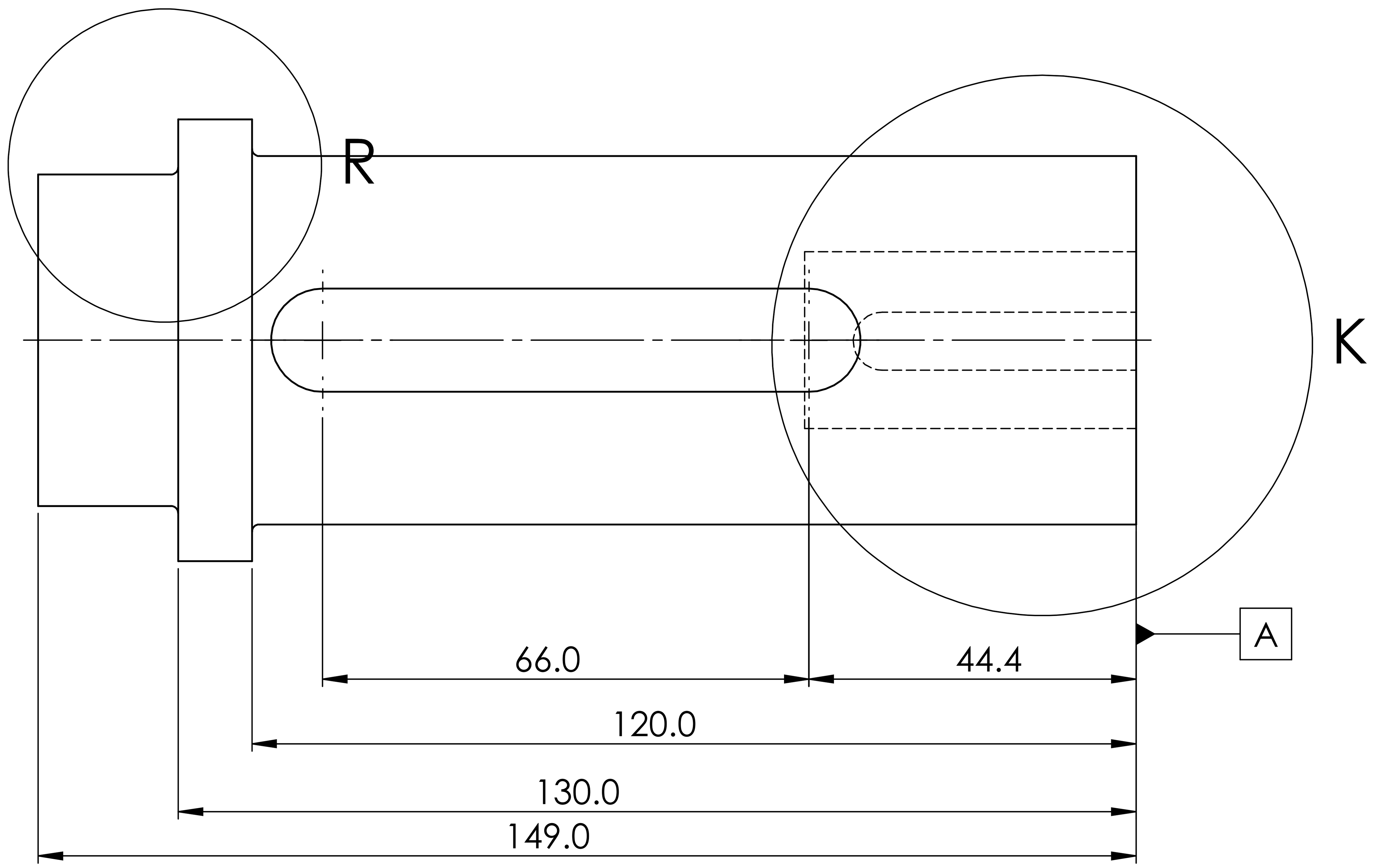
MATERIAL:
AISI 4130 Steel, normalized at 870C

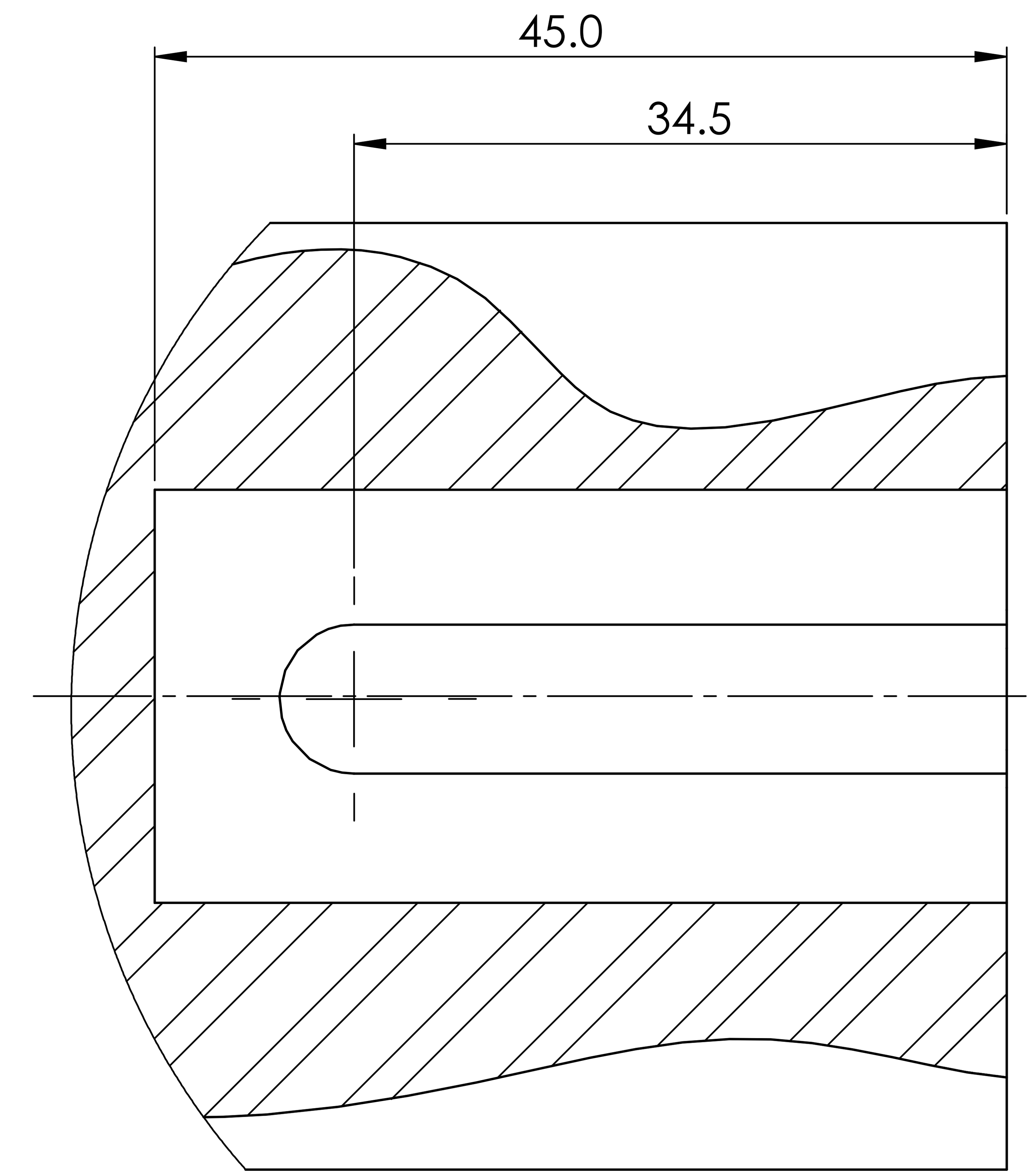
FILE NAME:
Driveshaft

SIZE **B** Assignment Number N/A REV 1

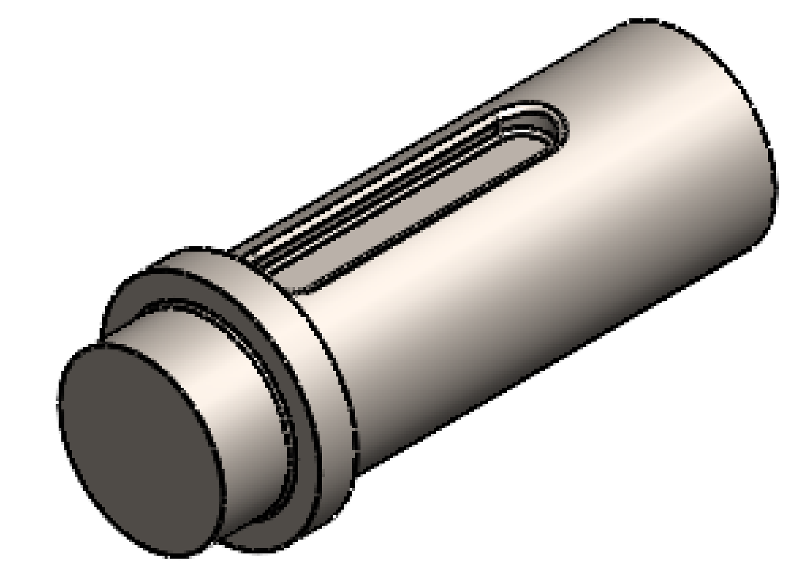
SCALE: 1:5 Mass: 2091.362 SHEET 1 OF 2

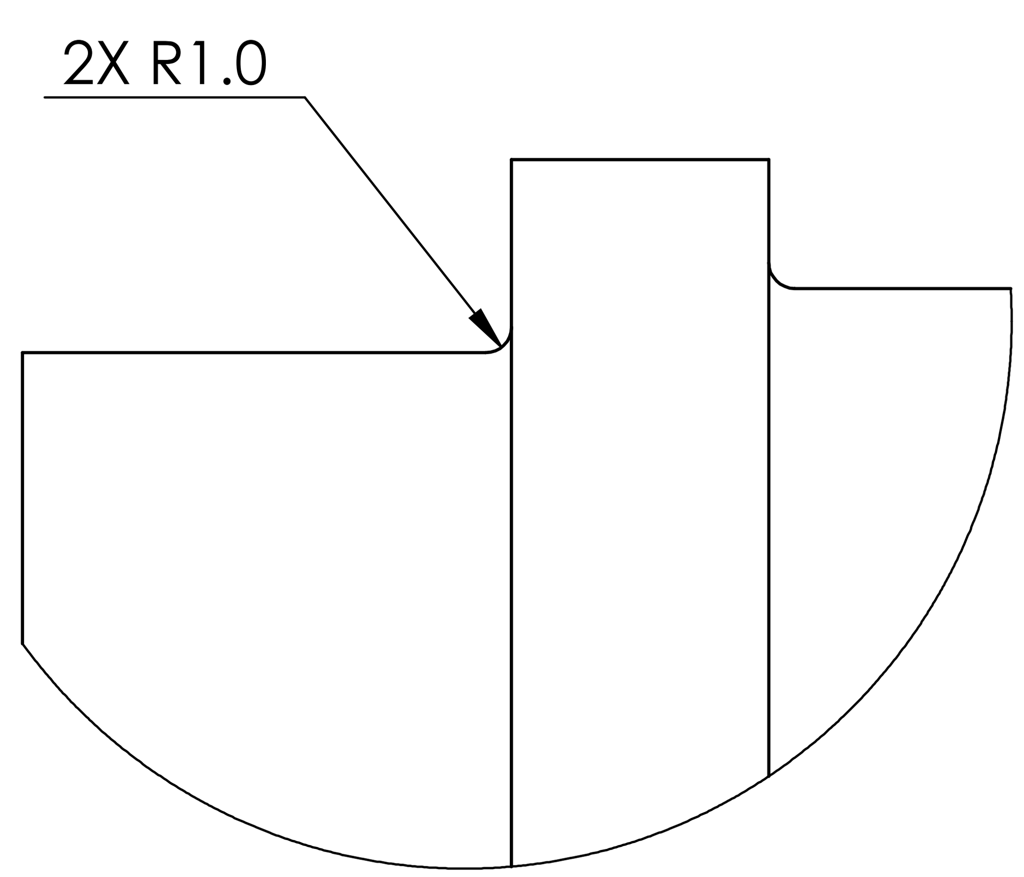
NOTES:
1. REMOVE ALL BURS AND BREAK SHARP EDGES





DETAIL K
SCALE 2 : 1





DETAIL R
SCALE 2 : 1

MEC E 460
Instructor:
Dr. NAKASHIMA
WINTER 2021
Comments:
N/A

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

OMIKRON ROBOTICS Group 15

Modeled By	**Calvin Chen**
Drawn By	**Liam Wolf**
Checked By	**Calvin Chen**

Liam
April 12, 2021 10:14:35 PM
March 14, 2021 7:05:48 PM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Driveshaft

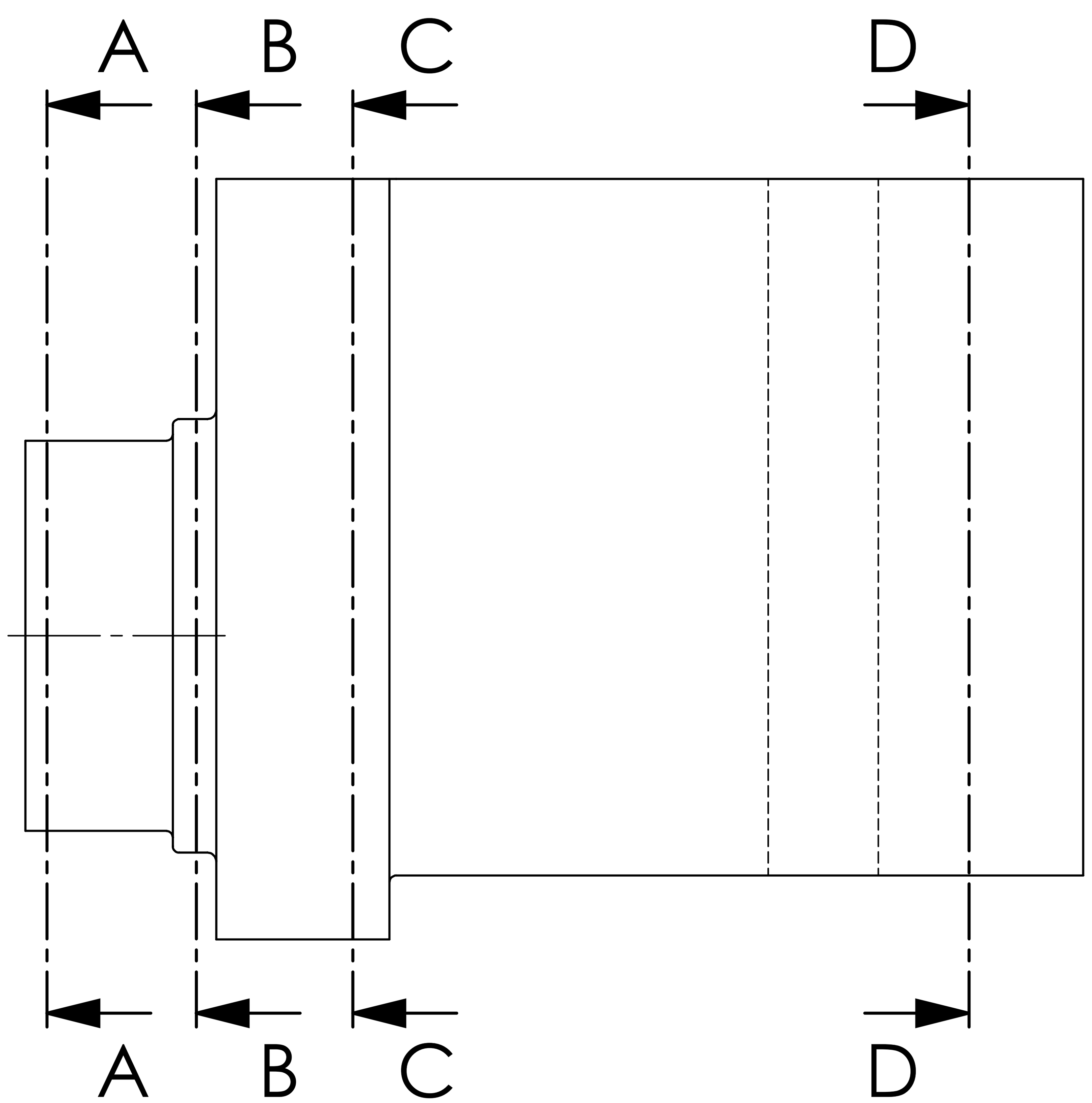
MATERIAL:
AISI 4130 Steel, normalized at 870C

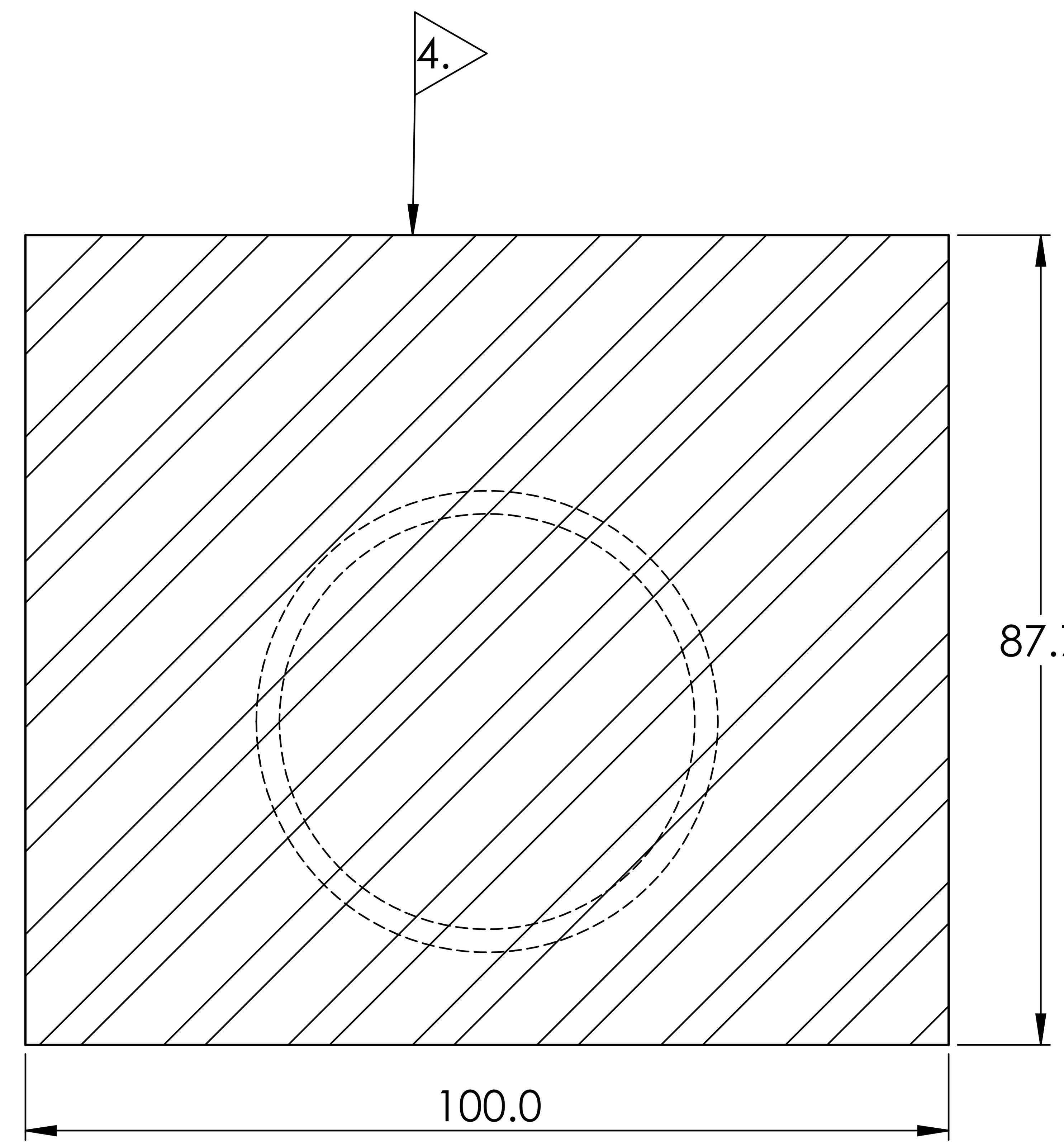
FILE NAME:
Driveshaft

SIZE	Assignment Number	REV
B	N/A	1

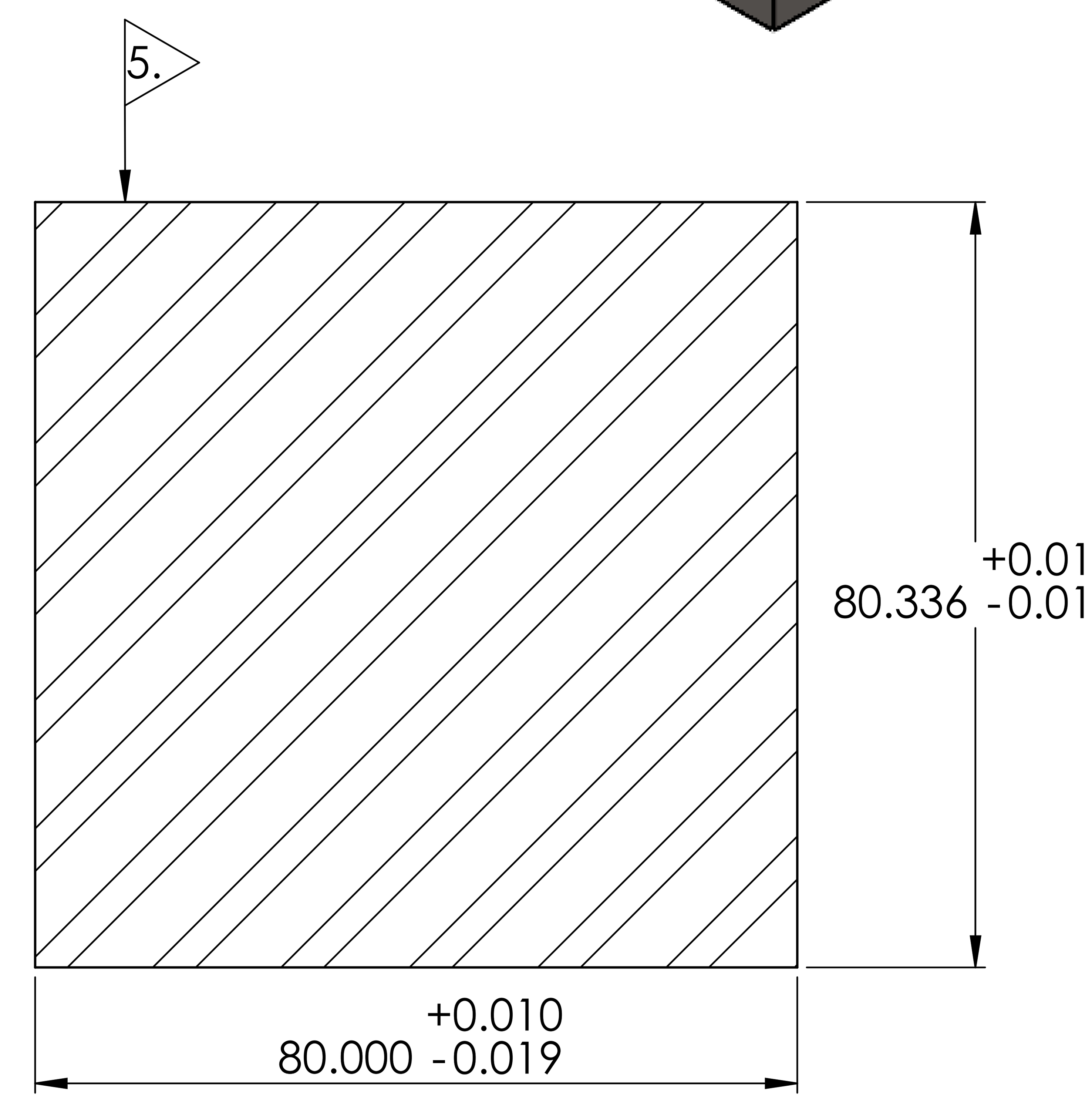
SCALE: 1:1 Mass: 2091.362 SHEET 2 OF 2

NOTES:
1. REMOVE ALL BURS AND BREAK SHARP EDGES
2. A-A FITS INTO BEARING ID (INTERFERENCE FIT)
3. B-B IS A LOCATING STEP
4. C-C IS A LOCATING STEP
5. D-D CONNECTS THE FIXED SHAFT TO THE CHASSIS (TRANSITION FIT)

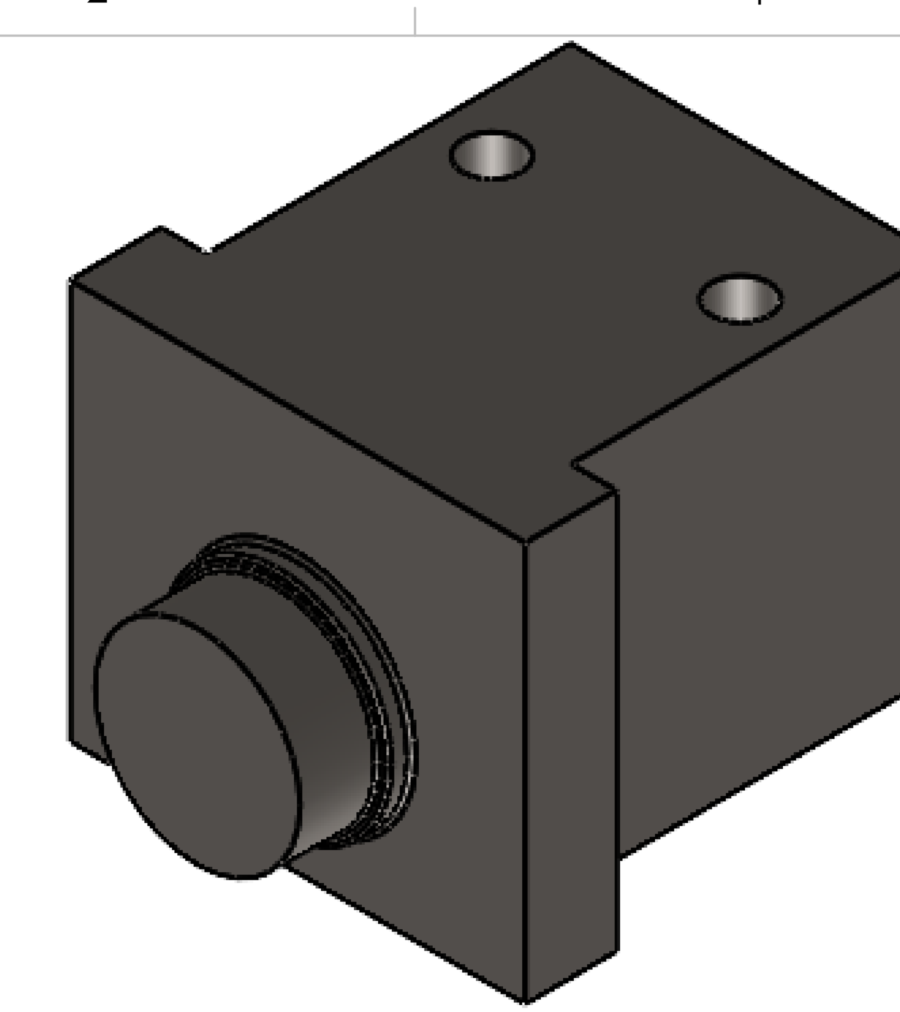


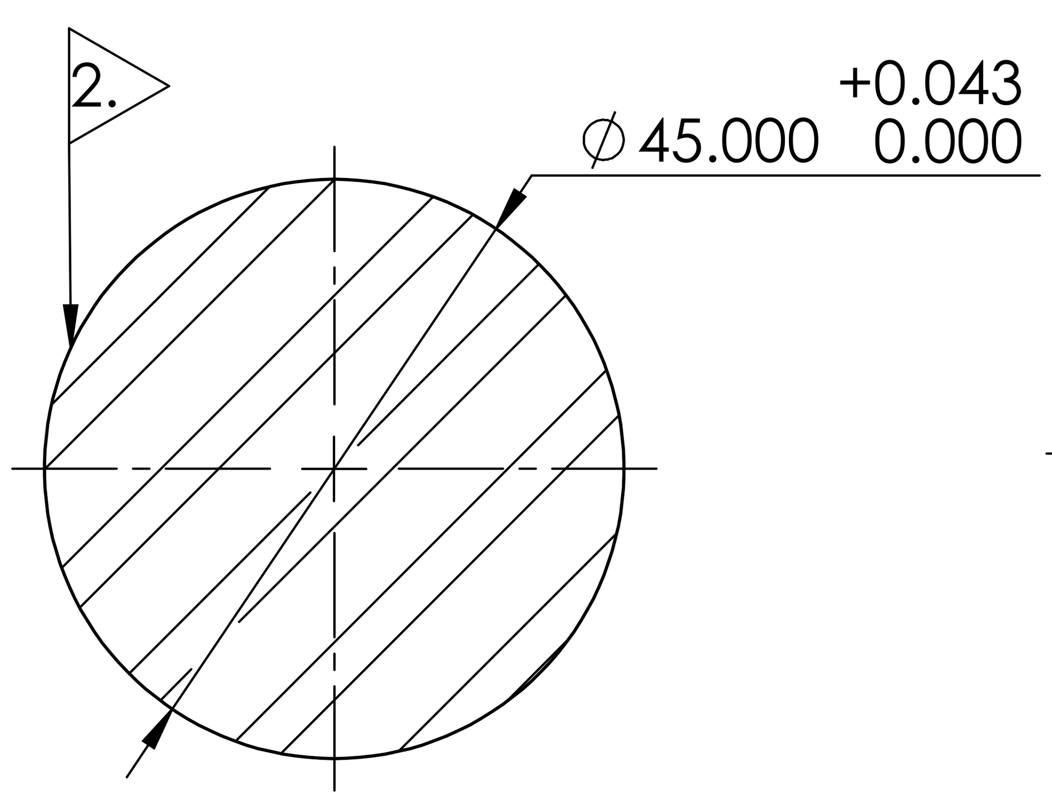


SECTION C-C

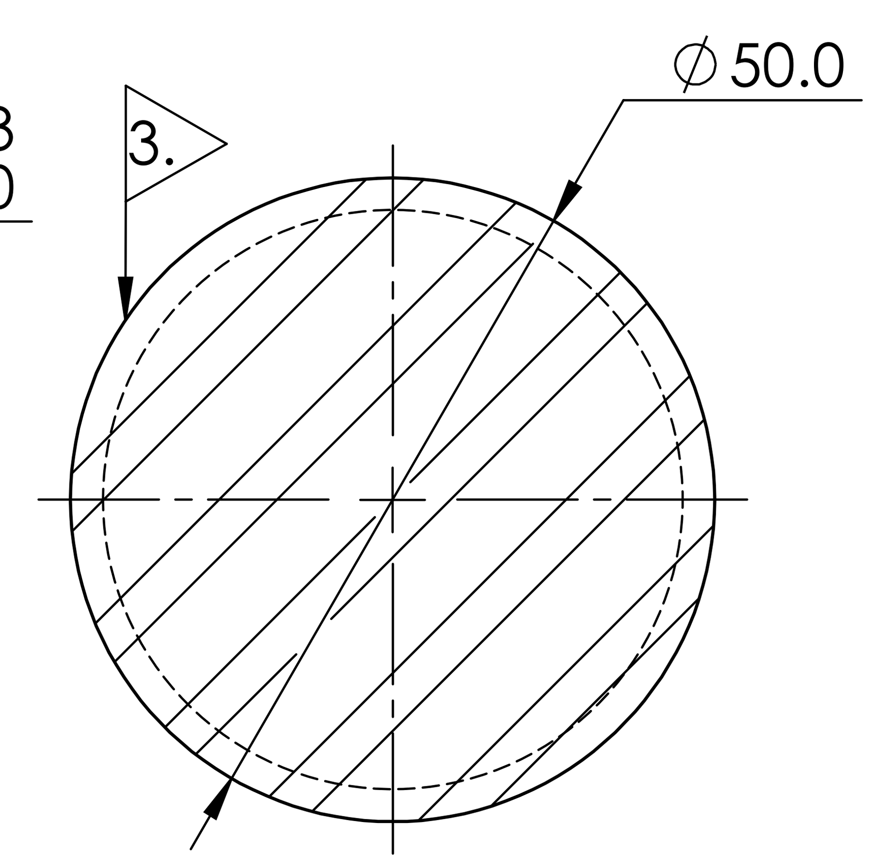


SECTION D-D





SECTION A-A



SECTION B-B

MEC E 460
Instructor:
Dr. NAKASHIMA
WINTER 2021
Comments:
N/A

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

OMIKRON ROBOTICS — Group 15

Modeled By	**Calvin Chen**
Drawn By	**Liam Wolf**
Checked By	**George Felobes**

Liam
April 13, 2021 12:13:34 AM
March 15, 2021 12:37:29 PM

MATERIAL:
AISI 4130 Steel, normalized at 870C

FILE NAME:
Fixed Shaft

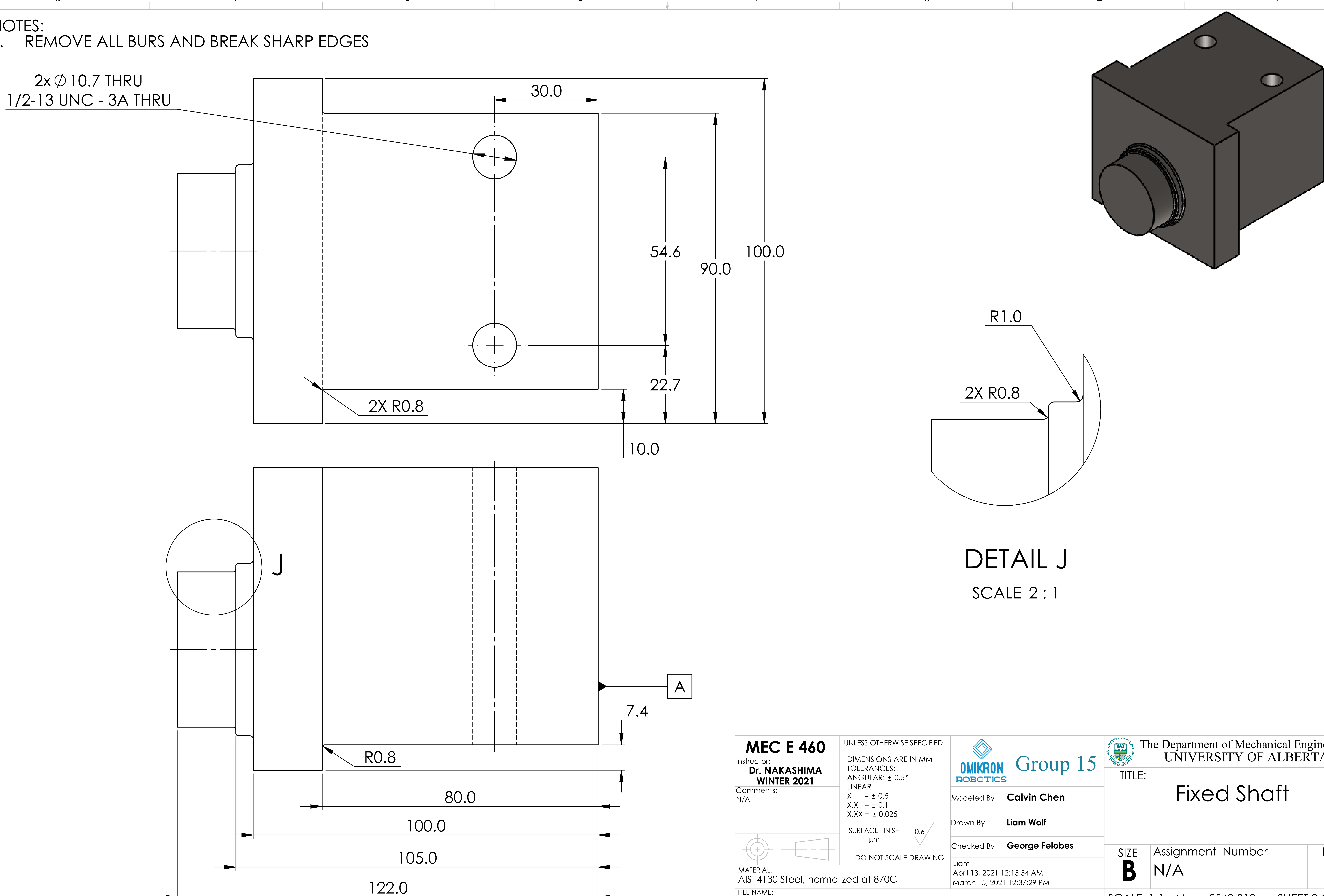
The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Fixed Shaft

SIZE	Assignment Number	REV
B	N/A	1

SCALE: 1:1 | Mass: 5543.013 | SHEET 1 OF 2

NOTES:
1. REMOVE ALL BURS AND BREAK SHARP EDGES



MEC E 460
Instructor:
Dr. NAKASHIMA
WINTER 2021
Comments:
N/A

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

OMIKRON ROBOTICS Group 15

Modeled By	**Calvin Chen**
Drawn By	**Liam Wolf**
Checked By	**George Felobes**

Liam
April 13, 2021 12:13:34 AM
March 15, 2021 12:37:29 PM

MATERIAL:
AISI 4130 Steel, normalized at 870C

FILE NAME:
Fixed Shaft

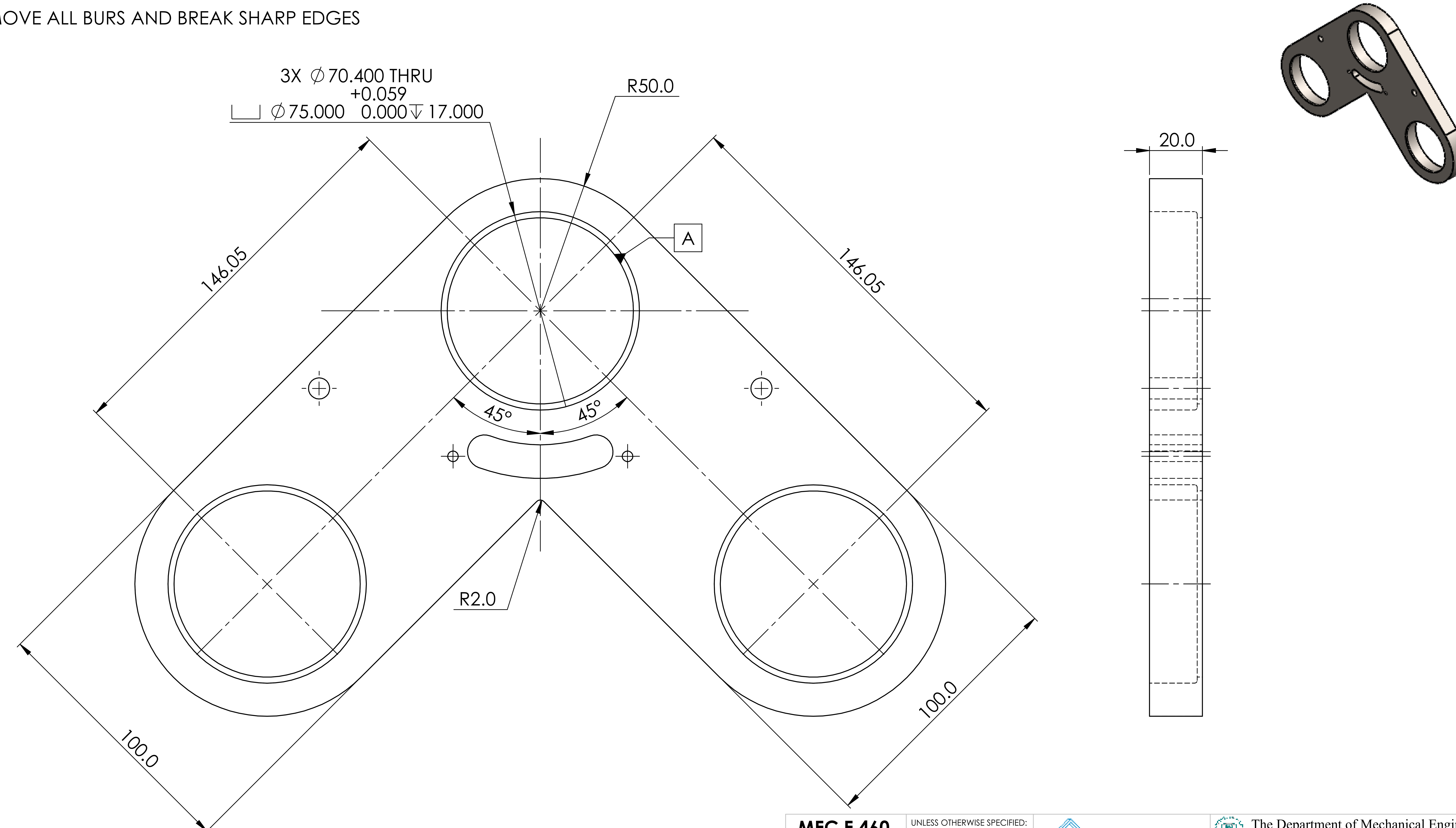
The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Fixed Shaft

SIZE **B** | Assignment Number N/A | REV 1

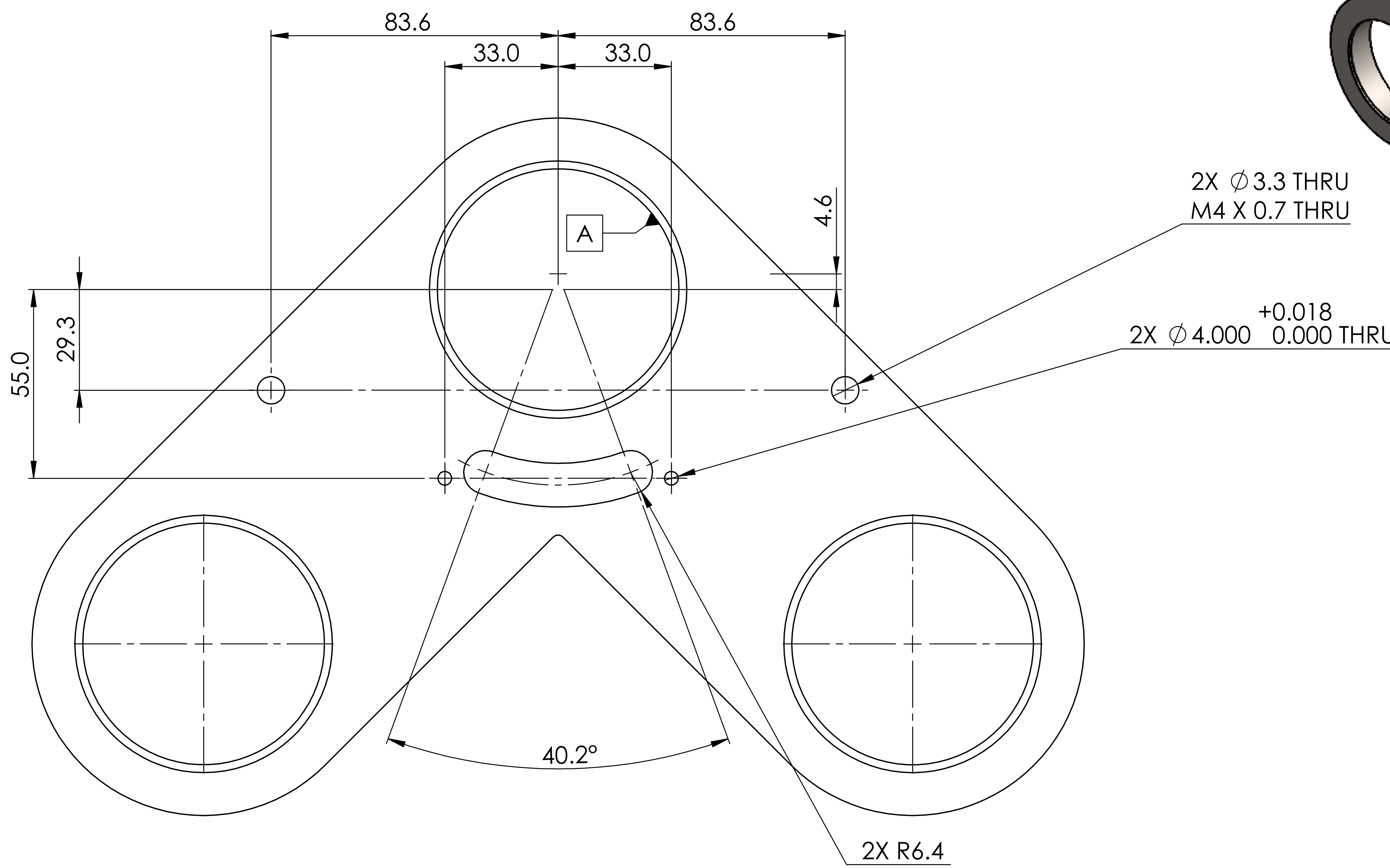
SCALE: 1:1 | Mass: 5543.013 | SHEET 2 OF 2

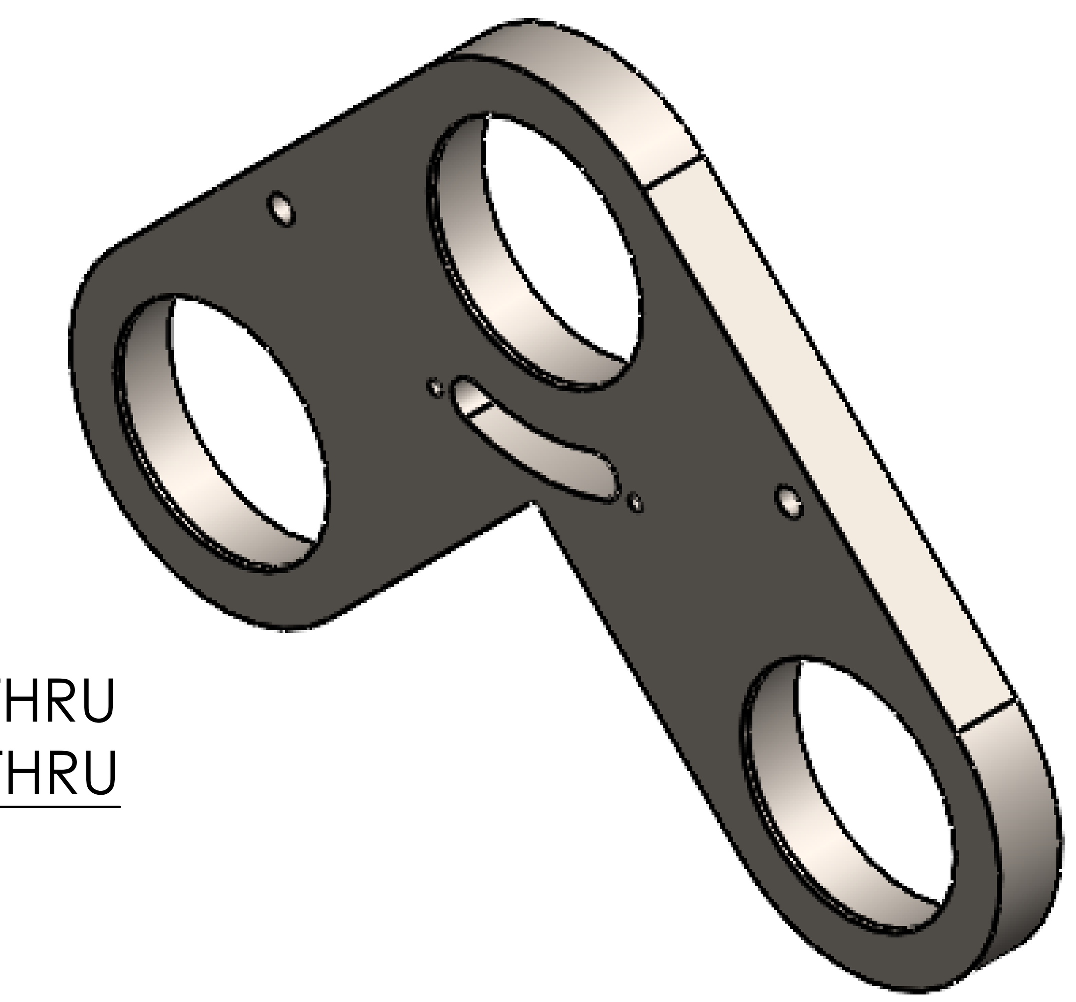
NOTES:
1. REMOVE ALL BURS AND BREAK SHARP EDGES



MEC E 460 Instructor: Dr. NAKASHIMA WINTER 2021 Comments: N/A	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	OMIKRON ROBOTICS Group 15 Modeled By: Liam Wolf Drawn By: Liam Wolf Checked By: Eric Wong Liam April 13, 2021 12:01:24 AM March 9, 2021 12:30:52 AM	The Department of Mechanical Engineering UNIVERSITY OF ALBERTA TITLE: V-Pivot SIZE B Assignment Number N/A REV 1
MATERIAL: AISI 4130 Steel, normalized at 870C			SCALE: 1:1.5 Mass: 3554.390
FILE NAME: V Bracket			SHEET 1 OF 2

NOTES:
1. REMOVE ALL BURS AND BREAK SHARP EDGES





MEC E 460
Instructor:
Dr. NAKASHIMA
WINTER 2021
Comments:
N/A

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

OMIKRON ROBOTICS Group 15

Modeled By	Liam Wolf
Drawn By	Liam Wolf
Checked By	Eric Wong

Liam
April 13, 2021 12:01:24 AM
March 9, 2021 12:30:52 AM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
V-Pivot

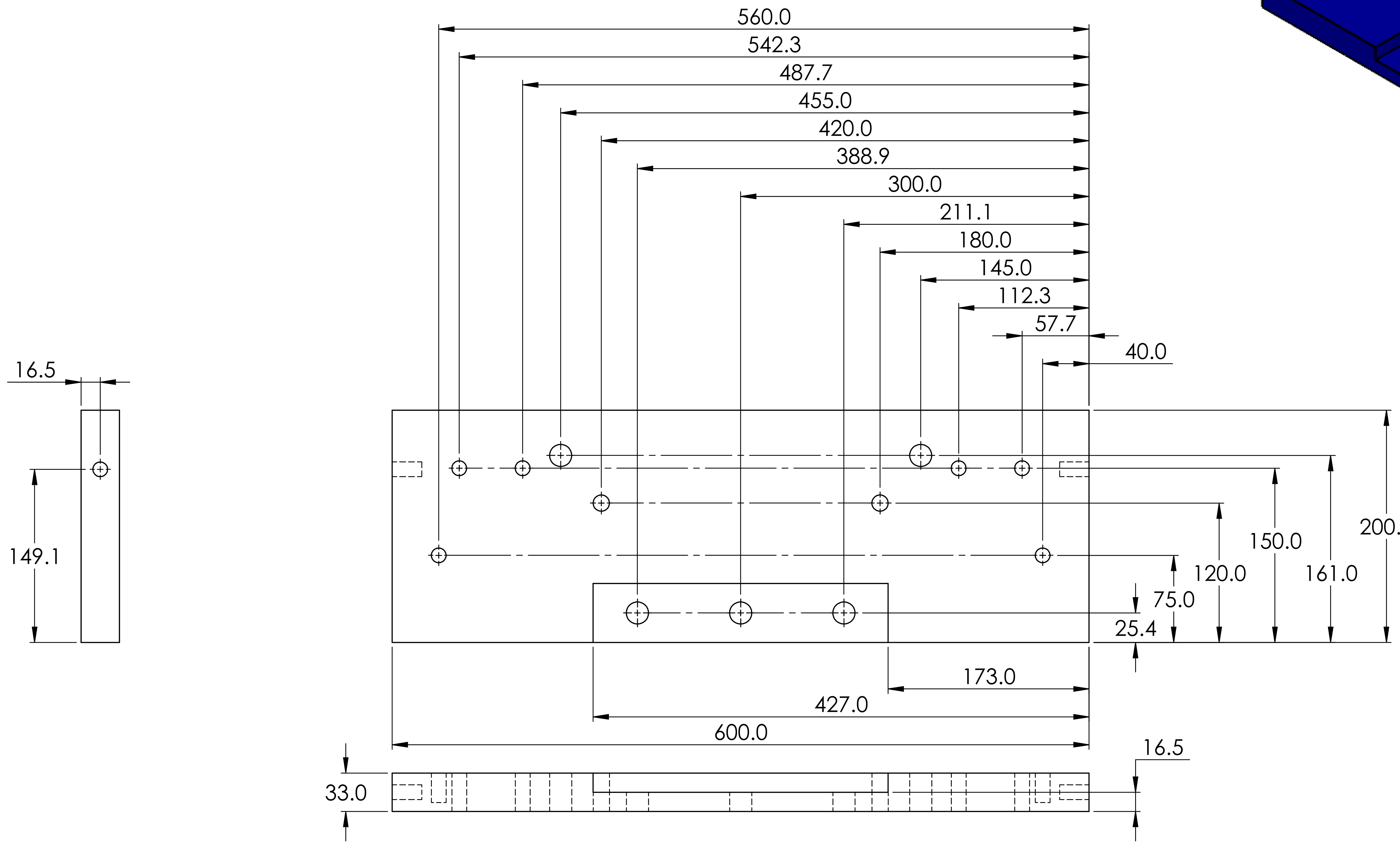
MATERIAL:
AISI 4130 Steel, normalized at 870C

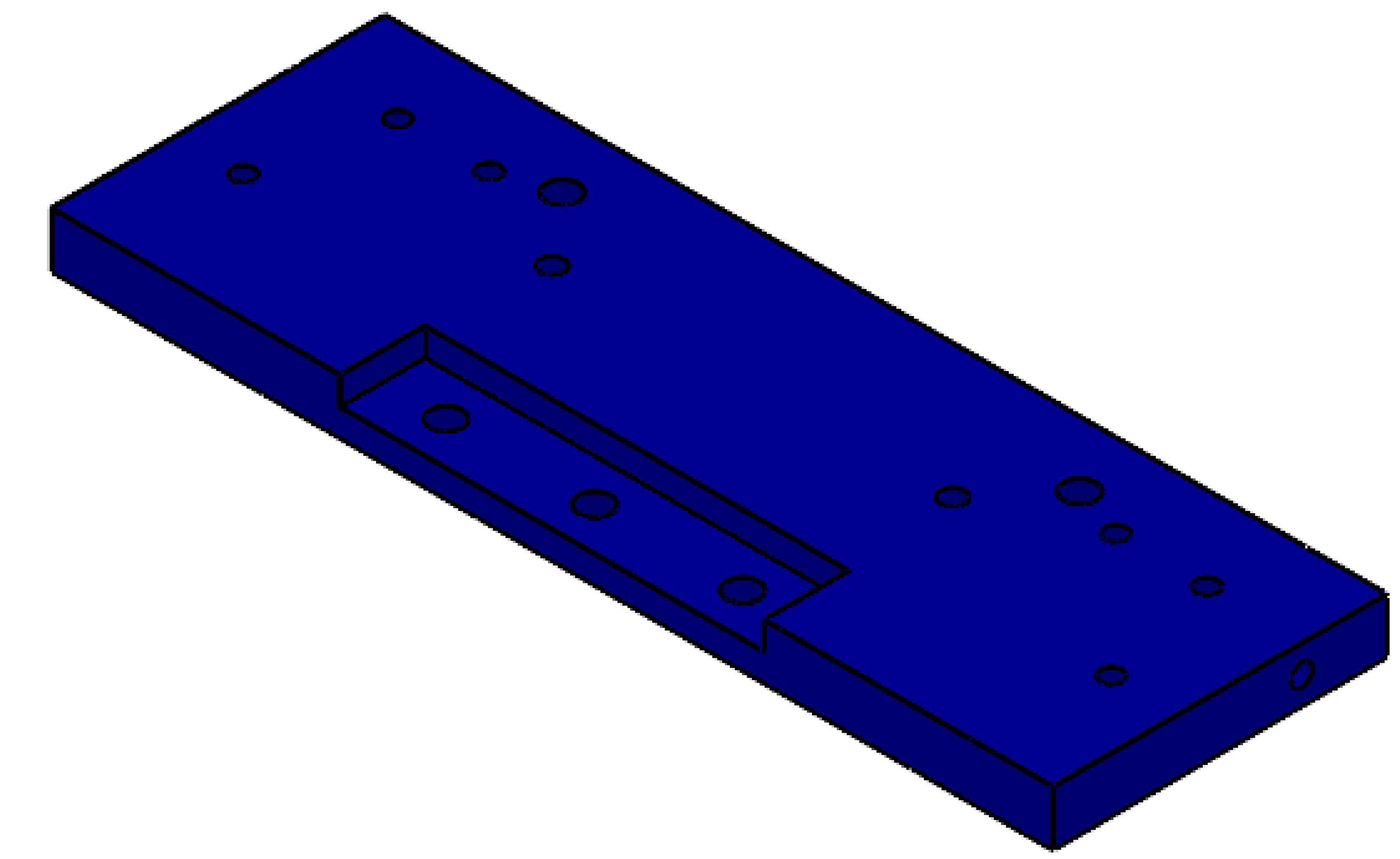
FILE NAME:
V Bracket

SIZE	Assignment Number	REV
B	N/A	1

SCALE: 1:1.5 Mass: 3554.390 SHEET 2 OF 2

NOTES:
1. REMOVE ALL BURS AND BREAK SHARP EDGES





MEC E 460
Instructor:
Dr. NAKASHIMA
WINTER 2021
Comments:
N/A

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

OMIKRON ROBOTICS Group 15

Modeled By **Liam Wolf**
Drawn By **Liam Wolf**
Checked By **Areej Khaddaj**
Liam
April 13, 2021 12:10:02 AM
February 25, 2021 1:10:09 AM

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Manipulator Platform

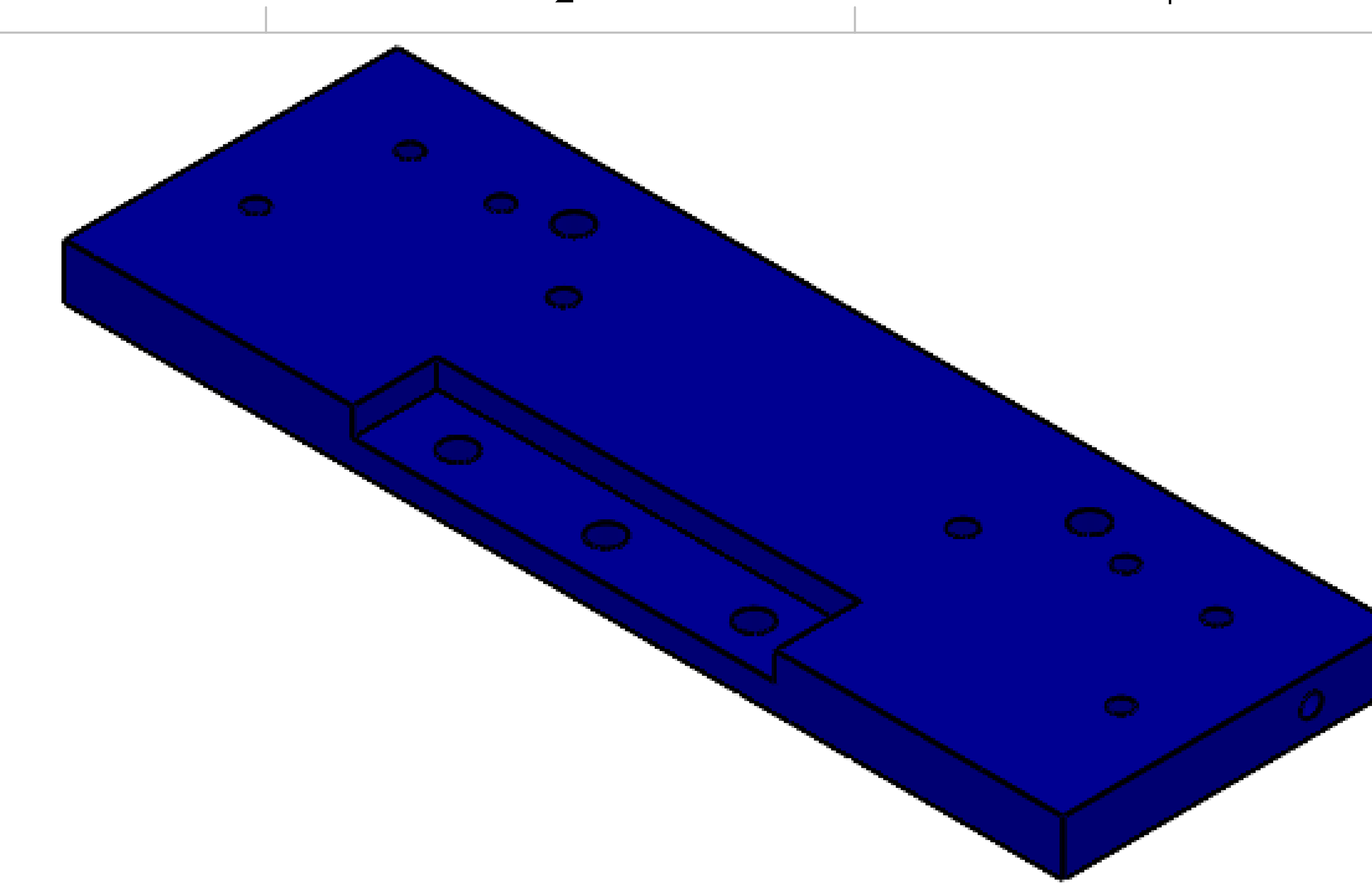
MATERIAL:
AISI 4130 Steel, normalized at 870C

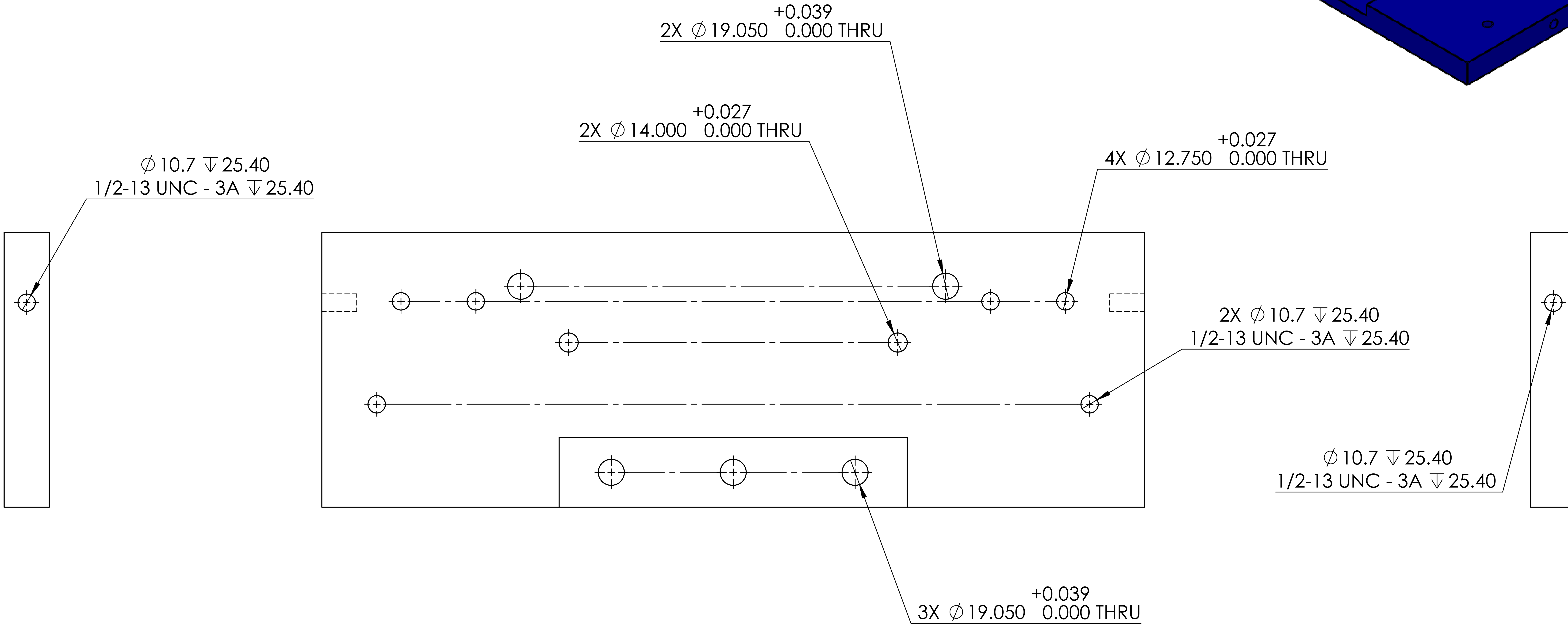
FILE NAME:
Manipulator Platform

SIZE **B** | Assignment Number N/A | REV 1

SCALE: 1:4 | Mass: 28843.463 | SHEET 1 OF 2

NOTES:
1. REMOVE ALL BURS AND BREAK SHARP EDGES





MEC E 460
Instructor:
Dr. NAKASHIMA
WINTER 2021
Comments:
N/A

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

OMIKRON ROBOTICS Group 15

Modeled By	**Liam Wolf**
Drawn By	**Liam Wolf**
Checked By	**Areej Khaddaj**

Liam
April 13, 2021 12:10:02 AM
February 25, 2021 1:10:09 AM

MATERIAL:
AISI 4130 Steel, normalized at 870C

FILE NAME:
Manipulator Platform

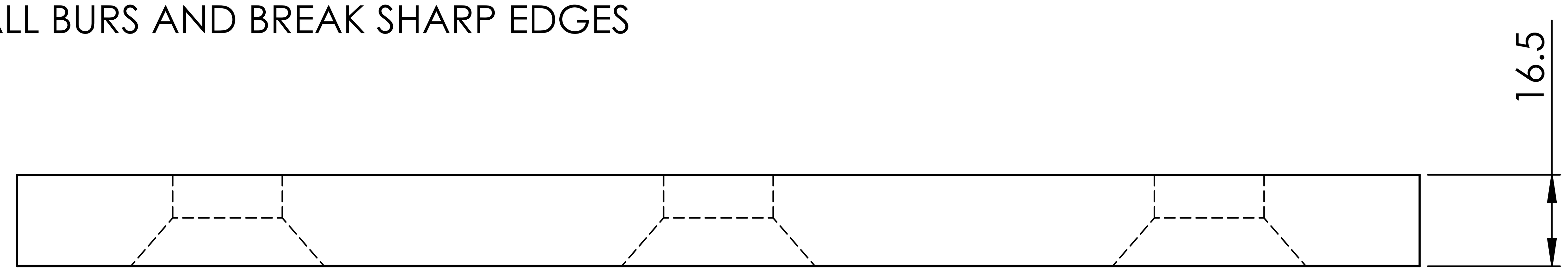
The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

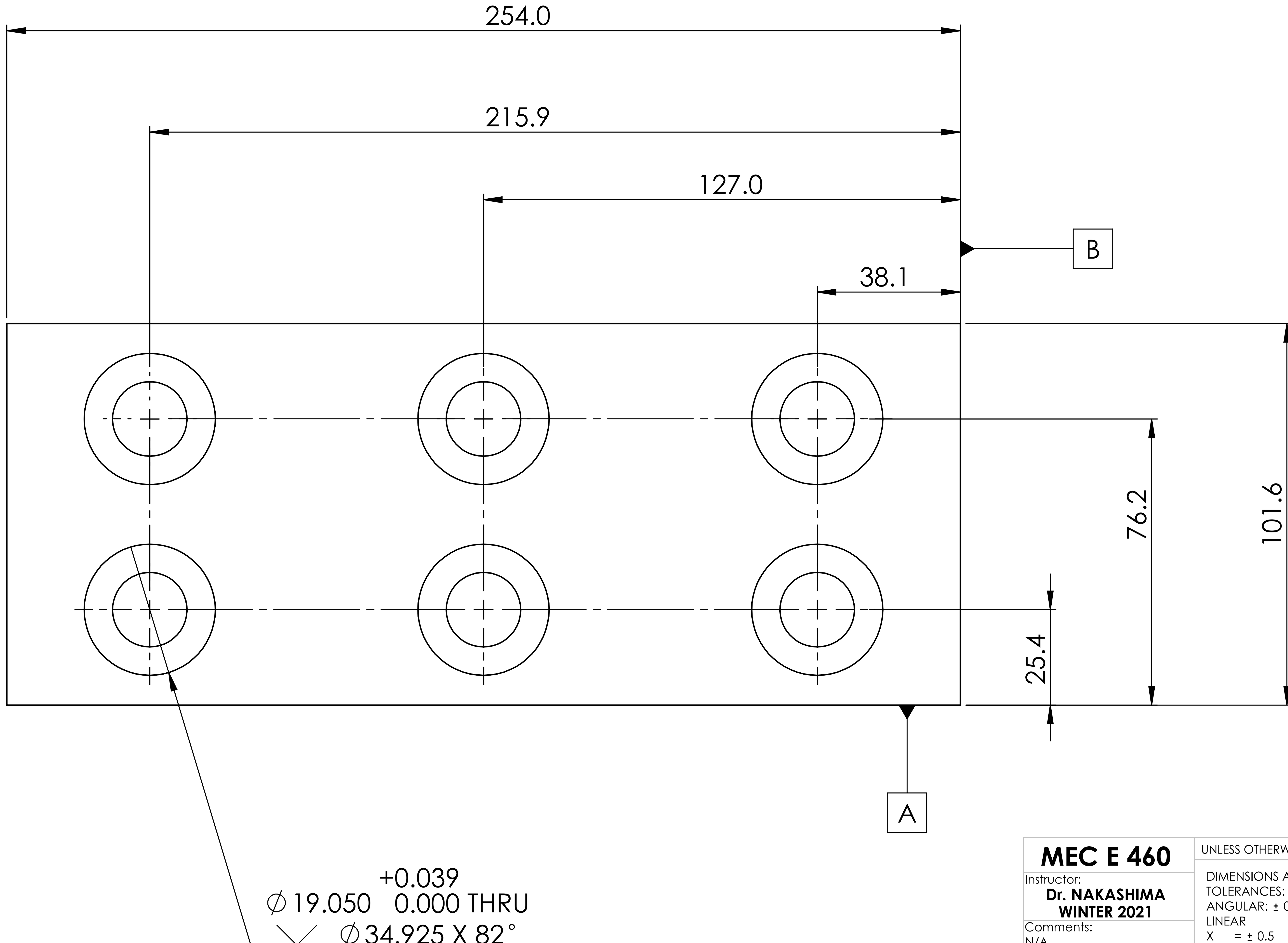
TITLE:
Manipulator Platform

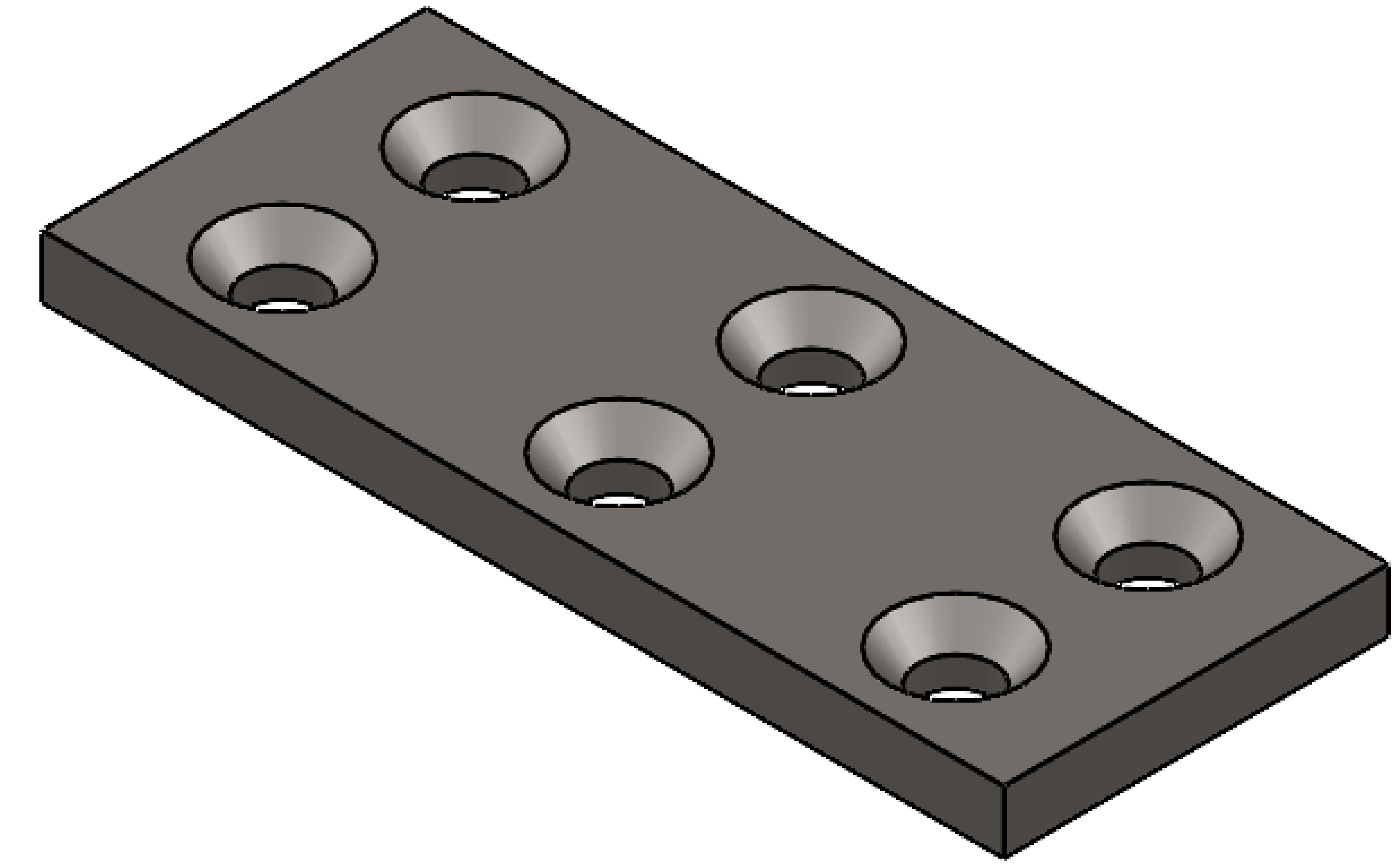
SIZE	Assignment Number	REV
B	N/A	1

SCALE: 1:3 | Mass: 28843.463 | SHEET 2 OF 2

NOTES:
1. REMOVE ALL BURS AND BREAK SHARP EDGES







MEC E 460
Instructor:
Dr. NAKASHIMA
WINTER 2021
Comments:
N/A

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

OMIKRON ROBOTICS Group 15

Modeled By	**Liam Wolf**
Drawn By	**Liam Wolf**
Checked By	**Kenny Okeke**

Liam
April 12, 2021 11:53:04 PM
February 25, 2021 1:22:06 AM

MATERIAL:
AISI 4130 Steel, normalized at 870C

FILE NAME:
Top Connector Plate

The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

TITLE:
Top Modular Connector Plate

SIZE	Assignment Number	REV
B	N/A	1

SCALE: 1:1.5 Mass: 2981.879 SHEET 1 OF 1