School of Engineering

MEC8099 Team Project Report

Institution of Mechanical Engineers Railway Challenge

Team 11

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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.

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

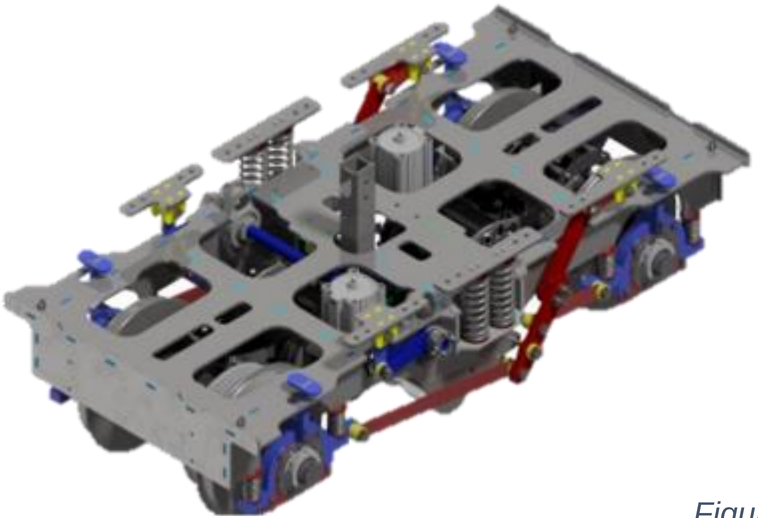


Figure 5-1: 2021 Bogie system design

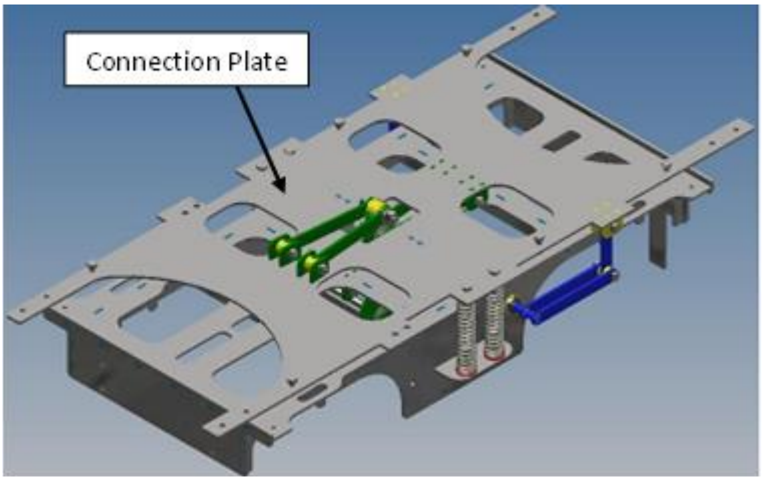


Figure 5-2: Year 3 bogie frame design with connection plate

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.

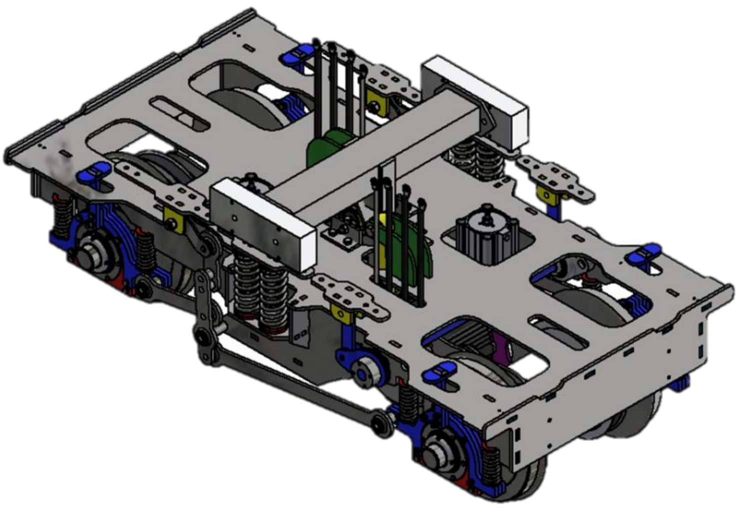


Figure 5-3: 2022 Railway Team final bogie system design

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.

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.

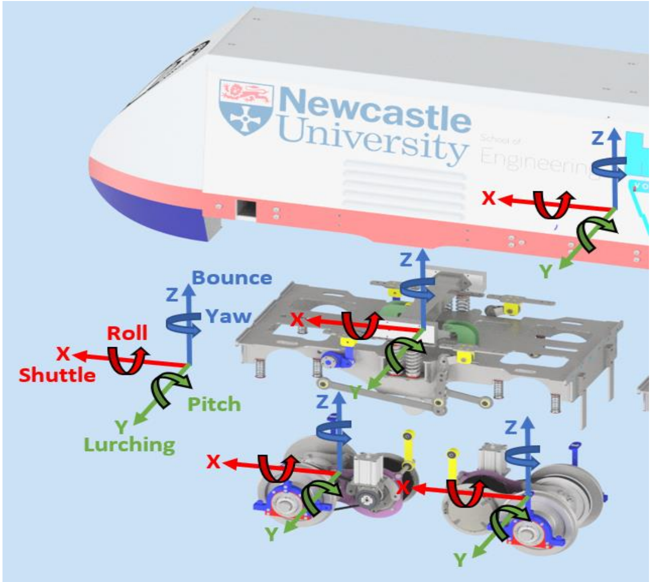


Figure 5-4. Degrees of freedom of the wheelset, bogie, and body.

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

5.2.3 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.

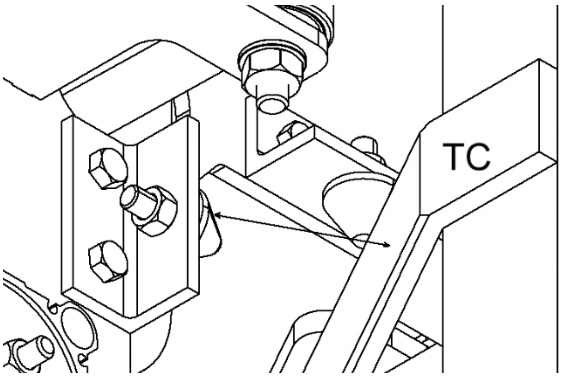


Figure 5-8: CAD Sketch - Bump Stops Positioned Allowing Contact with Traction Centre.

5.2.4. Frame Clearences (DS)

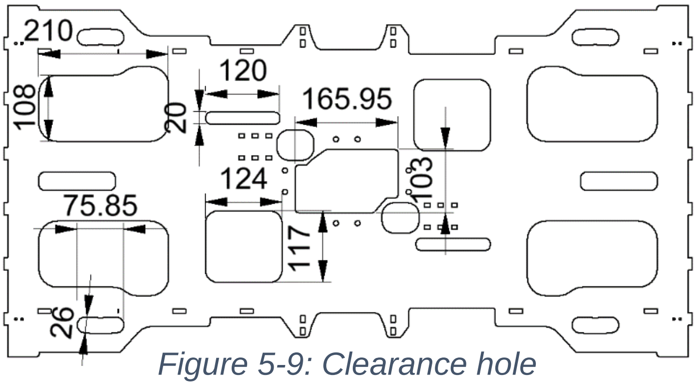


Figure 5-9: Clearance hole dimensions for bogie top plate

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 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

5.3.2.3 Anti-roll Bar Calculations (DS)

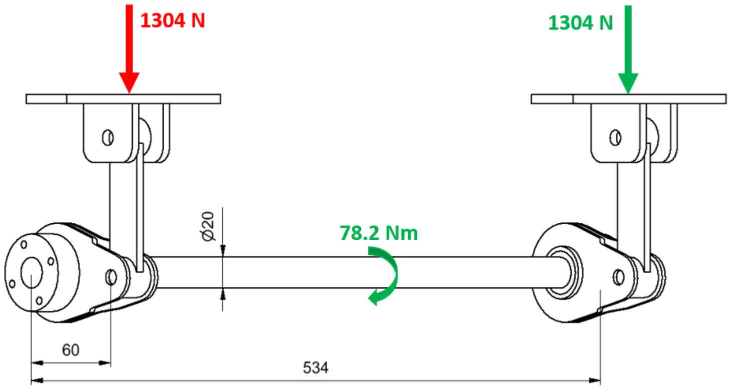


Figure 5-27: Anti-rollbar Torque & stress deflection sketch

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:

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

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.

5.5 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

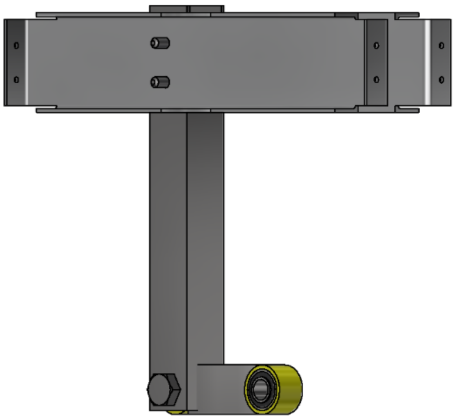


Figure 5-38: Inherited TC design from 2020-2021 locomotive team

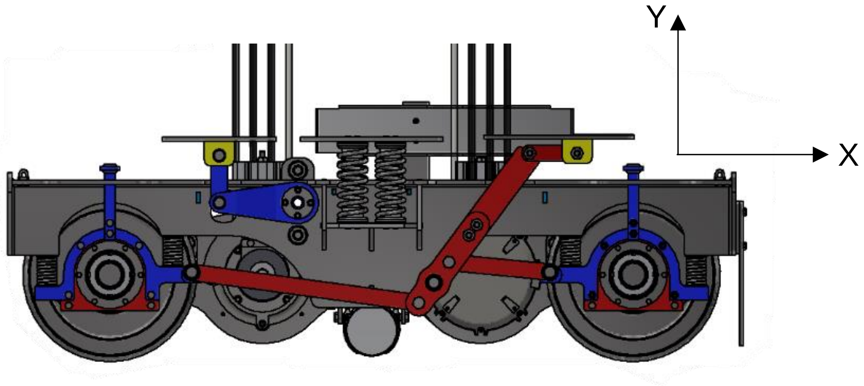


Figure 5-37: Bogie movement axis

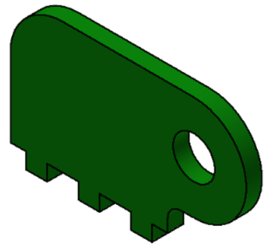


Figure 5-40: TC support bracket

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:

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

Subsequently, welding the TC assembly was deemed sensible.

Figure 5-48: Dimensioned Sketch of TC.

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.

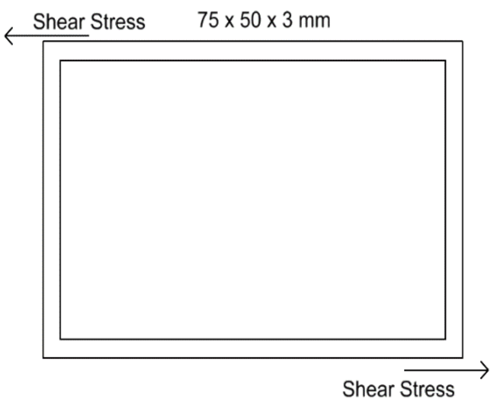


Figure 5-49: Sketched Shear Stress Behaviour.

Whilst shear stress is defined as: $\tau = \frac{Torque}{2 \times A \times t}$ 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 m^2.

Shear stress is then equal to: $\tau = \frac{2100\ Nm}{2\times0.003384\ mm^2\times3mm} = 103.428\ MPa.$

Even if the cheaper option of steel (S275) was used, this is still safe as the shear strength of S275 steel is: $\frac{275}{2} = 137.5\ MPa$, 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)

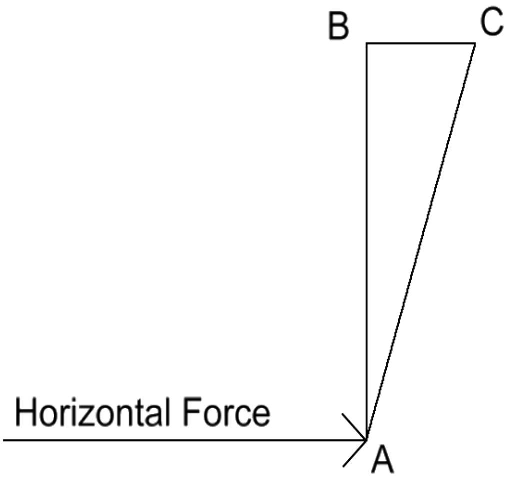


Figure 5-50: Force Illustration for Anti-Buckling.

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 x 10^{-9} m^4. Its critical load is:

$P_0 = \frac{\pi^2\times210\times10^9\times2.56\times10^{-9}}{(2\times0.098185)^2} = 146406.794\ N.$

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

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)	$\frac{M}{F} = 0.0586m$

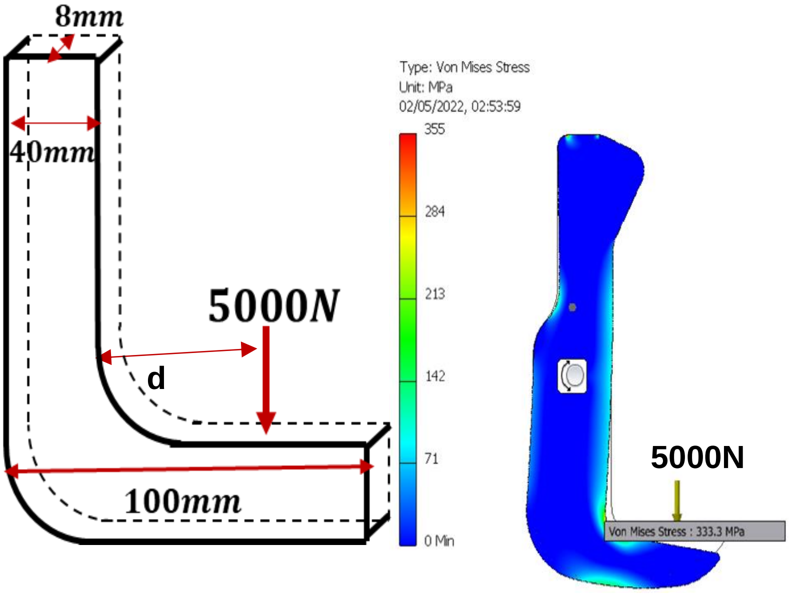


Figure 5-51: Maximum stress for the TC-hook analysis

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.

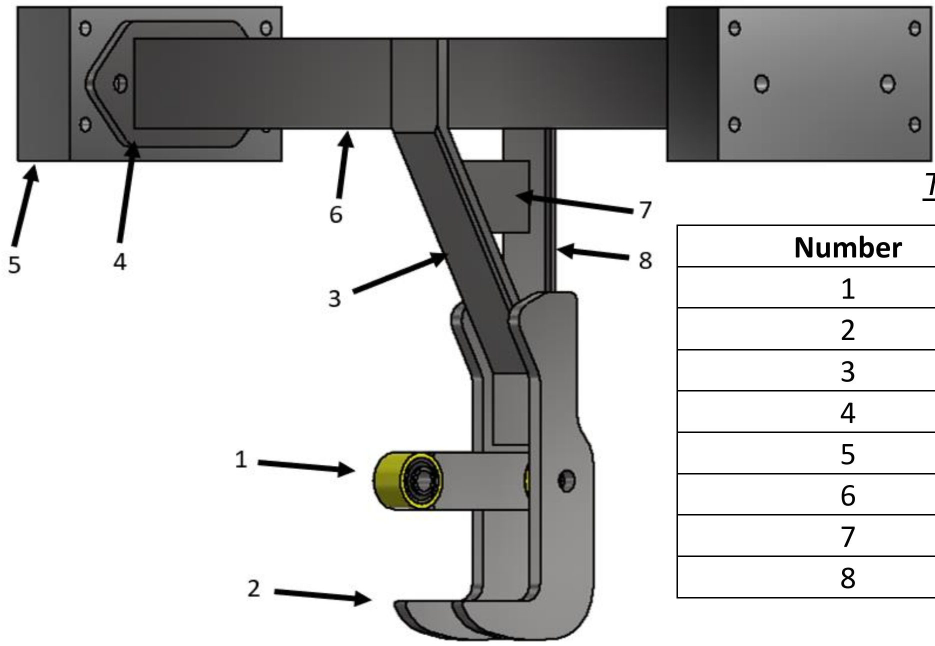


Figure 5.52: Final design

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

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

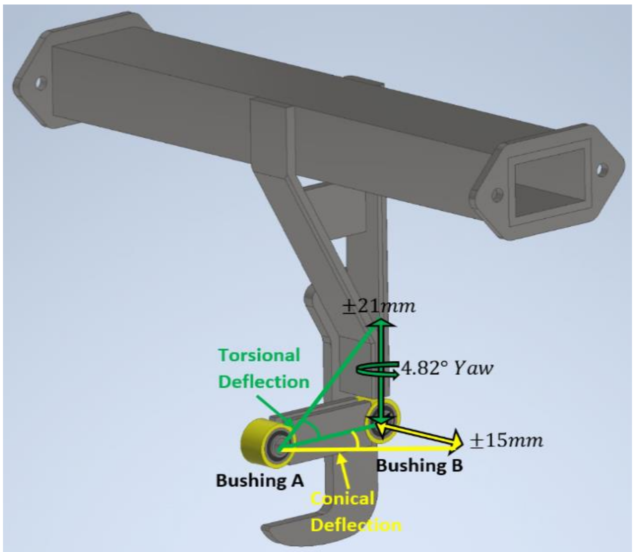


Figure 5-53. Degrees of freedom required by traction centre bushings.

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

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 $\pm 15\ mm$ which only occurs at maximum cornering. Bushing A will

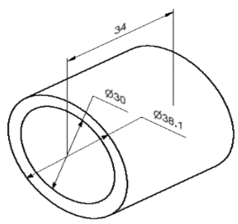


Figure 5-54. Traction centre

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

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.

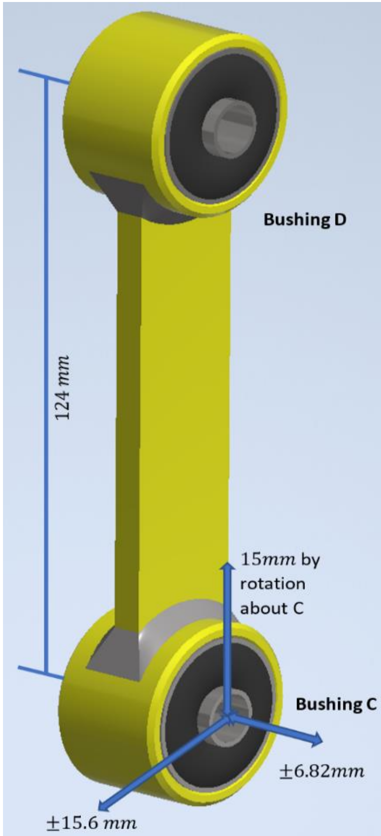


Figure 5-55. Degrees of freedom required by motor mount bushings

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.

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

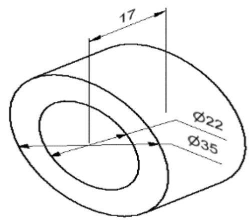


Figure 5-56. Motor Mount arm bushing

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.

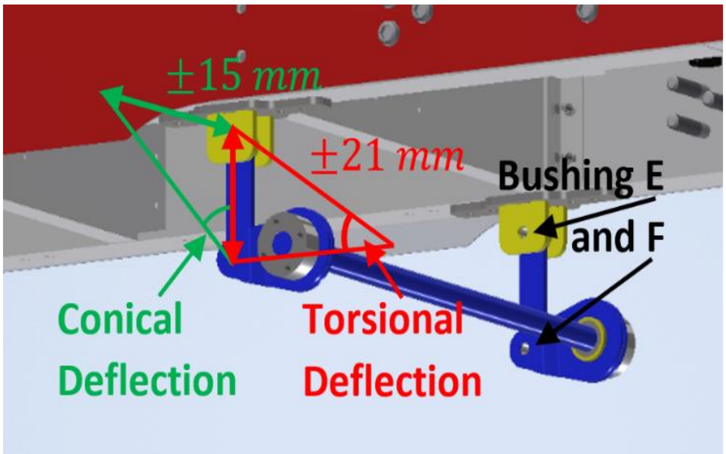


Figure 5-57. Degrees of freedom required by traction centre

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.

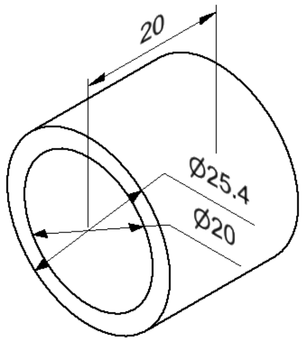


Figure 5-58. Anti-Roll Bar bushing lugs

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.

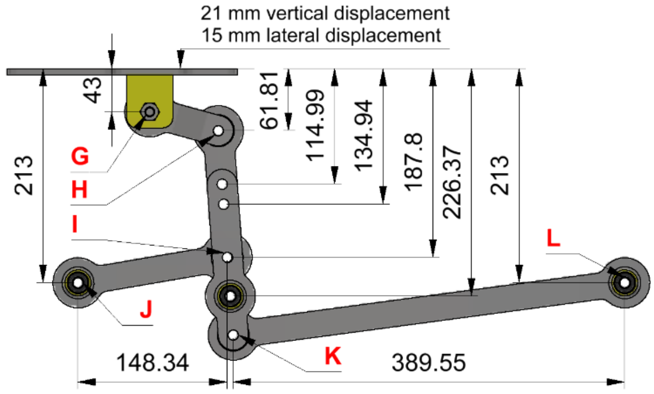


Figure 5-59. Degrees of freedom required by traction centre bushings.

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.

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 = gravitational acceleration = 9.81 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:

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

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):

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

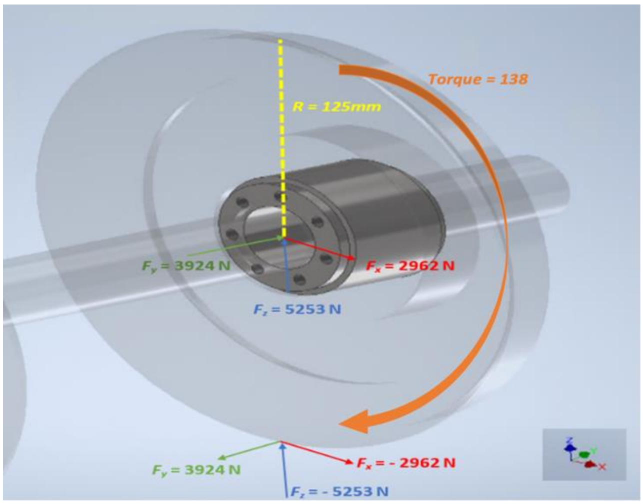


Figure 6-1. Wheel Contact Forces and Torque

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.

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

This value can't be used when calculating torque safety factors of the friction coupling components used. This is because an adhesion coefficient of μ = 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:

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

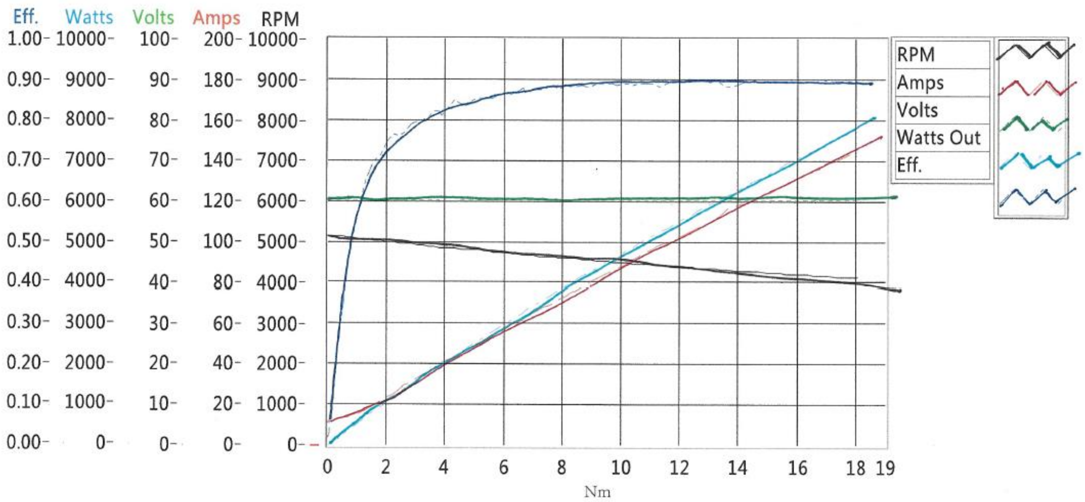


Figure 6-2. Table Displaying Force Multiplication factors

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

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

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

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

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.

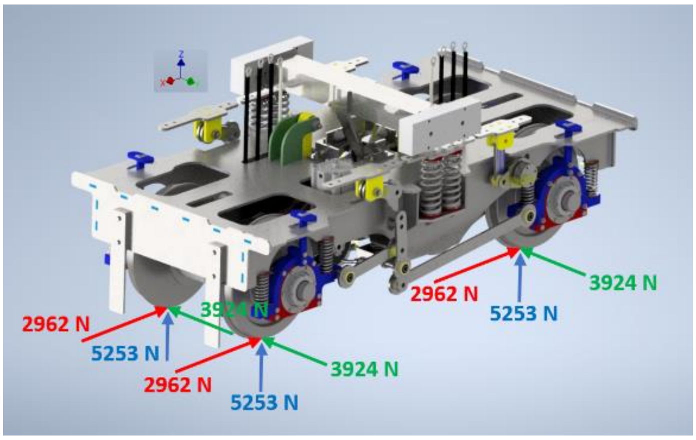


Figure 6-3. Wheel Contact Forces

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.

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.

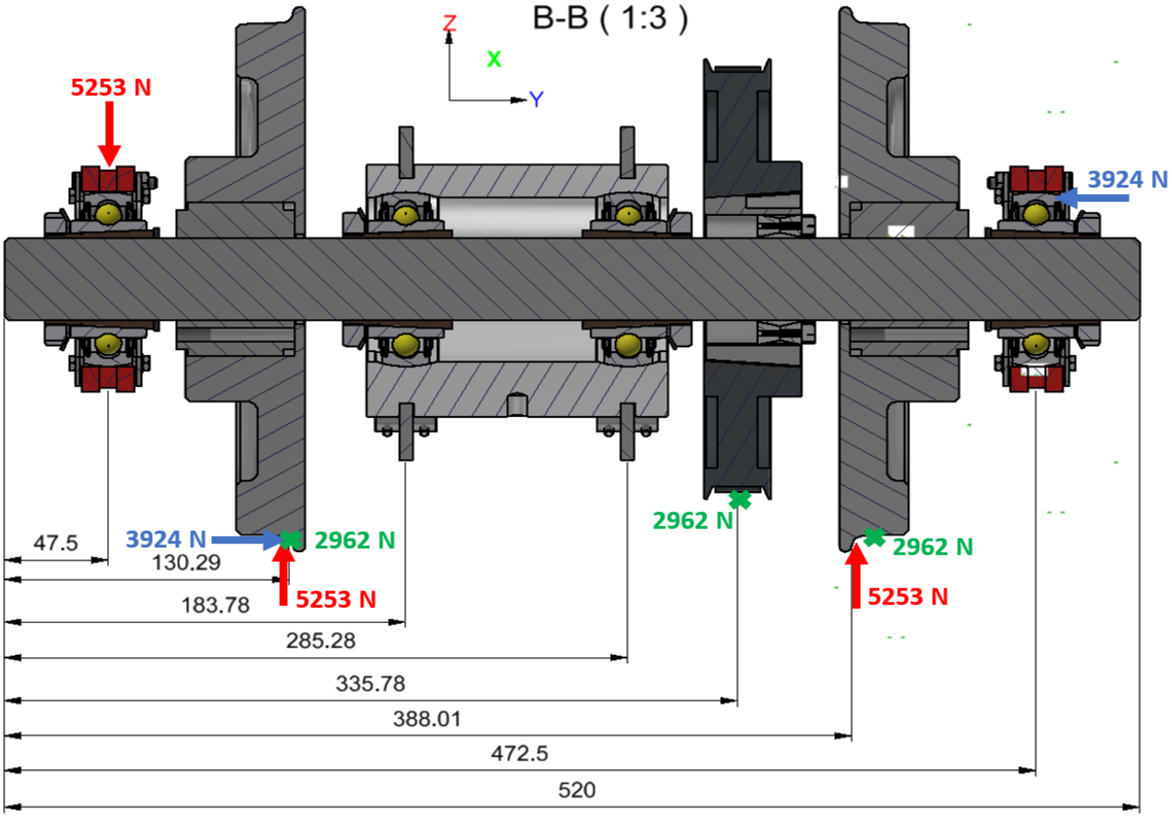


Figure 6-4. Lateral, Longitudinal and Vertical forces acting on shaft

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

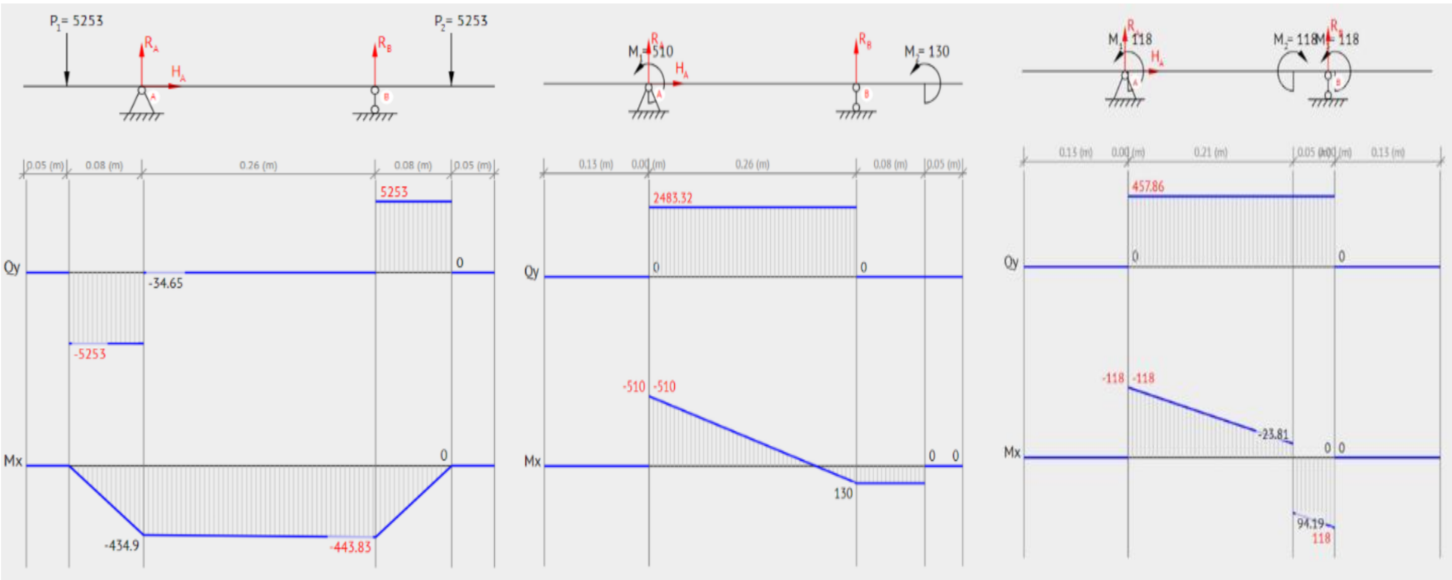


Figure 6-5. Shear Force and Bending Moment Diagrams in the Vertical, Longitudinal and Lateral directions (from left to right) on each Wheelset Axle

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

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.

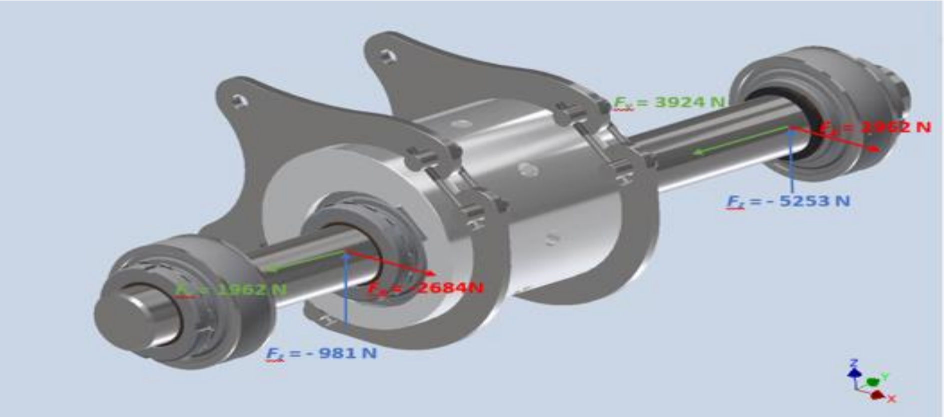


Figure 6-6. Lateral, Longitudinal and Vertical Forces acting on both bearing sets

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:

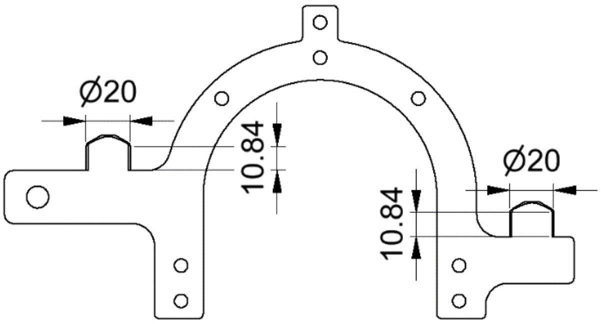


Figure 6-7. Axle Box Dimensions

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

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

6.1.6. Steering Link Calculations (DS)

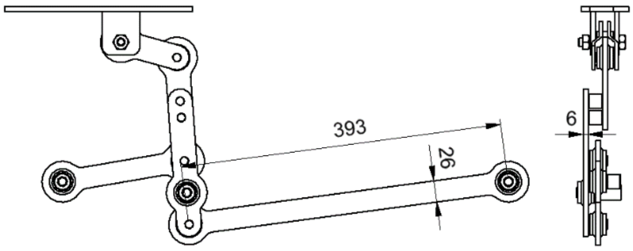


Figure 6-8. Steering Link Dimensions

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 bucking 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.

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 sever 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.

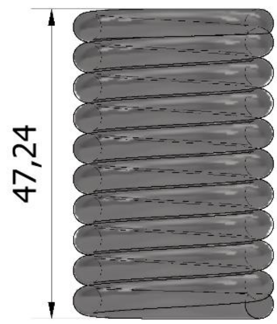


Figure 6-9. Bottomed Out Primary Suspension Spring

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.

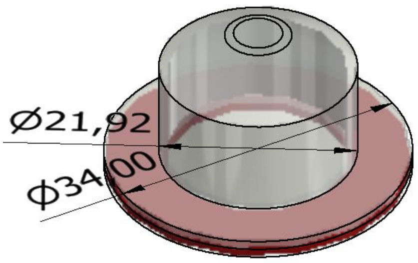


Figure 6-10. Old Primary Suspension Spring Seat

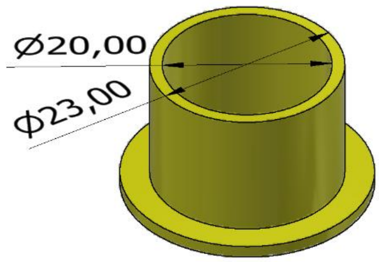


Figure 6-11. New Primary Suspension Spring Seat

6.2. <u>Belt Drive (JJ)</u>

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-

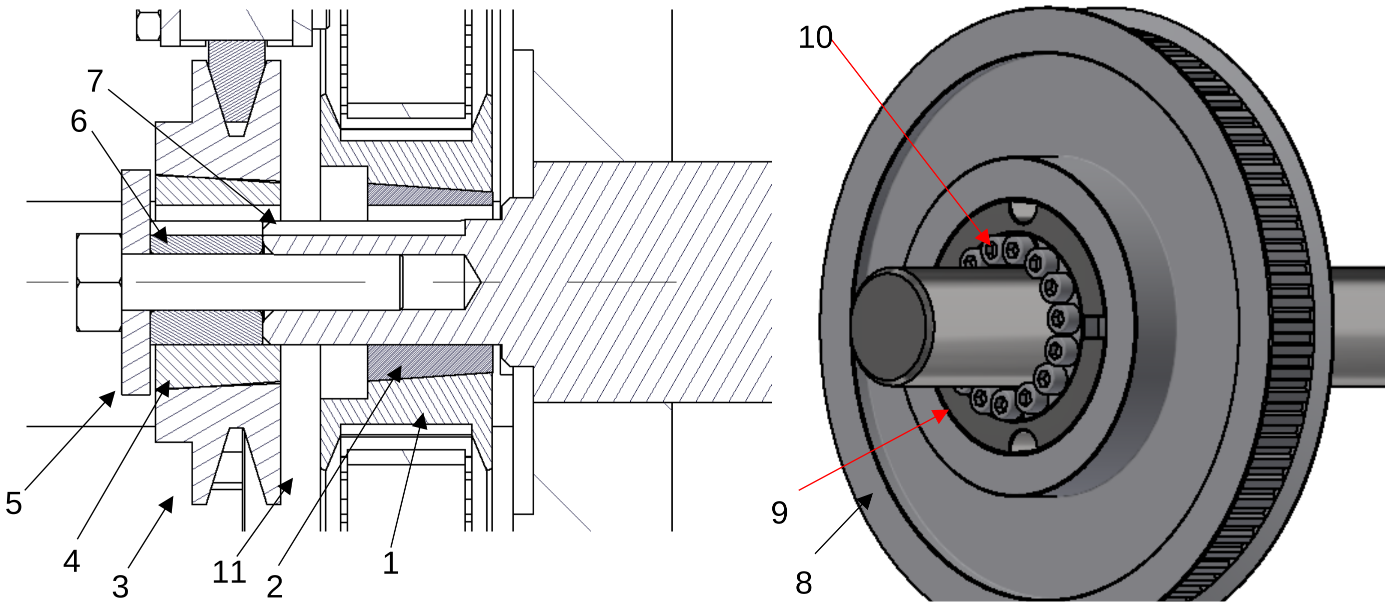


Figure 6.12: Cross section of the small sprocket assembly and design view of the big sprocket

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

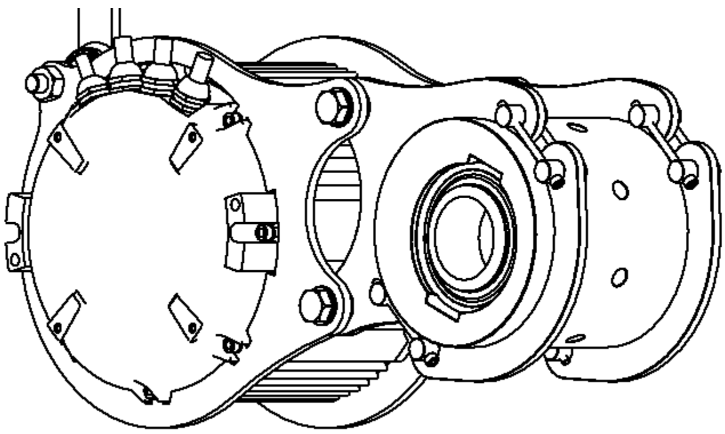


Figure 6-18 Motor Mount Assembly

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 $\pm 15\ mm$ 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.

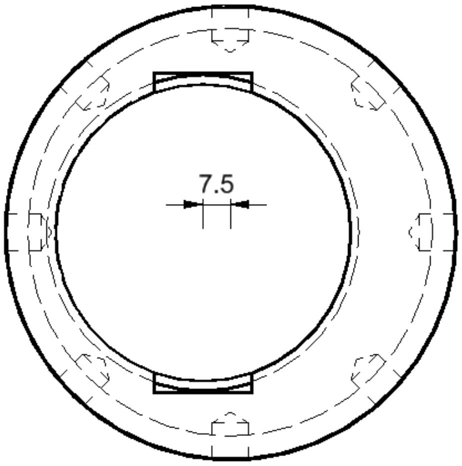


Figure 6-19 Eccentric Device

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:

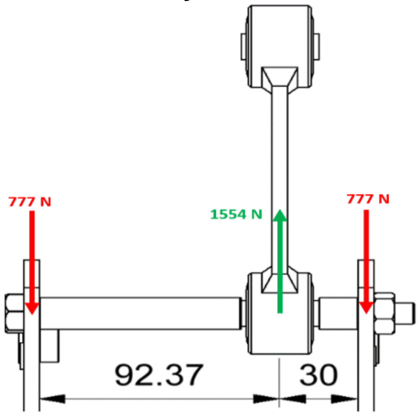


Figure 6-20 Connecting Bolt Three Point Bending

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.

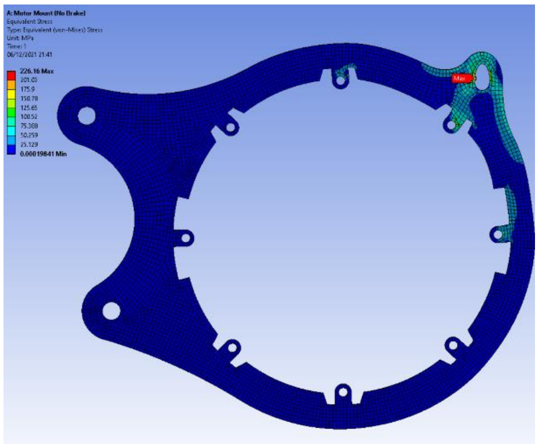


Figure 6-21 Motor Mount Panel FEA Stress Analysis

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.

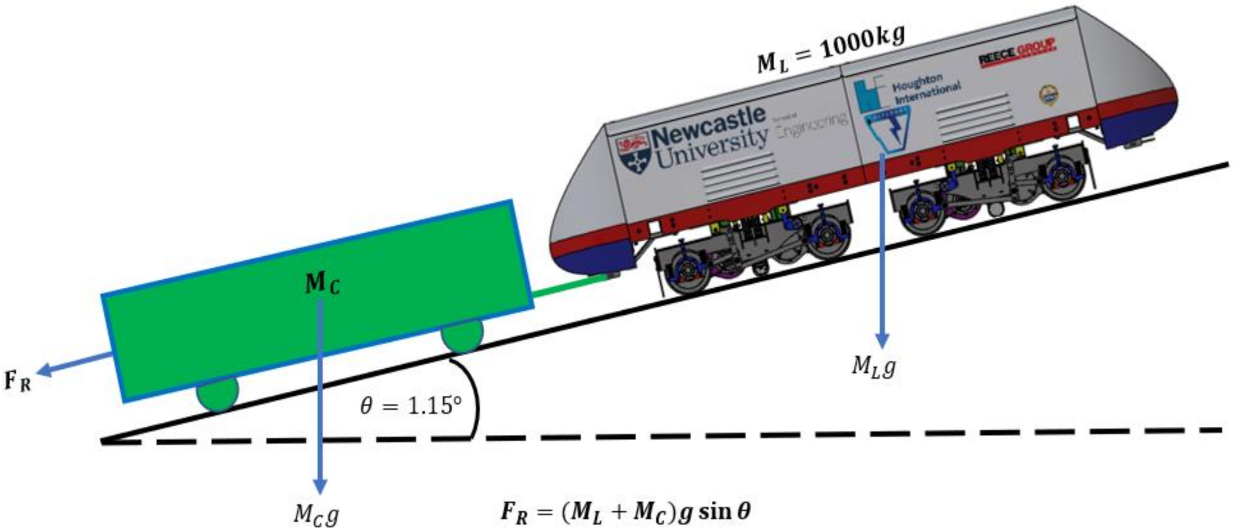


Figure 6-22 Free Body Diagram of Locomotive Trailing Loads at a 1:50 gradient

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
$M_c = 400\ kg$	1.99 A	23.9 A
$M_c = 18000\ kg$	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

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

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 $\pm 0.05\ mm$

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.

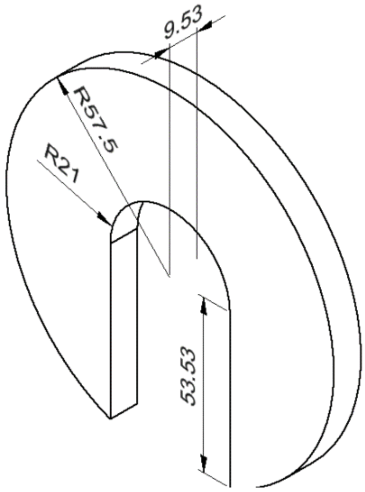


Figure 6-23 Axle Spacer Dimensions

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.

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.

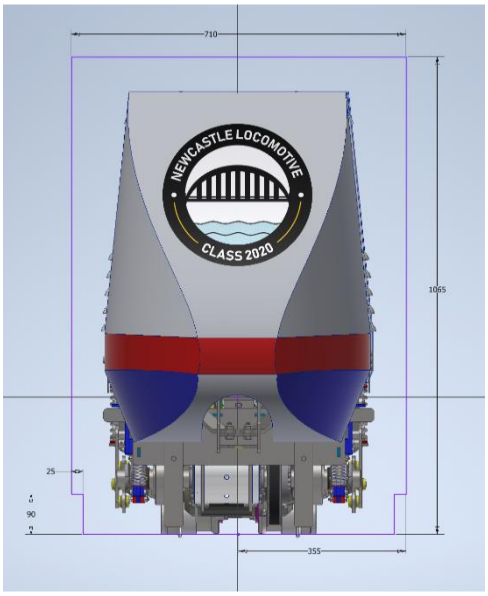


Figure 8-1-1 2022 Locomotive within Dynamic Loading Gauge

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.

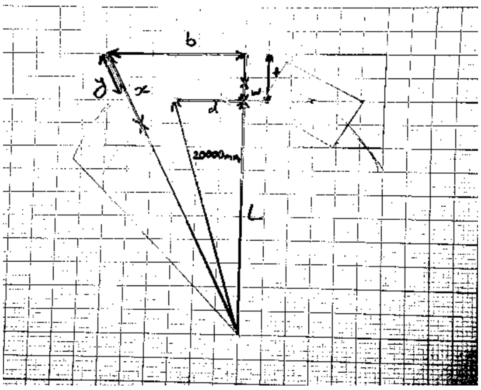


Figure 8-3-1: CAD sketch of dynamic loading gauge overhang.

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 ±15mm. 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.

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

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

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,

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

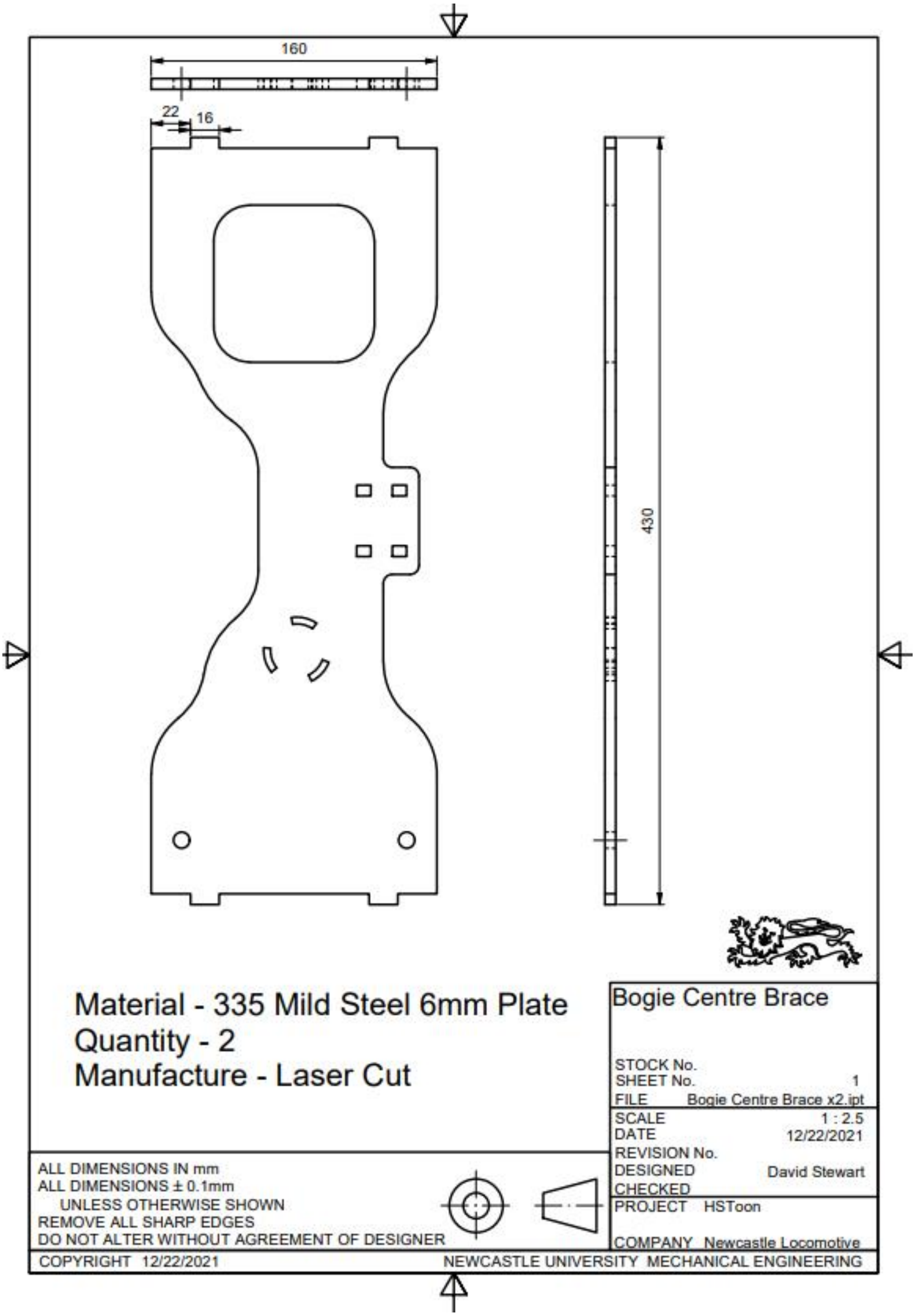
13.2.2 Body

- 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.

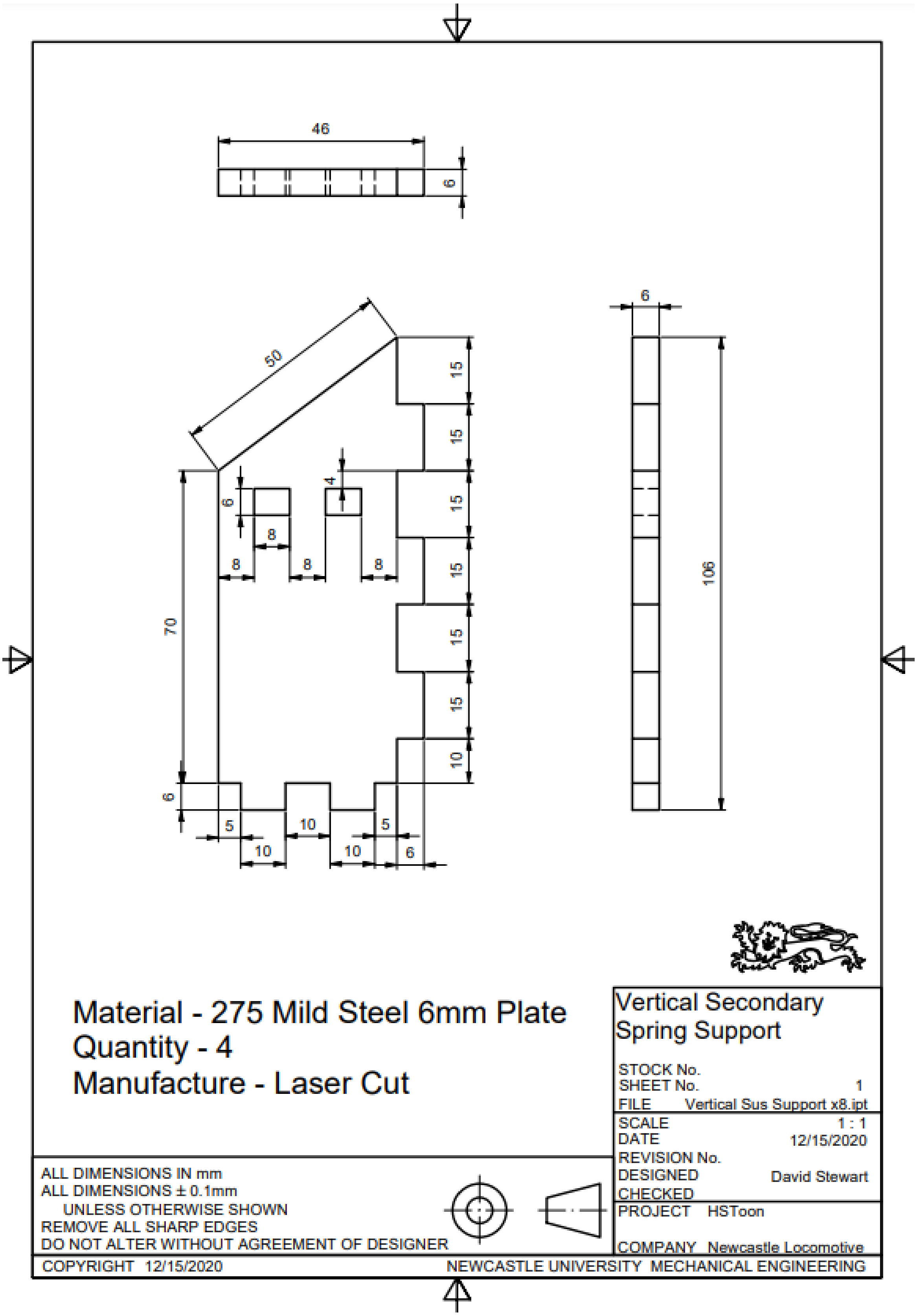
13.2.3 Low Voltage Circuitry (AH)

- 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.

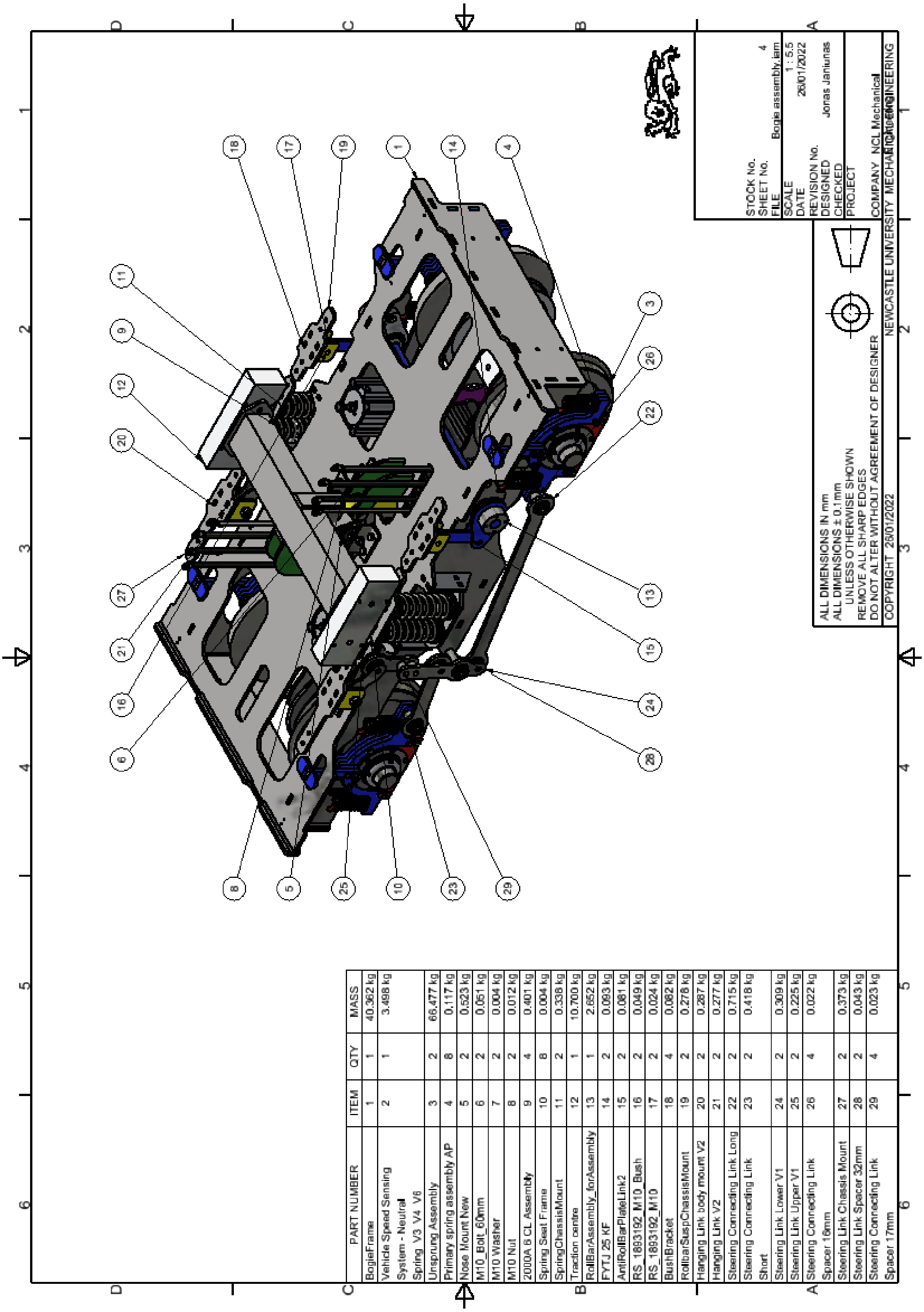
Appendix B-34: Bogie centre-brace (DS)



Appendix B-35: Vertical secondary-spring platform (DS)

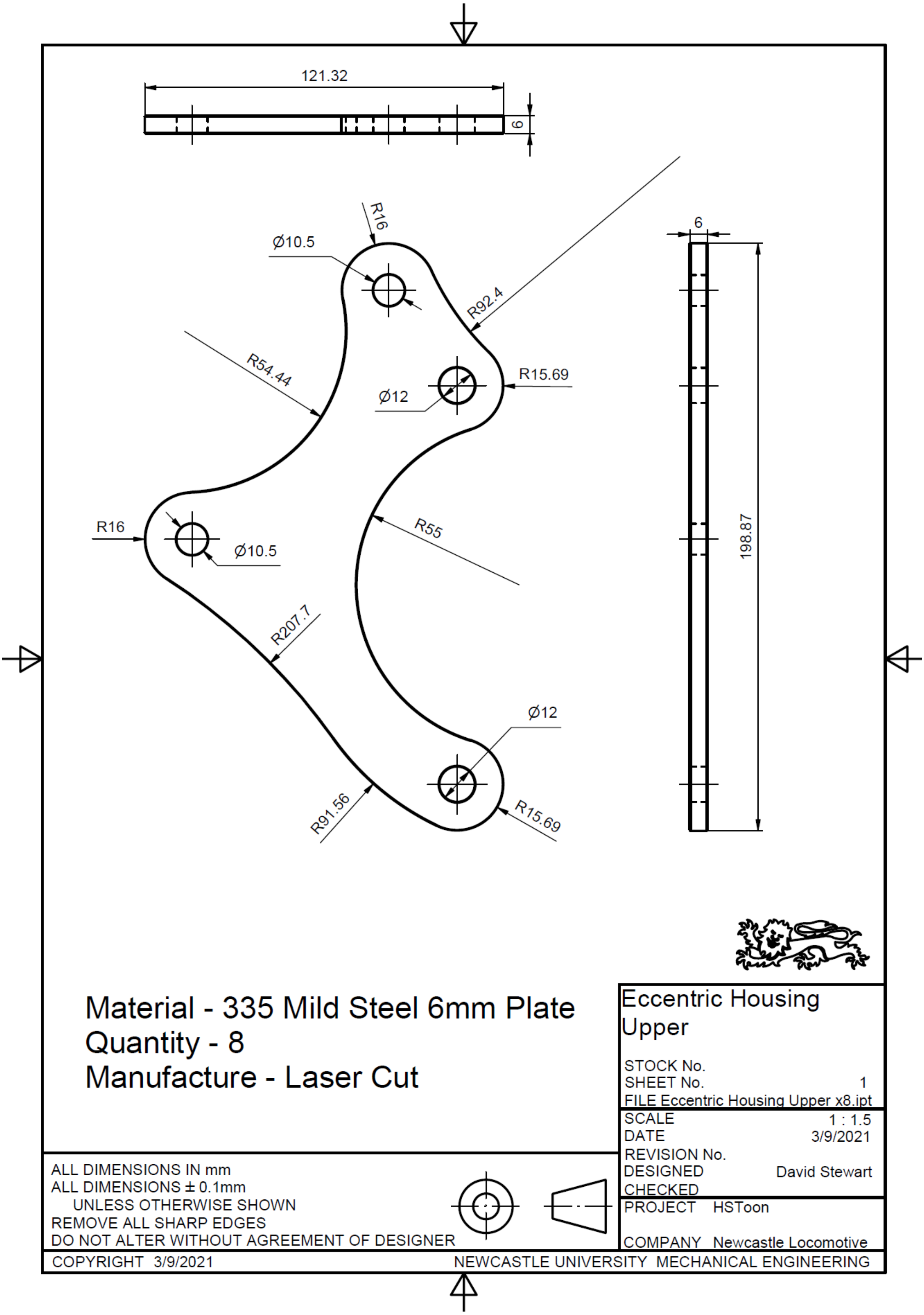


Appendix B-36: Bogie system assembly (DS, LK, JJ)

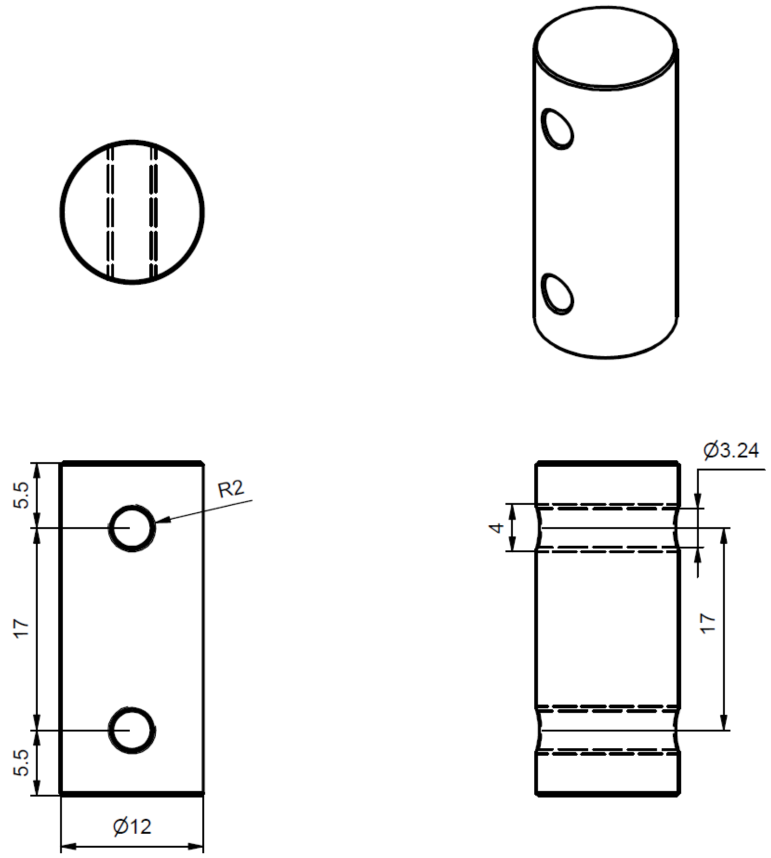


PART NUMBER	ITEM	QTY	MASS
BogieFrame	1	1	40.362 kg
Vehicle Speed Sensing System - Neutral Spring_V3 V4 V6	2	1	3.498 kg
Unsprung Assembly	3	2	66.477 kg
Primary spring assembly AP	4	8	0.117 kg
Nose Mount New	5	2	0.523 kg
M10_Bolt_60mm	6	2	0.051 kg
M10 Washer	7	2	0.004 kg
M10 Nut	8	2	0.012 kg
2000A 6 CL Assembly	9	4	0.401 kg
Spring Seat Frame	10	8	0.004 kg
SpringChassisMount	11	2	0.338 kg
Traction centre	12	1	10.700 kg
RollBarAssembly_forAssembly	13	1	2.652 kg
FYTJ 25 KF	14	2	0.093 kg
AntiRollBarPlateLink2	15	2	0.081 kg
RS_1893192_M10_Bush	16	2	0.049 kg
RS_1893192_M10	17	2	0.024 kg
BushBracket	18	4	0.082 kg
RollbarSuspChassisMount	19	2	0.278 kg
Hanging Link body mount V2	20	2	0.287 kg
Hanging Link V2	21	2	0.277 kg
Steering Connecting Link Long	22	2	0.715 kg
Steering Connecting Link Short	23	2	0.418 kg
Steering Link Lower V1	24	2	0.309 kg
Steering Link Upper V1	25	2	0.225 kg
Steering Connecting Link Spacer 16mm	26	4	0.022 kg
Steering Link Chassis Mount	27	2	0.373 kg
Steering Link Spacer 32mm	28	2	0.043 kg
Steering Connecting Link Spacer 17mm	29	4	0.023 kg

Appendix B-56: (DS)



Appendix B-57: (DS)



Geometry - Eccentric Clamping Pins
Material - S275
Quantity - x 32 pins

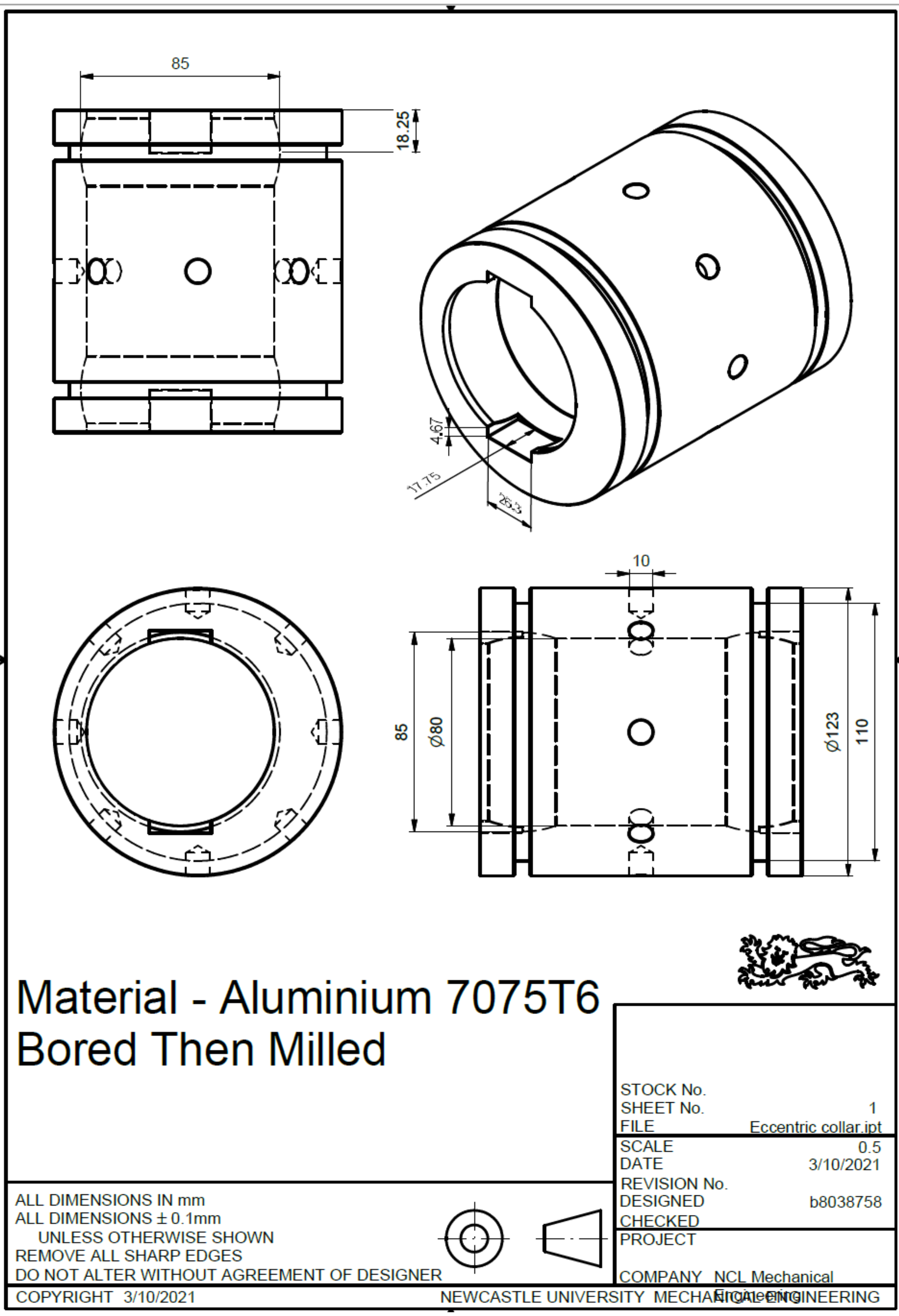
IMPORTANT
16 pins to have two M4 tapped holes
16 pins to have two M4 clearance holes

STOCK No.	
SHEET No.	1
FILE	Housing Clamp Pin - M4 Tapped.ipt
SCALE	2:1
DATE	3/10/2021
REVISION No.	
DESIGNED	b8038758
CHECKED	
PROJECT	
COMPANY	NCL Mechanical Engineering

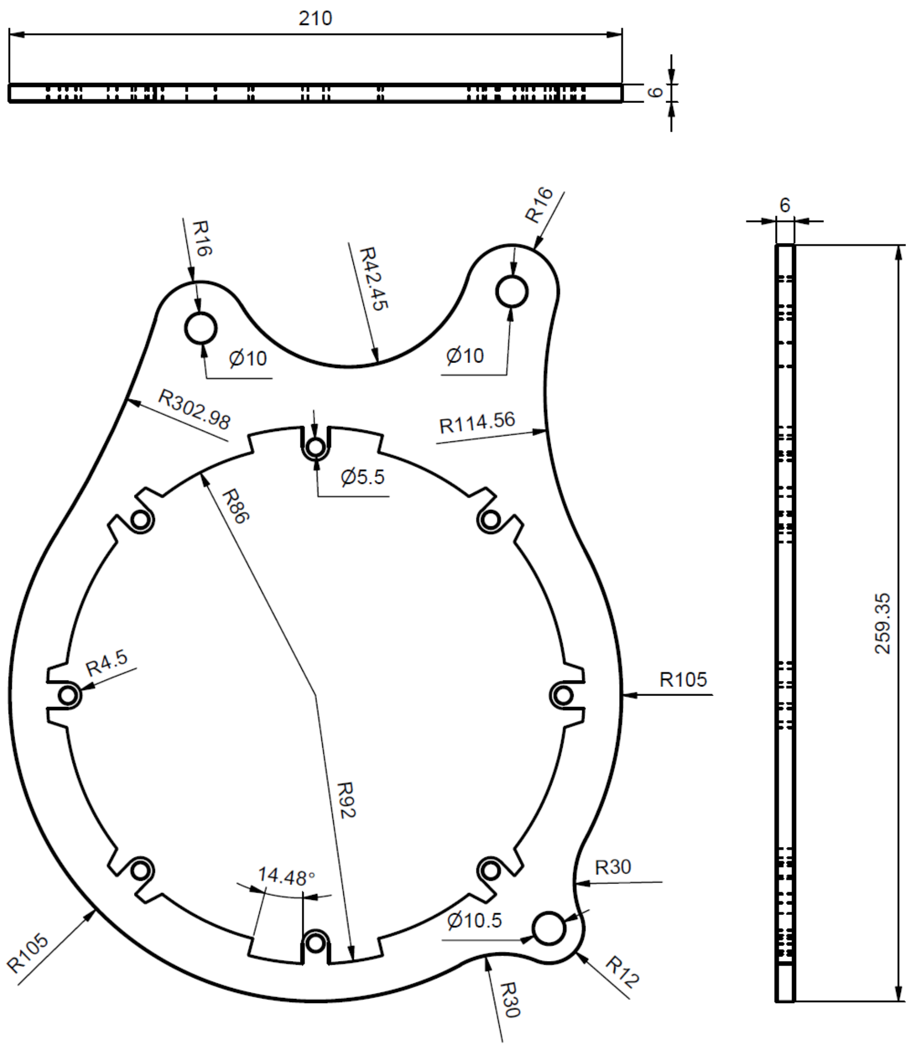
ALL DIMENSIONS IN mm
ALL DIMENSIONS ± 0.1mm
UNLESS OTHERWISE SHOWN
REMOVE ALL SHARP EDGES
DO NOT ALTER WITHOUT AGREEMENT OF DESIGNER

 NEWCASTLE UNIVERSITY MECHANICAL ENGINEERING

Appendix B-58: (DS)



Appendix B-59: (DS)



Material - 335 Mild Steel 6mm Plate
Quantity - 4
Manufacture - Laser Cut

Motor Mount Blank Side

STOCK No.	B059
SHEET No.	1
FILE	Motor Mount Blank Side x4.ipt
SCALE	1 : 2
DATE	10/22/2019
REVISION No.	
DESIGNED	David Stewart
CHECKED	M.Hoque
PROJECT	HSToon
COMPANY	Newcastle Locomotive

ALL DIMENSIONS IN mm
ALL DIMENSIONS ± 0.1mm
UNLESS OTHERWISE SHOWN
REMOVE ALL SHARP EDGES
DO NOT ALTER WITHOUT AGREEMENT OF DESIGNER

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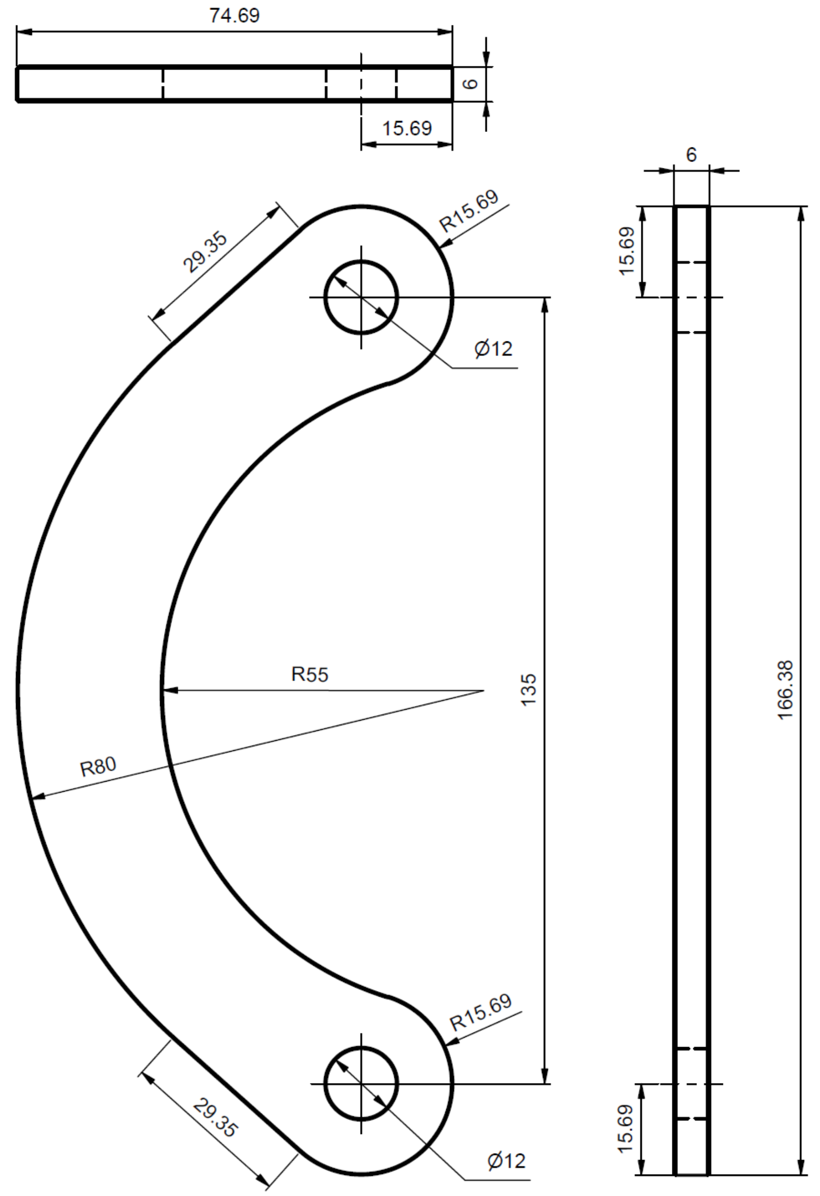
Appendix B-60: (DS)



Material - 335 Mild Steel 6mm Plate
Quantity - 8
Manufacture - Laser Cut

Eccentric Housing Lower

STOCK No.	
SHEET No.	1
FILE	Eccentric Housing Lower x8.ipt
SCALE	1 : 1
DATE	12/17/2021
REVISION No.	
DESIGNED	David Stewart
CHECKED	
PROJECT	HSToon
COMPANY	Newcastle Locomotive

ALL DIMENSIONS IN mm
ALL DIMENSIONS ± 0.1mm
UNLESS OTHERWISE SHOWN
REMOVE ALL SHARP EDGES
DO NOT ALTER WITHOUT AGREEMENT OF DESIGNER

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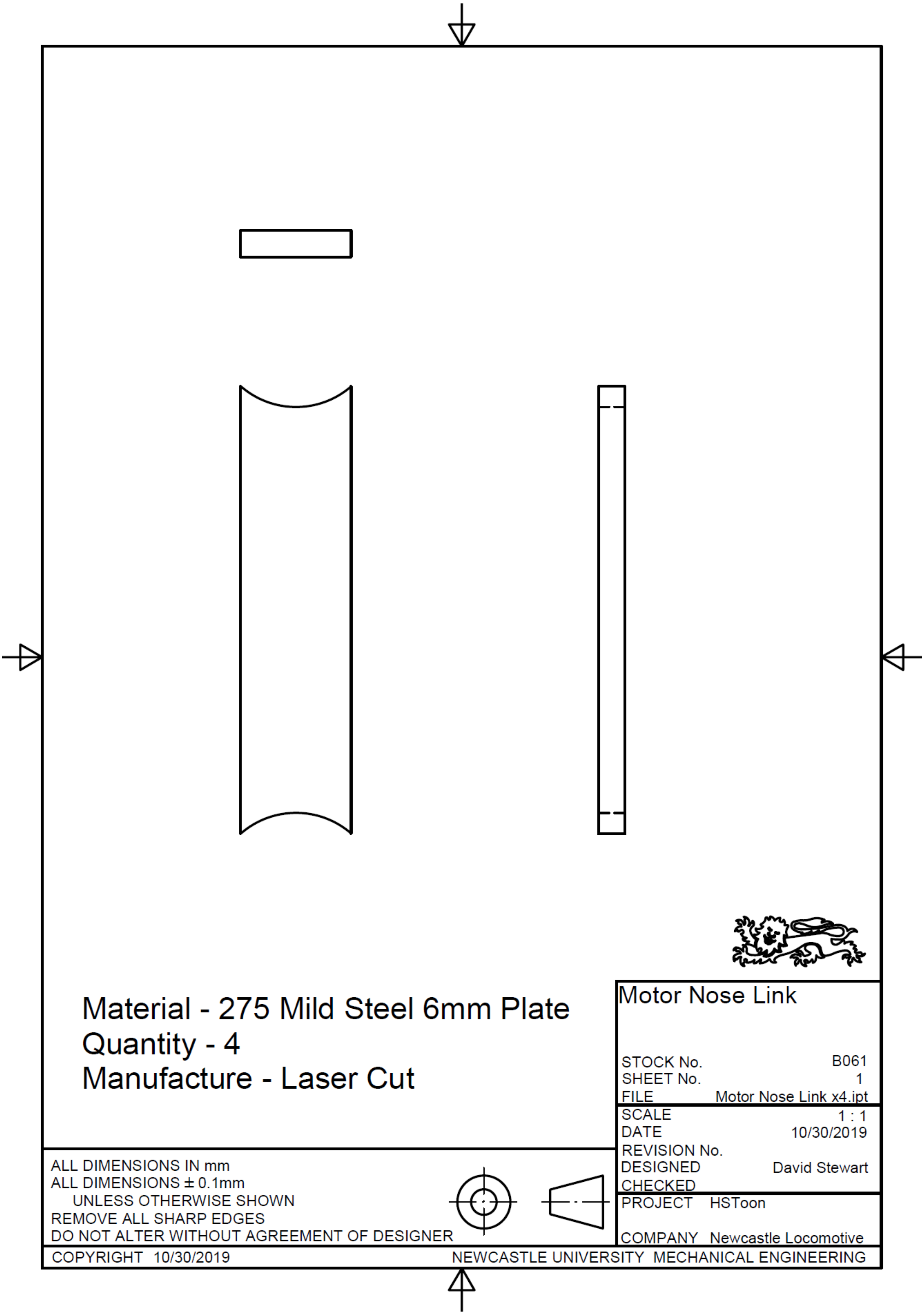
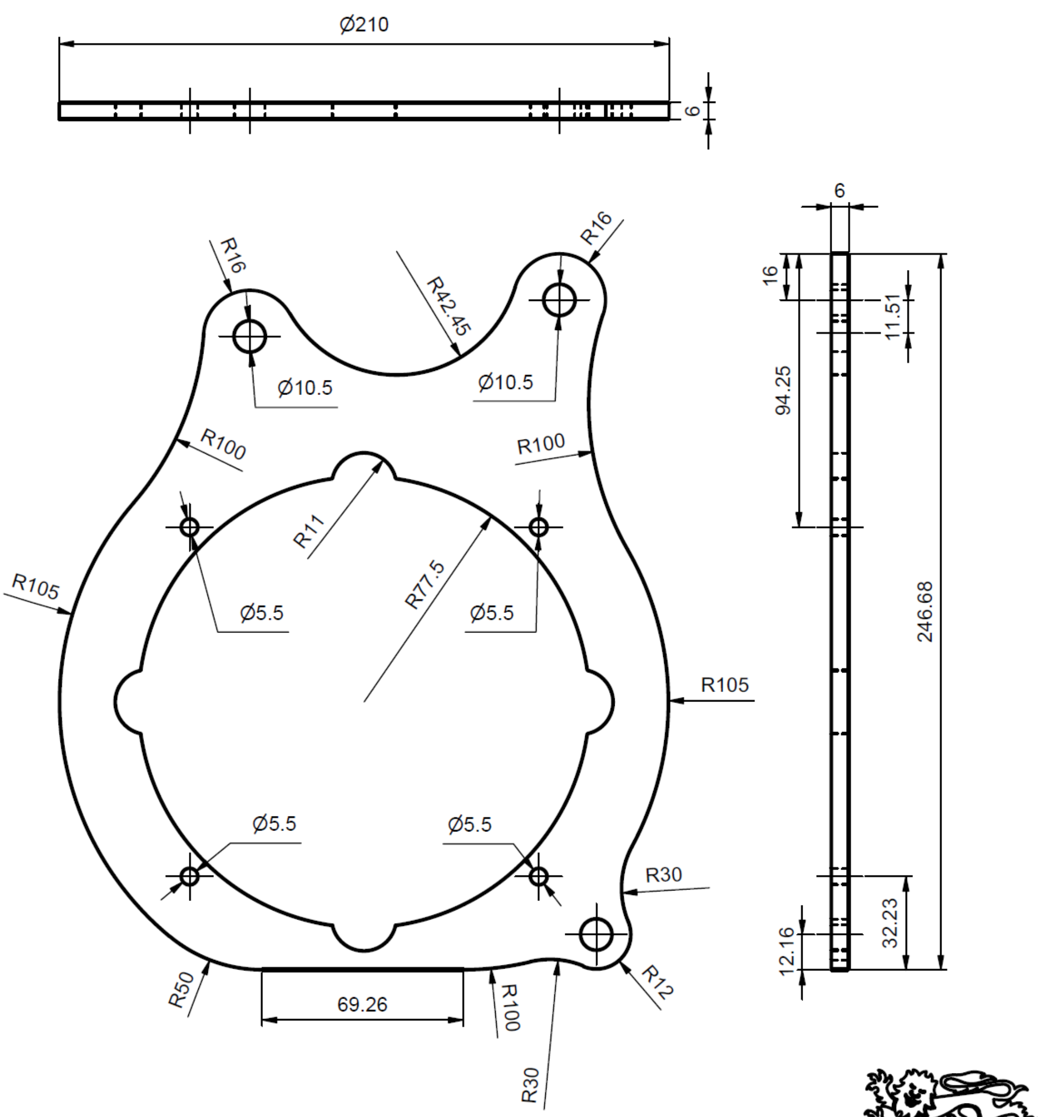
Appendix B-61: (DS)



Appendix B-62: (DS)



Material - 335 Mild Steel 6mm Plate
Quantity - 4
Manufacture - Laser Cut

Motor Mount Brake Side	
STOCK No.	
SHEET No.	1
FILE Motor mount Brake Side x4.ipt	
SCALE	1 : 2
DATE	4/14/2021
REVISION No.	
DESIGNED	David Stewart
CHECKED	
PROJECT	HSToon
COMPANY	Newcastle Locomotive

ALL DIMENSIONS IN mm
ALL DIMENSIONS ± 0.1mm
UNLESS OTHERWISE SHOWN
REMOVE ALL SHARP EDGES
DO NOT ALTER WITHOUT AGREEMENT OF DESIGNER

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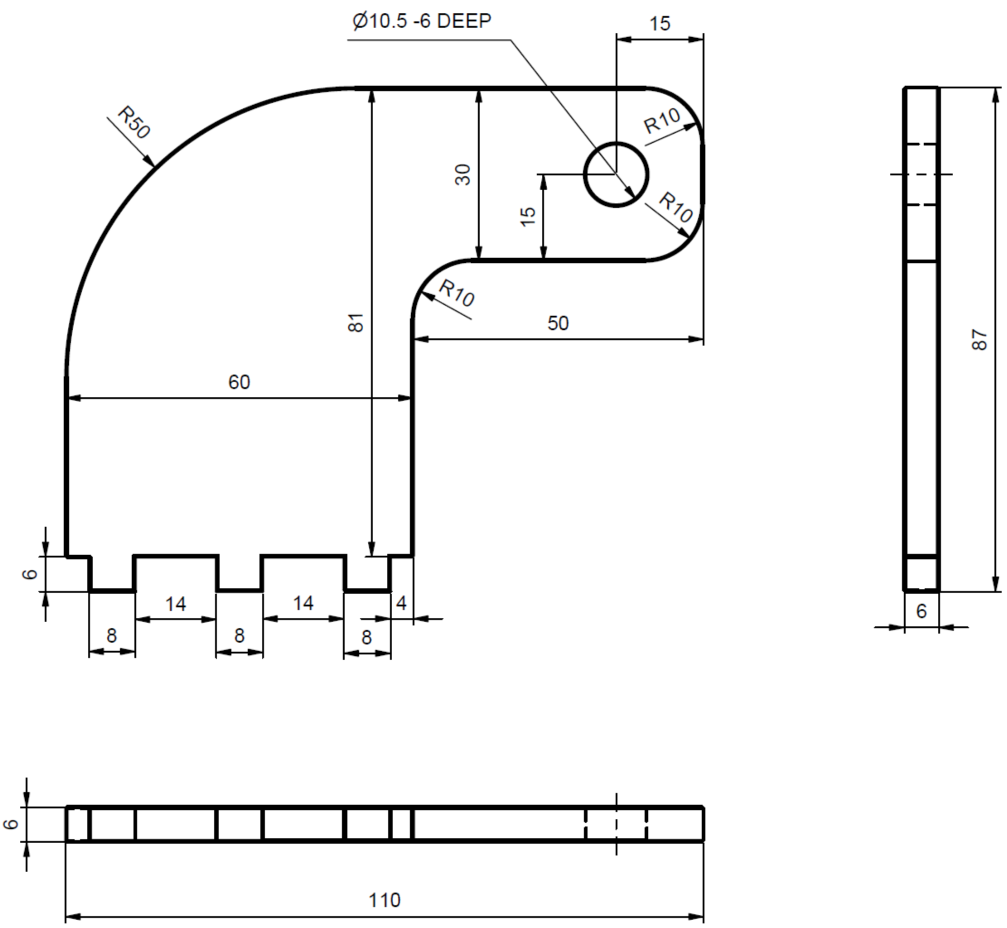
Appendix B-63: (DS)



Material - 335 Mild Steel 6mm Plate
Quantity - 8
Manufacture - Laser Cut

Motor-Bogie Bracket	
STOCK No.	
SHEET No.	1
FILE	Motor-Bogie Bracket x8.ipt
SCALE	1 : 1
DATE	12/2/2021
REVISION No.	
DESIGNED	David Stewart
CHECKED	
PROJECT	HSToon
COMPANY	Newcastle Locomotive

ALL DIMENSIONS IN mm
ALL DIMENSIONS ± 0.1mm
UNLESS OTHERWISE SHOWN
REMOVE ALL SHARP EDGES
DO NOT ALTER WITHOUT AGREEMENT OF DESIGNER

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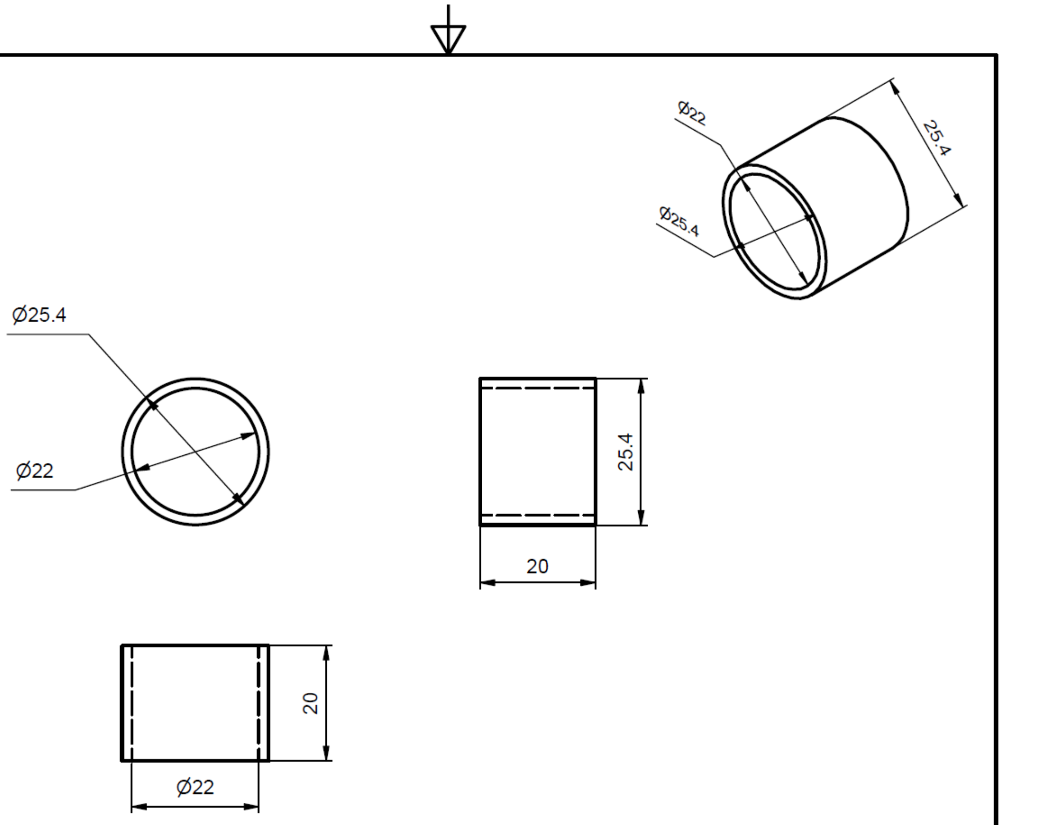
Appendix B-64: (DS)



Material - E275 Steel Round Tube

Quantity - x 10 (with some material left over)

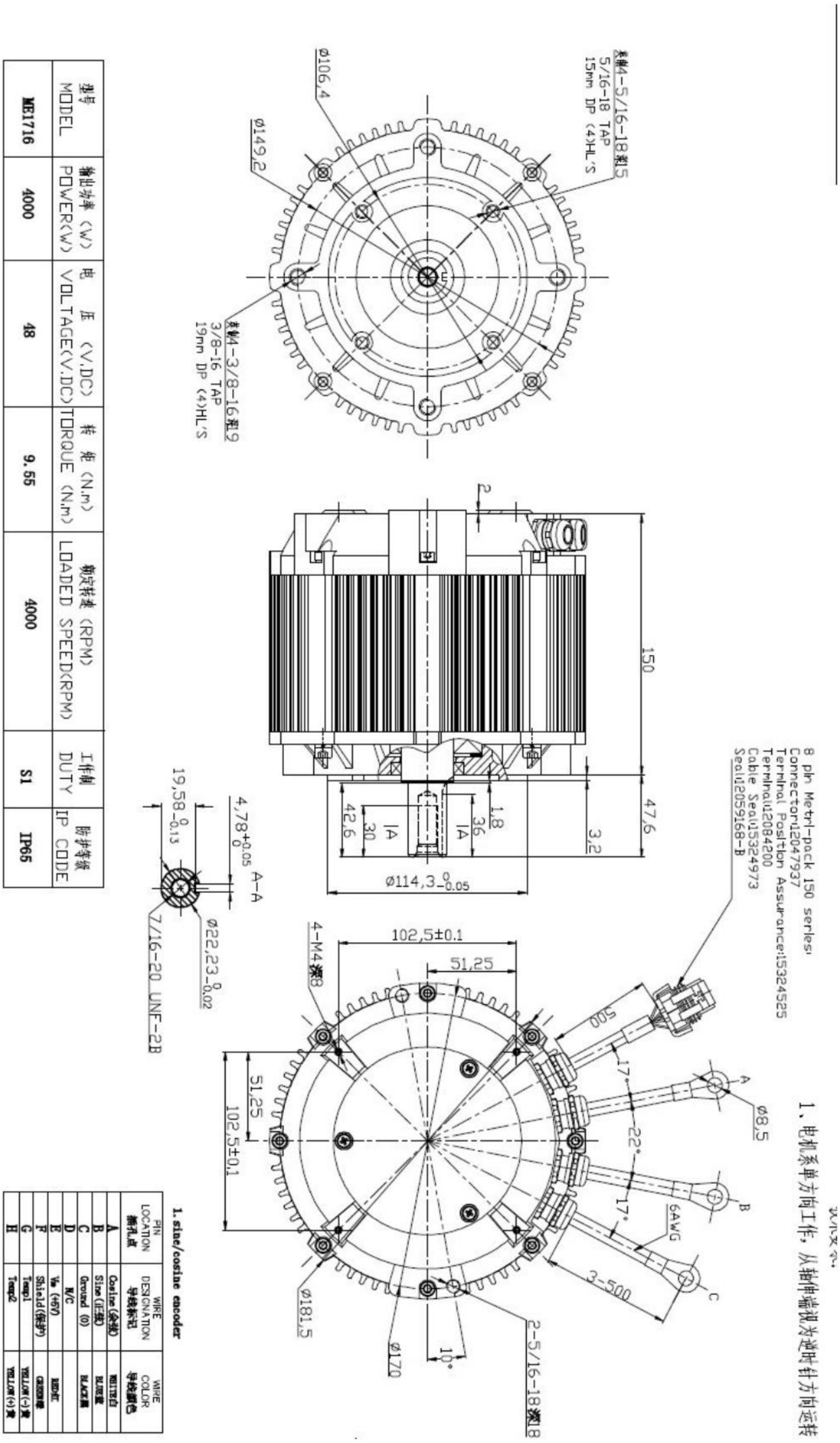
Bore inner diameter (around 20mm) so an RS Pro 189-3192 bushing interference fits inside.
(Bushing provided for reference)

ALL DIMENSIONS IN mm
ALL DIMENSIONS ± 0.1mm
UNLESS OTHERWISE SHOWN
REMOVE ALL SHARP EDGES
DO NOT ALTER WITHOUT AGREEMENT OF DESIGNER

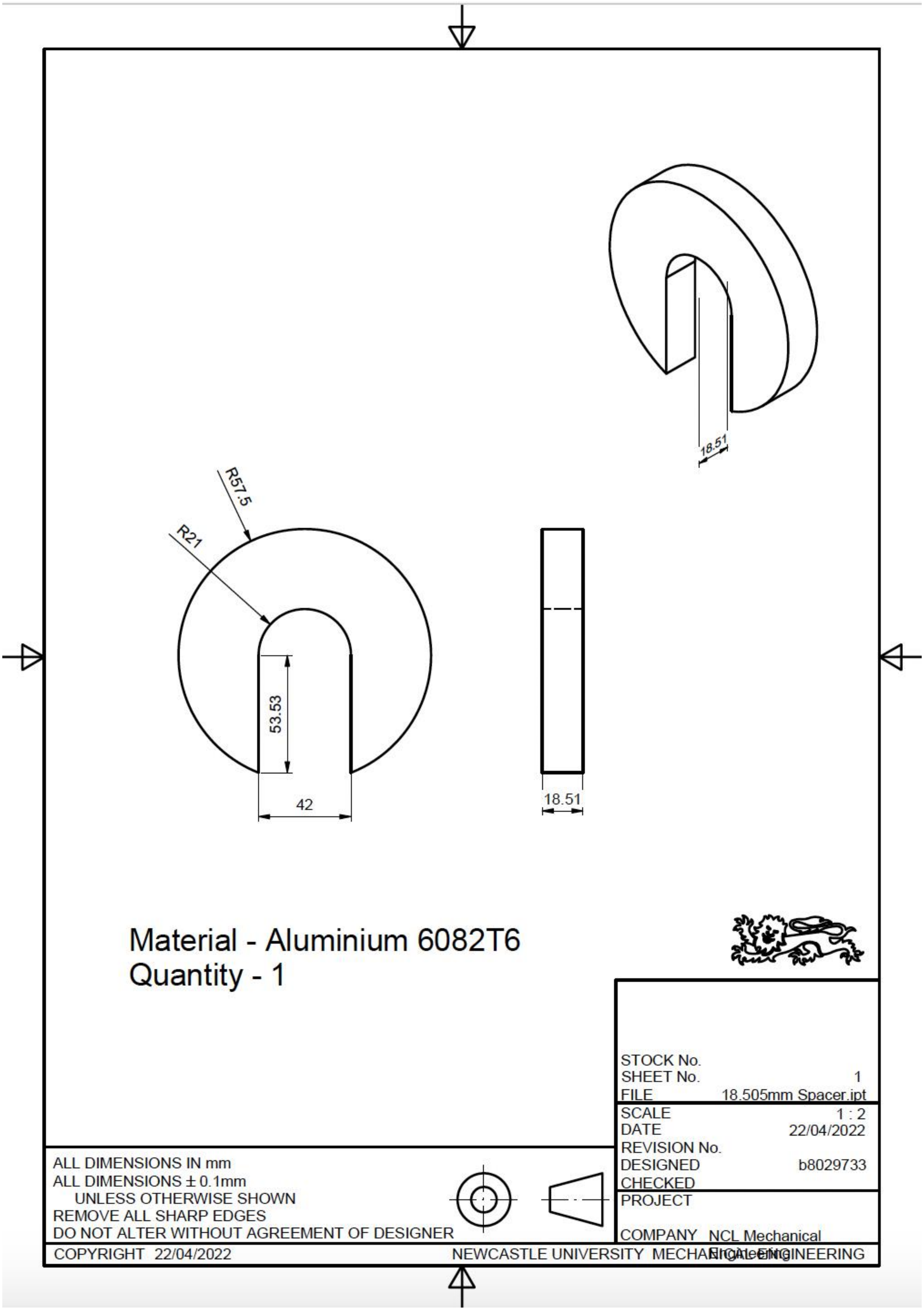
STOCK No.	
SHEET No.	1
FILE	RS Bushing Lug.ipt
SCALE	1 : 1
DATE	3/24/2022
REVISION No.	
DESIGNED	b8029733
CHECKED	
PROJECT	
COMPANY	NCL Mechanical Engineering

 NEWCASTLE UNIVERSITY MECHANICAL ENGINEERING

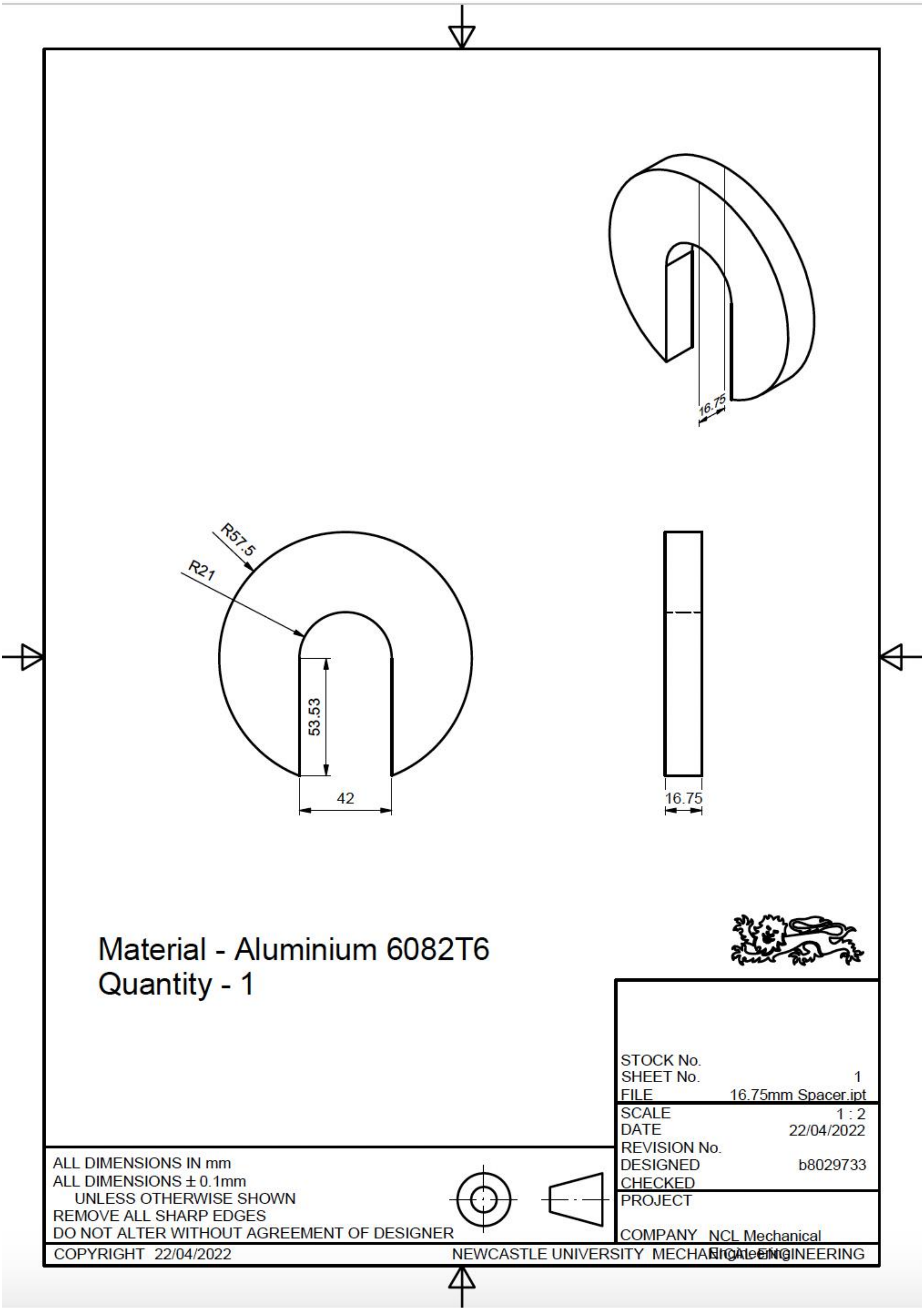
Appendix B-65: Motenergy 1718 Engineering Drawing (DS)



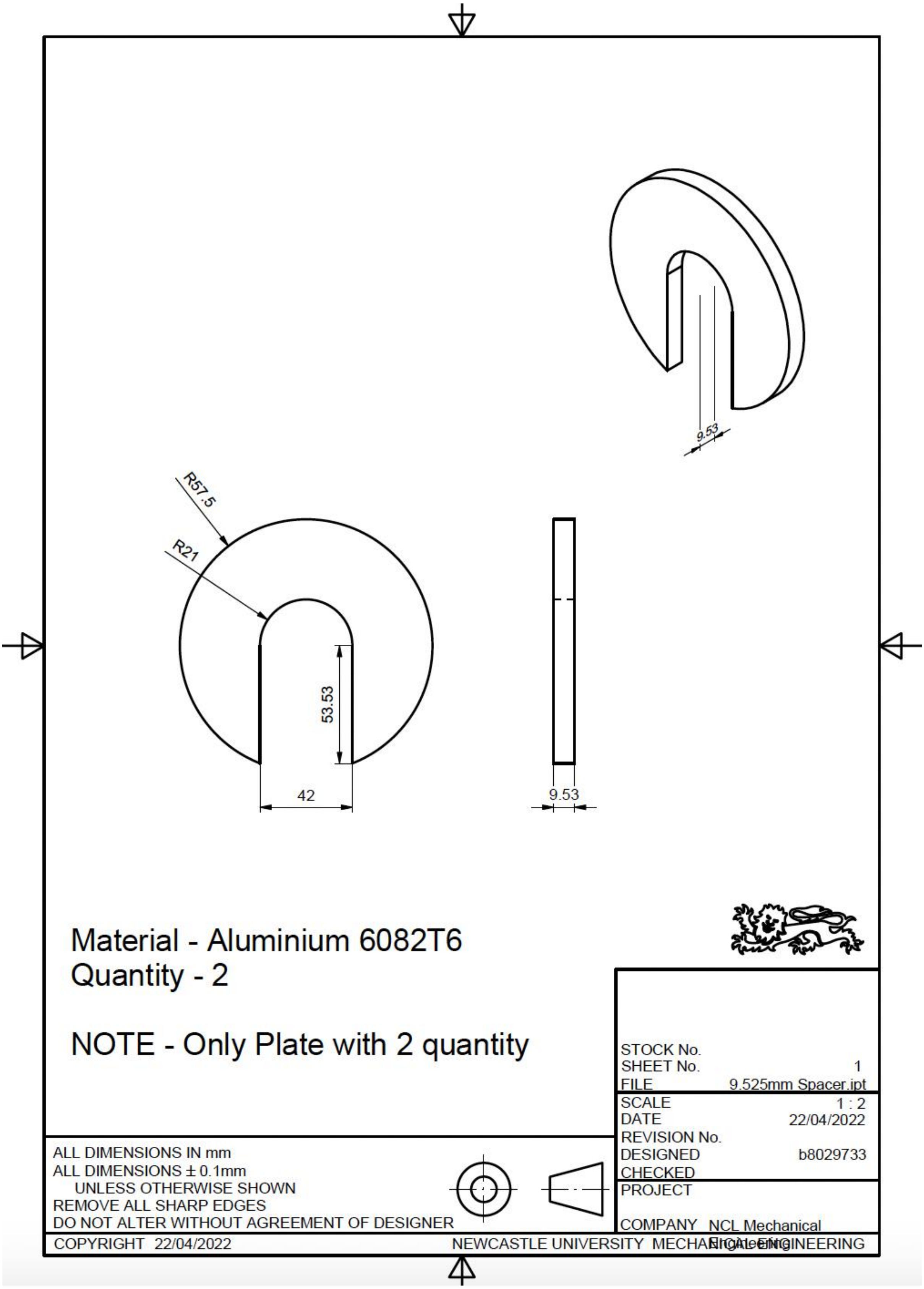
Appendix B-66: (DS)



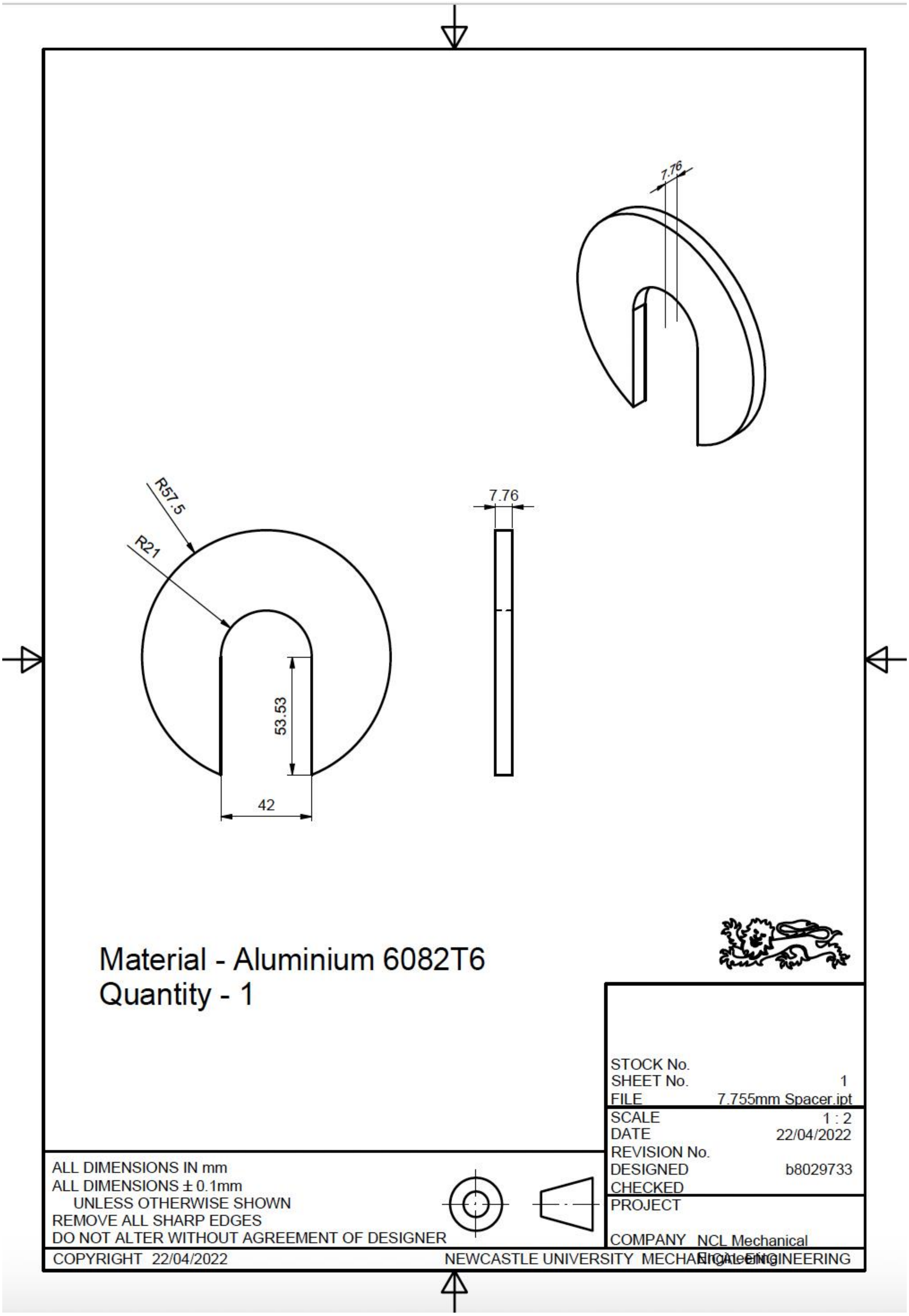
Appendix B-67: (DS)



Appendix B-68: (DS)



Appendix B-69: (DS)

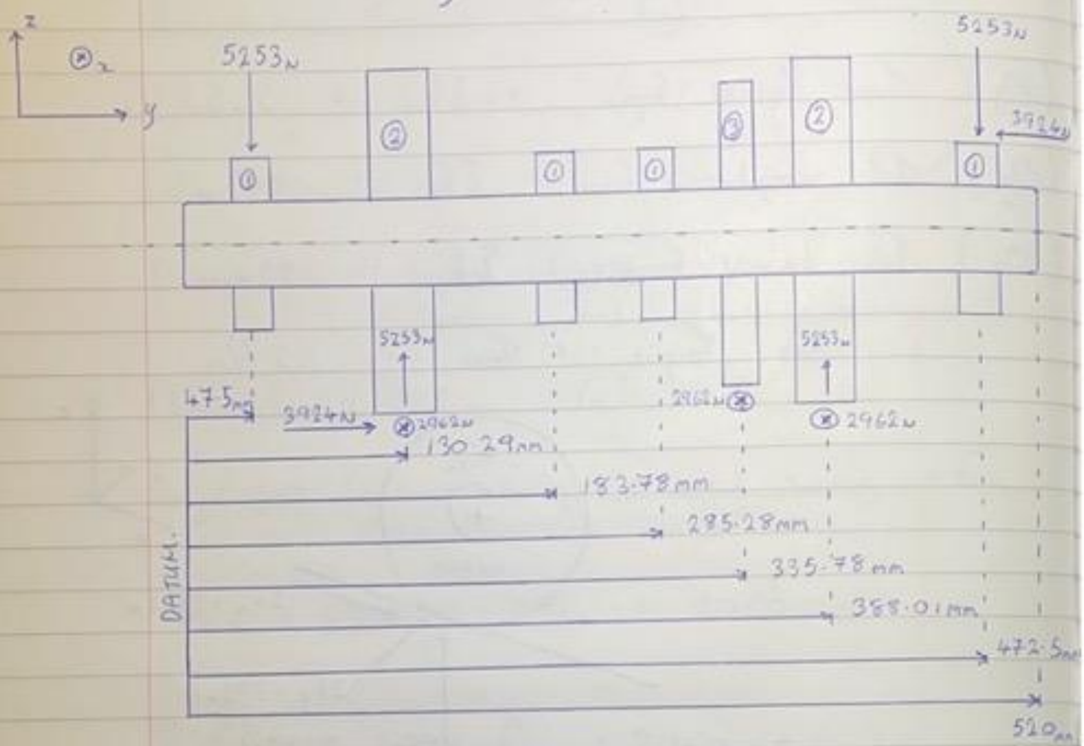


Appendix F: Unsprung Assembly Component Calculations (DS)

Appendix F-1: Shaft Calculations (DS)

12

Shaft Bending Moments & Shear Forces



① = Bearing ② = Wheel ③ = Pulley

Forces:

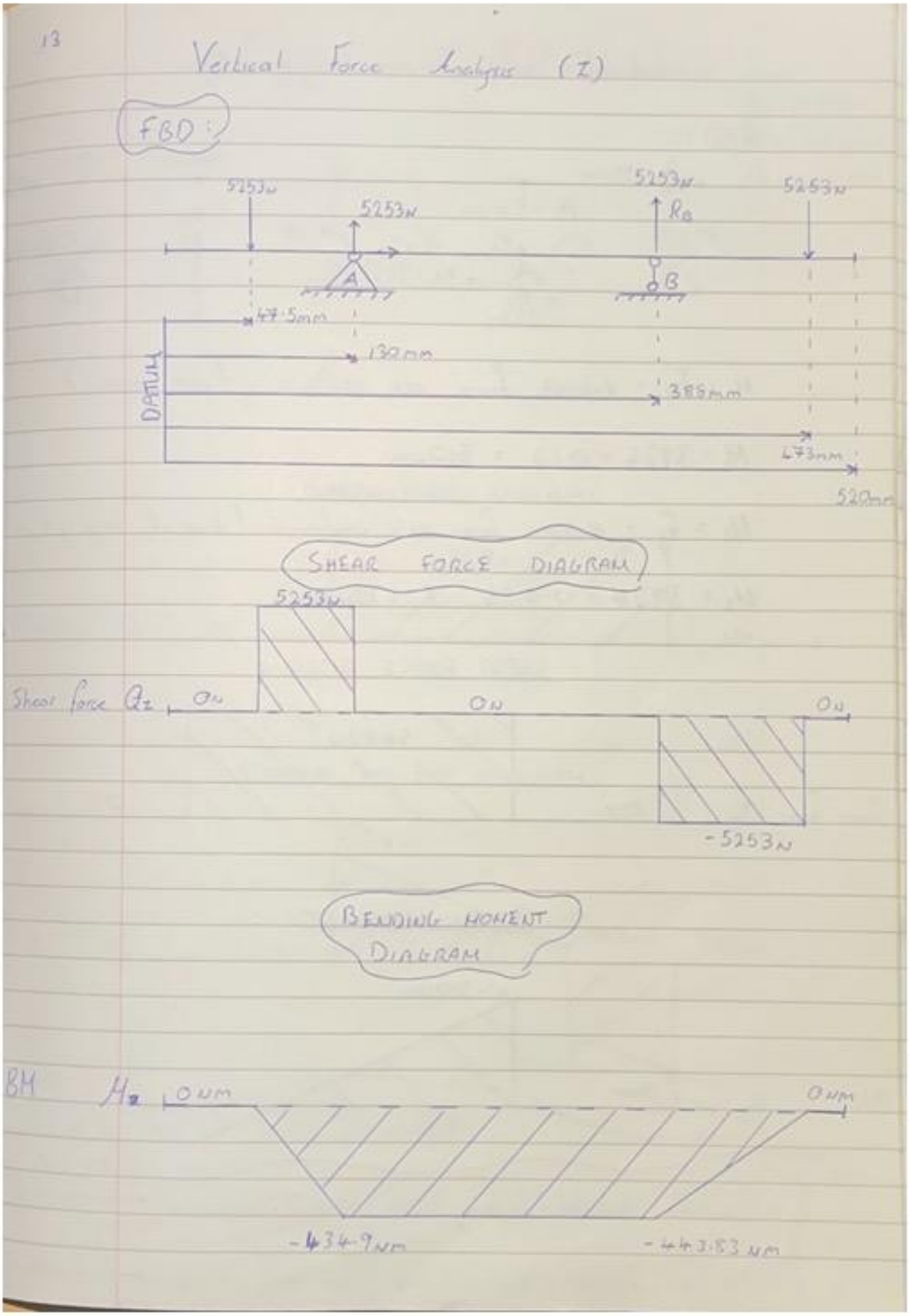
$\sum F_x = 2962N$ $\sum F_y = 3924N$ $\sum F_z = 5253N$

Method

Analyse forces in z direction (vertical forces)

Analyse forces in x direction (longitudinal forces)

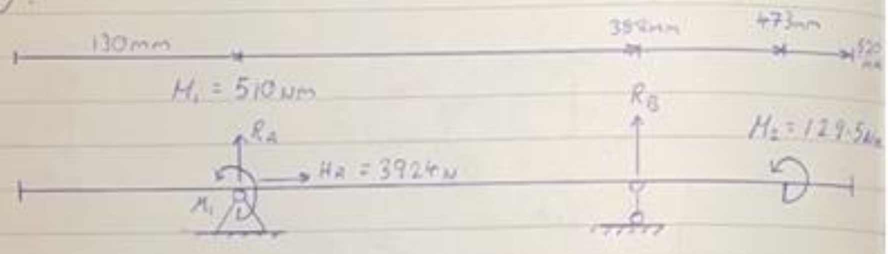
Analyse forces in y direction (lateral forces


13
Vertical Force Analysis (I)
FBD:
5253N
5253N
5253N
R_B
5253N
A
B
47.5mm
132mm
388mm
473mm
520mm
DATUM
SHEAR FORCE DIAGRAM
5253N
Shear force Q_z
0N
0N
0N
-5253N
BENDING MOMENT DIAGRAM
BM M_z
0NM
0NM
-434.9NM
-443.83 NM

14

Lateral Shaft Force Analysis (y)

FBD:



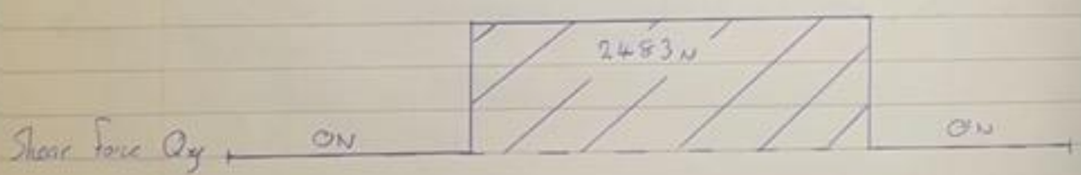
$M_1 = F_y \cdot$ distance from axle centreline (wheel radius)

$M_1 = 3924 \cdot 0.13 = 510$ Nm

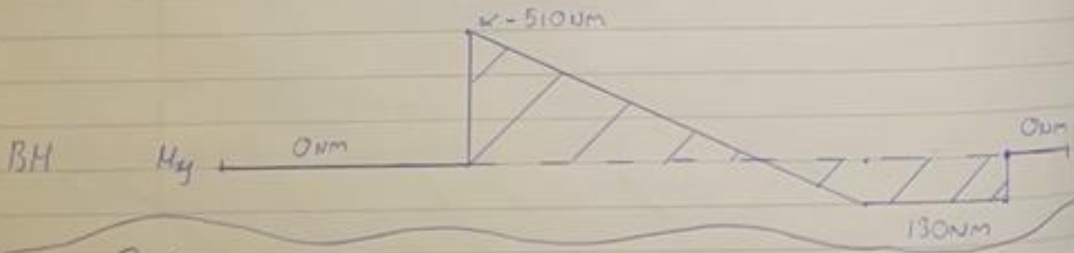
$M_2 = f_y \cdot$ distance from axle centreline (top of bearing)

$M_2 = 3924 \times 0.033 \approx 130$ Nm

SHEAR FORCE DIAGRAM



BENDING MOMENT



Due to vertical loading acting between wheel & bearing

Due to lateral loading acting on the wheels.

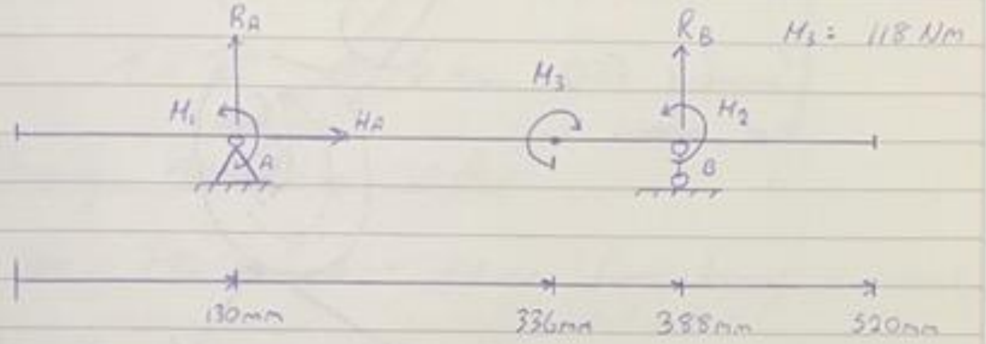
15

Torques on Shaft + Longitudinal forces

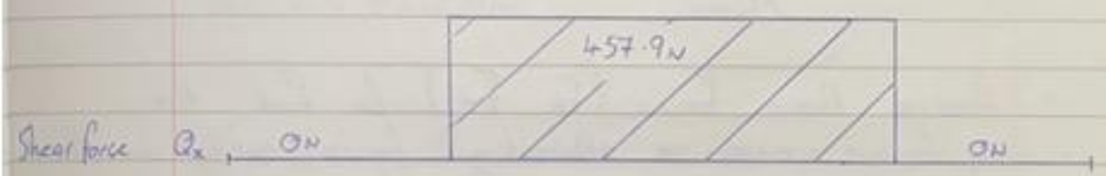
$M_1 = 118\,Nm$

$M_2 = 118\,Nm$

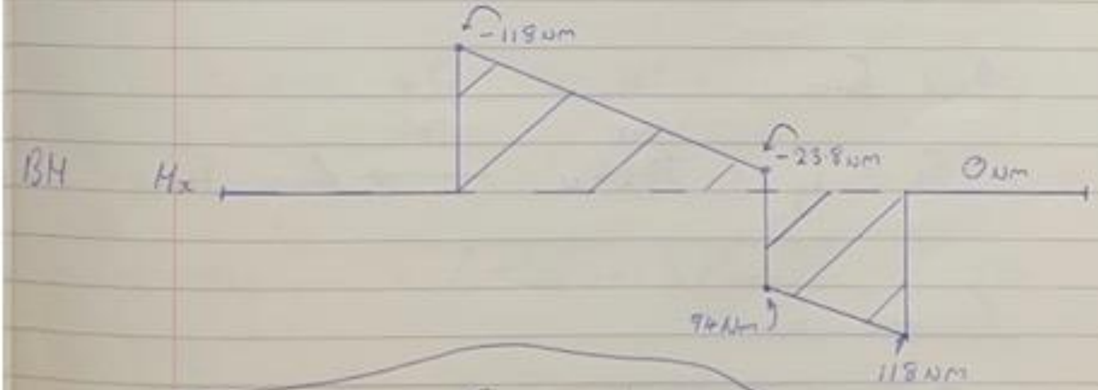
$M_3 = 118\,Nm$



SHEAR FORCE DIAGRAM



BENDING MOMENT DIAGRAM



SUMMARY

τ_{max} : MAXIMUM SHEAR FORCE = 5253N

$\sigma_{b,max}$: MAXIMUM BENDING MOMENT = 510Nm

21

Allowable Stress $\sigma_{Allowable}$:

$\sigma_{Allowable} = UTS \cdot k_1 \cdot k_2 \cdot k_3$

$= 660 \times 0.5 \times 0.75 \times 0.83$

Allowable σ = 205.4 MPa.

Factor of Safety (FOS) = $\frac{\text{Allowable Stress}}{\text{Maximum Stress}}$

$FOS = \frac{660 \div 205.4 \text{ MPa}}{8.9 \div 109 \text{ MPa}}$

Safety Factor = 1.88 ⟶ Allowable σ

= 74 ⟶ Mean σ

Summary

Current shaft geometry has a Safety factor of 1.88

Appendix F-2: Bearing Calculations (DS)

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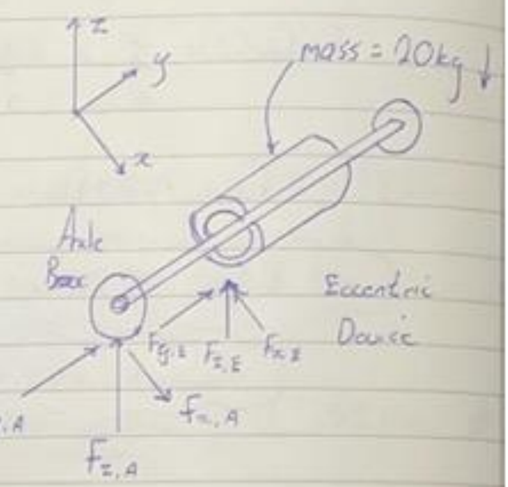
Bearing Calculations. Eccentric Housing?

mass of motor, mounts & brakes. :

- Breaks [0.865 kg]
- Mount [0.548 kg]
- Motor [10.9 kg]
- Eccentric [2.4 kg]
- Mount (motor) [4 kg]



$\Sigma \approx 19\ kg$

We will use 20kg for calculations.

Eccentric Bearing Forces

$$F_{z,e} = 20kg \times 9.81 \times 5g = 981N$$

$$F_{yc} = 20kg \times 9.81 \times 10g = 1962N$$

$$F_{x,e} = 20g \times 9.81 \times 10g = 1962N + 722N = 2684N$$

$$\text{Transverse force} = \sqrt{F_{xc}^2 + F_{zc}^2}$$

$$F_{Tr} = \sqrt{2684^2 + 1962^2}$$

Transverse force on eccentric housing bearings = 3324.7

$$\boxed{\text{Transverse force} = 3324.7\ N}$$

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Bearing Forces Axle Box

$F_{xA} = 2962\,N$

$F_{yA} = 3924\,N$

$F_{zA} = 5253\,N$

Transverse Force $= \sqrt{2962^2 + 5253^2}$

$F_{TA} = 6030.5\,N.$

Safety Factor of Eccentric Bearings

Static load rating $\geq$ safety factor $\times$ Equivalent Load

$$C_o \geq S_o P_o$$

Rearranging:

$$S_o = \frac{C_o}{P_o}$$

$C_o = 21.6\,kN$

$P_o = 3325\,N$

$$S_o = \frac{21600}{3325} = 6.5.$$

N

Safety Factor of Axle Box Bearings

$$S_o = \frac{21600}{6031} = 3.58$$

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Bearing Life

Formula provided by SKF[1]

basic life rating (90%) = $\frac{\text{Basic Dynamic Load Rating}}{\text{Equivalent dynamic bearing load}}$

$$L_{10} = \left(\frac{C}{P}\right)^3 \qquad C = 33.2\,kN$$

The equivalent dynamic bearing load is calculated as follows:

$$P = F_{Transverse} + Y_1 F_{Axial}$$

$$\left.\begin{aligned} F_T &= 6031\,N \\ F_A &= 2962\,N \\ Y_1 &= 2.2 \end{aligned}\right\} \text{Axle Box Bearings}$$

$$\left.\begin{aligned} F_T &= 3325\,N \\ F_A &= 981\,N \\ Y_1 &= 2.2 \end{aligned}\right\} \text{Eccentric Housing Bearings}$$

Axle Box

$P =$

$$P = 6031 + 2.2(2962)$$

$$P = 12547.4\,N$$

$$L_{10} = \left(\frac{33.2 \times 10^3}{12.55 \times 10^3}\right)^3$$

$L_{10} = 18.513$ million revolutions

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Given the wheel $\phi = 250mm$

Distance travelled D

$$D = \text{Circumference} \times \text{millions of revolutions}$$

$$D = 0.250 \times 18.5 \times 10^6$$

$D = 4625\,km$ before failure of 10%

Assuming average speed of 15km/h

$$\text{hours of operation} = \frac{4625km}{15km/h} = 308 \text{ hours}$$

Days of operation = 12.84 days of continuous operation

Eccentric Bearings

$$P = 3325 + 2.2\,(981) = 5483.2\,N$$

$$L_{10} = \left(\frac{33.2}{5.48}\right)^3 = 222.4 \text{ million cycles revolutions}$$

$$\text{Days of operation} = \frac{\pi \times 0.25 \times 222.4 \times 10^{3}\ km}{15\,km/h \times 24 \text{ hours}}$$

Days of continuous operation = 485.2 Days

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Bearing Summary

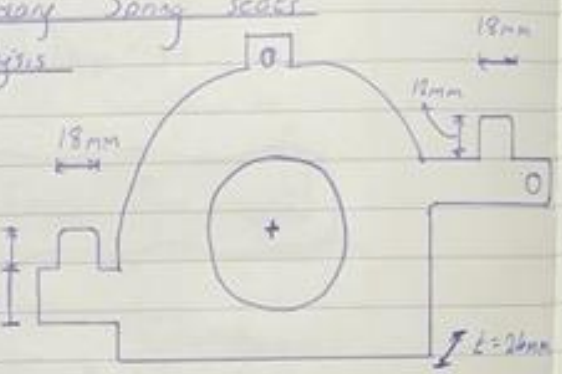
	Safety factor	Distance (km)	Days of continuous use
Eccentric	6.5	174000 km	485 days
Axle Box	3.5	4625 km	12.8 days

Axle Box & Primary Spring seats Stress Analysis

Assuming material yield strength of S275

$\sigma_y = 275\ N/mm^2$

Using hookes Law



Axle Box

$\sigma = F/A$ $\quad F_{shear} = \frac{\sigma_y}{2} \times Area$

Wheel Contact Forces :

$F_x = 5253.2\ N$ $\quad F_T = 6031N$

$F_{gx} = 2962\ N$

$F_y = 3924N$

x dir : $F_{shear} = {}^{275}/_{2} \times 12 \times 18 = 29700\ N$

y dir : $F_{shear} = 29700N$

transverse dir : $F_{shear} = 29700N$

Appendix F-3: Wheel Contact Forces (DS)

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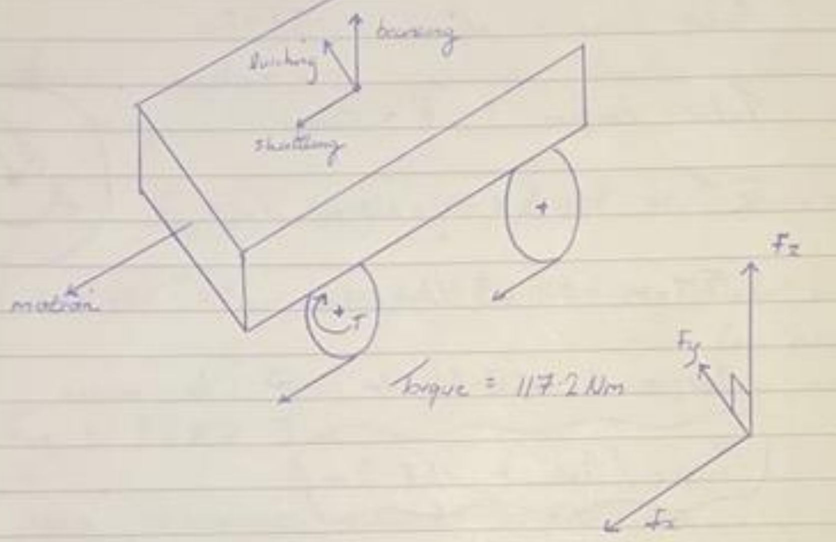
Wheel Contact Forces



<u>Wheelset dynamic loading case</u>

gravity = g = $9.81 ms^{-2}$

(Per Item)

wheelset weight = pulley ×1 [5kg]
+ wheel ×2 [11.206kg]
+ axle ×1 [5kg]
+ eccentric device ×1 [2.32kg]
+ bearing ×4 [0.77kg]
+ 2517 bushing ×1 [1.5kg]
+ ringfeder ×1 [0.340kg]
+ ringfeder ×2 [1.21kg]

☆ wheelset weight = 42.072 kg

Vertical (z) = 20g

Lateral (y) = 10g

Longitudinal (x) = 10g

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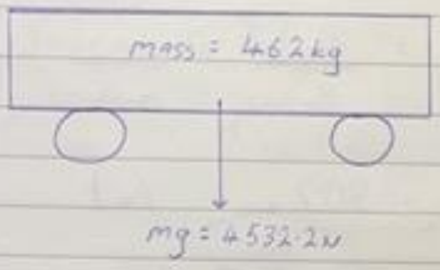
Mass of Loco = 924 kg

Vertical Static Force



(4 wheels)

Vertical Static Force $= \frac{4532.2}{4} = \boxed{1133\ N} = F_{S,Z}$

Vertical Dynamic Force $= \frac{\text{wheelset weight} \times 20g}{2 \text{ (wheels)}}$

$F_{Dynamic,Z} = \frac{M_{wheelset} \times 20g}{2}$

$F_{D,Z} = \frac{42 \times 20 \times 9.81}{2} = 4120.2\ N$

$F_{D,Z} = 4120.2\ N$ (Assuming 20g impact)

$F_{D,Z} = 2060.1\ N$ (Assuming 10g impact)

$\therefore$ Total Vertical Force = Dynamic + Static

$\Sigma F_Z = F_{D,Z} + F_{S,Z} = (1133 + 4120.2)\ N$

$\boxed{\Sigma F_Z = 5253.2\ N = \text{Total Vertical Force}}$

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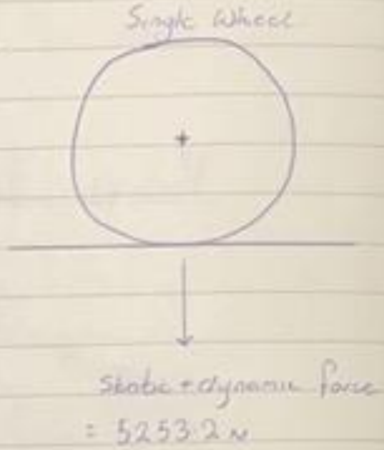


Longitudinal Tractive Force (F_x):

Driving torque = 117.2 N (pg 7)

Wheel Radius = 0.13 m

Driving force = $\frac{\text{Torque}}{\text{Radius}} = \frac{117.2}{0.13}$

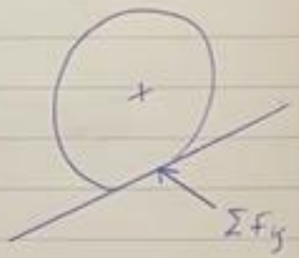
* Driving force$_x$ = 902 N. (x)

* Longitudinal Dynamic force$_x$ = $\frac{M_{wheelset} \times 10g}{\text{no of wheels}} = F_{D,x}$

$F_{D,x} = \frac{42 \times 10 \times 9.81}{2} = 2060.1\ N$

Σ Total Longitudinal force = 2060.1 + 902 N

$\Sigma F_x = 2962.1\ N.$



Lateral force F_y:

Σ Lateral force = $\frac{M_{wheelset} \times 10g}{(\text{no of wheels}) \cdot 1}$

Σ Lateral force = $\Sigma F_y = 42 \times 10 \times 9.81 = 3924\ N.$

Σ Total Lateral force = ΣF_y = 3924 N

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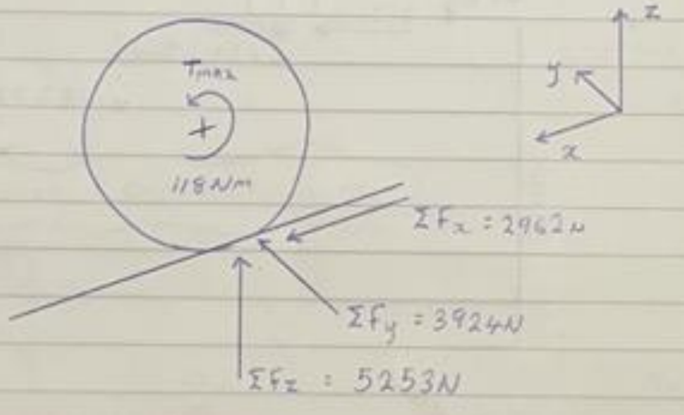
Summary of Wheel Contact Forces

(20g) Total Vertical Force $= \Sigma F_z = 5253.2\,N$

(10g) Total Longitudinal Force $= \Sigma F_x = 2962\,N$

(10g) Total Lateral Force $= \Sigma F_y = 3924\,N$

Max Torque $= T_{max} = 118\,Nm$.



Appendix F-4: Tractive Effort Calculations (DS)

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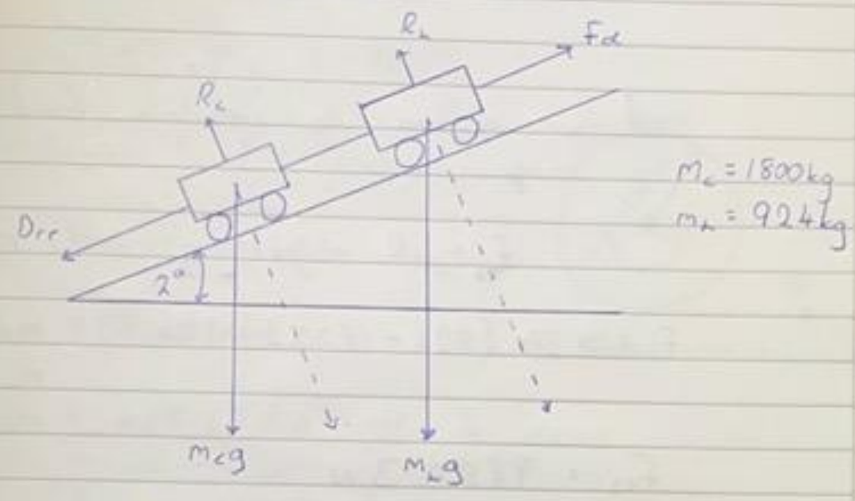
TRACTIVE EFFORT



$\sum F_x : F_D - \underbrace{D_{rc}}_{0} - m_c g \sin\theta - m_L g \sin\theta$

$\sum F_x : F_D = m_c g \sin\theta + m_L g \sin\theta$

$\sum F_x : 1800 \times 9.81 \times \sin(2) + 924 \times 9.81 \times \sin(2)$

$\sum F_x : 932.6\,N = F_g$

(~~1:50~~) 1:84

Required to trail 1800kg up (~~1:84~~) gradient

ASSUMING MAX ACCELERATION = $0.1g = a$
(comfortable riding)

$F_{acc} = (m_c + m_L)\,a$

$F_{acc} = 2724 \times 0.1 \times 9.81 = 2672\,N$

$\therefore$ $\boxed{\text{TRACTIVE EFFORT} = 3605\,N.}$

$TE = \sum F_g + F_{acc} = 2672 + 932.6 = 3604.8\,N$

(can optionally include rolling resistance force = 466N)

Appendix F-5: Motor Mount Linking Arm Calculations (DS)

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Motor Mount Arm length 03/12/21

☆ For Exhibition Park ☆

Arm length = 306mm

with 3° conical angle bushing

☆ For Railway Challenge ☆

Yaw displacement = 6.82mm

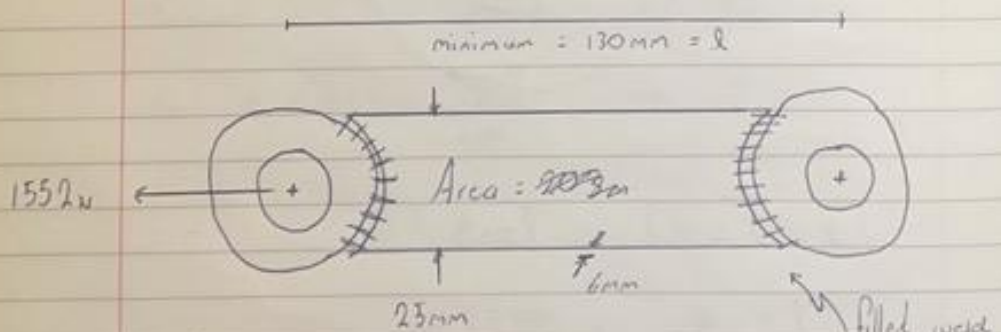
so with 3° connical freedom bushing

Arm length = 130mm !

forces in connecting arm

Force = 1552 N

$\sigma = F/A = 1552/A$ if $\sigma_y = 100\ N/mm^2$ Area of weld = $15.52mm^2$



Area = $1700mm^2$

☆

Appendix F-6: Friction Coupling Safety Factors (DS)

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Friction Coupling Forces & Safety Factors

The bogie utilises THREE locking assemblys and THREE bushes (taper locking)

Locking Assemblys

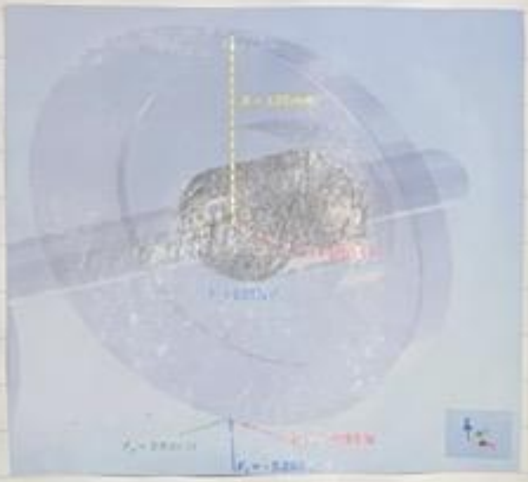
	Component	Location
(1)	OKBS11 - 40×75 Internal Locking Assembly	Wheel → Axle
(2)	RFN - 7012 - 40 × 65 Ringfeder Locking Assembly	Pulley (large) → Axle
(3)	TLK110 20 × 28 Locking Bush	Anti - Roll Bar

Tapered Bushings

	Component	Location
(1)	2517 65mm Taperlocking Bush $D_I = 65mm$ $D_o = 85.73mm$	Fills space between ringfeder & large pulley
(2)	1008 7/8 Taper Bush (Dunlop) $D_I = 22.23mm$ $D_I = 0.875"$ or $7/8"$ $D_o = 1.386"$	Motor shaft to → Small pulley sprocket.
(3)	1108 7/8 Taper Bush (OPTIBELT) $D_I = 7/8"$ $D_o =$	Motor Shaft to → Veebelt brake pulley.

OKBS11 - 40×75 Locking Assembly. 24

Using Forces calculated on page 11. of calculations logbook (4)



$F_T = \sqrt{F_x^2 + F_z^2}$

$F_T = \sqrt{2962^2 + 5253^2}$

$F_T = 6031\,N$

$F_x = 2962\,N \quad F_y = 3924\,N \quad F_z = 3924\,N \quad F_T = 6031\,N$

$T_{max} = 118\,Nm.$ FORCES ACTING ON OKBS11.

Safety Factor of OKS11.

Applied Torque	Max Torque Possible	Safety Factor
118 Nm (172 Nm) pg 33	2880 Nm	24.4

Pressure Safety Factors

Pressure Location	Pressure Exerted	Safety Factor
Hub	150 MPa	$400/150 = 2.7$
Shaft	281 MPa	$400/280 = 1.42$ $600/281 = 2.1$

Safety Factor = yield strength / surface pressure exerted by locking mechanism

25 RfN 7012 Ringfeder 40 × 65 Safety Factor

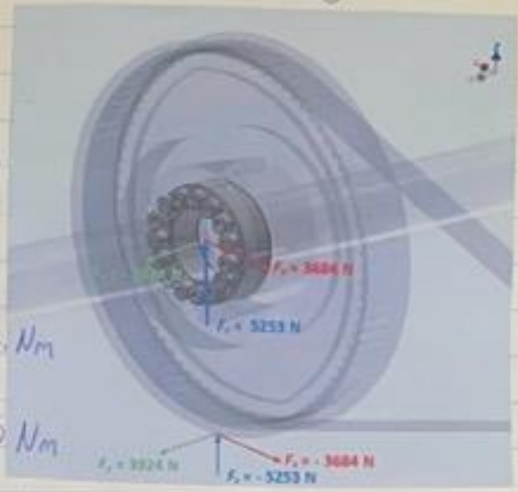
Ringfeder Specification:

Transmissible Torque T = 1154 Nm

Transmissible Axial Force = 58 kN

Max Transmissible Torque = 1404 Nm

Shaft Surface Pressure p_w = 225 Nm



Hub Surface Pressure p_N = 139 N/mm²

Safety Factor of Ringfeder 7012 40 × 65

	Factor	Applied Value	Allowable Value	Safety Factor
T	Torque	118 Nm (38 Nm)	1154 Nm	9.78
P_w	Shaft Pressure	225 N/mm²	600 N/mm²	2.67
P_N	Hub Pressure	139 N/mm²	480 N/mm²	3.45
$F_y = F_A$	Axial Force	3924 N	58000 N	14.78
T_{max}	Max Torque	3684 N × 0.1 m = 368.4 Nm	1404 Nm	3.81

Hub made of cast iron w/ σ_y = 480 MPa

Max applied torque = max force (longitudinal) (x) × 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	19816N	22000 Nm	1.11 (crushing)
Shaft Pressure	160 N/mm²	275 N/mm²	1.72
Hub Pressure	115 N/mm²	275 N/mm²	2.39

Appendix F-7: Maximum Motor Torque (DS)

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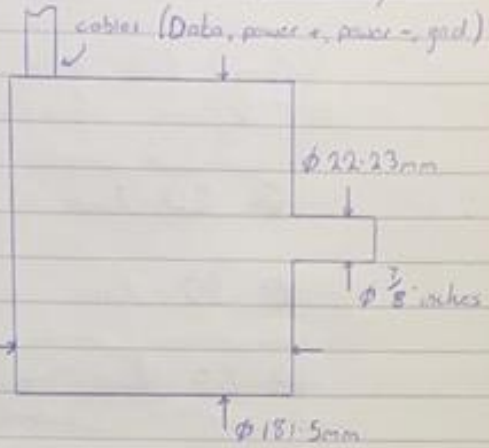
Motenergy 1918 Calculations for shaft torques.



Motor Characteristics PMAC

Mass = 10.9 kg

Continuous Current Rating = 100A

Peak current rating = 300A 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

$m = \frac{(y_2 - y_1)}{(x_2 - x_1)}$ using: (2, 20) ~~(12, 100)~~ (16, 130)

$$m = \frac{(130 - 20)}{(16 - 2)}$$

$$m = 7.86$$

$$y - y_2 = m(x - x_2) + \cancel{c}^{0}$$

$$\cancel{(y - 130 = 7.85(x -)}$$

$$y - 130 = 7.85(x - 16) + \cancel{c}^{0}$$

$$y = 7.85x + 4.4$$

$$x = \frac{y - 4.4}{7.85}$$

$x = 37.656$ Nm of torque / motor

using $y - y_2 = m(x - x_2) + c$

@ 300A Each motor exerts ≈ 38 Nm of Torque

^ MAX MOTOR TORQUE ^

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Using the reduction ratio of 3.63

$\frac{80}{22}$ teeth = 3.63 $(T = F \cdot d)$

MAX Torque exerted on shaft = 3.63 × 38 Nm

@ 300A. Max Torque by Large pulley = 138 Nm

☆ ∴ Max Torque on friction couplings = 138 Nm. ☆

Max Tractive Effort = 3605 N

Torque required to meet max TE = 450.6 Nm

Torque required / axle = 113 Nm ✓ possible

Torque @ motor shaft = 31.12 Nm

setting x = 31.12 Nm in $x = \frac{y - 4.4}{7.85}$

~~x = 3.14~~ $y = 7.85x + 4.4$

$y = 7.85 \times 31.12 + 4.4$

$y = 248.7$ A

so 249 Amps required for max tractive effort of 3605 N excerted by motors (each one)

33 Torque @ Large Pulley + Wheel.

Torque pulley (large) max = 138 Nm

$$\text{Force} = \frac{T}{d} = \frac{138}{0.1} = 1380\,\text{Nm}$$

$F_x = 1380\,\text{Nm}$

Torque @ wheel = 1380×0.125

$T_{wheel} = 172.5\,\text{Nm}$

Appendix F-8: Axle Box Calculations (DS)

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Safety factors of Axle Box

x direction :

$S_f = \dfrac{\text{Permissable Shear force}}{\text{Applied force}}$

★ $S_{f,x} = \dfrac{29700}{2962} = 10.0$

★ $S_{f,y} = \dfrac{29700}{3924} = 7.57$

★ $S_{f,T} = \dfrac{29700}{6031} = 4.92$

Safety factors of 2ndary suspension seats.

$F_{shear} = \dfrac{\sigma_y}{2} \times A$

$F_{shear} = \dfrac{275}{2} \times 60 \times 66.5 = 548.6\ kN$ (unrealistic)

$F_{shear} = 275/2 \times 16 \times 25 = 55kN$

$S_{fx} = \left[\dfrac{55000}{29700}\right]; \dfrac{55000}{2962} = 18.6$

★ $S_{fy} = \dfrac{55000}{202 \times 9.81 \times 10} = 2.78$

★ $S_{fx} = \dfrac{55000}{202 \times 9.81 \times 10} = 2.78$

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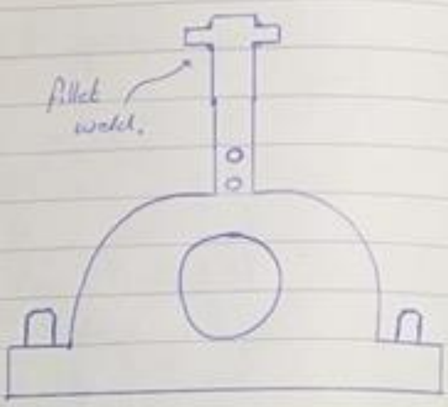
Weld Calculations for Axle Box.

Each T bar should support the weight of ½ the wheelset



Wheelset mass = 70kg

Force = 70 × 9.81 = 686.7 N

Assuming σ_y of weld = 100 N/mm²

$\sigma = \frac{F}{A}$ ∴ $A = \frac{F}{\sigma}$

Area = $\frac{686.7}{100}$ = 6.867 mm²

Area of weld required = 7mm²

Appendix F-9: Primary Spring Calculations (DS)

+5

Primary Suspension FoS

Primary Spring Characteristics

D_o = 30.48mm
Part number = LHL 1250A 06
Free length = 88.9mm
Rate (N/mm) = 24.519 N/mm
Δx = 45.97mm of compression allowed
Solid height = 42.93mm
Lateral Stiffness = 6.1 N/mm (From FEA)
Force to fully compress using Hookes: $F = k\Delta x$

$F = 24.519 \times 45.97$

Force = 1128 N

Force exerted on each spring:

$$\text{Force} = \frac{\text{body mass} \times 2g + 2(\text{bogie mass} \times 5g)}{16}$$

$$\text{Force} = \frac{(500 \times 2 \times 9.81) + 2(202 \times 5 \times 9.81)}{16}$$

Force = 1851.6 N (Force = 554 without acceleration

Safety factor of primary suspension = 0.609

Compression Δx = 75.5mm (bottomed out)

$$\text{FoS} = \frac{1128}{1851.6} = 0.609. \quad \text{FoS} = \left[\frac{554}{1128}\right]^{-1} = 2.04$$

Δx = 22.59mm compression

Appendix F-10: Motor Mount Connection Bolt (DS)

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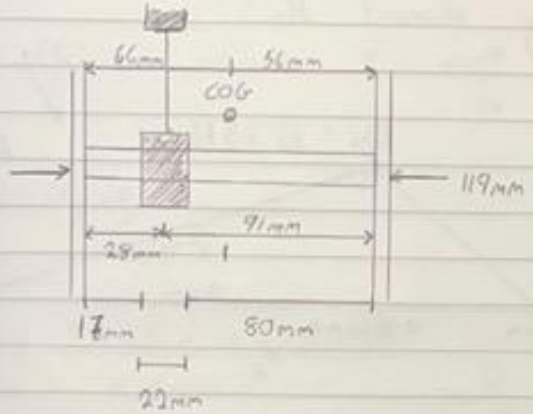
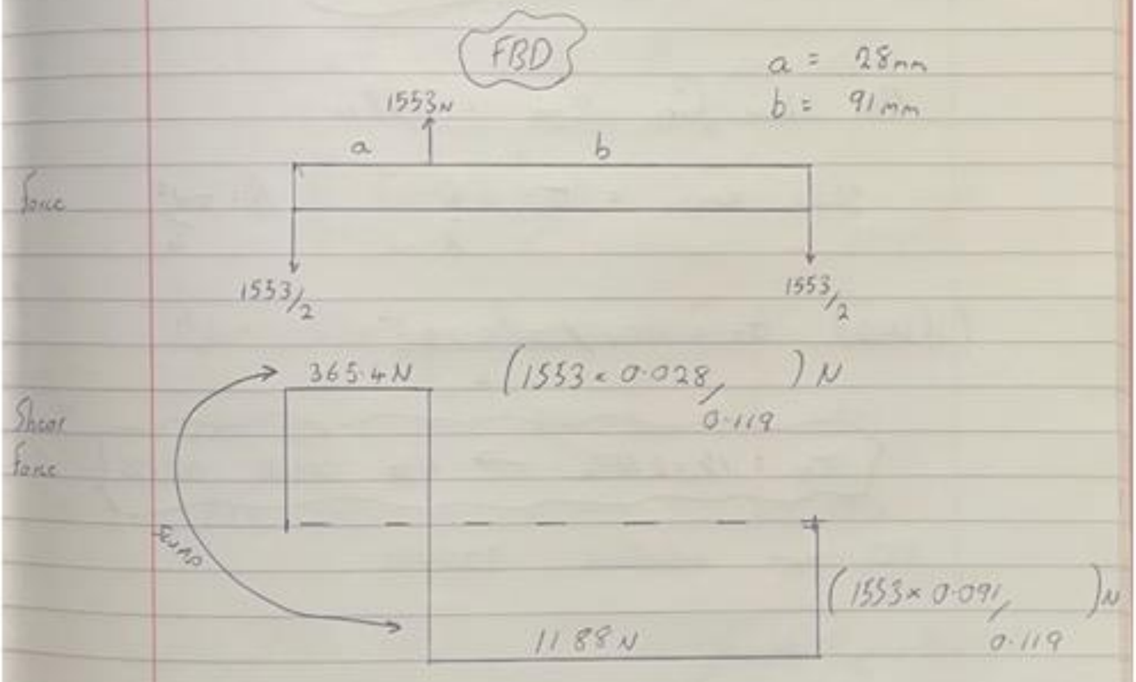
3 Point Bending of Motor Arm Link



We can neglect the effect of offcentered COG as distance is small



1553 = ω $\quad \frac{w \times a \times b}{L}$ = bending moment

For 3 point bending

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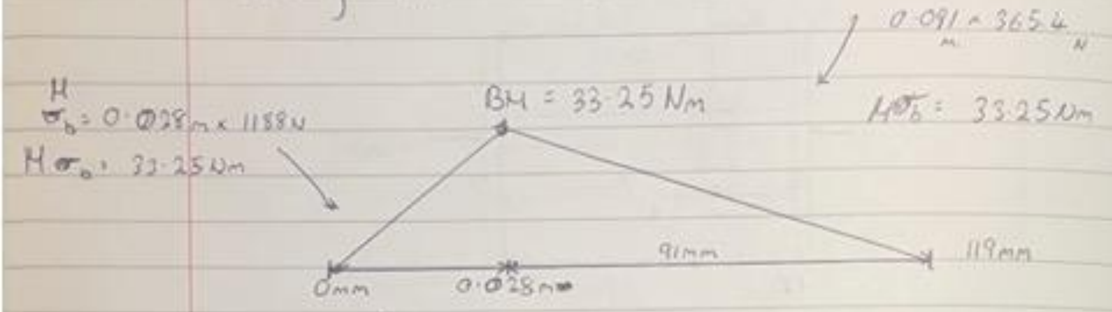
Bending Moment M_b:



Shear

BM = force × distance

BM = 33.25 Nm

Max Shear force τ_{max} = 1188 N

$\therefore$ Shear stress $= \sigma_\tau = \frac{F}{A}$ $A = \frac{\pi d^2}{4}$

(if M10) $\sigma_\tau = 1188 \Big/ \frac{\pi (0.010)^2}{4}$

$\boxed{\sigma_\tau = 15.12\ MPa}$ → MAX SHEAR STRESS

M10 can withstand 300MPa

Bending Stress:

$\sigma_B = \frac{My}{I}$ with ~~I =~~

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$\sigma_b = \frac{My}{I}$ where $I = \pi d^4 / 64$

$\sigma_b = My\,64 / \pi(d^4)$

$\sigma_b = \frac{33.25 \times 0.005 \times 64}{\pi (0.01)^4}$

$\sigma_B = 338.68$ MPa

Withstandable stress = 600MPa

$\therefore$ M10 is suitable for nose mount

SUMMARY

	Actual Stress	Allowable Stress	Check
τ	15.12 MPa	300 MPa	PASS
σ_B	338.68 MPa	600MPa	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

$H_1 =$
$H_2 =$
$H_3 =$
$H_4 = 594mm$

$W_1 = 425mm$
$W_2 = 484mm$

Secondary suspension $K_V = 41.2 N/mm$, $K_L = 25.7 N/mm$
Primary suspension $K_V = 24.5 N/mm$, $K_L = 6.1 N/mm$

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

$16 \times 6.1 N/mm \rightarrow 97.6 N/mm$

$1000 - (4 \times 45 kg) = 820kg$

$594 - 125 = 469mm$

$2g = 13988N \rightarrow 143mm \leftarrow$ use lim4

curving $15 kmph = 4.17 m/s$ $R = 20m$

$F = \frac{Mv^2}{R} = 713 \rightarrow \underline{7.3mm}$

$\boxed{\text{adjust to allow } \pm 10mm}$

Secondary lateral stiffness

$8 \times 25.7 N/mm = 205.6 N/mm$

$600kg$

$594 - 206 = 388$

$2g = 11772N \rightarrow 57.3mm$

$15 kmph = 4.17 m/s$ $R = 20m$

$F = \frac{Mv^2}{R} = 522N \rightarrow \underline{2.5mm}$

$\boxed{\text{limit arms } \pm 30mm \text{ adjust to } \pm 20mm}$

side of springs $\approx 11mm$ at $\frac{1}{2}$ height

$\approx 22mm$ at top

Roll

Primary roll stiffness Nm/deg

per side $8 \times 24.5\,N/mm$ @ 212.5mm

1° requires $212.5 \sin 1° = 3.71mm$

$727N \times 0.215m = 154.5Nm$

309 Nm/deg

H of CoG $594 - 125 = 469mm$

Will tip over 2318N @ CoG 594mm

2318N at 594mm = 1377Nm

$\frac{1377}{309} = 4.46° \rightarrow 77.2mm$

$993 \times \sin 4.46 =$

$F = \frac{MV^2}{R} = 750kg\ \frac{4.17^2}{20} = 652N$

$652 \times 0.594 = 387Nm$

$\frac{387}{309} = 1.25°$ $993 \times \sin 1.25° = 21.7mm$

Secondary roll stiffness Nm/deg

Per side $4 \times 41.2\,N/mm$ @ 242mm

1° : $242 \sin 1^\circ = 4.22mm$

1° force $4 \times 41.2 \times 4.22 = 696N$

moment $696 \times 0.242 = 168.4\,Nm$

337 Nm/deg

Torsional stiffness in anti roll bar

$$\frac{\text{Torque}}{\text{Angle}} = \frac{Nm}{\text{radius}} = \frac{T}{\theta}$$

$$\frac{T}{\theta} = \frac{GJ}{L}$$

G - shear modulus
J - polar moment inertia
L - length of torsion bar
θ - angle of rotation (rad)
T - torque applied

$G - 7.2 \times 10^{10}\,Pa$ steel 275
$J = 1.194 \times 10^{-8}\,m^4$
$L = 0.4\,m$

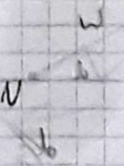
$$\frac{T}{\theta} = \frac{7.2 \times 10^{10} \times 1.194 \times 10^{-8}}{0.4} = 2149.2\,Nm/rad$$

hollow bar stiffness 2149.2 Nm/rad

solid bar 2827 Nm/rad

arm length 120mm

1 radian movement 120mm @ 2827N



$$\frac{2827}{0.12} = 23.6\,N/mm$$

400mm space

1 deg → $200 \sin 1 = 3.49mm$

$3.49mm \times 23.6 = 82.4\,N$

$82.4 \times 0.2 = 16.5\,Nm$

16.5 Nm/deg/bogie

33 Nm/deg.

Change L to 60mm

$\frac{2827}{0.06} = 47.2$ N/mm

3.49 mm × 47.2 N/mm = 164.5 N

164.5 × 0.2 = 32.9 Nm/deg/bogie

65.8 Nm/deg

65.8 + 337 = 402.8 Nm/deg

$\frac{600 \times 4.17^2}{20} = 521.7$ N   544 − 206 = 388 mm

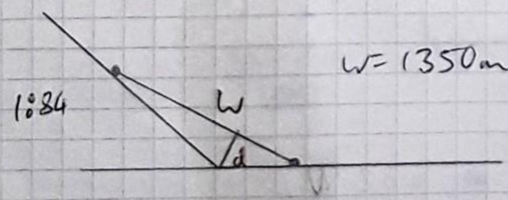
202.4 Nm $\frac{202.4}{402.8} = 0.50°$

993 − 206 = 787 mm

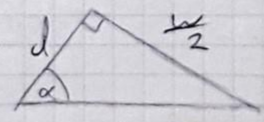
787 sin 0.5 = 6.9 mm

lateral displacement

Vertical curve



W = 1350 mm

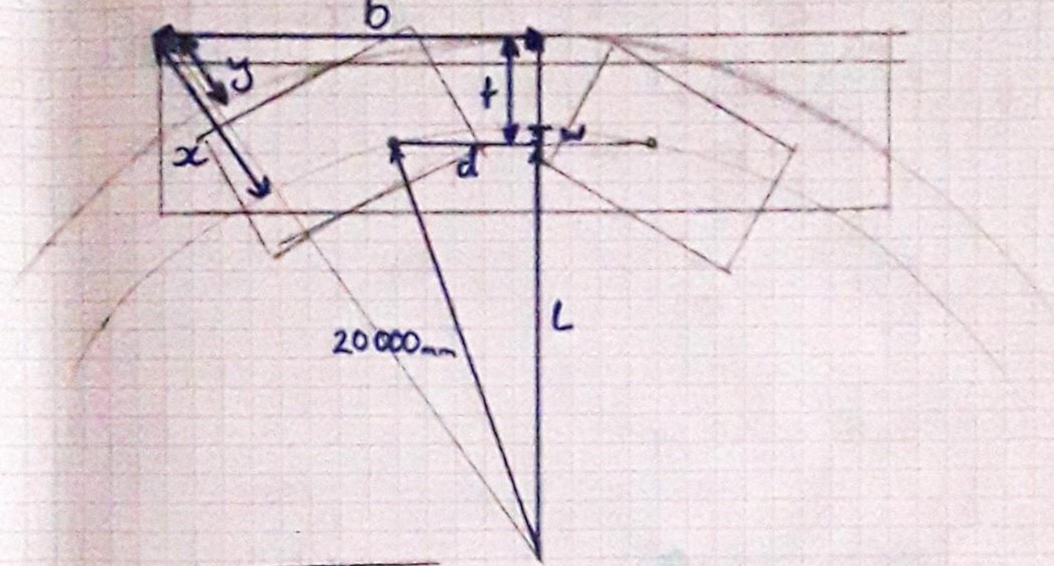


$\alpha = \frac{1}{84}$

$\frac{1}{84} \times \frac{1350}{2} = 8.04$ mm

vertical movement upwards

Horizontal curve



$L = \sqrt{20000^2 - d^2}$ $w = 20000 - L$

$(20000 + x) = \sqrt{(L+t)^2 + b^2}$

$y = x - t$

$d = 675mm$

optimum bogie distance, $w = y$

$L = 19988.6mm$ $w = 11.39mm$

$t = \frac{562.2}{2} = 281.1mm$

body at full extent

$b = 1350mm$ $y = 33.5mm$ ← overhang

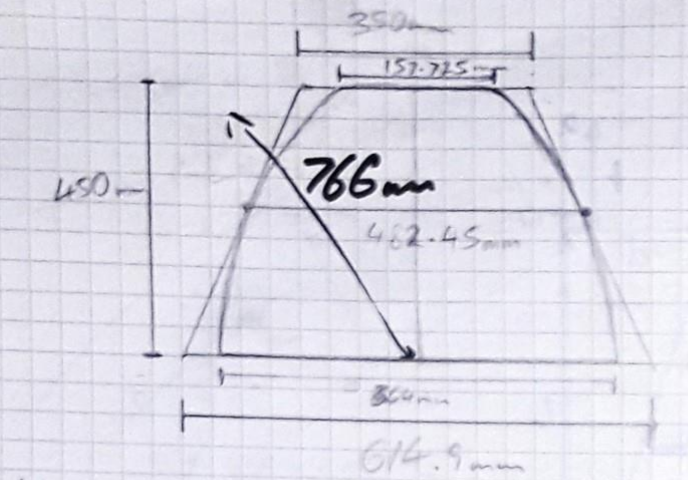
$x = 314.6mm$

body at max width (bolt heads on chassis)

$b = 1310mm$ $y = 30.9mm$

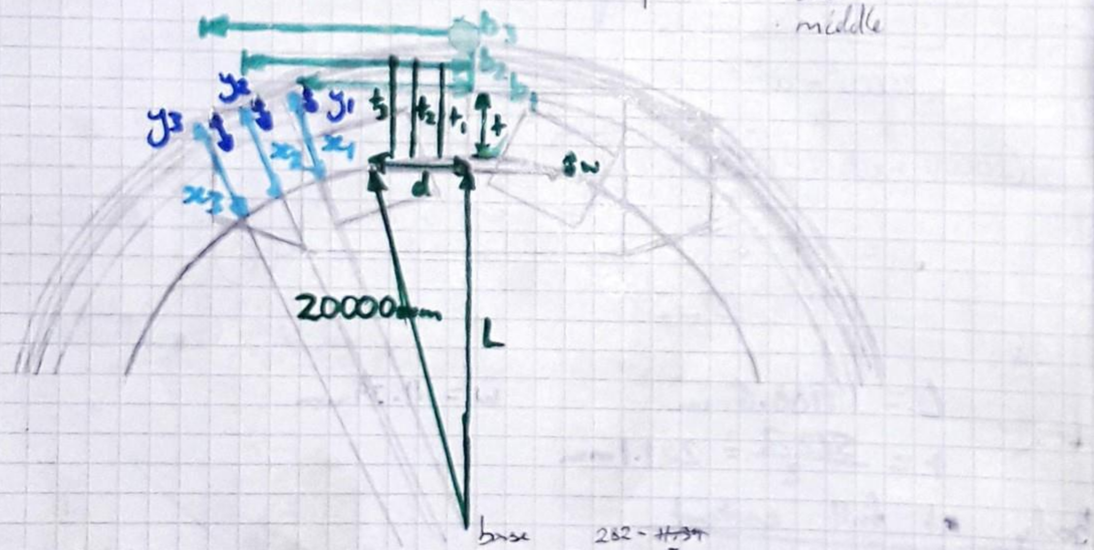
$x = 312mm$

x4
0.8mm



Checks : horizontal curve

Take at 3 points : base
· end
· middle



$L = 19983.6$ mm
$d = 675$ mm
$t = 281.1$ mm
$w = 11.39$ mm

base: $t_1 = 282 - \cancel{11.39} = 282$ mm
middle: $t_2 = 241.225 - 11.39 = 229.835$
end: $t_3 = 78.8625 - 11.39 = 67.4725$

$b_1 = 1350$ mm
$b_2 = 1575$ mm
$b_3 = 1800$ mm

$x_1 = 315.5$ mm
$x_2 = 291.04$ mm
$x_3 = 148.03$ mm

$y_1 = \cancel{33.5}$ 44.89 mm
$y_2 = 61.21$ mm
$y_3 = 80.56$ mm

Should be same as body full extent overhang

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 mm

Distance outside dynamic loading gauge 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 seem dependent 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
>156mm
min width >148mm

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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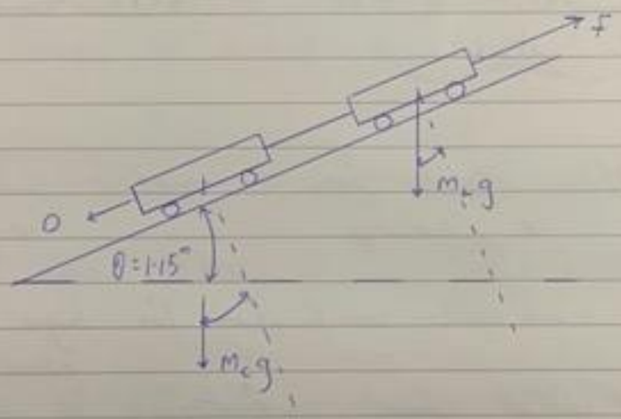
Energy Requirement for Challenge 2

The Locomotive shall be able to haul & start a trailing load of up to 1800 kg 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. ✓



$m_L = 1000\ kg$
$m_c = 1800\ kg$

For constant speed $\Sigma F_x = 0$

$$\text{Force} = m_c g \sin\theta + m_L g \sin\theta$$

$$\text{Force} = (m_c + m_L) g \sin\theta = (2800) \times 9.81 \times \sin(1.15)$$

$$\text{Force} = 551.28\ N.$$

551.28 N is required to pull 1800 kg up 2% gradient

230

Converting 551.28 N into Amps from the 12V batterys.

Force × Velocity = Power = Fv

Current × Voltage = Power = IV

$Fv = IV$ Assuming velocity = 15 km/h

15km/h $\therefore I = \frac{Fv}{V} = \frac{551.28 \times 4.17}{12 \times 4 \times 2} = 23.95$ Amps

15km/h Current Pulled = $\frac{47.89}{2}$ Amps MAXIMUM

5km/h $I = \frac{F \times V}{V} = \frac{551.28 \times 1.39}{12 \times 4 \times 2} = 7.98$ Amps

5km/h $I = \frac{15.96}{2}$ Amps REALISTIC

Velocity km/h	Velocity m/s	Current Amps
5	1.39	7.98 Amps
~~5~~	~~[illegible]~~	
15		23.95 Amps

Both values are well below 100 Amp continuous rating

231

Assuming operation trailing 1800 kg @ 5km/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

$y = x(7.85) + 4.4$ Amps = 140 Amps

Converting maximum tractive effort to current

Assuming 60 Volts input

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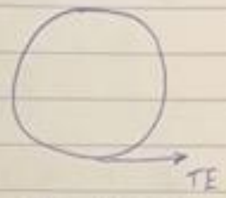
Converting Max Tractive Effort To Current

$F_{TE} = 3680_N$



F_{TE} / Axle = 3680/4 = 920 Nm

Max torque = 920 × 0.125 = 115 Nm

TE: 3680N

Using reduction ratio

Max torque @ motor before slip = $115 \times \frac{22}{80}$

Max motor torque = 31.625 Nm before slip

$$\text{Force} \times \left[\frac{\text{Distance}}{\text{time}}\right]_{\text{(Velocity)}} = \text{current} \times \text{voltage}$$

Current: $I = \frac{Fv}{V} = \frac{3680 \times 6.16}{8 \times 12}$

Power = force × velocity = 3680 × 4.16 = 15308.8 W

Locomotive Power = 15309 Watts

Power / Axle = 3827 Watts

∴ Max current @ tractive effort $= I = \frac{P}{V} = \frac{3827}{48}$

$I_{max} = 79.73$ Amps

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Summary of Required Challenges

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 : 11·96 Amps

③ • 1800kg trailing load @ 5km/h up 2% gradient

Current Pulled : 3·99 Amps

To find current : $\frac{\text{Force} \times \text{velocity}}{4 \text{ axles}} = \frac{\text{power}}{4}$

$\left[\frac{\text{power}}{4}\right] / \text{voltage}$

$\therefore$ Current $= \frac{\text{force} \times \text{velocity}}{4 \times \text{voltage}}$

④ • Max Tractive Effort (3680 N) @ 5km/h

Current pulled : 26·6 Amps

⑤ • Max tractive effort (3680 N) @ 15km/h

Current Pulled : 79·86 Amps

Appendix G-3: Battery Requirements (DS)

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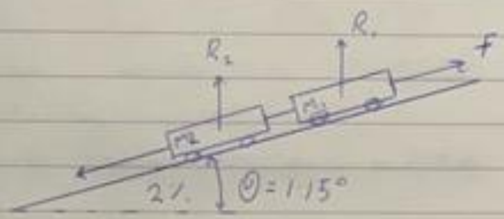
How much energy used?

How much battery energy will be used when:

☆ Operating for 3 hours travelling @ 5km/h on a 2% gradient trailing 400kg of cargo.



m_1 = Assume 1000kg
m_2 = 400 kg

$$\text{Force} = (1000 \sin(\overset{1.15}{2}) \times 9.81) + (400 \sin(\overset{1.15}{2}) \times 9.81)$$

$$\text{Force} = 196.9_N + 78.75_N$$

$$\text{Force} = 275.65\ N$$

☆ Force to maintain constant speed once @ 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 48 VOLTS

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Energy Consumed

Force = 275.65 N

Power = force × velocity

$P = F \times v = 275.65 \times 5\,km/h$

5000 m in 60 mins equates to...

83.3 m in 1 minute

1.388 m in 1 second

$\therefore\ 5\,km/h = 1.39\,m/s$

Power = 275.65 × 1.39

Power = 383.15 watts
= 0.514 horsepower

Energy = Power × time

$E = 383.15 \times 3 \times 60 \times 60$

$E_n = 4.138 \times 10^6$ Joules

Energy during 3 hour travel @ 5 km/h = 4.14 MJ

$\frac{4.14 \times 10^6}{8} = 518\,kJ$ per battery required for challenge

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Current Pulled from Battery

$P = F \times v = IV$

{ if $V = 48$ volts per loco ½ , force $= 276$ N, velocity $= 1.39$ m/s }

$$I = \frac{F \times v}{V} = \frac{276 \times 1.39}{48 \times 2} = \cancel{7.9925}\ 4 \text{ Amps @ 5km/h}$$

$= 12$ Amps @ 15km/h

$\boxed{\text{Current Pulled} = 8A}$ wrong.

The batterys installed in the locomotive allow 70 Amps

- 4 batterys in series per locomotive half

5km/h Current Pulled per locomotive half : 4A

correct.

15km/h Current Pulled per locomotive half : 12 Amps

Accelerations

$F = ma$

F = maximum tractive effort

$F = \mu R = 1000 \times 0.375 \times 9.81$

$F = 3680$ N

$$a = \frac{F}{m} = \frac{3680}{(1000 + 400)}$$

$\boxed{a = 2.629 \text{ m/s}}$

★ It will take 0.529 seconds to accelerate from 0 m/s to 1.39 m/s (5km/h)

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Battery Capacity Vs Battery Requirements

Energy = Power × time : $E = Pt$

Power = force × velocity : $P = Fv = IV$

Force = mass × acceleration : $F = ma$

Power = 3680 × 1.39 = 5115.2 Watts
to accelerate from 0 to 5 km/h

Energy = 5115.2 × 0.529 = 2705.94 J

Energy to accelerate to 5 km/h = 2706 J

$E = ITV$

of current batterys : Voltage = 12V
Amp hours : 75Ah

$Q = IT$, charge = 75Ah → Actually 70Ah
charge = 270000 Coulombs

Energy = charge × voltage : $E = QV$

$E = 270000 \times 12$

Energy / battery = 3.24×10^6 Joules

Energy / battery = 3.24 MJ

☆ OR 0.9 kWh ☆

Energy Required	Battery Capacity
4.14 MJ	3.24 × 8 = 25.92 MJ

∴ the locomotive can run for 18.8 hours trailing a 400kg load up a 2% gradient

$\frac{25.92}{4.14} = 6.26$

3 hours × 6.26 = 18.78 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)

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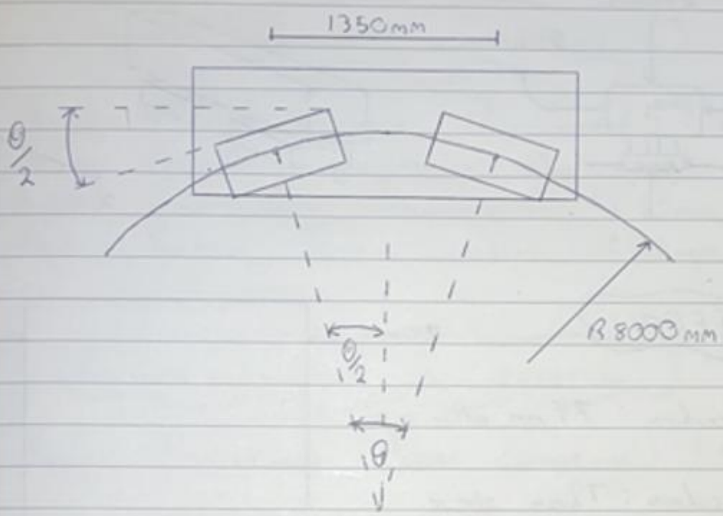
Secondary Suspension lateral displacement



$\theta = \tan^{-1}\left(\frac{1350}{8000}\right) = 9.646°$

$\therefore \theta/2 = 4.823°$

Distance between secondary suspension = 500mm

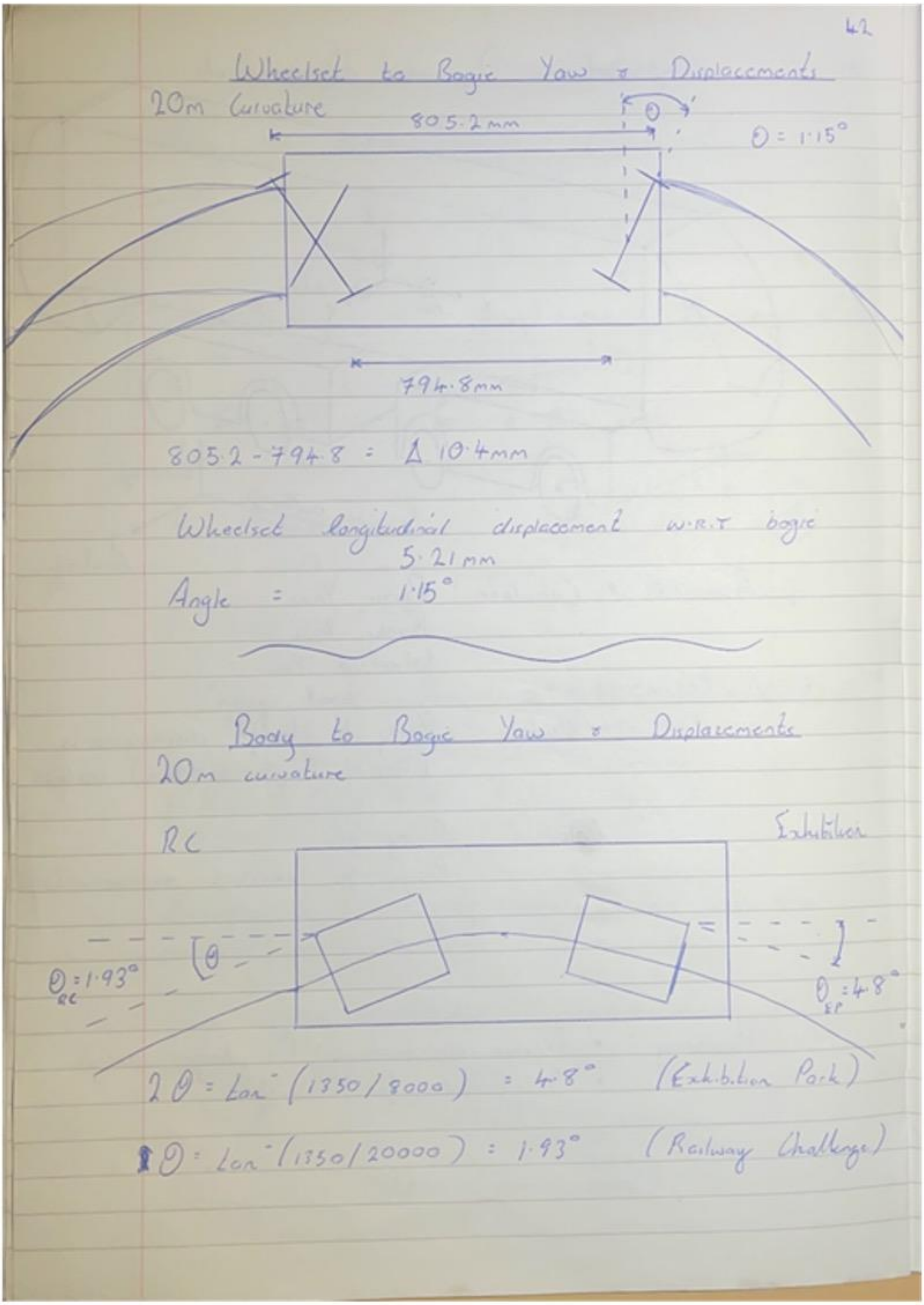
$\therefore$ Secondary suspension lateral displacement =

distance = sin (4.82) × 500/2

distance = 21.006mm @ R8000

distance = 8.423mm @ R20000

Appendix H-2: Wheelset - Bogie Yaw (DS)



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Wheelset to Bogie Yaw & Displacements

20m Curvature

805.2mm

θ

$\theta = 1.15°$

794.8mm

$805.2 - 794.8 = \Delta\ 10.4mm$

Wheelset longitudinal displacement w.r.t bogie

5.21mm

Angle = 1.15°

Body to Bogie Yaw & Displacements

20m curvature

RC

Exhibition

θ

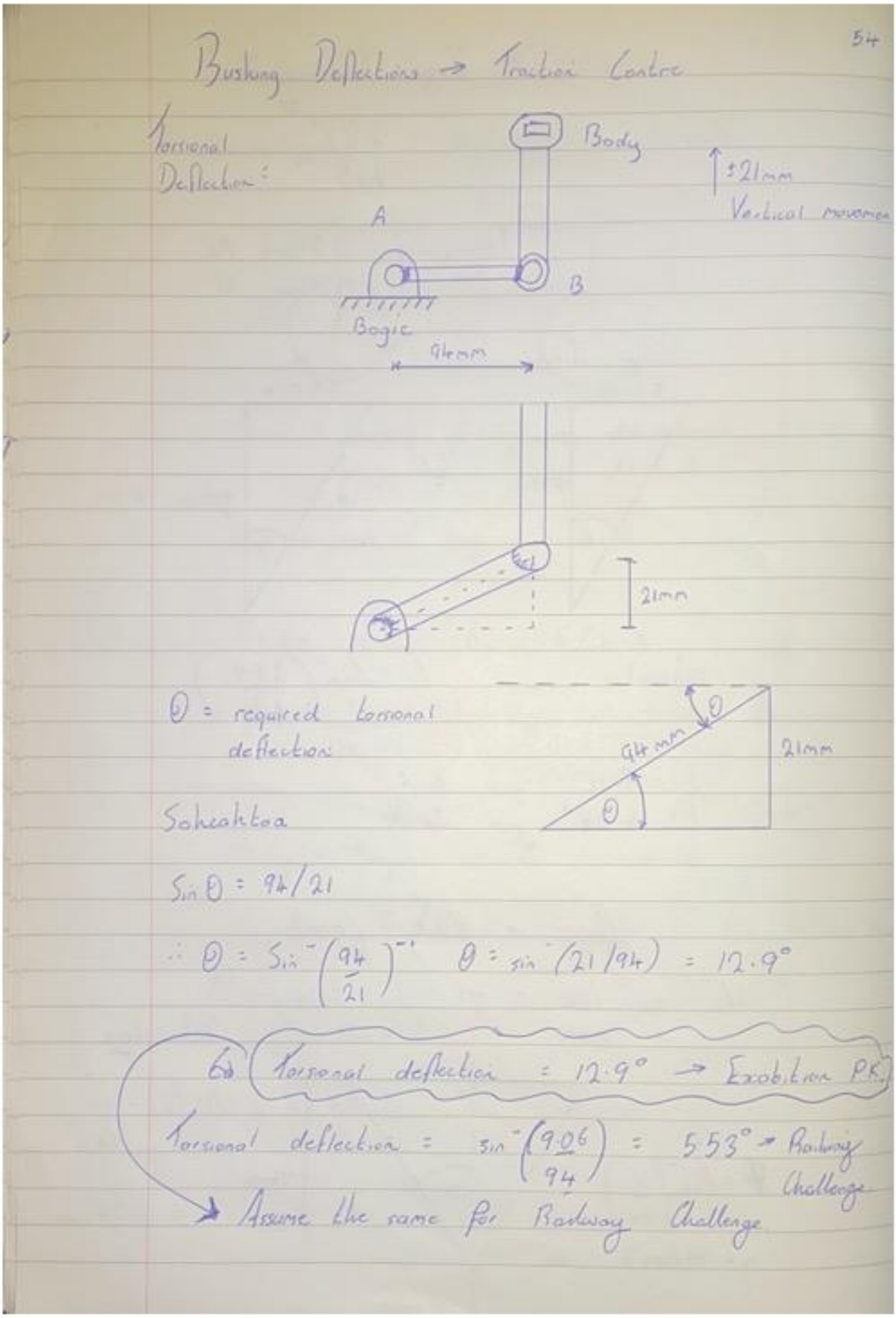
$\theta_{RC} = 1.93°$

$\theta_{EP} = 4.8°$

$2\theta = \tan^{-1}(1350/8000) = 4.8°$ (Exhibition Park)

$\theta = \tan^{-1}(1350/20000) = 1.93°$ (Railway Challenge)

Appendix H-3: Traction Centre Bushing Calculations (DS)



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Bushing Deflections → Traction Centre

Torsional Deflection:

θ = required torsional deflection

Sohcahtoa

$\sin\theta = 94/21$

$\therefore \theta = \sin^{-1}\left(\frac{94}{21}\right)^{-1} \quad \theta = \sin^{-1}(21/94) = 12.9°$

Torsional deflection = 12.9° → Exhibition P.K.

Torsional deflection = $\sin^{-1}\left(\frac{9.06}{94}\right)$ = 5.53° → Railway Challenge

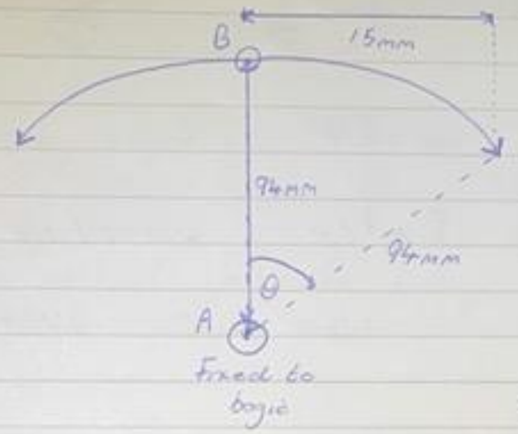
Assume the same for Railway Challenge

55

Traction Centre Bushing Conical Deflection

Secondary Suspension lateral displacement : ± 15mm limited to

Top View :



Sohcahtoa

$\sin\theta = 15/94$

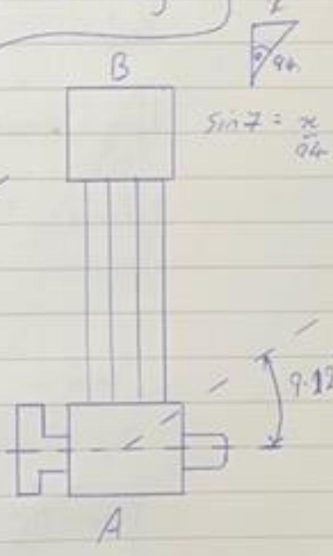
Say $\theta = 7°$
$\Delta x =$

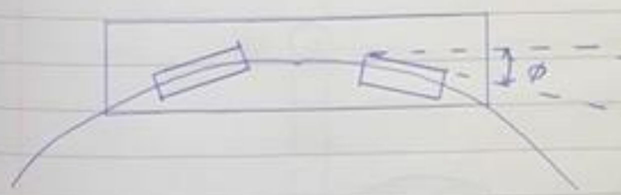
$\therefore \theta = 9.28°$ for bushing fixed to bogie

$\sin 7 = \frac{x}{94}$

Bushing A : Conical deflection of 9.12° minimum

Bushing B conical deflection :
(due to bogie yaw)





$\phi = 4.823$ (page 161)

Bushing B conical deflection of 4.823° minimum

Appendix H-4: Motor Mount Linking Arm Bushing Calculations (DS)

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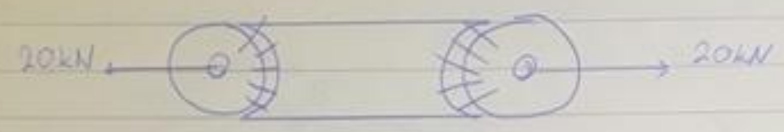
FC Weld Area



$\sigma = F/A$ Assume weld $\sigma = 100\,N/mm^2$

$A = F/\sigma = 200\cancel{00}/1\cancel{0}\cancel{0}$

$\therefore$ Weld area $= 200mm^2$

<u>MMLA Bushing</u>

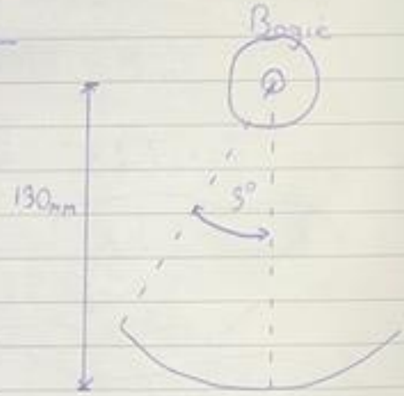
- Allow for:

Vertical = 15mm

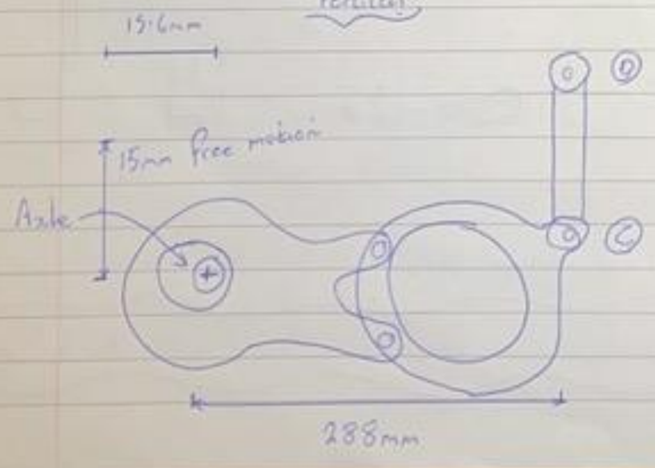
Longitudinal = 15.6mm

Lateral = 6.82mm

or 17.03 for EP.



<u>Vertical</u>



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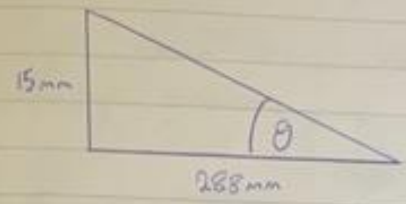

θ = Bushing torsional deflection (bottom bushing)

$\tan\theta = \left(\frac{15}{288}\right)$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 B's conical deflection.

Sohcahtoa

$\sin\theta = \frac{15.6}{124}$

$\theta = 7.23^\circ$

$\theta_0 = 7.23^\circ$ of torsional deflection for bushing D

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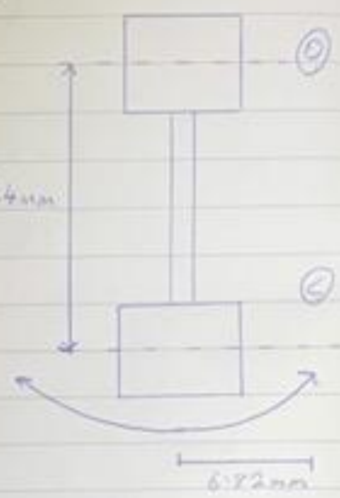
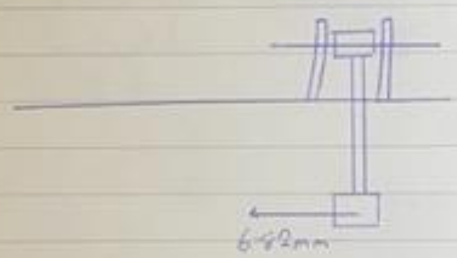
Lateral Displacement

lateral displacement = 6.82mm (17.03 for EP) due to bogie yaw, provided by D's conical deflection



$\tan\theta = 6.82/124$

θ = 3.15 degrees of conical deflection



$\phi = \phi$

Both bushing C & D need a conical ϕ deflection of 3.15°.

7.82° conical deflection for Exhibition Park

Say 6° = ϕ conical deflection

Soh: $\sin(6) = \frac{17.03}{x}$

$\therefore x = 17.03/\sin(6) =$

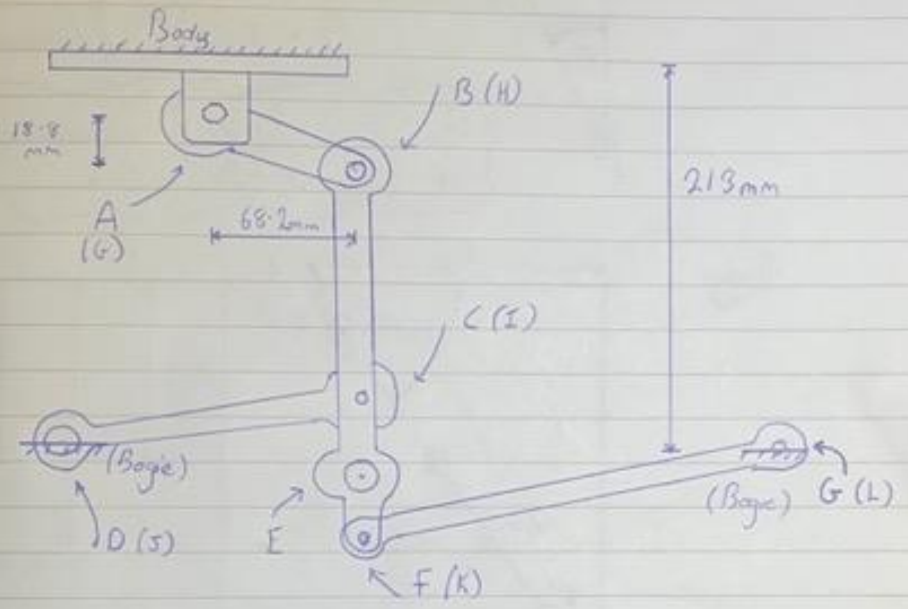
$x = 163$ mm

Appendix H-5: Steering Link Bushing Calculations (DS)

3

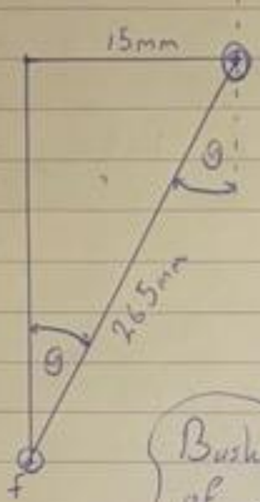
Steering Link Calculations Bushings

Specification :
- Allow for ± 21 mm vertical body displacement (torsional deflection)
- Allow for ± 15mm body lateral displacement (conical deflection)



- Bushing B is most effected by torsional deflection due to 21mm body ~~lateral~~ vertical displacement
- Bushings F&A suffers the highest conical deflection due to 15mm body lateral displacement
- Assume bushing E is unfixed to the bogie (calculates worst possible values integrating a safety factor in the design).

Steering Link Bushing Conical Deflection



Sohcahtoa

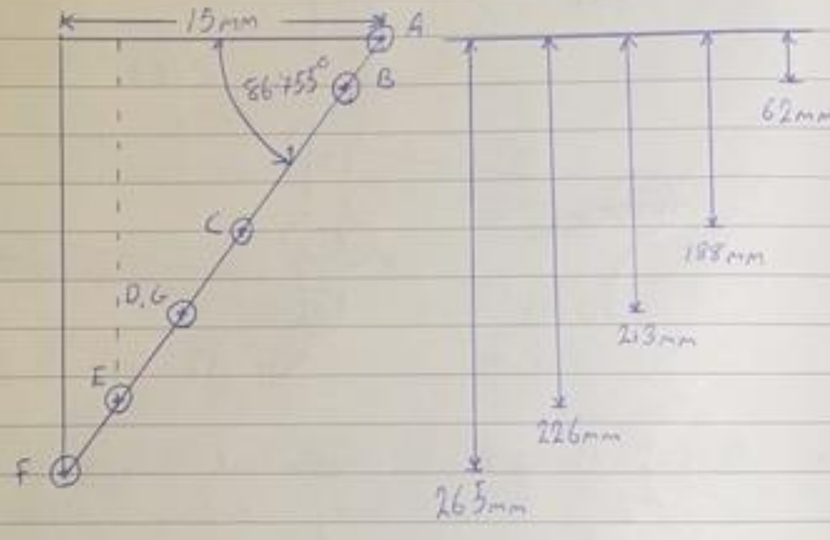
$\sin\theta = 15/265$

$\theta = \sin^{-1}(15/265)$

$\theta = 3.245^\circ$ Conical deflection

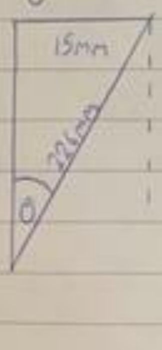
Bushing A & F experience 3.25° of conical deflection

Body:



Axle Box

Bushing E



$\theta = \sin^{-1}(15/226)$

$\theta = 3.8^\circ$ ~~mm~~ ×

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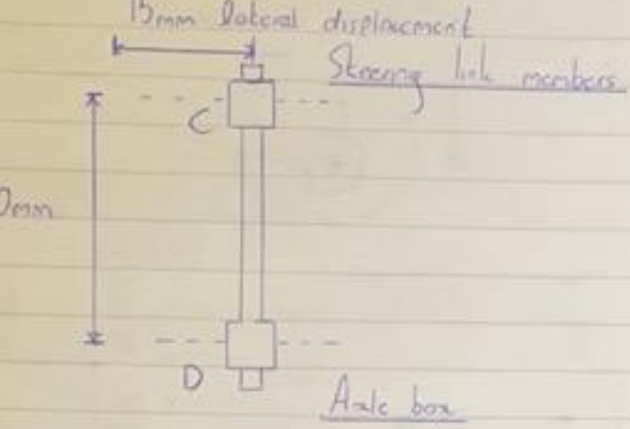
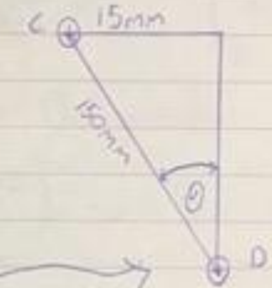
Steering Link Bushing Calculations

Bushing D & C will suffer severe conical deflection

Top view.



Sohcahtoa



$\sin\theta = {}^{15}/_{150}$

$\therefore \theta = \sin^{-1}({}^{15}/_{150}) = 5.74°$

Bushing D & C require a conical deflection of 5.74°

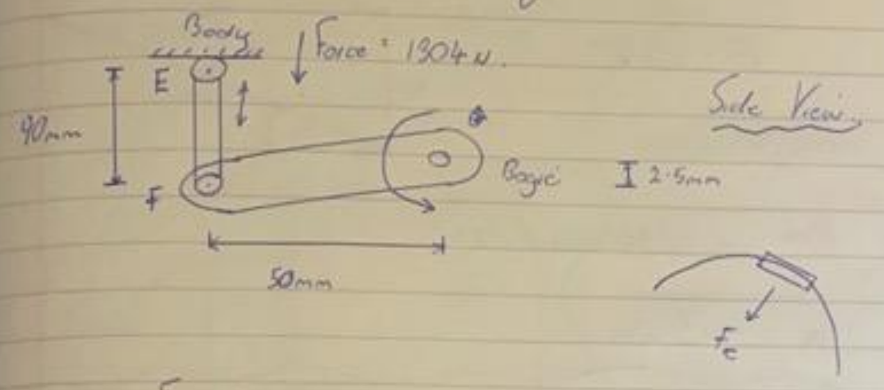
Summary of Steering link bushes

for Point	Bushing	Conical deflection	Torsional Deflection
(E)	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)

57.

Anti-Roll Bar Bushing Specification



$F = \frac{mv^2}{r}$ $v = 4.17 ms^{-2}$
$r = 8000$ mm (EP) / 20000 (RC) mm
$m = 600$ kg

Force $= \frac{600 \times (4.17)^2}{20000} = 522$ N

Force $= \frac{600 \times (4.17)^2}{8} = 1304$ N

A force of 1304 N acts 50mm about locking bushing G

Torsional moment @ G = $1304 \times 50 \times 10^{-3}$
" " " " = 65.2 Nm.
Around LTK110 locking bush.

Finding the deflection in bushing F.

Torsional Stiffness of ARB $\frac{\text{Torque}}{\Theta(\text{rad})} = 2827$ Nm/rad.

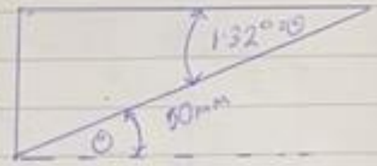
$\Theta = \frac{\text{Torque}}{\text{Stiffness}} = \frac{65.2 \text{ Nm}}{2827 \text{ Nm/rad}}$

$\Theta = 0.02306$ radians $= 1.32$ degrees

60.

Bushing F Torsional Deflection

$\theta = 1.32°$



$\tan\theta = x/50$

$x = 50\tan(1.32)$

$x = 1.15\text{ mm}$

* ∴ When cornering, body drops or rolls a vertical displacement of 1.15mm

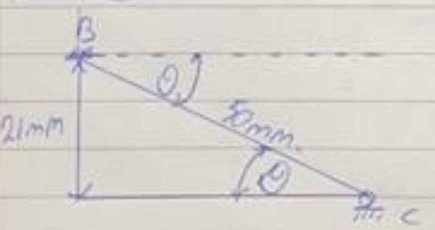
$\tau = 26.1\text{Nm}$ @RC

* Bushing F needs a torsional deflection of 1.32° τ a torsional moment of 65.2Nm

* Bushings E & F should withstand a radial force of 1304 N.

Body Vertical movement = 21mm
Body Longitudinal movement : 0mm
Body lateral movement = 15mm

Vertical movement



$\sin\theta = 21/50$

∴ $\theta = 24.83°$ torsional deflection for bushing F.

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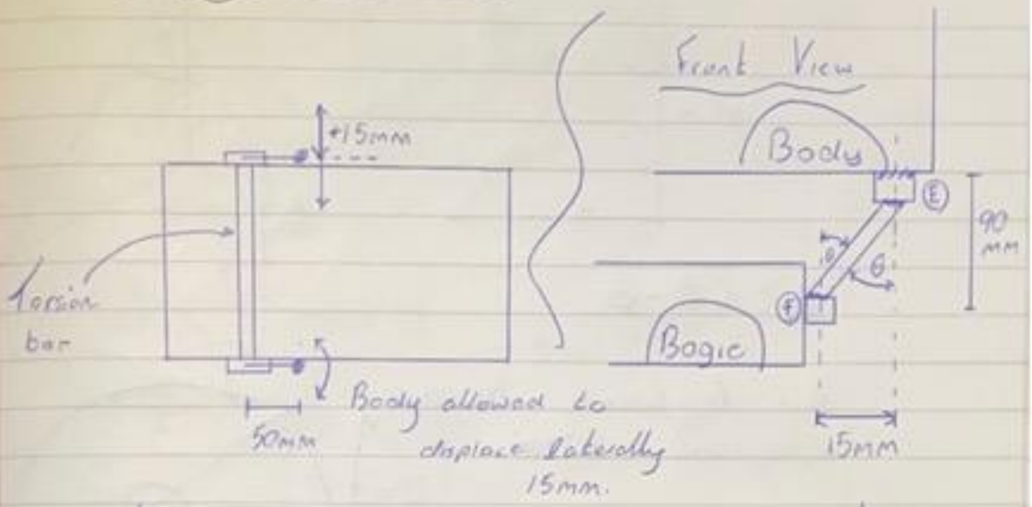
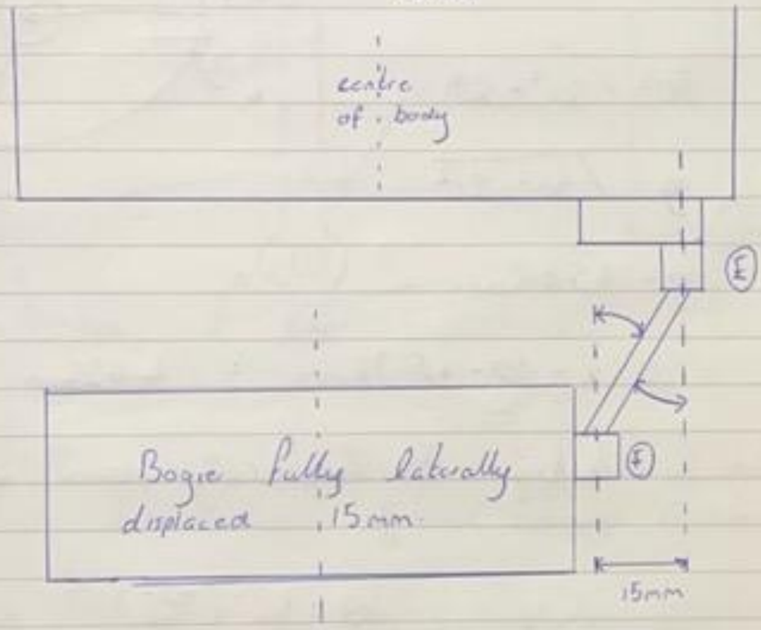
Bushing Conical Displacement.

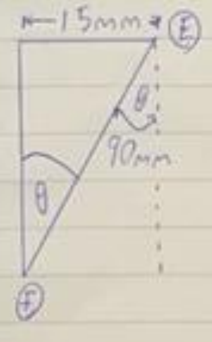
Body Lateral movement : 15mm





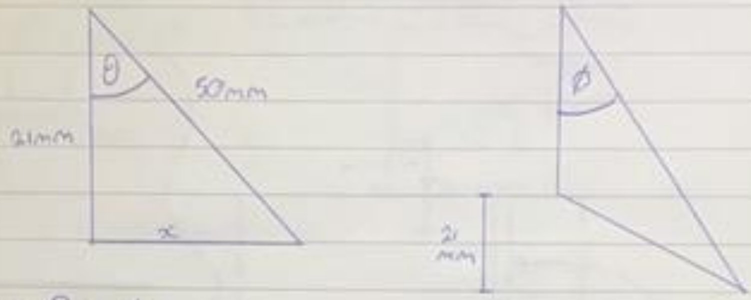
θ = Bushing E & F maximum necessary conical deflection

$\theta = \sin^{-1}\left(\frac{15}{90}\right) = 9.59^{\circ}$



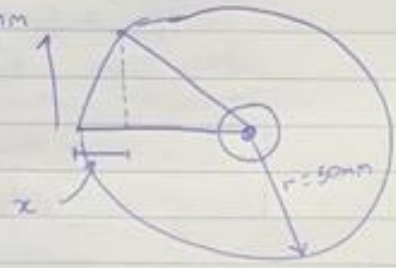
Bushings E & F conically deflect 9.59°

62.



$\Delta \Psi = \theta - \phi$

then longitudinal displacement $= 50 - x$



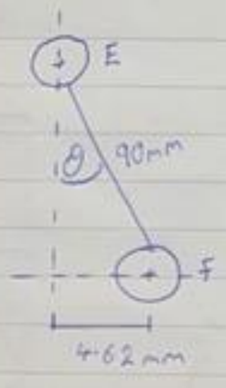
$50^2 = 21^2 + x^2$

$y = \sqrt{50^2 - 21^2}$

$y = 45.38\,mm$

$\therefore\ x = 50 - 45.38\,mm = 4.62\,mm.$

So bushing E longitudinally moves 4.62mm 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 bodys ±21 mm displacement vertical

Appendix H-7: Anti-Roll Bar Calculations (DS)

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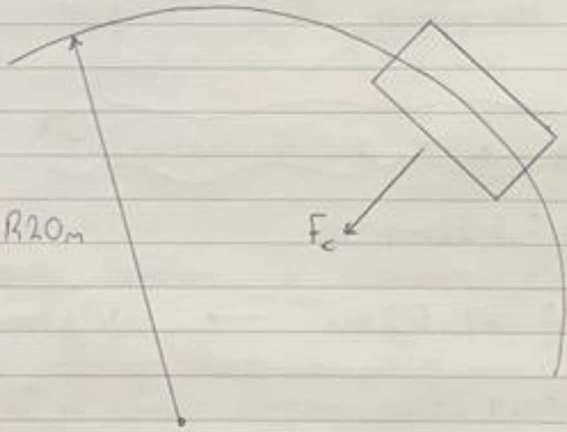
Anti-Roll Bar Deflection 10/12/21

- Find the centrapedal force exerted by the locomotive when cornering the tighest corner @ maximum velocity 15km/h.
- Use this value to calculate forces on ARB & find the corresponding deflection.
- This deflection value will replace the secondary suspension roll value.



$$F_c = \frac{mv^2}{r}$$

$$F_c = \frac{600 \times (4.17)^2}{20} \qquad F_c = \frac{600 \times (4.17)^2}{8} = 1304N$$

$F_c = 521.67N$ → Centrifugal Force @ railway challenge

$F_c = 1304N$ ⟶ @ Exhibition Park

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Torsion in ARB

Torsional Stiffness = $\frac{\text{Torque}}{\text{Angle}} = \frac{Nm}{\text{radians}} = \frac{T}{\theta}$

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

$G = 7.2 \times 10^{10}$ Pa (Steel) S275
$J = 1.194 \times 10^{-8}\ m^4$
$L = 0.4$ 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 Nm/rad

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Anti-Roll Bar Safety Factor

Rods characteristics : Torque = 78.2 Nm
Diameter = 40mm → 20mm
$\sigma_y = 355\,MPa \quad \tau_y = 177\,MPa$

Torsional Shear Stress :

$$\tau_t = \frac{Tr}{J} = \frac{Tr}{\frac{\pi(D^4)}{32}} = \frac{16TD}{\pi D^4}$$

$$\tau_t = \frac{16 \cdot 78.2 \cdot 20 \times 10^{-3}}{\pi (20 \times 10^{-3})^4}$$

$\tau_t = (6.22\,MPa) \qquad \tau_t = 49.78\,MPa$

τ_σ = Yield Shear of torsion bar

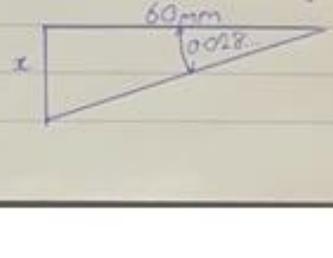
$\tau_\sigma = 177.5\,MPa \qquad \sigma_y = 355\,MPa.$

$$\therefore \text{Safety Factor} = \frac{\tau_\sigma}{\tau_t} = \frac{177.5}{49.78}$$

Safety Factor ARB = 3.56 → Shear

Torsional Stiffness = 2827 Nm/rad

ARB Deflection (vertical) due to 1304N centrifugal force :



Deflection in radians = 0.0277...
$x = 60 \tan(0.0277) =$
Vertical displacement = 1.66mm

Appendix H-8: Steering Link Calculations (DS)

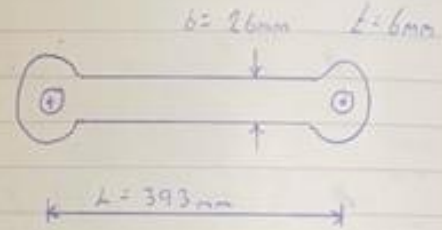
Steering Links Stress Analysis 40

Using the Euler Bernoulli equation

$$P_{cr} = \frac{\pi^2 E I}{L^2}$$

$$\text{Critical Buckling Load} = \frac{\text{pi}^2 \cdot \text{youngs modulus} \cdot 2^{nd}\text{ moment of Area}}{\text{Length}^2}$$

$I = \frac{bd^3}{12}$ for a thin rod.



$$I = \frac{0.026 \times (0.006)^3}{12}$$

$$I = 4.68 \times 10^{-10}\ m^4$$

Critical Load $P_{cr} = \frac{\pi^2 \times 206 \times 10^9 \times 4.68 \times 10^{-10}}{0.393^2}$

$P_{cr} = 6160.7\ N$ buckling load.

Assuming worst case this member must take the mass of the loco

Force = 202 × 9.81 = 1982 N

Safety factor = $\frac{6061}{1982}$

Steering Link Safety factor = 3.05

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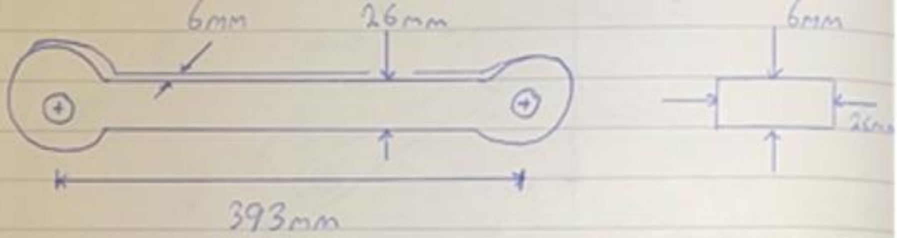
Further Steering Link Calculations

Finding the planar & shear stress safety factors.

Analyzing longest member: of S275 Steel



Force laterally = mass of bogie × 10g = F_y

Force vertically = mass of bogie × 10g = F_z

$F_y = F_z = 202 \times 9.81 \times 10 = 19816.2\,N.$

Planar Stress:

$\sigma = F/A \qquad \sigma = 19816 / A$

Area planar = $6mm \times 26mm = 156\,mm^2$

$\sigma_y = 275\,N/mm^2 \qquad \sigma_y = 275 \times 10^6\,MPa$

$\therefore$ yield force $= \sigma \times A = 156 \times 275 = 42906\,N$

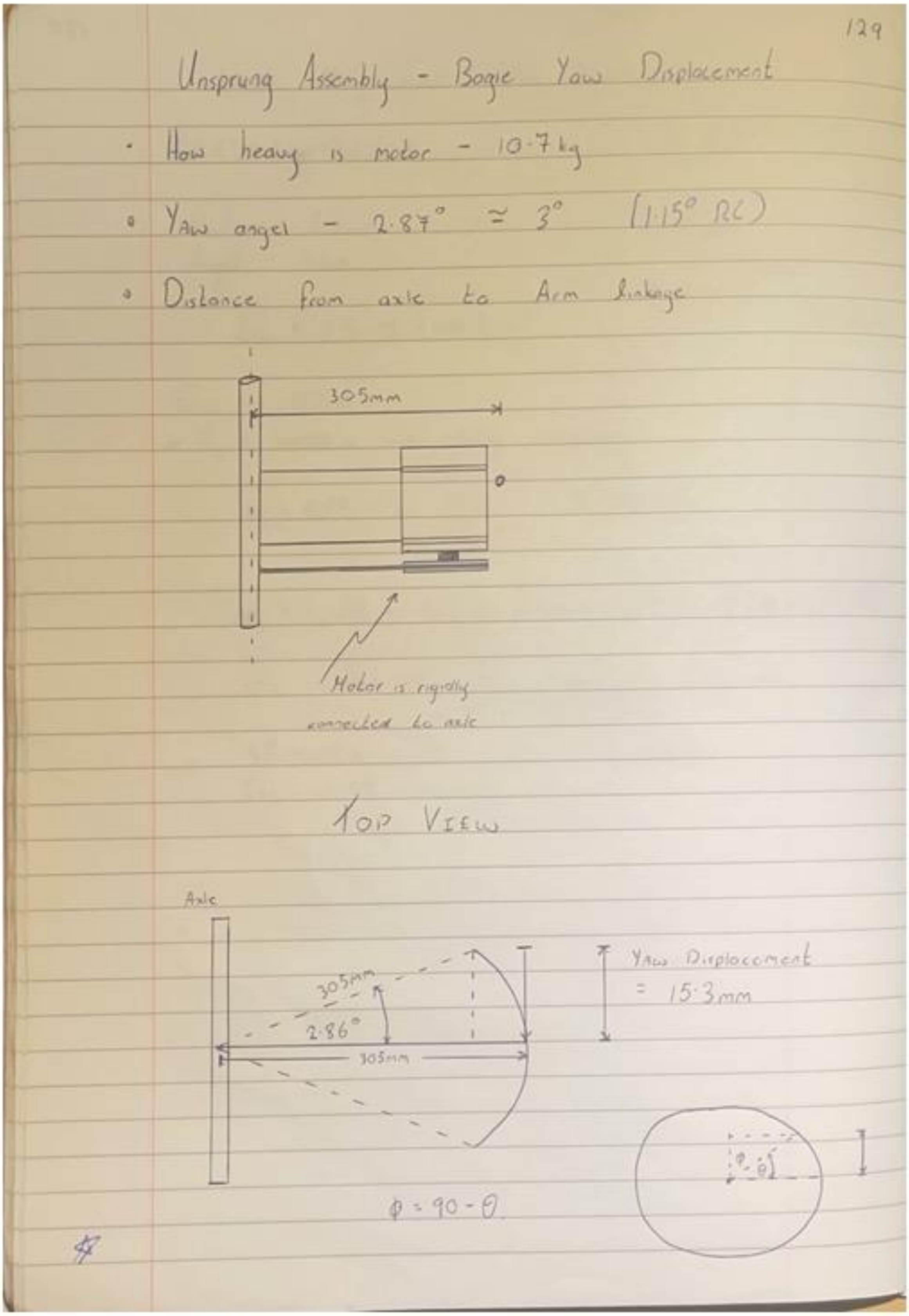
Area planar = $1.56 \times 10^{-4}\,m^2$

Stress applied $= \dfrac{F}{A} = \dfrac{19816.2}{1.56 \times 10^{-4}} = 127.03\,MPa$

Planar Safety Factor

$$SF = \frac{\sigma_{permissable}}{\sigma_{applied}} = \frac{275\,N/mm^2}{127\,N/mm^2} = 2.17$$

Appendix H-9: Bogie Frame Clearance Calculations (DS)



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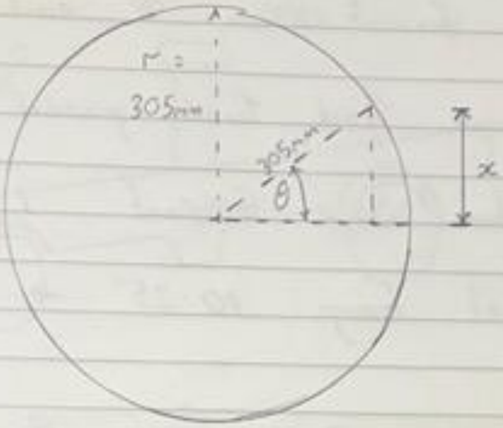
02/12/21

$r = 305mm$
$\theta = 2.87°$



$x = r \sin\theta$

$x = 305 \sin(2.87)$

$x = 15.27mm$ → 20mm to be safe

∴ ALLOW ± 20mm freedom either side of brake on upper bogie frame to allow for axle & unsprung assembly - bogie yaw.

$x = 612mm$ for Railway Challenge

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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

∴ Lateral Displacement = (Brake Distance) sin (Yaw ∠)

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 $D_1 = 340 \sin(6) = 35.5\,mm$

2 $D_2 = 340 \sin(2.87) = 17.03\,mm$

3 $D_3 = 340 \sin(1.15) = 6.82\,mm$

Appendix H-10: Degrees of Freedom (DS)

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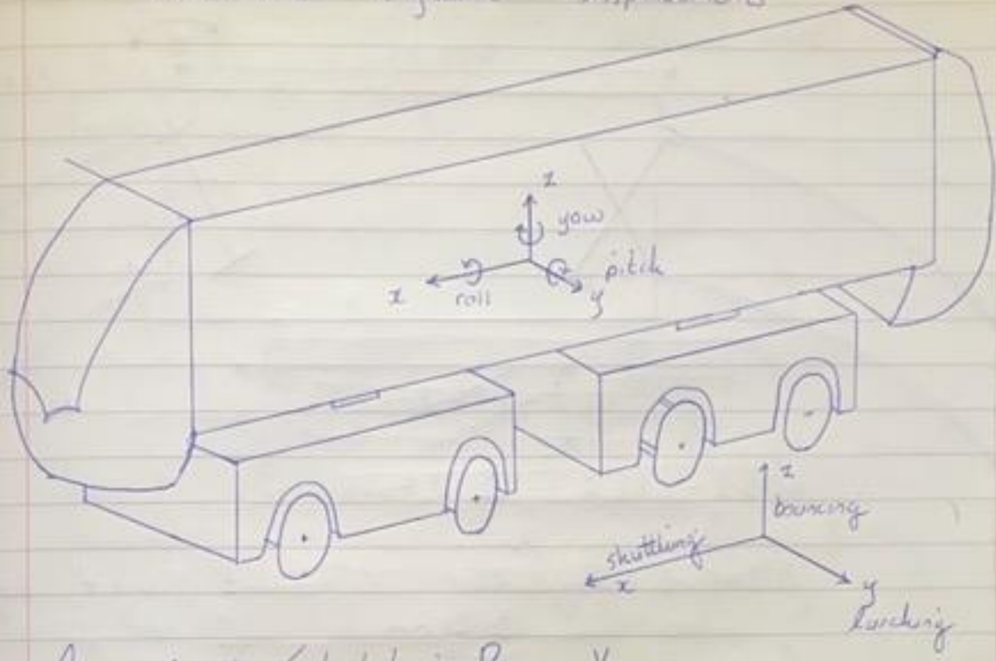
Locomotive Dynamic Displacements



Aims : + Calculate : Bogie Yaw
Body Yaw
Wheelset Yaw

Calculate : Body to ~~body~~ bogie displacements
lateral, longitudinal (0mm), vertical

Calculate : Body roll
Bogie to wheelset displacements

Reasoning : + Derive specification for bogie design
+ Specification for bushings

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Vertical Track Curvature 9/12/21
(Pitch)

Maximum slope = 1 : 84

Sohcahtoa

$\tan\theta = \frac{1}{84}$

$\therefore \theta = \tan^{-1}\left(\frac{1}{84}\right)$

$\theta_1 = 0.682°$ → RAILWAY CHALLENGE

$\theta_2 = 7.125°$ → Exhibition Park

Sohcahtoa

$\sin\theta = x / hyp$

$x = hyp \sin\theta$

$x = 800 \times \sin(0.682) = 9.523\,mm$

$x = 10\,mm$ Vertical Track Curve vertical displacement = 10mm

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Degrees of Freedom

Motion : Shuttle x-direction bogie - wheelset

deflection $\Delta x = \frac{F}{k(\text{shear})\ \text{primary suspension} \times 8}$!

Force = mass of ~~wheelset~~ bogie × 10g

Force = $202 \times 9.81 \times 10 = 19816.2$ N

$\Delta x = \frac{19816.2}{6.1\ N/mm} = 3248$ mm (unrealistic)

Force = $\frac{\text{mass of bogie} \times g}{\text{lateral stiffness of primary}}$

$\Delta = \frac{9.81 \times 202}{6.1} = 324.8$ mm (unrealistic)

$\Delta = \frac{9.81 \times 202}{6.1 \times 8} = 40.6$ mm

Shuttle : Bogie → Body

Unconstrained = $\frac{\text{mass of body} \times g}{\text{2nd spring lateral stiffness rate} \times 8}$

Unconstrained $\Delta x = \frac{600 \times 9.81}{25.7 \times 8} = 28.63$ mm

Constrained Bogie → Body shuttle (x motion)
= 0mm due to traction centre

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Roll about x-axes

Wheelset → Bogie roll

0_{mm}?

DLG

Body → Bogie roll

- Related to dynamic loading gauge
- Constrained by Anti-Roll-Bar

Lurching (motion in y-axis)

Bogie → Wheelset lurching

limit = 9_{mm}

unconstrained displacement = $\frac{\text{mass of bogie} \times 9.81}{\text{primary suspension lateral rate} \times 8}$

u, $\Delta x = \frac{202 \times 9.81}{6.1 \times 8} = 40.6mm$

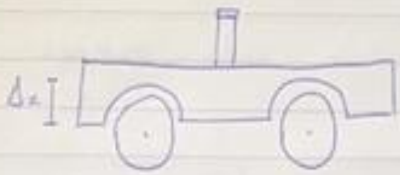
Body → Bogie lurching

Unconstrained $\Delta x = \frac{600 \times 9.81}{\cancel{6.1} \times 8 \times 25.7}$

Unconstrained Δx = 28.6 mm

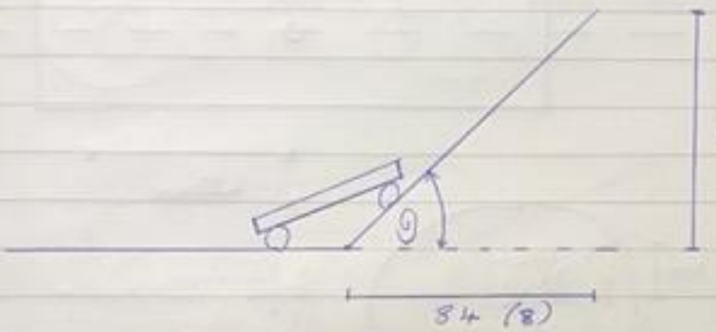
69

Pitch (about y axis)



Sohcahtoa

Pitch of wheelset relative to bogie



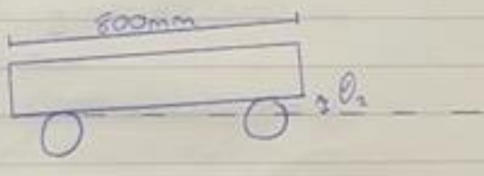
minimum slope = 1:84. ; max slope = 1:8

$\tan\theta = \frac{1}{84_1\ (8)_2}$ ∴ $\theta_1 = \tan^{-1}\left(\frac{1}{84}\right)$ $\theta_2 = \tan^{-1}\left(\frac{1}{8}\right)$

$\theta_1 = 0.682°$ for railway challenge
$\theta_2 = 7.125°$ for exhibition park.

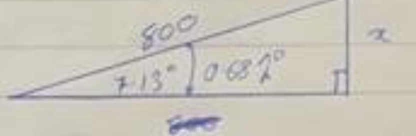
Bogie-wheelset constrained angle

Use θ_2 for calculations.



$\cancel{x = \tan}$

$\sin \cancel{\tan}(7.13) = \frac{x}{800}$



∴ $x = 800 \sin \cancel{\tan}(7.13)$

$x = 100mm$.

$x = 800 \sin(0.682) = 9.5mm$ ↑↓

say $x = 15mm$ $\theta = \sin^{-1}\left(\frac{15}{800}\right)$ $\theta = 1.07°$

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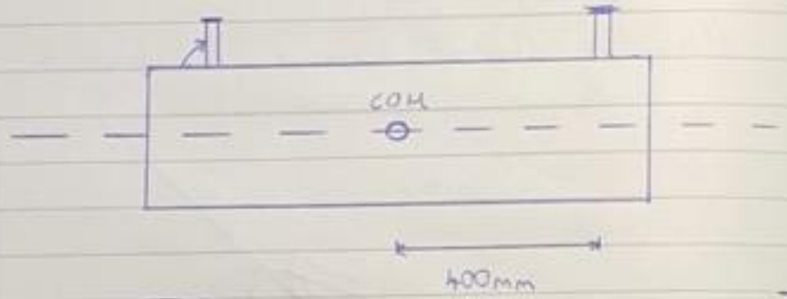
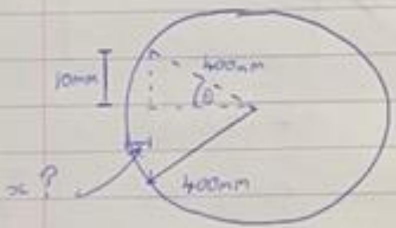
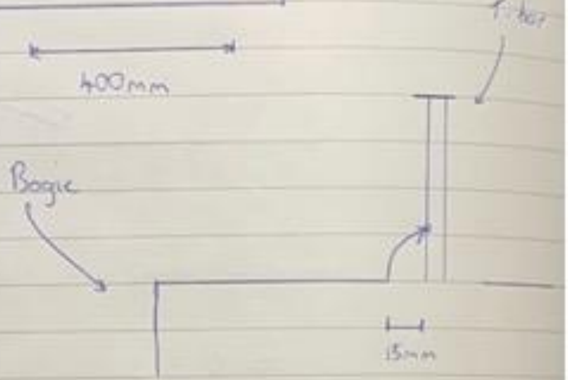
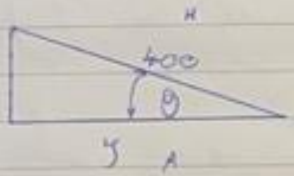
Bogie – Wheelset Pitch Constraint

Bogie will collide with axlebox T-bar









$x = 400 - y$

$\theta = 0.682^\circ$

Sohcahtoa

$\cos(0.682) = \frac{y}{400}$

$y = 400\cos(0.682) = 399.97\,mm$

$\therefore \; x = 0.028\,mm$

$\therefore$ pitch constraint = 15mm axle box T-bar

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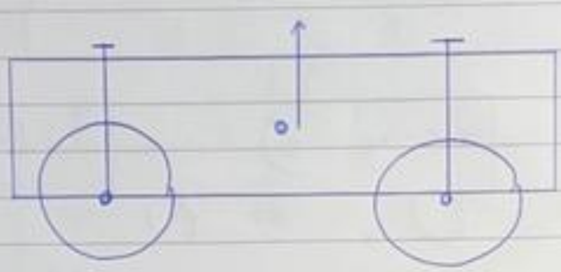
Vertical Displacement Wheelset → Bogie

Unconstrained :



Force = Vertical Primary Spring Stiffness × displacement Vertical

Force = mass of bogie × 9.81 ×

$F = kx \quad \therefore \quad \Delta x = F / k \times 8$

$$\Delta x = \frac{202 \times 9.81 \times 5g}{24.52 \times 8} = 50\,mm \text{ compression}$$

50.51mm

Body - Bogie vertical displacement

$\Delta x = F / k \times 8$ $k = 2^{nd}$ Sus Stiffness N/mm

$F = m_{body} \times 2g$

$$\Delta x = \frac{600 \times 2 \times 9.81}{41.156 \times 8}$$

$\Delta x = 35.73\,mm$

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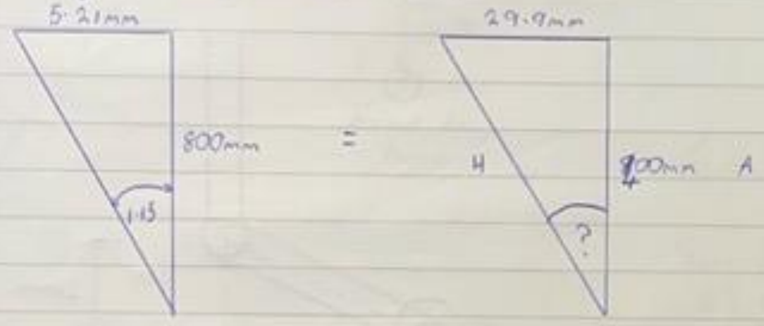
Wheelset → Bogie Yaw

Unconstrained : Angle = 2.87° EP
" 1.15 RC

Displacement : 5.21mm RC
" 13.02mm EP

Using Similar Δ :

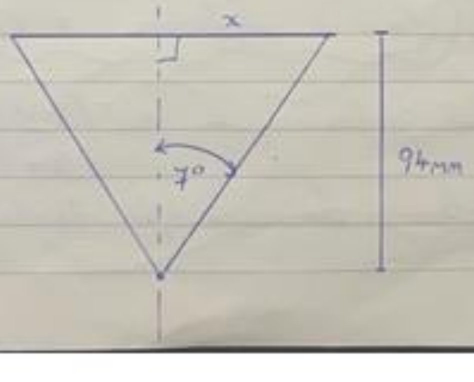


sohcahtoa $\theta = \tan^{-1}\left(\frac{29.9}{800}\right)$

$\theta = 2.14°$ yaw limit
θ 4.27° yaw limit

Bogie → Body yaw

Max deflection = 7° by bushings
∴ max



Sohcahtoa

$\tan(7) = \frac{x}{94}$

$x = 11.54$mm

$\theta = \tan^{-1}\left(\frac{x}{94}\right)$

$x = 94\tan 7$

Appendix H-11: Secondary Spring Calculations (DS)

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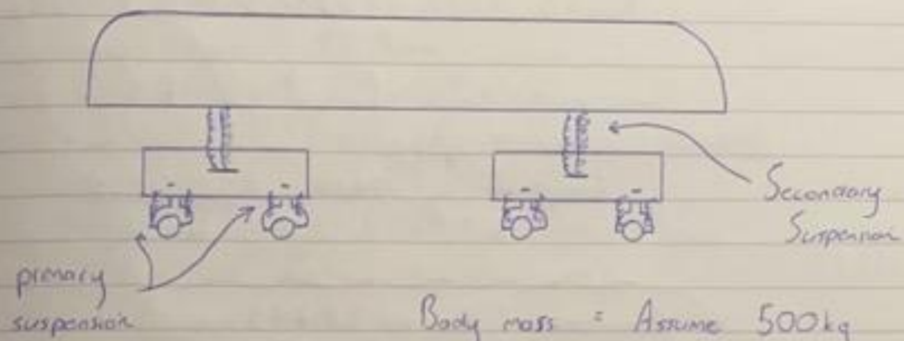
Suspension Springs Safety Factors

Secondary (~~Primary~~) Suspension Springs



Body mass = Assume 500 kg
Bogie mass = 202 kg

~~Primary~~ Secondary Spring Characteristics:

Part Number : LHL 2000A 06
D_o = 46·48 mm
Free length = 127 mm
Solid height = 60·96 mm
Δx = 66·04 mm compression
k Rate = 41·156 N/mm
Wire diameter = 6·35 mm
Lateral Stiffness = 25·7 N/mm (from FEA)
Force required to fully compress $F = kx$

$F = 41{\cdot}156 \times 66{\cdot}04$

Force = 2718 N

Force experienced by each spring

$\left[\frac{(\text{body mass} \times 2g)}{8} \right.$ ~~+ 2(bogie mass × 5g)~~ $\left. \right] / 8$

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Secondary Suspension FoS

Force experienced $= \dfrac{(500 \times 2 \times 9.81)}{8} \cancel{+ (802 \times 9.81 \times 8)} \atop \cancel{16}$

Force experienced $=$ ~~18516 N~~ $= 1226.25$ N

Compression from force ↑ : $F = kx$

$\Delta x = \dfrac{f}{k} = \cancel{18516} = \dfrac{1226.25}{41.156}$

Compression $\Delta x = 29.795$ mm

Safety factor $= \dfrac{\text{permissible force}}{\text{force}} = \dfrac{2718}{1226.25}$

Factor of Safety for 2ndary Sus $= 2.21$

Primary suspension static loading spring compression + height

Static Load $= (200 \times 2 + 600)/16$

613.13 N (force experienced by each spring)

$\Delta x = F/k = 613.13 / 24.52$

$\Delta x = 25$ mm

(inventor compression = ~~[illegible]~~ 20.4 mm)

(inventor force / spring = 500 N)

Appendix L: Data Tables

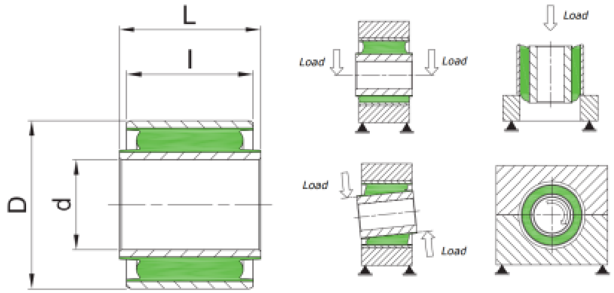
Appendix L-1 FIBET Fully Bonded Bushings Data Sheet

FIBET FULLY BONDED RUBBER BUSHES

https://www.fibet.co.uk/range/rubber-anti-vibration-mounts/bushes-bonded-suspension/?gclid=CjwKCAiAqIKNBhAIEiwAu_ZLDvHzzJQJx9Ud85g9scPOd12eiitVdeDHnTYDTqdiELQ6O6xQzJML3hoCGC8QAvD_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	External Diameter	Bore Length	External Length	Radial Stiffness	Radial Max. Defln.	Radial Max. Force	Axial Stiffness	Axial Max. Defln.	Axial Max. Force	Torsional Stiffness	Torsional Max. Defln.	Torsional Max. Moment	Conical Stiffness	Conical Max. Defln.	Conical Max. Moment	Mass	Rubber Hardness	Stock Item or On Request
	(mm)	(mm)	(mm)	(mm)	(dN/mm)	(mm)	(N)	(dN/mm)	(mm)	(N)	(Nm/deg.)	(+/- deg.)	(Nm)	(Nm/deg.)	(+/- deg.)	(Nm)	(g)	(IRHD)	
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

RdDes & MT2 R Mech Eng Compts/21