



School of Engineering

MEC8099 Team Project Report

Institution of Mechanical Engineers Railway Challenge



Team 11

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Date: 3rd May 2022

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Abstract (CC)

The word 'railway' is largely familiar to most people - we all had ridden one form of it or another. However, what are locomotives and what does it take to make one?

To answer these questions, HSToon was born – towing the important milestone of Newcastle's first physical locomotive. Spreading out 4 main aspects of the locomotive: bogie; body; electrical; manufacturing & logistics, this team of 8 engineers had dedicated a year to design; correlating components; finance liaisons; and a hands-on journey of manufacturing.

As engineers, the team did not just design a mechanically functional locomotive – the team management aspect is equally as important. Efforts were made to ensure information was updated to each other; anybody could brainstorm ideas in any way; efficient collaboration; and everybody had an enjoyable educational experience.

Acknowledgements

Newcastle Railway is delighted to express our thanks to Houghton International, who had kindly sponsored the team's ambitions on the locomotive. Without such funding, crucial parts of the locomotive could not have been ordered.

As quote estimator, Paul Blakemore from Acorn Laser and his colleagues liaised with different team members promptly and professionally. The laser profiling quality and lead time saved the team much valuable time which we are very grateful for.

The current team members would like to thank all previous Railway Challenge team members from Newcastle University, without their previous hard work and efforts, we would not have reached the position of manufacturing the university's first loco.

The team would like to offer a special thank you to Mr Peter Chapman of the School of Engineering for kindly donating £525 worth of DC motors to the team. To the technicians in the School of Engineering (especially Steve Farrell, Paul Scott, Paul Watson, Chris Brown, and Shay Taylor), the team would like to thank them for their insights of practical skills and aid throughout the project.

Lastly, the team must express our sincere regards to Robert Davidson, his valuable time, efforts, patience, and insight to guide us through the birth of HSToon, which would not have existed without Mr Davidson. His willingness to offer an utter education experience, as CAD designers, machinists, and good team members were treasured by every single team member, at any given time of the project, and much beyond. 'Nothing in engineering goes to plan, yet the people in it are the utmost priority'. His words shall be carried by the team in future as engineers, and participation in other projects. We would all like to wish him a happy retirement.

Declaration

This report is submitted as part of the requirements for the Degree of Mechanical Engineering at the University of Newcastle upon Tyne and has not been submitted for any other degree at this or any other University.

It is solely the work of Alexander Hooper, Alex Powell, Cyril Cheung, David Stewart, Halley Woods, James O'Brien, Jonas Janiunas and Lightness Kings except where acknowledged in the text or the Acknowledgements below.

It describes work carried out at the University of Newcastle upon Tyne which is entirely recorded in a Project Logbook and Team Project File which has been made available for examination.

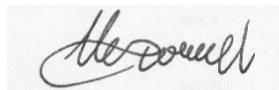
I am aware of the penalties for plagiarism, fabrication and unacknowledged syndication and declare that this report is free of these.

Alexander Hooper



3rd May 2022

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3rd May 2022

Cyril Cheung



3rd May 2022

David Stewart



3rd May 2022

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3rd May 2022

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1. Preface (AH)

1.1 Introduction and Team Structure (AH)

The Railway Challenge is an annual competition organised by the Institution of Mechanical Engineers (IMechE), hosted at the Stapleford Miniature Railway in Leicestershire. The challenge is to design and build a 10.25" Locomotive which is to compete against other teams of apprentices, students, or graduates in a series of track-based and design-oriented challenges. Each challenge is assigned several points which can be achieved based upon performance, with the winner of the competition decided as the team which accumulates the most points across the event.



Figure 1: 2021/22 Railway Challenge Team Members.

The current team consists of eight members, of which five members designed various aspects of the Locomotive during their University individual project during the 2020/21 Academic year. The preceding team were unable to build a Locomotive due to the COVID-19 pandemic. However, they entered the design-based challenges, winning the business case. Importantly though, the preceding team progressed the Loco design significantly. This, combined with the work undertaken by the five team members who had previously completed design work on the Loco, meant that the current team inherited a significant proportion of the Loco design to take forward and develop.

Despite the significant work from previous team members, design issues and conflicts still existed, as expected with a design of this complexity. The current team therefore had the task of reviewing the existing designs to bring them to the stage of manufacture, with the ambition of being the first team to represent Newcastle University with a Locomotive in the Railway Challenge. At the time of this report, the Locomotive is currently in the manufacturing stage.

1.2 Team Structure (AH)

To enable effective collaboration, team roles were elected at the beginning of the project. These allocated roles are listed in Table 1-1.

Table 1-1: Elected Team Roles

<u>Team Member</u>	<u>Elected Role</u>
Alex Hooper	Team Leader
Alex Powell	CAD Manager
Cyril Cheung	Secretary
David Stewart	Social Media Manager
Halley Woods	CAD Manager: Body
James O'Brien	Circuit Diagram & Schematics Manager
Jonas Janiunas	CAD Manager: Bogie
Lightness Kings	Treasurer

In addition to organisational roles, each member of the team was assigned to an area of the Loco of which they are responsible for the design of. This doesn't indicate that only the listed team member worked on that area, just that they were responsible. Many areas of the design were a result of the combined effort of multiple team members with many areas of the design heavily interconnected. The team member assigned responsible for each area is listed in Table 1-2.

Table 1-2: Responsible Team Member for Design Area

<u>Responsible Team Member</u>	<u>Areas of Design</u>
Alex Hooper	Control, Auto-stop, fire and heat detection, true ground speed monitoring
Alex Powell	Axle boxes, steering links, primary suspension, control box
Cyril Cheung	Transportation trolley, bump stops, Traction Centre, logistics
David Stewart	Unsprung assembly, bushings, calculations and verification, degrees of freedom
Halley Woods	Body and supporting structures, coupling system, vehicle jacking, energy regeneration
James O'Brien	High voltage circuits, energy regeneration, battery removal system, brake pipe routing, nose cones
Jonas Janiunas	Brakes, belt drive, bogie frame, traction centre
Lightness Kings	Bogie frame, secondary suspension, anti-rollbar linkage

Information about the management and organisation of the team throughout the project can be found in Section 12.

2. Project Relevance (AH)

This project has provided the team with a unique opportunity to be involved in a full design-to-manufacture process, allowing the team to develop practical skills, team-working ability and thus improve their employability. Throughout the project, the team had to carefully consider manufacturing constraints during design, replicating the limitations experienced in industry. The team have had to work together to source components and suitable manufactures, work within university finance policies and solve logistical problems, providing experience managing challenges team members will face daily in their future careers. Beyond the continued professional development of team members, the Railway Challenge presents Newcastle University an excellent opportunity to advertise the Mechanical Engineering course and potentially encourage aspiring engineers to apply to study. Success in the challenge not only reflects well on the team members personal achievements but also on the University's engineering department as whole.

3. Engineering Specification (AH, JO)

The project is to design and manufacture a 10.25" Locomotive for two purposes:

1. Compete in the IMechE 2022 Railway Challenge
2. Operate on a proposed miniature railway in Exhibition Park, Newcastle.

Of which key specifications include: (for full the technical specification please refer to the project file)

- Maximum speed: 15km/h
- Maximum mass: 2000kg
- Operable with a trailing load of 1800kg.
- Continually operate for 3 hours, with a maximum refuelling time of 2 minutes.
Assuming 5km/h continuous speed, 2% gradient, 400kg trailing load of negligible friction (1).

3.1 Railway Challenge, Stapleford Miniature Railway

The Locomotive must comply with the IMechE Railway Challenge Rules and Technical Specification (2022). An overview of the general requirements has been listed as follows:

- The locomotive will operate remotely via cable-based transmission (1).

- The locomotive shall be able to start and haul a trailing load of up to 1800kg (2% gradient).
- During operation at the Railway Challenge alone, the locomotive must not move out of the specified dynamic loading gauge (1). Features need to be provided to sufficiently limit the movement of the suspension systems.
- All components must be easily accessible and replaceable within 1 hour.
- Method statements and risk assessments must be submitted to the IMechE for activities including unloading and loading of the Locomotive from/in vehicles, work undertaken on the Locomotive during competition weekend, the refuelling, and maintainability challenge (2).

3.2 Testing, Lakeshore Railway (South Shields)

- Must be possible to adjust the gauge of the Locomotive to 9.5" to allow testing of the Locomotive on the Lakeshore Railway.
- The Locomotive must be tested for a minimum of four hours total, testing all functions and be continuously run for the last hour of which a recording is to be submitted to the IMechE as a condition of entry to the competition.

3.3 Future Operation, Exhibition Park (Newcastle)

- Traction power must be rated such that the Locomotive can haul a trailing load of 2000kg up a 1:8 gradient hill in all weather conditions.
- Smallest radius of curve at Exhibition Park: 7.76m.

3.4 Logistics and Manufacturing

- The Locomotive must be able to fit through a doorway of width 0.84m.
- The Locomotive must fit within the lift located in the Stephenson Building (Newcastle University) which has dimensions of 1.38 x 1m. It is acceptable to split the Loco into two and remove nose cones for this if required.
- The Locomotive will be transported in a 3.5 tonne van, where it must be secured during transit. Vans regularly have tailgates with 500kg capacity, which the locomotive and transporting platform must collectively comply with.

4. Trolley & Logistics (CC)

4.1 Trolley – Aims and Objectives

The locomotive manufacturing is situated in a carpeted office, on the 2nd floor of the Stephenson Building. The trolley is marketed as responsible for manufacturing and logistics – its job starts when the locomotive is getting manufactured and ends when it is safely transported to the Stapleford competition site. It is essential as the locomotive only has 1 degree of free movement (front & back), and it weighs about 1-tonne - manufacturing this on the workshop's carpeted floor will cause indentation and leave a track on the floor wherever the locomotive travels.

For safe manufacturing, the trolley must:

1. Allow no degree of movement whilst the Loco sits on.
2. Have an option to be fixed in position whilst assembling bogies.
3. Allow wheelsets spinning test.
4. Comply with H&S standard – each removable component weighs <25kg.
5. Withstand any pushes, toppling and slipping hazards during assembly.

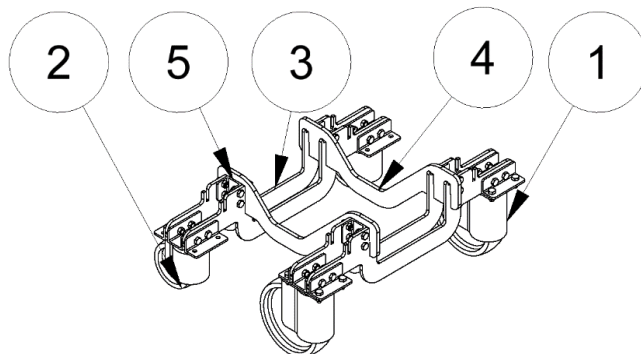
To allow safe & smooth transportation, the trolley must:

1. Fit through the workshop door – 840mm width.
2. Fit the lift in Stephenson Building – 1.4m length.
3. Be able to fit into Luton van tailgates – regularly 500kg capacity entailing the trolley + half loco (without removable parts) cannot exceed such.
4. Have capabilities to be rolled through any outdoor rough rounds.
5. Allow a slight degree of turning between bogies as the team turn it in corners.

4.2 Trolley – Design Iteration Process

4.2.1 Design Summary & Technical Specification

The trolley system consists of 4 identical wheelset trollies arranged as shown in figure 4-1. Each wheelset trolley is made of:



- 2x 200mm diameter castor wheels with brackets
- 2x 125mm diameter castor wheels with brackets
- 4x sideways connection panel
- 2x wheelset panels (for 10.25" rail gauge)
- 4x Anti-buckling U-channels

Figure 4-1: CAD Sketch of Trolley Annotation.

4.2.2 Initial Design

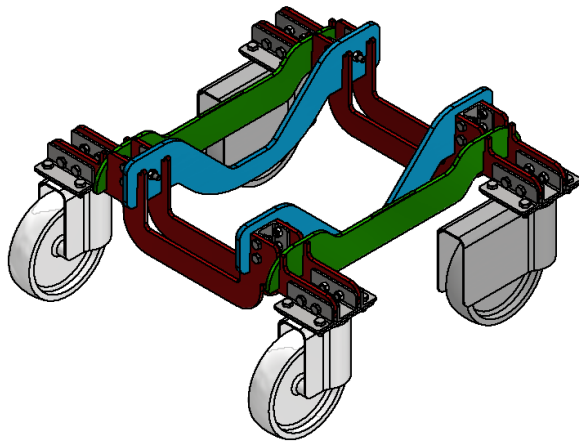


Figure 4-2: Initial Trolley CAD Design.

member does not coexist during any assembly process, as the green member is only needed for the loco sitting on the axle box during wheel spin tests.

Each wheelset trolley was originally symmetrical for an equal bending moment. The Height of the red panel was agreed to be kept consistent in between the wheelsets as the bending moment would be at maximum level throughout. The blue member catered an indent for the wheelset resting and restriction of movement for the loco. The green & blue member

Though this design has a symmetrical plane to all panels, the restriction of movement was controlled by 2 locked castors on one side. Locked castors for all would be more plausible but the spacing is restricted for trolley-loco interactions.

4.2.3 Initial Specification

In terms of technical specification, the initial design includes 4 wheelset trolleys:

1. 2x 200mm diameter BIL (Castor wheel specialist) locked castors (bigger swivel radius).
2. 2x 200mm diameter BIL swivel rubber castors.
3. Weight of 33.85kg.
4. All panels carry a safety factor of >2 for bending (the failure mode for all).
5. The team is safe to push this up to 0.82m from ground levels before toppling.

4.2.4 Final Design

The initial design was largely in good shape. However, there were several reasons it could not be a final signed-off version:

1. The red panel would have clashed into the locomotive motor, brake pulley, and very close to the steering links.
2. As much as the trolley fits well between the half loco (800mm apart), the distances between the two halves were less than such (530mm). The trollies would have clashed into each other.

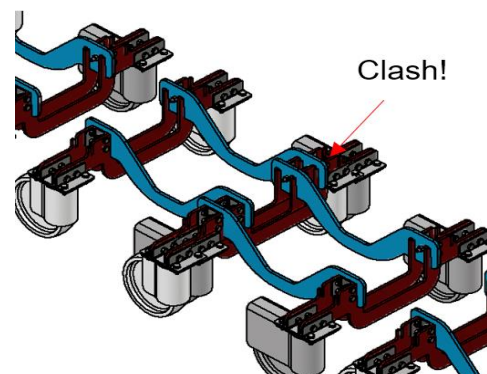


Figure 4-3: Indication of Trolley Clashing.

3. The wheelset panel (blue) has pushed the locomotive a touch too high. The centre of gravity would have been adjusted and easier for an accidental push-and-topple.
4. Panels can have weights further optimized – without compromising strength.

Subsequently, the full model of the trolley had such modifications:

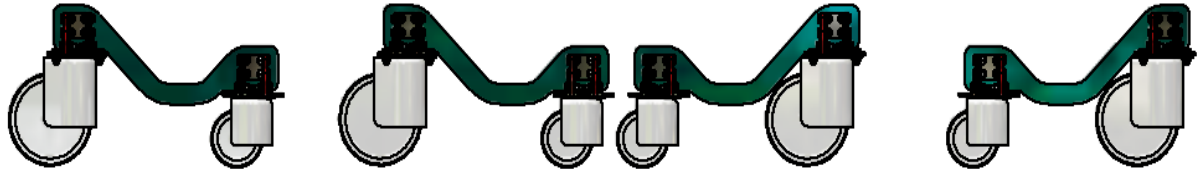
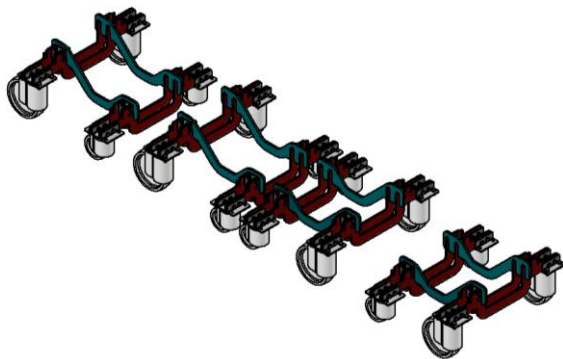


Figure 4-4: Side View of Final Trolley Design.



Each wheelset trolley is no longer symmetrical for tight spacing, though one set of castors was kept at 200mm diameter for push ease.

Figure 4-5: Projected View of Final Trolley Design.

4.2.4.1 Red Panel – Optimisation

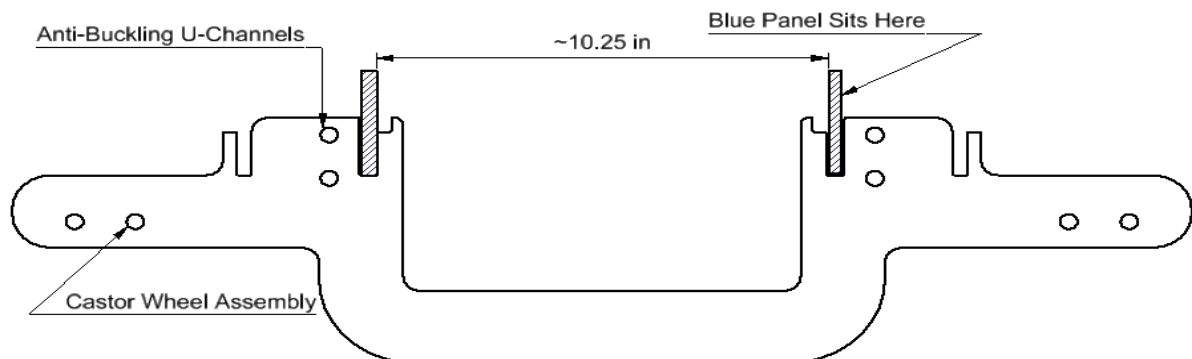


Figure 4-6: Annotated Diagram of Red Panel.

The red panel acts as a join for the two wheelset panels. It does so by a jigsaw slot-joining mechanism. As indicated in figure 4-6, blue panels get a deep slot (40mm). Not only has this given sufficient ease of removal and assembly; the jigsaw mechanism also provides good support for the blue panel. As the locomotive weighs about 1-tonne, the bounces to be experienced during transportation would not have made the blue panel leave its slots.

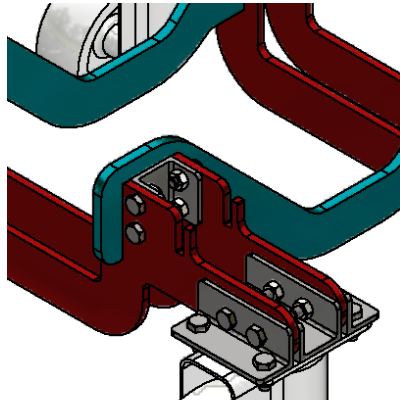


Figure 4-7: Castor-Bracket Set.

Though the 10.25" rail gauge is catered by the blue panel, for Lakeshore testing, a 9.5 gauge is needed. Another slot was created on the red panel such that the locomotive can sit on the jigsaw assembly comfortably as well. The shallower slots on the far left & right are for axle box panels (green). When the locomotive sits on the green panel, the blue panel is to be removed.

All the holes on the panel are for M10 bolts, to be laser cut through. The two lined up vertically are for anti-buckling channels, and the other set is for castor wheel sets.

The previous red panel design had issues with:

1. Motor mount – this clashed into the middle section. The bogie team tried to move the motors up instead but was unsuccessful. The minimum plate height was kept at 50mm but an indent of 120mm was made to cater for the low position of motors.
2. Steering links – too close to the far end slots for the green panel. As indicated the width of that panel slot was brought down from 10 to 8mm which is about as much as it structurally could be pushed. This increased the spacing with the links but was still insufficient. The decision was made that other trolley panels had to budge as well.
3. Brake pulley assembly – after removing material for the steering links and motor mount, it was still observed that the brake pulley assembly would clash with the red panels. It was deemed unsafe to remove the required materials, other proposals were developed for the team.

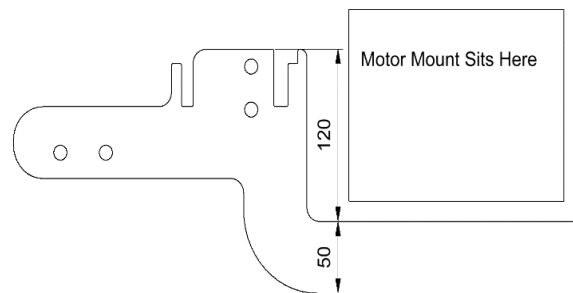


Figure 4-8: Illustration of Modification for Motors.

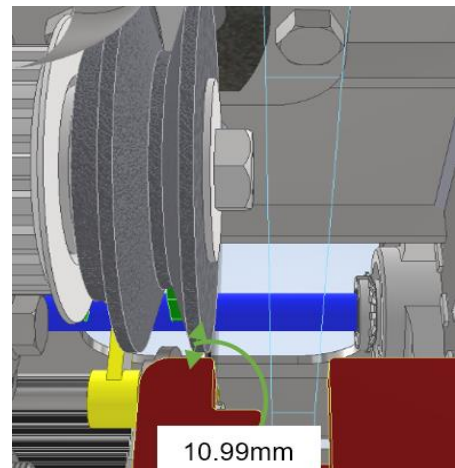


Figure 4-9: Illustration of Proximity Between Pulleys and Trolley.

Forces experienced by the panel can be shown in the figure below. This would have failed at bending, the bending moment validated as follows:

$$M = 1226.25 \text{ N} \times (0.13855 + 0.036 \div 2) \text{ m} + 100 \text{ Nm} = 291.97 \text{ Nm} \quad (4 - 1)$$

Where the extra 100 Nm is set moment for castor wheel pushes.

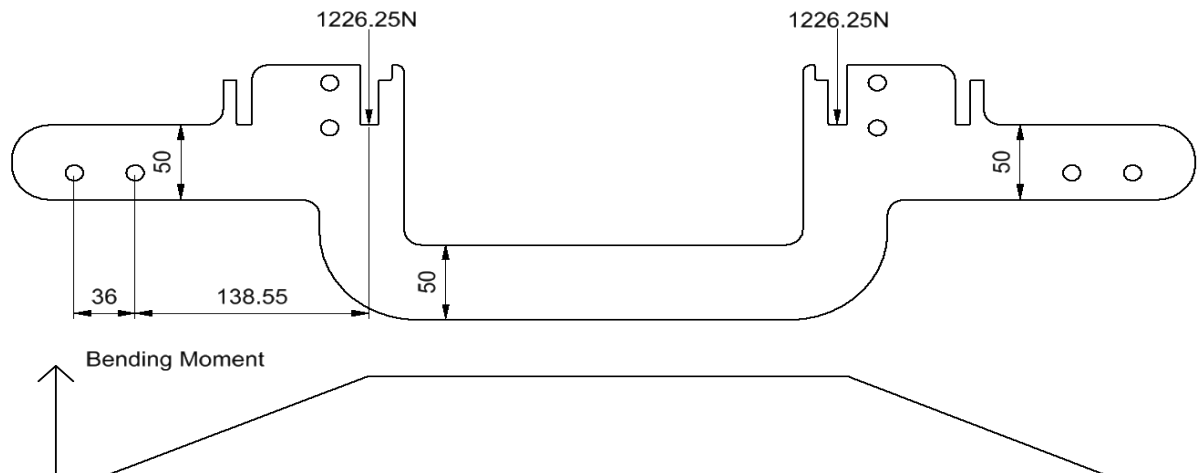


Figure 4-10: Force, Bending Moment, and Dimension Diagram for Red Panel.

$$\text{Stress } \sigma = \frac{My}{I} = \frac{M \times \left(\frac{0.05m}{2}\right)}{0.008 \times \frac{0.05^3}{12}} = 87.59 \text{ MPa} \quad (4-2)$$

$$\text{Safety Factor} = \frac{275 \text{ MPa (yield strength for S275 steel)}}{87.59 \text{ MPa}} = 3.13 \quad (4-3)$$

As an extra layer of validation, Finite Element Analysis was conducted to ensure the red panel does not experience unexpected stress:

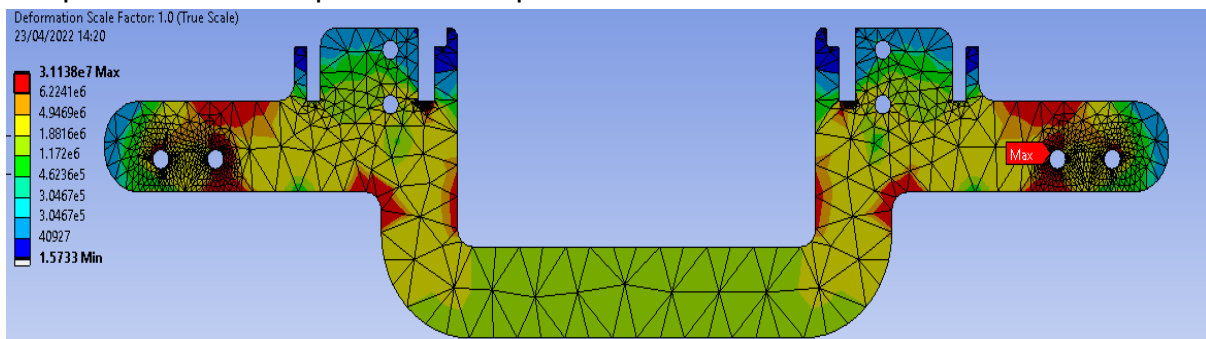


Figure 4-11: ANSYS FEA - Stress Behaviour of the Red Panel.

Deformation:

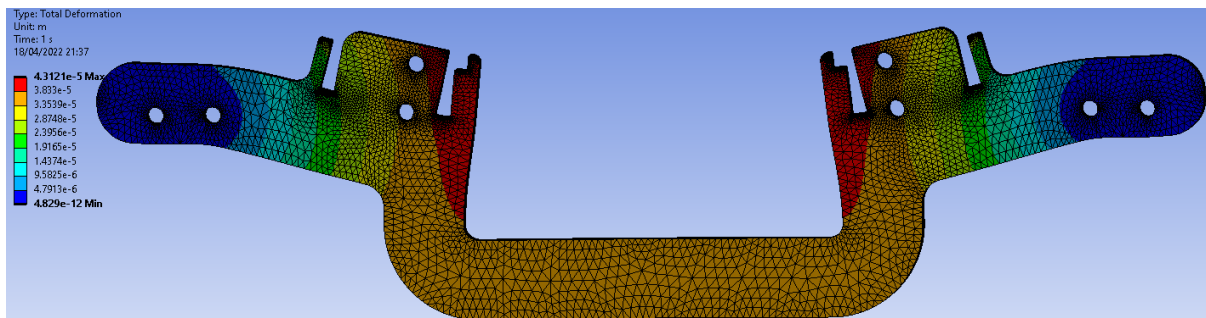


Figure 4-12: ANSYS FEA - Deformation Behaviour of the Red Panel.

Aside from the stress values, the deformation probe given by ANSYS was also of expectation. As the bending moment diagram initially drawn shows, maximum displacement would have occurred around the middle of the panel, where the blue panel is slotted in also observes maximum stress. This is a sensible behaviour as both slots are experiencing forces from the locomotive:

$$1000\text{kg} \div 4 \text{ trolley} \div 2 \text{ sides} \times 9.81 = 1226.25 \text{ N} \quad (4 - 4)$$

4.2.4.2 Blue Panel – Optimisation

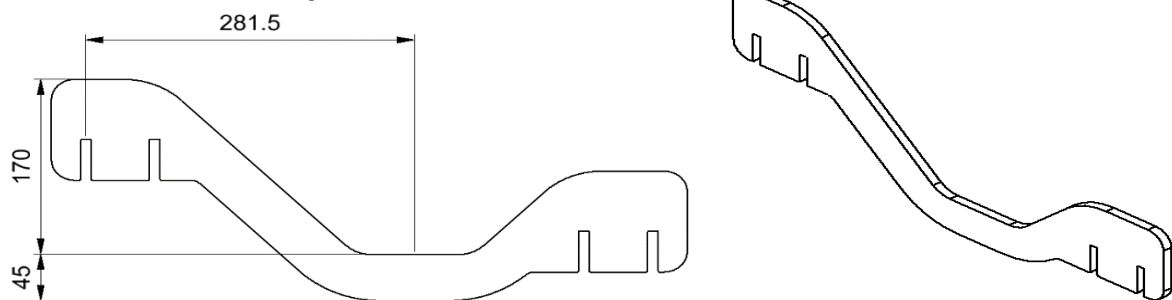


Figure 4-13: Dimensioned Sketch of the Blue Panel (left) and Side View (right).

The blue panel houses the wheelset on the deepened indent during manufacturing and Stapleford challenge weekend. It also has 4 jigsaw slots - to be inserted into 4 red panels. Despite no physical clamping mechanism for the wheelsets, this indent is sufficient for locking in the wheels as the locomotive's weight would have made any team member's day very difficult if they wished to push the loco out.

This panel also dominates the length of the wheelset trolley (whilst the red panel controls the width), it was believed the longer the blue panel, the less easy it is for toppling front/backwards. However, the trolley experiences restrictions on spacing in difficult aspects, including distances between per wheelset trolley; room for castor wheels to swivel in between as well as their radii itself; and any spacing issues with parts on the locomotive bogie.

To lower the centre of gravity, the indent was purposefully pushed down as much as possible, not only to provide more difficulties for the loco to escape but also to make sure toppling sideways does not occur as easy.

Some issues with the previous design had been rectified:

1. Brake pulley clashing into red panels – the clash was caused by the 100mm indentation on the blue panel. As much as this was safer for transportation, this created several issues.

3 Proposals were suggested to the team:

- A. Scrap the 9.5" panels – meaning the slot for it in the red panel can be removed as well. The pulleys would have just avoided the red panel. Though this further reduces weight and general trolley design complexity, not having a 9.5" panel brings difficulties for loco testing.
- B. Raise the blue panel (and the centre of gravity) – the bogie team can get away with having the blue panel lower the wheelsets by 55mm instead of 100mm. This does not compromise any panels nor require changes for the bogie team. Slight changes are needed to co-ordinate with other panels too.
- C. Scrap the blue panel completely and let the locomotive sit on axle-boxes instead – this would have reduced the materials needed to make the slots for the blue panel. Mass would have been reduced by at least 9kg. However, the axle box does not have much of a slot to be locked onto the trolley. This was deemed rather unsafe.

After discussions with the bogie team, it was agreed that it was most feasible to implement option B – raise the blue panel indent.

- 2. Wheelset distances – as mentioned above, the trolley design was based on misinformation about wheelset distances (530mm for the middle two). Modifications were made to the blue panel length, making a shorter 'B' trolley. A smaller castor wheel (125mm diameter) was also chosen to accommodate restricted spacing but not compromise push comfort as much. The other end had kept for a bigger castor for ease of pushing. The far ends still use a bigger 'A' trolley leading to an 'ABBA' approach:

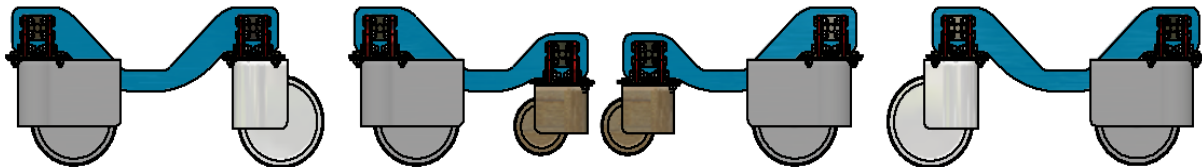


Figure 4-14: Trolley ABBA Arrangement.

- 3. Steering link clashes – after configuring the 'ABBA' approach, the team were also informed that the steering linkages would have clashed into the 'A' trolley too. The 'A' trolley was then scrapped, and the arrangement had changed to the 'BBBB' trolley:

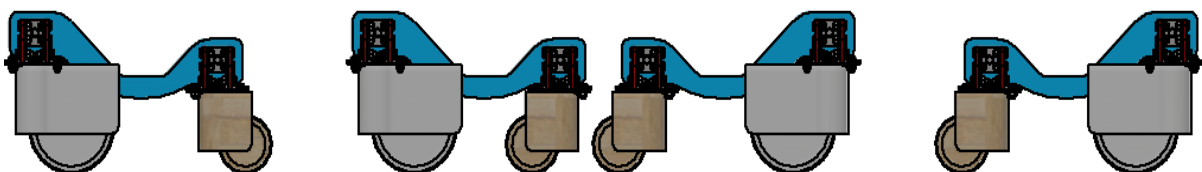


Figure 4-15: Trolley BBBB Arrangement.

This panel's failure mode is also bending. The longer side would experience a higher bending moment. Assuming worst caseload where all force acts on 1 point in the middle, its bending stress is as follows:

$$\text{Bending Moment } M = 1226.25N \times 0.2815m = 345.189 Nm \quad (4 - 5)$$

$$\text{Stress } \sigma = \frac{My}{I} = \frac{M \times \left(\frac{0.045}{2}\right)}{0.01 \times \frac{0.045^3}{12}} = 102.278 MPa \quad (4 - 6)$$

$$\text{Safety Factor} = \frac{275 MPa}{102.278 MPa} = 2.689 \quad (4 - 7)$$

Its Finite Element Analysis behaviour is as follows:

Stress Behaviour:

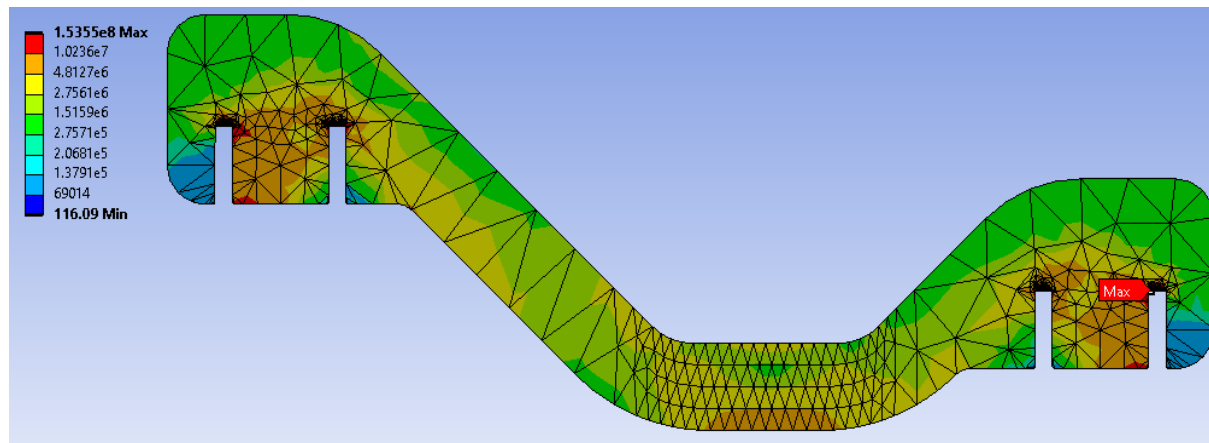


Figure 4-16: ANSYS FEA - Stress Behaviour of Blue Panel.

Deformation:

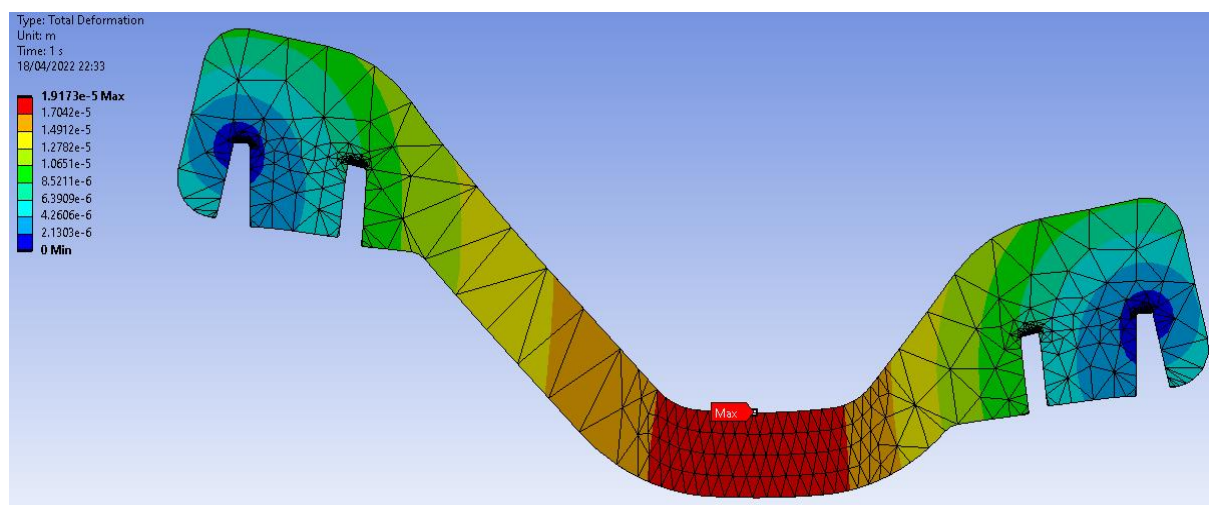


Figure 4-17: ANSYS FEA - Deformation Behaviour of the Blue Panel.

4.2.4.3 Green Panel – Optimization

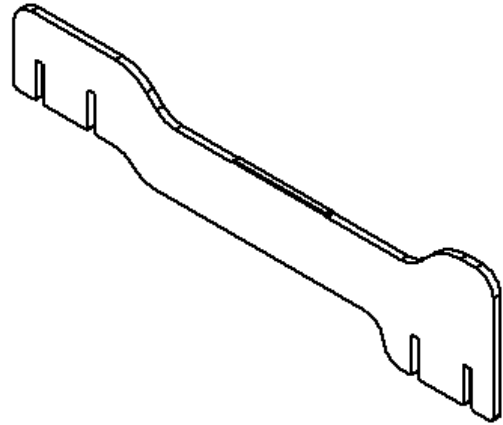
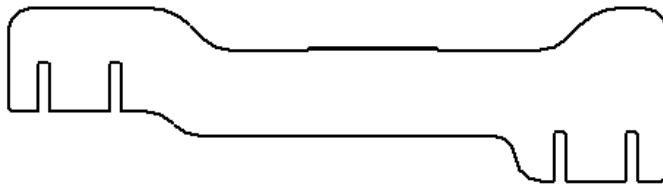


Figure 4-18: CAD Sketch of Green Panel.

The green panel will be utilised for axle box sitting. This is beneficial because before the locomotive enters its first testing stage in Lakeshore, the team needs a method to ensure the wheels are functional as they are. Also, during the electrical component assembly & fitting, there are hazards for controls connected wrong. If the locomotive was turned on by accident, the motors would have been powerful enough to get the loco away from the trolley and roll uncontrollably.

Dimensions of this panel were largely initiated from the projected geometry of the wheelset (blue) panel. This is to make sure the dimensions of the two are similar – the point where the blue panel meets the wheelset lines up with where the green panel meets the axle box as the two are both connected through the drive shaft.

As observed in figure 4-18, where the axle box sits had been notched up. The axle box has the same shape of indent on the underside such that no degree of movement could occur when they are locked together. Where the 4 jigsaw slots are has a slight incline from the axle box position such that if the locomotive does escape from its slot, it still cannot slide off the trolley.

This panel would share its stress behaviours with the blue panel, where a point force would have been at the axle box notch. The longer side of the panel induces higher bending stresses.

4.2.4.4 Panel for Lakeshore Testing (9.5")

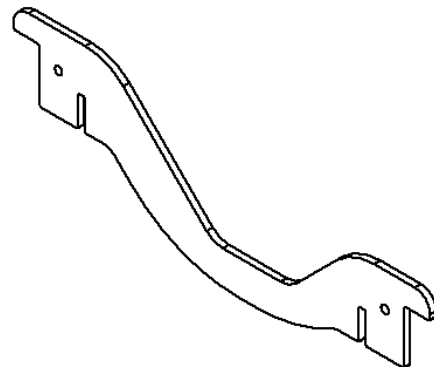
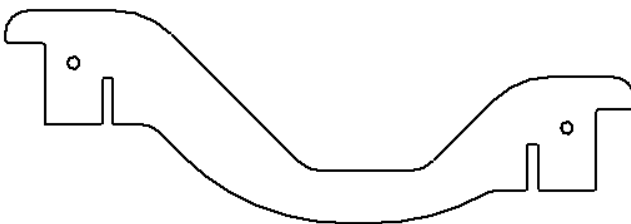


Figure 4-19: CAD Sketch of 9.5" Panel.

This panel is designed for Lakeshore Railway testing. As the locomotive wheelset gets its gauge changed to 9.5" the blue panel would be no longer suitable. An extra panel is needed on the inner side.

The 9.5" panel needs to be accompanied by the blue panel to function, hence the indent made on this panel is the same shape as the one with the blue panel – the two need to be flushed for the wheelset to sit steadily.

This panel's dimensions are like the blue panel, so as its stress behaviour.

4.2.5 Trolley Parts Purchasing & Liaison

The following is the list of merchants picked for manufacturing & acquiring parts:

Table 4-1: Table of Trolley Cost.

Parts?	Merchant?	Nature of Order?	Cost?
Castor Wheels (125mm dia. swivel)	RS Comp.	Purchase	£64.90
Castor Wheels (200mm dia. Braked)	RS Comp.	Purchase	£256.99
M10 Nuts & Bolts	RS Comp.	Purchase	£141.22
S275 Steel Angles & Channels	Metal Sup.	Purchase – to size	£140.16
Steel panels – all S275 Steel	Acorn Laser	Laser Cutting	£711.12
Grand Total:			£1314.39

Originally, a castor wheel specialist – BIL Castors was chosen for the trolley. They were at a low price so it would have aided the project's limited budget. However, the school office strongly refused the idea at late notice as it is not an active university vendor. Even though they have previously agreed to venture out and discover more vendors, it appears they have no intention of looking into the reason BIL Castors were chosen. If the same specification of castors were bought in RS Components, the cost would have gone up 3x. New castor wheels had to be picked from RS Components at the end after lengthy liaisons with the school office, even though not all technical specifications were clarified on their website (e.g.: plate dimension, swivel radius, brake pedal dimension, etc.) Other parts for the trolley were purchased relatively smoothly. Metal Supermarket had successfully delivered the brackets and channels in mid-December and Acorn Laser delivered the laser-cut panels on 3rd February.



Figure 4-20: Panels Arrival.

4.3 Physical Journey of the Trolley

Though the steel angles and channels arrived in advance (mid-December), steps of machining such as drilling, and bolting should be done with the steel panels present.

A manufacturing plan of the trolley was necessary, to factor in Lakeshore Testing's procedures as well. This was categorised into

several steps, including its interactions with the loco, testing and Stapleford Weekend:



Figure 4-21: Assembled Trolley.

Table 4-2: Stepped Approach of Trolley Manufacturing and Logistics.

Steps	Items Involved	Process	Tools & Specialists
Trolley Stage 1			
1	L-brackets & U-channels	Filling and Filletting	Linisher, hand file, vice
2	Steel angles & channels for castors, red panel	Hole drilling	Hand & Pillar drill, Centre Punch, Vice
3	Steel angles & channels for castors, red panel, castors	Fastening	17mm spanner, vice
4	U-channel (for anti-buckling), red panel-castor assembly, M10 nuts and bolts	Fastening	17mm spanner, vice
5	Blue Panel, Castor assembly	Slot panel to assembly	N/A
Trolley Stage 2			
6	Bogie, Assembled Trolley	Bogie assembled on trolley	N/A (for trolley)
7	Loco Wheelset, trolley, green panel	Jack up loco, remove blue panel and slot in green (axle box) panel instead, making the wheelset hanging	Jacks
Trolley Stage 3			
8	Assembled Trolley, blue (wheelset) & 9.5" panel	Jack up loco, remove green panel, change rail gauge to 9.5", insert blue (wheelset) and 9.5" panel, rest loco on trolley again	Jacks, Lathe
9	Trolley with the locomotive on	Bring to Lakeshore Railway for testing,	Jacks, Full team, Luton van

		practice load and unload there as well	
Trolley Stage 4			
10	Assembled Trolley, blue (wheelset) & 9.5" panel	Reserve Step 8 to return to 10.25" rail gauge, remove 9.5" panel	Jacks, Lathe
11	Trolley with the locomotive on	Bring loco to Stapleford	Jacks, Full team, Luton van

Like many manufacturing plans, unanticipated challenges were appearing throughout the duration of trolley assembly. This includes:

Table 4-3: Manufacturing Challenges, and Rectifications.

Problem Statement	Solution	Lesson Learnt
purchased castor wheels were different from the proposed ones caused by approved merchant issues plate sizes of the smaller castor wheel is too small	The castor plates were welded onto the bracket assembly.	As much as it is needed to purchase from university approved merchants, if the item does not have clarified technical specifications, it should not be purchased.
some inaccurate measurements on steel angles	Bigger sizes of holes were drilled to allow the M10 bolt to go through. Anticipating such issues might occur, lock nuts were purchased to further fix positions.	Holes are to be centre punched, marked with correspondent panels, and drilled through with a pillar drill to make sure it's straight.
castor wheel's specification was unclarified castor heights were and caused a 32mm lift	Rapid re-design was required, to fit on the locomotive model making sure no clashing occurs, and Acorn Laser to re-cut a new design.	

4.4 Logistics – Body Assembly Arrangement

4.4.1 Background and Objectives

In the case of the railway challenge for this team, the best part of the locomotive construction takes place in a carpeted office with limited room. Whilst the bogie is constructed on the trolley, which is mobile; the body is built on trestles ~750mm height – significantly higher than the trolley & bogie.

The question was soon raised – how would the team put the body & bogie together? And the answer certainly was not one group holding the body in position while the other bolt and rivet it onto the body. The body (per half) weighs up to 200kg and whoever holds it would have to keep it in position for prolonged period. This is not ergonomic nor safe – the risk of tripping and crushing electrical parts is vast.

Several solutions were discovered for lifting one of the items to attach it to another, they should all satisfy the following objectives:

1. Allow safe assembly of the locomotive body & bogie
2. Allow slow and smooth contact as both the body and bogie have significant weight, dropping either one causes uncontrollable damage.
3. This device should still allow the team to crawl underneath to access the bolting & fastening points between the body and bogie.
4. Must be mobile and light enough to be brought to Stapleford and Lakeshore.

4.4.2 Initial Thoughts

Some initial thoughts were gathered to approach this:

Idea 1: Hoist and Stands

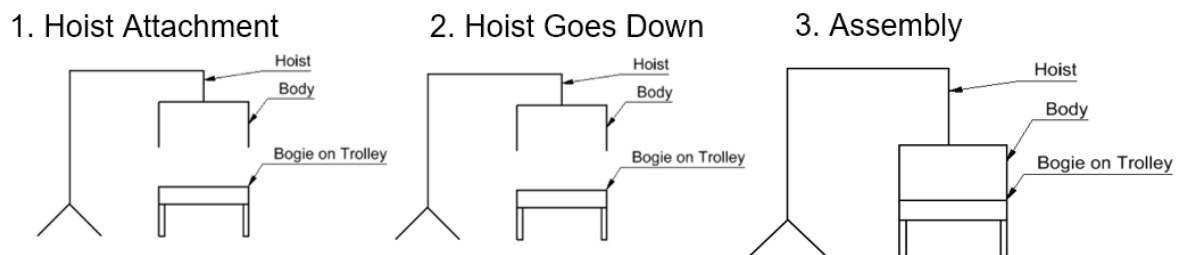


Figure 4-22: Idea 1 - Sketch of Proposed Steps.

This can take place in FSW. There is no need to fit the hoist/crane/ lifting device as those that are dismantlable are commercially available. As seen in figure 4-22, the hoist would have lifted the body up and slowly placed on top of the bogie. Team members in charge will then crawl under and tighten the designated constraints.

Advantages: Hoist does all the lifting, no manual labour for any personnel/ can stay in FSW for this/ hoist is a stable method to hold the body

Disadvantages: acquiring a hoist might be costly/ hoist can be rented instead but if the body needs to be disassembled and assembled several times attempting to rent a hoist every time adds unnecessary hassle & cost

Idea 2: Build the whole body on top of the bogie

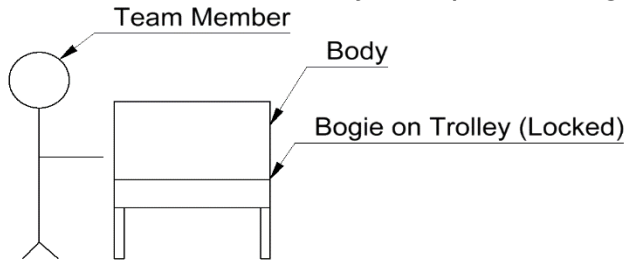


Figure 4-23: Idea 2 - Sketched Proposed Approach.

If the two components were not built separately in the first place, there would be no need for trestles nor acquiring any lifting methods.

Advantages: easy and quick/ no extra cost to buy or rent anything

Disadvantages: this meant the body can only be built after the bogie is finished which is not practical timewise. The body is largely riveted together which is time-consuming to un-do and re-do

Idea 3: Lift the bogie rather than the body for assembly

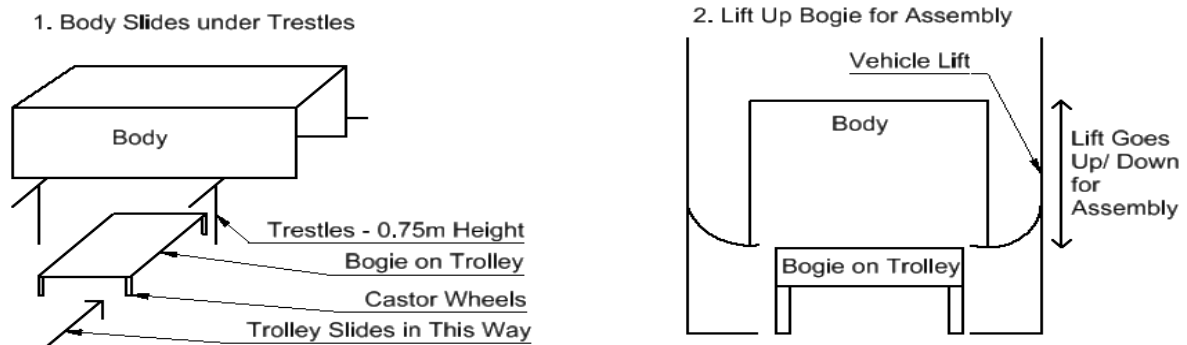


Figure 4-24: Idea 3 - Sketch of Proposed Steps.

Most ideas were circulated around the body being transplanted onto the bogie one way or another. However, the bogie is the mobile one (i.e.: on trolley). So, what if the bogie is to be moved in position for the body instead?

As figure 4-24 illustrated, the body could be standing on trestles (made sure it was spaced far enough to avoid the trolley & bogie), where the trolley can slide in enabling the bogie to be directly underneath. With a vehicle lift coming into play, the bogie could be lifted to level with the body. At this point the trolley could be temporarily removed, allowing the team to crawl underneath to constrain the two components – just like in garages where mechanics lift vehicles at height to inspect undersides. This idea was inspired by vehicle inspection lifts in garages, shown in figure 4-25.

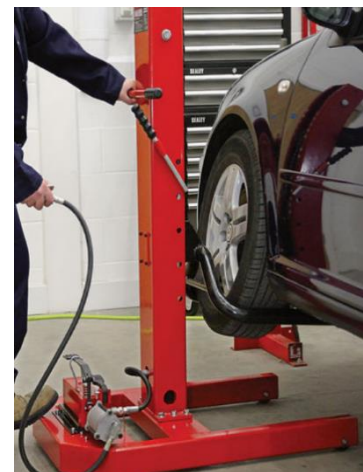


Figure 4-25: Idea 3 - Illustration of Vehicle Lift Usages.

Advantages: this is commonly seen in garages therefore deemed feasible/ vehicle lift could be compact enough to stay in FSW, and transportable to Stapleford

Disadvantages: When the bogie is level with the body, there are no constraints or assurances that the body would not go out of place from the trestles since it is merely sitting on it

Idea 4: Scissor Jacks to lift the bogie



Figure 4-26: Idea 4 - Scissor Jacks.

Like the vehicle lift idea, it might be a better idea to lift the bogie instead as the body has less mobility. Some garage equipment like a vehicle lift has been suggested. Scissor jacks make this mechanism even more mobile.

Advantages: even higher mobility than vehicle lifts/ cheaper option

Disadvantages: Scissor Jacks cannot lift as high as vehicle lifts/ there are also no constraints ensuring the body will not go out of place during such assembly

4.4.3 Chosen Arrangement – Design & Validation

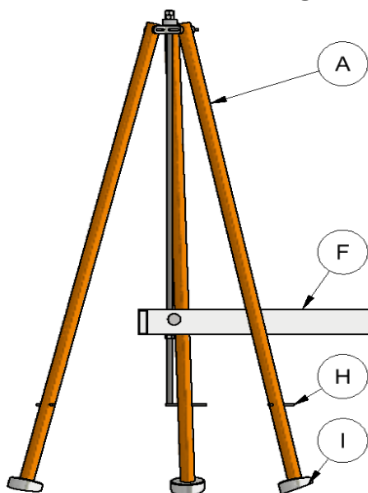


Figure 4-27: Full View of Body Lift.

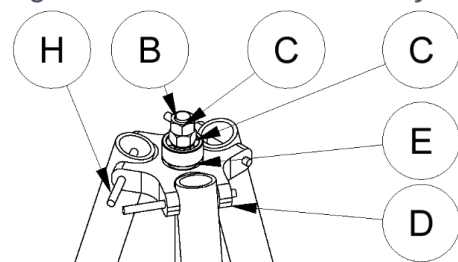


Figure 4-29: Detailed View of Body Lift Top Connection.

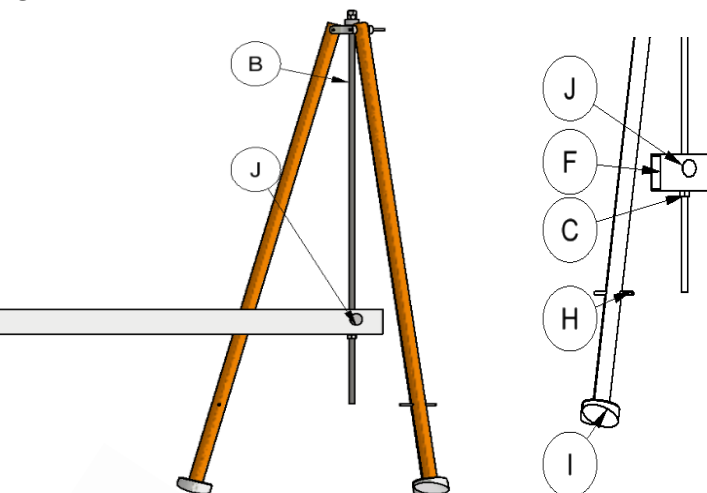


Figure 4-28: Detailed View of Body Lift Bottom.

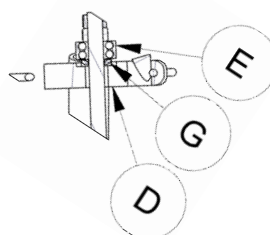


Figure 4-30: Sectioned View of Connection – Bearings & Washers.

The final arrangement for this matter is to design a body lift as shown in the images above. In general sense, this system comprises 2 tripods and 1 aluminium box tube in between where the body sits. Its lifting mechanism is controlled by the studding on both sides. Using 2 electric drills simultaneously allows the whole boxed tube to be lifted. The 2 of these systems are required for lifting, and collectively they have a capacity of 1000kg, which is how much the whole loco weighs. Even though this is only meant for lifting a 200kg half body at a time, it was anticipated that throughout the construction process it would be convenient to have a mobile device that lifts the entire loco. Its design features are as follows:

1. Tripods – aluminium round tubes at a 15° angle, connected by a steel connection plate on top.
2. Connection plate – laser cut plate with star-shaped extensions for tripod legs to be slotted in and pinned together. The centre allows studding to be through and houses a ball bearing, with a set of spherical washers spaced in between.
3. Deep Groove Ball Bearing – this allows the studding to spin freely but not the connection plate. The spherical washer spaces the bearing from the connection plate and allows slight degrees of play for the lifting mechanism.
4. Boxed tube – going through the studding on either side, the boxed tube has a slot on both ends with room for the studding to swing slightly. The slot also has a stainless-steel round block to restrict lateral movement.
5. Bottom stands – as the tripod legs are angled at 15°, the bottom stand is to be machined to size and at the same angle as well to fit well each tube.
6. Chain – this is to wrap around each tripod.

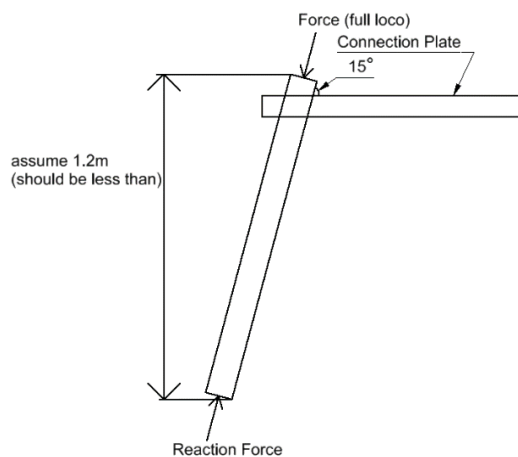
Table 4-4: List of Proposed Purchase Lift for Body Lift.

Item	Qty (per system/ needed in total)	Description	Related active vendor
A	6x/ 12x	Aluminium 6082 T6 Round Tube (1.25" diameter x 1.2m length)	AALCO
B	2x/ 4x	M12 Studding (Grade 8.8, 1m length)	RS Comp.
C	4x/ 8x	M12 Nuts (Grade 8.8)	ACCU
D	2x/ 4x	S275 Mild Steel Connection Plate (16mm thickness)	Acorn Laser
E	2x/ 4x	SKF 4201 ATN9 Deep Groove Ball Bearing	RS Comp.
F	1x/ 2x	Aluminium 6082 T6 Boxed Tube (2" x 2" x 0.25" x 1.5m length)	AALCO
G	2x/ 4x	Set of Male & Female Spherical Washers	ACCU

H	12x/ 24x	M6 Socket Cap Screws/ Pins (at least 70mm length)	RS Comp.
I	6x/ 12x	Aluminium 6082 T6 Round Block (3" diameter x 100mm length)	AALCO
J	2x/ 4x	Stainless Steel EN24T Round Block (30mm diameter x 65mm length)	Metal Sup.
K	1x/ 2x	Chain for Wrapping Legs	RS Comp.
L	6x/ 12x	M6 Grade 8.8 Nuts for Pins	ACCU

To ensure the body lift can withstand its said load capacity, a variety of validation calculations were carried out, the main ones are as follows (please refer to the appendix for further validation checks):

Buckling – Tripod Legs



As figure 4-31 illustrated, the two forces acting on the aluminium tube can result in the tube crushing/ buckling. Consequently, the load this tube can take before buckling (critical load) was calculated:

$$P_0 = \frac{\pi^2 \times E \times I}{4 \times L^2}$$

Figure 4-31: Sketched Illustration of Round Tube Buckling Condition.

Where E is Young's Modulus, 4 is safety factor, L is length of tube.

I is 2nd Moment of Area, $I = \frac{\pi}{64} (D^4 - d^4)$ where D & d are outer & inner diameters.

$$P_0 = \frac{\pi^2 \times 70 \times 10^9 \times \frac{\pi}{64} \times (0.030^4 - (0.030 - 0.003 - 0.003)^4)}{4 \times 1.2^2} = 2815.6364 \text{ N} (4 - 8)$$

Assuming this lift is taking the entire loco's load, and in the whole system (4 tripods, 12 legs) only 4 legs were in contact with the ground due to unlevel flooring in the workshop, each leg needs to take up to $1000\text{kg} \times 9.81 \div 4\text{legs} = 2452.5\text{N}$, this is lower than the critical load of the legs. Therefore, buckling should not occur.

Bending – Connection Plate Star-shaped Slot for Legs

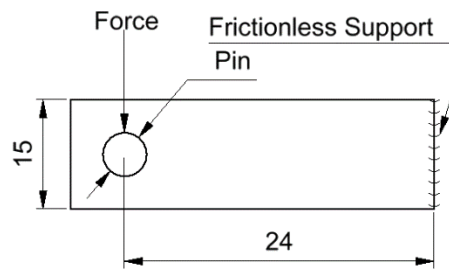


Figure 4-32: Sketched Illustration of Connection Plate Ears Bending.

The connection plate was made of 16mm mild steel, and the star-shaped extensions had a hazard of cracking off the plate through bending. It was made as light as possible (570g) to improve mobility. However, the studding can take up to 17000N (more than required).

So, each 'ear' must take:

$$17000 \text{ N} \div 6 \text{ ears} = 2833.33 \text{ N} \quad (4 - 8)$$

Each 'ear' was extended out for ~24mm, making the bending moment:

$$2833.33 \text{ N} \times 0.024 \text{ m} = 68 \text{ Nm} \quad (4 - 9)$$

Consequently, its bending stress is:

$$\sigma = \frac{68 \text{ Nm} \times 0.0075 \text{ m}}{\frac{0.010 \text{ m} \times 0.015 \text{ m}^3}{12}} = 181.33 \text{ MPa} \quad (4 - 10)$$

Considering the plate material, S275 mild steel yields at 275 MPa, the 'ears' could not have cracked off. This is also further confirmed by an FEA analysis:

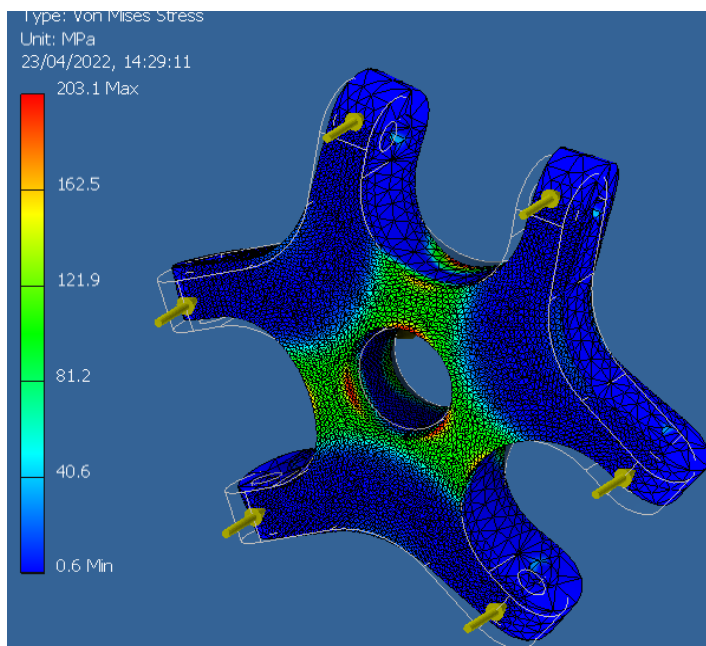


Figure 4-33: Inventor FEA - Stress Behaviour of Connection Plate.

Bending – Boxed Tube for Body/ Loco

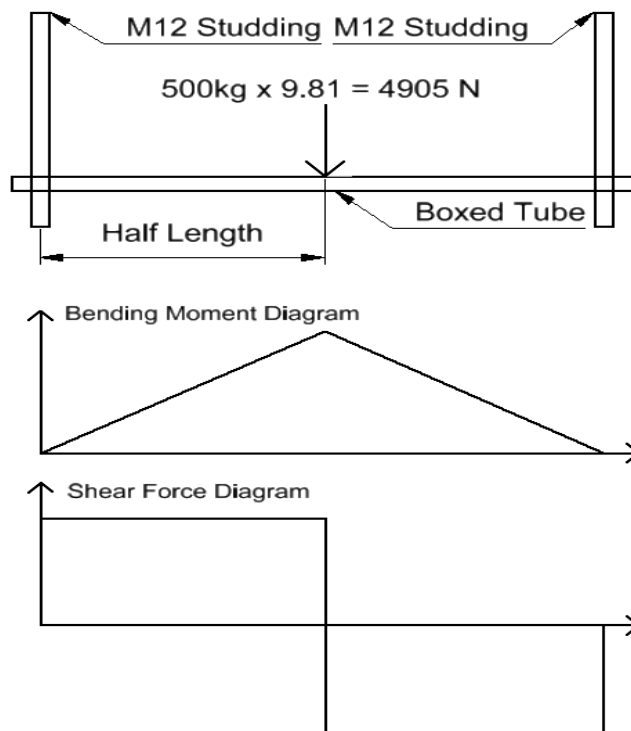


Figure 4-34: Force Distribution, Bending, and Shear Behaviour of the Aluminium Boxed Tube.

As much as the shorter the boxed tube, the lower the bending stress, the minimal requirement for its length is governed by the lateral distance of the tripod legs (0.30m); the bogie & trolley width for sliding underneath (0.70m). Making the tube's minimum length to be: $0.30m + 0.7m + 0.30m = 1.3m$. Its cross-section dimensions are 2" x 2" x 0.25" wall thickness. (2" = 50.8mm; 0.25" = 6.35mm)

Assuming the whole locomotive is on this stand (9810N of force, 4905N per system), and it is acting as a singular

point force in the middle of the boxed tube (realistically it should be 2-point forces acting closer to the studding), the boxed tube must be strong enough to withstand such worst-case scenario. Bending would be the failure mode for this as well.

$$\text{Bending Moment } M = 4905 \text{ N} \times 0.75 \text{ m} = 3188.25 \text{ Nm} \quad (4 - 11)$$

Its bending stress is: $\sigma = \frac{BM \times (0.0508 \times 0.5)}{I}$

$$\text{2nd Moment of Area } I = \frac{50.8 \times 50.8^3}{12} - \frac{(50.8 - 6.35 - 6.35) \times (50.8 - 6.35 - 6.35)^3}{12} \quad (4 - 12)$$

$$I = 379377.6 \text{ mm}^4 = 3.793776 \times 10^{-7} \text{ m}^4 \quad (4 - 13)$$

$$\sigma = 213.46 \text{ MPa} \quad (4 - 14)$$

Considering Aluminium 6082 T6 yields at 270 MPa, it was deemed safe as the worst load case does not exceed the material yield strength.

5. Bogie (LK)

Bogie is a system located underneath the locomotive vehicle body that contains the wheels, axles, suspensions, and motors allowing the locomotive to operate on tracks.

(3) The final bogie design was an adaptation of the work done in 3rd year as well as the work done by the previous 2021 railway team.

It was established very early at the start of the project that the bogie design that was to be modified with the aim of manufacturing and entering the competition with will be

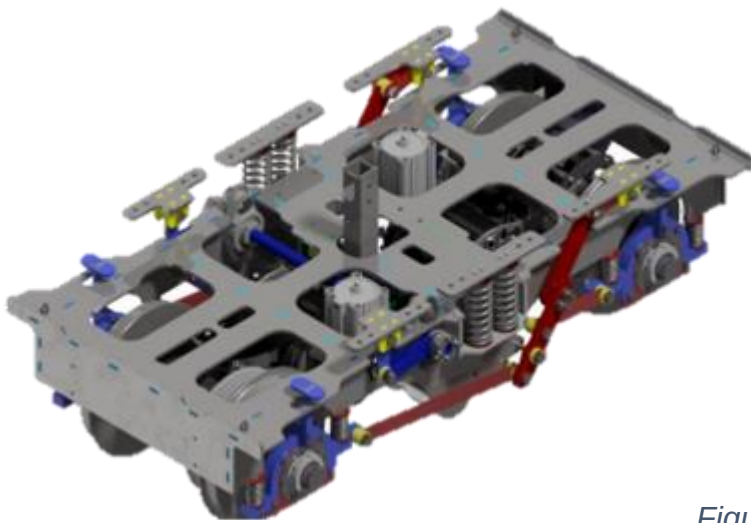


Figure 5-1: 2021 Bogie system design

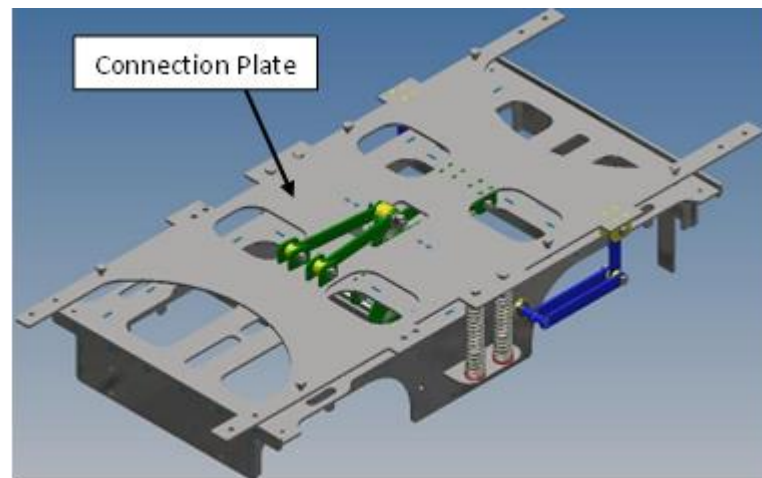


Figure 5-2: Year 3 bogie frame design with connection plate

the design developed by 2021 railway team; the main reason for such decision was that even though the design has some similar aspects to the 3rd year bogie design it was more superior as it did not require the use of the connection plate which helped to alleviate some weight off the bogie.

Figure 5-3 shows the final bogie system design that was achieved by the 2022 Railway team.

Below are the design justifications as well as designs of how the whole bogie system came to be together.

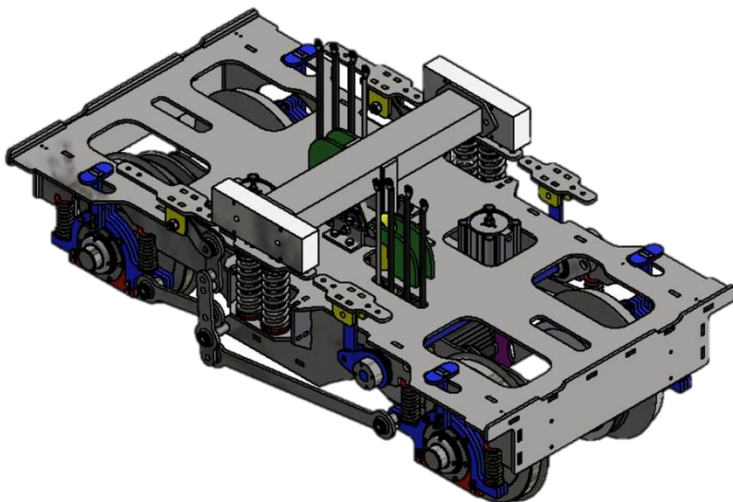


Figure 5-3: 2022 Railway Team final bogie system design

5.1 Bogies Degrees of Freedom (DS)

Due to the primary and secondary suspension, the wheelset and body is allowed to yaw, roll, and pitch with respect to the bogie. This movement must be limited to avoid extreme tipping moments on the body that can cause the locomotive to topple over and derail. The displacements must be enough to dampen vibrations from track irregularities and allow for cornering. The table below represents the locomotives displacement magnitudes and deflection angles of the degrees of freedom.

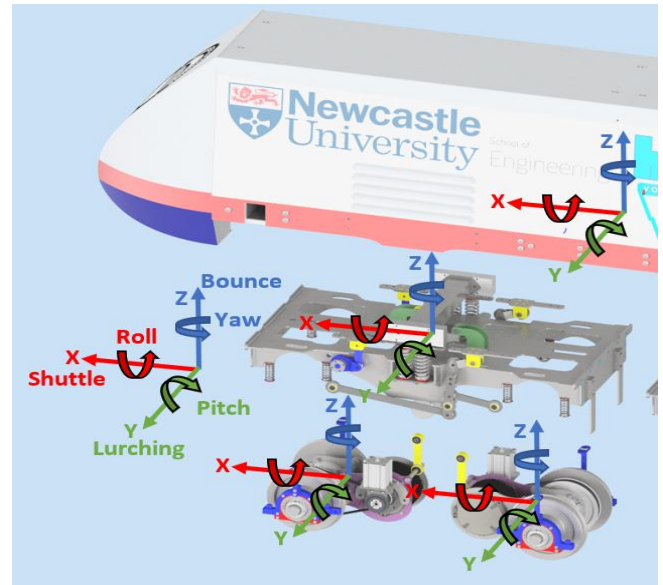


Figure 5-4. Degrees of freedom of the wheelset, bogie, and body.

Table 5-1. Displacements and angles of deflection in each axis of freedom for the wheelset and body relative to the bogie. (Calculations in appendix H-10)

Motion	Components in Relative Motion	Unconstrained Displacement		Constrained Displacement		Method of Constraint
		mm	Degree	mm	Degree	
Shuttle (X)	Wheelset-Bogie	40.6	-	15.6	-	Axle box T-Bar
	Bogie-Body	28.6	-	0	-	Traction Centre
Roll (About X)	Wheelset-Bogie	40.6	11.5	1.53	0.439	Axle box T-bar
	Bogie-Body	9.34	1.44	1.66	0.367	Anti-Roll Bar
Lurching (Y)	Wheelset-Bogie	40.6	-	9.0	-	Axle Box T-Bar
	Body-Bogie	21	-	15	-	Bump-stops
Pitch (About Y)	Wheelset-Bogie	9.5	0.682	15	1.07°	Axle Box T-Bar
	Body-Bogie	25.2	0.682	37.3	0.99°	No Constraint
Bouncing (Z)	Wheelset-Bogie	50.5	-	15	-	Axle Box T-Bar
	Body-Bogie	35.8	-	21	-	Traction Centre Hooks
Yaw (About Z)	Wheelset-Bogie	13.2	2.87	29.9	4.27°	Axle Box T-Bar
	Body-Bogie	21	4.82	11.5	7°	Bushing Conical Deflection Limit

5.2 Bogie frame (JJ)

The bogie frame has been reused from 2021 team's design with minor modifications for ease of manufacturing, assembling and optimization.

Changes made:

- The slots have been increased in size by 0.5mm on each side to allow easier assembly
- Motor mounts and traction centre bushing mounts have been optimized
- Bump stops have been added to have a soft limit for bogie movement
- Minor modifications made to fit the new designs
- FEA performed for bogie twist and static loading
- Centre brace has been moved up

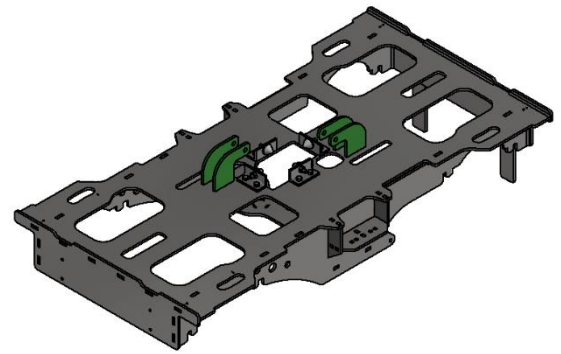


Figure 5-5: 2021-2022 locomotive bogie frame

5.2.1 Motor Mount Brackets (CC, JJ)

The motor mount bracket is an L-shaped inverted bracket, the two slots on one end are to be welded on the bogie frame; the other end houses bushings, where 1553N of force is exerted vertically. This would have failed in bending.

Its stress analysis behaviour is seen as figure 5-6, as expected, the max bending stress occurs at the bending pivot point.

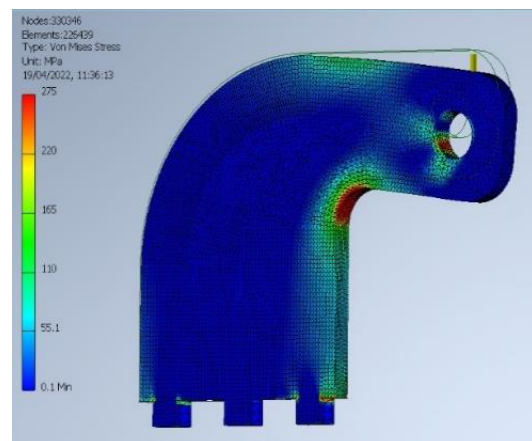


Figure 5-6: Inventor FEA - Mount Bracket Stress Behaviour.

5.2.2 Traction Centre Bushing Brackets (CC)

Like the motor mount bracket, the TC bushing bracket also has two slots welded onto the bogie frame. This would have taken 19816.2N longitudinally (X-axis) to transfer tractive effort to the body.

Its stress analysis behaviour is seen to at the image here, where the max bending stress would have been by the welding spots.

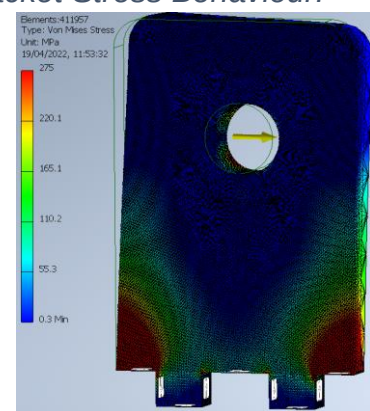


Figure 5-7: Inventor FEA - Traction Centre Bracket Stress Behaviour.

5.2.3 Bump Stops (CC, DS)

Bump stops are small rubber conical buffers that avoid metal-to-metal contact. As for the case on the bogie frame, bump stops were required and to achieve the following objectives:

- Avoid metal-to-metal contact between the bogie frame and bent plates of the traction centre.
- Simultaneously not restricting the movement for the Traction Centre.
- Provide a 15 mm soft limit and 20 mm hard limit to the body's lateral displacement.

As seen in the image above, each bump stop assembly consists of an S275 mild steel angle bracket (30x30x5mm), a rubber bump stops, 2 M8 bolts, and 3 M8 nuts for fastening.

Only the sides of the traction centre needed bump stops to limit lateral displacement because motion was restricted in the longitudinal direction. However, all four edges had a bump stop in place for further security. Due to the unusual shape of the TC slot on the bogie frame, to achieve the 15 mm lateral movement the TC originally had; holes were made on the bogie frame but all 4 had different distances from the edge as shown in the annotated image above.

All bump stops were stuck out of the frame slightly to ensure they could have contacted the TC. For the same reason, the rubber edge was deliberately positioned higher than the metal angle.

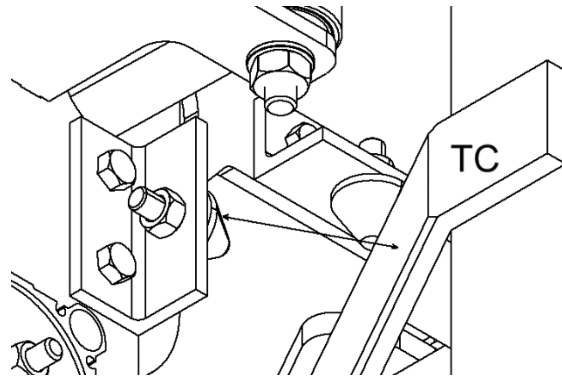


Figure 5-8: CAD Sketch - Bump Stops Positioned Allowing Contact with Traction Centre.

5.2.4. Frame Clearances (DS)

The design of the bogie frame top plate was the last component on the bogie to be finalised as its dimensions revolved around the displacements of the wheelset and traction centre during wheelset-bogie and bogie-body relative motion. Clearance holes were necessary for the traction centre, the motor

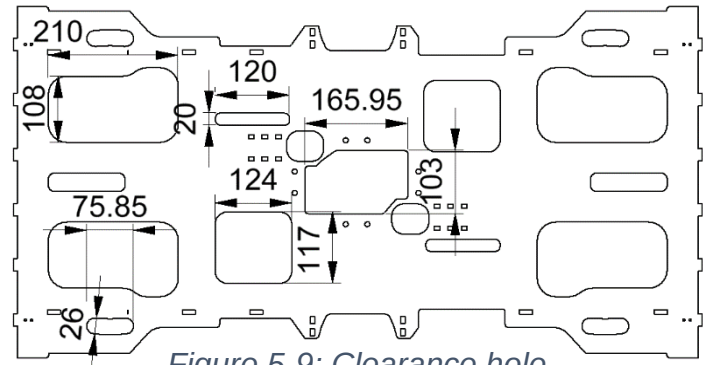


Figure 5-9: Clearance hole dimensions for bogie top plate

mount linking arm, the brake cylinder, the motors cables, and the wheels to ensure none of these components collided with the bogie frame during maximum relative displacements or maximum angles of yaw, roll or pitch. The axle box t-bar clearance hole was designed so it restricts extreme lateral bogie lurching to 10 mm. Calculations for the required frame clearances can be found in appendix H-9.

5.2.5 Bogie frame FEA (JJ)

The bogie frame has a complicated shape. Therefore, FEA analysis has been used to simulate the critical loadings – twist and static loading. To make the model work slots had to be made perfect fit instead of a 0.5 mm clearance fit. Also, welds, some brackets and bump stops have been removed to speed up the simulation, make it more accurate and remove errors. Simulations have been carried out for Von Mises Stress and displacement. Identical mesh has been used for both loadings

Static loading

One-half of the body weighs 200kg. Therefore, a force of 2 kN is distributed at the secondary spring mounting location and equality reacted by 8 primary spring mounts. The side panels which experience the highest stresses have 3mm mesh and the rest of the bogie has 5mm mesh (*Figure 5-9*). Under static loading, the stress is 38.72 MPa which is way below the yielding stress (*Figure 5-10*). Displacement is less than a millimetre (*Figure 5-10*). High-stress concentrations are marked in red circles. The design can be improved by increasing the corner radius in these areas. Red lines

indicate possible future improvements which would help to reduce the weight and the stress concentrations.

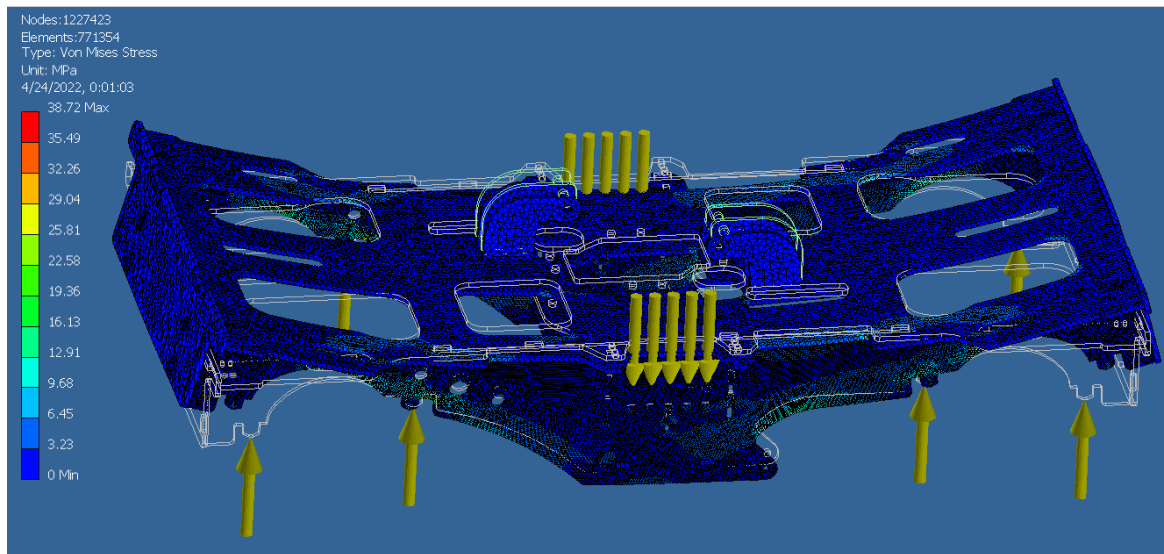


Figure 5-10: FEA model with the mesh view. The mesh has 1.2 million nodes. Same mesh was used for all the bogie frame simulations. Side plates are experiencing the highest stresses

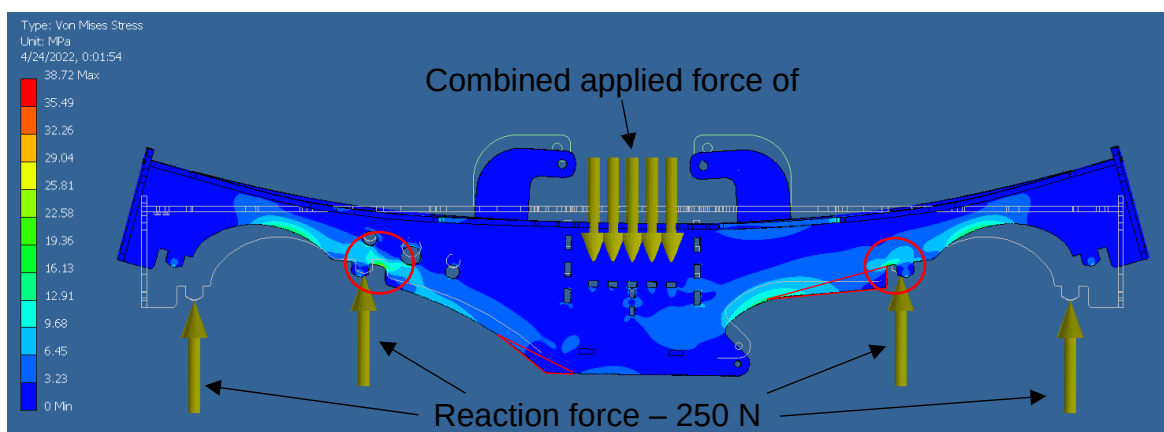


Figure 5-11: Bogie frame under static load showing von Mises Stress

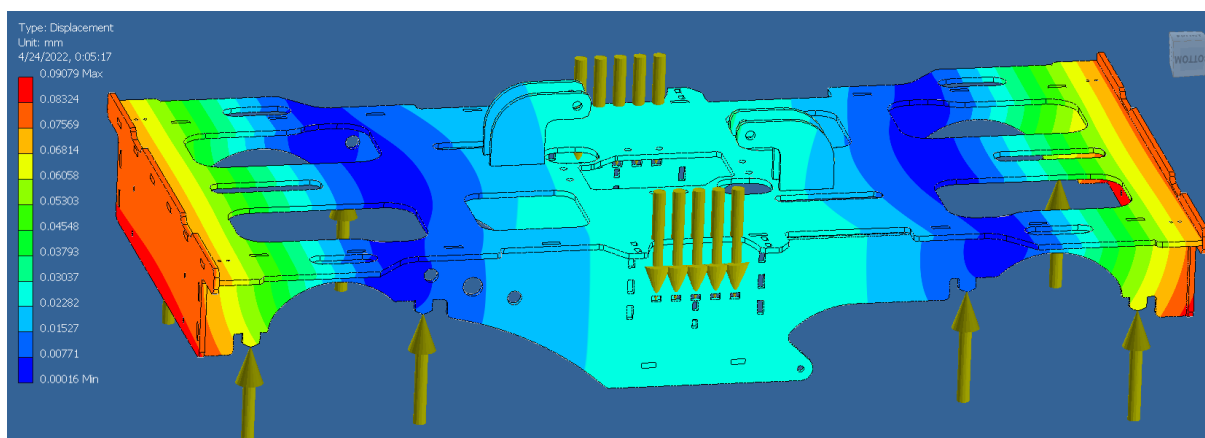


Figure 5-12: Bogie frame under static load showing deformation

Bogie twist

For the bogie twist, the same 2kN load has been applied and the same mesh has been used. However, the reaction forces have been scaled to 80% and 20% (*Figure 5-13*). This scaling has been chosen because if the reaction force on one side of the bogie drops below 20%, it means that the train has derailed. Maximum stress under this loading is 140.7 MPa (*Figure 5-13*). This gives a safety factor of 1.95. The highest stresses are circled in red. There is only 2.5 mm displacement at the corners of the bogie which is reasonable and safe (*Figure 5-14*).

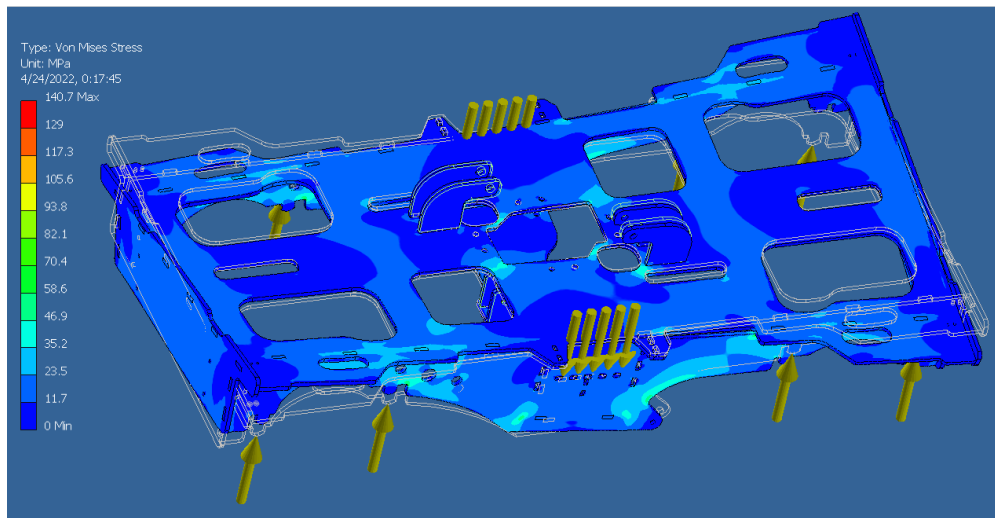


Figure 5-13: Bogie frame during extreme twist loading. Side plates experience highest stresses

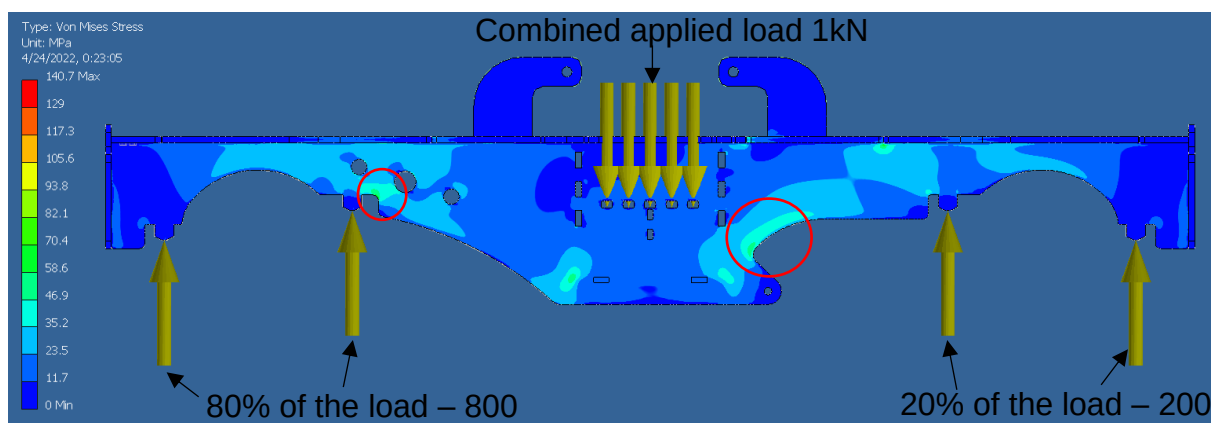


Figure 5-14: Bogie frame under twist load showing von Mises Stress

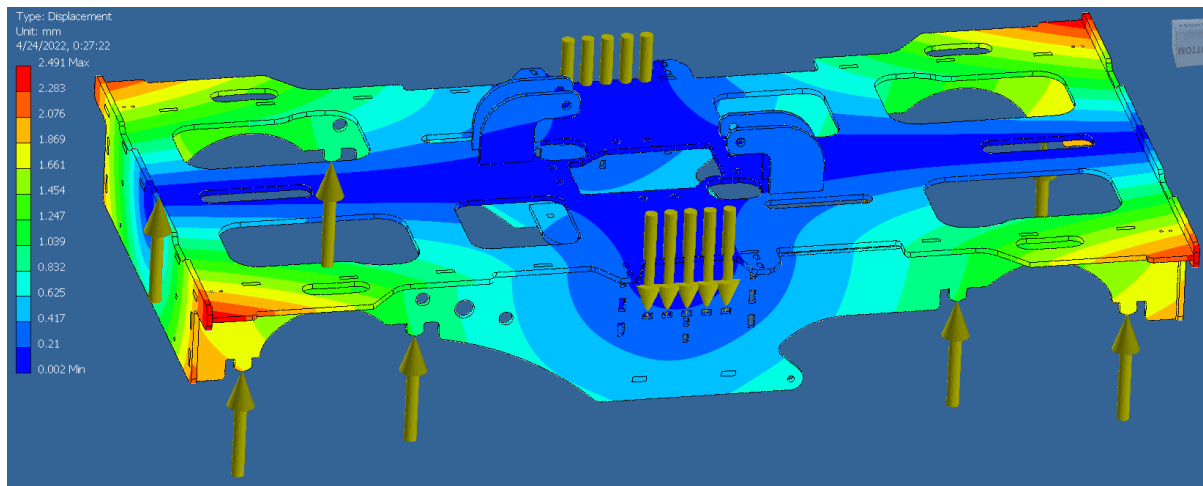


Figure 5-15: Bogie frame under twist load showing deformation

Future improvements

- Remove the slots and constrain flat faces to each other
- Model longitudinal and shear stresses coming from the traction forces and vehicle turning
- Fatigue testing – the structure has a lot of welds which makes it a subject to fatigue failure. Calculations for required weld is recommended

5.3 Secondary Suspension & anti-rollbar linkage (LK)

5.3.1 Secondary Suspension (LK)

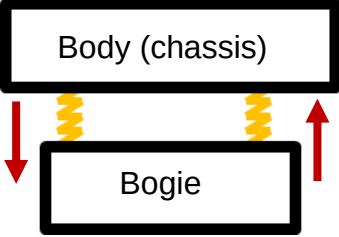
The secondary suspension is a system that connects the bogie frame to the body of the locomotive; as well as provide stiffness and damping force to the vertical, lateral, yaw and roll motions of the vehicle body. (4)

The 2021 Railway team elected to use coil springs as the secondary suspension. This design was complementary to the one applied in the 3rd year secondary suspension design, this can be observed in **Figure 5-1** and **Figure 5-2**. There was not much modification or change required for the secondary suspension other than determining the best coil spring for the locomotive and the connection from suspension to chassis.

5.3.1.1 Secondary suspension requirements & specifications

The main requirement for the secondary suspension is to be able to contribute towards the comfortability of the ride, as well as help lessen the noise from the wheelset and motors.

Table 5-2: Secondary suspension technical specifications (12)

Vertical (up & down) displacement	The secondary springs must be able to displace in an upwards and downwards motion under the chassis weight.  <i>Figure 5-16: Vertical movement</i>	When the locomotive is in stationary motion there needs to be a 60mm clearance between the body and the bogie.
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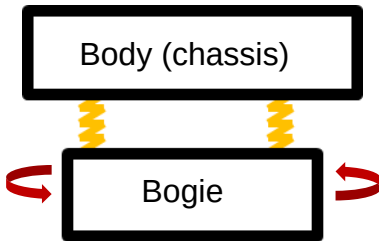
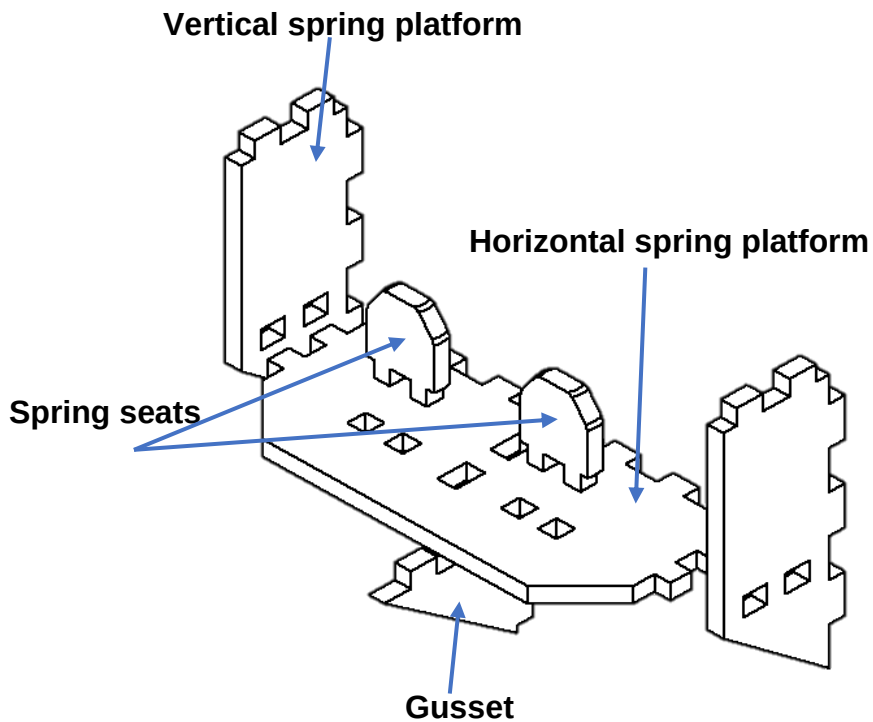
Lateral (side-to-side) displacement	The secondary springs must be able to displace sideways when the locomotive is in cornering/twisting motion.  <i>Figure 5-17: Lateral movement</i>	The secondary springs' stiffness needs to displace laterally during cornering/twisting to allow the locomotive to yaw.
Spring buckling	The secondary springs needs to handle extreme loads transmitted from chassis and must not buckle.	Failure of this will lead to suspension failure of the locomotive and derailment/toppling over of the locomotive.
Spring strength	The secondary springs must be able to handle the entire chassis weight.	Failure of this will cause the suspension to not function effectively.
Spring yaw clearance	There must be freedom for the secondary springs to perform displacements movements.	If the springs are blocked the locomotive will not be able to yaw and there will not be effective steering on the bogie system.

Table 5-2 are the technical specifications that governed the design/modifications of the secondary suspension.

5.3.1.2 Secondary suspension design

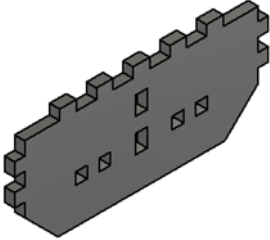
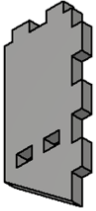
Most of the design work for the secondary suspension was done in 3rd year, and it helped greatly that both 2021 Railway team and the 3rd year secondary suspension design had some similarities.

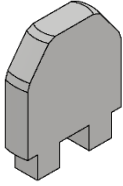


The 2021 Railway team design was preferred as it offered a better support for the springs. This design used spring seats in addition to having spring inner jackets to help support the secondary springs better and avoid buckling.

Figure 5-18: Secondary suspension bogie support

Table 5-3: design justifications

Part	Image	Design justification
Horizontal spring platform	 Size: 6mm S275 mild steel plate	The horizontal platform was designed to be able to carry the secondary springs as well as handle the motion/movements that are the suspension is subjected to.
Vertical spring platform	 Size: 6mm S275 mild steel plate	The vertical platforms were used to help ensure that in case of derailment/hard breaking
Gusset	 Size: 6mm S275 mild steel plate	These were used to help secure the horizontal spring platform, as the platform has to carry the shear force transmitted by the secondary suspension.

Spring seat		Size: 6mm S275 mild steel plate	These were used to help limit the longitudinal movement of the secondary suspension.
Future work/comments	The secondary suspension support design that was developed by the 2021 Railway team was more admirable as it constraints the longitudinal and vertical movements of the suspension whilst providing it freedom to manoeuvre as designed. The jigsaw-like slots provided ease of assemble given that a tolerance of $\pm 1\text{mm}$ if applied to account for weld.		

5.3.1.3 Secondary suspension spring analysis

To analyse the secondary suspension spring, the stiffness, displacement, buckling, and height calculations were required to help select the best coil spring for the locomotive. FEA analysis of the spring movement and forces carried was also simulated to help determine the safety and quality of the selected spring.

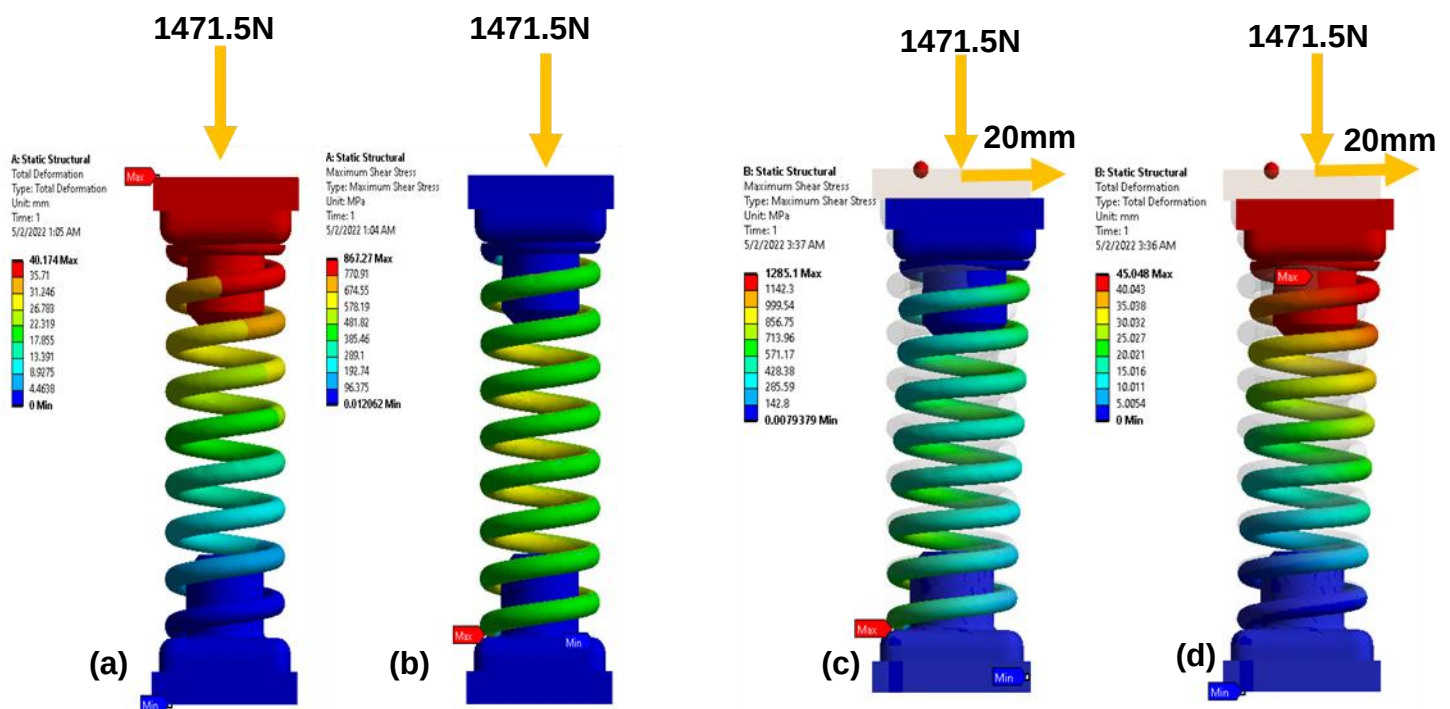


Figure 5-19: (a) Total vertical deformation (b) Maximum vertical shear stress (c) Maximum lateral shear stress (d) Total lateral deformation

Hand calculations were performed to determine the vertical and lateral forces acting on the secondary suspension, and FEA analysis was undertaken through ANSYS to verify the design of the suspension.

The forces highlighted on *Figure 5-19* are the ones for the worst-case scenario (for which the assumption taken was the curvature grade from the exhibition park). The reason behind using the worst-case scenario forces is to ensure that the secondary suspension will not fail under any condition unless when subject to a force higher than 2g that of the chassis weight.

5.3.1.3.1 Secondary suspension lateral displacement

Lateral displacement calculations were performed to determine the clearance that the springs required for its movements.

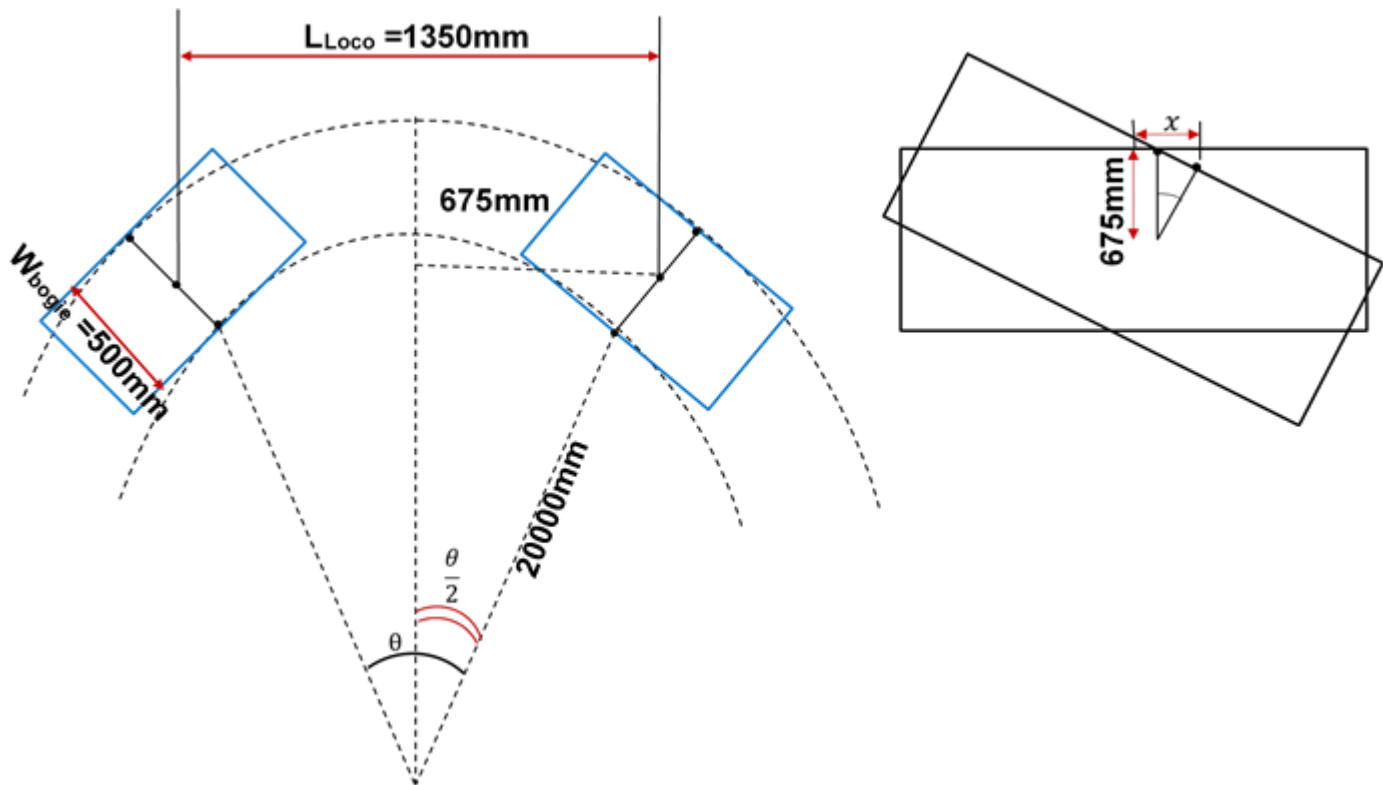


Figure 5-20: Lateral movement of bogie at 20m bend

Required data for lateral displacement calculations:

- Curvature radius for Stapleford (r): $20\text{m} = 20000\text{mm}$
- Bogie width ($W_{\text{sec.suspension}}$): $500\text{mm} = 0.5\text{m}$
- Length of half locomotive: $L_{\text{Loco}}/2 = 675\text{mm} = 0.675\text{m}$
- Using sine inverse:

Equation

$$\theta_{20m} = \sin^{-1}\left(\frac{0.675}{20}\right) \quad (5 - 1)$$

= **1.93°** this is the yaw angle for Stapleford, the lateral displacement will then be:

Equation

$$x_{1.93} = \tan(1.93) \times \frac{W_{sec.suspension}}{2} \quad (5 - 2)$$

= **8.424mm** where x = the distance required for cornering at 20m radius.

For worst case scenario, the radius of curvature taken was the one for the exhibition part which is $r_{exhibition} = 8m = 8000mm$

$\theta_{8m} = \underline{4.84^\circ}$ and the distance will be $x_{4.84} = \underline{21.169mm}$

A distance of $\pm 15mm$ was adapted at the top and bottom of the secondary suspension to allow the lateral displacement of the springs due to the use of bump stops for Stapleford. This was agreed to be the safest clearance that will allow the yaw displacement of the bogie and chassis.

The distance of $\pm 21mm$ was the worst-case scenario that will happen at exhibition park without having any bump-stops applied. 4 bump stops were applied to help limit the lateral displacement of the secondary suspension during cornering.

5.3.2 Secondary suspension spring buckling

To determine whether the springs will buckle under extreme loads, the slenderness ratio equation was calculated for both springs (the 2021 Railway Team spring and the spring chosen from 3rd year design).

Table 5-4: Slenderness ratio calculation

<u>Slenderness ratio equation 3rd year spring LHL 1500A 11 [Lee Spring]</u>	<u>Slenderness ratio equation 2021 Railway Team spring: LHL 2000A 06 [Lee Spring]</u>
Equation 1	Equation 2
$d_m = d_o - d_w \quad (5 - 3)$	$d_m = d_o - d_w \quad (5 - 3)$
Mean diameter ($d_{m(1)}$) = 30.17mm	Mean diameter ($d_{m(2)}$) = 39.83mm
Outer diameter ($d_{o(1)}$): 35.43mm	Outer diameter ($d_{o(2)}$): 46.48mm
Wire diameter ($d_{w(1)}$): 5.26mm	Wire diameter ($d_{w(2)}$): 6.65mm
Free length of spring ($L_{f(1)}$): 203.2mm	Free length of spring ($L_{f(2)}$): 127mm
Equation 3	Equation 4
Slenderness ratio = $\frac{L_{f(1)}}{d_m(1)} \quad (5 - 4)$	Slenderness ratio = $\frac{L_{f(2)}}{d_m(2)} \quad (5 - 4)$
= <u>6.735</u>	= <u>3.189</u>

The 2021 Railway Team spring was the better option as the ratio is less than 4 whilst the one from 3rd year it was going to buckle.

Going forward if compression springs are the selected secondary suspension, it is good to note that when designing the springs on CAD it needs to be the one with the static load operating on it.

Another design idea that can be taken forward in the future is the use of three compression springs aligned together, this can be an option when trying to find ways to save room on the bogie.

5.3.2.1 Secondary suspension spring height

The spring height is the desired static length at which the mass of the chassis is being exerted downwards producing a force under gravity. This is when the chassis is sitting on-top of the secondary springs without the additional forces that occurs during operation.

The specification stated there needs to be a 60mm clearance between the chassis and the bogie. For final design, it was highlighted that it is ideal to have the distance between the top frame of the bogie and the bottom plate of chassis as 40mm.

This distance was desirable as it not only allowed the locomotive to operate and remain within the dynamic loading gauge during motion, but it was also the minimum distance that can be achieved without compromising the functionality of the secondary suspension springs.

In future work it is best to start the dynamic loading gauge at the start (using the previous design), as this will help to determine the spacing within the locomotive as well as help with calculation in later days as this can be used as a guideline.

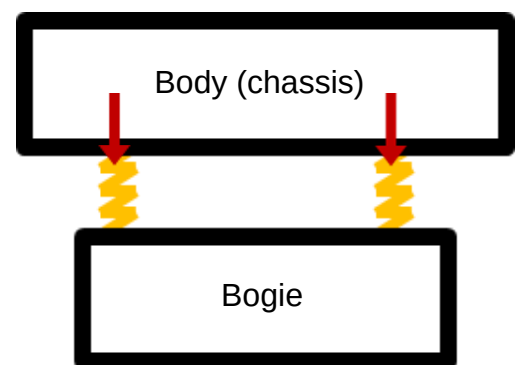


Figure 5-21: Chassis force acting on secondary springs

Table 5-5: Static length calculation

Static length equation 3rd year spring LHL 1500A 11 [Lee Spring]	Static length equation 2021 Railway Team spring: LHL 2000A 06 [Lee Spring]
Equation 4 Static length = $L_{f(1)} - (L_{f(1)} - L_{s(1)}) \times \frac{F}{F_{s(1)}} (5 - 5)$ = 154.04mm	Equation 4 Static length = $L_{f(2)} - (L_{f(2)} - L_{s(2)}) \times \frac{F}{F_{s(2)}} (5 - 5)$ = 105.55mm
Static load of the chassis per spring (F): $\frac{60}{100} \times \frac{600 \times 2 \times 9.81}{8} = 882.9N (5 - 6)$	
Solid length ($L_{s(1)}$): 96.01mm	Solid length ($L_{s(2)}$): 60.96mm
Free length of spring ($L_{f(1)}$): 203.2mm	Free length of spring ($L_{f(2)}$): 127mm
Load at solid length ($F_{s(1)}$): 1924.99N	Load at solid length ($F_{s(2)}$): 2717.86N

After all these calculations, it was preferred to use the 2021 Railway team secondary springs selection as they offered the optimal function for the suspension and allowed the locomotive to operate within the dynamic loading gauge.

5.3.2.2 Secondary suspension manufacturing & assembly

5.3.2.2.1 Manufacturing

In terms of manufacturing of secondary suspension, there was not much that required manufacture as the springs were to be bought in from Lee springs.

Table 5-6: Secondary suspension manufacturing plan

Part	Manufacturing plan	Quantity	Material	Price
LHL 2000A springs	Bought in	8	Steel, Mild	£119.68
Spring seats	Laser cut (Acorn)	16	Mild steel, S275	£70.96
Spring inner jackets	Machined (Newcastle university)	16	Acetal	£36.92
Spacers	3D printed (Newcastle university)	32		Free

5.3.2.2.2 Assembly

Table 5-7: Assembly plan for secondary suspension

Assembly order	How to assemble
1. Bottom plate spring seats	Welded to the horizontal spring platforms.
2. Bottom washers	Clearance fitted onto the bottom plate spring seats.
3. Bottom inner jackets	Interference fitted into the washers.
4. Springs	Clearance fitted outside the Bottom inner jackets.
5. Top inner jackets	Clearance fitted inside the springs (Top).
6. Top washers	Interferences fitted into the top inner jackets.
7. Top spring seats	Welded into the secondary suspension – to chassis connection plate and then inserted into the top inner jackets.

The assembly of the secondary suspension was dependent of the bogie frame assembly. The bogie was to be assembled first then the suspension can be assembled.

It was assembled on a Bottom-Top arrangement as the bottom spring's seats were welded together with the bogie frame.

5.3.2.2.3 Future work

For the secondary suspension future work that needs to be done is determining the best way to ensure that the springs do not buckle. The current use of spring seats is ideal for the time being, but it provides limitations on the type of springs that need to be used.

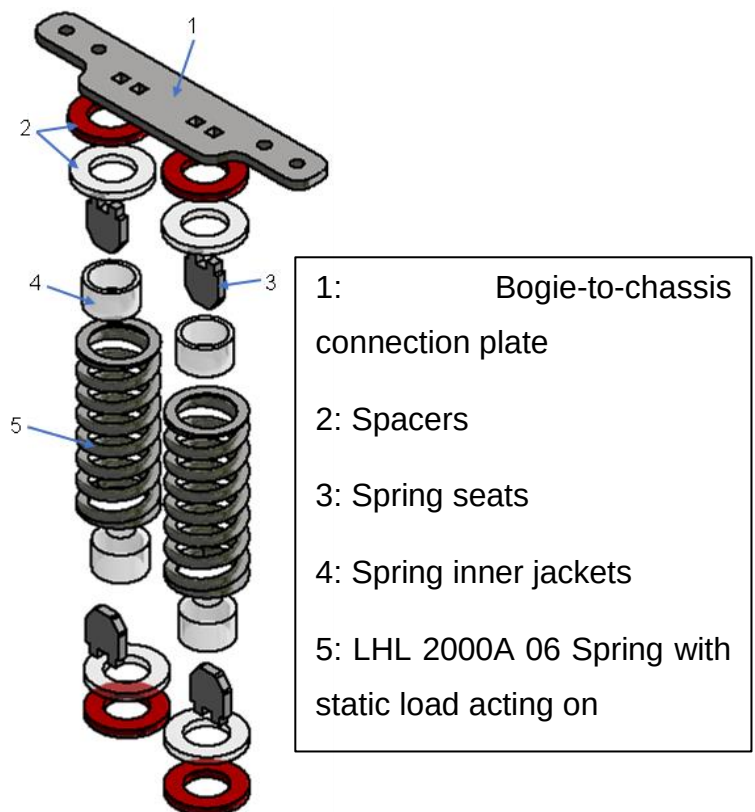


Figure 5-22: Secondary suspension assembly

It is difficult to replace the bogie-chassis connection plate as well as the horizontal and vertical platforms due to them having weldment, meaning that if the spring design changes then the bogie frame design must change as well as the connection.

5.3.3 Anti-rollbar linkage (LK)

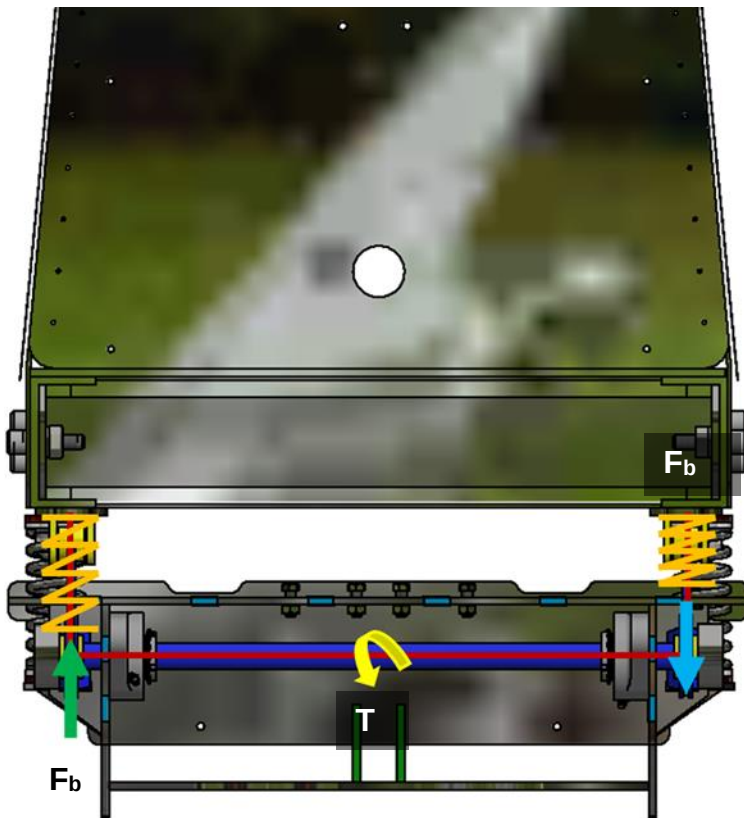


Figure 5-23: FBD of anti-rollbar

Anti-roll bar linkage helps to limit the body roll angle of the locomotive, it's aim is to prevent derailment when the locomotive is curving/twisting during operation.

When the locomotive is manoeuvring through bends, twists, and curves a torque force is produced causing the chassis of the locomotive to lean/tip onto one side due to change of direction, this also changes the velocity and momentum of the locomotive.

The stiffness of the anti-rollbar will cause the induced torque to be transmitted along the length of the

torsion bar onto the other side of the anti-rollbar and will be met with the other lever/arm that connects to the side of chassis that is leaning downwards in relation to the bogie, the transmitted torque will be induced to this side of the chassis causing an upward force that will try to equalize both sides of the chassis.

5.3.3.1 Anti-rollbar specification

Table 5-8: Anti-rollbar technical specification (12)

Anti-rollbar - chassis roll	Anti-rollbar must be able to limit the body roll that the chassis experiences.	This is to prevent the derailment of the locomotive when manoeuvring through curves/twists.
Anti-rollbar width	Anti-rollbar must not be wider than the chassis.	This is to help lessen the stresses acting on the chassis and anti-rollbar connection.
Anti-rollbar disassembly	The entire anti-rollbar assembly must be able to disassemble without damaging of other parts of the locomotive.	This is for ease of maintenance.

5.3.3.2 Anti-rollbar analysis

There was not much design submitted for the anti-rollbar in 3rd year, so the 2021 Railway team anti-rollbar design was taken forward for modifications.

The design did not require many changes apart from performing analysis and finding way to improve the performance as well as ensuring the design fit with the specification of the challenge.

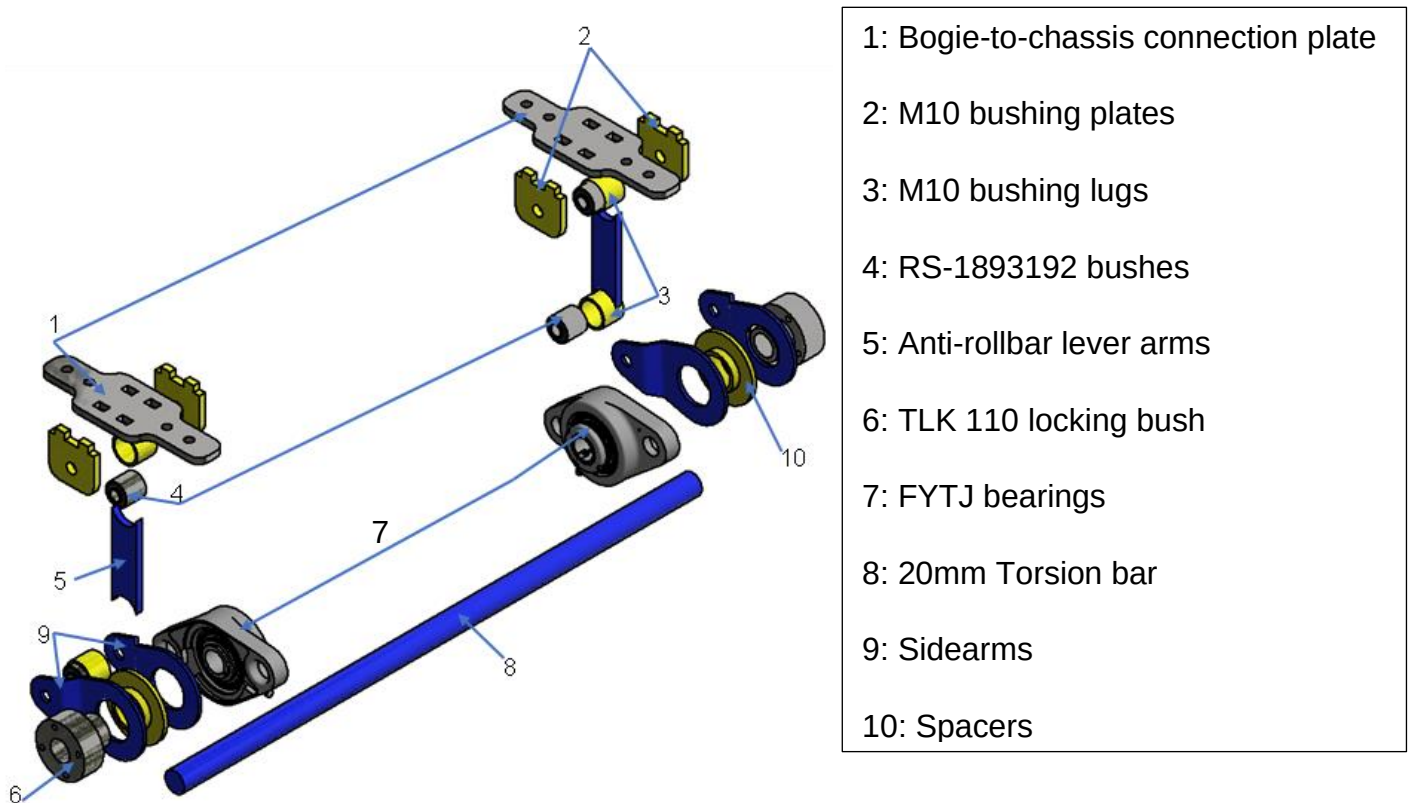


Figure 5-24: Anti-rollbar assembly & parts name

During first locomotive CAD assembly it was noted that the side arms of the anti-rollbar were longer than the space available, as well as the lever arms being longer than the 40mm distance between the bogie top frame and the chassis bottom.

Hand-calculations were made to obtain a suitable length of side arms and lever arm without causing the anti-rollbar to fail, and it was decided that the anti-rollbar arms were to be halved from the original length of 120mm to 60mm.

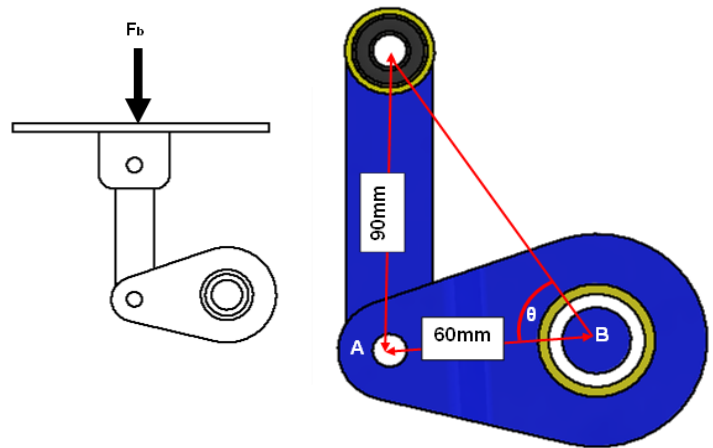


Figure 5-25: Anti-rollbar Torque & stress deflection sketch

Table 5-9: Anti-rollbar vertical displacement calculation

Force exerted by chassis (F_b)	$\frac{600kg}{4} \times 2g = 2943N$ (5 – 7)
Polar moment of inertia (J)	$\frac{\pi D^4}{32} = \frac{\pi \times 0.02^4}{32} = 1.5708 \times 10^{-8}m^4$ (5 – 8)
Torque per angular deflection ($\frac{T}{\theta}$)	$\frac{GJ}{L} = \frac{(80 \times 10^9) \times (1.5708 \times 10^{-8})}{0.534m} = \frac{2353Nm}{rad}$ (5 – 9)
Torque (T)	$F_b \times 0.06 = 2943 \times 0.06 = 176.58Nm$ (5 – 10)
Angular deflection (θ)	$\frac{176.58}{2353} = 0.075rads$ (5 – 11)
Vertical displacement (x)	$60 \times \tan \theta = 60 \times \tan 0.075 = 4.508mm$ (5 – 12)

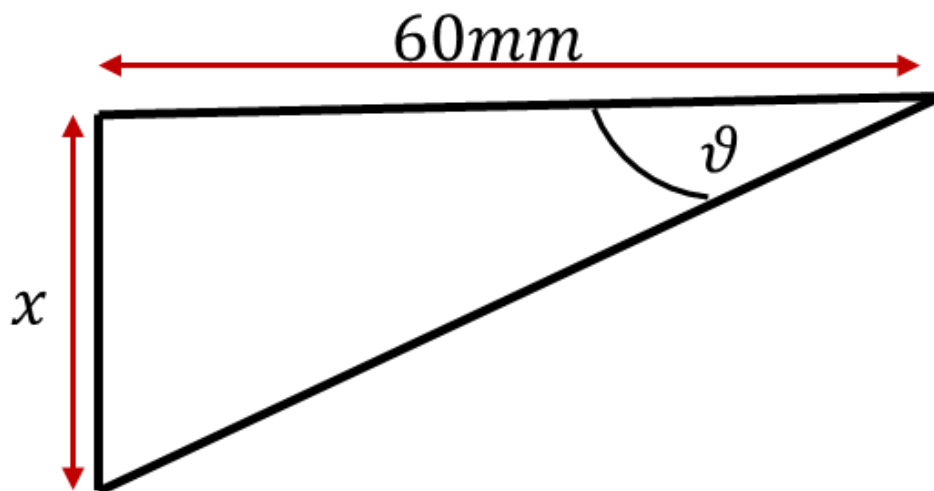


Figure 5-26: Anti-rollbar vertical displacement sketch

5.3.2.3 Anti-roll Bar Calculations (DS)

The anti-roll bar's S355 mild steel torsion bar of 20mm diameter (5) must withstand a torque of 78.2 Nm. This torque is the result of the 1304 N centripetal exerted by the body when the locomotive travels at 15km/h (4.17 m/s) around an 8m curve. The previous torsion rod was hollow with a 10 mm internal diameter whereas the new design is solid. The shear stress can be calculated using:

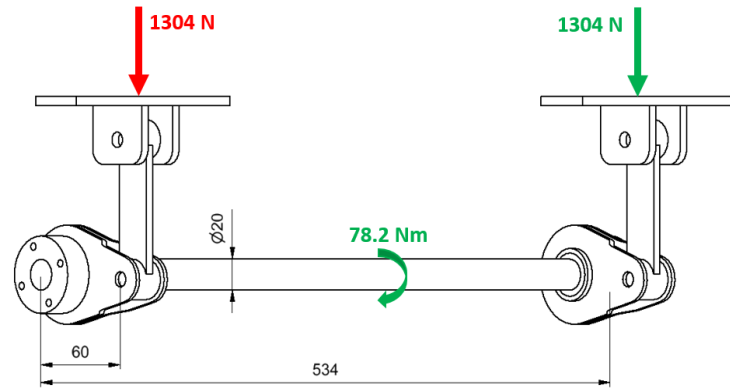


Figure 5-27: Anti-rollbar Torque & stress deflection sketch

$$\tau_t = \frac{16T}{\pi D^3} \quad (5 - 13)$$

Table 5-10. Safety Factors and torsional stiffness of Anti-Roll Bar

Design	Torsional Stiffness (Nm/rad)	Permissible Shear Stress (MPa)	Applied Shear Stress (MPa)	Safety Factor
Previous	2150	177	65.5	2.70
Current	2827		49.8	3.56

Calculations for the torsional stiffness and the factor of safety can be found in appendix H-7.

5.3.2.4 Manufacturing & assembly (LK)

The manufacturing process for the anti-rollbar assembly mainly consisted of laser-cutting the sidearms, the lever arm, the bush brackets as well as the anti-rollbar-to-chassis connection and then performing a weldment on them.

Parts such as the TLK-110 locking bush and the 1893192 bushing, and the FYTJ bearings were bought in from RS-components and Simply bearings respectively. The torsion bars were purchased from Metal supermarkets, and the spacers and bushing lugs were manufactured in-house at the University.

The plan was to weld the 2 sidearms with the spacers, as well as weld the lever arms with the bushing lugs then attach them all together using the torsion bar, that will pass through the bogie widths from one end to the other; Fig 5-24 shows the explosion view of how they all sit together.

5.4 Brakes (JJ)

5.4.1 Requirements and specifications

1. The locomotive must have an exclusive mechanical emergency brake which is applied by the driver
2. The mechanical brake must be a failsafe in case of a system malfunction
3. The emergency brake will operate whenever communication with the remote-control unit is lost, regardless of the current locomotive operating state, within 0.5 seconds of disconnection.
4. Emergency brake shall feature manual release and reapplication feature
5. No single fault shall reduce braking by more than 50%
6. Brakes should be able to have a deceleration of 1.15-1.3 m/s²
7. The locomotive must have an emergency brake
8. Brakes must allow to regulate speed downhill and help at the hill start
9. Brake design must allow to complete maintenance challenge

5.4.2 Concept design

Four possible concepts to operate the brakes have been considered.

Table 5-11: Four possible brake operating concepts considered. Air operated brakes proved to be the best choice

Vacuum operated brakes		Air operated brakes		Hydraulic brakes		Electrodynamic braking	
<i>Advantages:</i>	<i>Disadvantages:</i>	<i>Advantages:</i>	<i>Disadvantages:</i>	<i>Advantages:</i>	<i>Disadvantages:</i>	<i>Advantages:</i>	<i>Disadvantages:</i>
Simple	Large and slow	Simple and easy	Compressor and air reservoir is required	Consumes less power than air brakes	Leaks can get onto braking surfaces and suddenly reduce braking performance.	Power can be regenerated for later use or fed to the overhead wire – more sustainable and economical	Ineffective at low speeds
Pressure reservoir not required	Brake blending with electrodynamic braking would not be possible because of 1 to 0 output	Useful for other train systems such as doors, suspension	Consumes quite a bit of power	Small brake fluid reservoirs and master cylinder is smaller than air compressor	Brake fluid needs servicing and system needs bleeding from air bubbles	No high wearing components – easier maintenance, longer service life, more economical	Complex – microprocessors required to control the braking
	Low air pressure means large brake cylinders which would create fitting options	Air leaks are easy to find	High pressure required appropriate fittings and lines		Very high pressure in the system – requires appropriate handling and parts		Train must have electric generator/motor – expensive
	Difficult to find leaks	High pressure means compact brake cylinders			Leaks can damage other components		

From all the information listed above, it was decided to use electrodynamic braking blended with mechanical air-operated brakes. Electrodynamic brakes were chosen because the train has electric motors which can act as generators so there are no costs added to the system. Also, regenerating energy is one of the IMechE railway challenges, and using generators to convert braking energy to electrical energy is the most efficient and cost-effective way. Electrodynamic braking is combined with a mechanical air brake to solve the low-speed issue. The mechanical air brake would work as an emergency, parking and failsafe brake. The disadvantages of vacuum and hydraulic brakes seemed higher than air brakes. Also, investment has been already made by the previous year to buy an expensive air compressor and electro-pneumatic regulator.

Four possibilities were considered for the mounting of the mechanical brake

Table 5-12: Four possible brake mounting concepts considered. Motor mounted brakes proved to be the best

Wheel mounted brake discs		Wheel thread brakes		Axle mounted		Motor mounted	
Advantages	Disadvantages	Advantages	Disadvantages	Advantages	Disadvantages	Advantages	Disadvantages
Simple	Brake disc must be easily replaceable without the wheel being decommissioned	Compact and simple – used on motor bogies	Can damage the wheels, especially at high speeds	Multiple brakes can be mounted on one axle	Takes a lot of space and is usually used on trailer bogies	Due to gearing rotates faster which results in lower force required from braking	Load is carried by the drive system – must be considered during design
Compact – used on motor bogies		Cleans the wheel	Heats the wheels and makes them expand - design must be adjusted	Does not have the same problems as wheel mounted brakes	Harder to do maintenance	More compact than axle mounted brakes	Hard to maintain compared to wheel mounted as the access to the brakes is limited

Motor-mounted brakes have been chosen because this design is more compact than axle-mounted brakes due to the lower force required to produce the same deceleration. Wheel thread brakes had significant disadvantages and wheel-mounted brake discs are more complex than motor mounted. Also, wheels have been inherited from previous years without the required design considerations for wheel-mounted brake discs.

For mechanical air brakes, there were three options

- Brake calliper acting on the brake disc
- Brake shoe acting on a brake disc
- Brake shoe actuating into a vee pulley

The third option was chosen because it is compact. The Vee pulley and brake shoe is the smallest assembly. The vee pulley increases the braking force applied due to the 17-degree angle. Increased braking performance allows the design to use smaller parts such as a pneumatic cylinder and spring due to lower forces. This design was considered by previous years which meant that no fundamental changes were necessary.

5.4.3 Layout design

The inherited design had come to the same conclusion. However, the design had numerous design flaws (*Figure 5-26*):

1. Poor brake shoe design
2. Poor pneumatic cylinder mounting bracket design
3. Poor brake shoe anti-rotation bracket design
4. Impossible to remove the brake shoe for the maintenance challenge
5. Calculations required for rod, spring and braking force
6. Difficult manufacturing

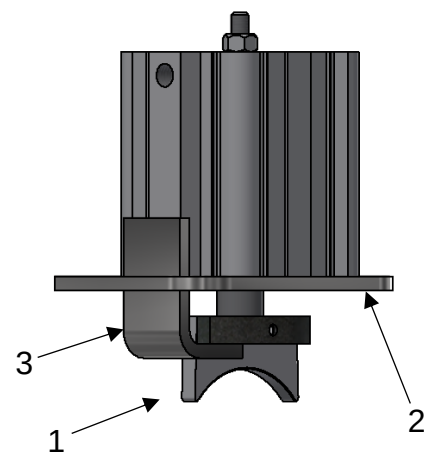


Figure 5-28: Brake design inherited from 2020-2021 locomotive team

To begin with, a new brake shoe was designed to maximise the contact area with the pulley and allow 5mm of wear before the replacement brake shoe is needed (*Figure 5-27*).

Design must ensure that only angled surfaces are touching the brake shoe and there is clearance in the centre. Also, with some simple changes, the design of the mounting bracket has been adjusted for easier welding to the motor plate.

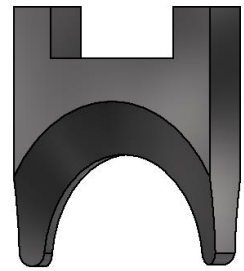


Figure 5-29: New brake shoe design

Then multiple solutions have been suggested for better brake shoe mounting to help with the removal of the brake shoe for the maintenance challenge.

Table 5-13: Brake shoe mounting concepts considered

Solution 1	Solution 2	Solution 3
The design used 2 bolts with a tapped hole on one of the brackets. The whole assembly slides down from the piston and can be rotated in the vee pulley to be fully removed. Parts could be laser cut. One of the brackets has an extension to prevent the brake shoe from rotating under braking. The design is ensured to work in the fully worn-out specification.	This design utilized a pin with wire to secure the brake shoe in place. The pin goes through the brake shoe and the piston. The pin is easier to remove than grub screws. All parts could be laser cut. However, it might be difficult to drill a hole in a 20mm piston and it could be unsafe. Pin wire could also cause some problems by getting stuck in unwanted places. The brake can still rotate.	Solution 3 utilizes a custom flange that can be secured with a pin to the piston or welded directly. If the flange is welded to the piston, then bolts must be undone to remove the brake shoe for maintenance challenge. If it is secured with a pin, then the whole assembly drops as the pin is removed. However, it is difficult to manufacture the flange, calculations are needed for pin strength and the assembly might not be stable under braking. Limited space is available to drill the holes into the brake shoe vertically.

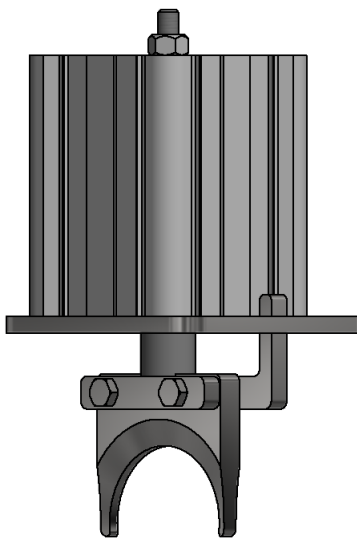


Figure 5-30: Brake shoe solution 1

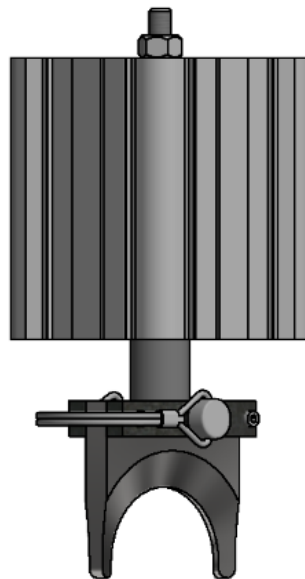


Figure 5-31: Brake shoe solution 2

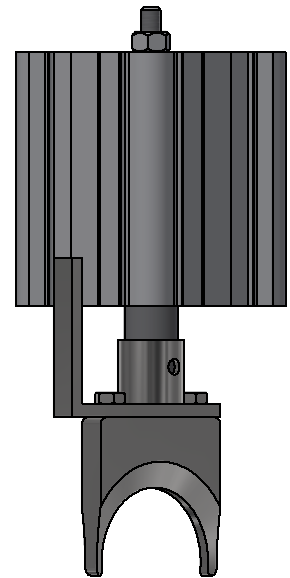


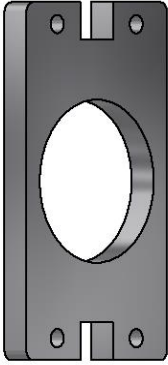
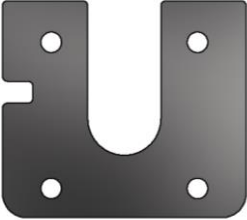
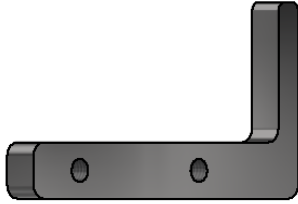
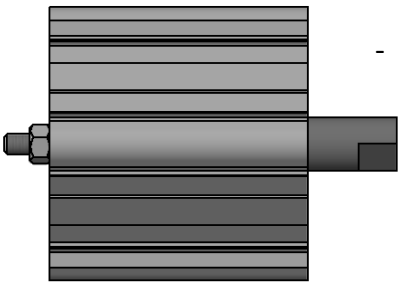
Figure 5-32: Brake shoe solution 3

Other ideas:

- Use a coupling to connect the piston and custom brake shoe
- Thread the piston

Solution 1 was pursued and further developed (*Figure 5-30*). The reasons were – that it was the easiest to quickly disassemble, the solution prevents brake shoe rotation, The design has been adjusted for manufacturing and ease of assembling.

5.4.4 Final design

Part	Manufacturing process and description
 - Brake shoe	• Laser cutting • Turning • Bandsaw cutting The profile gets laser cut. Then the brake shoe is put into the lathe and the inner hole get turned to have a tapered 17degree surface. The part gets cut in half to give two brake shoes. In total 6 brake shoes were made to have 2 spares The mild steel brake shoe is in contact with cast iron vee pulley to generate braking due to friction
 - Pneumatic cylinder support	• Laser cutting The profile is laser cut. The side cut for the extended brake shoe mount has been opened to avoid clearance issues and to make the assembly easier. This part is used to bolt the pneumatic cylinder and it is welded to the motor bracket
 - Spring seat	• The spring seat has been 3D printed • Specs: Cura Multimaker 3, PLA plastic, 40% fill and 0.1mm nozzle The spring seat prevents movement of the spring in the pneumatic cylinder, ensures good positioning during preloading and assembling of the brakes.
 - Brake shoe	• Laser cutting • Drilling This part secures the brake shoe to the piston rod
 - Brake shoe mount	• Laser cut • Drilling • Tapping The part secures the brake shoe to the piston rod and prevents the brake shoe from rotating under braking. The holes are tapped instead of using a nut due to limited space
 - Pneumatic cylinder	• Bought from RS components and modified • Drilling • Tapping • Milling The pneumatic cylinder actuates the piston with the brake shoe using compressed air. When air is supplied the spring inside is compressed and the brake is released.

5.4.4.1 Spring

LHL 1250B 02 spring from LeeSpings has been purchased with the following specifications:

Table 5-14: Spring specification

Part Number	LHL 1250B 02
Series	HEFTY Die Spring
Outside diameter (mm)	30.23
Hole diameter (mm)	31.75
Rod diameter (mm)	15.88
Free length (mm)	44.45
Rate (N/mm)	105.08
Solid height (mm)	27.43
Wire diameter (mm)	4.95
Material	Oil Tempered Chrome Silicon
Load at solid height	1788.19

The spring must provide enough force to stop the locomotive in case of power and air malfunction without the help of electric motors. The calculations were done for worst case scenario which occurs at Exhibition Park:

Maximum tractive effort

$$a = 3.63 \frac{m}{s^2} = 0.37g \quad (5-14)$$

Assume that the locomotive weights 1000kg, then total braking force

$$F = m * a = 1000 * 3.63 = 3630 N \quad (5-15)$$

There are 4 wheelsets with a brake each, therefore force per wheelset

$$F_w = \frac{3630}{4} = 907.5 N \quad (5-16)$$

The wheel diameter is 250mm. The force converted to torque is

$$T_w = F * r = 907.5 * 0.125 = 113.5 Nm \quad (5-17)$$

Assuming 100% belt drive efficiency and with a gear ratio of 22:80, then the torque on the motor shaft

$$T_m = T_w * \frac{N_1}{N_2} = 113.5 * \frac{22}{80} = 31.2 Nm \quad (5-18)$$

Assuming that the torque is applied at the centre of the pulley, then the tangential force at the brake shoe

$$F = \frac{T_m}{d} = \frac{31.2}{0.0325} = 960 N \quad (5-19)$$

The coefficient of friction between brake shoe (mild steel) and vee pulley (cast iron) = 0.23 (6)

$$F_{axial} = \frac{960}{0.23} = 4174 N \quad (5-20)$$

The shape of the vee pulley increases the applied force (Figure 5.31)

$$F_{applied} = \frac{F_{axial} * \tan(17)}{2} = \frac{4174 * 0.306}{2} = 639 N \quad (5-21)$$

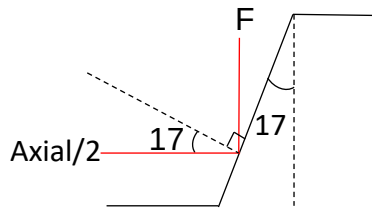
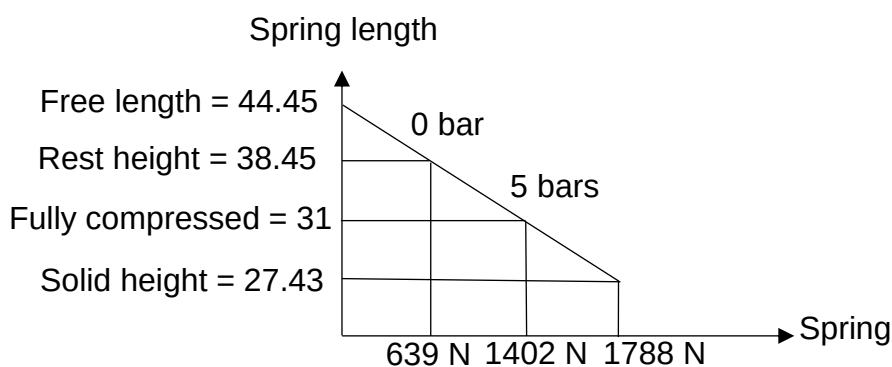


Figure 5-33: Vee pulley applied force amplification

For the system to be failsafe in case of system failure – loss of pressure or electrical problem, the spring must apply 639 N to the brake shoe to stop the locomotive going at the maximum tractive effort.



*The force of the compressed spring. Not the force acting on the brake

Figure 5-34: Spring force over spring length graph. Indicates the operating range

5.4.4.2 Piston bending

During braking, piston would experience bending. Therefore, the piston was treated as a cantilever experiencing bending to ensure its safety.

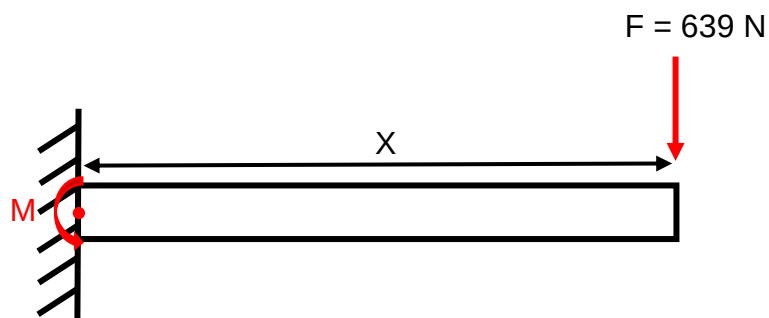


Figure 5-35: Brake piston loading free body diagram

Table 5-15: Brake piston loading parameters

Length L	55 mm
Diameter of the rod d	20 mm
Tangential force F	639 N
Yield strength of carbon steel (1008)	285 MPa (7)

$$\text{Second moment of area} \quad I = \frac{\pi * d^4}{64} = 7.854 * 10^{-9} m^4 \quad (5-22)$$

$$\text{Moment @ maximum length} \quad M = 639 * 0.055 = 35.145 Nm \quad (5-23)$$

$$\text{Maximum stress} \quad \sigma = \frac{M * y}{I} = \frac{35.145 * 0.01}{7.854 * 10^{-9}} = 44.75 MPa \quad (5-24)$$

The yield strength of carbon steel is much higher than 25.7 MPa. Therefore, the rod is safe.

5.4.4.3 Pneumatic ram

SMC pneumatic compact cylinders (CDQ2B63TF – 50 DZ) have been purchased. This cylinder has been purchased by previous years. However, it fits the purpose because it is compact, with correct stroke and force rating.

Some important specifications:

Table 5-16: SMC pneumatic cylinder specifications

Bore (mm)	63
Stroke (mm)	50
Max pressure (bar)	10
Dimensions (L x W x H in mm)	94 x 84 x 77
Area (m ²)	0.002803
Extra	Lubrication is not required and has a magnetic piston

The compressor can supply up to 5 bar pressures. Therefore, the maximum force exerted onto the piston would be:

$$F = P * A = 5 * 10^5 Pa * 0.002803 m^2 = 1402 N \quad (5-25)$$

The solid height force from the spring is higher than the maximum force from the piston. This means that the spring will never be fully compressed – up to 3.7 mm of compression left. This is good because a fully compressed spring would have metal to metal contact.

Modifications (*Figure 5-37*)

1. The rod has been cut to 53mm and the end face milled flat
2. M8 hole drilled in the centre for manual release bolt in case there is a malfunction with the air system

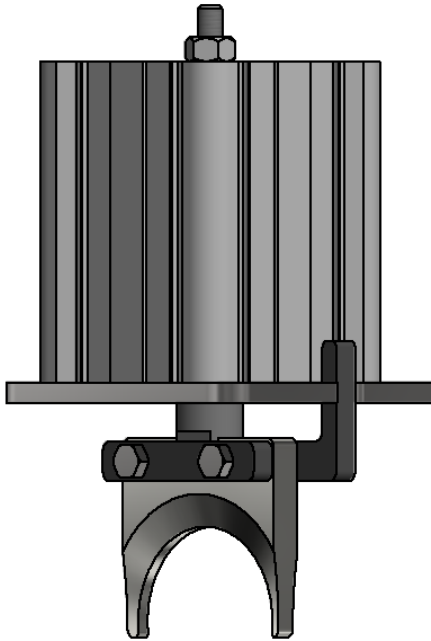


Figure 5-36: Final brake assembly design

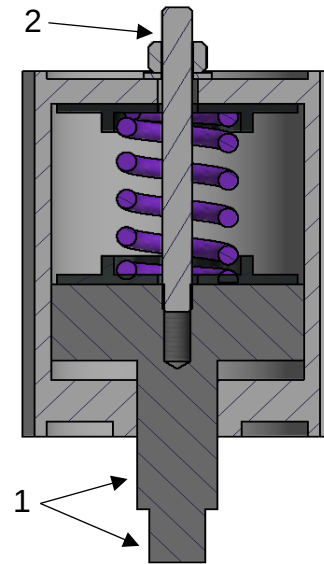


Figure 5-37: Cross-section of the final design with the modifications required

5.5 Traction Centre (TC) (JJ)

5.5.1 Requirements and specifications (DS)

The purpose of the traction centre is to transfer tractive effort and braking load from the wheelset through to the body. The traction centre should:

- Fully restrict body longitudinal motion
- Allow for 21 mm of body vertical displacement
- Allow for a free body lateral displacement of 11.5 mm and a damped maximum lateral displacement of 15 mm using hard and soft limits.
- Allow the body to yaw 4.82°
- Allow the bogie to pitch 0.682°
- Allow the body to roll 1.59°
- Have a clearance of 15 mm x 15 mm to allow for aforementioned freedoms
- Dampen yaw, pitch, and roll
- Be detachable from the body

5.5.2 Concept design (JJ)

The inherited design had flaws (*Figure 5-38*):

- The design was too weak to carry the load.
- The design was not centred
- The design was 44mm in the x-direction from the centre of the mass which affects the ride quality
- The design was 33 mm in the y-direction from centre of the mass which affects the ride quality
- Over-engineered support brackets (*Figure 5-38*)
- Bogie frame was limiting required movement x-movement
- The design was limiting the required z-movement
- Clearance issues with the motor
- Calculations required

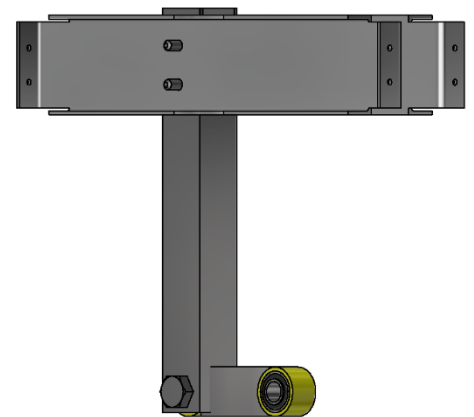


Figure 5-38: Inherited TC design from 2020-2021 locomotive team

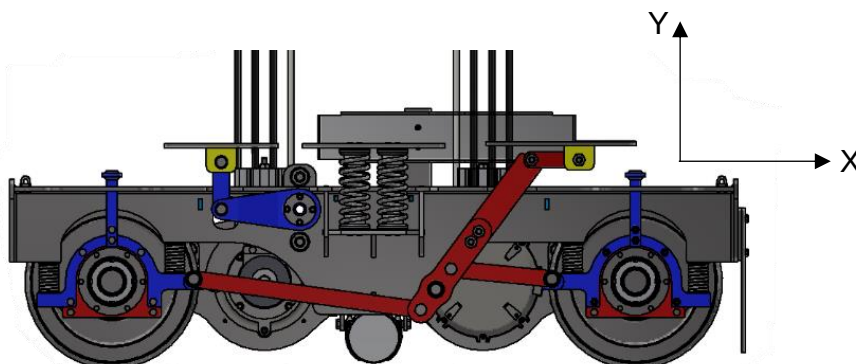


Figure 5-37: Bogie movement axis

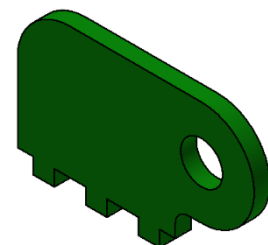


Figure 5-40: TC support bracket

Calculations have been performed to have a general idea of the design required

Mass of the bogie 200kg (5-26)

Maximum deceleration of the bogie relative to the body 10g (5-27)

Length of the TC 0.31 m (5-28)

Force acting on the TC $F = ma = 200 * 10g = 20000 \text{ N}$ (5-29)

Bending moment on TC at maximum length $BM = F * L = 20000 * 0.31 = 6200 \text{ Nm}$ (5-30)

Bending moment on TC at half length $BM = F * L = 20000 * 0.155 = 3100 \text{ Nm}$ (5-31)

Using the bending stress formula (5-24), it was determined that the tube must be at least 76mm in diameter and 5mm thick if the material is S335.

Three concepts have been considered for TC:

The first concept utilizes two round steel tubes welded together *Figure 5.40*. The tubes can be of different sizes because the moment decreases as the distance from load decreases *Equation (5-31)*. Therefore, using different size tubes can save weight and space. *Figure 5.41* shows how the assembly would be secured to the rectangle tube which is connected to the body. As the bolts are loosened, the whole assembly can slide out of the rectangle tube as the body is lifted of the bogie. This concept had a couple of flaws:

- Difficult to tighten the top bolts due to limited space
- Expensive aluminium billets
- Assembly is more than 40mm away from the centre of the mass in the x-direction

- Clearance issues with the motor and bogie frame

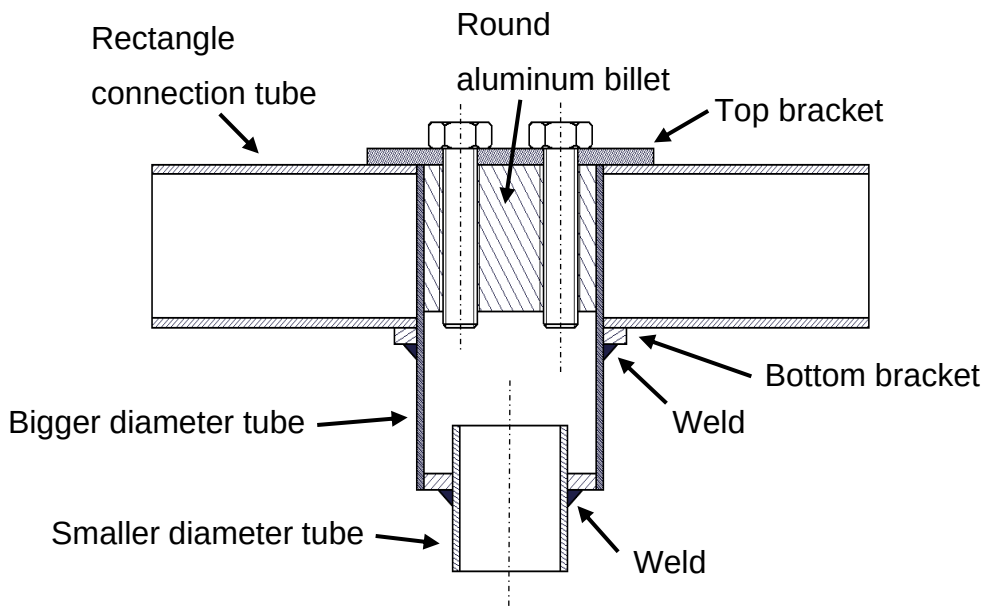


Figure 5.41: Cross-section of concept 1 for mounting to the connection tube

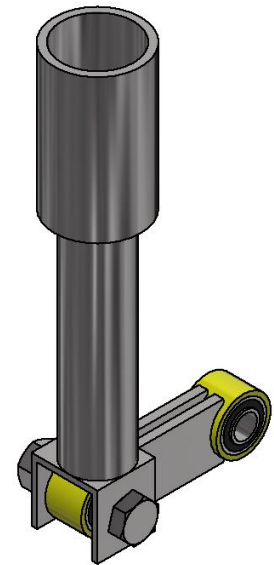


Figure 5.42: Concept 1 design

The concept has been altered to fix some of the flaws *Figure 5.44*. The design has been shifted to the centre of the mass. This provided more clearance for the motor mounts and the frame. However, traction centre bushing brackets had to be modified and TC sitting in the middle of the bushings has reduced the allowable movement. The weight of this design is just over 3kg. It would be difficult to manufacture such TC, especially connecting different size tubes.

The second concept was created trying to avoid the difficulty of the manufacturing the first concept. This design utilized a single round steel tube with the same mounting mechanism *Figure 5.44*. The design is about 300g heavier and has worse clearance with motor mounts. However, clearance issues can be solved and manufacturing of such a system is much more simple. The additional weight is negligible compared to the weight of the whole bogie.

The third concept uses laser cut, bent metal plates *Figure 5.44*. This concept is the easiest to manufacture and takes the same amount of space. Due to the shape of the design, the connection tube is mounted in the centre of mass to give good ride quality while keeping the solution mounted on top of the

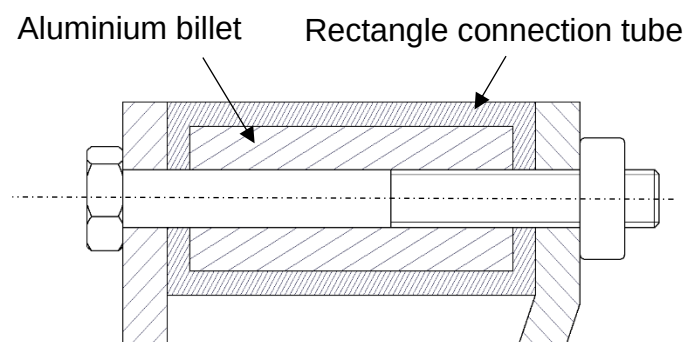


Figure 5.43: Cross-section of concept 3 for mounting to the connection tube

bushing to maximize allowed movement. However, this is the heaviest assembly – about 4.5 kg. The mounting of the assembly is quite difficult. The billet must be inserted into the tube to prevent tube crushing when the bolts are tightened and secured using epoxy. When the bolts are undone the whole assembly is ready to slide out when the body is lifted off the bogie.

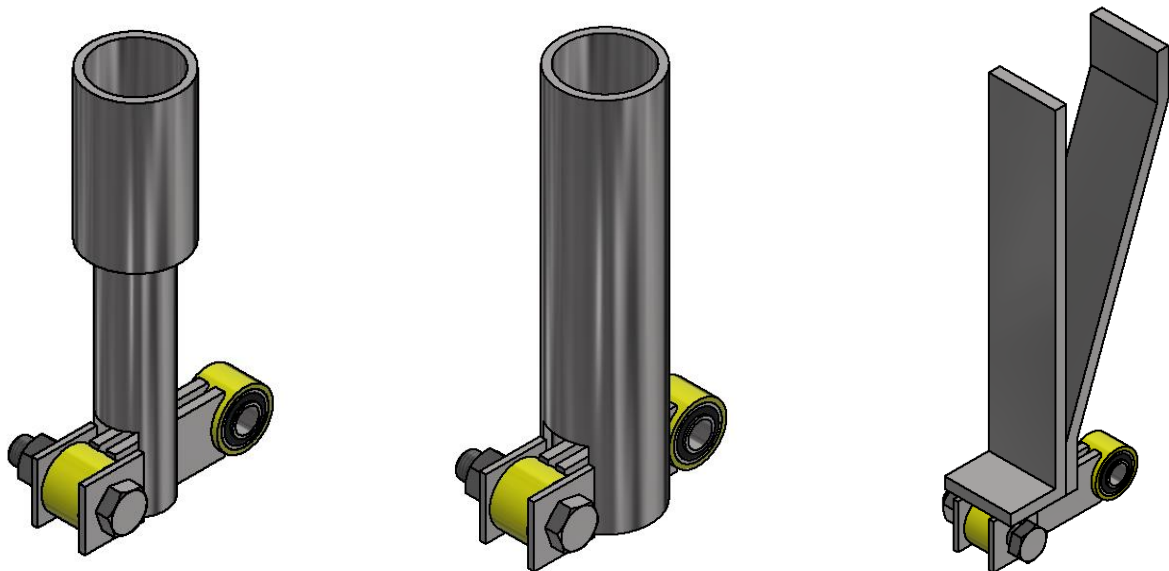


Figure 5.44: – fixed concept 1, concept 2 and concept 3 designs

After considering all three concepts – concept three has been chosen to progress. Ease of manufacturing and space occupied have been chosen as the most important criteria whereas mass and ease of assembly as the least important. The mass difference between designs becomes irrelevant in the scale of the whole locomotive and even if difficult the traction centre only needs be assemble once.

Table 5-17: Selection criteria table for traction centre

	Criteria					
Weighing criteria	5	1	4	3	2	
TC version	Ease of manufacturing	Mass	Space occupied	Cost	Ease of assembling	Total
1	1	3	2	2	1	24
2	2	2	1	3	3	31
3	3	1	3	1	3	37

*More points are better

5.5.3 Layout design (JJ)

Couple of solutions have been further investigated to prevent the crushing of the chassis connection tube while making the manufacturing and assembly easier. The first solution suggested using C-channel instead of the rectangle tube to connect to the chassis (*Figure 5.45*). This would give better access to the billet which would result in easier manufacturing. The channel would be supported with steel plates to prevent crushing. However, the channel is not as strong in bending as a beam. Therefore, a thicker metal must be used to have a safe design. Also, modifying the channel flanges reduces the strength of the channel significantly.

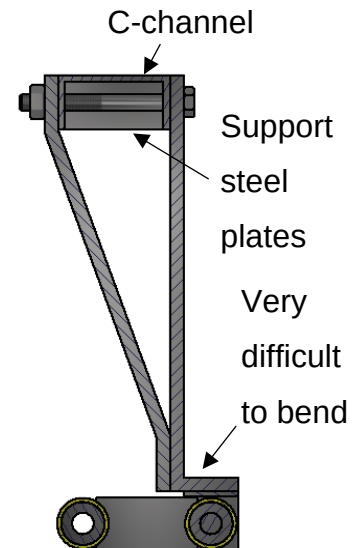


Figure 5.45: crushing prevention for the tube

It has been investigated using threaded inserts with steel tubes to replace the expensive and heavy billet. However, the steel tubes are difficult to assemble if a rectangle tube is used.

The second idea was to remove the TC with the rectangle tube connection all together. The TC plates would be permanently connected to the tube either by welding, rivets or hollo bolts by Lindapter. Rivets would require extra manufacturing time due to extra steps, hollo bolts have very limited space due to the small width of the traction centre plate. Therefore, welding seemed to be the most valid option since TC assembly is welded together it would be the easiest, quickest and cheapest. Removing TC with the rectangle tube connection imposed new problems as having L brackets or flanges at the end of the chassis connection meant that it was impossible to assemble.

Two solutions have been suggested how to tackle this problem (*Figure 5.44*):

1. To add blocks inside the body's C-channels to slide out the connection
2. Turn the body's C-channels the other way around

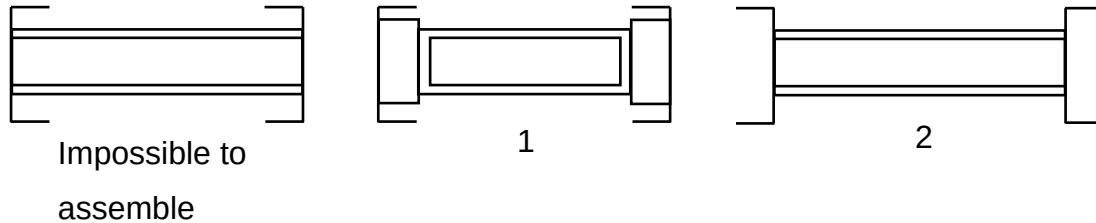
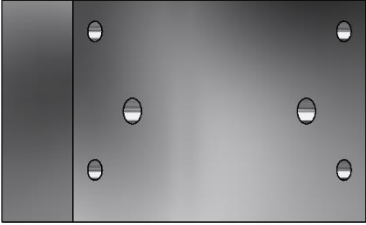
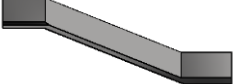
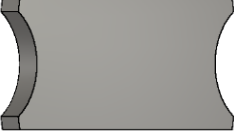


Figure 5.46: solutions to solve assembling problems with TC

The second option was not possible because the internal body walls attach to the outside of the c-channel. This change would require a redesign of the body which was too big of a change. Therefore, it was decided to go with solution 1. The design has been modified for ease of manufacturing and appropriate calculations have been carried out to ensure safety.

5.5.4 Final design (JJ)

Part	Manufacturing process and description
 - Straight plate	Laser cutting – 8 mm S275 steel The profile is laser cut and is carrying traction and braking force
 - Extension billet	- Aluminium 6082 T6 flat bar has been purchased - Faces milled to the correct dimensions - Holes drilled - Some holes tapped This extension billet sits in the body's c-channel and allows to disconnect the traction centre from the body
 - Connection	Laser cutting – 6 mm S275 steel The profile has been laser cut. Flange allows to connect TC to the body using 2 bolts.
 Chassis connection	- Bought in component 70x50x3 S275 steel
 - Anti-buckling	Laser cutting – 8 mm S275 steel Anti-buckling plate prevents the TC plates from buckling under load
 - Bent plate	- Laser cutting – 8 mm S275 steel - Bending The profile is laser cut and later bent. With the help of straight plate, this part carries the traction and braking force
 - Hook	Laser cutting – 8 mm S275 steel This part connects TC to the bushings which are attached to the bogie. It acts as a connection point and gives hard stop for vertical movement of the bogie
 - Bushing	Laser cutting – 6 mm S275 steel This part connects the TC bushings
 - Bushing	- Bought in component from Fibet - Part name: FBNA1230.3434 - Specification details in appendix H-1
 - Bushing lug	- Bought in component from metal supermarket - The lugs have been cut and bored to have an interference fit

In the final design, the TC has been raised by 33 mm to have TC bushing in the same plane as the centre of the mass. This improved the ride quality and reduced the bending moment.

L brackets and laser cut flange have been considered to attach chassis connection to the block. Flange proved to be easier to manufacture and weld. During assembly, it only requires 2 bolts to undo one side of the connection.

The bushings were kept around 100mm apart to have good ride quality. It was considered to use a tube to connect the bushings. However, it is harder to manufacture than laser-cut plates.

Bump stops have been introduced to have soft limits during the pitch and roll of the bogie. This prevents metal-to-metal contact in extreme situations.

90-degree bend on the straight plate was impossible to manufacture and it was redesigned along with the introduction of the hook for y-direction movement hard limit.

5.5.4.1 Finite element analysis (FEA) of the TC plates (JJ)

FEA analysis has been performed to optimize the design. It has been discovered that 35mm wide and 8mm thick plates works the best. Red colour in the pictures indicates yielding as the stress exceeds 275 MPa. However, as seen in *Figure 5.47* the colour is not all the way through the material which would indicate failure. This yielding is

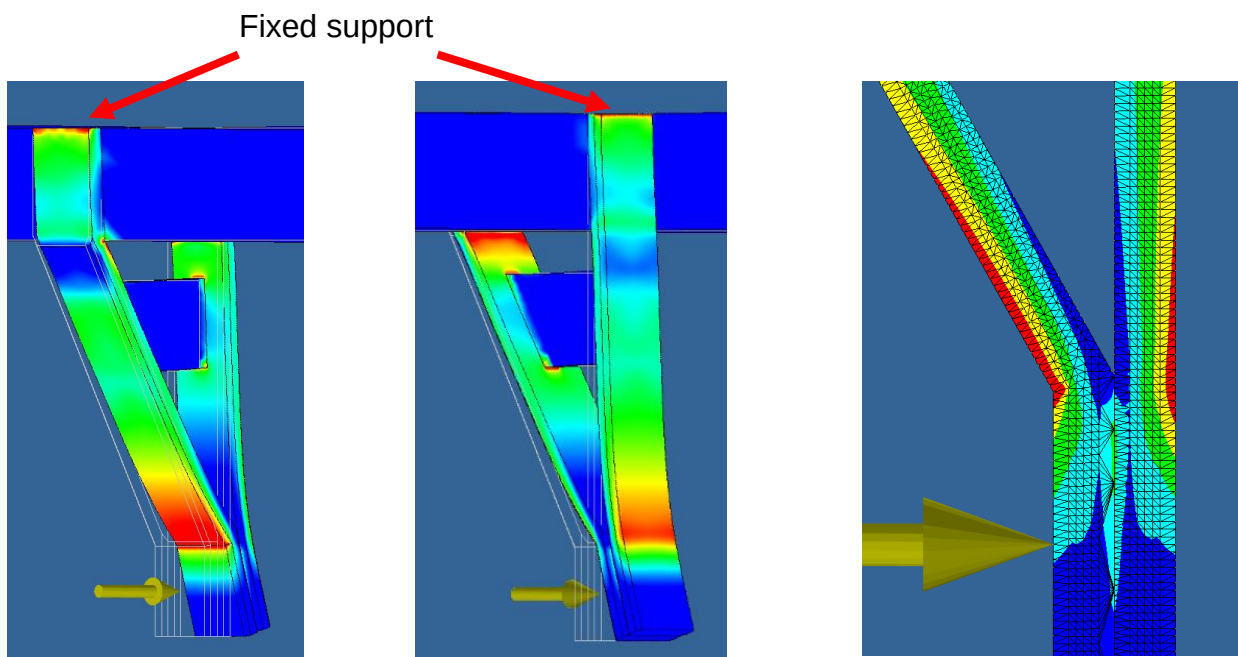


Figure 5.47: FEA analysis of the flat plates

local. Also, the force applied is equivalent to a heavy accident at 10g after which the train would have to be rebuilt anyway.

Table 5-18: FEA analysis setup and results

Analysis specifications	
Nodes	1133809
Elements	793009
Force applied	20 kN
Constrains	2 fixed supports at the top faces
Max deflection	0.7812 mm
Von Mises stress	Max 666.6 MPa (local) and low 0.1 MPa

5.5.4.2 Chassis Connection Validation (CC, DS)

The chassis connection consists of a rectangular steel bar connecting the two aluminium billets on the sides. At the centre, it is to connect the bent plates, preferably permanently and firmly. It was originally considered for the bent plates to be bolted onto the chassis connection. However, the traction centre needs to take up to 16kN of force, meaning 2x M16 bolts are needed at a minimum level, with the bent plates being a mere 35mm wide the stress concentration factor was driven up to 2.22. On the axis of the bolt holes the plate experiences:

$$\frac{60671N}{2.22 \times 35mm} = 3791 MPa, \text{ this is far too high.}$$

Subsequently, welding the TC assembly was deemed sensible.

As for the dimensions of the chassis connection bar, originally a 65x40x6mm rectangular tube was selected. However, this is not available for purchase. Consequently, a new tube dimension was required. The final iteration suggested 75x50x3mm is the next safe option. To validate this, it is important to note that this rectangular tube fails in shear, not bending.

Whilst shear stress is defined as: $\tau = \frac{\text{Torque}}{2 \times A \times t}$ where A is half-section area and t is wall thickness. The chassis connection experiences 2100 Nm of torque, and its half-section dimensions were 72x47x3mm, giving the half-section area: 0.003384 m².



Figure 5-48: Dimensioned Sketch of TC.

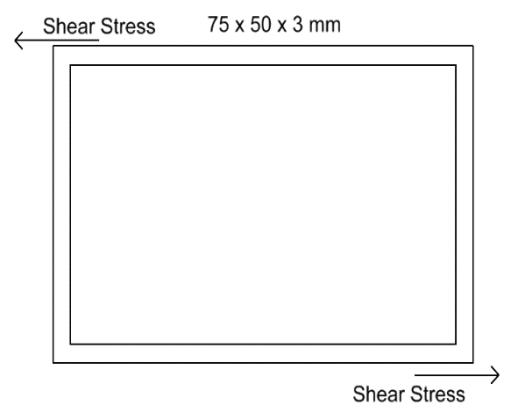


Figure 5-49: Sketched Shear Stress Behaviour.

Shear stress is then equal to: $\tau = \frac{2100 \text{ Nm}}{2 \times 0.003384 \text{ mm}^2 \times 3 \text{ mm}} = 103.428 \text{ MPa}$.

Even if the cheaper option of steel (S275) was used, this is still safe as the shear strength of S275 steel is: $\frac{275}{2} = 137.5 \text{ MPa}$, the required shear load would not exceed the material shear limit, with the safety factor of: $\frac{137.5}{103.428} = 1.33$.

5.5.4.3 Anti-buckling Bar – Validation (CC, DS)

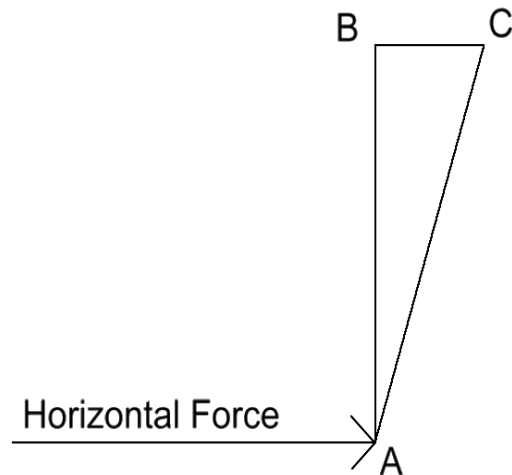


Figure 5-50: Force Illustration for Anti-Buckling.

S275 steel is used for this, however, a new thickness was required hence an extra step of validation iteration took place.

Taking pivot point at point A, the moment it experiences is 4182Nm. BC is in compression; AB is in tension. The member experiencing the highest force is AC (by Pythagoras Theorem, we get 63882.60595 N).

The critical load is defined as: $P_0 = \frac{\pi^2 \times E \times I}{K \times L^2}$

If the plate thickness is 8mm, which was the proposed new dimension it was deemed too thin, one gets a 2nd Moment of Area of $2.56 \times 10^{-9} \text{ m}^4$. Its critical load is:

$$P_0 = \frac{\pi^2 \times 210 \times 10^9 \times 2.56 \times 10^{-9}}{(2 \times 0.098185)^2} = 146406.794 \text{ N}.$$

$$\text{Safety Factor} = \frac{146406.7937}{63882.60595} = 2.292$$

The proposed plate thickness 8mm was deemed safe and implemented.

5.5.4.4 TC Hook

The TC hooks limits the body's vertical displacement 21 mm

Second moment of inertia (I)	$\frac{bd^3}{12} = \frac{0.008 \times 0.04^3}{12}$
Bending moment (M)	$\frac{I\sigma}{y} = \frac{I \times 275 \times 10^6}{0.02}$
Total Force (F)	$2kN \times 5$
distance (d)	$\frac{M}{F} = 0.0586m$

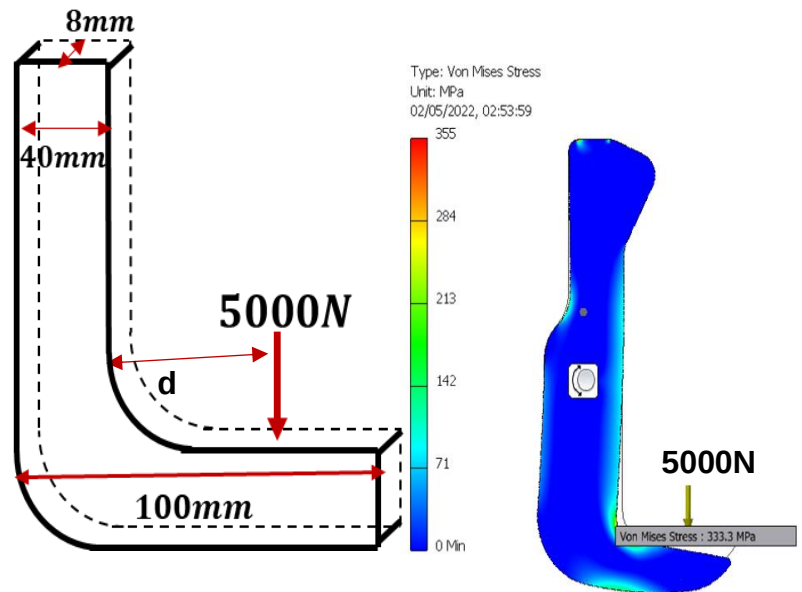


Figure 5-51: Maximum stress for the TC-hook analysis

The TC hooks must be able to carry a load of 10kN as well as limit a vertical displacement of 21mm with a safety factor of 5.

The final TC hooks have 35mm, this is less than the one calculated as the design of the hooks was taken at worst case scenario. 35mm should be more than enough to handle the bogie weight and stop the bogies from falling off when the chassis is lifted.

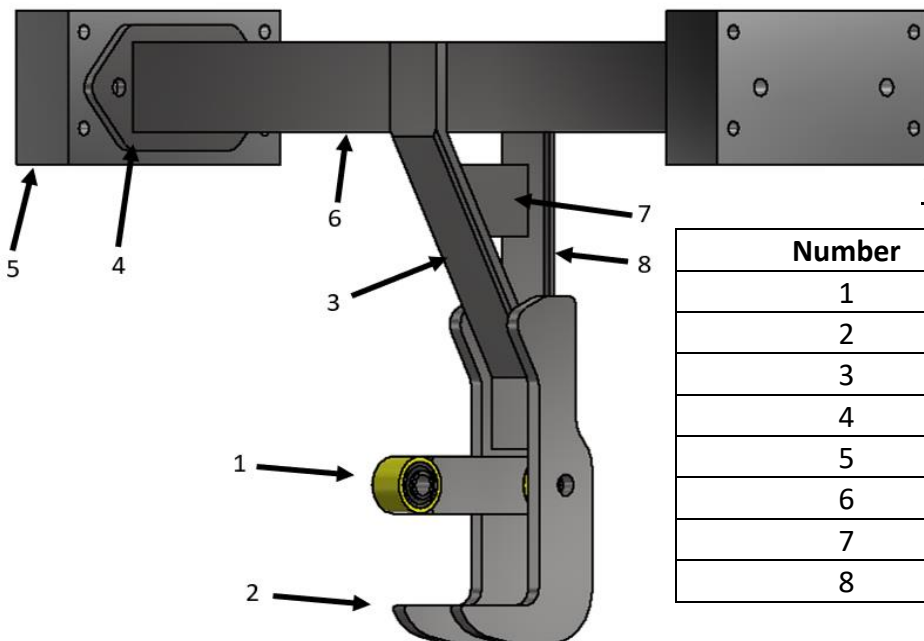


Table 5-19. Final design

Number	Part
1	Bushings
2	Hooks
3	TC bent plate
4	Connection flange
5	Aluminum extension billet
6	Chassis connection
7	Anti-buckling plate
8	TC straight plate

Figure 5.52: Final design

5.6 Bushings, Couplings and Lugs (DS)

Every mechanical component on the bogie uses bushings to couple their mechanisms. Bushings allow for conical and torsional motion whilst restricting radial and axial movement by absorbing energy through elastic deformation.

Every bushing has been interference fit into a E235 Mild Steel round tube. The tubes were welded to connecting plates then bored out to its respective diameter. Welding cannot be applied directly to the rubber bushes because the elevated temperatures would melt the rubber and destroy the bush.

5.6.1 Traction Centre Bushings and Lugs (DS)

The Traction Centre bushings should:

- Allow the body to vertically displace 21 mm and yaw 4.82° relative to the bogie
- Allow the body to laterally displace 15 mm and pitch 0.99°
- Allow the body to roll 1.59°
- Withstand a radial force of 1840N to transmit tractive effort.
- Withstand very small axial forces
- Be easily replaced

Below shows a table comparing the required specification against the FIBET FBNA 1230:3434 bushing used at both A and B.

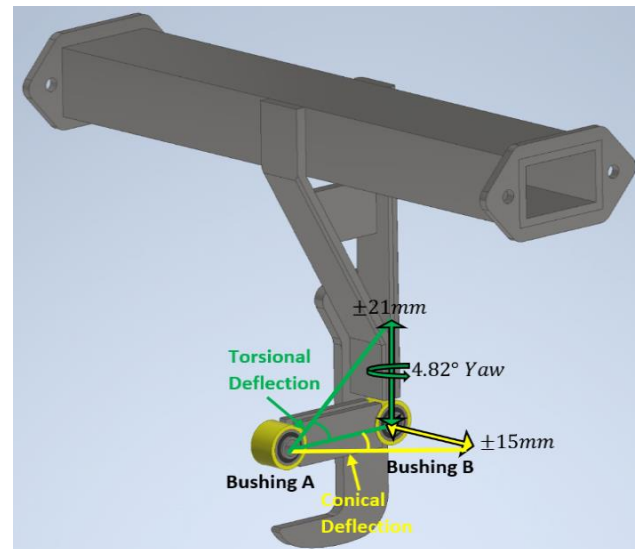


Figure 5-53. Degrees of freedom required by traction centre bushings.

Table 5-20: Traction Centre Bushing Loads and Deflections (appendix H-3)

Bushing	Loads		Deflections	
	Axial (N)	Radial (N)	Torsional (°)	Conical (°)
Bushing A	0	1840	12.9	9.12
Bushing B	0	1840	12.9	4.82
FBNA1230:3434	55	3400	30	7

It is clear from above that bushing A's conical deflection requirement exceeds that of the chosen bushing. This large deflection is to facilitate a maximum lateral body displacement of $\pm 15 \text{ mm}$ which only occurs at maximum cornering. Bushing A will

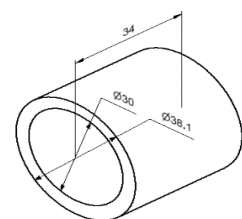


Figure 5-54. Traction centre

subsequently provide damping to the body's lateral displacement when it exceeds a value of $\pm 11.5 \text{ mm}$.

The two bushings are interference fit into E235 Mild Steel tubes connected to each other by two 6 mm plated welded on three sides with a weld area of 400 mm^2 giving a safety factor of **4.2**. It should be noted the traction centre bushing plates connected to the bogie were moved 40 mm apart to account for yaw.

5.6.2. Motor Mount Linking Arm Bushings and Lugs (DS)

The Motor Mount bushings should:

- Withstand a radial force of 777 N
- Withstand very small axial forces
- Be easily replaced
- Allow the unsprung assembly to yaw 1.15 degrees relative to the bogie
- Allow the unsprung assembly to displace vertically, laterally, and longitudinally a distance of 15 mm, 6.82 mm, and 15.6 mm respectively relative to the bogie.

A conical deflection of 7.82° would be needed for Exhibition Park where the wheelset yaws 2.87° , causing the bogie to laterally displace 17.03 mm . A method to resolve this would be to increase the distance between the bushing centres from 124 mm to 163 mm resulting in a 6° conical deflection which can be facilitated by the newly purchased FIBET bushing.

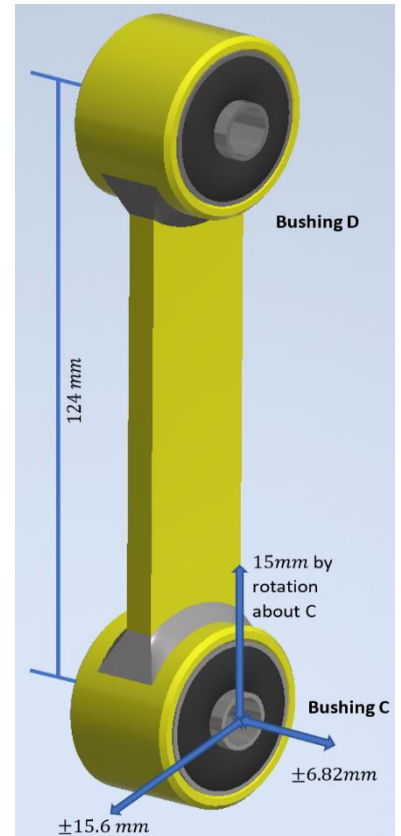


Figure 5-55. Degrees of freedom required by motor mount bushings

Table 5-21: Motor Mount Linking Arm Bushing Loads and Deflections (Appendix H-4)

Bushing	Loads		Deflections	
	Axial (N)	Radial (N)	Torsional (°)	Conical (°)
Bushing C	0	777	2.98	3.15
Bushing D	0	777	7.23	3.15
FBNA1022:1517	110	1035	25	6

The two bushings are similarly press fit into bored E235 Mild Steel tubes connected by a welded 6 mm plate with a weld area of 50 mm^2 giving a safety factor of 3.3. The S275 steel connection plate experiences a force of 31060 N acting under 20g acceleration and has an area of 164 mm^2 giving it a safety factor of 1.45. The bushing

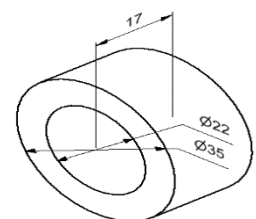


Figure 5-56. Motor Mount arm bushing

lugs have a safety factor of **19.6** without accelerations and **0.98** with 20g acceleration meaning they will need replacing in the event of a crash.

5.6.3. Anti-Roll Bar Bushings (DS)

The anti-roll bar (ARB) bushings should:

- Withstand a radial force of 1304 N
- Be easily replaced
- Allow the body to vertically displace 21 mm and yaw 4.82° relative to the bogie
- Allow the body to laterally displace 15 mm and pitch 0.99°
- Dampen the body roll below 1.6°
- Withstand the torsional moment resulting from cornering the minimum radius at maximum speed maximum cornering.

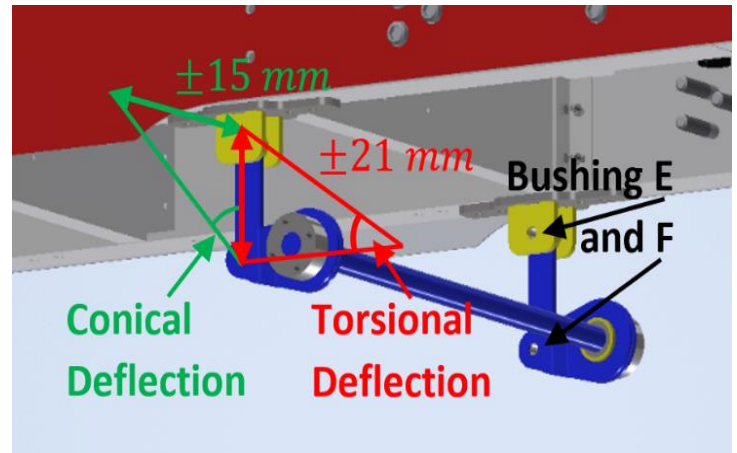


Figure 5-57. Degrees of freedom required by traction centre

Table 5-22: Anti-Roll Bar Bushing Loads and Deflections (Appendix H-6)

Bushing	Loads	Deflections		Moments
	Radial (N)	Torsional (°)	Conical (°)	Torsional (Nm)
Bushing E	1304	4.92	9.59	0
Bushing F	1304	24.8	9.59	26.1
FBNA1230:3434	3400	30	7	40.8

The bushing used in the anti-roll bar mechanism for the 2022 design is the RS PRO 1893192, however there is no datasheet for this bushing available online so the validity of it meeting this specification cannot be confirmed. It is **recommended for future years** to use a FBNA 1230:3434 bushing from FIBET to meet the specification, the same bushing used to couple the traction centre. This bushing will allow for 7 degrees of conical deflection, facilitating 11.0 mm of lateral body displacement. The remaining 4 mm of displacement will be dampened by bushing E and F.

The bushings are interference fit into bored out E235 Steel tubes that have been welded to a 6 mm thick, 25 mm wide S275 connection plate giving them a separation of 90 mm. The connection plate experiences a force of 1304 N and a bending moment of 26.1 Nm giving it a safety factor of **8.6** assuming 5g acceleration.

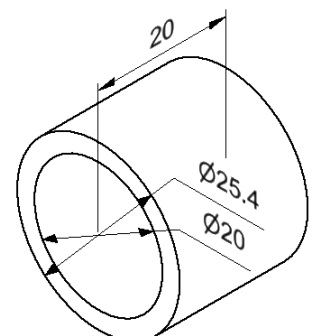


Figure 5-58. Anti-Roll Bar bushing lugs

5.6.4. Steering Link Bushings and Lugs (DS)

The steering bushings should:

- Allow the body to laterally displace 15 mm
- Allow the body to vertically displace 21 mm
- Have sufficient conical and torsional stiffness to ensure sufficient overall rigidity of the steering link to allow for active steering.

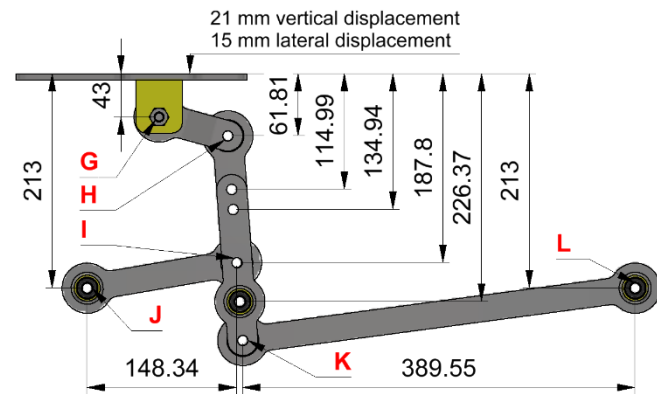


Figure 5-59. Degrees of freedom required by traction centre bushings.

The table below displayed the values of the conical and torsional deflections of bushings G to L in the steering link as labelled in figure 5-58. It is extremely important to note that bushing M was unconstrained from the bogie frame which isn't the case in reality. This incorrect assumption was used so the worst possible conical and torsional deflections would be calculated. A safety factor would be integrated if design was done to meet these extreme values.

Table 5-23: Steering Link Bushing Loads and Deflections (appendix H-5)

Bushing	G	H	I	J	K	L
Conical Deflection (°)	3.25	2.14	5.74	5.74	3.25	2.21
Torsional Deflection (°)	17.9	17.9	4.56	4.56	3.06	3.06
Max Radial Force	991					

The current bushings used in the steering link mechanism are the RS PRO 1893192. Since again no datasheet is available to verify their suitability to meet the dynamic specification, **FBNA 1022.1517 bushings** bought from FIBET **are recommended for future years for bushings G to E** shown in figure 5-59. Their limitations are outlined in table 18.

The bushing lugs used for the steering link are the same as those used for the anti-roll bar shown in figure 5-58. Further calculations on the steering link can be found in section 6.1.6. and appendix H-8. It is recommended for future years to increase the thickness of these lugs to endure larger impact forces.

5.7 True Ground Speed Monitoring System (AH)

5.7.1 Initial Design

The use of a motor was required to obtain the true ground speed of the Loco, this decision, motor selection and electrical configuration is explained in Section 9.2.1. The motor required a device to mount it to the underframe of the bogie. Jack Robson

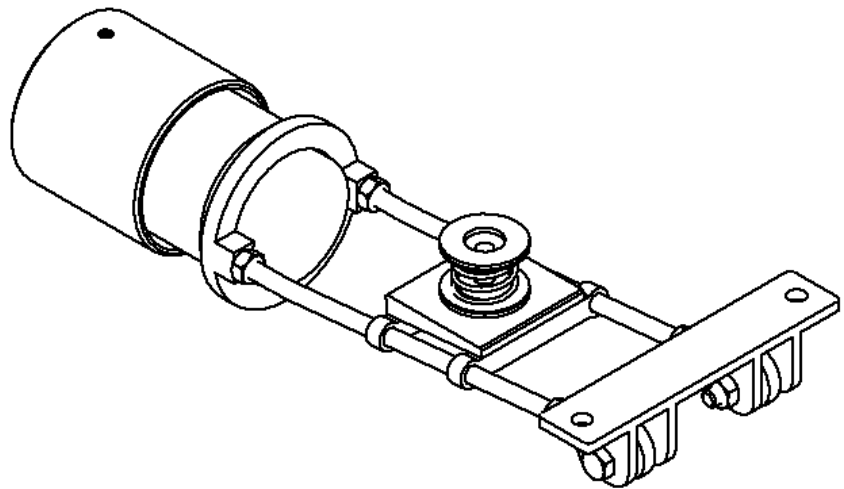


Figure 5-60: Inherited Ground Speed Design (8)

previously designed the mounting system shown in Figure 5-60 (8). This design was then adapted for the motors selected in Section 9.2.1. However, upon undertaking a manufacturing review, it was thought that design of the spring seat and motor connection were unnecessarily complex, both requiring CNC milling operations.

5.7.2 Simplified Mounting Bracket

The motors were inspected, and it was established that there were two M4 bolts which traverse the full length of the housing, allowing mounting of the motor from the rear plate a possibility. An aluminium 'cup' could be turned on a lathe to create a simple mounting point, which would then attach directly to the studding. The design was then simplified further to use three laser cut plates which would be bolted together using the studding, thus removing the need to machine the mounting bracket at all.

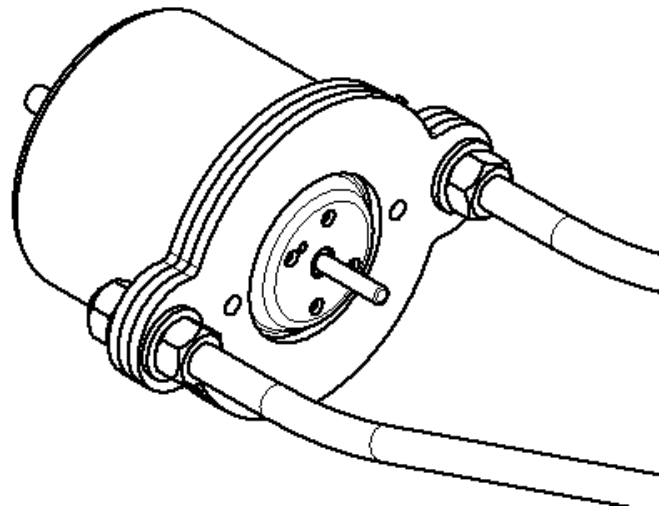


Figure 5-61: Simplified Mounting Bracket

5.7.3 Simplified Spring Seat

The spring seat was therefore the only remaining part requiring machining. The design was reviewed, and it was established that the location of the spring seat meant that the angles involved meant it required machining. The location of the spring was therefore considered, and it was thought that it made sense to locate the spring closer to the motor, on the un-bent section of studding. This is because the most susceptible failure mode of the motor was thought to be as a result of bending stresses on the motor shaft. Therefore, ideally the force of the spring would be located directly above the shaft, however this would be difficult to achieve due to the cylindrical shape and rotational movement of the roller and so the spring as situated as close to the shaft as possible.

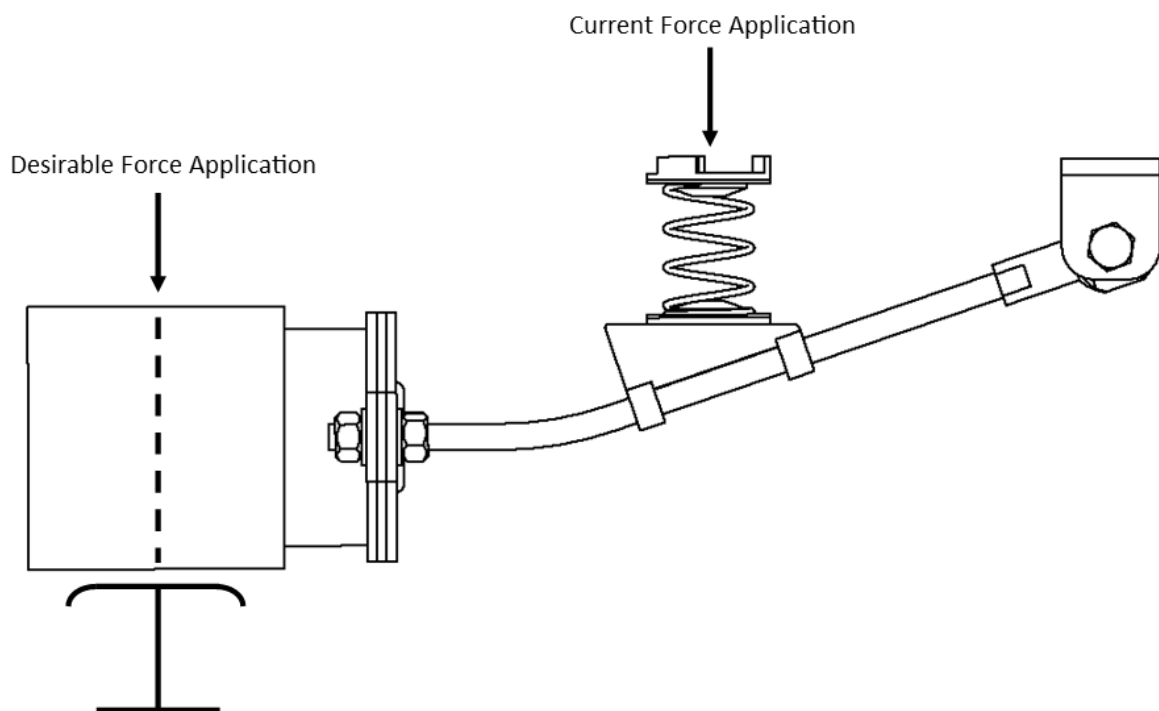


Figure 5-62: Current Force Application and Desired Force Application

Relocating the spring seat to the horizontal un-bent section of the studding allowed for a much simpler spring seat design to be developed. Using sheet metal and L-brackets bolted or riveted together, as shown in Figure 5-62. The spring seat was modelled on Inventor using the outer diameter of the springs, it was then 3D printed with all four spring seats printed in PLA in 2hours 27minutes, significantly quicker and cheaper than if they were to be machined.

5.7.4 Roller Design

The new mounting system offers the benefit of being simpler and cheaper to manufacture but at the disadvantage of no longer protecting the motor from the elements and from debris. The roller design was therefore modified to increase the protection it offers to the motor by creating two hollow cavities at either end of the cylinder, allowing the motor to sit inside one end such that the roller rotates around the motor housing. This reduces the weight which although should be taken by the railhead in most circumstances, would be taken by the motor shaft in unique circumstances. In these circumstances the roller would exert a bending moment on the shaft and so it's desirable to minimise the roller weight as much as possible.

5.7.5 Spring Selection

The spring was to be specified of a low stiffness rate as the weight of the motor and mounting system would maintain roller contact with the railhead. If the spring has too high a stiffness rate, then the motor shaft would be exposed to a higher bending stress

then necessary. In most situations, the spring will be obsolete with it only being required to push the roller onto the railhead on areas with poor track geometry or large cant. The spring selected was Lee Spring LCM200LM 03 S. Key technical data is available in Table 21.

Table 5-24: LCM200LM 03 S Data (14)

<u>Property</u>	<u>Value</u>
Material	Stainless Steel
Hole Diameter	28mm
Solid Height	21mm
Free Length	135mm
Rate	1.038N/mm

5.7.6 Final Design & Manufacture

The final design drawings can be found in Appendix B. During manufacture, a motor was broken as the brushes were damaged when removing the plate to fit one of the mounting plates to the motor. Figure 5-63 is an image of the mounting plates bolted to a motor.



Figure 5-63: Mounting plates bolted to motor

6 Unsprung Assembly (DS)

6.1. Calculations (DS)

It was vital that stress analysis was performed on all aspects on the locomotive to ensure it could withstand the applied loads during operation. Since the actual forces experienced by the locomotive are unknown, the team has designed components to withstand the force multiplication factors demonstrated below (assuming $g = \text{gravitational acceleration} = 9.81 \text{ ms}^{-2}$):

Table 6-1. Table Displaying Force Multiplication factors

Table Displaying Force Multiplication Factors			
Direction of Motion	Unsprung Assembly	Bogie	Body
Vertical Load z (N)	20g	5g	2g
Lateral Load y (N)	10g	10g	2g
Longitudinal Load x (N)	10g	10g	10g

The masses of each of the locomotives major assemblies can be found using:

$$Force_{x,y,z} = mass \times multiplication\ factor_{x,y,z} \quad (6 - 1)$$

Table 6-2. Resulting Forces acting on the Locomotive

Resulting Forces acting on the Locomotive			
Assembly	Unsprung Assembly	Bogie	Body
Mass (kg)	67	202	520
Vertical Load z (N)	13145	9908	10202
Lateral Load y (N)	6573	19816	10202
Longitudinal Load x (N)	6573	19816	51012

6.1.1. Tractive Effort and Axle Torques (DS)

The tractive effort of a locomotive is the maximum force it can exert at its wheels before slipping occurs. It is a function of the adhesion coefficient μ and locomotives mass m and can be calculated using (full calculations in appendix F-4):

$$F_{TE} = \mu mg = 0.375 \times 924 \times 9.81 = 3354 \text{ N} \quad (6 - 2)$$

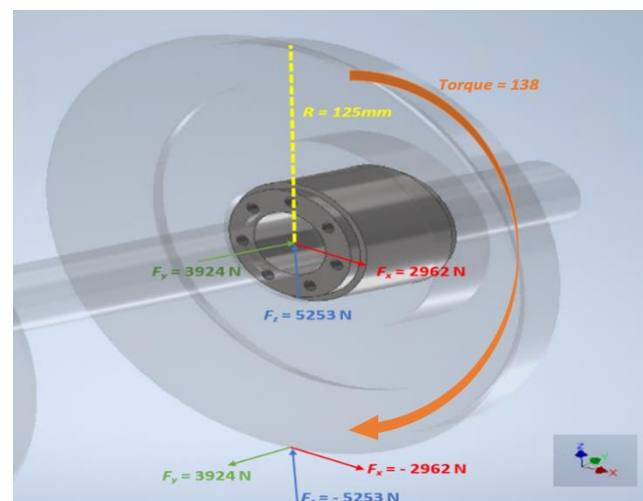


Figure 6-1. Wheel Contact Forces and Torque

The maximum torque on the driven axle before slip can be calculated by multiplying the maximum tractive force per axle by the wheel's radius.

$$\text{Driven Torque} = \frac{\text{Tractive Effort} \times \text{Wheel Radius}}{\text{Number of Axles}} = \frac{3354 \times 0.125}{4} = \mathbf{104.8Nm} \quad (6-3)$$

This value can't be used when calculating torque safety factors of the friction coupling components used. This is because an adhesion coefficient of $\mu = 0.375$ is assumed, however, depending on the environmental conditions this value could be higher resulting in a higher driven torque. The safest way to calculate torque safety factors is by analysing the maximum torque the motor is capable of outputting. This can be done by assuming a linear relationship between the torque and the current. The equation of the red line in the motors characteristics curve figure 6-2 below is:

Where y equals the current [Amps] and x equals the torque [Nm]. The maximum current the Motenergy 1718 motor can pull is 300 amps giving a maximum torque of:

$$\text{Max Motor Torque} = \frac{\text{Max Current} - 4.4}{7.85} = \frac{300 - 4.4}{7.85} = \mathbf{37.7Nm} \quad (6-4)$$

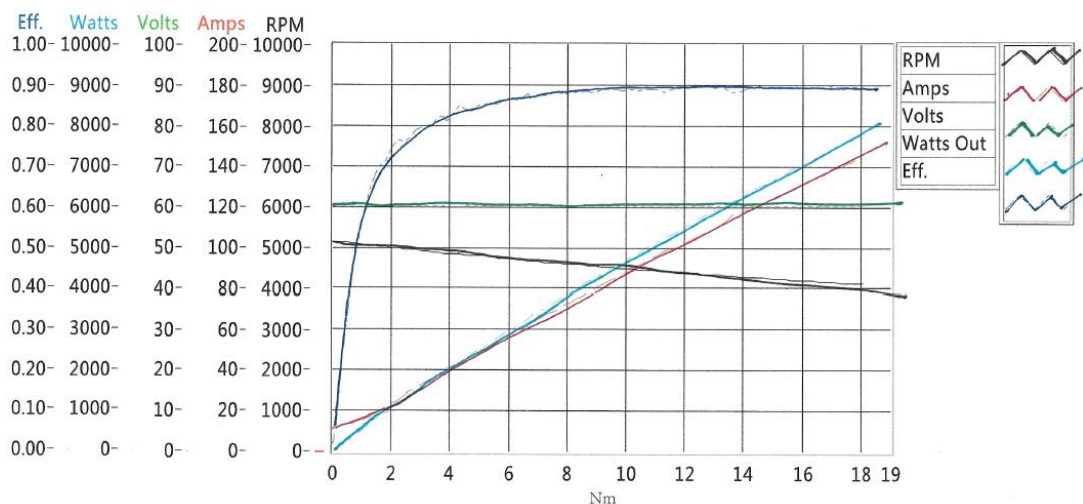


Figure 6-2. Table Displaying Force Multiplication factors

The maximum torque experienced at each wheel is the same as the torque experienced by the larger driven pulley. This torque can be calculated by multiplying the maximum motor torque by the belt drives reduction ratio. The reduction ratio can be calculated using:

$$\frac{\text{Large Pulley Teeth Number}}{\text{Small Pulley Teeth Number}} = \frac{N_2}{N_1} = \frac{80}{22} = \mathbf{3.64} \quad (6-5)$$

The maximum torque at the wheels and large pulley is therefore:

$$\text{Max Motor Torque} \times \text{Reduction ratio} = 37.7 \times 3.64 = \mathbf{137.3Nm} \quad (6 - 6)$$

The maximum torque in the anti-roll bar was rounded in section 5.2. to be 78.2 Nm.

Below is a table summarising the torque safety factors of all friction couplings used in the locomotive and full calculations can be found in appendix F-6:

Table 6-3. Friction Coupling Torques and Safety Factors

Friction Coupling	Location	Permissible Torque (Nm)	Applied Torque (Nm)	Safety Factor
OKBS11	Wheel-Axle	2880	138	16.5
Ringfeder 7012	Pulley-Axle	1154		8.36
TLK110	Torsion Rod	220	78.2	2.81

6.1.2. Wheel contact forces (DS)

The wheel contact forces need to be found to calculate the stresses experienced by the bearings, the shaft and the axle boxes. This was necessary to verify the structural soundness of the mechanisms and highlight necessary design changes.

These forces were found by multiplying the wheelset mass by their corresponding force multiplication factor from table 6-1, followed by the addition of torque effects for the longitudinal direction and static loading for the vertical direction. This value was then divided by the number of wheels the force was shared over. Detailed calculations can be found in appendix F-3 and the table below summarised the maximum contact forces experienced at each wheel.

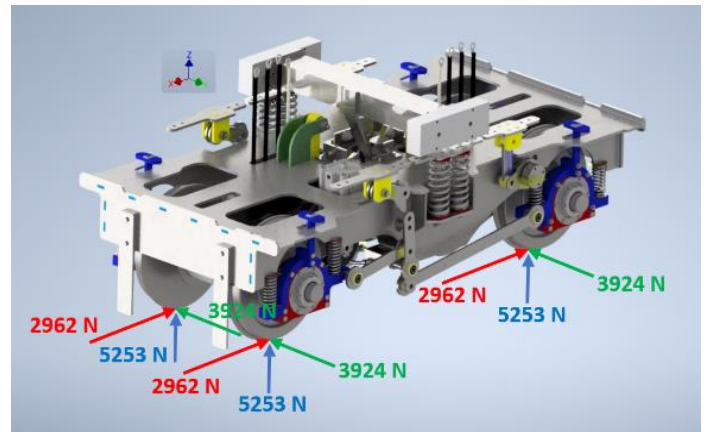


Figure 6-3. Wheel Contact Forces

Table 6-4. Table Displaying Wheel Contact Forces

Vertical Force (N)	Lateral Force (N)	Longitudinal Force (N)	Radial Force (N)
ΣF_z	ΣF_y	ΣF_x	ΣF_r
5253.2	3924	2962	6031

6.1.3. Axle Forces and Validation (DS)

It is vital to calculate the maximum stresses experienced by the shaft to validate its suitability for the locomotive and safety factors. Using the forces calculated in section 3.2 and the dimensioned cross section drawing of the shaft below, shear force and bending moment diagrams can be constructed.

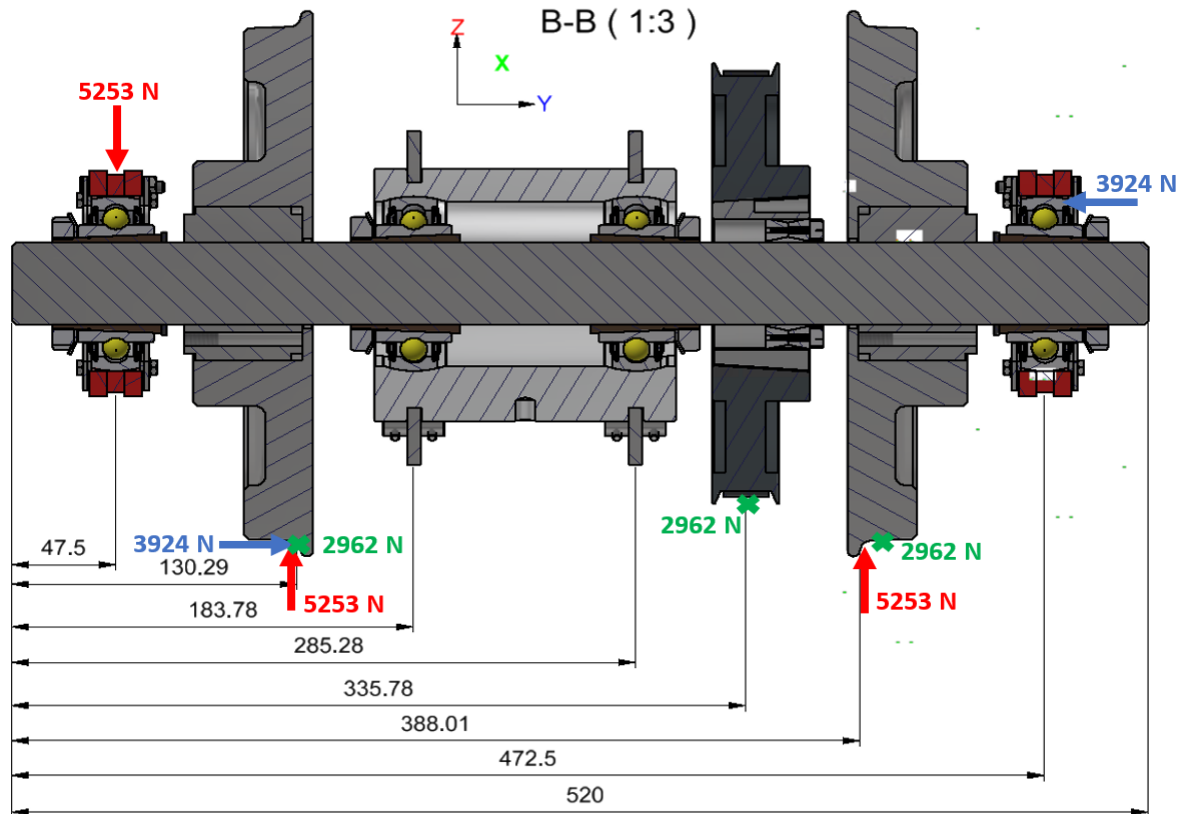


Figure 6-4. Lateral, Longitudinal and Vertical forces acting on shaft

From the shear force and bending moment diagrams below we can see that:

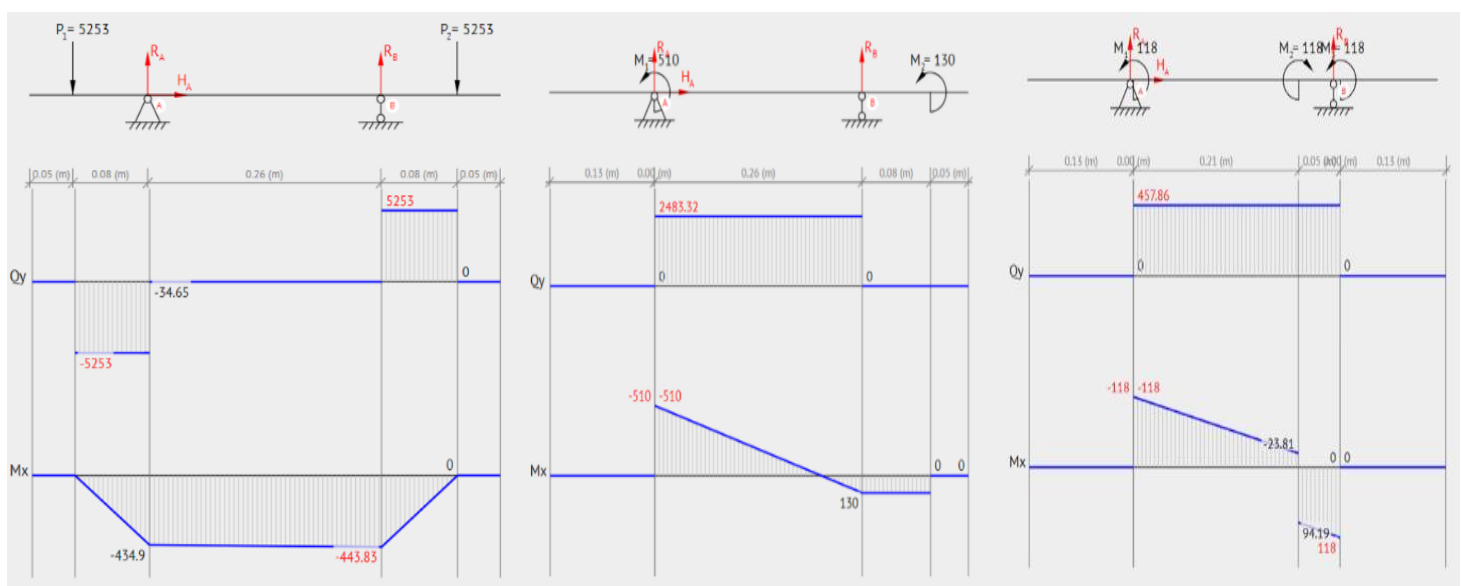


Figure 6-5. Shear Force and Bending Moment Diagrams in the Vertical, Longitudinal and Lateral directions (from left to right) on each Wheelset Axle

- The **maximum shear force is 5253 N** acting between the axle box bearings and the wheels.
- The **maximum bending moment is 510 Nm** acting between the wheels due to vertical loading.

Using the modified Goodman diagram method involves calculating the axial and transverse forces acting on the wheelset axle to find the equivalent stress. This is necessary in conjunction with the method above to find the largest stresses.

Table 6-5: Mean, Alternating and Equivalent Stresses Experienced by the Driven Shaft

Stress	Formula	Value
Worst Normal Stress σ_{worst}	$\frac{4F_a}{\pi d^2} + \frac{32M}{\pi d^3}$	$3.12 \pm 108 \text{ MPa}$
Worst Shear Stress τ_{worst}	$\frac{4F_T}{\pi d^2} + \frac{16T}{\pi d^3}$	$4.80 \pm 9.39 \text{ MPa}$
Equivalent Mean Stress $\sigma_{mean, equ}$	$\sqrt{\sigma_m^2 + 3\tau_m^2}$	8.9 MPa
Equivalent Alternating Stress $\sigma_{alt, equ}$	$\sqrt{\sigma_{alt}^2 + 3\tau_{alt}^2}$	$\pm 109.2 \text{ MPa}$
Equivalent Stress σ_{equi}	$\sigma_{equi} = \sigma_{mean, equ} + \sigma_{alt, equ}$	$8.9 \pm 109.2 \text{ MPa}$

The shafts strength is affected by three main factors:

- Endurance Limit of Steel $k_1 \approx 0.5UTS$
- Surface Finish (this shaft is turned) $k_2 \approx 0.75UTS$
- Size Factor $k_3 \approx 0.83UTS$

Stress concentrations don't need to be considered because there are no keyways or steps in the shaft due to the use of friction couplings only.

By multiplying the ultimate tensile stress of steel by the three values above that are found using a modified goodman diagram (in project folder), the alternating and mean stress factors of safety can be calculated. A detailed walkthrough all corresponding shaft calculations can be found in appendix F-1.

Table 6-6: Current Shaft Geometries Safety Factors

	Mean Stress	Alternating Stress
Permissible Stress (MPa)	660	± 205.4
Maximum Applied Stress (MPa)	8.9	± 109
Safety Factor	74	1.88

6.1.4. Bearing Forces and Validation (DS)

It was vital that bearing calculations were performed and the forces they would experience were analysed to create a specification to choose a safe, long-lasting bearing.

Figure 6.6 shows the forces

experienced by the eccentric device

bearings and axle box bearings. The eccentric device bearings have less mass to support (only the motor, mount and brakes), however they must resist the pulleys maximum belt tension force of 772N. The axle box bearings experienced much larger radial loads as they must support the bogie and bodies weight. Full bearing calculations can be found in appendix F-2.

Table 6-7: Predicted Bearing Life, Operational Distance and Safety Factors

Bearing Location	Safety Factor	Distance (km)	Days of Continuous Use
Axle Box	6.5	174000 km	485 days
Eccentric Device	3.5	4625 km	12.8 days

The bearings chosen for the axle box and eccentric device are SKF YSA 209-2FK bearings held to the 40mm diameter axle by a H2309 tapered adapter sleeve.

6.1.5. Axle Box Calculations DS

The axle box design was adopted from two years ago, but it was vital to verify the structural integrity of the design under the new forces experienced. Applying the longitudinal, lateral and tangential wheel contact forces to the axle box, the spring seats lowest safety factor was **4.92** using:

$$\text{Safety Factor} = \frac{F_{\text{shear}}}{F} = \frac{\sigma_Y \times A}{2F} \quad (6-7)$$

Calculations of the safety factor and weld areas can be found in appendix F-8.

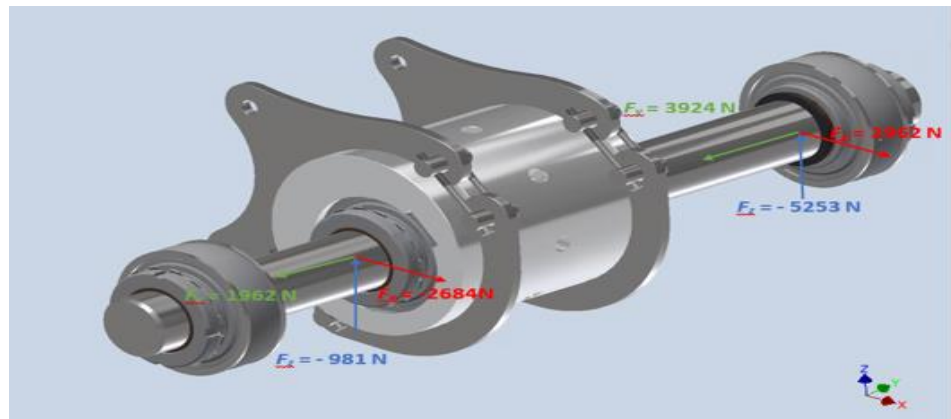


Figure 6-6. Lateral, Longitudinal and Vertical Forces acting on both bearing sets

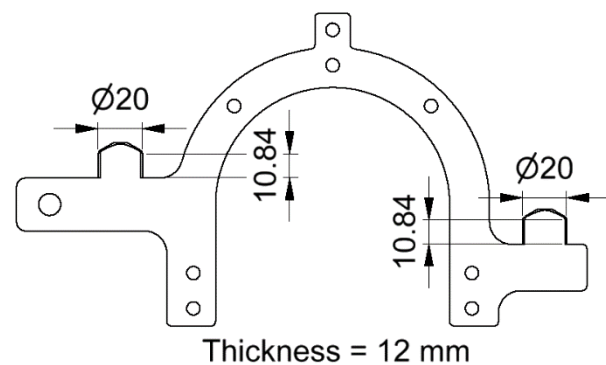


Figure 6-7. Axle Box Dimensions

6.1.6. Steering Link Calculations (DS)

Calculations were performed on the steering links to validate their dimensions, ensuring they can withstand the extreme accelerations and forces expected. This involved calculating shear and planar stresses as well as finding the critical buckling load. The shear stress and planar stress safety factors were found using Hooke's Law. The critical buckling load was found using Euler's theorem applied to the longest member where buckling is most prominent.

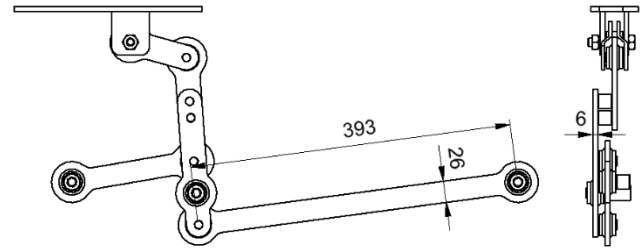


Figure 6-8. Steering Link Dimensions

Table 6-8: Steering Link Safety factors

Criteria	Applied Value	Permissible Value	Safety factor
Shear stress	127 MPa	275 MPa	2.17
Planar Stress		137.5 MPa	1.08
Critical Buckling Load	1982 N	6161 N	3.05

6.1.7. Springs and Spring Seat Calculations (DS)

Calculations were performed on the primary and secondary suspension springs to validate the spring choices and determine safety factors.

Table 6-9: Steering Link Safety factors

Spring Location	Stiffness Rate (N/mm)	Full compression Force (N)	Max Applied Force (N)	Factor of Safety
Primary Sus.	24.5	1128	1104	1.02
Secondary Sus.	41.2	2718	1226	2.21

It should be noted that in the table above, 2g vertical accelerations were used for both the body and bogie when calculating the safety factor of the primary suspension springs. If a vertical acceleration of 5g was used for the bogie as suggested in table, a safety factor of 0.61 would have been produced. This suggests the primary suspension springs will bottom out when subject to impact loads or severe track irregularities. An extensive sum of money would have to be spent if this issue was to be rectified for this year's locomotive as the Acorn order was placed before this issue was discovered. For future years, a stiffer and/or longer spring is suggested for the primary suspension along with an edit of the bogie frame.

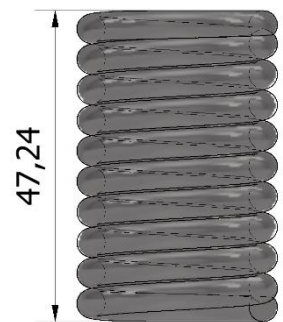


Figure 6-9. Bottomed Out Primary Suspension Spring

The primary and secondary suspension spring seat design and manufacture was greatly simplified by using a bought in Igus GFM-2023-16 and GFM-2730-20 flanged bushing respectively, both with a yield strength of 60 N/mm^2 . The initial design was an Acetal component with yield strength 30 N/mm^2 that required heavy machining resulting in long lead times and material wastage. The new design was stronger, cheaper, faster, and easier to manufacture.

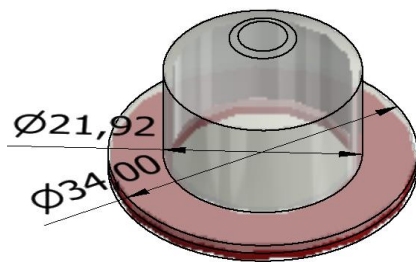


Figure 6-10. Old Primary Suspension Spring Seat

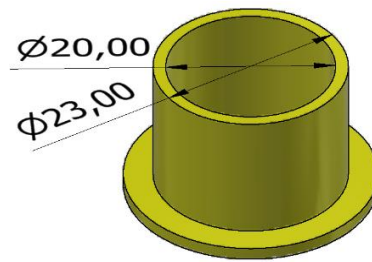


Figure 6-11. New Primary Suspension Spring Seat

6.2. Belt Drive (JJ)

The previous years decided to use belts and sprockets from Gates. Small sprocket with 22 teeth, big sprocket with 80 teeth and poly carbon belt. The gear ratio gives the required performance from the motor and carbon reinforced belts provide longevity, safety and strength required for the application. Gates catalogues have been checked to make sure that the design is correct. The length of the belt was chosen based on the centre distance between axle and motor shaft combined with the size of sprockets. The initial belt drive design was created using 8MX-22S-21, 8MX-80S-21 and 8MGT-

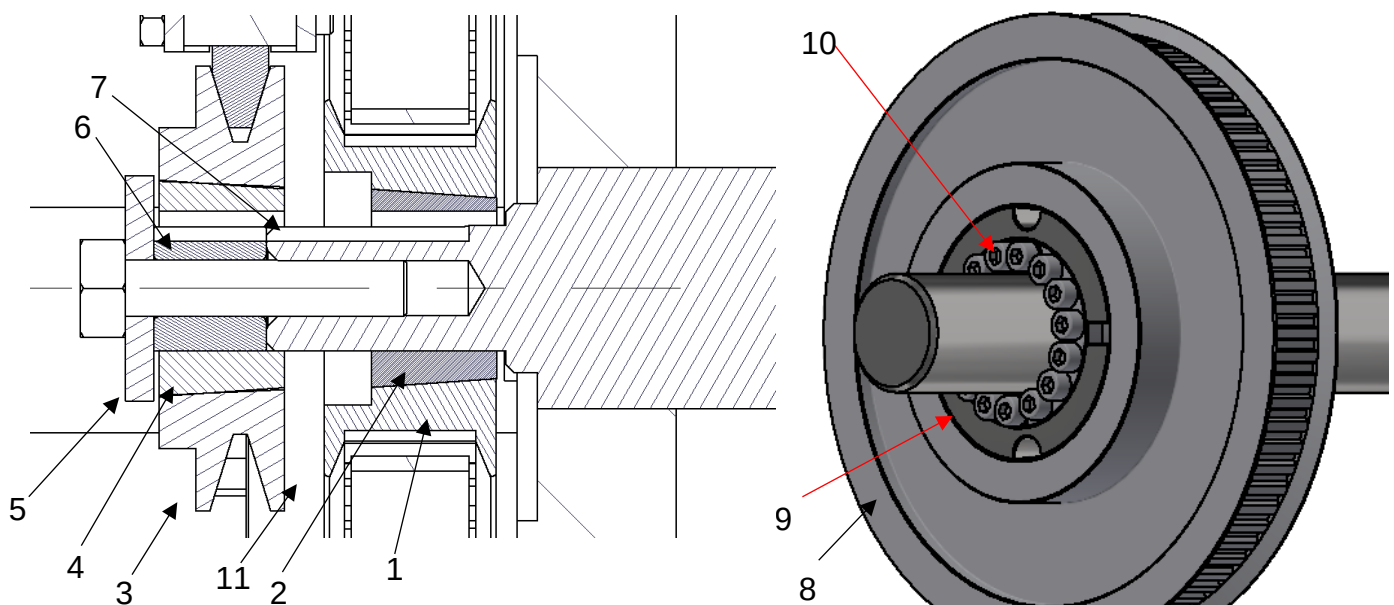


Figure 6.12: Cross section of the small sprocket assembly and design view of the big sprocket

896. The big sprocket is attached to the axle using a 2517- 65mm taper lock bush and locking/friction coupling. Friction coupling prevents the sprocket from moving on the shaft and can carry the torque. The small sprocket is attached using 1108 – 7/8inch taper lock bush which secures the sprocket from moving on the shaft and the key which carries the torque. Due to a short motor shaft, a custom shaft extension was required. The vee pulley for brakes was connected with 1108 – 7/8inch taper lock bush and key as well.

Table 6-10: Components for the belt drive system

Number	Part
1	22 teeth sprocket – 8MX-22s-21
2	Taper lock bush – 1108 7/8 inch
3	Vee pulley – SPZ75/1
4	Taper lock bush – 1108 7/8 inch
5	Washer to secure the shaft extension
6	Shaft extension – machined from steel
7	Keyway. Machined to fit the motor shaft
8	80 teeth sprocket – 8MX-80s-21
9	Taper lock bush – 2517 65mm
10	Friction coupling - Ringfeeder 7012-40x65mm
11	Unnecessary gap. Design flaw fixed later

Local Gates representative engineer was contacted, and it was brought to attention that selected sprockets and belts are only for the North American market. The sprockets and belts available in Europe are:

- 8M-22s-21 small sprocket – different design, comes with a pilot bore
- 8M-80s-21 big sprocket – identical design
- 8MGTV - 896 21mm – Poly chain carbon belt which is carbon reinforced belt

The small sprocket assembly had to be redesigned. The new small sprocket is much wider which resulted in virtually no overlap between the vee pulley and motor shaft. This leaves the key in shear with a small area. Calculations have been done to see if the key is capable of transferring the torque from the vee pulley under braking to the motor shaft. Key dimensions:

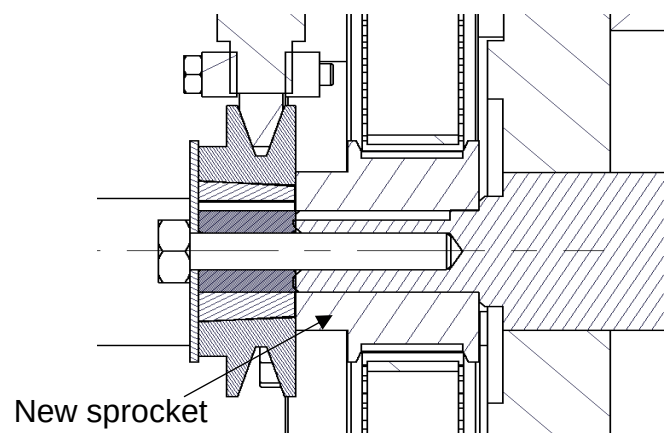


Figure 6.13: Redesigned cross-section of the European market small sprocket assembly

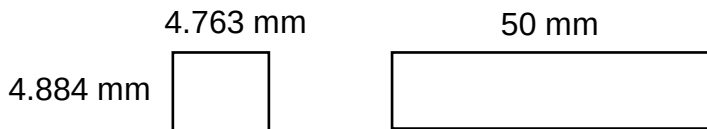


Figure 6.2.4 Shaft key dimensions

Area of the key in shear	$A = 4.884 * 4.763 = 23.26 \text{ mm}^2$	(6 – 8)
Maximum torque from braking	$T = 31.5 \text{ Nm}$	(6 – 9)
Shaft diameter	$d = 7/8 \text{ inch} = 22.225 \text{ mm}$	(6 – 10)
Shear stress	$\tau = \frac{\left(\frac{T}{d/2}\right)}{A} = 122 \text{ MPa}$	(6 – 11)

Allowable shear stress is 0.35 of the yield stress (ANSI B17.1 class 1). This means that the key material should have a yield stress of at least 350 MPa. Common materials used for keys are medium carbon steels and stainless steels which have a yield strength range of 200-400 MPa. To have a good safety factor two keys must be used or the overlap must increase. The compressive stress has a much higher safety factor compared to the shear stress due to a bigger contact area.

A design has been created where the flange of the 8M-22s-21 small sprocket was machined off to increase the overlap and as a result, increase the key shear area (marked in red). New maximum shear stress was calculated to be 69.5 MPa with the required yield stress of around 200 MPa which is commonly available.

Although it was expected to go with this design, it was spotted that the motor shaft and 1108 taper lock bushes have different size keyways. This added extra complexity to this design because the custom key must be machined. Therefore, this design has been revisited to come up with a design that requires less machining and is simpler.

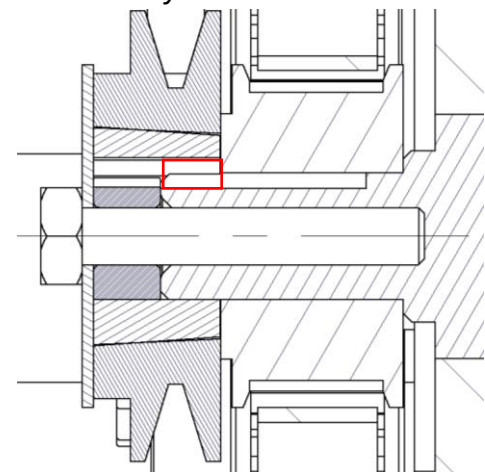


Figure 6.14: Cross section of the modified small sprocket assembly. Highlighted vee pulley and sprocket overlap

Final design

In the final design the sprocket has been bored to fit the motor shaft 7/8in and a keyway with a key was made to transfer torque from the motor. The design tried to avoid altering the sprockets too much as they are expensive and hard to get. Vee

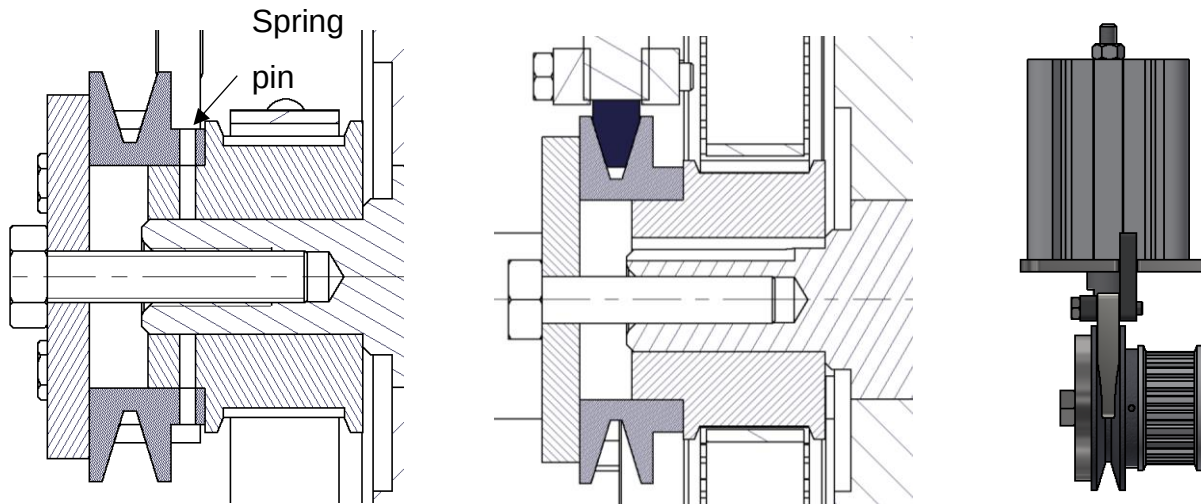


Figure 6.15: Final design of the belt drive system small sprocket assembly

pulley has been bored to have a heavy interference fit with sprockets shoulder. Although, an interference fit can carry some torque two 3mm diameter spring pins have been added to help transfer torque from braking to the motor shaft. It was considered to have horizontal spring pins. However, it proved to be easier for manufacturing to have them fitted vertically. The whole assembly is secured to the shaft using a 6mm washer and M10 bolt. For this assembly to work the pneumatic cylinder support bracket had to be shortened.

Spring pin calculations

Maximum braking force

$$F_{max} = \frac{T}{d} = \frac{31.2 \text{ Nm}}{0.0285 \text{ m}} = 1105 \text{ N} \quad (6 - 12)$$

ISO 8752 pin single shear strength

$$F_{pin} = 3.16 \text{ kN} \quad (6 - 13)$$

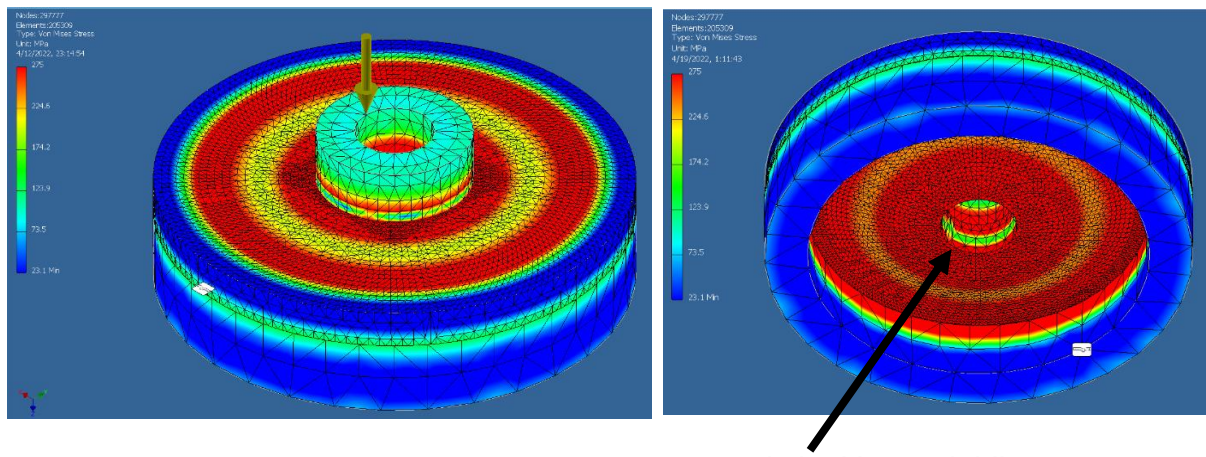
Safety factor

$$SF = \frac{F_{pin}}{F_{max}} = \frac{3160 \text{ N}}{1105 \text{ N}} = 2.86 \quad (6 - 14)$$

For balance two spring pins have been used which doubles the safety factor.

Washer FEA

FEA model has been created to choose the thickness of the washer. Additional parts have been used to simulate the loading. The load applied was the proof load of the M10 bolt – 34.8kN. The support for the washer was the overlap area with the vee pulley. 5mm washer showed significant yielding (indicated with red colour). Although, some yielding is visible on the 6mm washer, it is more local, not as deep and the loading chosen was extreme – in service, the bolt is not tightened to the proof load.



Deep and significant yielding for 5mm

Figure 6.16: FEA analysis of 5 mm washer loaded with M10 bolt

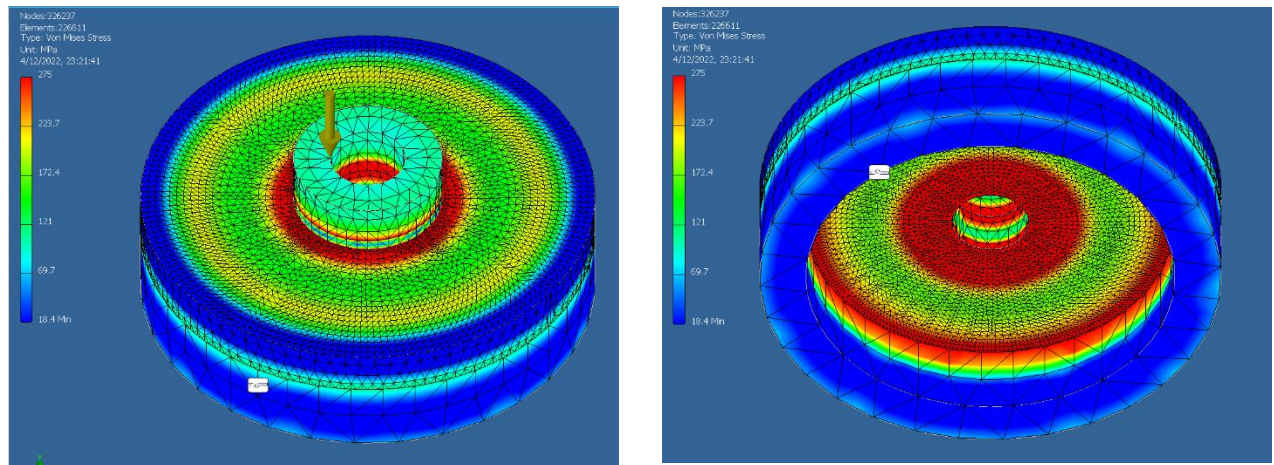


Figure 6.17: FEA analysis of 6 mm washer loaded with M10 bolt

6.3. Motors and Mounts (DS)

6.3.1. Specification (DS)

6.3.1.1 Railway Challenge

Below outlines the specification of the traction system to successfully operate in the Railway Challenge.

- Operate the locomotive for 3 hours at 5km/h on a 2% (1.15°) gradient trailing a 400kg load
- Haul and start a trailing load of 1800kg on a 2% (1.15°) gradient.
- The drive train should be capable of providing a maximum tractive effort of 3679N at the wheels before slip occurs.
- Fit between a 10.25" (260.35 mm) gauged wheelset.
- The motor should remain parallel to the driven axle during all driving conditions. This includes considering:
 - A minimum cornering radius of 20m
 - A maximum track twist of 6mm over 250mm
 - Allow for a 1.15° wheelset to bogie yaw
- The motor mount should allow for quick belt tensioning/relaxing facilitating the removal of the wheelset within 120 seconds for the Maintainability Challenge.
- The traction system should be as quiet as possible for the Noise Challenge.
- Vibrations should be sufficiently damped for the Drive Comfort Challenge.

6.3.1.2 Exhibition Park

Below outlines the specification of the traction system to successfully operate at Exhibition Park.

- All the requirements in the above section 2.1.1.
- Haul and start a trailing load of 1800kg on a 12.5% (7.13°) gradient.
- Fit between a 9.5" (241.3 mm) gauged wheelset.

6.3.2 Motor Mount (DS)

Minor changes were made to the previous year's design of the motor mount and eccentric device. The motor mount linking arm required a different bushing and separation distance. The diameter of the bearing clearance hole in the eccentric device needed enlarged by 50 microns and one of motor mount side plates needed modifying to fit over the motors power and

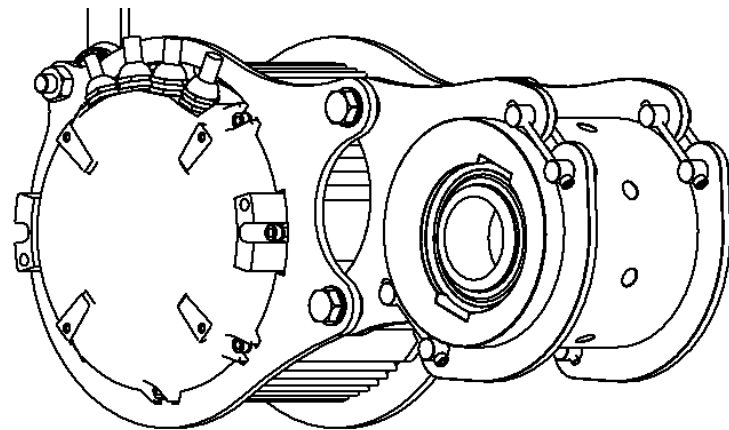


Figure 6-18 Motor Mount Assembly

data cables. The eccentric device is used as the pulleys tensioning mechanism. The centre hole of the bearings on the device is off-centre 7.5 mm meaning the axle can be displaced $\pm 15 \text{ mm}$ longitudinally from the pulley allowing easy tension and relaxation of the belt. Stress analysis was carried out on every component within the motor mount to validate its structural integrity against the accelerations expected.

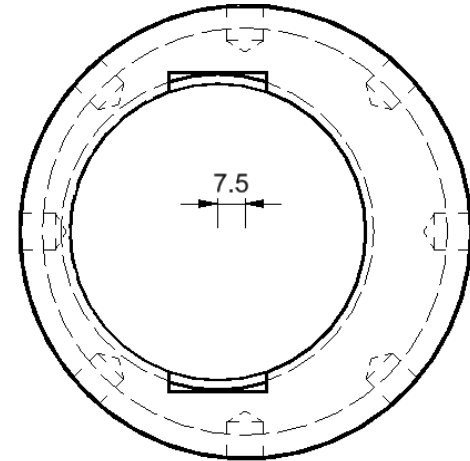


Figure 6-19 Eccentric Device

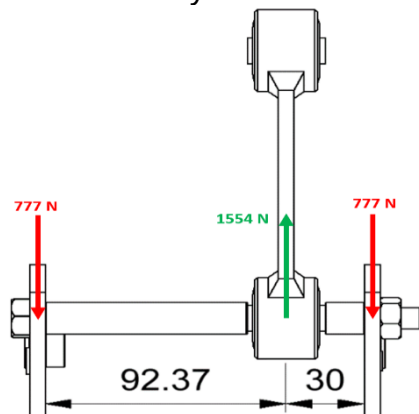


Figure 6-20 Connecting Bolt
Three Point Bending

Stress analysis was carried out on all components within the motor mount to ensure the design was sufficient to withstand the accelerations and forces expected. FEA was done on the motor mount side plates and hand calculations were performed on the connecting bolt to determine an appropriate grade and diameter. Assuming 3 points bending on the bolt and a force of 1553 N acting through the motor mount linking arm, the stresses are as follows:

Table 6-11 Safety Factors of Motor Mount Connection Bolt

Nature of Stress	Actual Stress (MPa)	Permissible Stress (MPa)	Safety Factor
Shear Stress	15.1	300	19.9
Bending Stress	338	600	1.78

Detailed hand calculations of the forces and moments experienced the linking arm and connecting bolt are shown in appendix F-10. The permissible stresses in the table above are for an M10 grade 8.8 bolt which was decidedly used in the final design. Validation of the 6 mm thick S275 motor side plates was done through FEA when a stress of 226 MPa was experience when acted upon by a 2 kN of force.

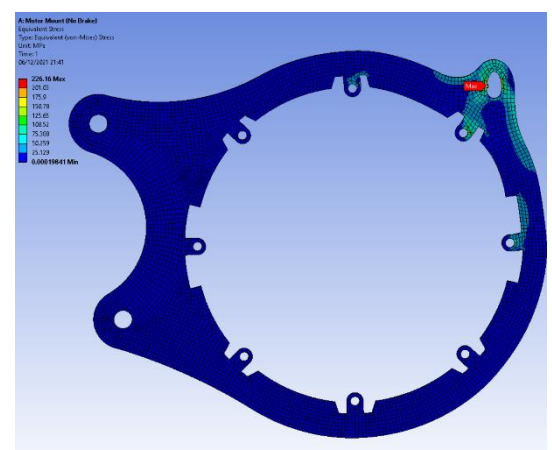


Figure 6-21 Motor Mount Panel
FEA Stress Analysis

6.3.3 Calculations (DS)

As with every year, calculations were required to be performed for the IMechE to ensure the designed locomotive was suitable to participate in the Railway Challenge. This involved ensuring the motors and batteries were capable of outputting enough power safely to satisfy items 1 and 2 in the specification above.

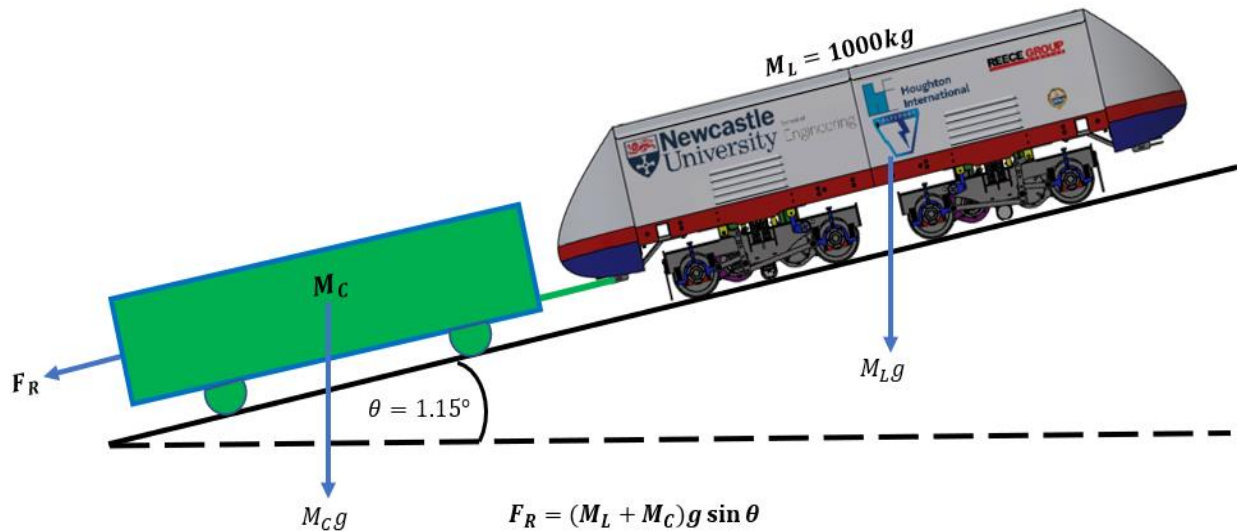


Figure 6-22 Free Body Diagram of Locomotive Trailing Loads at a 1:50 gradient

The Sevcon gen 4 Size 2 motor controller has a maximum current output of 110 Amps (for 60 minutes) meaning the motors must not pull a current that exceeds this at any point during the competition. Current calculations were performed for all challenge conditions using equation 6-4 with the voltage assumed to be 48 volts per locomotive half. The results are shown in the table below and verify the motors and motor controller are very capable. Full calculations in appendix G-2.

Amps Pulled by Motor		
Condition	5km/h	15km/h
$M_c = 400 \text{ kg}$	1.99 A	23.9 A
$M_c = 18000 \text{ kg}$	3.99 A	12.0 A
Max Tractive Effort = 3680 N	26.6 A	79.9 A

Table 6-12 Motor current Requirements for Various Operation Condition Appendix G-2

$$\text{Current Pulley} = I = \frac{\text{Force} \times \text{Velocity}}{4 (\text{number of axles}) \times \text{Voltage}} = \frac{Fv}{4V} \quad (6 - 14)$$

The total energy required for the locomotive to trail 400kg up a 2% gradient for 3 hours at 5km/h was calculated to be 4.14 MJ. Each battery carries an energy capacity of 1.94 MJ (assuming a 40% capacity loss to account for deterioration) meaning the total locomotive energy capacity of 15.6 MJ allows for challenge to be completed nearly 4 times before a recharge is necessary. Refer to appendix G-3.

6.3.4 Changing Wheel Gauge (DS)

It was necessary that a mechanism is in place to accurately change the wheel gauge between 10.25" and 9.5", allowing the locomotive to operate at both Stapleford and Lakeshore respectively. The simplest method involves using three spacers that slot on the axles between the eccentric device and 80 toothed pulley to provide necessary areas of clearance. The wheels and pulleys friction couplings are then tightened to the specified torque followed by the removal of the axial spacers. The axial spacers should:

- Ensure the wheels are constrained perpendicular to the shaft
- Allow the wheelset to be easily adjusted between a 10.25" gauge and a 9.5" gauge
- Be easily removeable from the shaft after component tightening
- Be machined to a tolerance of $\pm 0.05 \text{ mm}$

The initial design was quickly dismissed due to the unnecessary overcomplication of the design. The idea was to have the spacers fastened together with socket cap screws as shown in appendix B X but upon reflection, it was realised these screws were unnecessary. The final design shown in figure 6-23 consists of 5 horseshoe plates of varying thickness machined from a round bar of 6082T6 aluminium. Three of the plates are used to separate the wheels for a 9.5" gauge and all five plates are used for the 10.25" gauge.

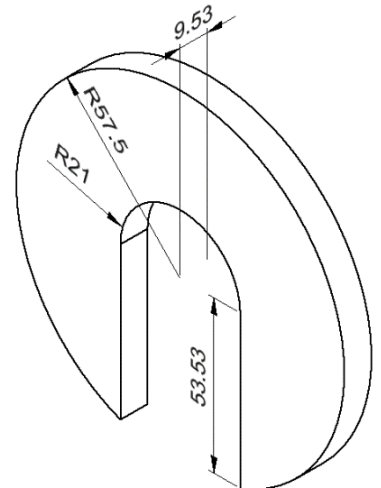


Figure 6-23 Axle Spacer Dimensions

Spacer Thickness	Quantity	10.25" Gauge	9.5" Gauge
7.76	1	✓	✓
9.52	2	✓	✗
16.8	1	✓	✓
18.5	1	✓	✓

Table 6-13 Spacer Thicknesses and Corresponding Gauges

6.4. Primary Suspension (AP)

Responsible team member has received an extension, section to follow shortly.

6.5. Axle Box (AP)

Responsible team member has received an extension, section to follow shortly.

7. Body, Structure and Nose Cones (HW, JO)

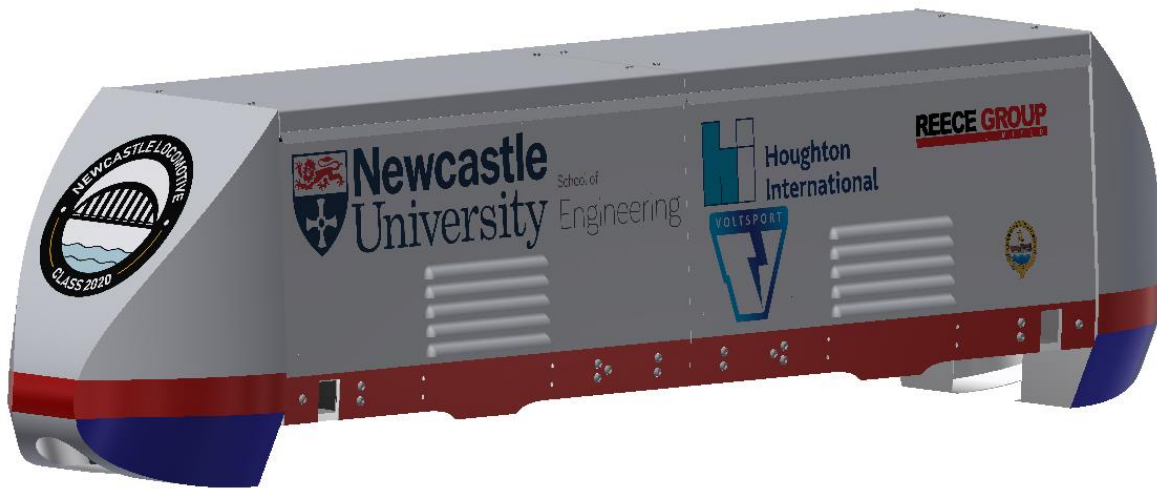


Figure 7-1: Full CAD model of body

7.1 Body Layout

The body houses most of the components that are required for the operation of the locomotive. This includes batteries, control systems and air compressor.

A modular design is used for the locomotive so that it can be split in half for easy transportation. Passive provision is in place to allow the inclusion of a passenger car between the two halves in the future. This also ensures reliability and easy fault finding as each half of the body can act largely independently, essentially acting as its own locomotive.

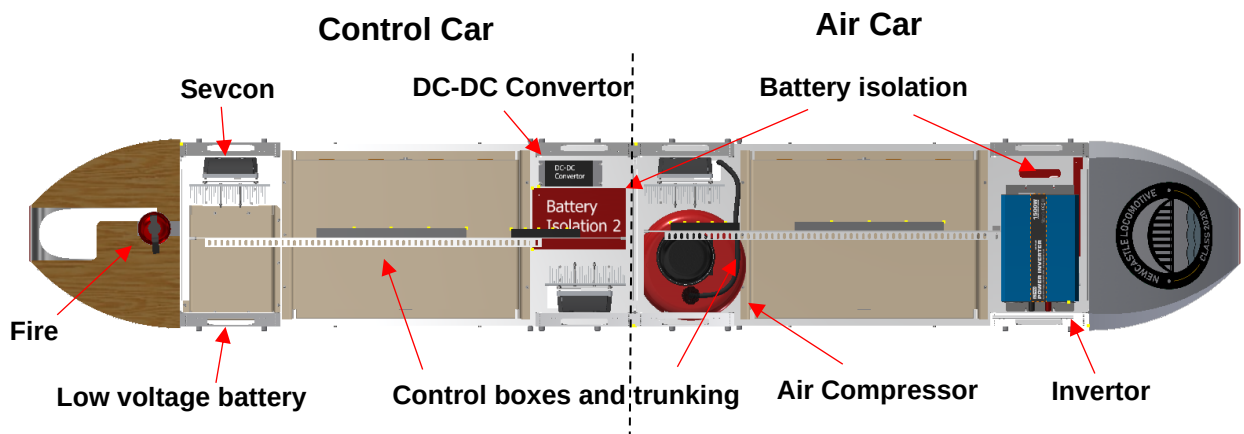


Figure 7-2: Layout of the body

7.1.1 Contents of the locomotive

Table 7-1: Table of contents of the locomotive body

Both Cars	Control hanger	A central plate for boxes (containing low voltage circuits) and cables to attach to.
	Sevcons	Each Sevcon is attached to a mount to keep it upright with its own heat sink. There is two sevcons per loco half (one for each motor).
	Lead Acid Batteries (48V)	Each loco halve contains four 12v car batteries to power the high voltage system. They are enclosed in a case with a clamping mechanism that is designed to allow for quick battery removal.
	Battery isolation switch & fuses	Each set of four battery's in series need an isolation switch to stop flow of charge in a case of emergency or when not in use. An adequate fuse is also needed to brake the circuit in the case of a short circuit.
	Autostop Sensor	Sensor to be used during the IMECHE autostop challenge. It detects when the locomotive has passed a reflective panel and brings the locomotive to a stop.
Air Car	Inverter	Converts the 48V battery supply to 230V AC for the compressor and horn. Located at the front for close proximity to these components.
	Compressor	The compressor is used to operate the brake system.
	Air Horn	Air horn is located in the front nose of the loco to warn anyone in front and to sound away from passengers riding behind the loco.
Control Car	Extra Battery	Extra battery for the low voltage circuits. Encased with an independent clamping system. A battery isolation switch and adequate fuse is also required.
	Fire Extinguisher	Fire extinguisher readily accessible by lifting gull-wing door in case of emergency and alerts from the fire detection system.

7.2 Connection block

7.2.1 Previous design work

The previous year's design took a two-pronged approach to connecting the locomotive halves – using two plates to connect the bottoms of the c-channels and plates higher

up on the internal wall assemblies. Validation of this design through calculations had largely not yet been conducted.

Discussion of this design identified that the strong points of the design were the methods used to connect the two locomotive halves as they were sound in principle. However, it was decided that if after further validation the connection could be supported using only the fishplate connection block the need for additional bolts and plates on the internal wall assemblies could be removed. Leading to less weight and removing potential difficulties of assembling the locomotive halves together as internal components and lack of space would make accessing these bolts for assembly difficult.

7.2.2 Design iteration

Upon completion of initial hand calculations [appendix C-2], it was found that the 3"x3/2"x1/4"x1/4" c-channels used in the previous design would not provide enough bending strength for use and not be able to accommodate a large enough connection block.



Figure 7-3: 4"x2"x1/4"x5/16" C-channel support on the body

Therefore, the decision to increase the c-channels to 4"x2"x1/4"x1/4" was made due to the increase bending strength and allowance for a larger connection block. This led to changes in size of many other components of the body, although these changes did not affect the locomotive structurally.

During purchase enquiry for the c-channels, difficulty in getting 4"x2"x1/4"x1/4" c-channels arose. This is due to a popular practice of making the channel flanges slightly thicker than the c-channel wall. The size of the flanges on the new c-channel would be 5/16" an increase of 1.5875 mm. This increase is minimal, and recalculation validated that the changes in size of the connection block and cross beam would be valid.

Vertical bolts that would hold the block in place in a tensile fashion and also provide the added benefit of bolting the floor securely in place were viewed as a good design

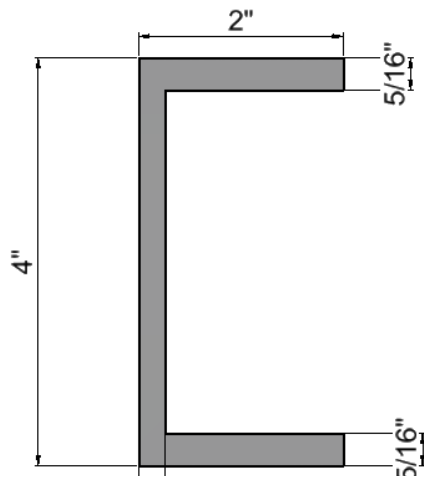


Figure 7-4: Cross section view of 4"x2"x1/4"x5/16" C-channel support on the body

decision – especially as bolt's strength in tension is double that in shear. However, taking a looking holistic approach it was identified that when attempting to bolt the two halves of the locomotive together these bolts would prove hugely difficult to line up, leading to opting for using additional horizontal bolts instead.

7.2.3 Final design validation

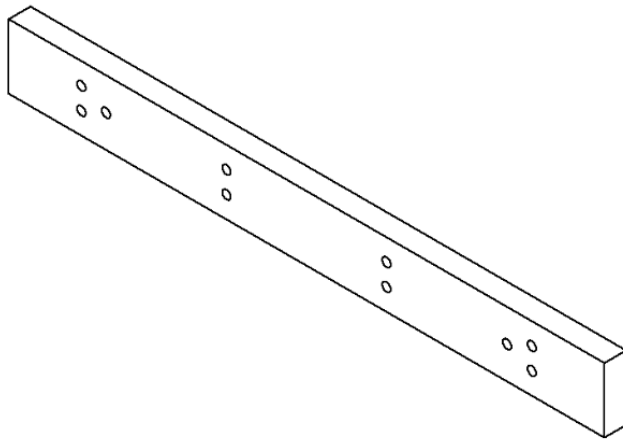


Figure 7-5: Final CAD design of connection block

The bending moment at the coupler due to forcing at the coupler (because of forcing on or due to carriages or collision): 100KN

This force is divided by two as it is reacted by two c-channel members: 50KN

Using a distance from the member to the force at the coupler as 0.1348m gives a moment applied at the end of the c-channel as:

$$M = F \times d = 100\text{KN} \times 0.1348 = 13.48\text{kNm} \quad (7 - 1)$$

7.2.3.1 Distance from member to force at coupler

- Height of bogie from point in contact with rail head to point in contact with c-channel: 338mm
- Height of c-channel: 4 inches [101.6mm]
- Height of centreline of coupler bracket from rail head: 10 inches [254mm]
- Distance from centre of member to force at coupler: 338mm+50.8mm -254mm = 134.8mm

Using this distance to find moment on the members:

$$M = F \times d = 100\text{KN} \times 0.1348 = 13.48\text{KNm}$$

Moment on each member:

$$\frac{13.48\text{KNm}}{2} = 6.74\text{KNm per member} \quad (7 - 2)$$

If the bolts are at a 200mm spacing the shear force in a single bolt is:

$$6740 \div 0.2 = 64.4\text{KN} \quad (7 - 3)$$

The proof loads a bolt should be able to withstand is 64.4KN (2 times the shear force needed to be withstood).

However, we will put each bolt in parallel (2 bolts) as above so the actual proof loads each bolt will need to take is 32.2KN.

Two M12 bolts would be sufficient for this spacing [Proof strength of M12 grade 12.9 socket cap bolt=81.8KN, shear strength=40.9KN.

Bolt crushing

This calculation checks that the wall of the c-channel will not be crushed by the force exerted from the bolts onto the c-channel which causes a contact stress. Note: this assumes that the wall of the c-channel will take all the force from the bolts; this is a very conservative assumption as in reality the connection block will also be in contact with the bolt and take a large amount of this force. Leading to a much lower stress in the walls of the c-channel. It is suitable to use this conservative assumption as the c-

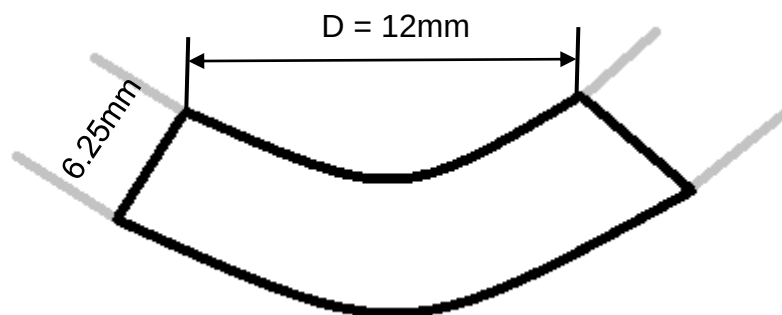


Figure 7.6 – Diagram of contact area of bolt on flange in crushing

channels support the structure of the locomotive and are therefore a critical component.

The contact patch between the bolt is assumed to be rectangular.

$$\text{Area in crushing} = 6.35 \times 10^{-3} \times 12 \times 10^{-3} = 7.62 \times 10^{-5} \text{m} \quad (7 - 4)$$

If we use two bolts the force in each bolt is half the shear force

$$\sigma_c = \frac{F}{A} = \frac{16850 \text{N}}{7.62 \times 10^{-5} \text{m}} = 221.12 \times 10^6 \text{Pa} \quad (7 - 5)$$

Which is valid as 221.12 MPa is below the yield strength of aluminium 6082-T6

Bending stresses on the connection block

If a 38.1mm (1.5inch) thick block is used

$$\sigma = \frac{My}{I} = \frac{6740 \times 0.0445 \times 12}{0.0381 \times (0.0889)^3} = 134.5 \text{MPa} \quad (7 - 6)$$

This is valid as stress is well under yield for aluminium 6082-t6

This block will be machined down from a 3.5-inch (88.9mm) aluminium grade 6082-t6 block (purchase reference C-4) as this is the closest readily available size that is able to be milled down.

7.3 – Body Structure

7.3.1 - Ladder Chassis

A ladder chassis uses two high strength beams to give the body a large amount of its



Figure 7.7 – Manufactured ladder chassis

strength especially in the longitudinal direction.

Perpendicular tubes between these two high strength beams help to provide lateral strength.

The benefits of this type of chassis are its easy assembly and high strength as well as ability to make with standard parts. The downside is its limited torsional strength – to counter this the internal wall assemblies and floor are designed to provide torsional strength to the structure.

The high strength beams used are aluminium 6082-t6 c-channel of dimensions 4"x2"x1/4"x5/16". 6082-t6 is a readily available grade of aluminium that is used for structural applications. Although expensive it is still in the expected price range of affordability for our applications and has the highest strength out of the 6000 series of aluminium with strength properties akin to s275 steel but is 1/3rd of its density. Therefore, saving a third weight from the locomotive. This is particularly important due to the c-channels large size contributing significant weight to the locomotive that needs to have its weight minimised.

Minimising the weight of the locomotive achieves two main objectives:

- Reducing the amount of energy needed to be fed to the motors to accelerate the locomotive or to keep a constant speed due to frictional forces and when the locomotive is climbing an incline.
- Keep each locomotive halve under the maximum weight for the Stevenson building lift (500kg including trolley) so that it is possible to move the locomotive from where it is assembled (floor 2) to a track.

60x60x3mm Steel box tubes(S275) were added to the ladder chassis to allow the locomotive to be jacked up, at the body, by using casters. S275 Steel is used as it has the required strength and has a high price to strength ratio. As well as this steels high weldability is required to weld to the mounting flanges that can be bolted to the c-channel for attachment, without having to bolt through the box tube. A requirement so that the jacking gear can be attached.

MECHANICAL PROPERTIES

BS EN 485-2:2008 Plate 6.00m to 12.5mm	
Property	Value
Proof Stress	255 Min MPa
Tensile Strength	300 Min MPa
Elongation A50 mm	9 Min %
Hardness Brinell	91 HB

Properties above are for material in the T6 and T651 condition

Figure 7.8 – Mechanical properties of 6082-t6 from Aalco (13)

- Density: 7800 KG/M³
- Electrical Resistance: 0.20 - 0.25 $\mu\Omega$ m
- Elongation (in 200mm): 20%
- Hardness Vickers: 115 - 154 Vickers – HV
- Melting Point: 1460 °C
- Modus of Elasticity: 210 GPa
- Shear Modulus: 80 GPa
- Tensile Strength: 370 - 530 N/mm²
- Thermal Conductivity: 50 W/m²°k
- Thermal Expansion: 11 μ m/m-k
- Yield Strength: 275 N/mm²

Figure 7.9 – Mechanical properties of mild steel S275 (5)

7.3.2 - Internal walls

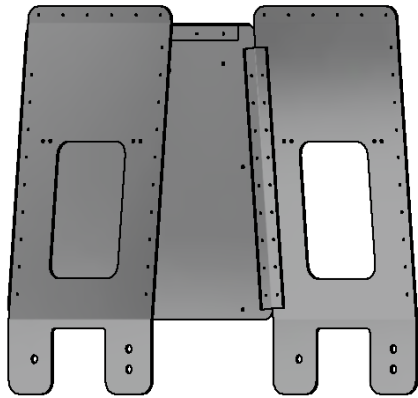


Figure 7.10 – Final CAD model of internal wall one of the internal wall assemblies

The internal walls of the locomotive give the locomotive its structure helping to support the sides of the locomotive, the control hangers, inverter shelf and top box that contains the jacking gear.

Previous years designs of the internal walls were to be assembled using bolts to connect the internal walls to the backplates. Unfortunately, this would lead to the issue of removing all these bolts after the halves of the locomotive were separated. This is not a viable

option due to issues regarding the accessibility and time taken to remove of over eighteen bolts.

Two alternate methods of assembling the internal wall assembly were suggested:

Welding- The limitations of welding the internal walls together are due to the difficulties of welding aluminium. Changing the internal walls to steel would lead to a large increase in weight and would be unviable.

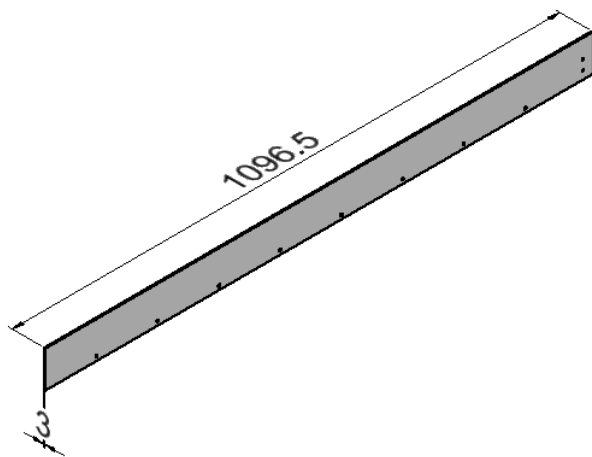
Riveting – Riveting of the brackets holding the internal wall assemblies would allow the assemblies to be placed back-to-back with one another without interference (as long as the rivets are filled down correctly). This was decided as the preferred option. Note was taken that this would likely result in the need for purchase of an air powered rivet gun to speed up manufacturing as well as prevent difficulties that would result from trying to hand rivet large rivets through relatively thick material. The completed



Figure 7.11 – Image of riveting setup

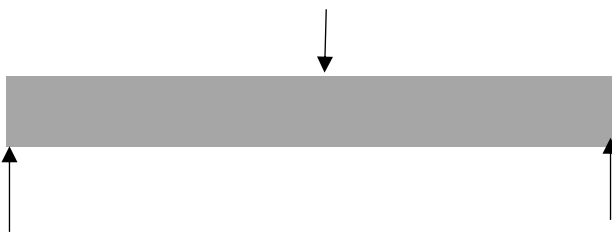
assembly of the internal walls used four 4.8mm stainless steel rivets per bracket and was manufactured by use of a large hand riveting device and air rivet gun.

7.3.3 - Control hanger

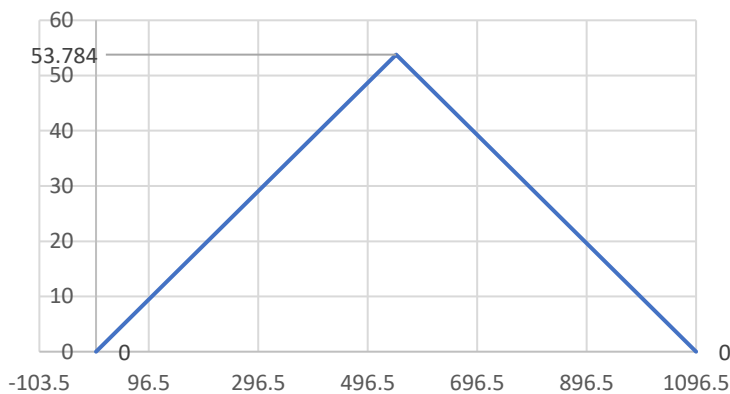


The control hanger is the home to all of the low voltage circuitry of the locomotive. Its objective is to store the low voltage circuitry away from the high voltage components and wires that could provide interference. These hangers are supported by the internal walls of the locomotive and make use of otherwise wasted space.

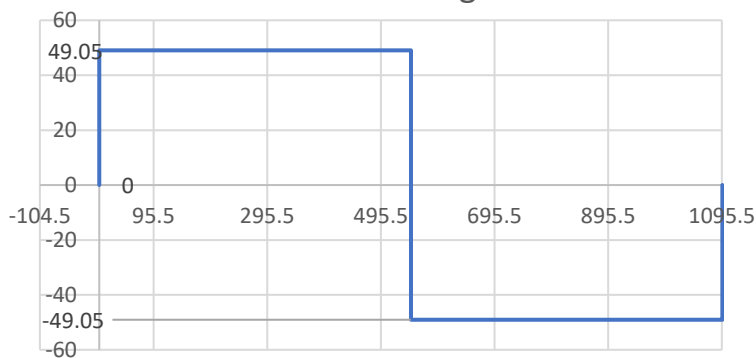
Figure 7.12 – Diagram of front control hanger



Bending Moment Diagram



Shear Force Diagram



The design underwent late changes to remove the channels as a result of a change to the boxes used for holding the low voltage electrical components. These component boxes increased in size and switched to aluminium boxes. This negated the need to have c-channels to hold the boxes as they could be bolted directly to the central hanger plate. This reduces cost and weight of the central hanger (a saving of 3.83kg per hanger.) As the central plates provided the hangers bending strength in the longitudinal direction where bending deflection (due to weight) will occur the bending strength of the control hangers will not be sacrificed.

Figure 7.13 – Diagram to represent control hanger in 3-point bending,

Figure 7.14 – Bending moment diagram for control hanger

7.15- Shear force diagram for control hanger

Hanger validation

Second moment of area for rectangle, $I = \frac{bd^3}{12} = \frac{(62 \times 10^{-3})^3 \times 3 \times 10^{-3}}{12} = 5.958 \times 10^{-8} \text{ (7-6)}$

$$\sigma = \frac{My}{I} = \frac{53.784 \times 31.375 \times 10^{-3}}{5.958 \times 10^{-8}} = 28.1 \text{ MPa} \quad (7-7)$$

The bending stress is well below the yield stress for aluminium 6082-T6 (255MPa) and therefore will not fail.

Bending deflection

$$\delta = \frac{FL^3}{48EI} = \frac{2.5g \times 1096.5 \times 10^{-3}}{48 \times 68 \times 10^9 \times 5.958 \times 10^{-8}} = 0.157 \times 10^{-4} \text{ m} = 0.16 \text{ mm} \quad (7-8)$$

7.3.4 – Sevcon mount design

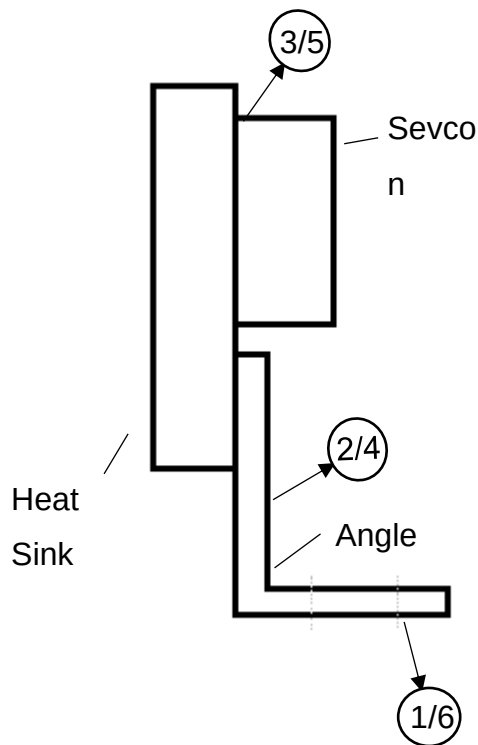
Limitations of the design for the previous sevcon mount are:

- Heatsink not in proper thermal contact with the sevcon as the aluminium stand sandwiched between the sevcon and heatsink will limit thermal conductivity and therefore cooling. The solution was to change the design for the heatsink to bolt directly to the sevcon.
- The radius of bending at the bottom of the sevcon mount will be very difficult with aluminium as it has a tendency to crack under tight bending radii especially as thickness of aluminium increases – Solution: use angle bought from manufacture to provide the 90-degree angle needed.
- Cost of getting the part externally bent out of aluminium sheet – solution: Use standard parts purchased from metal supermarket at much lower cost.



Figure 7.16 – Manufactured Sevcon mount

The new design of the second mounts is as below:



1) Drill holes on sevcon bracket angle for mounting to floor.

2) Drill holes between sevcon heatsink and sevcon bracket angle.

3) Drill holes on sevcon heatsink for mounting sevcon.

4) Bolt sevcon heatsink to sevcon bracket angle.

5) Bolt sevcon to sevcon heatsink (taking great care).

6) Bolt sevcon bracket angle to locomotive floor.

Figure 7.17 – Diagram to show the new design of the sevcon mount

7.3.5 – Door mounting

Hinges mounted to the top box allow for the side panels to be opened without tools allowing for access to the internals of the body that may need to be frequently accessed when on site at Stapleford railway.

Minor changes to the hinges included changing the hinge material from plastic to stainless steel to allow for stainless steels rivets to be used in attaching the hinges. Meaning that the doors can be opened fully.

7.4 - Internal components

The internal components of the locomotive were revised leading to the need for change in the layout of the body's internal components.

Some of these changes include:

- Changes to the storage of energy obtained during energy regeneration – leading to a removal of two large supercapacitors and the millipaks required to operate them. The energy is now stored back into the 48v battery assemblies.
- Changes to the positioning of low voltage components – all low voltage components moved to be positioned within the control hanger (made possible

due to adaptations made to the locomotive hanger). Freeing up more space on the floor of the locomotive.

- Addition of parts previously missing from the model – it was identified that the DC-DC converter for the sevcons, high voltage fuses and circuit isolation switches required for the high voltage had been missed off previous iterations of the body and needed to be added.



Figure 7.18 – Battery isolation switch

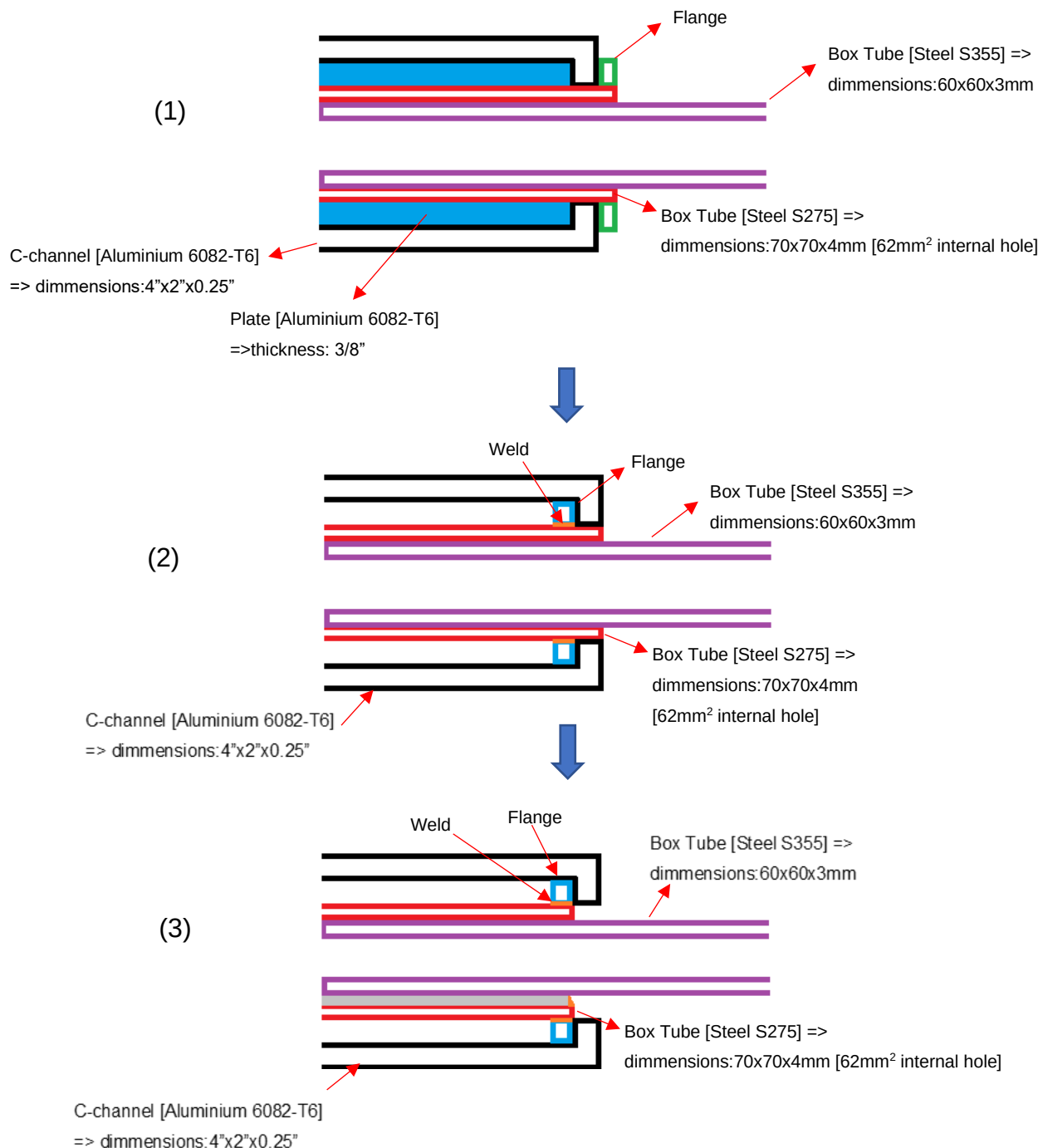


Figure 7.19 – Diagram showing the design stages to attach the jacks to the locomotive

7.5 – Jacking

7.5.1 Design Process for attaching the jacks to the locomotive body

Changes from (1) - (2): Flange moved onto the inside of the c-channel and increased in size, so it is supported by the c-channel (interference fit). This negated the need for the aluminium plate as a spacer

Changes from (2) – (3): The box tube attached to the body no longer sits on the flange of the c-channel. To allow for the welded assembly to be slid into position instead of welded in position, meaning welding is easier as the large c-channels do not need to be present.

Two 4mm aluminium plates were added to the bottom and left-hand side of the [70x70x4mm] box tube so that the jacking box tube [60x60x4mm] will not be able to rattle around and will sit inside the box tube flush.

Note: using a 2.5"x2.5"x1/4" steel box tube for the jacking tube was considered but later discarded as it was not readily available for purchase and the box had to be steel due to welding strength so no other materials could be used (e.g., aluminium)

7.5.2 Calculations to decide the size of the box section that is needed to attach the casters for jacking

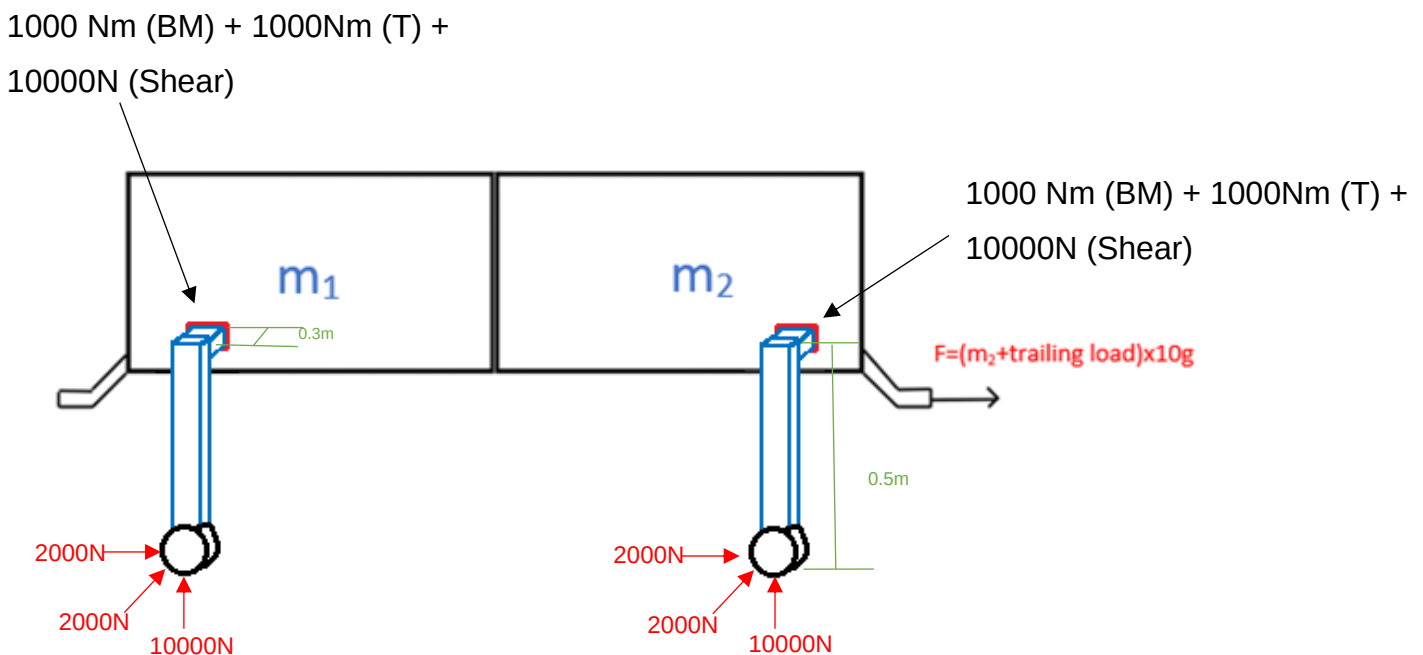


Figure 7.20 – Forces on the jacking gear box section due to jacking

Shear stress on the jacking gear box section

$$\tau_T = \frac{1000}{2 \times (0.056^2) \times 0.004} = 39.86 \text{ MPa} \quad (7 - 9)$$

$$\tau_{VF} = \frac{F}{A} = \frac{10,000}{4 \times (0.057^2) \times 0.003} = 5.58 \text{ MPa} \quad (7 - 10)$$

$$\text{Sum of shear forces} = \tau_T + \tau_T = 39.86 \text{ MPa} + 5.58 \text{ MPa} = 45.4 \text{ MPa} \quad (7 - 11)$$

Tensile Stress on the jacking gear box section

$$\sigma_{BM} = \frac{My}{I} = \frac{1500 \times 0.03}{4.707 \times 10^{-7}} = 95.6 \text{ MPa} \quad (7 - 12)$$

$$I = \frac{0.06^4 - 0.054^4}{12} = 4.707 \times 10^{-7} \text{ m}^4 \quad (7 - 13)$$

$$\sigma_{Tension} = \frac{2000}{0.24 \times 0.004} = 2.08 \text{ MPa} \quad (7 - 14)$$

Equivalent stress

$$\text{Stress equivalent} = \sigma_{eq} = \sqrt{\sigma^2 + (3 \times \tau^2)} = \sqrt{161.42^2 + 3 \times (45.4)^2} = 179.55 \text{ MPa} \quad (7 - 15)$$

The equivalent stress on the box section is 179.55 MPa this is well below the yield stress for mild steel S355 of 355 MPa and therefore is safe for use.

$$\text{Safety Factor} = \frac{\sigma_{eq}}{\sigma_y} = \frac{355 \text{ MPa}}{179.55 \text{ MPa}} = 1.98 \quad (7 - 16)$$

Note: at the welds for the box to the flange the bending moment is not as high as the maximum bending moment in the beam due to bending in the lateral direction, as a result of bending moment distribution in a cantilever beam. This means that the use of welds should be sufficiently strong especially with the safety factor used.

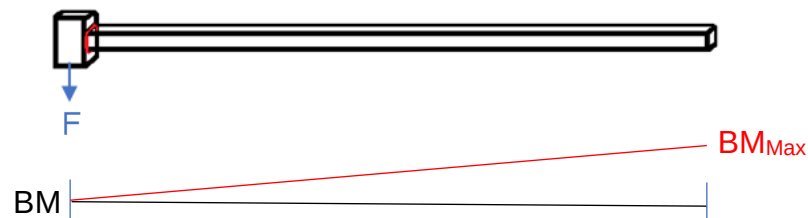


Figure 7.21 – Diagram and bending moment diagram of jacking box tube attached to the jacks

7.5.2 Is the box bracket flange in crushing?

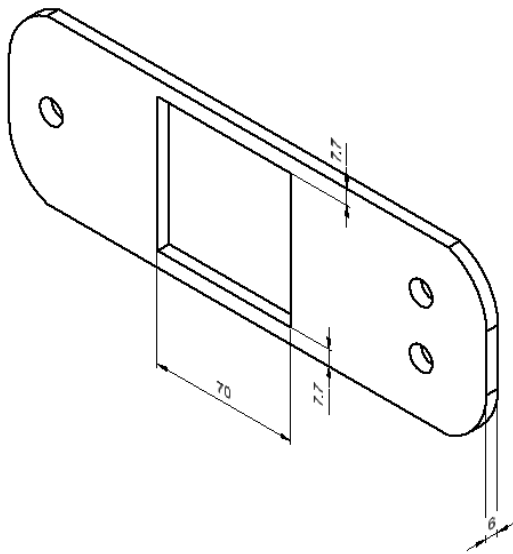


Figure 7.22 Drawing of box bracket flange

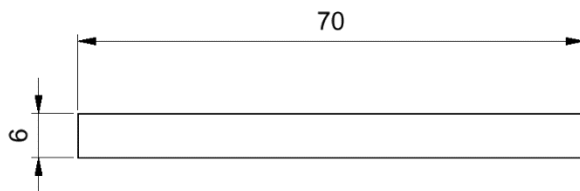


Figure 7.23 – Drawing of the area of the flange in crushing

$$A = (6 \times 10^{-3}) \times (70 \times 10^{-3})$$

$$= 4.2 \times 10^{-3} (7 - 17)$$

$$\sigma = \frac{F}{A} = \frac{10000}{6 \times 10^{-3} \times 70 \times 10^{-3}}$$

$$= 23.8 \text{ MPa} (7 - 18)$$

23.8 MPa < 270 MPa (yield of S275)
and therefore will not fail.

7.5.3 Removal of jacking tube at the centre of the locomotive

The jacking tubes at centre of the locomotive have been removed. This is due to the logic that by cutting large chunks of metal out of the wall of the c-channel you will greatly decrease its strength.

This has the drawback that each half of the locomotive can now not be jacked separately. However, the locomotive will still be able to be jacked as a whole locomotive. The main objective of jacking of the locomotive. Additional changes as a result of this is that the floor was then supported in less areas. Meaning a reversal of a previous change of the floors thickness back to 6mm from 3mm to increase its bending strength. More analysis of the connection between the 60x60x4mm box tube, and the flange to connect to the jacks is available in Appendix C-4.

7.6 - Manufacturing improvements (Body)

Table 7-2: Improvements made during manufacturing of the body

Lessons learned	Method to rectify	Design Changes
Box Tube flanges incorrect size as c-channel flange size changed after purchase	Milled 1.75mm off the sides of each flange.	The walls of the flanges then reduced in thickness. Crushing calculations were then conducted to

		ensure the flanges still have sufficient strength.
Cross member originally to be bent by acorn lasers. However, bends of aluminium would be weak and prone to breaking in bending.	The design of the cross member was changed to use standard aluminium parts from metal supermarket. Also reducing price.	A flat piece of aluminium sheet was rivetted to the aluminium c-channel that would create the flanges to mount the cross member in place.
The steel S275 box tube was too long as the decision was made to not have the box tube sit on the flange of the c-channel after purchase. To allow for the welded assembly to be slid into position instead of welded in position, allowing for easier welding.	A metal cutting band saw was used to shorten the length of the steel S275 box tube.	The box tube no longer sits on the flanges of the c-channels but results in no other design changes.
The bolts to attach the sevcons mount to the floor were below the heatsink. This did not matter as the holes were threaded but easier access could be provided by modification.	Moving the holes further apart on the angle.	Moving the holes further apart on the angle.
Lining up the M3 bolts on the top box to mount it to the internal wall assembly's proved to be difficult.	4.8mm Stainless steel rivets (16 per halve) were used to attach the top box instead. Bigger holes e.g. 4mm for the M3 bolts could have been used. However, this was against as the bolts would rattle about more and using rivets was deemed to be the easier method as inserting and removing 24 M3 bolts would already prove to be time consuming.	The top box could no longer be removed being a drawback. However as long as the control hangers were placed in before the top box was attached all other components can still be removed and inserted. It should be noted that accessibility is however decreased.

7.7 Nose Cones (JO)

A review of the overall shape of the nose was performed. The design needed to update to match the latest perimeter of the locomotive and it also had to fall within the kinematic loading gauge. Attempts were made to define the nose geometry with hand calculations though these proved unreliable with exception to the tip and backplate.

A different approach was taken to defining nose geometry within the dynamic loading gauge, using inventor to draw a model the locomotive and drawing the constraining lines on top:

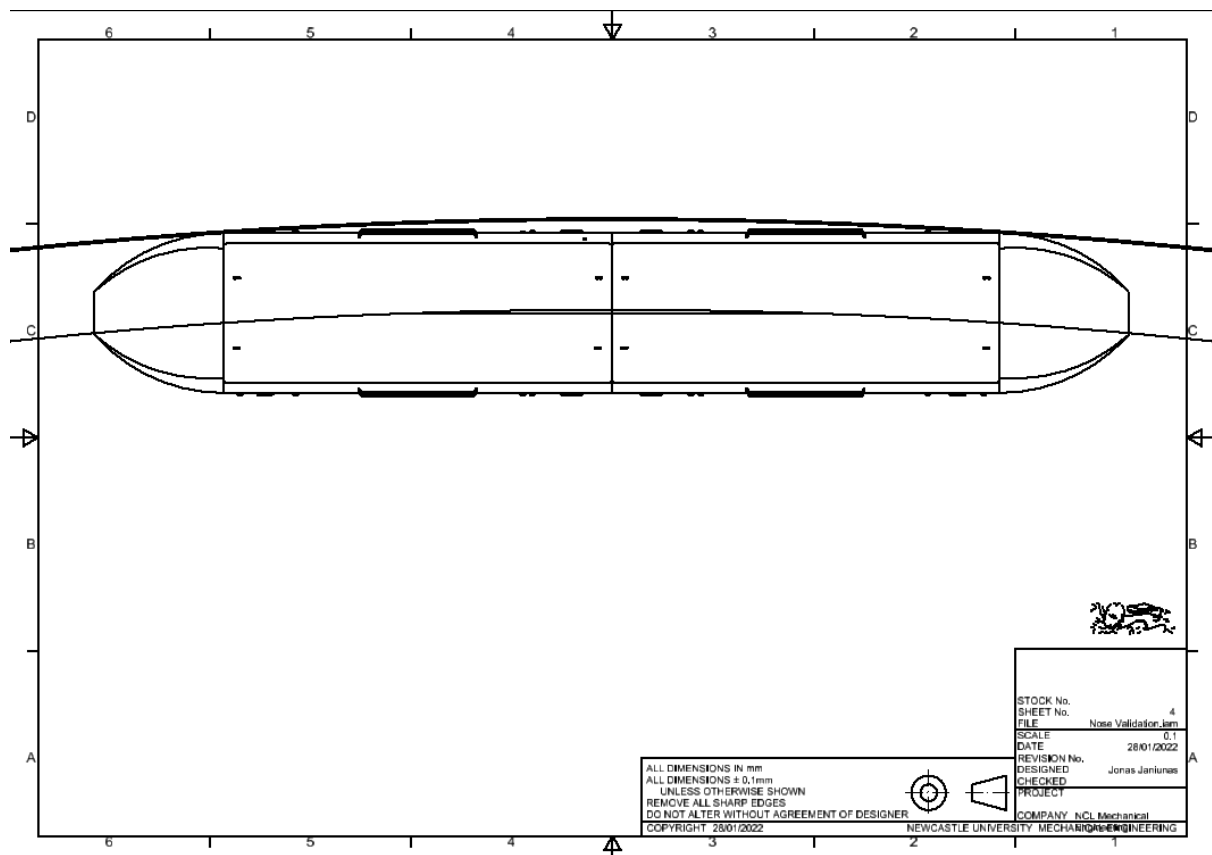


Figure 7-7-1: Autodesk Inventor drawing of body overhang.

It should be noted that the nose exceeds the dynamic loading gauge, though this is only for less than 100 microns, and upon consultation was determined to be a nonissue.

The CAD design of the nose was exceedingly complicated. Converting the nose to sheet metal was a more arduous task than first anticipated. The nose geometry is defined by a series of work planes and work axis, upon which drawings are created to

curve, extrude and cut the nose into shape. The size elevation and curvature of the nose cone were experimented with, creating a shorter, stubbed nose.

Upon review it was determined that the design was satisfactory and any change to the curve diameter and size elevation wouldn't be aesthetically pleasing. Due to the complex geometry of the nose, each one must be made from 8 parts of sheet metal, as opposed to the original intended 6.

The type of aluminium chosen was 5754 due to its weather resistance, weldability and most importantly, ductility. There were discussions about the nature of the nose, initially being two halves welded together, top and bottom, but this brought complications with access to the fire extinguisher and other tools residing inside the nose. The design was adjusted to include a hinged mount on the interior roof of the top plate. Since the shaping and cutting of the sheet metal was pre-paid for, the only assembly was the welding of the lower and upper portions of the nose, as well as mounting the assembled noses to the body, reducing manufacturing time and simplifying assembly. Both noses are identical and match the outline of the body, making incorrect assembly incredibly difficult.

Initially the nose was to be created by hand from aluminium sheet of approximately 1mm. A scale plot of the unfolded pattern for the respective parts of the nose would be created and fixed to the aluminium, using tin snips to cut the metal to shape. The parts would be rolled appropriately and then welded on the internal seam. This was deemed needlessly complicated and although much more adaptable to complications with geometry, would consume valuable time. Therefore, the components are to be order bent to shape from acorn laser, with the intention of welding remaining the same.

8. Dynamic Loading Gauge (JO, DS)

8.1 Significance of Dynamic Loading Gauge

Each time the dimensions pertaining to the lateral and vertical extension of the locomotive were adjusted, a review of the kinematic loading gauge had to be performed. It was imperative that the locomotive fell within the specified loading gauge as residing outside of these dimensions would result in a failure to compete. The constraints of the kinematic loading gauge can be seen in the IMechE handout appendix K-1. The dynamic loading gauge was critical as it helped define the allowable motion and exterior dimensions of the locomotive.

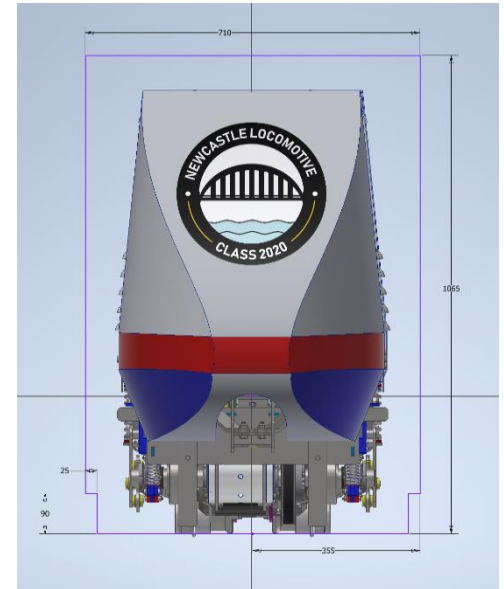


Figure 8-1-1 2022 Locomotive within Dynamic Loading Gauge

8.2 Initial calculations

Calculations began on December 7th when an entire model of the locomotive was assembled. The key areas of impactful motion were identified, laying the foundation for all further calculations. These were the horizontal and vertical components caused by curving, primary suspension, and secondary suspension.

Though the degrees of freedom and movement had been correctly identified on the locomotive, the calculations associated with them need to be revised.

Though the degrees of freedom and movement had been correctly identified on the locomotive, the calculations associated with them need to be revised.

8.3 Secondary calculations

Following the realisation that the calculations were incorrect, a meeting was held on December 13th with the project supervisor to amend these mistakes. The most complex of these calculations were primary and secondary roll.

During this review two important values were identified, the position of the body's widest extent and the transverse tipping force at the centre of mass. The dynamic loading gauge varied considerably from where it was measured in relation to height, identifying the worst heights on the locomotive which occurred at the body extremities allowed for a simplification of the calculations into 2 criteria: the locomotive had to fit within the dynamic loading gauge when measuring from the roof, and when measuring

from the bolt heads on the c-channels. If the locomotive met these parameters, it would meet all others. The transverse tipping force was also important as it helped constrain primary and secondary spring travel, as well as identifying the hazardous forces.

With the realisation that the model of the locomotive was outside of the dynamic loading gauge, several suggestions were made in order to assist in lessening the overhang and reduce displacement.

The first suggestion was to change the bogie centres, the distance of which impacted the overhang specifically. First the optimum centre distance for the bogies was determined, where the parameters w and y are approximately equal. This was achieved through trial and error. This distance was deemed unusable due to collision with rigid body parts, thus remained unchanged.

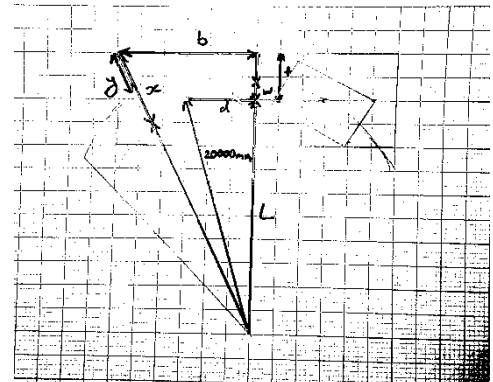


Figure 8-3-1: CAD sketch of dynamic loading gauge overhang.

Another suggestion was to increase the antiroll bar stiffness. The antiroll bar's initial design was a hollow steel tube, changing the hollow tube with a stiffness of 2149.2Nm/rad for a solid bar of stiffness 2827Nm/rad which corresponded to a change in lateral displacement of 0.17mm. This change alone was insufficient. Another way to increase the stiffness of the antiroll bar was to reduce its length, originally 120mm to 60mm reducing the lateral displacement by a further 0.61mm.

A third suggestion was to cut material from the body. By physically removing the exact amount of material at the points of overhang, it would eliminate all problems with the dynamic loading gauge immediately, though this would create other, more pertinent problems. This would have been a last resort after all other attempts.

8.4 Final Results

Modifications had to be agreed upon for the locomotive to be confined within the loading gauge. To allow the entire locomotive to fit, the anti-roll bar stiffness was increase by using a solid torsion bar, and the lateral displacement of the secondary suspension was limited to $\pm 15\text{mm}$. To accommodate the body at the maximum width, the types of bolts were changed from hex cap socket bolts to button head socket bolts, reducing the bolt distance on either side by 5.4mm. The adjustment of the bolts not

only assisted dynamically, but also improved the visual appearance with a slimmer, smoother figure; and structurally, distributing the large forces over a wider surface area. Detailed calculations of the displacements are found in appendix G-1.

Table 8-1. Causes and Magnitude of Displacements of Locomotive and Total Safety Clearance. Appendix G-1.

Dynamic Loading Gauge (DLG) Summary		
Cause of Movement	Deflections/mm	
	A (Top of Body)	B (Widest point on body)
Primary Suspension Laterals	10	10
Secondary Suspension Laterals	15	15
Primary Roll	21.7	6.8
Secondary Roll + ARB	6.9	4.1
Horizontal Curvature	31.5	31.5
Total Deflection	85.1	67.4
Clearance of DLG	26.9	6.5

9. Low Voltage Electronics & Control (AH)

9.1 Fire Detection & Battery Temperature Monitoring (AH)

9.1.1 Op-amp & Comparator Selection

Initially, LM741 op-amps were specified as the default op-amp for all low voltage circuits due to their relatively low cost, wide availability, and good overall functionality and performance. Specifying uniform component models was desirable, allowing lower prices to be accessed through bulk purchase and enabling simpler repair, requiring a smaller stock of reserve components. Alternative op-amps were used in specific circumstances, the reasons for which required justification.

Comparator integrated chips (ICs) were not previously specified. It is possible to use an LM741 op-amp as a comparator, however, comparators with multiple channels exist,

Table 9-1: Comparison of Key Data of LM393P (10) and LM741 (11)

	<u>LM741</u>	<u>LM393P</u>
Channels	1	2
Pins	8	8
Price	£1.02	£0.50
Supply Current	2.8mA	0.8mA

allowing multiple comparison operations to be undertaken by a singular IC. The specification of an LM393 comparator IC was compared to an LM741, of which key comparisons have been listed in Table 9-1. LM393 ICs were chosen as each one replaces two LM741 ICs, saves £1.54 and reduces supply current draw by 4.8mA, preserving battery capacity.

9.1.2 Fire Detection Circuit Testing

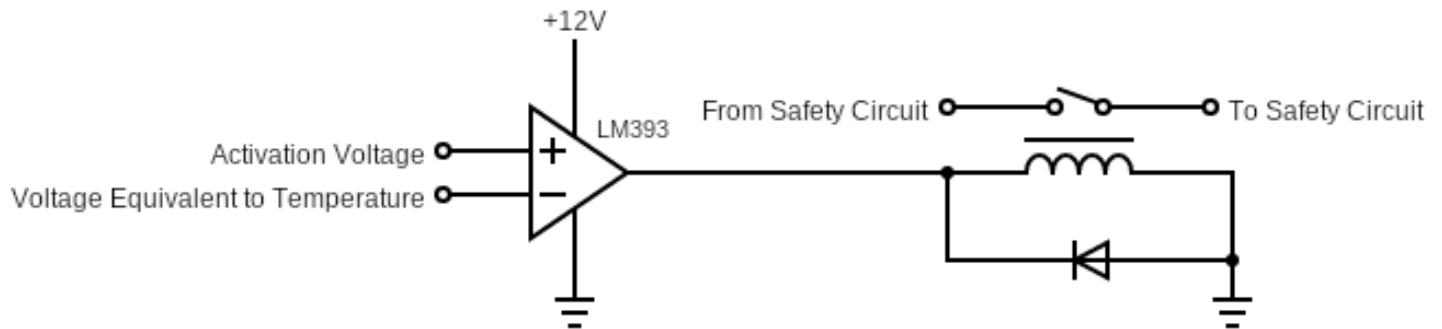


Figure 9-1: Using an LM393 for comparison of temperature voltage to activation voltage in the fire detection system

The fire detection circuit designed by Alex Hooper in the previous academic year was created on a breadboard for testing. Testing of the arrangement shown in Figure 9-1 found the use of an LM393 to be unsuitable as the high output was incapable of energising the relay coil. The LM393 output was measured as 11.8V and 20mA by a Multimeter. The relay is rated at 12V with a 50mA minimum switching current. Thus, the inability of the LM393 to energise the coil and activate the safety circuit alarm was determined to be due to an output current of 30mA less than required.

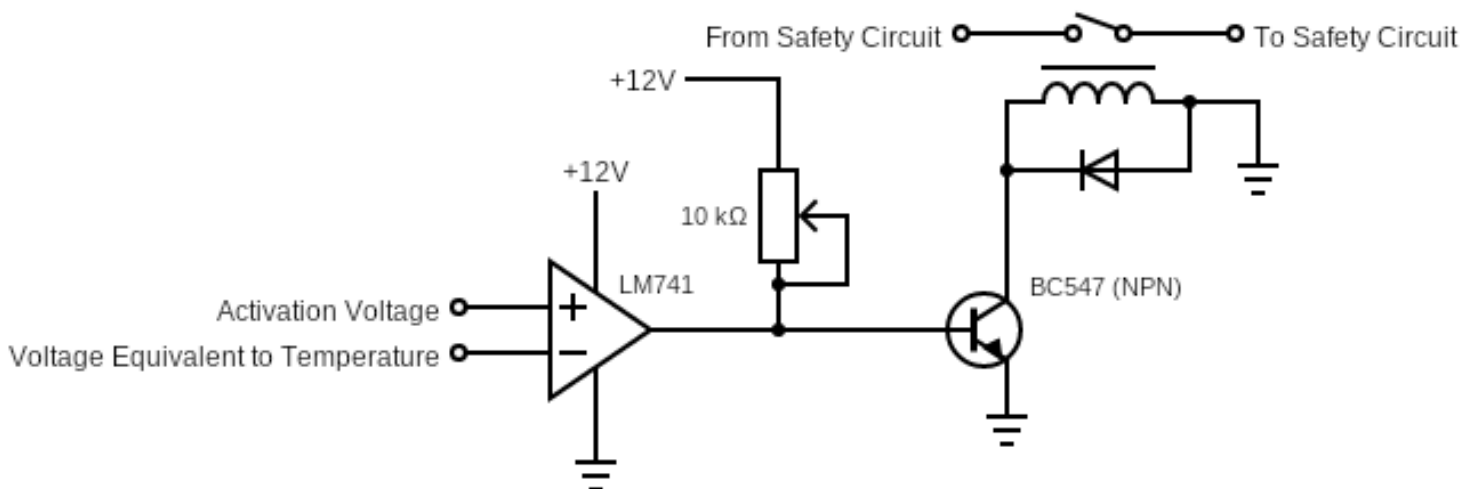


Figure 9-2: Replacing an LM393 with an LM741 outputting to a BC547 with 10kΩ potentiometer acting as a pull-up resistor

The LM393 IC was replaced with an LM741, connected by a pull up resistor (10k Ω potentiometer) to a 12V supply rail and then to the base of a BC547 transistor as shown in Figure 9-2. This configuration allowed the required 50mA switching current to be pulled from the 12V rail and through the relay coil when the base of the transistor is activated by the output of the LM741, energising the relay coil. However, upon the output returning to low, the relay coil did not de-energise causing the alarm to remain active despite the system not detecting the presence of a fire. It was observed that in the low state, the output voltage of 2.17V was preventing the contacts returning to the normally open position. Further research uncovered that an inherent feature of the LM741 is that it is incapable of outputting voltages close to either of its supply voltages. Therefore, an op-amp with 'rail-to-rail' output was sought, allowing an output up to the limits of the positive and negative supply. Therefore, if the negative supply is grounded, the output tends to 0V when low rather than 2.17V. An AD820 was specified as it can achieve a 'rail-to-rail' output. An AD820 is significantly more expensive (£5.84/unit) than an LM741 (£1.02/unit), however it enables the correction function of relay control. The transistor was replaced with a BS270 power MOSFET as it drew less current (168mA) compared to the transistor (209mA) when energising the relay coil. Figure 9-3 illustrates the adaptations, which were tested successfully.

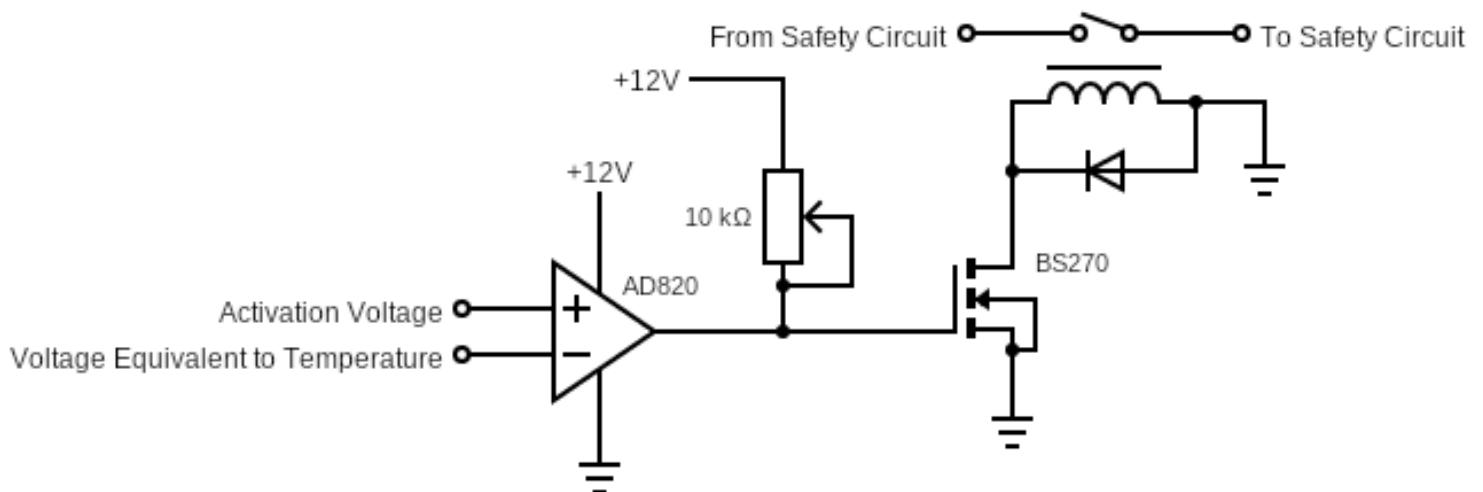


Figure 9-3: LM741 replaced with an AD820 and BC547 transistor replaced with BS270 power MOSFET

9.1.3 Combining Fire & Heat Detection Circuits (AH)

The fire detection and battery temperature monitoring circuits designed by Alex Hooper in the previous academic year were two independent circuits with each half of the Loco having one of each. The two circuit designs used a similar principle; both compared voltage across a NTC (Negative Temperature Coefficient) thermistor (equivalent to temperature) and compared it with a reference voltage to determine whether the temperature is above or below a threshold temperature.

Figure 9-4 illustrates the behaviours of NTC thermistors. The fire detection system energised a relay

to trigger the safety circuit, as explained in Section 9.1.2. The heat detection circuit utilised LM393 comparators to determine a range of which the voltage equivalent to temperature was within and illuminate a coloured LED as an indication.

The logical development of these two circuits was to combine them such that the systems share thermistors. This halves the number of thermistors required, reducing costs by £17 and halving the number of components requiring disconnection during battery removal, an important consideration for the refuelling challenge. The merge was simple, with the output from the thermistors feeding all the comparators and op-amps from both systems. Figure 9-5 illustrates the merge, where the red circuitry is that of the fire detection circuit and the blue circuitry of the battery temperature monitoring system. Pre-merge, the two circuits were completely independent and so the connection shown in green did not exist. Post-merge the green wire allows the use of a singular thermistor and voltage follower configuration as the temperature voltage for both the fire detection and battery temperature monitoring system, allowing the circuitry highlighted in yellow to be removed. The sections of circuitry shown would be scaled to accommodate either eight or nine monitoring ones in each half of the loco, while additional AND gates and comparators have been omitted for clarity. The fully, successfully, tested circuit is available in Appendix D-1 and D-2.

The merge increases the complexity of the fire detection system which, considering is a mandatory safety critical circuit, is desirable to minimise componentry and

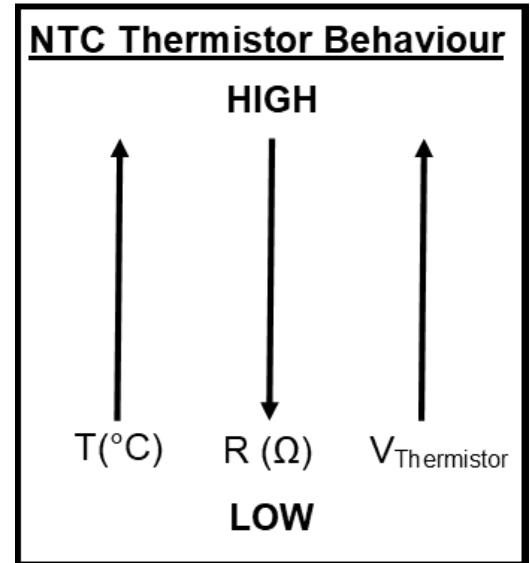


Figure 9-4: NTC thermistor behaviour

complexity to reduce the likelihood of a fault developing as any fault with the fire detection system would require operation of the Loco to cease immediately.

9.1.4 Fail-safe Adaption

Table 9-2: Circuit Logic Before & After Fail-safe Adaption

		<u>Logic Before Fail-safe Adaption</u>		<u>Logic After Fail-safe Adaption</u>	
<u>IF Statement</u>	<u>Indicating</u>	<u>AD820 Output</u>	<u>Relay</u>	<u>AD820 Output</u>	<u>Relay</u>
$V_{\text{Thermistor}} < V_{\text{Fire}}$	No Fire	LOW	De-energised (NC)	HIGH	Energised (NO)
$V_{\text{Thermistor}} > V_{\text{Fire}}$	Fire	HIGH	Energised (NO)	LOW	De-energised (NC)

Best practice dictates that the design of safety critical systems should be such that they are fail-safe, which in the case of fire detection means including mechanisms to identify faults with the system. Table 9-2 details the circuit logic pre-adaption where, in the event of activation, the relay coil is energised. However, if a fault resulting in loss of power to the thermistors or comparators were to develop, then a scenario could develop where a fire is not detected as the power fault would prevent the relay coil from energising. Figure 9-6 illustrates the circuitry before the adaption.

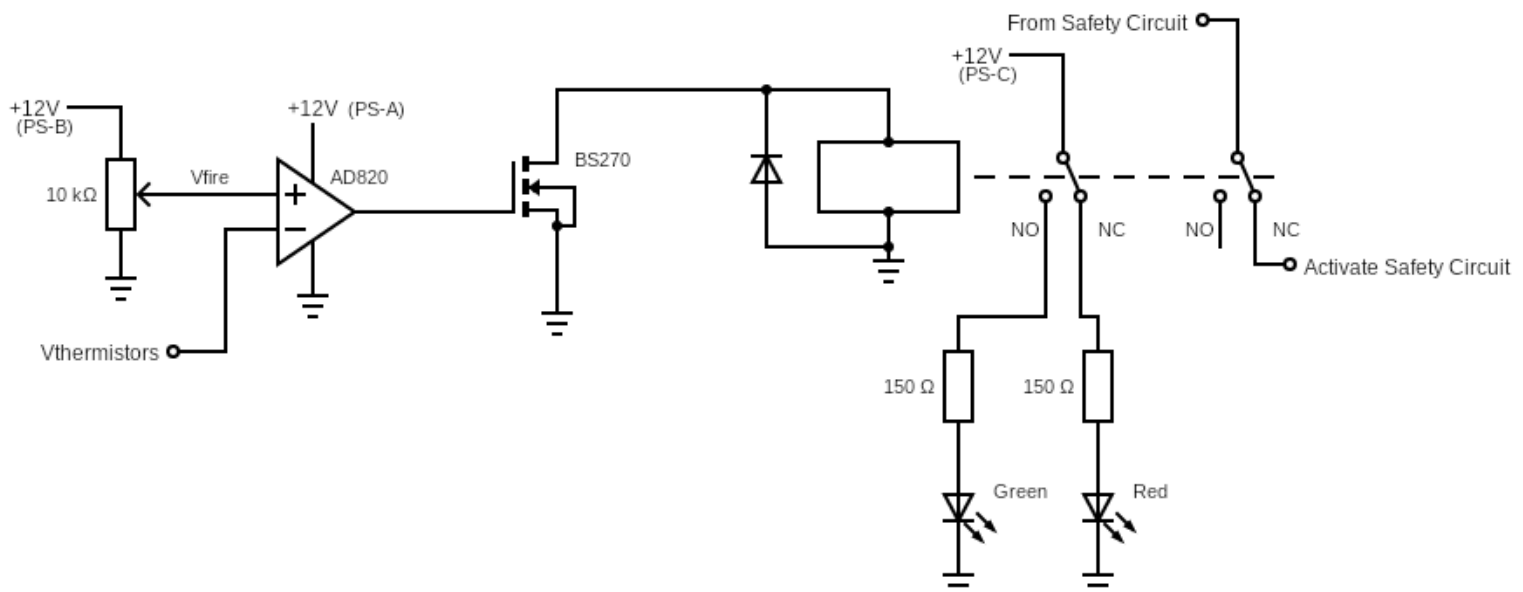


Figure 9-6: Fire detection circuit before fail-safe adaption

Therefore, the circuit was modified (shown in Figure 9-7) such that the relay coil is to be energised when the system is not activated such that any voltage drop across the thermistors or ICs will cause the relay coil to lose current, de-energise, and thus activate the alarm. This adaption does not distinguish a fault from a fire and so requires investigation by an operator upon alarm activation. This improvement also doesn't account for a fault with the power supply of the safety circuit; however, it much improves the fault detection capabilities of the fire detection system.

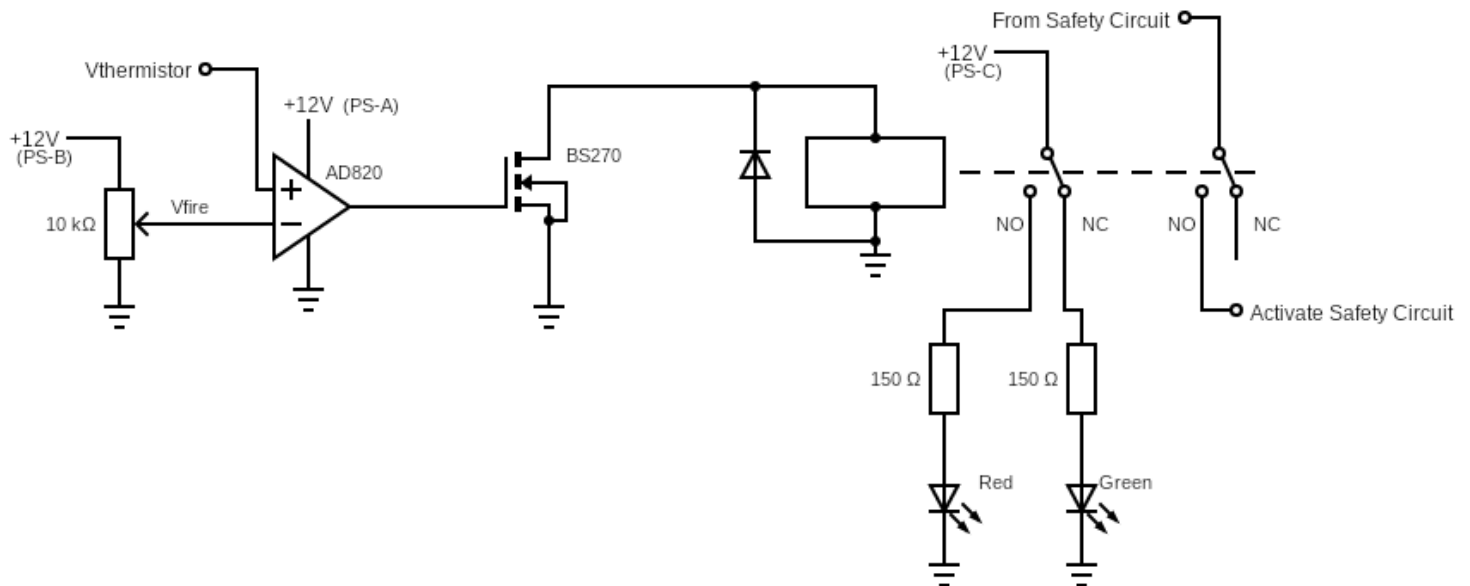


Figure 9-7: Fire detection circuit after fail-safe adaption

9.2 True Ground Speed Monitoring Method (AH)

9.2.1 Method Justification & Motor Selection

The research and development undertaken by previous team member Jack Robson determined the most cost effective and accurate method of true speed monitoring is to use a pair of motors operating as generators, with one for each rail. The output voltage of these motors, turned generators, can then be used to imply an operating speed based on the radius of the wheel and rating of the motors. To ensure accurate speed measurement, it was thought a high-quality motor would be required. However, high quality DC motors can cost upwards of £200. The decision was therefore made to purchase a Nidec 12V motor at a cost of £26.08. This motor was connected to an oscilloscope; however, the output was found to be unsuitable and so alternative motors were sought.

It was discovered that the University had high quality Trident 24V motors available which were, at the time, not in use. It was agreed, with thanks to Mr Peter Chapman, that two of these motors could be donated to the Loco project. Figure 9-8 displays the waveform output of one of these motors, which was significantly smoother compared to the cheaper Nidec motor. The two motors donated cost £175.04 each and so this kind of donation presented the team with a significant cost saving. The addition of a

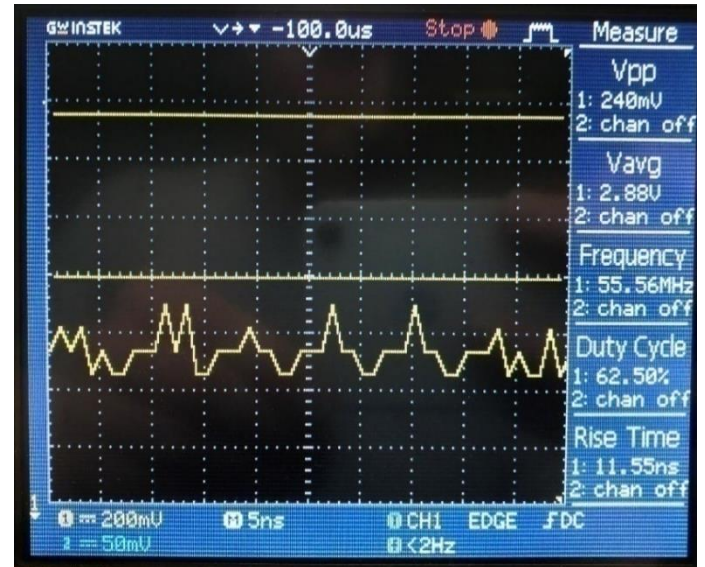


Figure 9-8: Trident 24V motor output

polarised capacitor at the output further improved the smoothness of the output, as shown in Figure 9-8 (200mV intervals). The generators output signal fed into an LM741 op-amp. However, the generators are rated at 24V while the maximum input voltage for an LM741 is 15V. The output voltage from the generators at 15mph is expected to be 16.7V. The decision was therefore made to use an AD820 on a single supply of 0-24V, as it's specified to supply voltages up to 30V. Zener-diodes were considered to 'clip' the voltage, however the ~0.7V potential drop would need to be considered, thus reducing the accuracy of the speed measurement as the true voltage drop would not be exactly known and may not be constant.

9.2.2 True Ground Speed Calculation

Assumptions:

- No load on generators
- Voltage output is linear to the rotational speed of the motor
- Roller has an external diameter of 80mm, as justified in Section #.

Based on the above assumptions, Equation 9-1 can be used to determine the true ground speed of the loco. Where: $\bar{V}_{gen}$ = Average voltage output of generators

$$LocoTrueGroundSpeed \left(\frac{m}{s} \right) = \frac{1.92\pi}{2900 \times \bar{V}_{gen}} \quad (9 - 1)$$

9.3 Control Circuit (AH)

9.3.1 Overview

The control circuit is based upon the original work undertaken by James Tordoff and modifications made by Alex Hooper during the previous academic year and is based upon analogue componentry on the basis that previous attempts to use micro-processors were unsuccessful. Voltages in the range of 0 to 10V are used to indicate an acceleration or brake demand. Full deceleration is equivalent to 0V, with 10V indicating a full acceleration demand, thus making 5V 'neutral'. Acceleration and deceleration demands are linear between the voltage extremities albeit with a small dead band around 5V. The control circuit has four main functions: driver specified acceleration/deceleration, cruise control, and WSP (Wheel Slip Protection).

The control circuit was reviewed, the following issues were highlighted:

- Outdated analogue multiplier selection for creep control
- Lack of brake blending and integration with EBV (Electronic Brake Valve)
- Lack of forward/reverse functionality
- Lack of functionality to set Pin 22 of the Sevcons to 5V during WSP activation
- Frequency to voltage conversion method not being validated with testing

9.3.2 Analogue Multiplier Selection

The operation of multiplication is required in both the control circuit and Auto-stop circuit. Op-amps are incapable of performing a multiplication operation and thus bespoke ICs were sought, specifically analogue multipliers. Previous team member James Tordoff recommended an AD534, however, this IC was specified two years ago and so it was decided that a review of the selected multiplier should be undertaken. Research discovered that an updated version of the AD534, the AD734, was available at a lower cost than the AD534 and with significant improvements. This multiplier and two others were compared, as listed in Table 9-3.

Table 9-3: Analogue Multiplier Key Data Comparison

	AD633	AD734	AD532	AD534
Price	£12.23	£32.84	£86.52	£52.87
Accuracy	2%	0.1%	1%	0.25%
Non-linearity	0.1%	0.05%	0.5%	0.2%
Frequency	1MHz	10MHz	1MHz	1MHz

An evaluation table (Table 9-4) determined an AD734 as the optimal (lowest scoring) selection. The evaluation factors were ranked as follows:

- Price: Most important due to lack of sponsorship and small budget at the time.
- Accuracy: Important as small voltages are to be multiplied, with accuracy to 0.01V expected.
- Non-linearity: Important for consistent behaviour of control system but not as important as accuracy.
- Frequency: High speed performance not deemed important.

Table 9-4: Evaluation Table for Creep Control Analogue Multiplier

	<u>AD633</u>	<u>AD734</u>	<u>AD532</u>	<u>AD534</u>
Price (1x)	1	2	4	3
Accuracy (0.8x)	4	1	3	2
Non-linearity (0.4x)	2	1	4	3
Frequency (0.1x)	2	1	2	2
Score	5.2	3.3	8.2	6

The Auto-stop multiplication operation has slightly different requirements and so the evaluation factors were modified. The voltage being multiplied is expected to be significantly larger and less sensitive to 0.01V changes, reducing the importance of accuracy. Linearity was also deemed to be less important as it is squaring a constant voltage and so the voltage output is expected to be near constant. Modified scaling factors were applied (Table 9-5), where an AD633 was specified for Auto-stop.

Table 9-5: Evaluation Table for Creep Control Auto-stop Multiplier

	<u>AD633</u>	<u>AD734</u>	<u>AD532</u>	<u>AD534</u>
Price (1x)	1	2	4	3
Accuracy (0.2x)	4	1	3	2
Non-linearity (0.2x)	2	1	4	3
Frequency (0.1x)	2	1	2	2
Score	2.4	2.5	5.6	4.2

9.3.3 Forward/Reverse Functionality

The initial circuit relied on the generators outputting a negative voltage in reverse and a positive voltage in forwards. This arrangement increased complexity of the subsequent control circuit and thus it was desirable for the output to be positive regardless of direction. Grounding the generators at 12V rather than 0V was considered, such that 0-12V would be reverse and 12-24V would be forward but this would still result in complexities in the control circuit. It was thought that a DPDT relay could be used to switch between forward and reverse, as shown in Figure 9-9. The

generators are to be wired such that the output is always negative, thus when passing through an inverting op-amp the voltage output from the circuit is to be positive.

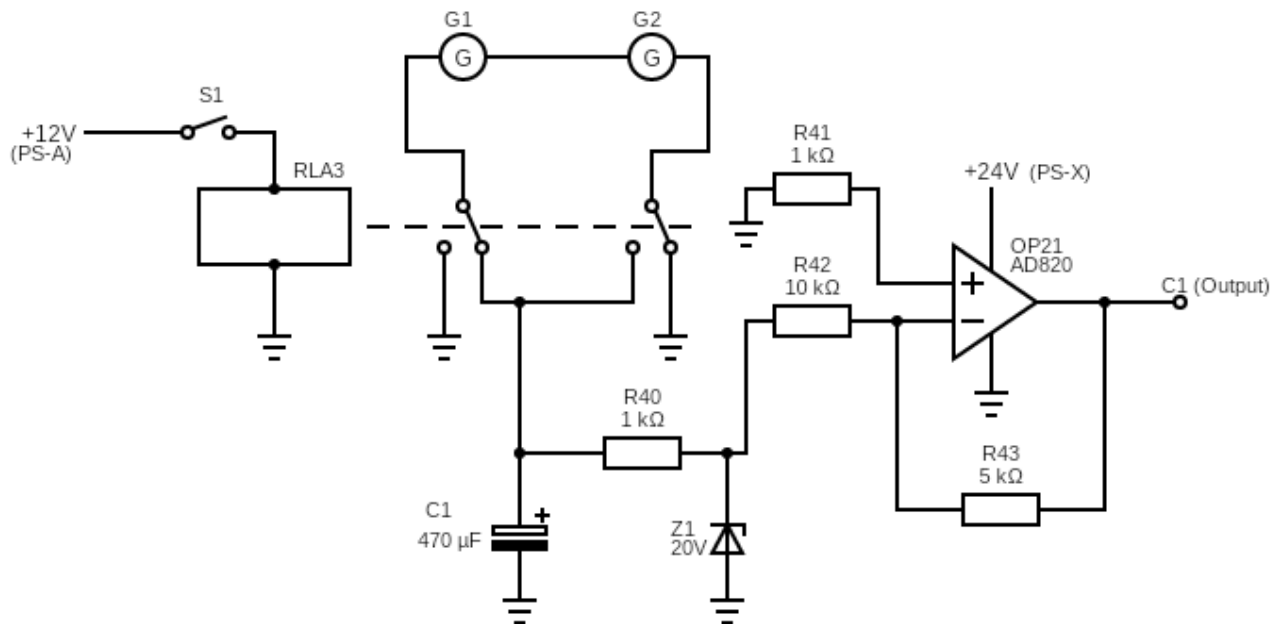


Figure 9-9: Forward and reverse functionality circuit

9.3.4 Fault Provision

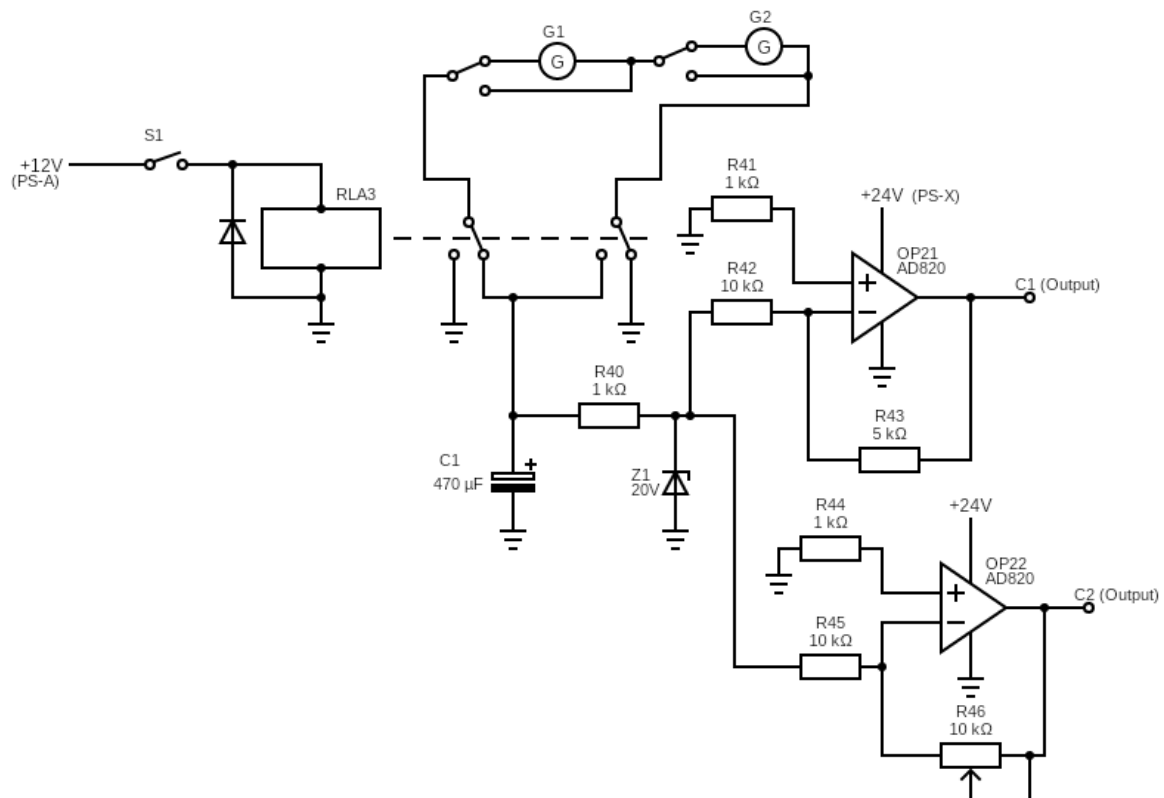


Figure 9-10: Fault provision and additional output modifications

The creep control and cruise control systems both rely on the output of the speed rollers and so in the event of either speed roller failing, the control system would be susceptible to malfunction. The true ground speed monitoring circuit was therefore modified to include two sets of switches which would allow one or both speed rollers to be bypassed in the event of a malfunction, as displayed by Figure 9-10. In the event of both rollers failing, a switch was added to the control circuit to allow the motor speed signal from the Sevcons to be used as an alternative. See Section 9.2.5 for details. The circuit was then modified to include a second output, C2. This output is to allow for a bespoke scaling factor to feed the capacitor in the Auto-stop challenge, to account for time taken to charge as discussed in Section 9.3.4.

9.3.5 Brake Blending

The EBV specified was supposed to be controlled using 0-5V DC input. However, the incorrect model was previously ordered and due to the cost (£495.53) it was prohibitive to order the correct model. The model ordered is controlled using 4-20mA current loop. From the datasheet, the input impedance was found to be 250Ω. Thus, substituting $I=4\text{mA}$ & $I=20\text{mA}$ into Ohms Law, it was found that the voltage was of the range 1V to 5V. Thus, voltage can be used to control the EBV, albeit in the range of 1 to 5V rather than 0 to 5V, where 1V is no brake demand and 5V is a full brake demand. Figure 9-11 shows the circuitry designed to control the EBV.

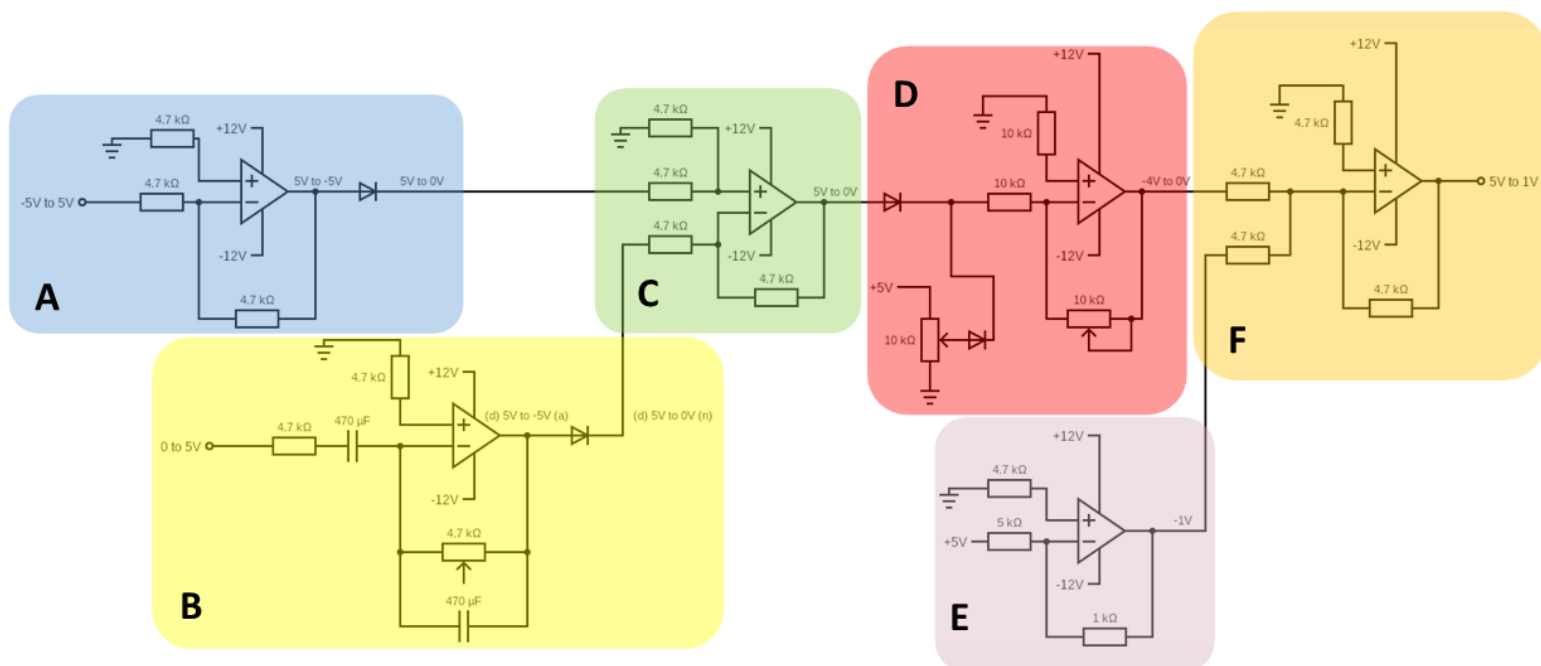


Figure 9-11: EBV control circuit

- **A:** Input of -5V to 5V from the power brake handle and cruise control system demand. The signal is inverted such that 5V is equivalent to full deceleration demand and -5V to full acceleration. The inversion allows the signal to be passed through the diode, effectively removing the negative control voltage thus only the deceleration demand voltage is considered.
- **B:** The average motor speed from the Sevcons is differentiated. The inverting input is used thus deceleration results in a positive output. Diode removes the acceleration (negative voltage) demand.
- **C:** The difference between the required deceleration and motor deceleration is output in the form of 5V to 0V, where 5V is equivalent to full brake demand.
- **D:** The deceleration difference is added to the deceleration demand from the brake handle and is scaled to 0 to 4V. The adding op-amp inverts the signal thus the output is negative.
- **E:** 5v input is used to generate a -1V output.
- **F:** The -1V is added to the -4 to 0V brake demand signal to shift it. The inverting input is used thus the output is 1 to 5V, where 5V is full brake.

9.3.6 WSP Issue

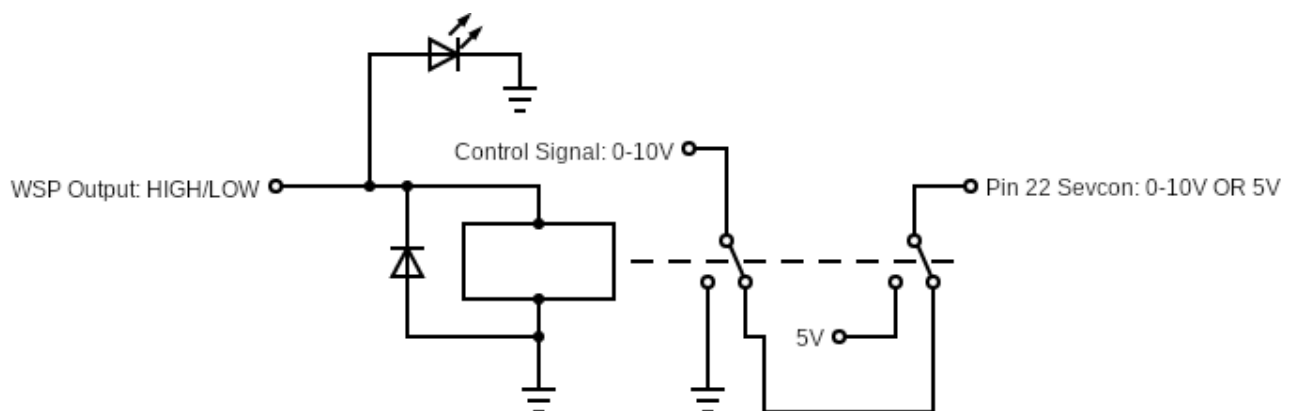


Figure 9-12: WSP output to Sevcon modification

The existing WSP system was successfully tested. However, the previous control circuit did not account for the fact that in the event of WSP activation, the voltage signal to Pin 22 of the Sevcons cannot be set to 0V as that is equivalent to full deceleration and thus needs to be set to 5V. Therefore, a DPDT relay was integrated into the control

circuit as shown in Figure 9-12, whereby the control signal passes through the first set of contacts such that if the WSP is activated, the motor controllers are set to 5V instead of 0V.

9.3.7 Frequency to Voltage Conversion

The Sevcon motor controllers are capable of outputting a frequency which is equivalent to the motor speed. The creep and cruise control systems require this measurement in the form of voltage to allow comparison with the true ground speed observed by the speed rollers. Therefore, a method of frequency to voltage conversion was required. The work of previous team members led to the recommendation on an LM2907 IC. Two of these chips were purchased at a cost of £2.71/unit to allow testing to determine suitability. The configuration in Figure 9-13 (9) was replicated for this testing. The frequency input at pin 1 was simulated using a function generator, outputting a sine wave of between 1 to 200Hz. As expected, the output at pins 4 and 7 was found to vary with input frequency.

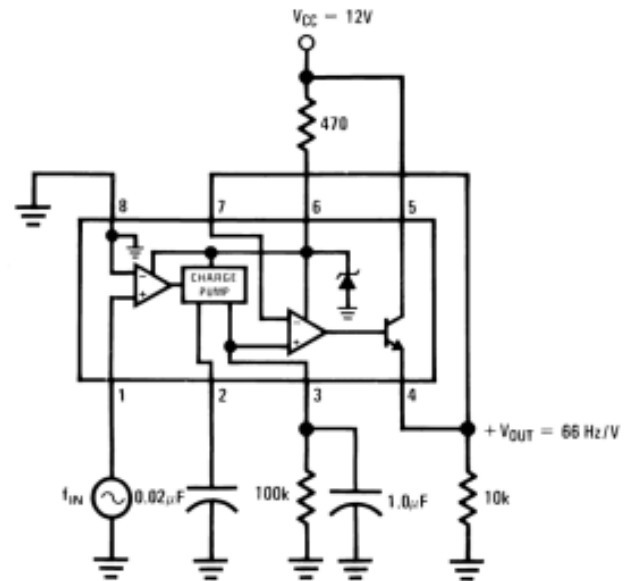


Figure 9-13: LM2907 frequency to voltage conversion circuit (9).

9.3.8 LM2907 Output Smoothing

The output was found to not be entirely smooth, see Figure 9-14 (yellow sine wave is the input and the blue wave is the output). For the same input, it was found that the addition of a 25V at the output pin improved the smoothness and reduced V_{pp} (peak to peak voltage) from 180mV to 80mV (Figure 9-15).



Figure 9-14: LM2907 output without polarised capacitor

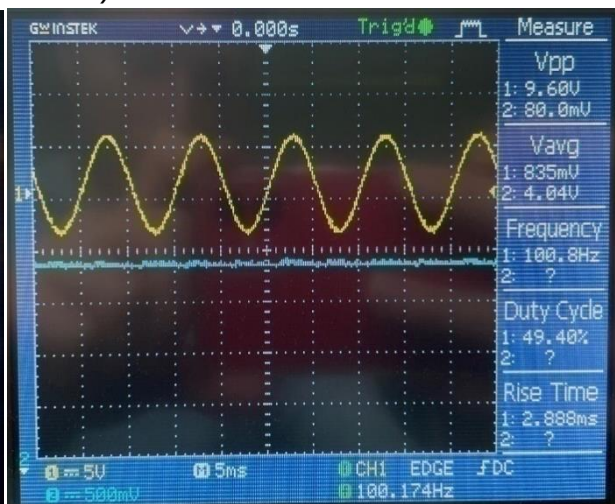


Figure 9-15: LM2907 output with polarised capacitor

9.3.9 LM2907 Linearity Testing

The use of a function generator was used to input sine waves of various frequencies while the output voltage was observed on an oscilloscope to calculate the linearity of the output. Output voltage was recorded for a range of frequencies from 0-180Hz at 10Hz intervals and was then plotted against input frequency. Linear regression r^2 analysis was undertaken on the data to quantify the linearity of the conversion. The r^2 value was found to be 99.9939%, indicating that the conversion is almost entirely linear, validating the decision to specify the use of an LM2907 for the frequency to voltage conversion.

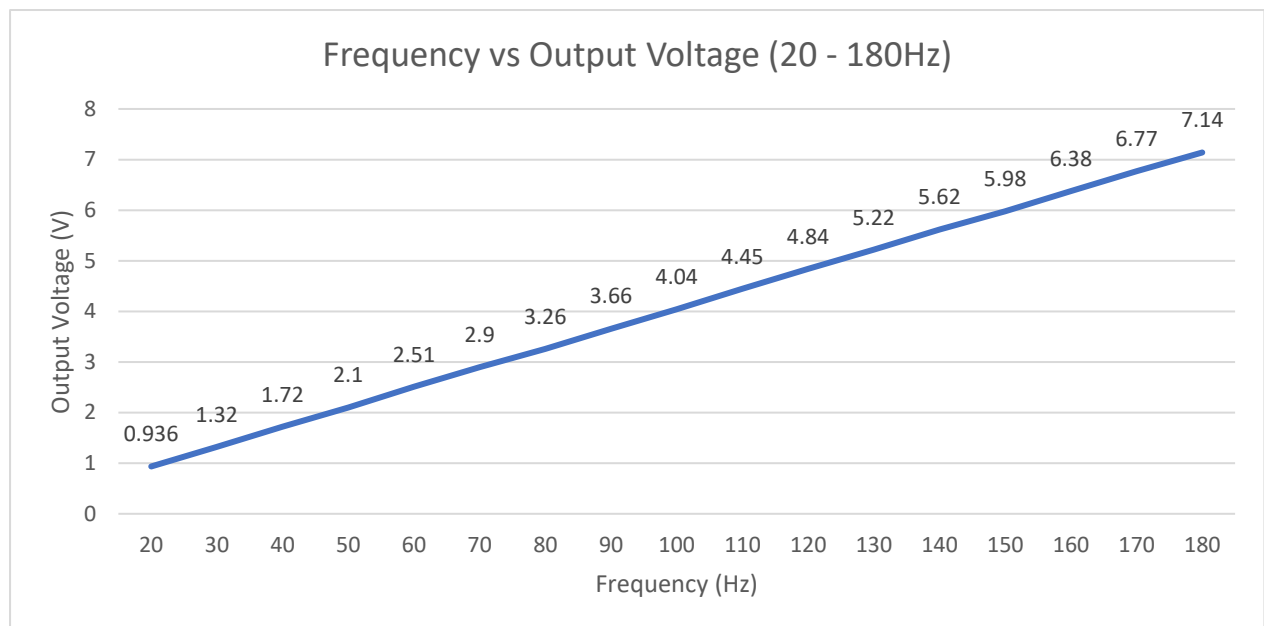


Figure 9-16: LM2907 frequency vs output voltage graph

9.3.10 Capacitance Values at LM2907 Pin 2

It was discovered during testing that adjusting capacitance values at Pin 2 scales the frequency range with respect to a set output voltage range. This allowed the capacitance value to be optimised such that the maximum output voltage is equivalent to the frequency of the motors which equates to the maximum speed of the Loco. Maximising the voltage output range improves the accuracy of motor speed monitoring. The capacitance was optimised for the Railway Challenge maximum speed of 15km/h, which resulted in three capacitors with a value of 47nF, 22nF and 4.7nF respectively. This however presented a problem in that any

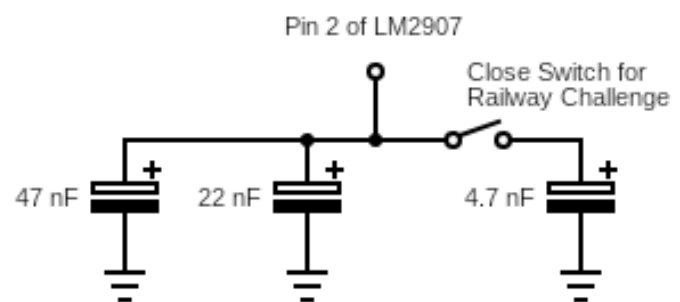


Figure 9-17: LM2907 Pin 2 capacitance adjustor

frequencies above 155Hz would output a voltage of 7.37V regardless. The specification for exhibition park is for running at 15mph and so any speeds in the range of 9.32mph to 15mph would be indiscriminate to the control circuit. Therefore, a single pole switch was incorporated to allow the removal of the 4.7nF capacitor, reducing the capacitance to 69nF, which is optimised such that the frequency equivalent to 15mph is equal to the highest possible voltage output.

9.4 Auto-stop Functionality (AH)

9.4.1 Voltage 'Storage' Functionality

The Auto-stop circuit inherited was designed by Alex Hooper in the previous academic year. The circuit was deemed to be mostly functional, with the main issue being the lack of proven method to 'store' a voltage and so this was the initial focus of research.

The Auto-stop circuitry uses the Equation 9-2 to determine the deceleration rate required to stop the loco 25m from strike in. Equation 9-2 requires the initial speed, u , to determine the required deceleration rate. The initial speed is a constant, it is the speed of the Loco at strike in, it cannot simply be implied from the voltage from the generators as this will vary with time and decrease as the Loco decelerates causing the wrong deceleration rate to be calculated. The voltage equivalent to initial speed at strike-in therefore needs to be 'stored' as a constant reference value.

$$a = \frac{u^2}{2 \times s} \quad (9 - 2)$$

9.4.2 Voltage 'Storage' Functionality: Latching Relay Concept

The initial concept used relay logic to 'store' a voltage (Figure 9-18). The relays were to be latching relays, fed from comparators (with pull-up resistors required to energise coil). The threshold voltage of the comparators would be set by a potentiometer such that they increase in small intervals of voltage. As the loco accelerates, the voltage from the generators would begin to exceed each of the threshold voltages, until it achieves its maximum speed. The latching relay energised by the highest threshold voltage would therefore correspond to the initial speed for the Auto-stop challenge. This system assumes that the Loco is at its maximum speed at strike-in and not before. This system is limited by tolerances as to how accurately the voltage can be stored. The interval between threshold voltages can be reduced to improve resolution, however this requires significantly more latching relays, increasing cost and size of the

circuit. The voltage drops ($\sim 0.7V$) across the diode would need to be accounted for using an adding op-amp configuration.

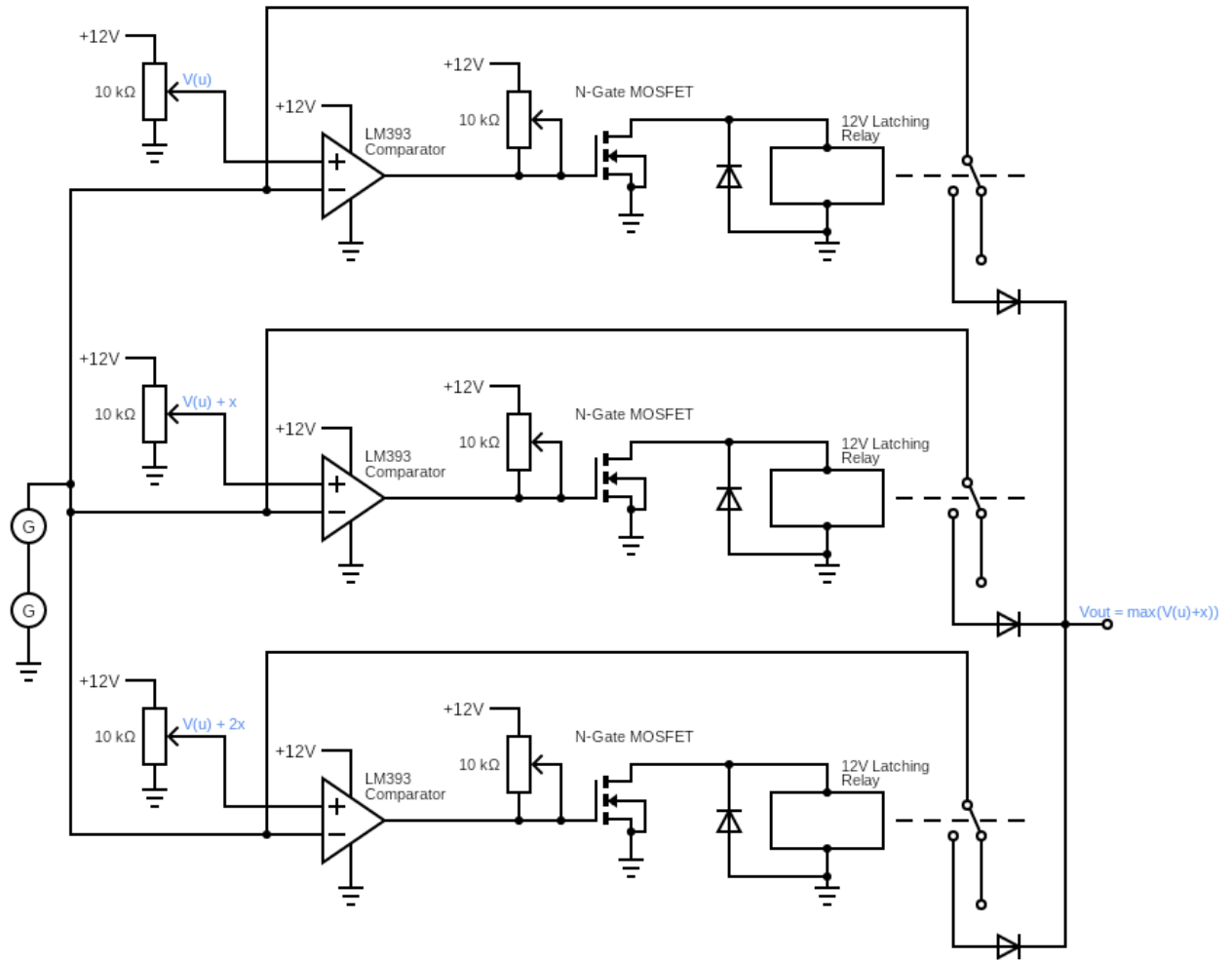


Figure 9-18: Latching relay voltage storage concept

9.4.3 Polarised Capacitor Concept

The concept of storing charge in a polarised capacitor was explored, where stored charge maintains a voltage across the capacitor. To minimise rate of voltage change, assuming constant current, Equation # dictates that, capacitance should be maximised as it is the denominator and so voltage change tends to zero as $C \gg i$.

$$i = C \frac{dV}{dt} \rightarrow \frac{dV}{dt} = \frac{i}{C} \quad (9 - 3)$$

Equation 9-3 was used to calculate the length of time the capacitor is required to hold its charge, and therefore, how long the voltage across it remains constant.

$$t = \frac{2s}{u + v} = \frac{2 \times 25}{0 + 4.17} = 11.99 \approx 12 \text{ seconds}$$

Therefore, the aim was to minimise voltage change for a period of 12 seconds. This theory was verified on Multisim, however testing discovered that in practice, the capacitor drained its charge much faster. An LM741 was placed in succession to the capacitor to act as a voltage follower. The voltage follower has high impedance and so was expected to reduce the charge drain of the capacitor. An oscilloscope was used to monitor the voltage over 12 seconds, it was found the total voltage drop to be 0.01V. Larger capacitors could have been specified however they take longer to charge and so if the loco doesn't run for long enough before the challenge then the stored voltage will not represent that at strike in.

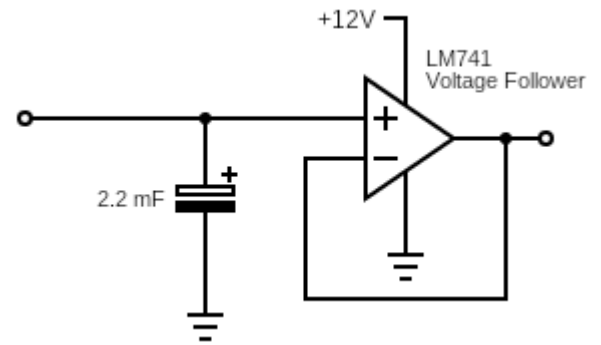


Figure 9-19: Voltage follower modification

9.4.4 Retroreflective Sensor Selection

Table 9-6: Retroreflective Sensors Considered Key Comparison

	<u>RS-Pro</u>	<u>Carlo Gavazzi</u>
Price (inc. VAT)	£38.56	£50.86
Detection Range	3m	5m
Adjustable?	NO	YES
Operating Temperature	-15°C to 55°C	-25°C to 60°C
Response Time	<8.2ms	<0.5ms
Protection	None	IP67

The work undertaken by Alex Hooper in the previous academic year determined the optimal method of triggering the Auto-stop system was to use a retroreflective sensor however, no specific sensor was recommended and so the requirements needed to be established. The required output was to be PNP (Positive-Negative-Positive) and light-off; the sensor outputs high when the reflector is detected. Multiple suppliers were consulted, and the selection of sensors was whittled down to two. These two were compared in Table 9-6, but the Carlo Gavazzi sensor was selected. The response time was considered important as slower response times would result in the sensor being more prone to missing the reflector. The IP67 rating improved the value of the Carlo Gavazzi sensor as it reduces the need to waterproof the lens. The sensor is to be

mounted to the left side of the body using an aluminium L-bracket purchased from Metal Supermarkets.

9.4.5 Auto-stop Holding Circuit

The retroreflective sensor only maintains a high output for the duration it detects a reflective surface, which in the case of the Auto-stop detection, will be less than one second, as the loco passes the beacon. The polarised capacitor requires charging before the system is triggered so that once the loco reaches the strike in speed, the capacitor is sufficiently charged such that the input voltage equivalent to loco speed is equivalent to voltage across the capacitor. Therefore, a system to stop the capacitor from receiving the loco speed signal needed to be devised. The circuitry in Figure 9-20 was created:

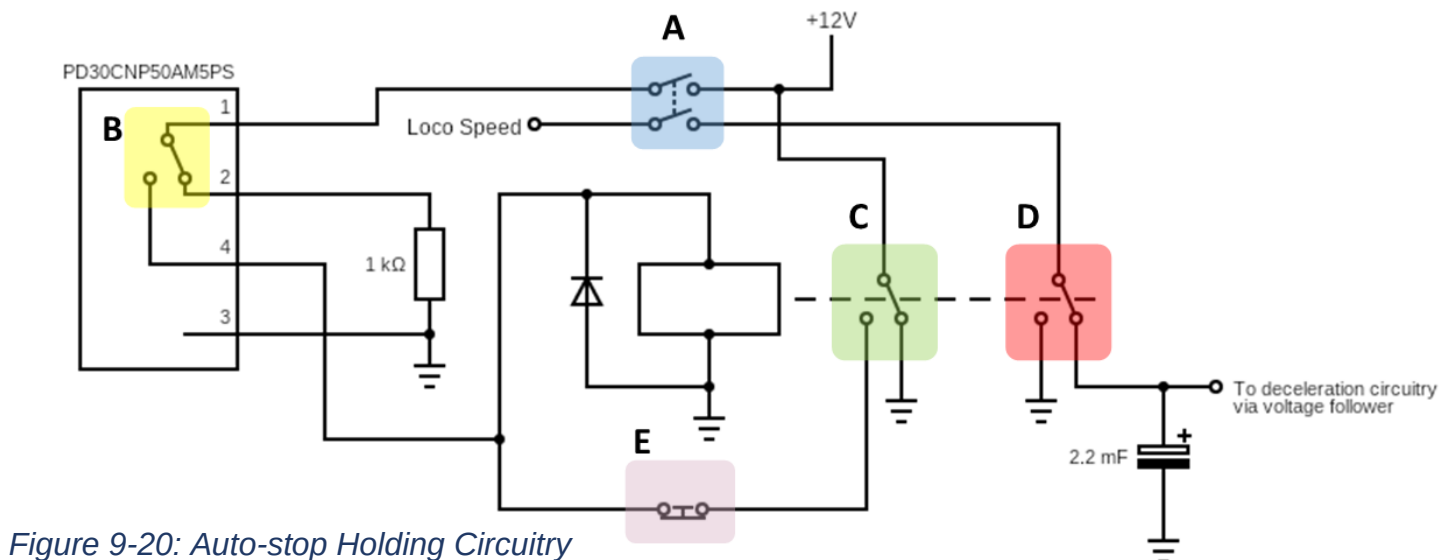


Figure 9-20: Auto-stop Holding Circuitry

- **A:** The double throw switch is closed, priming the Auto-stop system by providing power to the retroreflective sensor and completing the circuit by allowing the voltage equivalent to loco speed to feed into the capacitor to begin storing charge.
- **B:** Upon detection of a reflector, the sensor switches from contact 2 to 4, passing current through the holding circuit and energising the relay coil.
- **C:** The energised relay coil switches the first set of contacts to normally open, completing the section of circuit which connects the 12V rail to the relay coil such that the relay coil remains energised once the retroreflective sensor returns to its deactivated state.

- **D:** The second set of contacts is thrown to normally open such that the capacitor no longer receives a signal from the speed monitoring generators, thus the capacitor 'stores' the charge, equivalent to voltage at the point the relay is thrown, which will be equivalent to the speed at strike in.
- **E:** Once the challenge is complete, the push-to-brake button can be broken to break the connection between the power supply and relay coil, returning the relay to its deenergised state.

9.5 Electrical Enclosures, Wiring Management and Organisation (AH)

9.5.1 Low Voltage Circuit & Component Labelling System (AH)

Each low voltage circuit was assigned a code, as listed in Table 9-7. It was decided to situate all DC-DC isolated converters on one singular circuit in each half of the loco, these were named the power supply circuits PS1 & PS2. A coding system was devised to label each individual component. Referring to Figure 9-21:

Table 9-7: Circuit Codes & Locations

Circuit Name	Code	Location
Low Voltage Power Circuit - Front	P1	Front
Low Voltage Power Circuit - Rear	P2	Rear
Fire and Heat Detection - Front	A	Front
Fire and Heat Detection - Rear	B	Rear
True Speed Monitoring	C	Rear
Auto-stop	D	Front
Control	E	Rear

- Power supply code (red) indicates the op-amp is supplied from power supply two (PS2) output A,
- Component code (blue) text states this is an op-amp (OP – Full list of codes available in Appendix D-15), situated on circuit X with unique number 1.
- Component description (purple) text provides additional information, in this example it informs the reader specific component model, AD820.
- Wire number (yellow) informs the wire is from circuit Y to circuit X and has unique number 5.

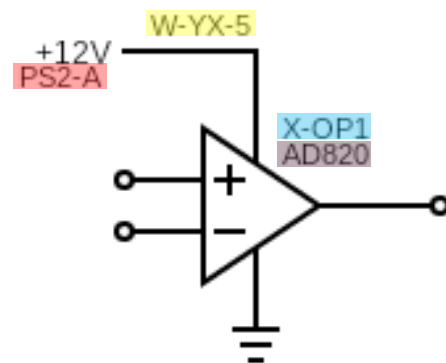


Figure 9-21: Labelling Example

The output current for from each DC-DC converter was checked and each component requiring a power supply from a converter was assigned a converter to be fed from,

denoted as PS#-X, and had its maximum output current summed to the other components fed from that converter to ensure the maximum load would not be exceeded. All component labels, values, current draw were tracked on a spreadsheet, an example extract of which can be found in Appendix D-14.

10. Control Box (AP)

Responsible team member received a submission extension, to be followed shortly.

This will be a description of how the design of the control box has progressed from the previously inherited design.

11. High Voltage Electrical Circuits (JO)

11.1 Traction Systems (JO)

The system is simplistic in nature, for both front and rear, power is taken from the batteries and used to power the Sevcon motor controllers, which in turn power the motors providing tractive effort.

Additional components exist for both front and rear ends of the locomotive. In the front, the air compressor and horn are powered by a power inverter turning DC into AC.

In the rear and additional 9th battery used for low voltage power supply is charged using a DC-DC convertor. Safety features exist to shut off the locomotive in the event of unforeseen failure both on the control desk and on the locomotive itself.

The traction system has seen minimal change, largely retaining components from previous years. The MOSFET switches and supper capacitors that originally were part of the circuit were removed due to their additional expense and needlessness. The Sevcons and lead acid batteries can handle volts going in. The MOSFET measuring of the regenerative charge has been replaced with a cheaper energy counting circuit.

Minor changes involved the inclusion of battery isolation switches and motor controller cut out switches. Unfamiliar with the model of either, a large space was claimed on the body model to represent a possible BIS, with the recommendation that the MCOS be bolted onto the Sevcon heat sink.

The wire thickness was calculated by determining the fuse rating for each circuit, which in turn was calculated from the absolute maximum theoretical power the batteries could provide. The wire thickness needed was 35mm², rated nominally to 240A, though for bend radii a smaller wire diameter of 30mm² was deemed sufficient.

To assist with the maintenance challenge for fast removal of the batteries, fast release terminals were investigated. The primary problem was electrically isolating each individual battery, to accomplish this individual quick release connectors for electrical vehicles were investigated. Surlock compression lugs, the same brand suggested by previous years, were optimal.

The selection of the terminal end was critical due to the limited space within the locomotive body. The use of both threaded and bus bar terminals was discussed, including whether to use a straight or angled plug. The final decision was ultimately to use Halfords stud wingnut terminals as a permanent fixture on the battery terminals which a bus bar terminal could be fastened to. This combined the security of the connection with reduced space and fast release.

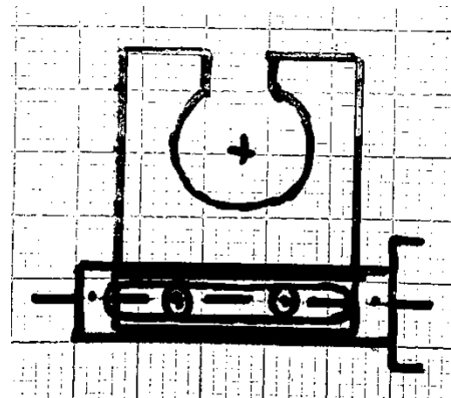


Figure 11-1: Proposed bus bar terminal attachment

11.2 Regenerative Breaking Challenge (JO, HW)

One of the challenges is regenerative braking, where the locomotive must run purely on recovered energy after braking a specified distance. The critical aspect here is providing proof of use of only the recovered energy. The solution was to count the recovered energy in and out of the system and use a relay that would cut the torque input command on the Sevcon. Regarding storing the recovered energy, the lead acid batteries and Sevcons themselves can handle what is deemed to be a sufficient value of current returning from the motors.

Any energy counter that would be bought would have to meet the specifications laid out. It would need to be electrically isolated, therefore would need a hall probe sensor, have a relay output, a sensitive input, and variable settings.

Originally, the search began by looking at solar power energy counters. These had a similar function to what was desired, they could measure bidirectional current and present these values on screen, with some models including variable settings. However, these models were very limited in their application, they often weren't electrically isolated, using a shunt as opposed to a hall probe, and few had a relay output. None could measure within the required current range.

A deeper investigation was looking at DC energy meters. Two models reviewed heavily were the Elmeasure EDC 4100D and the AcuDC 240. Below is a comparison of each model:

Table 11-1 Comparison of EDC 4100D and AcuDC 240

Properties	Elmeasure EDC 4100D	AcuDC 240
DC Hall probe	Yes	Yes
Number of input channels	4	7
Input voltage range	10-800V DC	20-160V DC
Variable settings	Yes	Yes
Programmable output relay	Yes	Yes
Relay tripping time	180sec	N/A
Set time	N/A	10ms
Switching voltage	N/A	30V DC
Power consumption	N/A	3W
Wiring diagrams	No	Yes

Although both met the criteria, the AcuDC 240 was superior due to the available information. Quotes were attempted for both devices, but responses were vague and untimely.

After realising the complexity of the required device and the lack of devices which met the criteria, a possible circuit of original design was presented. The circuit measures values of current and voltage into and out of the Sevcons, multiplying them and summing from each half of the locomotive, inverting the signal, and integrating to present power on a meter and storing the energy in a capacitor.

Once the maximum amount of energy has been recovered from braking, the capacitor releases the stored energy until all of it has been used to provide tractive effort, at which point a comparator triggers a relay which cuts the torque.

To measure current and voltage, specialised components were needed. To measure the current, a Hall probe sensor capable of measuring 300A DC was found. To measure the voltage, an IC chip operational was needed. It was determined that Integrated Amplifiers were the best solution, using a model, ACPL 782T, which could provide a measured value of voltage while remaining electrically isolated from the main circuit.

Tests were performed on the circuit to determine whether the original design was sufficient. The circuit was broken down in various stages and once each stage was correctly assembled, the next stage was attached. Potentiometers were used to simulate both voltage and current inputs. Due to complications with the summing op amp, the inverter was several stages later than expected.

Testing provided many obstacles, many of which were resolved when the op amps were provided the correct supply voltage. The most complex stage was the multiplier which was resolved once the provided with its recommended supply of $\pm 15V$.

It was noted that minor deviance from the expected value occurred at lower levels due to the oscilloscope, but the accuracy of the multiplier and other operational amplifiers decreased at higher voltage values. To resolve this, it was recommended that signals to op amps would need to be scaled down to reduce inaccuracy.

11.3 Battery Management, Containment, and Maintenance

11.3.1 Battery box (JO)

The battery box consists of a sheet of Tufnol material upon which acrylic side rails and an eccentric cam shaft clamp reside, situated underneath a sheet of acrylic.

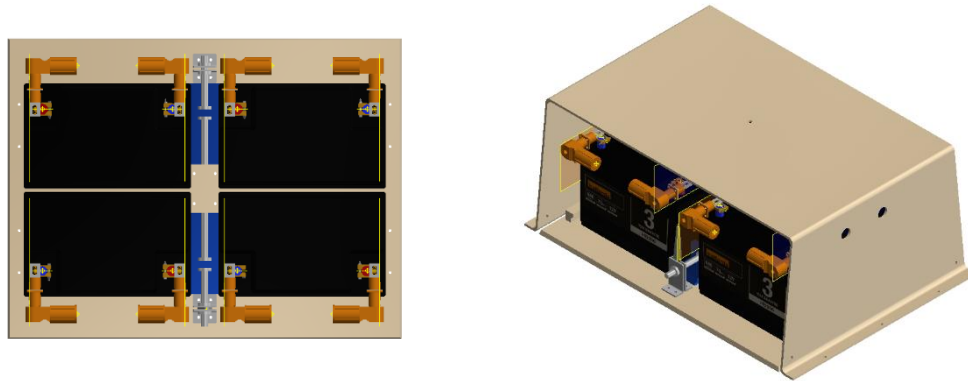


Figure 11-2: CAD Models of current battery box assembly.

For the IMechE competition, 2 challenges are related to the batteries: regenerative braking and maintenance challenge. Here the maintenance challenge takes precedence in design. The challenge requires the “discharge” and “refuelling” of the locomotive in under 120 seconds. Here the terms discharge will refer to battery removal and refuelling will refer to battery replacement.

The criteria for the battery box were categorised to essential and desirable features:

Essential

- Physically retain 4 batteries during travel.
- Electrically connect batteries to the high-power systems.
- Prevent battery leakage.
- Withstand 10g of acceleration in transverse direction.
- Release 4 batteries whilst stationary.

Desirable

- Electrically disconnect and reconnect all 4 batteries in under 30 seconds.
- Unilateral clamp release and application.
- Allow removal and replacement of 4 batteries in under 30 seconds.

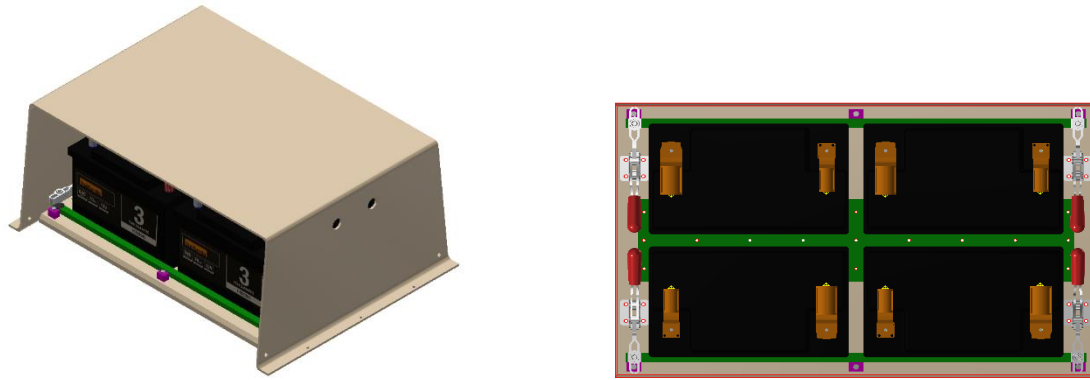


Figure 11-3: CAD Models of previous battery box assembly.

Previous designs by James Owen were like the final model as further changes unnecessarily complicated design. But the designs had many shortcomings. One glaring issue with the previous designs is the chained platform intended for use in battery removal. Another issue was the physical retention of the batteries themselves: 4 toggle clamps at either end of the box with a release bar.

The first battery box changes attempted to include speed of removal with security during travel. One idea was a spring assisted battery ejection device which after consideration and basic calculations on spring stiffness, was determined hazardous. After investigating, an idea inspired by battery maintenance on trains in Indonesia was considered. This was a simple box attached to a drawer slide mechanism that could retain the load of the batteries. The limitations of this design were the need for strong drawer slides and a redesign of the box and retention completely.

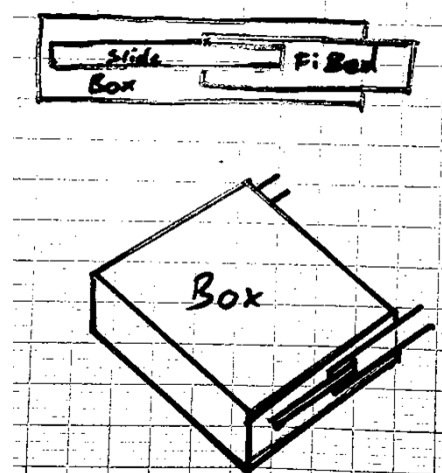


Figure 11-4: Sketch of draw trolley idea.

The box was design to fit within the confines of the battery cover, including a flapped front to allow removal without lifting, but the fixing points for the drawer slide were poorly conceived. Other ideas were to put the battery on rails, but this created clamping problems.

The designs migrated back towards the designs of previous years, a cover for the battery box and a single piece of plastic on which the batteries rested and were

clamped. One of the key differences is that the folding door mechanism for battery removal was replaced by a simple plastic sheet stored in the roof of the locomotive, propped up with pegs strong enough to take the load of all 4 batteries. The dividing bar that assisted in clamping the batteries in the middle was removed and the batteries were moved closer together meaning they could be removed theoretically from one side, though would need to be manually released from both.

Material selection was critical for this design. While the cover and side clamps remained as acrylic, the cover being clear to see inside the battery case, the floor needed to be a stronger more leak resistant material. A selection of laminated fabrics produced by Tufnol were presented with, Crow Brand Tufnol being the most suitable, but Carp being selected for ease of purchase.

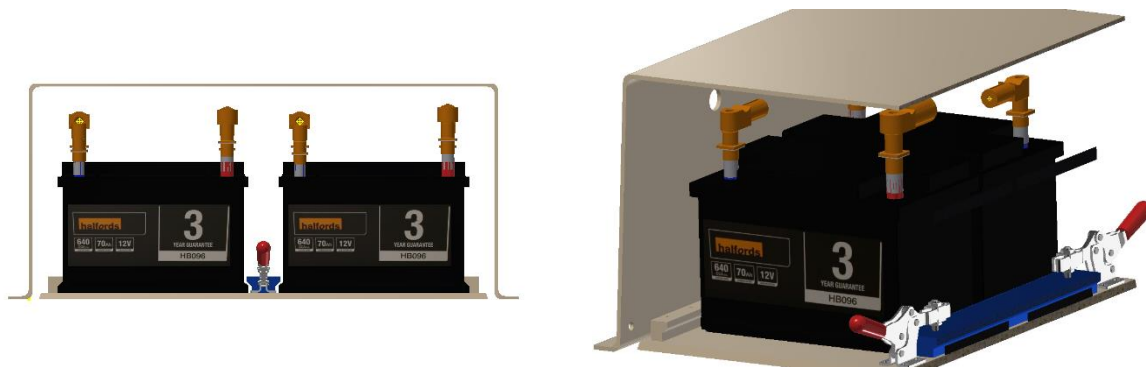


Figure 11-5: CAD Models of unrestrained battery removal box assembly.

A common theme between all previous designs is the type of clamping. All previously modelled designs required manual release from both ends using multiple toggle clamps, thus increasing the switch out time for the batteries. A single lever release mechanism was devised for unilateral clamping action using linkages. The design shared similarities with the toggle clamps already in use, though this would most likely be prone to shock loading from background noise, it was also overly complex.

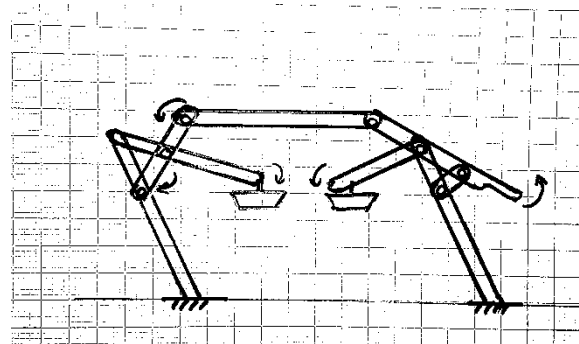


Figure 11-6: Sketch of unilateral clamp idea.

An improved idea was presented, using the forces of an eccentric cam on a shaft to lock the batteries in place with a single bar. This design achieved all previously desired criteria whilst remaining the mechanically simpler solution.

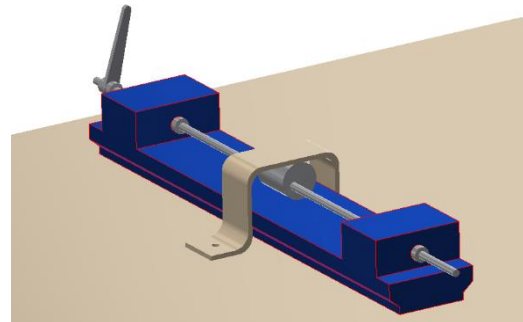


Figure 11-7: CAD Model of unilateral cam-based clamp.

Calculations were needed to determine the cam and shaft size, as well as whether the intended acrylic clamps and side railings could constrain the batteries under load.

Upon determining dimensions, the CAD model was updated appropriately. During the total locomotive CAD assembly, it was realised that the design of the clamp in its current form was insufficient, as the central flanged plate for the cam to rest on prevented unilateral battery removal.

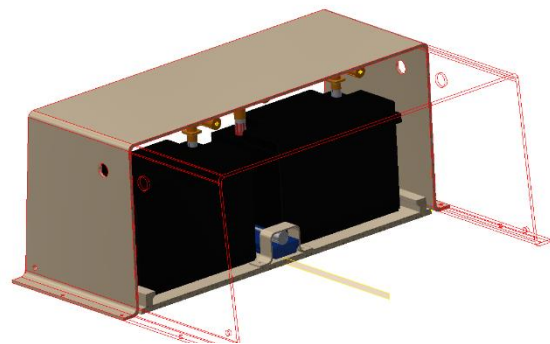


Figure 11-8: CAD Model illustrating issue with battery removal.

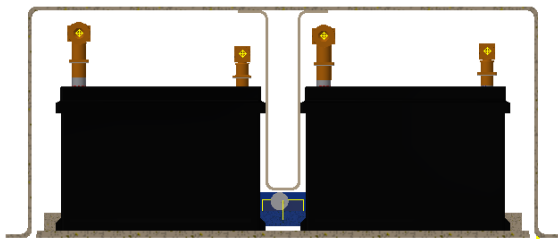


Figure 11-9: CAD Model illustrating roof hung plate.

One solution was to reverse the plate and have it rest from the roof of the cover. This idea allowed the batteries to be removed from one side, however due to the extreme length which would cause excessive forces, as well as interfering with wire routes for the traction system, this idea was discarded.

A simpler and better solution was a simple non-flanged plate of with 4 through holes that would match additional through holes on the clamping foot. This would allow the fastening of 4 bolts that avoided the path of the batteries.

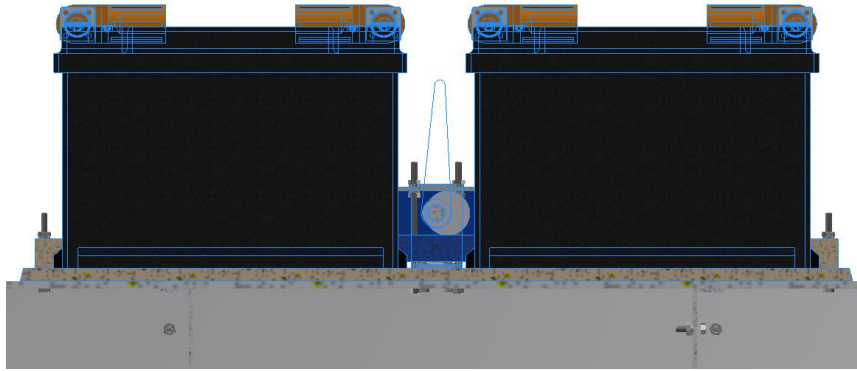


Figure 11-10: CAD Model illustrating current bolt and nut plate.

This design inadvertently resolved an additional problem. Due to the relatively obscure nature of the clamp, there were concerns that the clamp would be installed either too high or too low and this would depend on the length of the flanges of the plate. Removing the flanges and relying on nuts and bolts gives the plate an adjustable nature, allowing the installation of the clamp at the correct height without dependence on precise measurements.

There was another design suggestion that was too late to implement but simplified the clamp into a single bar bent at 3 points with clamping feet and the same roof plate in previous models. This would reduce costs and manufacturing time drastically if implemented correctly, though would have to be bent most likely out of house and was an incredibly late proposal.

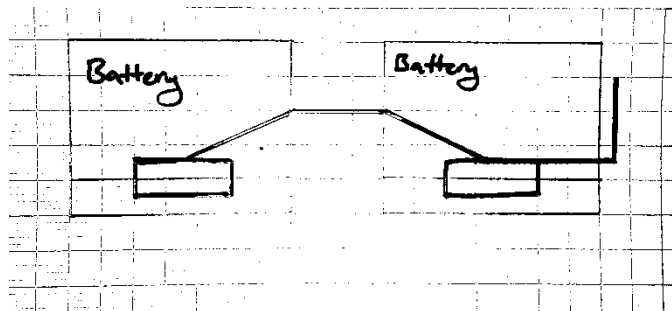


Figure 11-11: Sketch of single bar with 2 feet clamp.

The manufacturing plan for the battery box is simplistic, as most components need little machining. Both Tufnol plates require drilling, as do the holes on the flange of the cover, everything else requires lining up and fastening with nut and bolt. All components of the box are designed for ease of manufacture. After learning from the assembly of the trolley, several holes were enlarged or slotted, and some were removed to assist in assembly.

11.3.2 Battery trolley (JO)

Initial ideas were that trolleys with enough space and the correct height could be bought in. This required knowledge of the dimensions of the body in relation to the height off the track and the station floor. Using dimensions from the dynamic loading gauge it was assumed the distance of the station floor from the tops of the rail was 90mm, putting the distance of the station floor 10mm from the centres of the wheels.

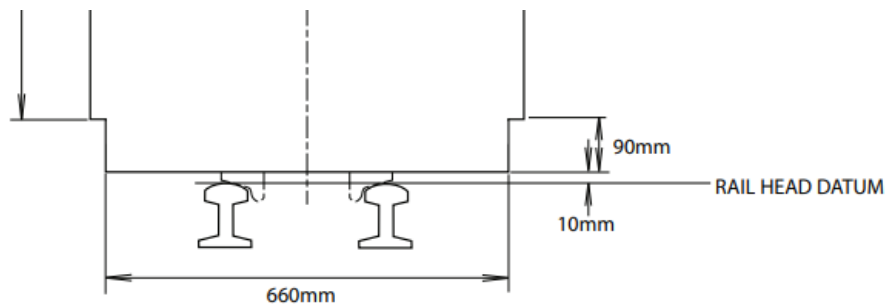


Figure 11-12: Lower portion of IMechE dynamic loading gauge.

Knowing this height allowed for a selection of trolleys in a range of heights that the batteries could be realistically loaded on. The primary problem with this solution was the lack of control over station floor to trolley height.

Another idea was that the lower body and bogie frame acted as a fixed trolley on which the batteries rested, and any change of batteries would involve jacking the top half of the body and moving a new undercarriage fixed with new batteries underneath while on tracks. This design was incredibly unfeasible as it would require large scale production of bogies, bodies and every component that would reside within them that was not rigidly attached to the upper portion of the locomotive.

The agreed upon design was a rotating platform trolley using a simple hand crank, a design loosely inspired by Japanese tower parking for cars. Upon more in depth thought it was realised that to rotate the platforms, they required stabilisation to prevent tipping,

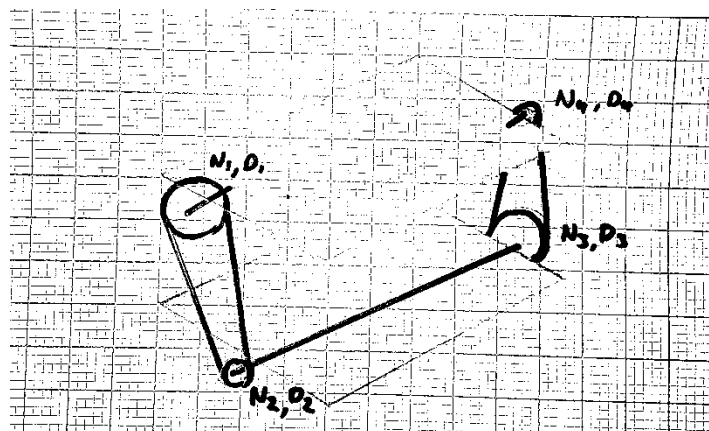


Figure 11-13: Sketch of rotating trolley idea.

but these stabilising bars could not intrude with the rotation of the mechanism.

Dimensions were calculated for each aspect of the trolley, including the diameter of rotation and base width of the physical trolley. After resolving these dimensions, calculations were performed to determine the loads experienced at trolley extremities and the bearings during the worst load case scenario. Following this the optimum pulley layout was determined.

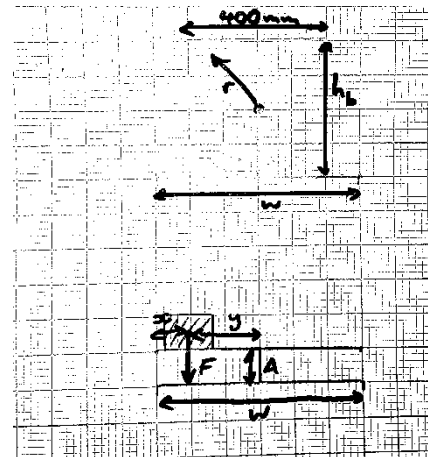


Figure 11-14: Drawings associated with calculations.

Further work done included the material choice for the trolley base, which was to be hollow steel box section that could be bought from metal supermarket, mild steel sheet of 2 mm thickness. Following design phase all files were passed on to Cyril for Caster selection and creation of CAD files.

12. Team Management & Organisation (AH, CC, JJ)

12.1 Day-to-day Management & Collaboration (AH)

The project was undertaken throughout the 2021/22 Academic year, beginning September 2021, and finishing May 2022, although with the Railway Challenge being at the end of June 2022, it is expected that team members will continue to work on the project after submission, in preparation for the challenge weekend. The year was broken into two semesters, with each semester consisting of three teaching blocks. Two of the three blocks in Semester 1 were allocated to project work, with no scheduled lectures, allowing the team to designate all their attention to the project, with the third block leaving the team to balance their time between lectures and project work. During the two blocks with no lectures scheduled, the team met bi-weekly, usually Monday and Thursday mornings. The Monday meeting was dropped during the blocks with lectures scheduled to allow the team to manage their time more effectively. The meetings were initially hosted in Mr Davidsons office but later moved to the Loco room, 2nd floor Stephenson Building. The meetings were led by the team leader, Alex Hooper, with minutes usually taken by team secretary, Cyril Cheung. The structure of the meetings was as follows:

1. Important information cascaded by team leader to other team members, for example important information about deadlines, change in procedure or significant design changes.
2. Everyone taking turns to share their work since the last meeting and talk about their planed work before the next meeting.
3. Technical support from Rob Davidson, for anyone that required it.
4. *Optional:* Team members remain behind after Mr Davidson leaves to discuss matters which can be resolved without him such as preparation for reports.

Meeting attendance was recorded separately by the team leader and team secretary to allow for comprehensive monitoring. Team members regularly arranged 'splinter' meetings with each other to discuss specific areas of the loco which didn't require input from unrelated team members. For example, the electronics team members regularly met separately, as did the bogie team members. These splinter meetings were recorded in logbooks with any significant developments shared in the main team meetings. General communication and brief questions were asked via Facebook Messenger group which all team members were part of, with more formal communication through the University email system.

12.2 Sponsorship (AH)

It was clear at the beginning of the project that the team would require significant sponsorship, with £30,000 being the initial estimated build cost. Previous team members had secured sponsorship from Reece group in the form of £2500, which assisted in the purchase of the motors and Sevcons. Lightness successfully applied for an award from the IMechE, securing the team £1500. The team inherited a spreadsheet of companies and organisations contacted in previous years to discuss sponsorship with this was reset and modified to a spreadsheet. Every team member was allocated companies they were to be responsible for contacting to ask for organisation. The spreadsheet included columns to allow the progress of this to be tracked so that organisations were not contacted twice, which would look unprofessional but also to ensure all team members were actively engaging in the search for sponsorship.

Alex Hooper, James O'Brien, and Halley Woods designed and created a sponsorship brochure, loosely based on a longer brochure inherited from the previous team. This two-page document can be found in the Appendix E. The first page included a CAD render of the loco at the time, with the key and unique features highlighted and briefly

explained, to 'sell' the loco to potential sponsors and to set it apart from competitors. The second page included an image of the team to put faces to names and a tiered sponsorship list, specifying levels of financial support expected to achieve each tier of sponsorship.

It quickly became clear that persuading companies to sponsor a team with no physical loco to show off very difficult, and so the team leader decided to change the approach and work to build a social media presence, where CAD renders and could be uploaded, allowing the manufacturing of the loco to be documented to show companies that the team are serious about building the loco. Contact was made to the existing social media channels in the hope that previous team members would reach out and share the log in details, to allow the team to build on the existing profiles started three years prior, benefiting from their followers. Courtney McQuillan kindly shared the details and recommended software to organise the posts. The team leader decided to create a new role within the team, social media manager, as it was expected that managing the social media would take a significant amount of time and effort. David Stewart kindly stepped forwards and volunteered to take up this role, becoming responsible for regular updates on the loco in the form of images and videos across Twitter, Facebook, Instagram and Tiktok.

Sponsorship has been successfully achieved from Britishvolt for the 2022/23 team. Britishvolt are kindly offering the team lithium-ion batteries, this is expected to be a significant improvement from the current lead acid batteries used for tractive power.

12.3 Finance Management (AH)

Lightness Kings was the designated treasurer for the team. Lightness set up an excel document to allow the team to track committed expenditure, requested expenditure and transactions. During the first semester when purchasing volume was low, all orders were sent off via the team leader. The volume of orders began to increase after the design freeze and so the treasurer took over managing all orders to Acorn laser, of which there were a lot, as it required a significant time commitment. Many components were ordered from RS and so an Excel spreadsheet was created to allow team members to add their required components, allowing orders to be sent off in large batches, making it easier to track orders and reduce workload on the school

office. Any specialist suppliers were contacted via the team member requiring the component, with the authorisation of the team leader.

12.4 CAD Management (JJ)

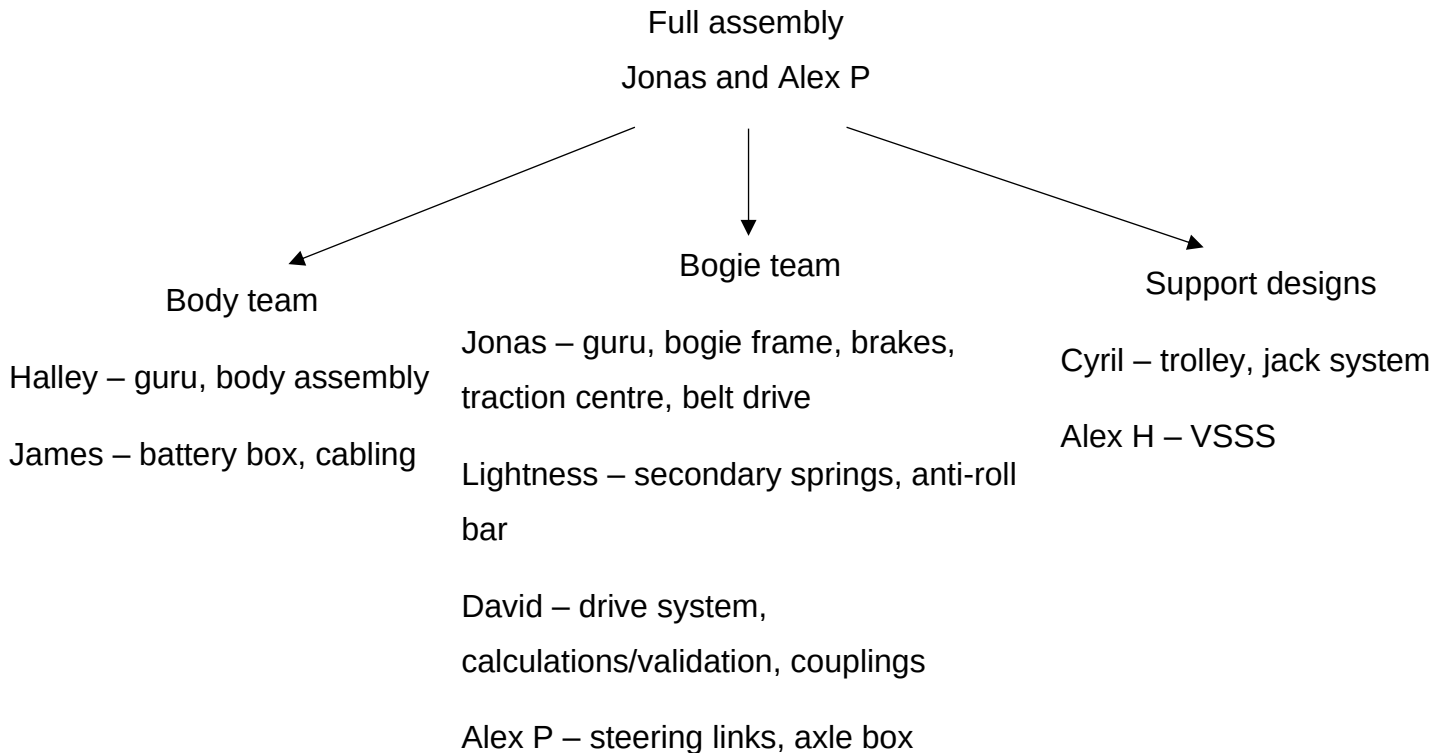
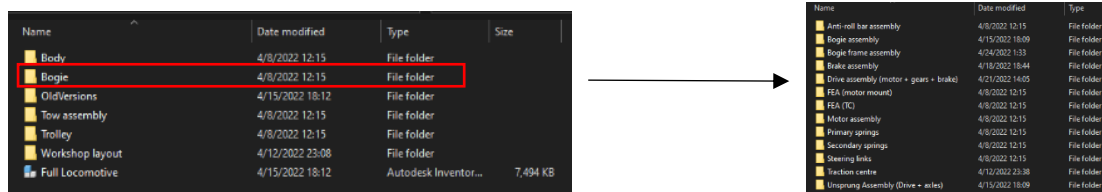


Figure 12-1: Flow chart of CAD assembly aid distribution.

Most of the design work was done before the design freeze on the 17th of December. Time before December was used for engineers to work on their individual designs on their own time using the previous year's locomotive design. In December the designs were merged into the final design which was scrutineered twice before the design freeze. Initially, there was one main CAD guru and deputy CAD guru that were responsible for full CAD assembly. However, it proved difficult to assemble a full locomotive for one engineer as it requires great knowledge about individual designs and time. Therefore, a new structure has been introduced. Halley was assigned as a body CAD guru because he had knowledge about body assembly and was a proficient Inventor user. Similarly, Jonas was assigned for bogie. Communication between these teams was important to ensure a smooth connection between body and bogie. The work on the separate assemblies was done simultaneously until the set deadline before scrutineering to allow time for the design merge. When the full assembly was worked on design changes to assemblies were not allowed. Working in three sub-teams has saved time, improved the quality of the assembly and made it easier.

Files

The files have been divided based on the team and subfolders have been created based on the individual designs. Some extra assemblies have been created to limit the effect of changes on the final design.



Name	Date modified	Type	Size
Body	4/8/2022 12:15	File folder	
Bogie	4/8/2022 12:15	File folder	
OldVersions	4/15/2022 18:12	File folder	
Tow assembly	4/8/2022 12:15	File folder	
Trolley	4/8/2022 12:15	File folder	
Workshop layout	4/12/2022 23:08	File folder	
Full Locomotive	4/15/2022 18:12	Autodesk Inventor...	7,494 KB

Name	Date modified	Type
Anti-roll bar assembly	4/8/2022 12:15	File folder
Bogie assembly	4/15/2022 18:09	File folder
Bogie frame assembly	4/24/2022 1:33	File folder
Brake assembly	4/18/2022 18:44	File folder
Drive assembly (motor + gears + brake)	4/21/2022 14:05	File folder
FEA (motor mount)	4/8/2022 12:15	File folder
FEA (TC)	4/8/2022 12:15	File folder
Motor assembly	4/8/2022 12:15	File folder
Primary springs	4/8/2022 12:15	File folder
Secondary springs	4/8/2022 12:15	File folder
Steering knuckle	4/8/2022 12:15	File folder
Traction centre	4/12/2022 23:38	File folder
Unsprung Assembly (Drive + axle)	4/15/2022 18:09	File folder

Figure 12-2: Files naming illustration.

Lessons and comments

- Initially, the copies of the design have been made using the copy function to keep the old versions of the design in case there is a need to go back on the design to see the alterations and the design would be uploaded to the cloud for all engineers to access. However, using this Inventor function caused problems with the file paths, made the file large and created long, unclear names. Solution – make modifications to the same assembly/ design and at the end of the change use pack and go function to gather all the active files into a workspace folder. Then the files from the workspace folder should be uploaded to the dated folder on the cloud. There should be a system (i.e. excel sheet) to indicate changes and who made them. Engineers should have access to download the files from the cloud. An attempt has been made to create guidance for future years in a file called “CAD file system solution”.

Important: no changes are overwritten on the existing files. A new folder shall be created. At the beginning of the project, there should be an attempt to ensure that the team uses the same design data and templates. Only the files from the pack and go workspace folder should be uploaded and not the full pack and go because uploading the full pack and go folder causes long file paths which result in errors when opening files.

It is recommended to have a mock-up attempt because the filing problems meant that the design had to be renamed and reassembled after the design freeze which was around 20 hours of additional work.

- Some of the individual designs were created and only tested in the assembly at the final stage to find there was something wrong. It is understandable as not every engineer was working with the latest design due to design changes

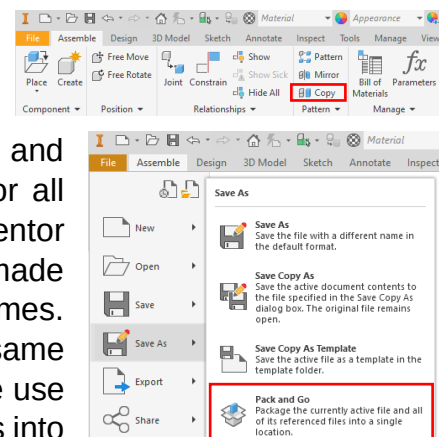


Figure 12-3: Inventor Pack & Go exportations.

happening simultaneously. A suggestion would be to have design deadlines based on design importance and influence on other designs. Also, it would be beneficial for the teams to introduce a rule that only one person can work on the design at a time. Combined with the file system that would ensure that everyone is working with the latest design.

- Although it seems obvious, it cannot be emphasized enough. The greater the detail of the design the easier assembly and a manufacturing process will be. Designs should really consider manufacturing and it is suggested that every step of the assembly is understood, and manufacturing plan is prepared before freezing the design.
- To get the full design ready and fixing scrutineering comments estimate to spend at least 80 hours or two weeks. Anticipate last minute changes that require big changes to the full design. It is beneficial to have well-structured and multiple sub-assemblies to limit the effect of changes from one assembly to another.

12.5 Health & Safety, Workshop Folder and Logbook (CC)

12.5.1 Health & Safety Forms

To ensure the team works in the workshop with the best knowledge of physical tasks carried out and the safety hazards alongside, Mr Davidson had constructed a comprehensive handout of Health & Safety information with forms to be signed off. The whole team were required to read through and to be questioned understanding of such. Those with full understanding of each form are to be signed off and could oversee safety of any given day; whereas those with slight confusion on details would have forms signed off but cannot be team members in charge.

The 4 main forms were:

1. Visiting & Working in Workshop Area
2. Specific Machinery Operation & Procedures (e.g.: Linisher, jigsaw, rivet guns)
3. Working on Running & Powered Vehicles
4. Working with Exposed High Power Electrical Systems

Each form had at least 4 team members approved to be in charge, to ensure the construction process was not stopped by insufficient safety officers.

To ensure forms and approval information are kept with clarity, the secretary used a binder to keep every signed-off form. This must be centrally located in the workshop, and electronically copied on the OneDrive folder as a backup.

12.5.2 Workshop Folder/ Binder

Aside from the Health & Safety forms, the workshop folder also keeps track of all receipts from the team's purchase flow. Orders come through frequently and keeping the receipts orderly ensures the team with full knowledge of what has arrived and what has not. A vast number of duplicate purchases were avoided with the help of such tracking system.

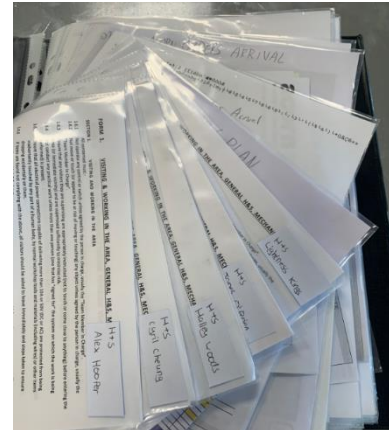


Figure 12-4: Image of H&S Forms in Binder.

12.5.3 Workshop Logbook

Meetings with the team, individual discussions, and any assembly work carried out in the workshop must be logged in. Not only does this keep track of how much time each team member had spent on the project, but it also ensures if accidents occur, any responsible person could be located. A typical log would include:

- Date, enter & exit time of activity.
- Team members present, and any guests they brought along with.
- Activity they were doing, and any safety hazards that they left before leaving. (e.g.: body is only loosely attached therefore need to handle with care)
- Team member in charge of safety of the day.

12.6 Manufacturing management (JJ)

Three things have been introduced to aid the team during manufacturing

1. **Detailed manufacturing plan** (Appendix J)

This plan included a Gantt chart with all the important deadlines, timeline, and detailed assembly plan. Design engineers were responsible for the manufacture of their individual designs with additional help available from colleagues. The work for the bogie was divided into preassembly work and assembling. As space and time are the limiting factors in the manufacturing process it was important to prepare well for the assembling. Preassembly work gave a good indication of the individual progress while allowing to control the time when the workshop was occupied with bogie assembly. Although, not fully executed but parts required for the beginning of the assembly were prioritized over other parts. Laser cut parts have the biggest lead time of about two weeks. Machining work takes about a week unless complicated machining is required.

Welding was quick – a couple of days. However, it was very dependent on the welder's and welding space availability. The painting was a day worth of work. The team was late with bogie assembly because important parts required for the begging of the assembly were ordered and manufactured too late. Suggested manufacturing order:

- Just after design freeze order parts to be laser cut
- Order all the necessary materials for machining
- Consult technicians and submit the machining work before the material is received. Some changes might be required!
- From detailed manufacturing plan order all the required tooling, fasteners, couplings, etc.

Note: after the design freeze all the listed work should be done within two weeks. This should leave enough time for modifications, assembling and problem solving. Success is dependent on how much detail is paid into each manufacturing step.

2. Workshop layout

Optimal workshop layout has been created to maximise the use of the space while keeping it safe and comfortable to work in. The bodies were worked on at the same time while bogies were assembled one at a time. It was decided this way because the body requires similar modifications. Therefore, working simultaneously on two bodies was more time efficient. Bogie preparation was much longer, and bogie occupies too much space to have two bogies worked at the same time. Bogie nr.1 is the first one to be built while the other one is kept out of the way

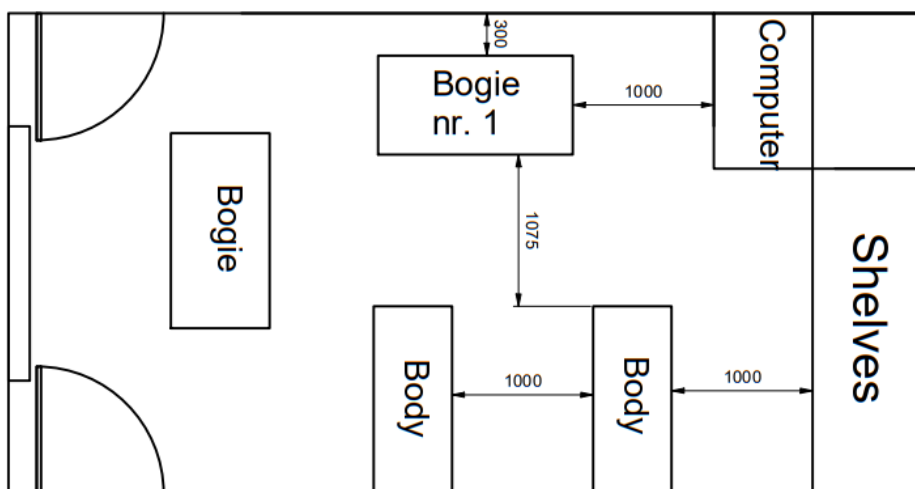


Figure 12-5: Proposed workshop layout for manufacturing.

3. Clock-in system

Later in the assembly process clock-in system has been introduced to better manage time and manufacturing. This form helped with accountability for the work done, time spent and effectiveness of the time in the workshop. The work timetable was visible to everyone at the workshop

HSTOON					Name					Work timetable				
Date	Clock in	Clock out	Total (h)	Jobs done	Date	Clock in	Clock out	Total (h)	Jobs done	Date	Clock in	Clock out	Total (h)	Jobs done
14/04/2022					05/05/2022									
19/04/2022					06/05/2022									
20/04/2022					09/05/2022									
21/04/2022					10/05/2022									
22/04/2022					11/05/2022									
25/04/2022					12/05/2022									
26/04/2022					13/05/2022									
27/04/2022					16/05/2022									
28/04/2022					17/05/2022									
29/04/2022					18/05/2022									
02/05/2022					19/05/2022									
03/05/2022					20/05/2022									

Figure 12-6: Clock-in sheet template.

13. Conclusion & Recommendations for Future Work

13.1 Conclusion (CC)

Newcastle Railway 2021-2022 has made progress in the modelling and physical journeys of HSToon forward. The main milestones achieved are:

- Trolley – Design of 4 wheelset trollies, manufacturing completed. It would be able to carry the full locomotive, with ease of transporting and method to constraint movement when necessary.
- Body Lift – Design of 4 tripod stands, parts ordered with comprehensive guidance to assemble. It is a mobile device that is easy to assemble and the whole system has a loading capacity of 1000kg.
- Bogie – Inheriting design from previous years, most parts had undergone new iterations of design and are all manufacturable as of now. It would be assembled for the year subject to the last resorts of certain components' assembly. It would have rolling capabilities on tracks.
- Body – Undergone several major design modifications and validation. Materials had all been ordered and mostly assembled. The following team would have full guidance of design thoughts and comprehensive manufacturing stepped approach to bring a sealed complete body to practice.
- Low Voltage Circuit – Undergone significant amount of testing, to validate and improve existing circuit design. The following team would have guidance on full circuit diagrams, mock-ups, and components specifications available. The circuit is manufacturable on current basis.
- High Voltage Circuit – Circuits tested, and wire routes proposed on AutoCAD. Circuit for regenerative braking tested for ease of future references.
- Dynamic Loading Gauge – With the current frozen design, lateral extensions were validated to ensure the locomotive is in gauge. Vertical and lateral motions were constrained by the Traction Centre & bump stops respectively.

The team's priority was to facilitate HSToon's capability to move as a locomotive. However, to enable this, efforts to keep the team organised were established and members had by large communicated with each other effectively and efficiently. The team's finance records are consistently updated, and sponsorship materials were established to keep the locomotive monetizable, and genuinely interesting.

13.2 Recommendations for Future Work

13.2.1 Bogie (DS)

- Increase the length of the anti-roll bars connecting arm to produce a larger resisting torsional moment. This would subsequently require more capable bushing in the mechanism.
- Perform rigorous FEA on the bogie frame to reduce the bogies overall mass by removing material in areas of low stress.
- Redesign the axle boxes for a lighter, cheaper, easier to assembled design.
- Stiffer and/or longer primary suspension springs for a larger safety factor
- Increase the distance between the reaction centre bushings.
- Reduce the thickness of the motor mount linking arm bushing lugs
- Increase the thickness of the anti-roll bar and steering link bushing lugs

13.2.2 Body

- Design of a passenger car to sit between the two powered cars of the locomotive.
- Changes to internal components of the locomotive because of the new components needed for use of lithium-ion batteries and any other additional changes.
- Complete the design to attach the jacking box tube to the flange to attach to the flange doing calculations and FEA where necessary.

13.2.3 Low Voltage Circuitry (AH)

- Learn how to use an industry standard software package to create circuit diagrams. Newcastle University Formula Student use Altium.
- Send the circuit designs to be printed on PCB circuit boards to reduce risk of short circuit and damage from vibrations.
- Investigate the use of multi-channel op-amps. For example, TL074ACN a singular IC with 4 op-amp channels.
- Investigate alternative methods of creep control to improve accuracy and reliability.
- Investigate methods of wireless control from loco to control box.

- Purchase a second retro-reflective sensor and connect using an OR gate to reduce the chance of a missed activation or design circuitry such that sensors must be triggered in a specific pattern to reduce sensitivity.

13.2.4 High Voltage Circuitry (JO)

- Purchase primary components needed to complete both the traction circuits and the energy counting circuits.
- Visit Voltsport for technical assistance in Sevcon motor controller calibration.
- Complete wiring looms for both traction circuits and assemble inside locomotive.
- Test traction systems to operate driveshaft and high voltage set up with batteries

13.2.5 Team Management & Structure (AH)

- Updates sponsorship brochure to include images of the loco so companies have a chance to see that the team already has an existing loco and are in a good place to enter the 2023 Railway Challenge.
- Contact as many companies as quickly as possible; it takes time getting through to the right people who have the authority to offer sponsorship and releasing funds can be slow.
- Remove pricing from tiers and find an alternative way to quantify sponsorship; smaller companies were quickly scared away from sponsorship when they saw the team were asking more than £1000.

References

1. **Poole, T.** Railway Challenge 2022 – Technical Specification V1. [Online] 15 September 2021. [Cited: 25 February 2022.] <https://www.imeche.org/docs/default-source/1-oscar/challenges/railway-challenge/railway-challenge-2022/railway-challenge-2022-technical-specification-v1.pdf?sfvrsn=2>.
2. **Fairbrother, Tim.** Institution of Mechanical Engineers Railway Challenge 2022 – Rules V1. *Institution of Mechanical Engineers Railway Challenge Team Resources*. [Online] 2022. [Cited: 25 February 2022.] https://www.imeche.org/docs/default-source/1-oscar/challenges/railway-challenge/railway-challenge-2022/rc2022_rules_v1-final.pdf?sfvrsn=2.
3. **Patch, The Contact.** Railway suspension. *Rail*. s.l. : The contact patch: Transport technology from the ground up, 2014.
4. **Iwnicki, Simon, et al.** *Handbook of Railway Vehicle Dynamics, Second Edition*. Milton, UNITED KINGDOM : Taylor & Francis Group, 2019. 9780429890635.
5. **Metal Supermarket.** Metal Supermarket. *Mild Steel Channel Black S275*. [Online] [Cited: 2 May 2022.] <https://www.metalsupermarkets.co.uk/product/mild-steel-channel-black-s275/>.
6. **Barret, Richard T.** *Fastener Design Manual*. Cleaveland, Ohio : NASA reference publication 1228, 1990 March.
7. **SMC.** SMC pneumatic cylinder Manual. [Online] [Cited: 14 04 2022.] https://www.distrelec.biz/Web/Downloads/_c/at/CDQ2B32TF-100DZ_eng_cat.pdf.
8. **Cussens, James, et al.** "Design of a Locomotive for the 2021 IMechE Railway Challenge Competition": *MEC8099 – Mechanical Engineering Team Project*. School of Engineering, Newcastle University. Newcastle Upon Tyne : s.n., 2021. p. 106.

9. **Texas Instruments.** LM2907 and LM2917 Frequency to Voltage Converter. June 2000.

10. **RS Components.** RS Components. *LM393P Texas Instruments, Dual Comparator, Open Collector O/P, 1.3μs 3 → 28 V 8-Pin PDIP.* [Online] [Cited: 2 May 2022.] <https://uk.rs-online.com/web/p/comparators/0810239>.

11. —. RS Components. *LM741CN/NOPB Texas Instruments, Op Amp, 1MHz, 8-Pin MDIP.* [Online] [Cited: 2 May 2022.] <https://uk.rs-online.com/web/p/op-amps/5359969>.

12. **Cussens, James.** *Design of a Locomotive for the 2021 IMechE Railway Challenge Competition - Mechanical engineering Team report.* Newcastle : Newcastle University, 2021. p. 66.

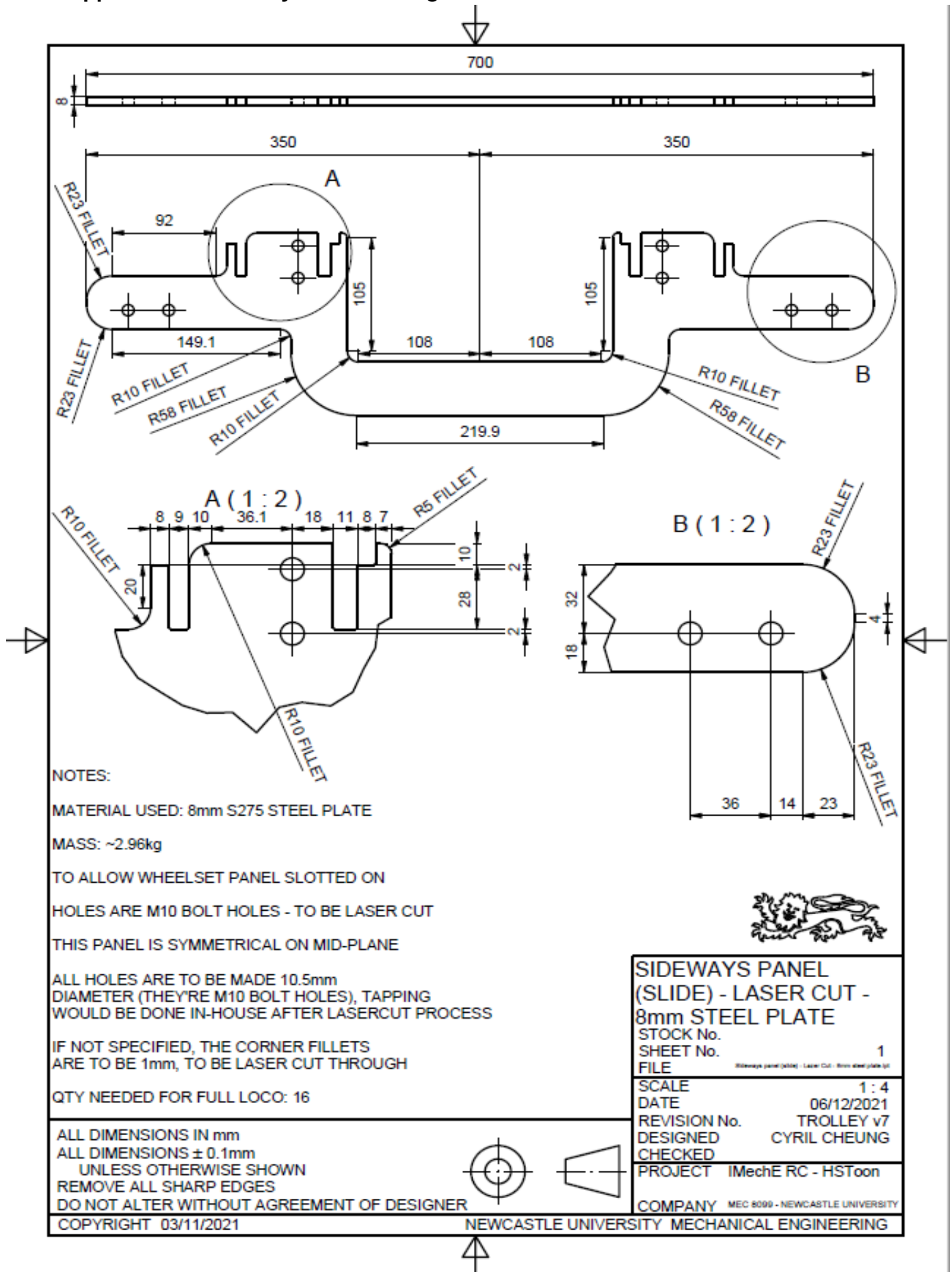
13. **Aalco.** Aalco. *Aluminium Alloy - Commercial Alloy - 6082 - T6-T651 Plate.* [Online] [Cited: 3 May 2022.] https://www.aalco.co.uk/datasheets/Aluminium-Alloy_6082-T6-T651_148.ashx.

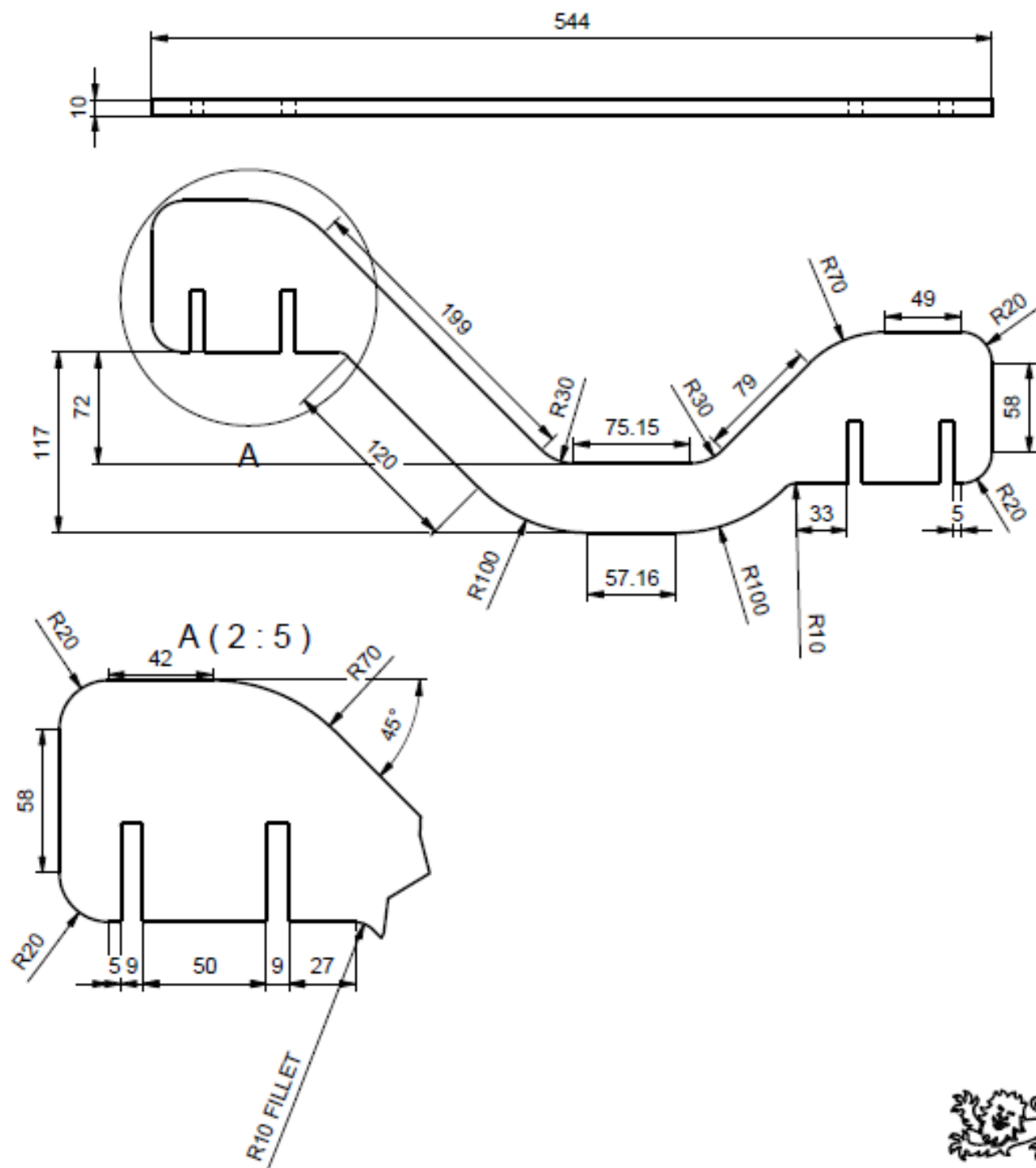
14. **Lee Springs.** Lee Springs. *Compression Springs - LCM200LM03S.* [Online] [Cited: 3 May 2022.] <https://www.leespring.co.uk/compression-springs?search=LCM200LM03S>.

Appendix

Appendix A: Trolley and Body Lift

Appendix A-1: Trolley CAD Drawings





NOTES:
MATERIAL: S275 MILD STEEL (10mm)

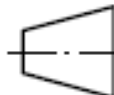
ALL CORNERS TO BE FILLET LASER CUT THROUGH (1mm RADIUS)
IF NOT SPECIFIED

QTY NEEDED: 8

THIS IS NOT A SYMMETRICAL PANEL

ALL DIMENSIONS IN mm
ALL DIMENSIONS $\pm 0.1\text{mm}$
UNLESS OTHERWISE SHOWN
REMOVE ALL SHARP EDGES
DO NOT ALTER WITHOUT AGREEMENT OF DESIGNER

COPYRIGHT 15/10/2021



SIDE PANEL VERSION B -
LASER CUT - 10mm R2D2

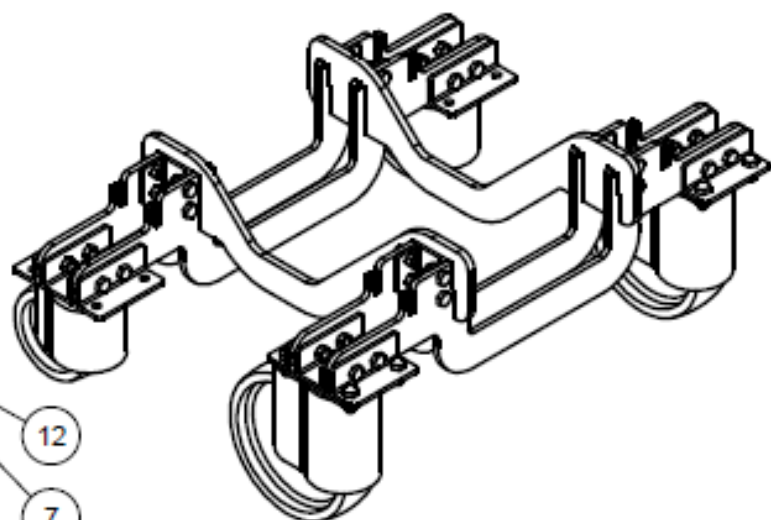
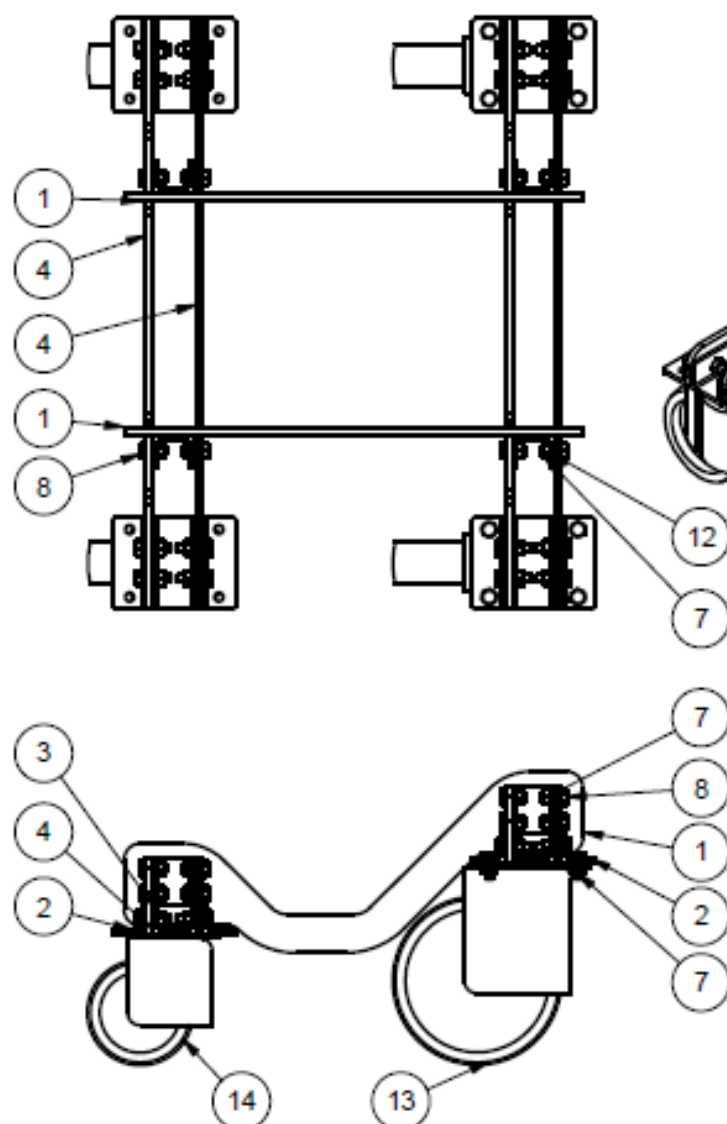
STOCK No.
SHEET No. 1
FILE

SCALE 1:4
DATE 15/10/2021

REVISION No.
DESIGNED CYRIL CHEUNG
CHECKED R I DAVIDSON
PROJECT IMechE RC - HSToon

COMPANY NCL UNI - Mech. Eng.

NEWCASTLE UNIVERSITY MECHANICAL ENGINEERING



PART NUMBER	ITEM	QTY
Side Panel version B (short) - Lazer Cut - 10mm steel plate	1	2
L Bracket (Bolted) - Buy In - 6mm steel plate	2	8
U Bracket (Bolted) - Buy In - 6mm steel plate	3	4
Sideways panel (slide) - Lazer Cut - 8mm steel plate	4	4
M10 Nut	7	40
M10 x 10mm bolt	8	24
M10 x 35mm bolt	9	16
U Channel (Bolted) 60mm length - Buy In - 5mm steel	12	4
Castor RD 200D	13	2
Castor CC 125D	14	2



TROLLEY v8 - PER WHEELSET

STOCK No.
SHEET No. 1
FILE Trolley v8 (short) - per wheelset (10-25 inch) (No Green Part) - assembly.jpg

SCALE 1 : 9

DATE 25/10/2021

REVISION No.

DESIGNED CYRIL CHEUNG

CHECKED R I DAVIDSON

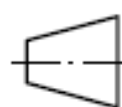
PROJECT IMechE RC - HSToon

COMPANY NCL UNI Mech. Eng.

NOTES:

THIS IS PER WHEELSET TROLLEY
FOR THE FULL LOCO 4 OF SUCH IS NEEDED

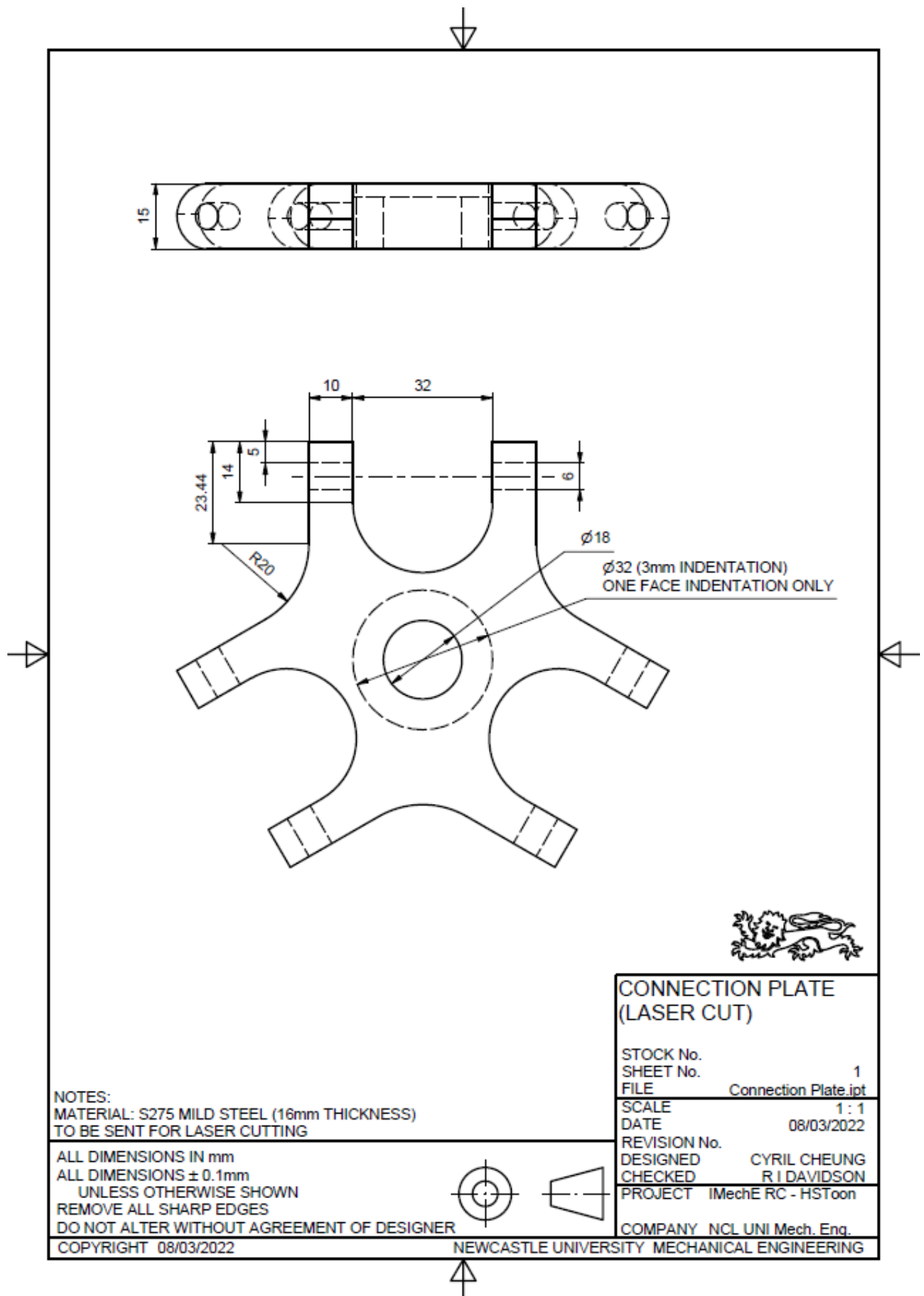
ALL DIMENSIONS IN mm
ALL DIMENSIONS ± 0.1 mm
UNLESS OTHERWISE SHOWN
REMOVE ALL SHARP EDGES
DO NOT ALTER WITHOUT AGREEMENT OF DESIGNER

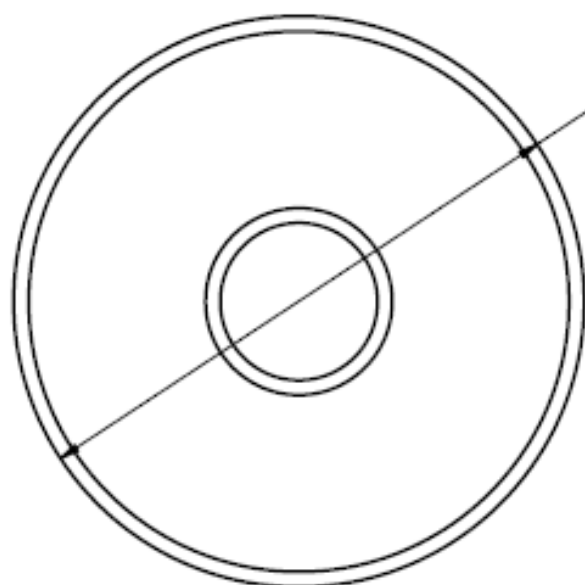


COPYRIGHT 25/10/2021

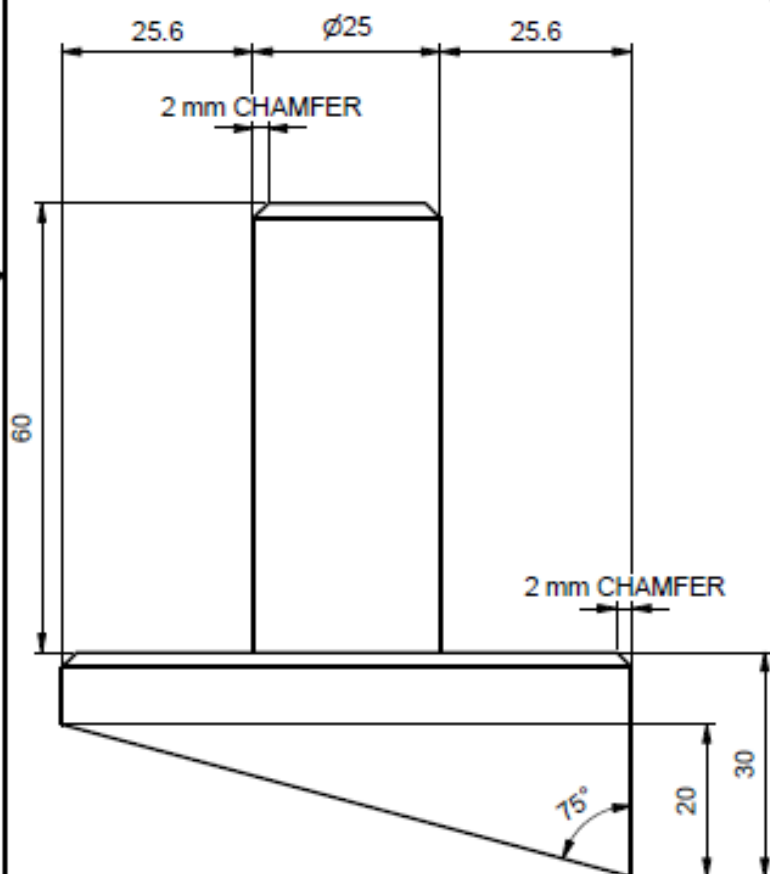
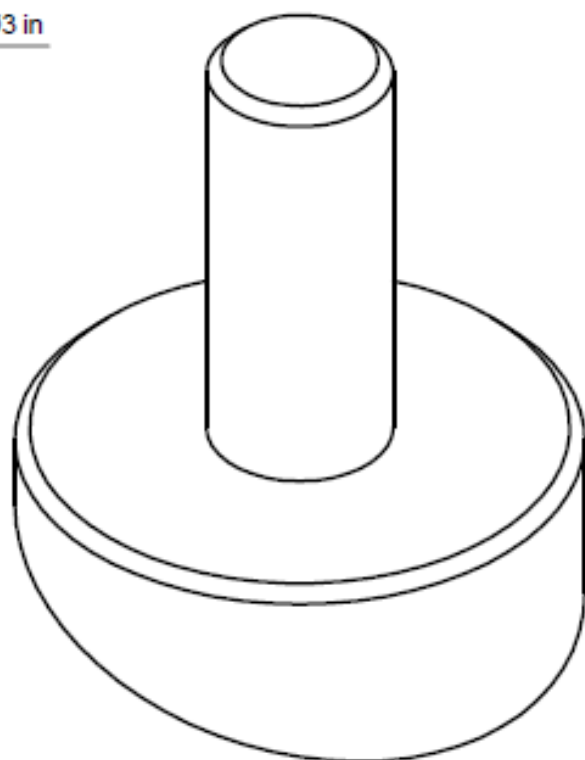
NEWCASTLE UNIVERSITY MECHANICAL ENGINEERING

Appendix A-2: Body Lift CAD Drawings





Ø3 in



**ALUMINIUM ROUND BLOCK
(BOTTOM STAND)
(3in BASE DIAMETER, 15deg)**

STOCK No.

SHEET No.

1

FILE Aluminium 6082 T6 Round Block - 3in bottom stand.ip

SCALE

1 : 1

DATE

29/03/2022

REVISION No.

DESIGNED

CYRIL CHEUNG

CHECKED

R.I DAVIDSON

PROJECT IMechE RC - HSToon

COMPANY NCL UNI Mech. Eng.

NOTES:

ALUMINIUM 6082 T6 ROUND BLOCK

TO BE BOUGHT IN MACHINED TO SIZE BY LATHE TURNING

ALL DIMENSIONS IN mm

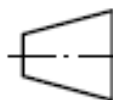
ALL DIMENSIONS $\pm 0.1\text{mm}$

UNLESS OTHERWISE SHOWN

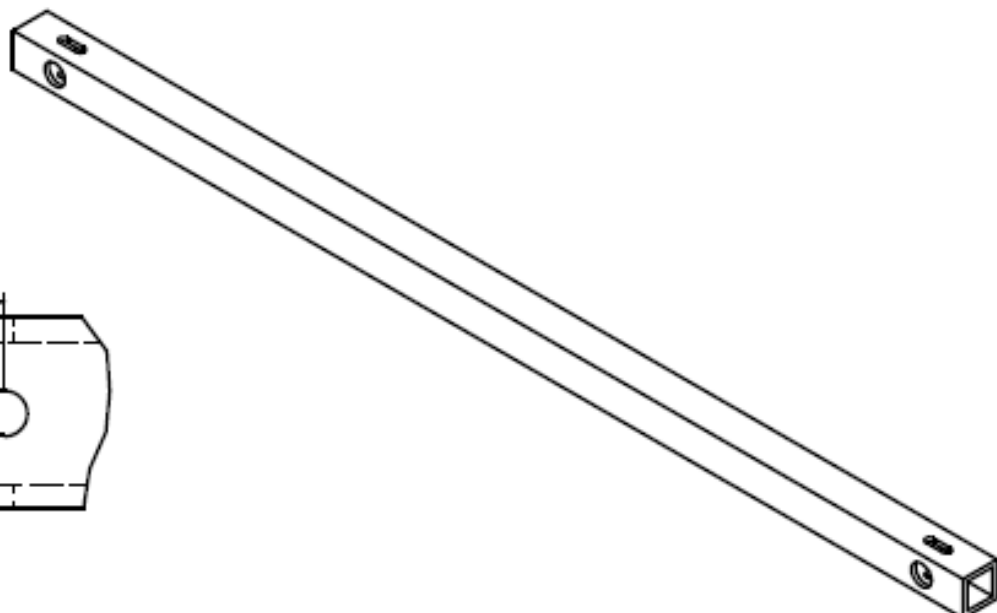
REMOVE ALL SHARP EDGES

DO NOT ALTER WITHOUT AGREEMENT OF DESIGNER

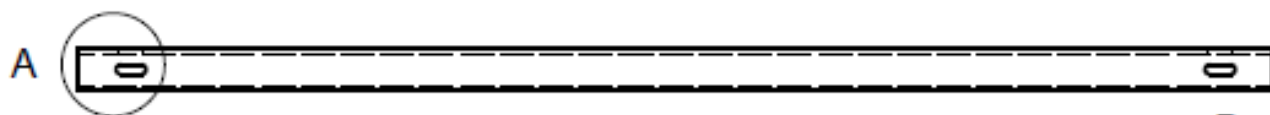
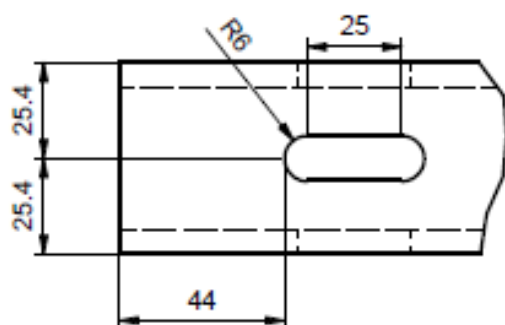
COPYRIGHT 29/03/2022



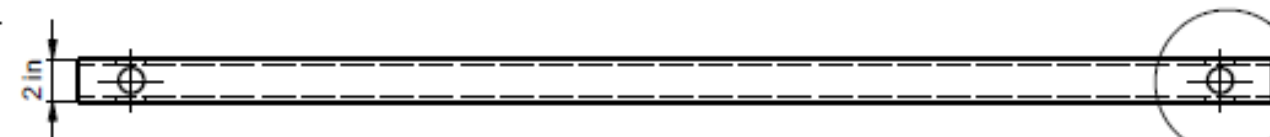
NEWCASTLE UNIVERSITY MECHANICAL ENGINEERING



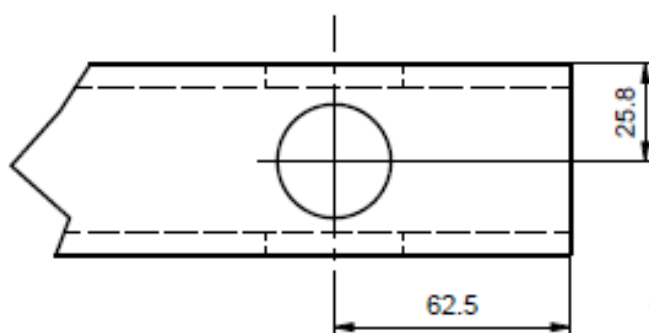
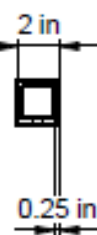
A (1 : 2)



B

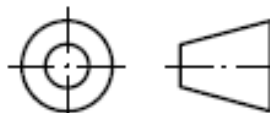


B (1 : 2)



NOTES:
ALUMINIUM 6082 T6 BOXED TUBE
TO BE BOUGHT IN AND HOLES DRILLED THROUGH MANUALLY

ALL DIMENSIONS IN mm
ALL DIMENSIONS ± 0.1 mm
UNLESS OTHERWISE SHOWN
REMOVE ALL SHARP EDGES
DO NOT ALTER WITHOUT AGREEMENT OF DESIGNER



COPYRIGHT 08/03/2022

NEWCASTLE UNIVERSITY MECHANICAL ENGINEERING

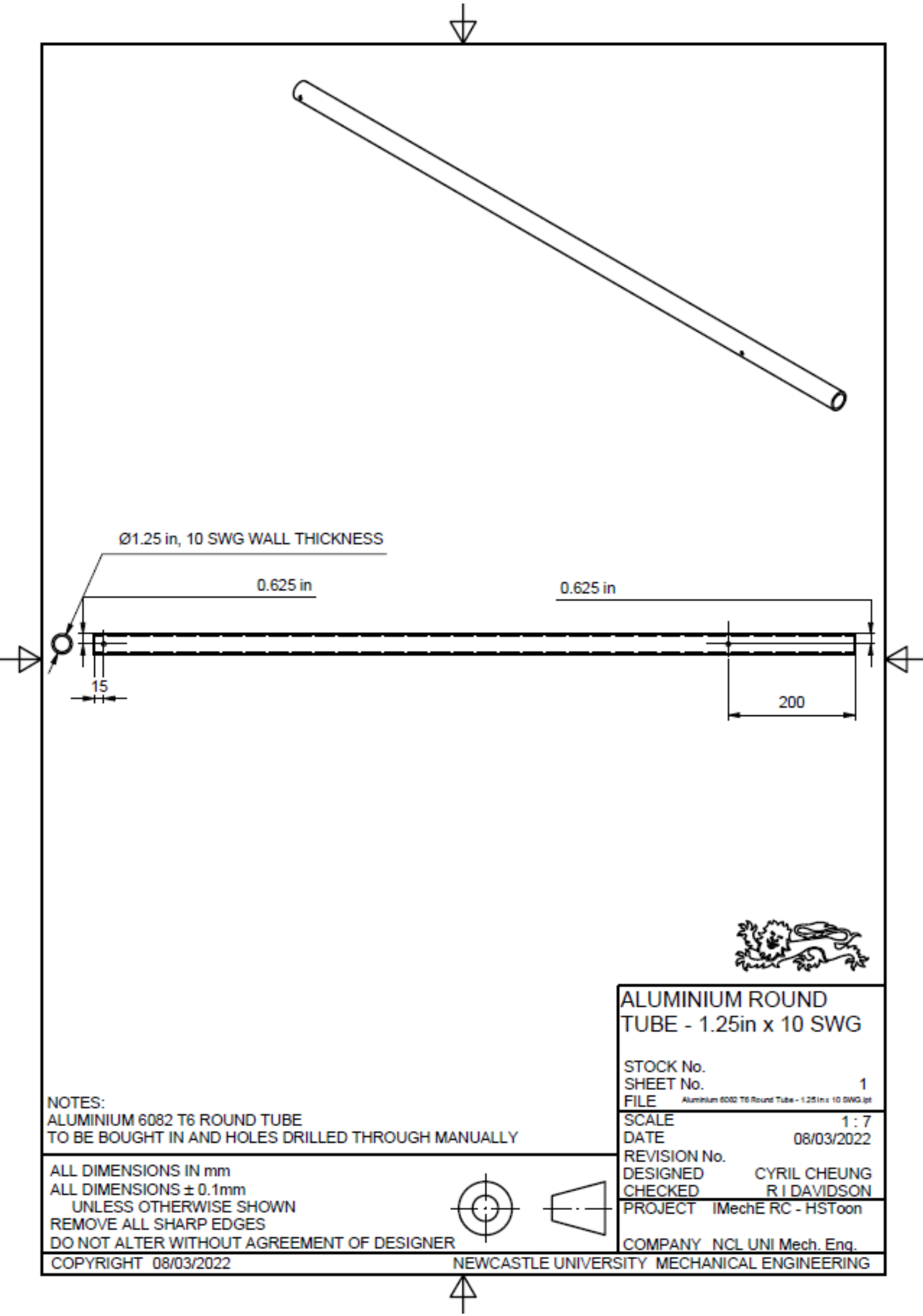
ALUMINIUM BOXED
TUBE - 2x2x0.25 in

STOCK No.
SHEET No. 1
FILE Aluminium 6082 T6 Boxed Tube - 2x2x0.25 in.dwg

SCALE 1 : 9
DATE 08/03/2022

REVISION No.
DESIGNED CYRIL CHEUNG
CHECKED R I DAVIDSON
PROJECT IMechE RC - HSToon

COMPANY NCL UNI Mech. Eng.



Ø1.25 in, 10 SWG WALL THICKNESS

0.625 in

0.625 in

15

200

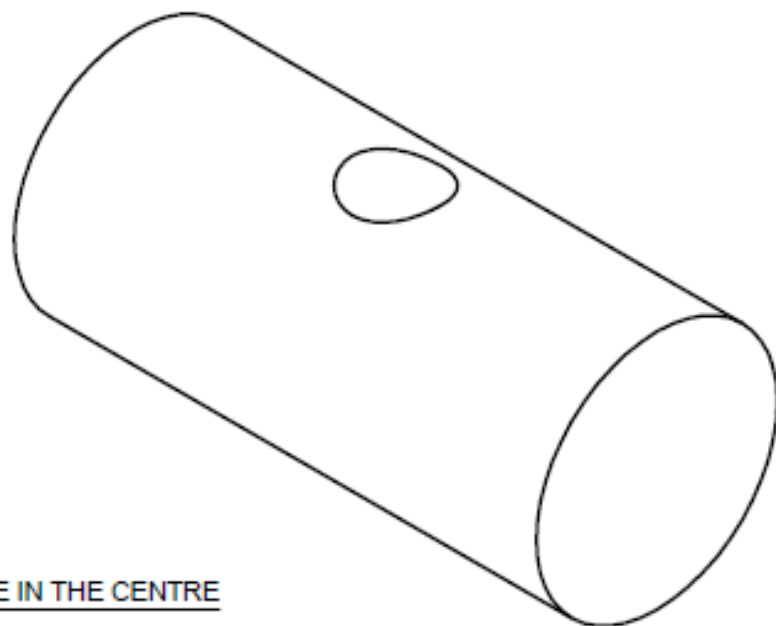


ALUMINIUM ROUND TUBE - 1.25in x 10 SWG	
STOCK No.	
SHEET No.	1
FILE	Aluminium 6062 T6 Round Tube - 1.25in x 10 SWG.dwg
SCALE	1 : 7
DATE	08/03/2022
REVISION No.	
DESIGNED	CYRIL CHEUNG
CHECKED	R I DAVIDSON
PROJECT	IMechE RC - HSToon
COMPANY	NCL UNI Mech. Eng.

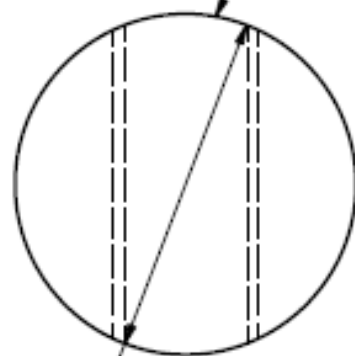
NOTES:
ALUMINIUM 6082 T6 ROUND TUBE
TO BE BOUGHT IN AND HOLES DRILLED THROUGH MANUALLY

ALL DIMENSIONS IN mm
ALL DIMENSIONS $\pm 0.1\text{mm}$
UNLESS OTHERWISE SHOWN
REMOVE ALL SHARP EDGES
DO NOT ALTER WITHOUT AGREEMENT OF DESIGNER

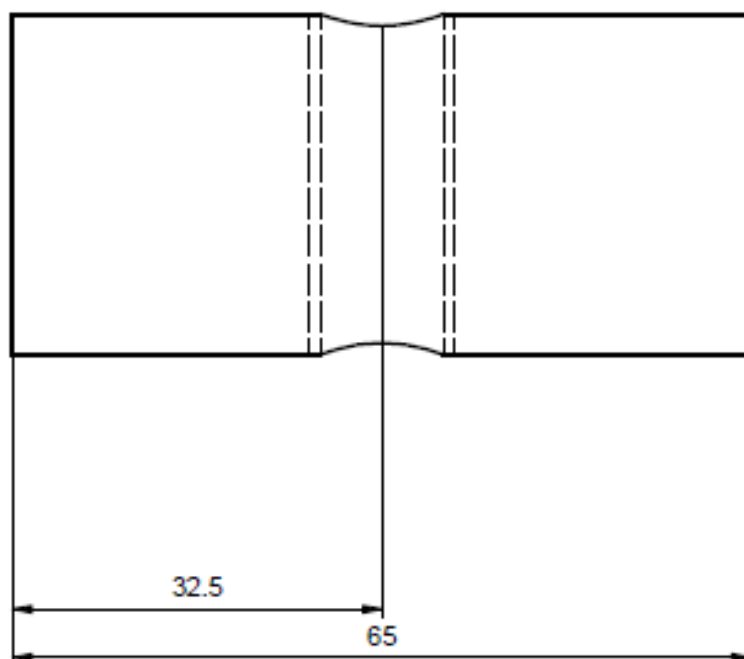




THROUGH M12 HOLE IN THE CENTRE



Ø30



NOTES:

MATERIAL USED: EN24T STAINLESS STEEL
QTY: 4 (IN MERZ WORKSHOP WITH OTHER FORMULA TRAIN PARTS)

DIAMETER 30mm, THROUGH M12 HOLE IN THE CENTRE

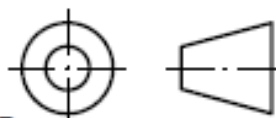
THANKS STEVE!

ALL DIMENSIONS IN mm
ALL DIMENSIONS $\pm 0.1\text{mm}$
UNLESS OTHERWISE SHOWN
REMOVE ALL SHARP EDGES
DO NOT ALTER WITHOUT AGREEMENT OF DESIGNER

COPYRIGHT 31/03/2022

STAINLESS STEEL
SAUSAGE - BODY LIFT

STOCK No.	
SHEET No.	1
FILE	Sausage - CC (sorry Rob).ipt
SCALE	1.5
DATE	31/03/2022
REVISION No.	
DESIGNED	CYRIL CHEUNG
CHECKED	R I DAVIDSON
PROJECT	IMechE RC - HSToon
COMPANY	NCL UNI Mech. Eng.



NEWCASTLE UNIVERSITY MECHANICAL ENGINEERING



BODY LIFT

COMPANY NCL UNI Mech. Eng.

NEWCASTLE UNIVERSITY MECHANICAL ENGINEERING

Appendix A-3: Body lift – fasteners strength justifications

DESIGN & MANUFACTURING 2
(MEC2007)

F. FASTENERS



BOLT, NUT & WASHER DIMENSIONS

Size (Thread diam.)	Pitch	Core Diam.	Tensile Stress Area	Hexagon Head Bolts & Screws			Socket Cap (Allen) Bolts & Screws				Nut Thick- ness	Washer	
				Proof Load for Grade 8.8 600N/mm ²	Across Flats	Head Height	Proof Load for Grade 12.9 970N/mm ²	Hex Socket Across Flats	Head Diam.	Head Height		Diam	Thick- ness
(mm)	(mm)	(mm)#	(mm ²)#	(kN)	(mm)*	(mm)*	(kN)	(mm)	(mm)*	(mm)*	(mm)*	(mm)*	(mm)*
M3	0.5	2.27	5.03	3.0	5.5	2.2	4.9	2.5	5.5	3	2.4	7	0.5
M4	0.7	3.00	8.78	5.2	7	3.0	8.5	3	7	4	3.2	9	0.8
M5	0.8	3.87	14.2	8.5	8	3.4	13.8	4	8.5	5	4	10	1.0
M6	1.0	4.60	20.1	12.0	10	4.2	19.5	5	10	6	5	12.5	1.6
M8	1.25	6.27	36.6	21.9	13	5.7	35.5	6	13	8	6.5	17	1.6
M10	1.5	7.94	58.0	34.8	17	7.2	56.3	8	16	10	8	21	2.0
M12	1.75	9.60	84.3	50.5	19	8.2	81.8	10	18	12	10	24	2.5
M16	2.0	13.3	144	86.4	24	10.2	140	14	24	16	13	30	3.0
M20	2.5	16.6	225	135	30	13.3	218	17	30	20	16	37	3.0
M24	3.0	19.9	324	194	36	15.3	314	19	36	24	19	44	4.0
M30	3.5	25.3	519	311	46	19.3					24	56	4.0
M36	4.0	30.6	817	490	55	23.3					29	66	5.0

* maximum values

minimum values

Note: Proof Load is that load at which the bolt begins to plastically deform,
not the Ultimate Tensile Load.

MECHANICAL PROPERTY CLASSES

For ISO, METRIC STEEL BOLTS, SCREWS & STUDS (selected grades only)

Property Class (grade)	Size range (mm)	Proof Stress (MPa)	Yield Stress (MPa)	Ultimate Tensile Stress (MPa)	Steel Type & Treatment (Typical Types)
4.6	M3 to M36	225	250	400	Low or medium carbon steel (most unmarked bolts & screws)
8.8	M3 to M36	600	635	830	Medium carbon steel, Quenched & Tempered. (most "High Tensile" bolts)
10.9	M5 to M36	830	895	1040	Low carbon martensitic steel, Quenched & Tempered. (some "Allen Screws")
12.9	M3 to M36	970	1100	1220	Alloy steel, Quenched & Tempered (most "Allen Screws" & similar)

Note: The property class numbers, **x.y**, are based on:-

x approx. UTS of material in 100's of N/mm²

.y proportion (¹/₁₀'s) of UTS at yield

e.g. Grade 8.8 means:- UTS of approx 800N/mm² and
Yield Stress of approx. 800N/mm² × 0.8 = 640N/mm²

According to Handout F of Fasteners (RID-MEC2007), a grade 8.8 M12 nut could hold 50.5 kN at 600 N/mm² where only *half loco's* 500kg × 9.81 = 4.905 kN is needed to withstand.

Grade 8.8 M6 can also take 12.0 kN at 600 N/mm² in tension, meaning it could take 6.0 kN in shear, which is the mode experienced by the pins on the body lift.

Appendix A-4: Body lift – aluminium tube material choice justification

ALUMINIUM ALLOYS												
Material Name	Material Grade	E GNm ⁻²	G GNm ⁻²	ρ kgm ⁻³	α 10 ⁻⁶ C ⁻¹	σ _{yield} MNm ⁻²	σ _{UTS} MNm ⁻²	σ _e MNm ⁻²	Elong %	Temp Limit °C	Cost £££££	Notes
Soft Sheet	1050-0	69	26	2700	24	60	80	40	30%		££	Pressed panels
Half Hard Sheet	1050-H14	69	26	2700	24	115	125	55	5%		££	Bent Panels, etc.
Fully Hard Sheet	1050-H18	69	26	2700	24	143	160	76	4%		££	Flat Panels, etc.
Extrusions	6063-T4	69	26	2700	24	90	155	79	14%		££	Lightly stressed parts. Bendable
Extrusions	6082-T6	69	26	2700	24	270	310	124	8%		££	Bridges, Vehicle structures.

According to Handout E of materials (RID-MEC2007) aluminium 6082 T6 has a yield strength of 270 MPa. The with 6mm pins inserted through the tubes, its contact surface area with the aluminium tube is $3mm \times 6mm$ (pin) = $18mm^2$.

Meaning for this aluminium tube to yield, it must experience:

$275 MPa \times 18mm^2 = 4860N$ per side. However, note that the critical load of buckling for this was 2815.6364N, and this load case does not observe any buckling. So, if the tube were to fail, it would have buckled before the pins deforms the tube.

Appendix A-5: Body lift – bearing choice

The bearing fitted onto the body lift are believed to take a large axial load in both directions, which is the only load it should experience (no radial load). Though a thrust ball bearing would have been the ideal choice, a cheaper, still suitable double groove ball bearing was chosen instead. According to its technical brochure:



CALCULATION DATA

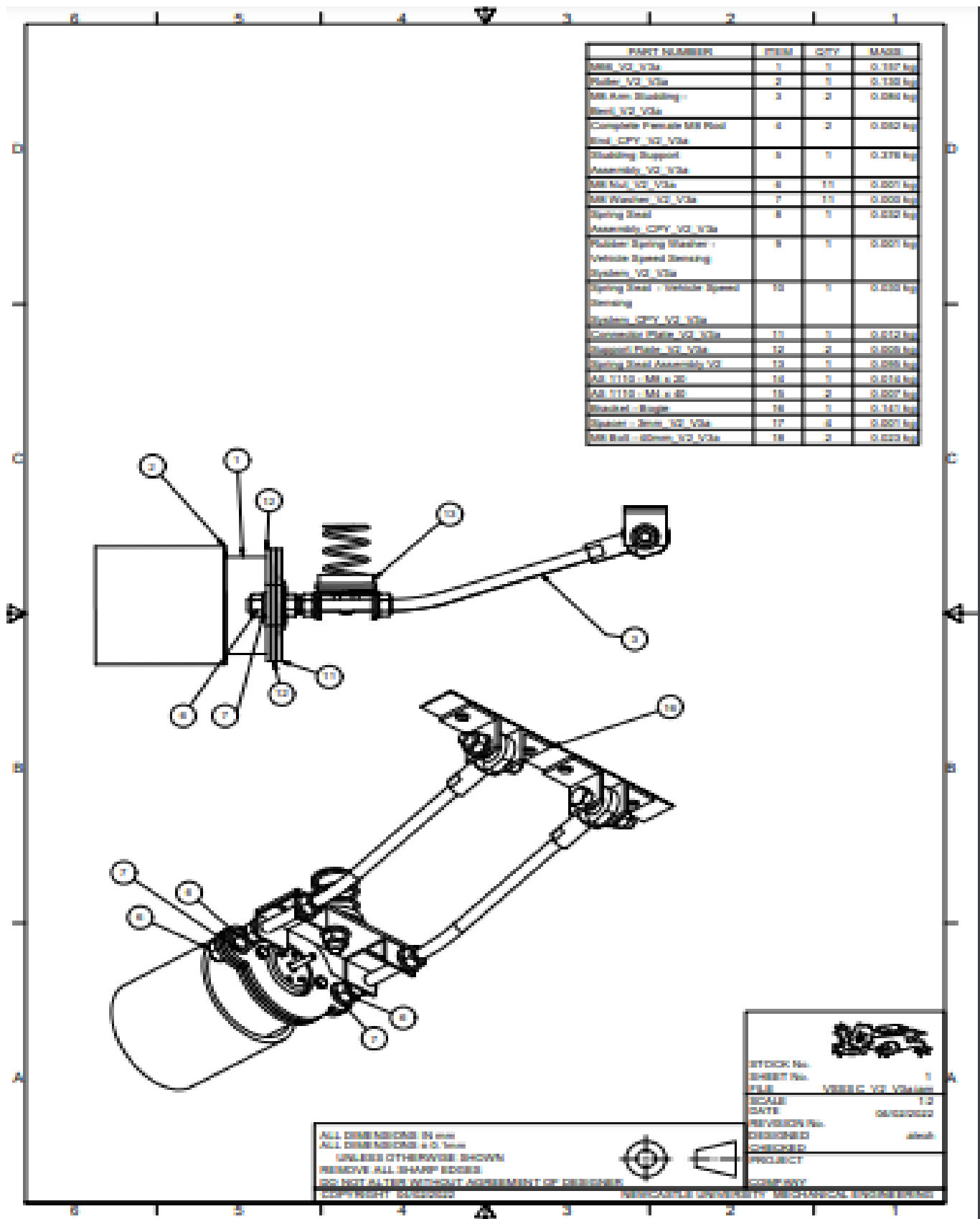
Basic dynamic load rating	C	10.6 kN
Basic static load rating	C ₀	6.2 kN
Fatigue load limit	P _u	0.26 kN
Reference speed		36 000 r/min
Limiting speed		20 000 r/min
Minimum load factor	k _r	0.05
Calculation factor	f ₀	12.2

This bearing has a static loading of 6.2 kN. Whereas in the worst load case, it only takes: $\frac{1000kg}{2} \times 9.81 = 4905N$. With a safety factor of: $\frac{6200}{4905} = 1.264$.

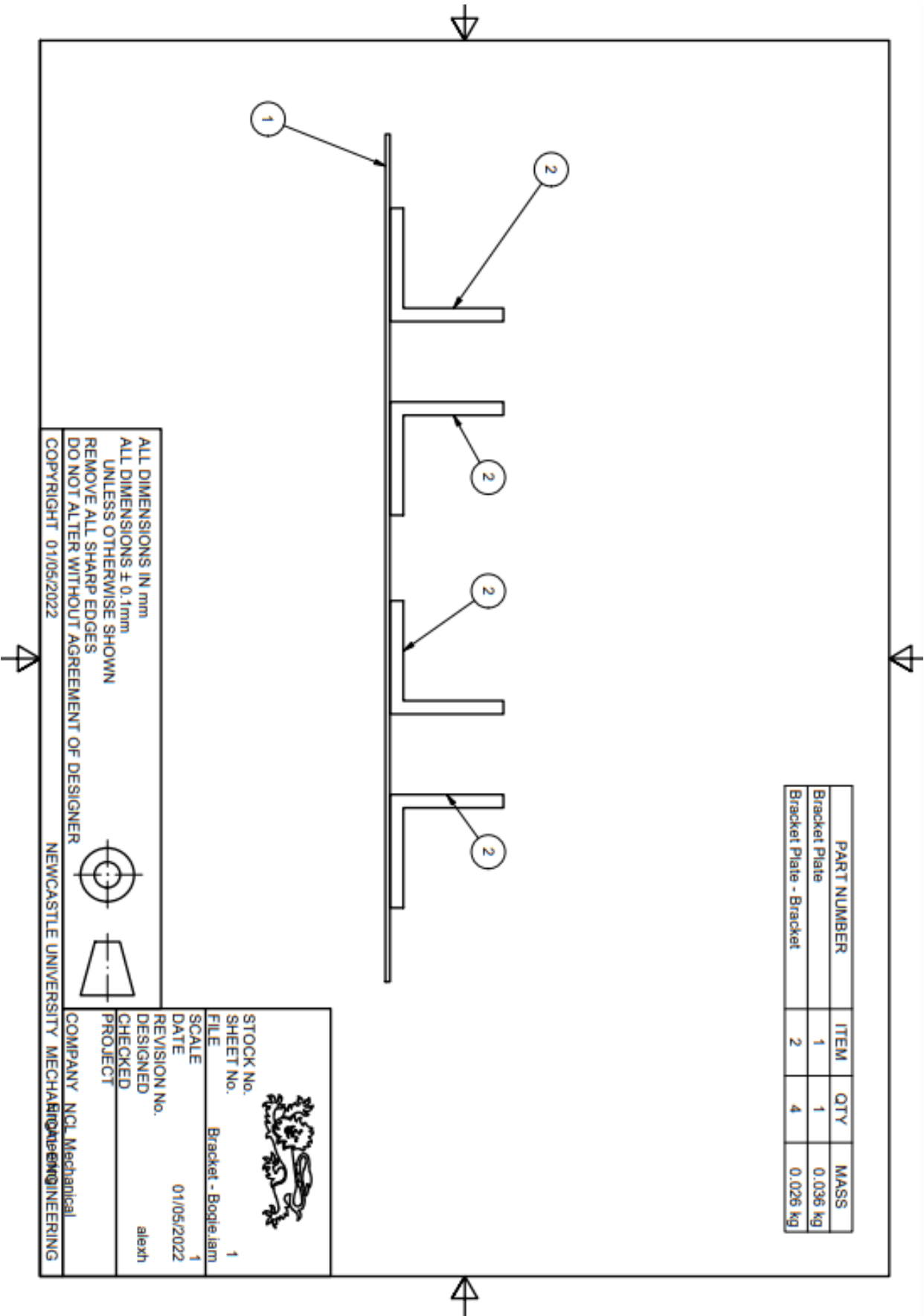
Therefore, this bearing being a cheaper option was still deemed safe.

Appendix B: Bogie

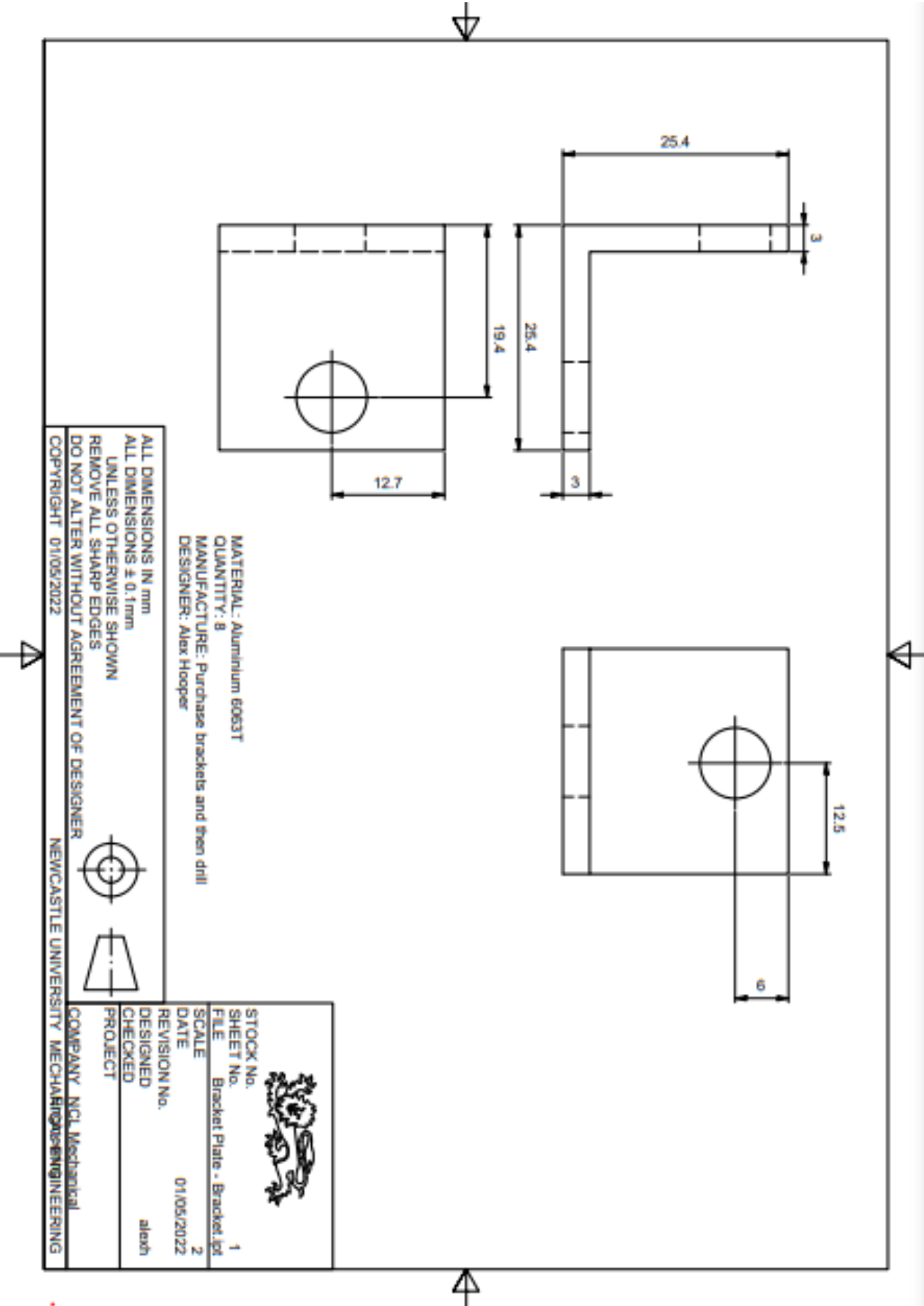
Appendix B-1: VSSS C Assembly (AH)



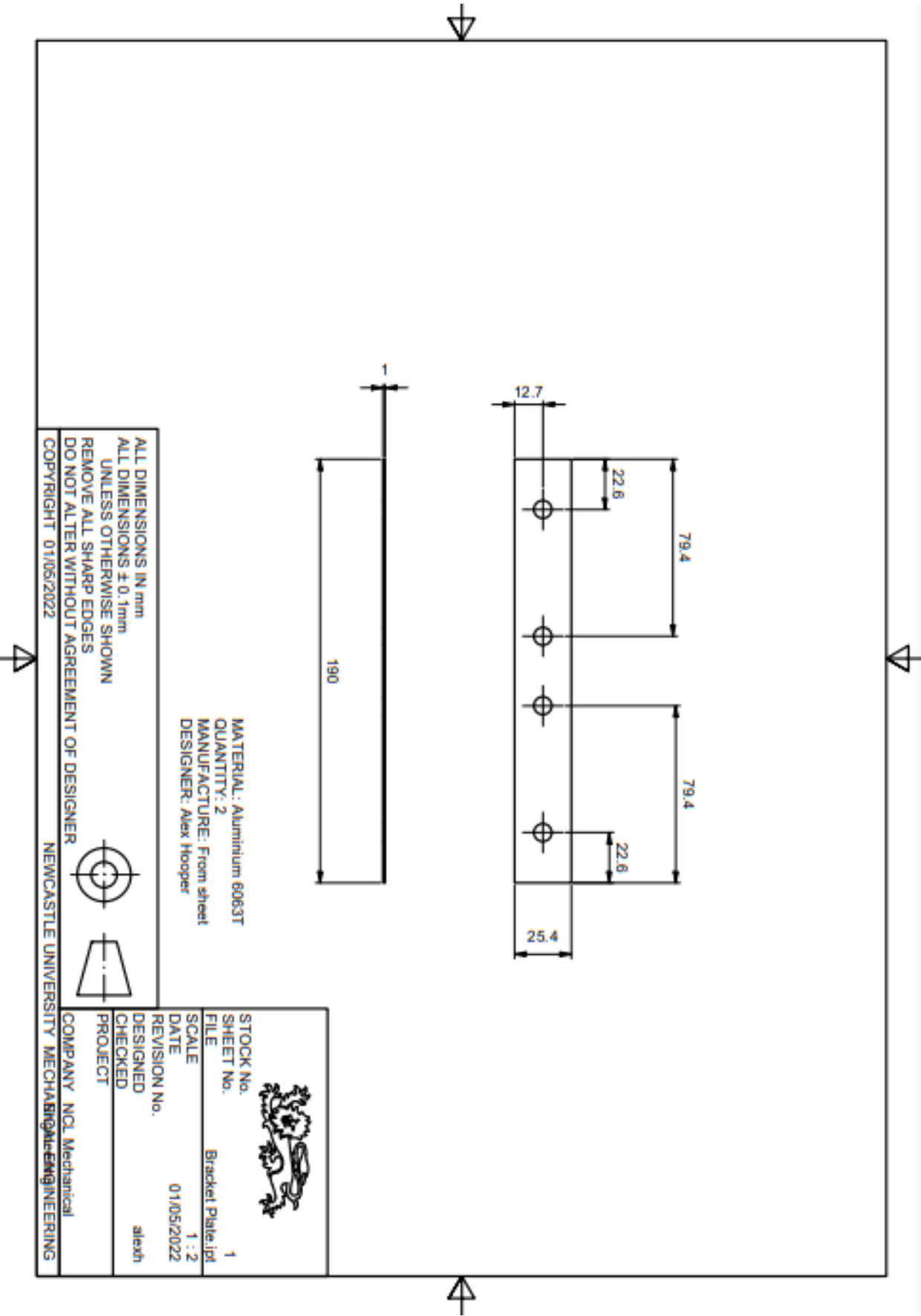
Appendix B-2: Bracket (Bogie) Assembly (AH)



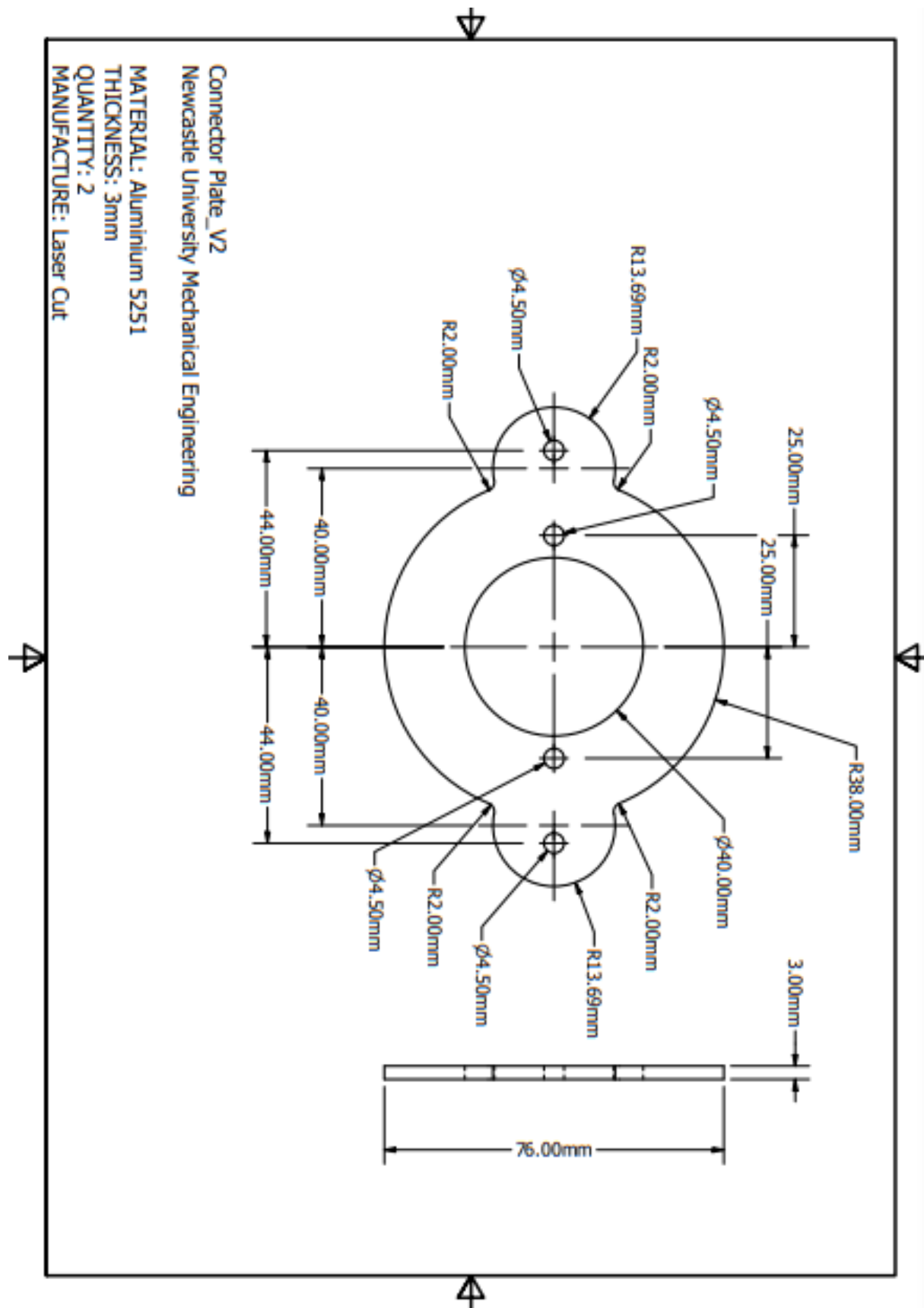
Appendix B-3: Bracket (AH)



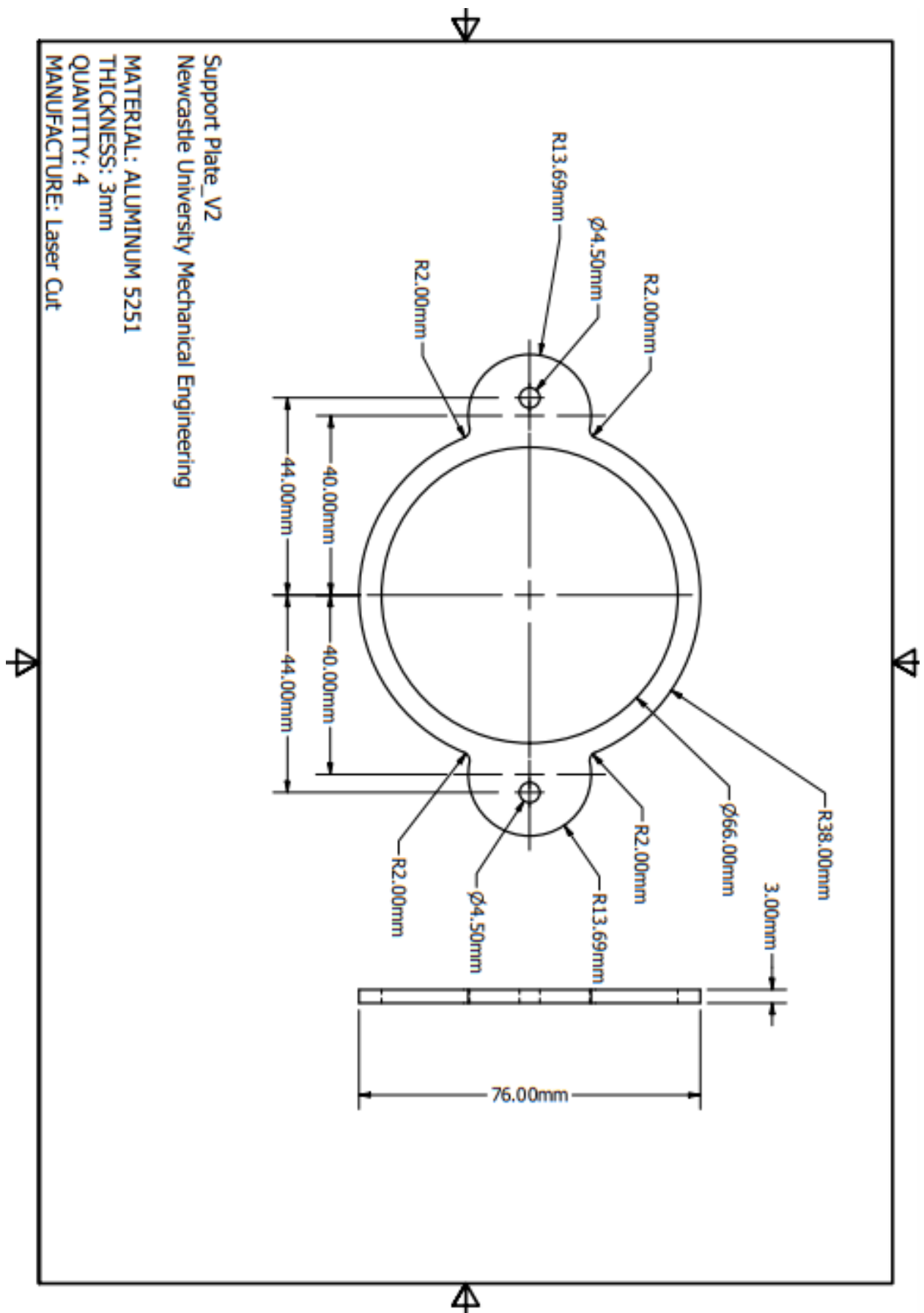
Appendix B-4: Bracket Plate (AH)



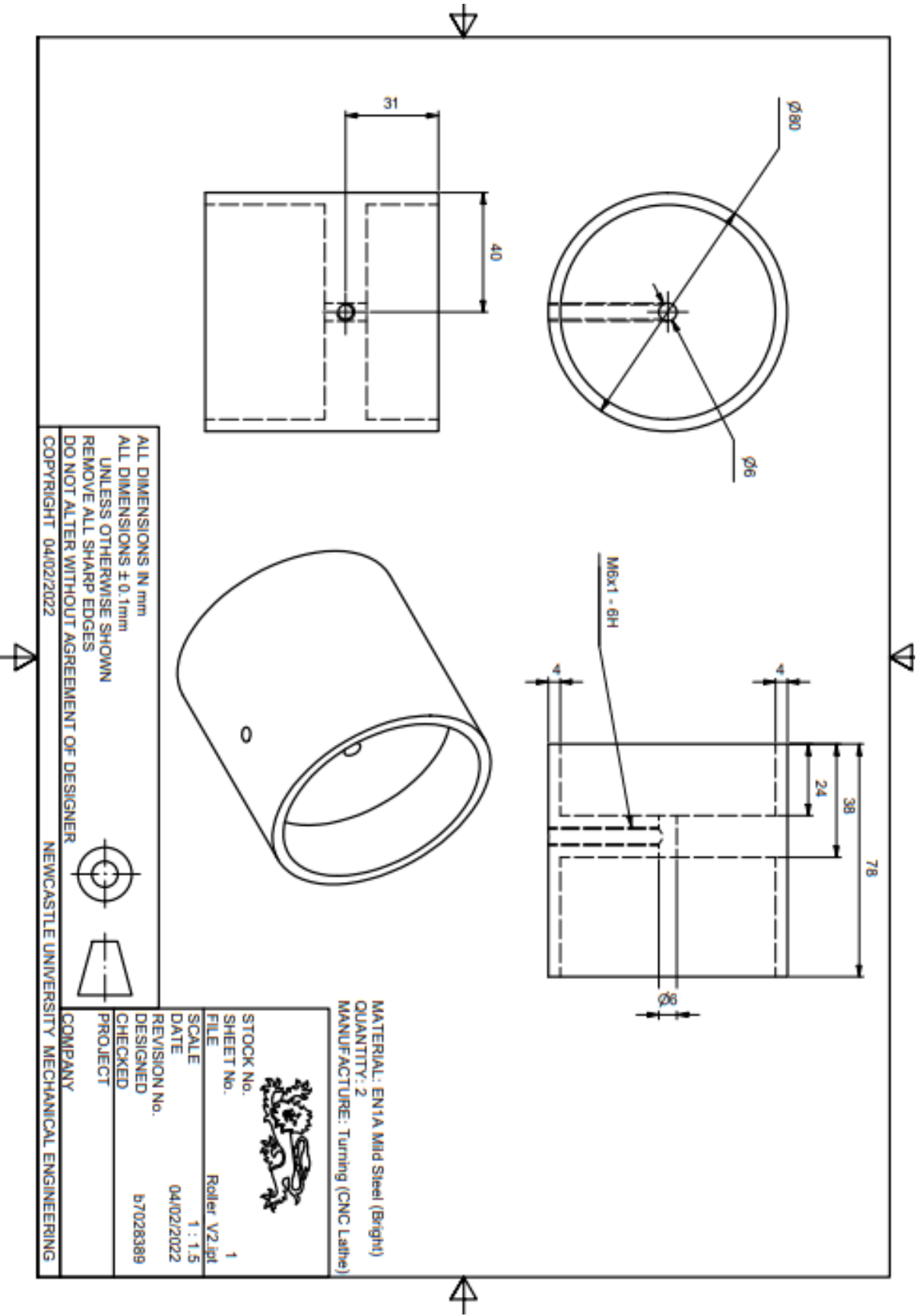
Appendix B-5: Connector Plate V2 (AH)



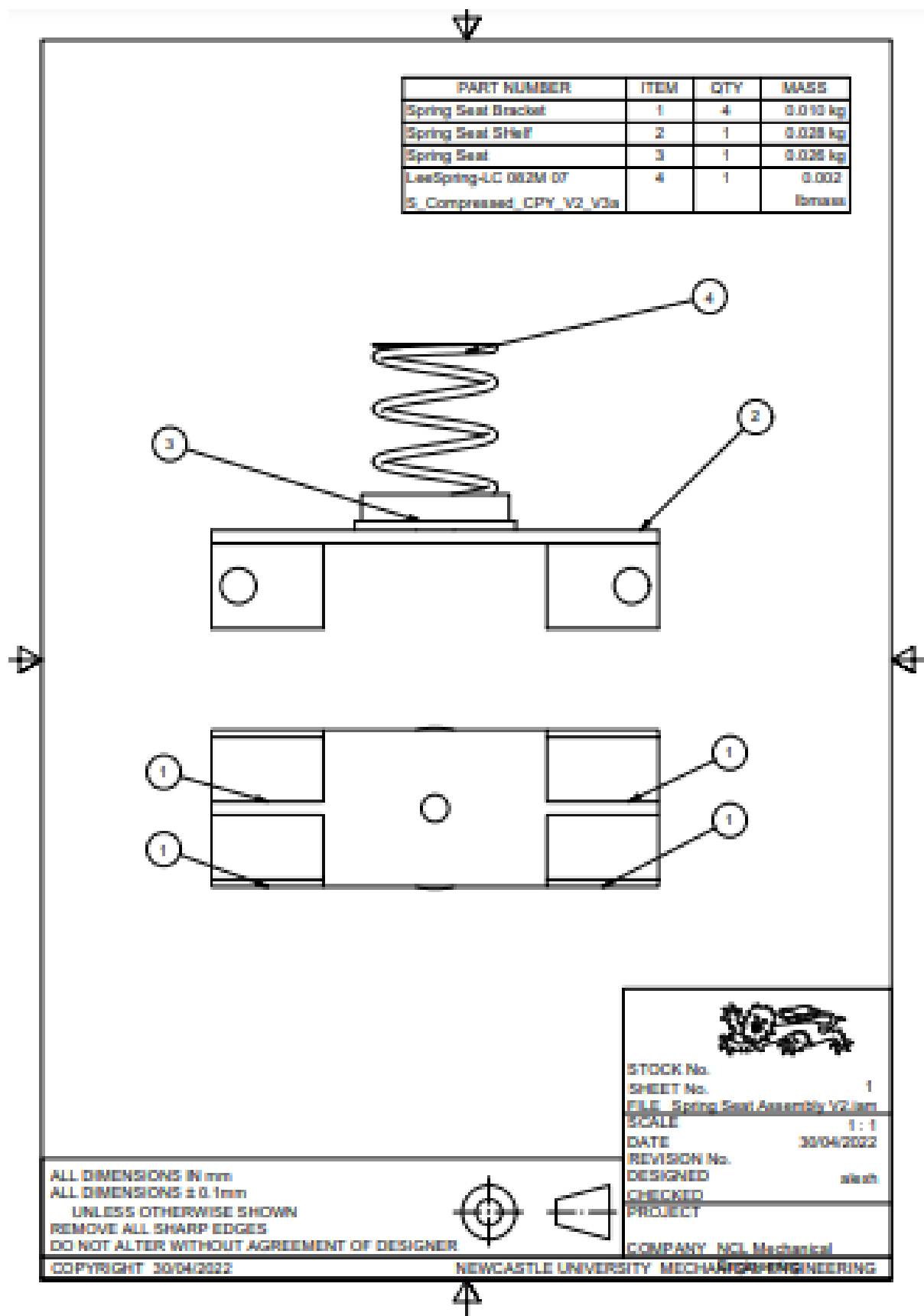
Appendix B-6: Support Plate V2 (AH)



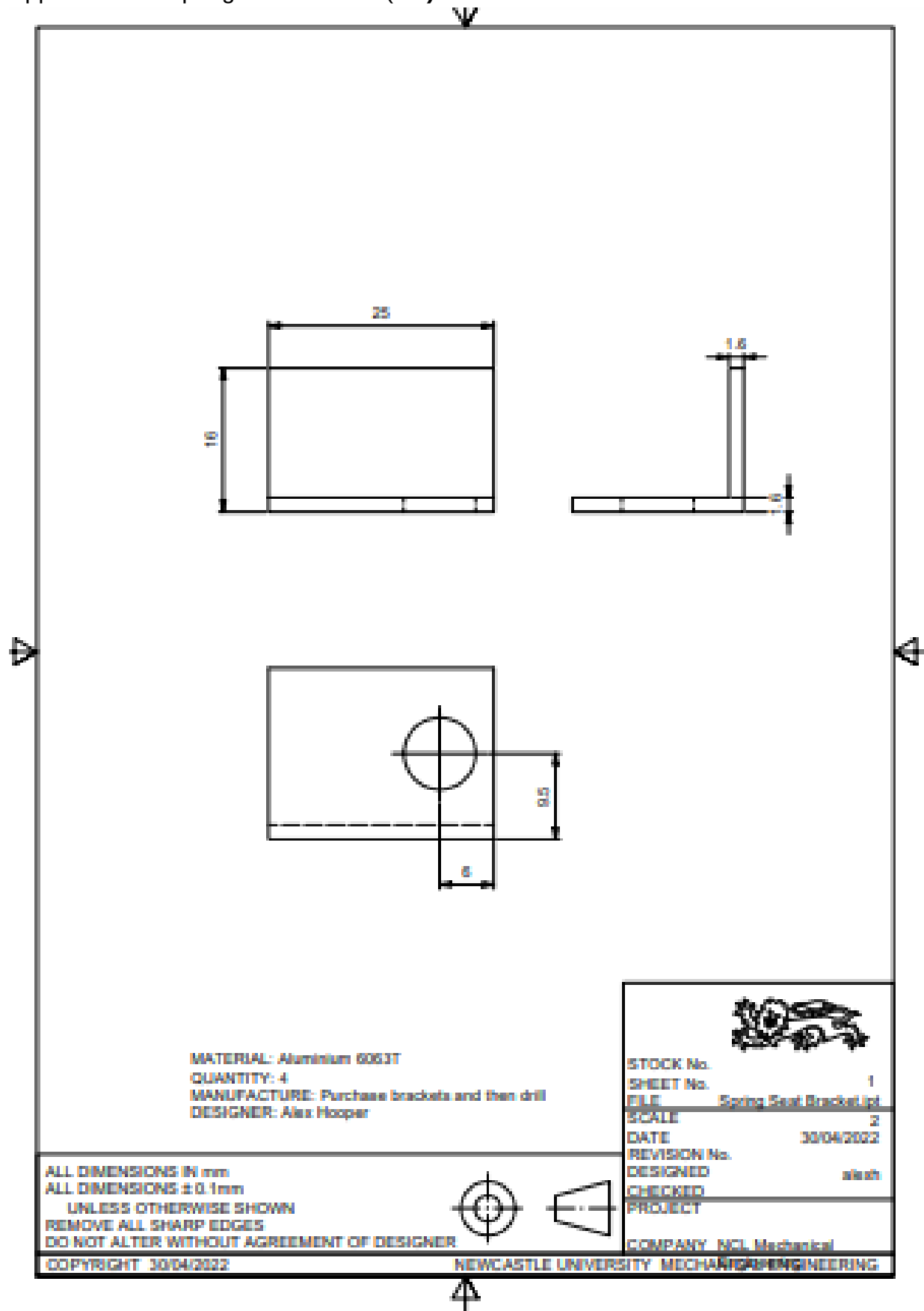
Appendix B-7: Roller V2 (AH)



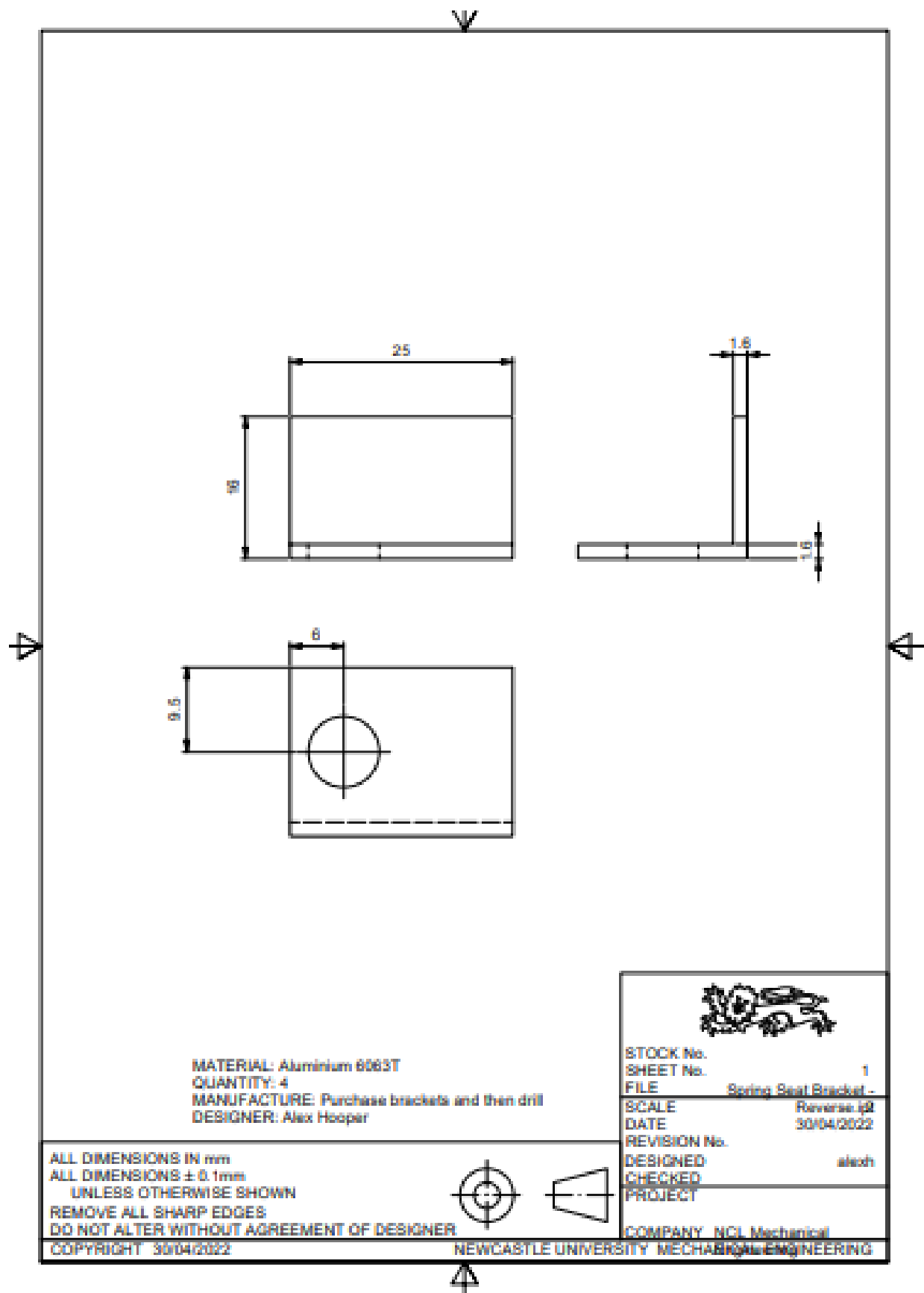
Appendix B-8: Spring Seat Assembly V2 (AH)



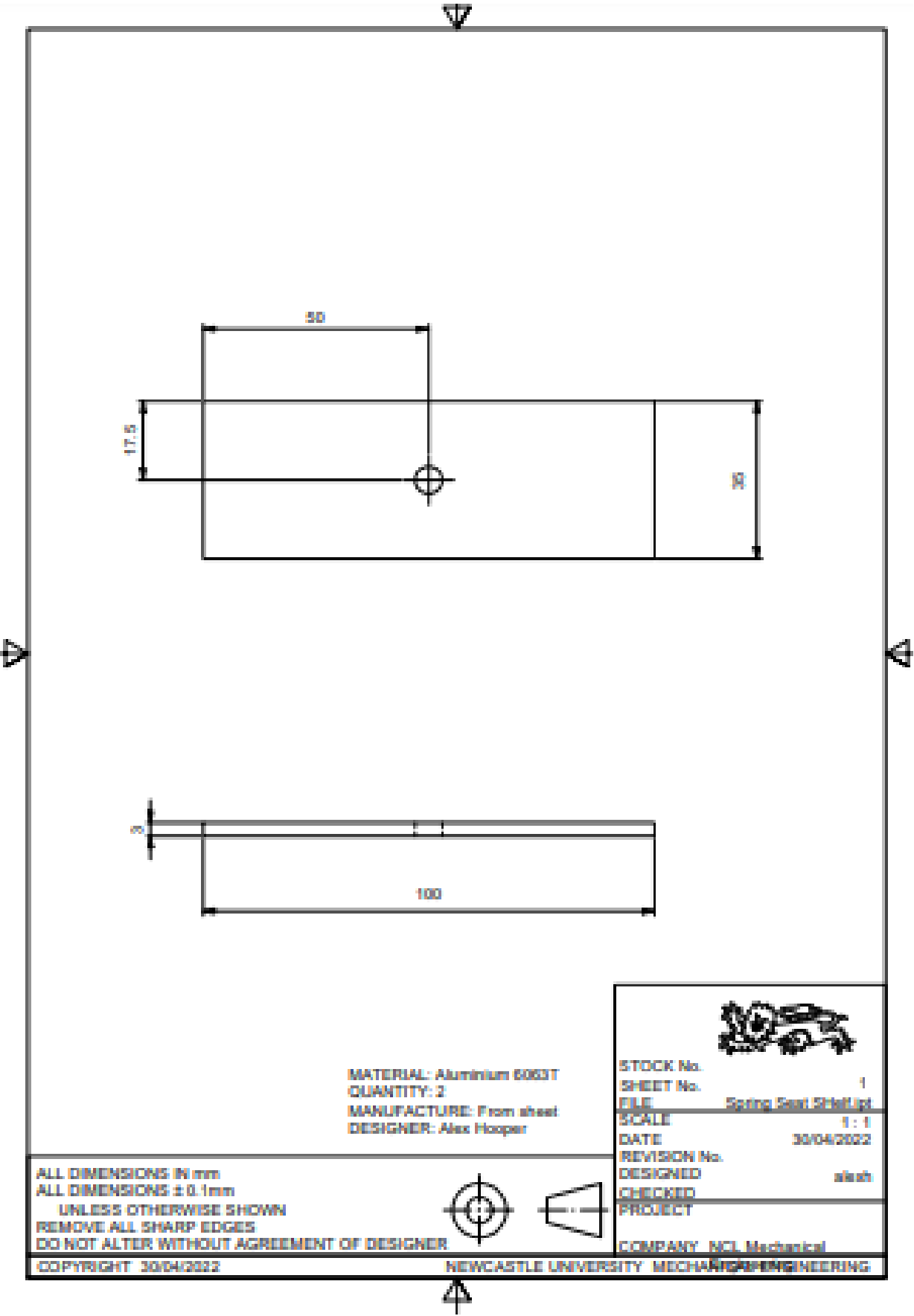
Appendix B-9: Spring Seat Bracket (AH)



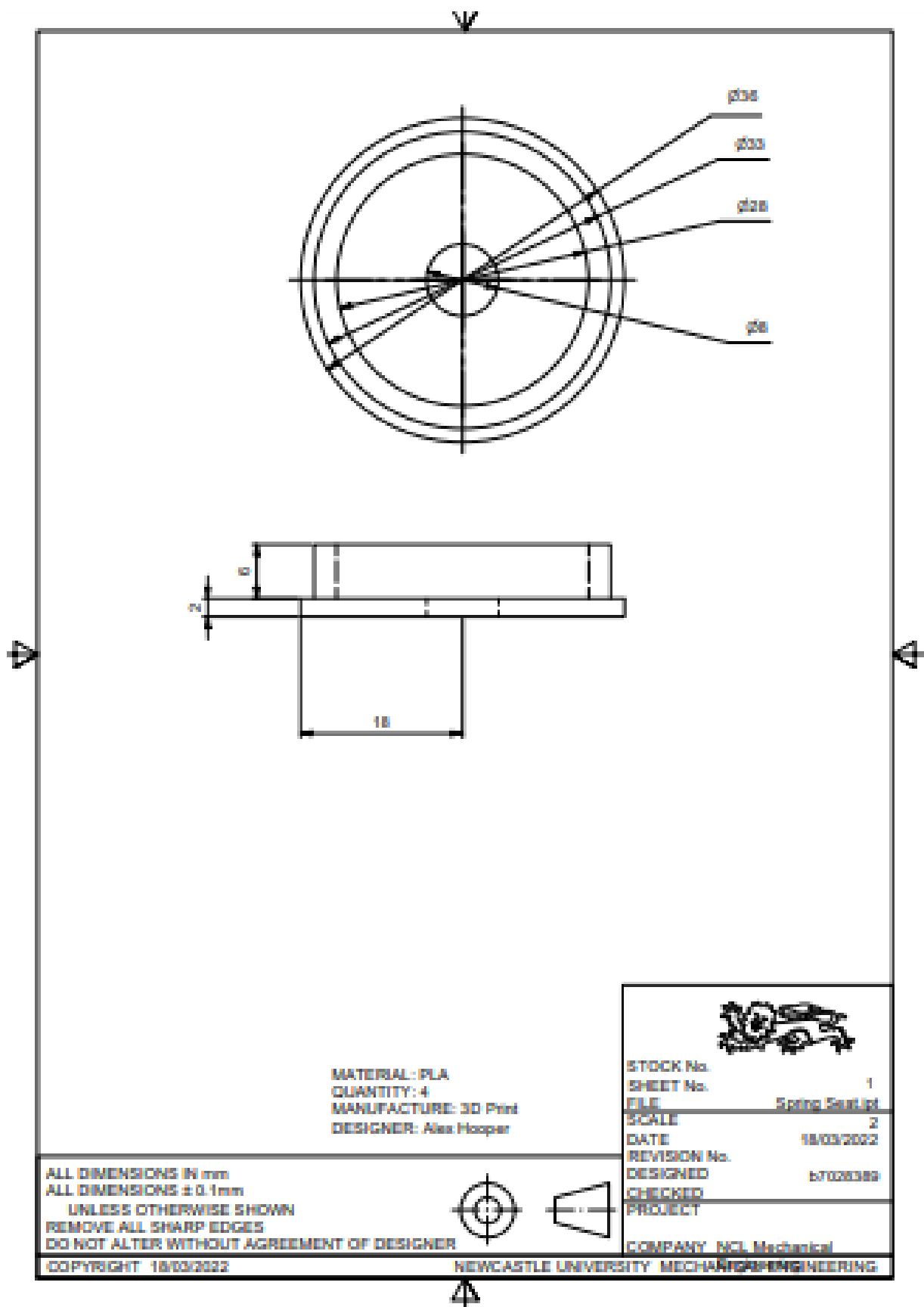
Appendix B-10: Spring Seat Bracket (Reverse) (AH)



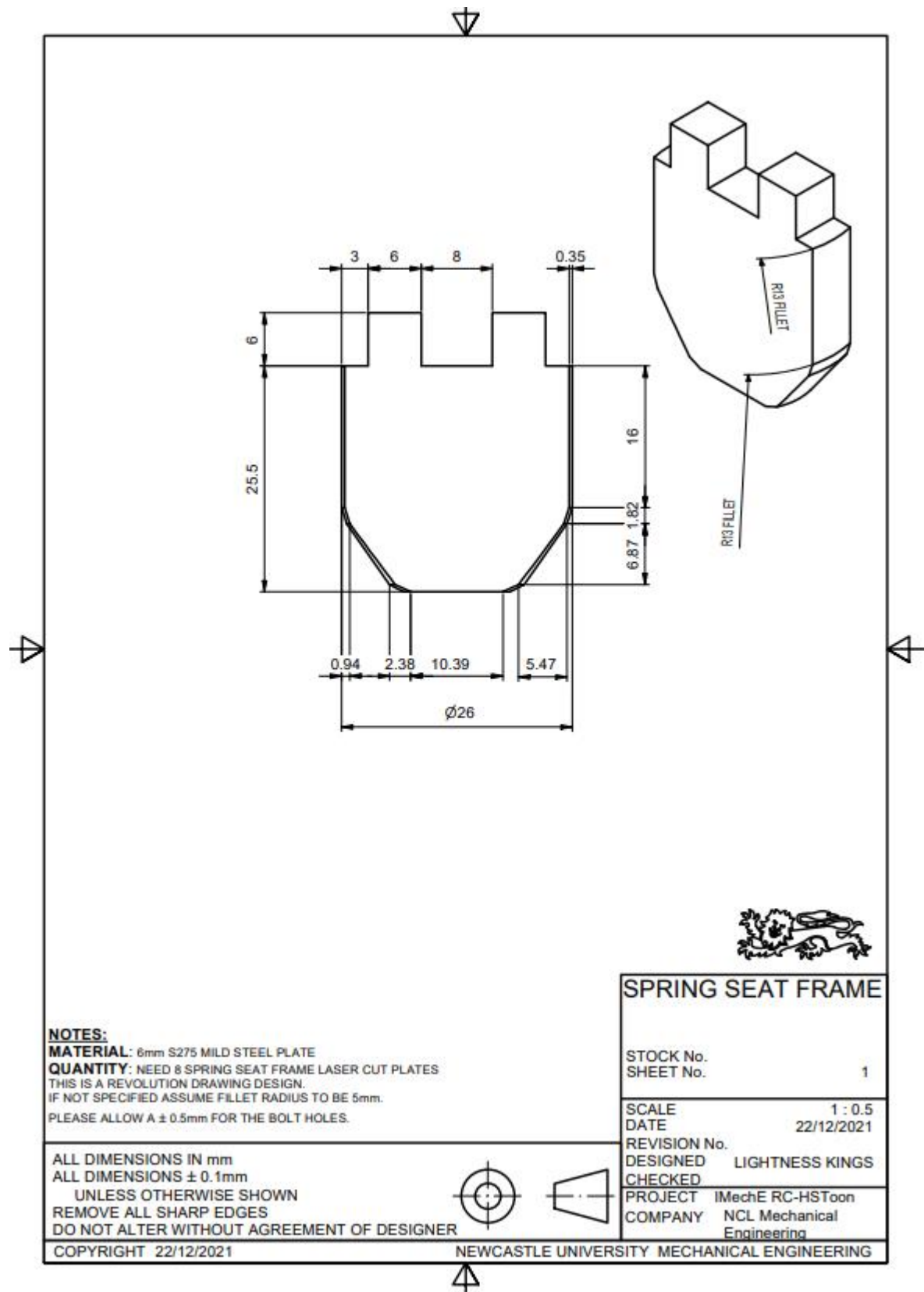
Appendix B-11: Spring Seat Shelf (AH)



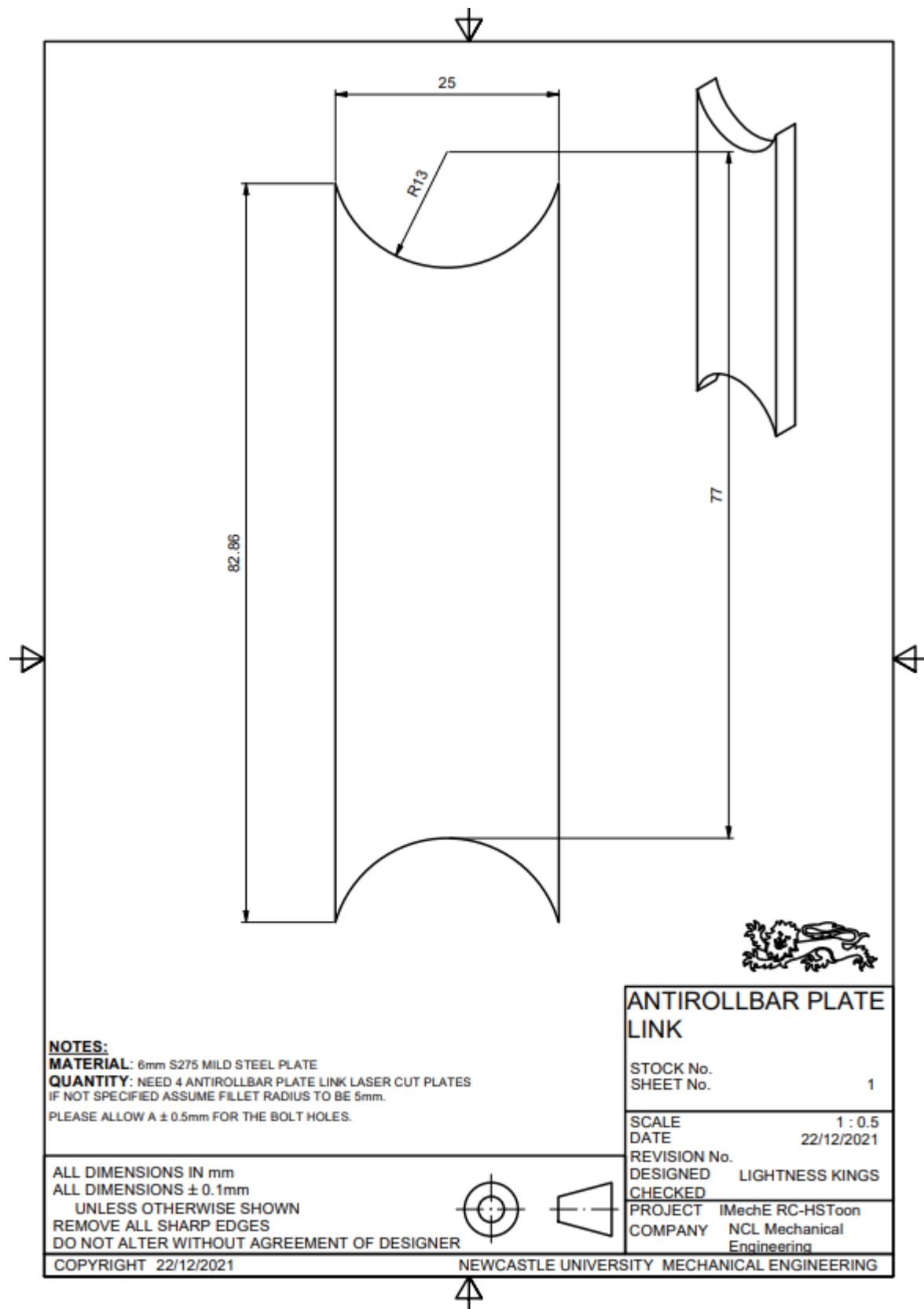
Appendix B-12: Spring Seat (AH)



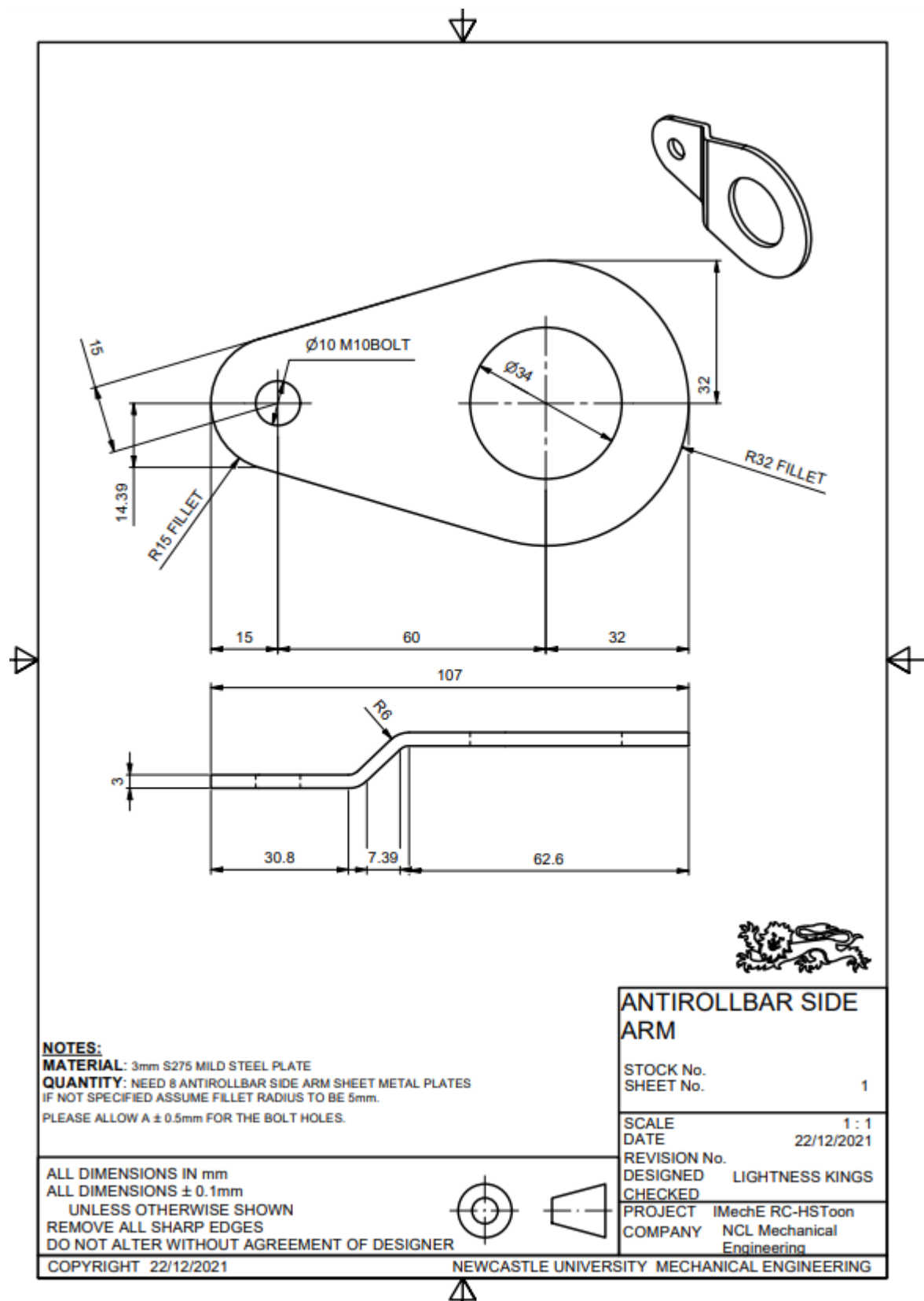
Appendix B-13 Spring Seat Frame (LK)



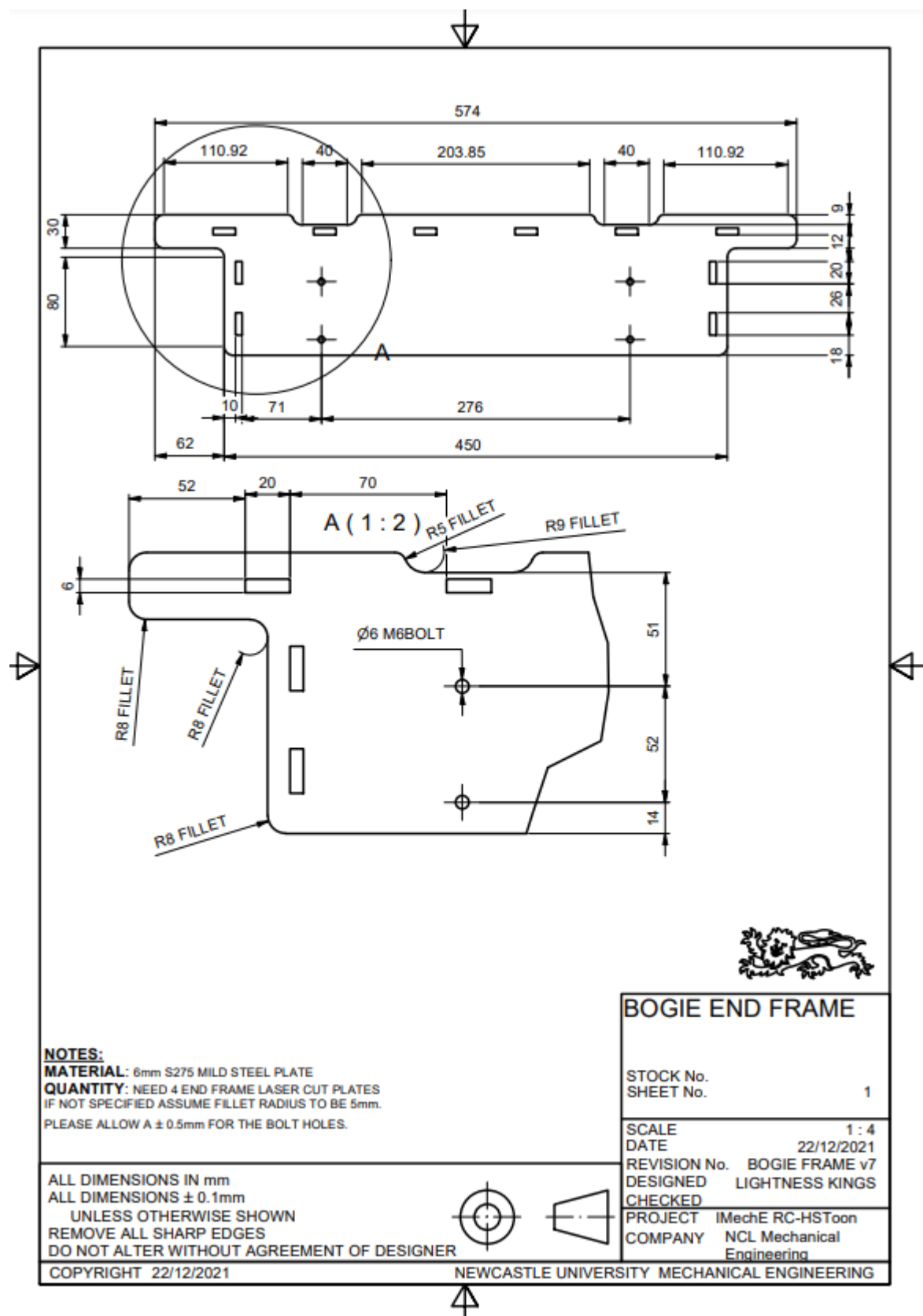
Appendix B-14: Anti-rollbar Linking plate (LK)



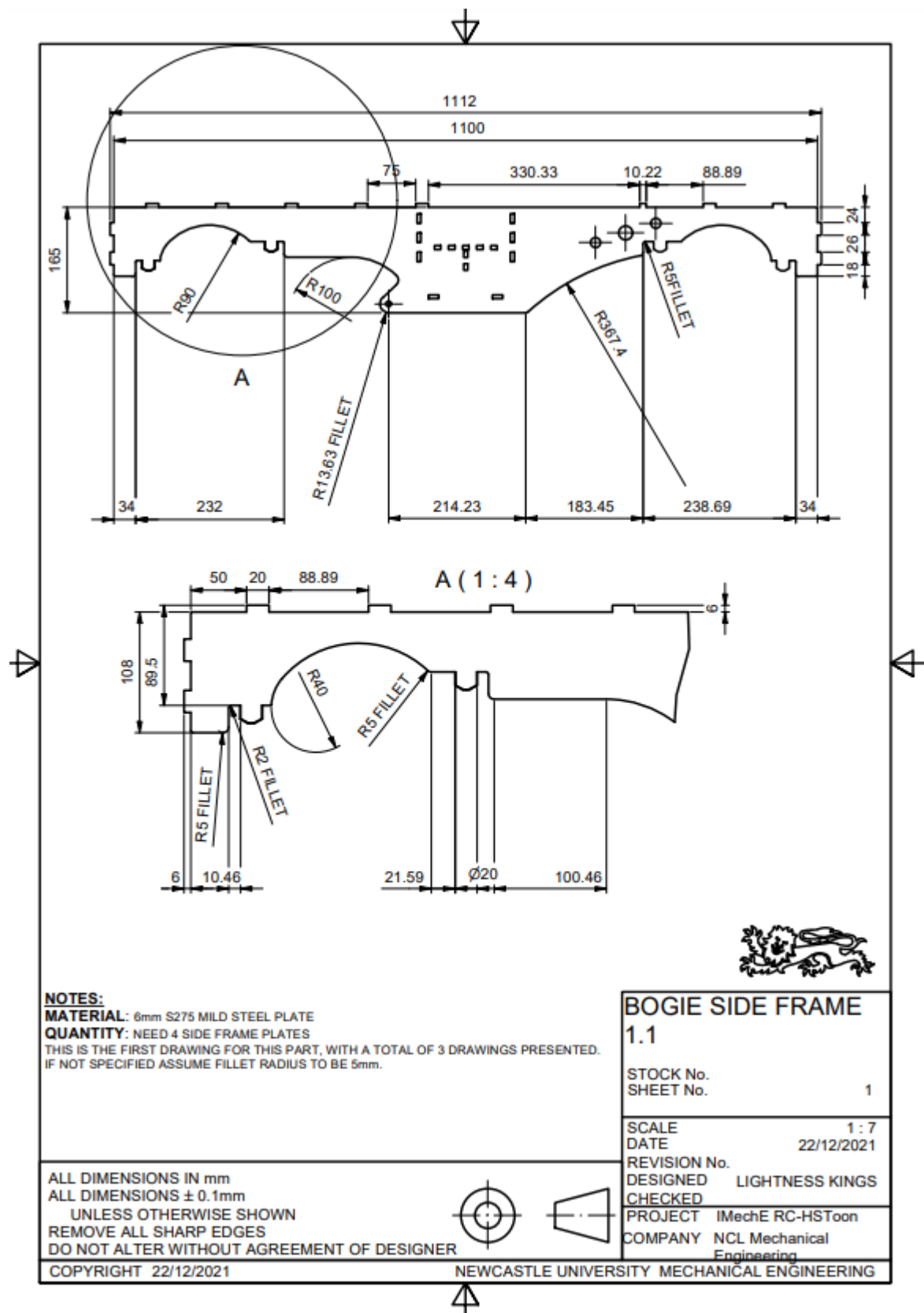
Appendix B-15: Anti-rollbar sidearm (LK)



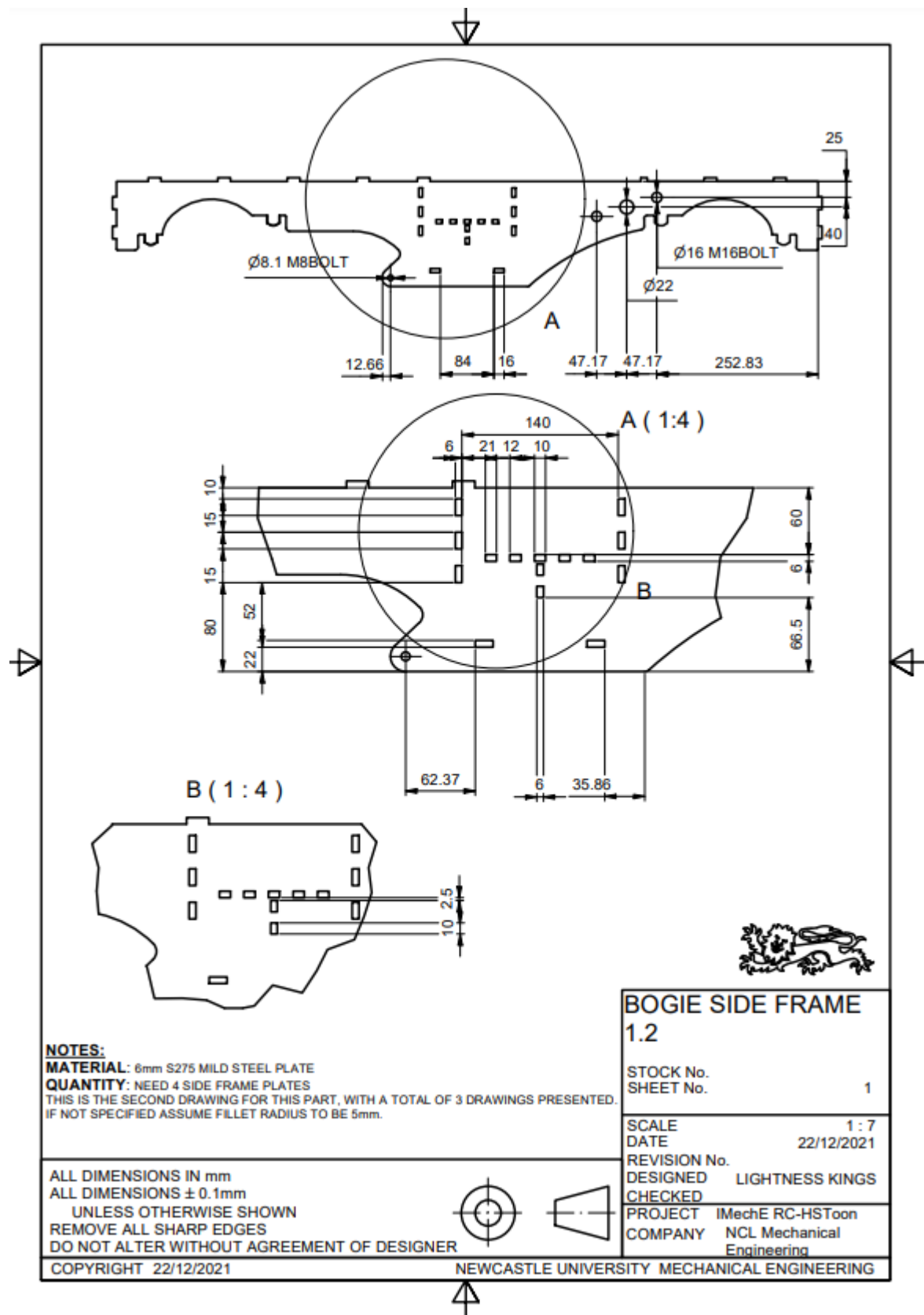
Appendix B-16: Bogie end-frame (LK)



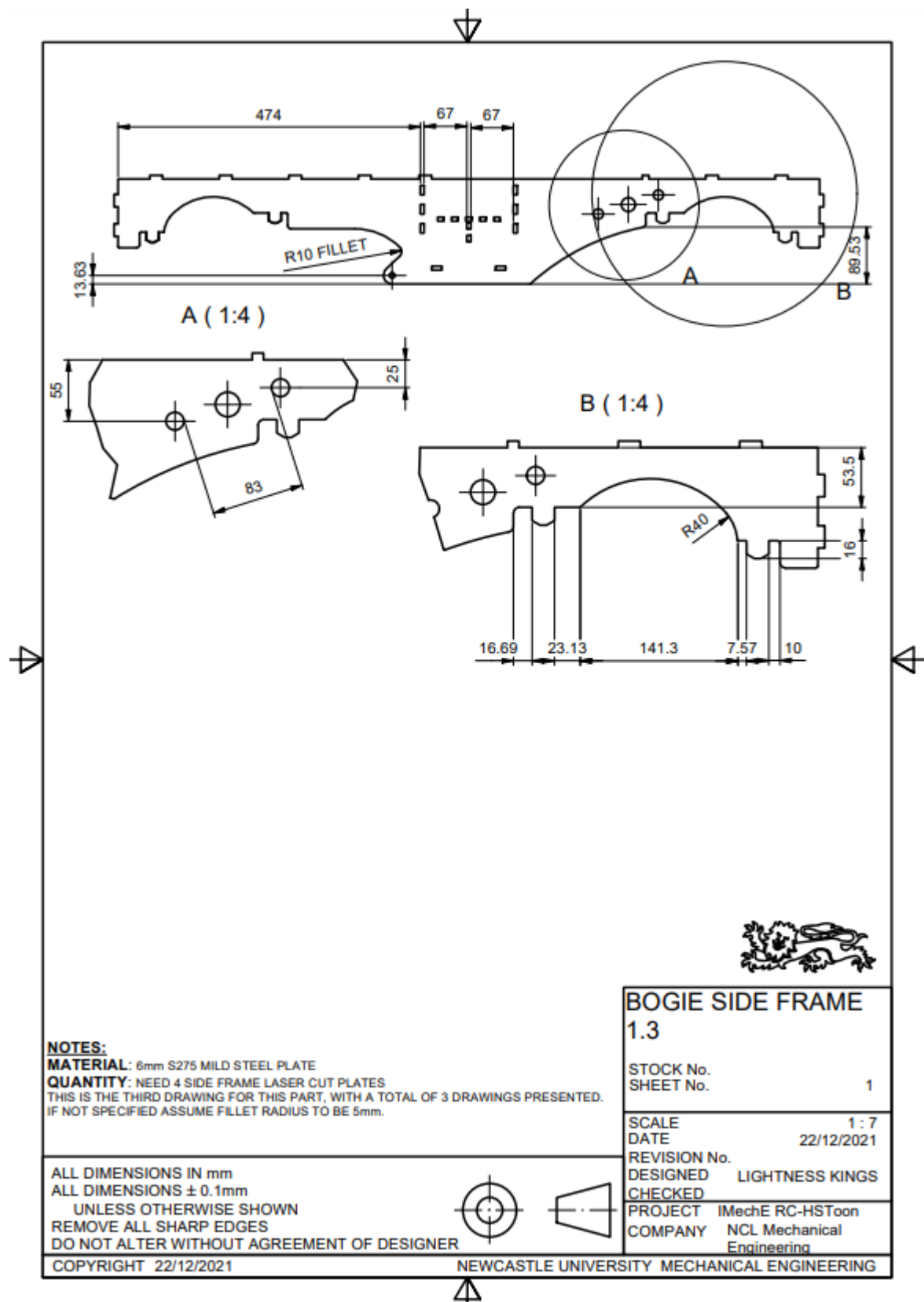
Appendix B-17: Bogie side-frame-1 (LK)



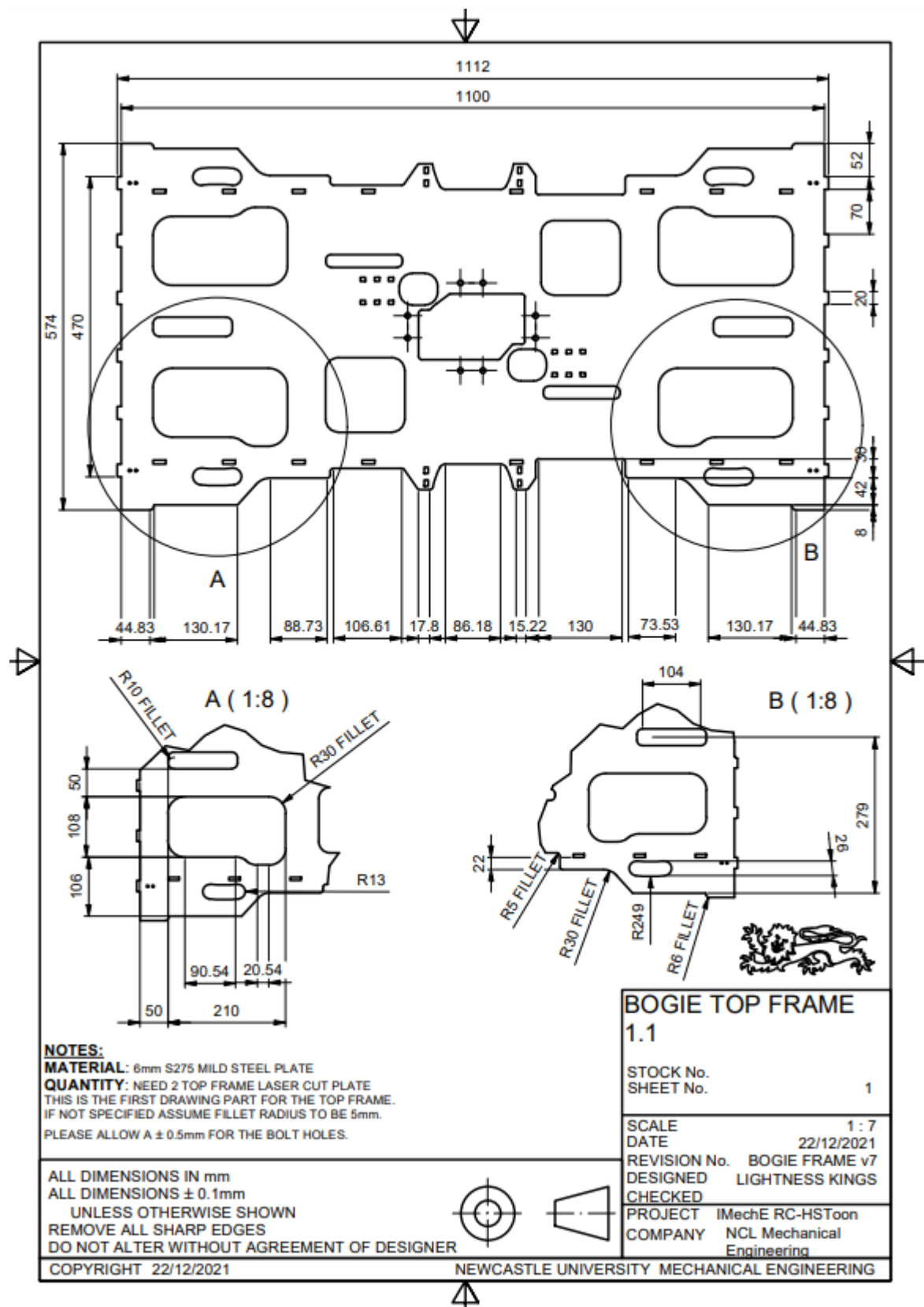
Appendix B-18: Bogie side-frame-2 (LK)



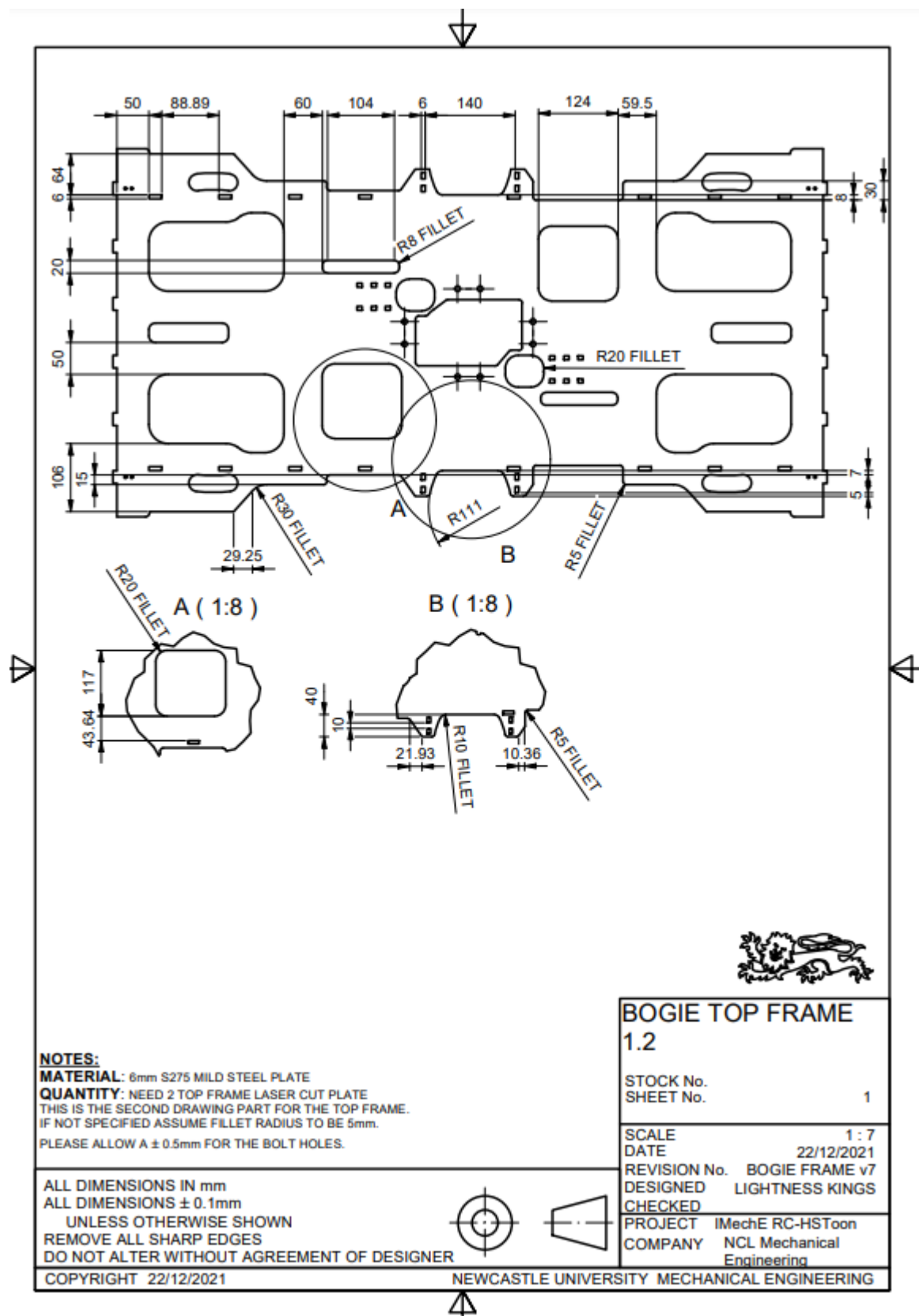
Appendix B-19: Bogie side-frame-3 (LK)



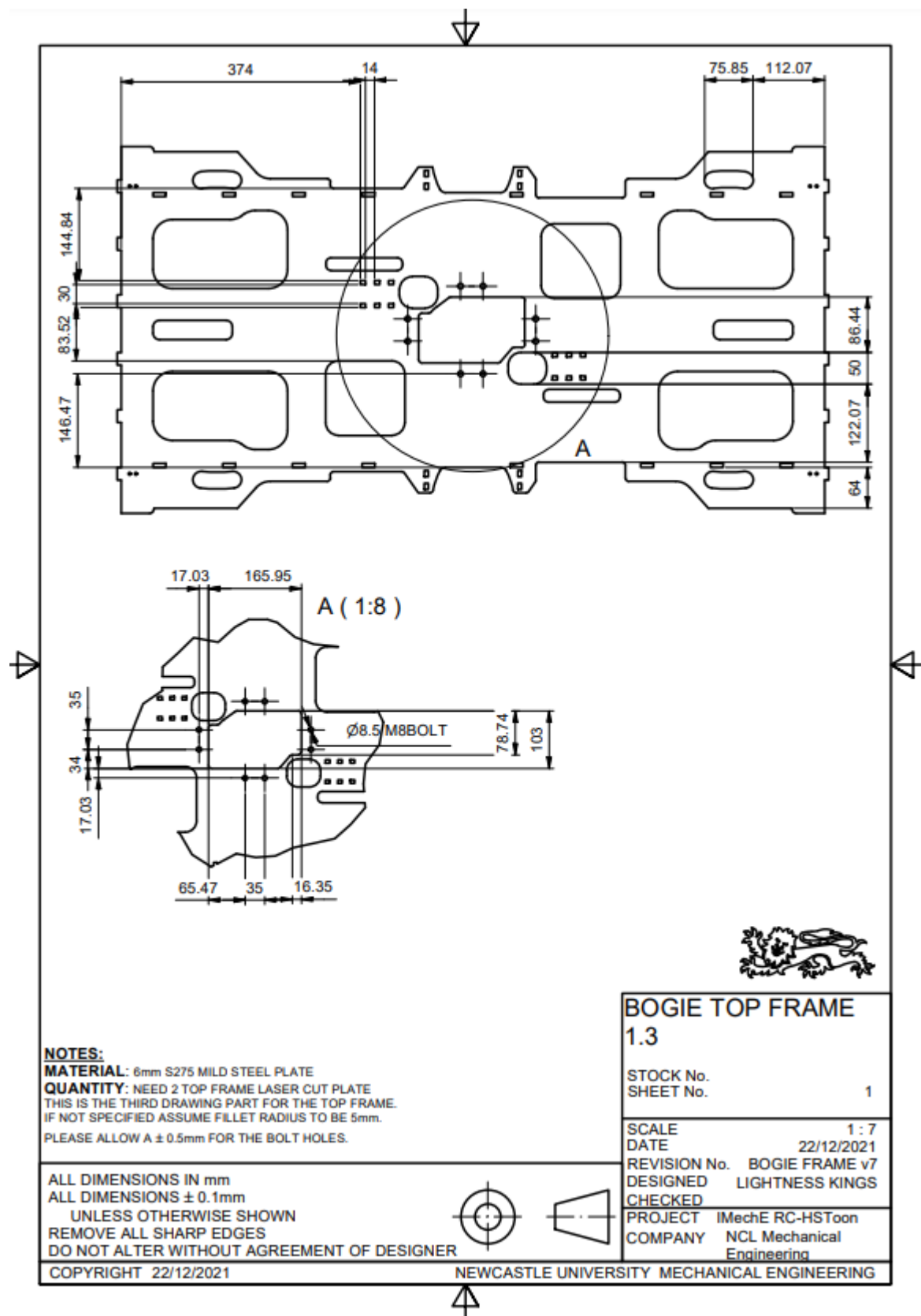
Appendix B-20: Bogie top-frame-1 (LK)



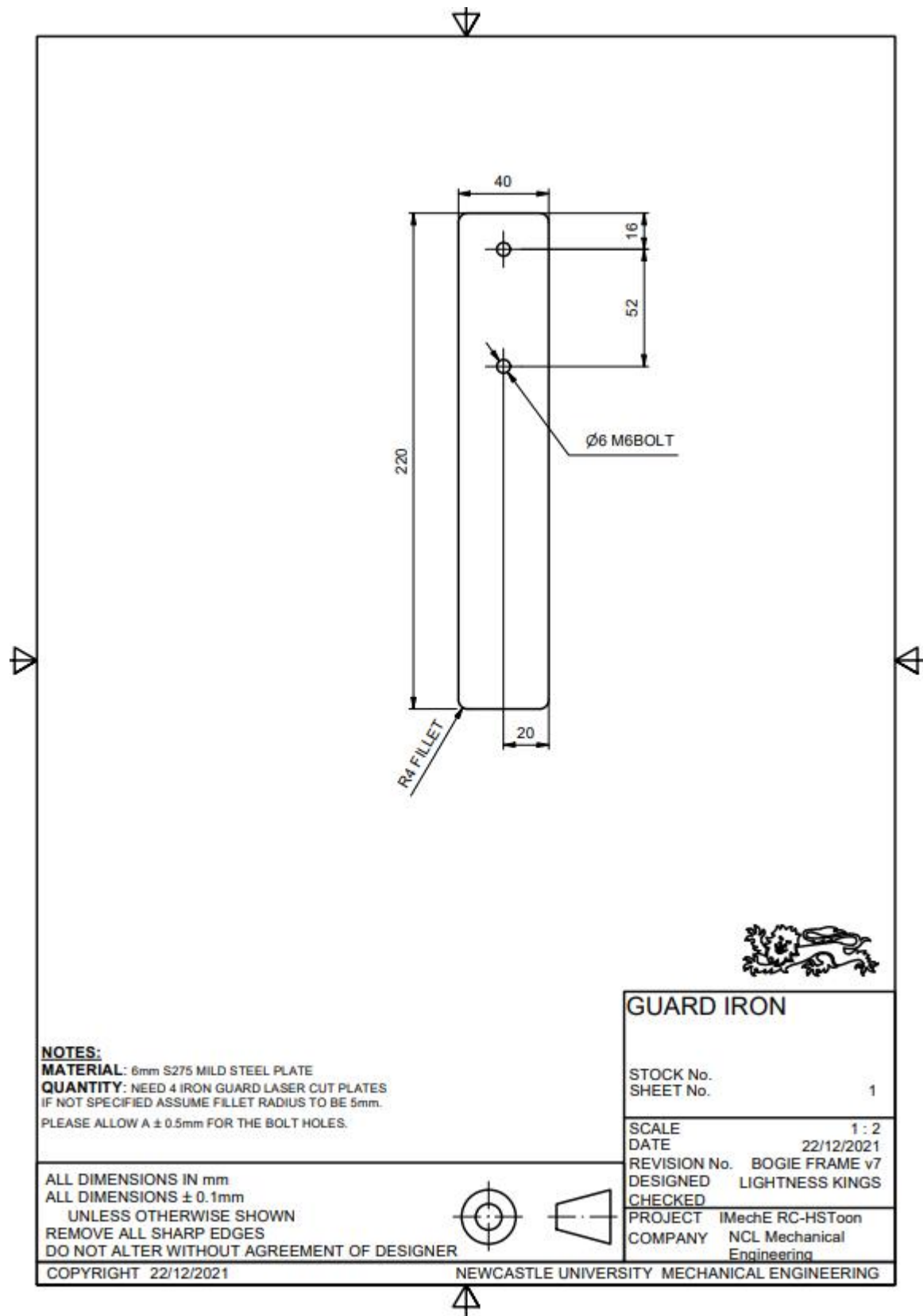
Appendix B-21: Bogie top-frame-2 (LK)



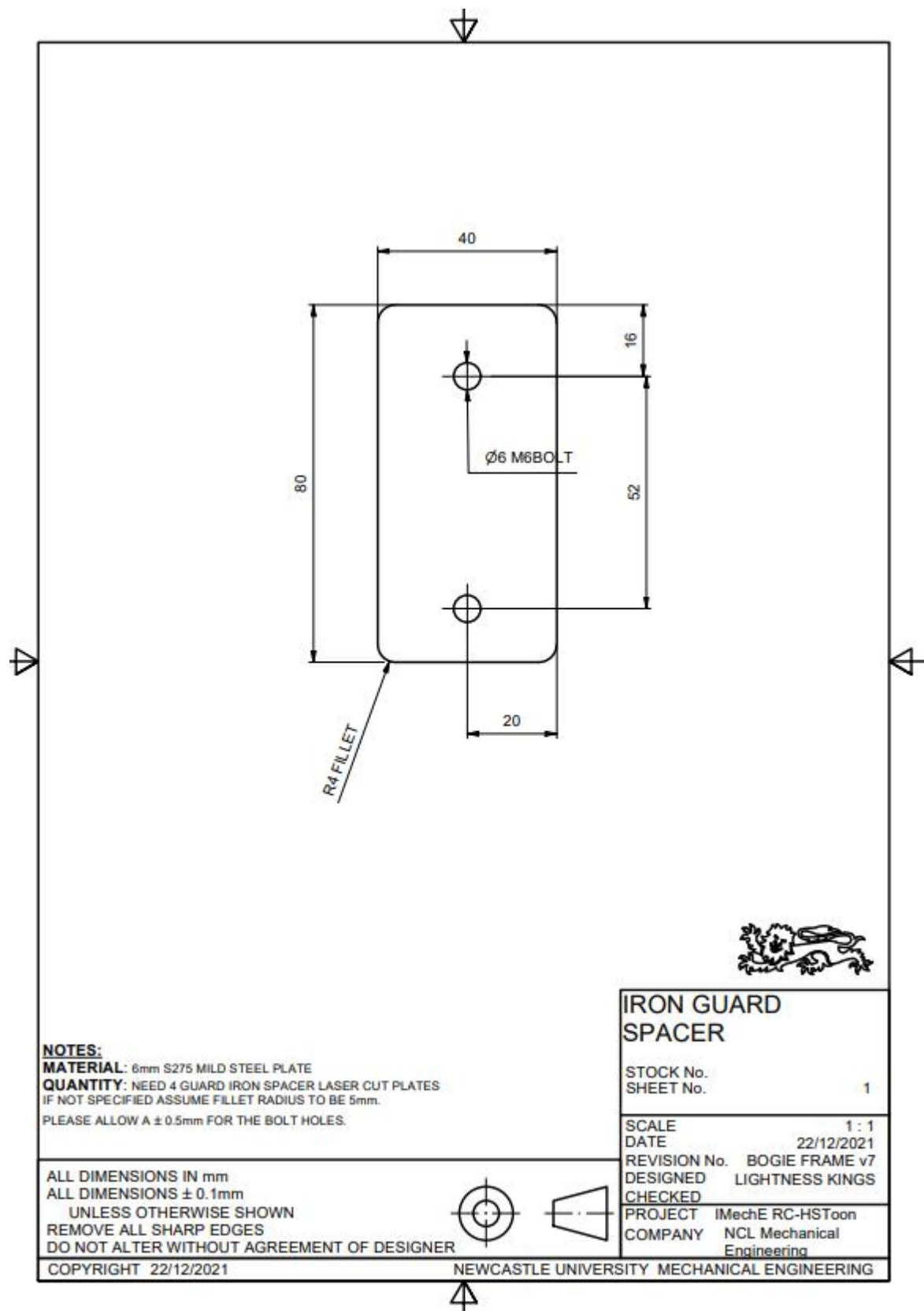
Appendix B-22: Bogie top-frame-3 (LK)



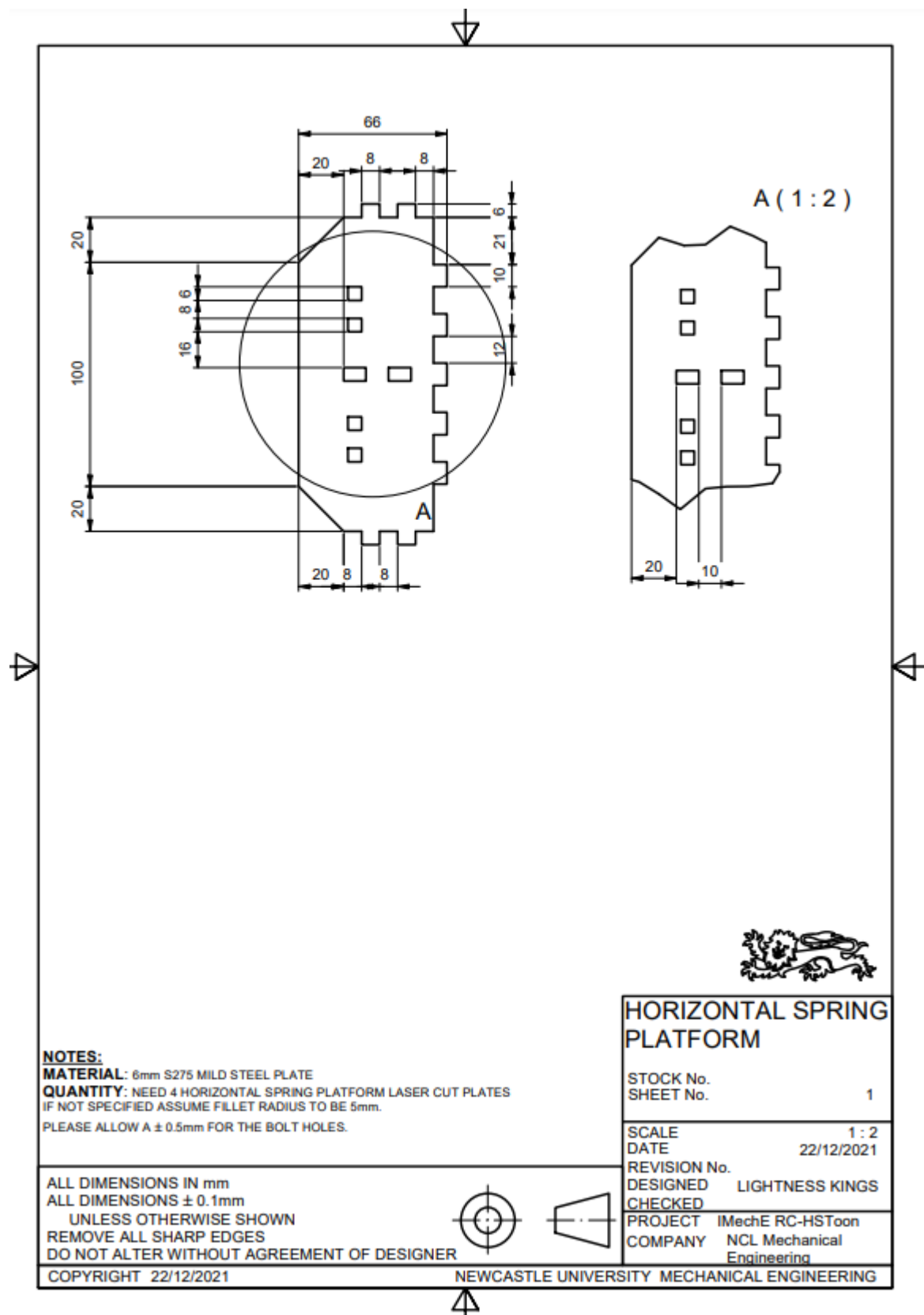
Appendix B-23: Bogie guard iron (LK)



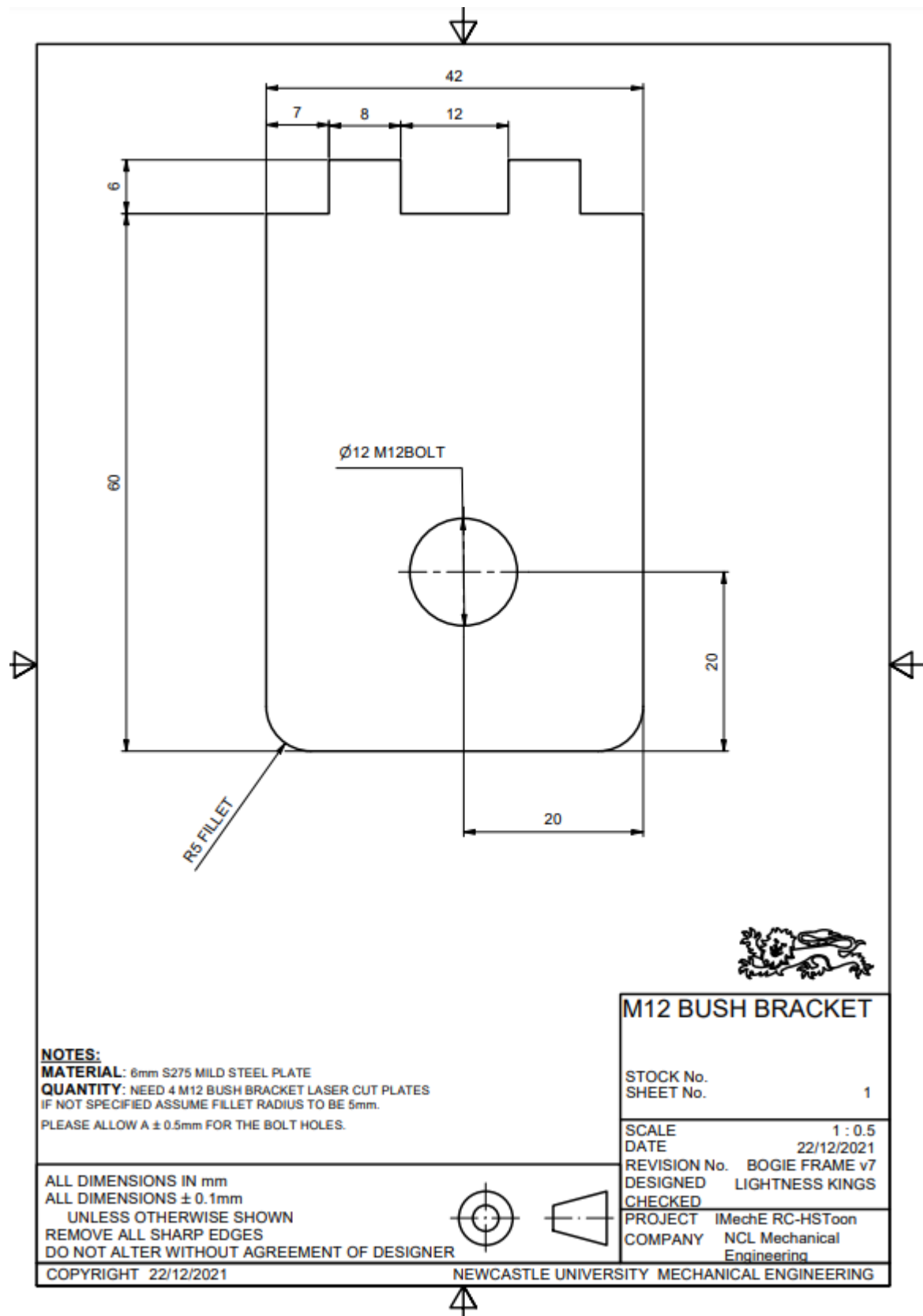
Appendix B-24: Bogie iron guard-spacer (LK)



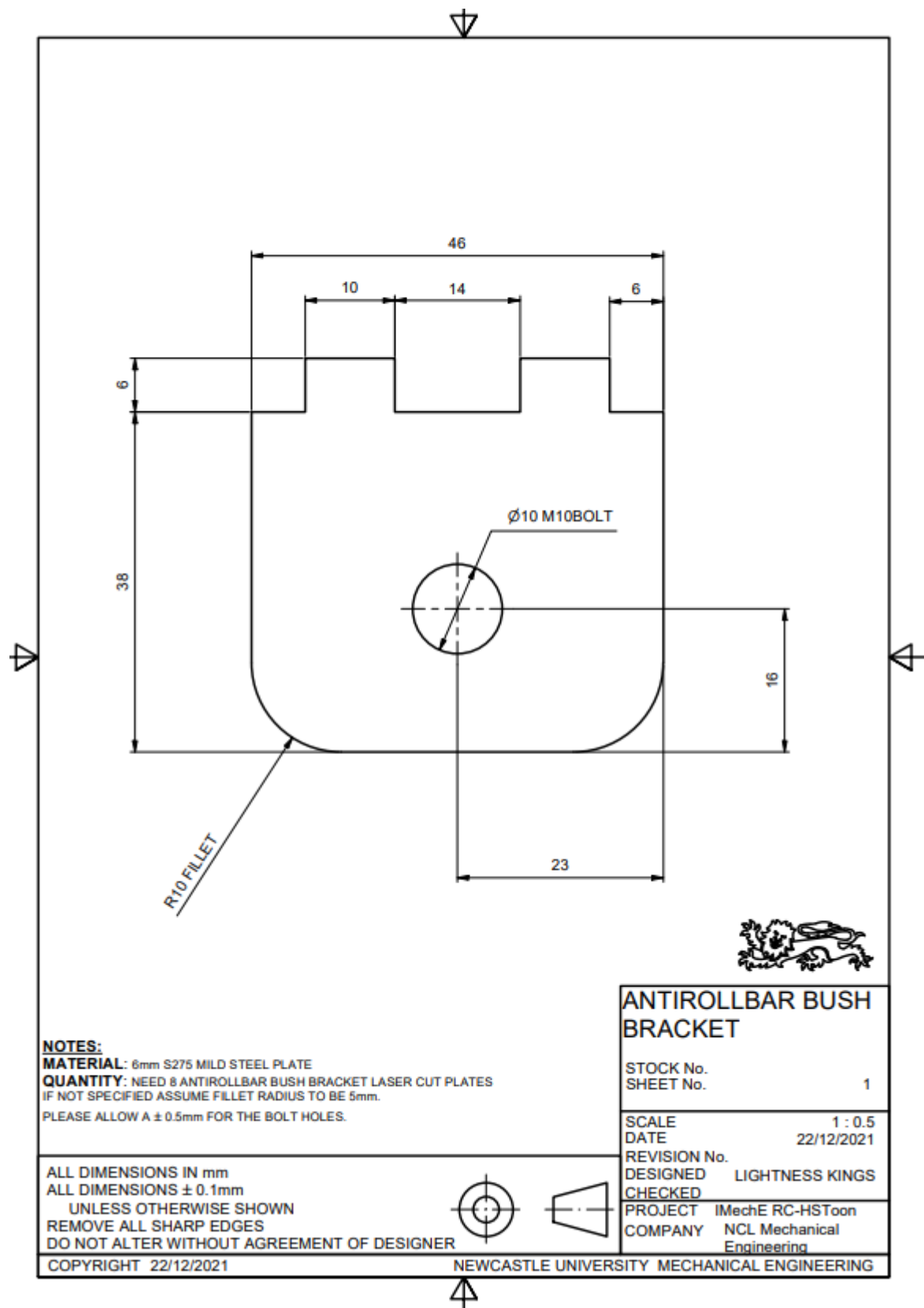
Appendix B-25: Bogie horizontal-spring platform (LK)



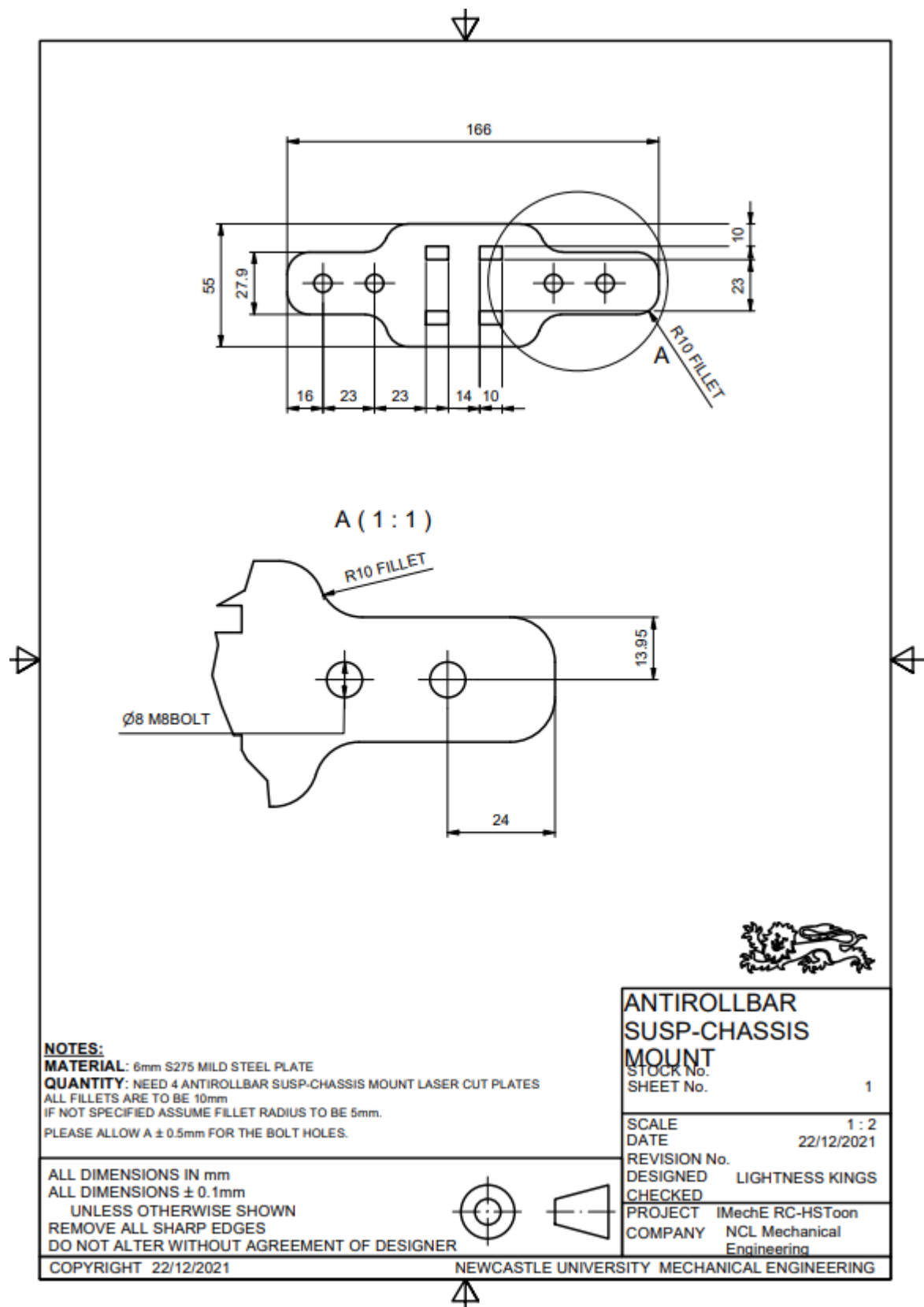
Appendix B-26: Bogie M12-bush bracket (LK)



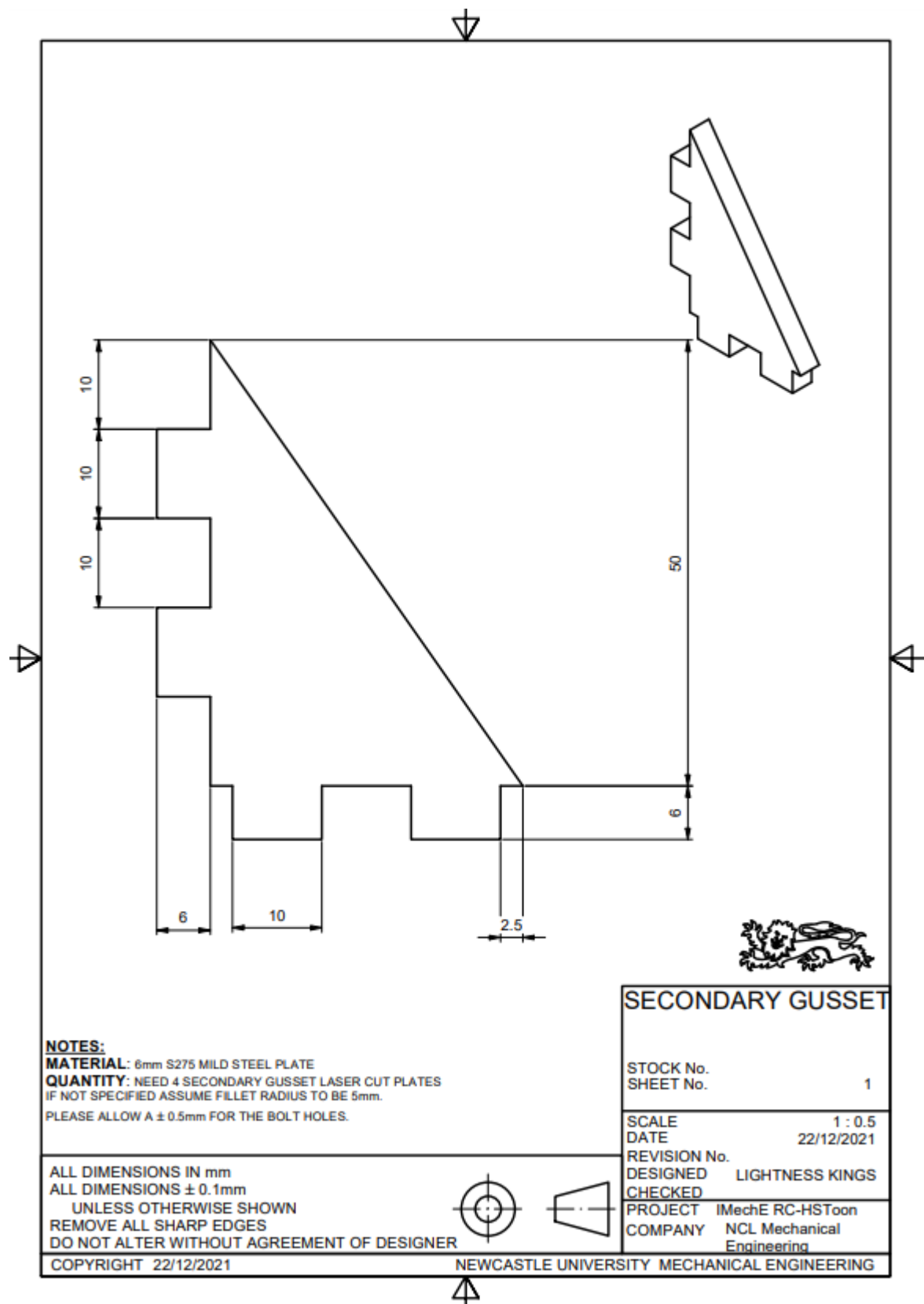
Appendix B-27: Anti-rollbar bush bracket (LK)



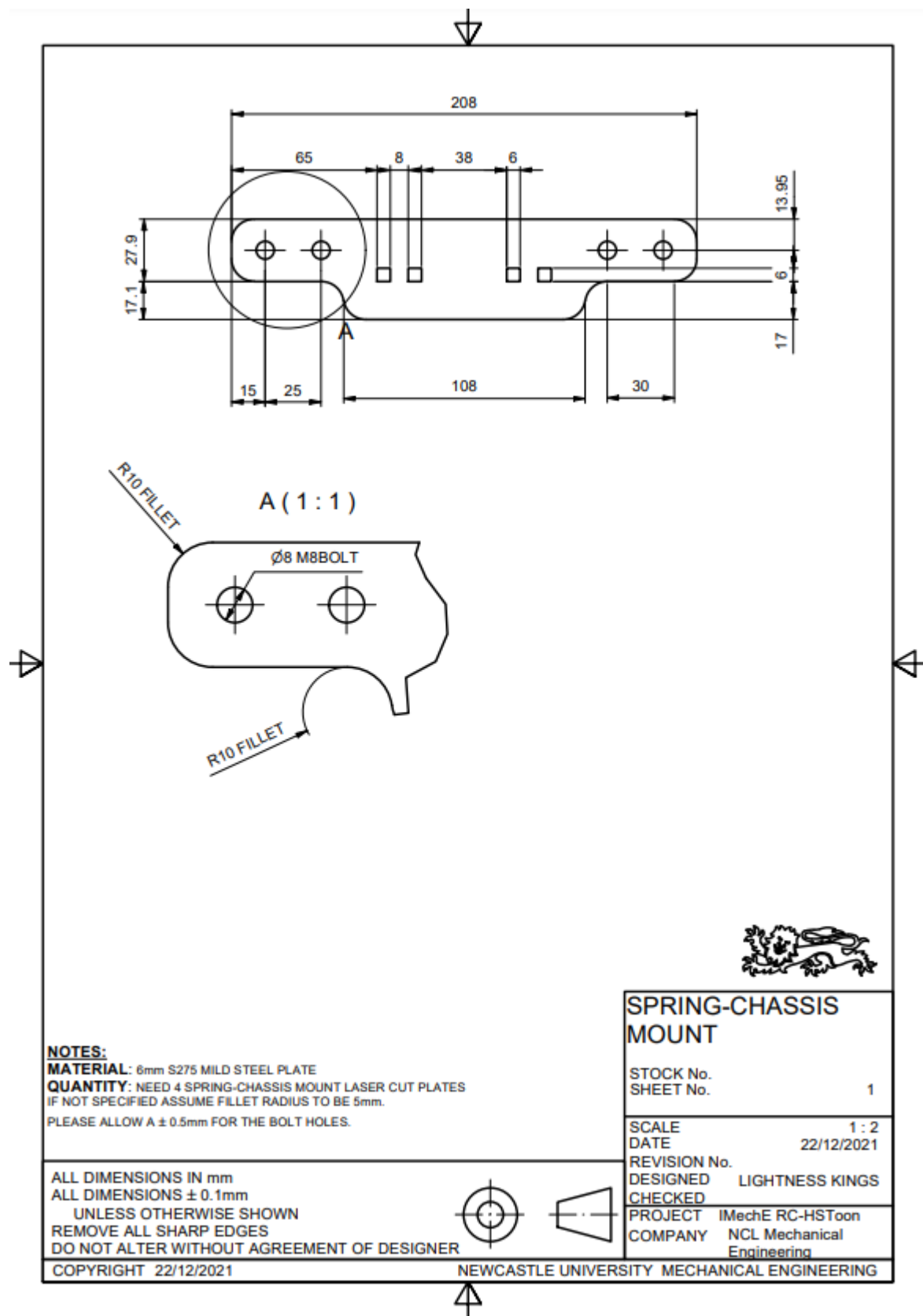
Appendix B-28: Anti-rollbar-to-chassis connection plate (LK)



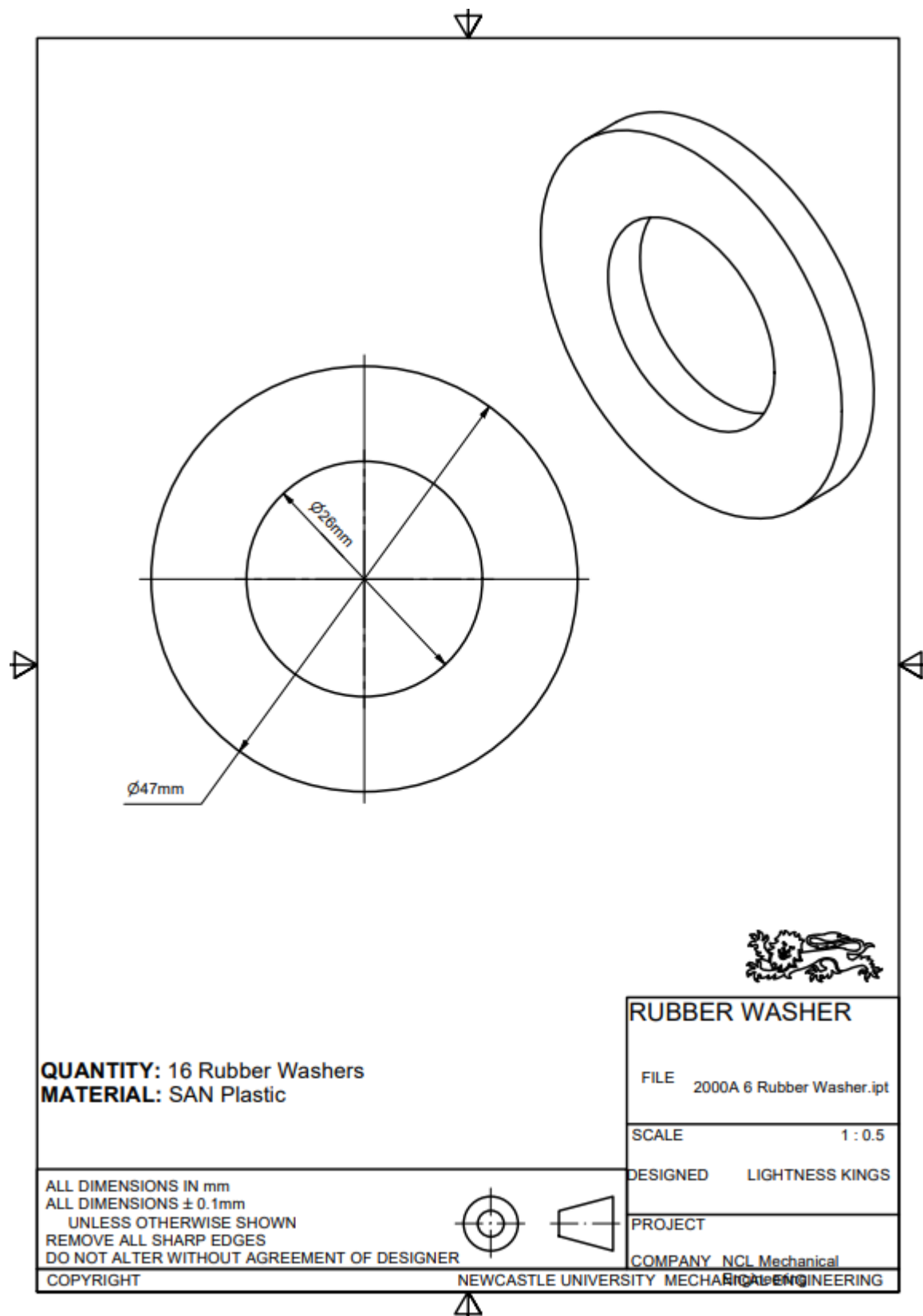
Appendix B-29: Bogie secondary springs gusset support (LK)



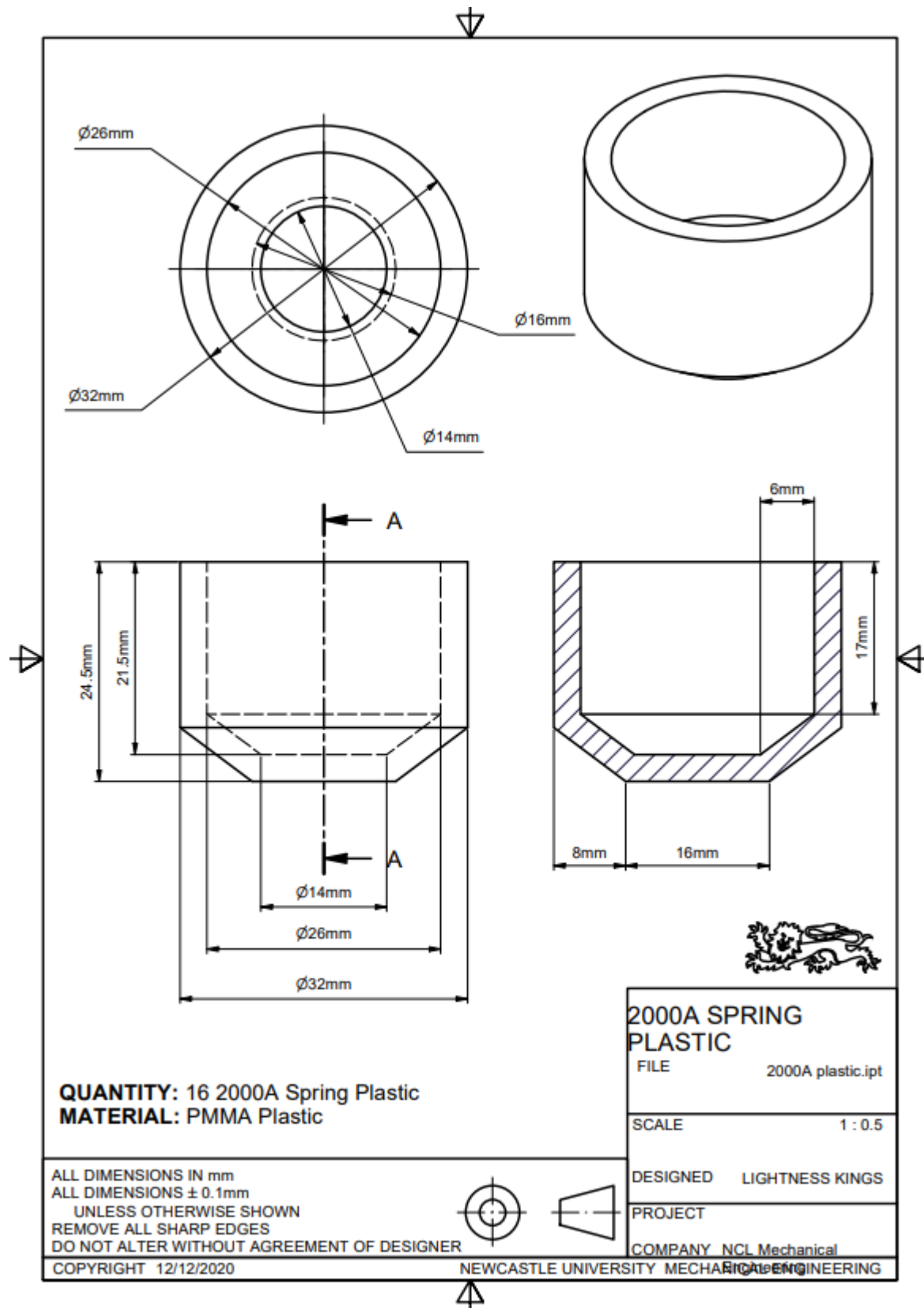
Appendix B-30: Secondary springs-to-chassis mount (LK)



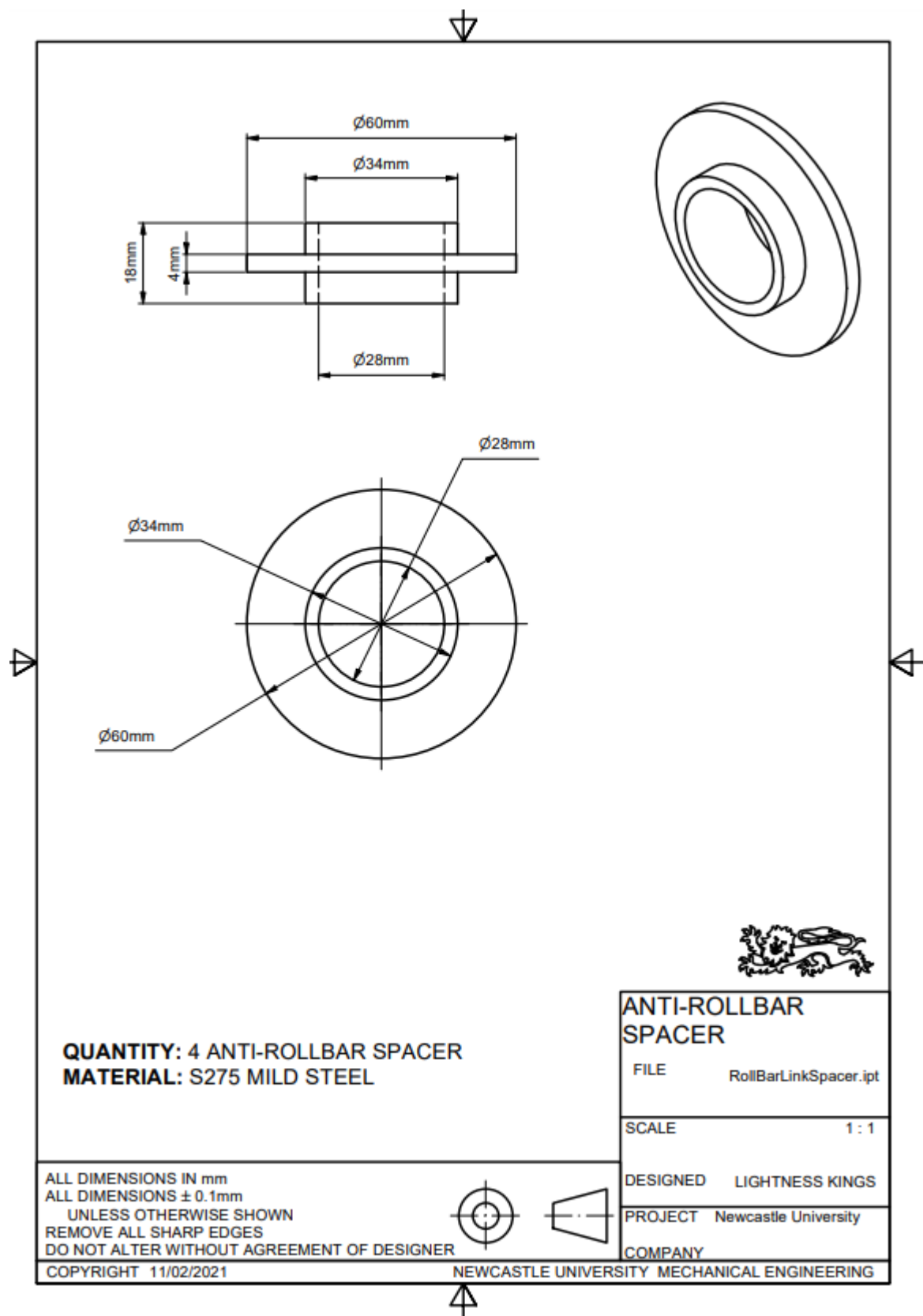
Appendix B-31: Secondary suspension rubber washers (LK)



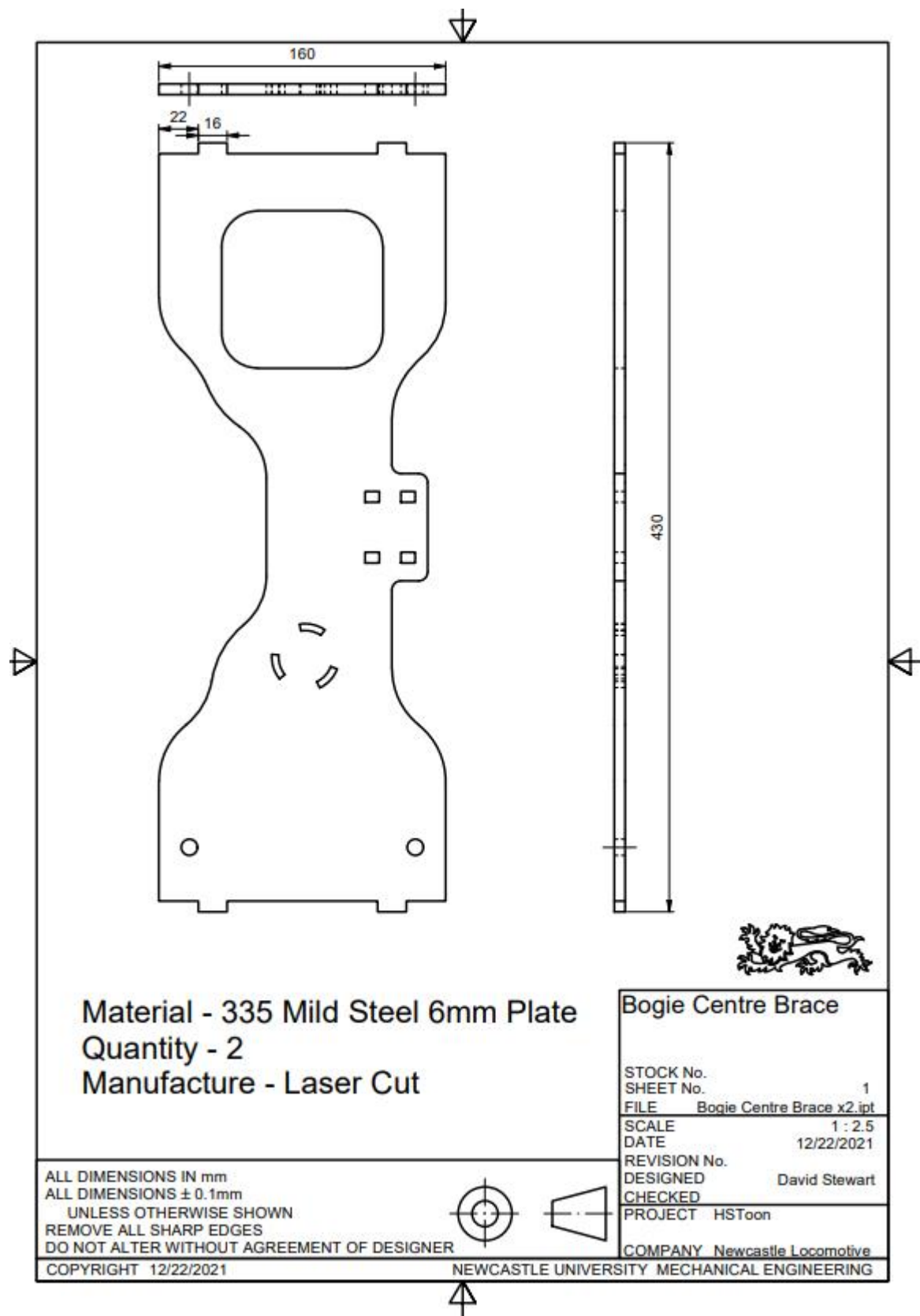
Appendix B-32: Secondary suspension spring inner jackets (LK)



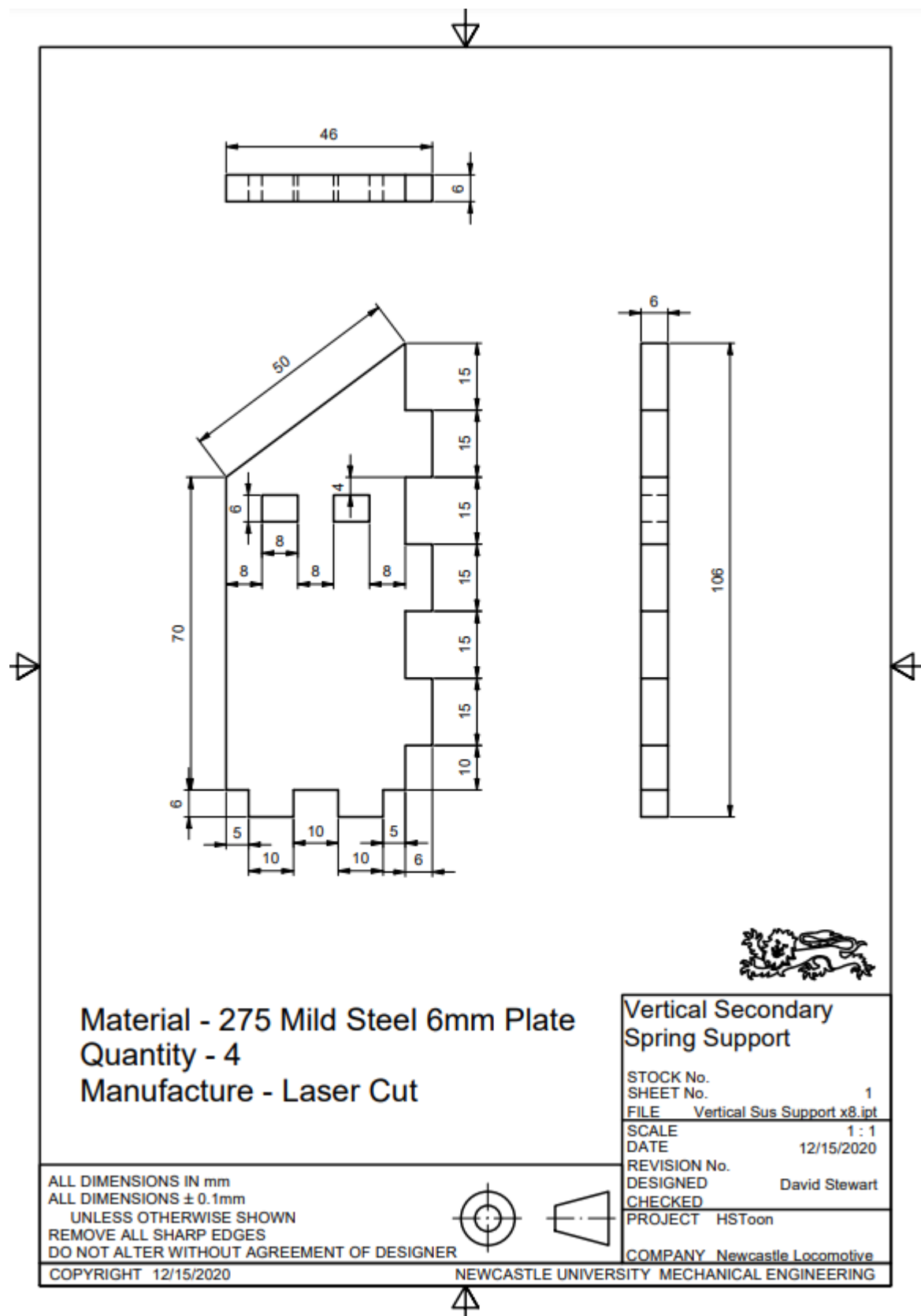
Appendix B-33: Anti-rollbar spacer (LK)



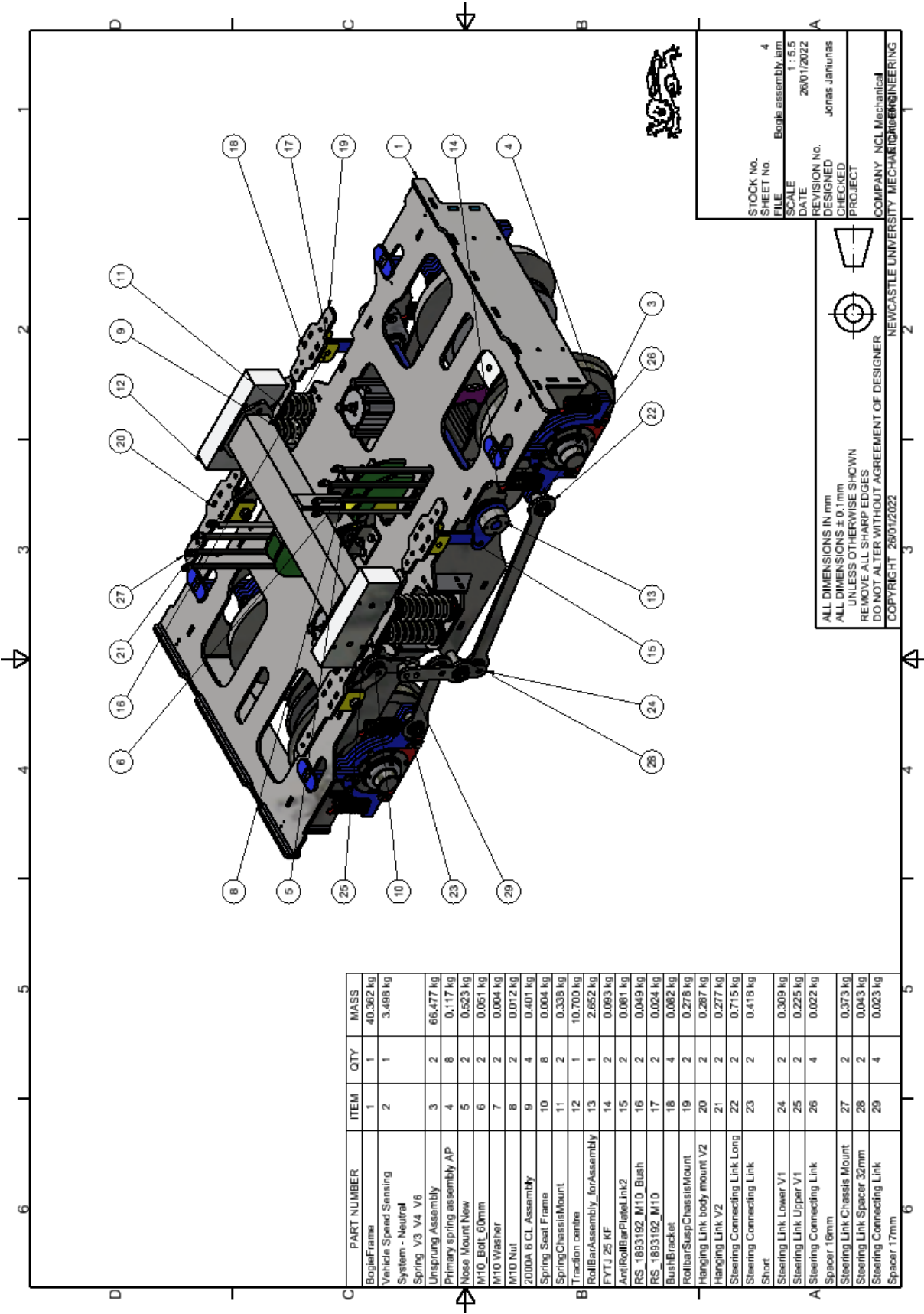
Appendix B-34: Bogie centre-brace (DS)



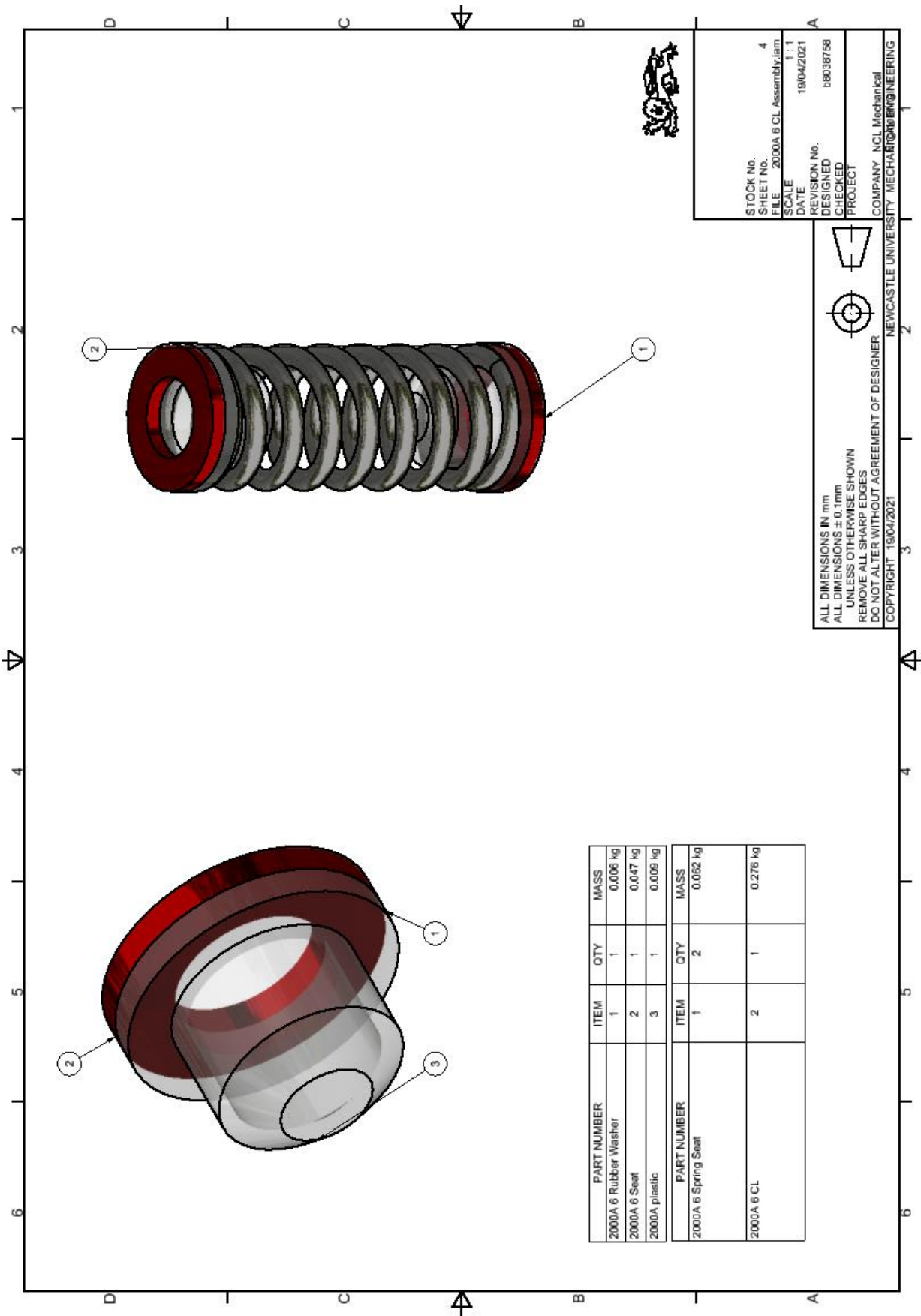
Appendix B-35: Vertical secondary-spring platform (DS)



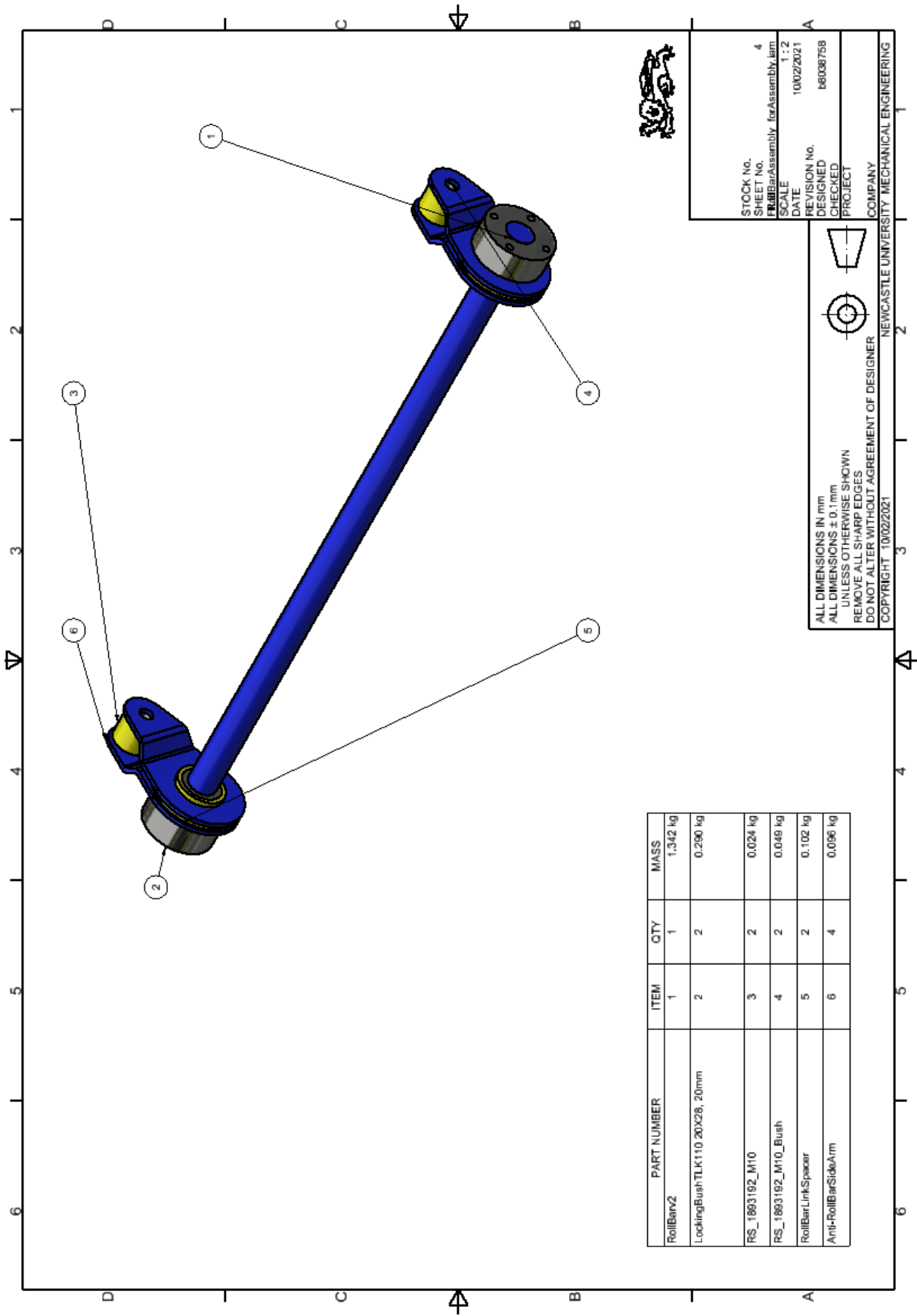
Appendix B-36: Bogie system assembly (DS, LK, JJ)



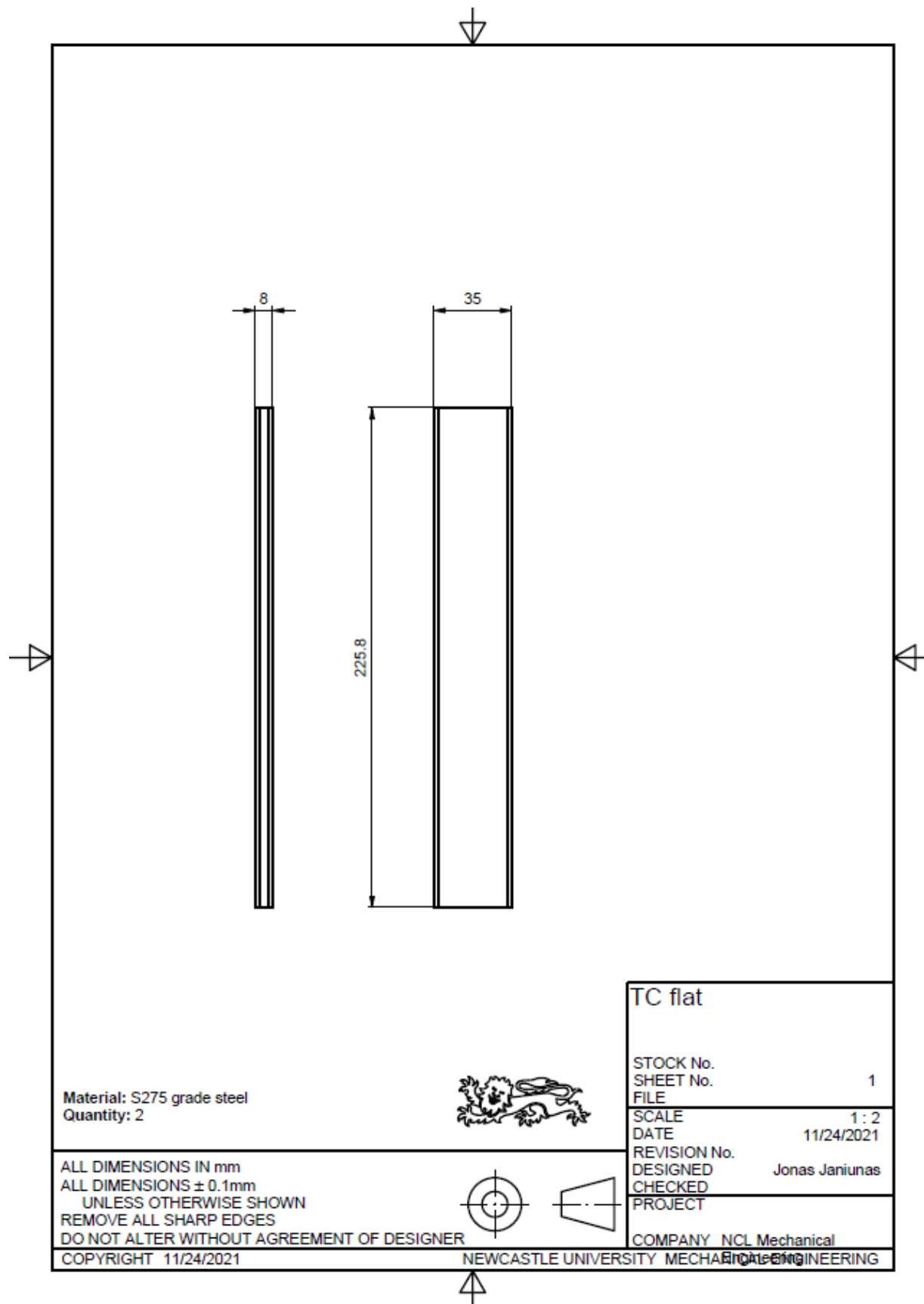
Appendix B-37: Secondary springs assembly (LK)



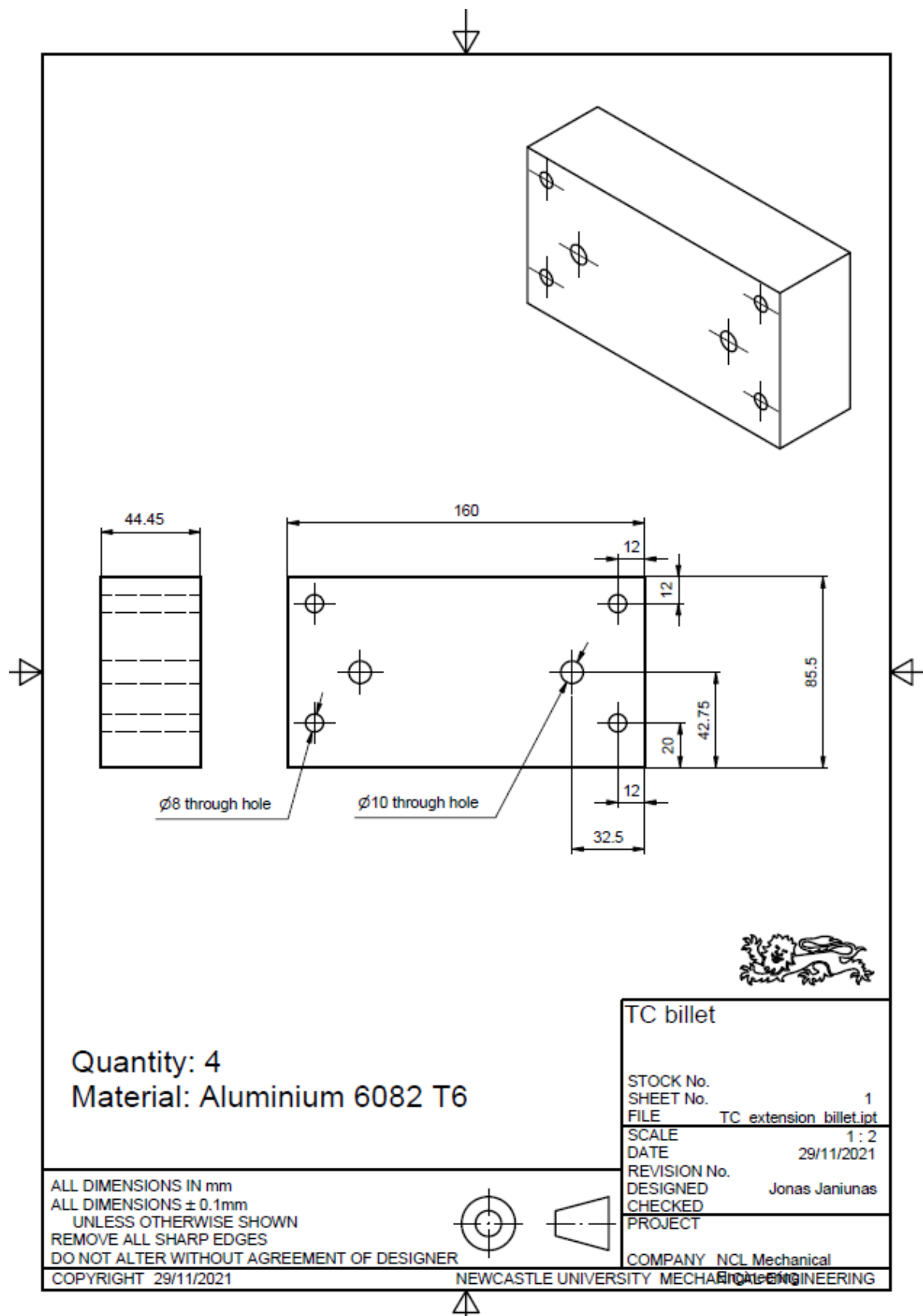
Appendix B-38: Anti-rollbar assembly (LK)



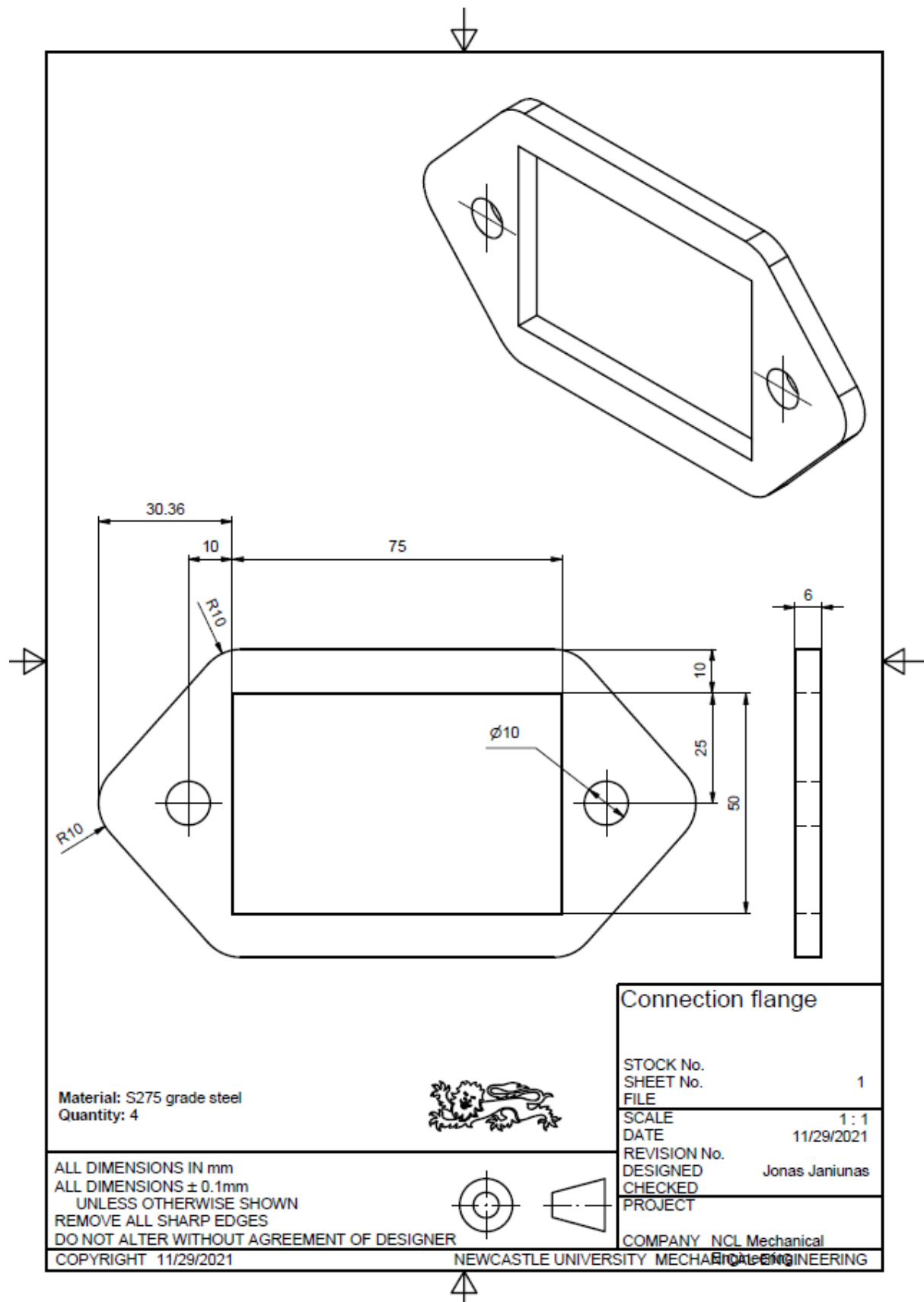
Appendix B-39: (JJ)



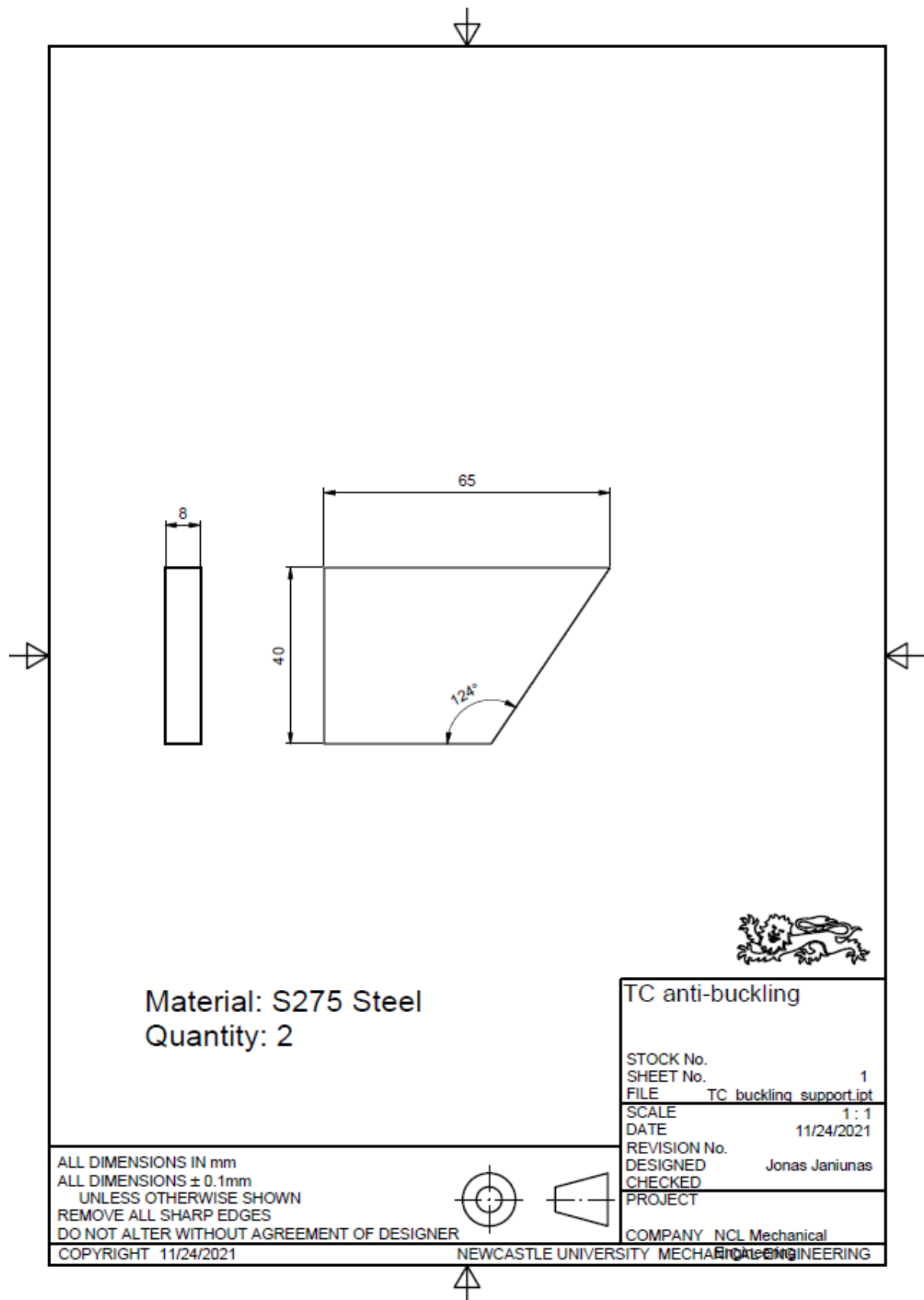
Appendix B-40: (JJ)



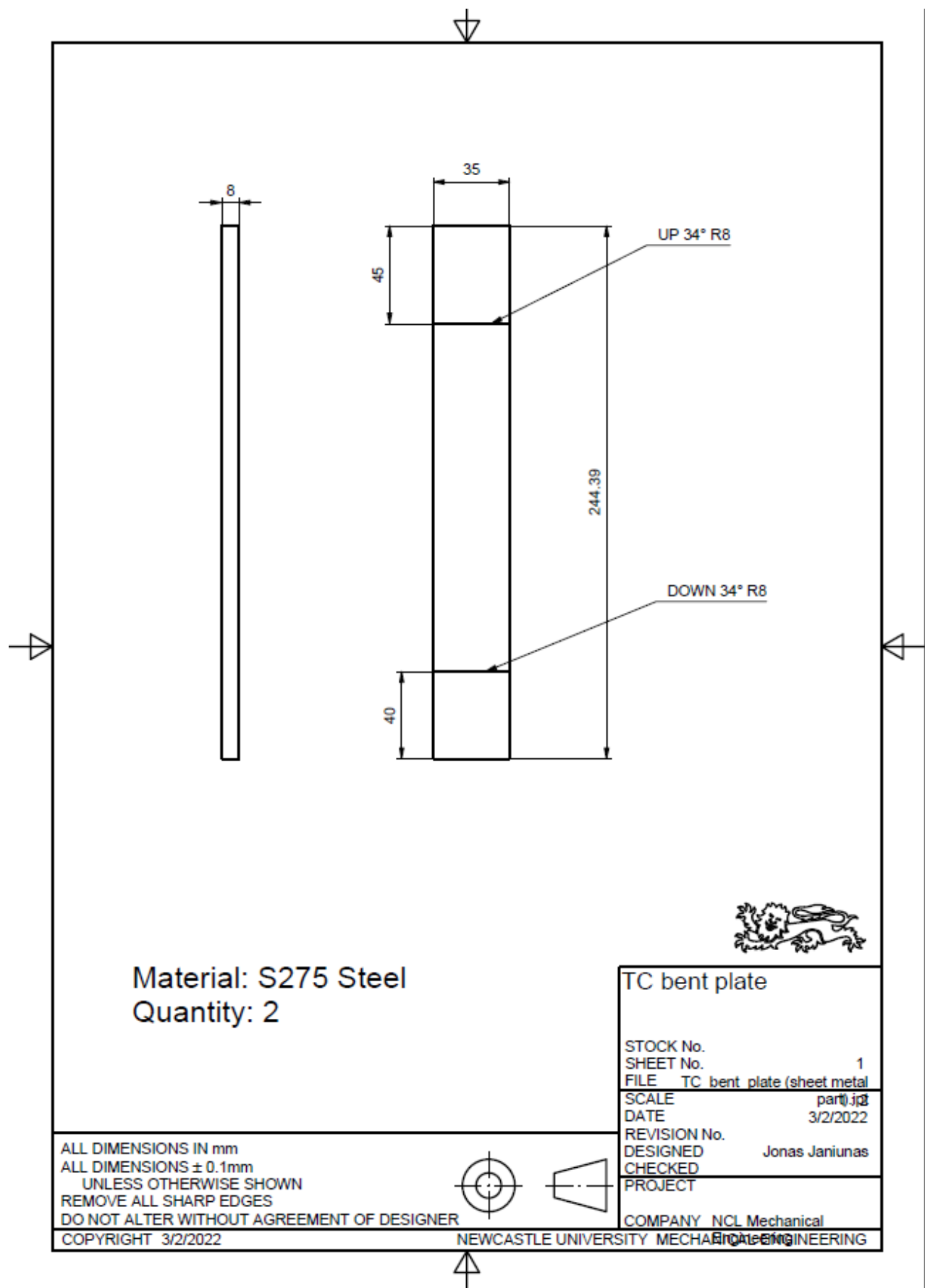
Appendix B-41: (JJ)



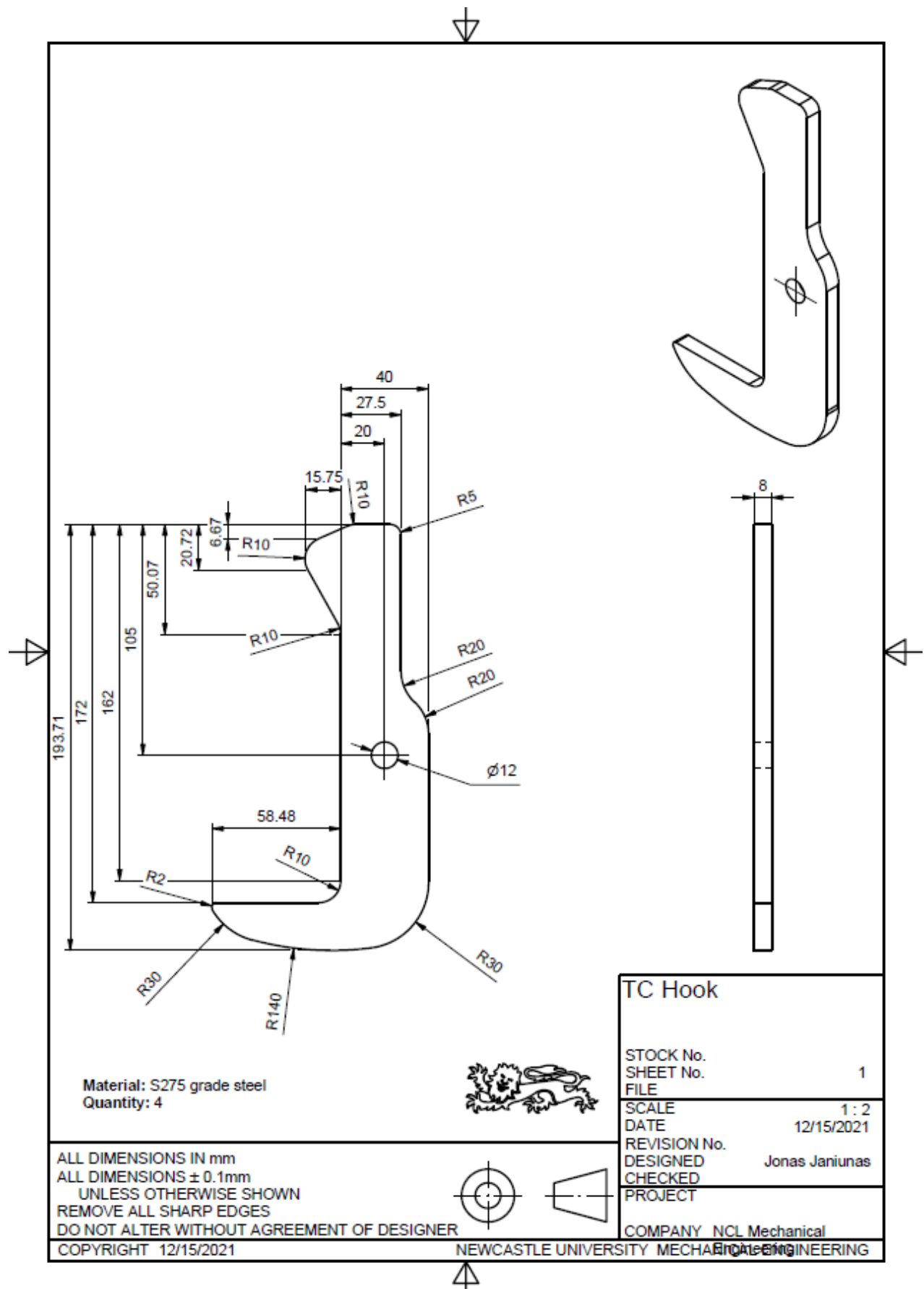
Appendix B-42: (JJ)



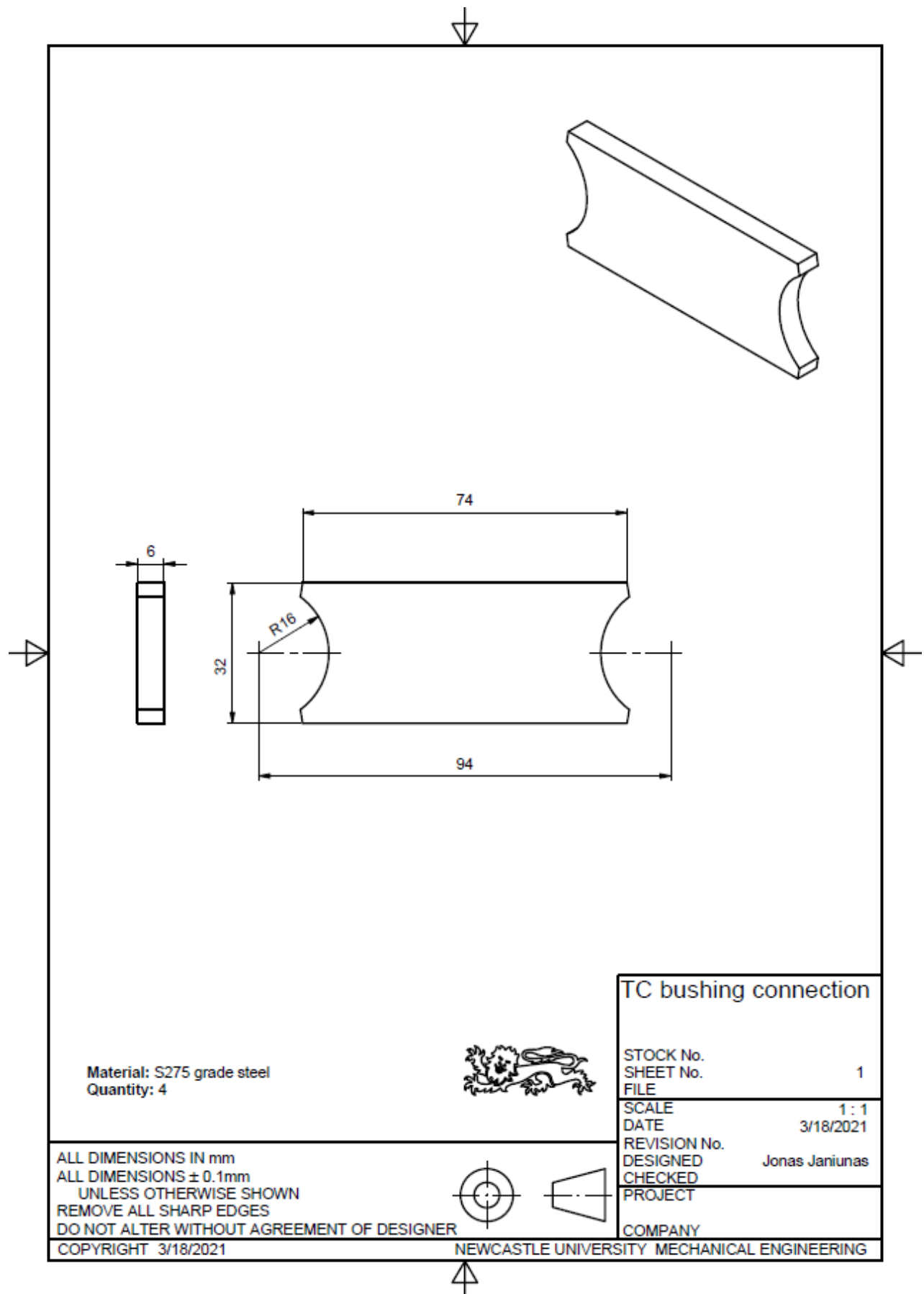
Appendix B-43: (JJ)



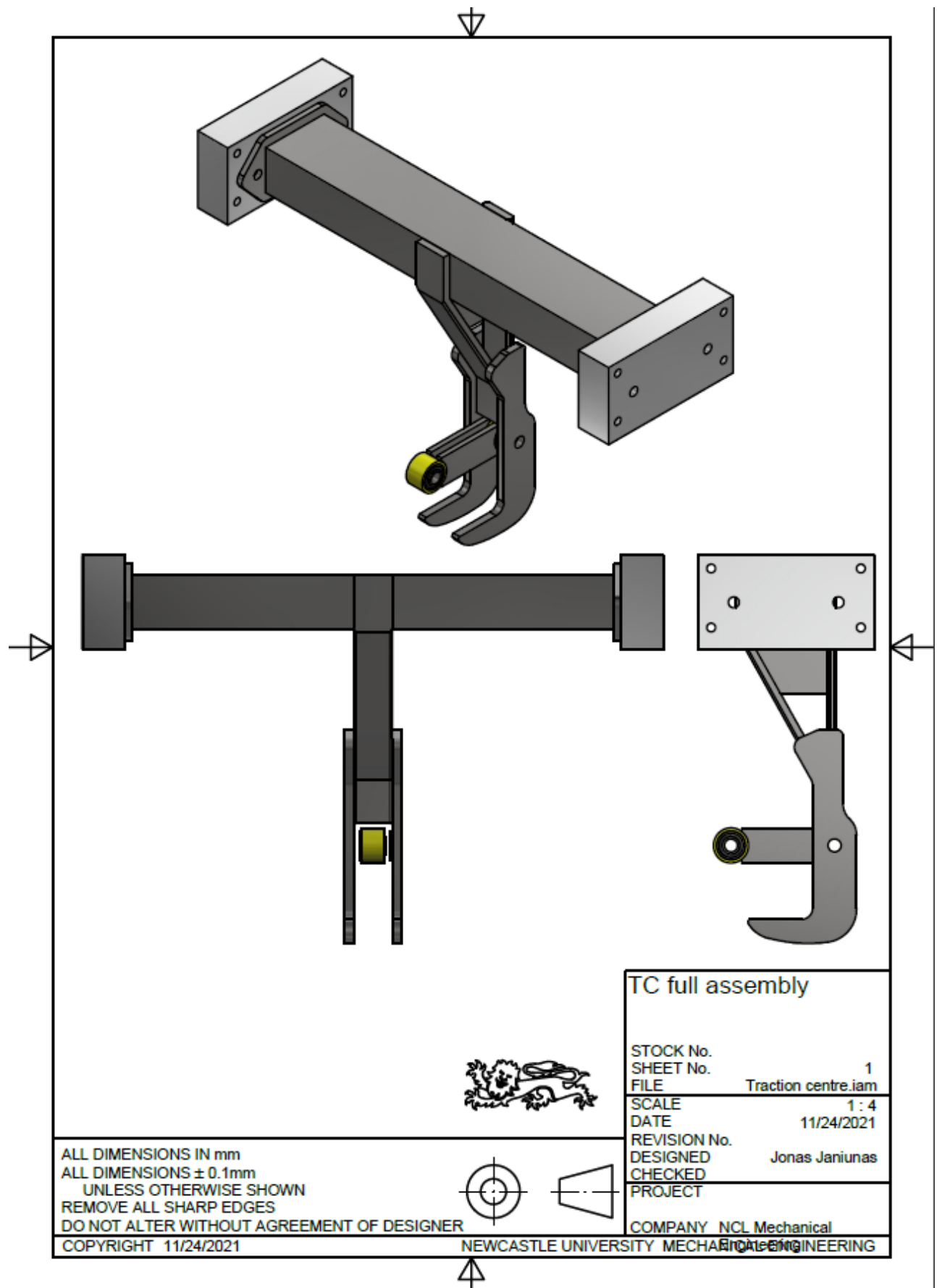
Appendix B-44: (JJ)



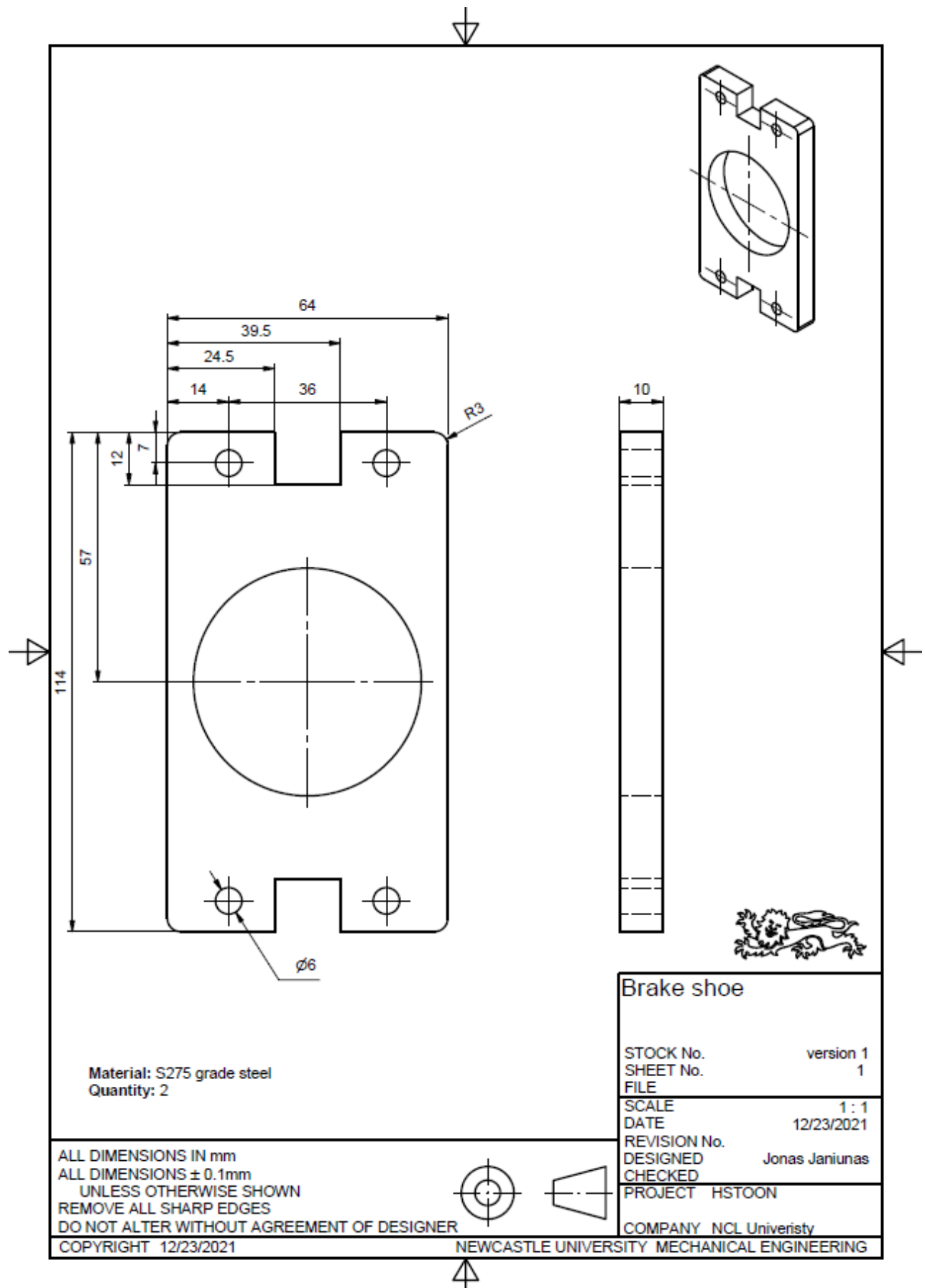
Appendix B-45: (JJ)



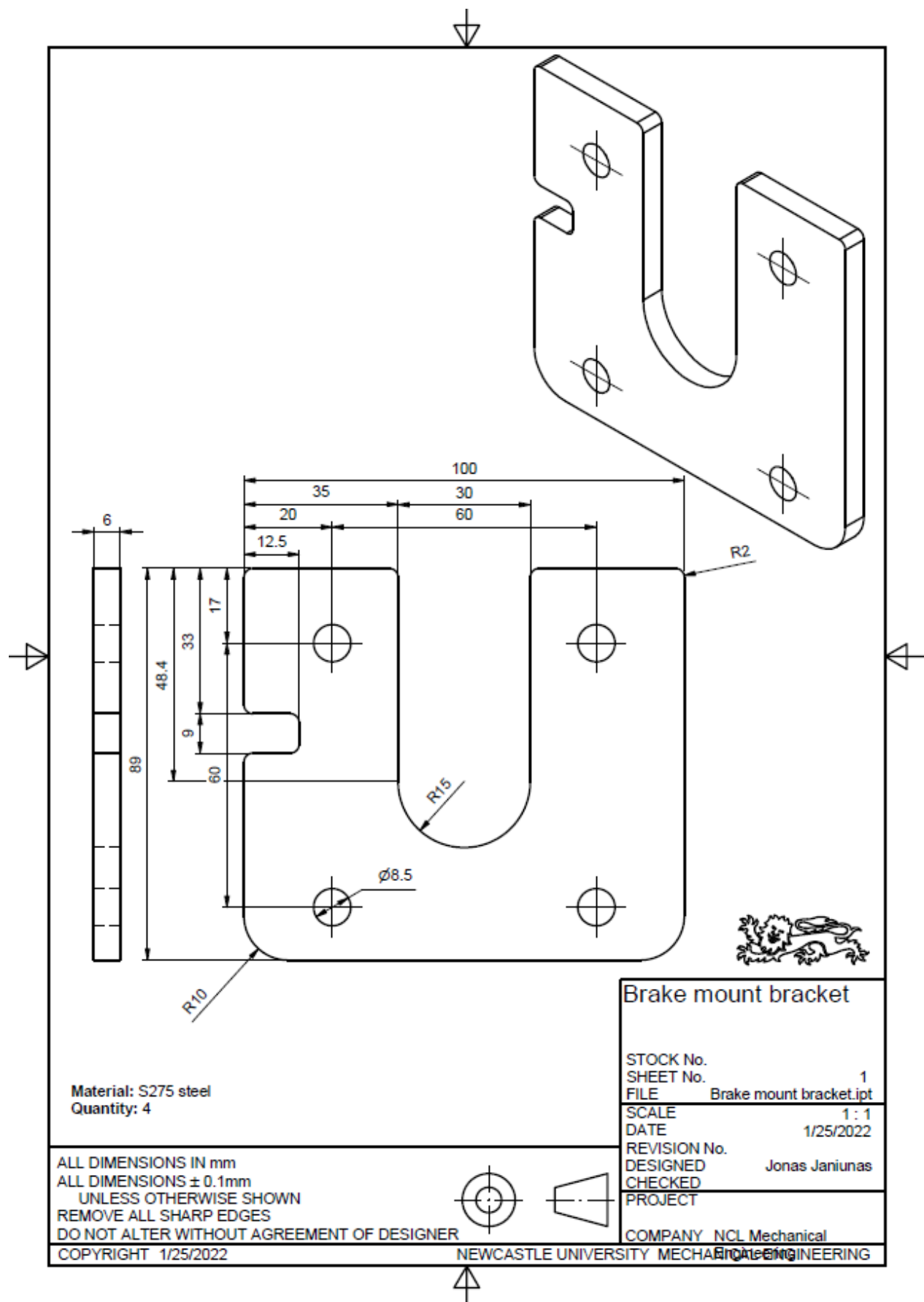
Appendix B-46: (JJ)



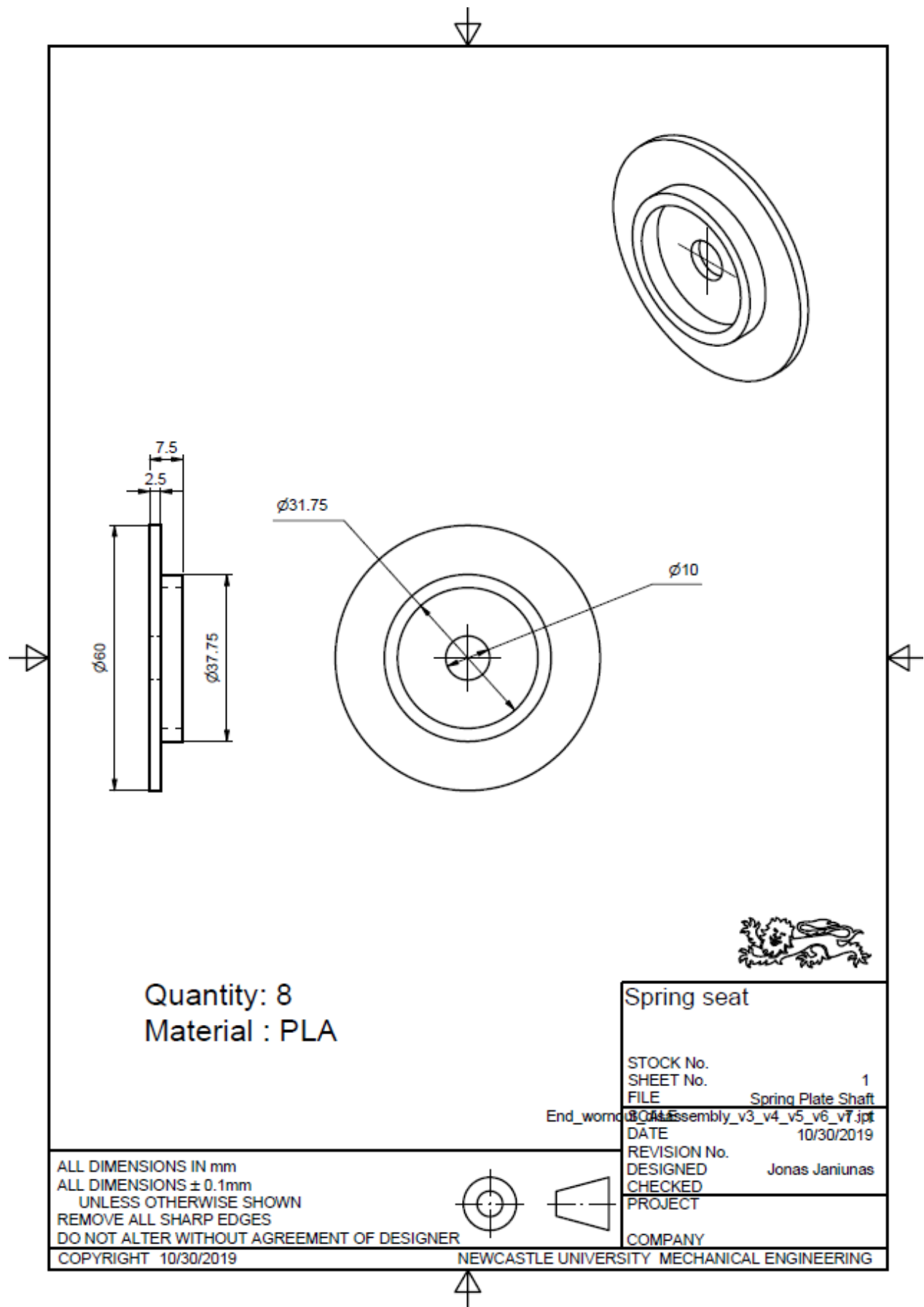
Appendix B-47: (JJ)



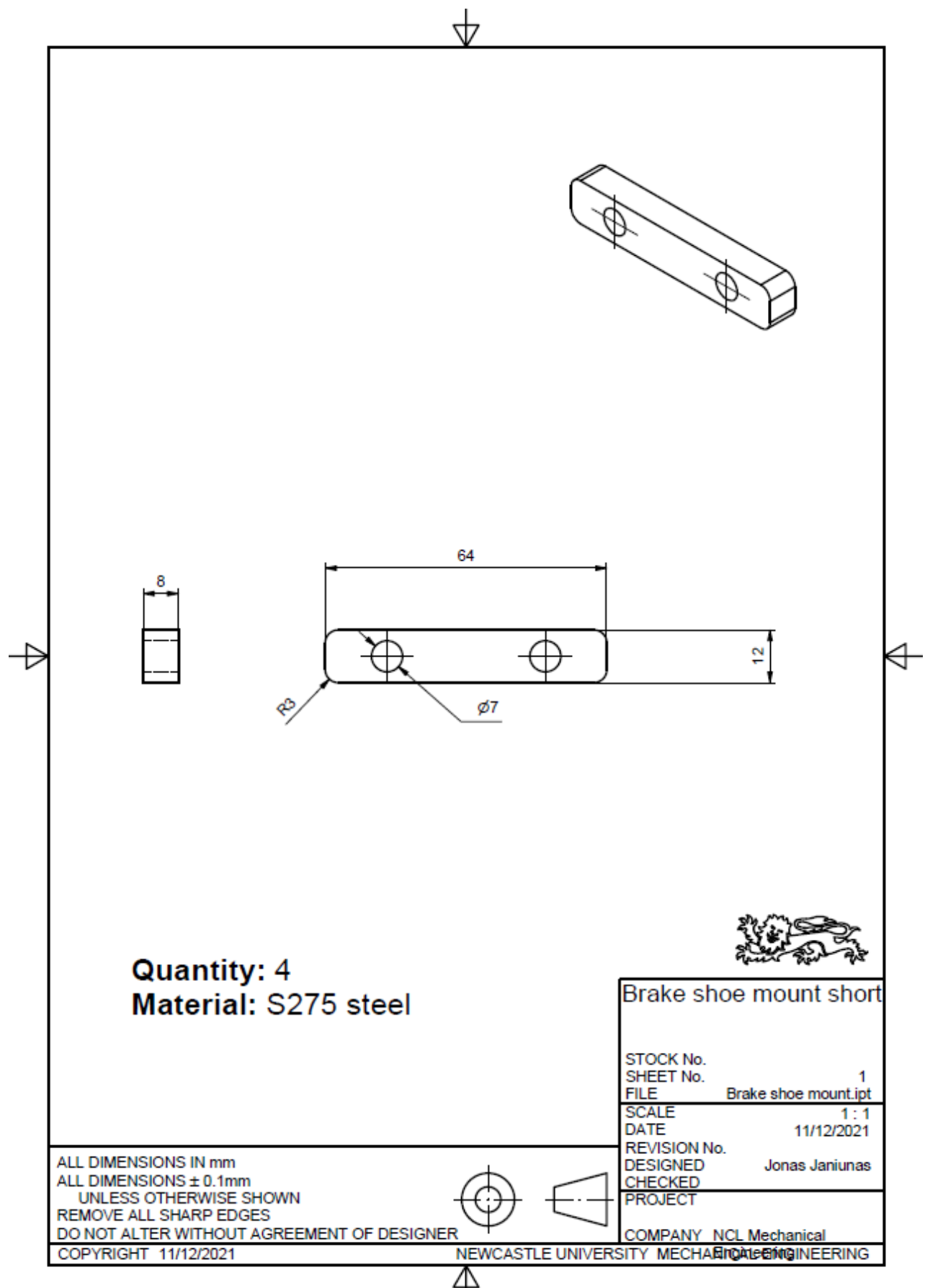
Appendix B-48: (JJ)



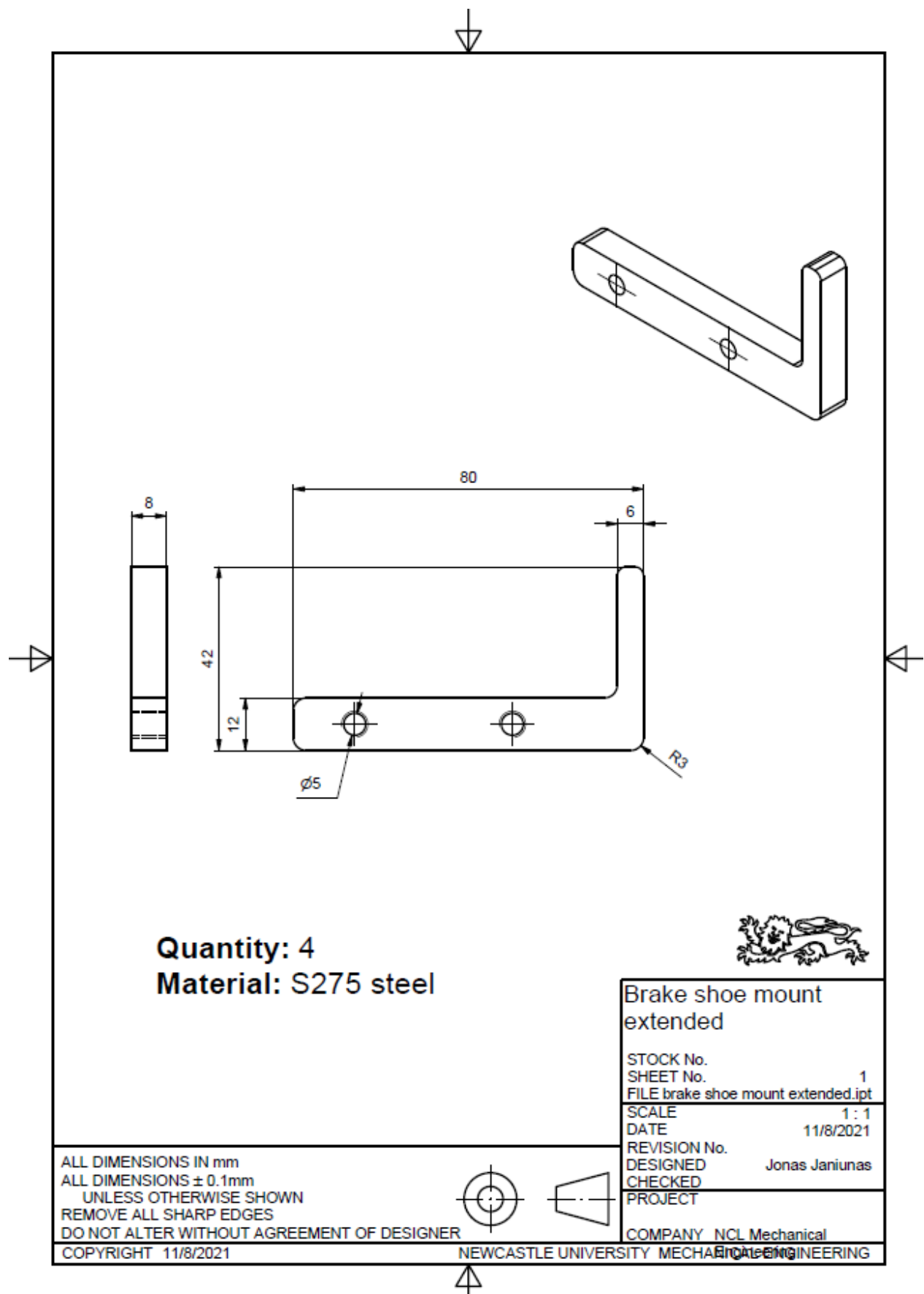
Appendix B-49: (JJ)



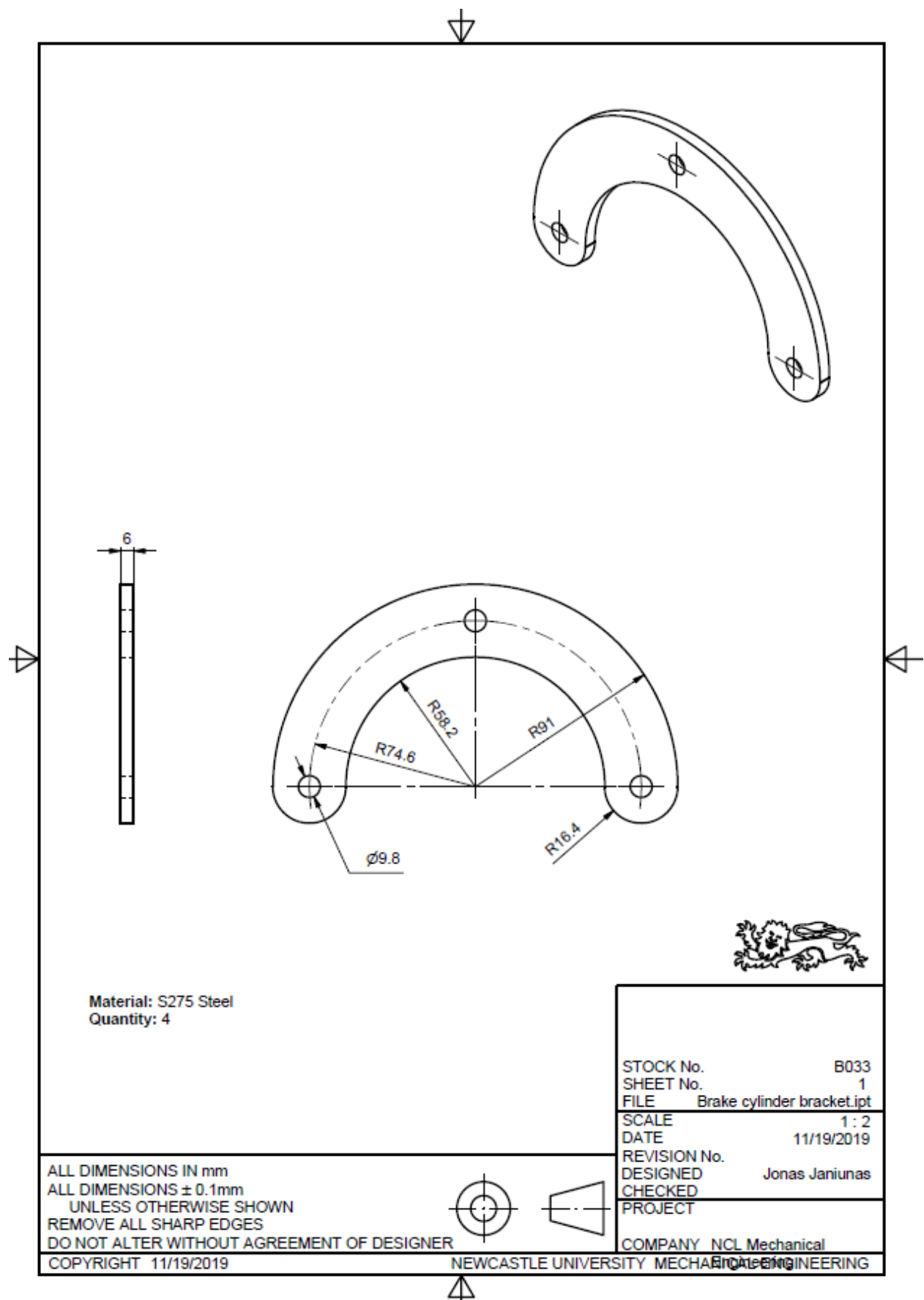
Appendix B-50: (JJ)



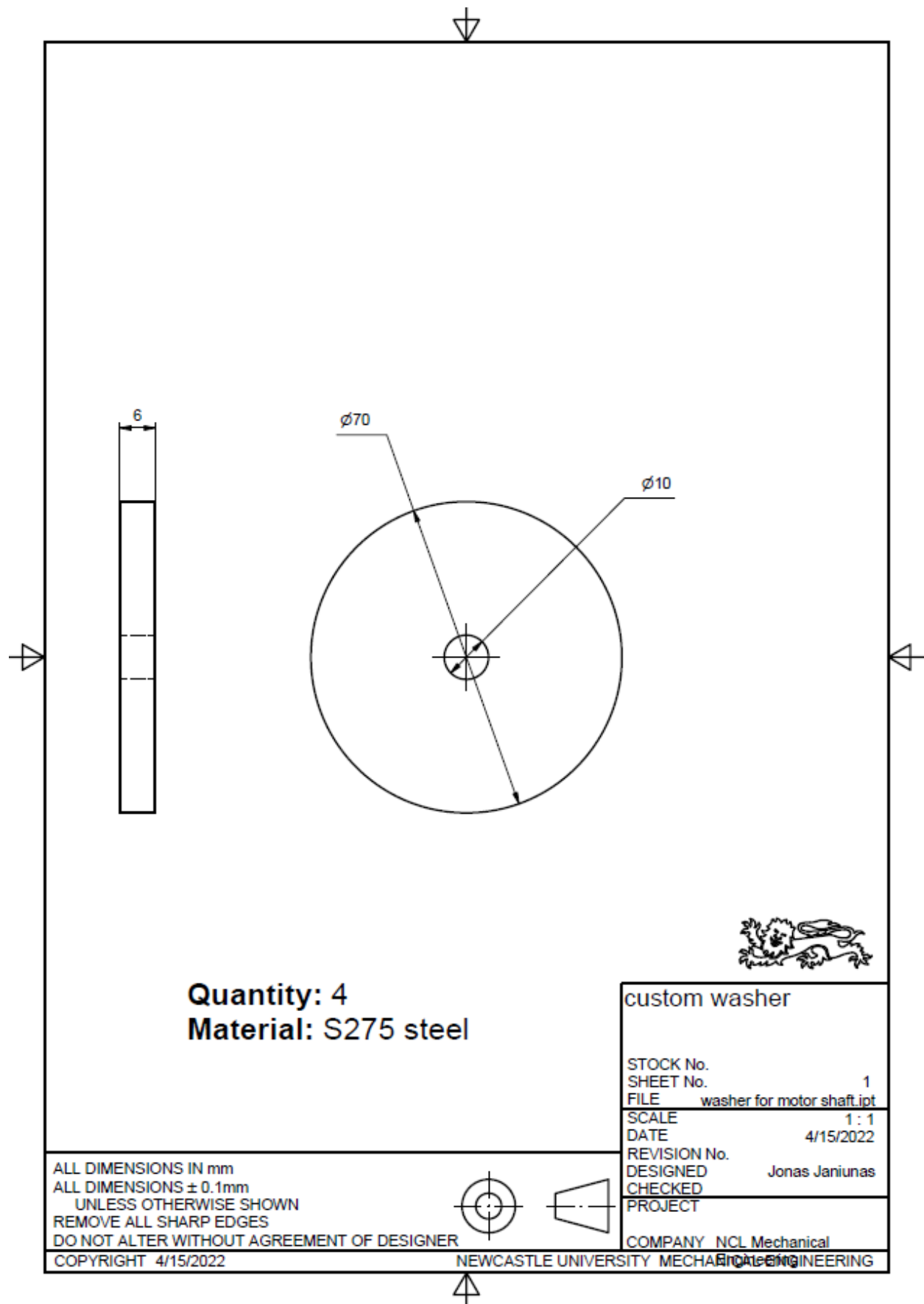
Appendix B-51: (JJ)



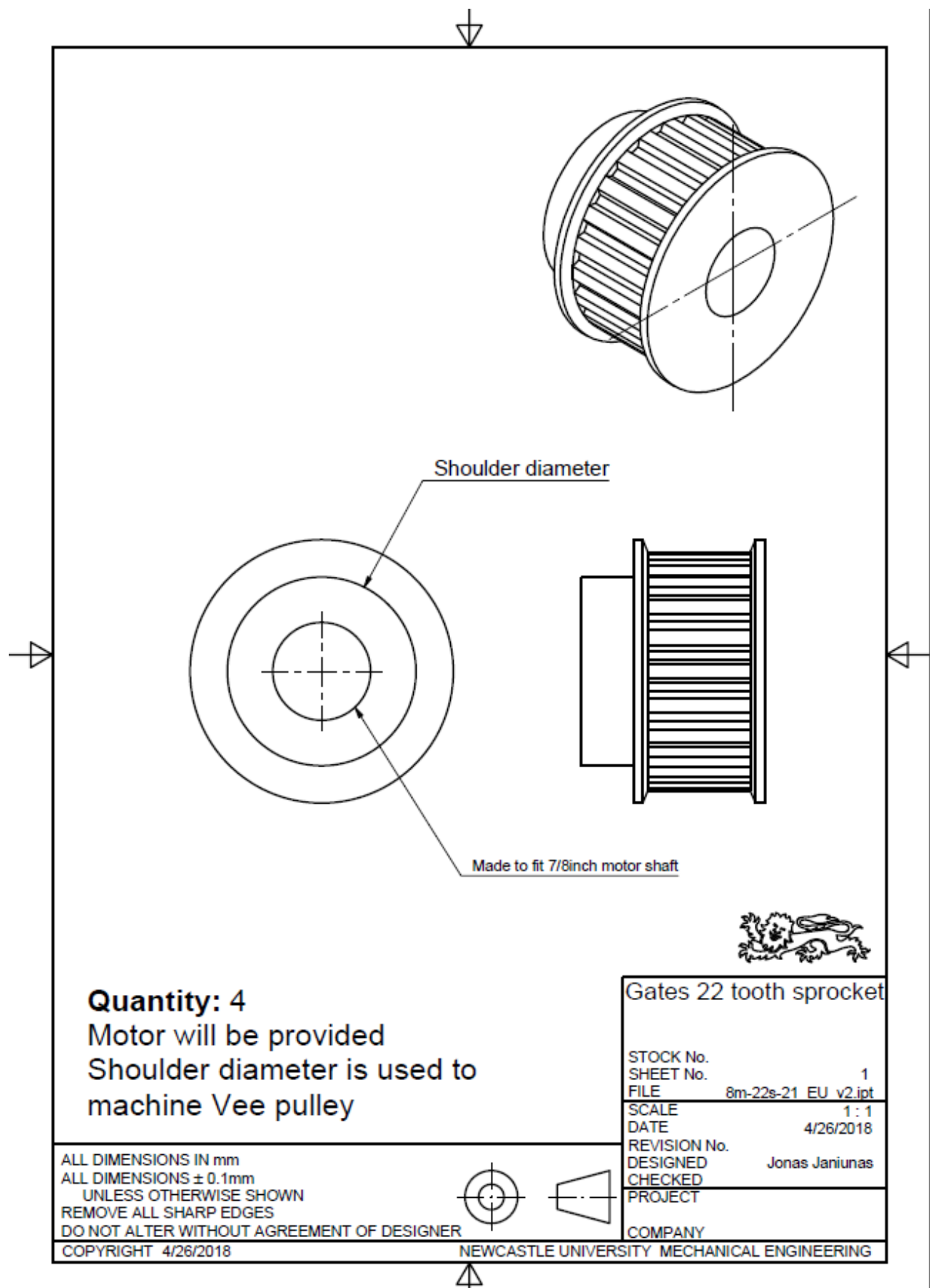
Appendix B-52: (JJ)



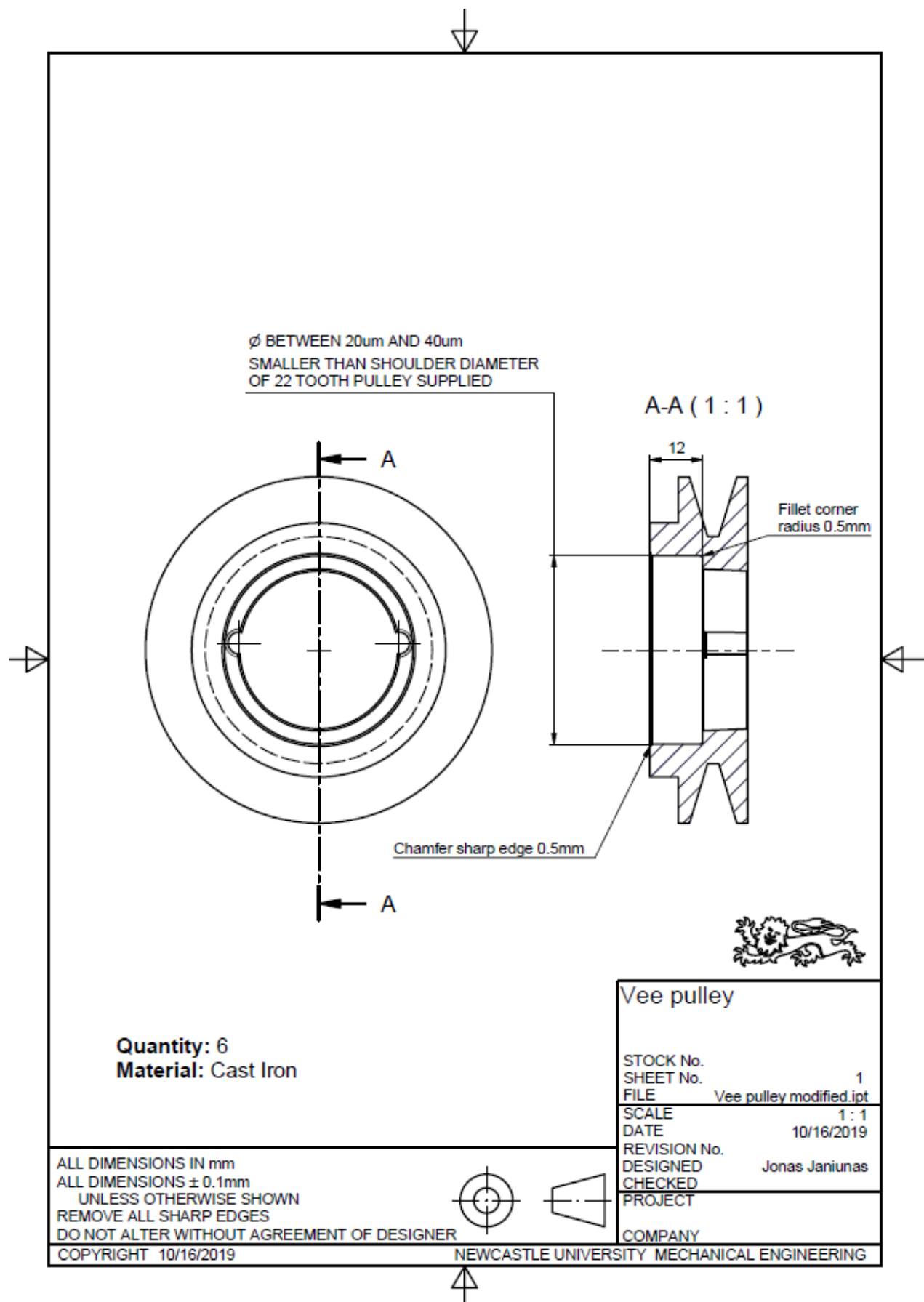
Appendix B-53: (JJ)



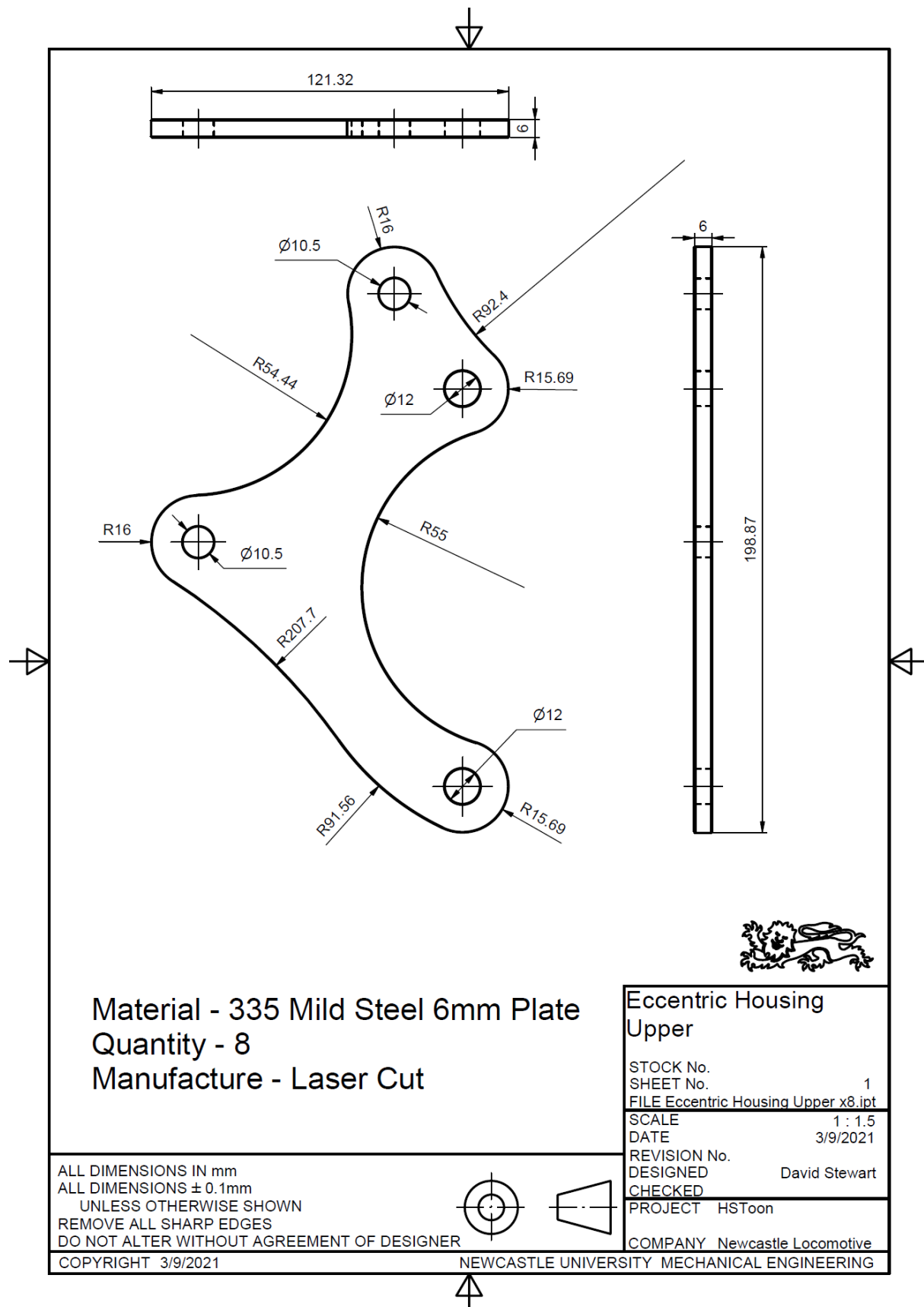
Appendix B-54: (JJ)



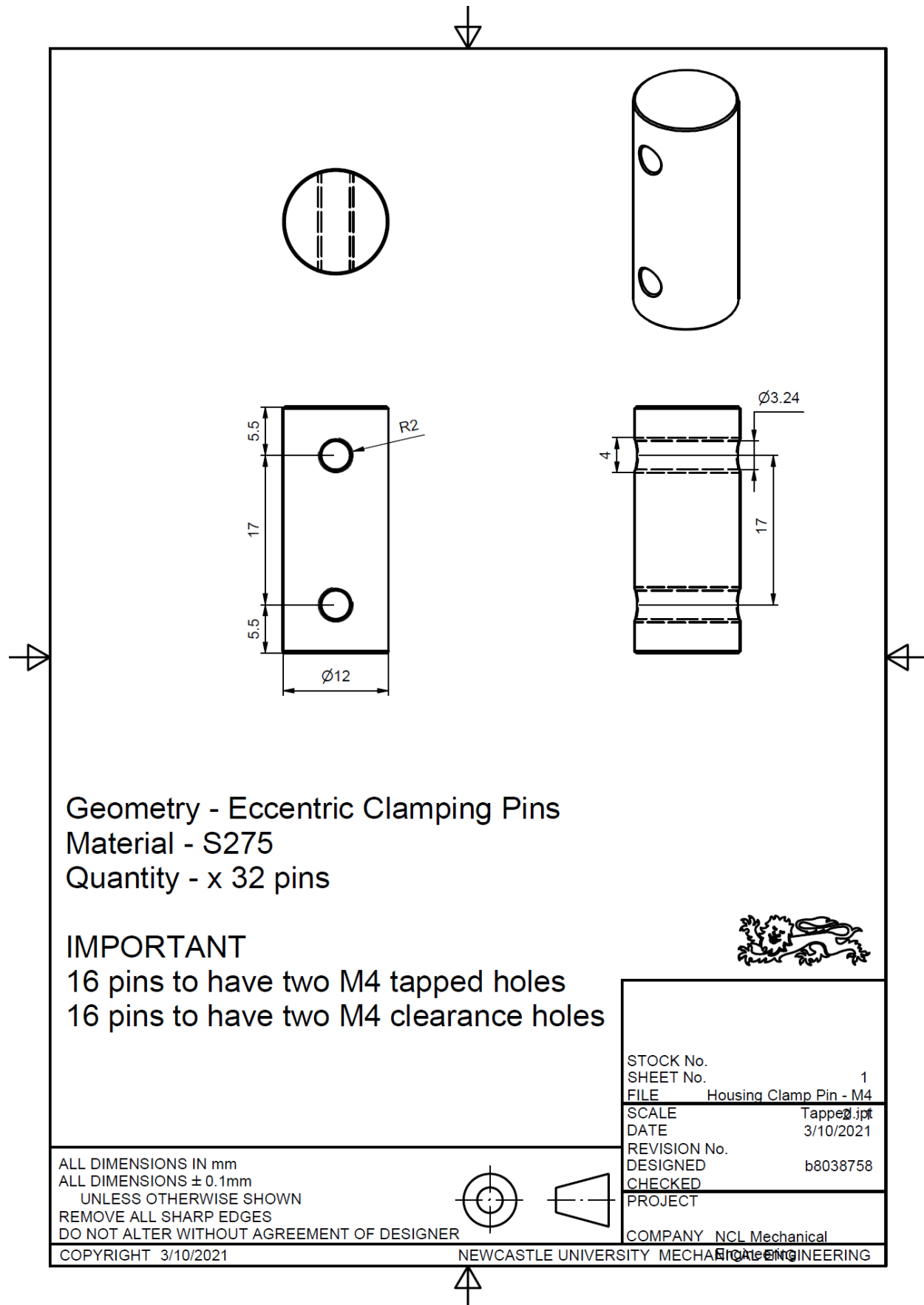
Appendix B-55: (JJ)



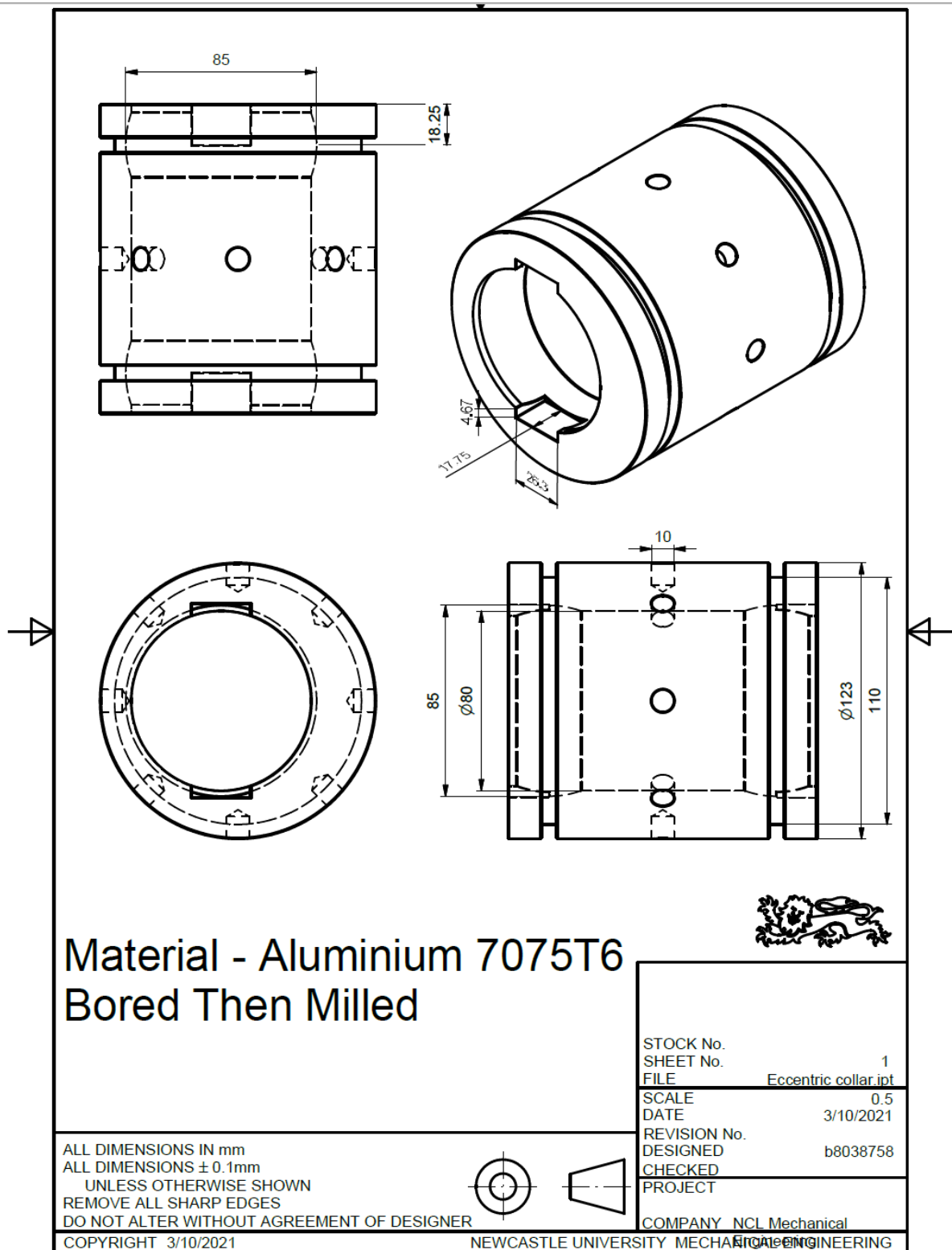
Appendix B-56: (DS)



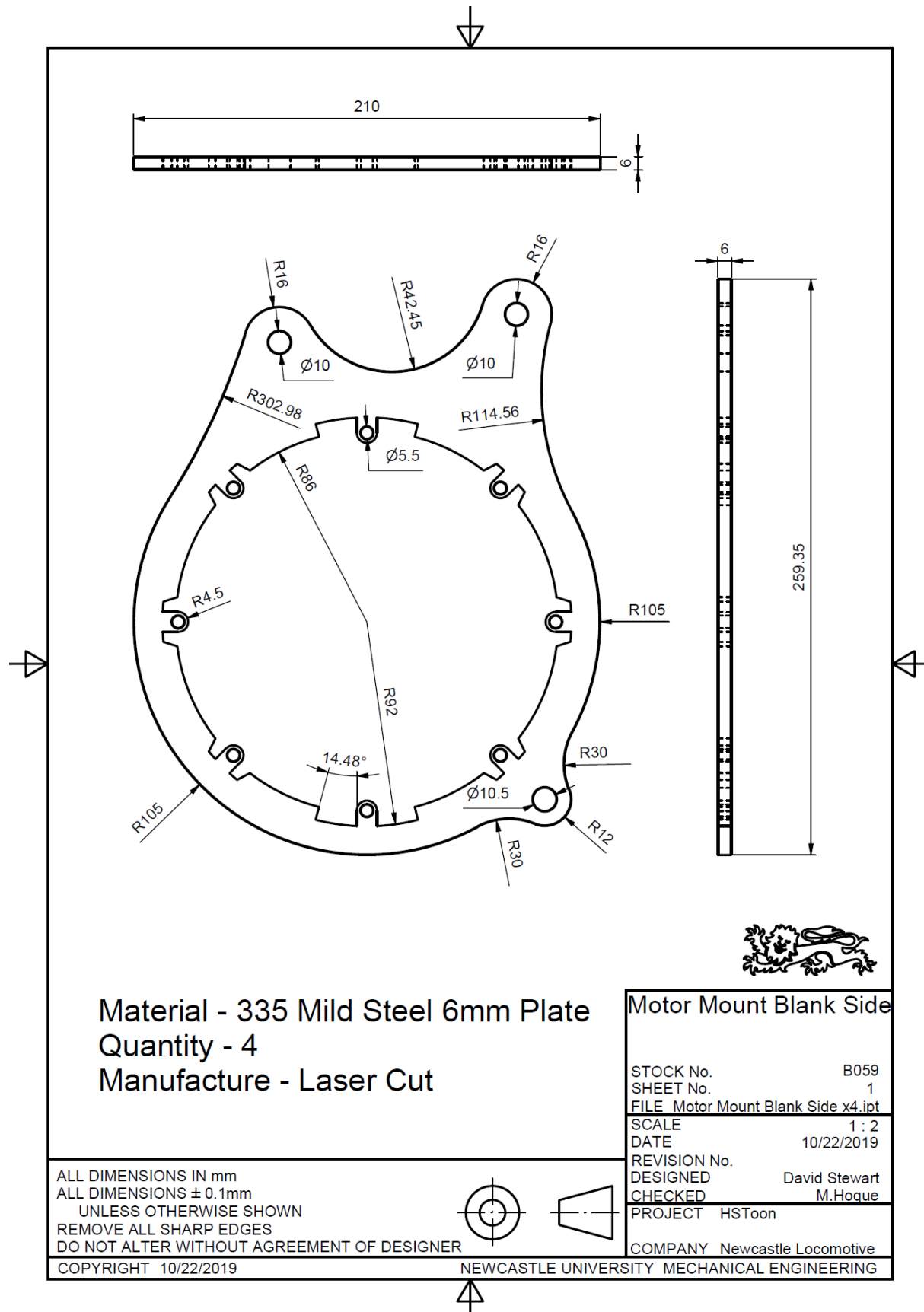
Appendix B-57: (DS)



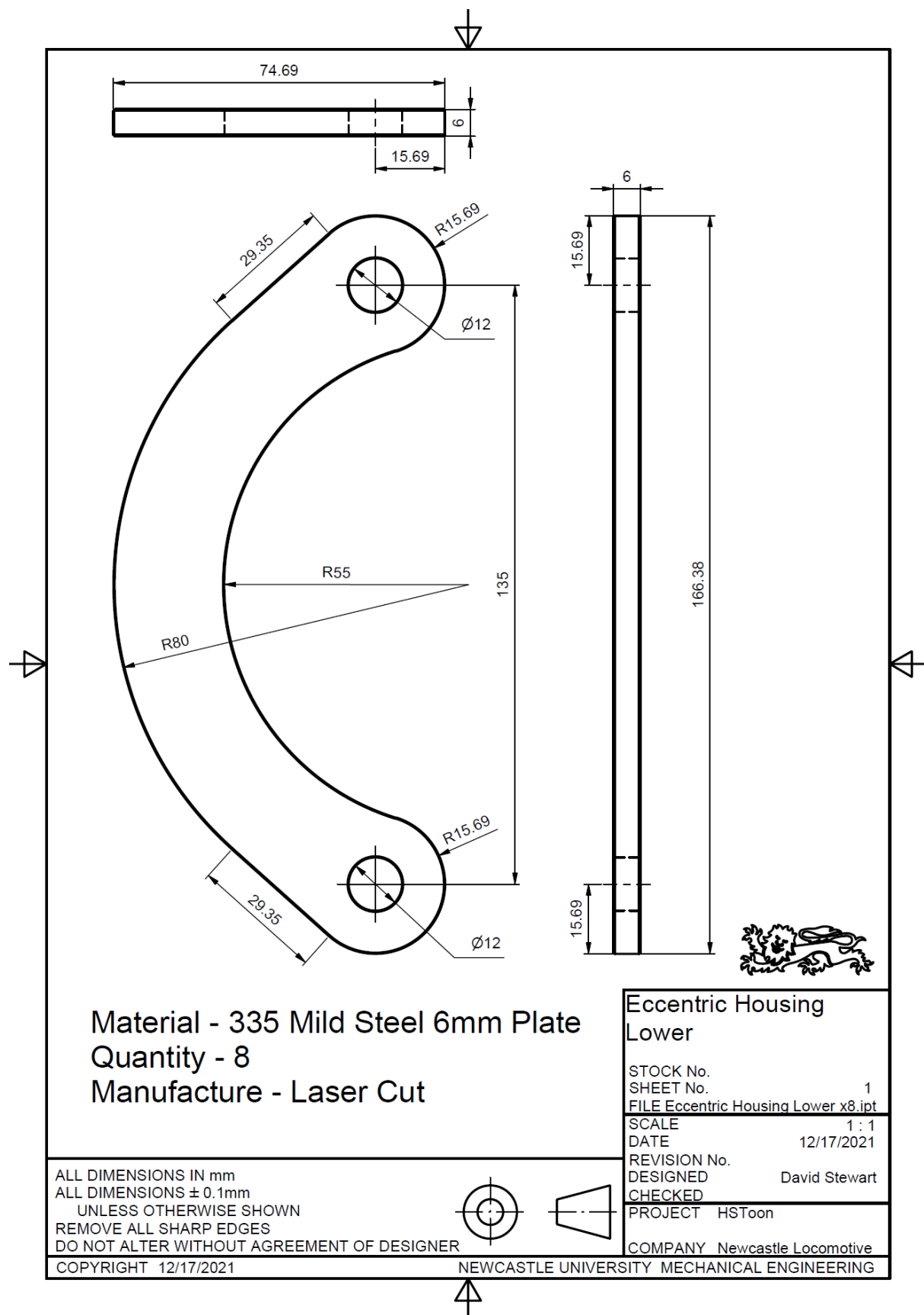
Appendix B-58: (DS)



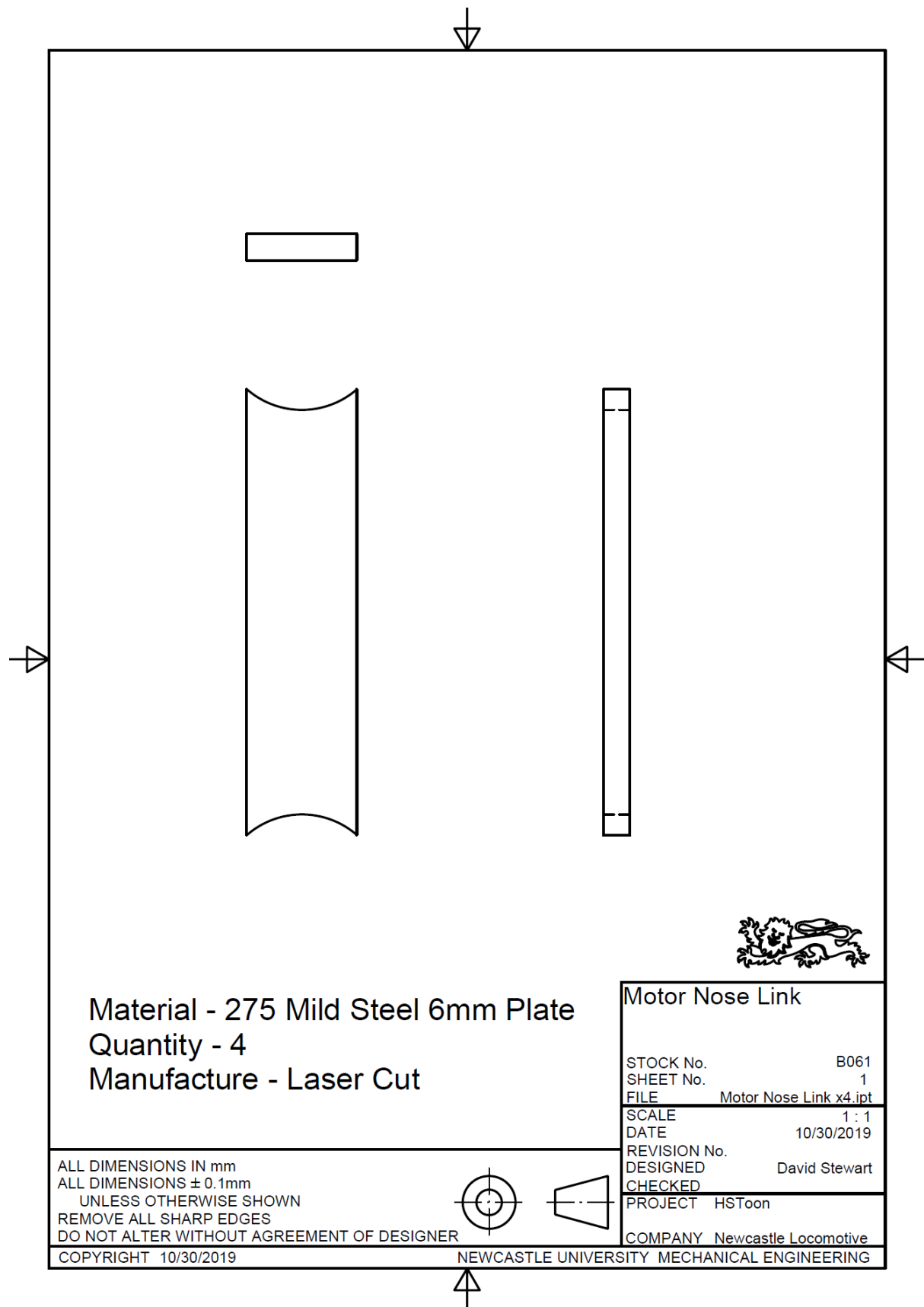
Appendix B-59: (DS)



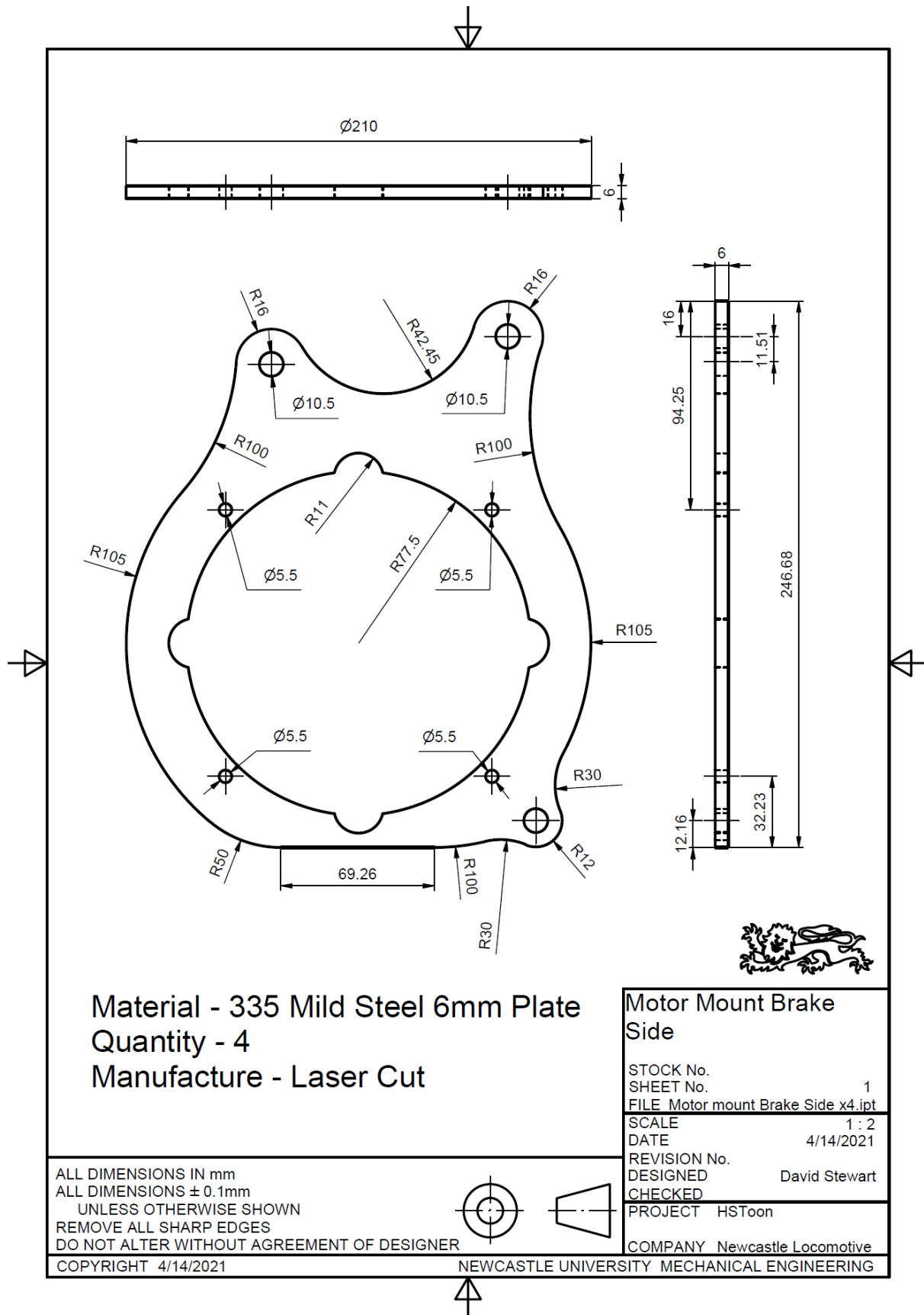
Appendix B-60: (DS)



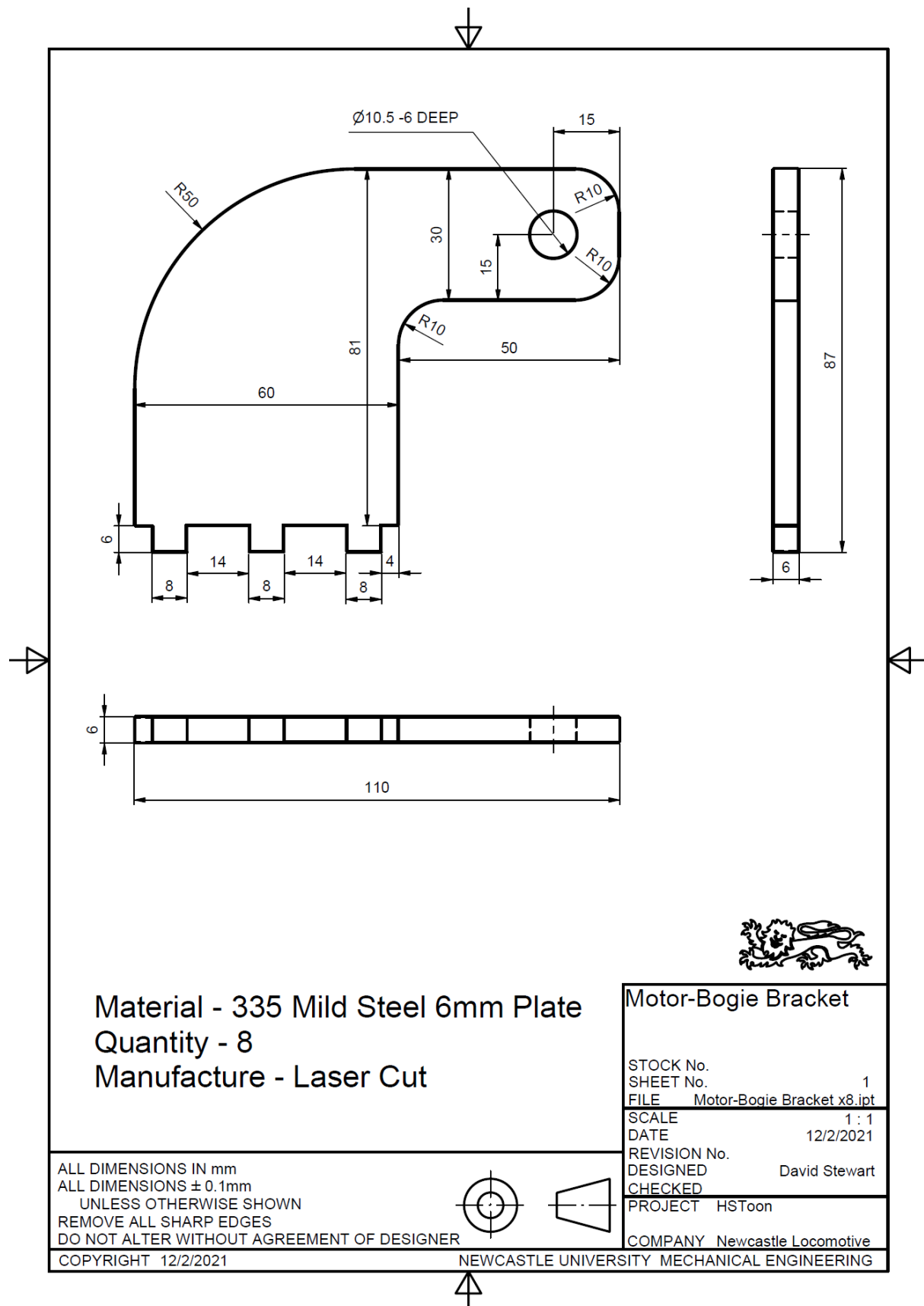
Appendix B-61: (DS)



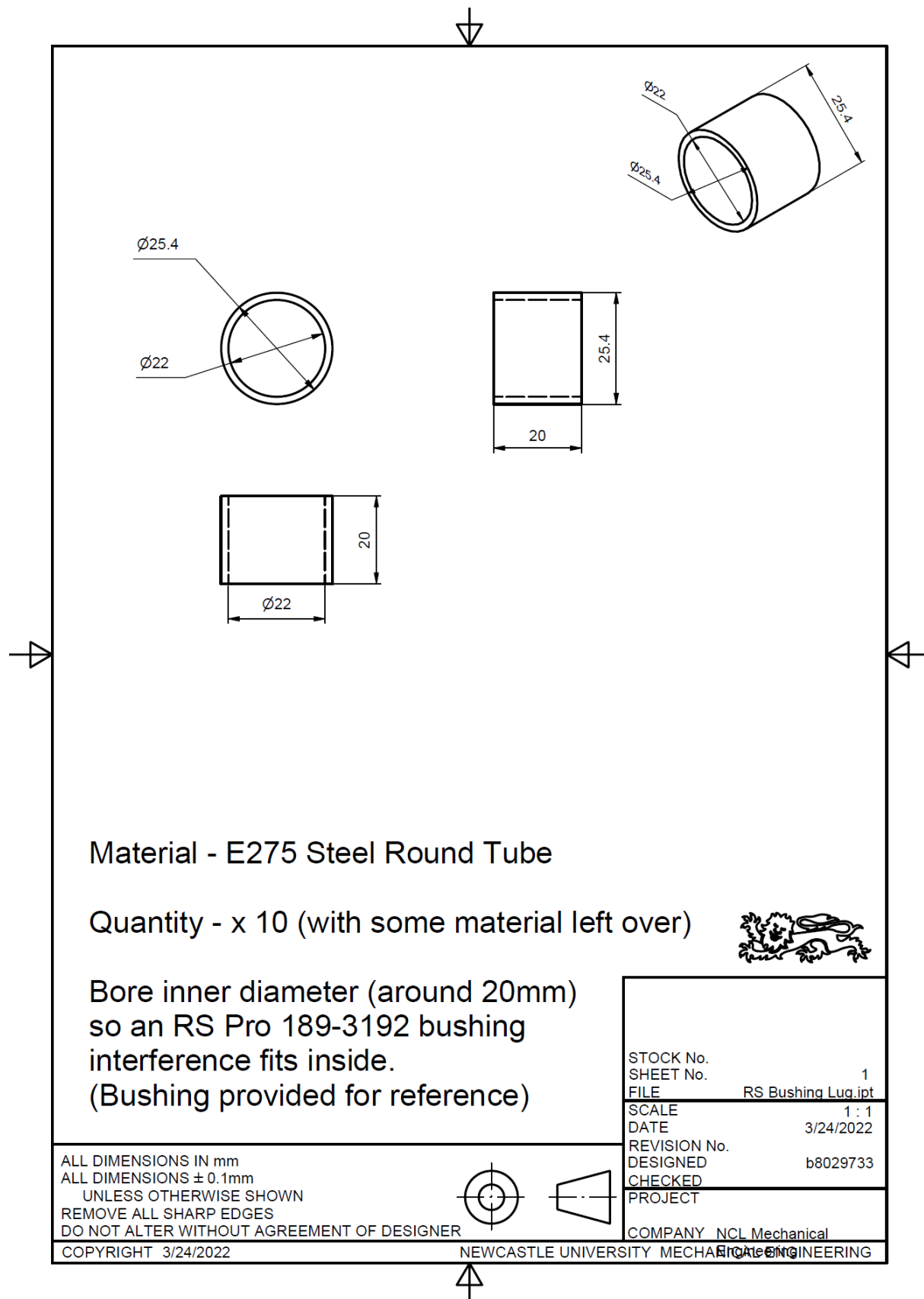
Appendix B-62: (DS)



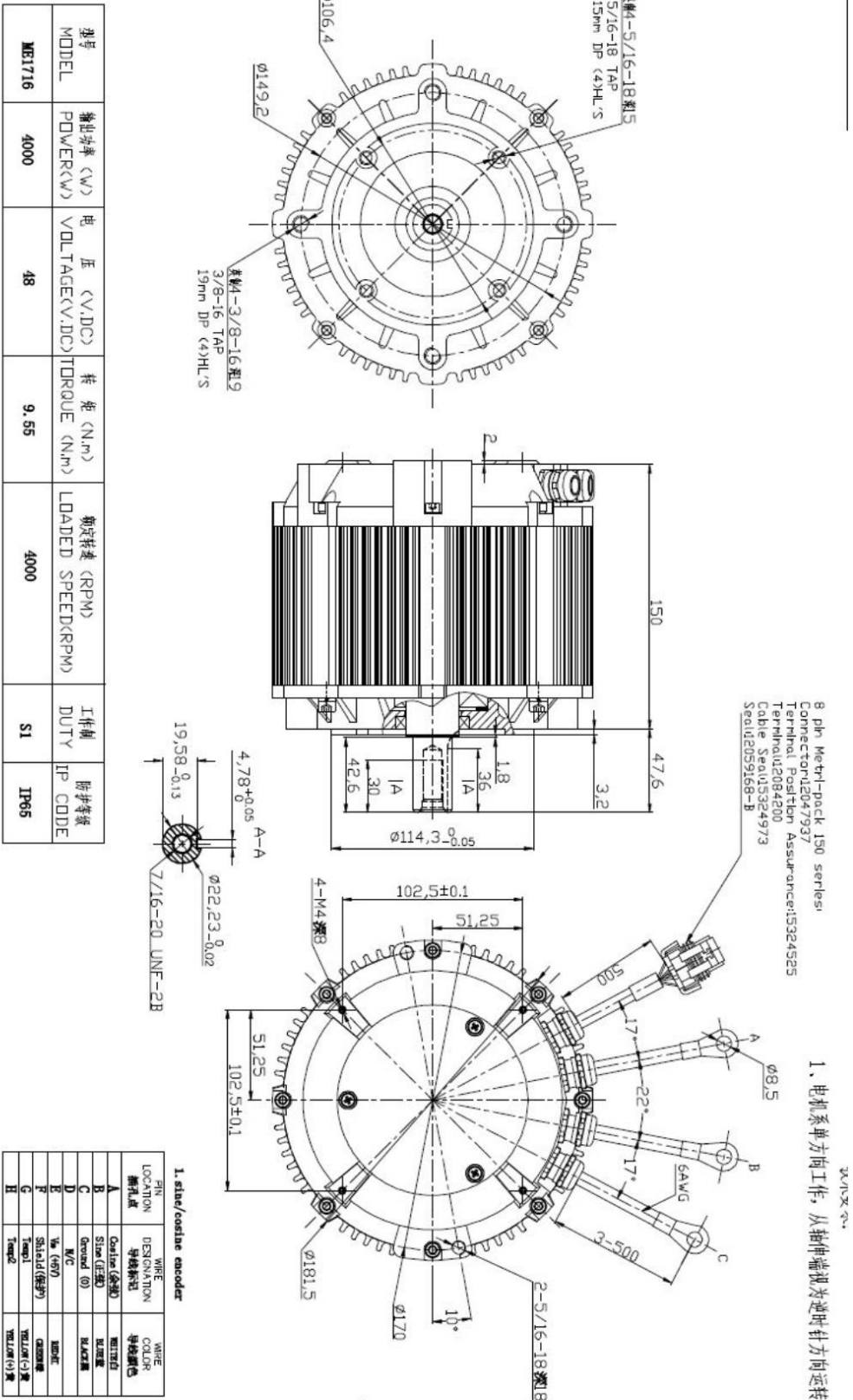
Appendix B-63: (DS)



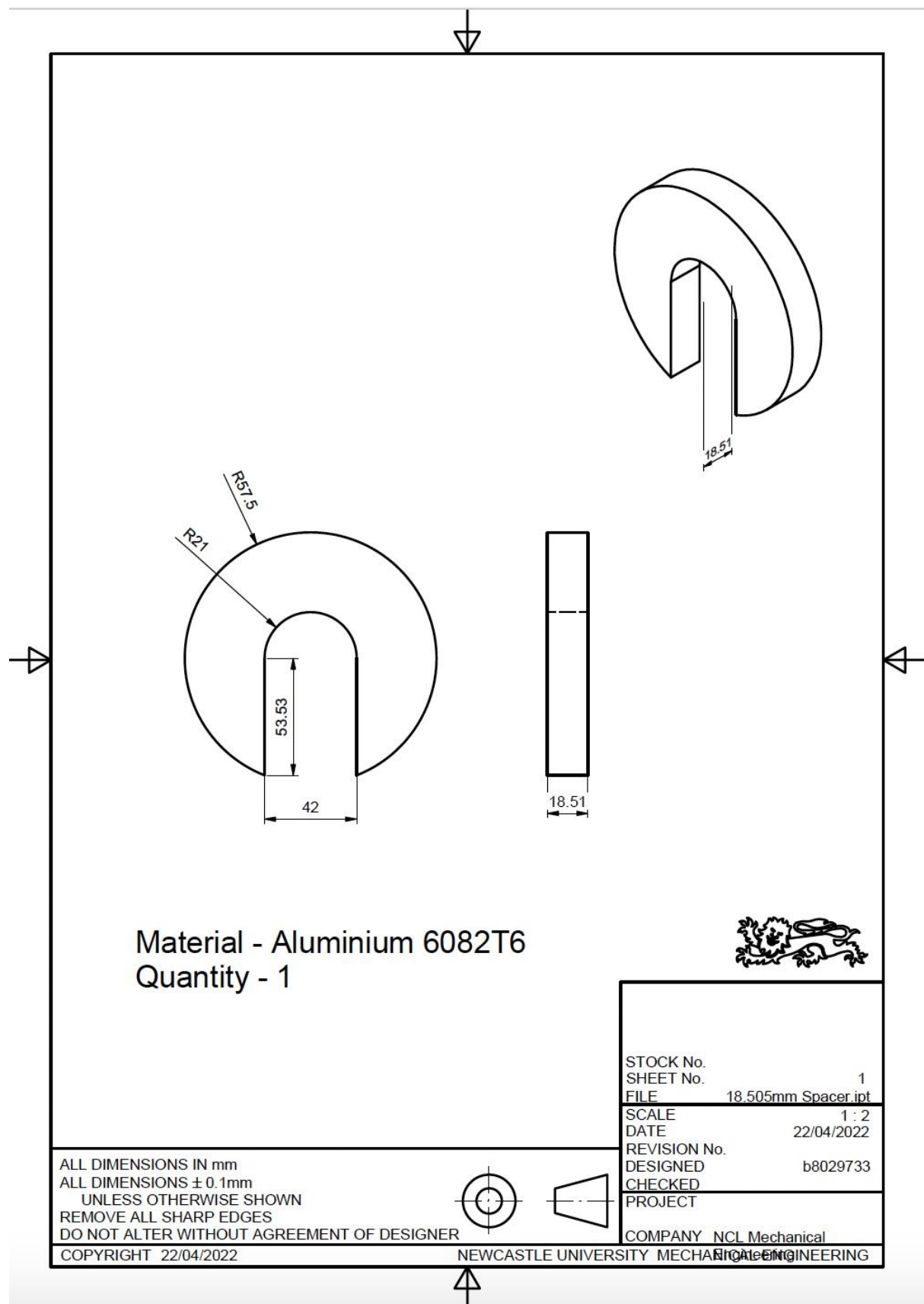
Appendix B-64: (DS)



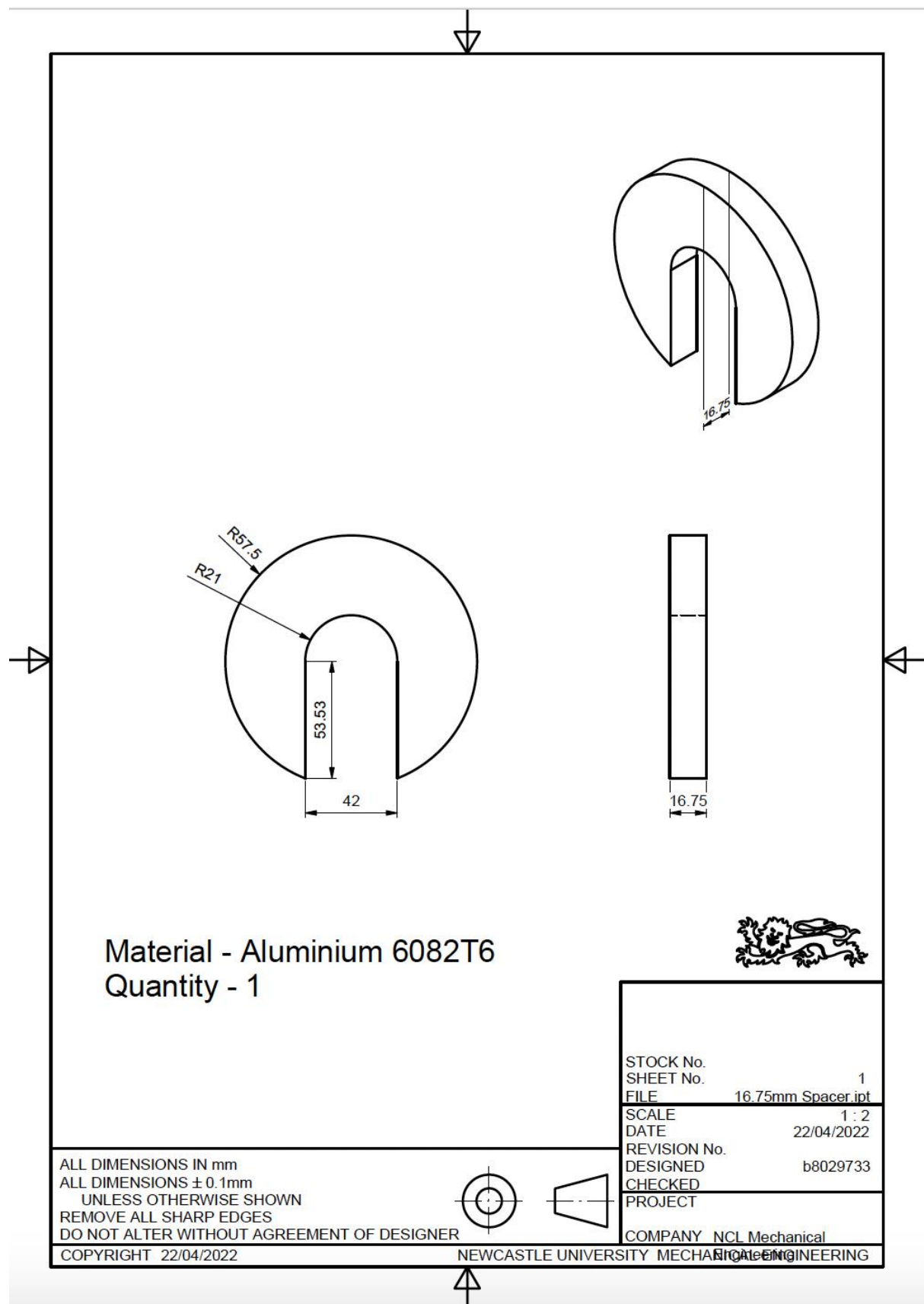
Appendix B-65: Motenergy 1718 Engineering Drawing (DS)



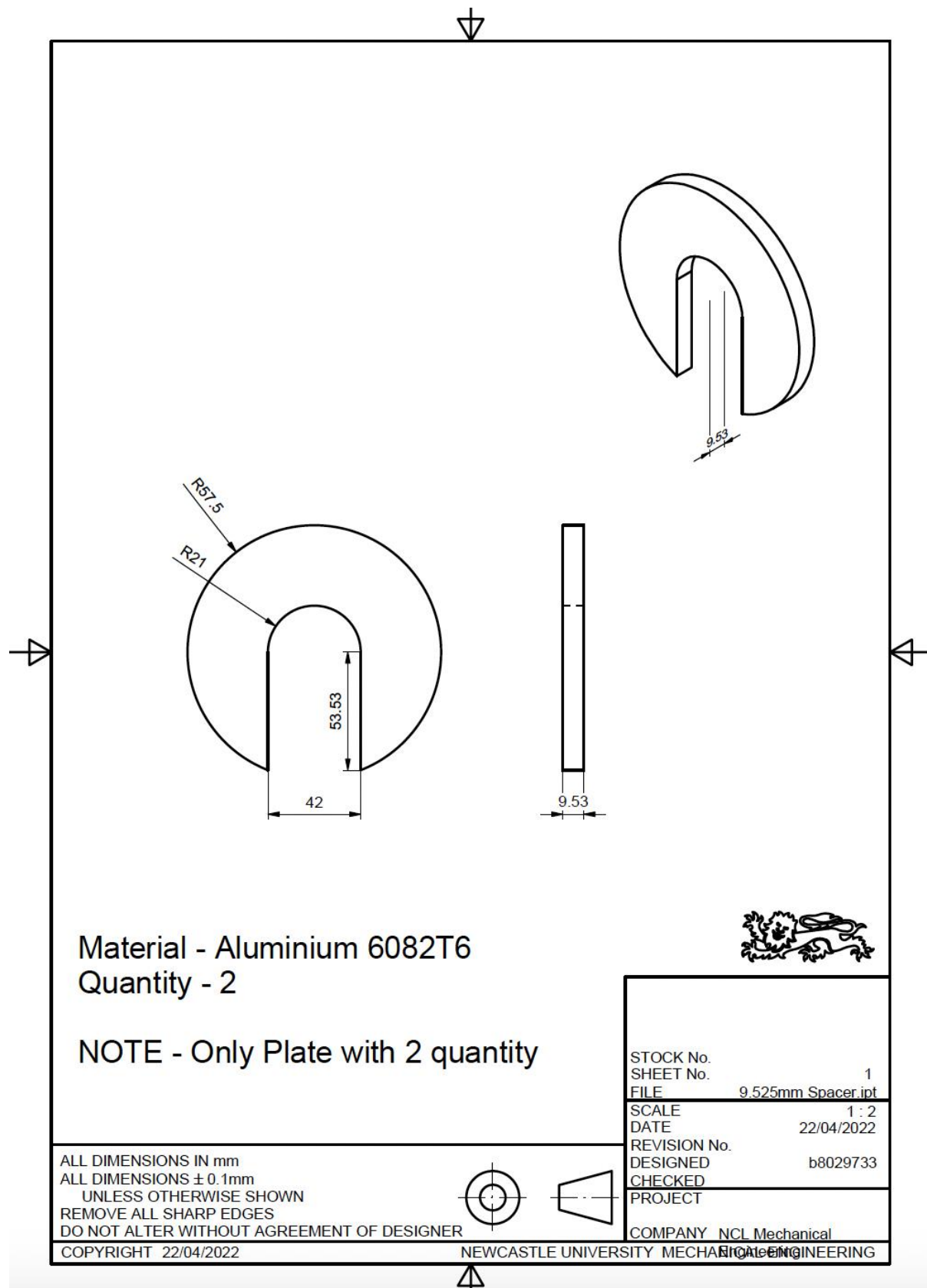
Appendix B-66: (DS)



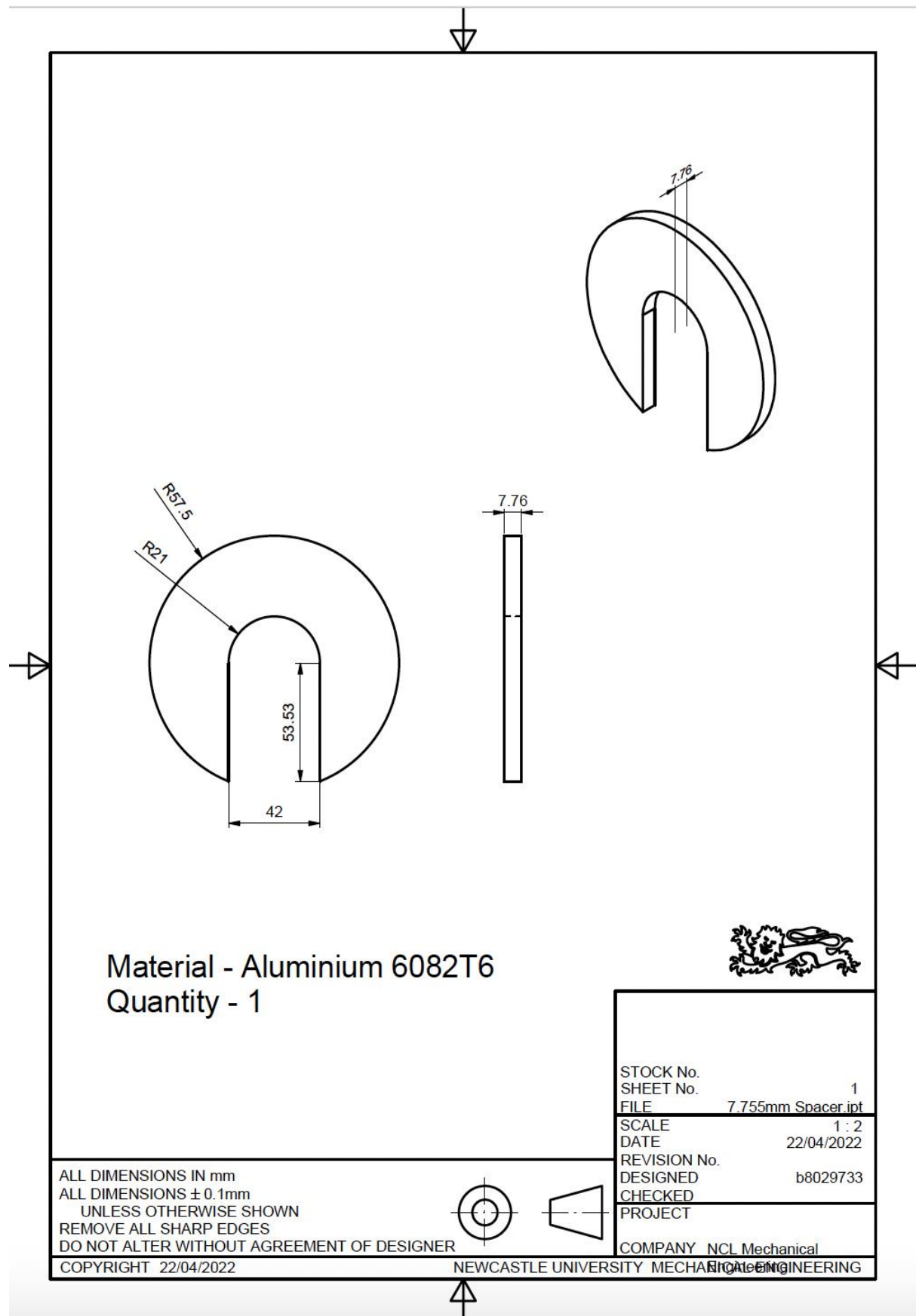
Appendix B-67: (DS)



Appendix B-68: (DS)

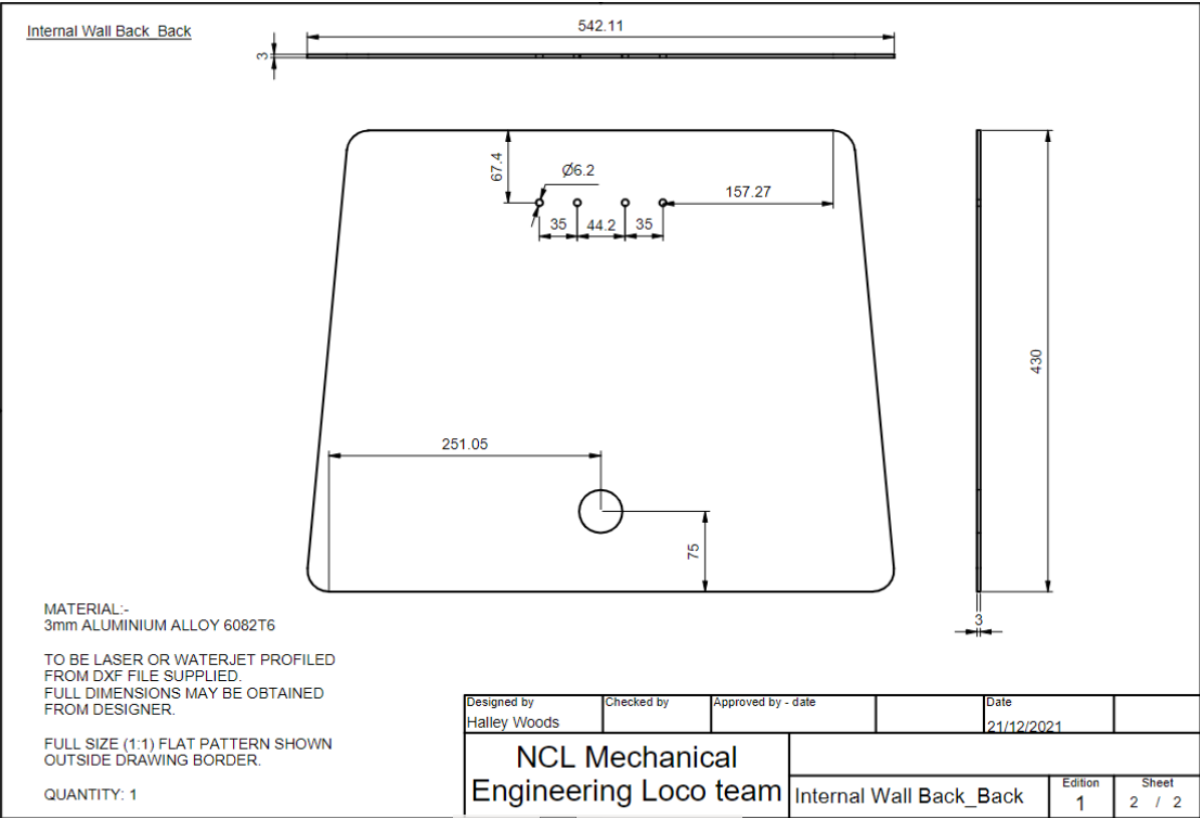
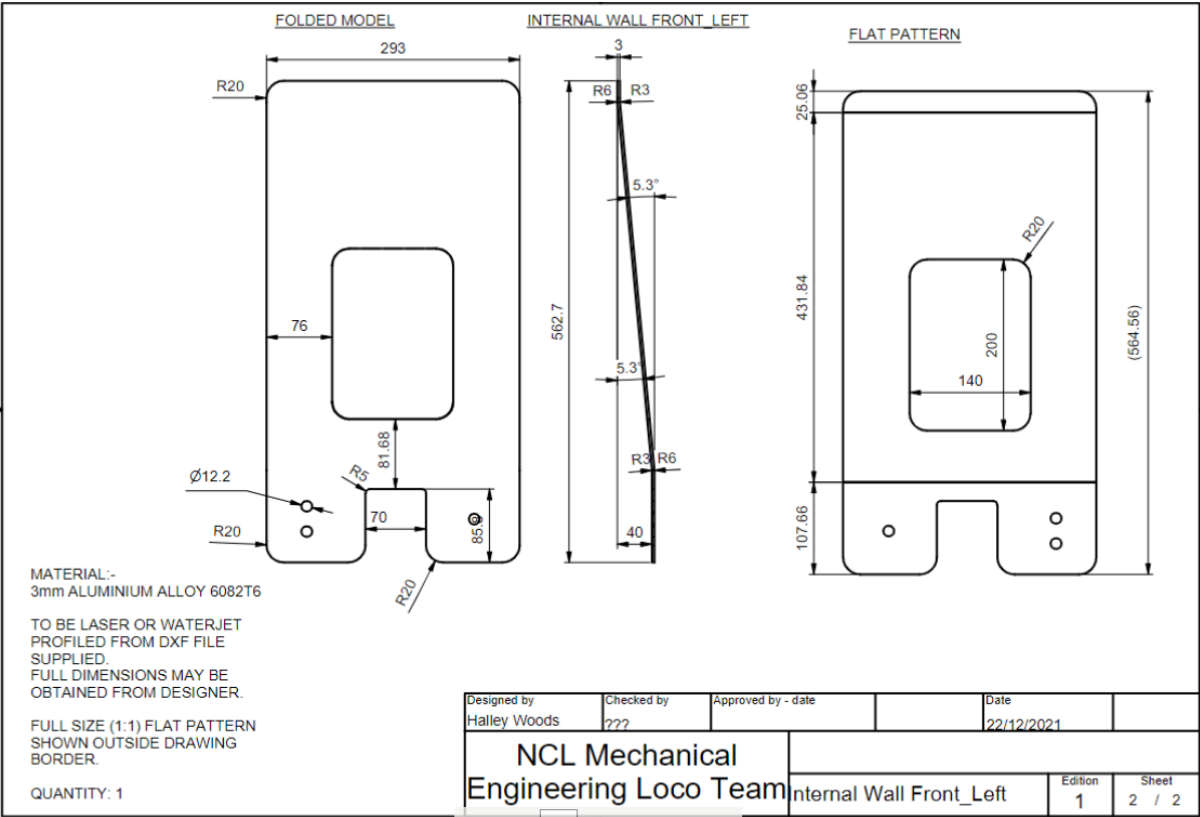


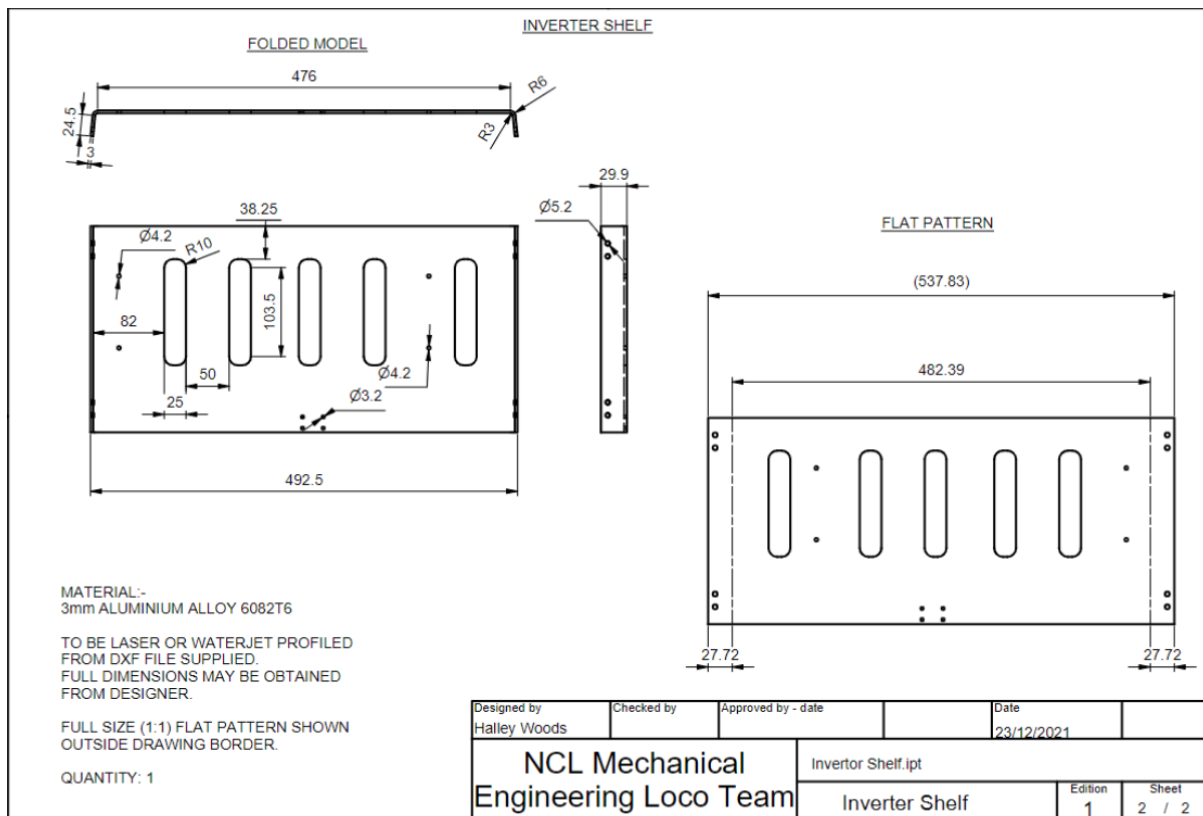
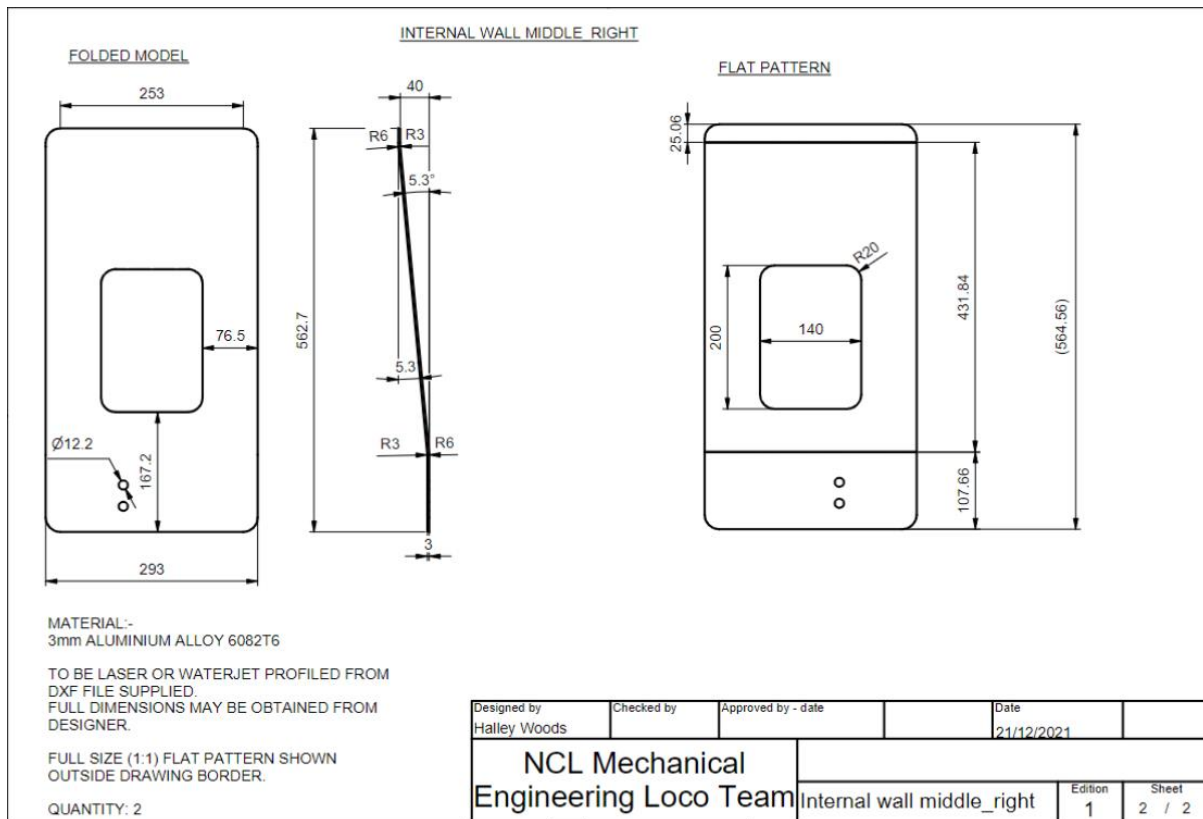
Appendix B-69: (DS)

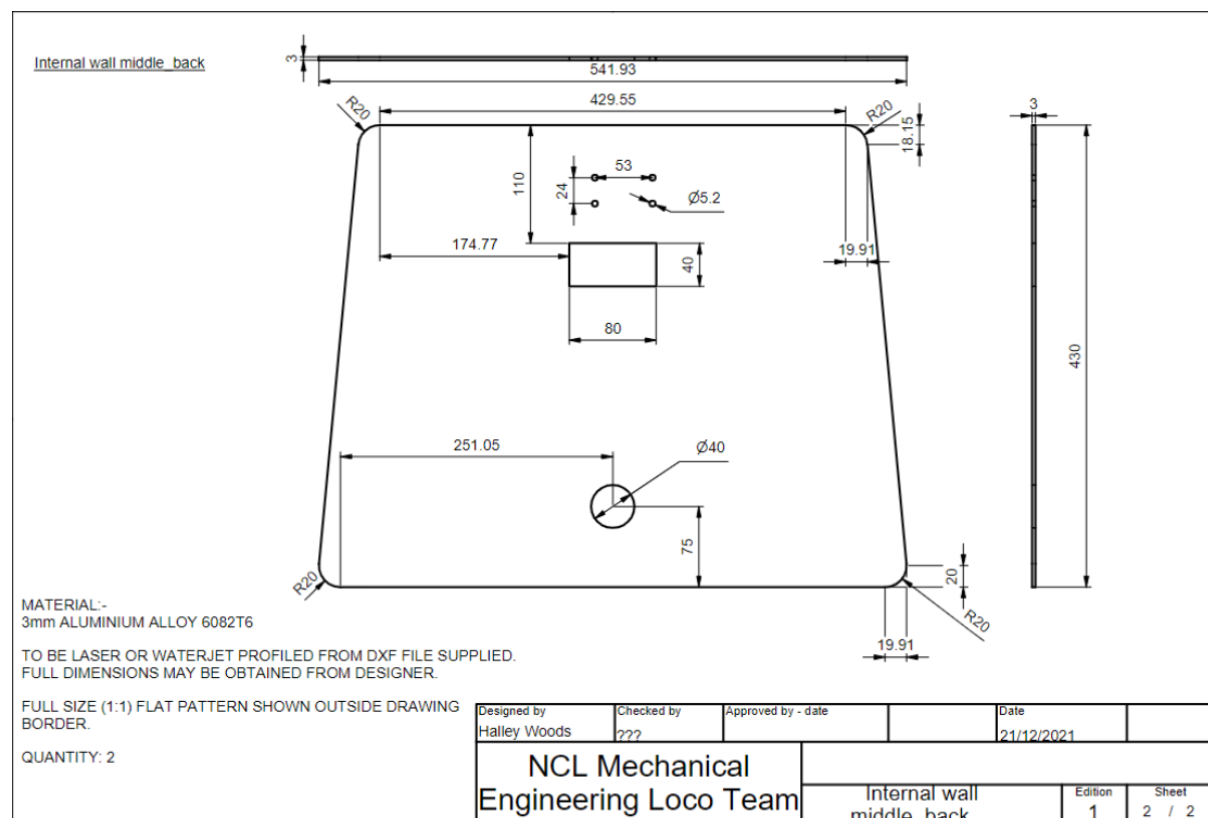
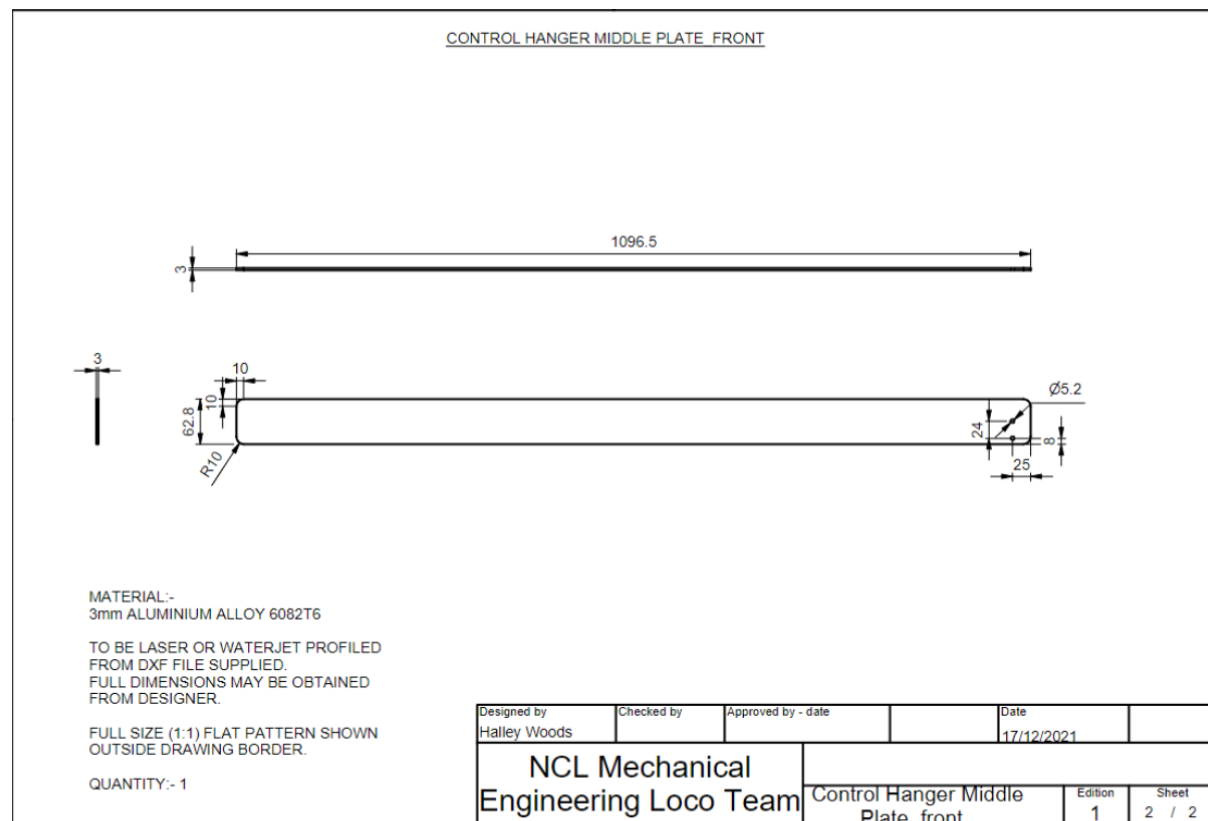


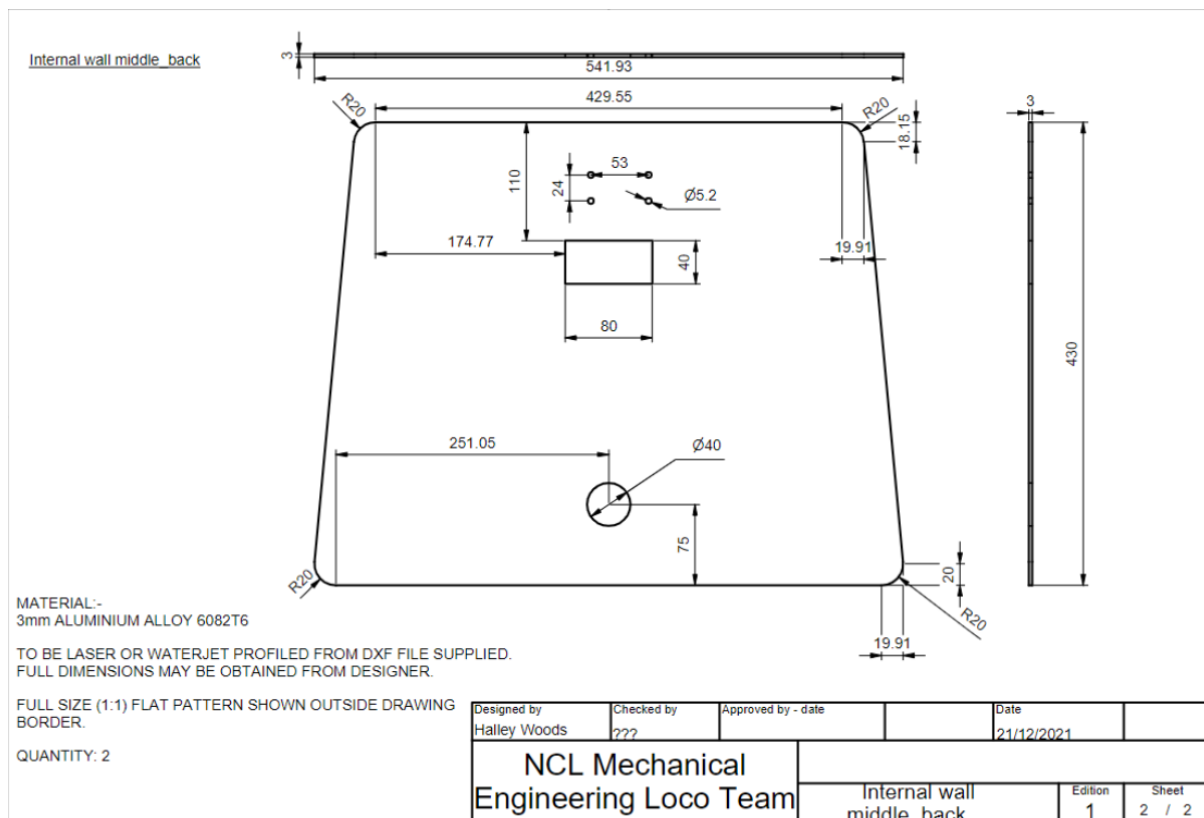
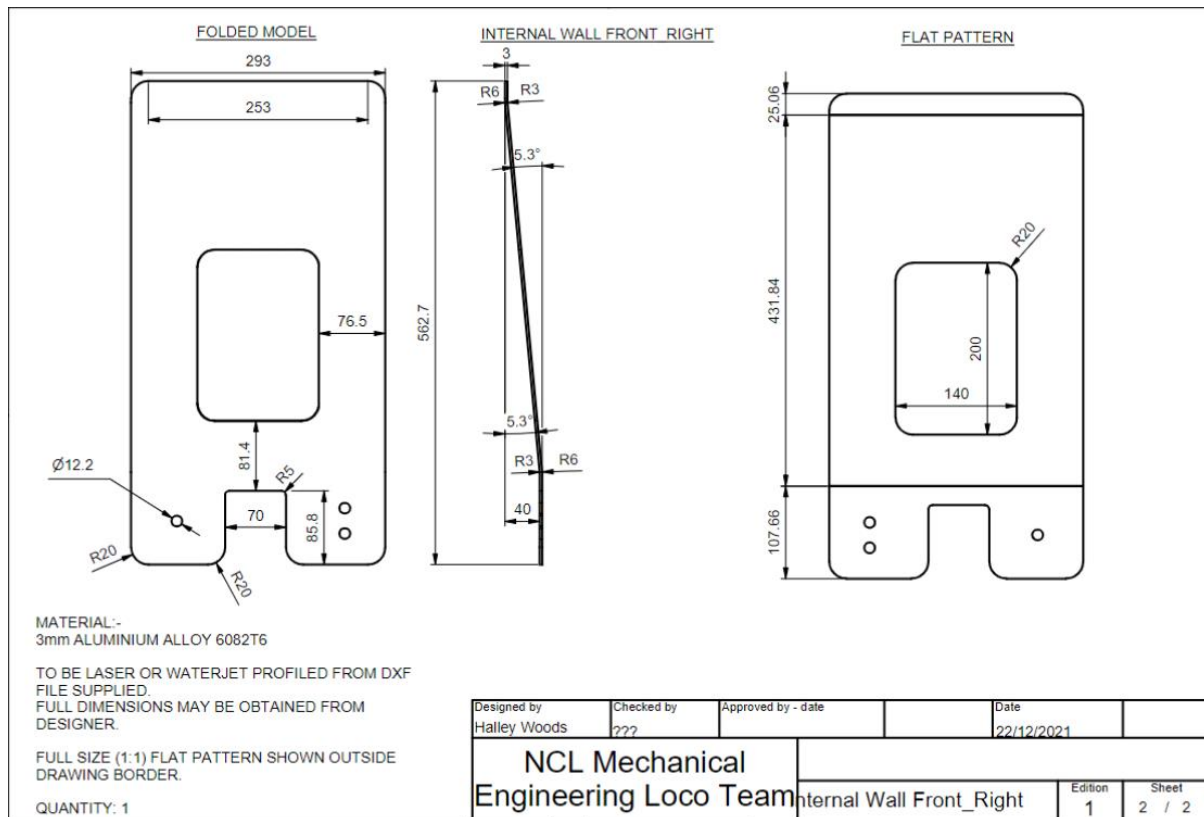
Appendix C: Body

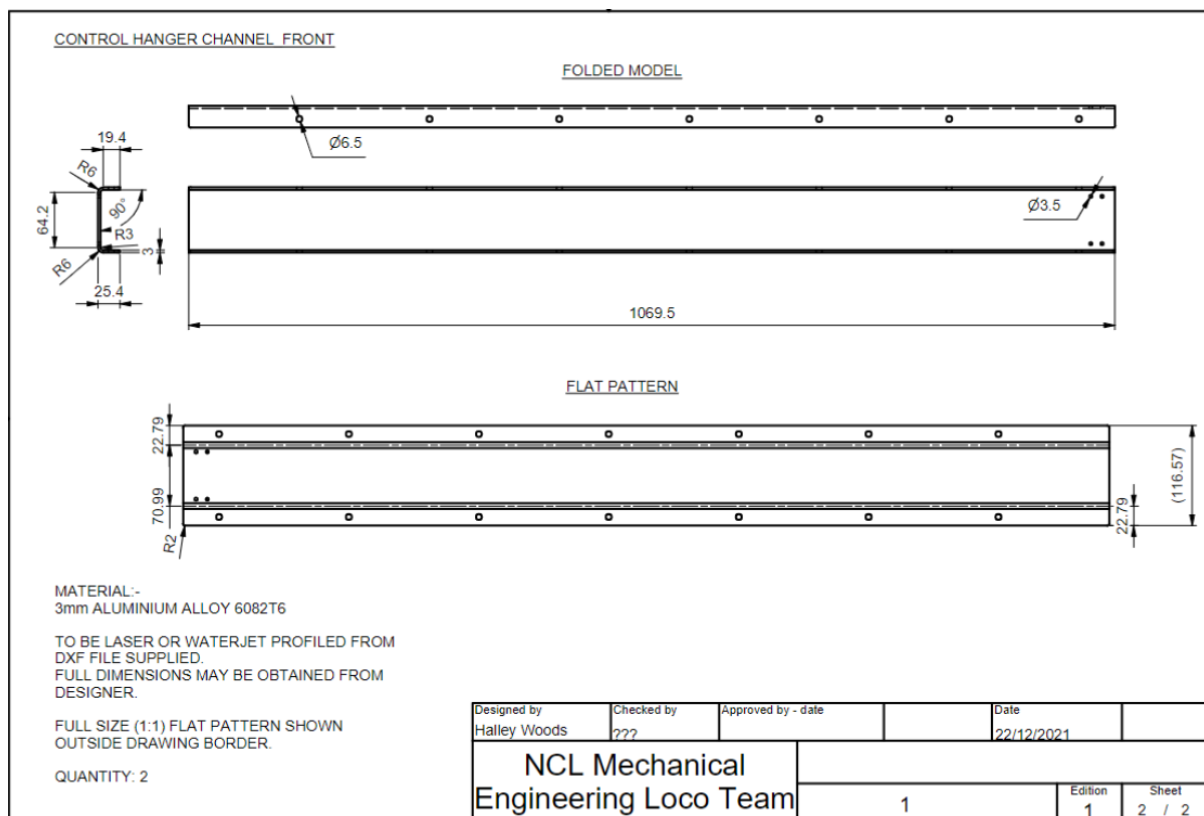
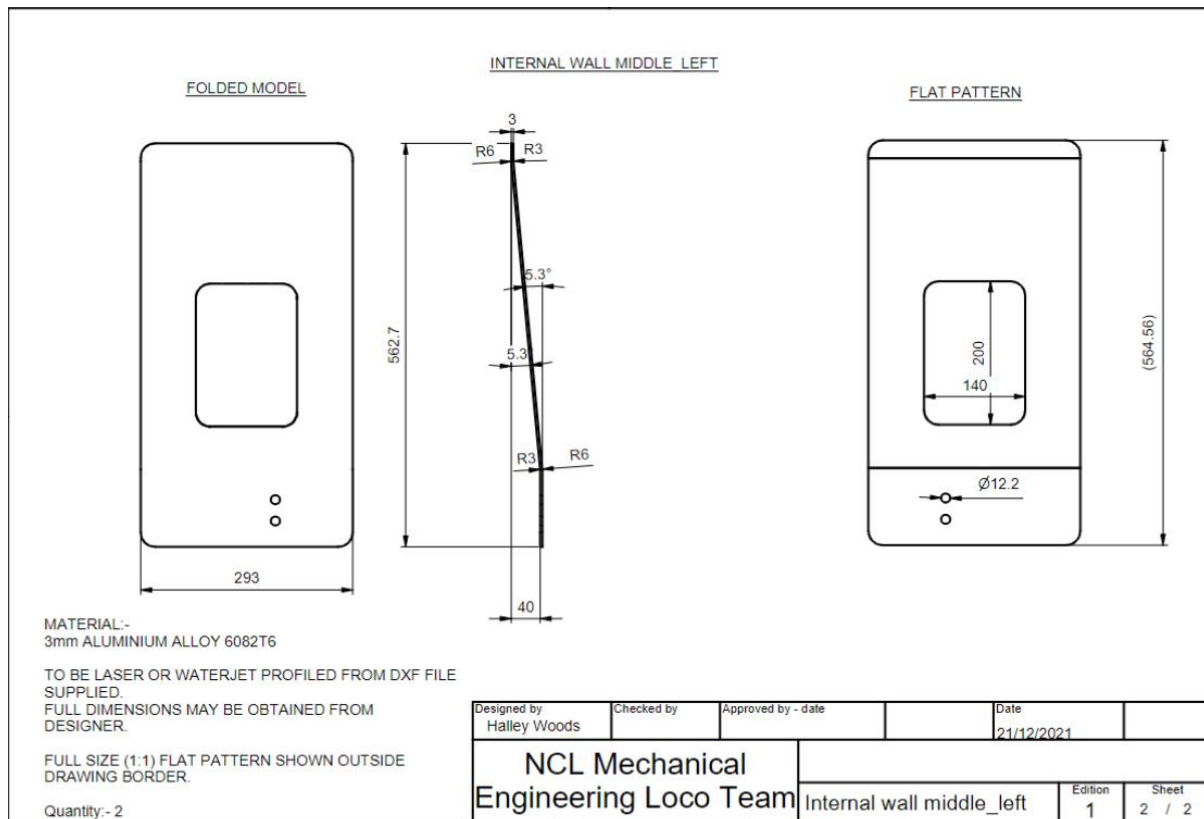
Appendix C-1: Body Drawings (HW)

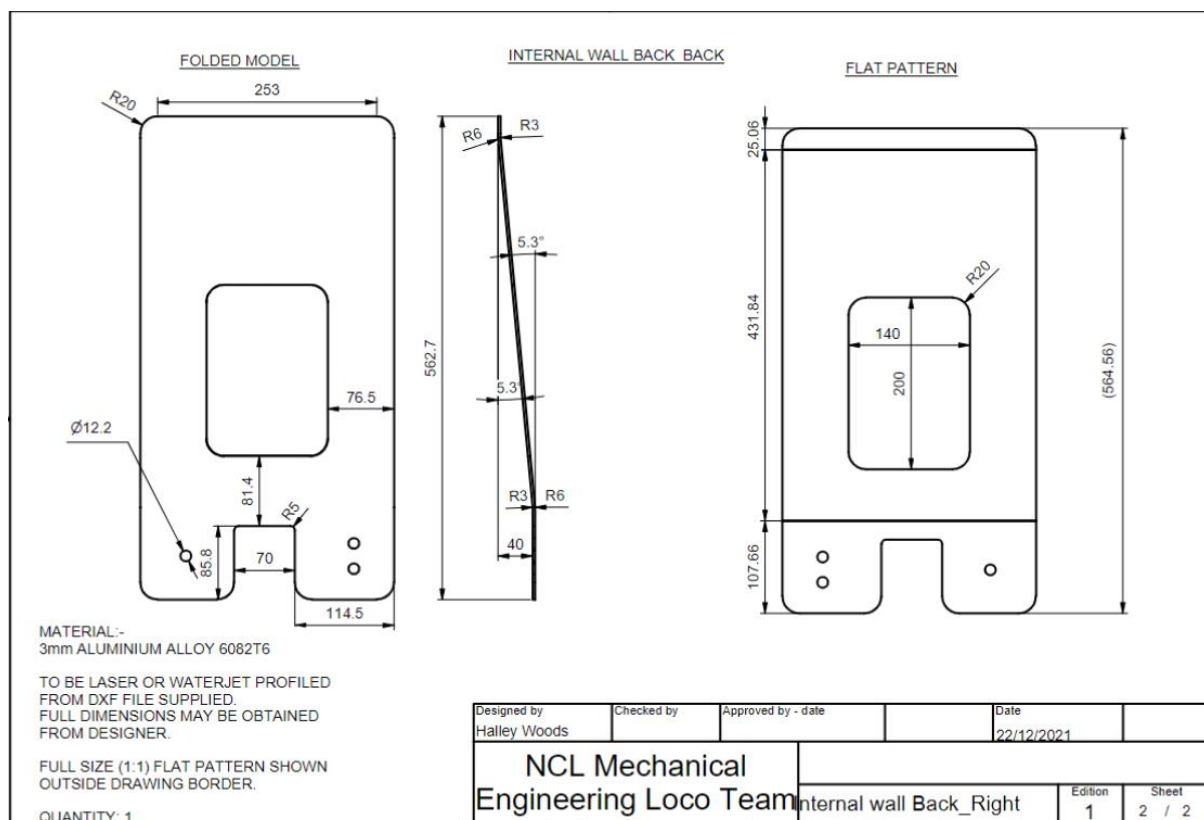
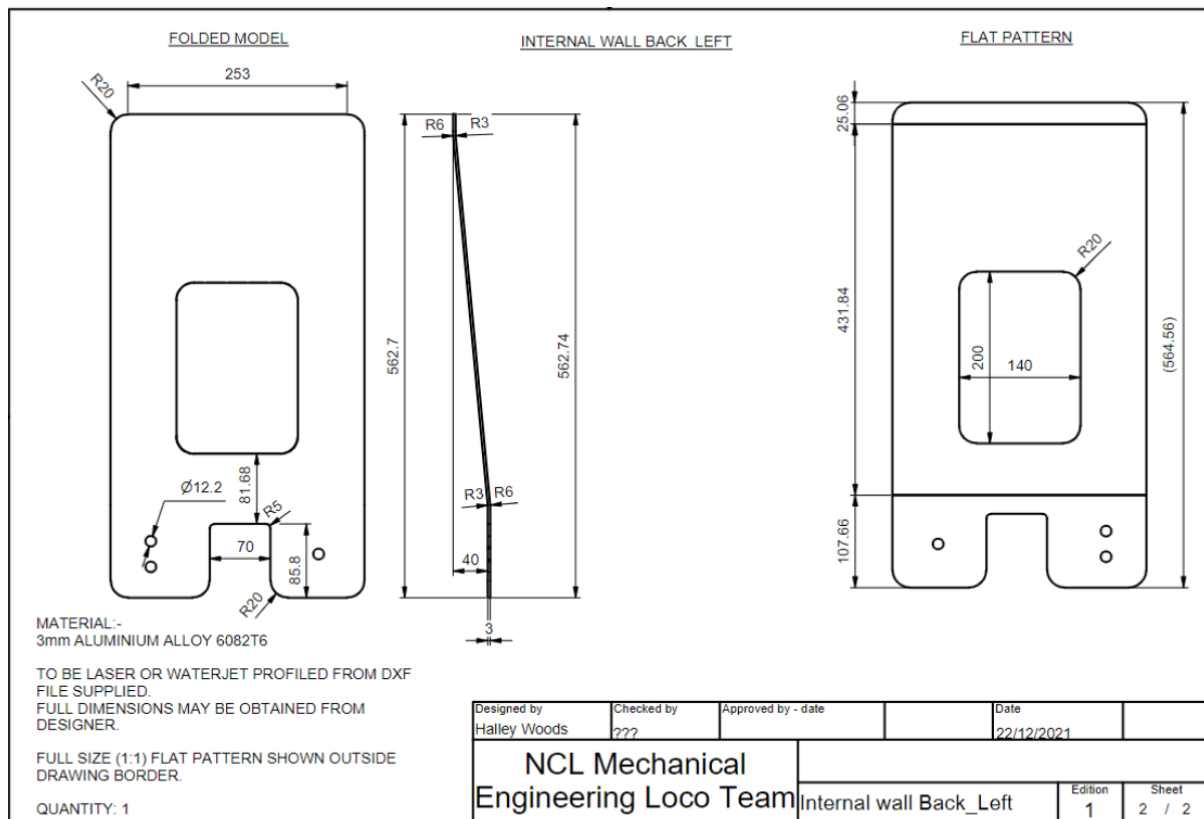


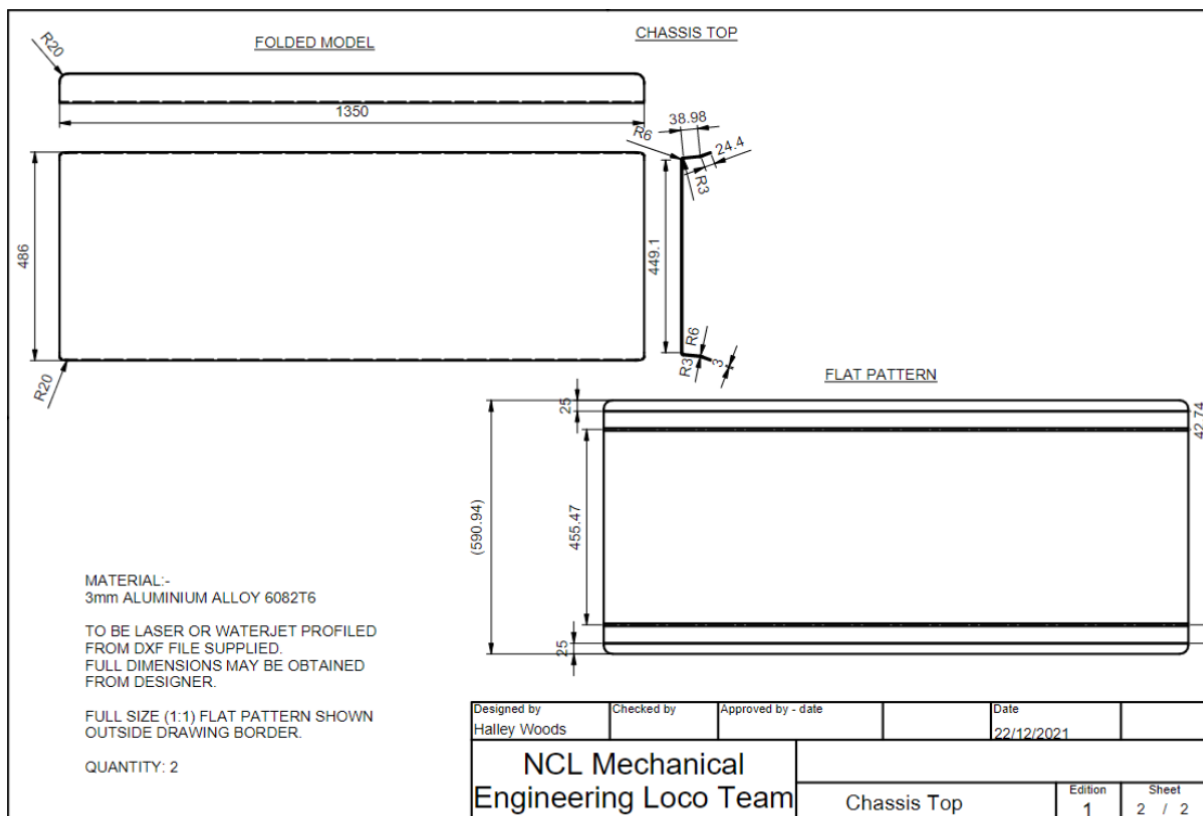
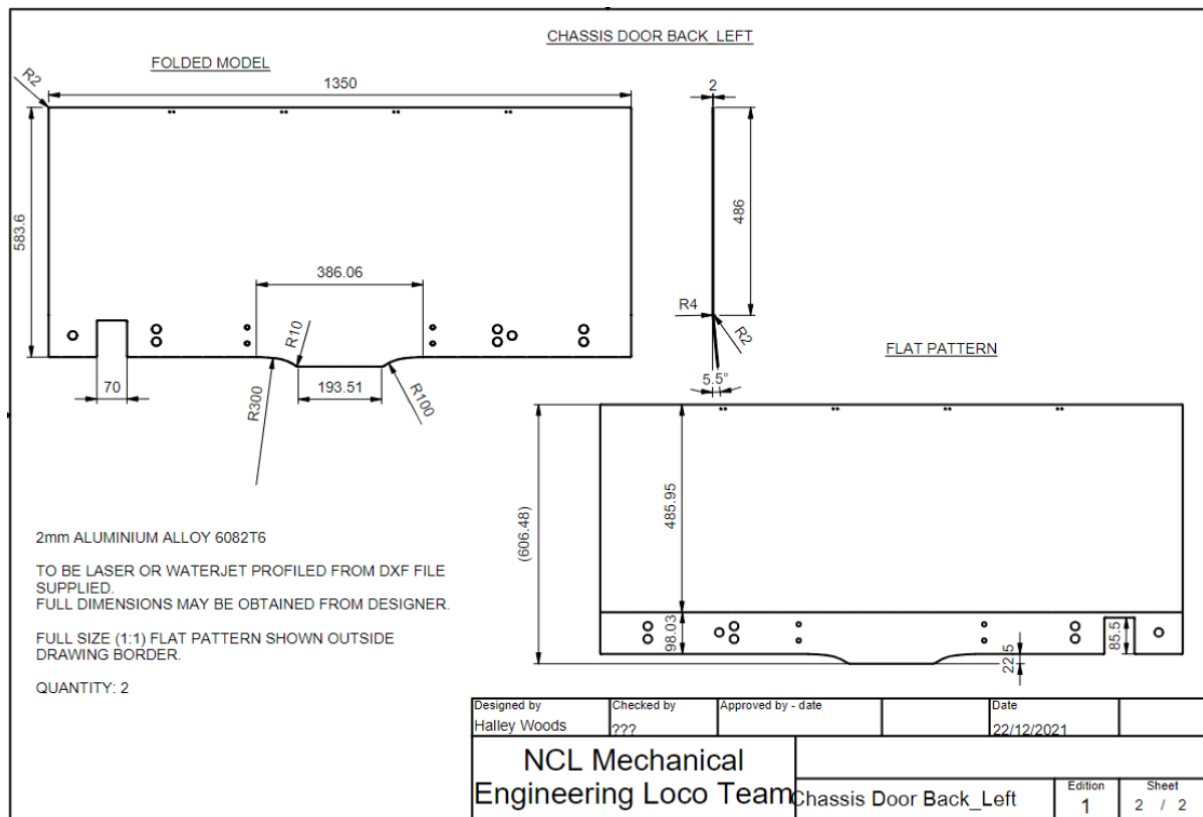


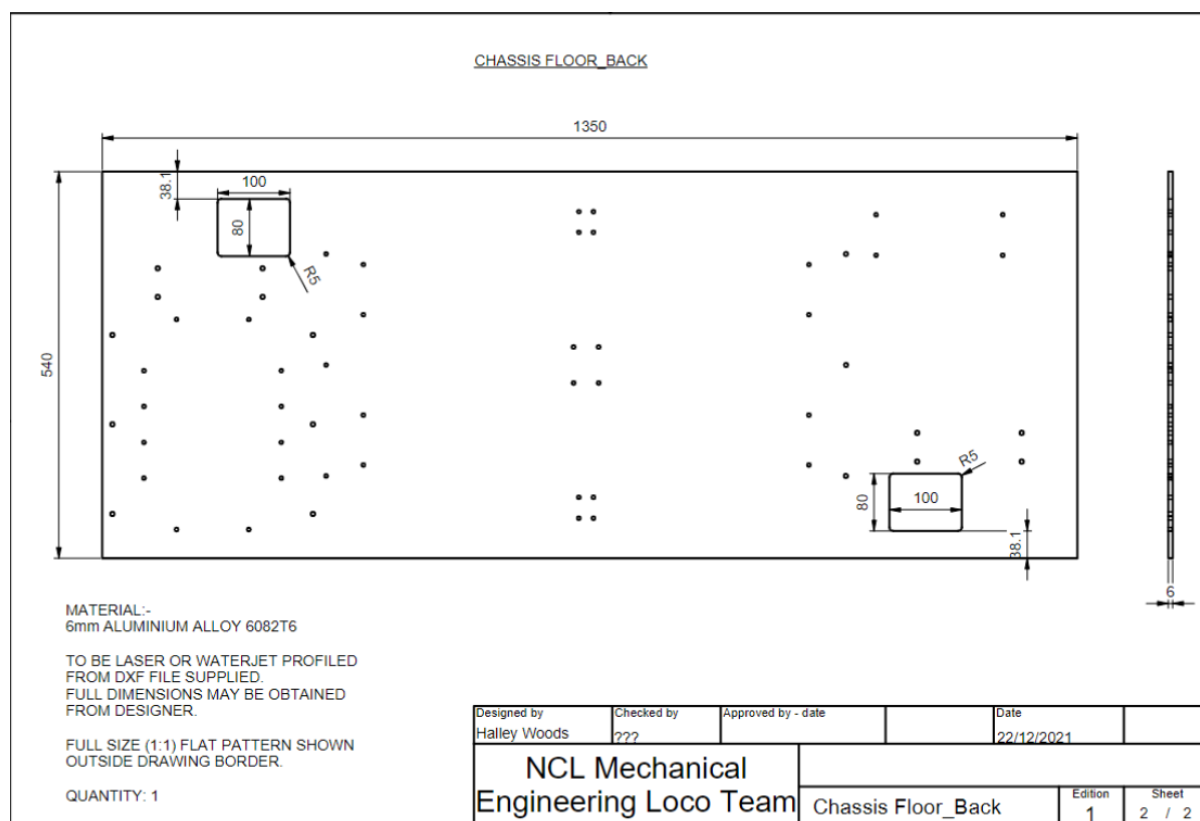
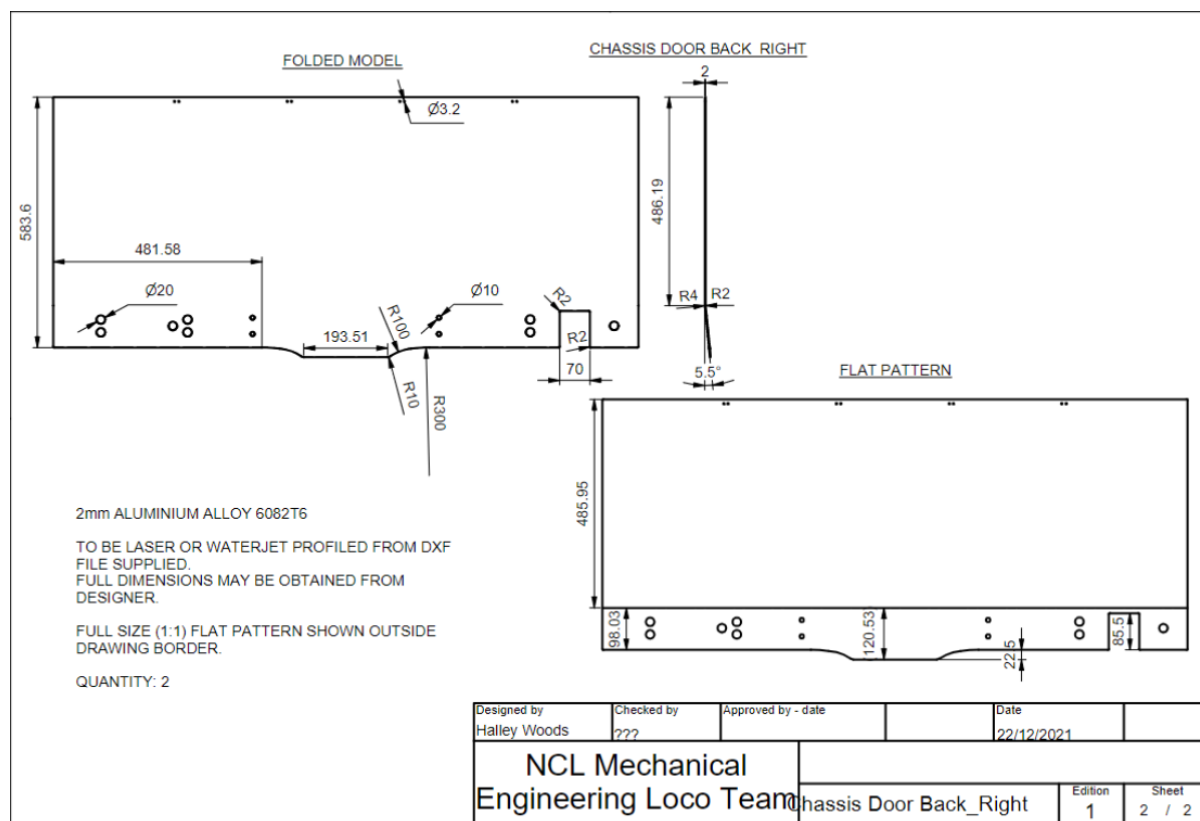


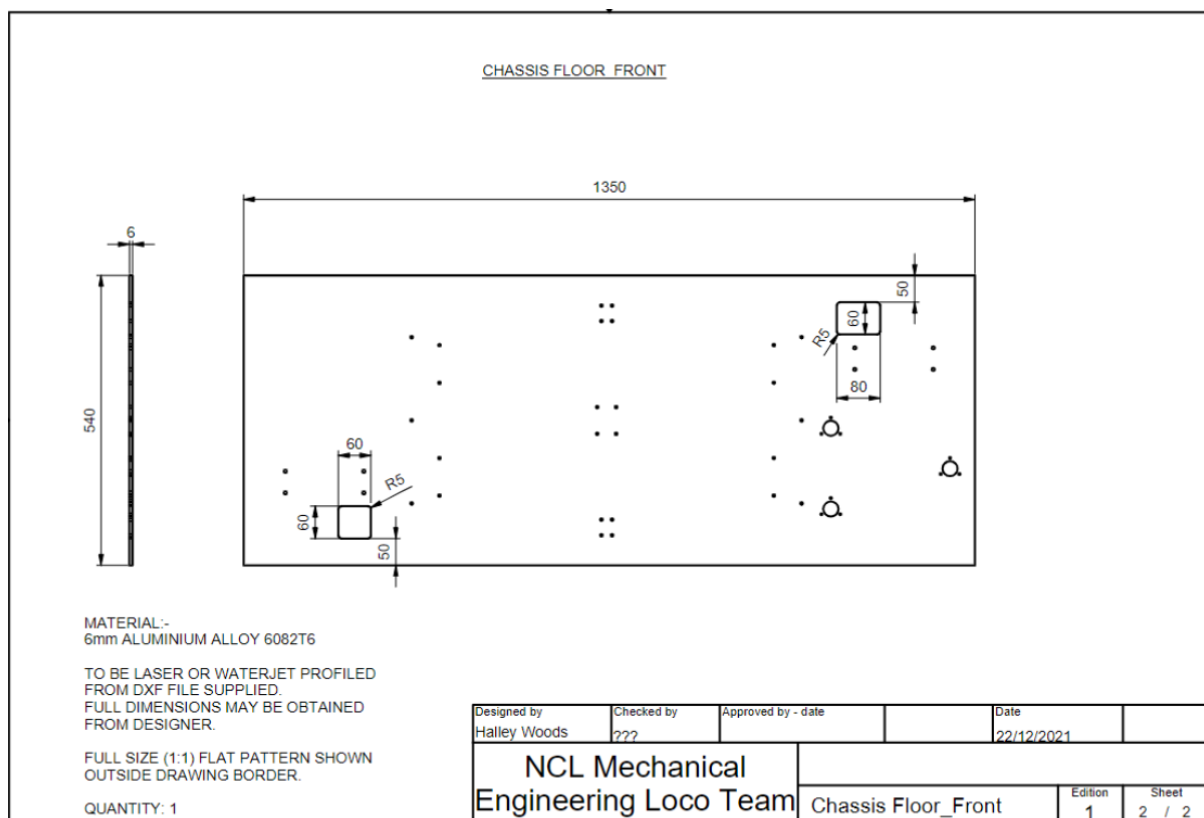
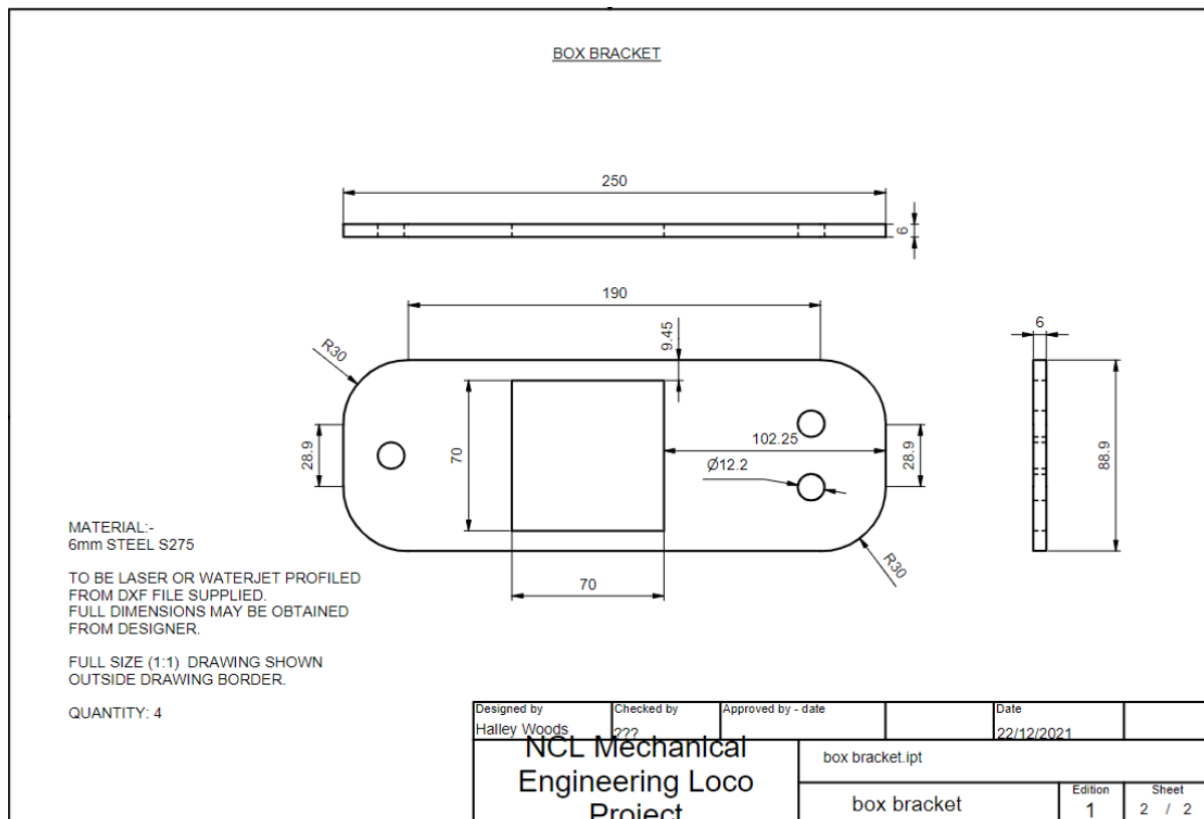




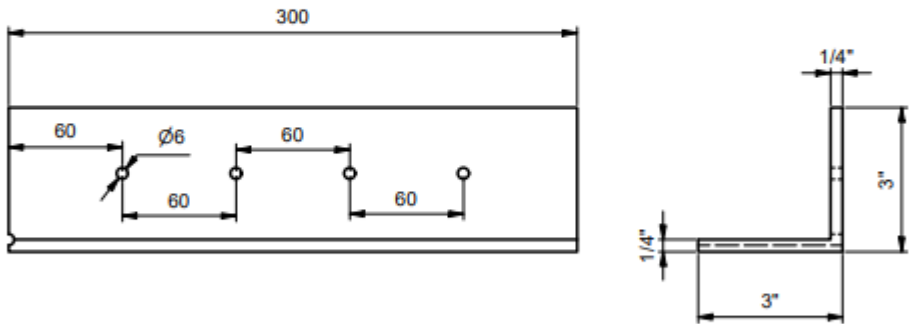






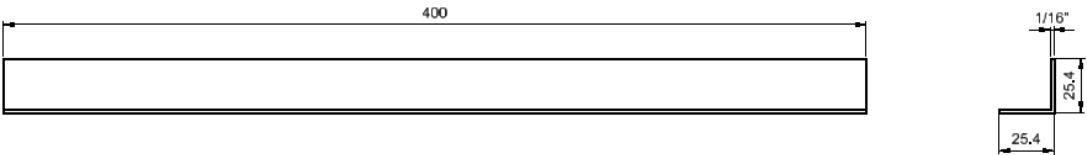


Sevcon L-bracket - 3"X3"X1/4"



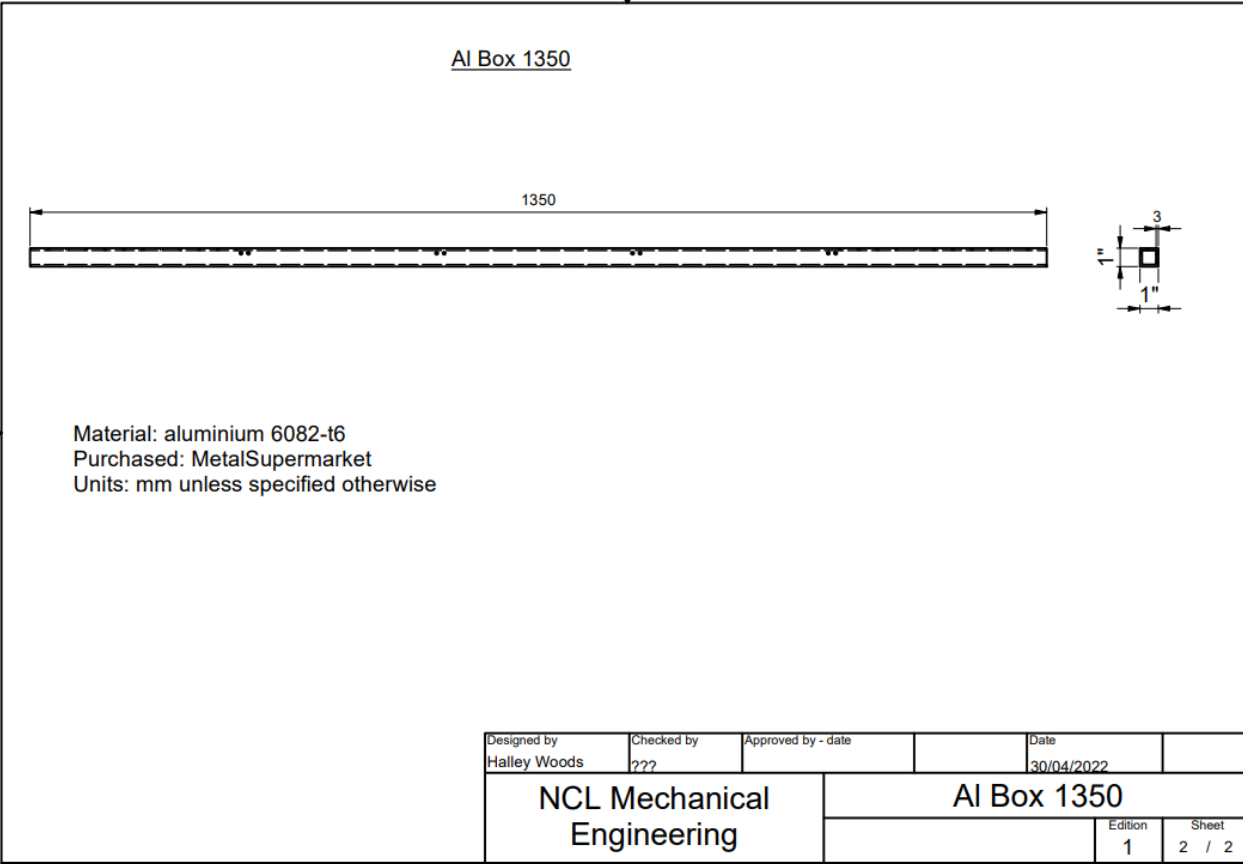
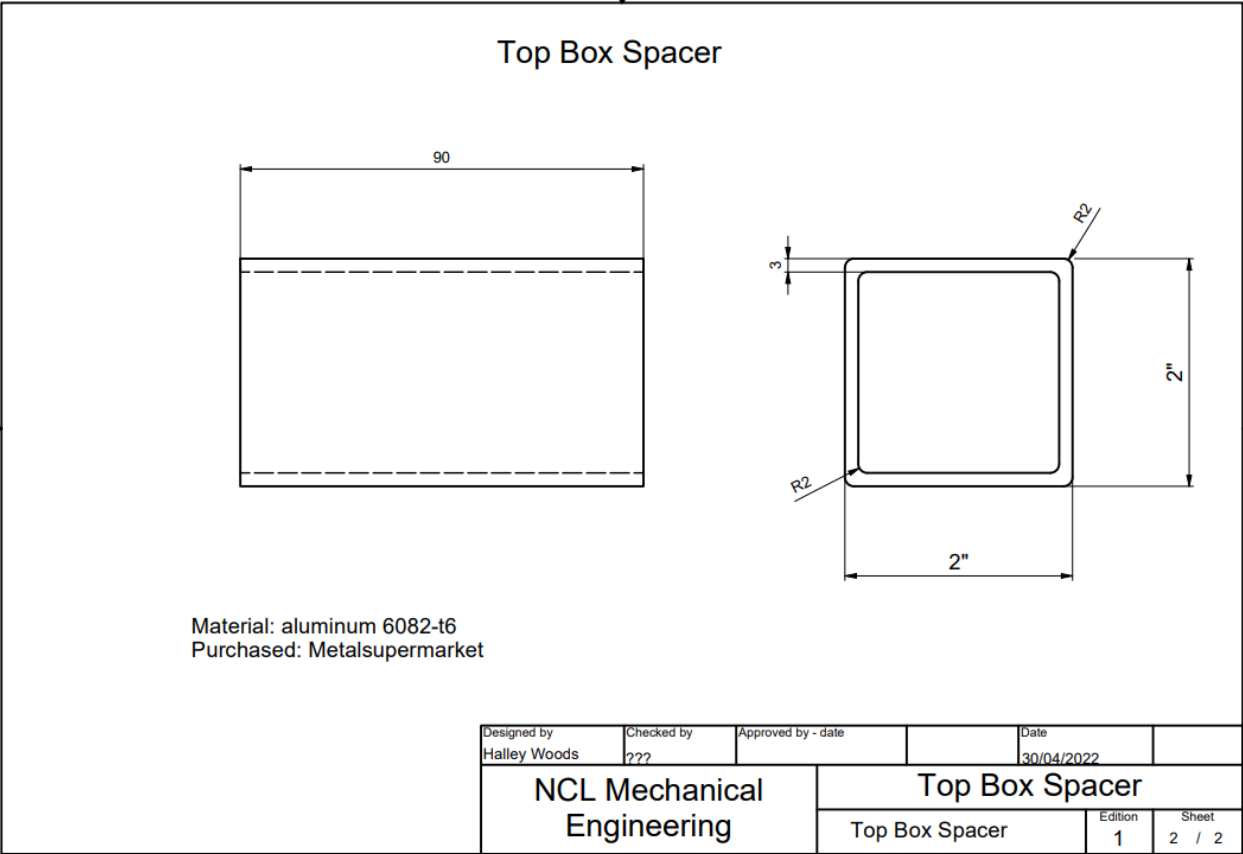
Material: 6082-t6
Purchased from: MetalSupermarket
Units: mm unless specified otherwise

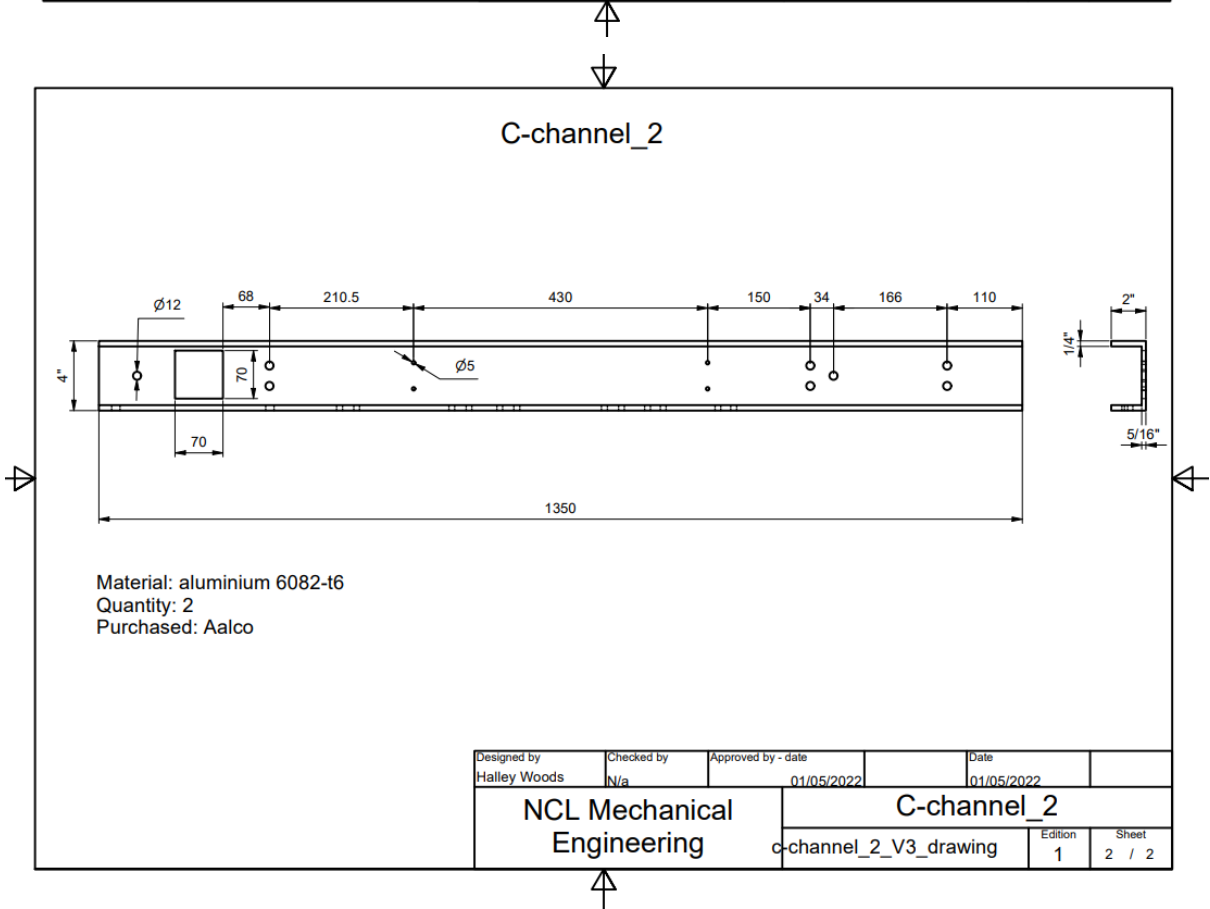
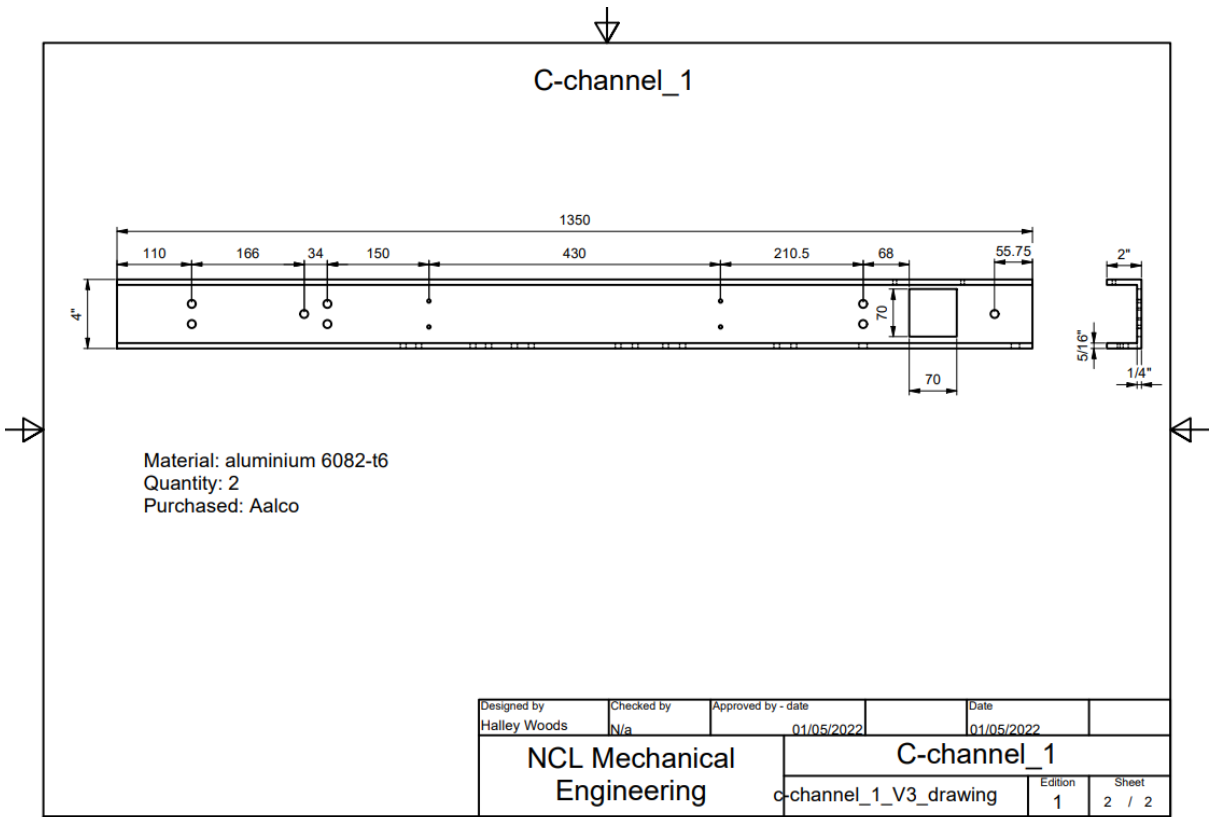
Intenal wall bracket - 1"X1"X1/16"

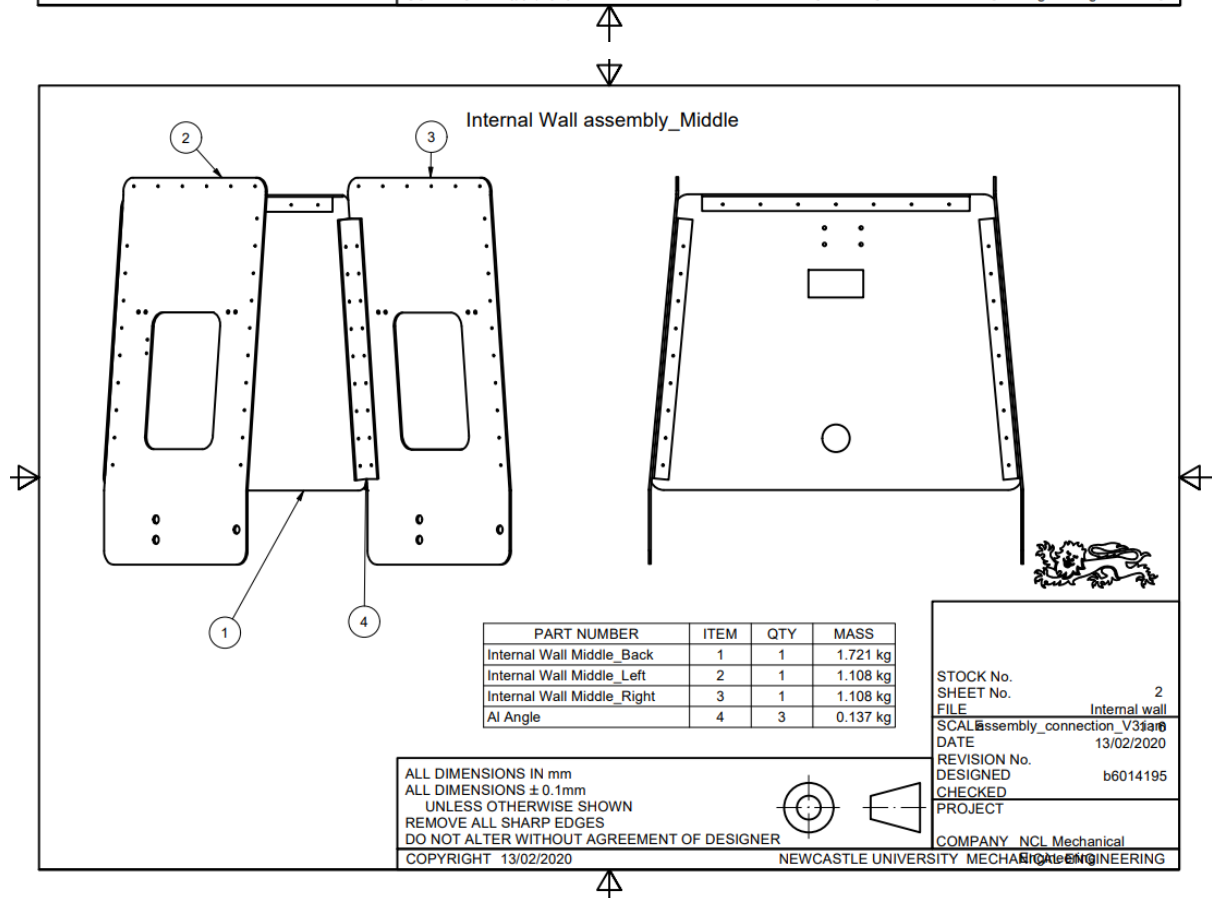
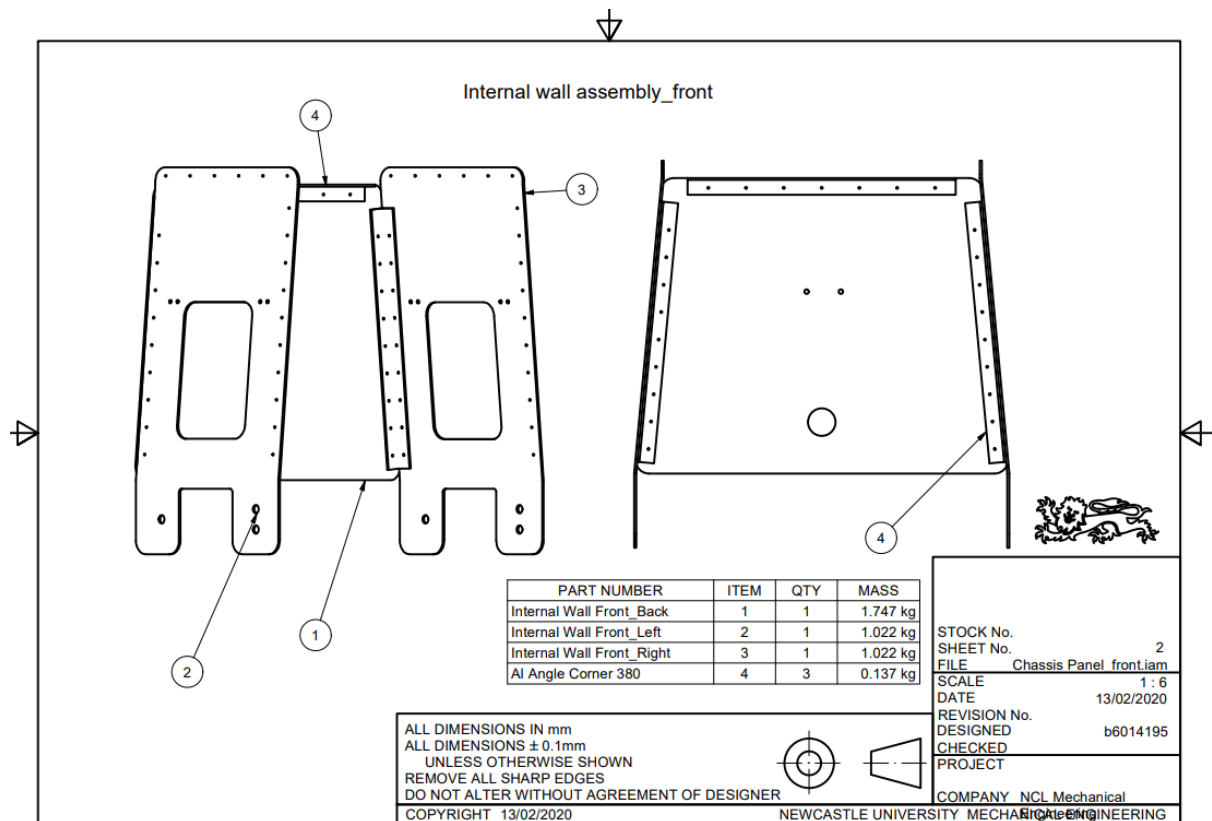


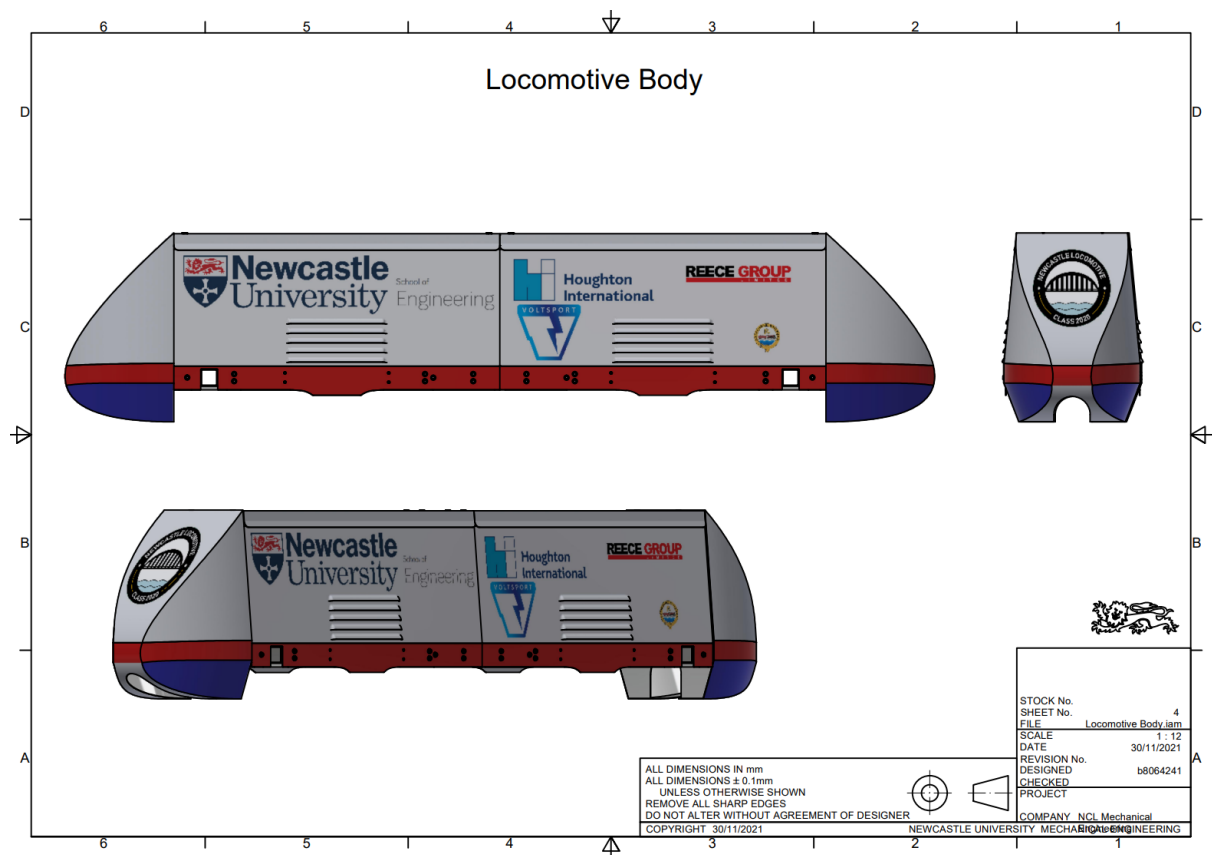
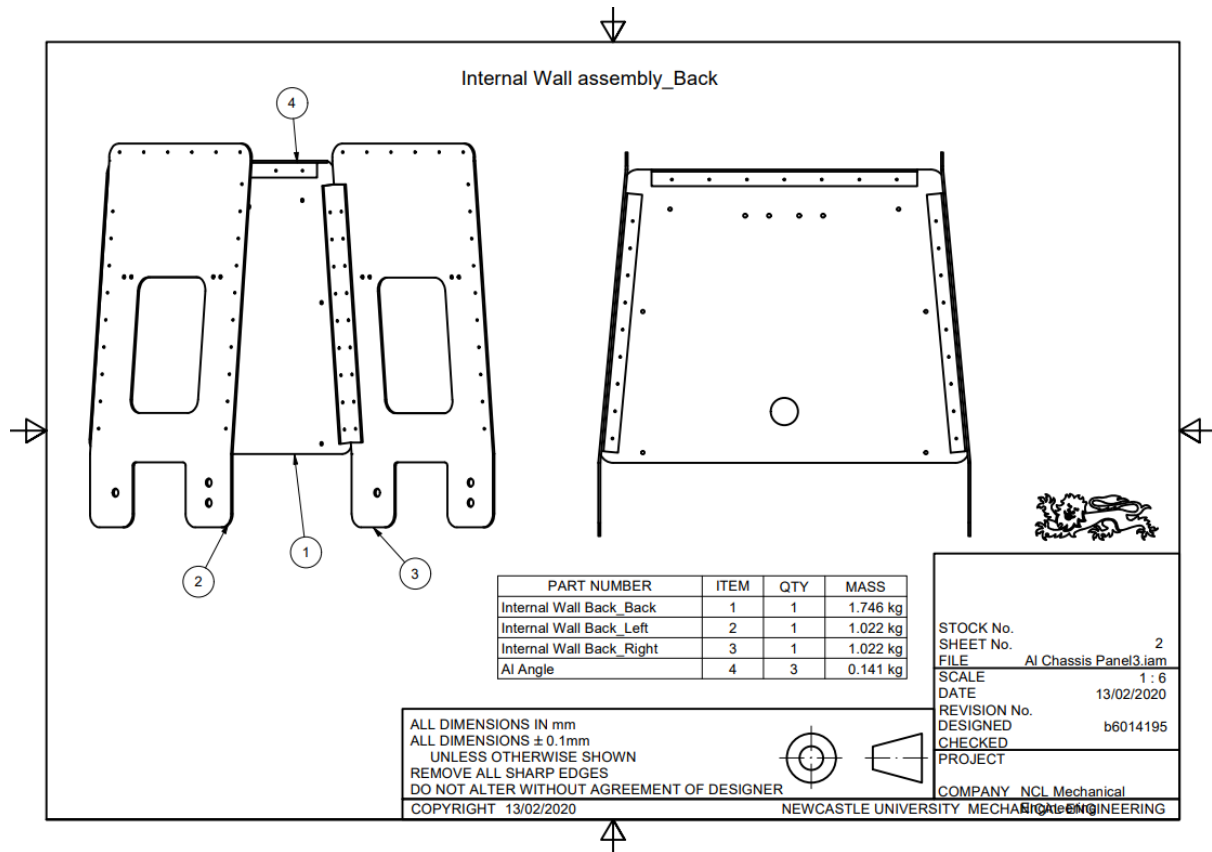
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Purchased from: MetalSupermarket
Units: mm unless specified otherwise

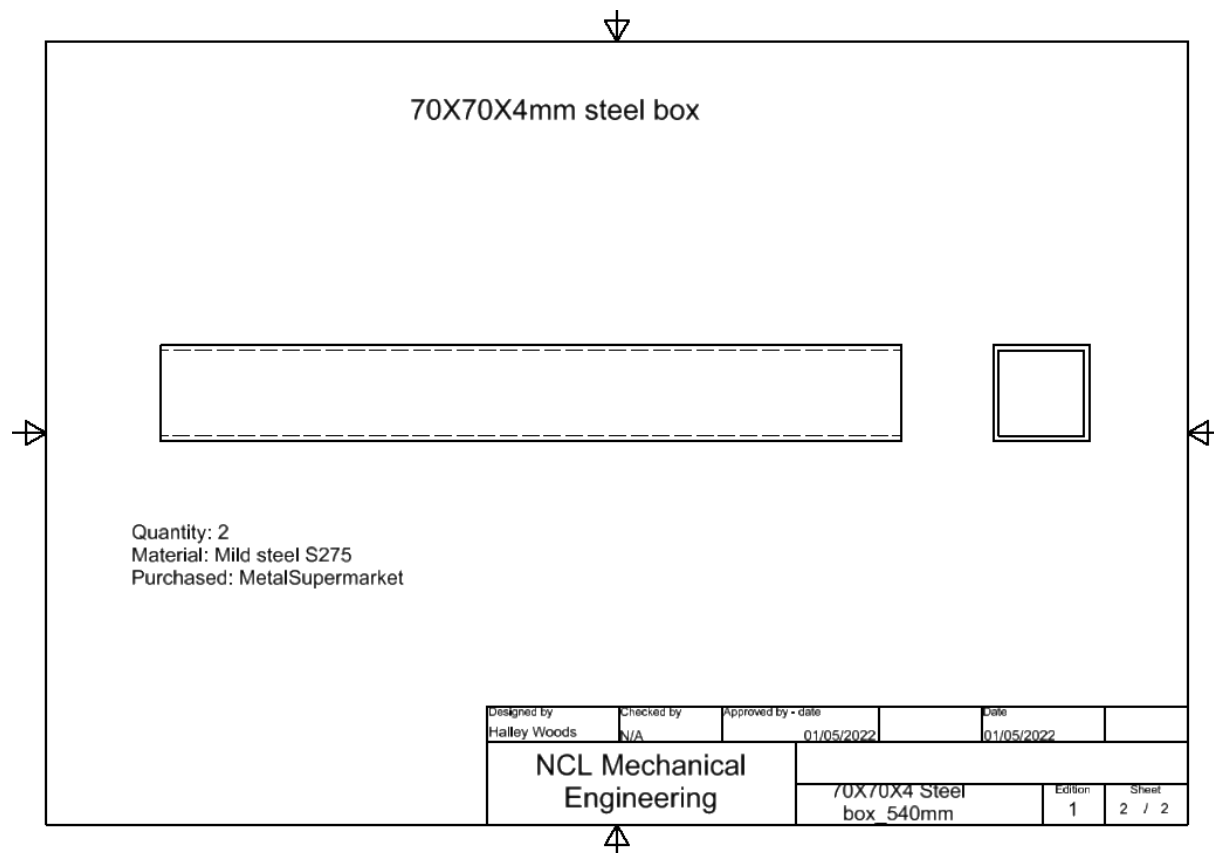
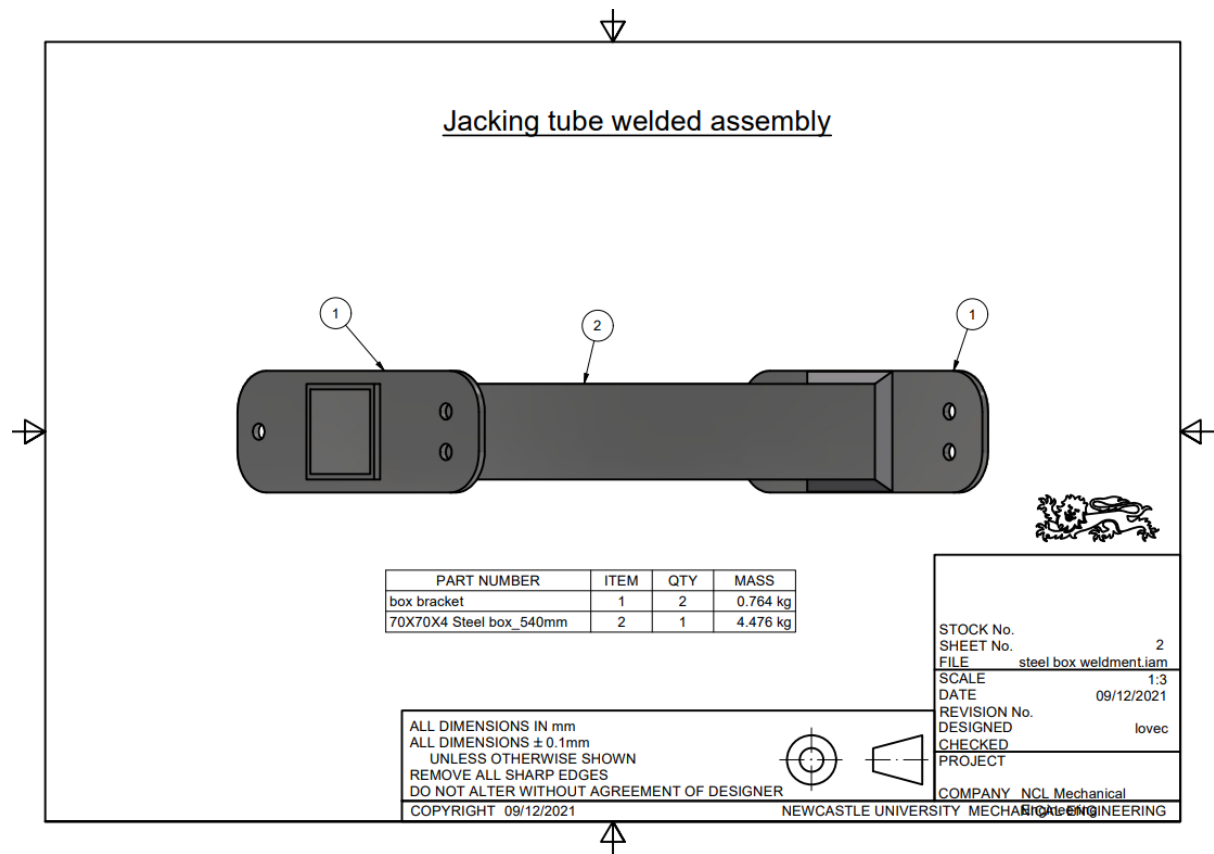
Designed by	Checked by	Approved by - date	Date	
Halley Woods	???		29/04/2022	
NCL Mechanical Engineering		Internal Wall bracket		
		Internal wall bracket	Edition 1	Sheet 2 / 2



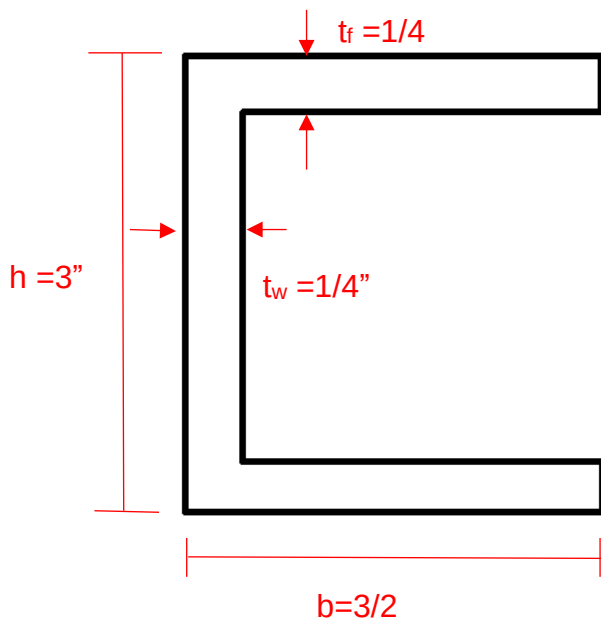








Appendix C–2: Bending strength of the large c-channels (HW)



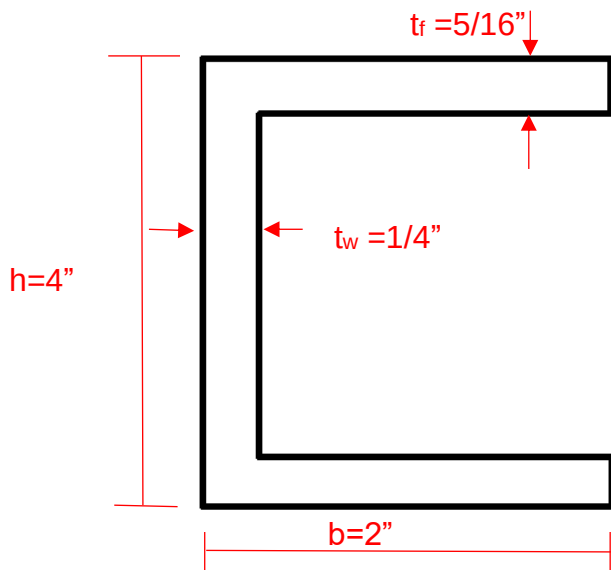
Inches	mm
3"	76.2
3/2"	38.1
1/4"	6.35

$$\begin{aligned}
 I &= \frac{bh^3}{12} - \frac{(b - t_w)(h - t_f)^3}{12} \\
 &= \frac{38.1 \times 10^{-3} \times (76.2 \times 10^{-3})^3}{12} \\
 &\quad - \frac{(38.1 \times 10^{-3} - 6.35 \times 10^{-3}) \times (76.2 \times 10^{-3} - 2 \times 6.35 \times 10^{-3})^3}{12} \\
 &= 7.275 \times 10^{-7}
 \end{aligned}$$

For Force in coupler = 100KN, Force in member = 50KN

$$\begin{aligned}
 \sigma &= \frac{my}{I} = \frac{788989 \times \frac{76.2 \times 10^{-3}}{2}}{7.275 \times 10^{-7}} = 408.2 \text{ MPa} \\
 &\geq 255 \text{ MPa} [\text{Tensile strength of aluminium 6082T6}]
 \end{aligned}$$

Therefore the c-channel will fail in bending due to forcing at the coupler and needs to be changed for a c-channel with a larger second moment of area.



Inches	mm
4"	101.6
2"	50.8
1/4"	6.35
5/16"	7.9375

$$I = \frac{bh^3}{12} - \frac{(b - t_w)(h - t_f)^3}{12}$$

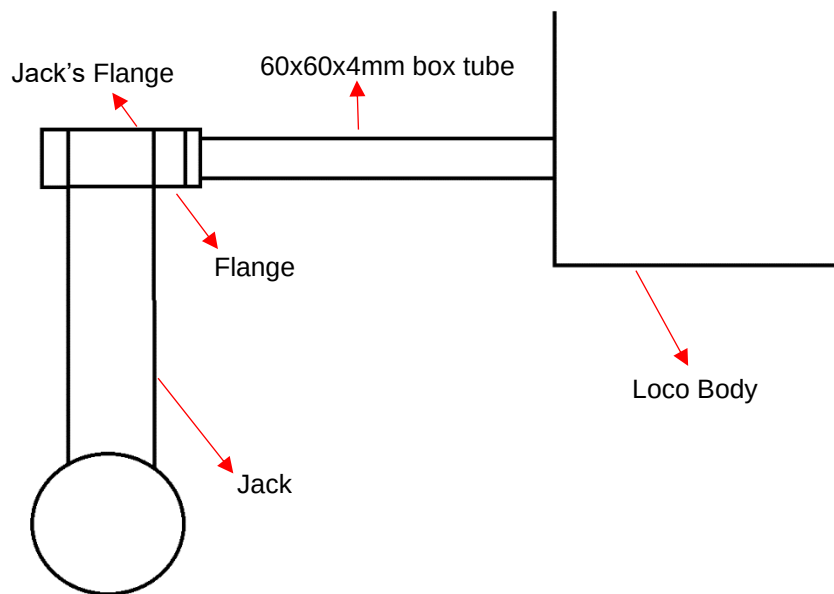
$$= \frac{50.8 \times 10^{-3} \times (101.6 \times 10^{-3})^3}{12} - \frac{(50.8 \times 10^{-3} - 7.9375 \times 10^{-3}) \times (101.6 \times 10^{-3} - 2 \times 6.35 \times 10^{-3})^3}{12}$$

$$= 1.837 \times 10^{-6}$$

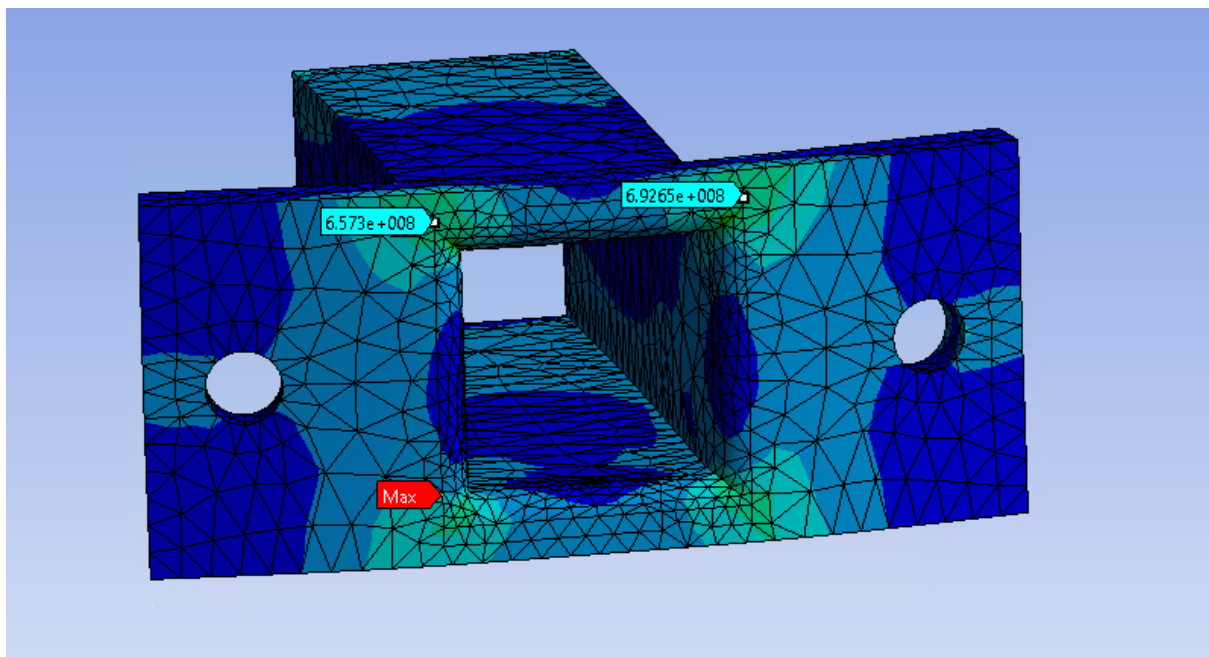
For Force in coupler =100KN, Force in member =50KN

$$\sigma = \frac{my}{I} = \frac{788989 \times \frac{101.6 \times 10^{-3}}{2}}{1.837 \times 10^{-6}} = 202.8 \text{ MPa} \leq 255 \text{ MPa} \text{ [Tensile strength of aluminium 6082T6]}$$

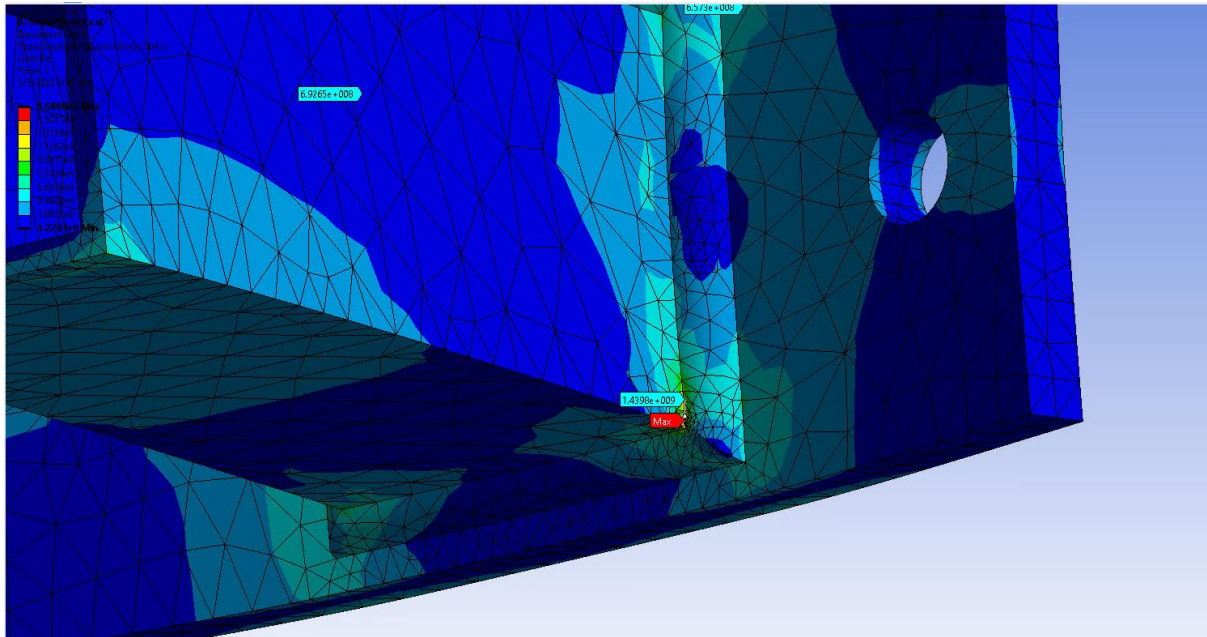
Therefore the c-channel will not fail in bending as a result of forcing at the coupler.

Appendix C-3 Jacking Gear FEA and Design Development (HW)

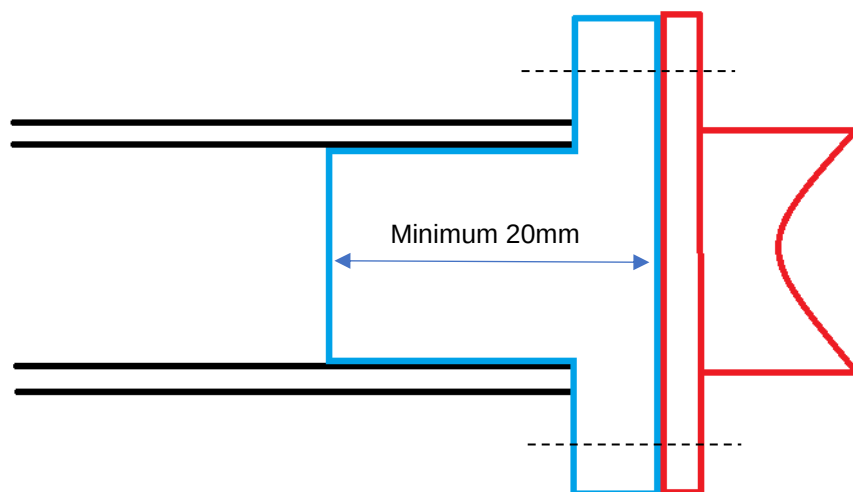
The forces on the jacking gear are the same as in section 7.5



The flange that connects to the jack's flange and box tube will fail due to the stresses exceeding the yield stress of steel at the welds. A objective of next years team should be to develop a flange that will not fail due to the stresses it undergoes.



A design which may could be investigated for feasibility is as below:



Appendix C-4: Aluminium 6082-t6 flat bar aalco purchase quote (HW)A

Page 1 of 1



Aalco Newcastle
First Avenue
Team Valley Trading Estate
Gateshead
Tyne and Wear
NE11 0NU
United Kingdom

Telephone: 0191 491 1133
Fax: 0191 491 1177

VAT Registration No. GB 711 1155 91

Sales Quote SQ0131627NE

Deliver To
NEWCASTLE UNIVERSITY
KINGS GATE
KINGS ROAD
NEWCASTLE UPON TYNE
NE1 7RU
United Kingdom

Invoice To
NEWCASTLE UNIVERSITY
KINGS GATE
KINGS ROAD
NEWCASTLE UPON TYNE
NE1 7RU
United Kingdom

Quote No.	Customer	Contact	Salesperson	Issue Date	Currency
SQ0131627NE	NEW51	Halley Woods	Charlie Dixon	08/02/2022	GBP
Your Reference			Terms		
			30 Days End of Month		

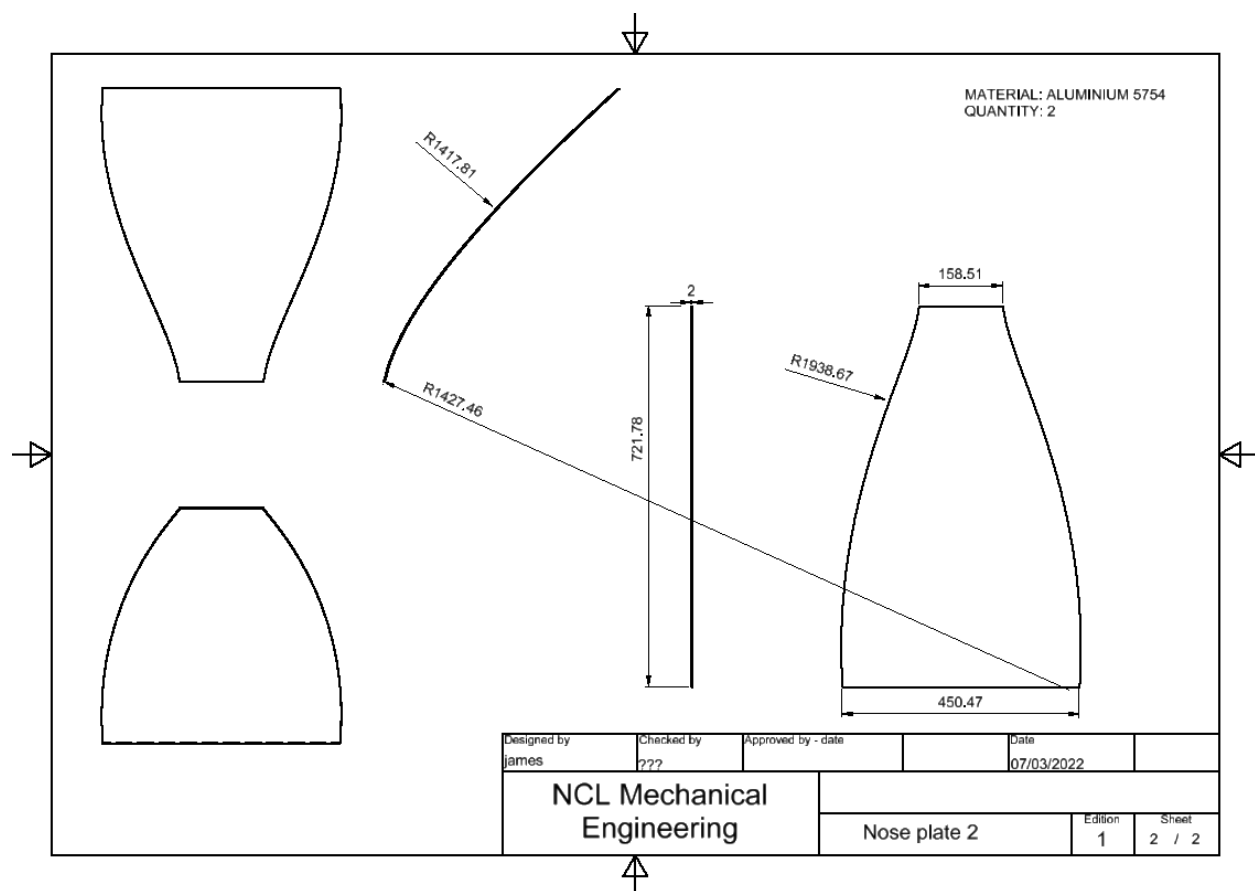
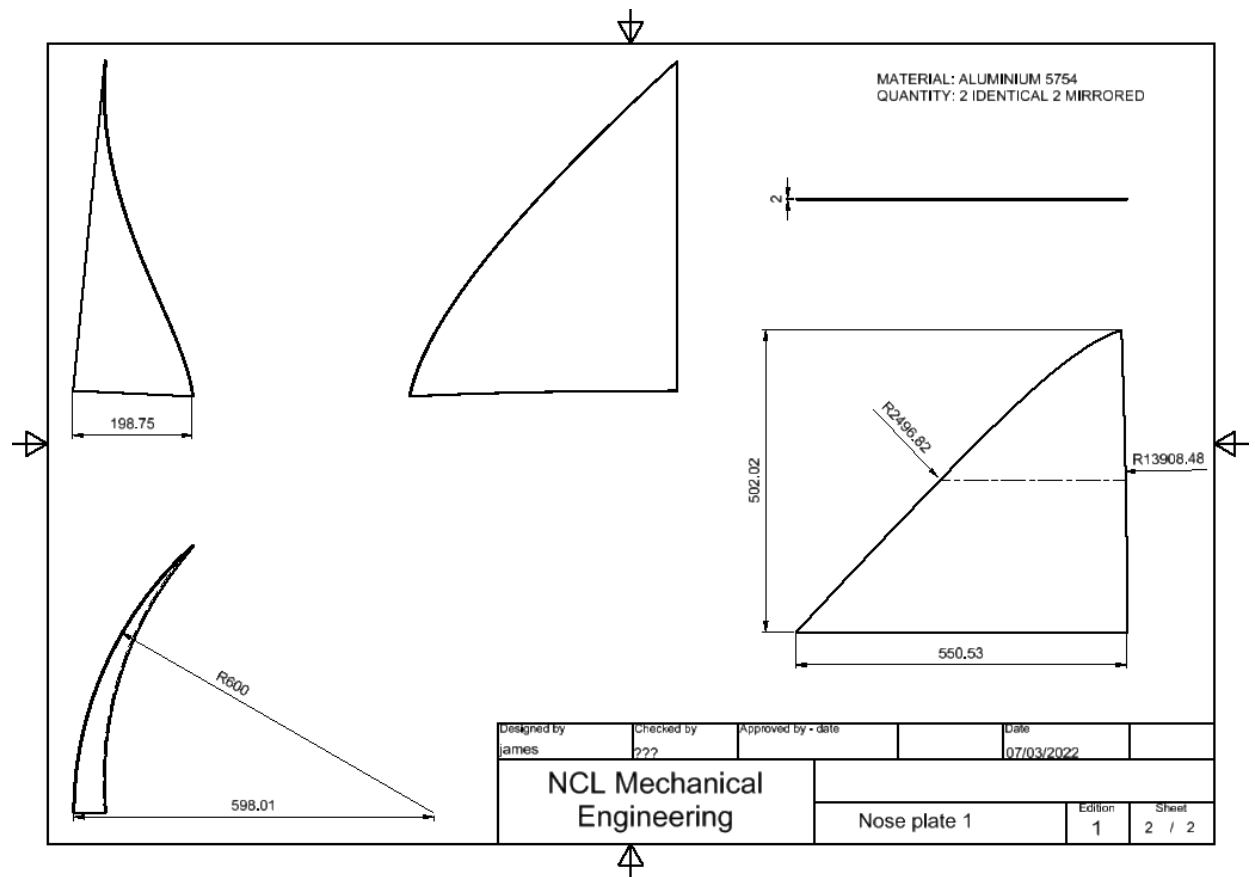
Line #	Quantity	Description	Price Quantity	Price GBP	Value GBP
1 017987NE	2 pcs	AL FLAT 6082T6 5Mtr 3.1/2" X 1.1/2" Processed to 800.0 mm EN 755-1:2016 2pcs 800mm long 4 days delivery. Prices ex vat	2 pcs	156.6125	313.23
	1 lot	Freight	1 lot	30.00	30.00
TOTAL:					343.23

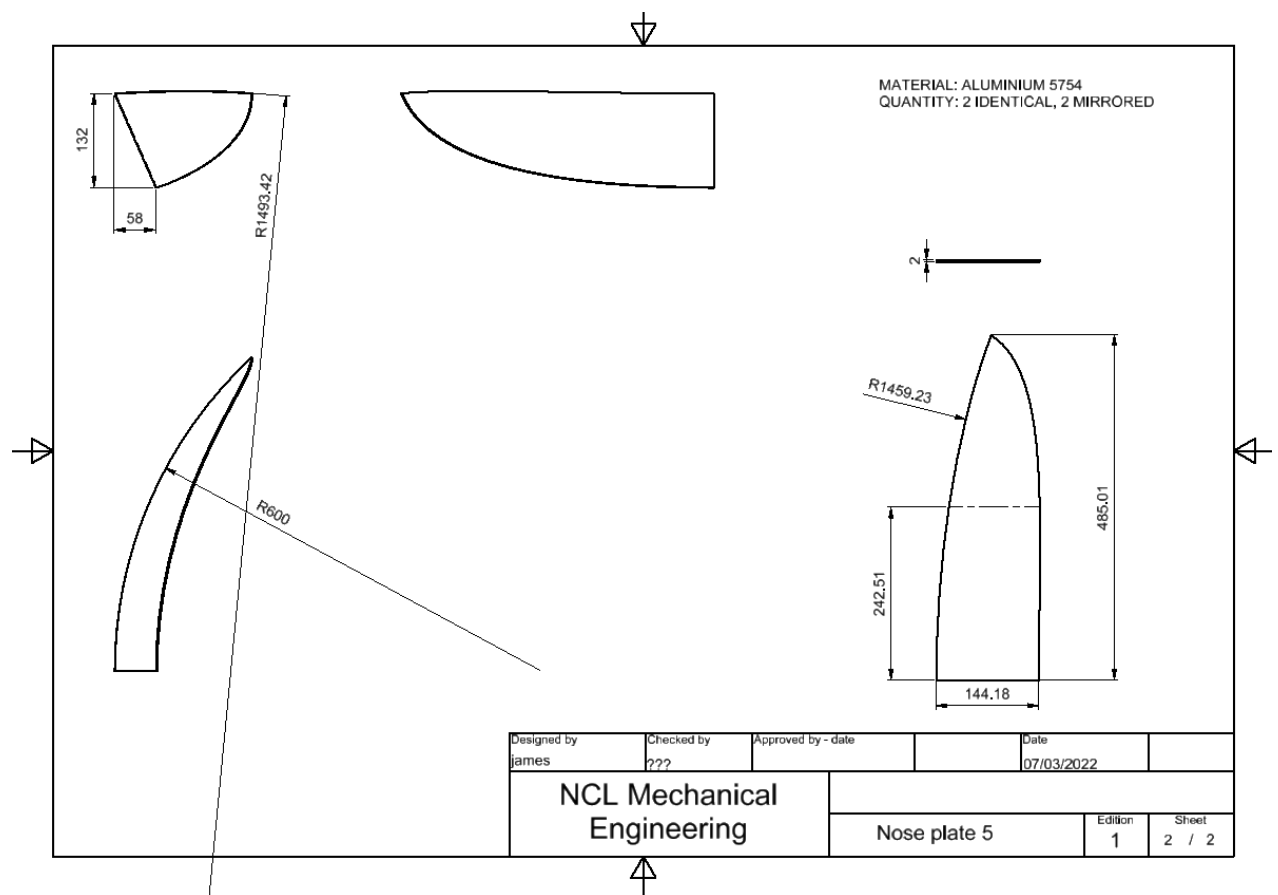
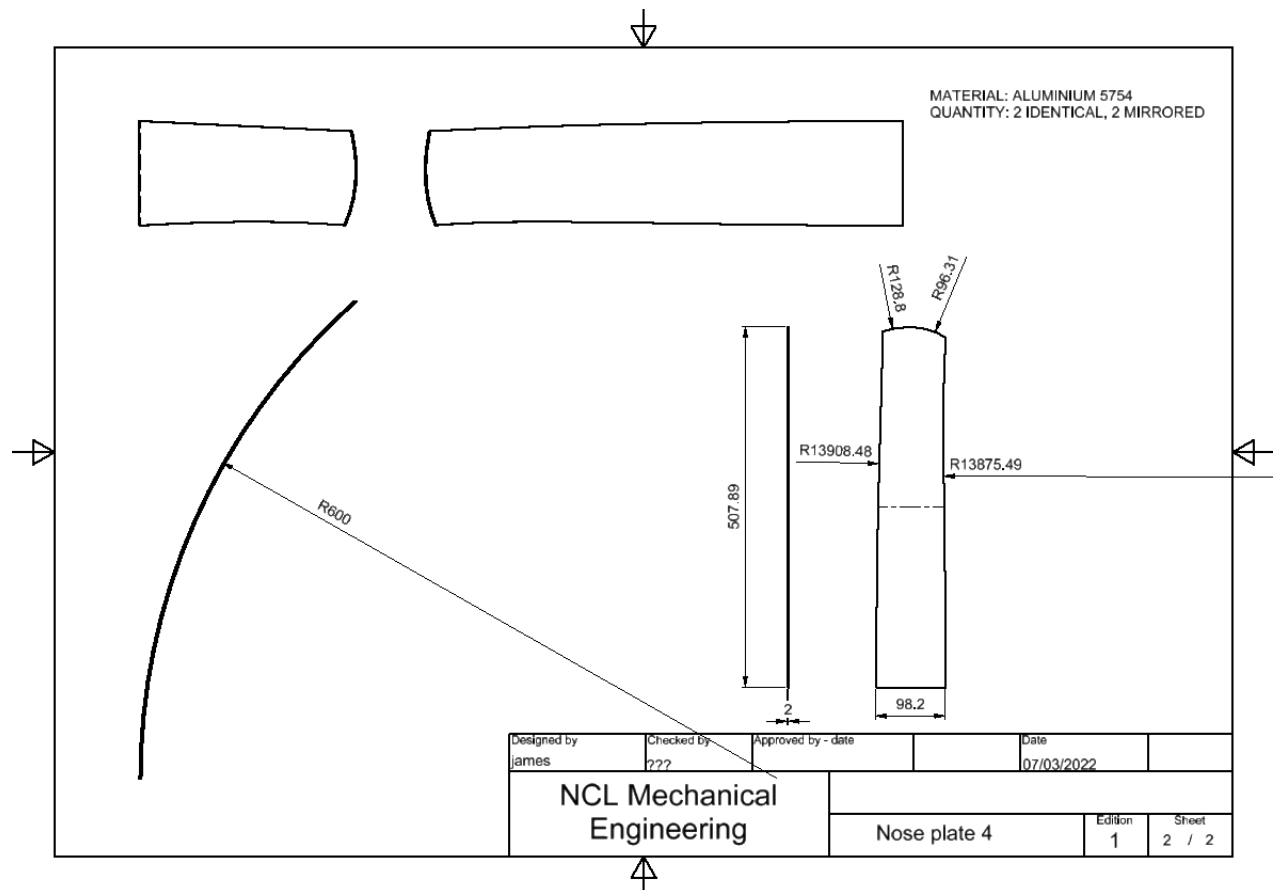
This contract is subject to our Conditions of Sale which are available on request or by visiting our web site (<http://www.aalco.co.uk>).
Prices quoted are based on the total quantities being purchased. This quotation remains subject to the daily metal price and exchange rate fluctuation meaning prices may change at time of order. Prices detailed are exclusive of VAT.
References to "The Company" overleaf refers to Aalco Metals Ltd.

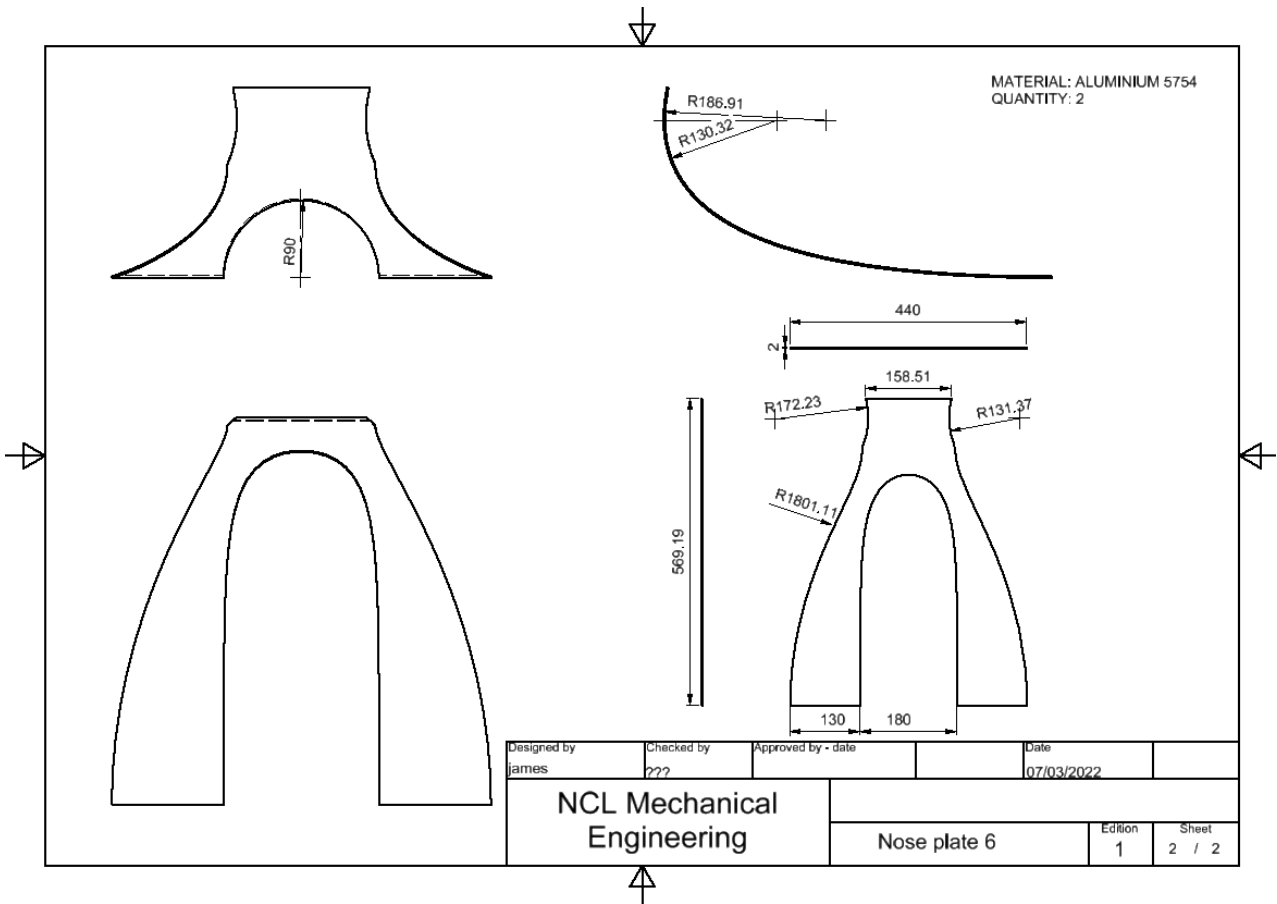
All orders are subject to VAT except for export orders, where proof of export will be required. Should you fail to provide us with your shipping documents and proof of material receipt you will be liable to the UK VAT applicable to this order.

Aalco Metals Ltd is a company registered in England No. 3551533
Registered Office: Parkway House, Unit 6, Parkway Industrial Estate, Pacific Avenue, Wednesbury, WS10 7WP

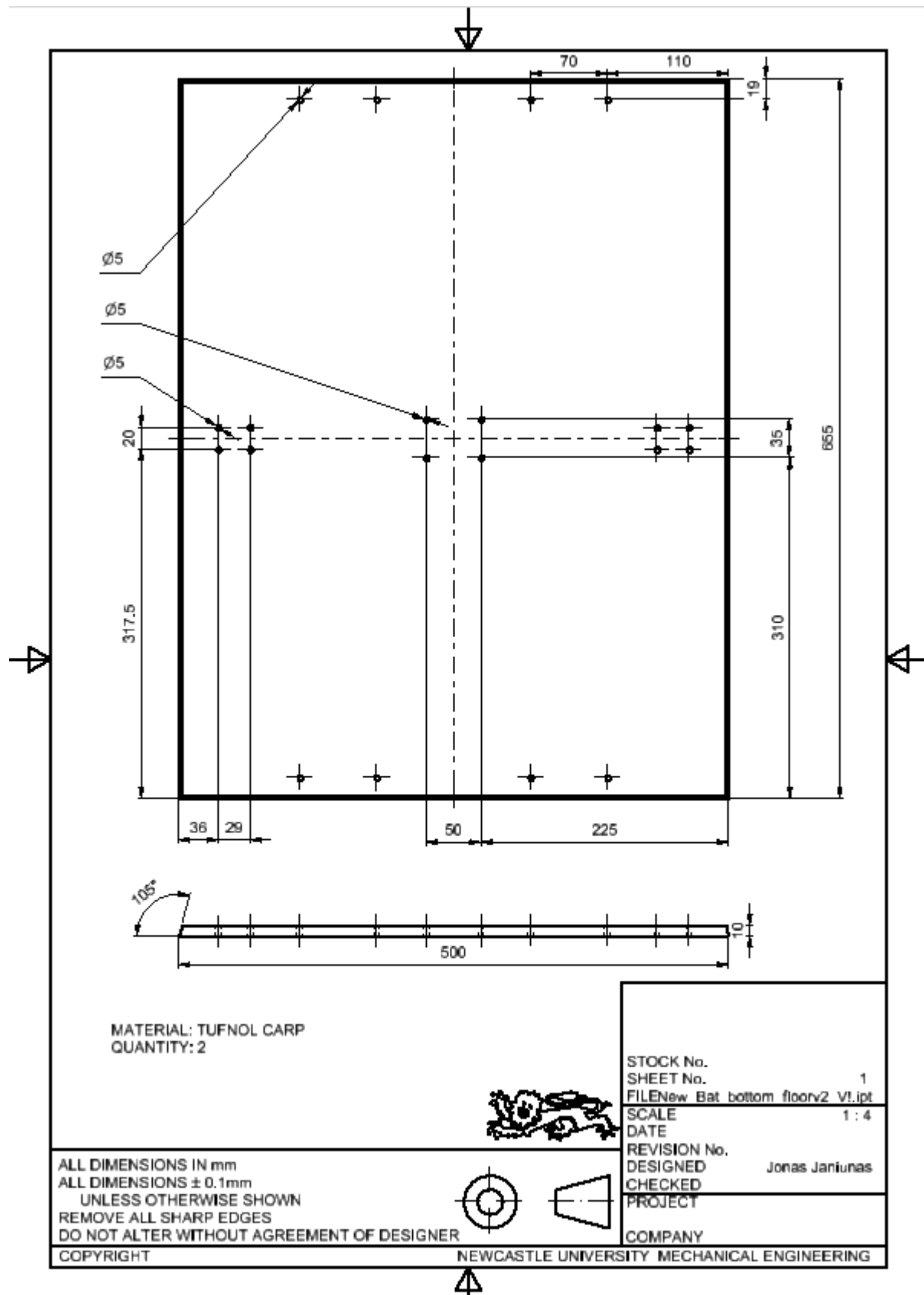
Appendix C-5: Nose Cone Drawings (JO)

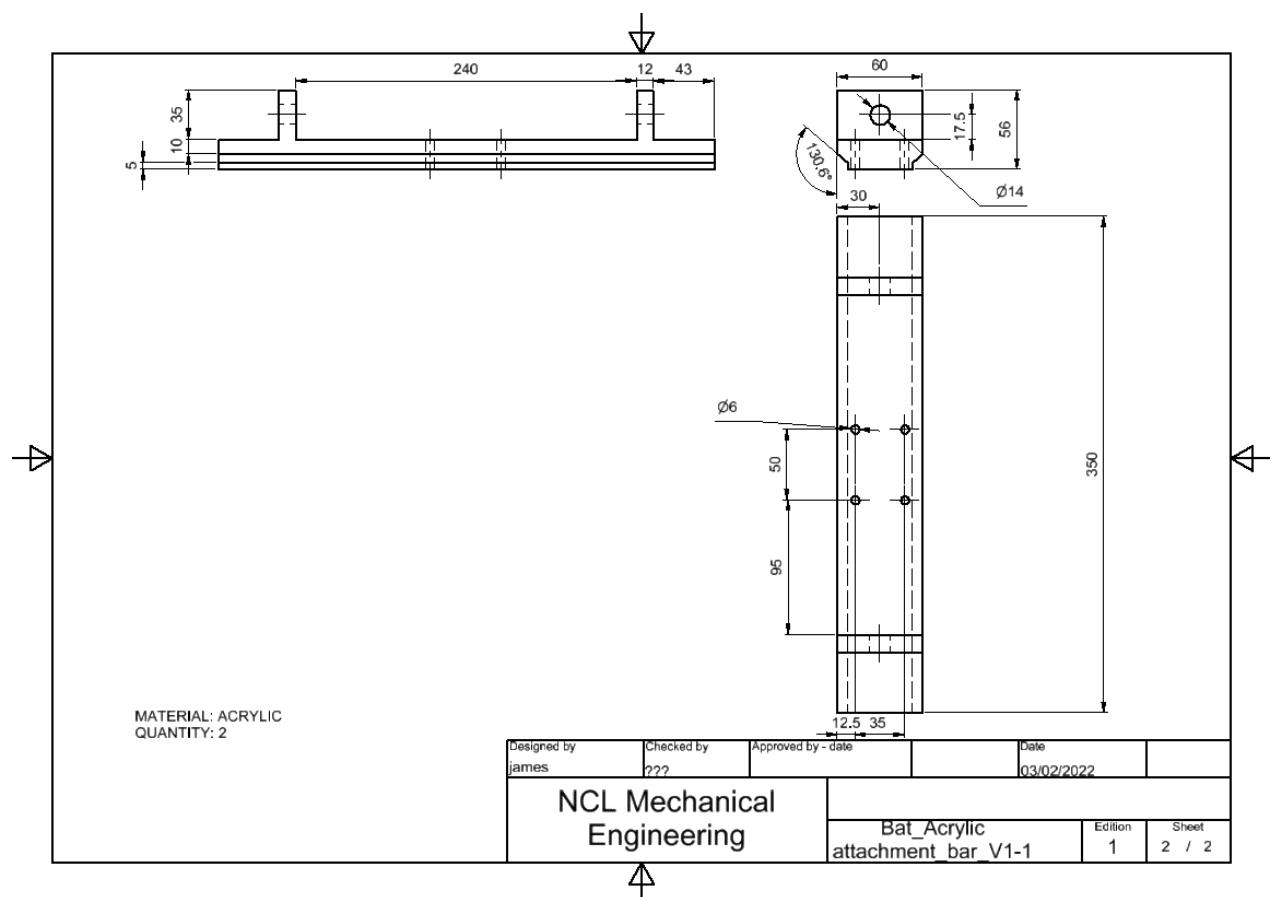
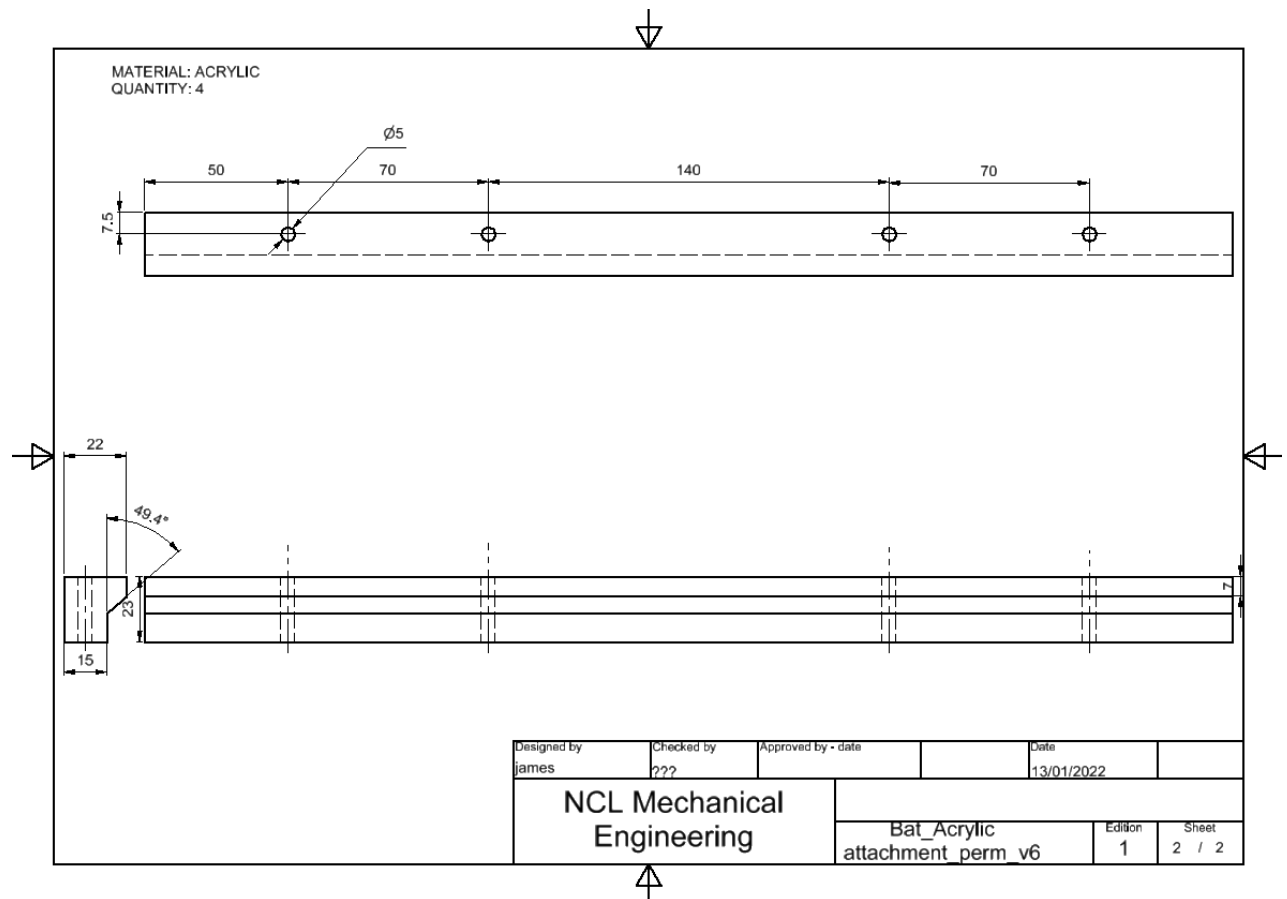


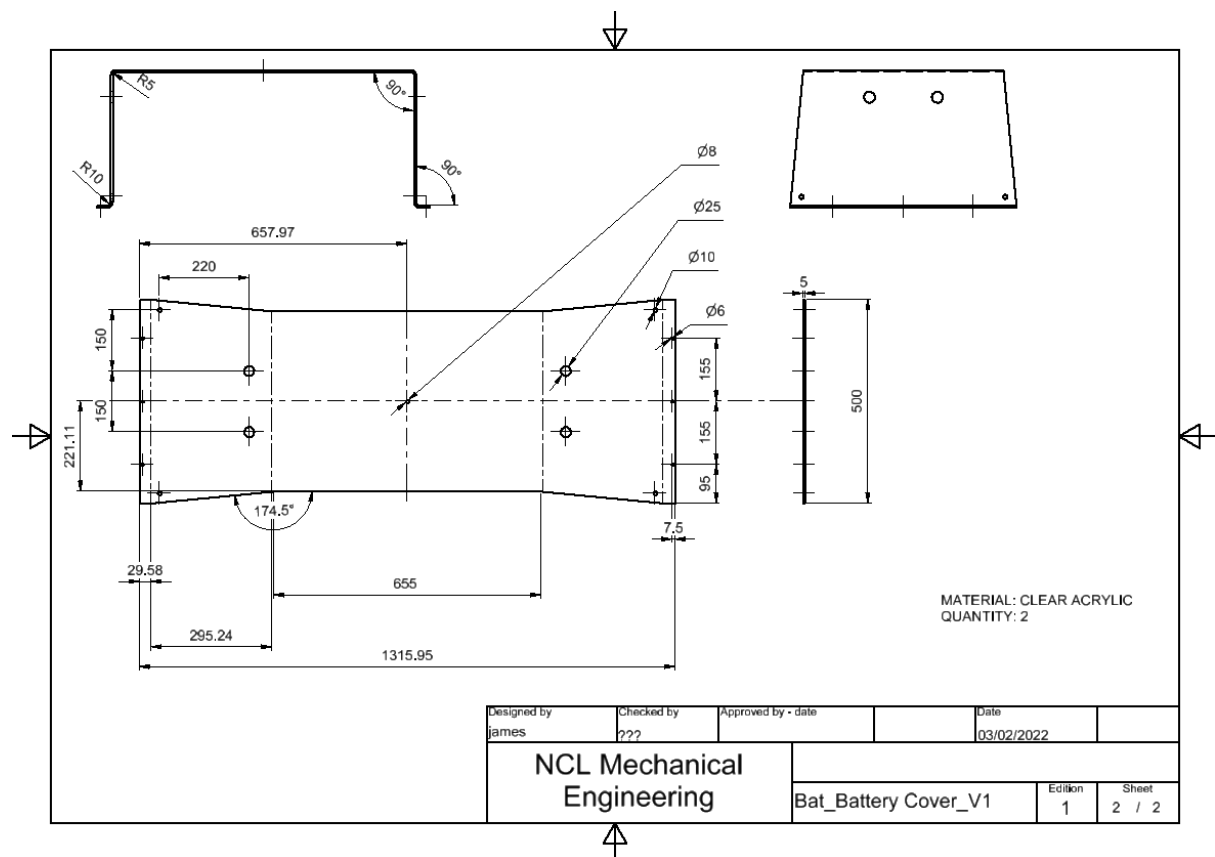
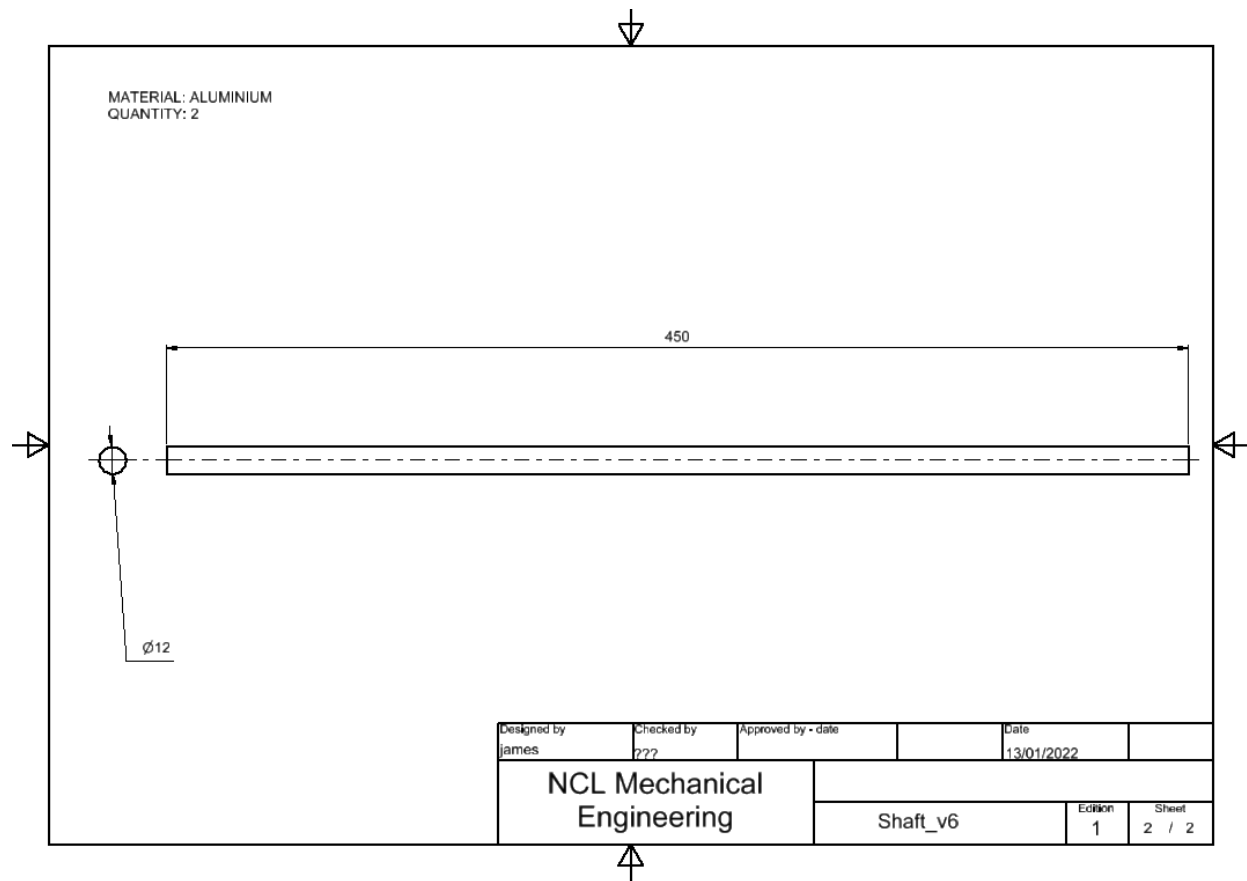


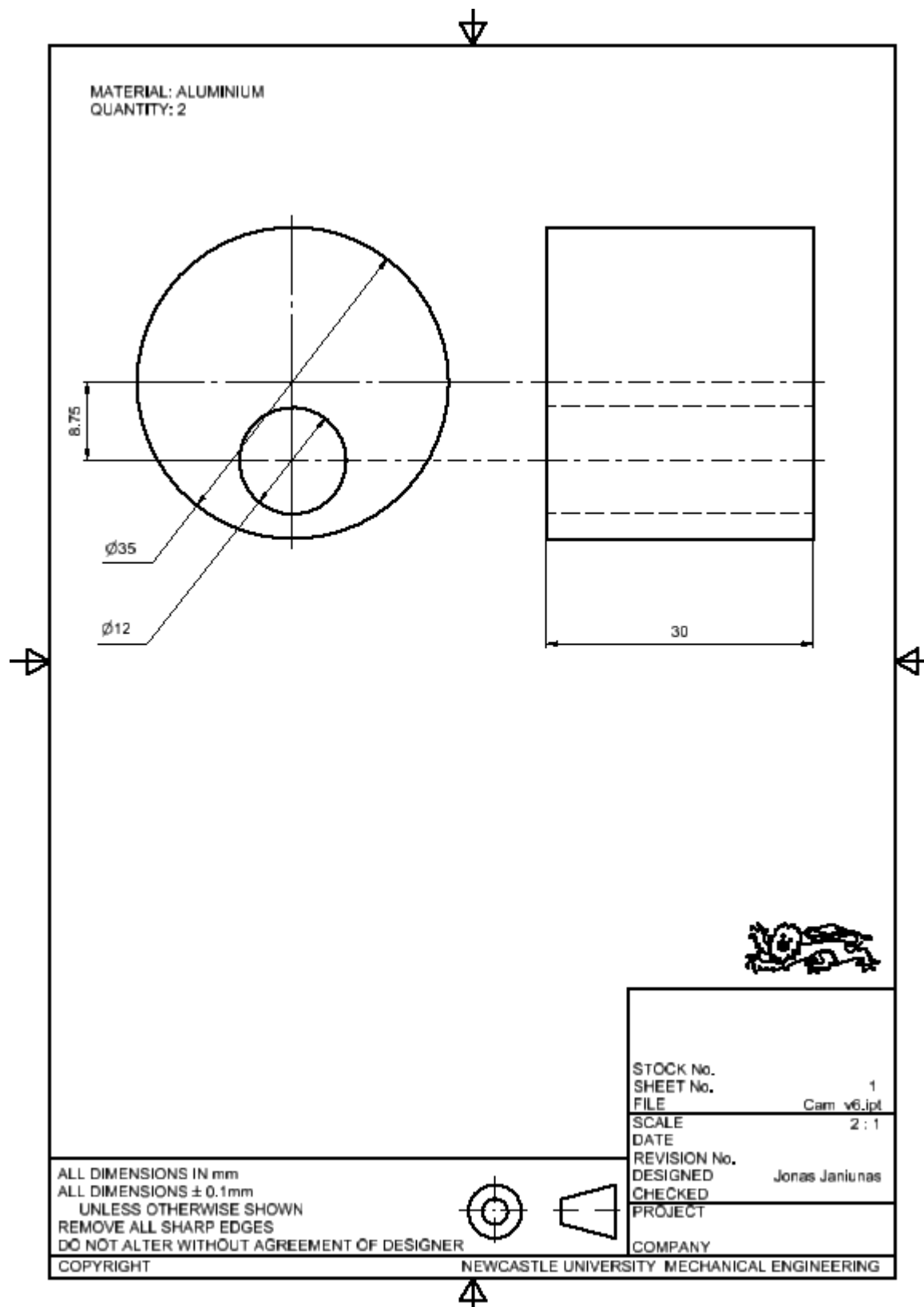


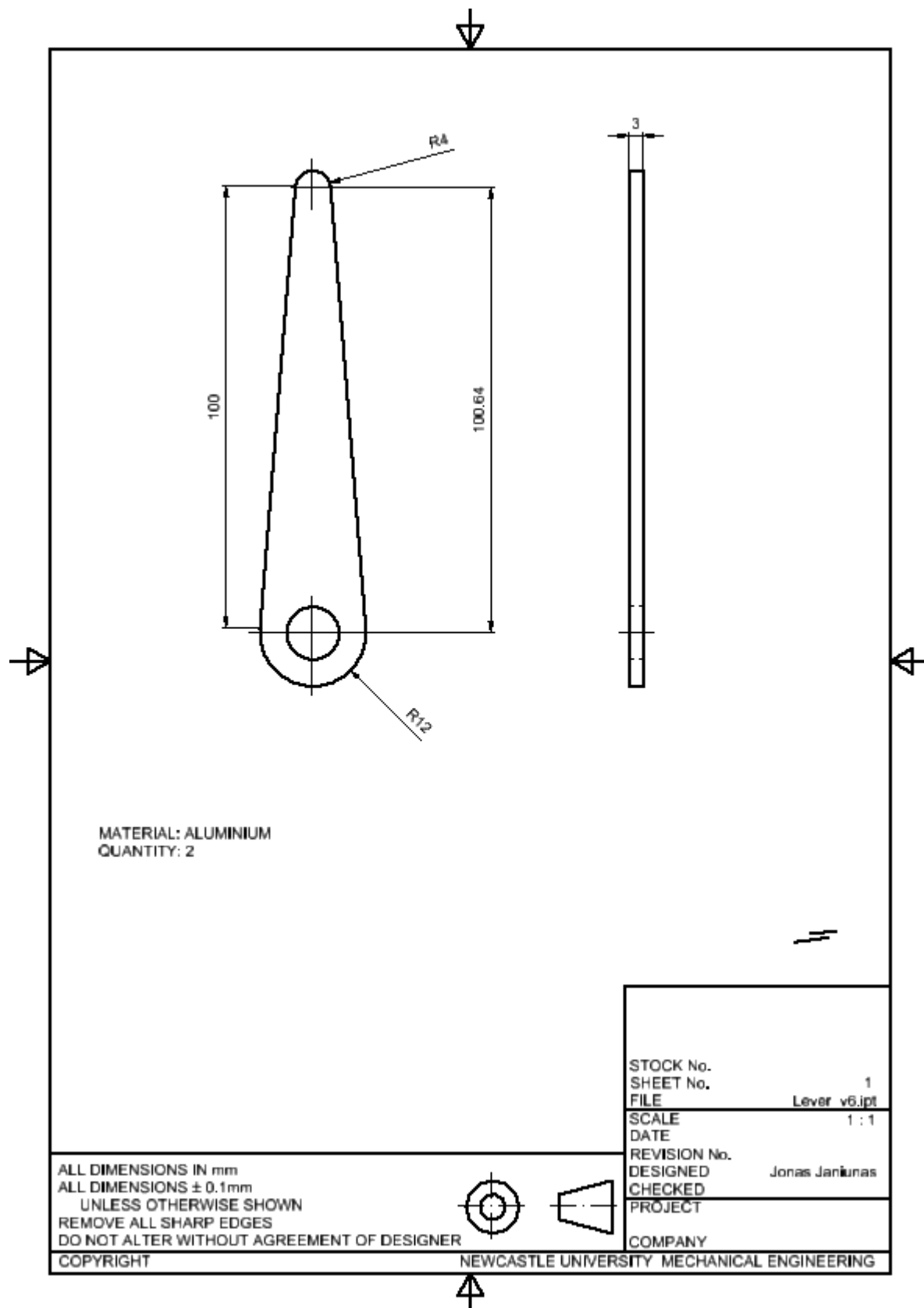
Appendix C-6: Battery Box Drawings (JO)

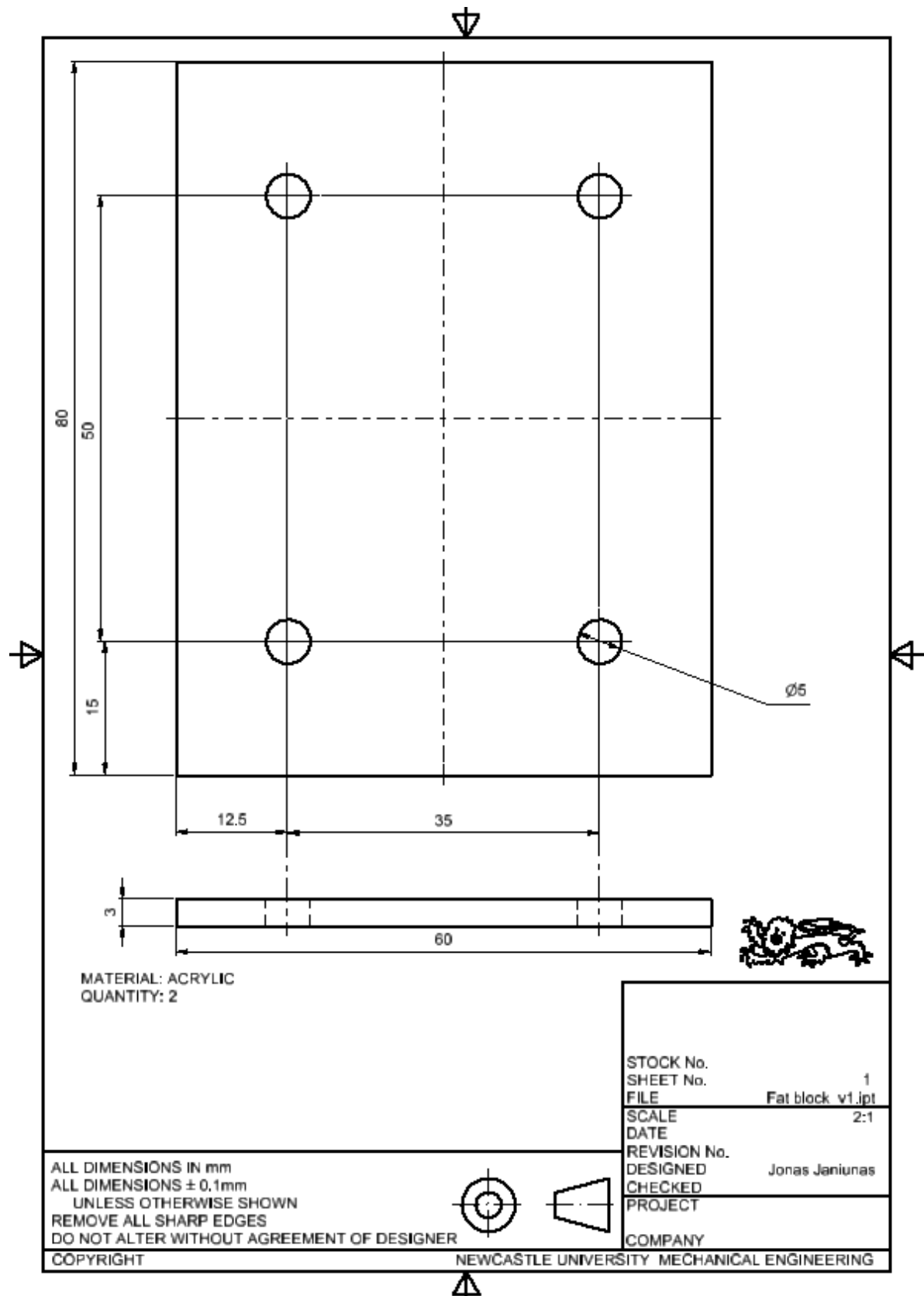






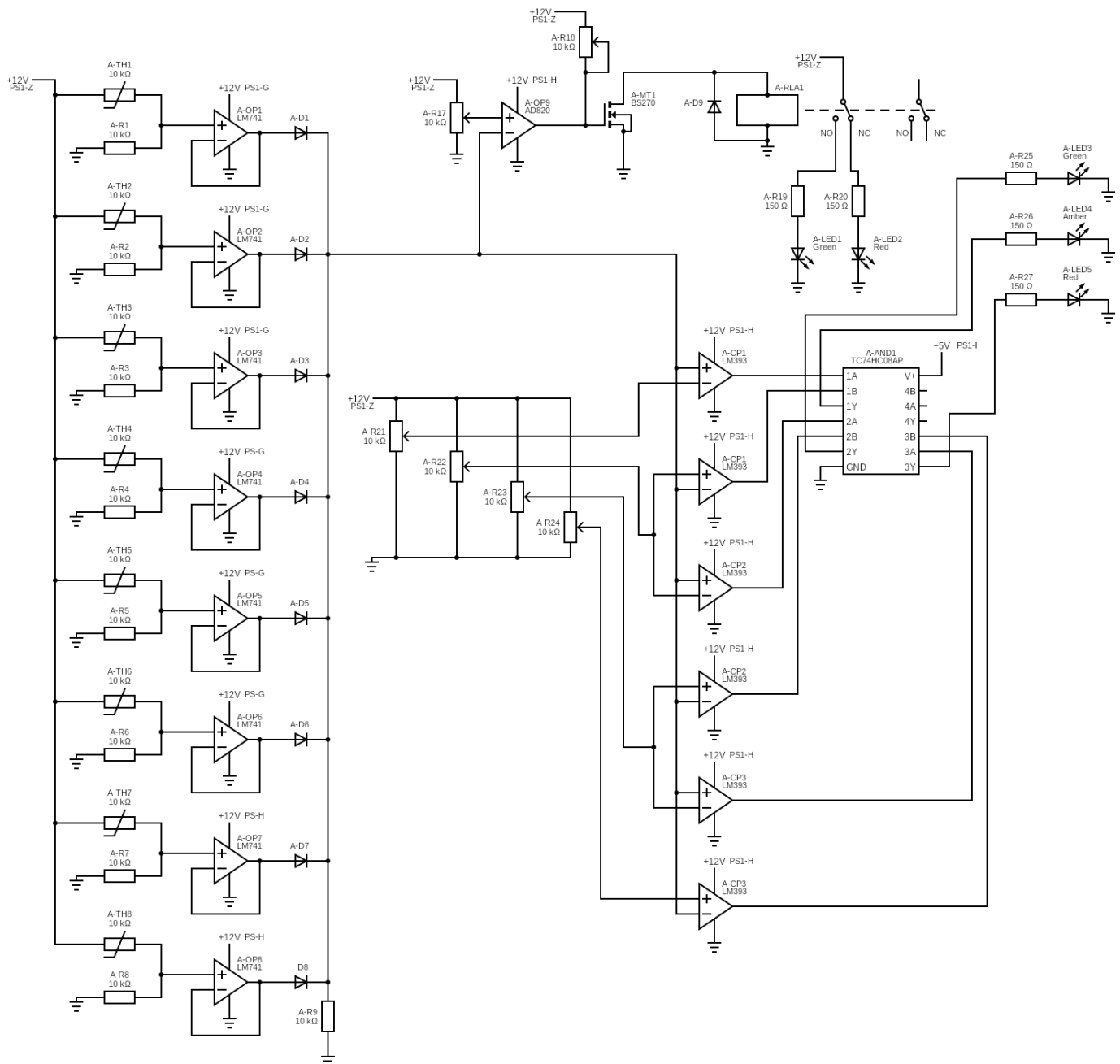




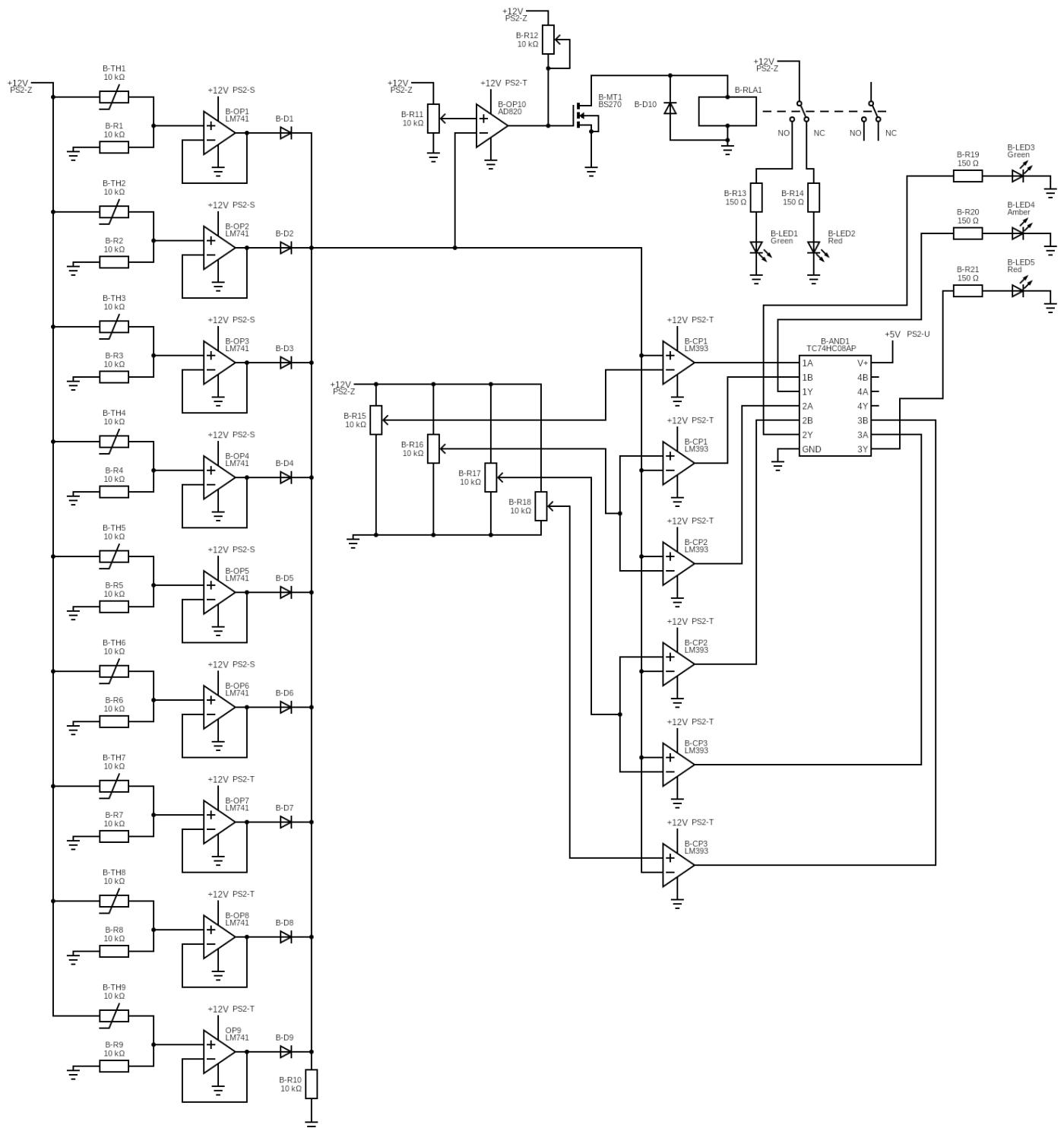


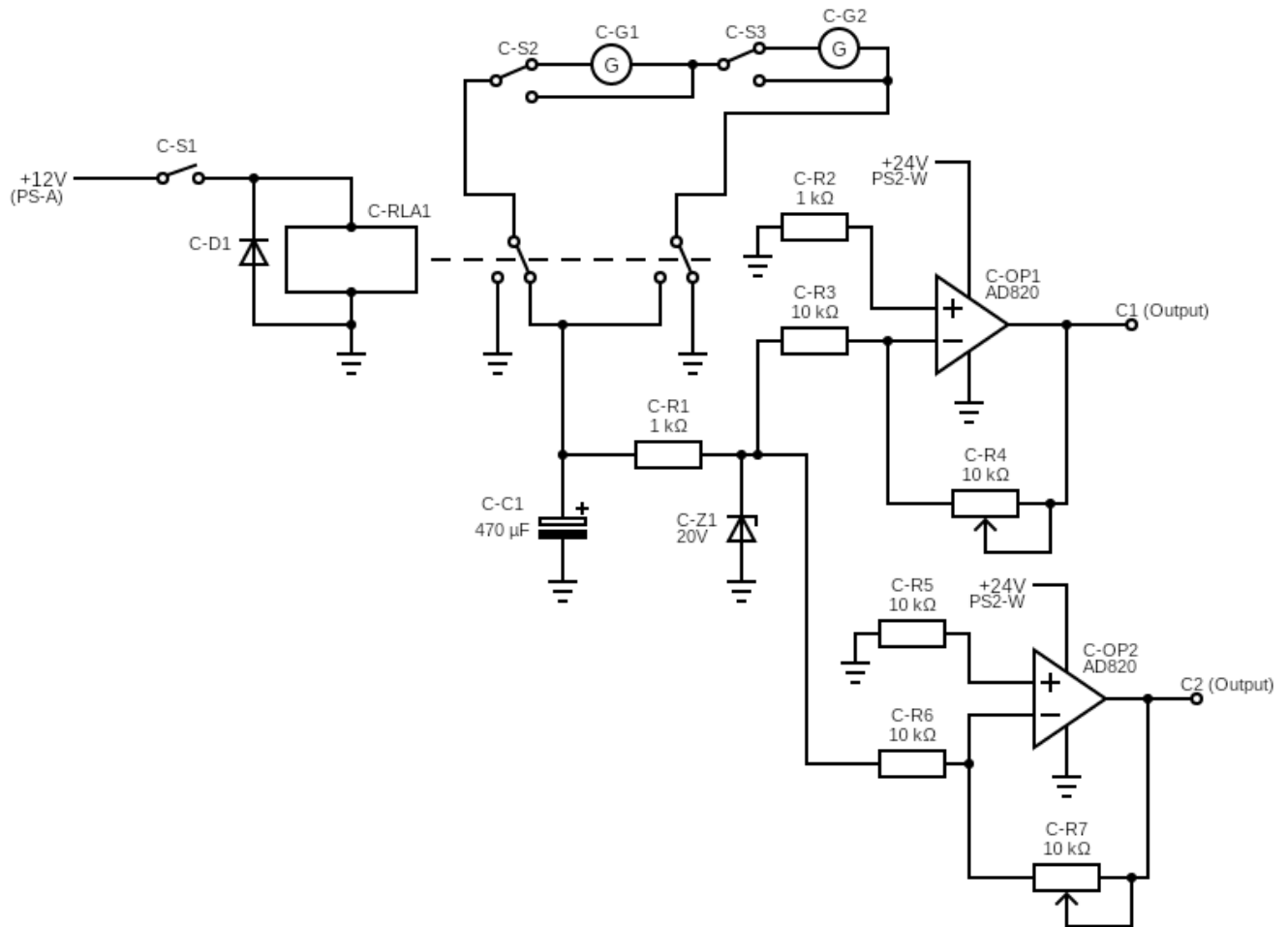
Appendix D: Low Voltage Electronics

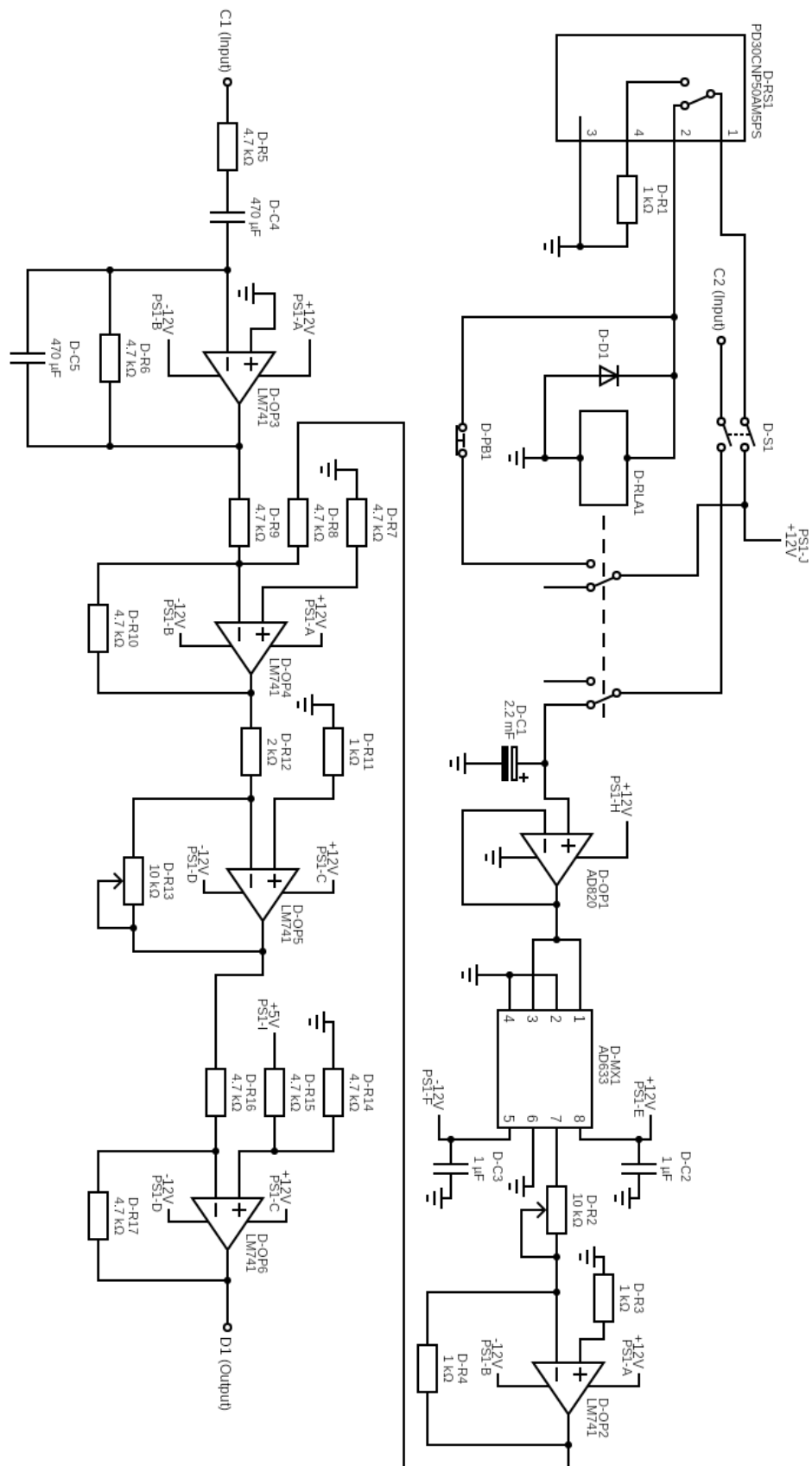
Appendix D-1: Fire and Heat (Front) Circuit A V5 (AH)

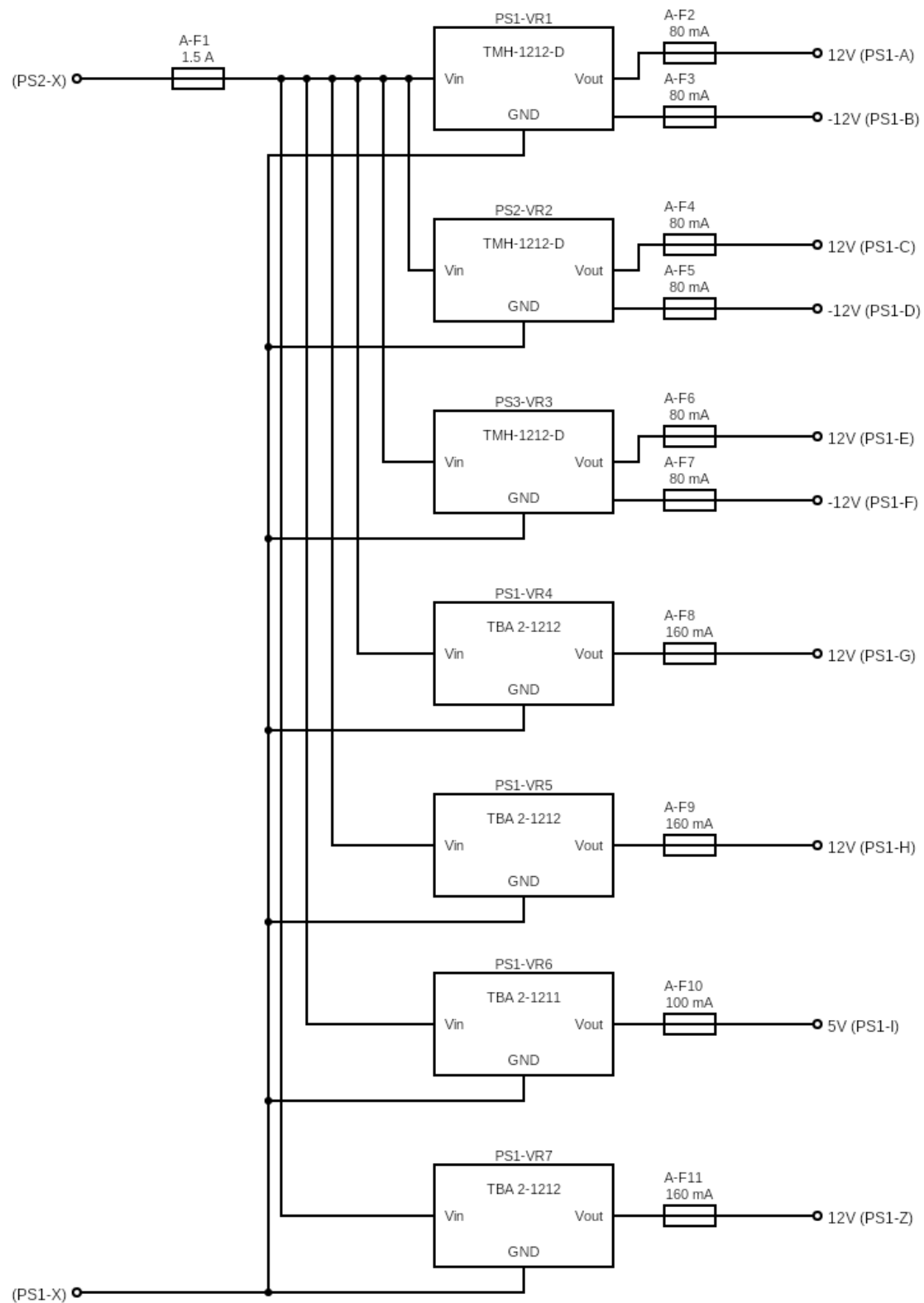


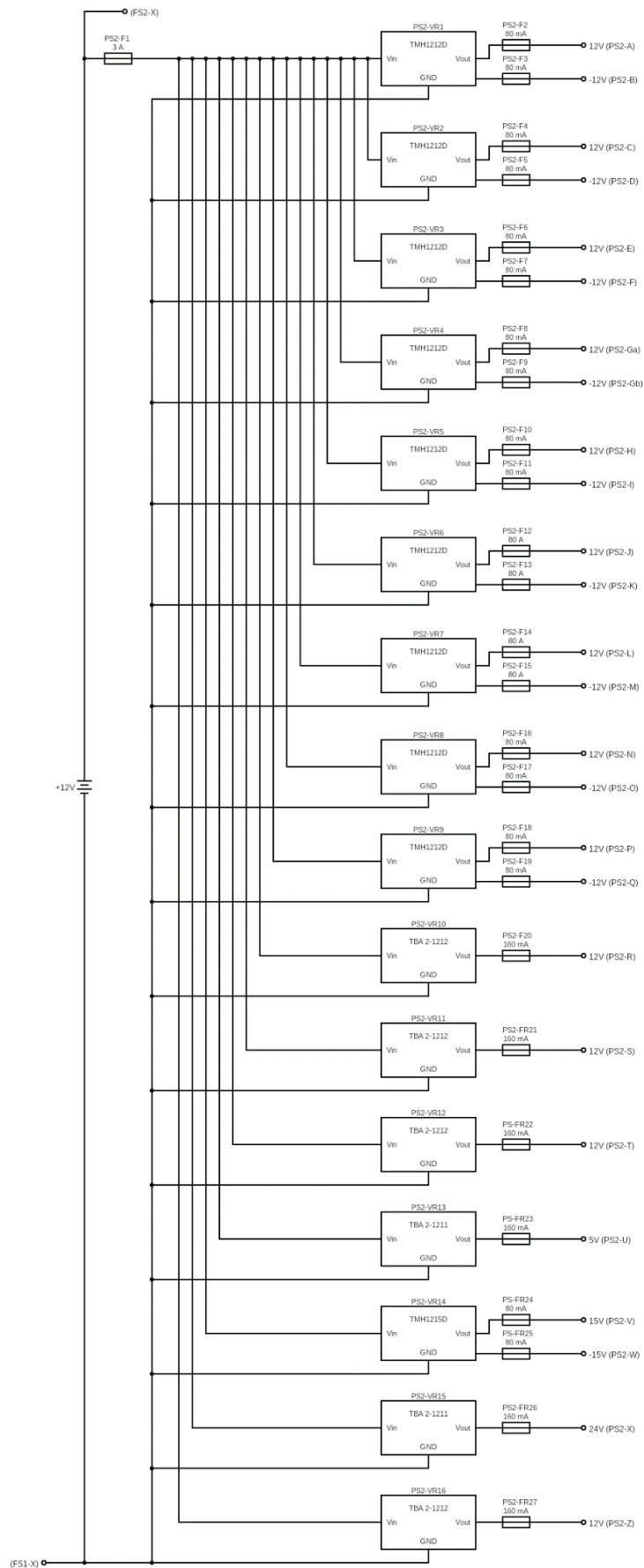
Appendix D-2: Fire and Heat (Rear) Circuit A V5 (AH)

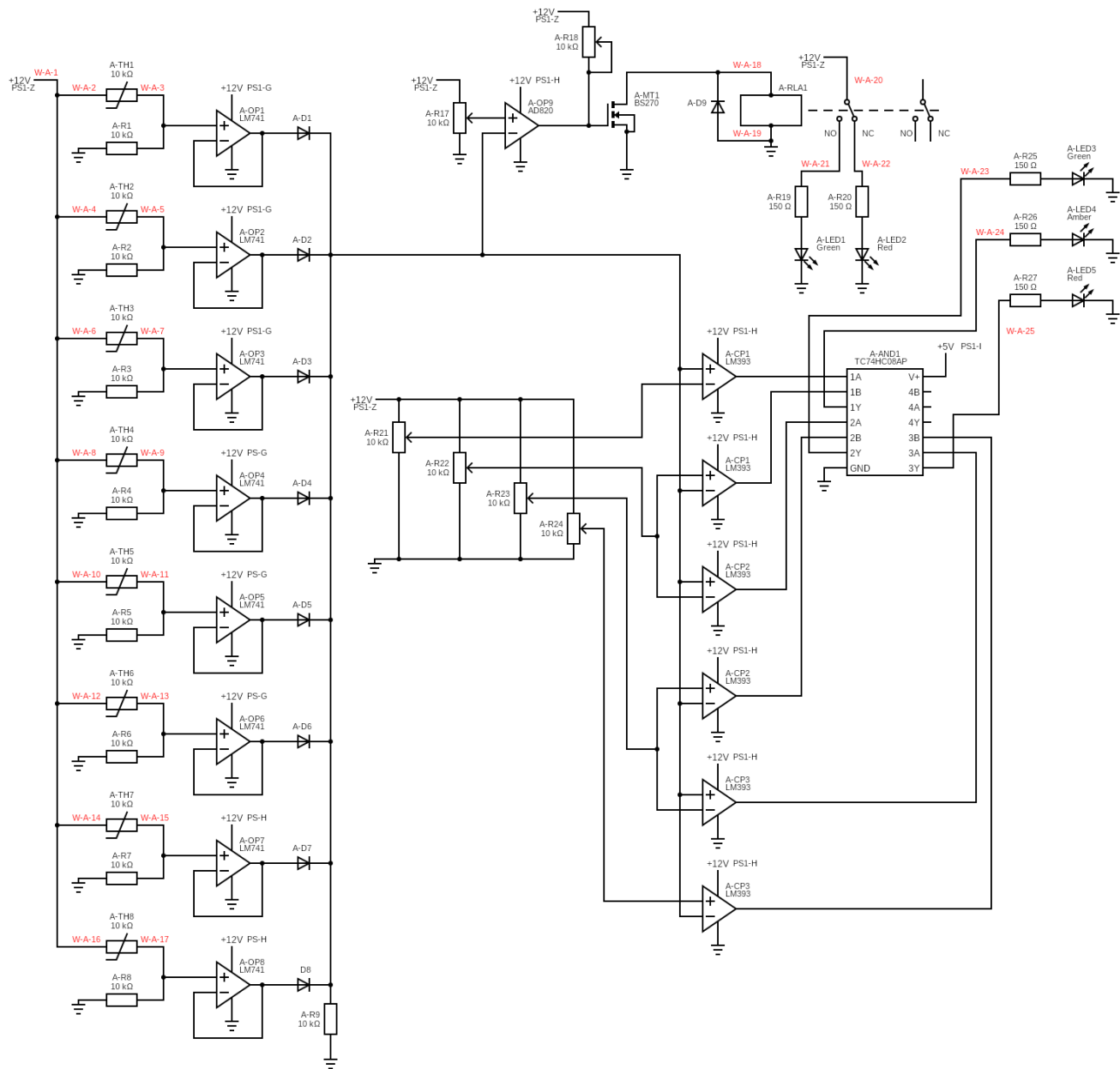


Appendix D-3: True Ground Speed Monitoring Circuit C V4 (AH)

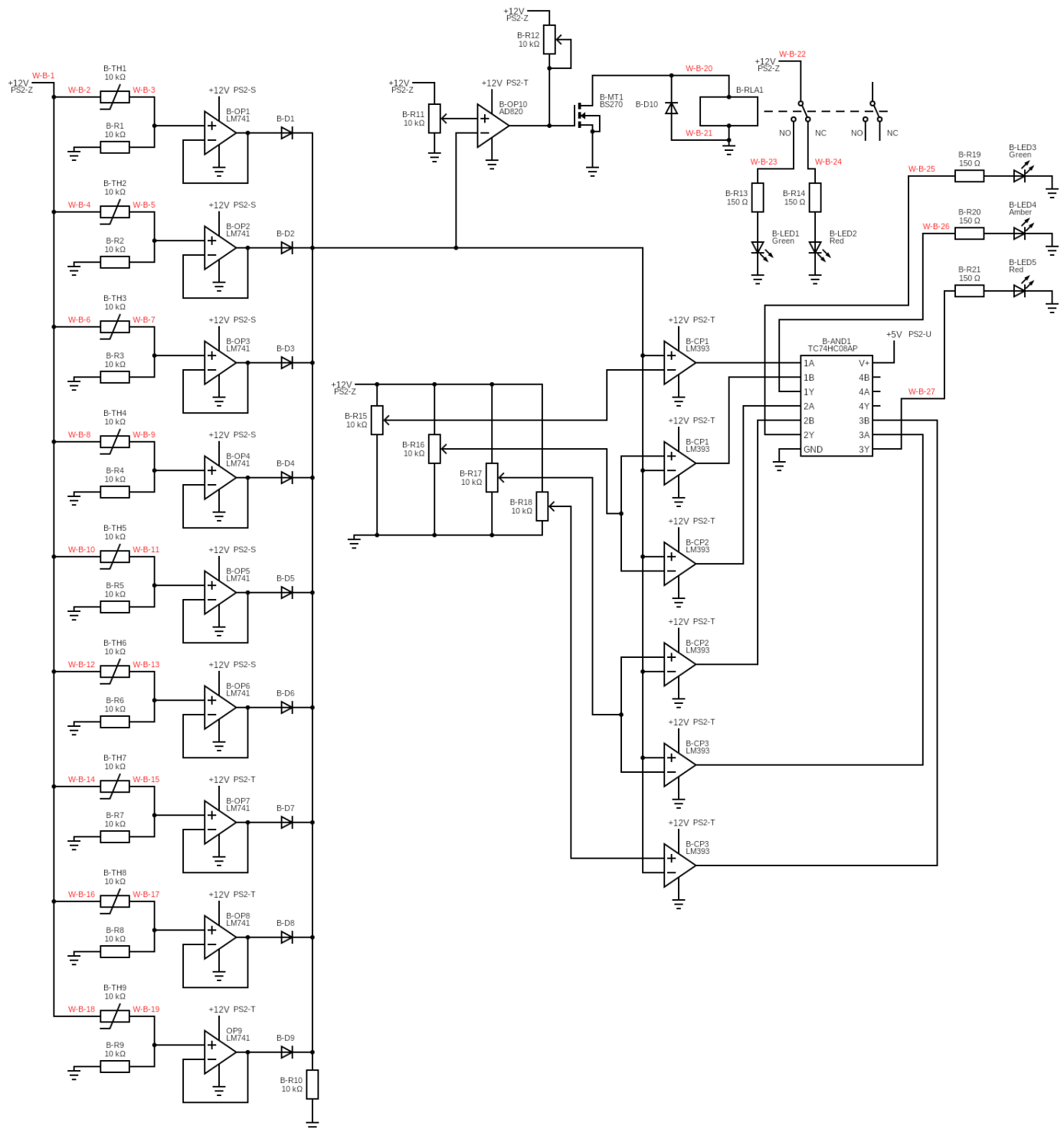


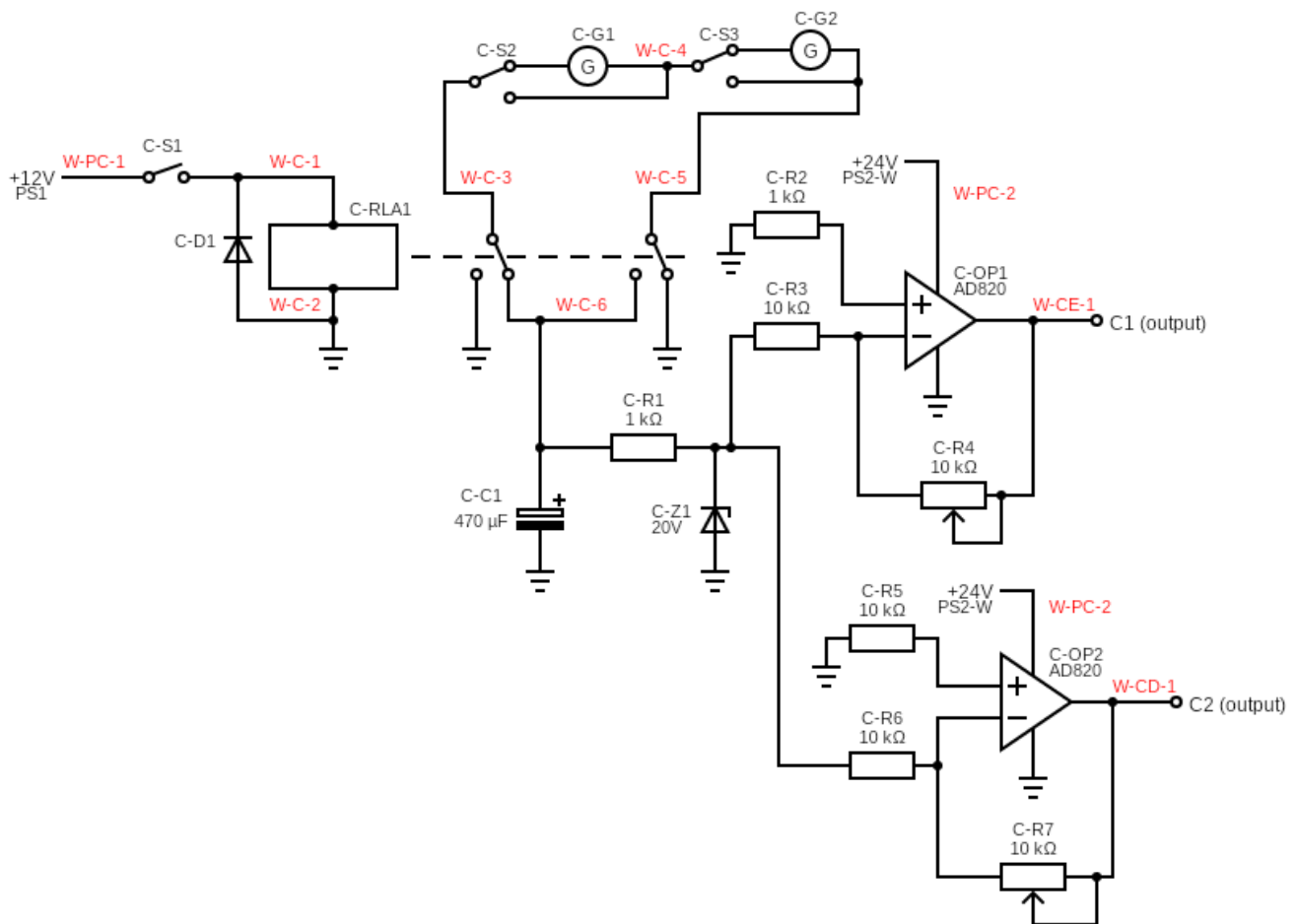
Appendix D-7: Low Voltage Power Supply (Front) Circuit PS1 V3 (AH)

Appendix D-8: Low Voltage Power Supply (Rear) Circuit PS1 V3 (AH)

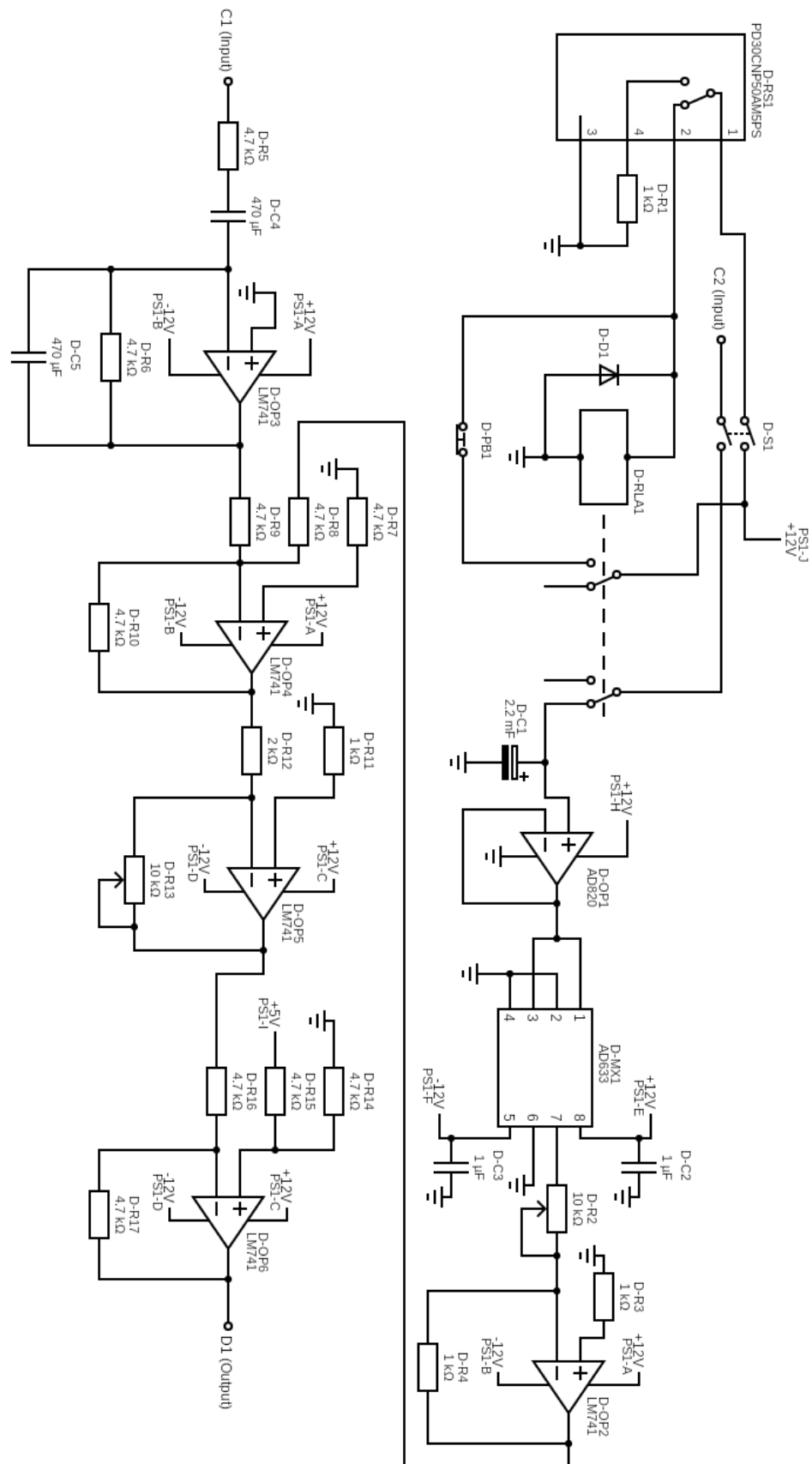
Appendix D-9: Fire and Heat (Front) Circuit A V5 [Wire Diagram] (AH)

Appendix D-10: Fire and Heat (Rear) Circuit A V5 [Wire Diagram] (AH)



Appendix D-11: True Ground Speed Monitoring Circuit C V4 [Wire Diagram]**(AH)**

Appendix D-12: Auto-stop Circuit D V5 [Wire Diagram] (AH)



The schematic diagram illustrates a 100W Class D audio amplifier. The power supply section at the top left shows a 100V AC input connected to a bridge rectifier (four diodes) and a large electrolytic filter capacitor (1000µF). The main amplifier stage is a differential input stage, followed by a voltage divider network (resistors and capacitors) and a push-pull output stage using MOSFETs. The output is connected to a speaker. The schematic is annotated with component values, part numbers, and frequency response curves.

Appendix D-14: Extract of Electronics Component Register (AH)

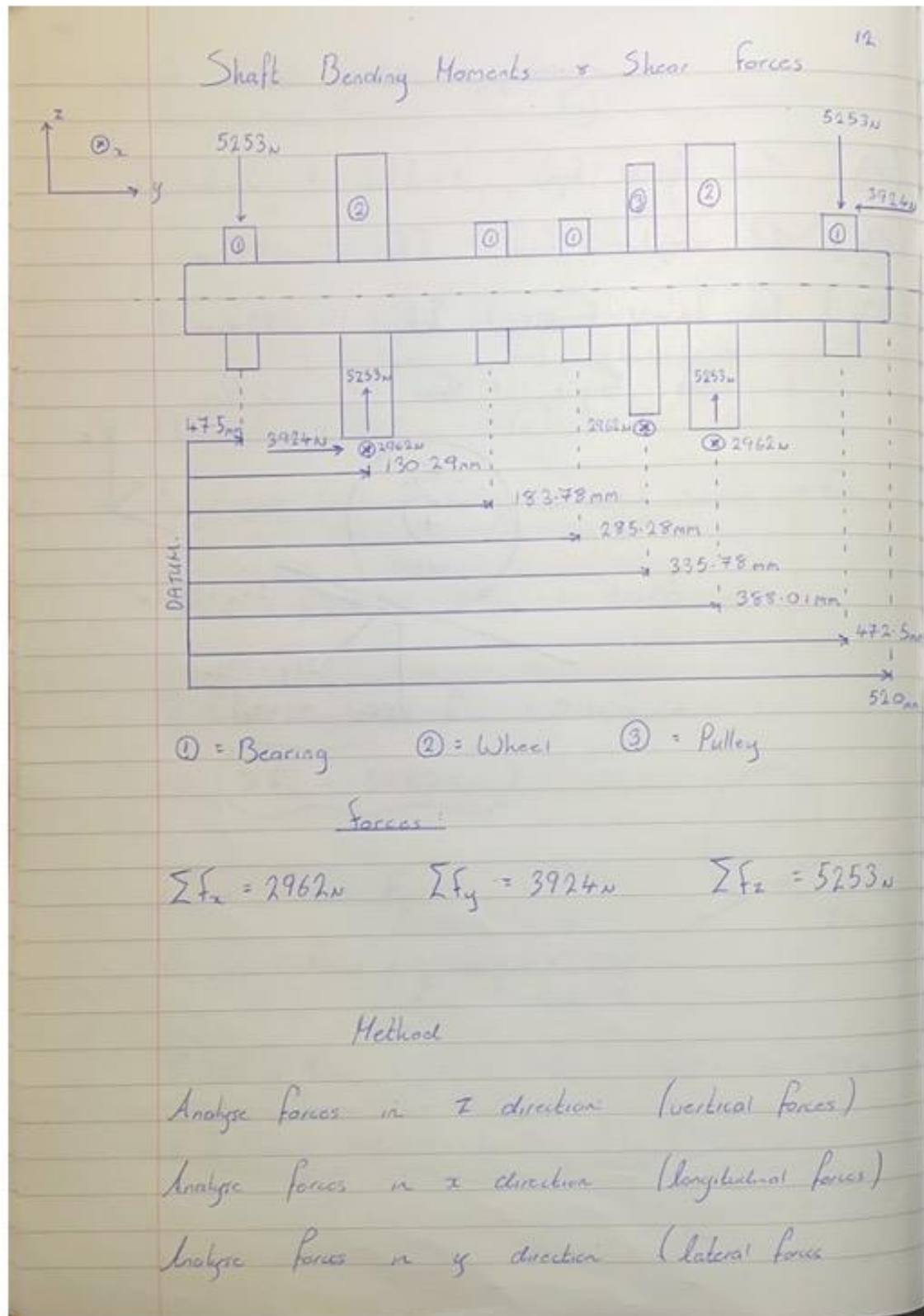
Circuit	Component	Code	Value	Unit
A	Resistor	A-R1	10000	Ω
A	Resistor	A-R2	10000	Ω
A	Resistor	A-R3	10000	Ω
A	Resistor	A-R4	10000	Ω
A	Resistor	A-R5	10000	Ω
A	Resistor	A-R6	10000	Ω
A	Resistor	A-R7	10000	Ω
A	Resistor	A-R8	10000	Ω
A	Resistor	A-R9	10000	Ω
A	Resistor (potentiometer)	A-R17	10000	Ω
A	Resistor (potentiometer)	A-R18	10000	Ω
A	Resistor	A-R19	150	Ω
A	Resistor	A-R20	150	Ω
A	Resistor (potentiometer)	A-R21	10000	Ω
A	Resistor (potentiometer)	A-R22	10000	Ω
A	Resistor (potentiometer)	A-R23	10000	Ω
A	Resistor (potentiometer)	A-R24	10000	Ω
A	Resistor	A-R25	150	Ω
A	Resistor	A-R26	150	Ω
A	Resistor	A-R27	150	Ω
A	LED	A-LED1	GREEN	
A	LED	A-LED2	RED	
A	LED	A-LED3	GREEN	
A	LED	A-LED4	AMBER	
A	LED	A-LED5	RED	
A	Diode	A-D1		
A	Diode	A-D2		
A	Diode	A-D3		
A	Diode	A-D4		
A	Diode	A-D5		
A	Diode	A-D6		
A	Diode	A-D7		
A	Diode	A-D8		
A	Diode	A-D9		
A	Thermistor (NTC)	A-TH1	10000	Ω
A	Thermistor (NTC)	A-TH2	10000	Ω
A	Thermistor (NTC)	A-TH3	10000	Ω
A	Thermistor (NTC)	A-TH4	10000	Ω
A	Thermistor (NTC)	A-TH5	10000	Ω
A	Thermistor (NTC)	A-TH6	10000	Ω
A	Thermistor (NTC)	A-TH7	10000	Ω
A	Thermistor (NTC)	A-TH8	10000	Ω

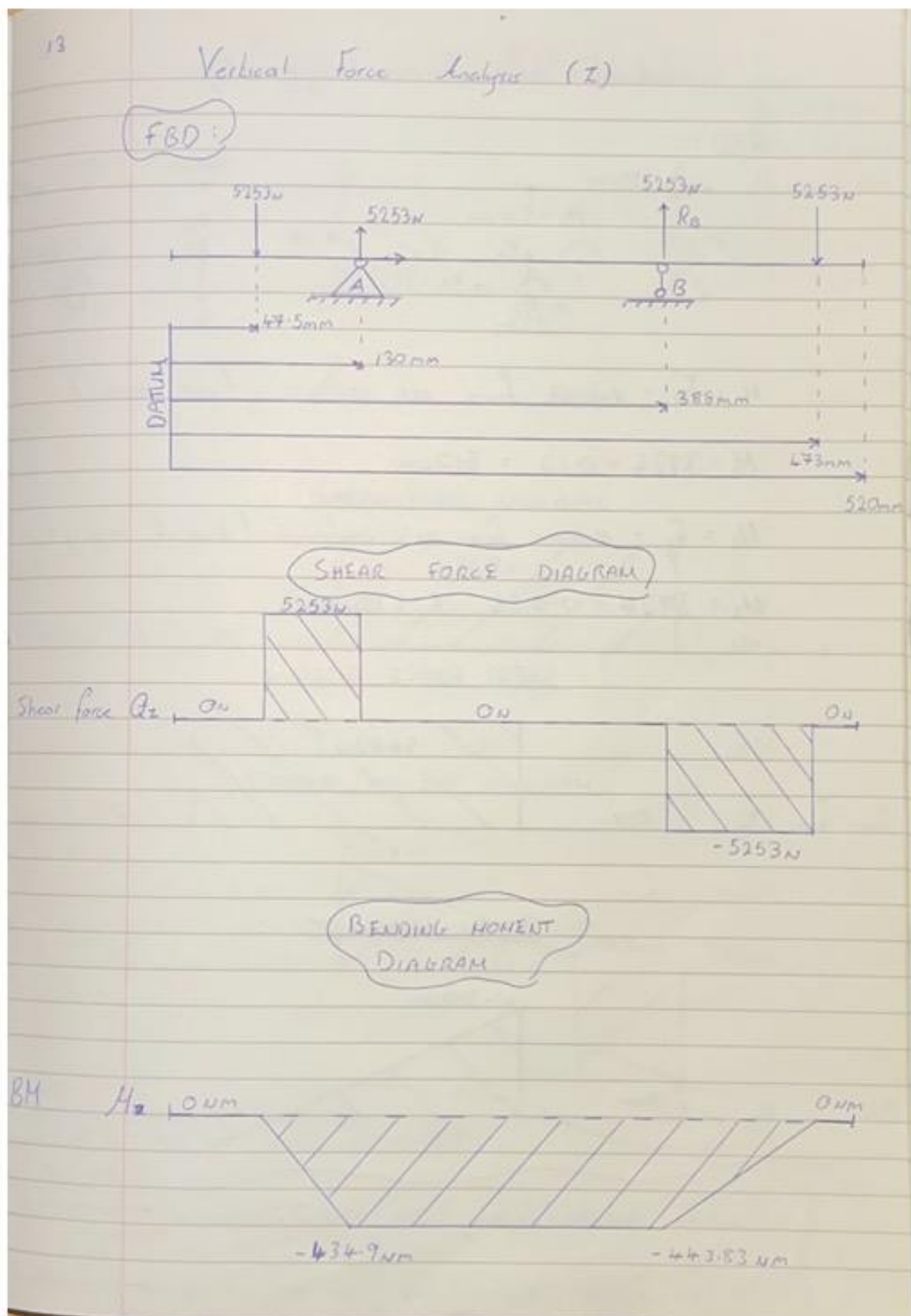
Appendix D-15: Electronic Component Register Code Abbreviations (AH)

ABBREVIATION	DEFINITION
D	Diode
R	Resistor
RLA	Relay
LED	Light Emitting Diode
AND	AND Logic Gate
CP	Comparator
OP	Operational Amplifier
MT	MOSFET
RS	Retroreflective Sensor
FTV	Frequency-Voltage Converter
Z	Zener Diode
MX	Analogue Multiplier
C	Capacitor

Appendix F: Unsprung Assembly Component Calculations (DS)

Appendix F-1: Shaft Calculations (DS)

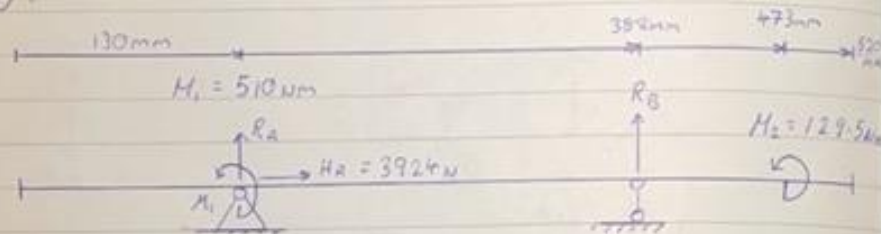




Lateral Shaft Force Analysis (4)

14

FBD:



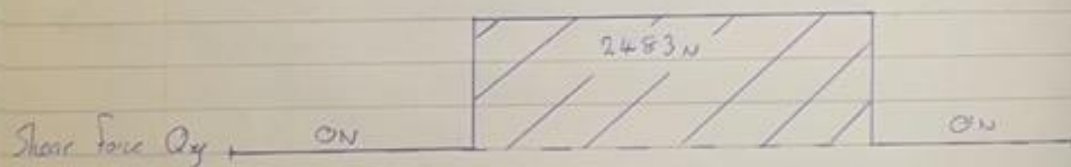
$$H_1 = F_y \cdot \text{distance from axle centerline (wheel radius)}$$

$$H_1 = 3924 \cdot 0.13 = 510 \text{ N}$$

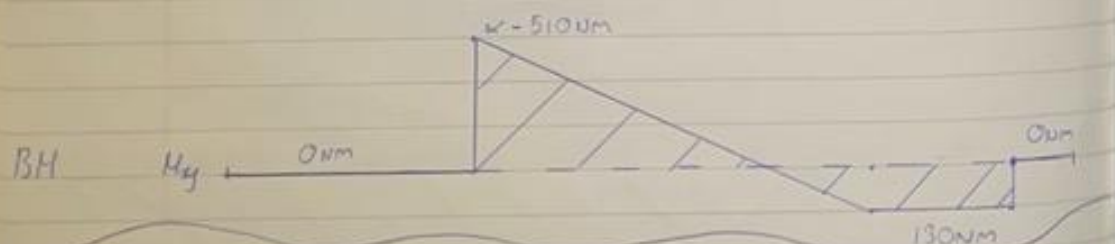
$$H_2 = F_y \cdot \text{distance from axle centerline (top of bearing)}$$

$$H_2 = 3924 \cdot 0.033 = 130 \text{ N}$$

SHEAR FORCE DIAGRAM



BENDING MOMENT



- Due to vertical loading acting between wheel & bearing
- Due to lateral loading acting on the wheels.

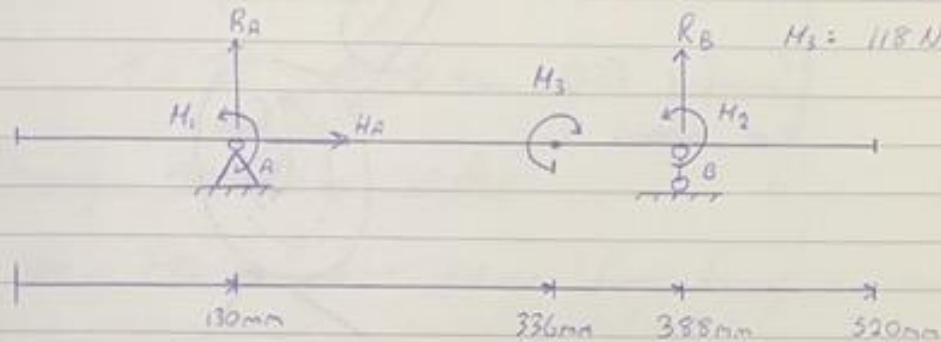
15

Torques on Shaft & Longitudinal Forces

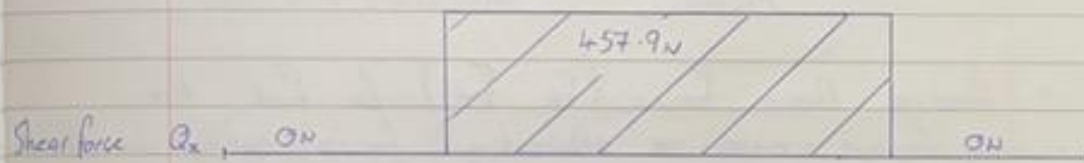
$$H_1 = 118 \text{ Nm}$$

$$H_2 = 118 \text{ Nm}$$

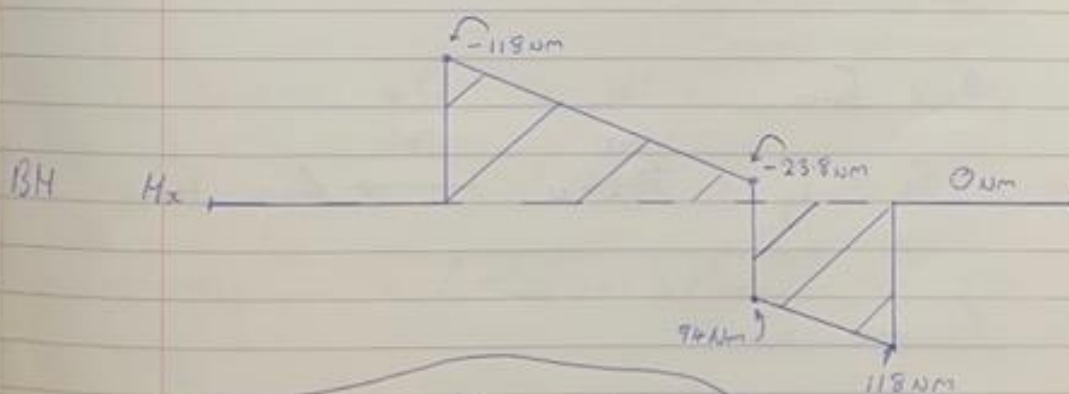
$$H_3 = 118 \text{ Nm}$$



SHEAR FORCE DIAGRAM



BENDING MOMENT DIAGRAM



SUMMARY

τ_{max} : MAXIMUM SHEAR FORCE = 5253 N

$\sigma_{b,max}$: MAXIMUM BENDING MOMENT = 510 Nm

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Allowable Stress: $\sigma_{\text{Allowable}}$

$$\begin{aligned}\sigma_{\text{Allowable}} &= \text{UTS} \cdot k_1 \cdot k_2 \cdot k_3 \\ &= 660 \times 0.5 \times 0.75 \times 0.83\end{aligned}$$

$$\text{Allowable } \sigma = 205.4 \text{ MPa}$$

Factor of Safety (FOS) = $\frac{\text{Allowable Stress}}{\text{Maximum Stress}}$

$$\text{FOS} = \frac{660 \pm 205.4 \text{ MPa}}{8.9 \pm 10.9 \text{ MPa}}$$

$$\begin{aligned}\text{Safety Factor} &= 1.88 \quad \longrightarrow \text{Allowing } \sigma \\ &= 74 \quad \longrightarrow \text{Mean } \sigma\end{aligned}$$

Summary

Current shaft geometry has a
Safety factor of 1.88

Appendix F-2: Bearing Calculations (DS)

34

Bearing Calculations. Eccentric Housing

mass of motor, mounts & brakes !

- Brakes $\{0.865 \text{ kg}\}$
- Mount $\{0.548 \text{ kg}\}$
- Motor $\{10.9 \text{ kg}\}$
- Eccentric $\{2.4 \text{ kg}\}$
- Mount (motor) $\{4 \text{ kg}\}$

$\Sigma = 19 \text{ kg}$

We will use 20 kg for calculations.

Eccentric Bearing Forces

$$\begin{cases} F_{z,e} = 20 \text{ kg} \times 9.81 \times 5g = 981 \text{ N} \\ F_{y,e} = 20 \text{ kg} \times 9.81 \times 10g = 1962 \text{ N} \\ F_{x,e} = 20g \times 9.81 \times 10g = 1962 \text{ N} + 722 \text{ N} \\ \quad \quad \quad = 2684 \text{ N} \end{cases}$$

Transverse Force $= \sqrt{F_{x,e}^2 + F_{z,e}^2}$

$$F_{te} = \sqrt{2684^2 + 1962^2}$$

Transverse Force on eccentric housing bearings: 3324.7 N

Transverse Force $= 3324.7 \text{ N}$

35

Bearing Forces Axle Box

$$F_{xA} = 2962 \text{ N}$$

$$F_{yA} = 3924 \text{ N}$$

$$F_{zA} = 5253 \text{ N}$$

$$\text{Transverse Force} = \sqrt{2962^2 + 5253^2}$$

$$F_{TA} = 6030.5 \text{ N}$$

Safety Factor of Eccentric Bearings

Static load rating $\geq$ safety factor $\times$ Equivalent Load

$$C_0 \geq S_0 P_0$$

Rearranging:

$$S_0 = \frac{C_0}{P_0}$$

$$C_0 = 21.6 \text{ kN}$$

$$P_0 = 3325 \text{ N}$$

$$S_0 = \frac{21600}{3325} = 6.5$$

N.

Safety Factor of Axle Box Bearings

$$S_0 = \frac{21600}{6031} = 3.58$$

Bearing Life

36

Formula provided by SKF:

$$\text{basic life rating (90\%)} = \frac{\text{Basic Dynamic Load Rating}}{\text{Equivalent dynamic bearing load}}$$

$$L_{10} = \left(\frac{C}{P} \right)^3 \quad C = 33.2 \text{ kN}$$

The equivalent dynamic bearing load is calculated as follows:

$$P = F_{\text{transverse}} + Y_1 F_{\text{axial}}$$

$$F_T = 6031 \text{ N}$$

$$F_A = 2962 \text{ N}$$

$$Y_1 = 2.2$$

} Axle
Box
Bearings

$$F_T = 3325 \text{ N}$$

$$F_A = 981 \text{ N}$$

$$Y_1 = 2.2$$

} Eccentric
Housing
Bearings

$$P = \underline{\text{Axle Box}}$$

$$P = 6031 + 2.2(2962)$$

$$P = 12547.4 \text{ N}$$

$$L_{10} = \left(\frac{33.2 \times 10^3}{12.55 \times 10^3} \right)^3$$

$$L_{10} = 18.513 \text{ million revolutions}$$

37

Given the wheel $\phi = 250\text{mm}$

Distance travelled D

$D = \text{Circumference} \times \text{millions of revolutions}$

$$D = 0.250 \times 18.5 \times 10^6$$

$$D = 4625 \text{ km before failure of } 10\%$$

Assuming average speed of 15 km/h

$$\text{hours of operation} = \frac{4625 \text{ km}}{15 \text{ km/h}} = 308 \text{ hours}$$

$$\text{Days of operation} = 12.84 \text{ days of continuous operation}$$

Eccentric Bearings

$$P = 3325 + 2.2 (981) = 5483.2 \text{ N}$$

$$L_{10} = \left(\frac{33.2}{5.48} \right)^3 = 222.4 \text{ million cycles revolutions}$$

$$\text{Days of operation} = \frac{\pi \times 0.25 \times 222.4 \times 10^3 \text{ km}}{15 \text{ km/h} \times 24 \text{ hours}}$$

$$\text{Days of continuous operation} = 485.2 \text{ Days}$$

Bearing Summary

38

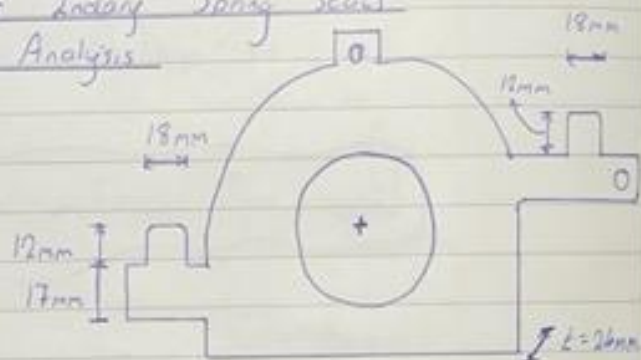
	Safety Factor	Distance (km)	Days of continuous use
Eccentric	6.5	174000 km	48.5 days
Axle Box	3.5	4625 km	12.8 days

Axle Box & Secondary Spring seats Stress Analysis

Assuming material yield strength of 527.5

$$\sigma_y = 275 \text{ N/mm}^2$$

Using Hookes Law



Axle Box

$$\sigma = F/A$$

$$F_{\text{shear}} = \frac{\sigma_y \times \text{Area}}{2}$$

Wheel Contact Forces :

$$F_z = 5253.2 \text{ N}$$

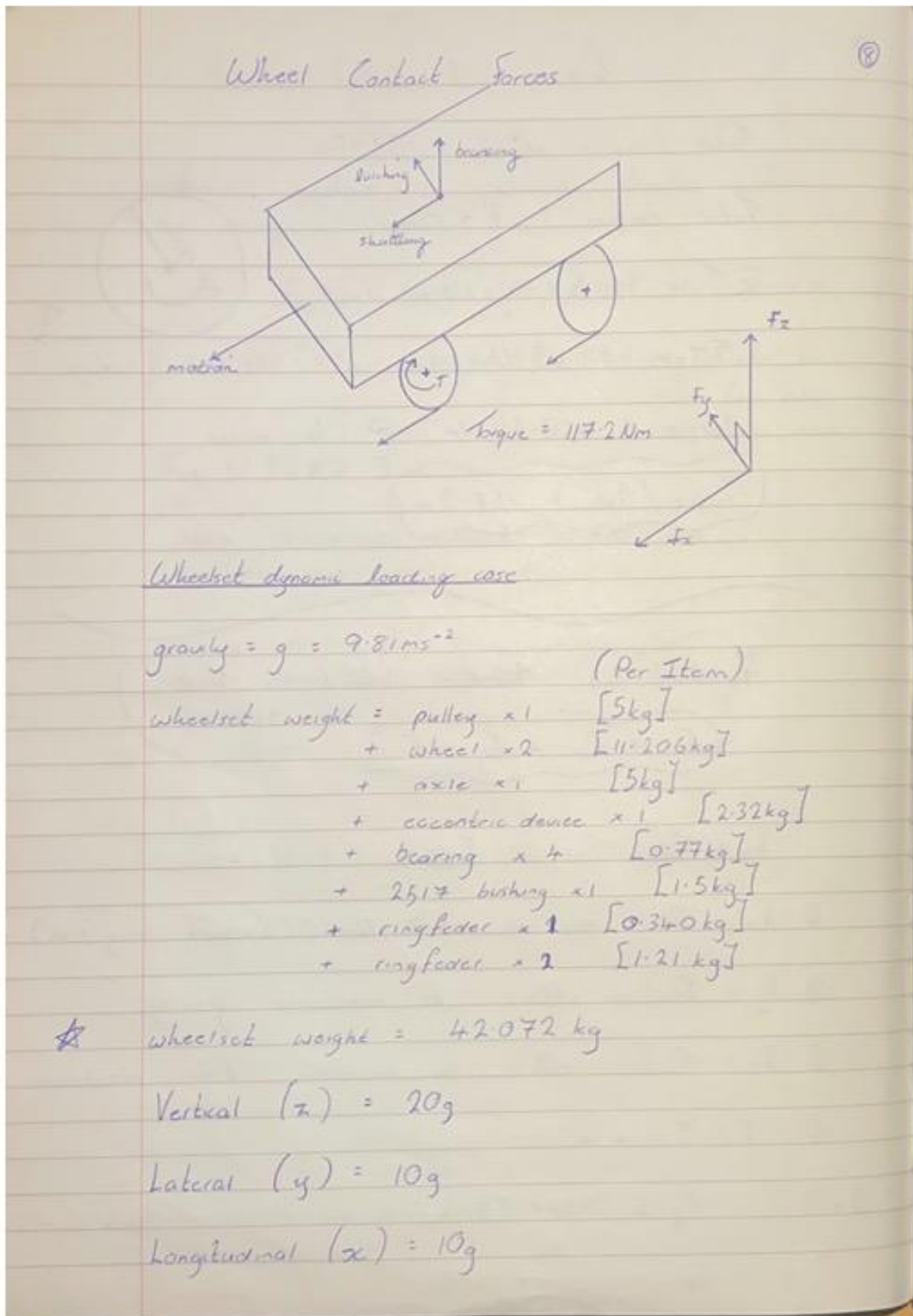
$$F_r = 6031 \text{ N}$$

$$F_{gz} = 2962 \text{ N}$$

$$F_{zy} = 3924 \text{ N}$$

x dir : $F_{\text{shear}} = \frac{275}{2} \times 12 \times 18 = 29700 \text{ N}$
 y dir : $F_{\text{shear}} = 29700 \text{ N}$
 Transverse dir : $F_{\text{shear}} = 29700 \text{ N}$

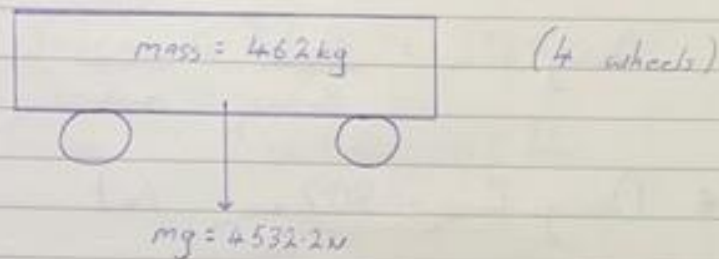
Appendix F-3: Wheel Contact Forces (DS)



⑨

Mass of Loco = 924 kg

Vertical Static Force



$$\text{Vertical Static Force} = \frac{4532.2}{4} = 1133 \text{ N} = F_{s,z}$$

$$\text{Vertical DYNAMIC Force} = \frac{\text{wheelset weight} \times 20g}{2 \text{ (wheels)}}$$

$$F_{\text{dynamic},z} = \frac{m_{\text{wheelset}} \times 20g}{2}$$

$$F_{0,z} = \frac{42 \times 20 \times 9.81}{2} = 4120.2 \text{ N}$$

$$F_{0,z} = 4120.2 \text{ N} \quad (\text{Assuming } 20g \text{ impact})$$

$$F_{0,z} = 2060.1 \text{ N} \quad (\text{Assuming } 10g \text{ impact})$$

$$\therefore \text{Total Vertical Force} = \text{Dynamic} + \text{Static}$$

$$\Sigma F_z = F_{0,z} + F_{s,z} = (1133 + 4120.2) \text{ N}$$

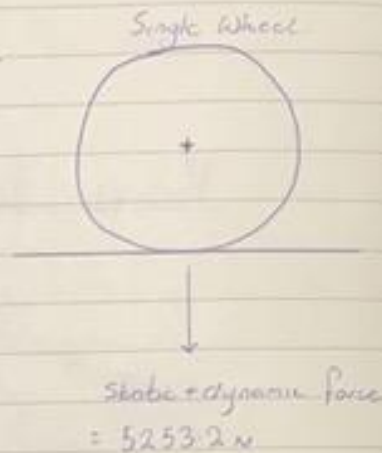
$$\star \quad \boxed{\Sigma F_z = 5253.2 \text{ N} = \text{Total Vertical Force}} \quad \star$$

Longitudinal Tractive Force F_x :

$$\text{Driving torque} = 117.2 \text{ N (pg 7)}$$

$$\text{Wheel Radius} = 0.13 \text{ m}$$

$$\text{Driving force} = \frac{\text{Torque}}{\text{Radius}} = \frac{117.2}{0.13}$$



$$\star \text{ Driving force}_x = 902 \text{ N}_x \quad (x)$$

$$\star \text{ Longitudinal Dynamic Force}_2 = \frac{M_{\text{wheels}} \times 10g}{\text{no of wheels}} = F_{0,x}$$

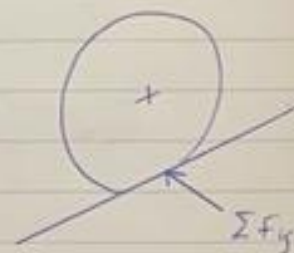
$$F_{0,x} = \frac{42 \times 10 \times 9.81}{2} = 2060.1 \text{ N}$$

$$\Sigma \text{ Total Longitudinal Force} = 2060.1 + 902 \text{ N}$$

$$\Sigma F_x = 2962.1 \text{ N}$$

Lateral Force F_y :

$$\Sigma \text{ Lateral Force} = \frac{M_{\text{wheels}} \times 10g}{(\text{no of wheels}) \cdot 1}$$



$$\Sigma \text{ Lateral Force} = \Sigma F_y = 42 \times 10 \times 9.81 = 3924 \text{ N}$$

$$\Sigma \text{ Total Lateral Force} = \Sigma F_y = 3924 \text{ N}$$

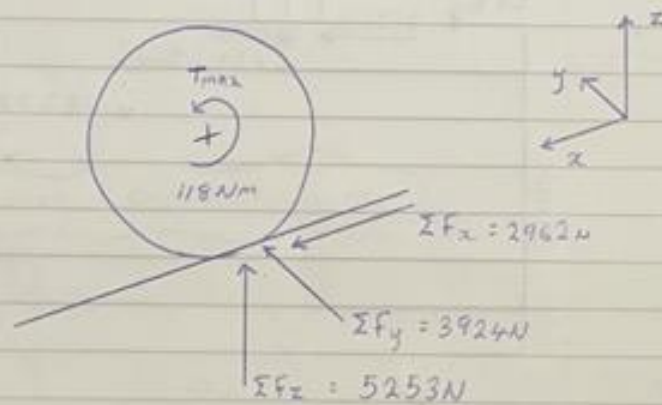
Summary of Wheel Contact Forces

(20g) Total Vertical Force = $\Sigma F_z = 5253.2 \text{ N}$

(10g) Total Longitudinal Force = $\Sigma F_x = 2962 \text{ N}$

(10g) Total Lateral Force = $\Sigma F_y = 3924 \text{ N}$

Max Torque = $T_{\text{max}} = 118 \text{ Nm}$



Appendix F-4: Tractive Effort Calculations (DS)

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TRACTION EFFORT

$m_c = 1800 \text{ kg}$
 $m_r = 924 \text{ kg}$

$$\Sigma F_x = F_d - D_{rr} - m_c g \sin \theta - m_r g \sin \theta$$

$$\Sigma F_x = F_d = m_c g \sin \theta + m_r g \sin \theta$$

$$\Sigma F_x = 1800 \times 9.81 \times \sin(2) + 924 \times 9.81 \times \sin(2)$$

$$\Sigma F_x = 932.6 \text{ N} = F_g$$

Required to trail 1800kg up ~~(1.84)~~ 1.84 gradient

Assuming MAX ACCELERATION = $0.1g = a$
(comfortable riding)

$$F_{acc} = (m_c + m_r) \cdot a$$

$$F_{acc} = 2724 \times 0.1 \times 9.81 = 2672 \text{ N}$$

$\therefore$ **TRACTION EFFORT = 3605 N.**

$$TE = \Sigma F_g + F_{acc} = 2672 + 932.6 = 3604.8 \text{ N}$$

(can optionally include rolling resistance force = 466N)

Appendix F-5: Motor Mount Linking Arm Calculations (DS)

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03/12/21

Motor Mount Arm length

★ For Exhibition Pack ★

Arm length = 306mm
with 3° conical angle bushing ★

★ For Railway Challenge ★

Yaw displacement = 6.82mm
so with 3° conical freedom bushing
Arm length = 130mm

Forces in connecting arm 

Force = 1552 N

$\sigma = \frac{F}{A} = \frac{1552}{A}$ if $\sigma_y = 100 \text{ N/mm}^2$
Area of weld = 15.52 mm²

minimum = 130 mm = l



Area = 923 mm

23 mm

6 mm

filled weld

Area = 1700 mm²

Appendix F-6: Friction Coupling Safety Factors (DS)

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Friction Coupling Forces & Safety Factors

The bogie utilises THREE locking assemblies and THREE bushes (taper locking)

Locking Assemblies

Component	Location
(1) OKBS11 - 40x76 Internal Locking Assembly	Wheel → Axle
(2) RFN - 7012 - 40x65 Ringfeder Locking Assembly	Pulley (large) → Axle
(3) TLK110 20x28 Locking Bush	Anti-Roll Bar

Tapered Bushings

Component	Location
(1) 2517 65mm Taperlocking Bush $D_i = 65\text{mm}$ $D_o = 85.73\text{mm}$	Fills space between ringfeder & large pulley
(2) 1008 $\frac{7}{8}$ Taper Bush (Dunlop) $D_i = 22.23\text{mm}$ $D_i = 0.875"$ or $\frac{7}{8}"$ $D_o = 1.386"$	Motor shaft to → Small pulley sprocket.
(3) 1108 $\frac{7}{8}$ Taper Bush (OPTIBELT) $D_i = \frac{7}{8}"$ $D_o =$	Motor Shaft to → Veebelt brake pulley.

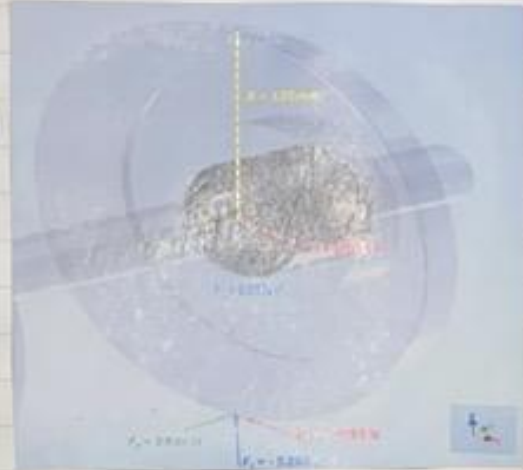
OKBS11 - 40x75 Locking Assembly. 24

Using forces calculated
on page 11. of
calculations logbook (4)

$$F_r = \sqrt{F_x^2 + F_z^2}$$

$$F_r = \sqrt{2962^2 + 5253^2}$$

$$F_r = 6031 \text{ N}$$



$$\left. \begin{array}{l} F_x = 2962 \text{ N} \quad F_y = 3924 \text{ N} \quad F_z = 3924 \text{ N} \quad F_r = 6031 \text{ N} \\ T_{\max} = 118 \text{ Nm} \end{array} \right\} \text{ FORCES ACTING ON OKBS11.}$$

Safety factor of OKS11.

Applied Torque	Max Torque Possible	Safety factor
118 Nm (172 Nm) pg 33	2880 Nm	$\frac{2880}{118} = 24.4$

Pressure Safety Factors

Pressure location	Pressure Exceeded	Safety factor
Hub	150 MPa	$\frac{400}{150} = 2.7$
Shaft	281 MPa	$\frac{400}{281} = 1.42$ $\frac{600}{281} = 2.1$

Safety Factor = yield strength / surface pressure exerted by locking mechanism

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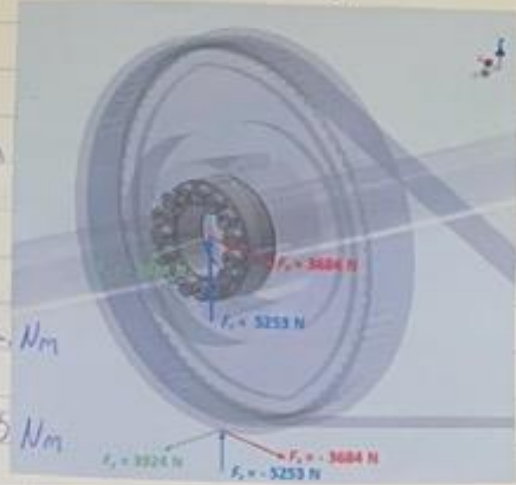
RfN 7012 Ringfeder 40 x 65 Safety Factor

Ringfeder Specification:

Transmissible Torque $T = 1154 \text{ Nm}$

Transmissible Axial Force = 58 kN

Max Transmissible Torque = 1404 Nm

Shaft Surface Pressure $p_w = 225 \text{ N/mm}^2$ Hub Surface Pressure $p_u = 139 \text{ N/mm}^2$ Safety Factor of Ringfeder 7012 40 x 65

	Factor	Applied Value	Allowable Value	Safety Factor
T	Torque	118 Nm (38 Nm)	1154 Nm	9.78
P_w	Shaft Pressure	225 N/mm ²	600 N/mm ²	2.67
P_u	Hub Pressure	139 N/mm ²	480 N/mm ²	3.45
$F_y = F_a$	Axial Force	3924 N	58000 N	14.78
T_{max}	Max Torque	$3684 \text{ N} \times 0.1 \text{ m}$ $= 368.4 \text{ Nm}$	1404 Nm	3.81

Hub made of cast iron w/ $\sigma_y = 480 \text{ MPa}$ Max applied torque = max force (longitudinal) $\times$ radius of pulley.

TLK 110 20x28 Safety Factors ²⁸

Factor	Applied Value	Allowable Value	Safety Factor
Torque	79 N/m	220 N/m	2.53
Axial Force	19816 N	22000 N	1.11 (crushing)
Shaft Pressure	160 N/mm ²	275 N/mm ²	1.72
Hub Pressure	115 N/mm ²	275 N/mm ²	2.39

Appendix F-7: Maximum Motor Torque (DS)

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Motenergy 1718 Calculations for shaft Torques

✓ cables (Data, power +, power -, gnd.)

Motor Characteristics PHAC

Mass = 10.9 kg

Continuous Current Rating = 100 A

168 mm

Peak current ratings = 300 A
for 30 → 120 seconds

Nominal Speed = 4000 rpm

Input Voltage = 24 → 72 VDC ★ 48 VDC ★

CALCULATIONS

Important Equations:

- (1) Power = Voltage × Current = $P = IV$
- (2) Power = Force × velocity = $P = Fv$
- (3) Torque = Force × radius = $T = Fr$
- (4) Angular Velocity = $\frac{\text{velocity}}{\text{radius}} = \omega = \frac{v}{r}$
- (5) → Reduction Ratio = $\frac{\text{teeth on driven}}{\text{teeth on pinion}} = \frac{N_2}{N_1}$

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Verifying Maximum Motor Torque

$$m = \frac{(y_2 - y_1)}{(x_2 - x_1)} \quad \text{using: } \begin{matrix} (2, 20) \\ (12, 100) \\ (16, 130) \end{matrix}$$

$$m = \frac{(130 - 20)}{(16 - 2)}$$

$$m = 7.86$$

$$y_2 - y_1 = m(x_2 - x_1) + c$$

$$(y - 130 = 7.85(x - 16))$$

$$y - 130 = 7.85(x - 16) + c$$

$$y = 7.85x + 4.4$$

$$x = \frac{y - 4.4}{7.85}$$

$$\left\{ \begin{array}{l} x = 37.656 \text{ Nm of torque / motor} \\ \text{using } y - y_2 = m(x - x_2) + c \end{array} \right\}$$

@ 300A

Each motor exerts $\approx 38 \text{ Nm}$ of Torque

• MAX MOTOR TORQUE •

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Using the reduction ratio of 3.63

$$\frac{80}{22} \text{ teeth} = 3.63 \quad (T = f \cdot d)$$

$$\text{MAX Torque exerted on shaft} = 3.63 \times 38 \text{ Nm}$$

@ 300A. Max Torque by Large pulley = 138 Nm

★ $\therefore$ Max Torque on friction couplings = 138 Nm. ★

$$\text{Max Tractive Effort} = 3605 \text{ N}$$

$$\text{Torque required to meet max TE} = 450.6 \text{ Nm}$$

$$\text{Torque required / axle} = 113 \text{ Nm} \quad \checkmark \text{ possible}$$

$$\text{Torque @ motor shaft} = 31.12 \text{ Nm}$$

$$\text{setting } \frac{x}{y} = 31.12 \text{ Nm} \quad \text{in} \quad x = \frac{y - 4.4}{7.85}$$

$$x = 31.12 \quad y = 7.85x + 4.4$$

$$y = 7.85 \times 31.12 + 4.4$$

$$y = 248.7 \text{ A}$$

so 249 Amps required for max tractive effort of 3605 N exerted by motors (each one)

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Torque @ Large Pulley & Wheel.

Torque pulley (large) max = 138 Nm

$$\text{Force} = \frac{T}{d} = \frac{138}{0.1} = 1380 \text{ Nm}$$

$$F_x = 1380 \text{ Nm}$$

$$\text{Torque @ wheel} = 1380 \times 0.125$$

$$T_{\text{wheel}} = 172.5 \text{ Nm}$$

Appendix F-8: Axle Box Calculations (DS)

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Safety Factors of Axle Box

x direction:

$$S_f = \frac{\text{Permissible Shear Force}}{\text{Applied Force}}$$

* $S_{f,x} = \frac{29700}{2962} = 10.0$

* $S_{f,y} = \frac{29700}{3924} = 7.57$

* $S_{f,z} = \frac{29700}{6031} = 4.92$

Safety Factors of Secondary suspension struts

$$F_{\text{shear}} = \frac{\sigma_y \times A}{2}$$

$$F_{\text{shear}} = \frac{275 \times 60 \times 66.5}{2} = 548.6 \text{ kN (unrealistic)}$$

$$F_{\text{shear}} = 275/2 \times 16 \times 25 = 55 \text{ kN}$$

$$S_{f,z} = \left[\frac{55000}{29700} \right]; \frac{55000}{2962} = 18.6$$

* $S_{f,y} = \frac{55000}{202 \times 9.81 \times 10} = 2.78$

* $S_{f,x} = \frac{55000}{202 \times 9.81 \times 10} = 2.78$

Weld Calculations for Axle Box 46

Each T bar should support the weight of $\frac{1}{2}$ the wheelset

Wheelset mass = 70 kg

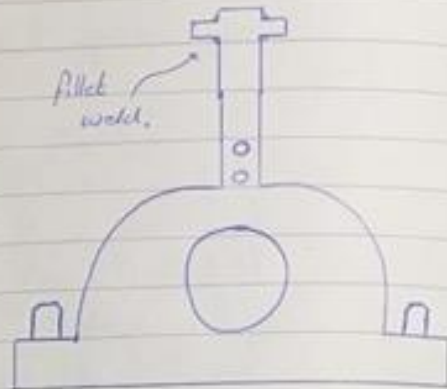
$$\text{Force} = 70 \times 9.81 = 686.7 \text{ N}$$

Assuming σ_t of weld = 100 N/mm^2

$$\sigma = \frac{F}{A} \quad \therefore \quad A = \frac{F}{\sigma}$$

$$\text{Area} = \frac{686.7}{100} = 6.867 \text{ mm}^2$$

Area of weld required : 7 mm^2



Appendix F-9: Primary Spring Calculations (DS)

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Primary Suspension FOS

Primary Spring Characteristics-

$D_o = 30.48 \text{ mm}$
 Part number = LHM 1250A 06
 Free length = 88.9 mm
 Rate (N/mm) = 24.519 N/mm
 $\Delta x = 45.97 \text{ mm}$ of compression allowed
 Solid height = 42.93 mm
 Lateral Stiffness = 6.1 N/mm (From FEA)
 Force to Pulley compress using Hookes: $F = k \Delta x$

$$F = 24.519 \times 45.97$$

$$\text{Force} = 1128 \text{ N}$$

Force exerted on each spring:

$$\text{Force} = \frac{\text{body mass} \times 2g + 2(\text{hogie mass} \times 5g)}{16}$$

$$\text{Force} = \frac{(500 \times 2 \times 9.81) + 2(202 \times 5 \times 9.81)}{16}$$

$$\text{Force} = 1851.6 \text{ N} \quad (\text{Force} = 554 \text{ without accelerations})$$

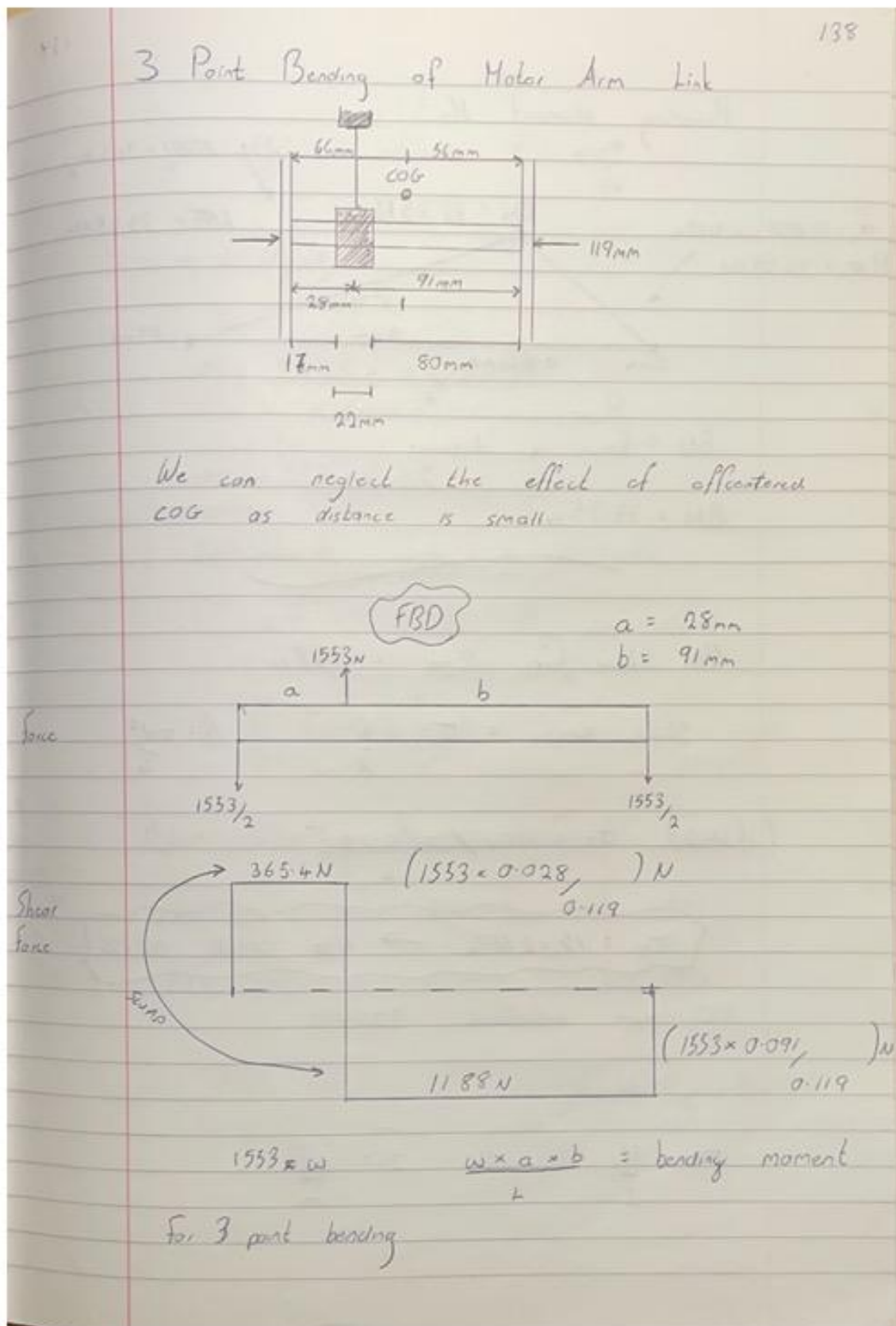
Safety factor of primary suspension = 0.609

Compression $\Delta x = 45.5 \text{ mm}$ (bottomed out)

$$Fos = \frac{1128}{1851.6} = 0.609 \quad Fos = \left[\frac{554}{1128} \right] = 2.04$$

$\Delta x = 22.59 \text{ mm}$ compression

Appendix F-10: Motor Mount Connection Bolt (DS)

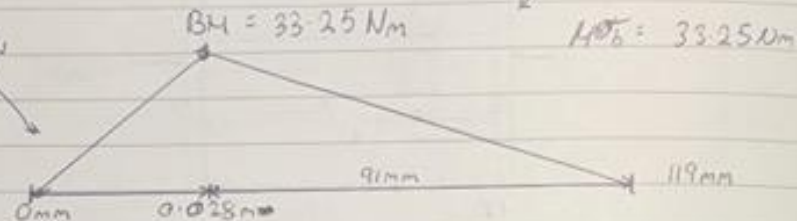


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Bending Moment M_b :

$$H = 0.028 \text{ m} \times 1188 \text{ N}$$

$$M_{\sigma_b} = 33.25 \text{ Nm}$$



Shear

$$BM = \text{force} \times \text{distance}$$

$$BM = 33.25 \text{ Nm}$$

$$\text{Max Shear force } F_{\text{max}} = 1188 \text{ N}$$

$$\therefore \text{Shear stress} = \sigma_r = \frac{F}{A} \quad A = \frac{\pi d^2}{4}$$

$$(\text{if H10}) \quad \sigma_r = 1188 / \frac{\pi (0.010)^2}{4}$$

$$\sigma_r = 15.12 \text{ MPa} \rightarrow \text{MAX SHEAR STRESS}$$

H10 can withstand 300 MPa

Bending Stress:

$$\sigma_b = \frac{M_y}{I} \quad \text{with} \quad I = \frac{\pi d^4}{64}$$

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$$\sigma_b = \frac{My}{I} \quad \text{where} \quad I = \frac{\pi d^4}{64}$$

$$\sigma_b = \frac{My \cdot 64}{\pi(d^4)}$$

$$\sigma_b = \frac{33.25 \times 0.005 \times 64}{\pi(0.01)^4}$$

$$\sigma_b = 338.68 \text{ MPa}$$

Withstandable stress = 600 MPa

$\therefore$ M10 is suitable for nose mount

SUMMARY

	Actual Stress	Allowable Stress	Check
τ	15.12 MPa	300 MPa	PASS
σ_b	338.68 MPa	600 MPa	PASS

M10 BOLT IS OKAY FOR MOTOR
MOUNT CONNECTION

Appendix G: Railway Challenge Required Calculations (JO, DS)

Appendix G-1: Dynamic Loading Gauge (JO, DS)

Dimensions

$H_1 =$
 $H_2 =$
 $H_3 =$
 $H_4 = 594 \text{ mm}$
 $W_1 = 425 \text{ mm}$
 $W_2 = 484 \text{ mm}$

Secondary suspension $K_V = 41.2 \text{ N/mm}$
 $K_L = 25.7 \text{ N/mm}$
 Primary suspension $k_V = 24.5 \text{ N/mm}$
 $k_L = 6.1 \text{ N/mm}$

Systems contributing to Kinematic envelope:

- Horizontal curve
- Vertical curve
- Secondary bounce
- Primary bounce
- Lateral primary
- Lateral secondary
- Angle primary
- Angle secondary

Lateral

Primary lateral stiffness

$16 \times 6.1 \text{ N/mm} \rightarrow 97.6 \text{ N/mm}$

$1000 - (4 \times 45 \text{ kg}) = 820 \text{ kg}$

$594 - 125 = 469 \text{ mm}$

$2g = 13988 \text{ N} \rightarrow 143 \text{ mm} \leftarrow \text{use limit}$

curving $15 \text{ km/h} = 4.17 \text{ m/s}$ $R = 20 \text{ m}$

$F = \frac{mv^2}{R} = 713 \rightarrow 7.3 \text{ mm}$

adjust to allow $\pm 10 \text{ mm}$

Secondary lateral stiffness

$8 \times 25.7 \text{ N/mm} = 205.6 \text{ N/mm}$

600 kg

$594 - 206 = 388$

$2g = 11772 \text{ N} \rightarrow 57.3 \text{ mm}$

$15 \text{ km/h} = 4.17 \text{ m/s}$ $R = 20 \text{ m}$

$F = \frac{mv^2}{R} = 522 \text{ N} \rightarrow 2.5 \text{ mm}$

limit arms $\pm 30 \text{ mm}$
adjust to $\pm 20 \text{ mm}$

side of springs $\approx 11 \text{ mm}$
at $\frac{1}{2}$ height
 $\approx 22 \text{ mm}$ at top

Roll

Primary roll stiffness N/m/deg
 per-side $8 \times 24.5 \text{ N/mm}$ @ 212.5 mm

$$1^\circ \text{ requires } 212.5 \sin 1^\circ = 3.71 \text{ mm}$$

$$727 \text{ N} \times 0.215 \text{ m} = 154.5 \text{ Nm}$$

$$309 \text{ Nm/deg}$$

$$U \text{ of cog } 594 - 125 = 469 \text{ mm}$$

Will tip over 2318 N @ $\text{CoG } 594 \text{ mm}$

$$2318 \text{ N at } 594 \text{ mm} = 1377 \text{ Nm}$$

$$\frac{1377}{309} = 4.46^\circ \rightarrow 79.2 \text{ mm}$$

$$993 \times \sin 4.46^\circ \text{ m}$$

$$F = \frac{mv^2}{R} = 750 \text{ kg} \frac{4.17^2}{20} = 652 \text{ N}$$

$$652 \times 0.594 = 387 \text{ Nm}$$

$$\frac{387}{309} = 1.25^\circ$$

$$993 \times \sin 1.25^\circ = 21.7 \text{ mm}$$

Secondary roll stiffness Nm/deg

Per side $L \times 41.2 \text{ N/mm}$ @ 242 mm

$$1^\circ = 242 \sin 1^\circ = 4.22 \text{ mm}$$

$$1^\circ \text{ force } L \times 41.2 \times 4.22 = 696 \text{ N}$$

$$\text{moment } 696 \times 0.242 = 168.4 \text{ Nm}$$

$$337 \text{ Nm/deg}$$

Torsional stiffness in anti roll bar

$$\frac{\text{Torque}}{\text{Angle}} = \frac{\text{Nm}}{\text{radius}} = \frac{T}{\theta}$$

$$\frac{T}{\theta} = \frac{GJ}{L}$$

G - shear modulus
 J - polar moment inertia
 L - length of torsion bar
 θ - angle of rotation (rad)
 T - torque applied

$$G = 7.2 \times 10^{10} \text{ Pa} \quad \text{steel 275}$$

$$J = 1.194 \times 10^{-8} \text{ m}^4$$

$$L = 0.4 \text{ m}$$

$$\frac{T}{\theta} = \frac{7.2 \times 10^{10} \times 1.194 \times 10^{-8}}{0.4} = 2149.2 \text{ Nm/rad}$$

hollow bar stiffness 2149.2 Nm/rad

solid bar 2827 Nm/rad

arm length 120 mm

1 radian movement $120 \text{ mm} \rightarrow 2827 \text{ N}$

$$\frac{2827}{0.12} = 23.6 \text{ N/mm}$$

$$1 \text{ deg} \rightarrow 200 \text{ sin } 1^\circ = 3.49 \text{ mm}$$

$$3.49 \text{ mm} \times 23.6 = 82.4 \text{ N}$$

$$82.4 \times 0.2 = 16.5 \text{ Nm}$$

$$16.5 \text{ Nm/deg/bogie}$$

$$33 \text{ Nm/deg}$$

change $\hookrightarrow$ to 60mm

$$\frac{2827}{0.06} = 47.2 \text{ N/mm}$$

$$3.49 \text{ mm} \times 47.2 \text{ N/mm} = 164.5 \text{ Nm}$$

$$164.5 \times 0.2 = 32.9 \text{ Nm/deg/bogie}$$

$$65.8 \text{ Nm/deg}$$

$$65.8 + 337 = 402.8 \text{ Nm/deg}$$

$$\frac{600 \times 4.17^2}{20} = 521.7 \text{ N}$$

$$202.4 \text{ Nm}$$

$$993 - 206 = 787 \text{ mm}$$

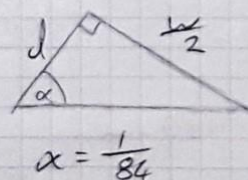
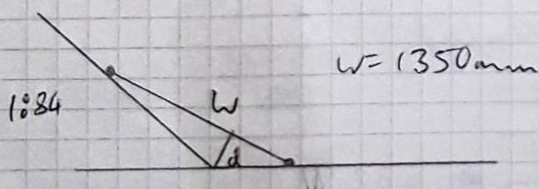
$$787 \sin 0.5 = 6.9 \text{ mm}$$

$$594 - 206 = 388 \text{ mm}$$

$$\frac{202.4}{402.8} = 0.501$$

lateral displacement

Vertical curve

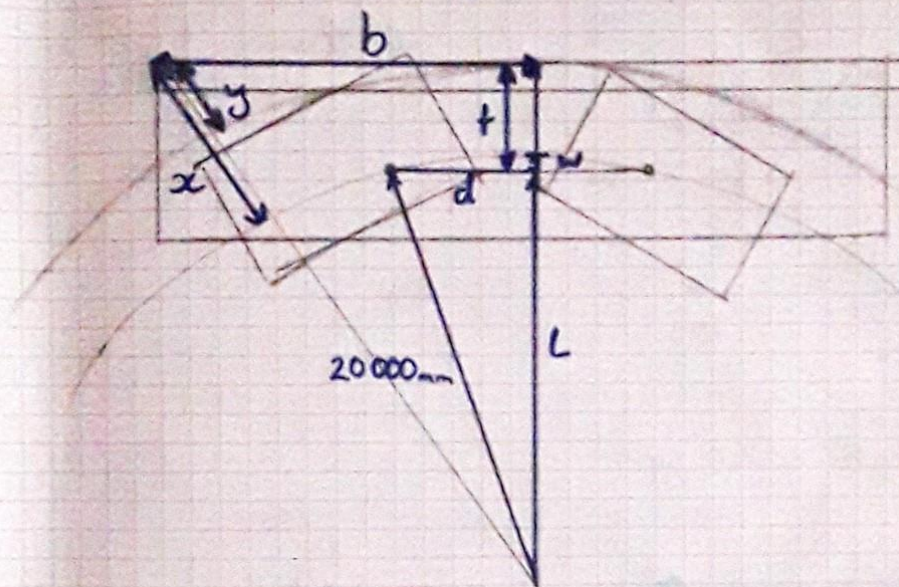


$$\frac{1}{84} \times \frac{1350}{2} = 8.04 \text{ mm}$$

$$\frac{1350}{2} \times \frac{1}{84}$$

vertical movement upwards

Horizontal curve



$$L = \sqrt{20000^2 - d^2}$$

$$w = 20000 - L$$

$$(20000 + x) = \sqrt{(L + t)^2 + b^2}$$

$$y = x - t$$

$$d = 675 \text{ mm}$$

optimum bogie distance
 $w = y$

$$L = 19988.6 \text{ mm}$$

$$w = 11.39 \text{ mm}$$

$$t = \frac{562.2}{2} = 281.1 \text{ mm}$$

body at full extent

$$b = 1350 \text{ mm}$$

$$x = 314.6 \text{ mm}$$

$$y = 33.5 \text{ mm}$$

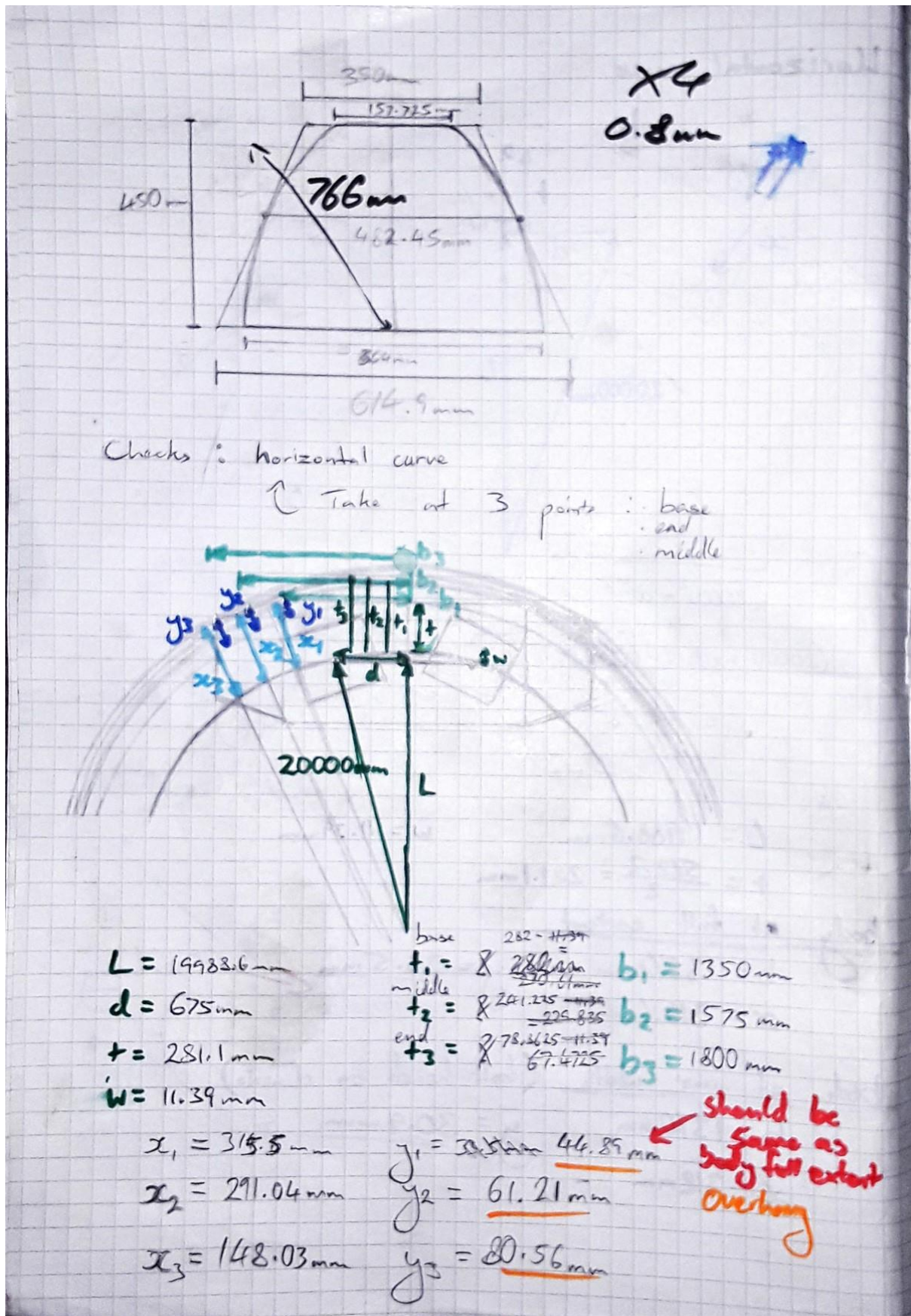
overhang

body at max width (balt heads on chassis)

$$b = 1310 \text{ mm}$$

$$x = 312 \text{ mm}$$

$$y = 30.9 \text{ mm}$$



Available space is 710mm

Body width (excluding bolt heads) - start of nose = 545.974mm

Maximum body width = 562.2mm

Space allowed in dynamic loading gauge = $\frac{710 - 562.2}{2} = 73.9 \text{ mm}$

Distance outside dynamic loading gauge $\frac{710 - 562.2}{2}$ or = 82.013mm

Start of nose

$$10 + 15 + 6.8 + 4.1 + 44.85 = \underline{80.79 \text{ mm}}$$

Mid nose

$$10 + 15 + 6.8 + 4.1 + 61.21 = \underline{97.11 \text{ mm}}$$

End nose

$$10 + 15 + 6.8 + 4.1 + 80.56 = \underline{116.46 \text{ mm}}$$

Values vary dependant on starting width of nose. Excessively over in both regards to dynamic loading gauge. Soln - shorten & thin nose

should be same as at edge of loco. Soln - change width of nose plate to match

Hard limits on length: Noses must be identical
 $> 156 \text{ mm}$ min width $> 148 \text{ mm}$
 Recalculating approximate dimensions

Assumption: best case scenario is true scenario
 i.e. body width is 545.974mm at ends of loco

Start of nose ✓

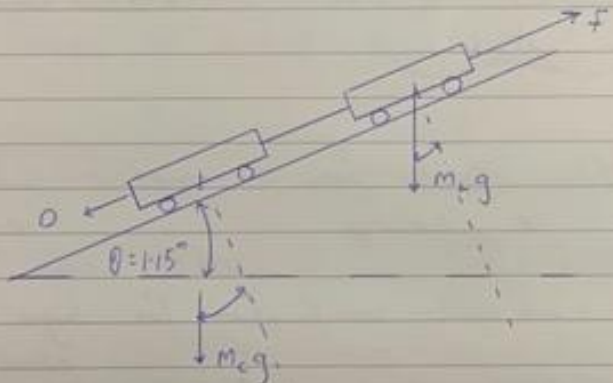
Appendix G-2: Power Requirements (DS)

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Energy Requirement for Challenge 2

The Locomotive shall be able to haul & start a trailing load of up to 1800kg on a 2% (1.15°) gradient.

What to do? → Find Amps pulled from motor ✓
 → How long it can sustain this for ✓
 → Torque required @ wheels & thus motor shaft. ✓



$m_L = 1000 \text{ kg}$
 $m_T = 1800 \text{ kg}$

For constant speed $\sum F_x = 0$

$$F_{\text{force}} = m_L g \sin \theta + m_T g \sin \theta$$

$$F_{\text{force}} = (m_L + m_T) g \sin \theta = (2800) \times 9.81 \times \sin(1.15)$$

$$F_{\text{force}} = 551.28 \text{ N.}$$

551.28 N is required to pull 1800kg up 2% gradient

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Converting 551.28 W into Amps from the 12V battery.

$$Force \times Velocity = Power = Fv$$

$$Current \times Voltage = Power = IV$$

$$Fv = IV \quad \text{Assuming velocity} = 15 \text{ km/h}$$

$$15 \text{ km/h} \quad \therefore I = \frac{Fv}{V} = \frac{551.28 \times 1.17}{12 \times 4 \times 2} = 23.95 \text{ Amps}$$

$$15 \text{ km/h} \quad \text{Current Pulled} = \frac{47.89}{2} \text{ Amps} \quad \text{MAXIMUM} \quad \text{WARRANT}$$

$$5 \text{ km/h} \quad I = \frac{F \times V}{V} = \frac{551.28 \times 1.39}{12 \times 4 \times 2} = 7.98 \text{ Amps}$$

$$5 \text{ km/h} \quad I = \frac{15.96}{2} \text{ Amps} \quad \text{REALISTIC} \quad \text{WARRANT}$$

✓	Velocity km/h	Velocity m/s	Current Amps
	5	1.39	7.98 Amps
✓	5 15	1.39	23.95 Amps

Both values are well below 100 Amp continuous rating.

231

Assuming operation braking 1800 kg @ 5 km/h
up a 2% gradient.

Force required = 552 N is required.

Torque required @ wheels.

552 N split over 4 axles = 138 N/
Axle

Torque / Axle = 17.25 Nm

Torque @ motor axle = $\frac{17.25}{3.63} = 4.74$ Nm

Using equation of motor characteristic curve

$$x = \frac{y - 4.4}{7.85}$$

y = current

x = torque = 1.86 Nm

$$y = x(7.85) + 4.4 \quad \text{Amps} = 14.0 \text{ Amps}$$

Converting maximum braking effort to current

Assuming 60 Volts input

232

Converting Max Tractive Effort To Current

$$F_{TE} = 3680 \text{ N}$$

$$F_{TE} / \text{Axle} = 3680 / 4 = 920 \text{ Nm}$$

$$\text{Max torque} = 920 \times 0.125 = 115 \text{ Nm}$$



Using reduction ratio

$$\text{Max torque @ motor before slip} = 115 \times \frac{22}{80}$$

$$\text{Max motor torque} = 31.625 \text{ Nm} \text{ before slip}$$

$$\text{Force} \times \left[\frac{\text{Distance}}{\text{Time}} \right] = \text{current} \times \text{voltage}$$

(Velocity)

$$\text{Current: } I = \frac{F \cdot v}{V} = \frac{3680 \times 4.16}{8 \times 12}$$

$$\text{Power} = \text{Force} \times \text{velocity} = 3680 \times 4.16 = 15308.8 \text{ W}$$

$$\text{Locomotive Power} = 15309 \text{ Watts}$$

$$\text{Power / Axle} = 3827 \text{ Watts}$$

$$\therefore \text{Max current @ locotive effort} = I = \frac{P}{V} = \frac{3827}{48}$$

$$I_{\text{max}} = 79.73 \text{ Amps}$$

Summary of Required Challenges

233

- ① + 400kg trailing load @ 5km/h up 2% gradient

Current Pulled = 2 Amps 23.9A @ 15km/h

- ② + 1800kg trailing load @ 15km/h up 2% gradient

Current Pulled = 11.96Amps

- ③ + 1800kg trailing load @ 15km/h up 2% gradient

Current Pulled = 3.99Amps

$$\text{To find current} : \frac{\text{Force} \times \text{velocity}}{\text{in miles}} = \frac{\text{power}}{4}$$

$$\left[\frac{\text{power}}{4} \right] / \text{voltage}$$

$$\therefore \text{Current} = \frac{\text{Force} \times \text{velocity}}{4 \times \text{voltage}}$$

- ④ + Max Tractive Effort (3680N) @ 5km/h

Current pulled = 26.6 Amps

- ⑤ + Max Tractive effort (3680N) @ 15km/h

Current Pulled = 79.86Amps

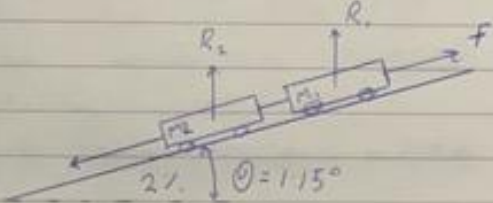
Appendix G-3: Battery Requirements (DS)

224

How much energy used?

How much battery energy will be used when:

- Operating for 3 hours travelling @ 5km/h on a 2% gradient trailing 1000kg of cargo.



$m_1 = \text{Assume } 1000 \text{ kg}$
 $m_2 = 100 \text{ kg}$

$$\text{Force} = (1000 \sin(2) \times 9.81) + (100 \sin(2) \times 9.81)$$

$$\text{Force} = 196.9 \text{ N} + 78.75 \text{ N}$$

Force = 275.65 N

- Force to maintain constant speed one @ 5km/h

We need to find the force to accelerate to 5km/h & how long this force is applied for to get energy during acceleration period.

★ WORK OUT POWER / AXLE &
 then USE 14.8 VOLTS

Energy Consumed

$$\text{Force} = 275.65 \text{ N}$$

$$\text{Power} = \text{Force} \times \text{velocity}$$

$$P = F \times v = 275.65 \times 5 \text{ km/h}$$

5000 m in 60 mins equals to...

83.3 m in 1 minute

1.388 m in 1 second

$$\therefore 5 \text{ km/h} = 1.39 \text{ m/s}$$

$$\text{Power} = 275.65 \times 1.39$$

$$\begin{aligned} \text{Power} &= 383.15 \text{ watts} \\ &= 0.514 \text{ horsepower} \end{aligned}$$

$$\text{Energy} = \text{Power} \times \text{Time}$$

$$E = 383.15 \times 3 \times 60 \times 60$$

$$E = 4.138 \times 10^6 \text{ Joules}$$

$$\text{Energy during 3 hour travel @ 5 km/h} = 4.14 \text{ MJ}$$

$$\frac{4.14 \times 10^6}{8} = 518 \text{ kJ per battery required for challenge}$$

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Current Pulled From Batteries

$$P = F \times v = IV$$

$$\left\{ \begin{array}{l} \text{if } V = 48 \text{ volts} \\ \text{per loco } \frac{1}{2} \end{array} \right. \quad \left\{ \begin{array}{l} \text{Force} = 276 \text{ N} \\ \text{velocity} = 1.39 \text{ m/s} \end{array} \right.$$

$$I = \frac{F \times v}{V} = \frac{276 \times 1.39}{48 \times 2} = \frac{797.25}{96} \text{ Amps} @ 5 \text{ km/h}$$

$$= 12 \text{ Amps} @ 15 \text{ km/h}$$

$$\text{Current Pulled} = 8 \text{ A} \quad \text{wrong.}$$

The batteries installed in the locomotive allow 70 Amps

• 4 batteries in series per locomotive half

5 km/h

$$\text{Current Pulled per locomotive half} = 4 \text{ A}$$

correct.

15 km/h

$$\text{Current Pulled per locomotive half} = 12 \text{ Amps}$$

Accelerations

$$F = ma$$

F = maximum tractive effort

$$F = \mu R = 1000 \times 0.375 \times 9.81$$

$$F = 3680 \text{ N}$$

$$a = \frac{F}{m} = \frac{3680}{(1000 + 400)}$$

$$a = 2.629 \text{ m/s}^2$$

★

It will take 0.529 seconds to accelerate from 0 m/s to 1.39 m/s (5 km/h)

227

Battery Capacity Vs Battery Requirements

$$\text{Energy} = \text{Power} \times \text{time} \quad E = Pt$$

$$\text{Power} = \text{Force} \times \text{velocity} \quad P = Fv = IV$$

$$\text{Force} = \text{mass} \times \text{acceleration} \quad F = ma$$

$$\text{Power} = 3680 \times 1.39 = 5115.2 \text{ Watts}$$

to accelerate from 0 to 5 km/h

$$\text{Energy} = 5115.2 \times 0.529 = 2705.94 \text{ J}$$

$$\text{Energy to accelerate to 5 km/h} = 2706 \text{ J}$$

$$E = ITV$$

of current batteries : Voltage = 12V
Amp hours = 75Ah

$$Q = IT \quad \text{charge} = 75 \text{ Ah} \rightarrow \text{Actually } 70 \text{ Ah}$$

$$\text{charge} = 270000 \text{ Coulombs}$$

$$\text{Energy} = \text{charge} \times \text{voltage} \quad E = QV$$

$$E = 270000 \times 12$$

$$\text{Energy / battery} = 3.24 \times 10^6 \text{ Joules}$$

$$\text{Energy / battery} = 3.24 \text{ MJ}$$

★

OR 0.9 kWh

★

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Energy Required

4.14 MJ

Battery Capacity

 $3.24 \times 8 = 25.92 \text{ MJ}$

$\therefore$ the locomotive can run for 18.8 hours
 carrying a heavy load up a 2% gradient

$$\frac{25.92}{4.14} = 6.26$$

$$3 \text{ hours} \times 6.26 = 18.78 \text{ hours}$$

* Notes Write Spec for traction centre
 & bushings page 55

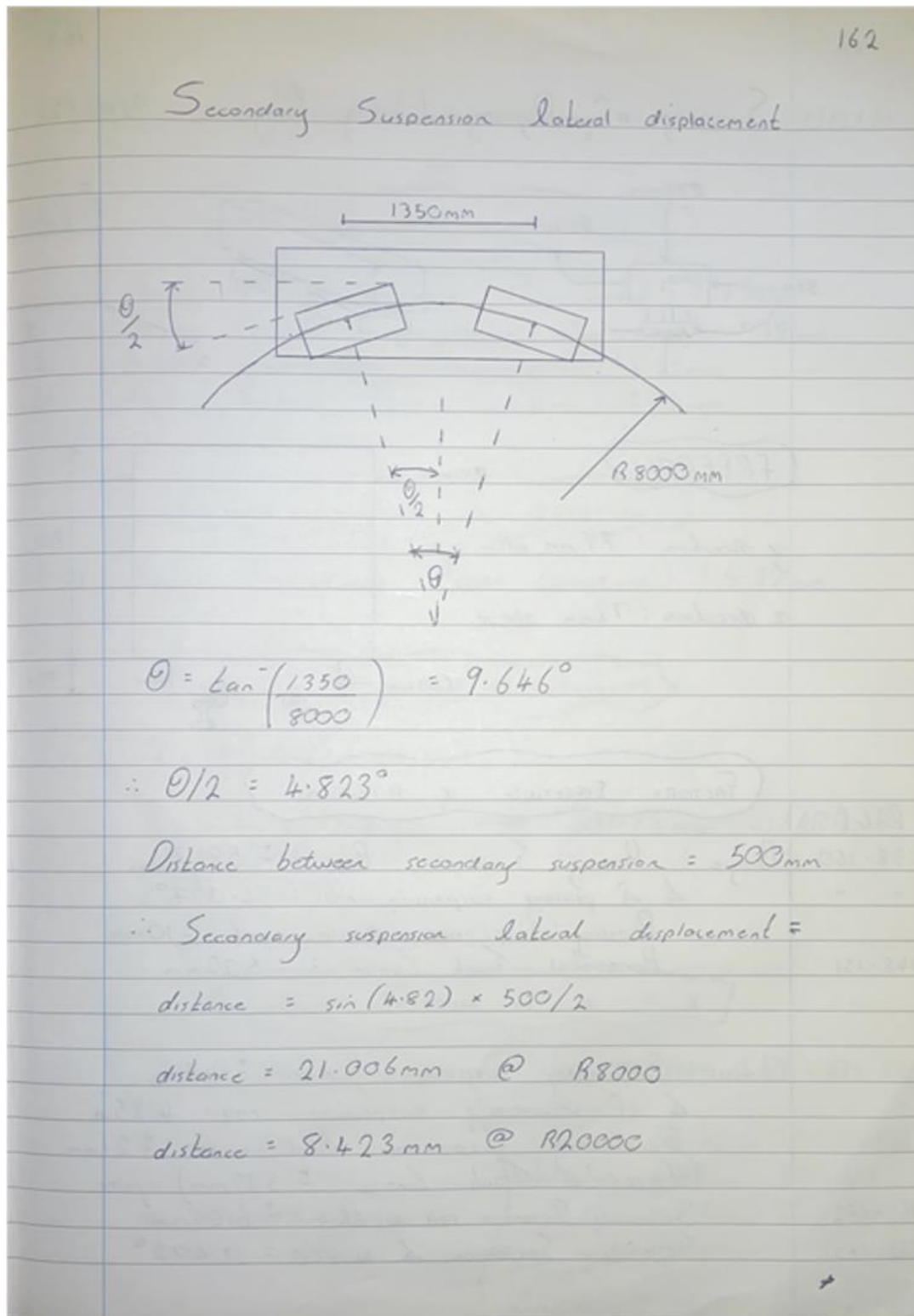
* write specification for bogie frame bump stops
 clearance holes

* traction centre specification

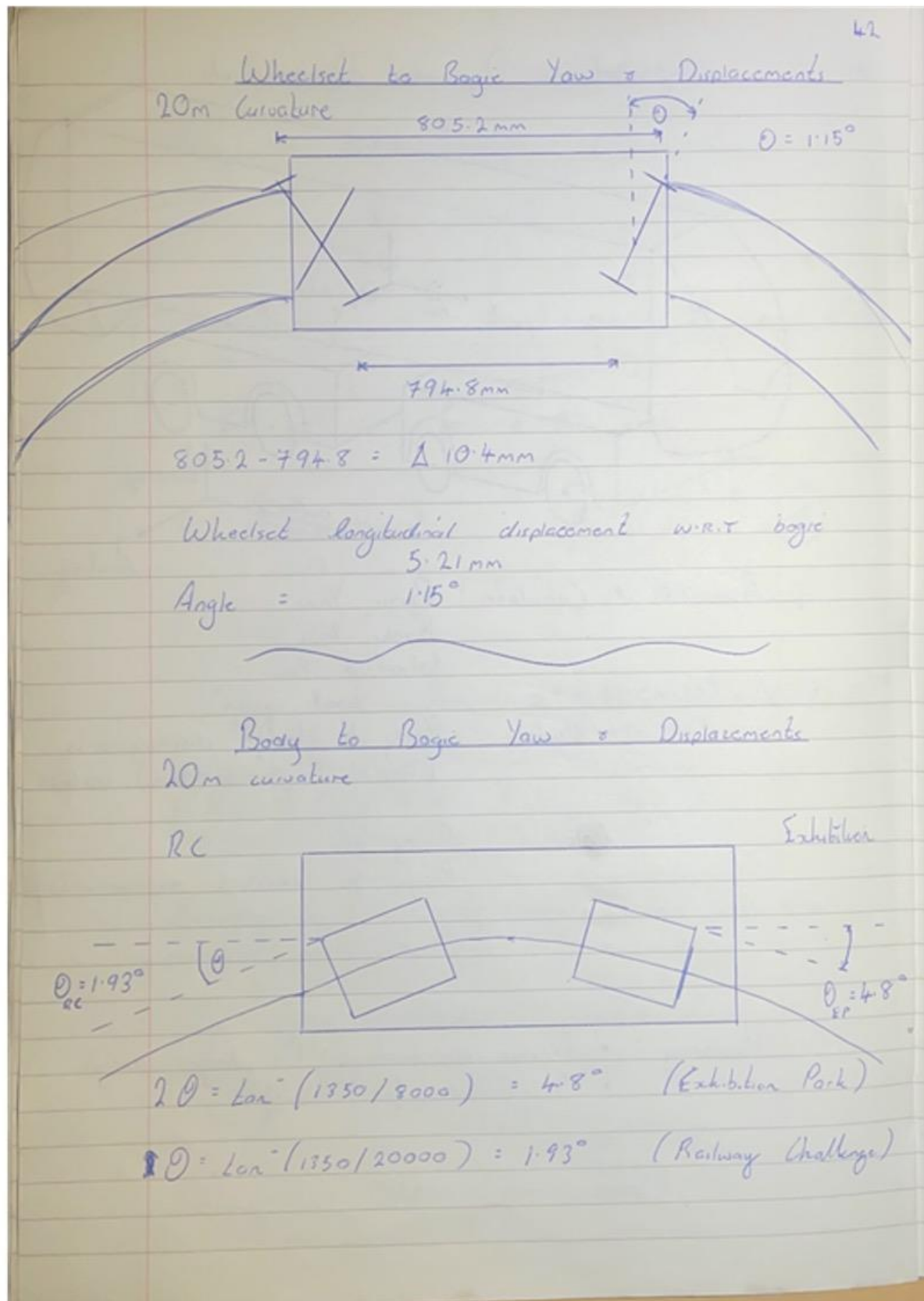
Bogie yaw & constraints

Body yaw

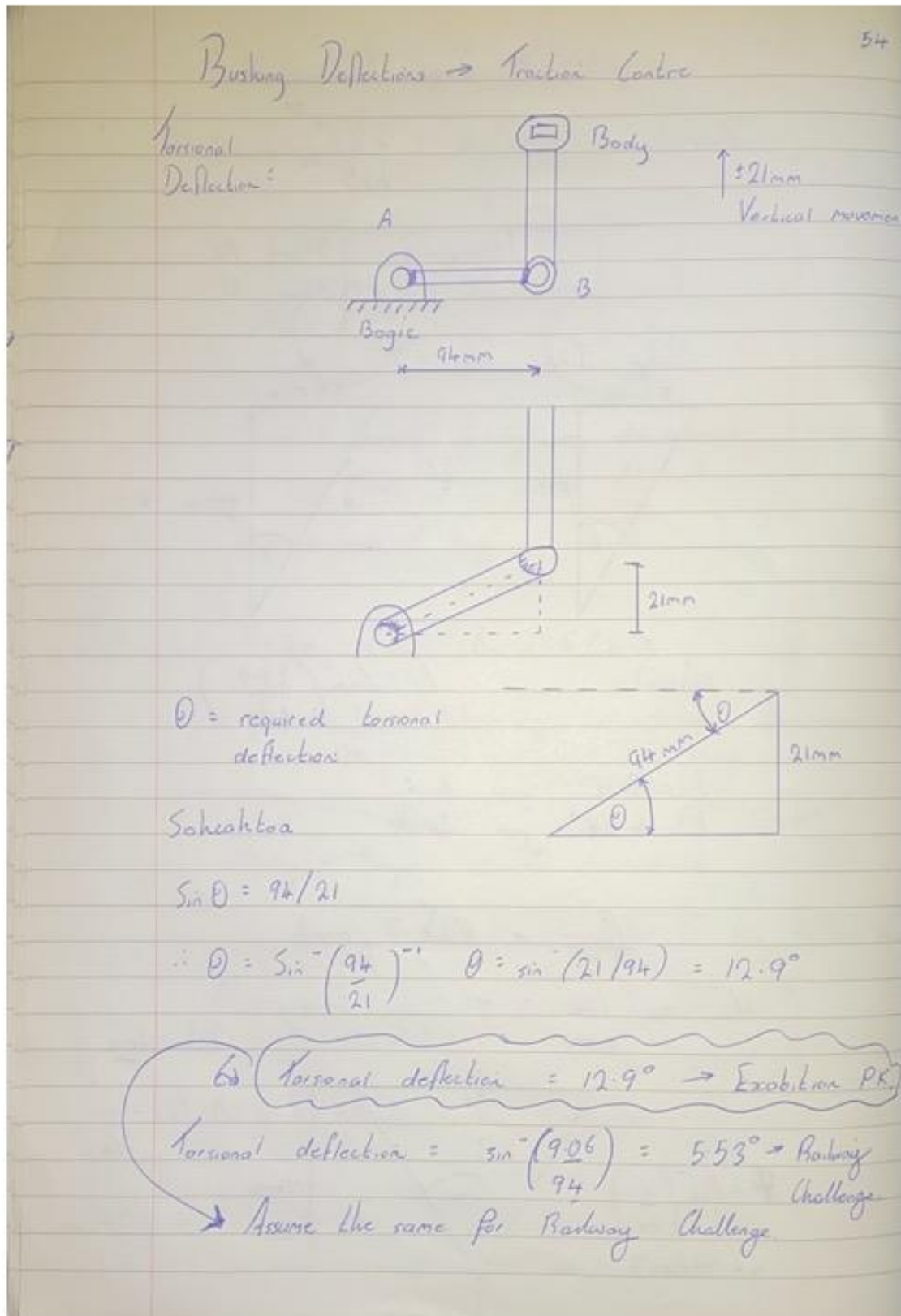
Wheelset yaw & constraints

Appendix H: Bogie Component Calculations (DS)**Appendix H-1: Bogie - Body Yaw (DS)**

Appendix H-2: Wheelset - Bogie Yaw (DS)



Appendix H-3: Traction Centre Bushing Calculations (DS)

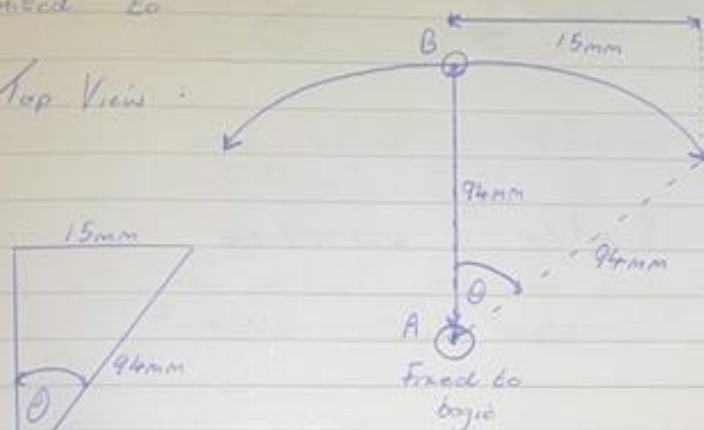


55

Traction Centre Bushing Conical Deflection

Secondary Suspension lateral displacement : $\pm 15\text{mm}$
limited to

Top View :

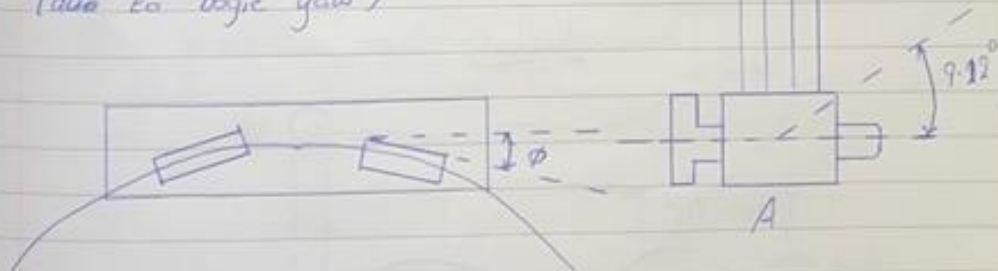


$$\sin \theta = 15/94$$

 $\therefore \theta = 9.28^\circ$ for bushing fixed to bogie

Bushing A : Conical deflection
of 9.12° minimum

Bushing B conical deflection :
(due to bogie yaw)



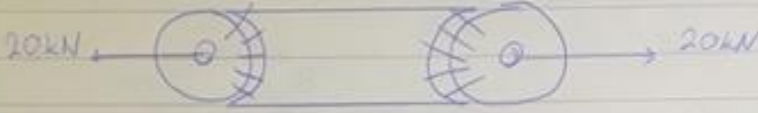
$$\phi = 4.823 \text{ (page 161)}$$

Bushing B conical deflection of 4.823° minimum

Appendix H-4: Motor Mount Linking Arm Bushing Calculations (DS)

56

WC Weld Area



$\sigma = F/A$ Assume weld $\sigma = 100 \text{ N/mm}^2$

$A = F/\sigma = 20000 / 100$

$\therefore \text{Weld area} = 200 \text{ mm}^2$

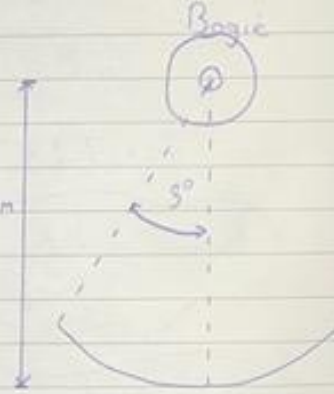
HMLA Bushing

• Allow Per :

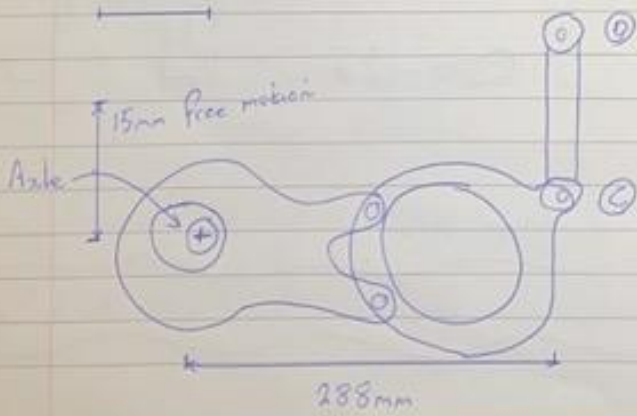
Vertical = 15mm

Longitudinal = 15.6mm

Lateral = 6.82mm
or 17.23 Per EP.



Vertical

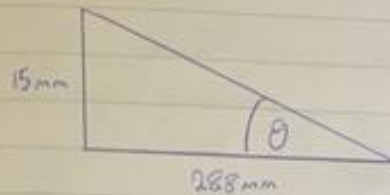


15.6mm

15mm Free motion

288mm

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θ = Bushing torsional deflection (bottom bushing)

$$\tan \theta = \left(\frac{15}{288} \right) \text{mm}$$

15mm vertical freedom

$\therefore \theta_c = 2.98^\circ$ of torsional deflection for bushing C.

Longitudinal

* 15.6mm longitudinal displacement is provided by bushing D's conical deflection.

Solution

$$\sin \theta = \frac{15.6}{24}$$

$$\theta = 7.23^\circ$$

$\theta_D = 7.23^\circ$ of torsional deflection for bushing D



Lateral Displacement

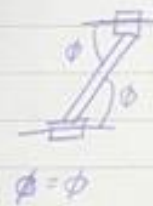
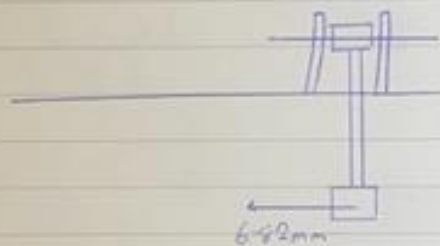
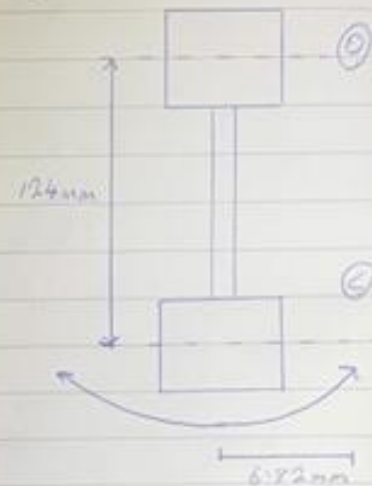
58

lateral displacement = 6.82mm due to bogie frame provided by D's conical deflection (17.03 for EP)



$$\tan \theta = 6.82 / 124$$

$\theta = 3.15$ degrees of conical deflection



Both bushing 1 & 2 need a conical ϕ deflection of 3.15° .

7.82° conical deflection for Exhibition Park

Say $6^\circ = \phi$ conical deflection

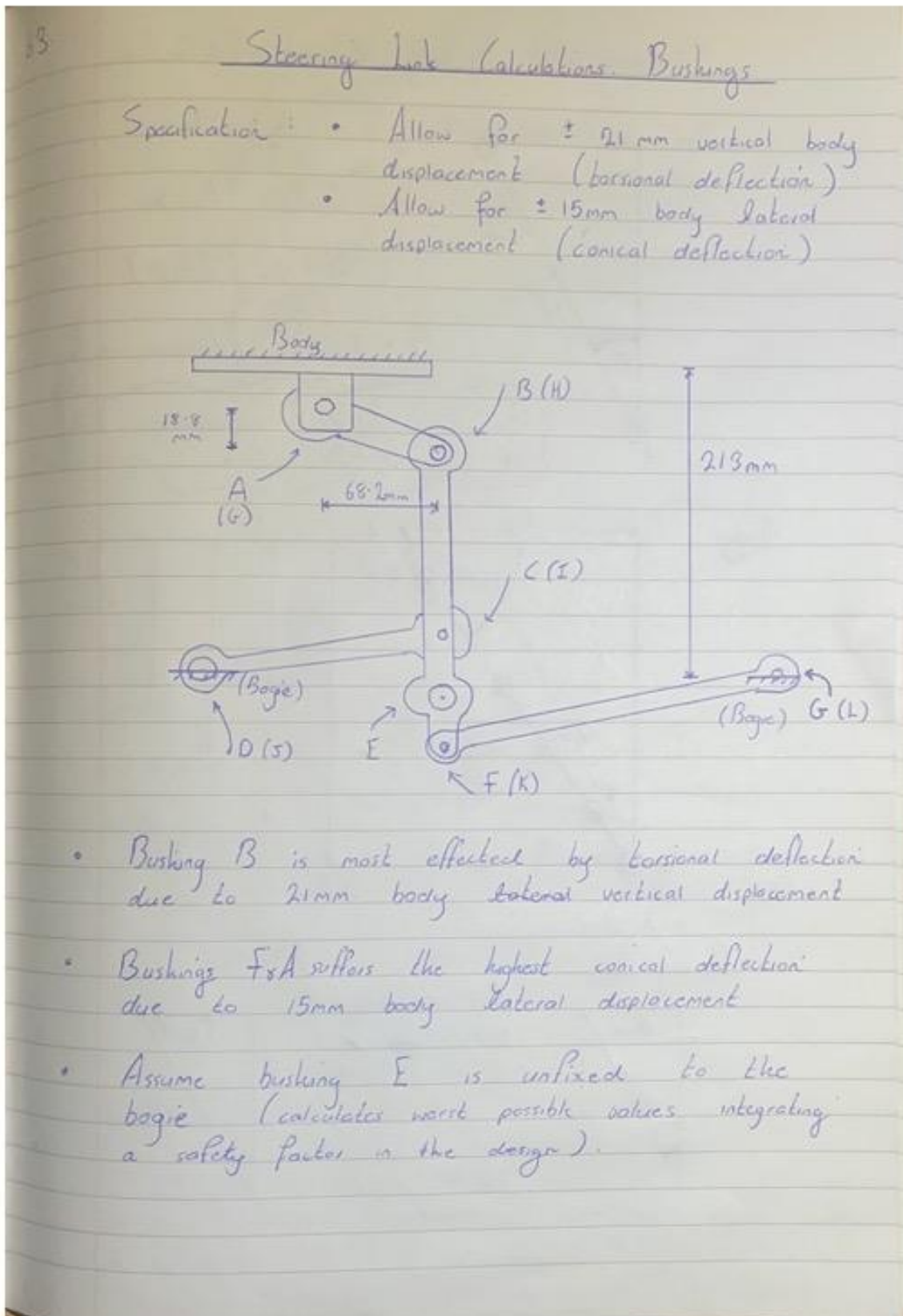
$$\text{Soh: } \sin(6) = \frac{17.03}{x}$$

$$\therefore x = 17.03 / \sin(6) =$$

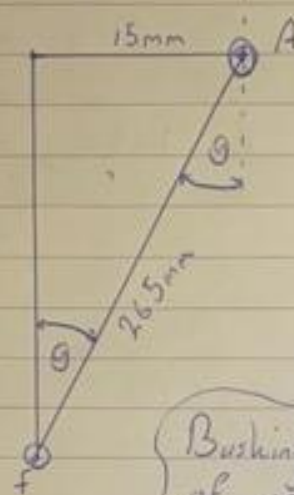
$$x = 163\text{mm}$$



Appendix H-5: Steering Link Bushing Calculations (DS)



Steering Link Bushing Conical Deflection



Solusio

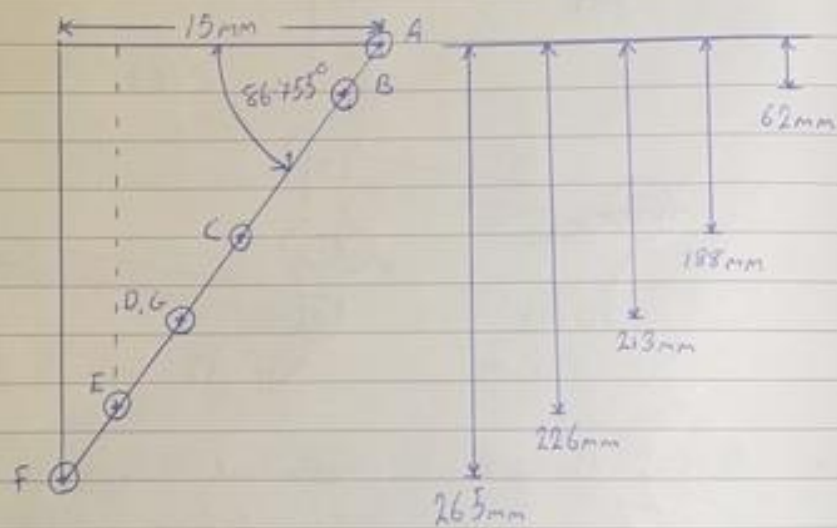
$$\sin \theta = 15 / 265$$

$$\theta = \sin^{-1}(15 / 265)$$

$$\theta = 3.245^\circ \text{ Conical deflection}$$

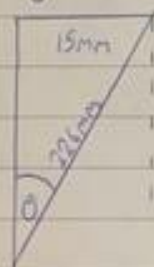
Bushing A & F experience 3.25° of conical deflection

Body:



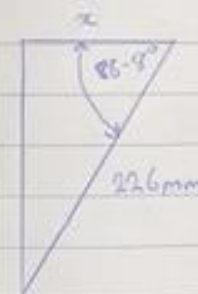
Axis
Body

Bushing E



$$\theta = \sin^{-1}(15 / 226)$$

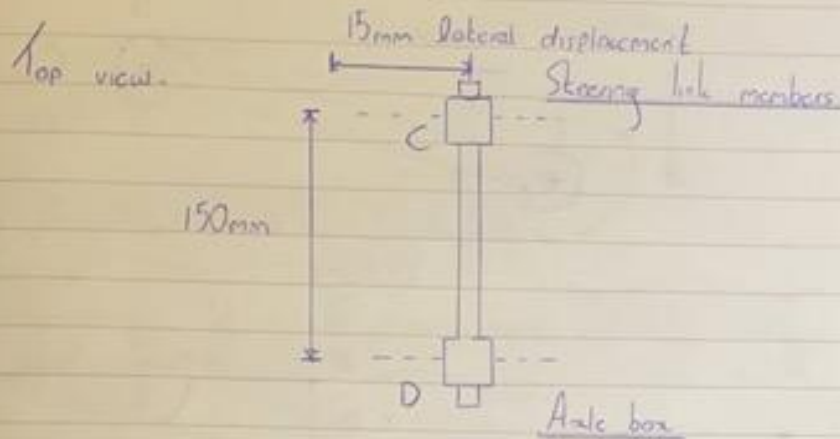
$$\theta = 3.8^\circ \times$$



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Steering Link Bushing Calculations

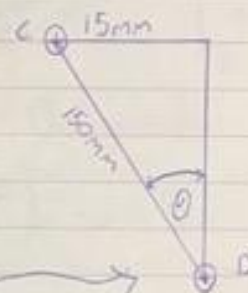
Bushing D & C will suffer severe conical deflection



Solution

$$\sin \theta = \frac{15}{150}$$

$$\therefore \theta = \sin^{-1}\left(\frac{15}{150}\right) = 5.74^\circ$$



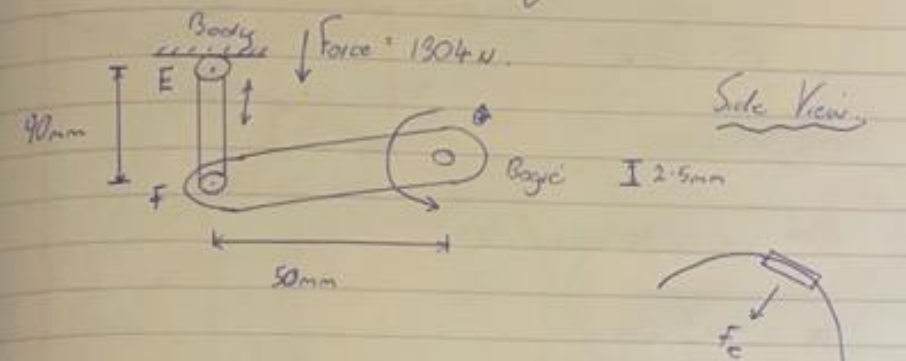
Bushing D & C require a conical deflection of 5.74°

Summary of Steering link bushes

Loc. Point	Bushing	Conical deflection	Torsional Deflection
(R)	E	- Ignore	- Ignore
(G)	A	3.25°	17.9°
(L)(K)	F	3.25°	3.06°
(H)	B	0° (Assume rigid)	17.9°
(J)	D	5.74°	4.56°
(I)	C	5.74°	4.56°
(L)	G	2.21°	3.06°

Appendix H-6: Anti-Roll Bar Bushing Calculations (DS)

59. Anti-Roll Bar Bushing Specification



Side View

Force = 1304 N

40 mm

25 mm

50 mm

Force = $\frac{mv^2}{r}$

$v = 14.17 \text{ ms}^{-2}$

$r = 8000 \text{ mm (EP)} / 20000 \text{ (RC) mm}$

$m = 600 \text{ kg}$

Force = $\frac{600 \times (14.17)^2}{20000} = 522 \text{ N}$

Force = $\frac{600 \times (14.17)^2}{8} = 1304 \text{ N}$

A force of 1304 N acts 50 mm about ^{Locking} bushing G

Torsional moment @ G = $1304 \times 50 \times 10^{-3}$

" " " " = 65.2 Nm.

Around LK110 locking bush.

Finding the deflection in bushing F.

Torsional Stiffness of ARB $\frac{\text{Torque}}{\theta(\text{rad})} = 2827 \text{ Nm/rad.}$

$\theta = \frac{\text{Torque}}{\text{Stiffness}} = \frac{65.2 \text{ Nm}}{2827 \text{ Nm/rad}}$

$\theta = 0.02306 \text{ radians} = 1.32 \text{ degrees.}$

Bushing F Torsional Deflection

60.

$$\theta = 1.32^\circ$$

$$\tan \theta = x/50$$



$$\text{Or } x = 50 \tan(1.32)$$

$$x = 1.15 \text{ mm}$$

∴ When cornering, body drops or rolls a vertical displacement of 1.15 mm

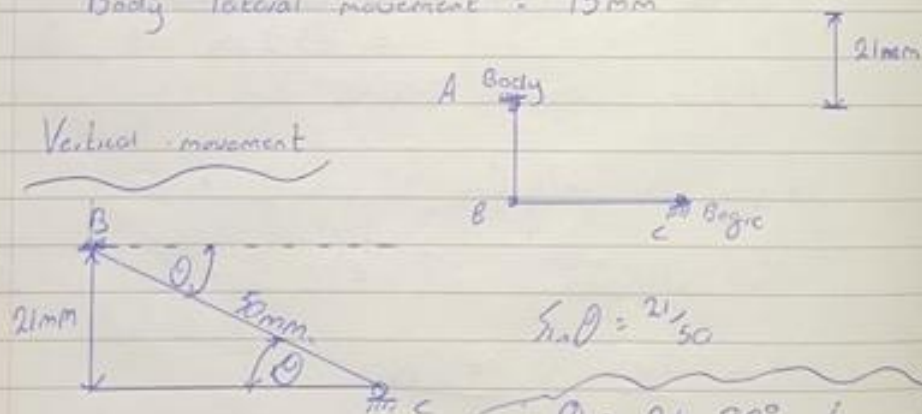
∴ Bushing F needs a torsional deflection of 1.32°
 ∴ a torsional moment of 65.2 Nm
 @ RC

∴ Bushings E & F should withstand a radial force of 1304 N .

Body Vertical movement = 21 mm

Body longitudinal movement = 0 mm

Body lateral movement = 15 mm

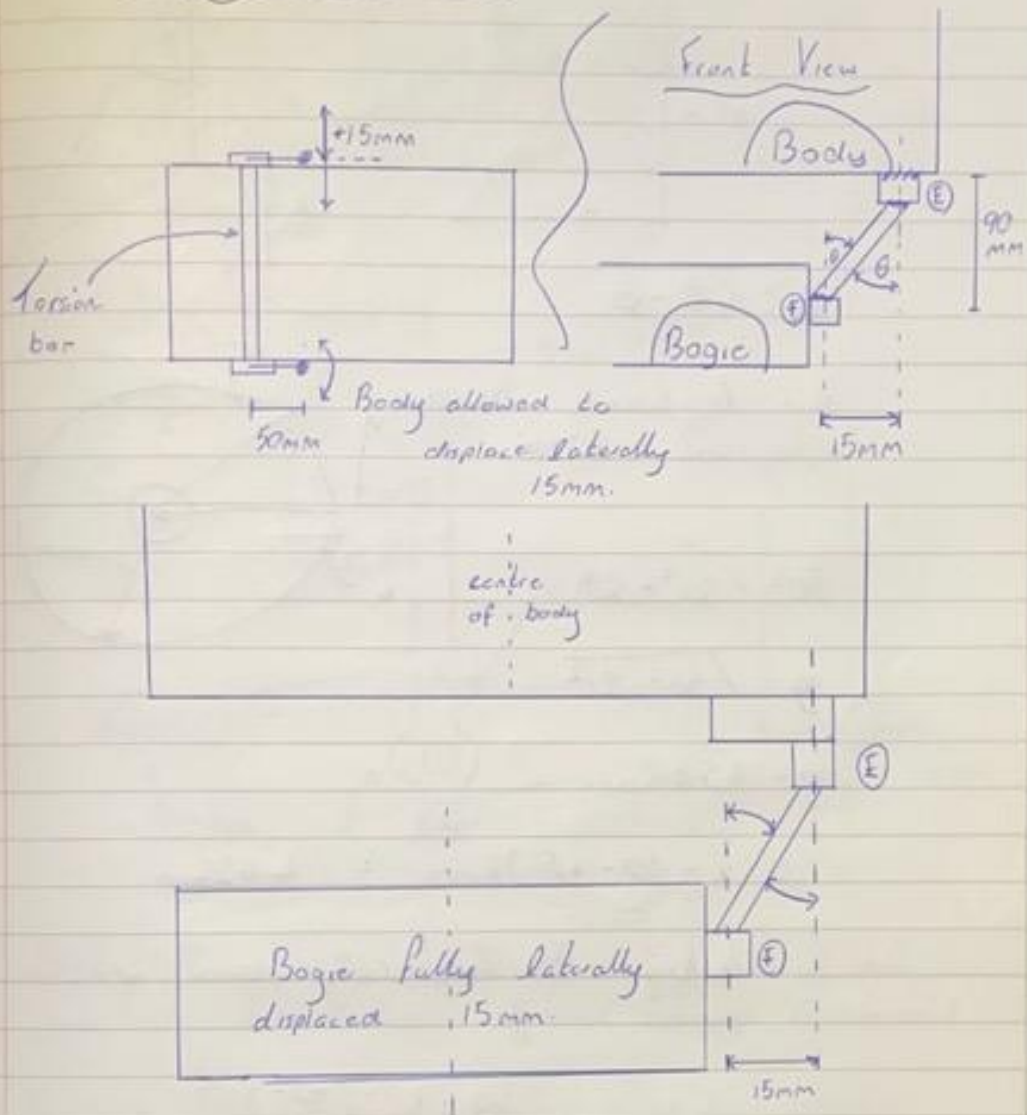


$$\tan \theta = 21/50$$

∴ $\theta = 24.83^\circ$ torsional deflection for bushing F.

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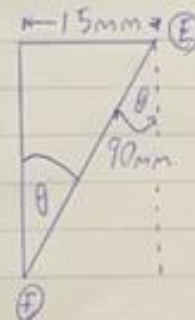
Bushing Conical Displacement Body lateral movement : 15mm



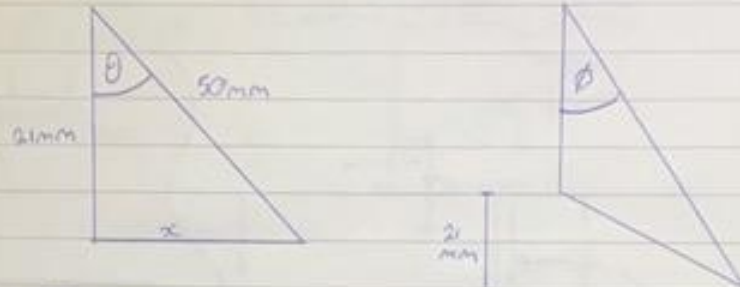
θ = Bushing E & F maximum necessary conical deflection

$$\theta = \sin^{-1}\left(\frac{15}{90}\right) = 9.59^\circ$$

Bushings E & F conically deflect 9.59°



62.



$$\Delta \Psi = \theta - \phi$$

then longitudinal displacement = $50 - x$

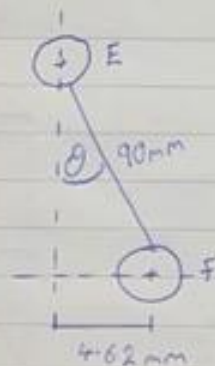
$$50^2 = 21^2 + x^2$$

$$x = \sqrt{50^2 - 21^2}$$

$$x = 45.38 \text{ mm}$$

$$\therefore x = 50 - 45.38 \text{ mm} = 4.62 \text{ mm}$$

So bushing E longitudinally moves 4.62 mm about its pivot



$$\theta = \sin^{-1}\left(\frac{4.62}{90}\right)$$

$$\theta = 2.94^\circ$$

Bushing E requires 2.94° of torsional deflection to account for the bush's $\pm 21 \text{ mm}$ displacement vertical

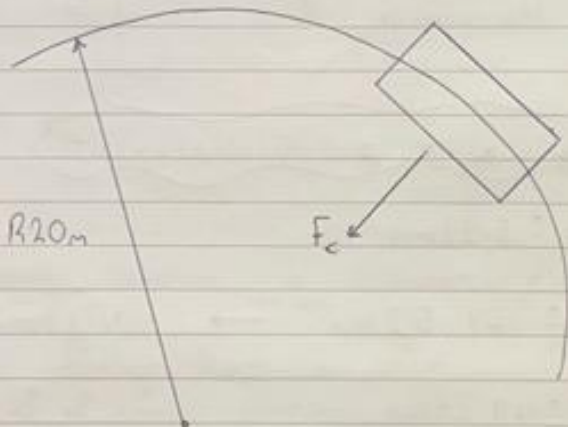
Appendix H-7: Anti-Roll Bar Calculations (DS)

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10/12/21

Anti-Roll Bar Deflection

- Find the centripetal force exerted by the locomotive when cornering the tightest corner @ maximum velocity 15 km/h.
- Use this value to calculate forces on ARB & find the corresponding deflection.
- This deflection value will replace the secondary suspension roll value.



$R = 20m$

F_c

$$F_c = \frac{mv^2}{r}$$

$$F_c = \frac{600 \times (4.17)^2}{20} \qquad F_c = \frac{600 \times (4.17)^2}{8} = 1304N$$

$$F_c = 521.67N \quad \rightarrow \text{Centrifugal Force.}$$

@ railway challenge

$$F_c = 1304N \quad \rightarrow \text{@ Exhibition Park}$$

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Torsion in ARB

$$\text{Torsional Stiffness} = \frac{\text{Torque}}{\text{Angle}} = \frac{\text{Nm}}{\text{radians}} = \frac{T}{\theta}$$

Torsion Equation:

$$\frac{T}{\theta} = \frac{GJ}{L}$$

G = Shear modulus

J = Polar moment of inertia

L = Length of torsion bar

 θ = angle of rotation (rad)

T = Torque Applied.

Torsional Stiffness for current rod

$$G = 7.2 \times 10^{10} \text{ Pa}$$

(Steel) S275

$$J = 1.194 \times 10^{-8} \text{ m}^4$$

$$L = 0.4 \text{ m}$$

$$\therefore \frac{T}{\theta} = \frac{7.2 \times 10^{10} \times 1.194 \times 10^{-8}}{0.4}$$

$$\frac{T}{\theta} = 2149.2 \text{ Nm/radian}$$

Torsional Stiffness of current bar = 2150 Nm/rad.

If the torsion bar was solid $\frac{T}{\theta} = 2827 \text{ Nm/rad}$

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Anti-Roll Bar Safety Factor

Road characteristics : Torque = 78.2 Nm
 Diameter = 40mm $\rightarrow$ 20mm
 $\sigma_y = 355 \text{ MPa}$ $\tau_y = 177.5 \text{ MPa}$
Torsional Shear Stress:

$$\tau_c = \frac{T_r}{J} = \frac{T_r}{\frac{\pi(D^4)}{32}} = \frac{16TD}{\pi D^4}$$

$$\tau_c = \frac{16 \cdot 78.2 \cdot 20 \cdot 10^{-3}}{\pi (20 \cdot 10^{-3})^4}$$

$$\tau_c = (6.22 \text{ MPa})$$

$$\tau_c = 49.78 \text{ MPa}$$

$\tau_y =$ Yield Shear of torsion bar

$$\tau_y = 177.5 \text{ MPa}$$

$$\sigma_y = 355 \text{ MPa}$$

$$\therefore \text{Safety Factor} = \frac{\tau_y}{\tau_c} = \frac{177.5}{49.78}$$

$$\text{Safety Factor} = 3.56$$

$\rightarrow$ Shear
ARB

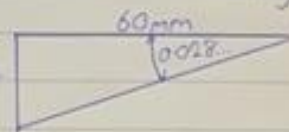
$$\text{Torsional Stiffness} = 2827 \text{ Nm/rad}$$

ARB Deflection (vertical) due to 1304N centrifugal force:

$$\text{Deflection in radians} = 0.0277 \dots$$

$$x = 60 \tan(0.0277) =$$

$$\text{Vertical displacement} = 1.66 \text{ mm}$$



Appendix H-8: Steering Link Calculations (DS)

Steering Links Stress Analysis 40

Using the Euler Bernoulli equation

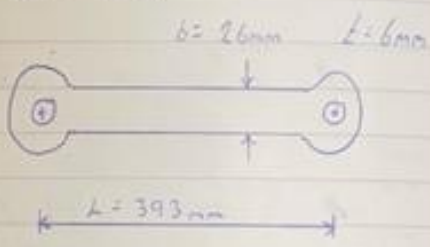
$$P_{cr} = \frac{\pi^2 EI}{L^2}$$

Critical Buckling Load = $\frac{\pi^2 \cdot \text{young's modulus} \cdot \text{2nd moment of Area}}{\text{Length}^2}$

$I = \frac{bd^3}{12}$ for a thin rod.

$I = \frac{0.026 \cdot (0.006)^3}{12}$

$I = 4.68 \times 10^{-10} \text{ m}^4$



Critical Load $P_{cr} = \frac{\pi^2 \times 206 \times 10^9 \times 4.68 \times 10^{-10}}{0.393^2}$

$P_{cr} = 6160.7 \text{ N}$ buckling load.

Assuming worst case this member must take the mass of the loco

Force = $202 \times 9.81 = 1982 \text{ N}$

Safety factor = $\frac{6061}{1982}$

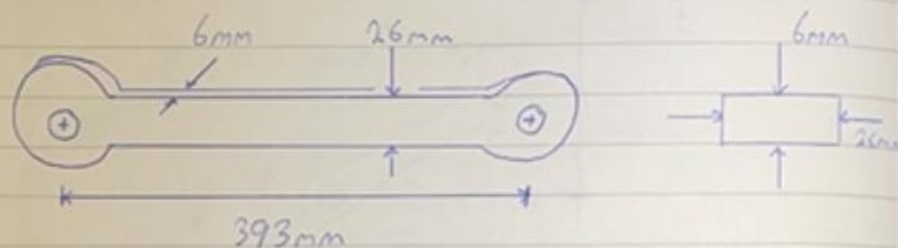
Steering Link Safety factor = 3.05

Further Steering Link Calculations

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Finding the planar & shear stress safety factors.

Analyzing longest member: of S275 Steel.



$$\text{Force laterally} = \text{mass of bogie} \times 10g = F_y$$

$$\text{Force vertically} = \text{mass of bogie} \times 10g = F_z$$

$$F_y = F_z = 202 \times 9.81 \times 10 = 19816.2 \text{ N}$$

Planar Stress:

$$\sigma = F/A \quad \sigma = 19816 / A$$

$$\text{Area planar} = 6\text{mm} \times 26\text{mm} = 156\text{mm}^2$$

$$\sigma_y = 275 \text{ N/mm}^2 \quad \sigma_y = 275 \times 10^6 \text{ MPa}$$

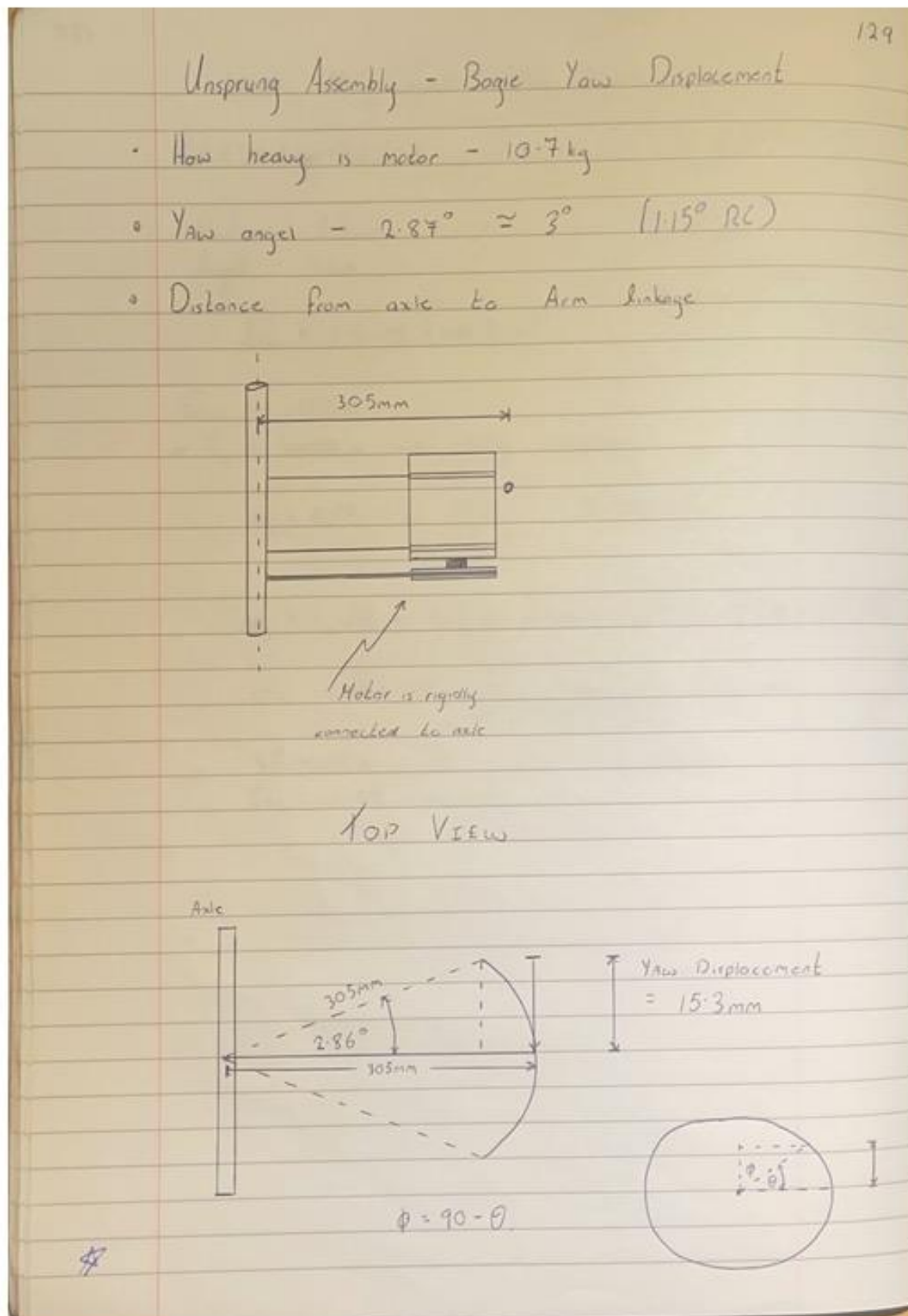
$$\therefore \text{yield force} = \sigma \times A = 156 \times 275 = 42906 \text{ N}$$

$$\text{Area planar} = 1.56 \times 10^{-4} \text{ m}^2$$

$$\text{Stress applied} = \frac{F}{A} = \frac{19816.2}{1.56 \times 10^{-4}} = 127.03 \text{ MPa}$$

Planar
Safety
Factor

$$SF = \frac{\sigma_{\text{permissible}}}{\sigma_{\text{Applied}}} = \frac{275 \text{ N/mm}^2}{127 \text{ N/mm}^2} = 2.17$$

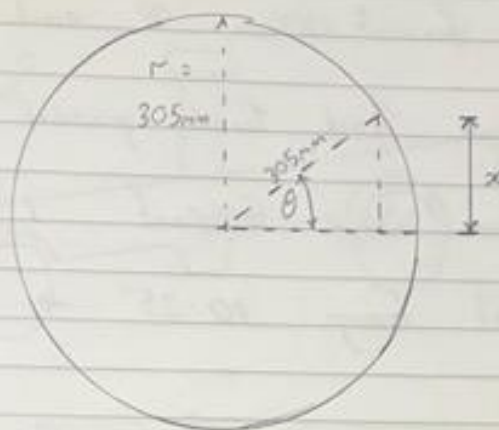
Appendix H-9: Bogie Frame Clearance Calculations (DS)

130

02/12/21

$$r = 305\text{mm}$$

$$\theta = 2.87^\circ$$



$$x = r \sin \theta$$

$$x = 305 \sin(2.87)$$

$$x = 15.27\text{mm} \rightarrow 20\text{mm to be safe}$$

.. ALLOW $\pm 20\text{mm}$ freedom either side of brake on upper bogie frame to allow for axle & unsprung assembly - bogie yaw.

$$x = 612\text{mm for Railway Challenge}$$

78

Yaw Angle for 20m x 8m Bend

Cornering Radius	Yaw Angle (°)	Brake Displacement
3.8m	6.00	34.4mm
8m	2.87	15.3mm
20m	1.15	6.22mm

Distance of brake to axle = 310mm

$$\therefore \text{Lateral Displacement} = (\text{Brake Distance}) \sin(\text{Yaw } \angle)$$

Motor Displacement

	Cornering Radius	Yaw Angle	(D) Motor Extreme Displacement
①	3.8m	6.00	35.5mm
②	8m	2.87	17.03mm
③	20m	1.15	6.82mm

Distance of end motor plate to axle = 340mm

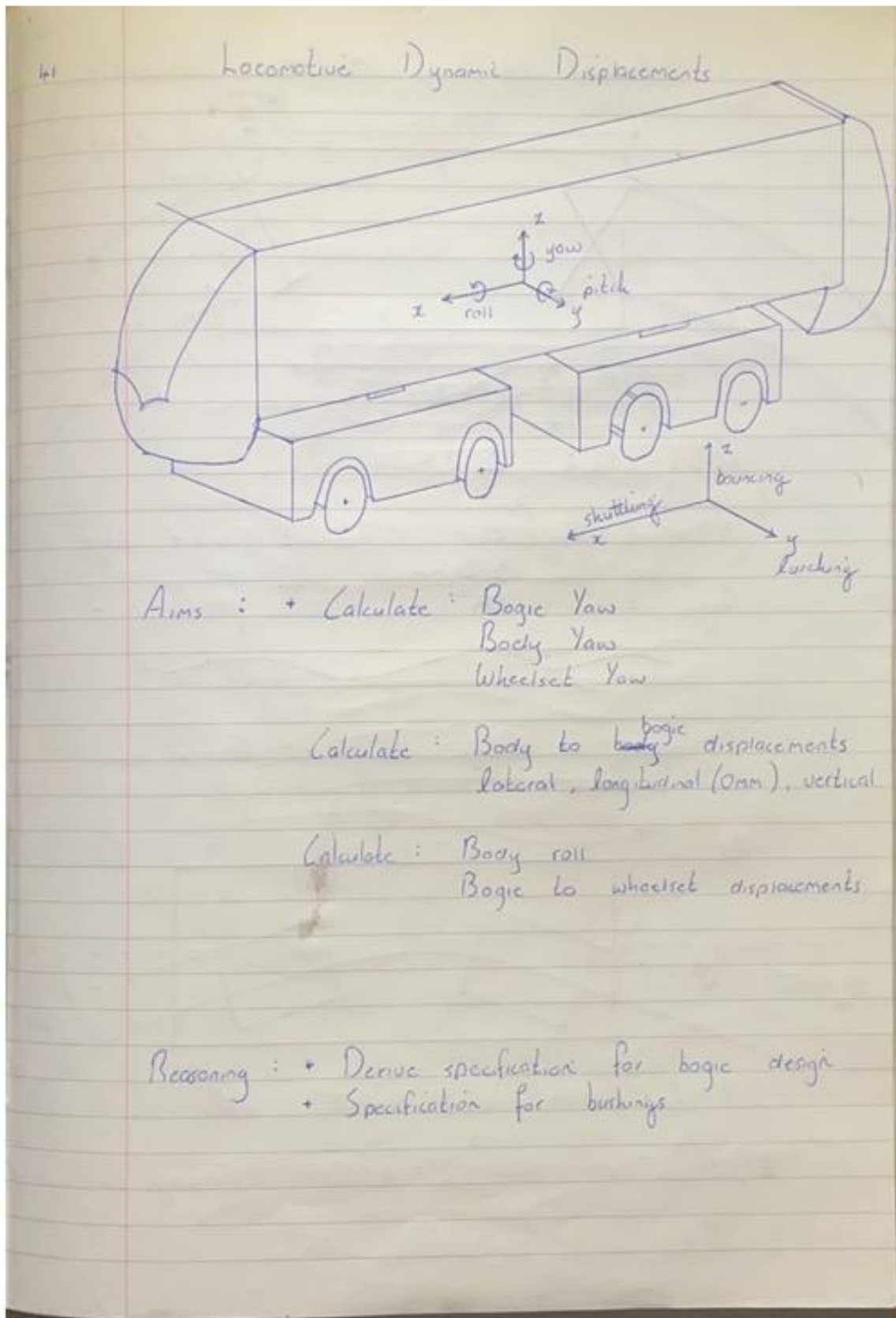
① $D_1 = 340 \sin(6) = 35.5\text{mm}$

② $D_2 = 340 \sin(2.87) = 17.03\text{mm}$

③ $D_3 = 340 \sin(1.15) = 6.82\text{mm}$

★

Appendix H-10: Degrees of Freedom (DS)



161

Vertical Track Curvature (Plich)

9/12/21



Maximum slope = 1:84

Solchehtog

$$\tan \theta = \frac{1}{84}$$

$$\therefore \theta = \tan^{-1}\left(\frac{1}{84}\right)$$

$$\theta_1 = 0.682^\circ \rightarrow \text{RAILWAY CHALLENGE}$$

$$\theta_2 = 7.125^\circ \rightarrow \text{Exhibition Park}$$

Spechtion

$$\sin \theta = x / \text{hyp}$$

$$x = \text{hyp} \sin \theta$$

$$x = 800 \times \sin(0.682) = 9.523 \text{ mm}$$

$x = 10 \text{ mm}$ Vertical Track Curve vertical displacement = 10mm



*

Degrees of Freedom

47

Motion : Shuttle x-direction bogie-wheelset

$$\text{deflection } \Delta x = \frac{F}{k(\text{shear})_{\text{primary suspension}} \times 8!}$$

$$\text{Force} = \text{mass of } \overset{\text{bogie}}{\text{wheelset}} \times 10g$$

$$\text{Force} = 202 \times 9.81 \times 10 = 19816.2 \text{ N}$$

$$\Delta x = \frac{19816.2}{6.1 \text{ N/mm}} = 3248 \text{ mm (unrealistic)}$$

$$\text{Force} = \frac{\text{mass of bogie} \times g}{\text{lateral stiffness of primary}}$$

$$\Delta x = \frac{9.81 \times 202}{6.1} = 324.8 \text{ mm (unrealistic)}$$

$$\Delta x = \frac{9.81 \times 202}{6.1 \times 8} = 40.6 \text{ mm}$$

Shuttle : Bogie $\rightarrow$ Body

$$\text{Unconstrained} = \frac{\text{mass of body} \times g}{2 \times \text{spring lateral stiffness rate} \times 8}$$

$$\text{Unconstrained } \Delta x = \frac{600 \times 9.81}{257 \times 8} = 28.63 \text{ mm}$$

Constrained Bogie $\rightarrow$ Body shuttle (x motion)
= 0 mm due to traction control

Roll about z-axes

48

Wheelset $\rightarrow$ Bogie roll

0mm?

DLG

Body $\rightarrow$ Bogie roll

- Related to dynamic loading gauge
- Constrained by Anti-Roll-Bar

Lurching (motion in y-axis)

Bogie $\rightarrow$ Wheelset lurching

limit = 9mm

unconstrained displacement = $\frac{\text{mass of bogie} \times 9.81}{\text{primary suspension lateral rate} \times 8}$

$$\text{u. } \Delta x = \frac{202 \times 9.81}{6.1 \times 8} = 40.6 \text{ mm}$$

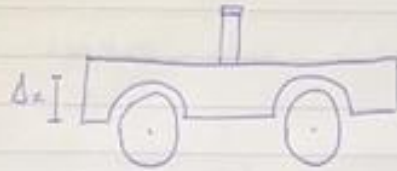
Body $\rightarrow$ Bogie lurching

$$\text{Unconstrained } \Delta x = \frac{600 \times 9.81}{6.1 \times 8} = 25.7$$

$$\text{Unconstrained } \Delta x = 28.6 \text{ mm}$$

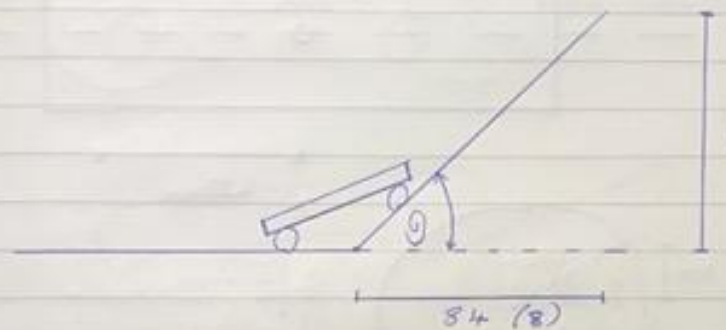
49

Pitch (about y axis)



Scheuchlin

Pitch of wheelset relative to bogie

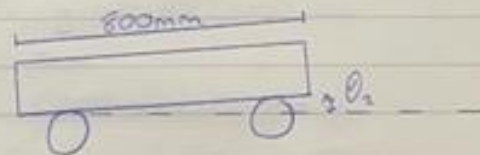


minimum slope = 1:84 ; max slope = 1:8

$$\tan \theta = \frac{1}{84, (8)_2} \quad \therefore \theta_1 = \tan^{-1}\left(\frac{1}{84}\right) \quad \theta_2 = \tan^{-1}\left(\frac{1}{8}\right)$$

$$\theta_1 = 0.682^\circ \quad \text{for railway challenge}$$

$$\theta_2 = 7.125^\circ \quad \text{for exhibition park}$$

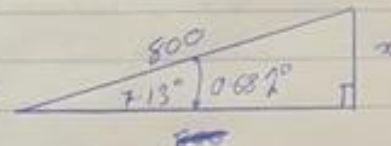
Bogie-wheelset
crabbed angle Use θ_2 for calculations.

$$x = 600$$

$$\sin \tan(7.13) = \frac{x}{800}$$

$$\therefore x = 800 \sin(7.13)$$

$$x = 100 \text{ mm}$$



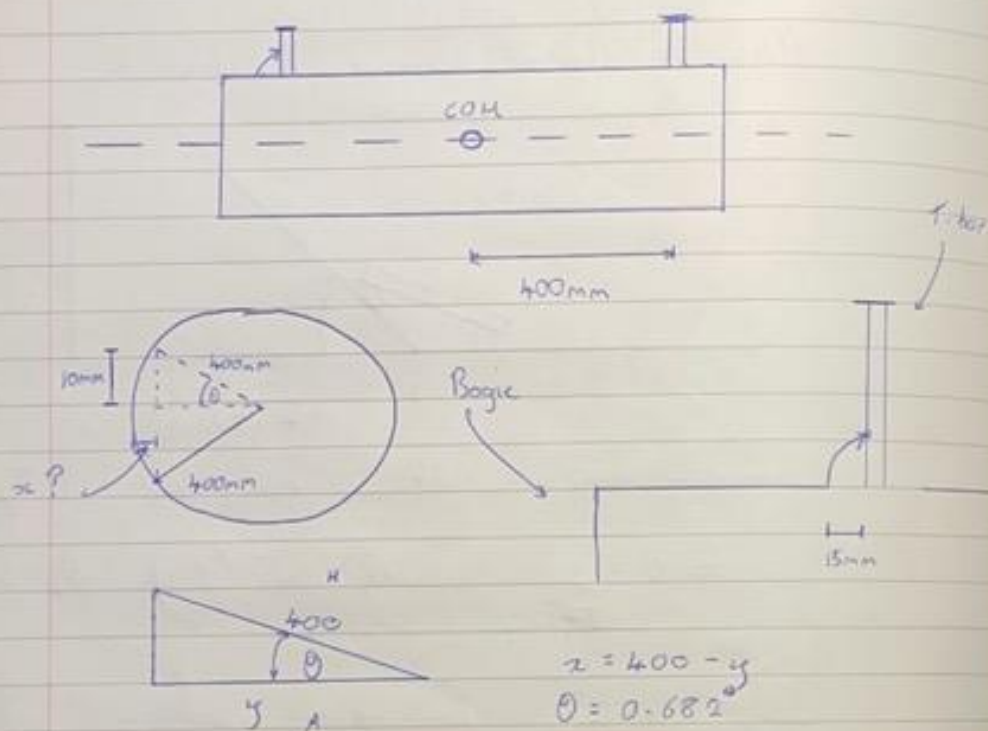
$$x = 800 \sin(0.682) = 9.5 \text{ mm} \quad \updownarrow$$

$$\text{say } x = 15 \text{ mm} \quad \theta = \sin^{-1}\left(\frac{15}{800}\right) \quad \theta = 1.07^\circ$$

Bogie - Wheelset Pitch Constraint

50

Bogie will collide with axlebox T-bar



Solution

$$\cos(0.682) = \frac{y}{400}$$

$$y = 400 \cos(0.682) = 399.97 \text{ mm}$$

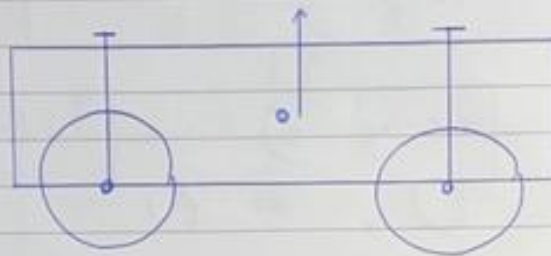
$$\therefore x = 0.028 \text{ mm} \quad \text{😊}$$

$\therefore$ pitch constraint = 15mm axle box T-bar

Vertical Displacement Wheelset $\rightarrow$ Bogie

52

Unstrained:



$$\text{Force} = \frac{\text{Vertical Primary Spring Stiffness} \times \text{displacement}}{\text{Vertical}}$$

$$\text{Force} = \text{mass of bogie} \times 9.81 \text{ s}$$

$$F = kx \quad \therefore \Delta x = F / k \times 8$$

$$\Delta x = \frac{202 \times 9.81 \times 5g}{24.52 \times 8} = \frac{50 \text{ mm compression}}{50.51 \text{ mm}}$$

Body - Bogie vertical displacement

$$\Delta x = F / k \times 8$$

$$k = 2^{\text{nd}} \text{ Sus Stiffness} \text{ Also}$$

$$F = m_{\text{bogie}} \times 2g$$

$$\Delta x = \frac{600 \times 2 \times 9.81}{41.156 \times 8}$$

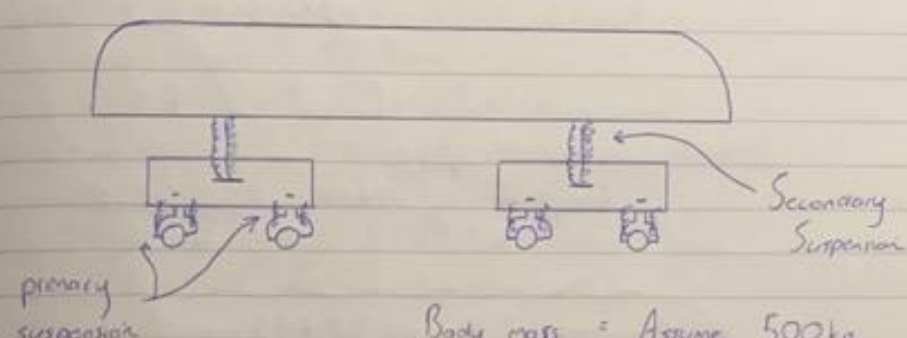
$$\Delta x = 35.75 \text{ mm}$$

Appendix H-11: Secondary Spring Calculations (DS)

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Suspension Springs Safety Factors

Secondary (Primary) Suspension Springs



primary suspension

Secondary Suspension

Body mass = Assume 500 kg
Bogie mass = 202 kg

Secondary Spring Characteristics

Part Number = LHL 2000A 06
 $D_o = 46.48 \text{ mm}$
 Free length = 127 mm
 Solid height = 60.96 mm
 $\Delta x = 66.04 \text{ mm}$ compression
 $k_{Rate} = 41.156 \text{ N/mm}$
 Wire diameter = 6.35 mm
 Lateral stiffness = 25.7 N/mm (From FEA)
 Force required to fully compress $F = kx$

$$F = 41.156 \times 66.04$$

$$\text{Force} = 2718 \text{ N}$$

Force experienced by each spring

$$\left[\frac{(\text{body mass} \times 2g)}{8} + \frac{(\text{bogie mass} \times 2g)}{8} \right] / 4$$

Secondary Suspension FOS

$$\text{Force experienced} = \frac{(500 \times 2 \times 9.81) + 2(200 \times 9.81 \times 8)}{8 \quad 16}$$

$$\text{Force experienced} = 1851.6 \text{ N} = 1226.25 \text{ N}$$

$$\text{Compression from force } \uparrow : F = kx$$

$$\Delta x = \frac{F}{k} = \frac{1851.6}{41.156} = \frac{1226.25}{41.156}$$

$$\text{Compression } \Delta x = 29.795 \text{ mm}$$

$$\text{Safety factor} = \frac{\text{permissible force}}{\text{force}} = \frac{2718}{1226.25}$$

$$\text{Factor of Safety} = 2.21$$

for 2ndary Sus

Primary suspension static locating spring compression & height

$$\text{Static Load} = (200 \times 2 + 600) / 16$$

$$623.13 \text{ N (force experienced by each spring)}$$

$$\Delta x = F/k = 613.13 / 24.52$$

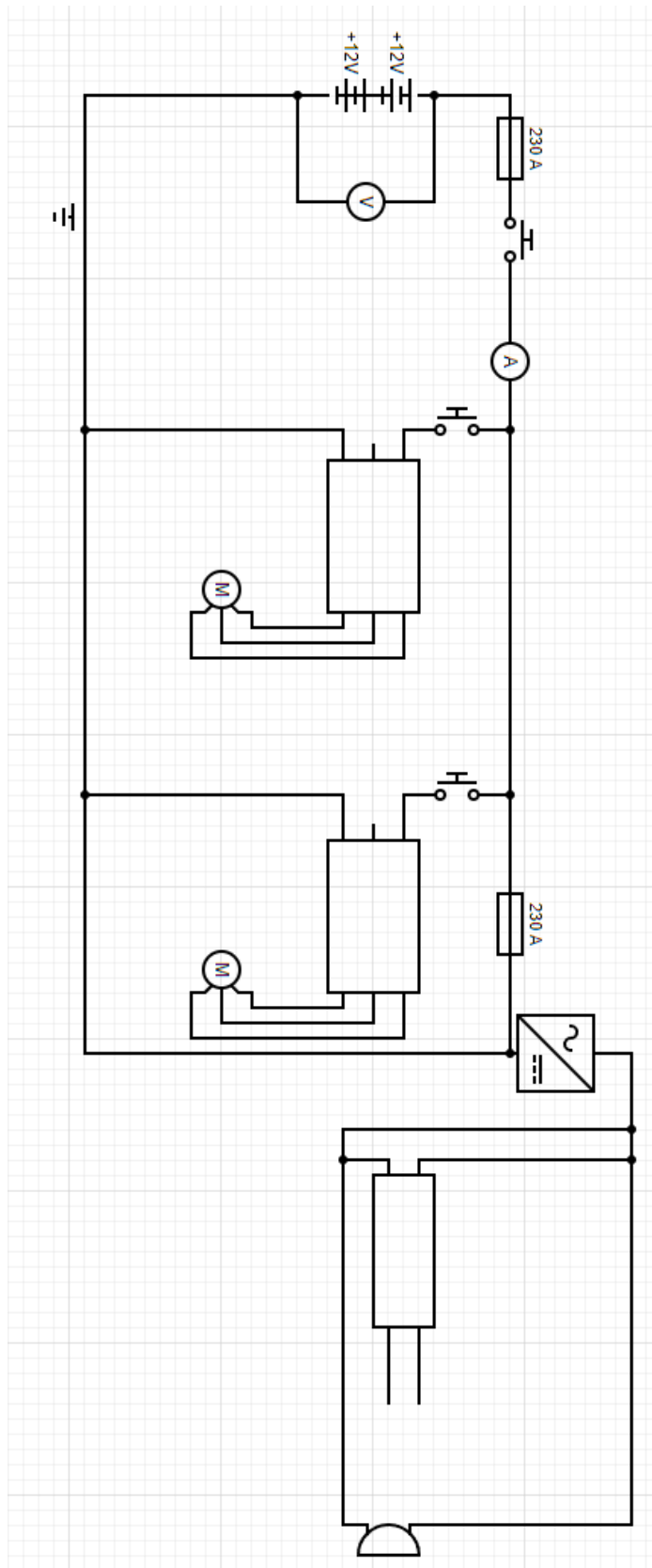
$$\Delta x = 25 \text{ mm}$$

$$\text{Inverter compression} = 20.4 \text{ mm}$$

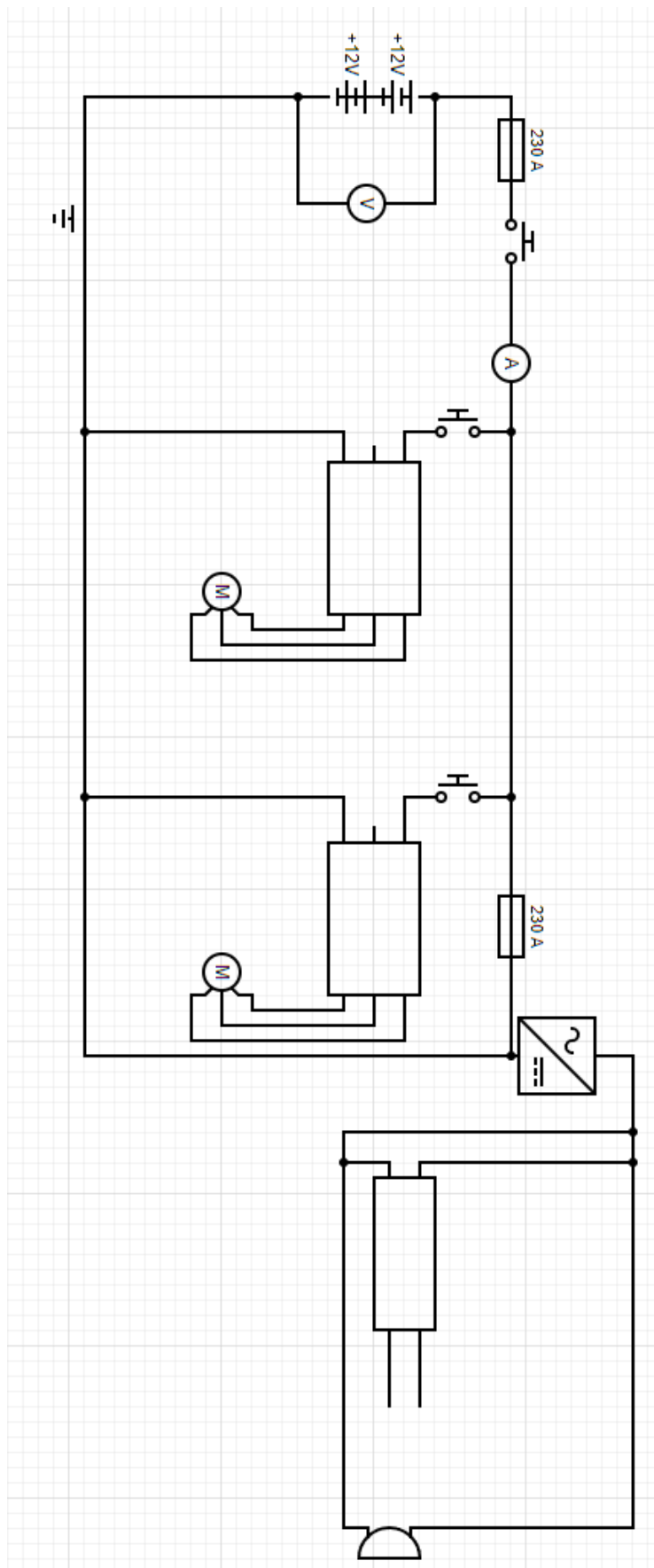
$$\text{Inverter force / spring} = 500 \text{ N}$$

Appendix I: High Voltage and Traction Related Circuits

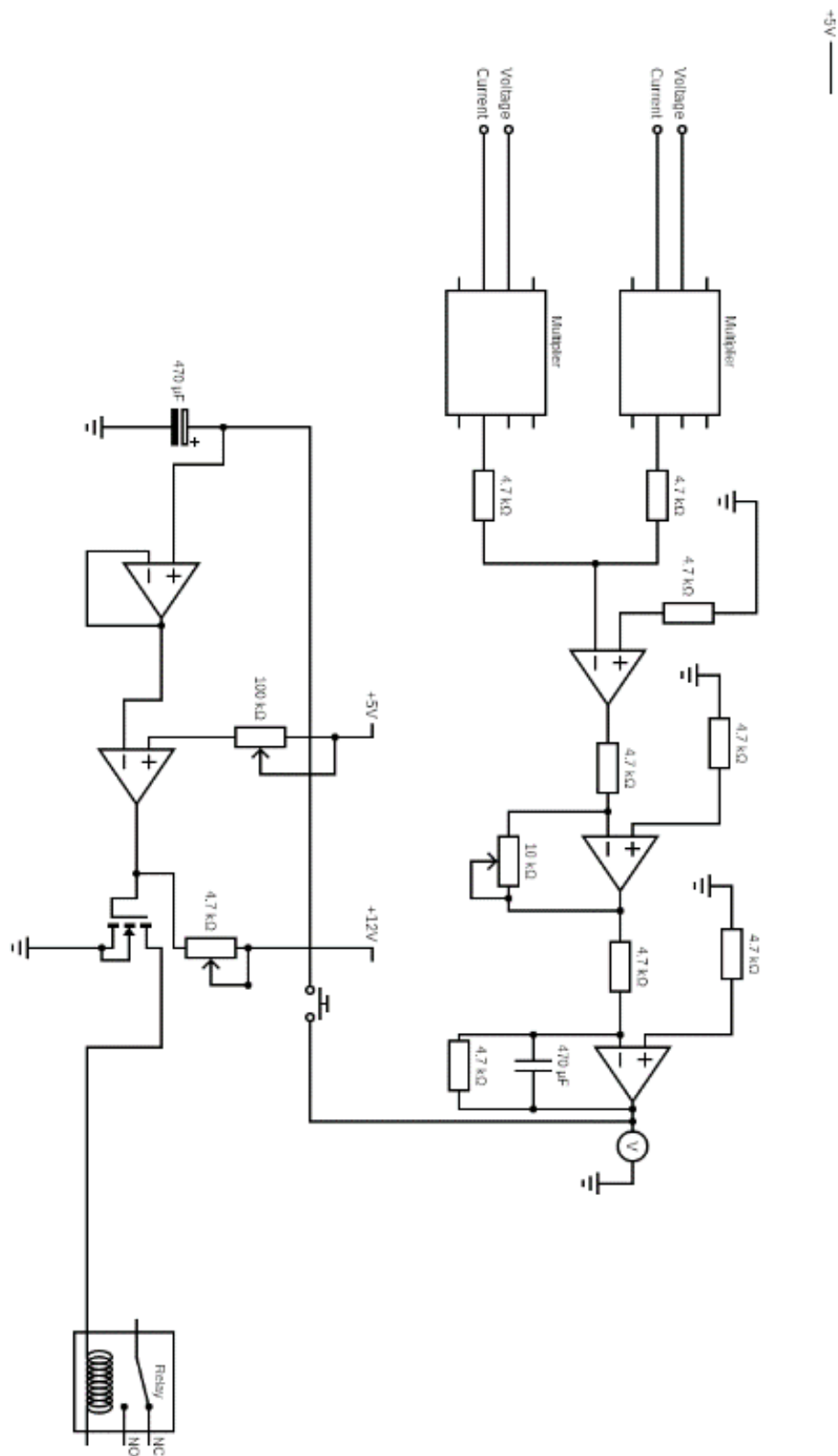
Appendix I-1: Locomotive Traction Circuit Front



Appendix I-2: Locomotive Traction Circuit Rear



Appendix I-3: Energy Counting Circuit



Appendix I-4: ACPL 782T 000E Voltage Measuring Circuit

Appendix J: Sponsorship Document (AH, HW, JO)

Institution of
**MECHANICAL
ENGINEERS**

Newcastle University School of Engineering

Railway Challenge — HSToon

We are HSToon, a group of students from Newcastle University competing in the IMechE Railway Challenge in June 2022. We are designing and manufacturing a 10 ¼" gauge locomotive to compete against other teams in a series of challenges. Newcastle's entry last year won the business case, we now want to build on this achievement and win more this year.

Our Loco Design is Unique

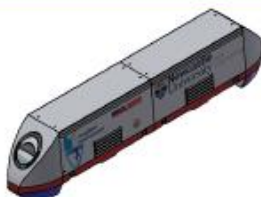
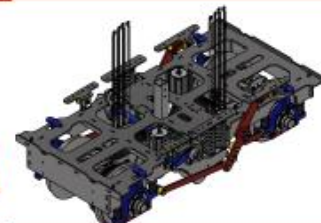


- Features **Auto-stop** functionality, allowing the loco to automatically control its braking.
- Track and wheel wear is minimised through the inclusion of an **active steering system**.
- The bogies and body are of a **modular design** to ensure reliability. The control system is completely analogue, enabling **simple fault-finding** and replacement.
- An analogue **creep control** system actively maximises tractive effort (for a particular loco weight), allowing the loco to haul trailing loads up **1:8 gradients**.



Four brushless permanent magnet motors are being used for traction. The **regenerative braking** system captures almost all braking energy, even at speeds as low as 1mph. Improving energy efficiency and increasing run time.

A pair of bogies is to be constructed from laser cut steel with minimum welding to allow for **simple construction** and repairs. Coupling is provided to allow towing, where it will be possible to haul loads of up to 1800kg.



Reliability is ensured through the use of a modular design. The loco can be split in half for **easy transportation**. Passive provision is in place to allow the inclusion of a passenger car between the two halves in the future.

Our Friendly Team



Left to right: Alex Powell, Halley Woods, James O'Brien, Lightness Kings, Jonas Janiunas, Alex Hooper, David Stewart, Cyril Cheung

Sponsorship Opportunities

Platinum Tier : Donation or support to the value of £2500+

Prominent large logo on the locomotive, a full description of your company on our social media accounts and website, an invite to the challenge weekend in Leicestershire. Plus all benefits included in lower packages.

Gold Tier: Donation or support to the value of £1500+

Large logo on the locomotive, large logo on the footer of our website. Invite to our manufacturing facility to meet the team and view the locomotive during construction. Plus all benefits included in lower packages.

Silver Tier : Donation or support to the value of £1000+

Medium logo on the locomotive, medium logo on footer of our website. Large logo on team merchandise which will be worn to the event. Plus benefits included in the Bronze tier.

Bronze Tier : Donation or support to the value of £500+

Small logo on the locomotive and footer of our website. Small logo on team merchandise which will be worn to the event.

Contact Details

Alex Hooper (Team Leader) - A.Hooper@newcastle.ac.uk

Rob Davidson (Academic Supervisor) - r.i.davidson@ncl.ac.uk



@newcastlelocomotive



@newcastleloco

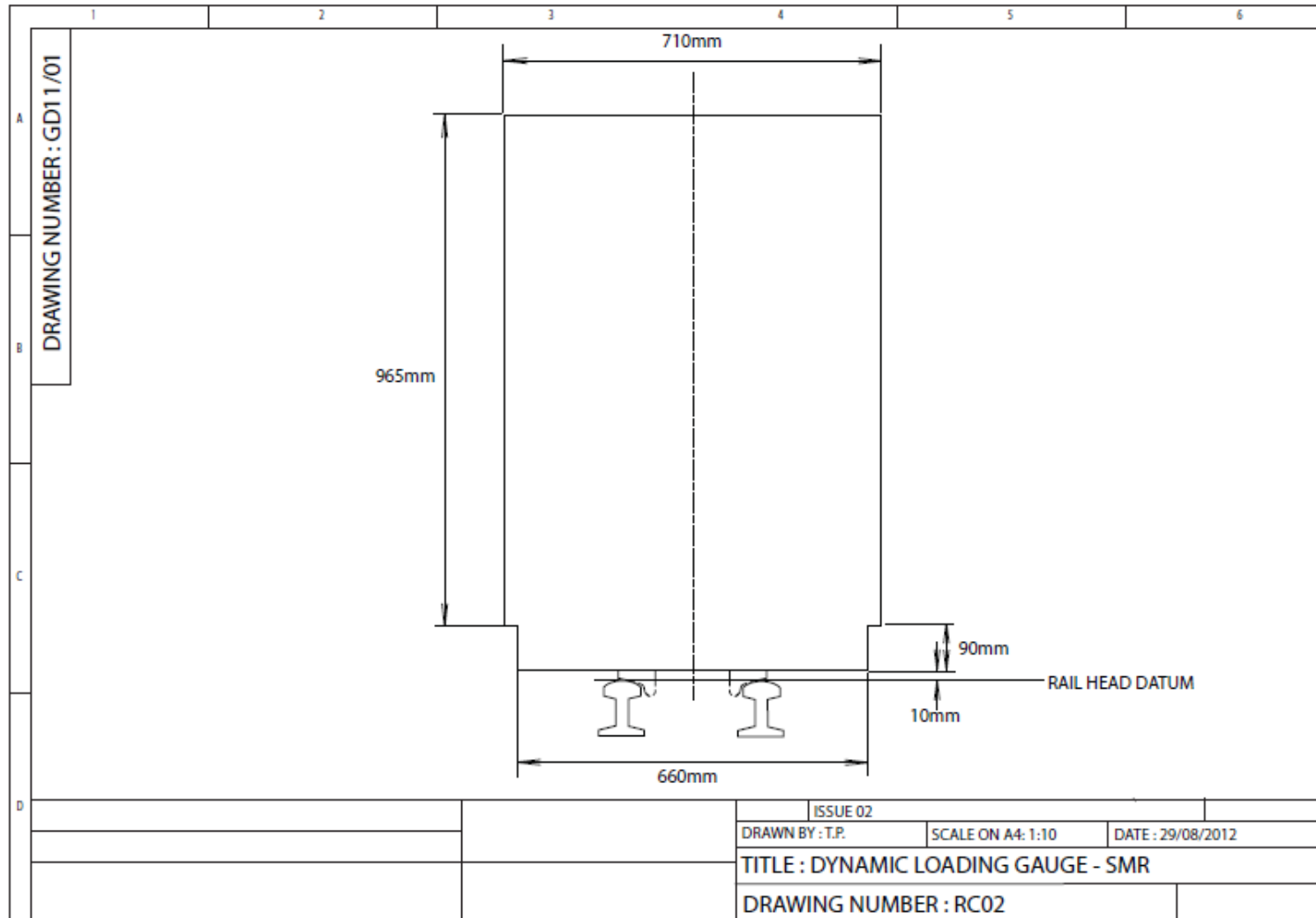


@newcastleloco

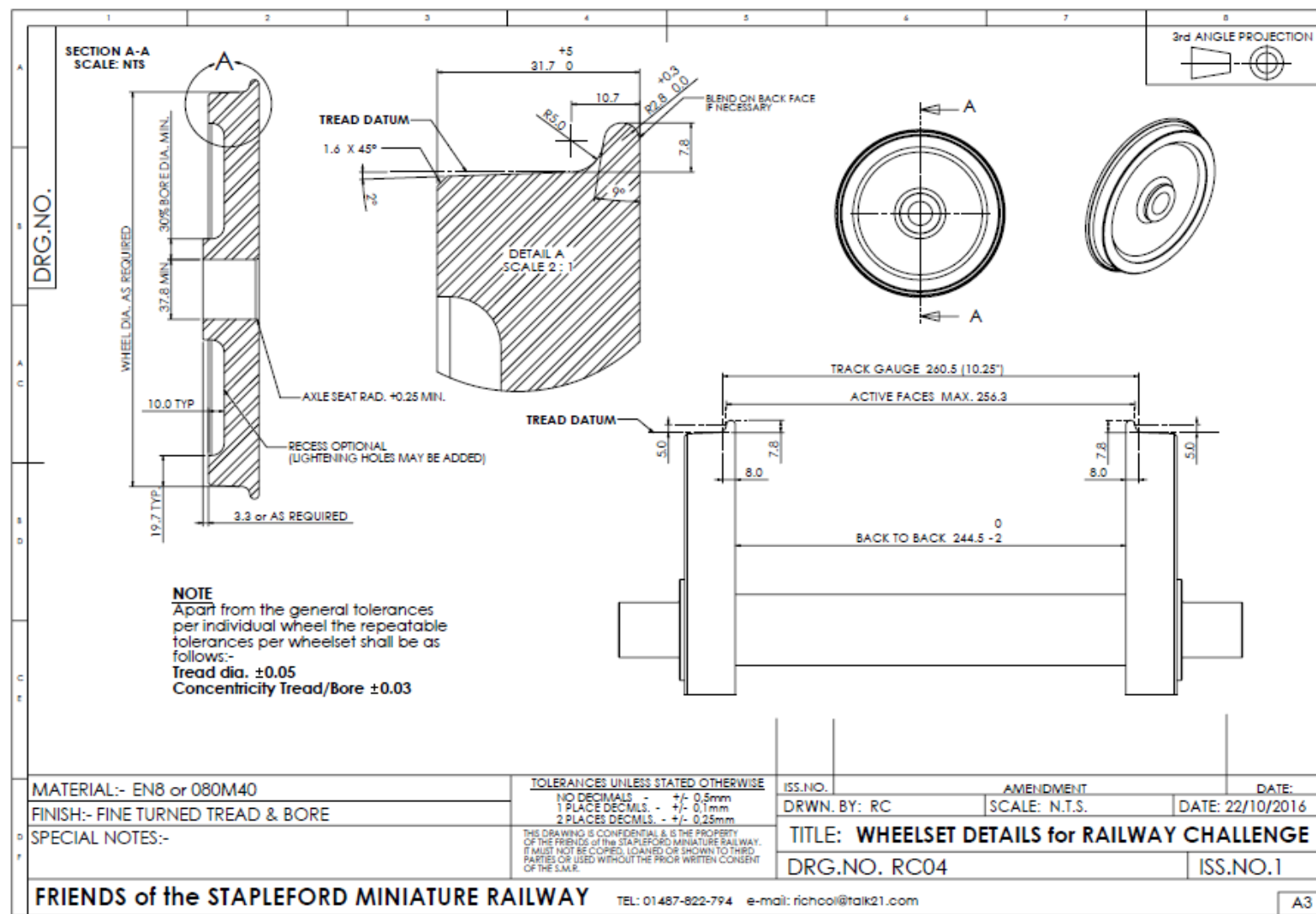
Visit our website: <https://research.ncl.ac.uk/newcastlelocomotive/>

Appendix K: IMechE Supplied Documents

Appendix K-1 Dynamic Loading Gauge Stapleford



Appendix K-2 Wheelset details for Railway Challenge



Appendix L: Data Tables

Appendix L-1 FIBET Fully Bonded Bushings Data Sheet

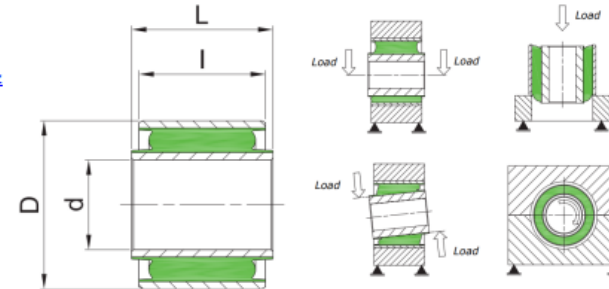
FIBET FULLY BONDED RUBBER BUSHES

https://www.fibet.co.uk/range/rubber-anti-vibration-mounts/bushes-bonded-suspension/?qclid=CjwKCAIAqIKNBhAIEiwAu_ZLDvHz_zJQJx9Ud85g9scPOd12eitVdeDHnTYDTqdiELQ6O6xQzJML3hoCGC8QAvD_BwE

<https://www.fibet.co.uk/range/rubber-anti-vibration-mounts/bushes-bonded-suspension/fully-bonded-bushes-type-fbna/?s=>

The maximum loads given are for loading in one mode only.

If loading is in more than one mode simultaneously, the maximum loads will be reduced.



Fibet Item Number	Bore (mm)	External Diameter (mm)	Bore Length (mm)	External Length (mm)	Radial Stiffness (dN/mm)	Radial Max. Defln. (mm)	Radial Max. Force (N)	Axial Stiffness (dN/mm)	Axial Max. Defln. (mm)	Axial Max. Force (N)	Torsional Stiffness (Nm/deg.)	Torsional Max. Defln. (+/- deg.)	Torsional Max. Moment (Nm)	Conical Stiffness (Nm/deg.)	Conical Max. Defln. (+/- deg.)	Conical Max. Moment (Nm)	Mass (g)	Rubber Hardness (IRHD)	Stock Item or On Request
FBNA0832.1823	8.2	32	18	23	29	1.4	406	8	1.0	80	0.2	35	6.7	0.1	6	0.5	35	55	Stock
FBNA0921.1922	9.5	20.6	19	22.2	207	0.5	1035	23	0.7	161	0.3	25	6.8	0.5	6	3.2	25	55	Stock
FBNA1022.15.17	10	22	15	17	207	0.5	1035	22	0.5	110	0.3	25	8.0	0.3	6	1.9	25	55	Stock
FBNA1025.3541	10	25	35	41	1483	0.5	7415	104	1.4	1456	1.8	20	35.0	13.8	3	41.3	75	65	Stock
FBNA1032.1520	10	32	15	20	32	1.1	352	9	1.0	90	0.2	30	7.2	0.1	7	0.4	35	55	Stock
FBNA1032.2430	10	32	24	30	350	1.1	3850	35	1.0	350	0.7	20	14.0	3.5	3	10.5	105	65	Stock
FBNA1225.2528	12	25	25	28	764	0.5	3820	67	0.7	469	1.2	20	24.4	3.5	3	10.6	45	65	Stock
FBNA1230.3434	12	30	34	34	425	0.8	3400	55	0.1	55	1.4	30	40.8	4.0	7	27.9	75	65	Request
FBNA1230.3444	12	30	34	44	425	0.8	3400	55	2.1	1155	1.4	30	40.8	4.0	3	11.9	80	65	Request
FBNA1230.3640	12	30	36	40	414	0.9	3726	55	0.8	440	1.4	30	42.6	4.4	3	13.2	75	65	Stock
FBNA1234.3541	12	34	35	41	249	1.1	2739	43	1.4	602	1.3	20	26.4	2.6	3	7.8	100	65	Stock
FBNA1250.5062M	12	50	50	62	102	2.3	2346	24	2.7	648	1.5	30	43.5	2.4	7	17.1	235	55	Stock
FBNA1274.4566	12.5	74.5	45	66	126	3.4	4284	35	4.4	1540	2.8	30	84.9	1.5	7	10.6	600	65	Request
FNBA1330.4040	13	30	40	40	396	0.8	3168	44	0.0	0	1.1	30	33.9	5.1	3	15.4	85	55	Stock
FBNA1474.4566	14.5	74.5	45	66	126	3.4	4284	35	4.2	1470	4.7	30	140.7	2.5	7	17.6	580	65	Stock
FBNA1638.6472	16	38	64	72	2070	0.7	14490	127	1.6	2032	4.9	20	98.0	67.1	1	67.1	300	55	Stock
FBNA1650.6082	16	50	60	82	390	1.9	7410	68	4.5	3060	4.4	20	88.6	12.9	3	38.8	400	65	Request
FBNA1651.3847	16	51	38	47.5	102	2	2040	23	2.0	460	1.7	30	49.5	1.4	7	9.5	190	55	Request
FBNA1651.4454	16	51	44.3	53.8	128	2	2560	27	2.1	567	1.9	30	58.2	2.3	7	16.2	220	55	Request
FBNA1738.3338	17	38	33	38	449	1.1	4939	60	1.2	720	2.6	25	65.8	4.0	3	11.9	120	65	Stock
FBNA1857.6072	18	57	60	72	159	2.4	3816	34	2.4	816	3.0	30	90.3	5.5	7	38.6	350	55	Request
FBNA1857.7086M	18	57	70	86	393	2.3	9039	73	3.3	2409	6.7	30	201.6	17.9	7	125.6	450	65	Request
FBNA2866.5569	28	66	55	69	234	2.1	4914	43	2.8	1204	5.9	20	117.2	6.4	3	19.3	450	55	Stock
FBNA3882.7090	38	82	70	90	720	2.1	15120	89	2.8	2492	20.4	15	306.0	30.3	1	30.3	1.27	55	Stock
FBNA4082.7090	40	82	70	90	732	1.9	13908	90	4.0	3600	20.5	15	306.9	30.8	1	30.8	1.2	55	Stock

[illegible]

Appendix M-2 Detailed bogie manufacturing preparation plan

To do	Process	Progress	Equipment required	Place	People	specialist help required?	Note
1.1 Eccentric Device		In progress					
1.1.1 Order Solid Aluminium Round Bar (do=125mm L=138mm [x4])	Order	Completed					
1.2 Eccentric Housing Clamp Pins [x32]		In progress					Medium
1.2.1 Order Solid round S303 Bar (do=12mm L=28mm [x32])	Order	Completed					
1.2.2 Clamp pin at either side	Clamping	Not started	Clamp	MFL1/2		1 No	
1.2.3 Drill [x2] M4 through hole 5.5mm from either flat side of clamp	Drilling	Not started	M4 Drill, M4 bolt, M4 Nut, wash	MFL1/2		1 No	
2.1 Stock 8mm Gates Poly Chain GT2 Sprocket (8M-225-21)	Order	Completed	None	None	Jonas	No	https://www.gates.com/gb/en/power-transmission/po
2.2 Stock 8mm Gates Poly Chain GT2 Sprocket (8M-805-21)	Order	Completed	None	None	Jonas	No	
2.3 Poly Chain carbon volt belt (Pitch - 896)	Order	In progress	None	None	Jonas	No	
2.4 1008-7/8in Taper lock bush	Check	Completed	None	FSW	Jonas	No	Check size, key and design
2.5 2517-60mm Taper lock bush	Bore to 65mm	Completed	Lathe, chuck	FSW	Jonas	Maybe	
2.6 Ring feeder/fric locking assembly	Order	Completed	None	FSW	Jonas	No	
2.7 Machine the 22s sprocket	machine	In progress					
4.1 Motor Mount Spacers [x4]		Not started					
4.1.1 Order Solid S303 Solid Round Bar (do=20mm, L=32mm [x4])	Order	In progress					
4.1.2 Clamp Solid Bar to Drill Machine	Clamping	Completed					
4.1.3 Drill M10.5 through hole	Drilling	Not started	Clamp	MFL1/2		1 No	
4.1.4 Hack Saw Tube into 4 equal segments of L = 8 mm	Sawing	Not started	M10.5 Drill Tip, Clamp, Ruler	MFL1/2		1 No	
4.1.5 File Segment to L = 7.435mm	Filing	Not started	Hack saw, clamp	MFL1/2		1 No	
5.1 Pneumatic cylinder	Purchased	Completed	Micrometer, file, ruler, clamp	MFL1/2		1 No	
5.2 Spring seat	3D printing	Completed	none	not required	not required	No	
5.3 Brake shoe		In progress	3D printer	Unsure	Charlton	Yes	Leave space for tolerances?
5.3.1 Brake shoe	Cut and turn the surface to fit vee pulley	Completed	Hacksaw, lathe	Unsure	Technicians	Yes	
5.3.2 Brake shoe	Drill holes, holesaw for piston	Completed	Holesaw, 6mm drill head, drill,	FSW, outside FSW or Jonas and team*		No	
5.4 Brake shoe mount	drill holes, cut	Completed	Hole saw, 7mm holes, clamps	FSW, outside FSW or Jonas and team*		No	
5.5 Brake shoe mount extended	drill holes, cut, thread	Completed	Hole saw, 5mm hole and thread	FSW, outside FSW or Jonas and team*		No	
5.6 Pneumatic cylinder piston	drill and tap	Completed	M8x1.25	FSW, outside FSW or Jonas and team*		No	Take a look at cylinder for more modifications
5.7 Brake mount bracket	Weld	Not started	Welder, welding prep, safety equipment, file, engineering squares, clamps, ruler	Wallsend	Jonas + technicians	Yes	
5.8 Motor shaft extension	Machine	In progress	Lathe, mill	Unsure	Charlton	Yes	Needs to be extended, check if motor shaft is threaded
5.9 Shaft key	Machine/buy	In progress	Mill	Unsure	Unsure		
5.10 Pneumatic piston machining	Machine	Completed	Mill	Merz workshop		Yes	
6.1 Motor Mount Linking Arm [x4]							High
6.1.1 Order Outer Lugs (do=38.1mm, di=32.82, l=24mm [x12])	Order	Completed					
6.1.2 Order Bushings (ERMA 10251025101511)	Order	Completed					

Combined **Bogie parts preparation** Drive assembly Bogie Frame Anti-roll Bar Traction centre Secondary Suspension Primary Suspension Steering links Control Box Trolley Body Lo +

Appendix M-3 Example of detailed single system assembly process (Drive assembly)

Part	to do	Process	Equipment required	Place	People	specialist
team* - depends on availability						
ORANGE - take more care planning/ priority						
1. Belt drive detailed assembly						
1.1 22t pulley	Secure 22t pulley to the motor shaft using taper bush by tightening grub screws. Insert key into the keyway	assembly	Allen key	FSW	Jonas + team*	No
1.2 80t pulley + taper bush + friction coupling	Assemble 80t with taper bush. Put this assembly with loose locking assembly onto the axle	assembly	Wrench (spanner) or electric drill	FSW	Jonas + team*	No
1.3 80t pulley	Make sure to align well with motor pulley and tighten locking assembly bolts to secure the assembly onto the axle	assembly	Allen key	FSW	Jonas + team*	No
1.4 Belt	Put the belt on and using essentric tensioner to tension the belt	assembly		FSW	Jonas + team*	No
2. Axle detailed assembly						
2.1 "A" axle box		assembly	Wrench (spanner) or electric drill	FSW	AP	No
2.2 "A" axle box + bearing		assembly	Wrench (spanner) or electric drill	FSW	AP	No
2.3 "B" axle box		assembly	Wrench (spanner) or electric drill	FSW	AP	No
2.4 "A" and "B" axle boxes connected		assembly	Wrench (spanner) or electric drill	FSW	AP	No
3. Tensioner detailed assembly						
3.1 Bearings		assembly	Wrench (spanner) or electric drill	FSW	DS	No
3.2 Collar + upper housing + lower housing		assembly	Wrench (spanner) or electric drill	FSW	DS	No
4. Full drive assembly						
4.1 Axle + tensioner + belt drive + wheels	Secure tensioner assembly, wheel sprocket onto the axle. Put and secure the wheels on the axle after. Lift the assembly onto the trolley	Assembly	Wrench (spanner) or electric drill	FSW	Bogie team	No
4.2 Axle box (everything except stop link)	On trolley assembly put the axle box on to the axle. Stop links should not be put on as it will be impossible to assemble with bogie frame	Assembly	Wrench (spanner) or electric drill	FSW	Bogie team	No
4.3 Motor assembly	Motor assembly is bolted to the tensioner assembly	Assembly	Wrench (spanner) or electric drill	FSW	Bogie team	No
4.4 Belt drive	Pinion is secured on to the motor shaft and belt is put on and tensioned	Assembly	Wrench (spanner) or electric drill	FSW	Bogie team	No
4.5 Brake assembly	Vee pulley is secured on to the motor shaft extension. After, brake assembly is attached to the motor side while watching brake shoe fit into the vee pulley	Assembly	Wrench (spanner) or electric drill	FSW	Bogie team	No
4.6 Motor mount	Motor mount is attached to the motor and left hanging until required to connect to bogie frame	Assembly	Wrench (spanner) or electric drill	FSW	Bogie team	No

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Combined
Bogie parts preparation
Drive assembly
Bogie Frame
Anti-roll Bar
Traction centre
Secondary Suspension
Primary Suspension
Steering links
Control Bo