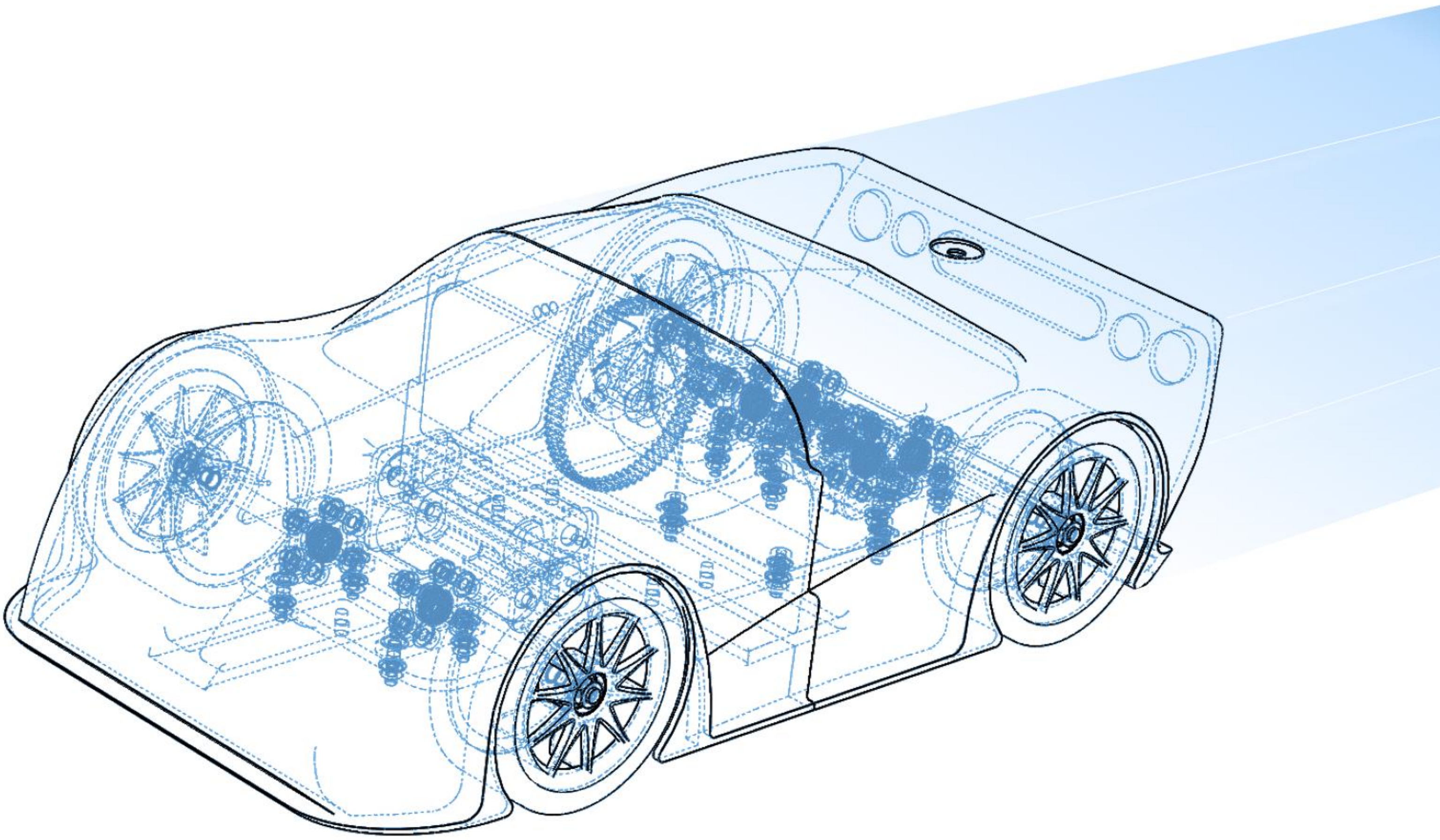
Designing

SONIC

ME2 Autumn Design & Manufacturing Project



Group 3

Brindhiini **Perera**
James **Lum**
Yilong **Huang**
Ionas **Alexopoulos**
Rohit **Nag**

Imperial College London

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Design Justification

Introduction

Given the task of designing, building and testing a miniature motorised car, the team devised a project design specification (PDS) taking into consideration the limitations, constraints and the desired objective. The main objective was to design a product which would be fast, achieved by reducing its mass. This factor along with the manufacturing constraints and possible inefficiencies were considered when making design decisions. This resulted in the choice of a rear wheel driven car with a spur gear drive transmission. In order to reduce the mass, the lengths of the three shafts were kept to a minimum and plastic gears were chosen. To reduce the number of parts that would have to be manufactured individually, 6 identical bearings were used so that the bearing housings could be designed to be almost identical and be manufactured using the CNC machine. The base plate was produced by laser cutting the acrylic material available and was the structure to which all the key components of the design would be bolted on to.

This report illustrates the decisions and considerations the team made to result in the development and final product, 'Sonic.'

Design Timeline and Conceptualization

To start turning the brief into a realistic Product Design Specifications, the team broke up the project into parts – drive transmission, car body style, bearings, amongst others. The team made clear the bounds of the project, such as track size, car size, and what materials could be used.

When sorting out each part, the team started by brainstorming ideas and then presented them in meetings and gave feedback to each other's designs to produce secondary designs. A most realistic design was then selected. Details were then considered, figuring the right balance between simplicity and performance, and most importantly its manufacturability within the workshop constraints. The team also tried to figure out possible problems arising from the design and how they might be mitigated

Product Design Specification (PDS)

Element	*Statement or criteria*	*Verification by*
Customer		
Needs	• Create a miniature motorized car using the materials provided and allowed.	• Design review.
Competition	• 31 other groups.	• Testing in the concourse at the end of the summer term.
Aesthetics	• No exposed moving parts. • Required to have a double curved geometry. • Cover to be made using additive manufacturing.	• Visual inspection.
Operation		
Performance	• Reach an approximate speed of 4m/s. • Should work as soon as the switch is turned on.	• Testing to be done once the product is made.
Environment	• Racing track in concourse with a width of 200mm.	• Testing to be done once the product is made.
Size	• 160mm width • 280mm length	• Design review. • Measurement after the product is made
Weight	• Approximately 1kg • Needs to be as light as possible.	• Approximation to be done before design review. • Measurement of complete product.
Ergonomics	• Should not require any manual force once the switch is turned on.	• Test on the concourse once the product is complete.
Life		

Product Life	• 1 year	• Will receive feedback.
Service Life	• Should operate for a minimum of 100 hours before failure	• Testing.
Maintenance	• Parts may have to be re-lubricated before the final testing.	• Testing in the concourse at the end of the summer term.
Producer		
Quantity	• 1	• Design brief.
Product Cost	• Affordable to build • Would include the cost of the components ordered from RS/HPC.	• Calculate the cost of each component (i.e. bearings and gears)
Manufacturing Constraints	• Manufacture in STW • CNC only one part • Additive layer manufacturing for the cover. • Acrylic laser cutting.	• Design review.
Regulatory		
Safety standards	• CE product safety standard. • Check the risk of moving parts. • Check for sharp edges.	• Design review. • Inspection once the product is complete.
Product Regulations	• No exposed moving parts.	• Design brief. • Design review.
End of Life disposal	• Recyclable plastic and metal parts. • Reusable/recyclable electronics.	• Design review.

Drive Options

What is the best way to power the car?

AWD

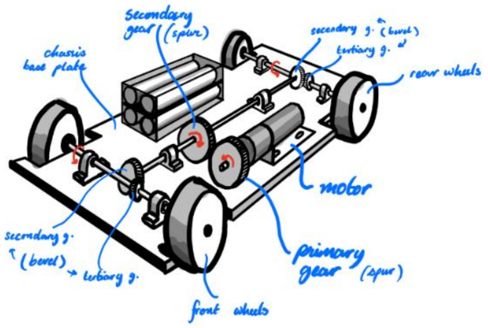
An all-wheel-drive allows for the best traction possible but requires a more complex transmission system and extra weight. Having maximum traction reduces the likelihood of the wheels slipping, which may lend greater acceleration and the ability to transmit torque to all wheels.

One possible AWD setup shown in figure 1 was considered.

Figure 1. Concept AWD setup

FWD and RWD

Front-wheel-drive (FWD) and rear-wheel-drive (RWD) are options to simplify the drive transmission system and reduce weight drastically. Although done so to save weight and simplify manufacture, it is at the expense of having less traction.



A conventional FWD setup feature the transmission and other components being placed behind the driven wheels. This means that the least amount of weight, between the 3 drives, are placed on the driven wheel. This would be further amplified during forward motion which causes more weight to transfer towards the rear. Overall, this option would provide the least traction and thus the lowest acceleration.

On the other hand, an RWD system would support more weight on driven wheels, affording more traction and therefore greater acceleration. A realistic approach of mounting the transmission components slightly in front of the wheels would be ideal to account for any weight transfer once in motion.

Furthermore, a 2-wheel drive introduced the possibility of using only 3 wheels. While helping in weight reduction, the lack of a wheel came at the expense of reduced traction and stability. Figure 2 shows a considered initial 3-wheel concept, powered by either an FWD or RWD.

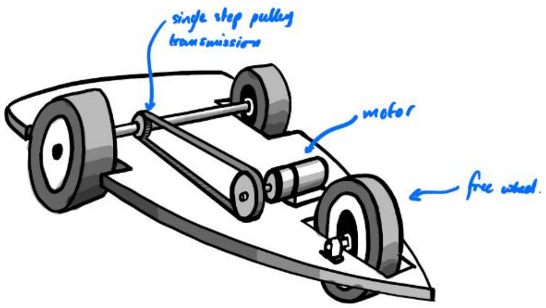


Figure 2. 3-Wheel concept

Transmission Designs

The given motor had a prebuilt 30:1 step down gearbox attached which meant that the stock motor was not enough in driving the car to the intended speeds mentioned in the PDS.

Thus, a custom transmission design was necessary to perform the speed increase function. The following ideas were considered.

Flexible Drives

One of the first ideas was a belt drive; a one-step speed increase would reduce the number of parts as well as simplify shaft designs. The motor axis would be parallel to the drive shaft. However, to achieve a high enough gear ratio, the driver pulley would have to be large and heavy. To accommodate for the size, an additional platform would be required to raise the pulley clear of the ground as shown in figure 3.

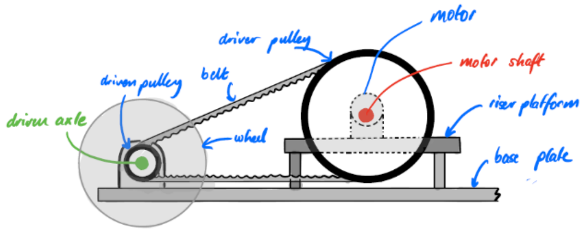


Figure 3. Single-step pulley design

Another method would be to introduce a secondary belt. This would however increase the longitudinal footprint and introduce the added complexity of designing a tensioning method for two of the three shafts, shown in red on figure 4.

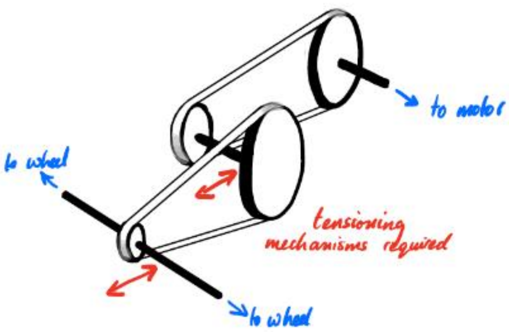


Figure 4. Double-step pulley design

Combined Drive

To overcome the issue of pulley sizes, a combined rigid and flexible drive mechanism was also considered where the motor drove a pulley increasing the speed to a pair of gears which further increased the speed to the drive shaft. However, it was soon realized that the design, as illustrated in figure 5, would share the disadvantages of both the two types of drives.

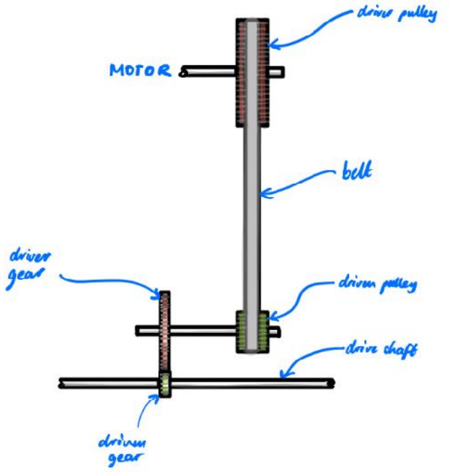


Figure 5. Combined drive design

Rigid Drive

Establishing the complications associated with having a belt drive, the decision to use gears was well justified to achieve a compact design.

Much alike the belt drive designs, one of the gears would be excessively larger than the other in a single step design. This poses another risk of excessive stress on the teeth of the smaller gear. Therefore, only double steps were considered.

Bevel Gears:

Bevel Gears offered an attractive solution to make the car as narrow as possible. This is because power could be transmitted along orthogonal shafts. However, this shaft arrangement also introduces axial forces which is undesired. Positioning is also very difficult due to small clearances between shafts as demonstrated by the concept sketch in figure 6.

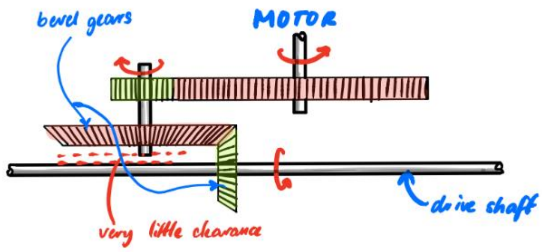


Figure 6. Bevel gears concept setup

Spur Gears:

Compared to bevel gears, spur gears are easier to assemble and only have radial forces. However, the arrangements aren't as compact. This prompted the design a stacked spur gear setup which utilizes a platform to support the raised intermediate shaft shown in figure 7.

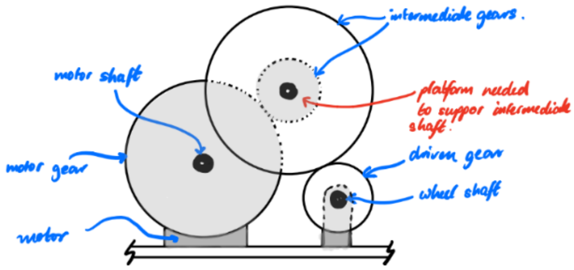


Figure 7. Stacked spur gear arrangement concept

The final chosen transmission design which considered the drawbacks of the concept designs is shown in figure 8.

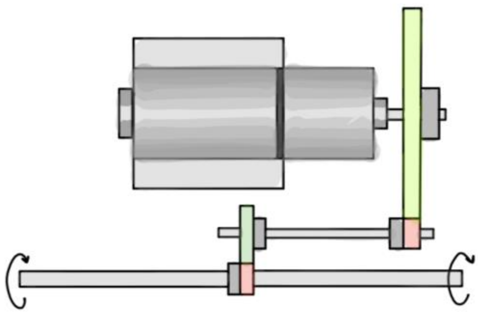


Figure 8. Final spur gear arrangement schematic

This simpler design featured coplanar shafts and a simple 2 step transmission.

Chassis and Weight Distribution

It was important that most of the weight be supported by the driving wheels as this would achieve greater traction on the driving wheels. The motor and transmission account for most of the weight that could be shifted. A low centre of gravity spread evenly throughout the body would also lead to greater stability. Should the weight be concentrated above the driving wheels or spread out evenly? How can a low centre of gravity be achieved?

Initially, a parallel plate design shown in figure 9 was considered.

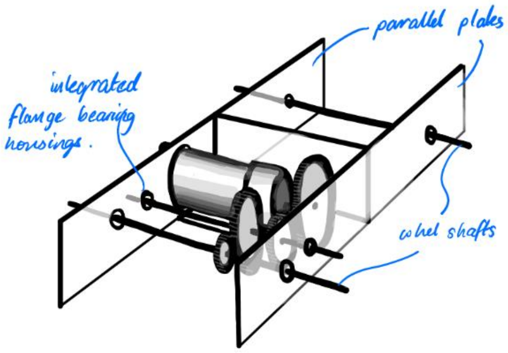


Figure 9. Parallel plates chassis design

While being relatively simple, achieving perfect parallel geometry would be difficult when manufacturing and could have catastrophic results if otherwise, such as non-meshing gears, or misaligned bearings.

After some basic calculations, equation 1 was formulated. This stated that a car with weight, (W) with lower centre of gravity (h) and a longer wheel base (L) was preferable to reduce longitudinal weight transfer (ΔW) and maintain a smoother ride when moving forward with an increasing acceleration (a)

$$\Delta W = \frac{h}{L}(Wa) \qquad (1)$$

As a result, any elevated platform designs were disregarded to have a lower C.G height. Thus, a much simpler design was formulated which relied on having a low base plate where all shafts could be mounted on using simple pillow block bearing housings as shown in figure 9. All components could be directly bolted on the base plate. This automatically raised the wheel shafts higher than the plate, thus lowering the chassis towards the ground as shown in figure 10.

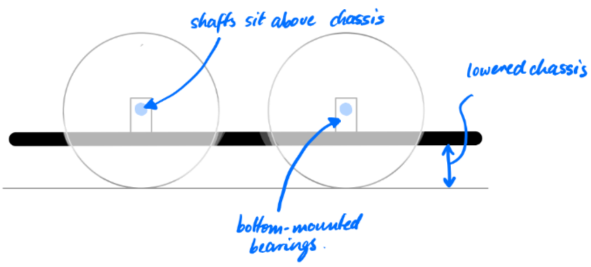


Figure 10. Side view of updated chassis design

Final Design Analysis

General Theme

The final design (figure 11) was heavily influenced by keeping the ease of manufacturing in mind. Ultimately, a more complicated design would multiply the human error associated with the physical manufacturing and assembly of the parts. Thus, the final design featured minimal number of self-made parts while maximizing the opportunity to use additive manufacturing where possible. The following sections demonstrates how this, and other important considerations were carried across the whole design.

Figure 11. Final design CAD render

Important Design Decisions

Drive

As the sole testing parameter was the time taken to complete a straight-line sprint, the team felt that the increased acceleration gained from having a greater weight concentration over the driven wheels was paramount. Hence, a *rear-wheel drive* was chosen due to the ease of placing most of the weight, the transmission components, over the driven wheels. An RWD system also eliminated the consequences of weight transfer towards the rear, a result of forward motion of the vehicle, an aspect where an FWD would particularly suffer.

Transmission

Rigid Drive:

Due to the relatively large gear ratio of approximately 10 (calculations in section 4), a rigid drive allowed to constrict the footprint of the transmission to within the edges of the chassis. Additionally, a more compact design meant that more load could be concentrated over the rear wheels.

Spur Gears:

To maintain the highest efficiency while avoiding any axial forces on the shafts, spur gears were chosen. Another advantage of using spur gears as opposed to bevel or worm, was that the motor and gear shafts could be mounted parallel to the axle (see figure 1).

Double-Speed Increaser:

It was found that an optimal gear ratio for pure acceleration bias, without considering gear efficiency would be 1:16.4, and 1:9.3 when using gears of 96% efficiency. With the later setup, the car would reach its top speed much before covering any significant distance of the concourse. Thus, a more conservative *ratio of 1:10.9* was chosen to sustain the period of acceleration for longer and a maintain a higher average speed.

A double-speed increaser setup was chosen to achieve this ratio, rather than a single stage for two main reasons:

1. Avoiding the use of gears larger than 120 teeth (minimum size to achieve ratio while keeping good practice of not having any spur gears with less than 12 teeth), in a single stage setup. Such a large gear would need a complex elevated riser to keep moving components from contacting the ground.
2. To reduce tooth wear of smaller gears associated with large gear ratios.

The solution involved having 2 steps, with each induvial step having a lower ratio to counter the above problems, as shown in figure 12.

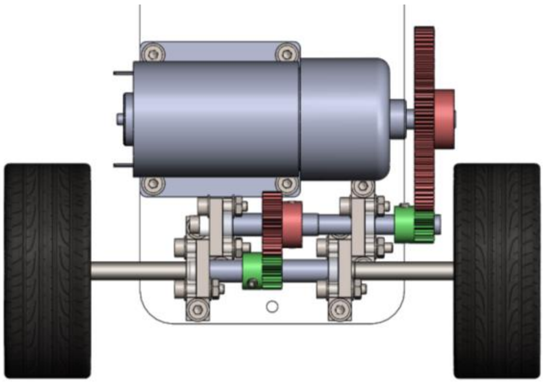


Figure 12. Transmission and Motor CAD

Delrin:

With minimal load acting on the gears itself, plastic gears were implemented due to their significant light-weight advantage and enough load ratings.

Central drive:

A key focus of the final design was trying to ensure straight motion of the car as any contact with the walls would be detrimental in maintaining speed. One method used to minimize deviation was to have the driving gears positioned exactly in the middle of the shaft (rear axle) which connected the driven wheels. This eliminated a phenomenon called “torque drive,” which is when a rolling object deviates from its intended path of motion because of the drive being misaligned to the central axis.

Wheel Fastening and Engaging Drive

Ensuring the complete transfer of torque from the shaft to the wheels by avoiding slip was a fundamental requirement for the car. This prompted the decision to manufacture bespoke hex nuts to slot into the non-standard 12mm hexagonal given wheel hubs.

Figure 13 shows the final fastening method used on both the front and rear axle.

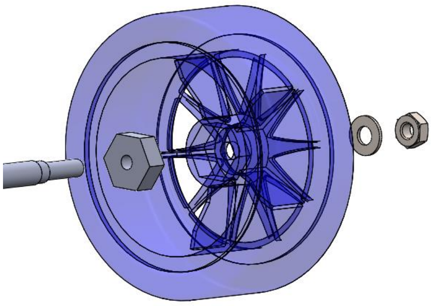


Figure 13. Wheel fastening method on threaded axles

Key Parts

Bearing Housing

A total of six bearings were mandatory for fully supporting the 3 shafts featuring in the design. Thus, six bearing housing were needed to be designed and manufactured individually. To avoid the complex manufacturing processes involved, this part was chosen to be made on the CNC machine. However, our design criteria restricted the use of CNC machines to only parts that were identical.

This posed the challenge of creating a design which could both accommodate for differences in bearing float while being identical to take advantage of additive manufacturing. Figure 14 shows the multipurpose solution.

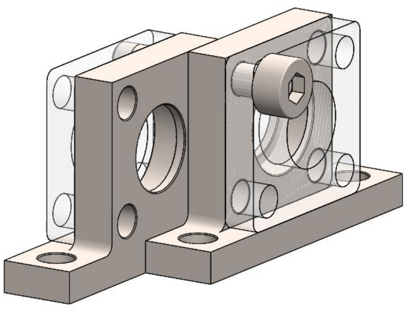


Figure 14. Combined bearing housings for rear axle and intermediate shaft. (All other components hidden for clarity)

Key bearing housing design features:

1. A singular shouldered bearing housing unit when requiring to only fix one side of the bearing outer face.
2. A modular design involving the same housing and an external acrylic back plate when requiring to fully constrain the outer face of the bearing.
3. Bolted mounts to interlink different housings to maintain parallel geometry and ease the placement onto the chassis.

Chassis Base Plate

Being the single largest part of the whole assembly, a light and rigid acrylic base plate was designed. Material choice deferred to the more conventional aluminium or steel option. However, assuming a perfectly flat track, the chassis didn't require any flexibility properties to withstand unexpected bumps. Moreover, this presented another opportunity to lower manufacturing time by using laser cutting.

The chassis dimensions were selected to give the car a long and slender profile, which would further assist in maintaining a straight path and have a smoother ride if there are bumps on the track.

Fasteners were also standardised across the part by only having M3 holes for all bolted mountings as shown in figure 15.

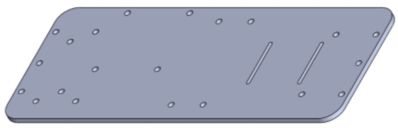


Figure 15. Final chassis base plate CAD

External Shell

The shell had to cover any moving transmission parts and have a double-curved geometry. The part was also designed to act as a shield against potential wall collisions by having front and rear protrusions. Furthermore, a small offset between the wheels and the cover eliminated the risk of the rubber tires getting caught up with the wall. In the event of a collision with the side walls, the shell would help the car slide back into a preferable orientation.

To accommodate for the long chassis, the external shell had to be printed separately in 2 parts (figure 16). The 3d printed shell also featured a small slot which provided a power switch mounting solution to be accessible from the outside.

The team had the most liberty in designing this part as it was 3d printed, thus apart from its basic functionality, the body also served an aesthetic purpose. The general shape was inspired by various classic endurance vehicles.

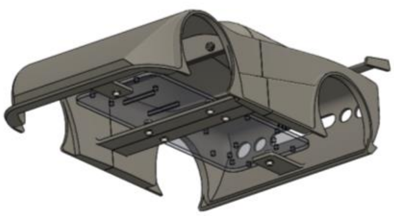


Figure 16. External shell subassembly

Performance Calculations

Determining Gear Ratio

Required Torque

First, we began by finding the maximum traction from the tyres (i.e. when starting from rest) and the corresponding torque which the motor would need to provide.

Let the mass of the car be m. The one half is introduced since there are two rear wheels and it is assumed that they each support half of the weight. A new variable χ, weight fraction, is introduced to account for the fact that the centre of gravity of the car does not coincide with the geometrical centre. It is defined as the fraction of the weight of the car which falls on the rear wheel (i.e. a car with more weight on the rear would mean $\chi > 0.5$, while a car with more weight on the front side would mean $\chi < 0.5$). Figure 17 shows a free body diagram of the wheel and the corresponding forces acting on it.

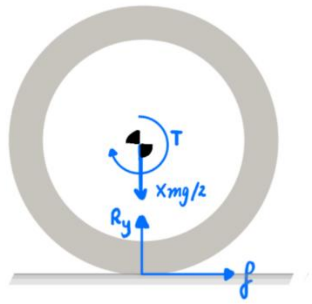


Figure 17. Forces on rear wheel

As a result, the vertical reaction force and the frictional force would be:

(2) $$R_y = -W = \frac{mg\chi}{2}$$

(3) $$f = \mu_s R_y = \frac{\mu_s mg\chi}{2}$$

With the grip each tyre can provide, the corresponding motor torque could be found by multiplying the force with the radius of the wheel r, then multiplying it by 2 since both rear wheels would be driving wheels, as shown in equation 4.

(4) $$T_g = 2fr = \mu_s mg\chi r$$

Motor Characteristics

Moving on to motor characteristics, it was given that, on the motor by itself, the stall torque $T_s = 65\,\mathrm{Nm}$ (equation 5) and the no load speed was 7500 RPM (equation 6). There was already a 30:1 reduction gear box installed on the motor. Assuming we designed the transmission with a gear ratio of 1:x, the overall gear ratio from the motor's perspective would be 30:x. From this, we could work out the new characteristics with consideration of the gears.

(5) $$T_s = 65 \times \frac{30}{x} = \frac{1.950}{x}$$

(6) $$n.l.s = 7500 \times \frac{x}{30} = 250x$$

Operating Conditions

The actual operation point of the motor and the added transmission would be on a straight line, with torque equal to T_s when speed is 0 and speed equal to n.l.s. when torque is 0. This could also be parametrized as the following, where T and ω represent the operating torque and angular velocity:

$$(7) \qquad \frac{\omega}{\text{n.l.s}} + \frac{T}{T_s} = 1$$

However, to make the model more realistic, gear inefficiencies were also considered. The gears were assumed to have an efficiency (η) of 96%. This slightly altered the above equation to equation 8. This was due to the shift of the motor curve as demonstrated in figure 18.

$$(8) \qquad \frac{\omega}{\eta \cdot \text{n.l.s}} + \frac{T}{T_s} = 1$$

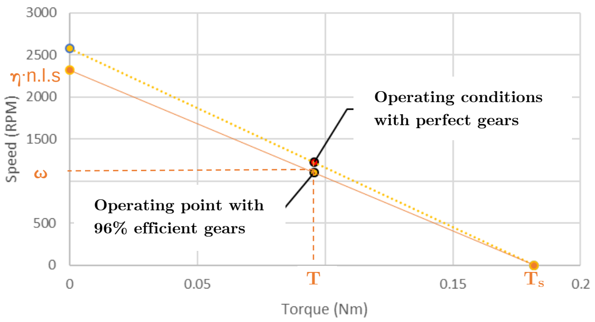


Figure 18. Axle characteristic curves, perfect vs real gears.

Gear Ratio Optimization

Substituting the expression for stall torque and no-load speed, and expressing the equation in terms of ω, equation 9 was derived.

$$(9) \qquad \omega = 240x - \frac{1600T}{13}x^2$$

Our goal was to optimize this equation to find the best gear ratio, x, that maximizes the performance in terms of speed. Therefore, we differentiate ω with respect to x and make that equal to 0.

$$(10) \qquad \frac{d\omega}{dx} = 240 - \frac{1600T}{13}x = 0 \longrightarrow x = \frac{39}{40T}$$

This equation had 3 variables present, however for our analysis T was treated as a constant. This was because for maximum initial acceleration and no slip to occur, the operating torque had to equal the tractive torque derived in equation 4. We could now select the optimal gear ratio by combining the two equations:

$$(11) \qquad x = \frac{39}{40\mu_s m g \chi r}$$

In this expression, the values of g and r are fixed. χ was estimated to be between 0.6 and 0.7, μ_s was found to be around 0.6 to 0.7 from various sources, and the mass of the car was estimated to be between 1 to 1.2 kg. This gave a range of gear ratios between 6.2 to 10.6.

Having a lower gear ratio would mean a higher output torque, hence faster acceleration but lower maximum speed, and a higher gear ratio would mean lower acceleration but higher maximum speed. Given the length of the track compared to the size of the car, we decided to go for higher maximum speed rather than acceleration. The acceleration phase only takes the first 1.5 seconds and the initial 2.8 metres even in the worst circumstance, 5% of the entire track.

Potential Failure Modes

Shaft Stress Analysis

Stress analysis on the shaft was judged to be essential, being one of the most critical components of our design. Reducing the diameter of the shaft was crucial in weight reduction, the fundamental element of a faster car. Difficulty in these calculations arose as the shaft under analysis had varying diameters and furthermore, had a hole for torque transmission from the gears.

Schematic of Rear Axle

Since this part of the analysis was conducted in an early stage of the project, a simplified schematic of the design shaft shown in figure 19, was used.

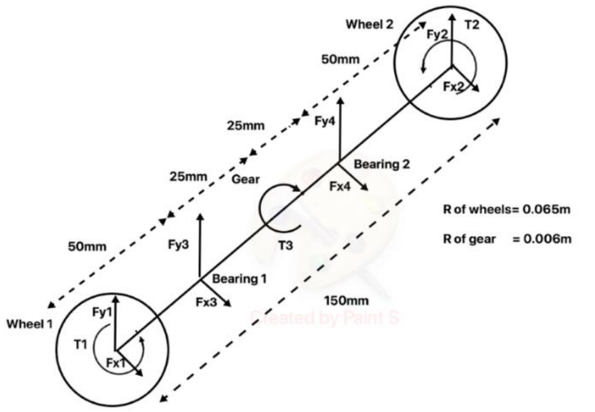


Figure 19. Rear Axle Diagram with all loads present

In order to measure the force applied by the driving gear in the horizontal axis of the shaft, the concept of a pressure angle was introduced. Using this, the force subject to the shaft was found by applying equation 12:

$$(12) \qquad F1 = \frac{Torque}{Radius} \cdot tan(\theta)$$

Where ϑ is the pressure angle. In our case a value of ϑ=20 ° was used corresponding to HPC gear catalogues. By performing a horizontal force balance, the following diagram (figure 20) was derived which could then be easily solved.

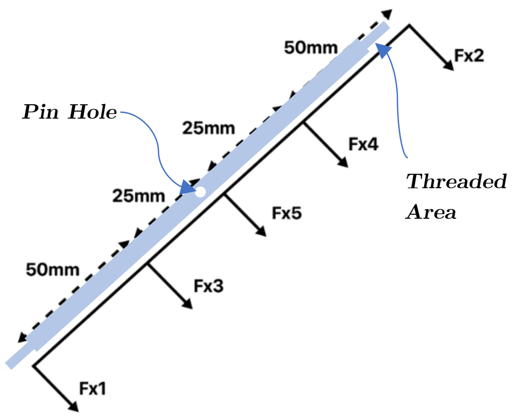


Figure 20. Rear Axle horizontal Force Balance

Table 1. Values of forces on Diagram 2

Force	Value (N)
Fx1	-2.37
Fx2	-2.37
Fx3	-2.13
Fx4	-2.13
Fx5	9

Finding Maximum Stresses

Using figure 20, the shear stress and bending moment diagrams shown in figures 21 and 22, could be drawn for the horizontal axis. For the vertical (y) axis such an analysis seemed unreasonable as forces were much smaller and would not be critical determinants during potential buckling of the shaft.

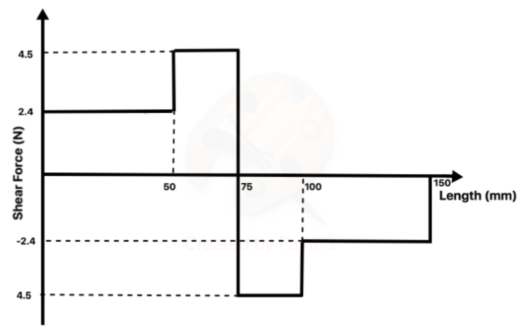


Figure 21. Shear force diagram in the horizontal plane of the rear axle shaft

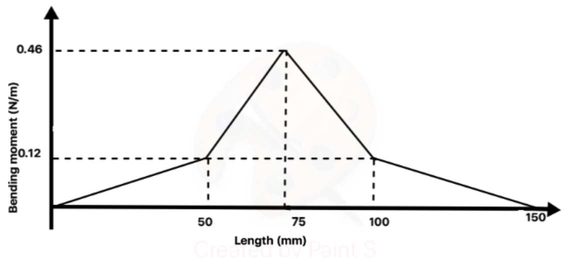


Figure 22. Bending moment diagram in horizontal plane of rear axle shaft

Having drawn the SE and BM diagrams, the moment of inertia at each cross section of the shaft had to be calculated. After that, the maximum stress the safety factor of the structure could be determined.

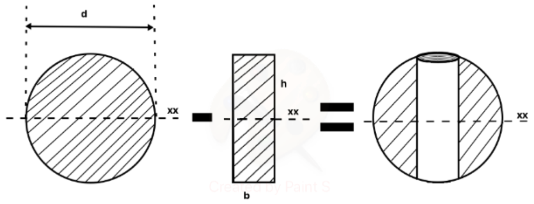


Figure 23. Second Moment of Area derivation of the cross-sectional area of the hole

From the above diagram (figure 23), the following equations could be understood:

(13) $$I_{pin\ hole} = \frac{\pi d^4}{4} - \frac{bh^3}{12} = 9.64 \times 10^{-10} m^4$$

(14) $$I_{thread} = \frac{\pi d^4}{4} = 2.01 \times 10^{-10} m^4$$

It was known that the moment of inertia would be minimum either at the location of the hole or at the part with the smallest diameter (thread). Thus, only these locations were considered in the calculations. Furthermore, by looking at the bending moment diagram it can be deduced that the bending moment was maximum at the point of the hole, hence the stress calculation at this point was deemed essential.

Using the above results, the maximum stress at the two points of interest could be calculated using equation 15 and 16, where d1 was the diameter of the shaft where the hole was located and d2 was threaded shaft diameter.

(15) $$\sigma_{max,pin\ hole} = \frac{d_1 M}{2 I_{pin\ hole}} = 1.46\ Mpa$$

(16) $$\sigma_{max,thread} = \frac{d_2 M}{2 I_{thread}} = 1.46\ Mpa$$

To further enhance the depth of the analysis, the maximum stress on the vertical axis was calculated.

(17) $$\sigma_{ymax,pin\ hole} = \frac{2 \cdot [F_{y2} \cdot x - (F_{y2} - 0.5W) \cdot (x-y)] \cdot 10^{-1}}{D^3 \cdot \pi} = 0.30 MPa$$

(18) $$W = L \cdot \pi \cdot \left(\frac{D \cdot 10^{-3}}{2}\right)^2 \cdot \rho \cdot g = \begin{cases} W_{thread} = 0.34N \\ W_{shaft} = 0.15N \end{cases}$$

(19) $$\sigma_{ymaxhole} = \frac{D \cdot [F_{y2} \cdot x - (F_{y2} - 0.5W) \cdot (x-y)] \cdot 10^{-1}}{2 \cdot I_{hole}} = 0.31 MPa$$

Where D was the diameter of the shaft, x was the distance of the gear from the left edge of the shaft (75 mm) , W was the weight of the shaft given its length (L), diameter (D) and density (ρ), y was the distance of 'bearing 1' from the left edge of the shaft (50 mm).

Having derived the maximum stresses on both vertical and horizontal axis on two points of maximum stress concentration and by assuming no shear stress, the maximum value of direct and shear stress could be computed.

(20) $$\sigma_{1,2} = \frac{1}{2} \cdot (\sigma_x + \sigma_y) + \frac{1}{2}\sqrt{(\sigma_x + \sigma_y)^2 + 4\tau_{xy}^2} = \begin{cases} \sigma_1 = \sigma_x = 1.49\ MPa \\ \sigma_2 = \sigma_y = 0.31\ MPa \end{cases}$$

(21) $$\sigma_{shear1,2} = \pm \frac{\sigma_1 - \sigma_2}{2} = \pm 0.59\ MPa$$

Failure Risk and Safety Factor

From the above results, Mohr's Circle for stresses was draw, which gave a graphical representation of the state of stress at any specific plane angle.

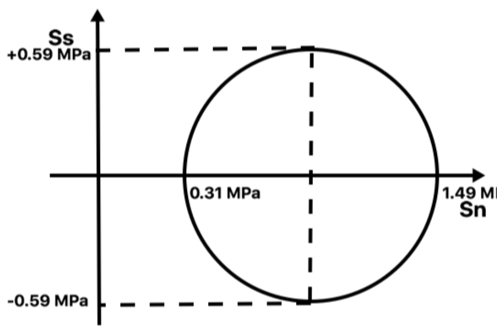


Figure 24. Mohr's circle for stresses on the rear shaft

The Von Misses equivalent stress (σ_{yield}) for the rear shaft was the final step of stress analysis carried out to give an overall safety factor of all the shafts.

$$\sigma_{yield} = \sqrt{(\sigma_1^2 + \sigma_2^2 - \sigma_1 \cdot \sigma_2)} = 1.36\, MPa \quad (22)$$

From literature, the yield stress of steel is $\sigma_{yield} = 300MPa$ and thus the safety factor of the shaft, under the specific working conditions mentioned above, could be computed.

$$Safety\ Factor = \frac{300}{1.36} \approx 220 \quad (23)$$

From the above result it is evident that our structure will be safe from failing under its regular working conditions. It is also evident from the exceptionally large safety factor that the diameter of the shaft could be reduced even further.

However, due to manufacturing restrictions in our workshop, the shafts diameter could not be further moderated.

Overall, the shaft under investigation was assumed to be the one more prone to failing, being the shaft with a hole and the smallest diameter. Thus, the other two shafts were subject to a smaller risk of failing. The maximum stress in the weaker shaft was found to be well below the tolerable amount, when referencing Von Misses criterion for failure, and the design was deemed very safe.

Base Plate Failure Analysis

Due to the brittle nature of acrylic a Stress Analysis was deemed to be essential since any fracture will result in an irreparable damage setting the project back by a lot of time.

Base plate Stress Analysis

The most simplistic approach of analysing the Base Plate was by reducing it to only one dimension, the most critical one, as performed on the shaft.

Having said that, the Second Moment of Area, Maximum Bending moment and the Safety Factor had to be calculated.

$$I_{plate} = \frac{b \cdot h^3}{12} = 1.8 \cdot 10^{-10} m^4 \quad (24)$$

The bending moment diagram of the base plate is shown in Figure 25.

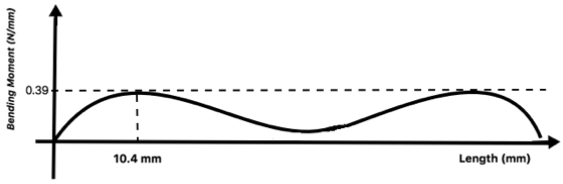


Figure 25: Bending Moment Diagram on Base Plate

$$BM_{max} = 3.98 \cdot 10^{-4} Nm^{-1} \quad (25)$$

Using the above data, the maximum stress can be calculated and thus the safety factor.

$$\sigma_{maxplate} = \frac{h \cdot BM}{2 \cdot I} = 0.0033\, MPa \quad (26)$$

$$Safety\ Factor_{plate} = \frac{10.4}{0.0033} \geq 3000 \quad (27)$$

Judging by the results, our Base Plate was marked as safe from failing.

Maximum fastening of bolts

A simple manner of failing of our plate is if the bolts are fastened too much and the compression from them exceeds the maximum compressive stress of acrylic.

$$N = \frac{E \cdot A_{bolt} \cdot Pitch}{L \cdot A_{washer} \cdot \sigma_{maxcompression}} = 30\, Turns \quad (28)$$

In order for the acrylic to fail from extreme stress due to the bolts, the bolts need to be fastened as much as 30 turns. For our design to be safe and reliable, each bolt will be fastened with 5 turns.

Technical Realization

Manufacturing Considerations

Each part had to be designed to be made in the student teaching workshop (STW). As a result, there was a constraint on how the product was made. These constraints had to be considered in order to be able to manufacture each part.

Shafts

All three shafts had to be produced by turning on a lathe. Therefore, the material chosen for this component was steel to prevent deformation. Steps were added for ease of assembly. Due to small diameters, the shafts had to be supported on both ends during the turning process.

Bearing constraints and base plate

These components were produced by laser cutting the acrylic material made available. The acrylic plates for the bearing housings were designed to constrain the bearing. Therefore, they too were identical. The base plate was made of acrylic so that it would not have to be fabricated from aluminium, which is a tedious and time-consuming process. This material was chosen as it was a light and effective choice.

Hex-nuts

As the hex hubs in the provided wheels were not to exact dimensions, the manufactured custom hex nuts had to be tolerance to a smaller size.

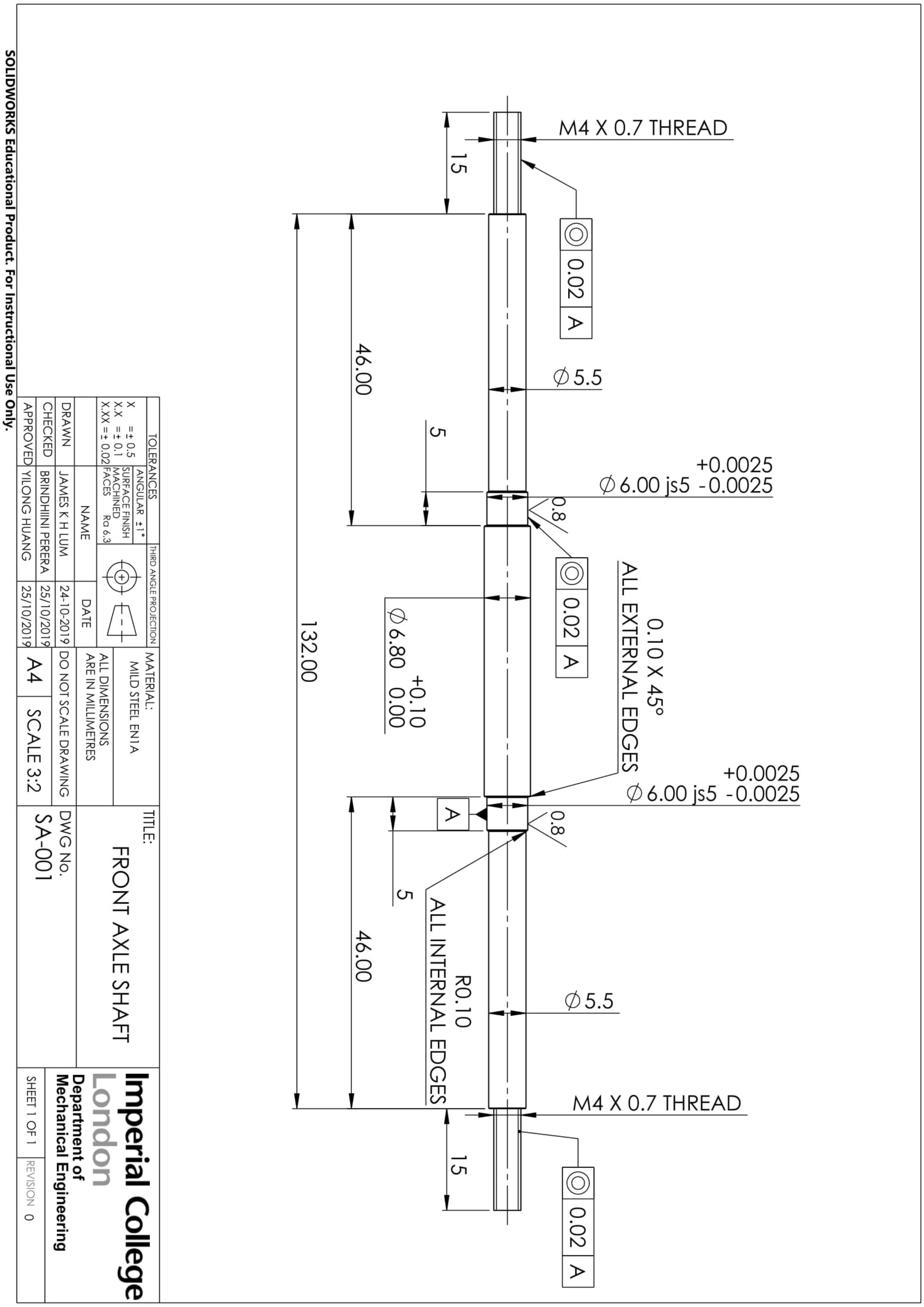
Cover

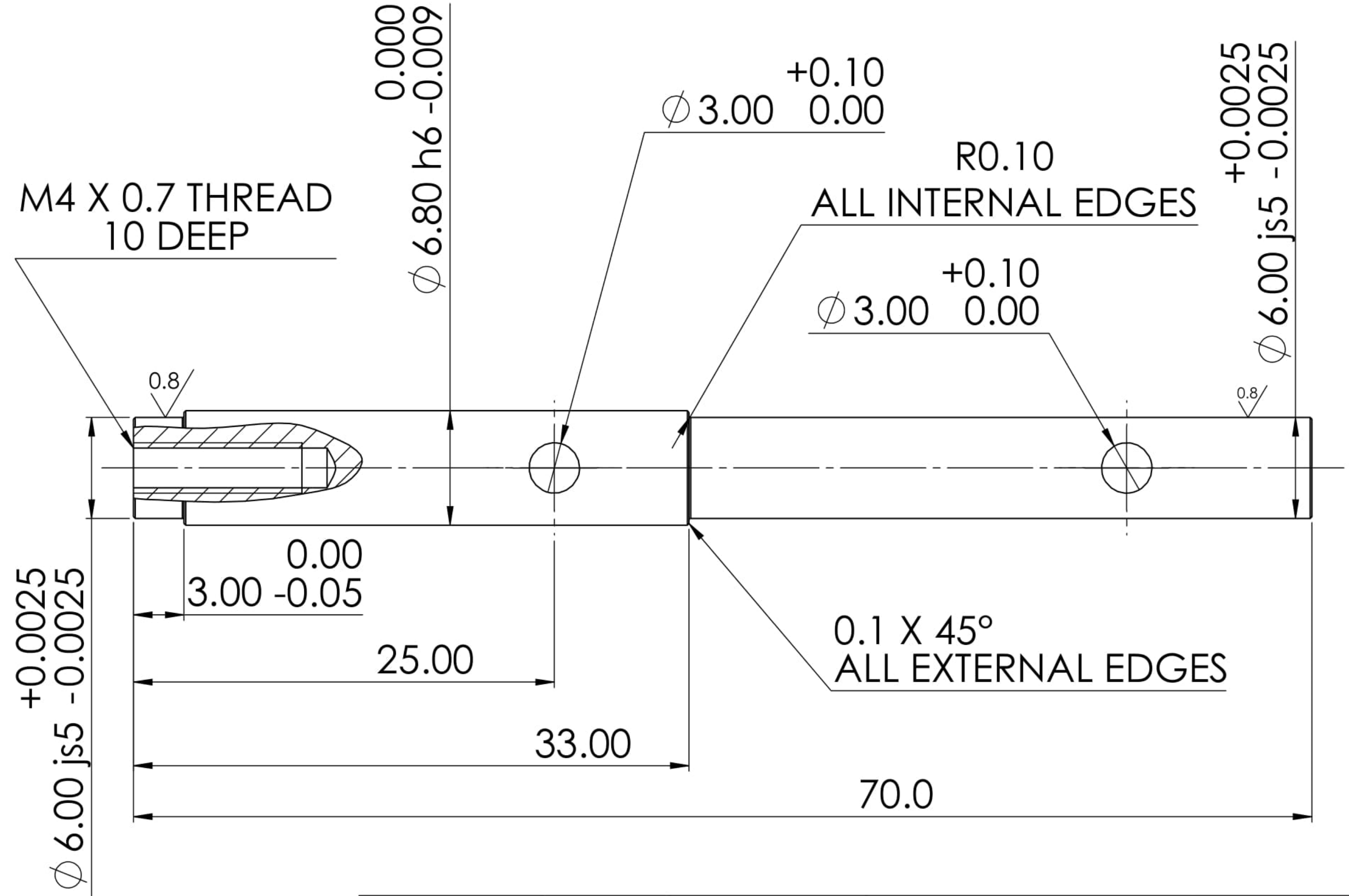
The shell had to be printed in 2 sections as the dimensions exceeded that of the printer. Due to certain geometric features on part had to be printed vertically while the other was printed horizontally. This was to prevent the failure of support structured when 3d printing.

Gears

Although the gears were sourced from HPC, they had to be altered for the design. These alterations included milling holes through the gears or the hubs to fix them onto the shafts using spring pins and increasing the bore diameter using the lathe. Special care had to be taken to ensure the plastic gears did not deform under excessive load when inserting spring pins.

Engineering Drawings





TOLERANCES		THIRD ANGLE PROJECTION	MATERIAL: MILD STEEL EN1A	TITLE: INTERMEDIATE SHAFT	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		
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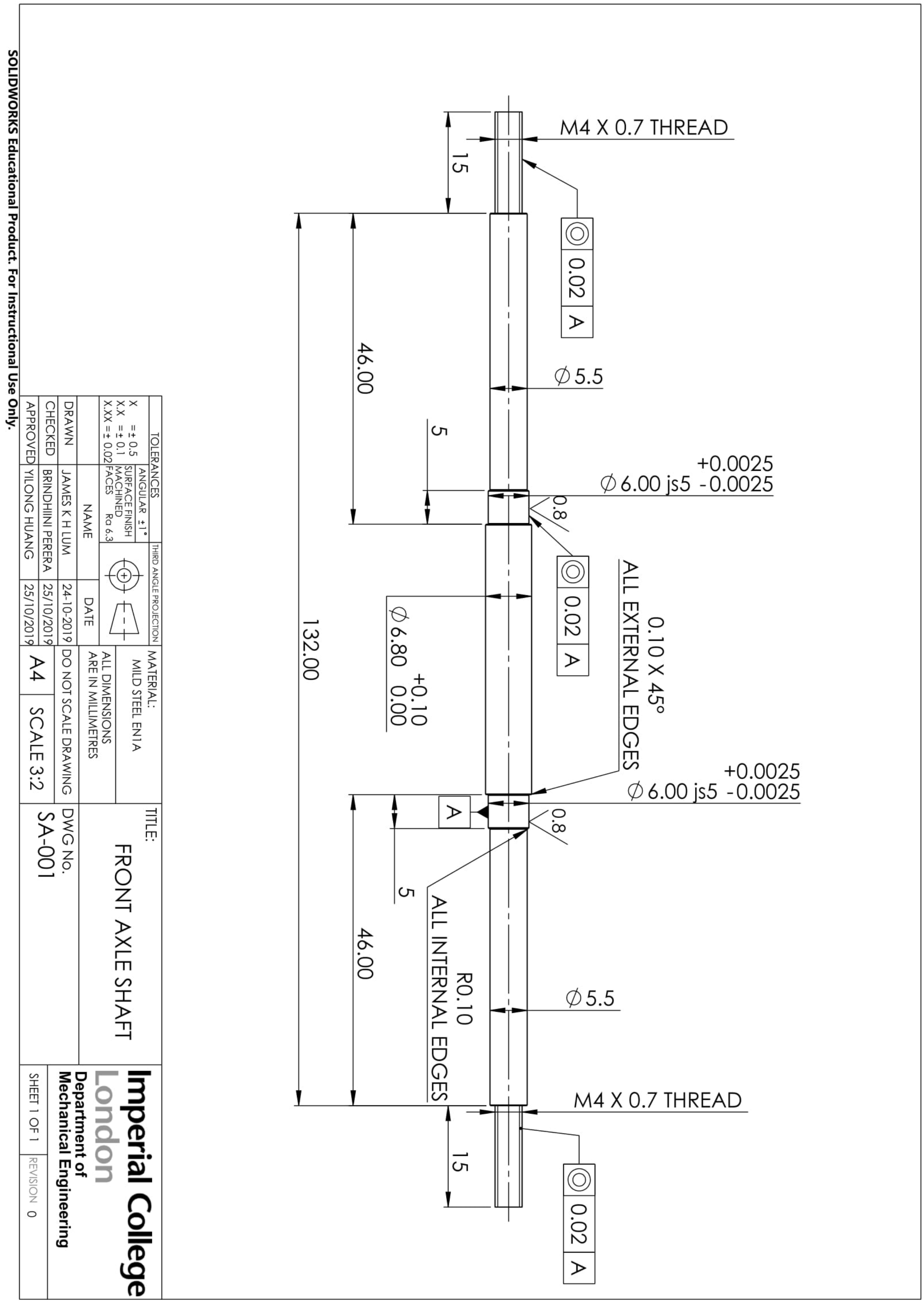
APPROVED	ION ALEXOPOULOS	25/10/2019			REVISION 0


SOLIDWORKS Educational Product. For Instructional Use Only.
M4 X 0.7 THREAD
15
0.02 A
46.00
Ø 5.5
5
Ø 6.00 js5 +0.0025 -0.0025
0.8
132.00
Ø 6.80 +0.10 0.00
0.10 X 45°
ALL EXTERNAL EDGES
A
R0.10
ALL INTERNAL EDGES
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
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CHECKED BRINDHIINI PERERA 25/10/2019
APPROVED YILONG HUANG 25/10/2019
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DO NOT SCALE DRAWING
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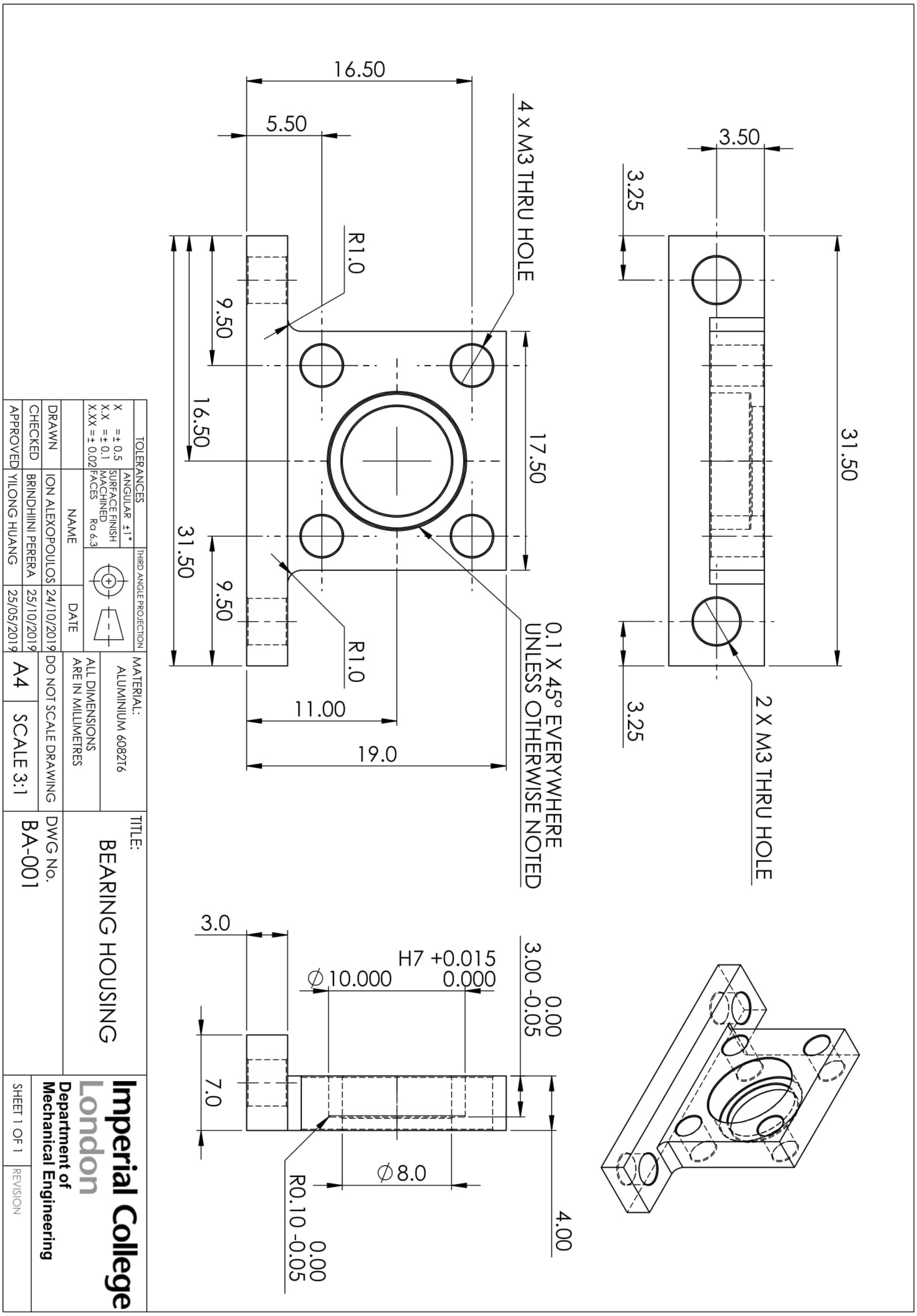
TITLE: FRONT AXLE SHAFT
DWG No. SA-001
Imperial College London
Department of Mechanical Engineering
SHEET 1 OF 1
REVISION 0


16.50
5.50
4 x M3 THRU HOLE
R1.0
9.50
16.50
31.50
17.50
9.50
R1.0
0.1 X 45° EVERYWHERE
UNLESS OTHERWISE NOTED
11.00
19.0
3.50
3.25
31.50
3.25
2 X M3 THRU HOLE
3.0
H7 +0.015
∅10.000 0.000
3.00 -0.05 0.00
7.0
∅8.0
4.00
R0.10 -0.05 0.00
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 ION ALEXOPOULOS 24/10/2019
CHECKED BRINDHINI PERERA 25/10/2019
APPROVED YILONG HUANG 25/05/2019
MATERIAL:
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ALL DIMENSIONS
ARE IN MILLIMETRES
DO NOT SCALE DRAWING
A4
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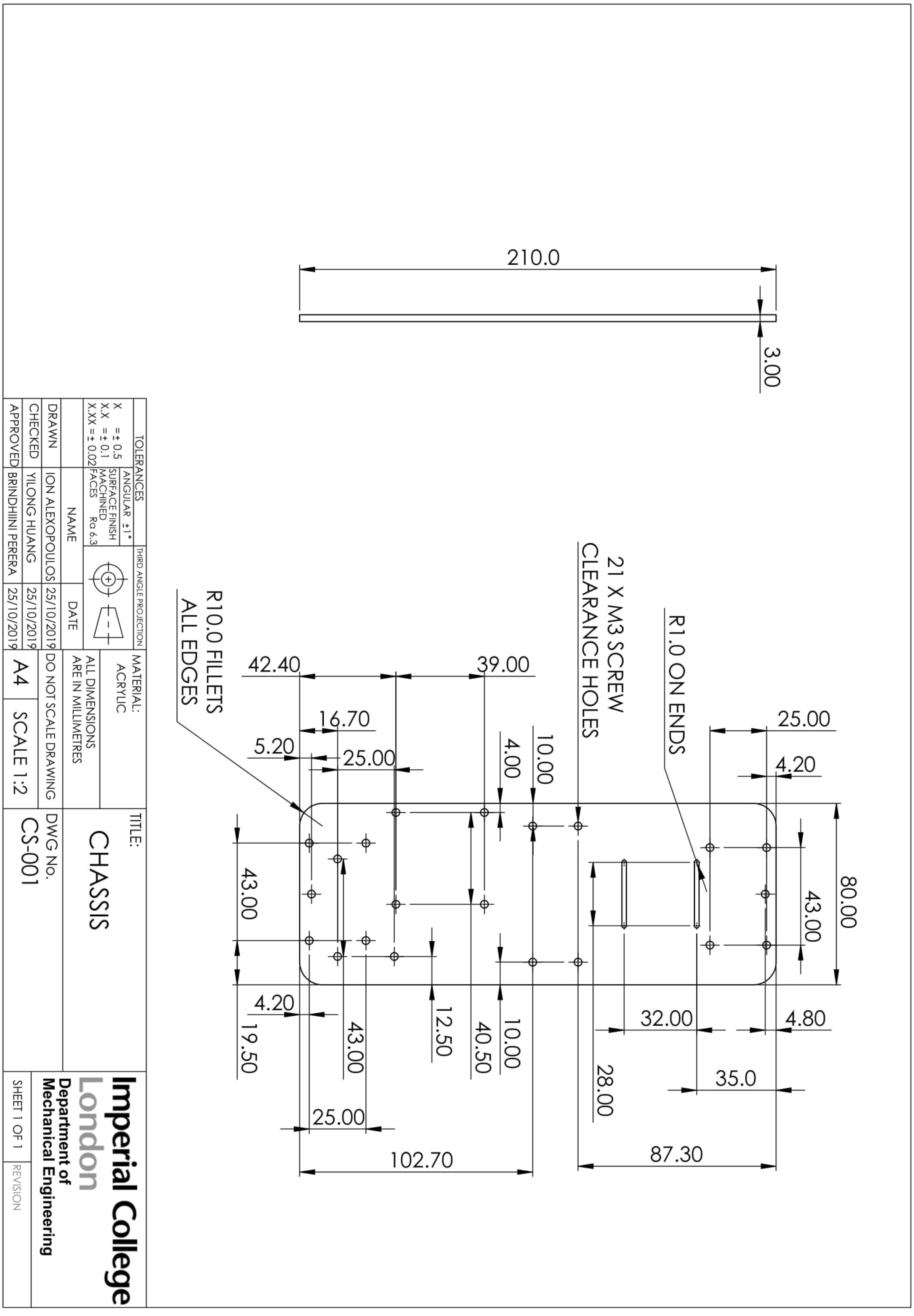
TITLE:
BEARING HOUSING
DWG No.
BA-001
Imperial College
London
Department of
Mechanical Engineering
SHEET 1 OF 1
REVISION



210.0
3.00
R10.0 FILLETS
ALL EDGES
21 X M3 SCREW
CLEARANCE HOLES
R1.0 ON ENDS
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39.00
16.70
5.20
25.00
4.00
10.00
25.00
4.20
43.00
80.00
43.00
4.20
19.50
43.00
12.50
40.50
10.00
28.00
32.00
4.80
35.0
25.00
102.70
87.30
TOLERANCES
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X.X =± 0.1
X.XX =± 0.02
ANGULAR ±1°
SURFACE FINISH
MACHINED
FACES Ra 6.3
THIRD ANGLE PROJECTION
NAME
DATE
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CHECKED YILONG HUANG 25/10/2019
APPROVED BRINDHIINI PERERA 25/10/2019
MATERIAL:
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ALL DIMENSIONS
ARE IN MILLIMETRES
DO NOT SCALE DRAWING
A4
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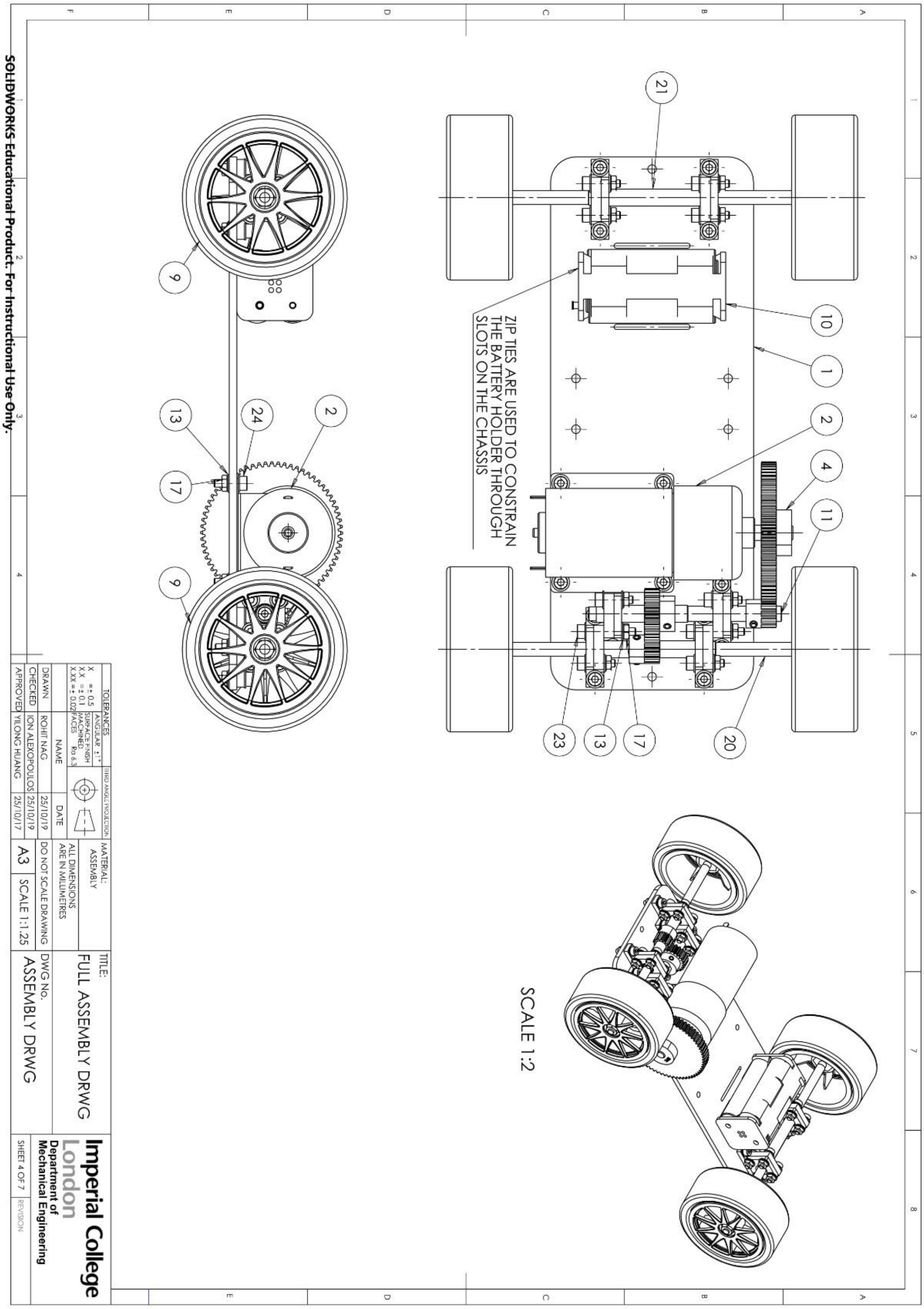
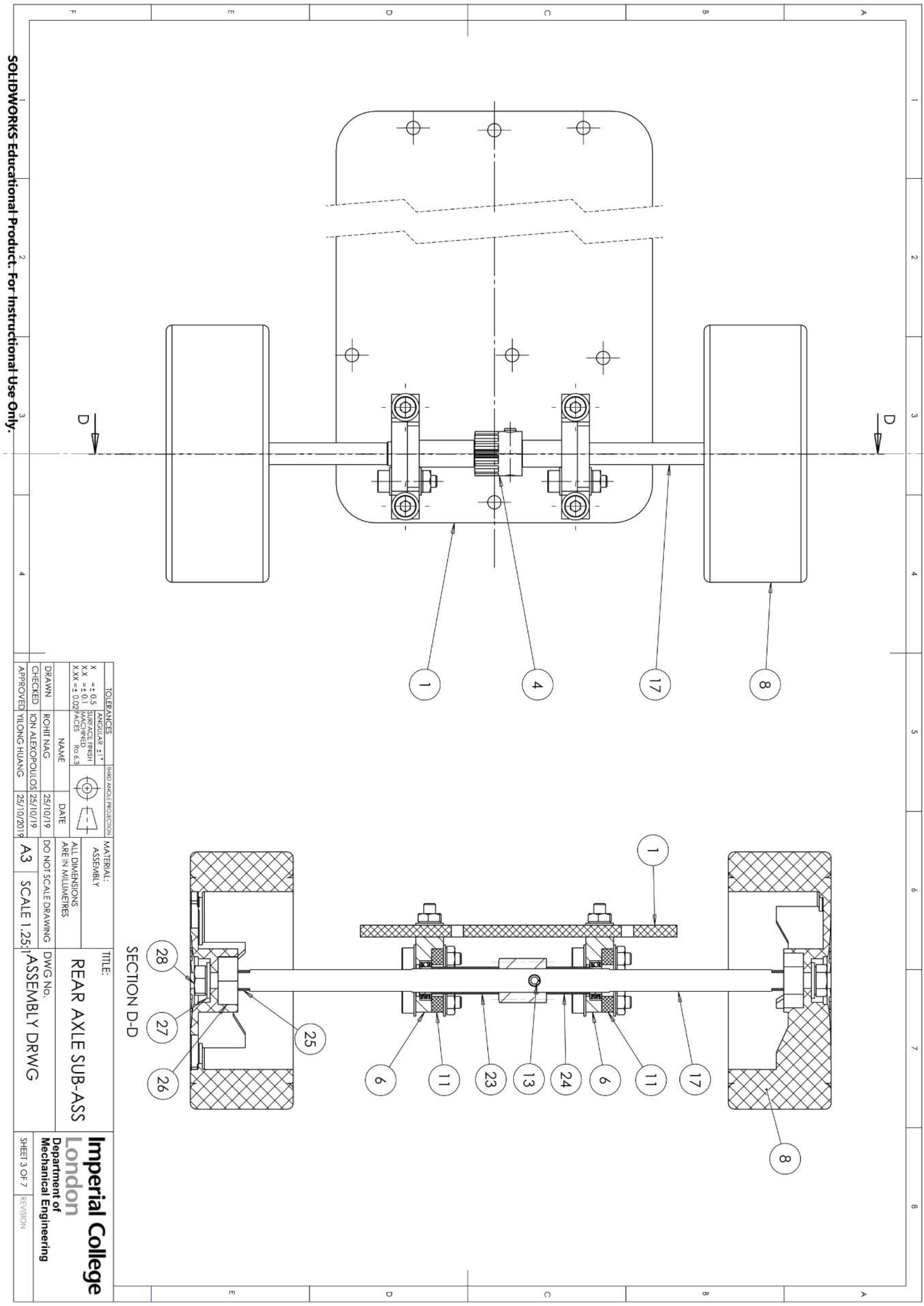
TITLE:
CHASSIS
DWG No.
CS-001
Imperial College
London
Department of
Mechanical Engineering
SHEET 1 OF 1
REVISION

General Assembly and Sub-Assemblies




SOLIDWORKS Educational Product. For Instructional Use Only.
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ANGULAR ±1°
SURFACE FINISH MACHINED FACES Ra 6.3
THIRD ANGLE PROJECTION
NAME
DATE
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CHECKED ION ALEXOPOULOS 25/10/19
APPROVED YILONG HUANG 25/10/2019
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ALL DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING
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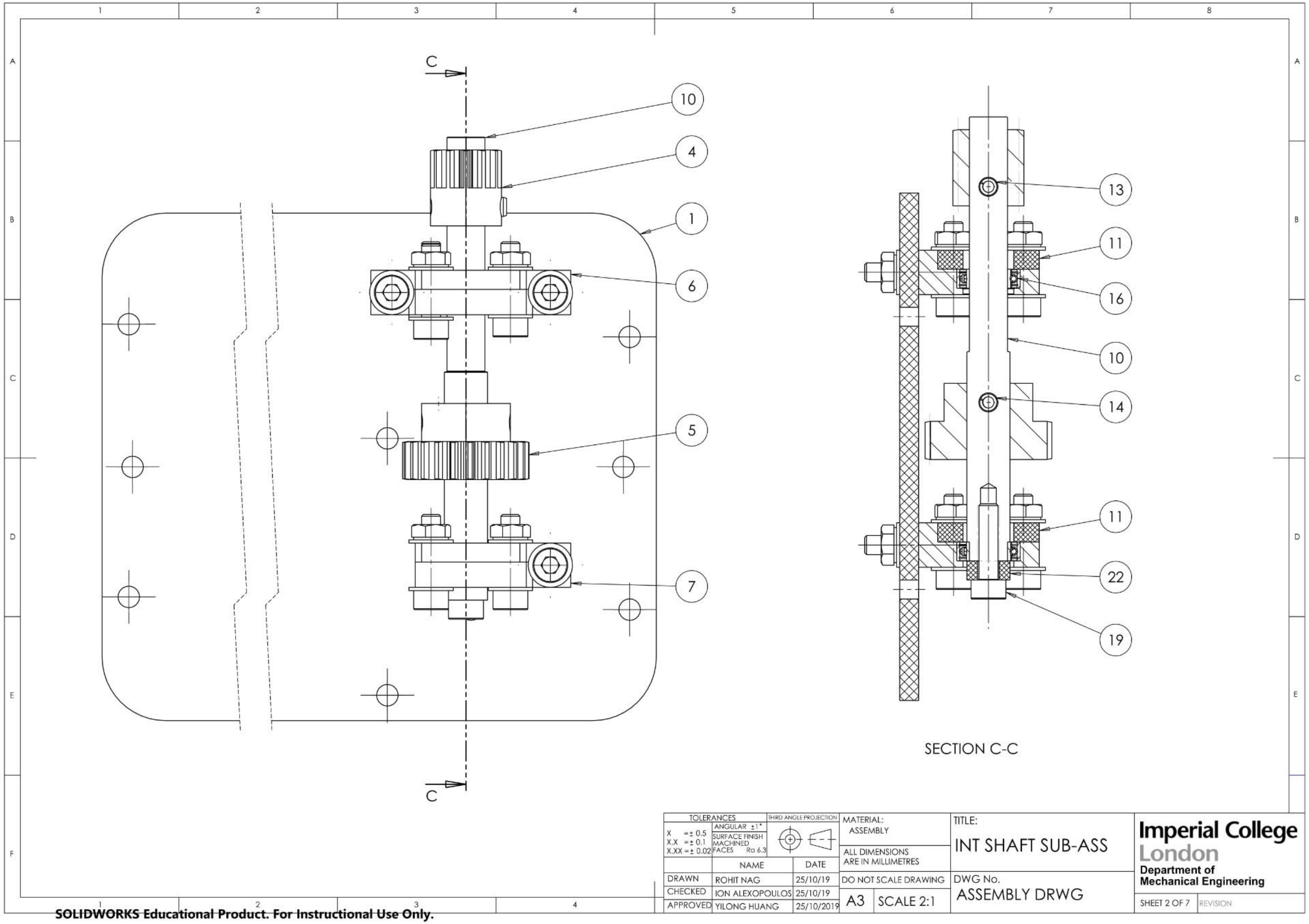
DWG No.
ASSEMBLY DRWG
Imperial College London
Department of Mechanical Engineering
SHEET 3 OF 7
REVISION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19
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X.X =± 0.1
X.XX =± 0.02
ANGULAR ±1°
SURFACE FINISH MACHINED FACES Ra 6.3
THIRD ANGLE PROJECTION
NAME
DATE
DRAWN ROHIT NAG 25/10/19
CHECKED ION ALEXOPOULOS 25/10/19
APPROVED YILONG HUANG 25/10/2019
MATERIAL: ASSEMBLY
ALL DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING
A3
SCALE 2:1
TITLE:
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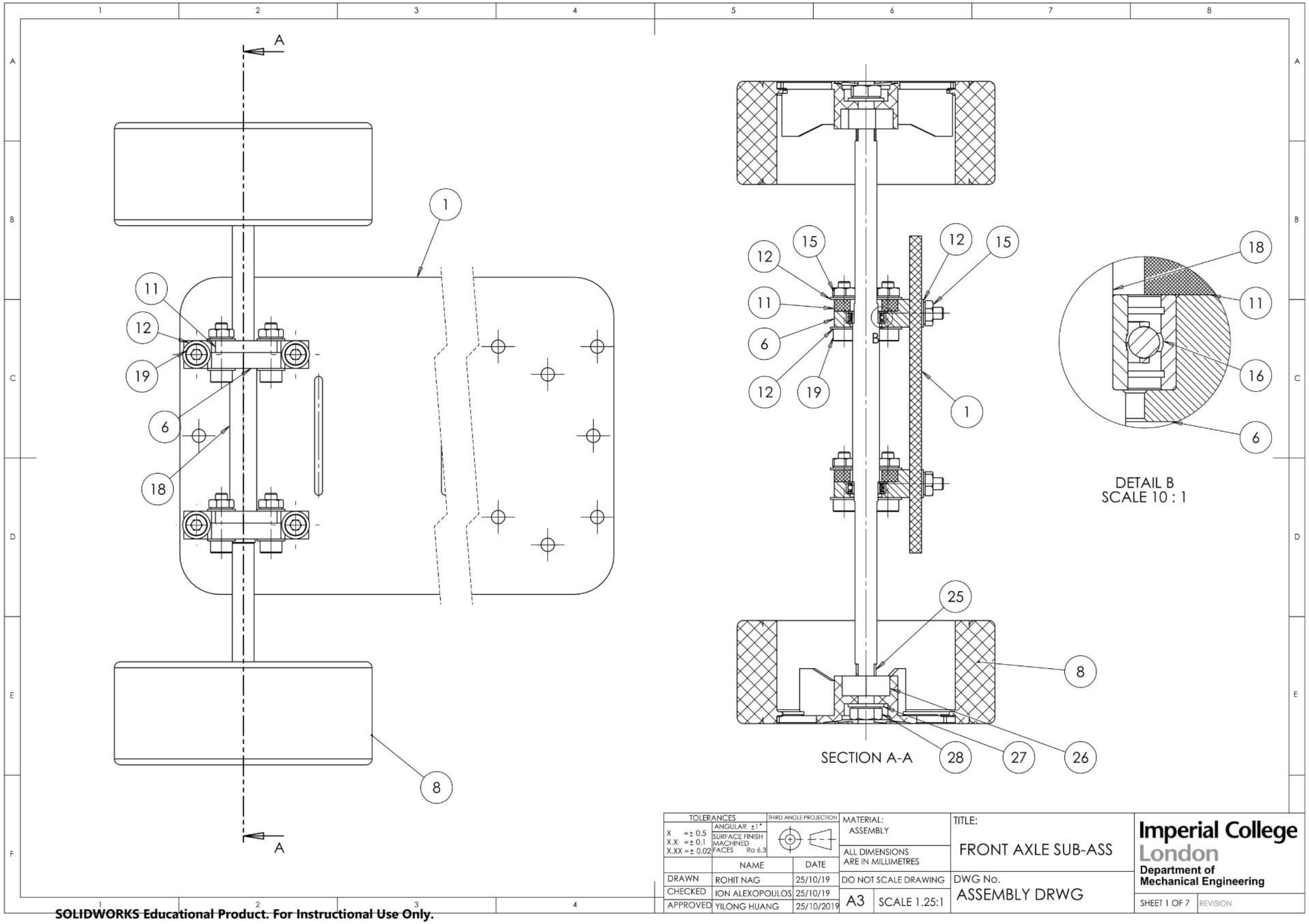
DWG No.
ASSEMBLY DRWG
Imperial College London
Department of Mechanical Engineering
SHEET 2 OF 7
REVISION
SOLIDWORKS Educational Product. For Instructional Use Only.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X.X = ± 0.1
X.XX = ± 0.02
ANGULAR ±1°
SURFACE FINISH MACHINED FACES Ra 6.3
THIRD ANGLE PROJECTION
MATERIAL:
ASSEMBLY
ALL DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING
TITLE:
FRONT AXLE SUB-ASS
DWG No.
ASSEMBLY DRWG
NAME
DATE
DRAWN ROHIT NAG 25/10/19
CHECKED ION ALEXOPOULOS 25/10/19
APPROVED YILONG HUANG 25/10/2019
A3
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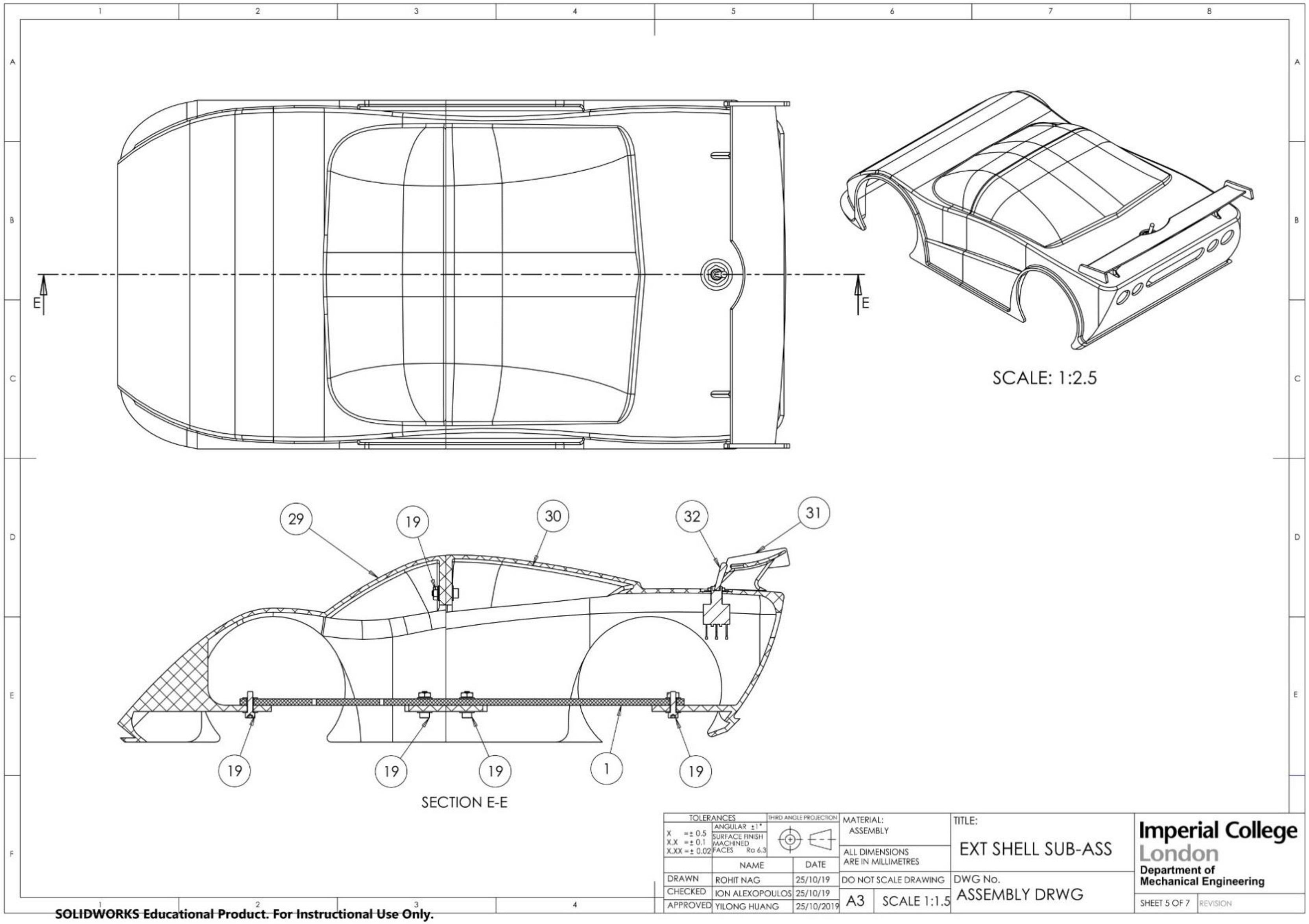
Imperial College London
Department of Mechanical Engineering
SHEET 1 OF 7
REVISION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X.XX =± 0.02
ANGULAR ±1°
SURFACE FINISH MACHINED FACES Ra 6.3
THIRD ANGLE PROJECTION
MATERIAL:
ASSEMBLY
ALL DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING
NAME
DATE
DRAWN ROHIT NAG 25/10/19
CHECKED ION ALEXOPOULOS 25/10/19
APPROVED YILONG HUANG 25/10/2019
A3
SCALE 1:1.5
TITLE:
EXT SHELL SUB-ASS
DWG No.
ASSEMBLY DRWG
Imperial College London
Department of Mechanical Engineering
SHEET 5 OF 7
REVISION
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ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	CS-001	CHASSIS BASE PLATE	1
2	M-001	MOTOR	1
3	HPC8-68	0.8 MOD 68 TEETH GEAR	1
4	HPC8-12	0.8 MOD 12 TEETH GEAR	2
5	HPC8-23	0.8 MOD 23 TEETH GEAR	1
6	BA-001	BEARING HOUSING	5
7	BA-001S	FLANGELESS BEARING HOUSING	1
8	WH001	WHEEL	4
9	B001	BATTERY ASSEMBLY	1
10	SH-002	INTERMEDIATE SHAFT	1
11	BA-002	ACRYLIC SHOULDER PLATE	6
12	WM3	M3 WASHER	87
13	SP3M12	M3 SPRING PIN x12 LENGTH	2
14	SP3M14	M3 SPRING PIN x14 LENGTH	1
15	HNM3	M3 HEX NUT	43
16	B6D3W10D	BEARING	6
17	SH-003	REAR AXLE SHAFT	1
18	SH-001	FRONT AXLE SHAFT	1
19	SHM3X12	M3 BOLTS x 12 LENGTH	40
20	SHM3X20	M3 BOLTS x 20 LENGTH	4
21	GS10M3	GRUB SCREW	1
22	SA-004	END CAP	1
23	SP11	11MM SPACER	1
24	SP17	17MM SPACER	1
25	SP03	WHEEL SPACER	4
26	SA-005	WHEEL HEX NUT	4
27	WM4	M4 WASHER	4
28	HNM4	M4 HEX NUT	4
29	SH-001	SHELL PART 1	1
30	SH-002	SHELL PART 2	1
31	SH-003	SPOILER	1
32	B002	SWITCH	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

	NAME	DATE
DRAWN	ROHIT NAG	25/10/19
CHECKED	ION ALEXOPOULOS	25/10/2019
APPROVED	YILONG HUANG	25/10/2019

MATERIAL: YILONG HUANG
ALL DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING
A3 SCALE 1:5

TITLE: BILL OF MATERIALS
DWG No. ASSEMBLY DRWG

Imperial College London
Department of Mechanical Engineering
SHEET 6 OF 7 REVISION

Extended Bill of Materials

ITEM NO.	PART NUMBER	DESCRIPTION	MATERIAL	QTY.	ADDITIONAL NOTES
1	CS-001	CHASSIS BASE PLATE	ACRYLIC 3MM	1	LASER CUTTED PART
2	M-001	MOTOR	ASSEMBLY	1	GIVEN PART
3	HPC8-68	0.8 MOD 68 TEETH GEAR	DELRIN	1	HPC ORDER 4115787
4	HPC8-12	0.8 MOD 12 TEETH GEAR	DELRIN	2	HPC ORDER 4115787
5	HPC8-23	0.8 MOD 23 TEETH GEAR	DELRIN	1	HPC ORDER 4115787
6	BA-001	BEARING HOUSING	MILD STEEL EN1A	5	CNC MILLED PART
7	BA-001S	FLANGELESS BEARING HOUSING	MILD STEEL EN1A	1	MANUALLY ALTERED ON TOP OF CNC PART
8	WH001	WHEEL	N/A	4	GIVEN PART
9	B001	BATTERY ASSEMBLY	ASSEMBLY	1	GIVEN PARTS
10	SH-002	INTERMEDIATE SHAFT	MILD STEEL EN1A	1	
11	BA-002	ACRYLIC SHOULDER PLATE	ACRYLIC 3MM	6	LASER CUTTED PART
12	WM3	M3 WASHER	STEEL	87	
13	SP3M12	M3 SPRING PIN x12 LENGTH	STEEL	2	CUT TO CUSTOM LENGTH
14	SP3M14	M3 SPRING PIN x14 LENGTH	STEEL	1	CUT TO CUSTOM LENGTH
15	HNM3	M3 HEX NUT	STEEL	43	
16	B6D3W10D	BEARING	N/A	6	RS ORDER 152934613
17	SH-003	REAR AXLE SHAFT	MILD STEEL EN1A	1	
18	SH-001	FRONT AXLE SHAFT	MILD STEEL EN1A	1	
19	SHM3X12	M3 BOLTS x 12 LENGTH	STEEL	40	
20	SHM3X20	M3 BOLTS x 20 LENGTH	STEEL	4	
21	GS10M3	GRUB SCREW	STEEL	1	
22	SA-004	END CAP	ACRYLIC 3MM	1	LASER CUTTED PART
23	SP11	11MM SPACER	MILD STEEL EN1A	1	
24	SP17	17MM SPACER	MILD STEEL EN1A	1	
25	SP03	WHEEL SPACER	ACRYLIC 3MM	4	LASER CUTTED PART
26	SA-005	WHEEL HEX NUT	MILD STEEL EN1A	4	MANUALLY MILLED FROM SMALL BAR
27	WM4	M4 WASHER	STEEL	4	
28	HNM4	M4 HEX NUT	STEEL	4	
29	SH-001	SHELL PART 1	ABS PLASTIC	1	3D PRINTED PART
30	SH-002	SHELL PART 2	ABS PLASTIC	1	3D PRINTED PART
31	SH-003	SPOILER	ABS PLASTIC	1	3D PRINTED PART
32	B002	SWITCH	N/A	1	GIVEN PART

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

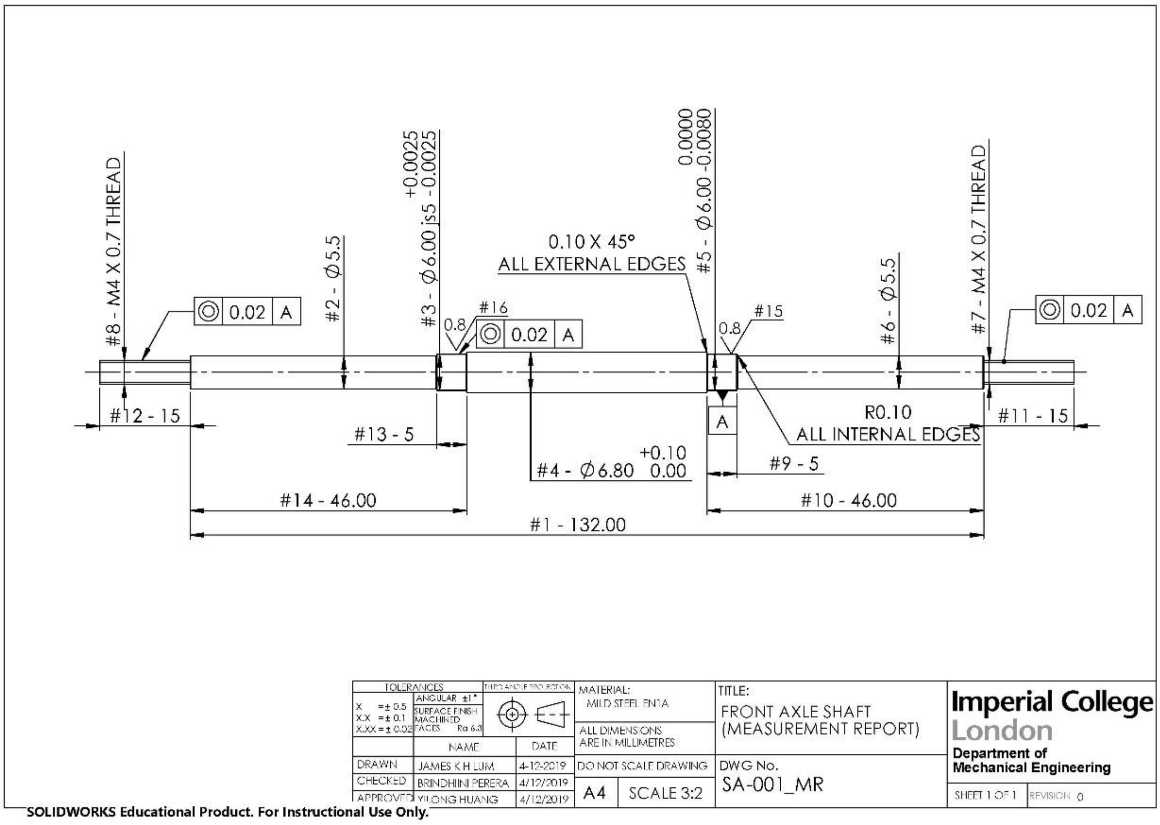
	NAME	DATE
DRAWN	ROHIT NAG	25/10/19
CHECKED	ION ALEXOPOULOS	25/10/2019
APPROVED	YILONG HUANG	25/10/2019

MATERIAL: YILONG HUANG
ALL DIMENSIONS ARE IN MILLIMETRES
DO NOT SCALE DRAWING
A3 SCALE 1:5

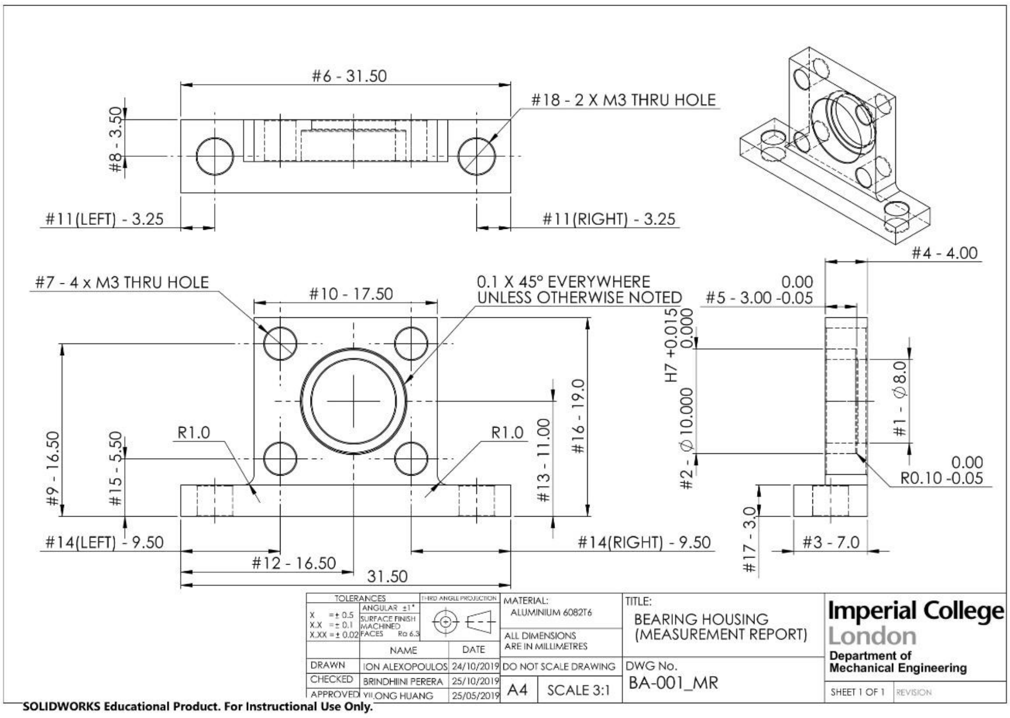
TITLE: LONG FORM BOM
DWG No. ASSEMBLY DRWG

Imperial College London
Department of Mechanical Engineering
SHEET 7 OF 7 REVISION

Measurement Reports



Measurement report						
Part:	FRONT AXLE SHAFT					
Part number:	SA-001					
Measurement	Dimension(mm)			Tool & Method, Average of 3 measurements	Recorded Measurement	Conclusion
	Nominal	Min	Max			
1	132	-0.02	+0.02	Measuring tape	132	No significant variation detectable.
2	5.50	-0.10	+0.10	MSG	5.56	Within tolerance
3	6.00 js5	-0.025	+0.025	MSG	5.95	Small variation outside of tolerance, bearing may fit loosely.
4	6.80	+0.00	+0.10	MSG	6.95	Small variation of a non-critical dimension.
5	6.00	-0.008	+0.000	MSG	5.980	Small variation outside of tolerance, bearing may fit loosely.
6	5.50	-0.10	+0.10	MSG	5.52	Within tolerance.
7	M4 X 0.7			MSG	3.89	Part fits well.
8	M4 X 0.7			MSG	3.98	Part fits well.
9	5	-0.5	+0.5	Vernier calliper	5.02	Within tolerance.
10	46.00	-0.02	+0.02	Vernier calliper	45.96	Small variation a non-critical dimension.
11	15	-0.5	+0.5	Vernier calliper	15.49	Within tolerance.
12	15	-0.5	+0.5	Vernier calliper	15.72	Small variation of a non-critical dimension.
13	5	-0.5	+0.5	Vernier calliper	4.90	Within tolerance
14	46.00	-0.02	+0.02	Vernier calliper	45.54	Significant variation for a non-critical dimension.
15	0.8Ra			Surface finish chart	3.2Ra	Significant variation, bearing may have trouble fitting.
16	0.8Ra			Surface finish chart	3.2Ra	Significant variation, bearing may have trouble fitting.



Measurement Report							
Part:	**BEARING HOUSING**						
Description:	**CNC BEARING HOUSING WITH FLANGES**						
Part number:	**BA-001**						
Measurement	**Dimension(mm)**				**Recorded Measurement**		**Conclusion**
	Nom-inal	**Min**	**Max**		**LEFT**	**RIGHT**	
1	**8.0**	**-0.1**	**+0.1**		**7.99**		**Within tolerance, should not be any problems.**
2	**10.00**	**+0.000**	**+0.015**		**10.02**		**Exceeded tolerance by a small amount, bearing may be slightly loose.**
3	**7.0**	**-0.1**	**+0.1**		**7.12**		**Small variation of a non-critical dimension.**
4	**4.00**	**-0.02**	**+0.02**		**4.08**		**Small variation of a non-critical dimension.**
5	**3.00**	**-0.05**	**+0.00**		**3.05**		**Small variation outside tolerance. Bearing might be loose.**
6	**31.50**	**-0.02**	**+0.02**		**31.52**		**Within tolerance. Should not be any problems.**
7	**3.00**	**+0.006**	**+0.020**		**3.19**		**Large variation from tolerance, should not be any problem for through hole part.**
8	**3.50**	**-0.02**	**+0.02**		**3.43**		**Large variation from tolerance, could displace bearing position slightly.**
9	**16.50**	**-0.020**	**+0.020**		**16.51**		**Within tolerance, should not have any problems.**
10	**17.50**	**-0.02**	**+0.02**		**17.53**		**Slightly outside of tolerance for a non-critical dimension.**
11	**3.25**	**-0.02**	**+0.02**		**3.30**	**3.05**	**Significant variation from tolerance of a non-critical part.**
12	**16.50**	**-0.02**	**+0.02**		**16.02**	**17.52**	**Significant variation from tolerance of a non-critical part.**
13	**11.00**	**-0.02**	**+0.02**		**11.04**		**Small variation outside of tolerance. Could angle shaft slightly.**
14	**9.50**	**-0.02**	**+0.02**		**9.52**	**11.00**	**Significant variation of a non-critical dimension.**
15	**5.50**	**-0.02**	**+0.02**		**5.47**		**Small variation outside of tolerance. May have alignment issues.**
16	**19.0**	**-0.1**	**+0.1**		**19.07**		**Within tolerance.**
17	**3.0**	**-0.1**	**+0.1**		**3.07**		**Large variation outside of tolerance for a non-critical dimension.**
18	**M3 X 0.5 F8**	**+0.006**	**+0.020**		**3.05**		**Large variation outside of tolerance for a non-critical dimension.**