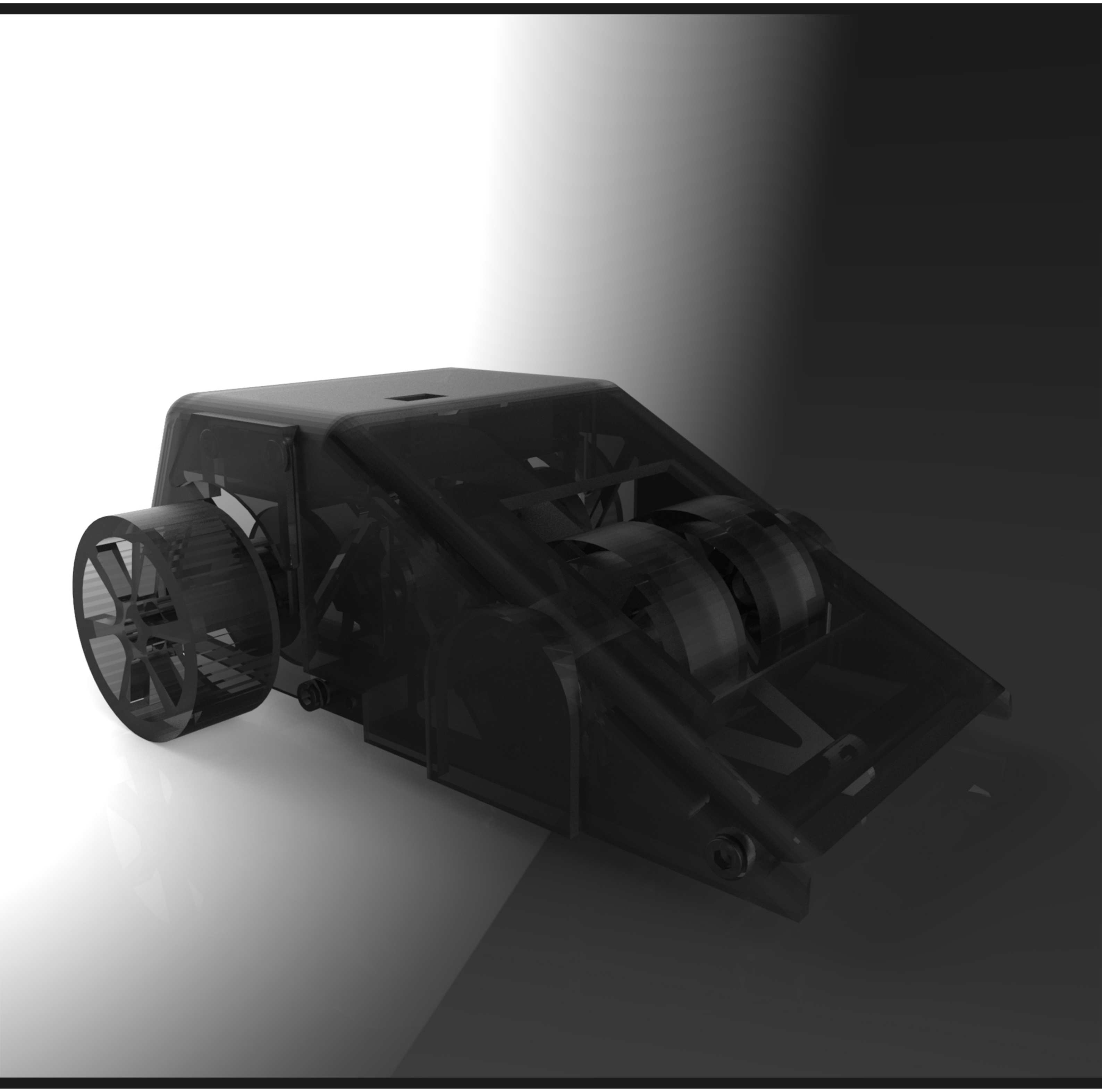
Imperial College London

DEPARTMENT OF MECHANICAL ENGINEERING



GROUP P6

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Executive Summary

Miniature vehicles are perhaps one of the most popular forms of toy and the production of them is a well-trodden path. The aim of the project was to produce a toy vehicle, from idea to tangible product. The focus was on a simple and effective design that would overcome three challenges: win races of speed, a tug of war challenge, and to endure being accidentally stepped on. Compared to its competitors, the vehicle is very lightweight and aerodynamic.

The most labour intensive tasks, such as manufacture and CAD design of parts, were split evenly between the group members. Smaller tasks were allocated to each according to ability and area of interest. These included stress analysis calculations on the chassis and axles and modelling of vehicle dynamics.

The first stage of production included concept sketching and brainstorming to create a fully defined model on paper for each of the necessary parts. This was followed by virtualising each component in *Solidworks* to produce a full set of engineering drawings.

The second stage involved the manufacture and assembly of all parts. This was time consuming and many setbacks were encountered along the way, such as cancelled STW sessions and the need to manufacture some parts twice (due to poor quality of initial production).

This was enriching as an exercise in planning and realising a product in a team and offered insight into the methodology of producing a group project, as well as the difficulties one is likely to encounter doing so.

The car accelerates to roughly 4.75m/s under no-load and can pull a load of 10.5N. It is powered by an electric motor, which is powered by batteries. The switch is accessible through the hole on the top of the vehicle. All critical parts are cheap and easy to manufacture. Since it is light-weight (1.8kg) it is unlikely to damage something or get damaged in case of a collision.

Contents

1. Introduction

The task assigned by the client was to design, build and test a miniature motorised car to perform a speed-race as well as a torque challenge on a track of length 30m. The car is to be used multiple times and have a service life of minimum 100 hours. It must also withstand a force of up to 80kg applied on the top, simulating being accidentally stepped on. The maximum width restriction for the car was 200mm.

Standard components were provided: a set of 4 wheels, an electric motor, a battery holder and batteries. Additional constraints included use of stock materials available in the STW workshop, and purchase of components from the RS online catalogue.

The key design parameters and targets taken from the project task description are laid out in the Design Specification. Based upon this, initial design decision choices were taken and summarised in the Morphological Chart. Concepts were developed and an overall design was decided, and the parts split up to be designed individually. Regular meetings ensured parts worked together to create the most effective design. Stress calculations on the shafts and key components ensure PDS requirements are met and are summarised in this brief. Part progression and the final design can be found in Concept Development and Final Design Breakdown. The calculated theoretical performance characteristics are shown, followed by a conclusion of the project. Drawings for the assembly, bill of materials and key parts can be found in the Technical Realisation Document along with a measurement analysis document of manufactured parts.

2. Project Product Design Specification (PDS)

Element	***Statement or Criteria***	***Verification By***
Customer		
Needs	Win race on 75% of concourse (30m) Win tug of war race Car with a service life in excess of 100 hours.	Product Test Results
Competition	Multiple other group designs.	
Aesthetics	No sharp edges, with a 3-D printed double curved cover. Streamlined	Design Review
Operation		
Performance	Speed of 4m/s based on market research Complete course in under 10 seconds Leads to a gear ratio of 1:2 Reasonable torque Maintain speed for distance 75% length of concourse.	Testing & Calculation
Environment	Indoors. Concourse floor. Flat.	Testing
Size	Width less than 200mm, light and to reduce bending moment Max length 300mm to reduce weight and bending moment. Design optimises COM to be in front of back shaft 4 wheels to provide stability and an accurate direction	Calculation & Measurement
Weight	Maximum weight 1.5kg- lower weight, higher acceleration. Not too light as this will reduce traction.	Testing
Ergonomics	On/Off switch must be located on the outside of the vehicle. A M6 eyebolt should be attached to back. Car must withstand 80kg force from top. No exposed moving transmission parts.	Design Review
Service Life	Life in excess of 100 hours.	Testing
Maintenance	Easily change battery, remove cover.	Design Review

3. Morphological Analysis

Features	Option 1	Option 2	Option 3	Option 4	Option 5
Wheel arrangement	3 wheels	4 wheels	Caterpillar tracks	Wheels in-line with chassis	
Front or back drive	Front	Back	Both		
How to resist bending moment/ weight	Thicker, Stronger Axle	Suspension/ Springs	A retractable stand		
Transmission	Gears - 2 Stage	Chain + Gear	Belt + Gear	Gear- single (if possible)	
Chassis Design	No Internal Side Plates, parts attached to Cover.	Thick or thin plates	Box shaped design		
Motor Mountings	Foot mounted	Face Mounted			
Bearing Constraints	Circlips	Grub Screws	Taper Locks	Shoulders/ Housings	Threaded washer
Bearings	Rolling Bearings	Sliding Bearings	Bush Bearings		

4. Concept development

Chassis-

From figure 1, it is obvious that the similarities between the concepts conceived lies in the trapezoidal shape. The leading edge was angled to improve the aerodynamics of the car while the flat top shape would distribute the pressure of a person's foot on the car.

Using SolidWorks to develop the model, a few adjustments were made to include the eyebolt mount, battery plate and motor mount. The first version of the chassis assembly, in figure 2, included two connecting rods with the battery plate bolted on the angled edge. However, to utilise space efficiently changes were made to the final version of the chassis assembly. One of the connecting rods was removed and the battery plate was placed on the flat edge. This reduced the manufacture workload and made it easier to assemble.

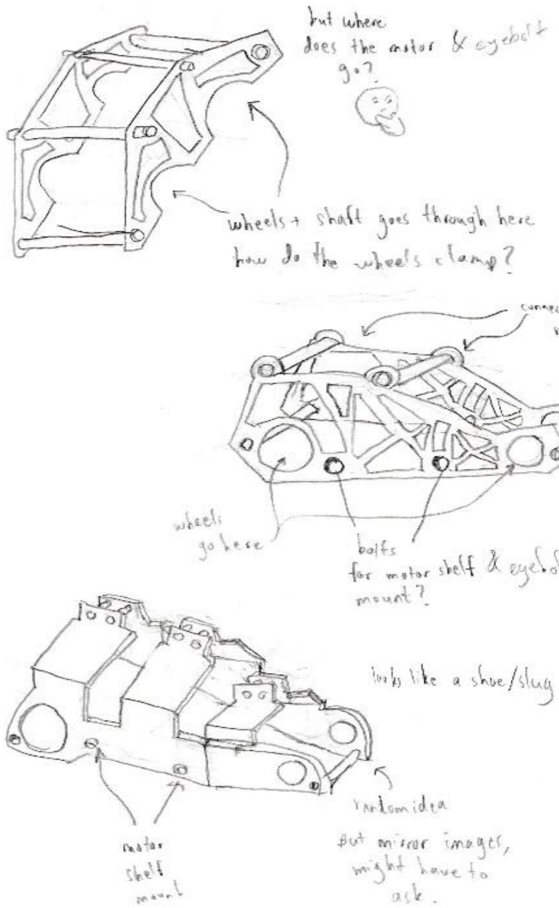

Figure 1: Sketchbook concepts of chassis assembly

Wheel Arrangement-

There were several concerns regarding the wheel arrangement designs. It requires effective use of the space within the dimensional restrictions, as well as maintain stability and balance of the car.

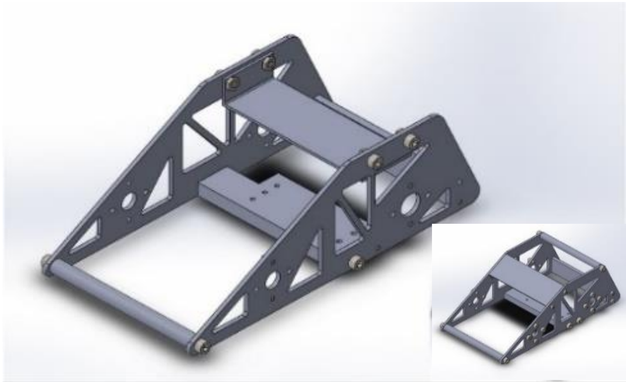

Figure 2: First version of CAD chassis assembly (inset) and the final version of the CAD chassis assembly

The first option considered was a three-wheel vehicle. The concept was having two evenly spaced wheels on the rear shaft outside the car and a single wheel at the centre of the front shaft. This reduces the weight and minimises the size of the car. However, a single wheel at the front of the car might reduce balance and cause the car to not drive straight. In addition to this, stress would concentrate on the single wheel increasing the chance of fracture.

A 4-wheel arrangement was deemed the most appropriate as it provided greatest stability. As the front of the vehicle has plenty of empty space, the front wheels were designed within the vehicle chassis to slightly improve aerodynamics and impart an interesting aesthetic.

3D Cover-

The cover was progressively edited to comply with alterations made to the side plates and battery box shelf. The battery box shelf was initially angled but was made parallel to the ground to make more efficient use of internal space. A hole was cut to allow access to the switch. The cover is located by two screws on each side that also function as constraints for the front most connecting rod and the eyebolt mounting plate. A much more aesthetically pleasing form was considered (supercar or 'bat mobile') but due to time constraints the simplest functional form was chosen.

Front Shaft-

This project was not limited by the need to steer. Therefore, other design options were considered. From research into the record holder of the Pinewood Derby Car Challenge, the idea was put forward to tilt the axis of the shaft to 2 degrees, tilting the wheels reducing friction through minimising contact area with the floor. This concept would be manufactured using 2 long bolts, creating 2 cantilevered front shafts. This would reduce weight, thus helping to achieve a faster top speed. The idea was discounted due to worry of the car deviating off course. Although an interesting concept, it was deemed too risky.

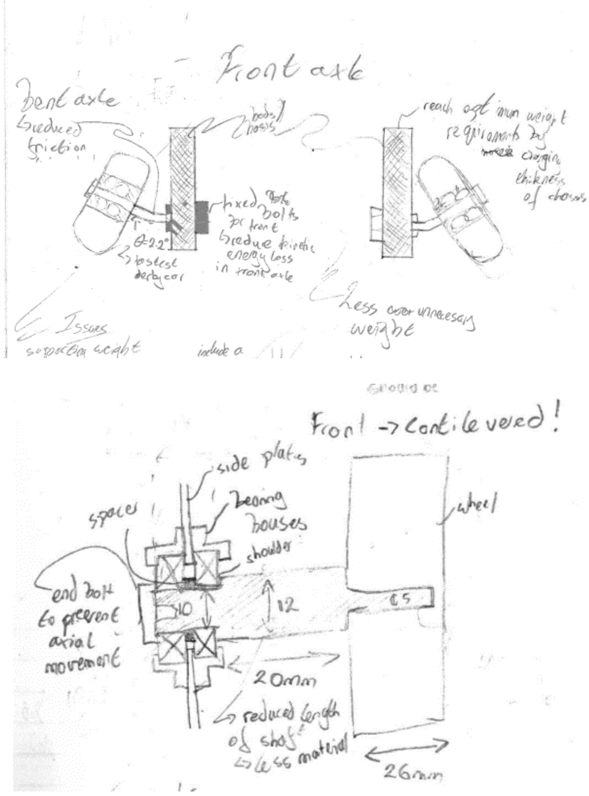

Figure 1: Concept sketch of dual cantilever front axles

Rear Shaft-

It was initially decided to use rear wheel drive. Initially a general drawing without dimensions was sketched. There were several iterations, each with added dimensions and improved placements of features, but there weren't any significant changes, and the final version was not dissimilar from the rough sketch as it was always a goal to keep the design simple and effective to aid manufacture and assembly.

Bearing Mounts-

Custom bearing mounts were made to constrain the bearings. The bearing mounts were a compromise of a need to be lightweight while also being durable, and simple to manufacture. Initially the four holes were only M3 clearance holes, but it was decided that the location could not be accurate enough to achieve concentricity between the pairs of bearings to the tolerance specified in the general assembly document. Two opposite M3 clearance holes on each mount were therefore changed to 3mm dowel holes.

3D Cover-

The cover was progressively edited to comply with alterations made to the side plates and battery box shelf. The battery box shelf was initially angled but was made parallel to the ground to make more efficient use of internal space. A hole was cut to allow access to the switch. The cover is located by two screws on each side that also function as constraints for the front most connecting rod and the eyebolt mounting plate. A much more aesthetically pleasing form was considered (supercar or 'batmobile') but due to time constraints the simplest functional form was chosen.

Transmission-

Deciding on the transmission ratio was one of the first challenges that was met. Originally a step-down ratio of 1:4 was decided upon by researching the speeds that vehicles with similar power sources (4XAA batteries) could reach. However, the ratio was halved when it was realised that the vehicles would likely accelerate to full speed in a relatively short amount of time compared to the total duration of the race. This reduced the advantage of high torque but would result in a much greater speed. It was also desirable for the vehicle to reach the point of maximum power earlier on in the race rather than later, as the latter scenario would mean that very little of the course would be run at the vehicles highest speed.

Gears-

The gear selection process was built around two things. Firstly, the RS part catalogue as they were the only ones available and in-stock. Secondly, the decided transmission ratio. From there the minimum face width, W_f, was determined:

$$W_f = \frac{W_t}{K_v m \sigma_p Y} \qquad (Equation\ 1)$$

Where W_t is the transmitted load, m is the module, the maximum permissible stress for the material σ_p = 245MPa (EN8 Steel), Y is the Lewis Form Factor and K_v is the dynamic factor given by:

$$K_v = \frac{6.1}{6.1 + V} \qquad (Equation\ 2)$$

Where V is the pitch line velocity

Bearings-

To decide on the bearings, the minimum dynamic load rating, C_{min}, was calculated. This was done using the following formula :

$$C_{min} \geq P \times (L_{10})^{\frac{1}{k}} \qquad (Equation\ 3)$$

Where k depends on the type of bearing and L_{10} the basic life rating. P can be calculated as follows:

$$P = XF_R + YF_A \qquad (Equation\ 4)$$

Where X is the radial load factor, F_R is the radial load, Y is the axial load factor and F_A is the axial thrust force. The bearings were selected once again from the RS catalogue and adjusted based on the shaft sizes.

5. Final Design Breakdown

Our final design is shown on the right. It features a wedge-shaped design, with a slot in the cover for the front wheels. The design is small and compact, with space and weight being saved where possible. Below shows a detailed in-depth breakdown of individual components.

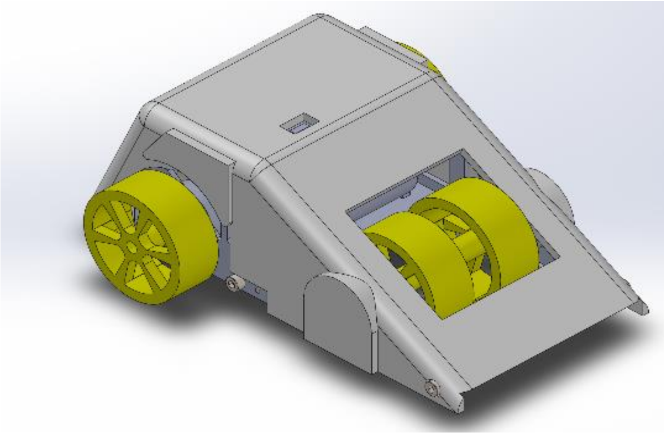

Figure 4: Final Design

a. Chassis design

The side plates of the chassis had triangular cut-outs in order to reduce the overall weight of the vehicle. The battery would be taped on the battery plate while two nuts would constrain the eyebolt on its mount. The motor would be constrained on the base plate by two plates either side. Dowel holes and clearance holes for bolts were placed to ensure correct location of all crucial parts. If the plates were not parallel, it would risk the gears not meshing properly, and the bearing mounts needed to be parallel and concentric to a high degree of tolerance.

Figure 5: Chassis design

b. Front Shaft-

The front shaft was designed so that the wheels go inside the chassis plates. This was done to reduce air resistance off the wheels and so that if stepped upon, the wheels would transfer the force, directly to the ground, allowing us to have a lighter shaft with a smaller diameter. The aesthetics of inside wheels, such as is done with the Batmobile, was also considered. The bearings were constrained using shoulders, 2 circlips and a bolt at the end to constrain axial movement. A floating surface was left between the shaft and bearing.

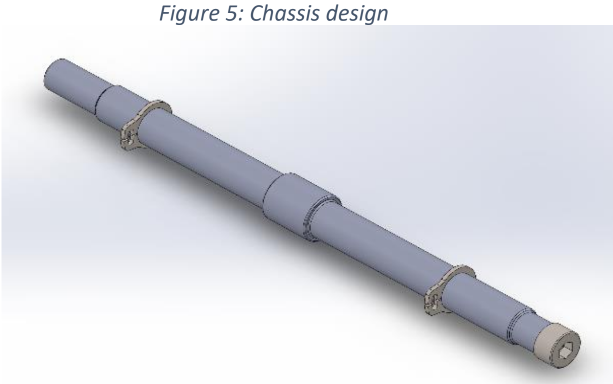

Figure 6: Front shaft

c. Rear Shaft-

The rear shaft is directly connected to the motor via a single pair of gears. Since the wheels are made of plastic, it was undesirable to bore them to an acceptable diameter to fit onto the shaft. Hence, the ends of the shaft (the place where we intend to attach the wheels) were modified so that they had the exact dimensions as the interior of the wheel. The bearings are constrained by a shoulder and circlip on both ends (the unconstrained edge is the outer edge of one bearing). The gear is constrained axially by a shoulder and a circlip, and rotationally by a keyway. The positioning of the shoulder is such that the length of the tolerance cut is minimal (only 15.5mm) and center of mass is roughly in the center of the shaft to aid balance.

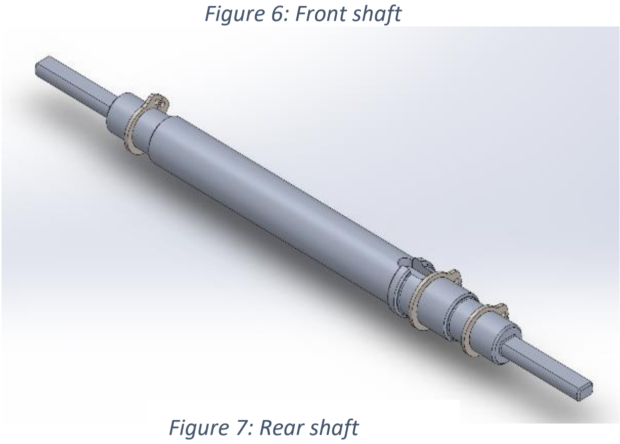

Figure 7: Rear shaft

d. Bearing Mounts-

The bearing mounts are 10mm wide discs of 35mm outer diameter and 19mm (front) and 22mm (rear) inner diameter. Each has a shoulder to constrain the outer edge of the bearing axially and four holes running through the width, two 3mm dowel holes for accurate location and two M3 clearance holes to constrain the mounts to the side plate. The side plate is flush with the other edge of the bearing.

e. Drive transmission-

The final design involved a single stage transmission consisting of a 50-tooth gear and a 25-tooth gear, both spur gears of modulus 1 and width 6mm. Steel gears of these dimensions adequately satisfied the Lewis equation and increased torque to an acceptable level without being excessively heavy or complex to manufacture. The chosen ratio of motor to rear axle speed was 1:2. It was decided that this was a good compromise between vehicle speed and wheel torque.

The design was intended to be small and light and will benefit from greater acceleration because of this. If the torque is lower than some vehicles with greater step-down ratios, the reduction in mass should compensate for it.

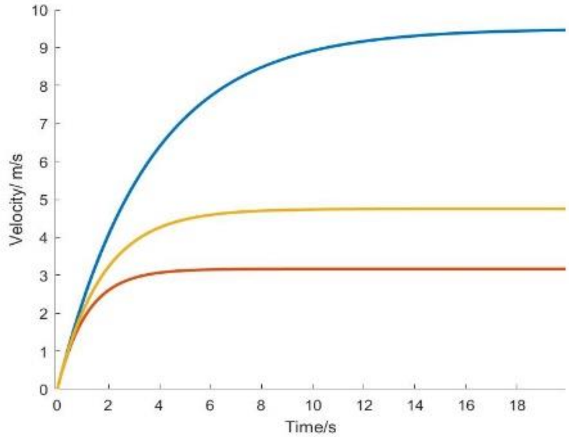

Figure 8: Graph showing a velocity-time plot for 3 different drive transmission ratios. (Red 1:1, Yellow 1:2, Blue 1:3)

A *Matlab* script was used to plot graphs of velocity vs time & torque vs time with different transmission ratios. The graphs below show the velocity-time trace for transmission ratios of 1, 2 and 3 respectively. A transmission ratio of 2 was chosen as decreasing the ratio causes the time taken to reach top speed to increase whilst increasing the ratio compromises the car's top speed. For detailed calculations, refer to Appendix A.

f. Bearings

The bearings were chosen with knowledge of the C_{min} and shaft diameters.

Characteristics	Rear bearings	Front bearings
Inner diameter, d (mm)	*10*	*6*
Outer diameter, D (mm)	*22*	*19*
Width, B (mm)	*6*	*6*
Basic dynamic load rating, C_d (N)	*2160*	*1474*
Basic static load rating, $C_{s\,(N)}$	*1040*	*558*

Table 1: Selected Bearing characteristics

g. ABS plastic cover design

The 3D cover was the final component to be designed and was dimensioned to fit over the entire side plates' assembly, shielding all moving transmission elements, and be relatively unobtrusive. Should the vehicle receive a large normal force, such as being accidentally stepped on, the cover will withstand the force. The switch is accessed from a hole in the top which aligns with the battery pack.

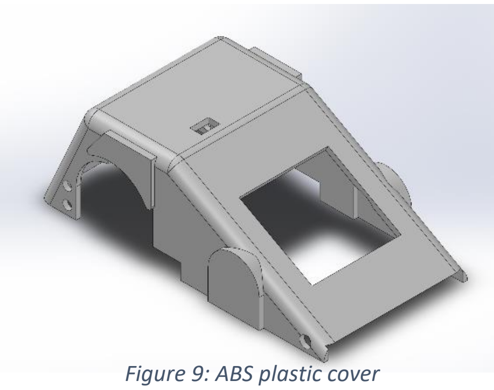

Figure 9: ABS plastic cover

6. Potential Failure Modes & Mitigation

a. Manufacture Error

Front Shaft – It was necessary to bore through the entirety of the wheel hub. This could potentially snap due to the low tensile strength of the ABS plastic if the bore was too large. Therefore, the front shaft was modified to a diameter of 7mm to minimise the bore hole.

Gears -The RS gears were made of hardened steel. Due to the diameters of the keyless bush and rear shaft, the hubs were removed from both gears. There was extra precaution needed to machine hardened steel, but it met the criteria for the file test. The gears were bored with caution nonetheless.

b. Gear stress

From Equation 1 and 2, the minimum face width of the selected gear was calculated to be 0.551mm for the smaller gear and 0.492mm for the larger gear. Both gears have a face width of 5mm which gives a safety factor of 9.07 for the smaller gear and 10.16 for the larger gear.

c. Bearing life

The product is designed to operate at a minimum of 100 hours before failure. The basic life rating, L_{10}, was calculated to be 14.22×10^6 revolutions. The planned maximum mass of the vehicle was 2 kg. To assume the worst case scenario, each bearing withstands a radial load equivalent to half of the weight of the vehicle. Since the car is moving in a straight line, there is no axial thrust acting on the bearings.

For the loading conditions used, X = 1 and Y = 0 for Equation 4, which agrees with our intuition that only the weight of the car gives the equivalent load. Hence, using Equation 3 with k = 3 for deep groove bearings, The C_{min} was calculated to be 29.7N which was way below the chosen bearings.

d. Weight failure

The first point of contact when a person steps on the car would be on the plastic cover. There are two possibilities. Firstly, the person's foot would land directly on the flat part of the plastic cover. The second scenario would involve the person stepping directly onto the front wheels. However, regardless of wherever the foot lands on, the weight of the person would also be transmitted to the side plates and rear shaft too. For detailed calculations, refer to Appendix B.

Plastic cover-

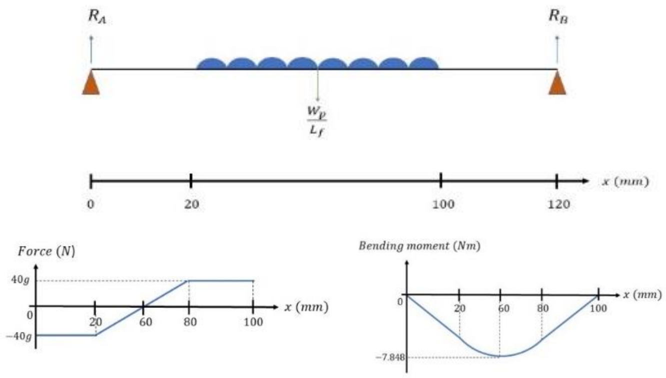

Figure 10: Diagrams showing the free body diagram, force diagram and bending moment diagram of the plastic cover

As shown in Figure 9, the plastic cover was modelled as a beam and the foot as a distributed load over 80 mm. The average adult shoe width was rounded off to the lower tenth, estimated at 80 mm. The maximum bending moment was calculated to be 7.848 Nm. The second moment of area was calculated as 7.29×10^{-10} m^4.

Thus, using equation 5, the σ was found to be 26.91 MPa. The flexural strength of ABS plastic is known to be 75.84 MPa. Together with the σ, and equation 6, the safety factor was calculated to be 2.8. This stress factor was considered adequate for the design.

$$\sigma = \frac{My}{I} \qquad (Equation\ 5)$$

Where σ is the maximum stress, M the bending moment, y the distance from the centre of the beam to the top and I the second moment of area.

$$Safety\ Factor = \frac{Maximum\ allowable\ stress}{Operating\ stress} \qquad (Equation\ 6)$$

Wheels-

If a person steps directly onto the front wheels, the load would be transmitted through plastic. Area of contact is estimated by multiplying shoe-width and diameter, which amounted to roughly 5.6 cm^2. Dividing the weight of the person this by, the stress exerted is about 1.4MPa. The flexural strength of ABS plastic is known to be 75.84 MPa which gives a safety factor of 54.

Side plates-

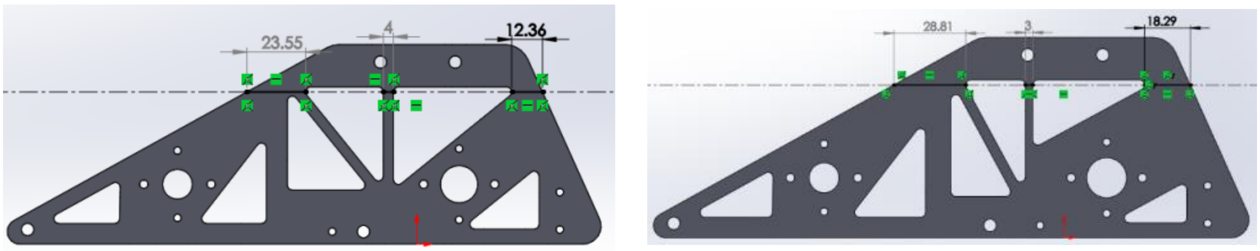

Figure 11: Side plates and shortest lengths taken into calculation

The irregular geometry complicates the identification of stress pathways along the part. We modelled the part as a flat plate of the same volume, width and height. The smallest lengths of the original side plate are added to form the length of the modelled part (as shown in Figure 11). This is to maximise stress acting on the plate. The whole weight of the person is assumed to act on one plate.

For mild steel EN1A, the σ_y is known to be 250 MPa. The stress acting on plate 1 (left) and 2 (right) were calculated as 6.55 MPa and 5.22 MPa respectively. The safety factor was found to be reasonable at around 38.

Rear shaft-

The whole weight of the person is assumed to act on the rear shaft. At the point where the bending moment is maximum, the second moment of area was found to be 4.10 x 10^{-10} m^4. Using equation 1, the σ_{max} was found to be 67.9 MPa. For mild steel EN1A, the σ_y is known to be 250 MPa. Using equation 2, the safety factor was calculated to be around 3.7 which was considered sufficient.

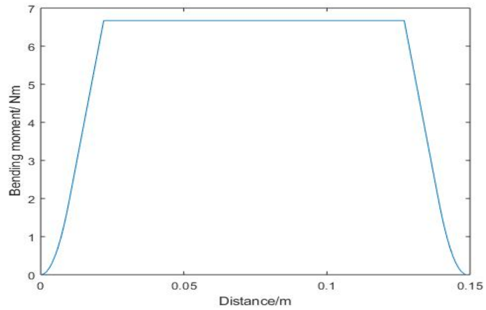

Figure 13: Bending moment diagram of rear shaft

At the point where the second moment of area is minimum, the second moment of area was found to be 1.45 x 10^{-11} m^4 . Using equation 1, the σ_{max} was found to be 237 MPa. The safety factor was found to be low at around 1.05. Considering that most of the assumptions were exaggerated (eg. Full weight falling on rear, weight modelled as point force, taking lower estimate of second moment of area), the rear shaft is unlikely to fail unless it is repeatedly stepped on. However, a cautionary note should be attached.

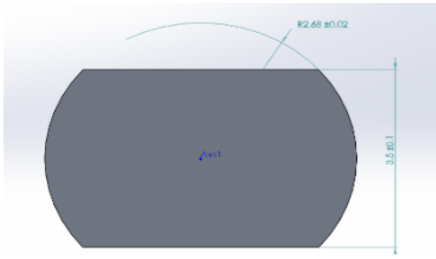

Figure 12: Cross section of thinnest part of axle

7. Capability Prediction

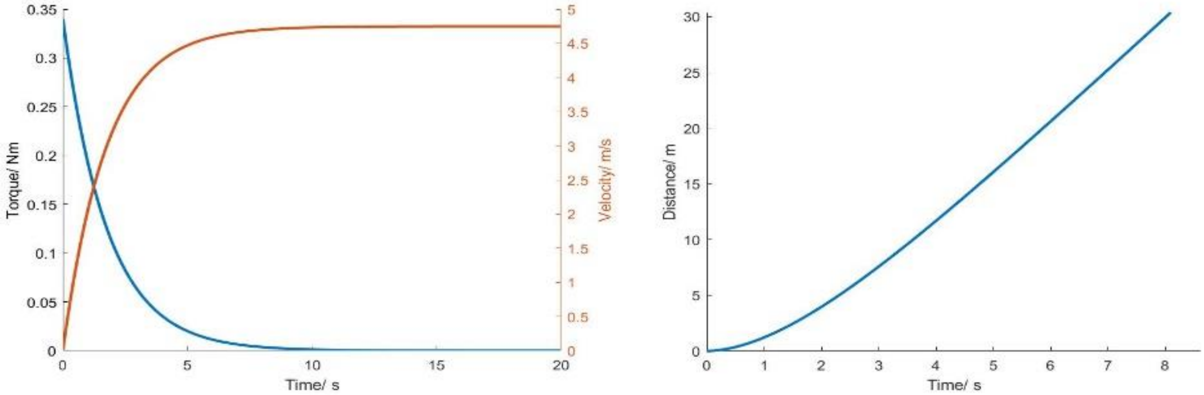

Figure 14: Torque over time and distance over time graphs

As can be seen from the graphs above, our car's expected top-speed is roughly 4.75m/s. It is predicted to finish the race in roughly 8(.0136…) seconds and has a stall torque of 0.34 Nm. Since the diameter of the wheel is 65mm, the max expected load it can move is roughly 10.5N or a 1kg mass.

$$F = \frac{2T_s}{d} = \frac{2 \times 0.34}{0.065} \approx 10.5N \qquad (Equation\ 7)$$

8. Conclusion

The entire team is proud of the final design and a promising final product will be delivered on schedule. The team worked well together, quickly learning to capitalise on each team member's skillset. The product fits the criteria set in the PDS efficiently. Our design focuses on being simple and small, whilst also finding ways to innovative, such as keeping the front wheels inside the plates for aerodynamic and aesthetic effect.

Throughout the process of design, capability calculations and manufacturing, challenges were met and overcome. Time management is an area in which our team initially struggled with. After realising the importance of this in week 5 during the gateway review more effective planning methods were put in place. This made it possible to proceed on schedule during manufacturing, including having added time in case human error demanded parts be remade.

Looking to the future, it is believed that the design will perform well in both torque and speed tests. Speed has been somewhat prioritised, theoretically topping at roughly 4.75m/s and taking 8 seconds to finish the race. Although the aim of the challenge is to win, the difficulty lies in the sacrificial relationship between speed and torque. With the low mass of the car, it is anticipated that the car will have an edge over heavier vehicles in the race. With regards to performance in the tug of war, it will have to be seen on the day.

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10. Appendices

Appendix A-

Characteristics of a DC Motor are characterized by the following equation:

$$T_m = T_s\left(1 - \frac{\omega}{\omega_{max}}\right) \qquad (Equation\ 8)$$

Where T_m is the motor torque, T_s is the stall torque, ω is the angular velocity, and ω_{max} is the no load angular velocity.

Given linear velocity v = rω, the above equation can be re-written as:

$$T_m = T_s\left(1 - \frac{v}{v_{max}}\right) \qquad (Equation\ 9)$$

Since the driving force is coming solely from the friction from the wheels,

$$a_w = \frac{F_w}{m} = \frac{2T_w}{D \times m} \qquad (Equation\ 10)$$

Where a_w is the acceleration of vehicle, F_w is the force driving the vehicle, m is mass, D is diameter of the wheel, and T_w is the torque on the wheel.

$$T_w = X \times T_m \qquad (Equation\ 11)$$

Where X is the transmission ratio.

Combining the equations 9, 10 and 11, we get:

$$a_w = \frac{2XT_s}{Dm}(1 - \frac{v}{v_{max}}) \qquad (Equation\ 12)$$

$$dv = adt \qquad (Equation\ 13)$$

$$v = \int adt \qquad (Equation\ 14)$$

Equations 12, 13 and 14 were entered into a MATLAB script and used to plot graphs.

Appendix B-

Plastic cover

The bending moment equation is expressed as:

$$M(x) = -R_A x + \frac{W_p}{L_f} < x - 0.02 >^2 - \frac{W_p}{L_f} < x - 0.01 >^2 \qquad (Equation\ 15)$$

Rear Shaft

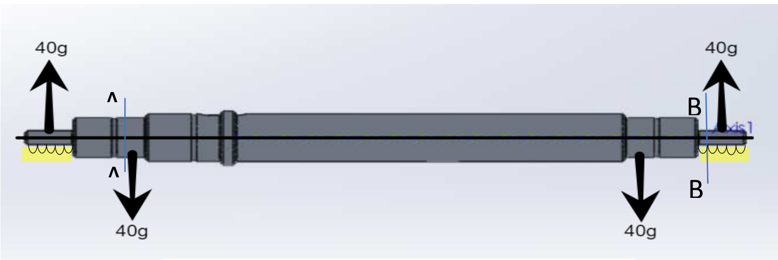


Figure 15: Force diagram of rear shaft

Maximum bending moment-

Second moment area for cylindrical cross-sections,

$$I_A = \frac{1}{4}\pi r^4 = \frac{1}{4}\pi 0.005^4 \approx 4.909e^{-10}m^4 \qquad (Equation\ 16)$$

Using equation 5,

$$\sigma = \frac{6.6708 \times 0.005}{4.909e^{-10}} \approx 67.9MPa \qquad (Equation\ 17)$$

Minimum second moment of area-

A lower estimate of its second moment of area was taken by treating it as a rectangular cross-section, finding width using Pythagoras' Theorem

$$b = \sqrt{(2 \times 0.00268)^2 - (0.0035)^2} \approx 0.00405m \qquad (Equation\ 18)$$

where b is the thickness of the beam

Second moment area for rectangular cross-sections,

$$I_B = \frac{bd^3}{12} = \frac{0.00405 \times 0.0035^3}{12} \approx 1.447e^{-11}m^4 \qquad (Equation\ 19)$$

where d is the width of the beam

Using equation 5,

$$\sigma = \frac{1.962 \times 0.00175}{1.447e^{-11}} \approx 237.3MPa \qquad (Equation\ 20)$$

Side Plates

Stress is defined as force divided by area,

$$\sigma_{max} = \sigma_z = \frac{F_{weight}}{Area} \qquad (Equation\ 21)$$

The area of the side plate is the sum of the smaller lengths multiplied by the width,

$$Area = l_{sum} \times w \qquad (Equation\ 22)$$

Where l_{sum} is the sum of the smaller lengths expressed in the following equation,

$$l_{sum} = l_1 + l_2 \ldots + l_n \qquad (Equation\ 23)$$

Equation 20 is then used to find the stress acting on the side plates,

$$\sigma_1 = \frac{80g}{(0.02355 + 0.004 + 0.01236) \times 0.003} = 6.55MPa \qquad (Equation\ 24)$$

$$\sigma_2 = \frac{80g}{(0.02881 + 0.003 + 0.01829) \times 0.003} = 5.22MPa \qquad (Equation\ 25)$$

Using equation 6 to identify the safety factor by taking the higher stress of the two,

$$Safety\ Factor = \frac{250}{6.55} = 38.14 \qquad (Equation\ 26)$$

Technical Realisation Document

Manufacturing Considerations and Design Features for Key Components

Motor Front Support-

The motor front plate was a toleranced component, that had few features. The plate started off as a 5mm aluminium plate which was cut down to rough size using a hacksaw. The only difficulty was ensuring that the holes on the base of the plate were positioned correctly in relation to the holes on the front of the plate, as the motor position is vital in meshing the gears correctly. Therefore, the position of the holes were toleranced at ±0.05, a milling machine was used to ensure high precision. The dowel holes on the base of support plate were drilled at a smaller size of 2.8mm before being reamed at the correct size to ensure a snug fit. Dowels were used either side of a bolt on the base of the plate, as they are more positionally accurate than bolts. Manufacturing constraints meant that the curved top surface of the plate could not be achieved by the hand-operated milling machine. Therefore, the corners were filed down by hand.

Front Shaft-

The front shaft is a critical component and as such it was manufactured early on, allowing time in case of potential manufacturing error. The shaft, which began as a length of 10mm steel rod was initially cut to rough length using a band saw, before being faced off on a lathe, held by a collet chuck. The lathe was also then used for producing the shoulders, circlip grooves, chamfers and fillets on one half of the piece. It was then rotated in the chuck and the opposing half was manufactured.

The piece was small and therefore there was a risk of breaking during manufacture due to large forces produced by removal of material with the lathe tool. A centre hole was drilled, and the piece was held in place with a revolving centre combatting this manufacturing constraint. A hole was drilled into the end and tapped. The piece was checked using a micrometre and a Vernier calliper for the diameters and lengths respectively.

a. Floating surface- Allow axial movement for shaft. Bearing constrained in its housing but allowed to move on the shaft. A tolerance of band js5, found from bearing tolerance tables.
b. - Chamfer on outside edges- To ensure the bearings and wheels are pressed up against the shoulder.
   - Fillets on inside edges- Modelled due to manufacturing constraints of lathe tool.
c. Circlip grooves- To prevent axial movement of wheels. Circlips deemed appropriate due to low axial force on wheels. Appropriate circlip tolerance applied.
d. Central shoulder- To prevent axial movement of wheels.
e. Shoulder and bolt to constrain bearing axially. High tolerance on length of ±0.02mm to ensure bearing fit.
f. Wheel shaft tolerance such that the wheel will fit onto the shaft.

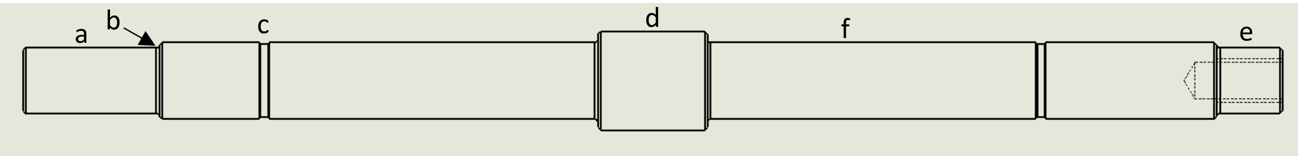


Figure 1: Front Shaft Drawing

Rear Shaft-

The rear shaft was made by cnc lathing. The keyway and custom made edges were milled using appropriate cutting tools. Most measurements were checked by a Vernier Caliper. High tolerance measurements were measured using a Micrometer.

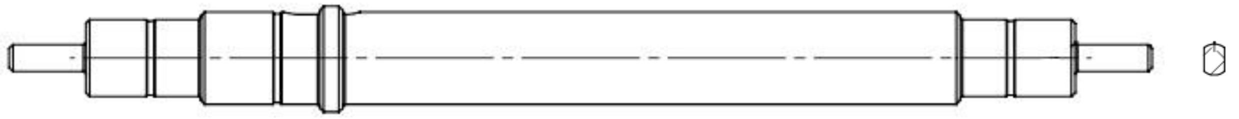


Figure 2: Rear Shaft Drawing

The two outside edges are where we plan to fit the wheel. Cross-section is shown on right. All edges are chamfered for ease of assembly. Additionally fillets applied so that bearings/gears would fit in comfortably. The bearings are on the ends (right before the wheels), and the gear is located in the center. Gear and bearings are constrained by combination of shoulder and circlip.

Motor mount baseplate-

A 40mm x 10mm flat plate was cut using a hacksaw to a length of 120mm. It was then faced-off using the milling machine to a length of 115mm. The 16mm x 25mm section was then milled off. Although the milled off section had a rounded corner because of the geometry of the tool, it was not critical to the whole assembly. The dowel holes were drilled at a smaller size of 2.8mm and reamed while the threaded holes were drilled at a smaller size of 2.4mm and then tapped. To check the measurements and confirm they were within tolerances, a Vernier calliper was used.

A dowel hole was necessary on the right motor mountplate because it had to line up the gears. The dowels and bolts constraining the baseplate to the side plates ensured the side plates were parallel. This was possible by assigning suitable tolerances.

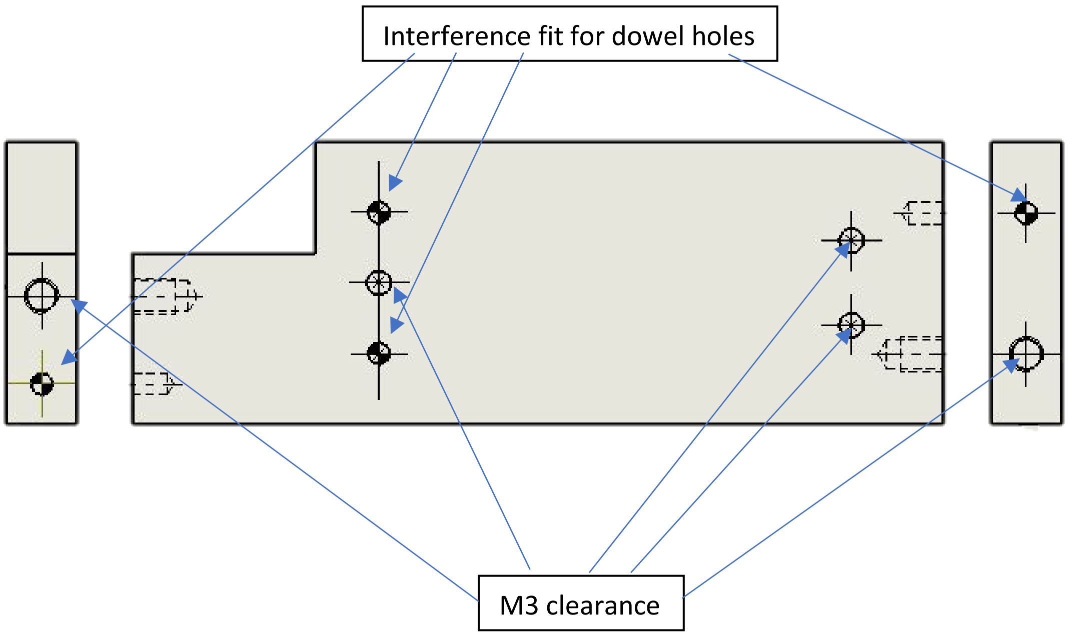
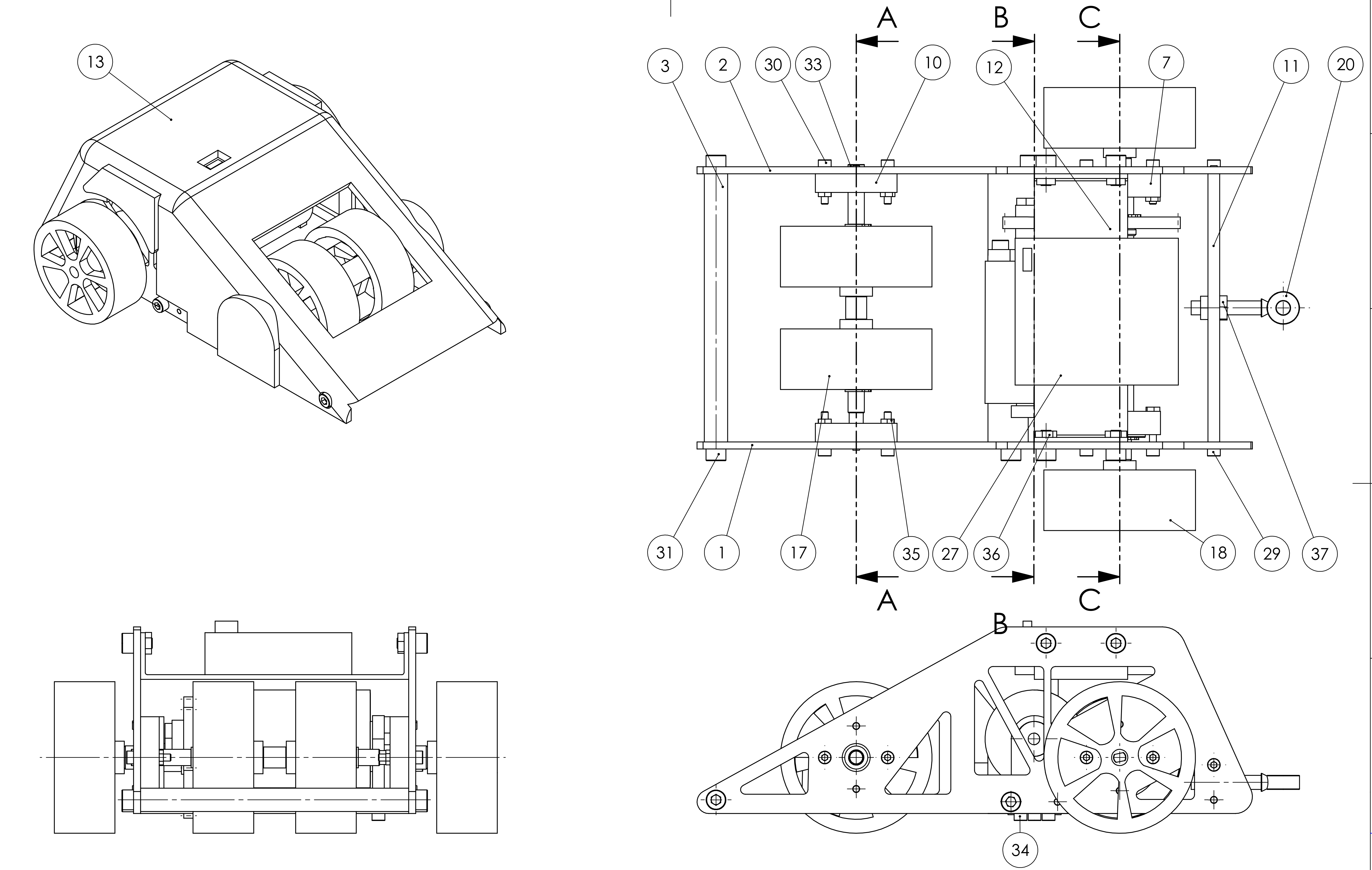
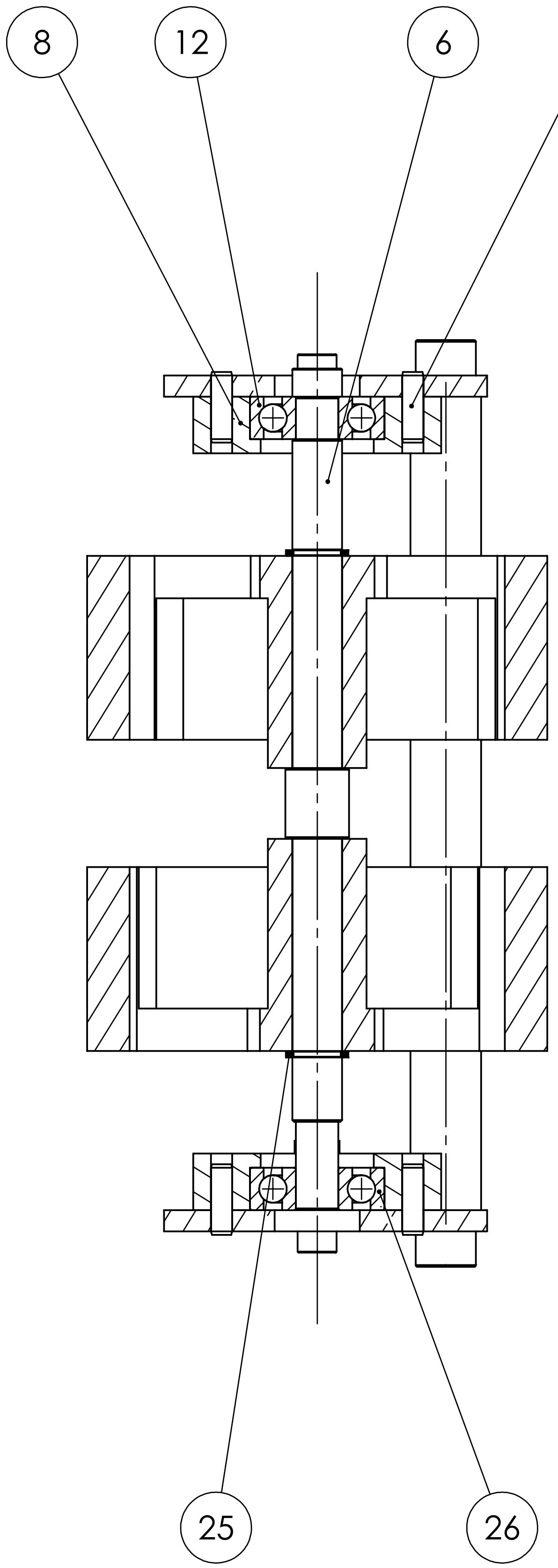


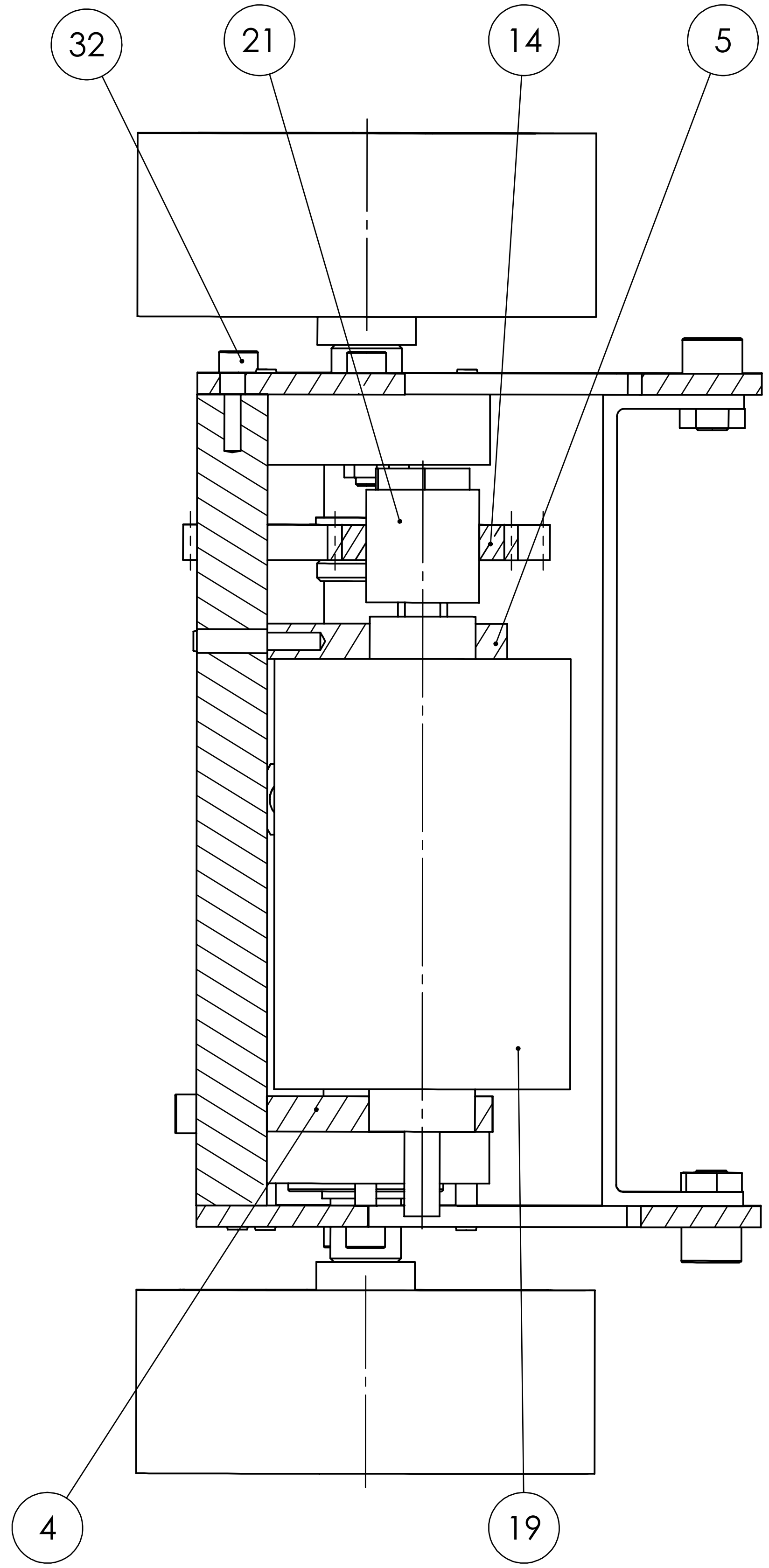
Figure 3: Motor Mount Base Plate drawing



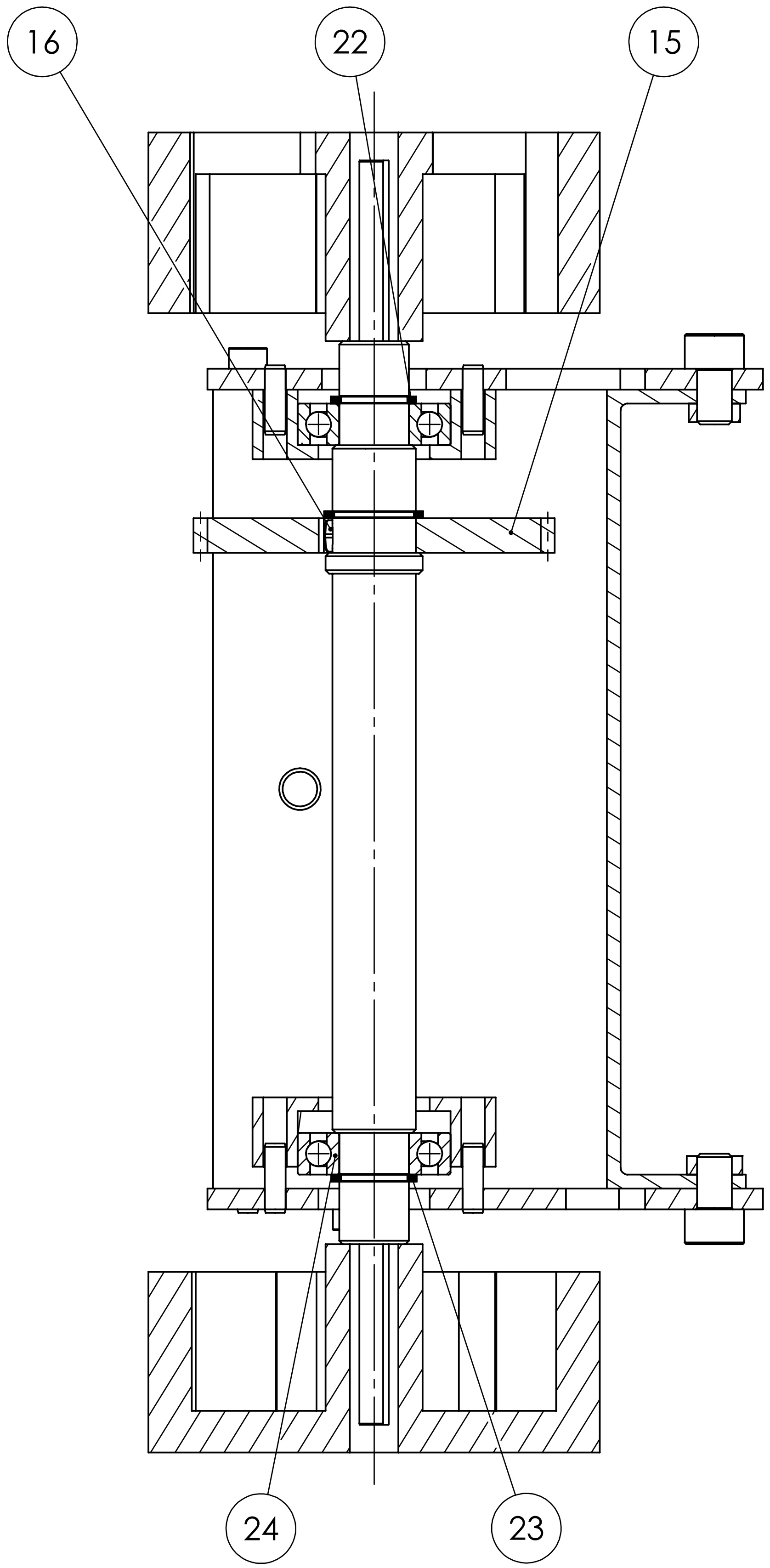
TOLERANCES		THIRD ANGLE PROJECTION		MATERIAL:	TITLE: Assembly drawing	Imperial College London Department of Mechanical Engineering
X =± 0.5 X.X =± 0.1 X.XX =± 0.02	ANGULAR ±1° SURFACE FINISH MACHINED FACES Ra 6.3			ALL DIMENSIONS ARE IN MILLIMETRES		
	NAME		DATE			
DRAWN	SYABIL HUD		01/11/2018	DO NOT SCALE DRAWING	DWG No.	
CHECKED	Rajasegaran Rohit		07/12/2018	A3 SCALE 2:3	P6_GA001	
APPROVED						SHEET 1 OF 3 REVISION



SECTION A-A



SECTION B-B



SECTION C-C

TOLERANCES X =± 0.5 X.X =± 0.1 X.XX =± 0.02	ANGULAR ±1° SURFACE FINISH MACHINED FACES Ra 6.3	THIRD ANGLE PROJECTION	MATERIAL: ALL DIMENSIONS ARE IN MILLIMETRES
	NAME	DATE	DO NOT SCALE DRAWING
DRAWN	SYABIL HUD	01/11/2018	
CHECKED	Rajasegaran Rohit	07/12/2018	A3 SCALE 1:1
APPROVED			

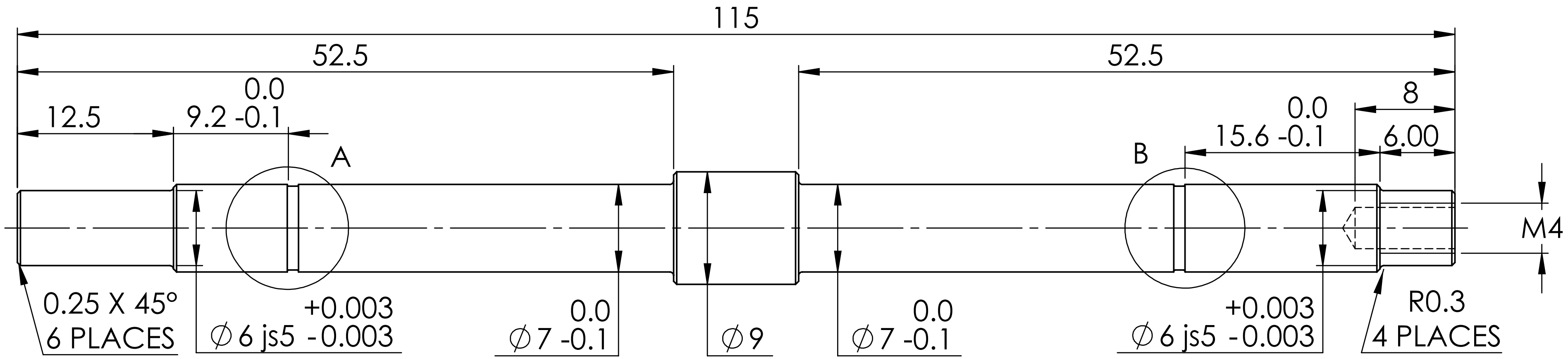
TITLE: Assem

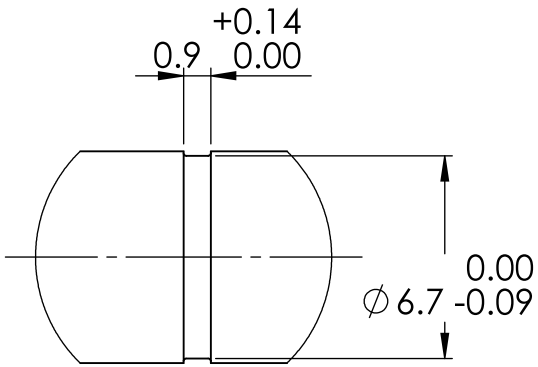
DWG No. P6_GA002

Imperial College London
Department of Mechanical Engineering

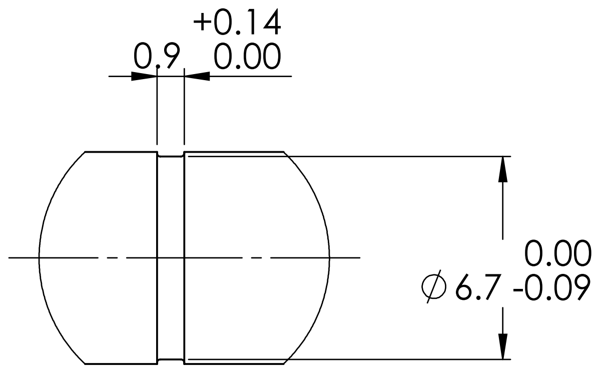
SHEET 2 OF 3 REVISION

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	P6_LSP001	LEFT SIDE PLATE, MOTOR MOUNT SCREW HOLES ARE FURTHER AWAY	1
2	P6_RSP002	RIGHT SIDE PLATE, MOTOR MOUNT SCREW HOLES ARE CLOSER TOGETHER	1
3	P6_CR02	CONNECTING ROD	1
4	P6_LMM001	LEFT MOTOR MOUNTPLATE	1
5	P6_RMM002	RIGHT MOTOR MOUNTPLATE	1
6	P6_FS001	FRONT SHAFT	1
7	P6_RS002	REAR SHAFT	1
8	P6_FBM02	FRONT BEARING MOUNTS	2
9	P6_RBM02	REAR BEARING MOUNTS	2
10	P6_MMBP001	MOTOR MOUNT BASE PLATE	1
11	P6_EM001	EYEBOLT MOUNT	1
12	P6_BP001	BATTERY PLATE	1
13	P6_PC001	3D PRINTED COVER	1
14	P6_GT25	SPUR GEAR 25 TEETH, RS Part No. 878-7913 (MOTOR GEAR)	1
15	P6_GT50	SPUR GEAR 50 TEETH, RS Part No. 878-7922 (REAR SHAFT GEAR)	1
16	P6_K10	KEY 10MM (REAR SHAFT GEAR)	1
17	P6_FW02	FRONT WHEELS	2
20	P6_EB001	EYEBOLT	1
21	P6_KB001	KEYLESS BUSH, RS Part No. 815-262 (MOTOR GEAR)	1
22	P6_EC12	EXTERNAL CIRCLIP 12MM (REAR SHAFT AND GEAR)	1
23	P6_EC10	EXTERNAL CIRCLIP 10MM (REAR SHAFT AND BEARINGS)	2
24	P6_SKB10	BEARING 10MM, RS Part No. 893-7461 (REAR SHAFT)	2
25	P6_EC07	EXTERNAL CIRCLIP 7MM (FRONT SHAFT AND WHEELS)	2
26	P6_SKB08	BEARING 6MM, RS Part No. 286-7849 (FRONT SHAFT)	2
27	P6_BB001	BATTERY BOX	1
28	P6_DP001	DOWEL PINS (MOTOR MOUNT BASE PLATE, EYEBOLT MOUNT, BEARING MOUNTS)	14
29	P6_SM310	M3 X 10 SCREW (EYEBOLT MOUNT AND SIDE PLATES)	2
30	P6_SM316	M3 X 16 SCREW (BEARING MOUNTS, SIDE PLATES AND MOTOR MOUNT BASE PLATE, MOTOR MOUNTPLATE)	11
31	P6_SM58	M5 X 8 SCREW (BATTERY PLATE, CONNECTING RODS)	6
32	P6_SM510	M5 X 10 SCREW (MOTOR MOUNT BASE PLATE, SIDE PLATES)	2
33	P6_SM46	M4 X 6 SCREW (FRONT SHAFT)	1
35	P6_NM3	M3 NUT (BEARING MOUNTS)	8
36	P6_NM5	M5 NUT (BATTERY PLATE SCREWS)	4





DETAIL A
SCALE 4 : 1



DETAIL B
SCALE 4 : 1

TOLERANCES
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.02
ANGULAR ±1°
SURFACE FINISH MACHINED FACES Ra 6.3
THIRD ANGLE PROJECTION

MATERIAL:
Steel

ALL DIMENSIONS ARE IN MILLIMETRES

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Rohan Popat	31/10/2018
CHECKED	Aman Didwania	2/11/2018
APPROVED		

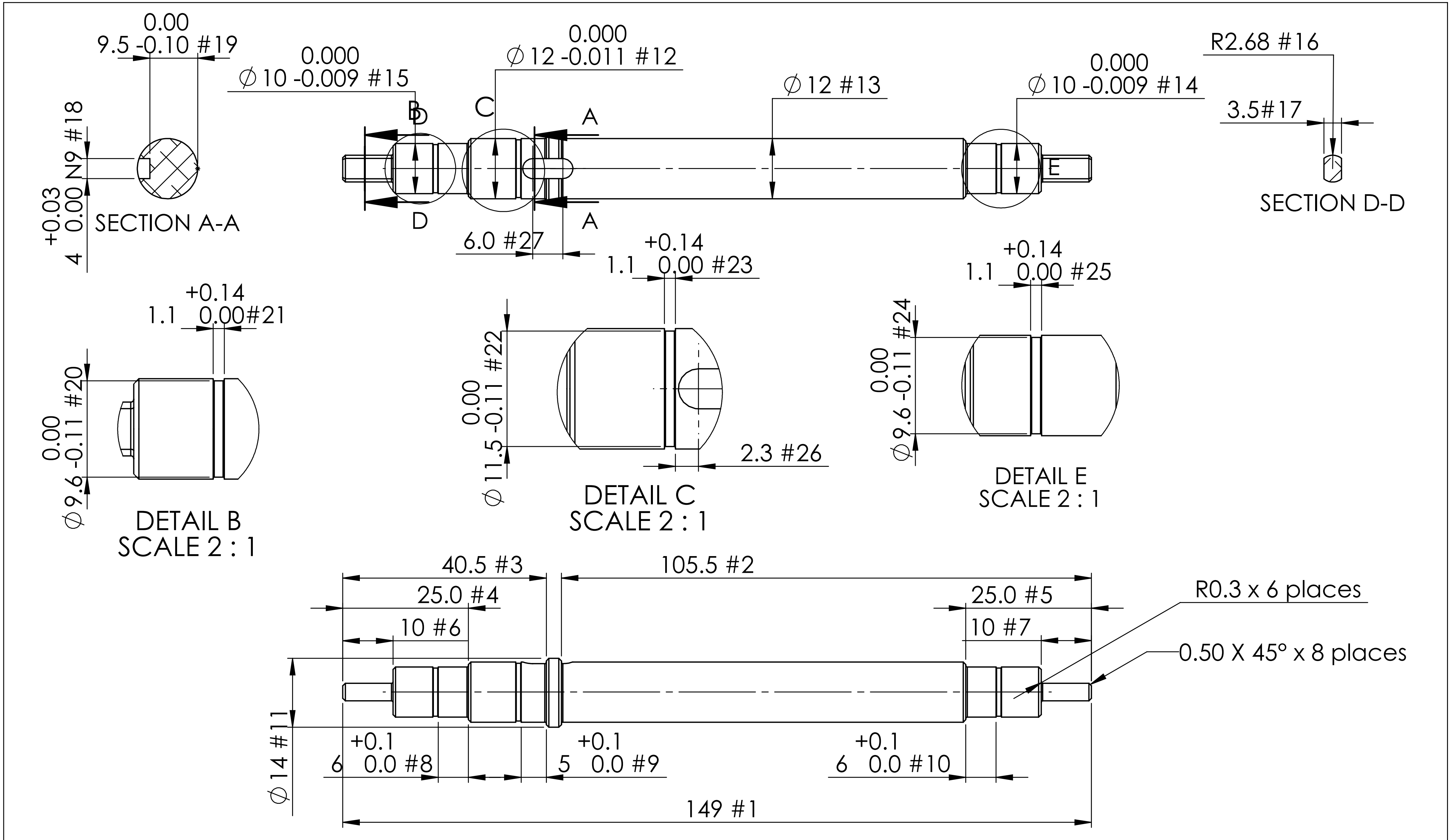
A4 SCALE 2:1

TITLE:
Front Shaft

DWG No.
P6_FS001

Imperial College London
Department of Mechanical Engineering

SHEET 1 OF 1 REVISION



All Shaft Tolerances are h6

TOLERANCES		THIRD ANGLE PROJECTION	MATERIAL: Mild Steel	TITLE: Rear Shaft
X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.02	ANGULAR ±1° SURFACE FINISH MACHINED FACES Ra 6.3		ALL DIMENSIONS ARE IN MILLIMETRES	
	NAME	DATE		
DRAWN	Aman Didwania	6/11/2018	DO NOT SCALE DRAWING	DWG No. P6_RS002
CHECKED	ROHAN POPAT	14/11/2018		
APPROVED	RICHARD VAN ARKEL	16/11/2018	A4 SCALE 1:1	

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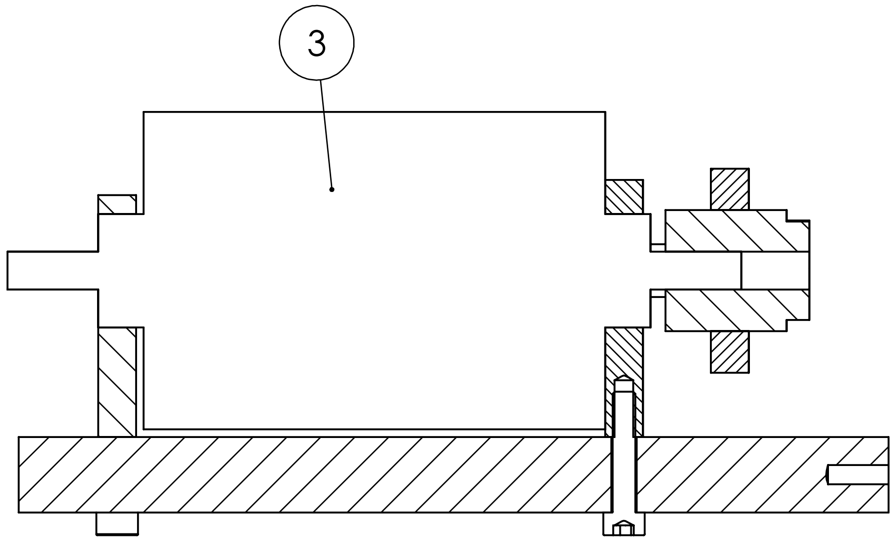
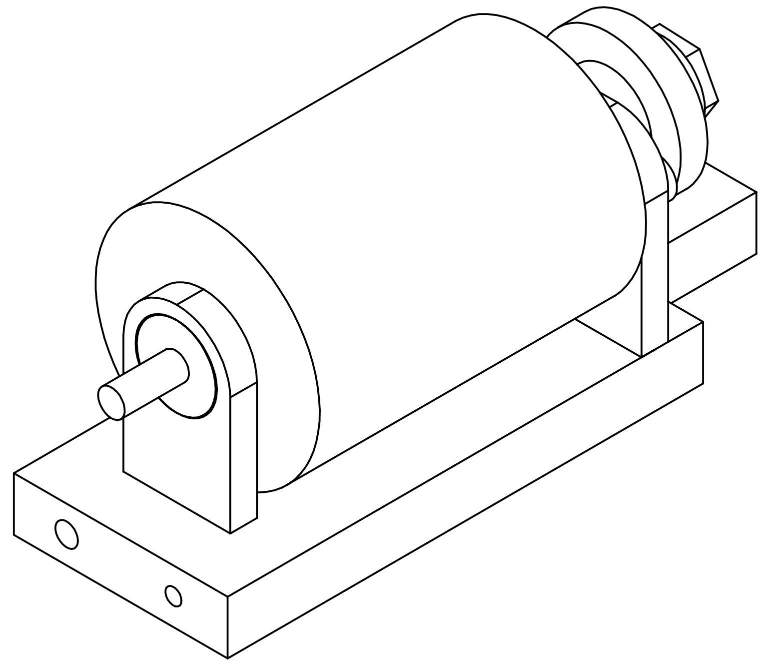
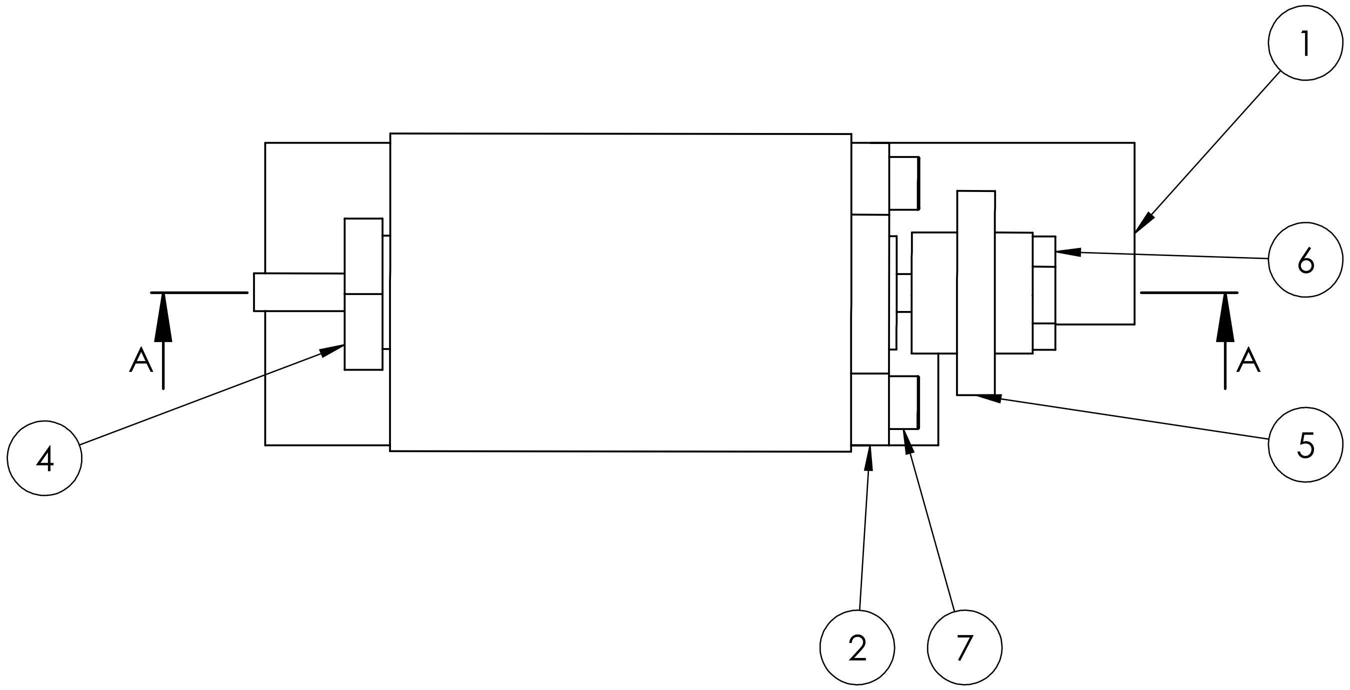
Department of Mechanical Engineering

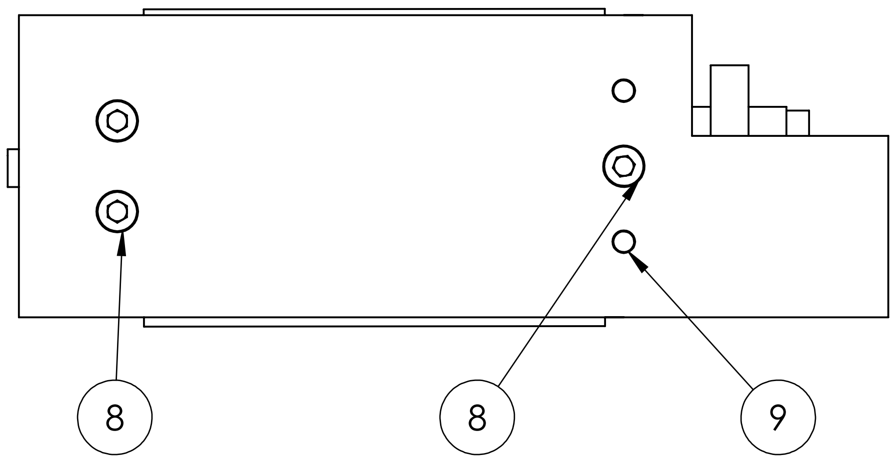
SHEET 1 OF 1 REVISION



SECTION A-A
SCALE 1 : 1







ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	p6_MMBP001	Motor Mount_base_plate	1
2	P6_LMM001	LEFT MOTOR MOUNT PLATE	1
3	P6_M001	Motor	1
4	p6_RMM001	RIGHT MOTOR MOUNT PLATE	1
5	P6_GT25	SPUR GEAR 25 TEETH, RS Part No. 878-7913 (MOTOR GEAR)	1
6	P6_KB001	KEYLESS BUSH, RS Part No. 815-262 (MOTOR GEAR)	1
7	P6_SM45	M4 X 5 SCREW (LEFT MOTOR MOUNTPLATE)	2
8	P6_SM46	M3 X 16 SCREW	3
9	P6_DP001	DOWEL PINS (MOTOR MOUNT BASE PLATE)	2

TOLERANCES
X =± 0.5
X.X =± 0.1
X.XX =± 0.02
ANGULAR ±1°
SURFACE FINISH MACHINED FACES Ra 6.3
THIRD ANGLE PROJECTION

	NAME	DATE
DRAWN		
CHECKED		
APPROVED		

MATERIAL:
ALL DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING
A3 SCALE 1:2

TITLE:
MotorAssembly
DWG No.
MotorAssembly_A3

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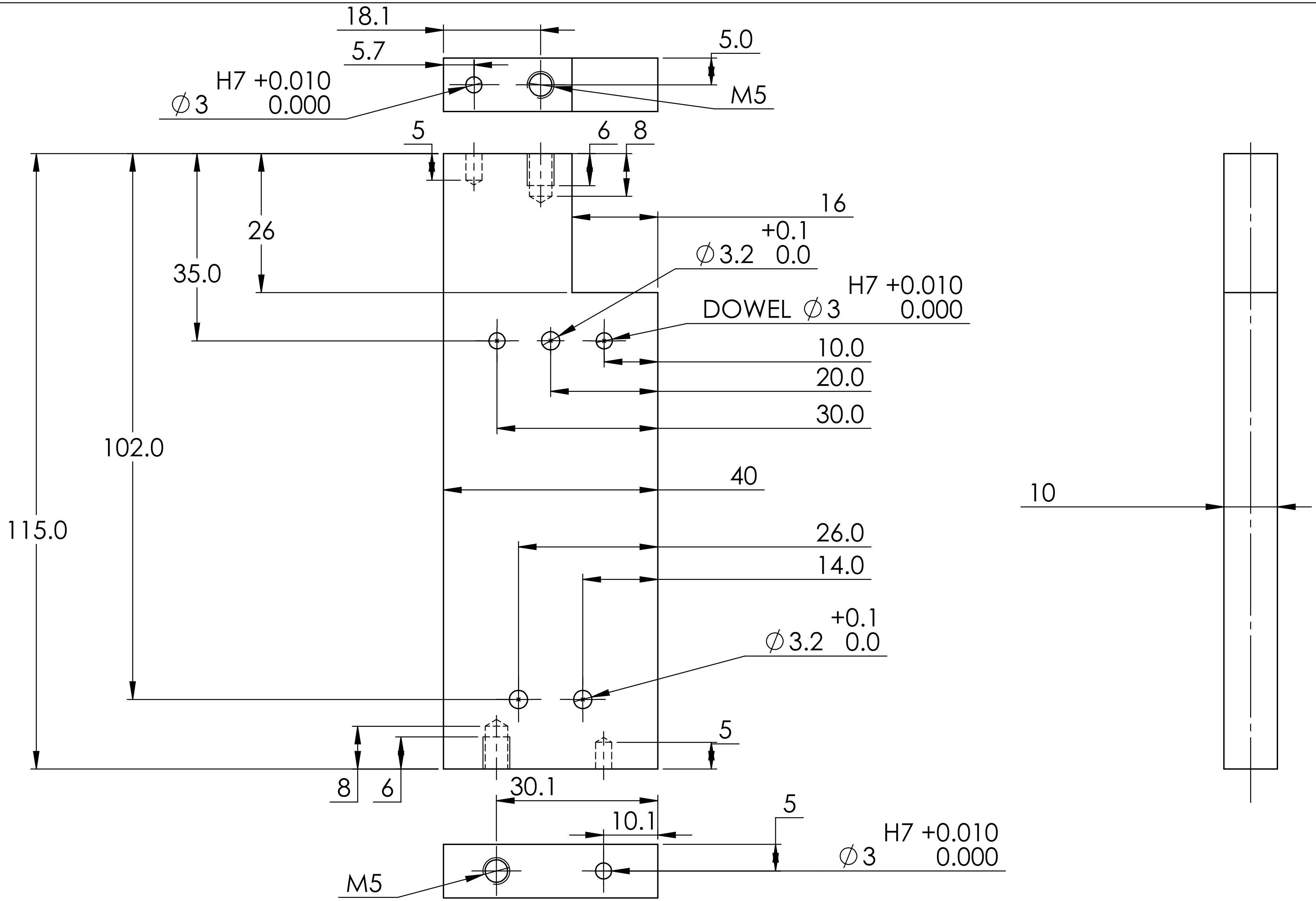
SHEET 1 OF 1 REVISION



TOLERANCES		THIRD ANGLE PROJECTION	MATERIAL: Aluminium	TITLE: Motor Mount-Base	Imperial College London Department of Mechanical Engineering
X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.02	ANGULAR ±1° SURFACE FINISH MACHINED FACES Ra 6.3		ALL DIMENSIONS ARE IN MILLIMETRES		
	NAME	DATE			
DRAWN	Rohan Popat	31/10/18	DO NOT SCALE DRAWING	DWG No. P6_MMBP001	
CHECKED	Aman Didwania	2/11/2018	A4 SCALE 1:1		SHEET 1 OF 1
APPROVED					REVISION

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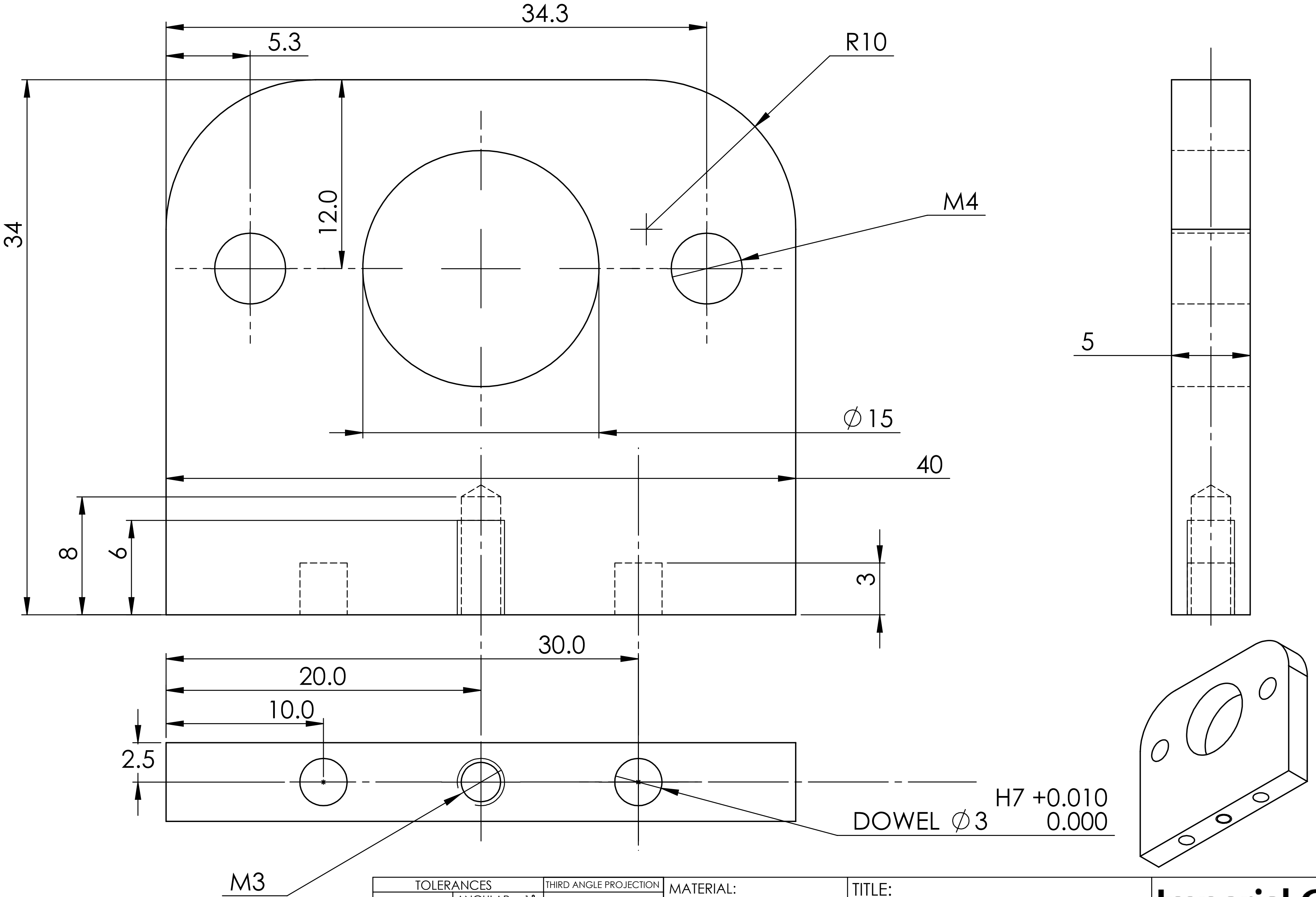
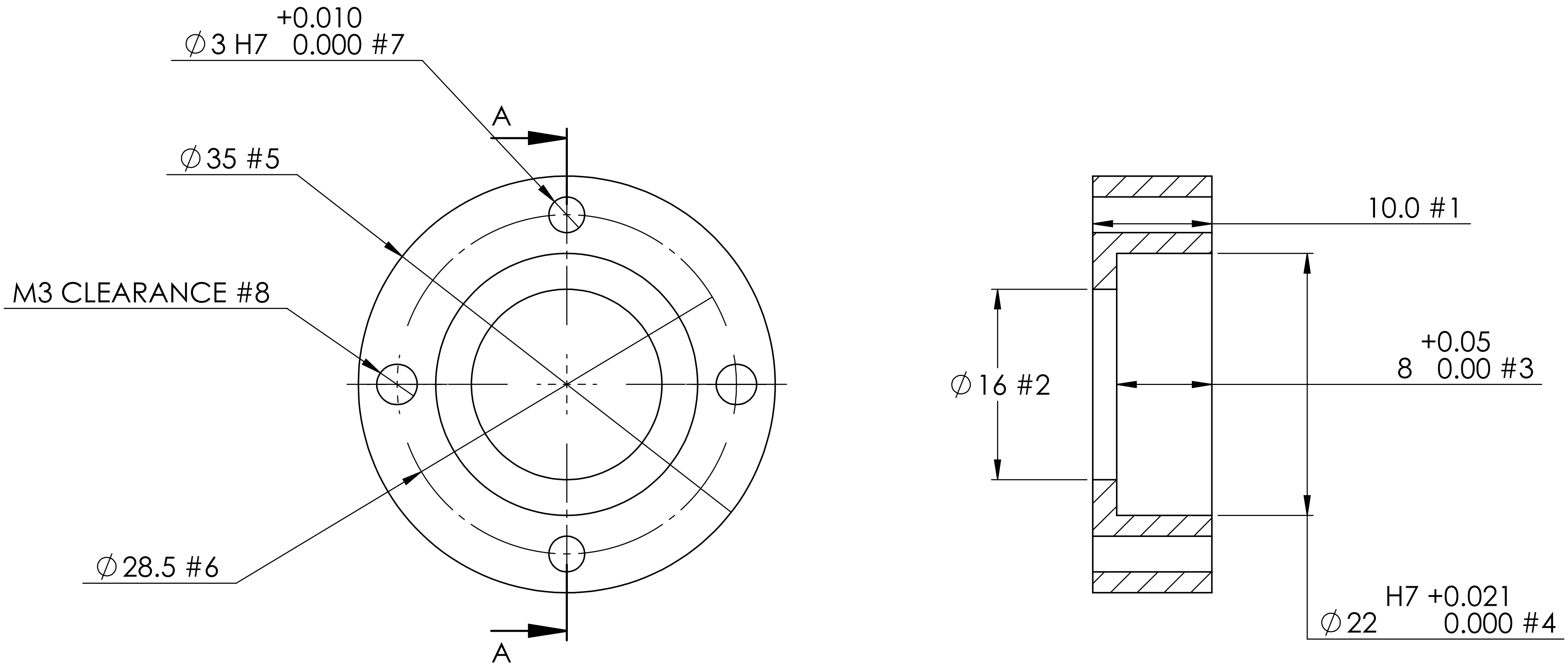


TOLERANCES X = ± 0.5 X.X = ± 0.1 X.XX = ± 0.02	ANGULAR ±1° SURFACE FINISH MACHINED FACES Ra 6.3	THIRD ANGLE PROJECTION	MATERIAL: ALUMINIUM ALL DIMENSIONS ARE IN MILLIMETRES	TITLE: Motor Mount-Front Support
	NAME	DATE		
DRAWN	Rajasegaran Rohit	28/11/2018	DO NOT SCALE DRAWING	DWG No. P6_LMM001
CHECKED	Aman Didwania	30/11/2018	A4 SCALE 3:1	
APPROVED				

Imperial College London
Department of Mechanical Engineering

SHEET 1 OF 1 | REVISION



SECTION A-A

TOLERANCES: X = ± 0.5; X.X = ± 0.1; X.XX = ± 0.02	ANGULAR ±1°; SURFACE FINISH MACHINED FACES Ra 6.3	THIRD ANGLE PROJECTION	MATERIAL: ALUMINIUM	TITLE: REAR BEARING MOUNT
			ALL DIMENSIONS ARE IN MILLIMETRES	
	NAME	DATE		
DRAWN	T FEATHERSTONE	08/11/18	DO NOT SCALE DRAWING	DWG No.
CHECKED	AMAN DIDWANIA	28/11/18	A4	P6_BM_R_001
APPROVED	RICHARD VAN ARKEL	28/11/18	SCALE 2:1	

Imperial College London
Department of Mechanical Engineering

SHEET 1 OF 1 | REVISION

Measurement for critical components

Inspection report

Part number: RS002				
Description: Rear Shaft				
Dimension ID	Value and Tolerance	Measurement tool & method	Recorded measurement	Conclusion
#1	149 ± 0.5	Vernier Caliper	149.33	Within Tolerance
#2	105.5 ± 0.1	Vernier Caliper	105.88	Outside Tolerance, but not critical to performance
#3	40.5 ± 0.1	Vernier Caliper	40.50	Within Tolerance
#4	25.0 ± 0.1	Vernier Caliper	25.07	Within Tolerance
#5	25.0 ± 0.1	Vernier Caliper	25.06	Within Tolerance
#6	10 ± 0.5	Vernier Caliper	10.05	Within Tolerance
#7	10 ± 0.5	Vernier Caliper	10.08	Within Tolerance
#8	6 + 0.1	Vernier Caliper	6.09	Within Tolerance
#9	5 + 0.1	Vernier Caliper	5.03	Within Tolerance
#10	6 + 0.1	Vernier Caliper	6.02	Within Tolerance
#11	14 ± 0.5	Vernier Caliper	14.04	Within Tolerance
#12	12 – 0.009	Micrometer	11.99	Within Tolerance
#13	12 ± 0.5	Vernier Caliper	11.99	Within Tolerance
#14	10 – 0.009	Micrometer	10.00	Within Tolerance
#15	10 – 0.009	Micrometer	10.00	Within Tolerance
#16	2.68 ± 0.02	Vernier Caliper	2.68	Within Tolerance
#17	3.5 ± 0.1	Vernier Caliper	3.42	Within Tolerance
#18	4 ± 0.03	Vernier Caliper	4.02	Within Tolerance
#19	9.5 – 0.1	Vernier Caliper	9.45	Within Tolerance
#20	9.6 – 0.11	Vernier Caliper	9.56	Within Tolerance
#21	1.1 + 0.14	Vernier Caliper	1.17	Within Tolerance
#22	11.5 – 0.11	Vernier Caliper	11.45	Within Tolerance
#23	1.1 + 0.14	Vernier Caliper	1.13	Within Tolerance
#24	9.6 – 0.11	Vernier Caliper	9.56	Within Tolerance
#25	1.1 + 0.14	Vernier Caliper	1.19	Within Tolerance
#26	2.3 ± 0.1	Vernier Caliper	2.37	Within Tolerance
#27	6.0 ± 0.1	Vernier Caliper	5.96	Within Tolerance

All dimensions conform to tolerances except one. However, that dimension isn't critical to the working of the part.

Part number: BM_R_001				
Description: Rear Bearing mount				
Dimension ID	Value and Tolerance	Measurement tool & method	Recorded measurement	Conclusion
#1	10.0 ± 0.1	Vernier Caliper	10.07	Within Tolerance
#2	16 ± 0.5	Vernier Caliper	16.01	Within Tolerance
#3	8 + 0.05	Vernier Caliper	8.02	Within Tolerance
#4	22 + 0.021	Vernier Caliper	22.01	Within Tolerance
#5	35 ± 0.5	Vernier Caliper	35.12	Within Tolerance
#6	28.5 ± 0.1	Vernier Caliper	28.43	Within Tolerance
#7	3 + 0.010	Vernier Caliper	3.00	Within Tolerance
#8	M3 Clearance	Vernier Caliper	3.30	Within Tolerance

All dimensions conform to tolerance.