Urban Transport Vehicle

MECHENG 236, T04-G08

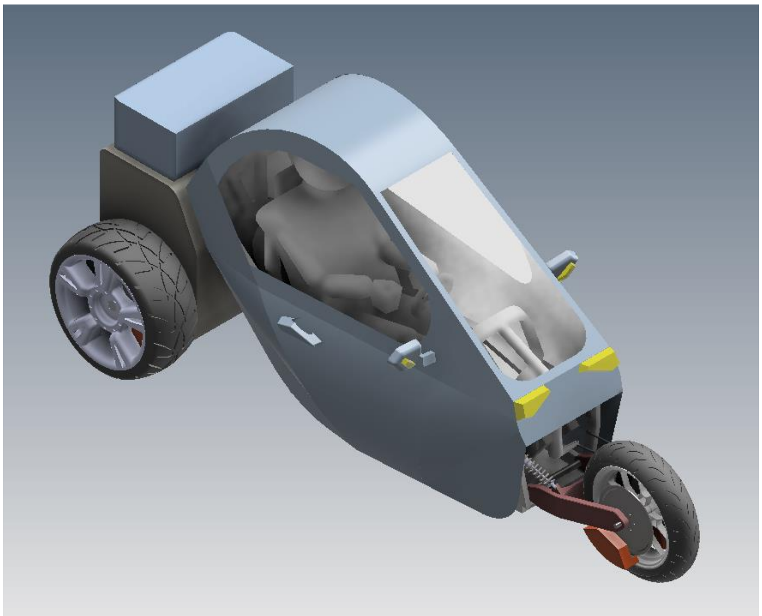

Figure 1: Final Design

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Executive Summary

Daily commuters of busy cities face numerous problems related to their travel. Students and working professionals in cities such as Auckland are often running late as a result of their commute. Parking and traffic in urban areas tend to force commuters towards public transport. While public transport may be a good alternative for some, it introduces too many restrictions on many others.

This proposal is targeted towards working professionals and students, who travel to and from the heart of the city daily. Our proposal provides a simple solution to issues such as shortage of parking, extended travel times and the hassle of public transport. A four-wheeler is the optimal way of transport, except whilst travelling to the city. Excessive parking fees, decreased manoeuvrability and traffic congestion are some of the biggest problems of travelling with a four-wheeler. An alternative is to use a two-wheeler such as a motorbike or moped. However, alongside the swiftness and manoeuvrability of the common two-wheeler, comes a substantial safety risk. Another alternative is use public transport. A clear advantage of public transport vehicles is that they can take advantage of facilities such as the bus lane. This allows the vehicle to bypass the traffic during rush hour, reducing travel time. On the contrary, restrictions such as limited routes, inadequate bus stop locations and irregular frequencies often frustrate our daily commuters.

Our undergraduate engineers have designed a three wheeled, Urban Transport Vehicle which combats the daily hassles of a commuter. Our design combines the rigidity and stability of a four-wheeler whilst attaining the swiftness and manoeuvrability of a two-wheeler. Our aim was to guarantee a certain degree of safety whilst not compromising on the power and manoeuvrability. Alongside the functional requirements one of the fundamental goals was to optimise user experience and tailor it to the needs of the daily traveller.

The specifications include a protective casing of the chassis, manufactured from lightweight aluminium, with great energy absorption properties. This shell is mounted on a cost-effective yet strong steel frame using stainless steel mounting brackets and plates, to birth a rigid chassis. As the vehicle is designed for one person, a scissor door is installed to improve accessibility and maintain the modern look. The braking system consists of dual front disk brakes, and dual rear brakes. The braking system is split to allow for front and rear braking systems to be independent of each other. The vehicle is a rear wheel drive as the KTM 690 engine is mounted onto the rear chassis. As promised the engine provides a top speed of 212 km/h and produces 75Nm of Torque at 6500 RPM. The vehicle can travel 250 km on a single tank and gives an average of 8.7 L/100km. The swiftness and the manoeuvrability of the vehicle is achieved via a smart hydraulic system. The semi hydraulic actuator outputs a constant torque of 847Nm which tilts the front end of the chassis to a maximum of 45° from the vertical within 2 seconds.

Our vehicle is an ideal solution to daily commuting problems. It is a three wheeled vehicle which can be driven on a regular car licence because of the LE1 class rating. This enables the driver to occupy the bus lanes and the transit lanes to bypass the traffic in a legal and more efficient manner. The compact design allows the user to occupy parking spaces delegated to bikes. The rigidity of the chassis and the robustness of the brakes ensures the safety of the driver as well as the vehicle. To optimise the user, experience the vehicle also features charging ports, localised GPS and Bluetooth hands-free technology.

Features

Chassis

Specifications and Design Approach

Our main goals for the chassis were to design a structure which encases and protects the driver, while also housing and arranging the other subsystem assemblies. We also needed to provide fast and easy access which didn't require any heavy lifting. We initiated the design by exploring the possible variations in chassis elements such as seating style, opening, storage, and overall look and shape. This exploration was facilitated by examples of similar products that have been developed, such as Carver, CLEVER, and Toyota i-ROAD. We then selected the preferred combination of element options, based on their compatibility with other subsystems, and the fulfilment of criteria such as safety, accessibility, and effectiveness. The overall design of the chassis has been developed and iterated from our original concept, to one better suited to the frame provided and accommodating of other the other subsystems, resulting in a cohesive, optimal design as demonstrated in Figure 2. Our final proposed chassis is a compact, sleek, eye-catching design that any urban commuter would be proud to traverse around town in.

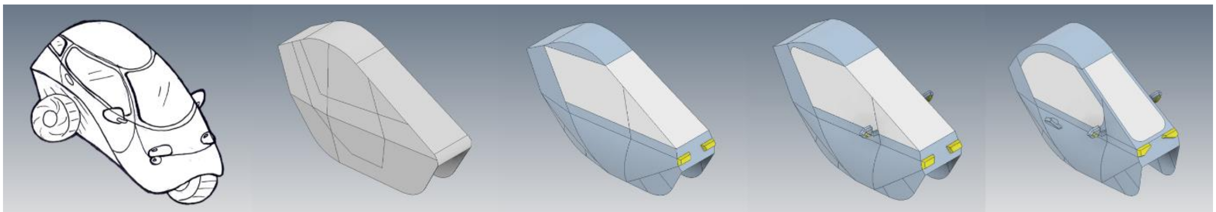


Figure 2: Front chassis shell original concept, CAD iterations and final design

Frame and Body

It is a body-on-frame model which consists of an aluminium shell mounted on a steel frame using stainless steel brackets and plates. The steel frame provides strength and rigidity to the vehicle, while also being cost-effective, and easy to work with in manufacturing stages (sections are easily shaped and welded). We've added overhead beams and cross members to the frame provided in order to help support the windows and roof of the vehicle, as detailed on page 35. A modern aluminium body was chosen for its weight reduction benefits (improving fuel economy), while retaining great energy absorption properties to protect the driver in the event of a crash. Details of this body are specified in the Front Chassis Shell drawing on page 36. Stainless steel mountings help to prevent corrosion between the frame and the body, to maintain the integrity and longevity of the vehicle. We believe these materials find the balance between affordability, availability, and function, compared to other materials that are used in these applications such as fibreglass, magnesium, and carbon fibre.

Opening

Our chosen opening is a single scissor door, which provides easy operation from the low vehicle height, while also maintaining the desired compactness of a motorcycle. This allows the driver to enter and exit the vehicle in tight parking spots as the door shifts upwards, instead of outwards like a traditional car door. One of the concerns brought up in the management meeting was the amount of heavy lifting required to manually open the scissor door. To address this, the scissor door hinge design involves a gas strut to assist in the lifting of the door, meaning it should be as easy to open as a traditional car door. The gas strut will have spherical pivots at either end to allow the door to be manoeuvred to the desired angle. A representation of the size and location of the scissor door hinge and gas strut is shown in Figure

3. In the final product, these components will sit inside the body, rather than directly on the frame as shown.

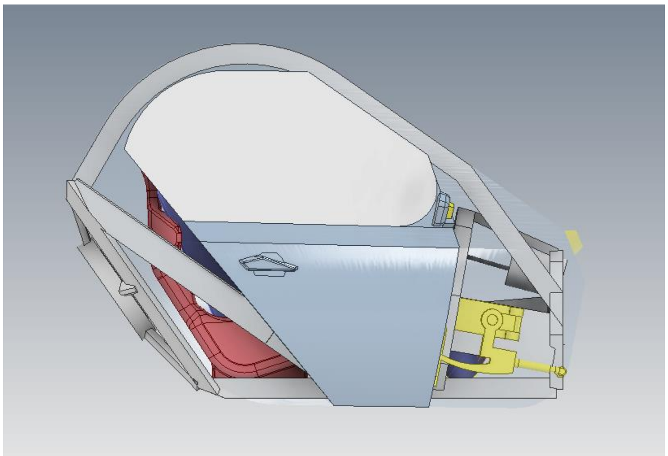


Figure 3: Scissor door hinge and gas strut placement

Windows and Interior

One of the main focuses in the development of this design was ensuring the driver has a wide field of vision. Our large windshield and side windows leave the driver's view to the front, left, and right of the vehicle completely unobstructed. Our model also includes a representation of the side mirrors which can be used for a rear view. In line with the common car design, the windshield will be made of laminated glass for its resistance to cracks and can involve UV filters to protect the driver's vision. The side windows are constructed from tempered glass, which is easier to shatter in the case of an emergency if the door becomes inoperable. The large windows along with the spacious interior help to improve driver comfort in the compact vehicle. We've left plenty of space along the sides and overhead so the vehicle can accommodate a range of body types, with an overall shell height of 1.34 m and width of 0.86 m. There is additional space available in the dashboard area, which could be used to house safety features like airbags, additional driver interfaces such as a stereo system or hands-free technology, and/or convenient storage compartments for personal items like a wallet or sunglasses.

Rear Chassis and Storage

Powertrain and turning system assemblies are housed in a separate frame atop the rear axle connected to the front chassis via the provided bearing behind the driver's seat. This allows the rear chassis to remain level with the rear wheels as the front chassis tilts. The separate chassis system means the driver can experience a smooth ride, undisturbed by the operation of the engine and hydraulic components. These components are assembled in a frame which maximises the space provided, as well as the efficiency of the subsystems, detailed in the Rear Chassis drawings found on page 45. The structure is fully covered by a rear chassis shell which is mounted on the rear axle frame, with additional support above the fuel tank. This shell has permanent openings for the chassis connection, exhaust pipe, and rear axle and suspension. There is also a fuel tank opening on the right side, and a maintenance opening which allows complete access to the top, and partial access on the left side, to the powertrain and hydraulic assemblies. A hood prop should be included inside the shell to assist in holding the lid open during inspection and maintenance. The shell also provides the mounting location for a 70L removable aluminium storage case. The case can connect to the shell using a locking system similar to motorcycle monolock. This allows the user to add or remove storage at their leisure for particular ride or maintenance purposes. This amount of storage can easily fit a couple school bags, a few briefcases and folders, or a medium grocery haul. It may be beneficial to implement a firewall behind the driver's seat and below the storage space, to protect the driver and their luggage from the heat of the engine. Drawings for the Rear Chassis Shell and Removable Storage Case are provided in the appendices.

Verification

We identified the added overhead beams and the rear wheel axle as critical members in the structure of the chassis, each supporting the biggest loads. To verify the structural integrity of the chassis, we modelled these members as simply supported beams with loads distributed at specific mounting points, with a safety factor of two. From these calculations we attained the maximum deflections, axial stresses, and shear stresses through each member. For the overhead beams, we approximated a total load (for the body, door, and windshield) of 100kg, distributed evenly between the two members, and acting at the three mounting points. We attained a maximum deflection of 1.55 mm, across the 2.42 m, 25 by 75 by 6 mm beam. The load on the rear axle (including engine, fuel tank, hydraulic components, frame, and shell) was approximated at 150kg, acting at three points of contact between the frame and the axle. For this loading we calculated a maximum deflection of 1.2 mm, across the 1.11 m, 48 mm diameter axle. All the maximum axial and shear stresses were found to be well below the assumed tensile and shear yield stresses of steel (350 MPa and 200 MPa respectively). The full calculations, diagrams, and results of these verifications can be found under Appendix A, Chassis Calculations 1 to 6.

Powertrain

Specifications

The goal of the powertrain was to provide the user with a reliable system of components that work efficiently to maintain an amazing and unique driving experience whilst still being practical, and easy to maintain. The wet clutch and hydraulic actuation give the user a smooth and easy driving experience that maintains a consistent clutch engagement point over time. The KTM 690 engine was specifically chosen for this vehicle partly because of its compact and lightweight design, but also its above average fuel economy. The rear subframe is designed so that components like the engine, that must be accessed for regular maintenance, require no disassembly of frame housings. For more complex tasks, bolted connections simply need to be removed to extract individual housings and the required parts can be accessed. This makes overall maintenances even for difficult tasks far less complicated.

Design Approach

This vehicle is aimed towards daily commuters who want the luxury and safety of a car, as well as the manoeuvrability, simple maintenance and fuel efficiency of a motorcycle. When designing the drivetrain there were three things that were always referred back to: efficiency, accessibility and practicality. The engine needed to be large enough to provide sufficient power for the speed requirement, whilst maintaining a good fuel efficiency. Components needed to be easily accessible for regular maintenance and parts like the clutch needed to be long lasting and require minimal upkeep. Finally, the vehicle should be practical for the user and space must be allocated for storage.

Engine

During the early stages of the design process, a 1000cc GSXR engine was chosen to power the vehicle. However, during the later stages of the design process, we realized an engine of this size was unnecessarily powerful, and large for our intended submarket, therefore we had to iterate back and find a new one. Fortunately, we were downsizing the engine, so the chassis did not require many changes. After reviewing all the 600cc engines it was decided that the KTM 690 engine was the most viable option for this vehicle. It combined a stellar fuel economy of 8.76L/100km with a torque of 75Nm and peak power of 54Kw giving the vehicle a top speed of 212km/hr. Being a 1 cylinder 4 stroke the engine is small, simple and easy to maintain especially in comparison to the inline 4 counterparts.

Fuel Tank

The previous engine chosen (1000cc GSXR engine) had a fuel economy of 12.43L/100km in our vehicle, which meant a minimum fuel tank size of over 31 Litres. With the later iteration and implementation of the KTM engine, a fuel tank capacity of only 21.9L is required this is roughly a 30% decrease in size to achieve the same distance for one tank of fuel. Unlike a regular car the fuel tank is located close to the engine which means long piping connection is not needed and accessing the fuel tank for cleaning is a simple task.

Clutch

Throughout the design process, choosing whether we go ahead with a dry or wet clutch was one of the more difficult decisions. After some discussion with the engineering team we decided to go ahead with a multi disk, wet friction clutch, similar to a motorcycle. Being doused in engine oil means the clutch is far more durable, long lasting and quieter in comparison to a traditional dry clutch. During stop and start traffic, a dry clutch will often undergo a lot of wear and can potentially burn out, which is an unpleasant experience for the driver. With a wet clutch this issue is resolved and there is next to no chance of this happening. On top of this, the wet clutch also makes considerably less noise which means less soundproofing is required for the driver's cabin.

Clutch Actuation

The two actuation methods in question were hydraulic and cable actuation. For this scenario the hydraulic actuation was clearly a far better option than cable, and that is what is featured in this vehicle. This means there is no need for messy cables which require constant maintenance in the form of cable adjustments and lubrication to mitigate inevitable corrosion and cracks. Hydraulic actuation solves this problem, there is next to no constant upkeep and the fluid in the reservoir allows the system to self-adjust according to the wear on the clutch, which will provide a consistent clutch engagement point throughout its useful life. On top of this, the clutch pedal won't be as heavy which means a smoother engagement and less fatigue for the driver during daily commutes.

Rear Chassis Subframe

This part of the design was the one that required the most iteration and changes. This is due to component changes like engine downsizing and switching from linear hydraulics to rotary hydraulics for the turning system. The chassis has been designed so that accessibility for critical components during regular maintenance is simple. Most of the individual housings like motor, fuel tank, and gearbox housings are bolted onto the main frame (refer to Figure 20) and can be easily removed or re-attached. Only the accumulator housing (refer to Figure 21) and pump/motor housing (refer to Figure 22) are welded to the main frame because these components can be easily lifted out when the chassis lid is open. When the chassis lid is open it reveals the engine bay and fuel tank as well as most of the turning system. This ease of access means that for regular maintenance, whether it be on the engine or turning system components, there is no need to disassemble parts of the chassis. For more complex maintenance tasks like clutch changes or a gearbox replacement you simply remove the bolted connections and access the necessary components. The hydraulic tank is located underneath the vehicle and can be accessed by unlocking the latches which will allow the tank to come down.

Exhaust Silencer

In order to preserve the simplicity of design and manufacturing, the same exhaust silencer used in the KTM 690 motorcycle will be used in this vehicle, which is the "KTM 690 SMC Remus Hexacone exhaust". It will provide the necessary sound damping to be below 90 decibel and meet the legal road requirements in New Zealand. As well as this, it is an inexpensive silencer system which is readily available and can be easily installed due to its “slip-on” design and compatibility with the KTM 690 engine.

Verification

The fundamentals for the Power Train System are listed below

- The vehicle must be powered by an internal combustion engine
- The maximum force the driver can exert on the pedal is 250N
- The vehicle must be fitted with an exhaust silencer system that operates constantly
- The vehicle must have a range of 250KM per tank of gas
- The vehicle should have a top speed of at least 125km/hr
- The design must have a foot pedal clutch

An important assumption made was the vehicle mass including driver is 500kg, this value has been approximated by comparing the mass of large motorcycles, small cars and rickshaws and deciding on a reasonable mass of 500kg.

The internal combustion engine used is a KTM 690 engine which produces 75Nm of peak torque and 54Kw of peak power. Calculation (10) in Engine Calculations was used to calculate the maximum speed of the entire vehicle which ended up being 212 Km/hr which is roughly 40% greater than the minimum requirement.

Three contact surfaces are present for the wet clutch in this vehicle (refer to Clutch Calculations (8)) Calculation (9) shows Force Requirement on clutch pedal which is 3200N. However, the max force exerted on the clutch pedal is only 250N. In order to achieve the necessary force ratio of 1:13 a pivot mechanism is installed at the clutch pedal as well as the clutch fork which will provide the necessary ratio to engage the clutch.

The silencer system is a Remus Hexacone exhaust, which was used on the KTM 690 motorcycle and will be able to meet the maximum exhaust sound specification by the road safety specification.

The KTM 690 engine has a fuel consumption of 3.87L/100km when used on the original motorcycle. By comparing the mass ratio to the fuel consumption, this vehicle achieves a fuel efficiency of 8.76L/100km as detailed in Calculation (7) in Fuel Tank Calculations. Therefore, the minimum fuel tank size is 21.9L in order to meet the requirement. The fuel tank featured in this vehicle is 23.5L, which has a large enough capacity for the vehicle to drive 250Km on a full tank.

Turning System

Specifications

The overarching goal of the turning system is to achieve the swiftness and manoeuvrability of a traditional motorbike whilst preserving the rigidity of a four-wheeler. According to our research, a motorbike is designed to achieve a maximum tilting angle of 65 to counter the centrifugal moments. However due to the size differences our vehicle can achieve similar results at a tilting angle of 45˚. A smart tilting system is integrated into the design to tilt the front chassis. A constant torque of 847 Nm is applied by a semi rotary hydraulic actuator which permits the front chassis to lean 45˚ either side from the vertical position within 2 seconds. The mechanism is capable of carrying out three consecutive sweeps (total of 270˚) whilst driving in less than 1 minute.

Design Approach

As the fundamental requirement was for the vehicle to be a three-wheeler, the design process of the turning system was separated into two parts; the front-end steering system and the rear end tilting system. This segment of the proposal focuses primarily on the tilting system and how its integrated with the other sub-systems.

In-hub Steering system

The front wheel uses an in-hub steering mechanism which connects to the driver's steering wheel. This is a common system used in most motorbikes and two-wheelers. The biggest advantage of the in-hub steering mechanism is that it separates the suspension, brakes and the steering, allowing more finer adjustments to the overall design. Conversely, the mechanism turning on its own produces centrifugal forces, which in turn produce a moment about the pivot point located at the wheels (refer to Figure 13). The moment produced is dangerous as it's capable of toppling the entire vehicle in certain conditions.

Tilting system

To prevent toppling of the vehicle our engineers have developed a hydraulic tilting system which resolves this problem. Figure 14 shows that when the front chassis leans into a turn, the dangerous moment due to centrifugal forces is balanced by the moments due by gravitational forces. This in turn reduces the risk of toppling significantly.

During the early design stages of this subsystem, many configurations of the hydraulic mechanism were discussed. Most of the alternatives utilised linear hydraulic actuators to tilt the front chassis. This idea was eventually scrapped because the torque required to tilt the chassis would be dependent on the angle of tilt, and therefore it would be difficult to precisely control the leaning angle. A scenario considered was if the vehicle over tilts, it could topple the vehicle and put the health and safety of the driver at risk.

After more research and iterating backwards, the team came up with the idea of using a semi-rotary hydraulic motor. The current tilting system uses an HTR series, semi-rotary hydraulic motor of a rack and pinion type from Parker. The motor is configured to have a maximum tilting angle of 90° (45° either side from the vertical position) and a constant output torque of 847 Nm. Figure 16 shows the full configuration of the hydraulic motor utilised in the tilting system.

To supply the hydraulic fluid to the actuator, a geared hydraulic pump is used and to drive the pump an electric motor draws power from the battery. The pump selected is PG640 fixed displacement gear pump from Parker. The gear pump has 75 cm^3 displacement which rotates clockwise. To achieve a constant output torque by the actuator, the pump is required to run at 66.6 RPM. Therefore, it produces 47.7 Nm of torque. To drive the pump, an electric motor is selected of the shelf which runs at the same RPM using the power from the battery. An alternative option was to utilise the power of the engine to drive the fluid around the circuit. However, this was quickly dismissed as the engine fluctuates its RPM leading to a discontinuous flow rate, which in turn will fluctuate the output torque.

The hydraulic circuit also consists of a bladder accumulator and a relief valve, which work to provide a quick response from the system to tilt the vehicle. The accumulator is chosen from Parker which is configured to have hold 2.3L of hydraulic fluid and is made from carbon steel. It has a maximum flow rate of 450 L/min and accommodates 8MPa of pressure. These values are far less than the operating values which implies smooth functioning. As the maximum pressure within the circuit is restricted to 4MPa a relief valve is also included as a precaution.

Other components of the hydraulic circuit include a tank/reservoir which holds the hydraulic fluid, a filter to maintain the purity of the fluid and a directional control valve to ensure the correct input and output of from the motor.

Functioning (Refer to Figure 15 whilst reading)

As the vehicle is started, the power from the battery is used to run the electric motor which in turn drives the gear pump at 66.6 RPM. The shaft of the electric motor is coupled with the gear pump, using a manufactured coupling. The accumulator is then charged up so it can supply the fluid when necessary. The relief valve is incorporated to relieve the pressure from the system once the 4MPa threshold has been reached. This ensures the pressure doesn't exceed 4MPa. When the driver approaches a turn, the ECU (electrical control unit) will take read values such as the speed of the vehicle and the angle of the steering wheel using inbuilt sensors to make a calculated judgement of the tilting angle. The accumulator will then prompt the fluid towards the directional control valve. The valve will then supply the fluid to the correct port and actuate the turning system. The hydraulic motor is connecter to the front chassis using another manufactured coupling.

Verifications

The fundamental requirements for the turning system are listed below.

- The design must be able to lean from the vertical to the maximum lean angle of 45°, within 2 seconds.
- The design must be able to turn at least three ±45° turning sweeps within one minute.
- The tilting mechanism must exert 500Nm of torque at all angles.
- The pressure within the system shouldn't exceed 5MPa.

For the system to tilt 45° within 2 seconds a semi-rotary hydraulic actuator is selected, which has a maximum angle of 90° and an angular velocity of $\frac{\pi}{8}\ {rad}/{s}$. The selection of the actuator started from power calculations. Calculation (11) was used to calculate the minimum power requirement and therefore the actuator had to exceed the 500 Nm of torque and 196.35 W of power. The selected actuator has a torque rating of 847 Nm and therefore calculation (12) shows that the output power is 332.8 KW which exceeds the minimum.

The flow rate within the circuit was determined using calculation (13), which implies that the mechanical power output by the actuator must equal to the fluid power within the circuit. The flow rate of the fluid was therefore calculated to be approximately 3.95 L/min.

An inversely proportional relationship between pressure and flow rate meant the pressure could be reduced by increasing the flow rate. The flow rate was therefore increased to 5 L/min ($8.33 \times 10^{-5}\ {m^3}/{s}$) so, the pressure was not anywhere close to the upper limit. Calculation (14) shows the maximum pressure within the circuit doesn't exceed approximately 4 MPa.

The bladder accumulator that is chosen has a total volume of 2.3 L. Given the flow rate and volume of the cylinder, the time taken to fully charge the accumulator was calculated. Calculation (17) shows that the time taken to charge the tank is 27.6 seconds.

The total time for 3 sweeps was calculated using Calculation (18). This shows that the time taken for the for the actuator to tilt 270° plus the time taken for the accumulator to charge is equal to the total time taken to accomplish 3 consecutive sweeps. The upper limit was 1 minute, whereas our design can achieve this within 39.6 seconds.

Braking system

Specifications

The goal with the braking system was to allow the rider to react to any dangerous scenario and prevent harming themselves or others. This would mean that our brake design would need to operate at a high level regardless of the environment or situation. Our answer to this was to design powerful yet controllable braking, which would ask very little force from the foot pedal but would provide large amounts of braking force at the wheels. Our three-wheel vehicle is designed to have dual disk braking at the front and single disks at the rear. With all disks being accompanied by 2 piston callipers and a split braking circuit which can apply brake force to the front and rear independent of one another (Refer to Figure 19). The vehicle can decelerate from 60km/hr to 0km/hr in under 2 seconds, applying 1263 N of force at each front disk, and 842 N of brake force on each rear disk, resulting in a total of 4210 N of brake force (Supporting calculations are shown in Front and Rear brakes segment). The driver will be able to apply a maximum of 400 N of force onto the pedal which will be converted to the required braking force by a mechanical enhancer (refer to Mechanical Enhancer section in Appendix A). The parking brakes are also designed to withstand the vehicles forces up to an angle of 45°, with the rear brakes providing the braking force for the entire vehicle (refer to the Parking brakes section in Appendix – A).

To achieve these high levels of braking, we decided to operate with Harley Davidson 43 mm diameter piston callipers, made from tufram coated aluminium alloy. They are light weight but very strong and provide a 22.1 cm^2 brake pad area (refer to Figure 18). They are paired with two radial master cylinders which connect to the back and front of the vehicle (Refer to Figure 17). They can provide up to 5.25 MPa of pressure within the cylinder and are able to withstand the mechanical enhancers increase factor of 3.2 N of force form the pedal to the brake pads (As shown in Front and Rear brakes section).

We also took into consideration the anti-lock braking system or ABS. The ABS will function by taking readings from the rear brakes and being constantly monitored by the ECU. Because our product has only a single front wheel, the imbalance between the front steering and the rear driving force could cause trouble if the driver had to make an abrupt stop. However, due to the sensors placed within the rear frame of the product, the ECU will be able to provide a controlled and safe deceleration for the driver. Within the attachment of the callipers, we have integrated smart sensor which measure the distance from one plate of the calliper to the other. Once the calliper plates sensor reaches 2 mm, a warning light will be displayed next to the speedometer to indicate that the brake pads need changing (Refer to Brakes Maintenance for a detailed summary).

Design Approach

Force and Dimension Analysis

The initial process of design began with the research and development of the subsystems. The major points that our braking system had to achieve were foot actuated brakes, a method of maintenance for the brakes at the wheel, and numerical specifications. The initial two ideas were to implement either drum or disk brakes as the main system of braking. However, analysing and comparing the two systems it was clear the disk brakes offered better range for customising the sizes of components. Disk brakes are also smaller and more open by design, allowing natural deterrents such as water to slide pass them, avoiding water build up in crevasse. This is essential as water is a very potent corrosive agent, which makes the maintenance and life span of the system harder to measure when large build up is present. We passed the two systems through a morphological matrix, and it resulted that the disk brake had better compatibility with the other subsystem. From this we were able to select disk brakes as being the optimal braking system.

Once the braking system was chosen, the sizing of the brake callipers and disk was next. Using the ISR brakes catalogue, we found multiple possible sets of brakes that would work. However, with further numerical analysis and researching the possible dimensions that would fit the CAD model, we found that the brake sets would not fit with our dimensions, so we decided to find the individual parts and match them with the corresponding sizes of their pairs (e.g. calliper size must match cylinder). After the dimensions had been sorted, we continued with the braking ratio for our vehicle.

Braking Ratio

Because the anatomy of our vehicle was un-even in contact points (has one wheel at the front and two at the rear), we opted to find a similar vehicle to use the ratios that had already being experimented on. From our research we found a similar vehicle, the Toyota i-ROAD, which is a three-wheeled vehicle with two wheels at the front and a roller type wheel at the rear. From analysis of the body, we arrived at a 60% front 40% rear split. The reason being that the centre of mass of our three wheeled vehicles is located on top of the rear axle, as the engine and other power train components are there. Because there is a greater amount of mass in the rear, we need a greater amount if frontal force to prevent the vehicle from wanting to rotate on the rear axle when braking. By applying a greater amount of brake force to the front, we force the system to want to shift its mass forward, allowing for a more even distribution of moments on the body.

Mechanical Enhancer

Finally, it was clear that a multiplying factor had to be applied to the pedal to achieve the required braking forces on the wheels. The design of the mechanical enhancer was calculated by using the ratio between the required braking force over the maximum appliable force on the pedal. This gave as a ratio for the scaling factor (Refer to Mechanical Enhancer calculations).

Overall Design

Although our subsystems were each optimised for their specific task, we've also managed to develop a complete vehicle concept with the subsystems working together in harmony, thanks to group feedback and iteration. We managed this by focusing on efficient use of space and avoiding interference between components. One example of this is in the development of the chassis to better fit the frame, in order to avoid taking up space for the hydraulic assembly. The final front chassis shell is flush with the back of the frame, allowing the turning system to move freely without interference with the rear chassis. The rear chassis was also organised to improve the efficiency of both the powertrain and the turning system. Components for each subsystem are assembled in such a way that keeps respective components separate for better transmission and simpler maintenance, while also making the most of the space provided, and leaving generous room for storage.

The designed braking system was compact which made other subsystems easier to implement. The braking system works perfectly in harmony with the hydraulic turning system, allowing our drivers a swift, safe and satisfactory driving experience, with minimal needed input by the driver. While the braking and turning systems work well together, they are placed with enough distance for maintenance to be done on both without either one being in the way. This allows for quick and clean check-ups, which are both easy on the driver, and the professional servicing the vehicle. We found that the differential housing would be a perfect structure to bolt the callipers to, as it is strong and is located close enough to the wheels to provide the support the callipers need to function accordingly. All these subsystems are brought together and protected by our double chassis design, which gives the overall vehicle assembly a cohesive and streamlined appearance.

Conclusion

The purpose of our design is to provide urban commuters such as students and workers with a convenient, accessible, and cost-effective form of transport that prioritises safety and driver comfort. The tilting three-wheeler manages to find the happy medium between a compact, swift motorcycle and the enclosed, rigid car. Our chassis design protects the driver with a lightweight aluminium body fitted to a strong steel frame, while remaining open and spacious with the help of large windows. Materials were selected based on fit for purpose, as well as ease of manufacture. The scissor door provides quick and easy accessibility from the driver's position, retaining a narrow profile, without requiring heavy lifting thanks to gas struts included in the door hinge assembly.

The combustion engine in this vehicle allows it to reach a max speed of 212 km, and its great fuel efficiency along with optimal tank size give it the ability to drive over 250 km on one tank of fuel. An easy to install and compatible KTM exhaust silencer means there is minimal engine noise produced. The clutch pedal featured allows for a maximum force of 250N by using a 1:13 ratio between the pedal and clutch discs.

For swiftness and manoeuvrability, the turning system had to be able to lean 45˚ either side of the vertical. This requirement also entailed an output torque of 500 Nm. Originally the plan was to incorporate two linear hydraulic cylinders, however, after some iteration a semi-rotary hydraulic actuator was utilised. The ECU has sensors, which attain two key pieces of information: the speed of the vehicle and the rotational position of the steering wheel. These two inputs are used to make a quick decision on the leaning angle. This smart turning system is how our vehicle achieves swiftness and manoeuvrability.

The requirements for the brake system was to provide a foot actuated system, that would notify the driver when maintenance on the brakes was needed. To meet the specifications, we located a foot pedal with a mechanical enhancer and placed sensor systems located on each plate of the callipers that would indicate the driver when the distance had been reduced to 2 mm. A further specification was to design or choose braking components that would allow our vehicle to have a minimum stopping distance of 33 m, as the vehicle travelled 60 km/hr, while also having the pressure within the braking system not exceed 7 MPa. We met the specification by researching and numerically solving the dimensions required of the components. Using the IRS brakes catalogue, we were able to find braking components that would be implemented to have the specifications met.

Each of these four subsystems come together to form a cohesive, completed vehicle concept which meets the set requirements. The performance and placement of each component has been verified in calculations and CAD. We developed our designs by exchanging regular feedback and taking on the concerns brought up in the management meeting, in regards to the detailing of the turning system, and the assembly of the vehicle as a whole. Each subsystem specification has challenged our teams engineering creativity and analysis skills to provide the optimal solution for the singular subsystem as well as their conjunction towards a cohesive mechanical product. The creative engineering used to conserve space, and produce a smoother operating vehicle meets all our stakeholders wants and needs, providing the space, safety, power and handling of a modern automobile, in the sleek, sharp body of a three wheeled motorcycle.

There is room for further development and specialisation towards our submarket through the implementation of a convertible roof, hands-free technology, localised GPS, in-housing electronics, seat heating, and other interior detailing.

Maintenance Instructions

Powertrain Maintenance

Engine

- Engine oil needs to be topped up if running low. Access the engine bay by disconnecting the storage space and lifting the rear chassis lid. The engine will be located beside the fuel tank with an air intake connected to it.
- Engine oil and filter needs to be replaced every 4000km. To replace oil and filter reach under the vehicle and drain the oil and remove filter. Connect new oil filter in place of the old one, open oil cap atop the engine and pour fresh oil.
- Spark plugs should be replaced every 50,000 km. Simply remove the storage, lift open the chassis lid and the spark plugs will be located atop the engine.

Clutch

- Clutch will need to be replaced every 60,000km. In order to access the clutch, the fuel tank and fuel tank housing will need to be detached from the main housing by unscrewing the bolts using either a power tool or torque wrench. After this has been done the clutch casing should be visible between the gearbox and engine and can be extracted by disconnecting the latches. Drain oil from clutch, connect back by locking latches and reassemble the removed parts.
- Clutch fluid will need to be replaced every two years. Lift the chassis lid and the fluid housing is located atop the engine near the air intake. Replace the old fluid with fresh fluid.

Brakes Maintenance

- Brake pads need to be changed every 80 000 km - 100000 km, or when brake pads get bellow 2 millimetres in thickness.

- Wheels will need to be removed using a power tool or lug wrench to replace the brake pad. It is advised to take it to a mechanic or a professional.

- If maintenance is done by owner, we advise using an impact drill or automotive tool that fits bolt size.

- Brake fluids need to be changed every 2 years (around 60 000km – 80 000km).

Turning System Maintenance

- Hydraulic Fluid will need to be replaced every 50,000 Kms. The latches holding the tank underneath the vehicle allows the tank to drop down for easy access during maintenance.
- The filter on the hydraulic fluid will need to be replaced every 5000 Km as it is prone to contamination. This can be accessed similarly to the tank.
- At each service (roughly six months) the gear pump housing should be checked, and the clearance should be less than 0.005”. If it exceeds the limit, wear will necessitate replacement. Any wear on the gear hubs and/or nicking, grooving or fretting of teeth necessitates replacement. Wear of more than .002” on drive shaft indicates oil contamination and must be replaced.
- During each servicing check the surface of the bladder or damage, check around the accumulator shell for corrosion and foreign bodies. If found use correct replacement parts for maintenance.

Teamwork Reflection

Our group initially found that this project was an amalgamation of the Warman Design and Window Washer projects. With aspects of the Warman Design Project being that it was a very open-ended project and was up to the individual groups to continue the design using the conventional processes. Conversely, the individual subsystem aspect mirrored that of the Window Washer project. However, when it came to the specifics of designing our subsystems, a lot of the calculations and critical information was left for us to find and decide if it was reliable or not. The content covered over the lectures were valuable hints towards the general direction of this the information, but with lectures being more tangentially related to the project, it became increasingly difficult to know if the direction that we took was correct. The scope and independent research required seemed a little overwhelming, but in the end we all acquired specialised knowledge for our individual subsystem. To create the single cohesive product, we had to utilise our communication and teamwork skills. We worked very well as a team, even during the lockdown period where we lost momentum and drive, we managed to get back on top by arranging Zoom meetings over the weekends. These meetings worked both as motivation to stay on top of the project and to give each other the feedback and better communicate our ideas and our thinking. CAD-ing ideas and transferring CAD files also proved to be challenging, however with the weekend calls and tutorials we managed to convey our concepts well.

Our initial design process led us to creative and malleable ideas which we were able to develop as we further researched into our own subsections. The morphological analysis and Pugh's matrix were very well utilised to figure out what would work well for each individual subsystem, as well as the vehicle as a whole. We often found that space and interference was a constraint that appeared often further into the designing and CAD aspects of the project. Iteration and simultaneously developing ideas also made compatibility a continuous issue. For example, around halfway through the project timeline, we decided to change our hydraulic system from linear actuated to rotary actuated for ease and efficiency. This was due to the linear actuated hydraulic system needing twice as much space as the rotary, making the rotary a much more desirable change to our restricted quantity of space to allocate our sub systems, but the late switch costed us time as we needed to rearrange our powertrain and chassis assemblies to accommodate the change, and most work on the turning system had to start from square one. We also found that the chassis and engines had to undergo iterations on size and dimensions. The engine was changed from a GSXR 1000cc Suzuki, to a smaller and more compact 690cc KTM. This allowed us to allocate more space for compartments and other luxuries. The chassis underwent iterations to guarantee a good field of vision, this change did not impact any of the other subsystems, and overall fitted the frame in a better way. The more we developed our design, the more work we realised had to be done to produce the best possible product.

Overall, we managed to produce a well-designed and thought through product which we are proud of. In the next project we hope to tackle our goals with improved efficiency, through coordinating tasks and coming to a mutual agreement of what to work on. This way, we won't each end up at different stages in the design process throughout the project. We've also learned not to underestimate the difficulties that can arise when trying to merge CAD models from multiple people, so such tasks should be initiated earlier in the project timeline. We would also benefit from being more prepared during tutorial times and discussing which questions to ask before going to the tutorials. We have learnt to target these weaknesses by approaching the problems as a team and finding a mutual answer to these problems.

Appendix A – Calculations

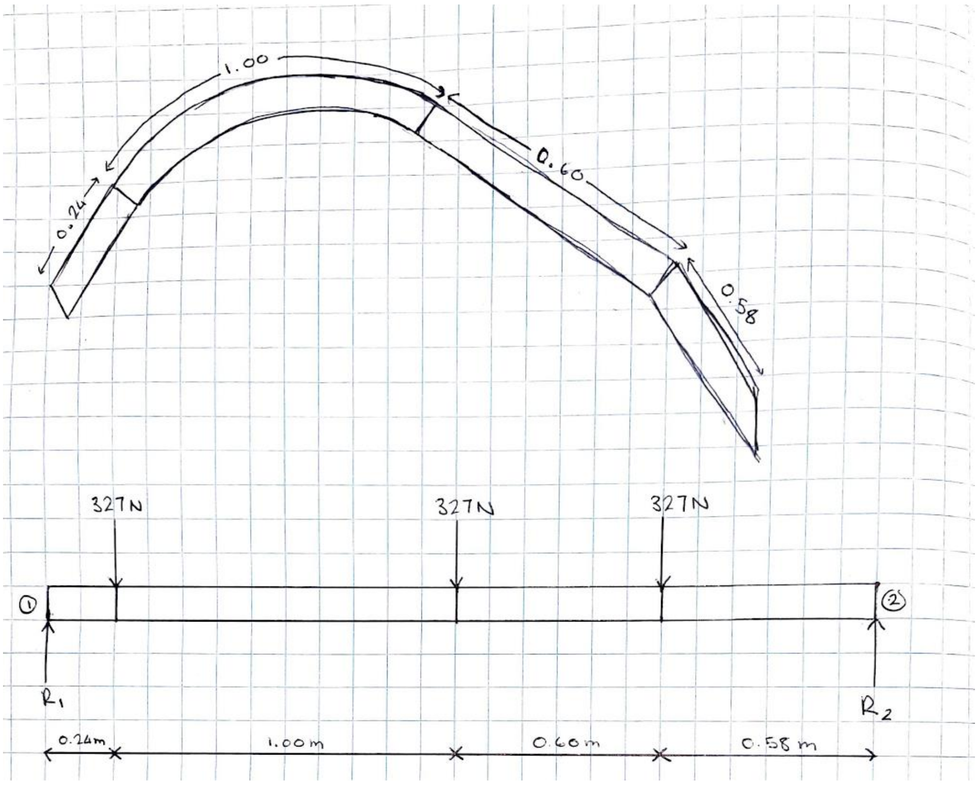
Chassis Calculations

Overhead Beam Stress Analysis

Approximating the overhead beams as simply supported 2.42 m, 25 $\times$ 75 $\times$ 6 mm steel beams (E ≅ 200 GPa), with 100kg load (aluminium vehicle body with windows, door components, and interior features) evenly distributed across the three mounting points. Assume each beam will take half the load. A safety factor of two is applied.

$$Total\ load\ for\ one\ beam\ = 100 \times 9.81 \times 2 \times 0.5\ =\ 981\ \text{N}$$

$$Load\ at\ each\ mounting\ point\ =\ 981 \div 3 = 327\ \text{N}$$



(1)

Figure 4: Dimensions and forces acting on overhead beam

$$\sum F_y = 0, \therefore\ R_1 + R_2 = 981\ N$$

$$\sum M_1 = 0, \therefore\ (327 \times 0.24) + (327 \times 1.24) + (327 \times 1.84)\ =\ 2.42R_2$$

$$1085.64\ = 2.42R_2$$

$$R_2 = 449\ N$$

$$\therefore R_1 = 532\ N$$

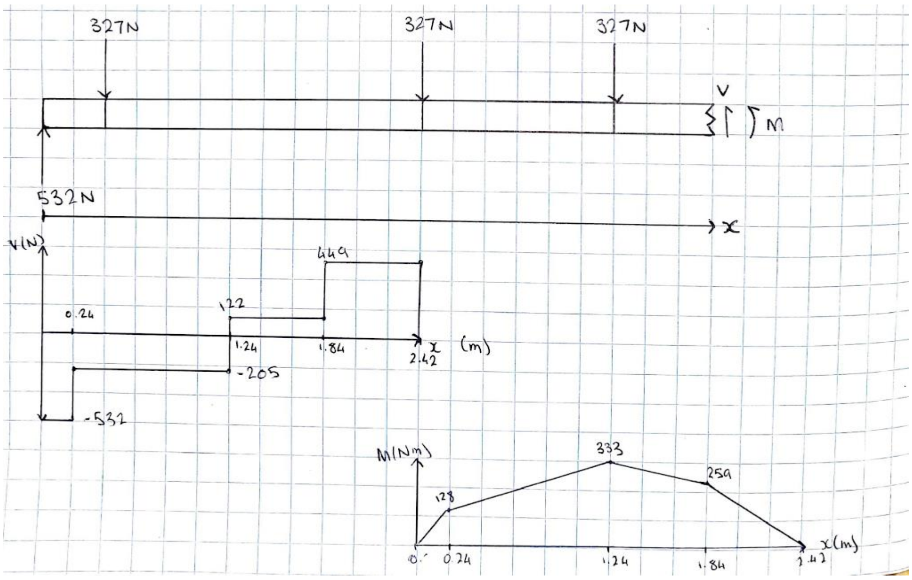


Figure 5: Shear force and beaning moment diagrams for overhead beam

$$M_{xz} = 532x - 327\langle x - 0.24\rangle - 327\langle x - 1.24\rangle - 327\langle x - 1.84\rangle = \mathrm{EI}_z \frac{d^2v}{dx^2} \quad (2)$$

$$\mathrm{EI}_z \frac{dv}{dx} = 266x^2 - 163.5\langle x - 0.24\rangle^2 - 163.5\langle x - 1.24\rangle^2 - 163.5\langle x - 1.84\rangle^2 + c$$

$$\mathrm{EI}_z v = \frac{266}{3}x^3 - 54.5\langle x - 0.24\rangle^3 - 54.5\langle x - 1.24\rangle^3 - 54.5 - \langle x - 1.84\rangle^3 + cx + d$$

v = 0 at x = 0 and x = 2.42, substitute x values into deflection equation:

$0 = 0 + d$

$\therefore d = 0$

$0 = 591.82 + 2.42 \times c$

$\therefore c = -244.5$

Assume maximum deflection ($\frac{dv}{dx} = 0$) when $x < 1.24$

$0 = 266x^2 - 163.5(x^2 - 0.48x + 0.0576) - 244.55$

$0 = 102.5x^2 + 78.48x - 253.97$

$\therefore x = 1.237\ m$

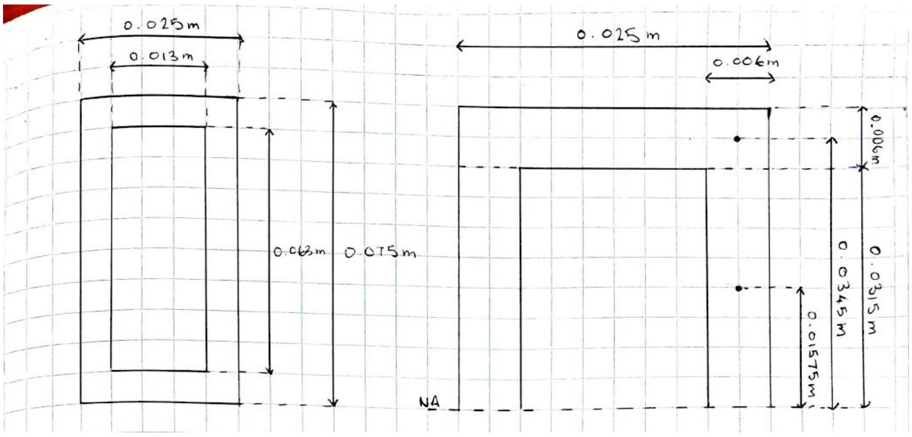


Figure 6: Cross-sectional dimensions of overhead beam

$$I_Z = \frac{0.025 \times 0.075^3 - 0.013 \times 0.063^3}{12} = 6.08 \times 10^7 \text{ m}^4$$

Substitute x, E, and Iz into deflection equation: (3)

$$\text{v}_{max} = \frac{1}{EI_Z}(-188.69) \ = \ -0.00155 \text{ m}$$

Maximum axial stress:

$$\sigma_{max} = \frac{\text{M}_{XZ,max} \times \text{y}'_{max}}{I_Z}, \qquad \text{M}_{XZ} = 332.68 \text{ Nm}, \qquad \text{y}'_{max} = 0.0375 \text{ m}$$

$$\therefore \ \sigma_{max} = 20.5 \times 10^{\ 6}\text{Pa}$$

Maximum shear stress:

$$\tau_{max} = \frac{V_{max} \times \text{A} \times \text{y}'}{I_Z \times t}, \qquad V_{max} = \ -532 \text{ N}, \qquad \text{t} \ = \ 0.0012 \text{ m}$$

$$\text{A} = 0.025 \times 0.006 + 2 \times 0.006 \times 0.0315 \ = 0.000528 \text{ m}^2$$

$$\text{y}' = \frac{0.025 \times 0.006 \times 0.0345 + 2 \times 0.006 \times 0.0315 \times 0.01575}{0.000528} = 0.021 \text{ m}$$

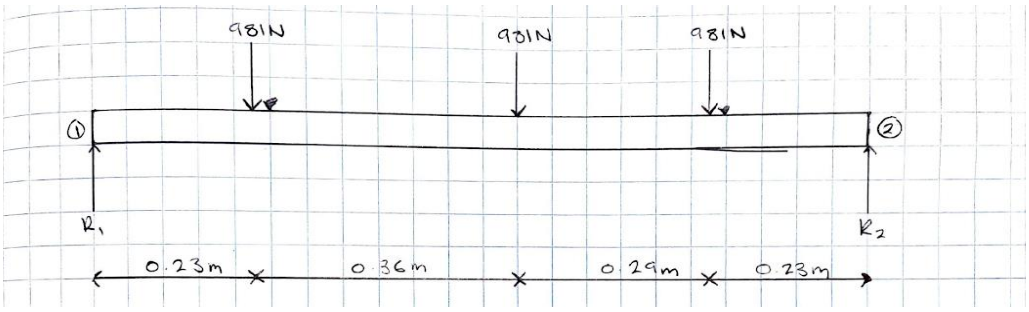
$$\therefore \ \tau_{max} = 811 \times 10^3 \text{ Pa}$$

Rear Axle Stress Analysis

47.625 mm diameter, 1.11 m steel axle (E ≅ 200 GPa) with 150kg load (powertrain components, turning system components, steel frame) evenly distributed across three contact points. Safety factor of two is applied.

$$Total\ load\ = 150 \times 9.81 \times 2\ =\ 2943\ \mathrm{N}$$

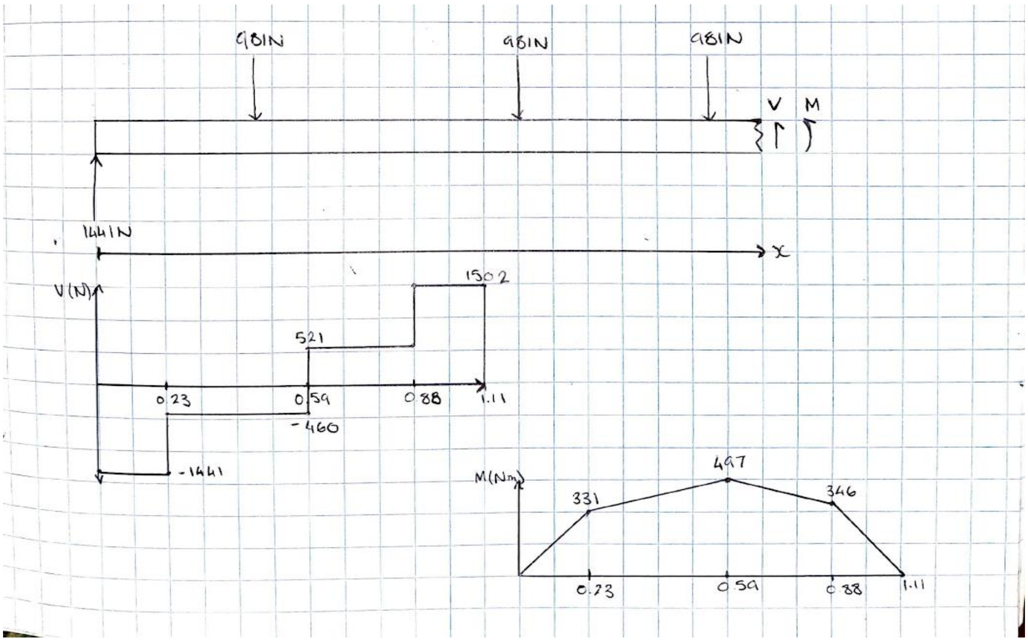
$$Load\ at\ each\ mounting\ point\ =\ 2943 \div 3 = 981\ \mathrm{N}$$



(4)

Figure 7: Dimensions and forces on rear axel

$$\sum F_y = 0, \qquad \therefore\ R_1 + R_2 = 2943\ N$$
$$\sum M_1 = 0, \therefore\ (981 \times 0.23\ ) + (981 \times 0.59) + (981 \times 0.88)\ =\ 1.11R_2$$
$$1667.7 = 1.11R_2$$
$$R_2\ = 1502\ N$$
$$\therefore R_1\ = 1441\ N$$



(5)

Figure 8: Shear force and bending moment diagrams

$$M_{xz} = 1441x - 981\langle x - 0.23\rangle - 981\langle x - 0.59\rangle - 981\langle x - 0.88\rangle = \mathrm{EI}_z \frac{d^2v}{dx^2}$$

$$\mathrm{EI}_z \frac{dv}{dx} = 720.5x^2 - 490.5\langle x - 0.23\rangle^2 - 490.5\langle x - 0.59\rangle^2 - 490.5\langle x - 0.88\rangle^2 + c$$

$\mathrm{v} = 0$ at $\mathrm{x} = 0$ and $\mathrm{x} = 1.11$, substitute x values into deflection equation:

$0 = 0 + d$

$\therefore d = 0$

$0 = 192.06 + 1.11c$

$\therefore c = -173.03$

Assume maximum deflection $\left(\frac{dv}{dx} = 0\right)$ when $x < 0.59$

$0 = 720.5x^2 - 490.5(x^2 - 0.46x + 0.0529) - 173.03$

$0 = 230x^2 + 225.63x - 198.97$

$\therefore x = 0.561\ m$

2nd Moment of Interia

$$I_Z = \frac{\pi}{4}(\frac{0.047625}{2})^4 = 2.53 \times 10^7\ \mathrm{m}^4$$

Substitute x, E, and I_Z into deflection equation:

$$\mathrm{v}_{max} = \frac{1}{EI_Z}(-60.59) = -0.00120\ \mathrm{m}$$

Maximum axial stress:

$$\sigma_{max} = \frac{\mathrm{M}_{XZ,max} \times \mathrm{y'}_{max}}{I_Z}, \qquad \mathrm{M}_{XZ} = 497.03\ \mathrm{Nm}, \qquad \mathrm{y'}_{max} = 0.238125\ \mathrm{m}$$

$$\therefore\ \sigma_{max} = 46.9 \times 10^{\,6}\mathrm{Pa} \tag{6}$$

Maximum shear stress:

$$\tau_{max} = \frac{V_{max} \times \mathrm{A} \times \mathrm{y'}}{I_Z \times t}, \qquad V_{max} = 1502\ \mathrm{N}, \qquad \mathrm{t} = 0.047625\ \mathrm{m}$$

$$\mathrm{A} = \frac{\pi}{8}(0.047625)^2 = 0.000891\ \mathrm{m}^2$$

$$\mathrm{y'} = \frac{2 \times 0.047625}{3\pi} = 0.0101\ \mathrm{m}$$

$$\therefore\ \tau_{max} = 1.12 \times 10^6\ \mathrm{Pa}$$

Drivetrain Calculations

Fuel Tank Calculations

$$Mass\ Ratio = 2.26{:}1$$
$$Fuel\ Consumption\ of\ KTM\ 690 = \frac{3.87\ L}{100\ Km}$$
$$Fuel\ Consumption = 2.26 \times 3.87$$
$$Fuel\ Consumption = \frac{8.76\ L}{100\ Km} \tag{7}$$
$$Tank\ size = Fuel\ consumption \times Distance$$
$$Tank\ size = \frac{8.76}{100} \times 250$$
$$Tank\ size = 22\ L\ (minimum)$$

Clutch Calculations

$$Safety\ Factor = 1.5$$
$$Torque = 75\ Nm$$
$$Maximum\ Pressure = 5000000$$
$$Friction\ coefficient =$$
$$Inner\ radius = \sqrt{\frac{1}{3}} \times Outer\ radius$$
$$S.F \times \tau = P_{max} \times \pi \times \mu \times r_i \times (r_o{}^2 - r_i{}^2)$$
$$\therefore \sqrt{\frac{1}{3}} \times r_o \times \left[r_o{}^2 - \left(\sqrt{\frac{1}{3}} \times r_o \right)^2 \right] = \frac{S.F \times \tau}{P_{max} \times \pi \times \mu} \quad (8)$$
$$r_o \times \left(r_o{}^2 - \frac{1}{3} r_o{}^2 \right) = 1.836 \times 10^{-4}$$
$$\frac{2}{3} r_o{}^3 = 1.836 \times 10^{-4}$$
$$r_o = 65.1\ mm$$
$$\therefore r_i = 37.6\ mm$$

Force Requirement on clutch pedal

$$F = 2 \times \pi \times P_{max} \times r_i (r_o - r_i)$$
$$F = 2 \times \pi \times 500000 \times 0.0376(0.0651 - 0.0376)$$
$$F = 3245.5N \quad (9)$$
$$Mechanical\ Advantage = \frac{3245.5}{250} = 13$$
$$Ratio = 1:13$$

Engine Calculations

$$
\begin{gathered}
Engine\ Power = 54\ KW \\
Mass = 500\ kg \\
Drag\ Coefficient = 0.3 \\
Coefficient\ of\ rolling\ resistance = 0.015 \\
Frontal\ Area = 1.2m^2 \\
Transmission\ efficiency = 90\% \\
Density\ of\ Air = 1.225\ {}^{kg}/_{m^3} \\
P_u = \eta \times P_{Engine} \\
P_u = 0.9 \times 54 \\
P_u = 48.6\ KW \\
F_{Normal} = m \times g \\
F_{Normal} = 500 \times 9.81 \\
F_{Normal} = 4905\ N \\
F_D = \frac{1}{2} \times \rho \times C_D \times A \times V^2 \\
F_{rr} = C_{rr} \times F_N \\
P_u = (F_D + F_{rr}) \times V \\
48600 = \left[\left(\frac{1}{2} \times 1.225 \times 0.30 \times 1.2 \times V^2\right) + (0.015 \times 4905)\right] \times V \\
48600 = (0.2205 \times V^3) + (73.575 \times V) \\
\therefore V = 59\frac{m}{s} = 212\frac{km}{hr}
\end{gathered} \tag{10}
$$

Turning System Calculations

Power of actuator

$$
\begin{gathered}
Power = Torque \times Angular\ Velocity \\
P = \tau \times \omega,\ \text{where } \omega = \frac{\theta}{t} \\
P = 500 \times \left(\frac{\pi}{4} \times \frac{1}{2}\right) \\
P = 196.35\ W
\end{gathered} \tag{11}
$$

$$
\begin{gathered}
P = 847.4 \times \left(\frac{\pi}{4} \times \frac{1}{2}\right) \\
P = 332.8\ W
\end{gathered} \tag{12}
$$

Flowrate in the system

$$
\begin{gathered}
\Delta P \times \dot{V} = \tau \times \omega \\
\therefore\ \dot{V} = \frac{\tau \times \omega}{\Delta P} \\
\dot{V} = \frac{847.4 \times \left(\frac{\pi}{4} \times \frac{1}{2}\right)}{5 \times 10^6} \\
\dot{V} = 6.65 \times 10^{-6}\ {}^{m^3}/_{s}
\end{gathered} \tag{13}
$$

Pressure within the system

$$
\begin{gathered}
Change\ in\ Pressure = \frac{Mechanical\ Power}{Flow\ rate} \\
\Delta P = \frac{P}{\dot{V}} \\
\Delta P = \frac{332.8}{8.33 \times 10^{-5}} \\
\Delta P = 3.995\ MPa \\
\Delta P \approx 4\ MPa
\end{gathered} \tag{14}
$$

Pump

$$
\begin{gathered}
RPM = \frac{Flow\ rate\ [GPM] \times 231}{Displacement\ [in^3]} \\
RPM = \frac{1.32 \times 231}{4.58} \\
RPM = 66.6 \\
66.6\ RPM = 6.97 \frac{rad}{s}
\end{gathered} \tag{15}
$$

$$
\begin{gathered}
P = \tau \times \omega \\
\therefore \tau = \frac{P}{\omega} \\
\tau = \frac{332.8}{6.97} \\
\tau = 47.7\ Nm
\end{gathered} \tag{16}
$$

Accumulator

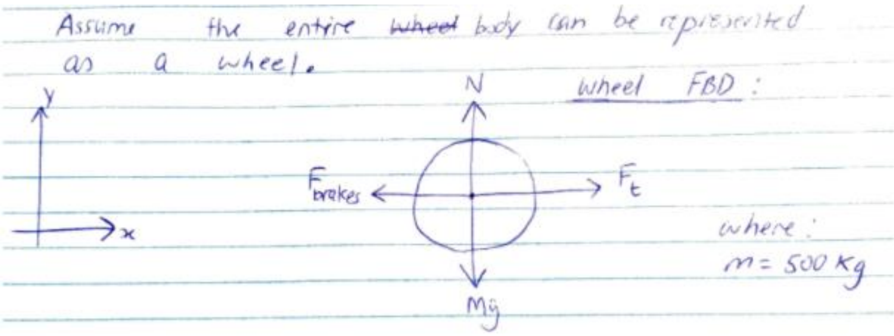
$$
\begin{gathered}
\dot{V} = \frac{V}{t_{accumulator}} \\
t_{accumulator} = \frac{V}{\dot{V}} \\
t_{accumulator} = \frac{2.3 \times 60}{5} \\
t_{accumulator} = 27.6\ s
\end{gathered} \tag{17}
$$

Timing

$$
\begin{gathered}
\omega = \frac{\theta}{t_{270^\circ}} \\
\therefore t_{270^\circ} = \frac{\theta}{\omega} \\
t_{270^\circ} = \frac{3\pi}{2} \times \frac{8}{\pi} \\
t_{270^\circ} = 12\ s \\
t_{Total} = t_{270^\circ} + t_{accumulator} \\
t_{Total} = 12 + 27.6 \\
t_{Total} = 39.6\ s
\end{gathered} \tag{18}
$$

Brake Calculations

Front and Rear brakes



(19)

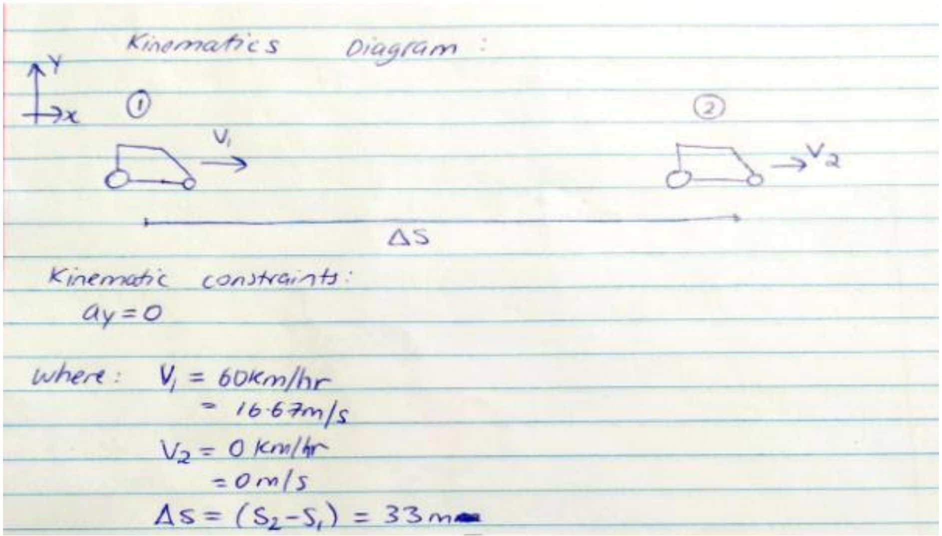
Figure 9: Force analysis of the wheel

$$Assuming\ \text{wheel does not move in the y} - \text{direction}\ \therefore\ \text{a}_\text{y} = 0$$

$$\sum F_y = ma_y = N - mg$$

$$N = (500) \times (9.81)$$

$$N = 4905\ N$$



(20)

Figure 10: Kinematics diagram

$$Where: v_1 = 60\frac{km}{hr} = 16.67\frac{m}{s}$$

$$v_2 = 0\frac{m}{s}$$

$$s_1 = 0\ m$$

$$s_2 = 33\ m$$

Using kinematic equations:

$$v_2^2 - v_1^2 = 2 \times a \times (s_1 - s_2)$$

$$\therefore a = \frac{v_2^2 - v_1^2}{2 \times (s_2 - s_1)}$$

$$\therefore a = -4.210 \ \frac{m}{s^2}$$

Continue force calculations:

$$\sum F_x = ma_x = F_{brakes}$$

$$\therefore F_{brakes} = (500) \times (4.21)$$

$$\therefore F_{brakes} = 2105 \ N$$

Apply a safety factor of 2 to account for wet roads or emergencies: $F_{brakes} = 2105 \times 2 = 4210 \ N$

Assume front wheel takes 60% of total brake force:

$$\therefore F_{frontBrakes} = 4210 \times 60\%$$

$$\therefore F_{frontBrakes} = 2526 \ N \tag{21}$$

Front brakes have 2 disk brakes; each disk takes 1263 N of brake force

Therefore, rear brakes take 40% of total brake force.

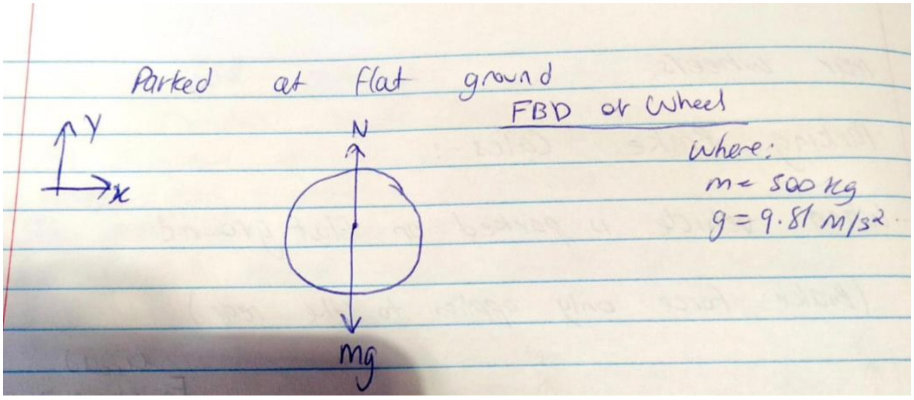
$$\therefore F_{rearBrakes} = 4210 \times 40\%$$

$$F_{rearBrakes} = 1684 \ N \tag{22}$$

The rear brakes only have a single disk each; each disk takes 842 N of force.

Parking brakes

Parked on flat ground



(23)

Figure 11: FBD of stationary wheel on flat ground

When on flat ground, there are no horizontal forces ∴ if there was a force applied, it would need to overcome weight force.

$$\sum F_y = 0 = N - mg$$

$$N = (500) \times (9.81)$$

$$N = 4905\ N$$

Therefore, braking force required is 4905N at the rear OR 2452.5N on each rear disk.

Parked on a 45° incline:

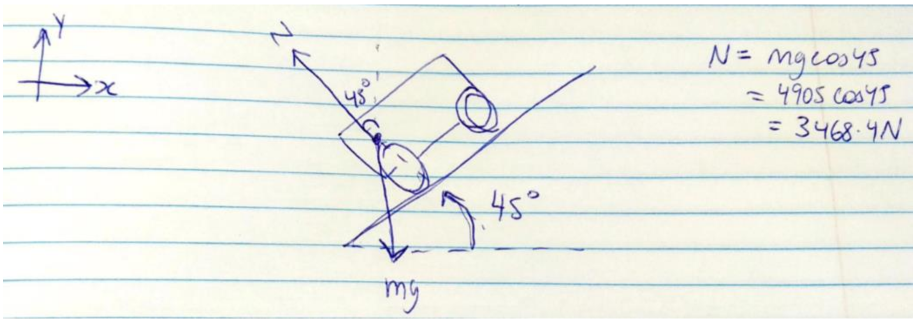


Figure 12: FBD of stationary car on 45° incline

Assume vehicle is stationary:

$$\sum F_y = 0 = F_{brakes},y + N \times cos\emptyset - mg$$

$$F_{brakes},y = mg - N \times cos\emptyset$$

$$F_{brakes},y = 2452\ N \tag{24}$$

$$\sum F_x = 0 = F_{brakes},x - N \times\ sin\emptyset$$

$$F_{brakes},x = N \times sin\emptyset$$

$$F_{brakes},x = 3468.4 \times \sin 45$$

$$F_{brakes},x = 2452\ N$$

$$\therefore F_{brakes} = \sqrt{(F_{brakes},y)^2 + (F_{brakes},x)^2}$$

$$F_{brakes} = \sqrt{(2452)^2 + (2452)^2}$$

$$F_{brakes} = 3467\ N$$

Therefore, for a 45° jill, the rear brake force required is 3467 N OR 1734 N to each rear disk.

Mechanical Enhancer

$$Mechanical\ Enhancer\ applicable\ force\ =\ Required\ braking\ force$$

Due to front braking force being greater than the rear braking force, calculating front mechanical enhancer can also apply to the rear.

$$M.E \times 400 = 1263\ N \tag{25}$$

$$\therefore M.E = \frac{1263}{400}$$

$$M.E = 3.2$$

Verification

Braking force at the front is greatest, therefore, we analyse the front to find maximum pressure at the cylinder.

$$Cylinder\ diameter = 0.0175\ m$$

$$Cylinder\ area = \pi \times \left(\frac{d}{2}\right)^2$$

$$\therefore = \pi \times \left(\frac{0.0175}{2}\right)^2$$

$$Cylinder\ Area = 2.405 \times 10^{-4}\ m^2 \tag{26}$$

$$F_{braking} = 1263\ N$$

Therefore, the pressure calculation proceeds as:

$$Pressure\ = \frac{Brake\ force}{Cylinder\ area}$$

$$P\ = \frac{1263}{2.405 \times 10^{-4}}$$

$$= 5.25 \times 10^6 = 5.25\ MPa$$

Appendix B – Figures

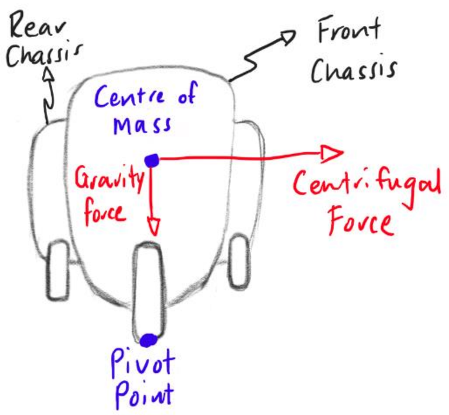


Figure 13: FBD showing cause of toppling

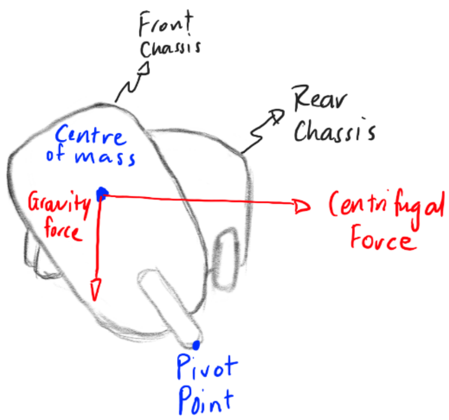


Figure 14: FBD showing solution to toppling

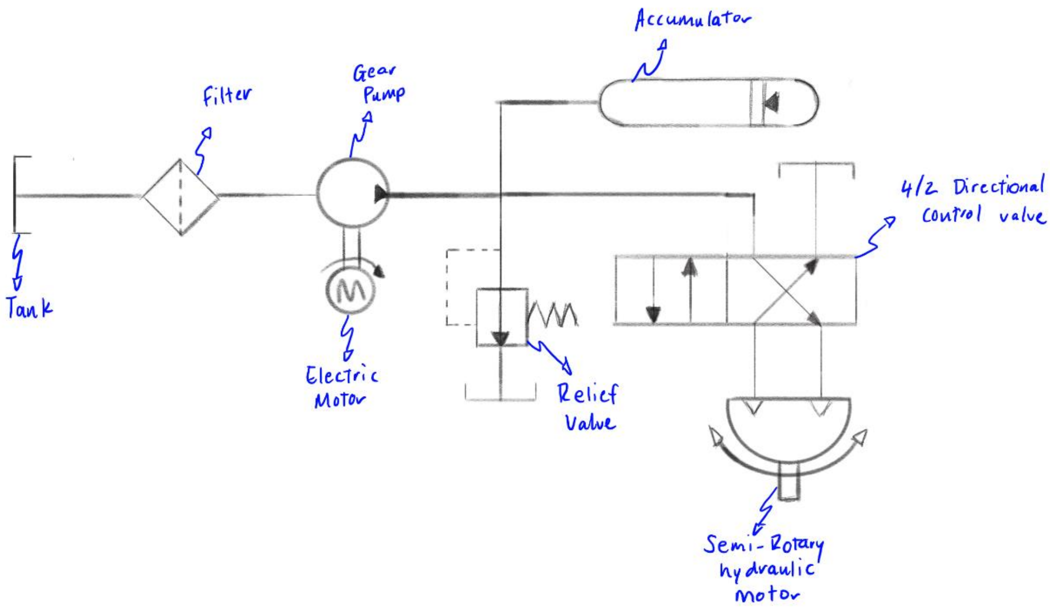
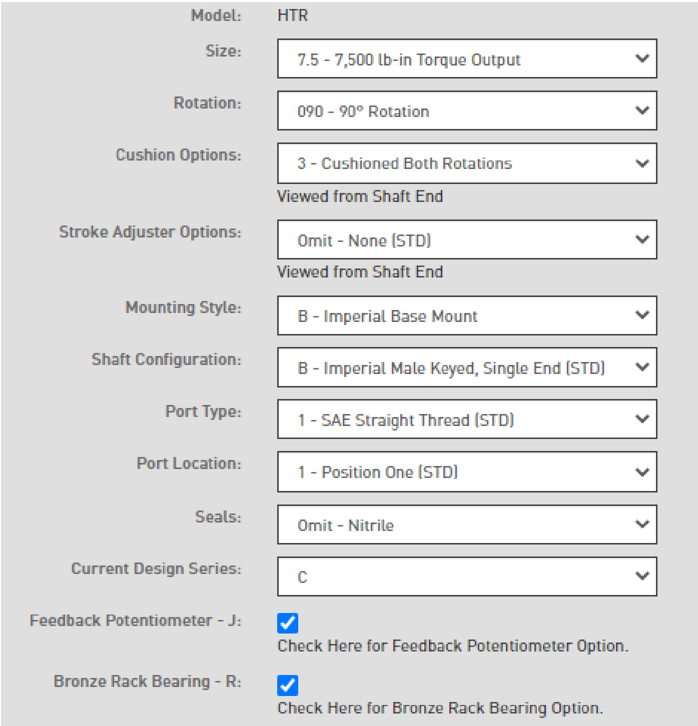


Figure 15: Hydraulic circuit Schematic diagram



Model:	HTR
Size:	7.5 - 7,500 lb-in Torque Output
Rotation:	090 - 90° Rotation
Cushion Options:	3 - Cushioned Both Rotations Viewed from Shaft End
Stroke Adjuster Options:	Omit - None (STD) Viewed from Shaft End
Mounting Style:	B - Imperial Base Mount
Shaft Configuration:	B - Imperial Male Keyed, Single End (STD)
Port Type:	1 - SAE Straight Thread (STD)
Port Location:	1 - Position One (STD)
Seals:	Omit - Nitrile
Current Design Series:	C
Feedback Potentiometer - J:	Check Here for Feedback Potentiometer Option.
Bronze Rack Bearing - R:	Check Here for Bronze Rack Bearing Option.

Figure 16: Configuration of semi-rotary hydraulic actuator

Figure 18: Brake caliper from ISR catalogue

Figure 17: Master Cylinder from ISR catalogue

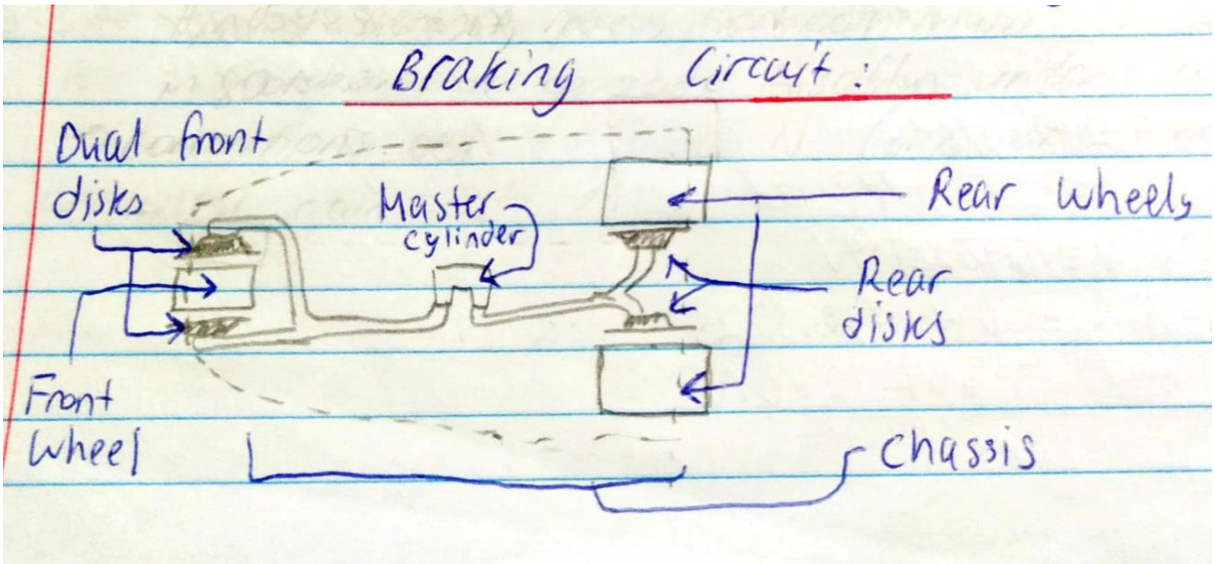


Figure 19: Braking Circuit

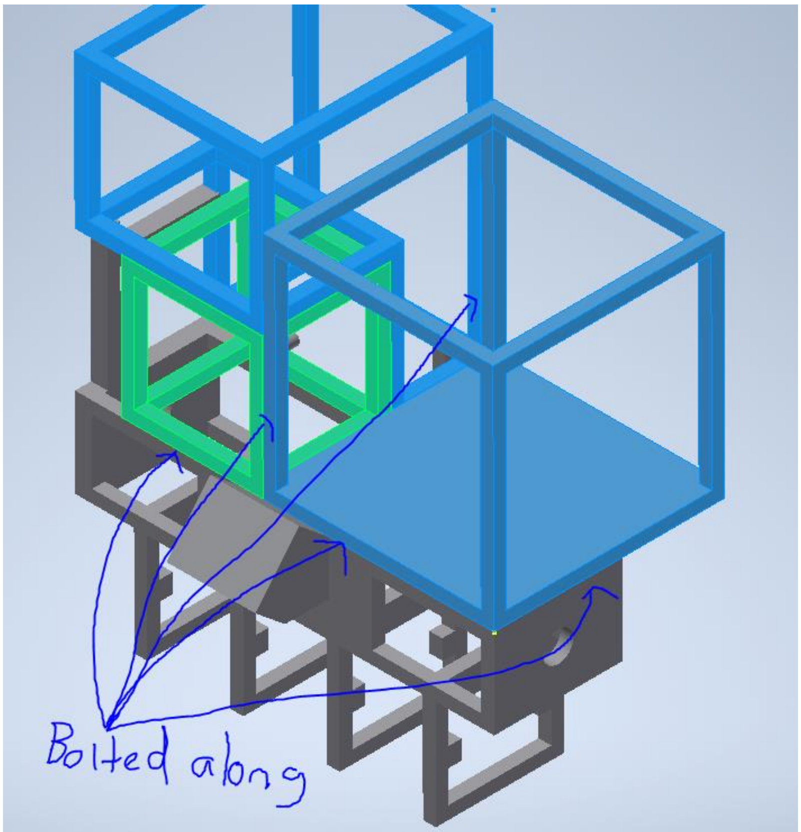


Figure 20: Motor Housing (blue right), Fuel Tank (blue left), Gearbox Housing (green)

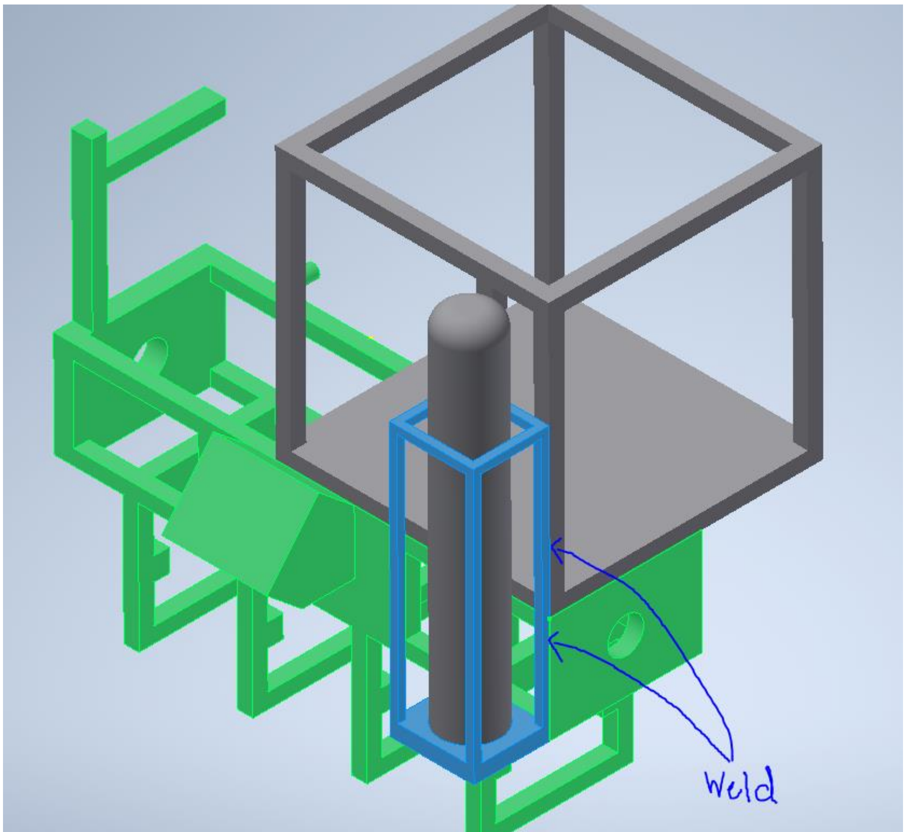


Figure 21: Motor Housing (grey), Accumulator Housing Blue, Main Subframe (green)

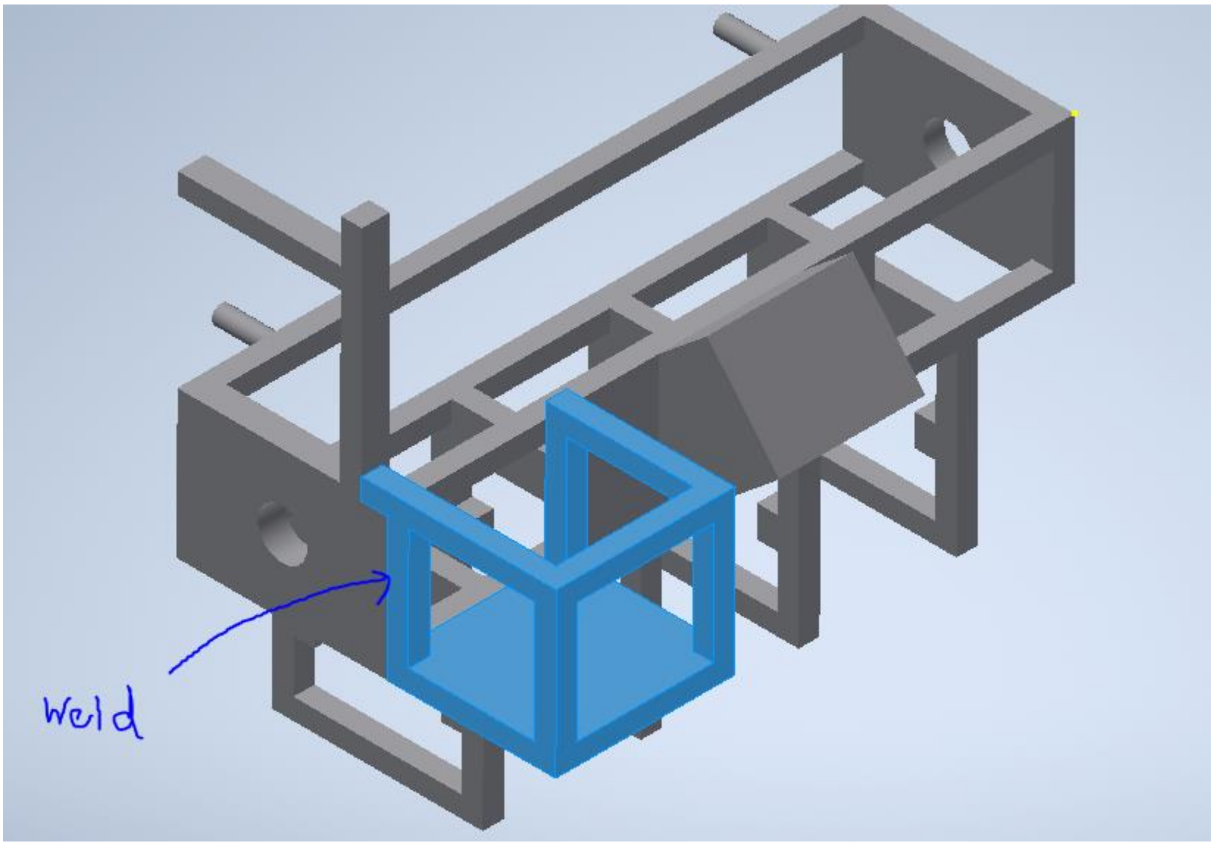
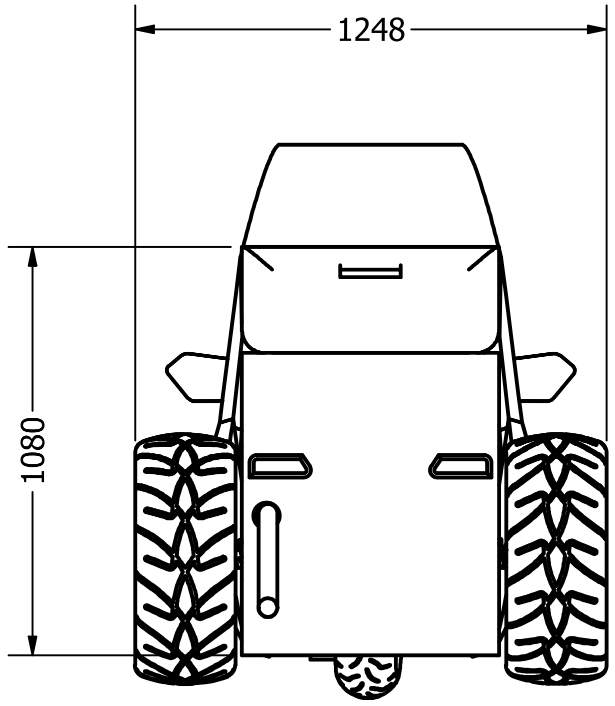


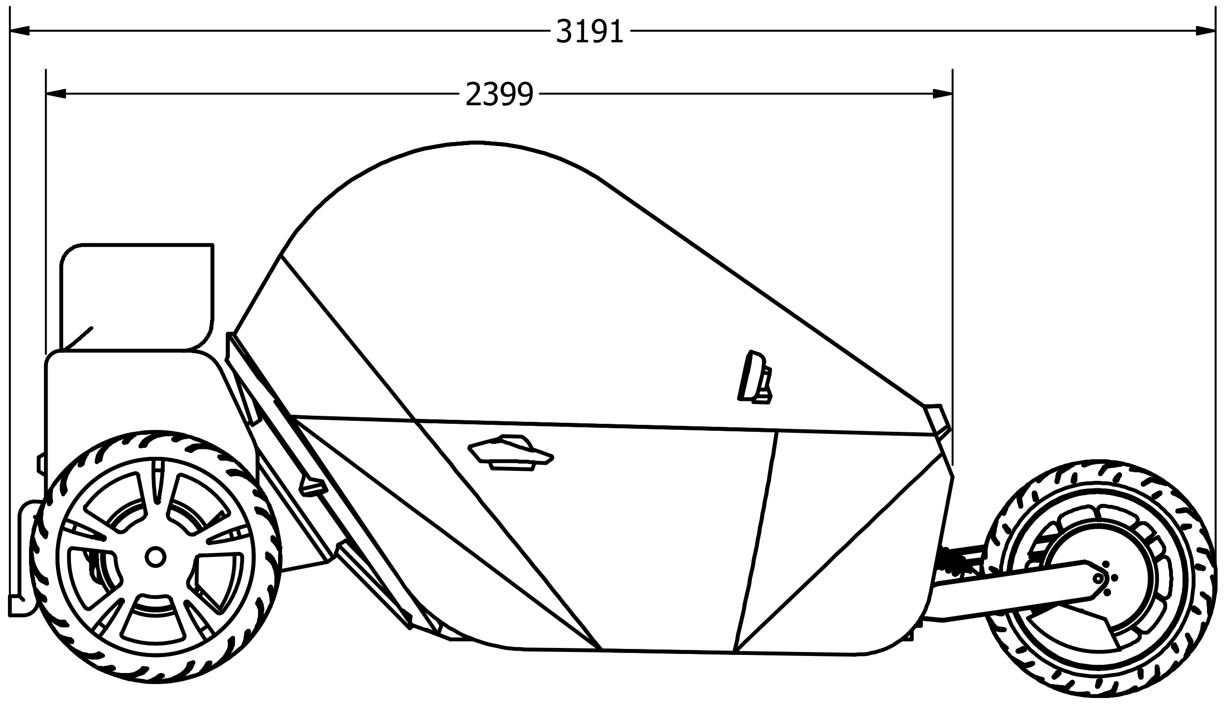
Figure 22: Main Subframe (grey), Accumulator Housing (Blue)

Appendix C – Engineering Drawings

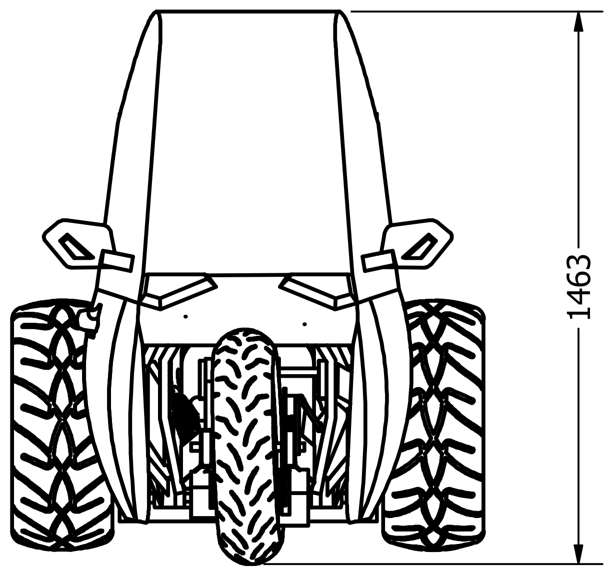
The engineering Drawings are all attached on the preceding pages. References in the text are given of the page number rather than the figure number.



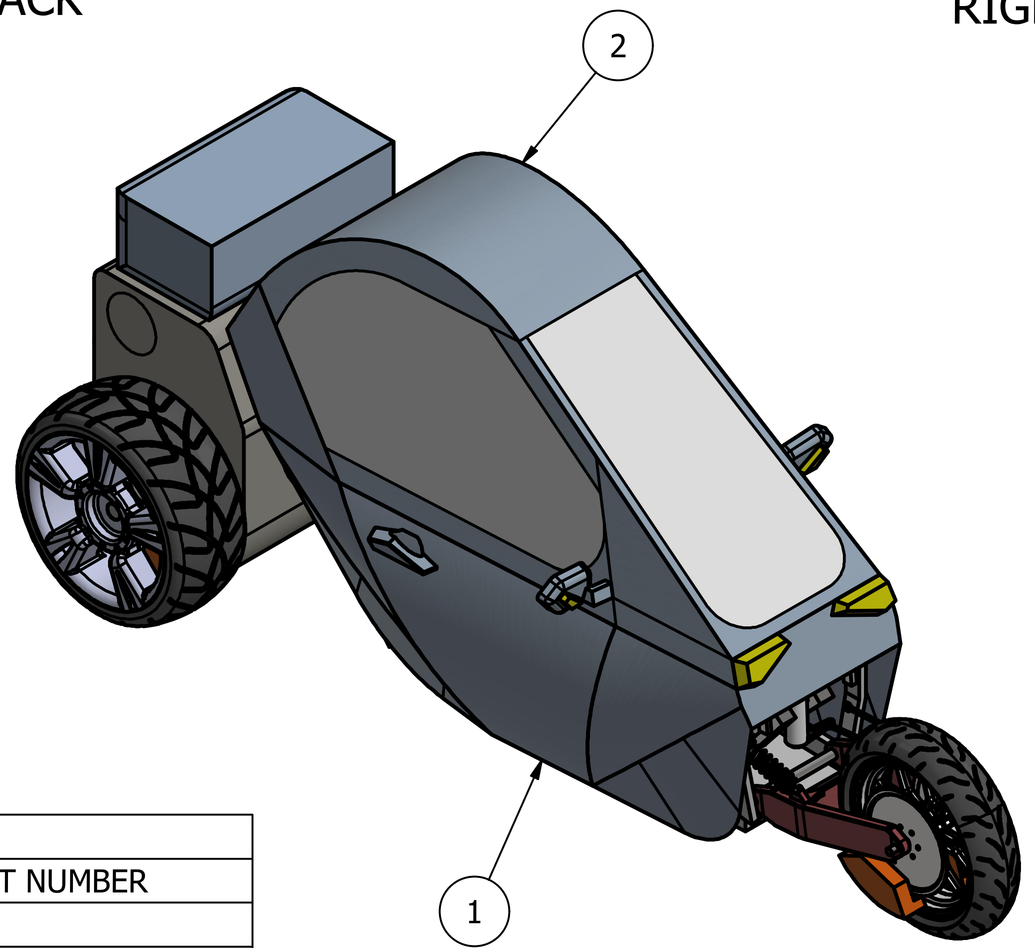
BACK

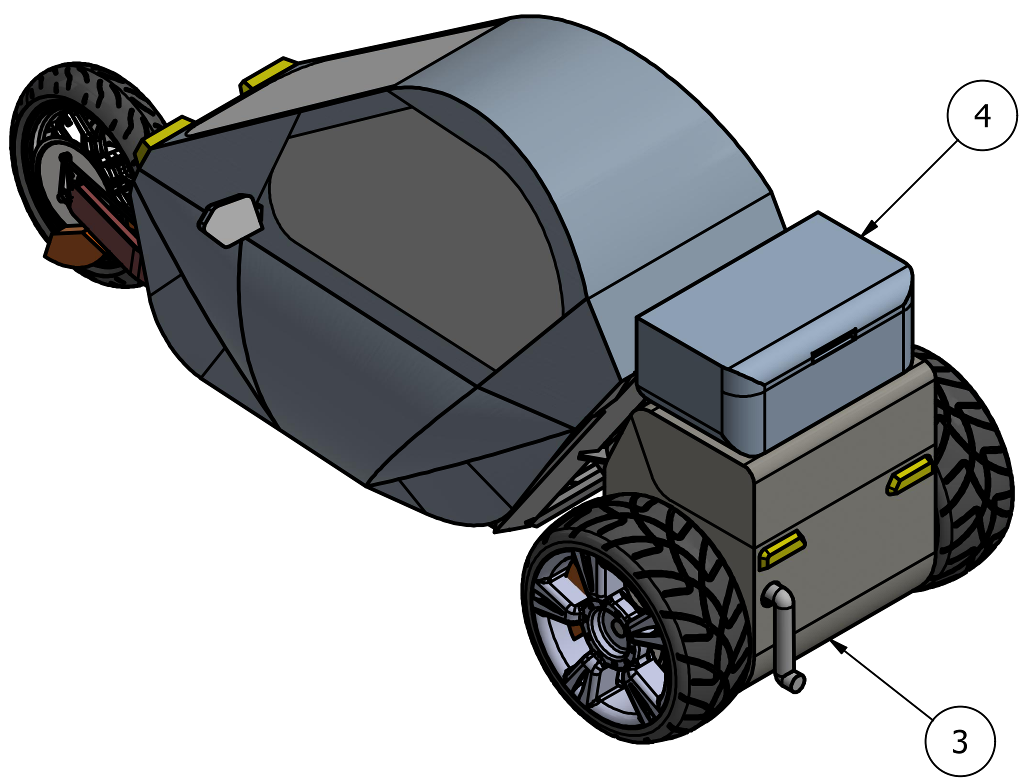


RIGHT SIDE



FRONT





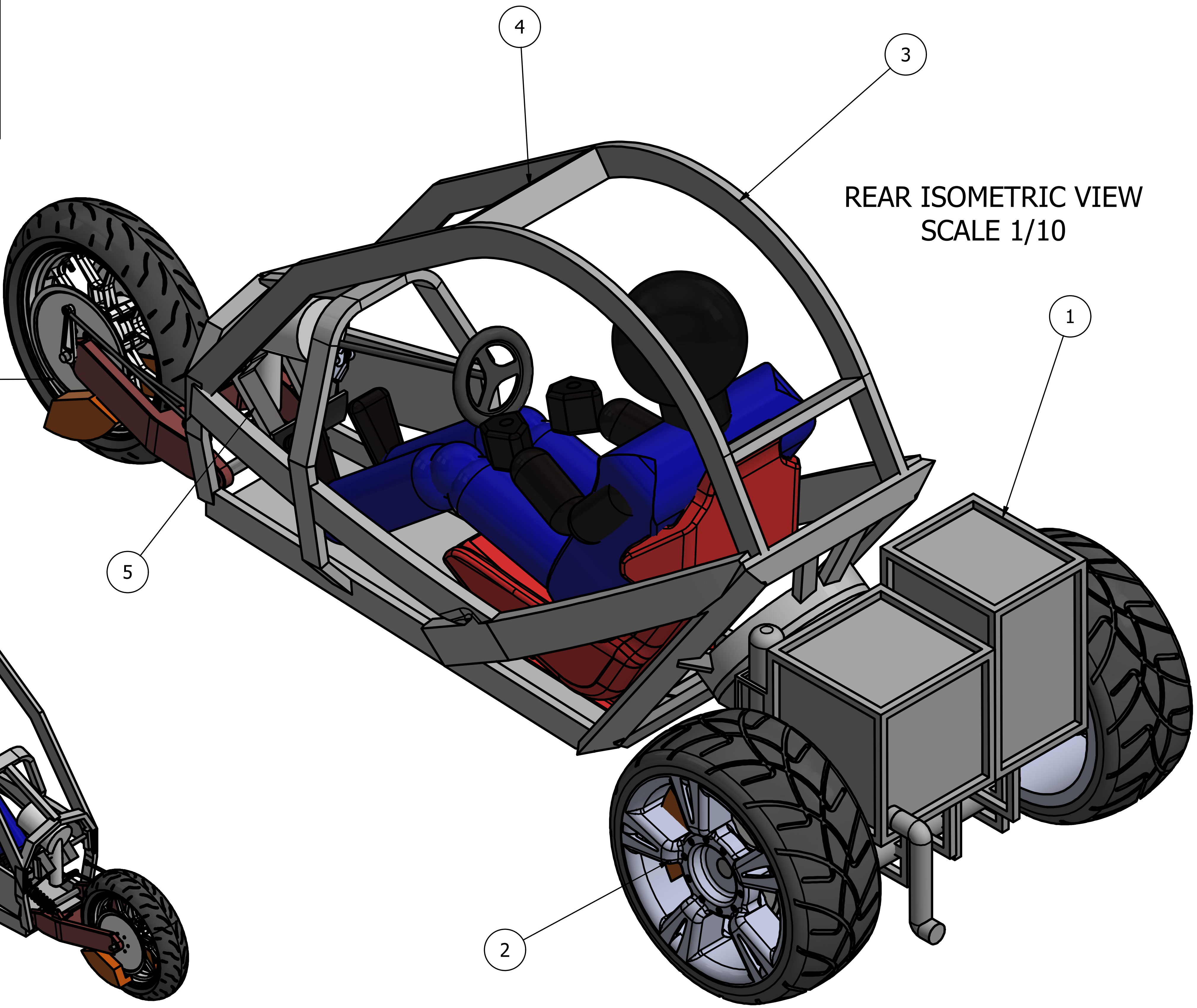
PARTS LIST	
ITEM	PART NUMBER
1	Door
2	Front Chassis Shell
3	Rear Chassis Shell
4	Removable Storage Case

THE UNIVERSITY OF AUCKLAND
NEW ZEALAND
Te Whare Wananga o Tamaki Makaurau

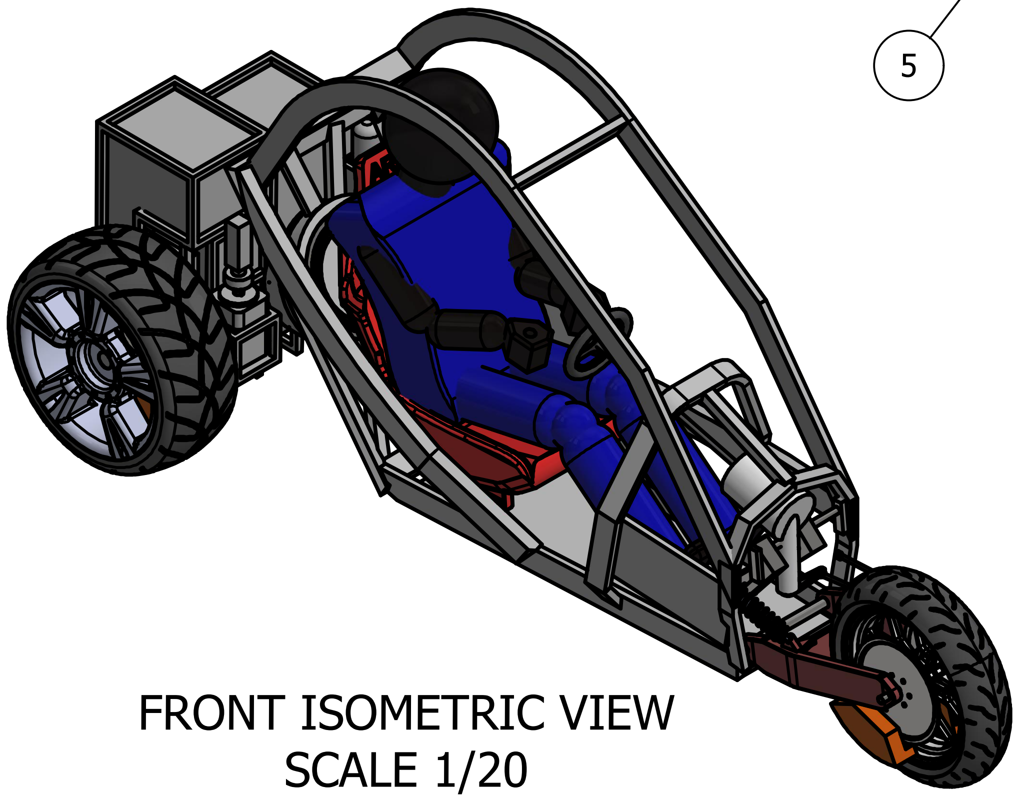
REV A	DESIGNED BY Emma Sio	TOLERANCES .X ±0.5mm .XX ±0.25mm .XXX ±0.10mm ANGLE ±0.5°	MATERIAL	PART NAME
REV B	APPROVED BY Nabeel Azafer		PROJECT Urban Transport Design Project	FILE NAME Full assembly with shell
REV C				3RD ANGLE PROJECTION

ALL DIMENSIONS IN MILLIMETRES UNLESS STATED | DO NOT SCALE DRAWING | SCALE 1:20 | SHEET 1 OF 1 | REV | A3

PARTS LIST		
ITEM	QTY	PART NUMBER
1	1	Rear Chassis Assembly
2	2	Rear Brake Assembly
3	2	Overhead Beam
4	2	Cross Beam
5	2	Brake and Clutch Pedals
6	2	Front Brake Assembly



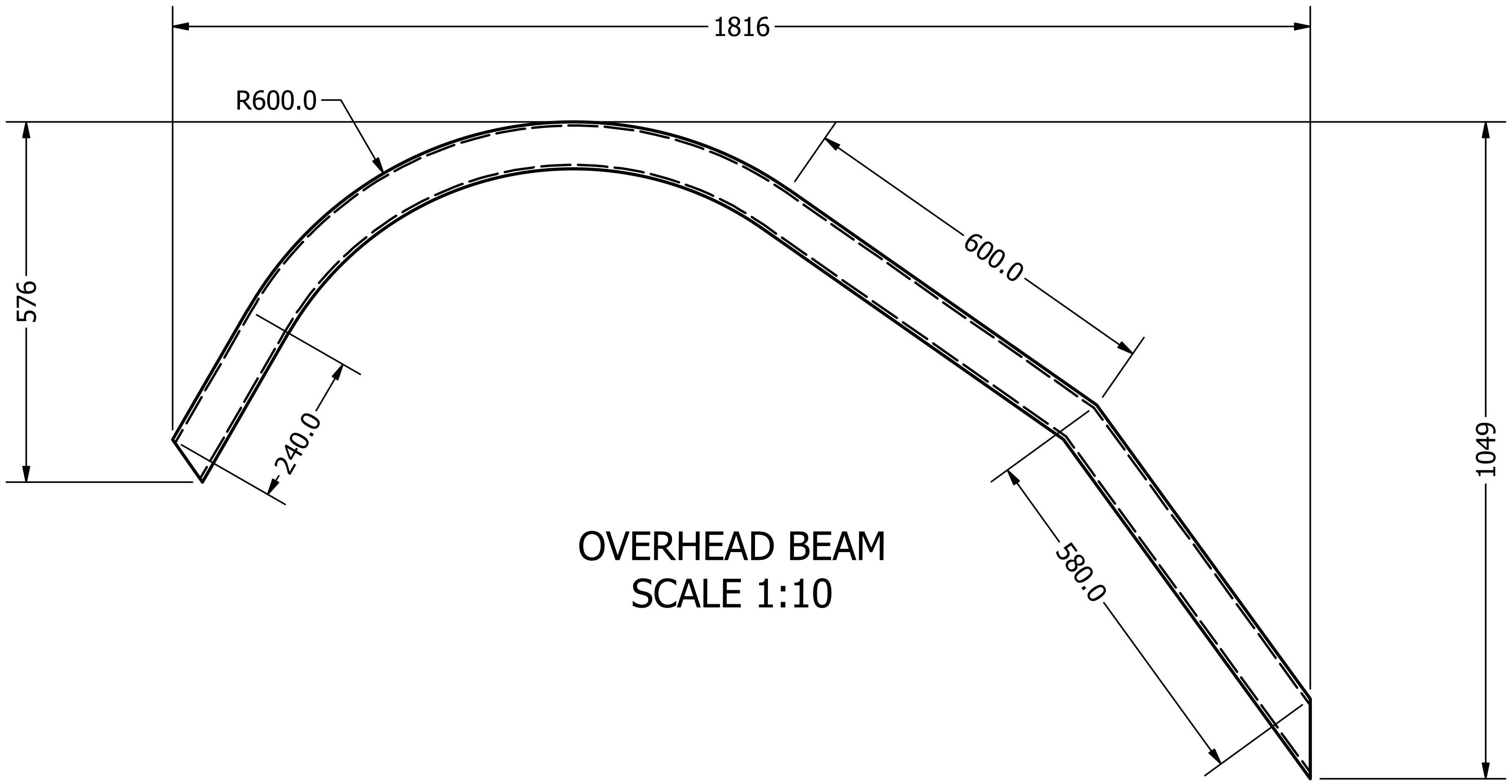
REAR ISOMETRIC VIEW
SCALE 1/10



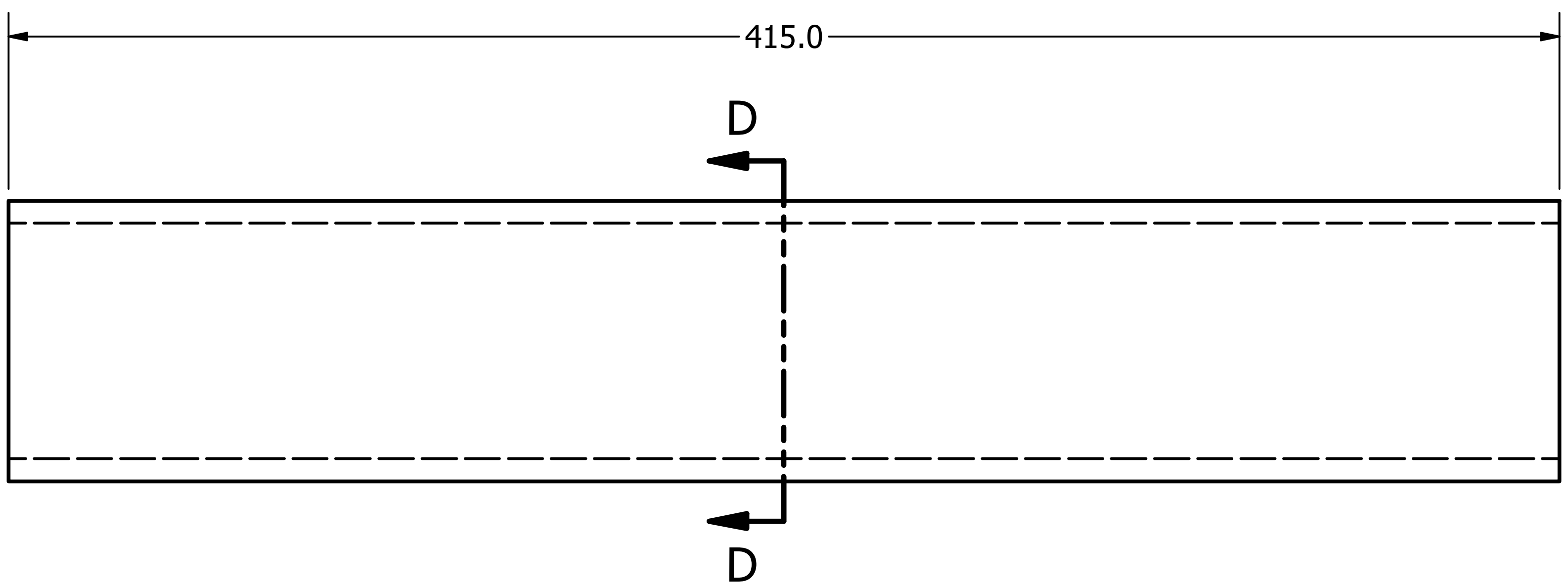
FRONT ISOMETRIC VIEW
SCALE 1/20

THE UNIVERSITY OF AUCKLAND NEW ZEALAND Te Whare Wānanga o Tāmaki Makaurau	REV A	DESIGNED BY Emma Sio	TOLERANCES .X ±0.5mm .XX ±0.25mm .XXX ±0.10mm ANGLE ±0.5°
	REV B	APPROVED BY Nabeel Azafer	MATERIAL
	REV C		PROJECT Urban Transport Design Project

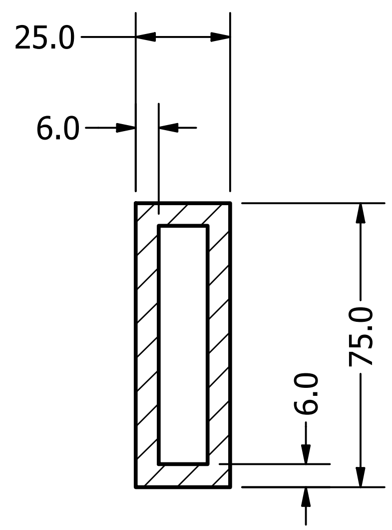
PART NAME		3RD ANGLE PROJECTION
FILE NAME Full assembly without shell		
ALL DIMENSIONS IN MILLIMETRES UNLESS STATED	DO NOT SCALE DRAWING	SCALE 1/20
SHEET 1 OF 1	REV	A3



OVERHEAD BEAM
SCALE 1:10



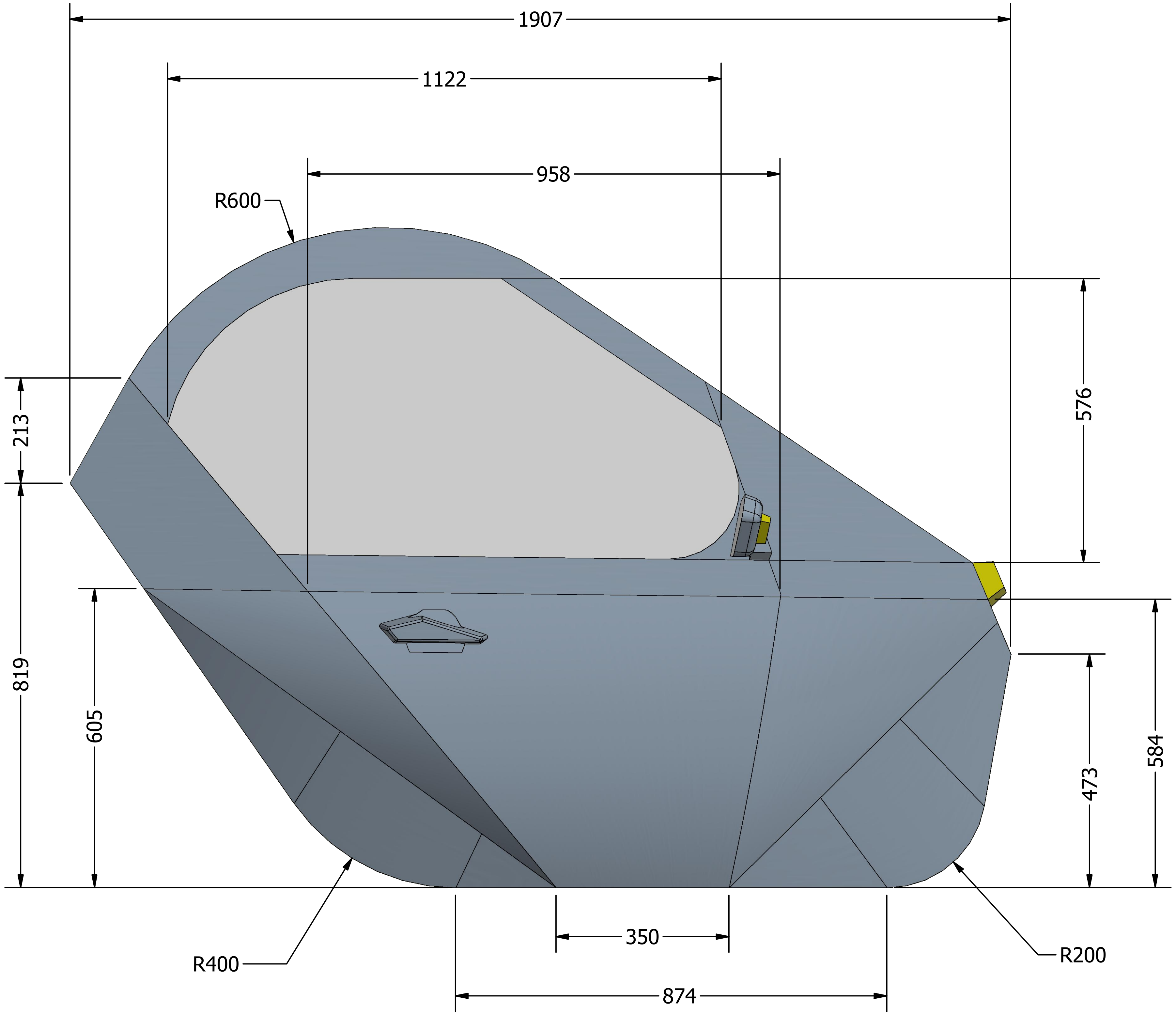
CROSS BEAM
SCALE 1:2



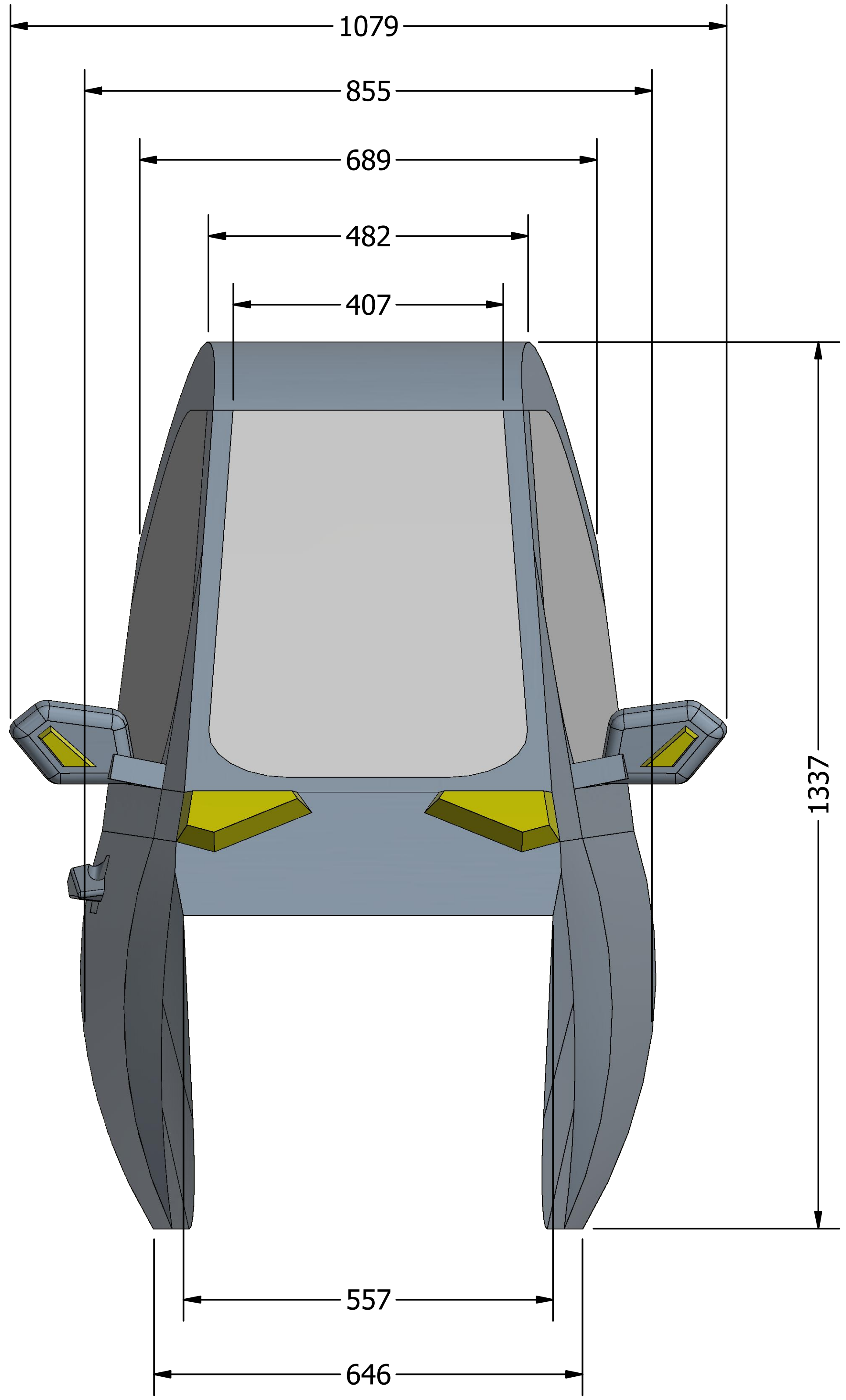
SECTION D-D
SCALE 1:2

NOTE: Both beams have same section

THE UNIVERSITY OF AUCKLAND NEW ZEALAND Te Whare Wānanga o Tāmaki Makaurau	REV A	DESIGNED BY Emma Sio	TOLERANCES .X ±0.5mm	MATERIAL Steel	PART NAME Overhead and cross beam
	REV B		.XX ±0.25mm .XXX ±0.10mm	PROJECT Urban Transport Design Project	FILE NAME Beams
	REV C	APPROVED BY Nabeel Azafer	ANGLE ±0.5°		3RD ANGLE PROJECTION
			ALL DIMENSIONS IN MILLIMETRES UNLESS STATED	DO NOT SCALE DRAWING	SCALE 1:10 · SHEET 1 OF 1 · REV · A3



RIGHT SIDE



FRONT

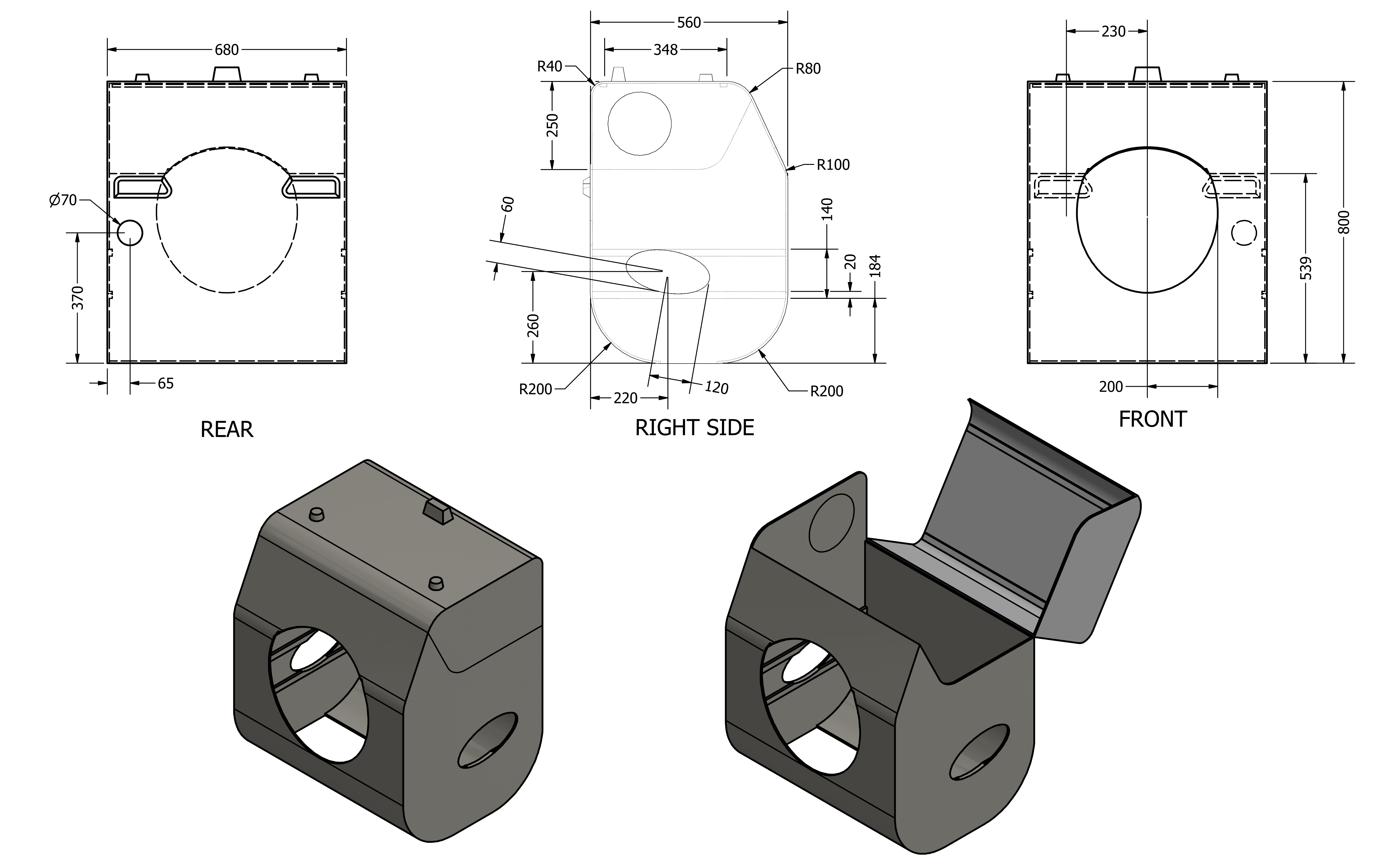
THE UNIVERSITY OF AUCKLAND NEW ZEALAND Te Whare Wānanga o Tāmaki Makaurau	REV A	DESIGNED BY Emma Sio	TOLERANCES .X ±0.5mm .XX ±0.25mm .XXX ±0.10mm ANGLE ±0.5°	MATERIAL
	REV B	APPROVED BY Nabeel Azafer		PROJECT Urban Transport Design Project
	REV C			

PART NAME Front chassis shell and door

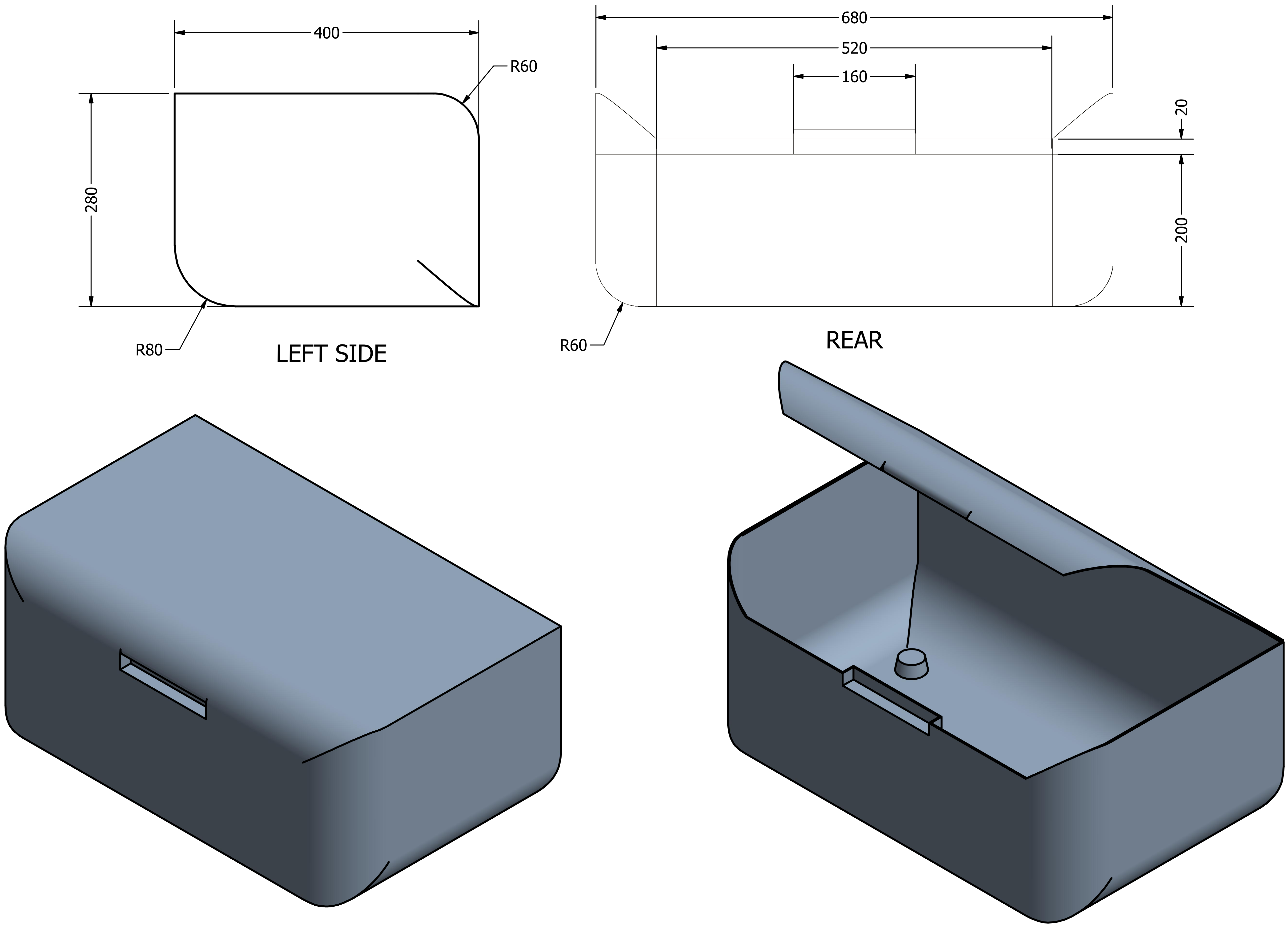
FILE NAME Front Chassis Shell

3RD ANGLE PROJECTION

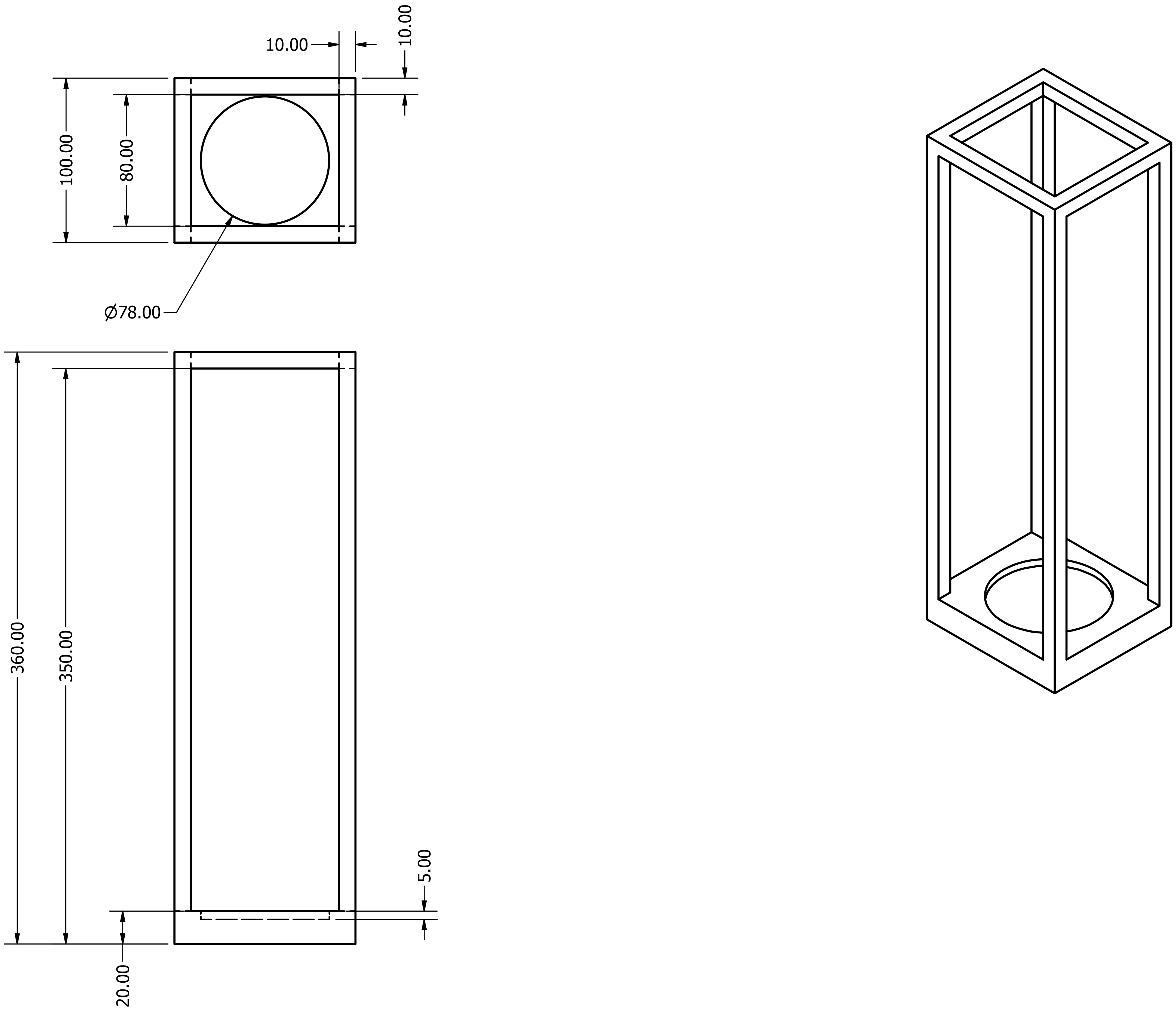
ALL DIMENSIONS IN MILLIMETRES UNLESS STATED | DO NOT SCALE DRAWING | SCALE 1/10 | SHEET 1 OF 1 | REV | A3



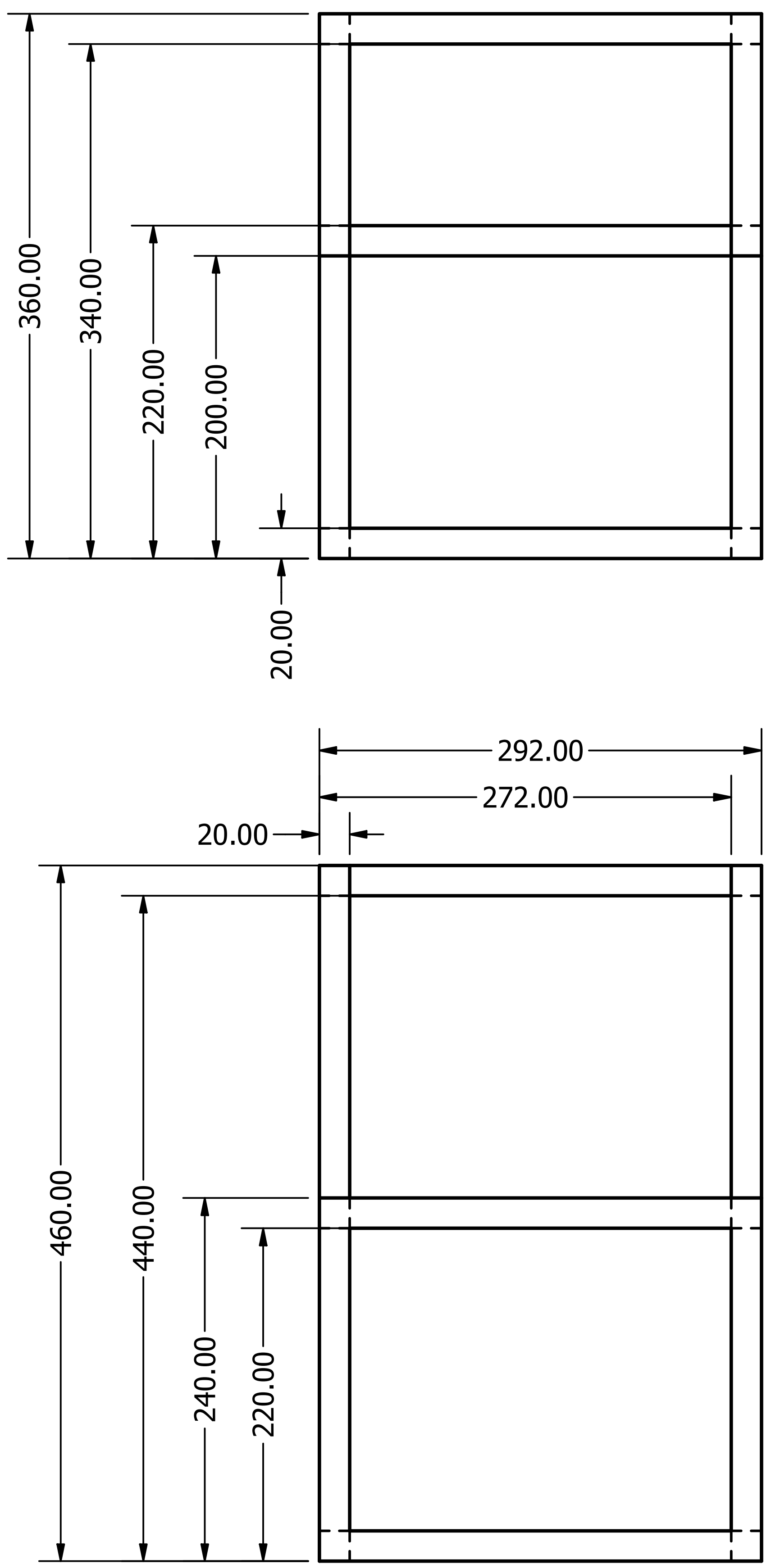
THE UNIVERSITY OF AUCKLAND NEW ZEALAND Te Whare Wānanga o Tāmaki Makaurau	REV A	DESIGNED BY Emma Sio	TOLERANCES .X ±0.5mm .XX ±0.25mm .XXX ±0.10mm ANGLE ±0.5°	MATERIAL Aluminium	PART NAME Rear Chassis Shell
	REV B			PROJECT Urban Transport Design Project	FILE NAME Rear Chassis Shell
	REV C	APPROVED BY Nabeel Azafer			3RD ANGLE PROJECTION
			ALL DIMENSIONS IN MILLIMETRES UNLESS STATED	DO NOT SCALE DRAWING	SCALE 1/10 · SHEET 1 OF 1 · REV · A3

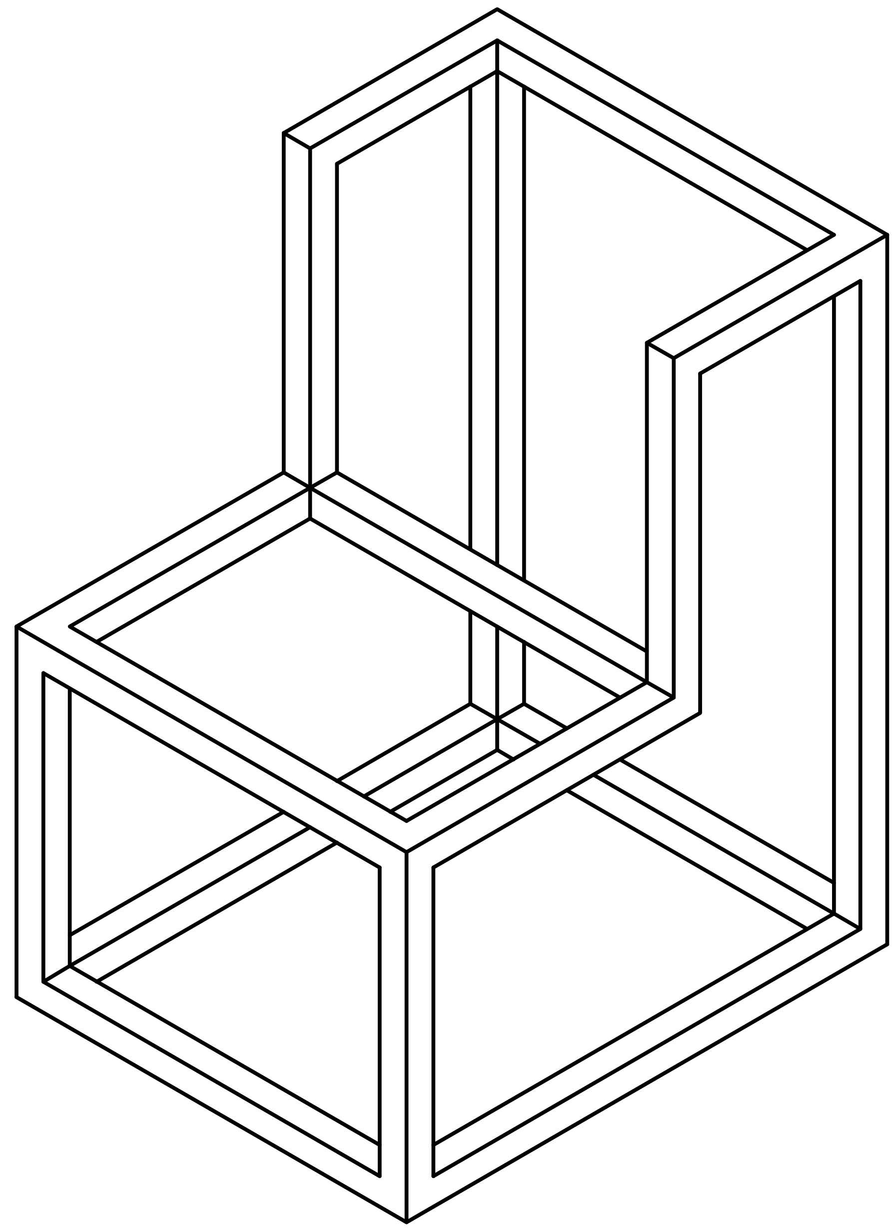


THE UNIVERSITY OF AUCKLAND NEW ZEALAND Te Whare Wānanga o Tāmaki Makaurau	REV A	DESIGNED BY Emma Sio	TOLERANCES .X ±0.5mm .XX ±0.25mm .XXX ±0.10mm ANGLE ±0.5°	MATERIAL Aluminium	PART NAME Removable storage case	3RD ANGLE PROJECTION
	REV B	APPROVED BY Nabeel Azafer		PROJECT Urban Transport Design Project	FILE NAME Removable Storage Case	
	REV C		ALL DIMENSIONS IN MILLIMETRES UNLESS STATED		DO NOT SCALE DRAWING	SCALE 1:5 / SHEET 1 OF 1 / REV / A3



THE UNIVERSITY OF AUCKLAND NEW ZEALAND Te Whare Wānanga o Tāmaki Makaurau	REV A	DESIGNED BY Nabeel Azafer	TOLERANCES .X ±0.5mm .XX ±0.25mm .XXX ±0.10mm ANGLE ±0.5°	MATERIAL Alluminium Alloy	PART NAME Accumalator Housing
	REV B			PROJECT Urban Transport Project	FILE NAME Accumalator Housing
	REV C	APPROVED BY Emma Sio			3RD ANGLE PROJECTION
			ALL DIMENSIONS IN MILLIMETRES UNLESS STATED	DO NOT SCALE DRAWING	SCALE 1/2.5 · SHEET 1 OF 1 · REV · A3

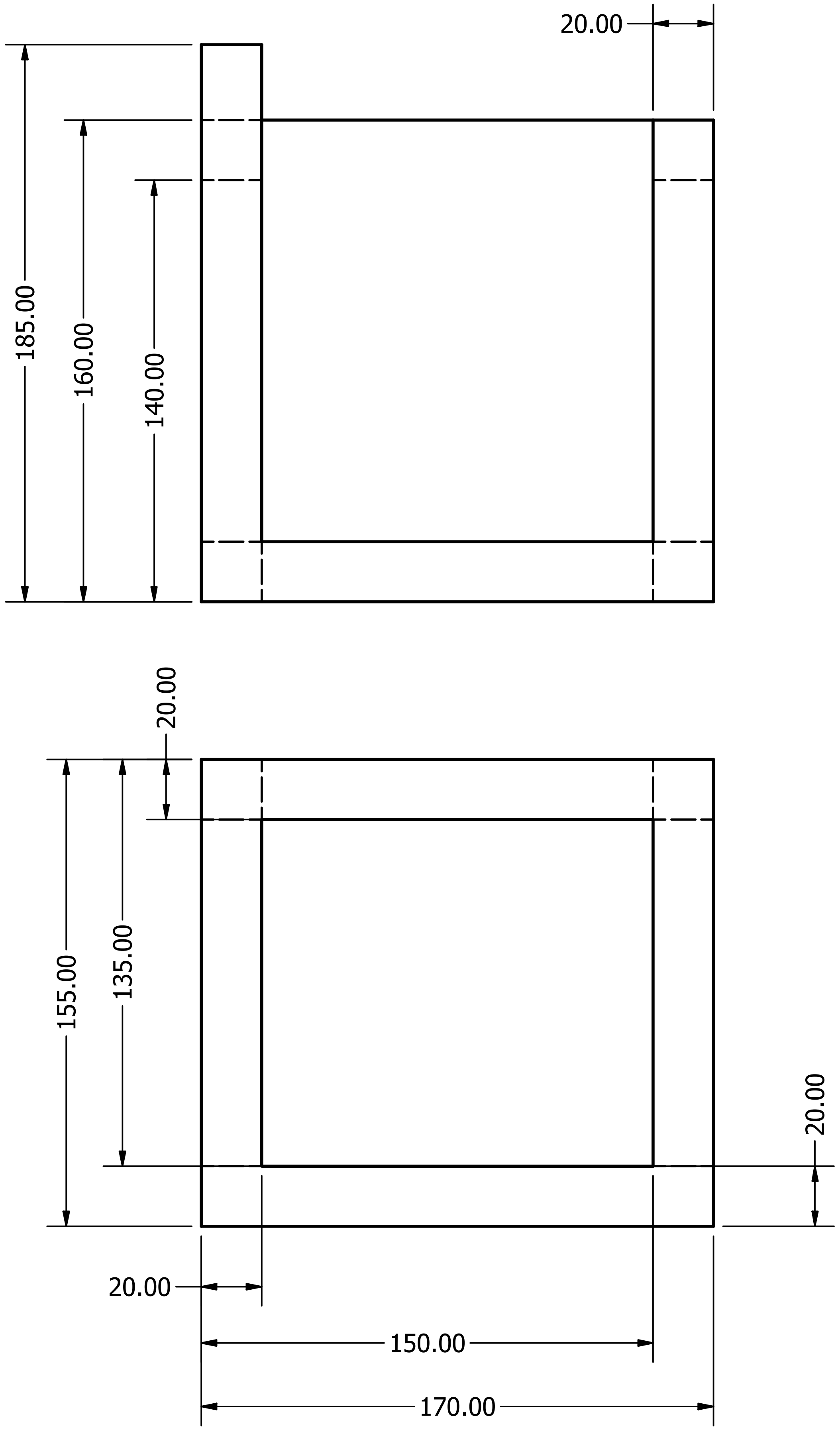


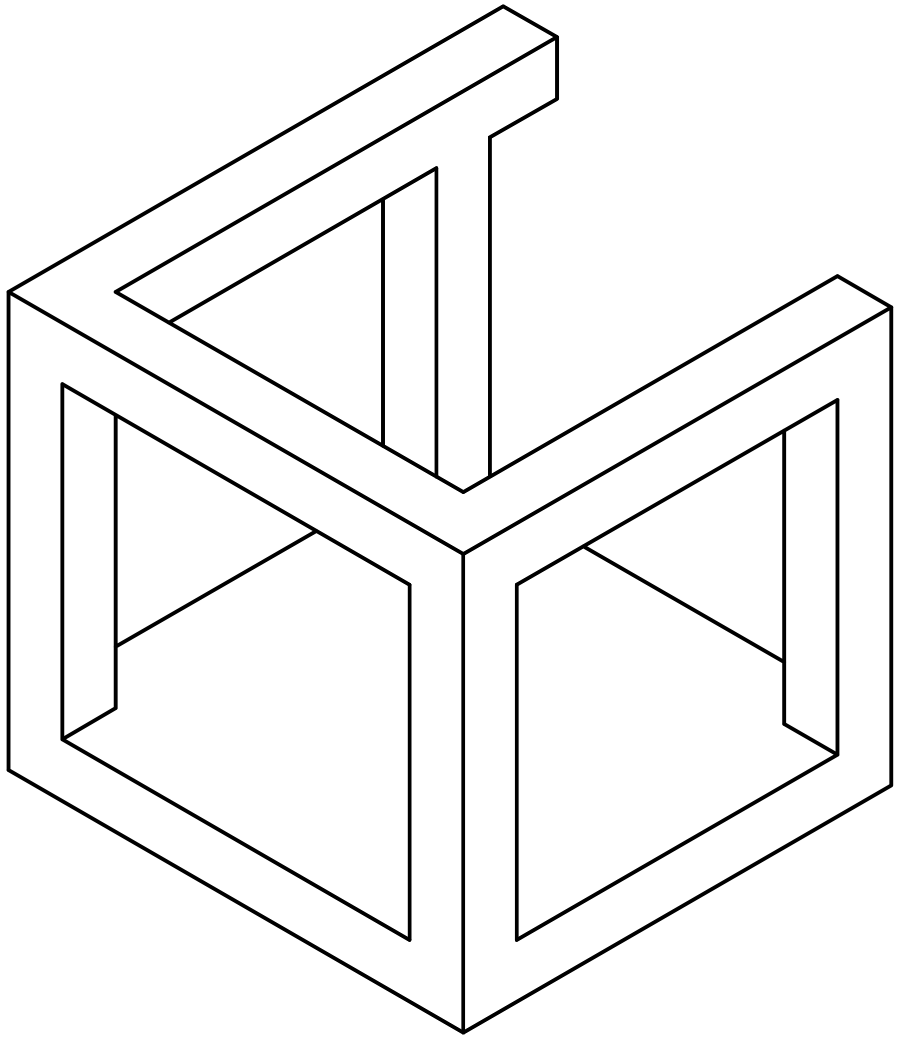


THE UNIVERSITY OF AUCKLAND NEW ZEALAND Te Whare Wānanga o Tāmaki Makaurau	REV A	DESIGNED BY Nabeel Azafer	TOLERANCES .X ±0.5mm .XX ±0.25mm .XXX ±0.10mm ANGLE ±0.5°	MATERIAL Alluminium Alloy	PART NAME Fuel Tank Housing
	REV B	APPROVED BY Emma Sio		PROJECT Urban Design Project	FILE NAME Fuel Tank Housing
	REV C			ALL DIMENSIONS IN MILLIMETRES UNLESS STATED	DO NOT SCALE DRAWING

3RD ANGLE PROJECTION

SCALE 1/4.5 | SHEET 1 OF 1 | REV | A3

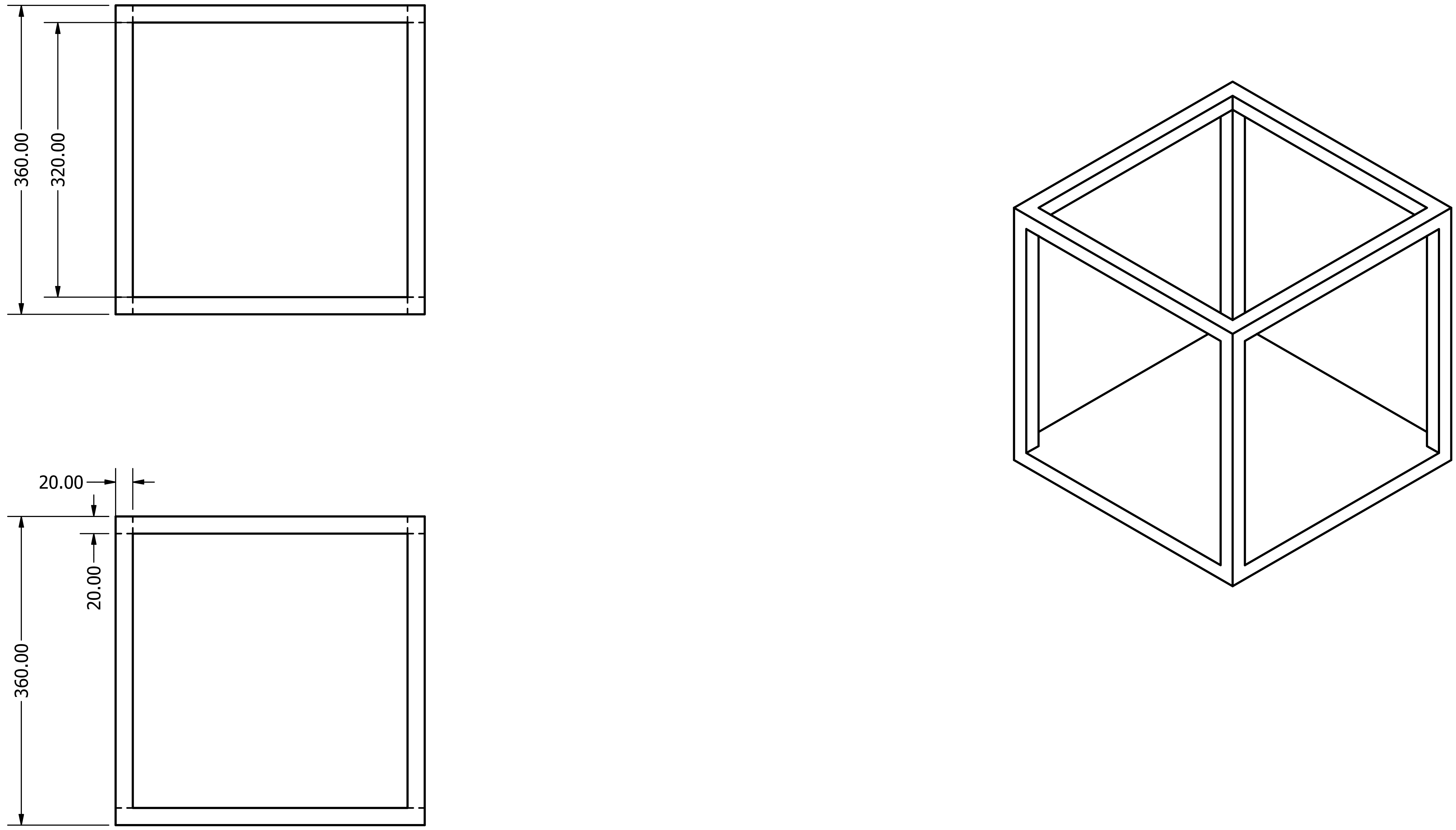




THE UNIVERSITY OF AUCKLAND NEW ZEALAND Te Whare Wānanga o Tāmaki Makaurau	REV A	DESIGNED BY Nabeel Azafer	TOLERANCES .X ±0.5mm .XX ±0.25mm .XXX ±0.10mm ANGLE ±0.5°	MATERIAL Alluminium Alloy	PART NAME Pump and Motor housing
	REV B	APPROVED BY Emma Sio		PROJECT Urban Transport Project	FILE NAME Pump and Motor Housing
	REV C		ALL DIMENSIONS IN MILLIMETRES UNLESS STATED		DO NOT SCALE DRAWING

SCALE 1 / 2 | SHEET 1 OF 1 | REV | A3

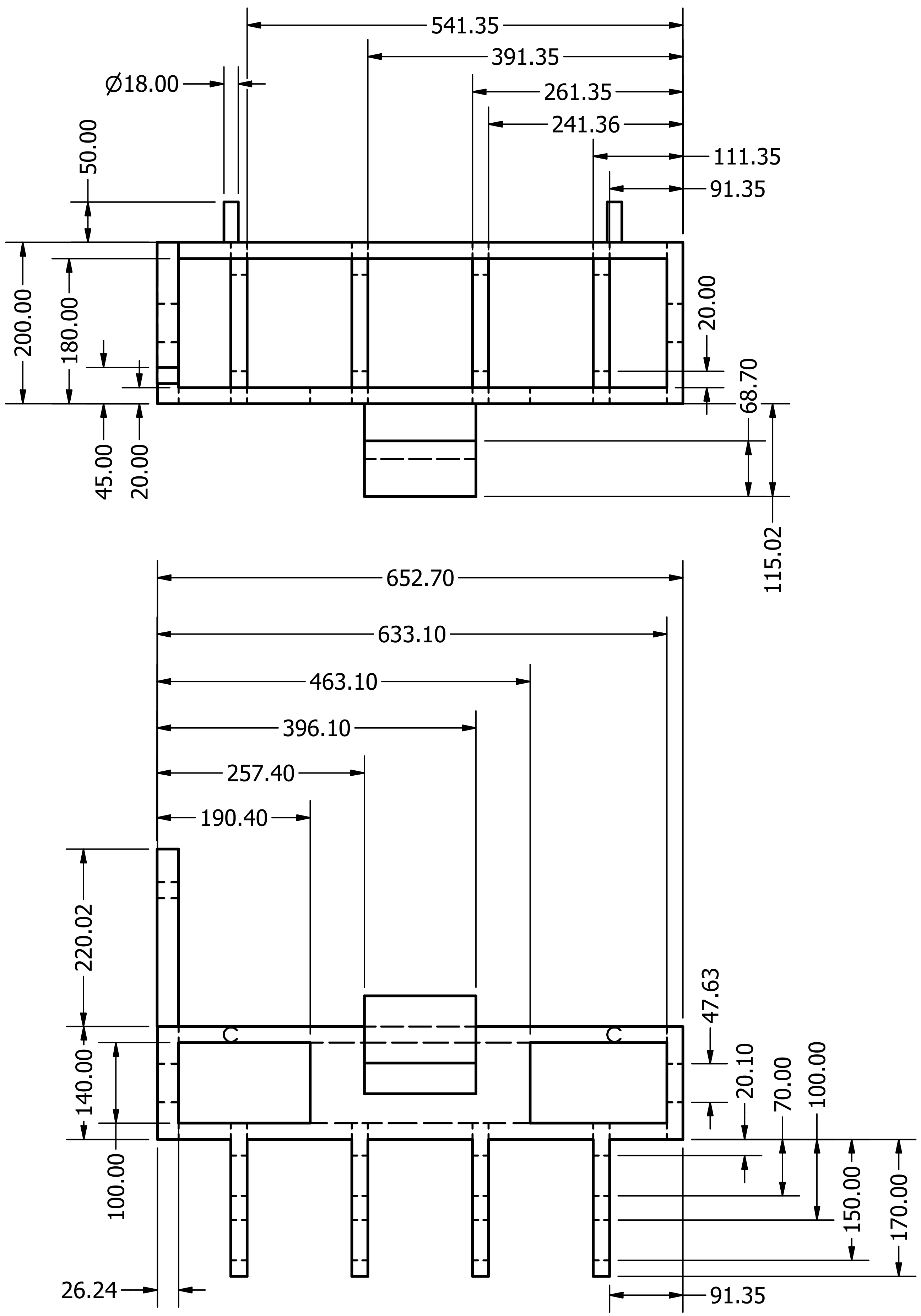
3RD ANGLE PROJECTION

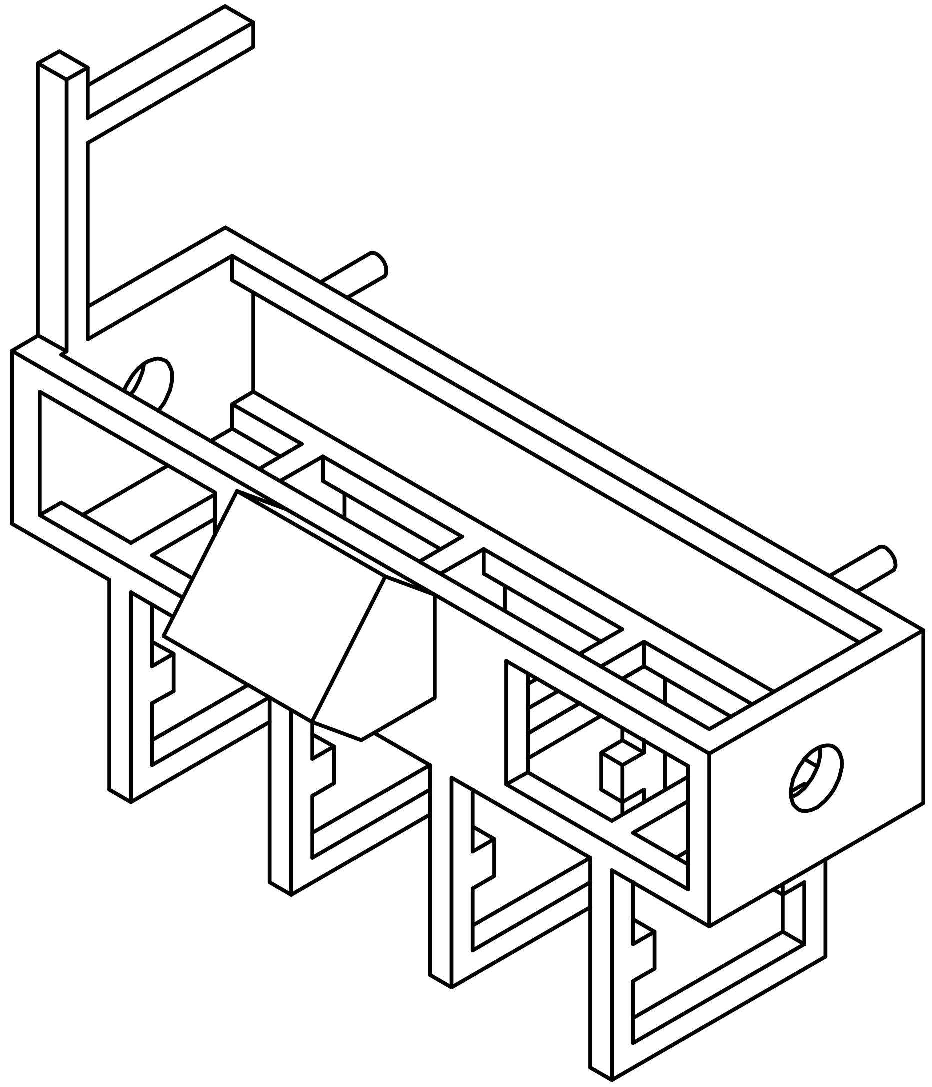


THE UNIVERSITY OF AUCKLAND NEW ZEALAND Te Whare Wānanga o Tāmaki Makaurau	REV A	DESIGNED BY Nabeel Azafer	TOLERANCES .X ±0.5mm .XX ±0.25mm .XXX ±0.10mm ANGLE ±0.5°	MATERIAL Alluminium Alloy
	REV B	APPROVED BY Emma Sio		PROJECT Urban Design Project
	REV C			PART NAME Motor Housing
				FILE NAME Motor Housing

3RD ANGLE PROJECTION

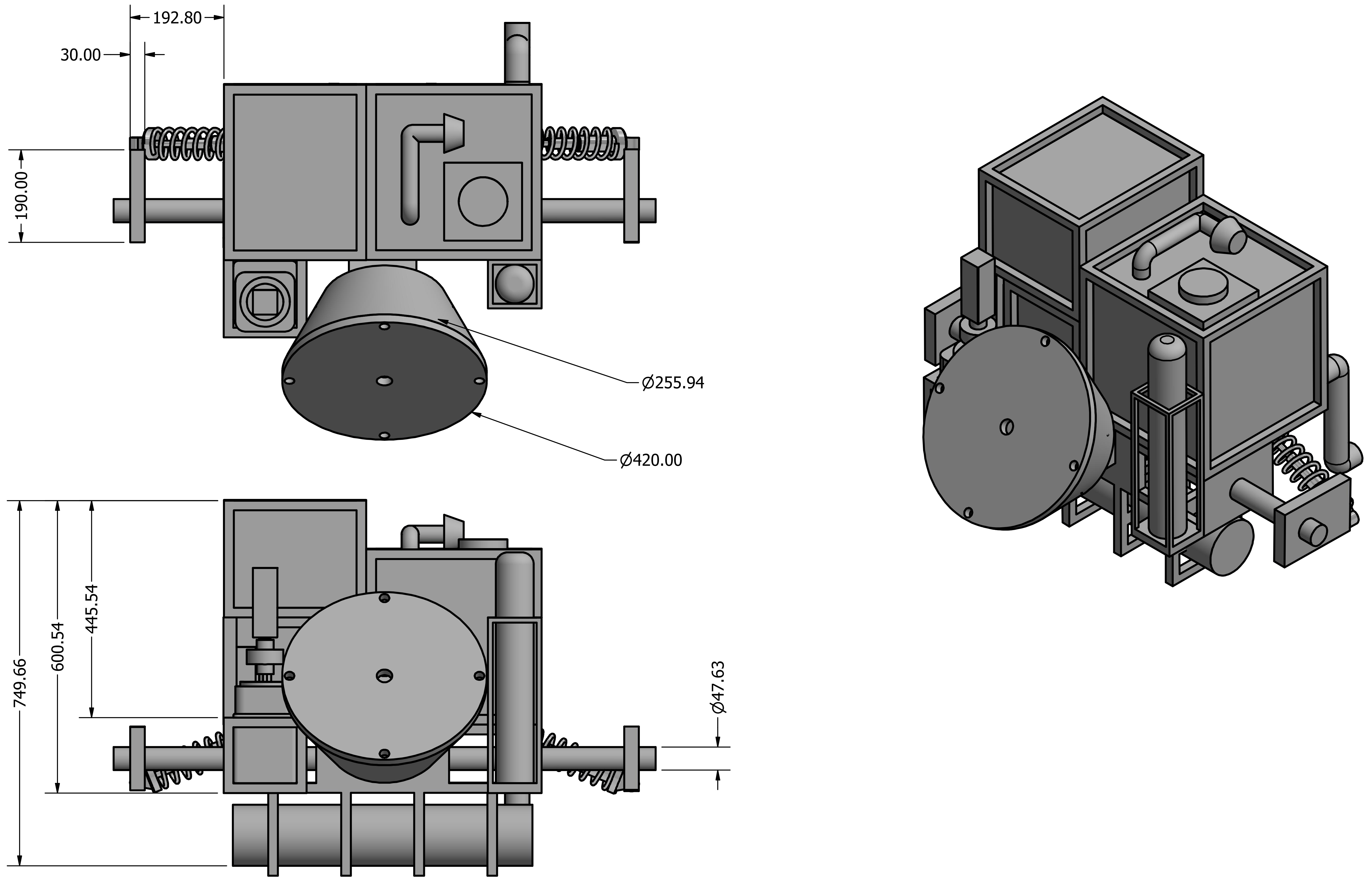
ALL DIMENSIONS IN MILLIMETRES UNLESS STATED | DO NOT SCALE DRAWING | SCALE 1/5 | SHEET 1 OF 1 | REV | A3





THE UNIVERSITY OF AUCKLAND NEW ZEALAND Te Whare Wānanga o Tāmaki Makaurau	REV A	DESIGNED BY Nabeel Azafer	TOLERANCES .X ±0.5mm .XX ±0.25mm .XXX ±0.10mm ANGLE ±0.5°	MATERIAL Aluminum Alloy	PART NAME Main Frame
	REV B	APPROVED BY Emma Sio		PROJECT Urban Transport Vehicle	FILE NAME Main Housing
	REV C				3RD ANGLE PROJECTION

ALL DIMENSIONS IN MILLIMETRES UNLESS STATED | DO NOT SCALE DRAWING | SCALE 0.15:1 | SHEET 1 OF 1 | REV | A3

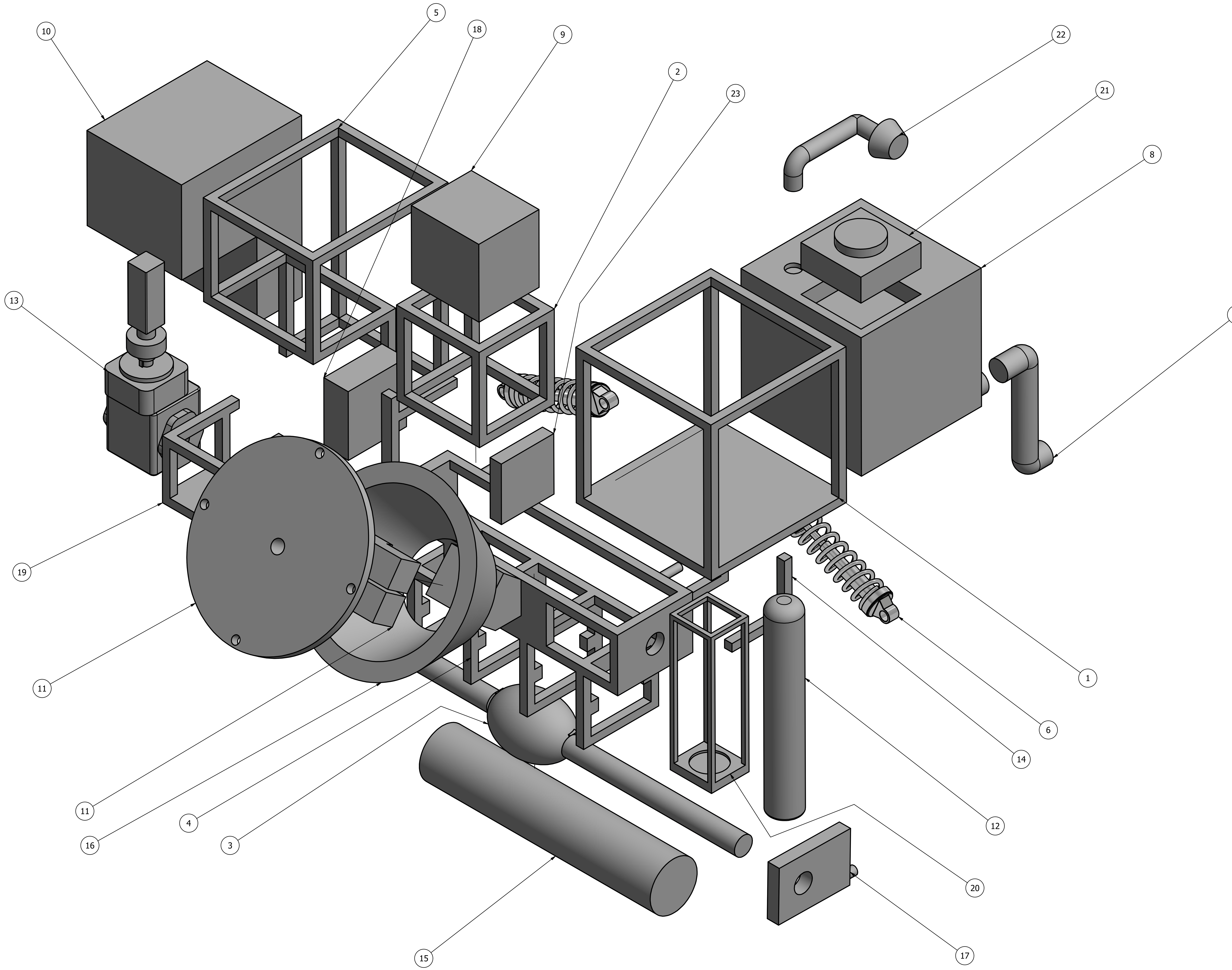


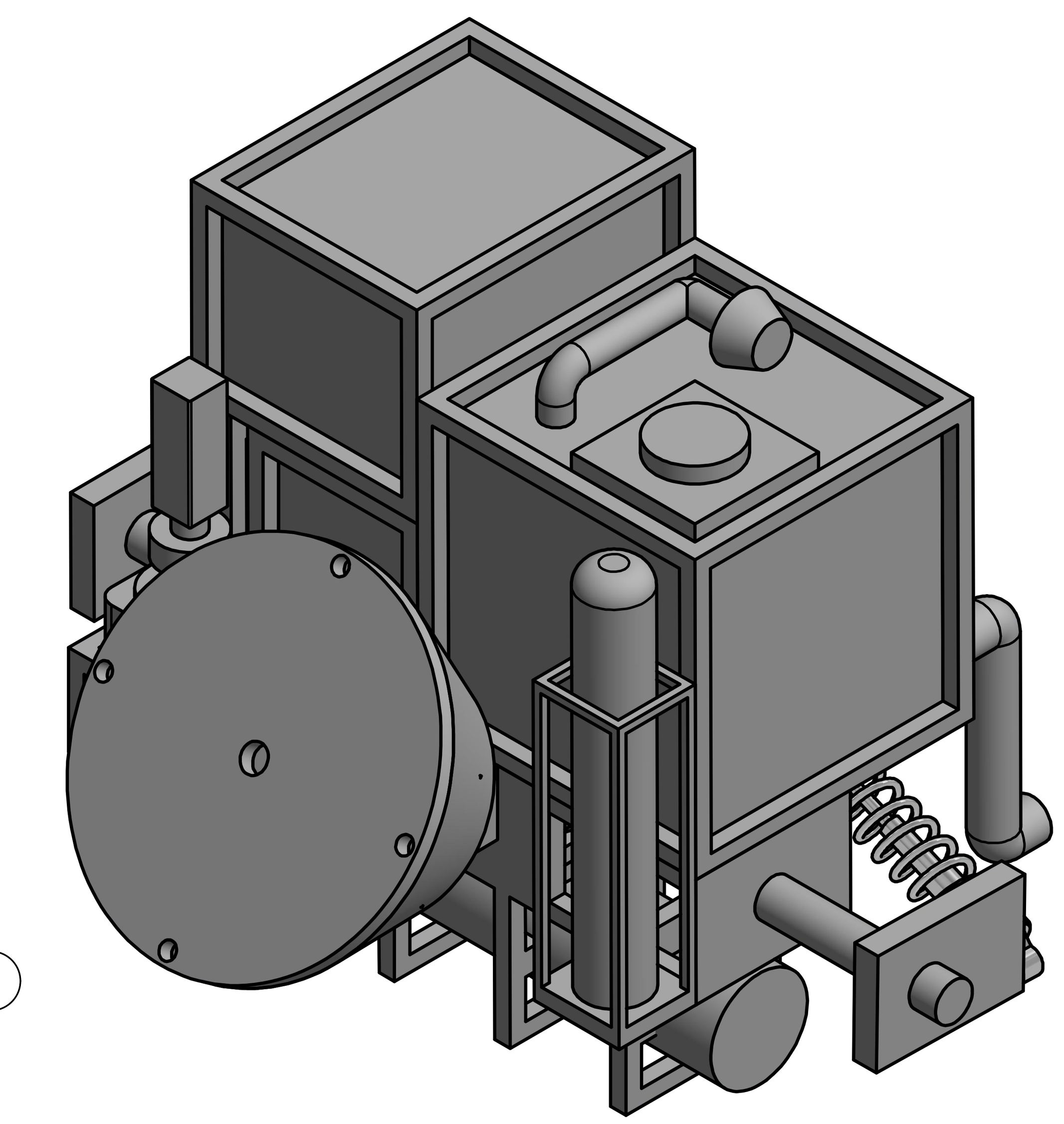
THE UNIVERSITY OF AUCKLAND NEW ZEALAND Te Whare Wānanga o Tāmaki Makaurau	REV A	DESIGNED BY Nabeel Azafer	
	REV B	APPROVED BY Emma Sio	
	REV C		

TOLERANCES		MATERIAL Alluminium Alloy	PART NAME Rear Chassis
.X	±0.5mm	PROJECT Urban Design Project	FILE NAME Rear Chassis
.XX	±0.25mm		
.XXX	±0.10mm		
ANGLE	±0.5°		

ALL DIMENSIONS IN MILLIMETRES UNLESS STATED | DO NOT SCALE DRAWING | SCALE 1/8 | SHEET 1 OF 1 | REV | A3

3RD ANGLE PROJECTION

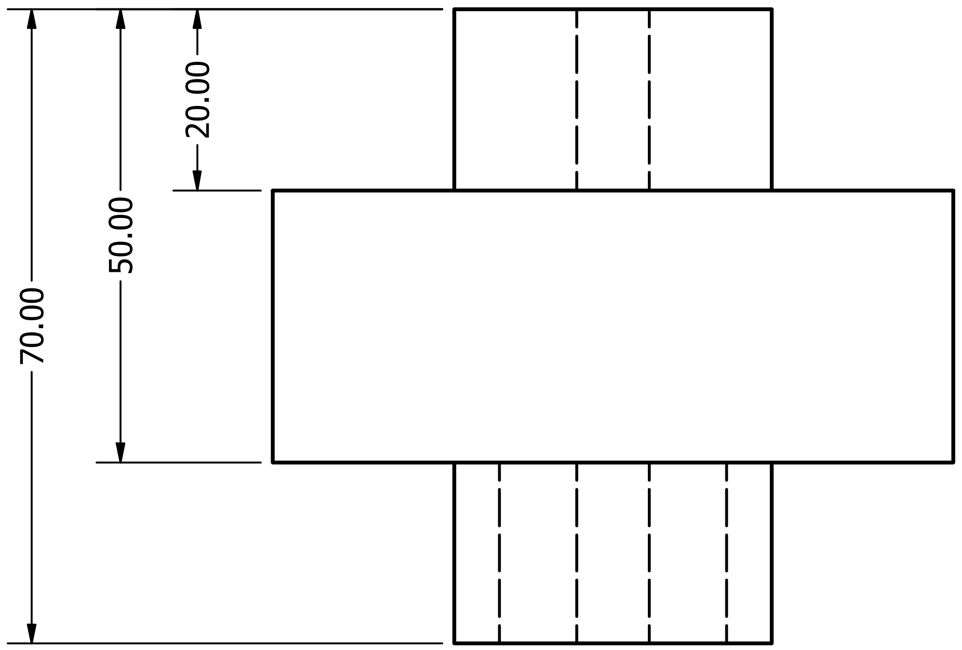


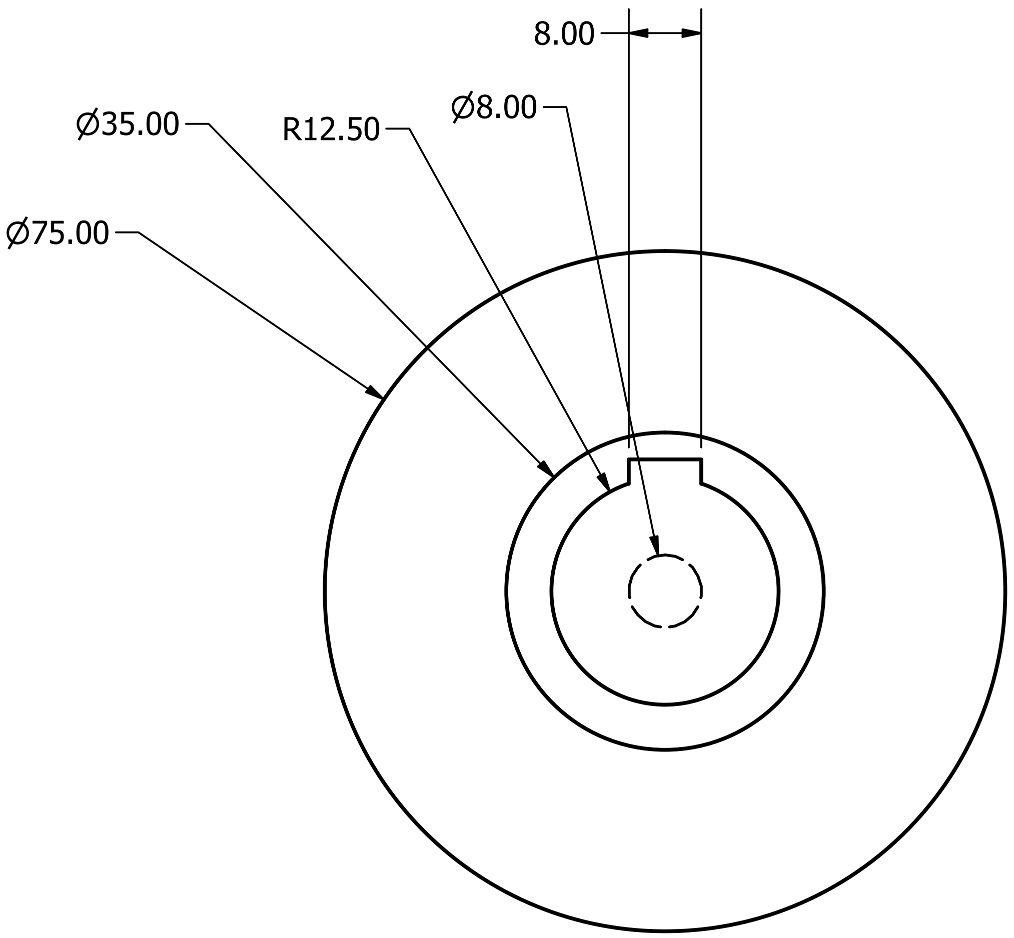


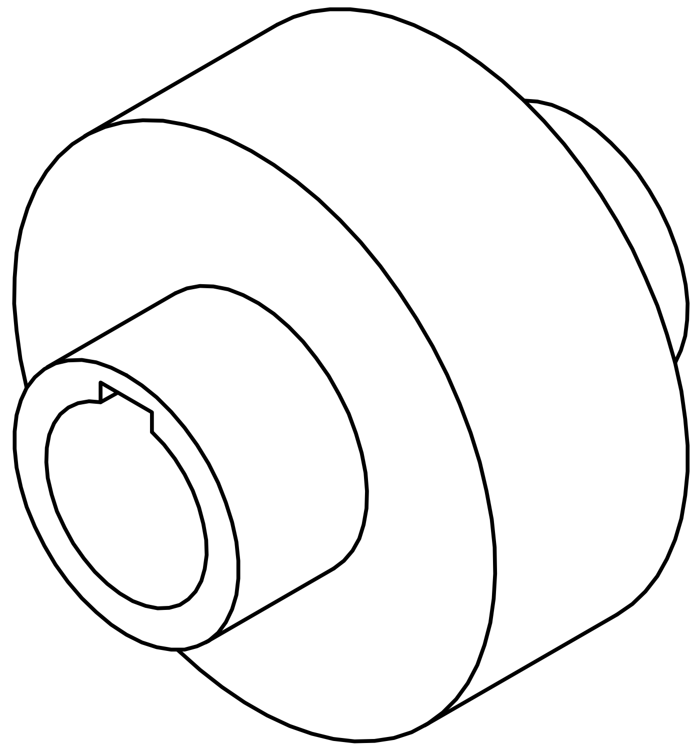
PARTS LIST

ITEM	QTY	Part Name
1	1	Motor Housing
2	1	Gearbox Housing
3	1	Differential
4	1	Diff housing
5	1	fuel tank housing
6	2	Shock Absorber
7	1	Exhaust and Silencer
8	1	Engine ktm 690
9	1	Gearbox
10	1	Fuel Tank
11	1	Hydraulic Assembly
12	1	Accumulator
13	1	Pump and motor Coupled
14	2	Main Housing Support
15	1	Hydraulic tank
16	1	Main Chassis Connection
17	2	Spring Connection
18	1	Battery
19	1	Pump and Motor Housing
20	1	Accumalator Housing
21	1	Carburetor
22	1	Air Intake
23	1	Clutch Casing

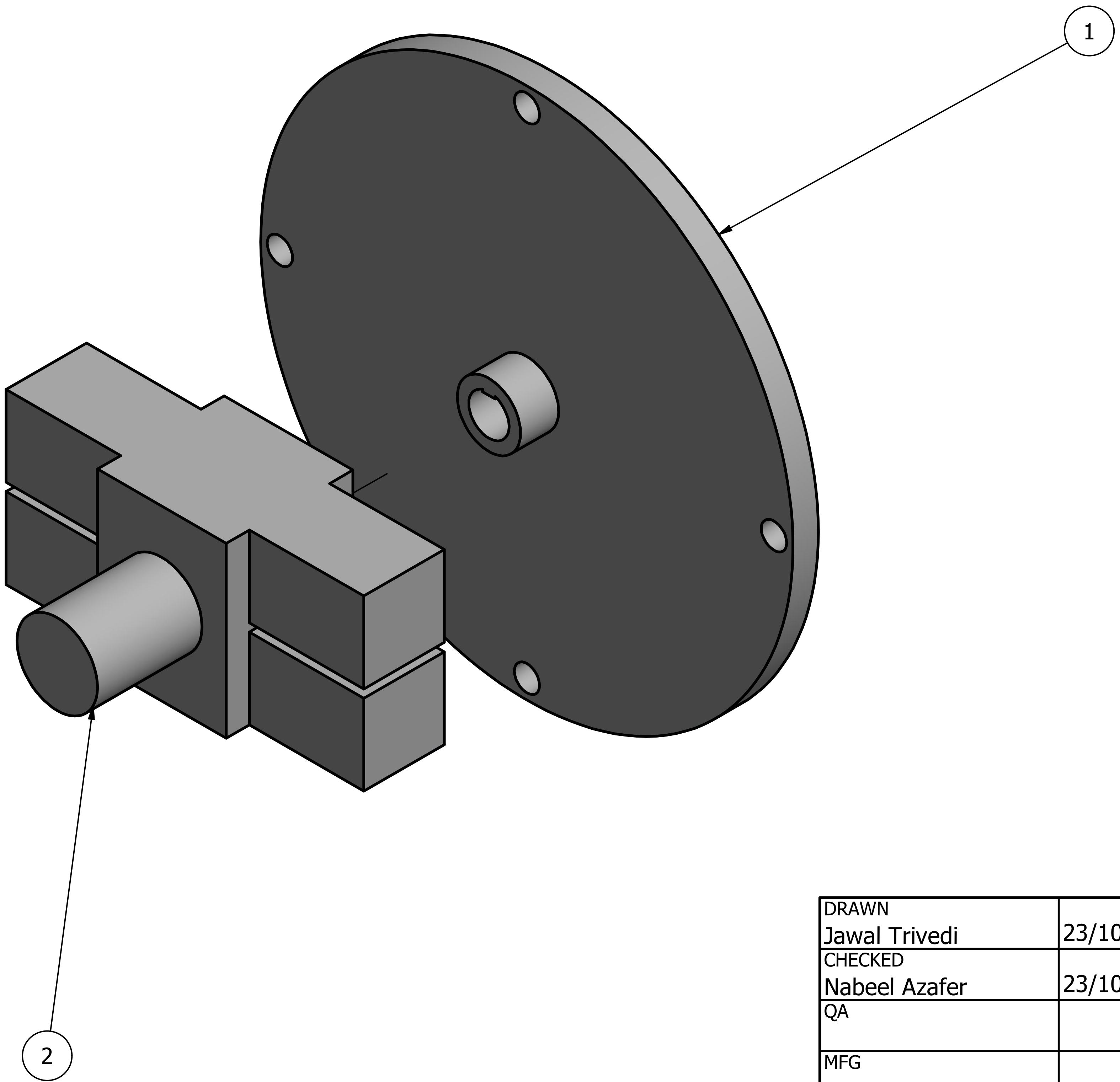
DRAWN Nabeel Azafer	22/10/2020	Urban Design Project
CHECKED Emma Sio	22/10/2020	TITLE
University of Auckland		Rear Chassis Exploded View
		SIZE A3 · Third Angle Projection · DWG NO 6 · REV
All Dimensions in mm		SCALE 1/3.75 · SHEET 1 OF 1





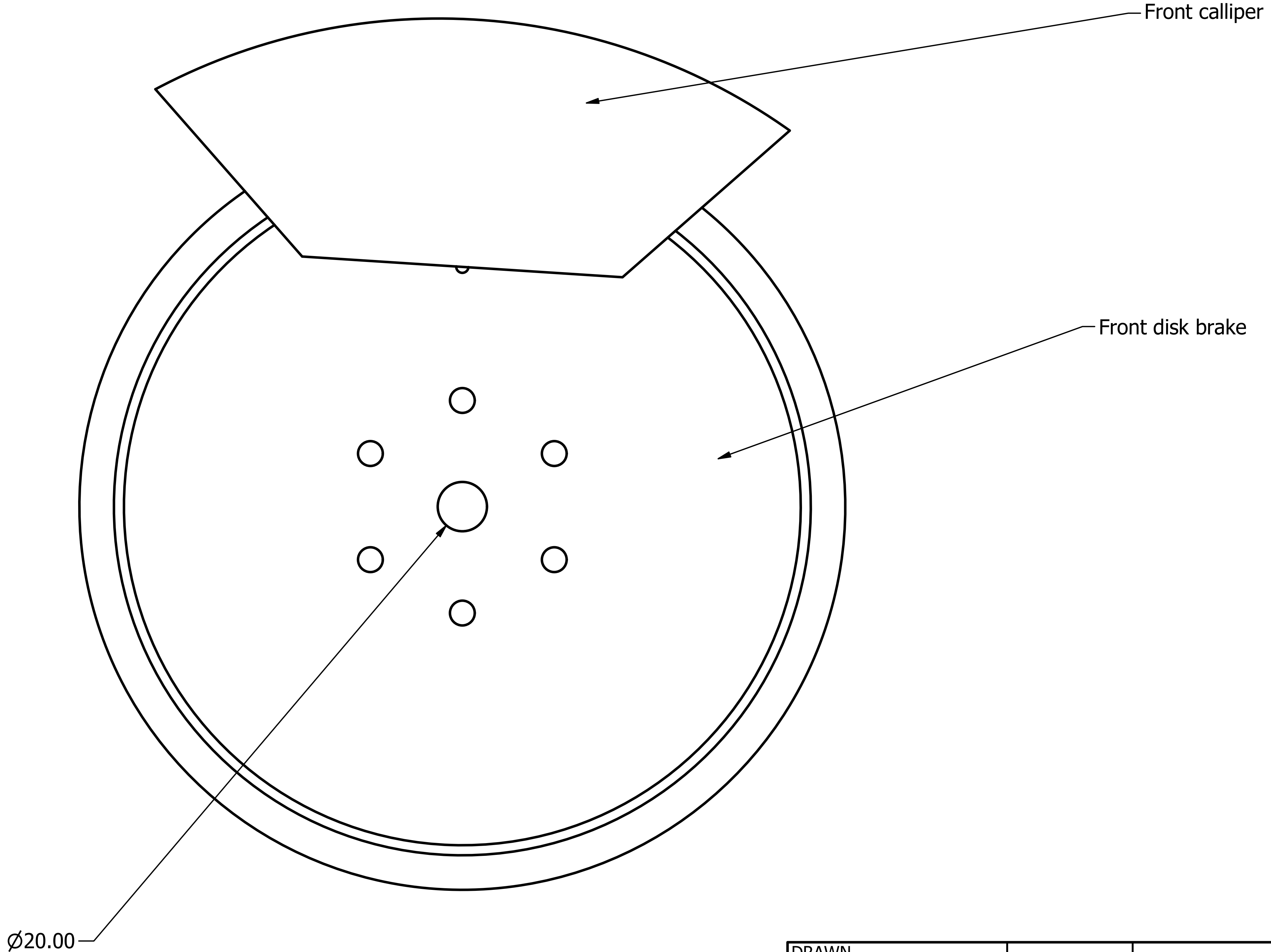


DRAWN Jawal Trivedi	23/10/2020	Urban design Projection		
CHECKED Nabeel Azafer				
QA		TITLE Pump and E-Motor Coupling		
MFG				
APPROVED				
		SIZE A3	Third Angle Projection	DWG NO Pump and e-motor coupling — REV
All Dimensions in mm		SCALE 2.2 : 1		SHEET 1 OF 1

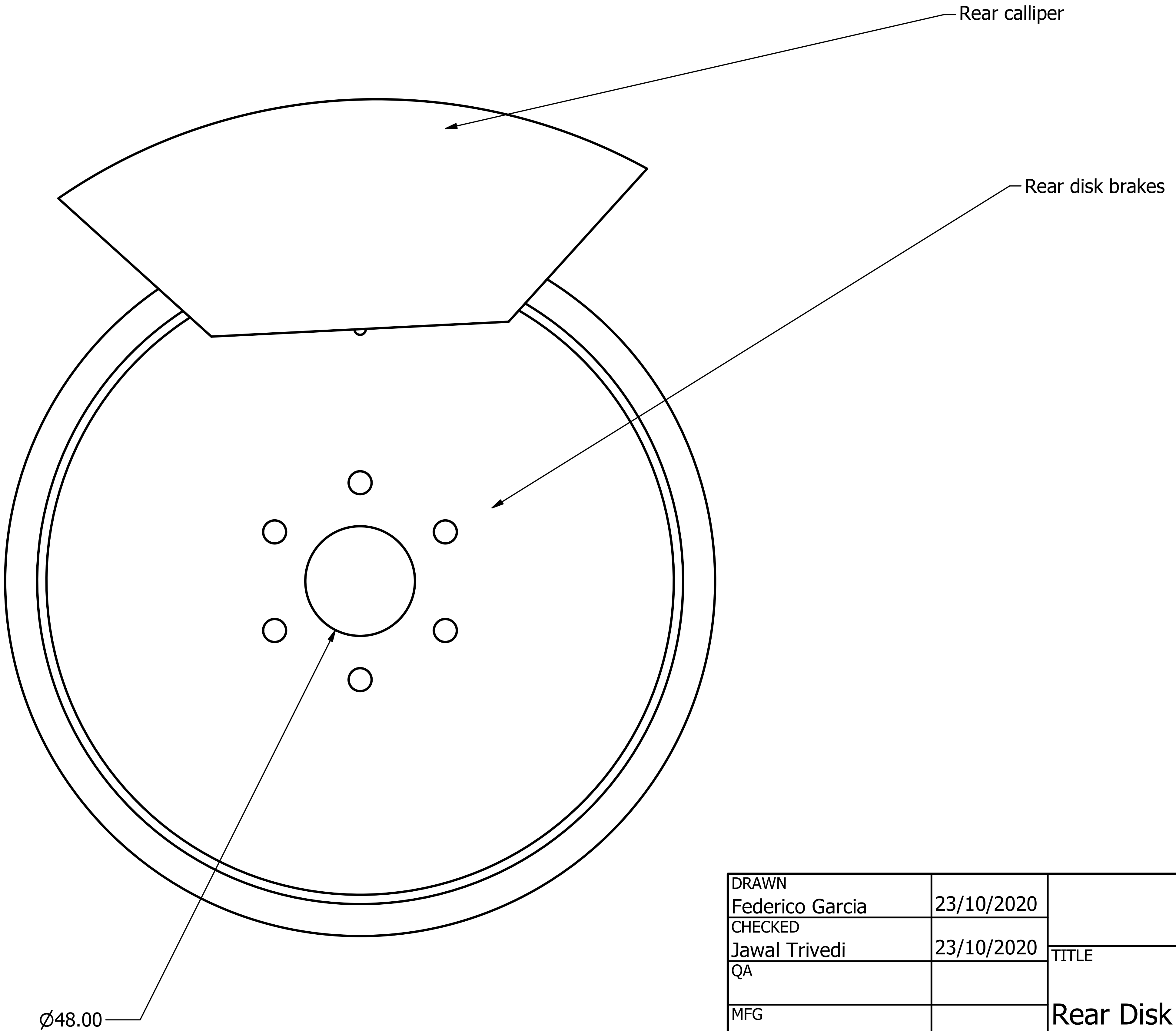


PARTS LIST		
ITEM	QTY	PART NUMBER
1	1	Rotary Motor Coupling
2	1	Semi Rotary Hydraulic Motor

DRAWN Jawal Trivedi	23/10/2020	
CHECKED Nabeel Azafer	23/10/2020	TITLE
QA		Hydraulic Actuator Connection
MFG		
APPROVED		
		SIZE A3 · Third Angle Projection · DWG NO Hydraulic Assembly · REV
All Dimensions in mm		SCALE 1/3 · SHEET 1 OF 1



DRAWN Federico Garcia	23/10/2020	University of Auckland		
CHECKED Jawal Trivedi	23/10/2020			
QA		TITLE		
MFG		Front Brakes Drawing		
APPROVED				
		SIZE A3	DWG NO FrontBrakeAssembly	REV
		SCALE 1/2		SHEET 1 OF 1



DRAWN Federico Garcia	23/10/2020	University Of Auckland
CHECKED Jawal Trivedi	23/10/2020	TITLE Rear Disk Brake Drawing
QA		
MFG		
APPROVED		SIZE A3 / DWG NO RearBrakesAssembly / REV
		SCALE 1 / 2 / SHEET 1 OF 1