EGB111: PROJECT REPORT

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Author Declaration:

"In signing this cover sheet, each author attests that they have made a fair contribution to the conduct of the design project and the generation of this document. Each signed author also attests that they acknowledge the fair contribution of all other listed authors, and that if a team member is not included in the signed authorship list, the team has discussed this with the unit coordinator and this course of action is supported by the unit coordinator."

Statement of Contribution

Team Member	Report Sections Written	Technical Contributions	Percentage Contribution
Jordan Thuo	Evaluation of design concepts, Conceptual & Final Design - Fork, Truss and Base.	CAD designs, Laser cutting, Assembly of base	20%
Kiara Maier	Balance equation, Costing Breakdown, Design Evaluation & Recommendations	Materials testing, initial CAD structure & gear designs	15%
Nigel Siong	Mechanical Design – torque, speed, current, power & efficiency	Gear design, Motor testing, Mechanical Design	20%
Jiang Pass	Truss analysis, Gears Motors, and Axles	Mechanical Design	15%
Einstein Roi Dimalibot	Conceptual Design & Final Design - Fork, Truss and Base	Truss Builder and Assembly of base	15%
Deena Al Shahwani	Executive Summary, Initial Concepts & Meeting Minutes	Structural design	15%

Executive Summary

After careful evaluation of the three initial design concepts, the Trebuchet was selected to be the most ideal. During the final testing of the Trebuchet, all four of these criterions were met. These included an innovative design weighing under 1.5kg, an electromechanical and structural safety system, and to successfully carry the 0.5kg object from pedestal A to B in less than 45 seconds on the first attempt.

The Trebuchet design was an unfixed structure placed in line diagonally in the centre of the testing board, between the pedestals. It included a vertically rotating arm to lift the object from Pedestal A to B in a circular clockwise motion, driven by a motor. A counterweight and a square based A-frame were integrated in the design. However, with further calculations and prototype testing, complications in both the base of the structure and its counterweight were found. Hence, adjustments were made to the final concept of the Trebuchet, by re-designing base to simply stand on 4 leg supports 600mm in width apart.

The Warren truss design was selected as it resulted in a linear increase in force through the horizontal beams and internal truss members. Each of the internal members was 45mm long. The cross-section dimensions of the members had to be determined to withstand the force of compression during the lift, as it was discovered that member lengths did not affect tensile strength and were all capable of withstanding the force provided by the 0.5kg load. Calculations showed that the truss design would fail at the 1.5kg load as per safety factor between 2 and 3 requirements.

Whilst the initial design included a counterweight on the rotating end of the arm, the final concept discarded that component, as the motor proved to provide enough torque with the gears to fully rotate and have the structure remain balanced.

Through calculations, a nominal voltage of 12V DC gear motor was chosen in the final design due to its low power rating and the torque it would provide to lift the 0.5kg mass. Operating at its most efficient state, the mechanical system of the motor ran at 15% efficiency. When tested under a 1kg load the gears began to skip, resulting in a failure to lift the load as desired. The rotational speed of the motor was calculated using the speed and torque motor curve, however as the gears provided greater torque to the motor, the calculated speed of 25rpm proved too fast. Therefore, a gearbox at a gear ratio of 10 was added, resulting in a much lower rate of $2.5rpm$ capable of handling the load. Using the current and torque motor curve, the current draw was estimated to be approximately 0.4-0.5A using the operating torque of $3806gcm$ at a gear ratio of 10. With this, the power supply calculated remained well below the power budget, resting at 6.05W. Due to the singular rotational movement in the design only one mechanical system was required and therefore the circuit only included a DPDT to control the motor's on and off function.

Despite achieving all objectives and requirements for this project, more constructive and practical solutions remain that would improve the overall design of the Trebuchet. These include but are not limited to; removing the rotating end of the arm. As the counterweight was no longer needed, the arm served no purpose and only added extra weight to the structure. By removing the arm, the base must increase in size to reduce the risk of toppling during lift, while extra cross-bracing support can be added to the base to limit twisting. An additional circuit design can be included to generate an electrical failure, rather than a mechanical failure, through the addition of fuses and resistors to the circuit.

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Introduction

The aim of this design project was to create a model lifting device capable of moving a 0.5kg load from a lower pedestal of 50mm to a higher pedestal of 100mm, repositioning the load diagonally across the testing board. This report assesses the process of developing the lifting device from a preliminarily design and prototyping, to the final product. The process includes the initial conceptual planning of ideas, developing designs and lastly implementing the technical applications through the structural, mechanical, and electrical factors to develop the final design.

The objective of this project was to achieve all the optimal requirements specified in the design brief. These included: a design model weighing under 1.5kg, an electromechanical system with a safety factor between 1 and 2, the main truss designed with a factor of safety between 2 and 3, and to successfully carry the 0.5kg load from pedestal A to B in less than 45 seconds on the first attempt. Ideally, through fulfilling all the required specifications, the design would satisfy the preliminary requirements for the Grand Challenge, where the final design can be showcased to a panel of professional industry personnel. To achieve this, an additional criterion of providing an innovative and unique design, as an alternative to the fundamental crane design must be fulfilled.

Design Concepts Considered

Multiple designs were initially considered during the conceptual planning stage:

1. Gantry Idea: Weights are lifted vertically, then translated horizontally across a track.
2. Tower crane idea: The structure is fixed to the board. It carries the weights with a pulley system and pivots around a centre point.
3. Trebuchet idea: It is placed in the centre and lifts the load in a vertical circular motion from pedestal A to B.

The positive and negative outcomes of each design concept were identified during this stage. This allowed for a more sophisticated overview of the conceptual design process and evaluated potential future roadblocks.

Table 1: Overview of the positive and negative outcomes of each design concept.

Design Concept	**Positives**	**Negatives**
Gantry Design	- Low stress on components - Easy to balance, even if unfixed due to large size.	- Unfixed structure has possibility of not fitting within testing board. - Requires more material to build; is heavier. - Electrical wiring moves with the load across, is impractical
Crane design	- Easy to build. - Fixed device: balance is non-issue.	- Conventional - Requires strong cantilever support. - More complex stress calculation - Requires two mechanical system for pully and translation
Trebuchet design	- Simple concept - Simple torque calculations - Less materials - Unique aspect - Requires only one motor, can fulfil lift in one movement	- High torque required. - Adding counterweight to account for torque may exceed weight limit. - Chance of toppling due to increased dynamic load.

Evaluation of Design Concepts

During the process of ideation, these design concepts were assessed against an evaluation matrix (table 2) developed specifically for this project. This matrix was constructed to ensure the design would fulfill all the optimal requirements, as well as the unique design component.

The performance of each design was considered, examining the potential power supply needed, the time to complete the lift, and the potential to exceed the weight limitation. Whereas the effectiveness criterion demonstrates how successfully the design would transport the load from pedestal A to pedestal B without causing the load to swing and the structure to become unbalanced. In addition, the overall cost considers the amount of material and the cost of the mechanical and electrical components required. Furthermore, the complexity of the design identified whether the time for construction will be within the time constraints and finally, the uniqueness of the design was determined by the originality of the design concept.

Table 2: Evaluation matrix for determining the best design concept.

	Performance	Effectiveness	Cost	Complexity	Unique	x/25
Gantry	4	3	4	4	3	18
Tower crane	3	4	3	3	2	15
Trebuchet	5	4	5	4	5	23

After careful consideration of each of the evaluation criteria, the trebuchet design exhibited the highest grade of 23/25 and was selected. This was due to the aesthetic features of its design- taking into account its innovative and unique model as well as the elegance of its motion, and its ability to achieve all the optimal requirements of this project.

The Initial Concept

The initial concept of the Trebuchet-like structure would reposition the load from pedestal A to B diagonally across the board, rotating the arm in a circular clockwise motion 180 degrees to the plane.

For this concept, the structure would need to be freestanding and placed in the middle of the test board as pictured below. The orthographic diagram (figure 1) illustrates the initial design structure. This design included:

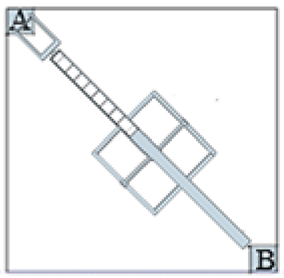

Figure 1: Board placement.

- A Warren truss design
- A triangular square base made of balsa wood.
- A 400g counterweight to reduce the torque requirements of the motor to fit to power budget specifications.
- A fork to which the mass is hooked to and allows the mass to swing through upon rotation.
- Axle and bearings for the arm to rotate at

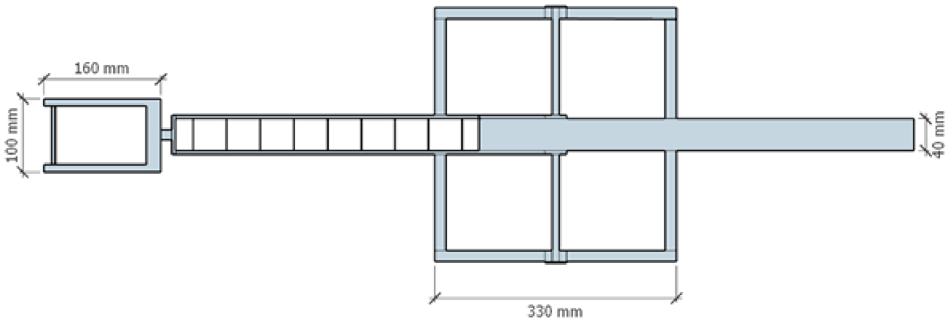

Figure 2: Top view of initial concept

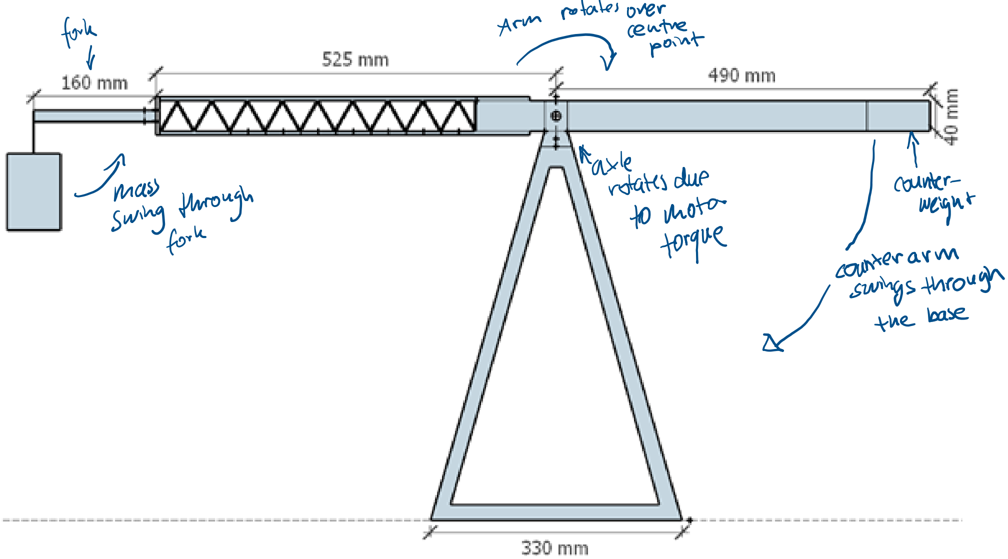

Figure 3: Side view of initial concept

In the initial design, one high torque or geared motor would turn the axle which in turn, would rotate the arm over and across. The mass swings through the fork which has been fitted to the dimensions of the mass of 88mm in width and height of 122mm as outlined in the brief.

Conceptual design components

The initial design consists of five main components:

1. The base structure: The base structure is non-fixed and supports the rotating components. Additionally, it is wide to ensure the stability of the trebuchet design concept.

2. Turning axle (attached within bearings) powered by motor: The turning axle is fixed to the arm and is driven by the motor with an attached gear. It is also attached to the base structure with bearings to allow vertical rotational movement of the arm with less friction.

3. Rotating arm: The arm consists of the 525mm truss structure on the carrying side and 490mm of thick balsa on the opposite end with the counterweight. At the end of the truss structure a ‘fork’ is attached to act as the load carrier, allowing for the load to swing through the fork as the arm completes its circular rotation.

4. Motor and gears: The motor consists of a worm gear to prevent it from rotating backwards. It drives on multiple gears with sufficient gear ratio to rotate the axle. It is designed to fail at carrying the 1kg load, in order to meet the optimal criteria.

5. Circuit: The design requires only one motor, so a simple circuit is implemented with DPDT switch to drive the arm forward and reverse and control its on and off function.

Evolution of Initial Concept to the Final Concept

The construction stage of the design was an iterative project. Two prototypes were built throughout the process.

Issues that arose with the initial design were:

→ The base of design would crossover a pathway when placed in the centre of the board due to its square base shape and dimensions of 330mm length and width.
→ Dimensions of mass did not account for the height added by the hook of the mass, therefore would not swing through.
→ Needed room to attach gears and motor.

Improvements made to the concept:

→ Modified the base material to be made of laser cut high density fibreboard (HDF), this was chosen as it reduced the construction time significantly. This allowed for experimentation with the design to occur at a much faster rate, and ultimately sped up the production process. This was useful as it was found with the first prototype that the base proved too heavy however, this issue was quickly resolved with minor adjustments to the CAD file by adding triangular cut-outs to reduce the weight of the HDF.
→ Along with the improvement of the base design four legs were added at 600mm apart, removing the previous square base concept with an A-Frame. Through this, could cross over the pathway without obstructing it.
→ The dimension of the fork size was increased to 180mm to allow space for the mass to rotate through.
→ A suitable method to attaching the fork to the truss was to use a balsa block as a support between the fork and truss components.

Detailed Design

The structure is comprised of structural, mechanical, and electrical components.

Structural Design

Truss calculations

A Warren Truss design was chosen. The equilateral triangles resulted in a linear increase in force through the horizonal top and bottom members of the truss and the same force through the internal truss members.

The desired truss height of 40mm and total truss length of 415mm resulted in the truss containing 18 bays, 37 members and 20 joints. With one pin joint and one roller joint, the truss had a total of 3 reaction forces. Therefore, the truss was determinate.

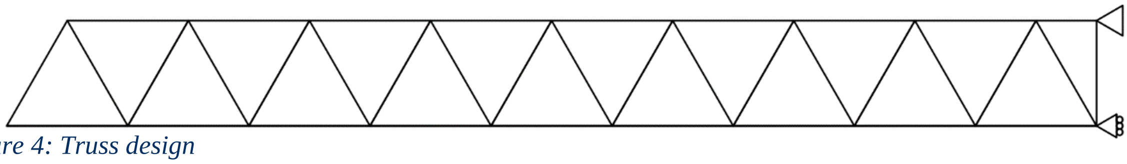

Figure 4: Truss design

The joint method was used to calculate the forces experienced by members.

$$2J = M + R$$

The load used to calculate member forces was the 0.5kg load. Figure 5 shows the compressive and tension forces for each member in the truss. Sample force calculations are present in Appendix D.

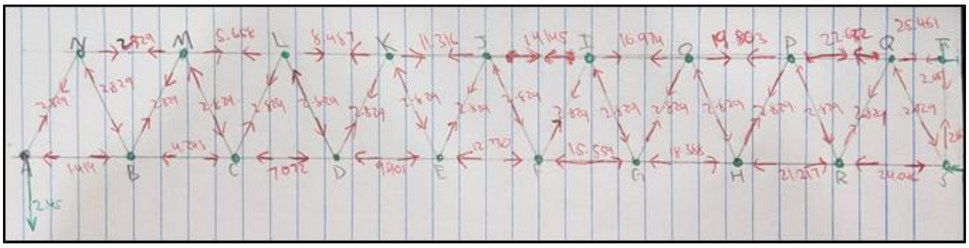

Figure 5: Member compressive and tensile forces

The greatest compression force at this load was 24.046N. The greatest tension force at this load was 25.461N. Therefore, for the truss to fall between the factor of safety of 2 and 3, the truss members must be able to withstand 48.092N in compression and 50.922N in tension, while failing at 72.138 compression or 76.383N tension.

Material Testing & Selection

Balsa wood was selected as the most suitable material for the truss design as it is extremely lightweight and versatile (The Basic Woodworking, 2016).

Multiple lengths and two widths were tested using the testing rig. To reduce random error three test trials were undertaken and the average values were used to determine the maximum force that size member could withstand. Results are shown in Appendix A1.

It was discovered that all members tested were capable of withstanding 76.383N in tension, as changing member length does not affect tensile strength. Therefore, only compressive forces were considered when determining member size and length.

At 1.5kg the greatest compressive force would be 72.138N. Therefore, it was determined that the 45mm length with a 3x3mm cross section would provide the strength needed to carry the load of 1kg, while remaining within the factor of safety of 3.

The 5x5mm Balsa was also tested however it was found that it would provide too much strength and therefore would not buckle at the 1.5kg load as per design specifications.

The average Young's Modulus of balsa was calculated using the Elastic Modulus formula:

$$E = \frac{F_{cr}(KL)^2}{\pi^2 I}$$

Both ends are pinned and therefore the effective length factor K equal to 1 was used. The calculation was done for various lengths and gave an average value of 4145.6MPa. Therefore, the forces in the truss members would not be able to significantly alter the length of the members (see Appendix B for sample calculation). This calculated Young's Modulus corresponds well with the book value for balsa wood of 3710 megapascals (The Wood Database , 2014).

Balance calculations

The structure is a freestanding structure and therefore must be balanced when the 0.5kg load is lifted. If the structure were to tip, it would tip at point A, rotating anti-clockwise.

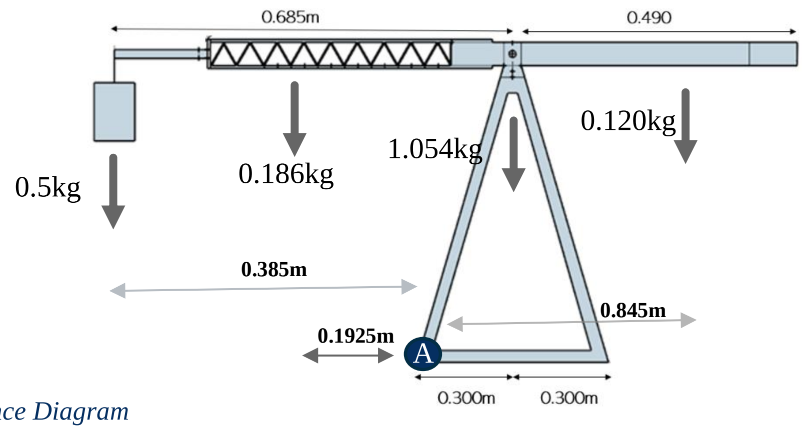

Figure 6:Balance Diagram

Table 3: Weights of Structure Components

Structural Component	Weight
Base	590g
Motor + Gears + Axle	464g
Left Arm	186g
Right Arm	120g
Total Structure Weight	**1.36kg**

To ensure that the design was balanced in the state where the 0.5kg load is the furthest away from the base, the moment around A was calculated. The main components of the structure were weighed and are shown in Table 3. The base, motor, gears, and axle weighed in total 1.054kg at the centre of structure.

$$\sum M_A = -0.5 \times 9.8 \times 0.385 - 0.186 \times 9.8 \times 0.1925 + 1.054 \times 9.8 \times 0.300 \times + 0.120 \times 9.8 \times 0.845 = 1.85509\text{Nm}$$

As the moment acts clockwise, it opposes the anti-clockwise moment created by the 0.5kg load and therefore the structure will remain balanced. This countering moment is created due to the large base width and self-weight of the structure, mechanical and electrical components at the centre.

Testing of truss

The first truss design tested was built entirely with 3x3mm balsa members and only horizontal support beams connecting the two 2-dimensional truss frames. This design failed at the 0.5kg due to torsion. To correct this, the main horizontal members were replaced with 5x5mm balsa. Diagonal cross-bracing was added to the top of the truss to both reduce twisting and assist with supporting the compression load.

Testing of the final truss was undertaken by trialling the truss with an increasing load from 0.5kg to 1.5kg adding a mass of 100g incrementally.

It was found that the truss was able to hold the 0.5kg and 1kg, however, failed due to buckling at the 1.5kg load. The member buckled at the furthest end of the load (See Figure 7: Tested Member Failure, however this was expected as they are the members that experience the greatest compressive force.

Figure 7: Tested Member Failure

Other Structural Components

- → The counterweight side of the crane arm was made using a 38x38mm balsa beam. This was chosen as it is very light and excessively capable of withstanding the resultant forces from the counterweight.
- → The axle used to attach the crane arm to the base was an 8mm threaded steel rod.
- → The counterweight was made of 38x38mm washers, with 13 washers and attaching screw totaling the weight to 400g, however this was not implemented in the final design as discussed later in this report.
- → The support base was laser cut from 3mm high density wood fiber board (HDF). The 3mm thickness was not an issue, as most forces would travel perpendicular to this thickness. These forces were instead supported by the 600mm of width in the legs, which was far more width than necessary.
- → HDF was chosen for the base structure as it is a material generally unaffected by warping and cracking. In comparison, the acrylic is very prone to cracking due to its molecular structure and resistance to expansion and contraction (Ruck Cabinet Doors, 2013). Limitations of HDF include: sensitivity to heat and moisture damage, however, are considered negligible to the scope of the task.

Mechanical Design

A 12V DC gear motor (YG2736, see Appendix D for the data sheet) was used in the final design. This was chosen as it had a low power rating and would be able to provide the torque required to lift the 0.5kg mass.

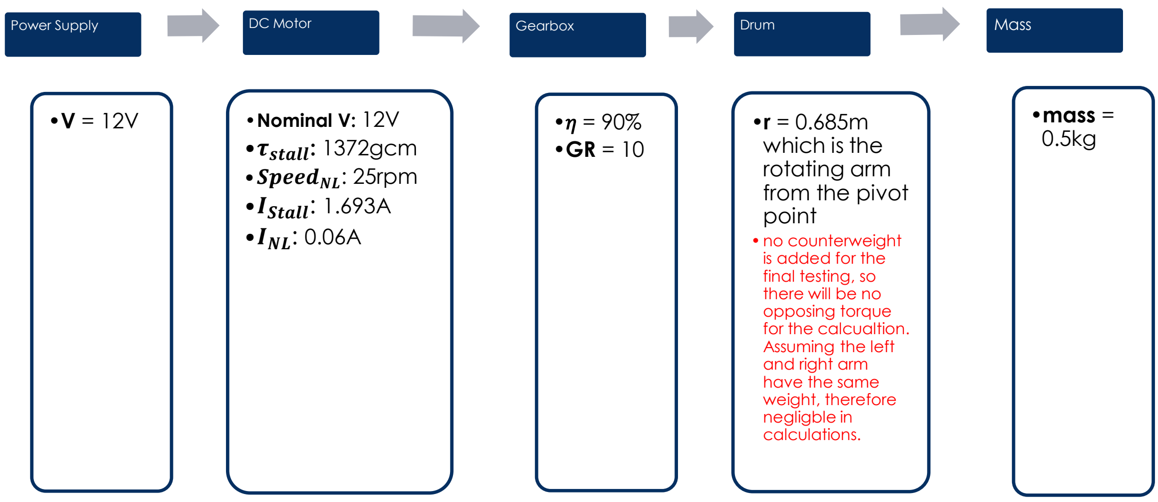

Figure 8: Electromechanical Systems Diagram

Torque Calculations

The torque required for the 0.5kg load at distance 0.685m from the pivot was calculated:

$$\tau_l = 0.5 \times 9.8 \times 0.685 = 3.3565\ Nm$$

The required torque output of the motor given a gear ratio of 10 and with a gear efficiency assumed to be 90% efficient:

$$GR = \frac{\tau_{out\ by\ gear\ box}}{\tau_{in\ by\ motor} \times\ efficiency}$$

$$\tau_{in\ by\ motor} = \frac{\tau_{out\ by\ gear\ box}}{GR \times efficiency}$$

$$\tau_{in\ by\ motor} = \frac{3.3565}{10 \times 0.90} = 0.37294Nm$$

$$= 3805.5555gcm$$

This torque required from the motor is far less than the stall torque of 13990.61gcm which was provided in the data specifications of the motor. Therefore, the motor will be able to lift the 0.5kg mass successfully.

Rotational Speed of the Motor

The rotational speed of the motor was calculated using the speed and torque motor curve equation:

$$Rotational\ Speed_{motor} = -\frac{rotational\ speed_{no\ load}}{\tau_{stall}} \times \tau + \ rotational\ speed_{no\ load}$$

$$Rotational\ Speed_{motor} = -\frac{35}{13990.61} \times 3805.5555 + 35 = 25.4797\text{rpm}$$

Not only do the gears provide the greater torque required for the motor, but also assist in slowing down the rotation speed. The calculated rotational speed of approximately 25rpm is too fast.

The rotational speed of the rotating arm with the gearbox at a gear ratio of 10 is far slower:

$$\omega_{out} = \frac{\omega_{in}}{GR} = \frac{25.4797\text{rpm}}{10} = 2.54797rpm$$

The rotating arm will move at the speed of $2.54797rpm$ which is now slow enough to handle the dynamic load.

The Expected Current Draw

Using the current and torque motor curve, the current draw was estimated using the operating torque of $3806gcm$ at the gear ratio of 10. The motor datasheet provided had the no load current as $I\_no\ load = 0.06$ and the stall current to be $I_{stall} = 1.693$.

$$I = \frac{I_{stall} - I_{no\ load}}{\tau_{stall}} \times \tau + I_{no\ load}$$

$$I = \frac{1.693 - 0.06}{13990.61} \times 3805.5555 + 0.06 = 0.504\ Amps$$

At the 0.5kg load the current draw was expected to be at 0.5A, corresponding to the test results which were recorded to fluctuate around 0.4-0.5A.

With this low current draw of the motor, the power supply was calculated to be well below the power budget of 10W:

$$P = IV = \ 0.504 \times 12 = 6.05\text{W}$$

Efficiency of Mechanical System

The overall efficiency of the mechanical system is hard to estimate. The efficiency of the gear box was assumed to be 0.9 however in reality this would probably be far less due to friction between the gear teeth. The motor was operated at its nominal voltage of 12V and therefore operated in it’s most efficient state.

$$\begin{aligned} \eta &= \eta_{motor} - \eta_{gear\ box} \\ &= 0.169172 - 0.9 \\ &= 0.1522 \end{aligned}$$

The sum of efficiency for the mechanical system is 15%.

Testing of mechanical systems

The mechanical system can operate the arm under no load condition at 12V drawing a current of 0.04A.

After having succeeded with the no load translation the motor was then tested with the 0.5kg load, throughout the transfer the motor drew 0.39A which was the current expected with the calculated value of 0.5A.

The motor was unable to handle the 1kg as the gears began to skip and therefore was unable to complete the full lift of the mass.

Circuit Design

As the structure concept only required one motor the electrical circuit was a very simple setup. As well as this, no electrical system failure had to be included as mechanical failure was implemented instead, failing to lift the 1kg mass.

The circuit only incorporated the DPDT switch so that the motor could be turned on and off and reversed. With no resistors in the circuit there was no voltage drop or power dissipation that had to be accounted for, so the 12V from the power supply was the total voltage supplied to the motor.

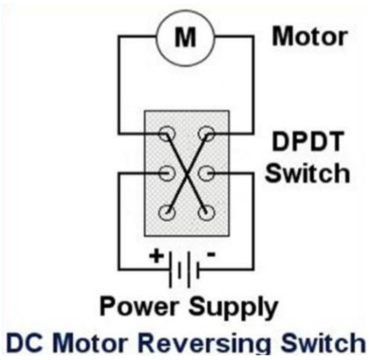

Figure 9: Electric circuit with switch

The Final Design

After several modifications and trials, a final design for the trebuchet was achieved (Figure 10).

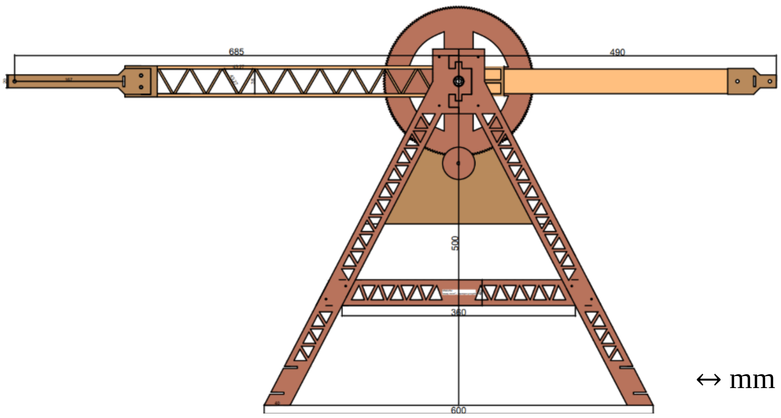

Figure 10: Sideview of the Structure

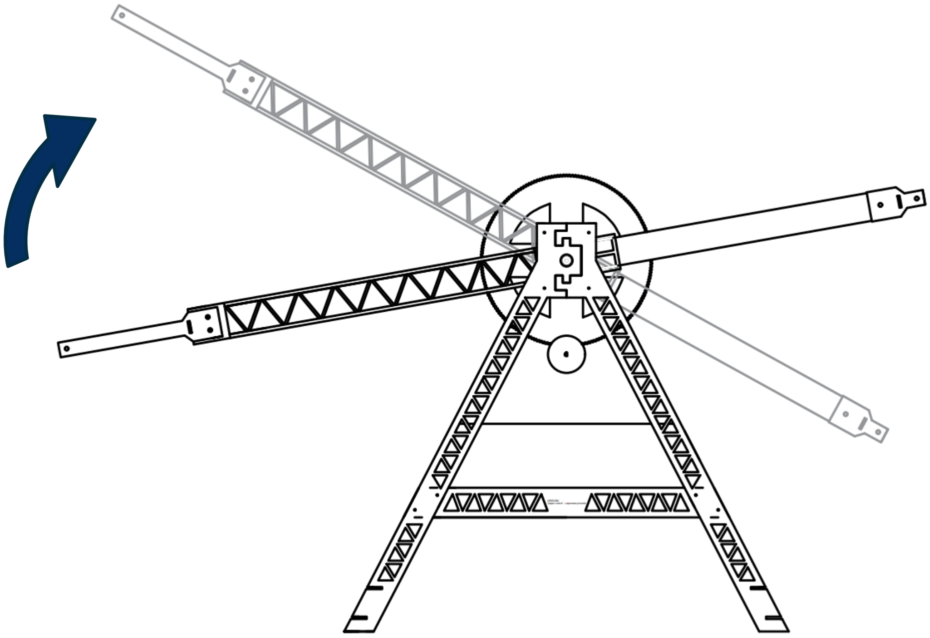

Figure 11: Motion diagram of the structure.

This trebuchet stands at 0.616m from the top of the big gear down to its feet, with the length of the arm being 1175m from one end to another. This main structure consists of 3 main components, the arm, the base and the gears.

The Lifting Arm

The arm of the trebuchet included the fork, truss, and balsa to attach the counterweight. The fork is made of laser cut HDF, a brass rod, washers, and balsa.

The Fork

Laser cut HDF was used for the sides of the fork which were connected to the balsa block with two brass rods for a strong support. Two washers were glued to the centre of the brass rod to ensure that when the load is applied, it remains stable and does not undergo twisting when in motion as it swings through the width of the fork. The balsa block was conveniently used to attach the fork to the truss, and HDF plates to secure the truss and balsa block together (Figure 12).

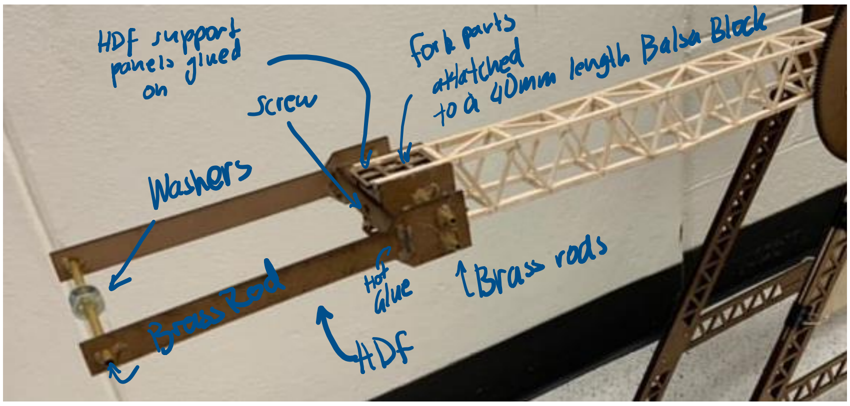

Figure 12: The fork.

This was all secured together with glue to support the load mass and ensure failure does not occur within the fork and rather the truss members.

The Truss

The second component of the arm is the truss. The truss itself was made to be 415mm long but the 5x5mm framing balsa was extended an additional 110mm; 40mm to allow for attachment of the wood and fork at the front of the arm, and an additional 70mm at the end of the arm to attach to the counterweight balsa.

The 5x5mm balsa was also used for the cross bracing. The diagonal bracing as well as the vertical members were made of the 3x3mm balsa. All the truss members were put together with hot glue. The height of the truss was 38mm as the balsa for the counter arm was 38mm in height and 40mm in width.

Using basic trigonometry, the member length to form an equilateral triangle was found. Knowing the height had to be 38mm and the angle was 60°, the member length was calculated to be 45mm.

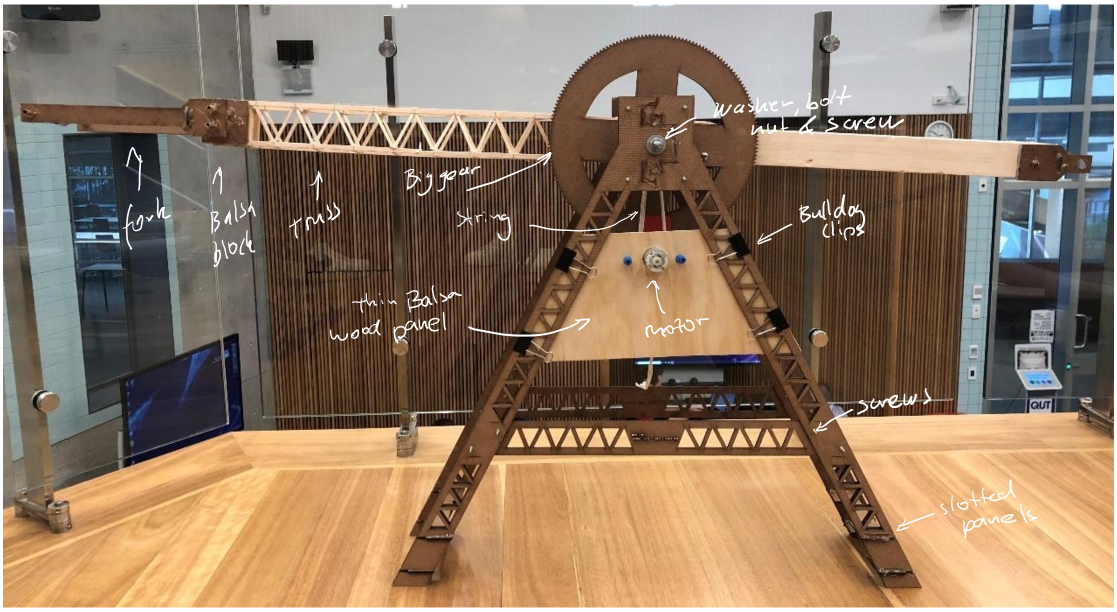

Figure 13: The completed structure.

The Base

The base was made of two layers of laser cut HDF on each side. The laser cut design made the assembly of the structure very simple. The two layers were screwed together with the pre-cut holes and the panels were slotted between the legs to combine the two sides together. The two separate legs were hot glued.

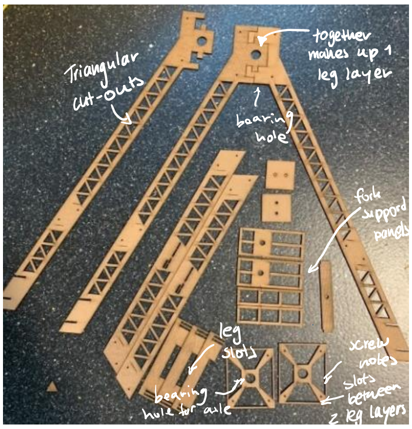

Figure 14: Laser cut components for the base.

The Gears, Motors, and Axle

The 36rpm 12V DC Reversible Gearhead Motor (YG2734) is used as the motor to power the mechanical system. It is securely screwed to the plywood and attached between the thin gap of the base. Due to it's high torque, it causes the whole motor to lean downwards as seen in figure 10, so a string is attached as a fan belt to hold the gear upright while running, ensuring the gears stay in contact.

Two spur gears with teeth size of 6mm were used; a small one attached to the gear has 18 teeth while the large one has 180 teeth, giving a gear ratio 10. The gears were constructed of HDF due to its durability and lightweight material, in comparison to the acrylic. Two layers of 3mm gears were glued together to increase the thickness and increase surface contact area, reducing possibility of skipping. Safety features were added on either side of the smaller gear to prevent the gears from slipping at the teeth. Additionally, holes were added to the larger gear to reduce the weight in order to meet the optimum maximum 1.5kg criteria. Furthermore, the small gear was replaced with new ones before the final testing as damage due to grinding was evident after several trials.

The large gear was attached to a nut which was secured on the threaded axle and screwed in multiple points on the arm to fix the rotation with respect to the arm. The rotating arm was tightly secured on the axle with nuts and washers to prevent slipping when carrying the load.

The nuts and washers were tightened to the axle at the sides of the structure to hold it in position and ensure appropriate room between the two structural base components. It also supports the stability of the structure to prevent significant movement while operating.

In addition, the bearing was added where the axle is attached to the base to ensure smooth rotation of the arm.

Figure 15: The mechanical components including the gears and motor configuration.

Costing Breakdown

Due to time constraints and access to resources components of the structure had to be compromised through aesthetic and strength factors.

Initially the structure was constructed with an 490mm long counter arm to hold a 400g mass. Having the mass placed at this distance gave it a greater moment and therefore the mass had greater effect on the torque required. However, before the test day it was found that the motor would provide enough torque with the gears and therefore the 400g counterweight became obsolete. Due to time constraints and the limited availability of Launchpad, there was no time to remove the arm and to upgrade the base support for stability. It was determined that removing the counter arm allowed for additional cross bracing between the legs as there would be no need to leave room for the arm to swing through them. This would have significantly reduced the twisting experienced in the base during testing.

The shipping time for geared motors was an issue and these requirements of high torque and low speeds were hard to find in local hobby stores. Two motors were ordered online and after testing the Tamiya gearbox motor, significant deformation in the plastic gears was found and therefore caused the motor to skip. Considering this was also for the no load lift this was not a viable option. The low rpm and high torque motor was set to arrive a day before test day; however, due the unreliability of online shipping, the YG2736 motor was used as it was readily available for purchase at Jaycar. Having only two days to test the motor, left little time for laser cutting of the gears and assembling the motor, gears and axle together. Therefore, the motor attachment slightly compromised the aesthetic while using bulldog clips and a plywood panel to secure the motor, which ultimately still fulfilled its structural purpose.

Design Performance Evaluation

There were four optimal requirements that were aimed to be achieved with the final design. During testing all four of these criterions were achieved.

One of the hardest criterions to overcome was the maximum weight limit of 1.5kg. The initial design exceeded this limit due to the additional mass of the counterweight. However, in the final design this was removed due to the fact that, with the gear ratio, the motor provided enough torque to complete the rotation. Ultimately, the final design weighed 1.36kg, which is under the optimal requirement of 1.5kg.

With the gears and voltage supply, the mass moved at a preferable speed, taking 12 seconds in total and remaining well under the 45 second time constraint, as per the optimal design requirement and under the group goal of 30 seconds.

The factor of safety between 1 and 2 was fulfilled through a mechanical failure in the gears. With the 1kg load attached, the gears began to skip and failed at lifting it. However, at this load the structure was observed to almost topple with no counterweight balancing the structure. This aspect can be considered as a potential improvement in the future to ensure maximum security.

The truss was designed with factor of safety between 2 and 3 through the compression calculations for the 45mm member lengths. Calculations revealed that the truss would have a member buckle at the 1.5kg load. As expected, during testing the truss experienced torsion and eventually lead to failure in a member due to buckling.

The additional group goal of creating a unique and innovative design was also fulfilled, despite the many challenges faced throughout the design process. The trebuchet idea concept was pursued and through persistence, successfully constructed.

Recommendations and Conclusions

Whilst the final design structure fulfilled all the criterions and optimal requirements specified in the Project brief, more effective and viable solutions to achieve these are possible through further design and implementation. As previously mentioned, the final design rendered the use of the counterweight obsolete. As a result, the balsa wood arm created with the intention to carry the counterweight proved useless. As such, in order to increase the practicality of the structure, removing the arm and adding additional cross-bracing support to the design can increase both stability and reduce the overall weight of the structure. However, taking into consideration the removal of the arm, the base of the structure must then be altered to increase in size in order to accommodate the loss of weight and reduce the risk of toppling whilst carrying the 1kg and 1.5kg load. An additional improvement to the circuit can also be made. Altering it so that the structure can undergo electrical rather than mechanical failure, through the addition of either fuses or resistors is possible, in order to ensure the 1kg load fails to be lifted.

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Appendix

Appendix A: Material Testing

A1: Compressive Force Testing

Balsa 3mmx 3mm, Material Cross-sectional Area: 9

Length (mm)	Trial 1 (N)	Trial 2 (N)	Trial 3 (N)	Average (N)
40	169.2	167.7	80.9	139.3
45	61.9	78.8	166.2	102.3
50	184.3	136.9	156.9	159.4
55	147.4	110.2	118.5	125.4
60	132.3	131.1	175.4	146.3
65	170.5	148.3	175.4	164.7
70	81.2	57.6	149.5	96.1
75	59.7	52.3	37.9	50.0
80	94.8	98.8	141.2	111.6
85	81.2	97.5	96	91.6
90	124.6	78.2	77.9	93.6

Balsa 5mm x 5mm, Material Cross-sectional Area: 25

Length (mm)	Trial 1 (N)	Trial 2 (N)	Trial 3 (N)	Average (N)
40	517.2	360.3	429.5	435.7
45	379.1	353.5	423.7	385.4
50	536.6	482.7	517.2	512.2
55	461.5	420	348.6	410.0
60	435.7	145.2	396.9	325.9
65	695.9	820.9	459.7	658.8
70	483	621.8	461.8	522.2
75	352.6	480.9	305.5	379.7
80	394.7	118.2	552.2	355.0
85	424.9	491.4	445.2	453.8
90	545.8	411.7	443	466.8

A2: Youngs Modulus Calculations:

Example calculation for 3mmx 3mm balsa with length of 45mm, pin jointed K=1:

$$E = \frac{F_{cr}(KL)^2}{\pi^2 I} = 102.3 \times \frac{(1 \times 45)^2}{\pi^2 \times \frac{3^4}{12}} = 3109.547MPa$$

Length(mm)	Failure load	young modulus
40	139.3	3345.550431
45	102.3	3109.547126
50	159.4	5981.702471

	Average E	**4145.600009**

Calculations for the 5mm x 5mm balsa:

Length(mm)	Failure load	Young modulus
40	435.7	1356.162731
45	385.4	1518.241998
50	512.2	2491.058035
	Average E	**1788.487588**

Appendix B: YG2736 Motor Data Sheet

B1: Data sheet

Performance (in an ambient temperature of 25-30 ° C)
Motor tested rapidly to prevent significant temperature rise.

At a constant voltage of :	12.00	Volts
Direction:	CW	
At No Load		
Speed :	35	RPM
Current :	0.06	AMPS
At stall (Extrapolated)		
Torque :	1372.01	mN-m
Current :	1.693	AMPS
At maximum efficiency		
Efficiency:	65.9	%
Torque :	217.350	mN-m
Speed :	29	RPM
Current :	0.319	AMPS
Output :	2.52	Watts
At maximum Power output		
Output :	4.726	Watts
Torque :	686.005	mN-m
Speed :	19	RPM
Current :	0.82	AMPS
Characteristics		
Torque constant :	840.007	mN-m/AMP
E.M.F. constant :	87.921	mV/rad/sec
Dynamic resistance:	7.0866	Ohms
Motor regulation :	0.03	RPM/mN-m

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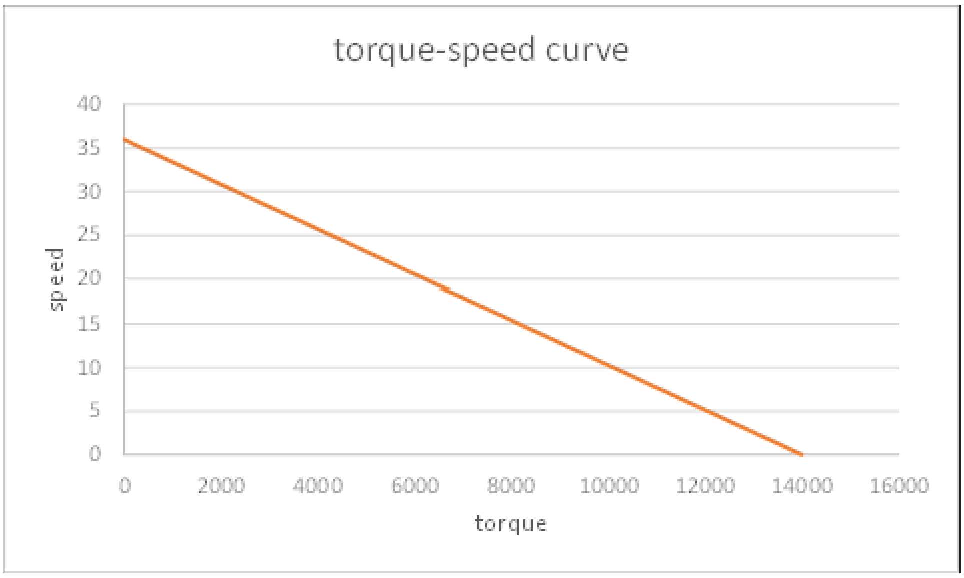
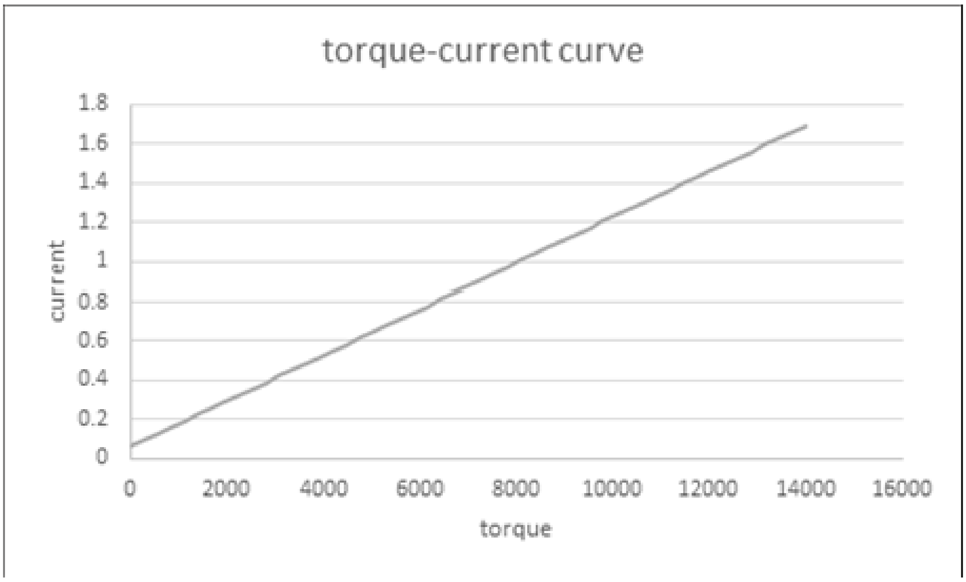
NOMINAL MOTOR CURVES
Performance and characteristics are measured based on limited motor samples only

Appendix C: Motor Curves

C1: Torque speed curve for DC gear motor (YG2736)

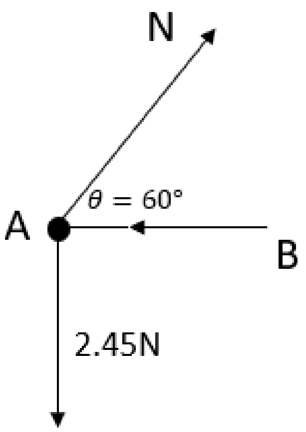


C2: Torque current curve for DC gear motor (YG2736)



Appendix D: Truss Member Analysis

Sample calculation for Joint A:



$$\sum F_y = 0$$

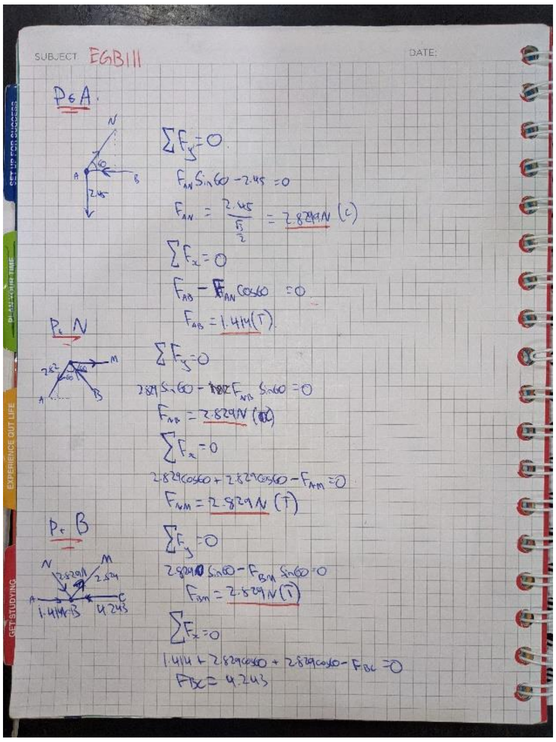
$$F_{AN} = sin60 - 2.45 = 0$$

$$F_{AN} = \frac{2.45}{sin60} = 2.82994N\ (Compressive)$$

$$\sum F_x = 0$$

$$F_{AB} - F_{AN}cos60 = 0$$

$$F_{AB} = 1.414\ (Tensile)$$



SUBJECT: EGB111 DATE:

Pt A

$\sum F_y = 0$

$F_{AN} \sin 60 - 2.45 = 0$

$F_{AN} = \frac{2.45}{\frac{\sqrt{3}}{2}} = 2.829N\ (C)$

$\sum F_x = 0$

$F_{AB} - F_{AN} \cos 60 = 0$

$F_{AB} = 1.414\ (T)$

Pt N

$\sum F_y = 0$

$2.829 \sin 60 - F_{NB} \sin 60 = 0$

$F_{NB} = 2.829N$

$\sum F_x = 0$

$2.829 \cos 60 + 2.829 \cos 60 - F_{NM} = 0$

$F_{NM} = 2.829N\ (T)$

Pt B

$\sum F_y = 0$

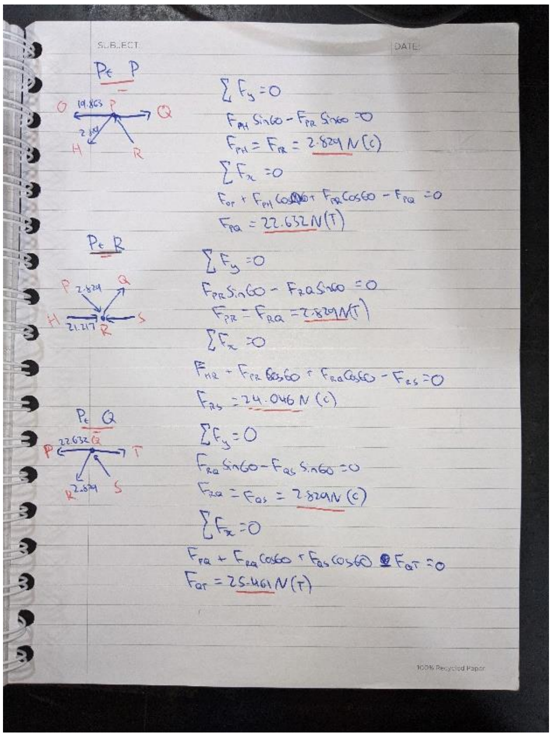
$2.829 \sin 60 - F_{BM} \sin 60 = 0$

$F_{BM} = 2.829N\ (T)$

$\sum F_x = 0$

$1.414 + 2.829 \cos 60 + 2.829 \cos 60 - F_{BC} = 0$

$F_{BC} = 4.243$



SUBJECT: DATE:

Pt P

$\sum F_y = 0$

$F_{PH} \sin 60 - F_{PR} \sin 60 = 0$

$F_{PH} = F_{PR} = 2.829N\ (C)$

$\sum F_x = 0$

$F_{OP} + F_{PH} \cos 60 + F_{PR} \cos 60 - F_{PQ} = 0$

$F_{PQ} = 22.632N\ (T)$

Pt R

$\sum F_y = 0$

$F_{PR} \sin 60 - F_{RQ} \sin 60 = 0$

$F_{PR} = F_{RQ} = 2.829N\ (T)$

$\sum F_x = 0$

$F_{HR} + F_{PR} \cos 60 + F_{RQ} \cos 60 - F_{RS} = 0$

$F_{RS} = 24.046N\ (C)$

Pt Q

$\sum F_y = 0$

$F_{RQ} \sin 60 - F_{QS} \sin 60 = 0$

$F_{RQ} = F_{QS} = 2.829N\ (C)$

$\sum F_x = 0$

$F_{PQ} + F_{RQ} \cos 60 + F_{QS} \cos 60 - F_{QT} = 0$

$F_{QT} = 25.461N\ (T)$

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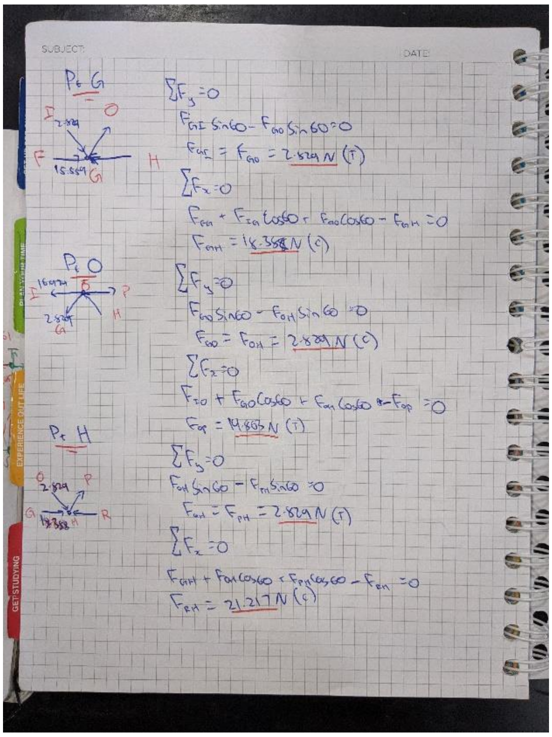


SUBJECT: DATE:

Pt G

$\sum F_y = 0$

$F_{GI} \sin 60 - F_{GO} \sin 60 = 0$

$F_{GI} = F_{GO} = 2.829N\ (T)$

$\sum F_x = 0$

$F_{FG} + F_{IG} \cos 60 + F_{GO} \cos 60 - F_{GH} = 0$

$F_{GH} = 18.388N\ (C)$

Pt O

$\sum F_y = 0$

$F_{GO} \sin 60 - F_{OH} \sin 60 = 0$

$F_{GO} = F_{OH} = 2.829N\ (C)$

$\sum F_x = 0$

$F_{IO} + F_{GO} \cos 60 + F_{OH} \cos 60 - F_{OP} = 0$

$F_{OP} = 19.863N\ (T)$

Pt H

$\sum F_y = 0$

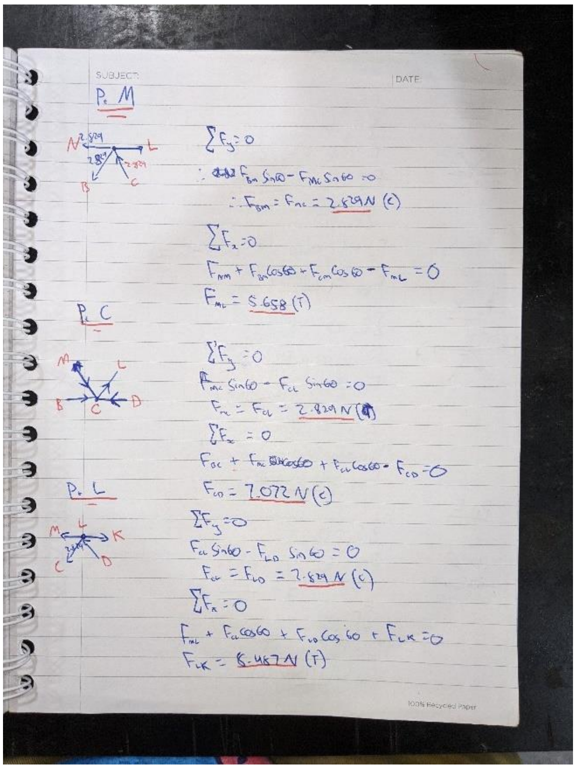
$F_{OH} \sin 60 - F_{PH} \sin 60 = 0$

$F_{OH} = F_{PH} = 2.829N\ (T)$

$\sum F_x = 0$

$F_{GH} + F_{OH} \cos 60 + F_{PH} \cos 60 - F_{RH} = 0$

$F_{RH} = 21.217N\ (C)$



SUBJECT: DATE:

Pt M

$\sum F_y = 0$

$\therefore F_{BM} \sin 60 - F_{MC} \sin 60 = 0$

$\therefore F_{BM} = F_{MC} = 2.829N\ (C)$

$\sum F_x = 0$

$F_{NM} + F_{BM} \cos 60 + F_{CM} \cos 60 - F_{ML} = 0$

$F_{ML} = 5.658\ (T)$

Pt C

$\sum F_y = 0$

$F_{MC} \sin 60 - F_{CL} \sin 60 = 0$

$F_{MC} = F_{CL} = 2.829N$

$\sum F_x = 0$

$F_{BC} + F_{MC} \cos 60 + F_{CL} \cos 60 - F_{CD} = 0$

$F_{CD} = 7.072N\ (C)$

Pt L

$\sum F_y = 0$

$F_{CL} \sin 60 - F_{LD} \sin 60 = 0$

$F_{CL} = F_{LD} = 2.829N\ (C)$

$\sum F_x = 0$

$F_{ML} + F_{CL} \cos 60 + F_{LD} \cos 60 + F_{LK} = 0$

$F_{LK} = 8.487N\ (T)$

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