MONASH University

MALAYSIA

MEC3416 ENGINEERING DESIGN II

FINAL REPORT SUBMISSION

TEAM NUMBER : ST35

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TEAM MEETING SESSION : WED 12-1PM

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EXECUTIVE SUMMARY

This report provides a detailed process of designing a hoist, with the purpose of lifting and stacking concrete cubes with a volume of 1m^3 to generate and store energy, while having the ability to have translation motion along an I-beam fixed to the lower edge of a crane. The hoist is designed to be made up of components that are either purchased or manufactured.

Free Body Diagrams of parts that are critically loaded are illustrated in this report to show the forces acting on them, with relevant calculations included. The components are then put through the Design Analysis Framework, where they are analysed using various methods such as tensile stress analysis, shear stress analysis and bending stress analysis, based on an estimation on the source of the components' possible failure. Components with complex geometrical features that are tough to be analysed through hand calculations are put through the Finite Element Analysis on the Solidworks software. With respect to relevant formulas and data sheets, the minimum safety factor of the critical components that make up the hoist can be found from the Design Analysis Framework. Upon analysing, the components are optimised to have a minimum safety factor of above and as close as possible to 5. The optimisation of these components also takes into consideration reducing the cost of each individual component.

While taking into account the costing, tolerances and suitable motors and alternator to be selected to operate the hoist, a hoist that is of optimal functionality, efficiency, safety and cost effectiveness is produced. This report also includes assembly and detail drawings of the hoist and several of its components, along with relevant catalogues of purchased parts used to build the hoist, found in the appendices section.

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1.0 INTRODUCTION

1.1 BACKGROUND OF SOLUTIONS

Humans as a species have been daily consuming enormous amounts of energy in the form of electricity, with the purpose of completing various tasks. In recent days, electricity is utilised in homes, buildings, transportation, manufacturing and for just about any major or minor activity possible. This has led to an increase in electricity consumption globally, year by year. On average, the global electricity consumption rate increases by 3% each year, and in the year 2018 itself, the total global electricity consumption was 22315 terawatt-hours, which is 4.0% higher than that recorded in the year 2017. [1]

Therefore, experts are in constant search of ways to produce reliable and sustainable forms of energy to keep up with the increasing amount of energy consumption globally. A solution to this is to design a contraption that would allow concrete cubes to be lifted and stacked in order to provide an outlet to store energy in the form of an ‘energy vault’. The energy can then be expelled by the contraption when needed and stored when excess energy is produced.

The objective of this project is to demonstrate this contraption in the form of a hoist system which can travel horizontally along an I-beam fixed to the lower edge of a crane and can also hoist $1m^3$ concrete cubes vertically. This is also while ensuring that the hoist module is able to lift the concrete cubes to a height of up to 50 metres. The hoist module consists of two motors, an alternator and includes several shafts, bearings, gears and pulley drums. In the designing process of the hoist, the safety factor of the selected hoist components was kept above a minimum value of 5, while the costs of individual components in the hoist were adjusted so that they are minimal.

This report outlines in detail the design and capabilities of the hoist module and all manufacturing cost estimates, all while adhering to the above limitations of the design.

1.2 <u>STACKING STRATEGY</u>

In this project, a staggered stacking method is used to create stacks made up of 15 concrete cubes each to generate and store energy. When operating at 8 hours/day, a total of 53.963 Megajoules of energy can be generated using this stacking strategy. The relevant calculations done to obtain the amount of energy generated using this stacking strategy can be found below:

STACKING STRATEGY ASSUMPTIONS:

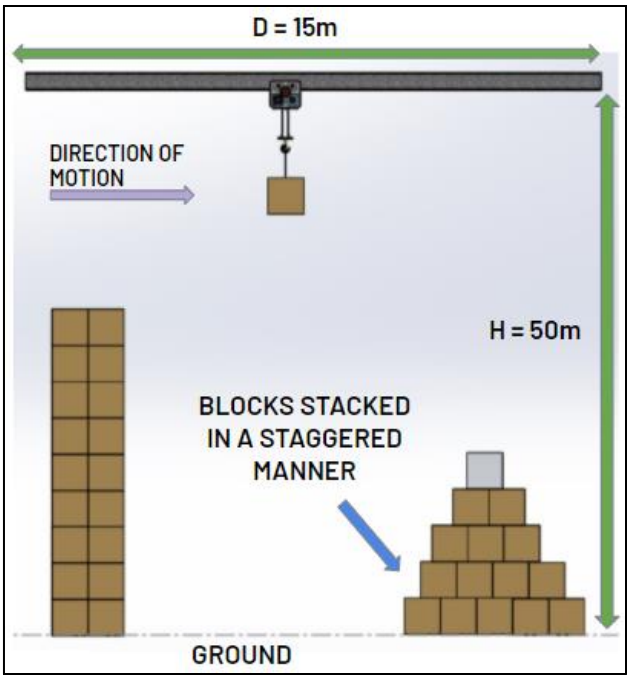


Figure 1: Concrete Loads Stacked in A Staggered Manner

- Volume of concrete load is 1 m^3
- Density of concrete is 2400 kg/m^3
- Gravitational acceleration is 9.81 m/s^2
- Translation distance of hoist on the I-beam, D
  = 15 m
- Vertical height of the I-beam from the ground, H
  = 50 m
- Mass of a single concrete cube
  = ρ x V
  = 2400 kg
- Stacking method
  = Staggered stacking
- Total number of cubes in a single stack
  = 15 cubes
- Stacking Operational Hours
  = 8 hours / day

HOISTING SPEED CALCULATIONS:

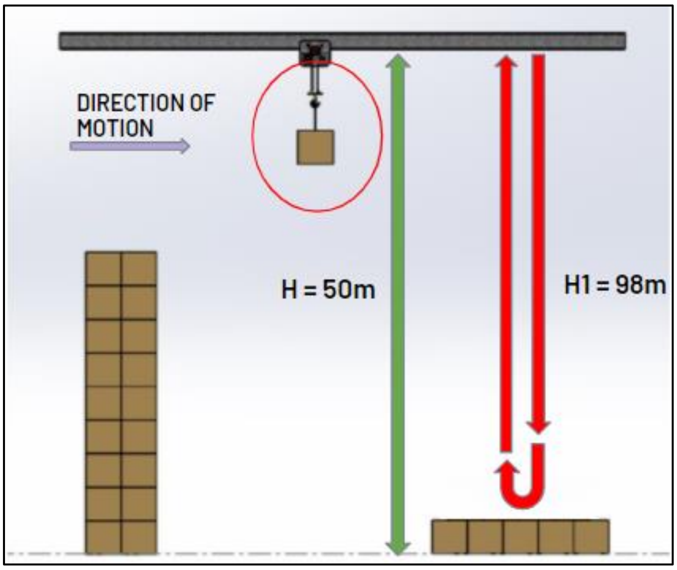


Figure 2: Vertical Hoisting of Concrete Cube Loads

- Vertical Distance Travelled by Hoist Hook to Hoist One Concrete Cube, H1:
  = 49 m + 49 m
  = 98 m
- Hoisting Speed:
  = 40 rpm
  = 0.461 m/s
- Estimated Time Taken to Attach & Detach the Hook to One Concrete Cube
  = 5 mins
- Estimated Time Taken to Vertically Hoist One Concrete Cube:
  = 98 m / 0.461 m/s
  ≈ 212.69 s
  ≈ 3.54 mins

TRANSLATING SPEED CALCULATIONS:

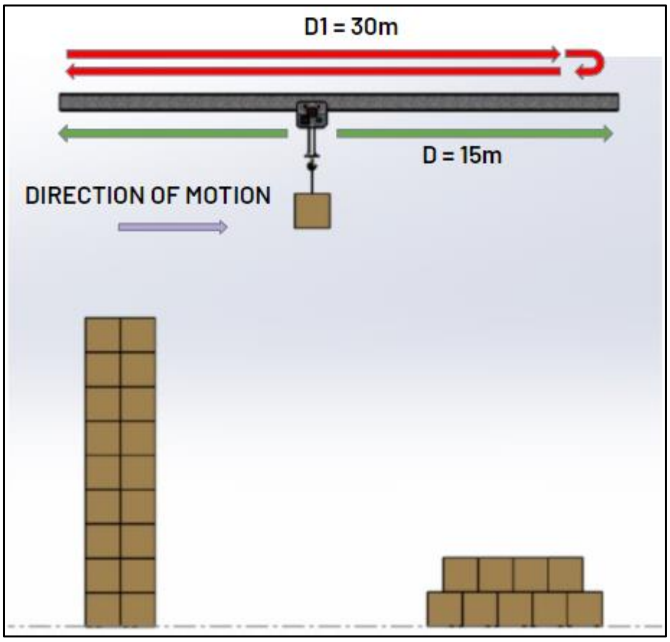


Figure 3: Horizontal Translation of Hoist When Stacking Concrete Cube Load

- Horizontal Distance Travelled by Hoist Hook to Hoist One Concrete Cube, D1:
  = 15m + 15m
  = 30m

- Translation Speed
  = 30 rpm x (2π /60) x 0.115 m
  = 0.361 m/s

- Estimated Time Taken to Horizontally Hoist One Concrete Cube:
  = 30m / 0.361 m/s
  ≈ 83.102 s
  ≈ 1.39 mins

ENERGY STORAGE CALCULATIONS:

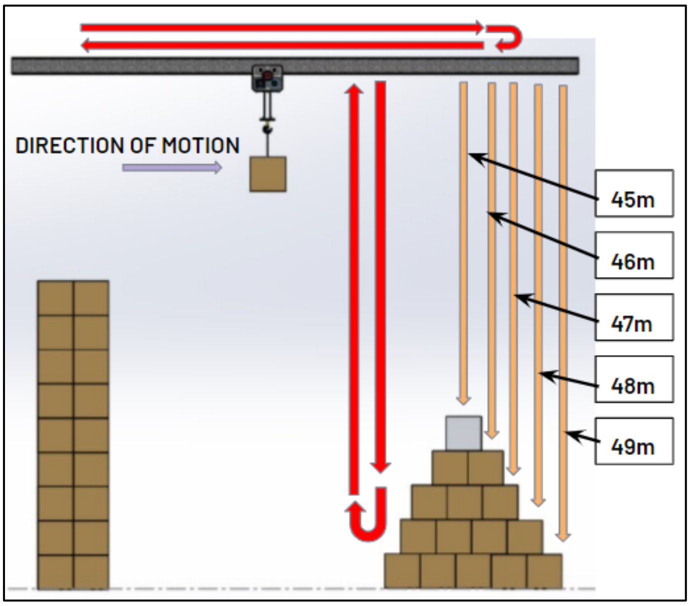


Figure 4: Gain in Height By Concrete Cube Loads When Stacked Staggered

- Total Time Taken to Hoist One Concrete Cube
  = 3.54 mins + 5 mins + 1.39 mins
  = 9.93 mins

- Number of Concrete Cubes Hoisted Within 8 Hours (480 minutes):
  = 480 mins / 9.93 mins
  ≈ 48 Concrete Cubes Hoisted

- Number of Stacks of Cubes That Can Be Made Within 8 Hours:
  = 48 Concrete Cubes / (15 Concrete Cubes/Stack)
  = 3.2 Stacks
  = 3 Complete Stacks & 1 Incomplete Stack

- Total Gain in Height by Blocks that Leads to the Generation of Potential Energy, H:
  = Gain in Height During First Stack + Gain in Height During Second Stack + Gain in Height During Third Stack + Gain in Height During Fourth Stack
  = 3 x [(5x49m) + (4x48m) + (3x47m) + (2x46m) + (1x45m)] + (3x49m)
  = 2292 m

- Estimated Potential Energy Gained in 8 hrs/day:
  = m x g x H
  = 2400 kg x 9.81 m/s² x 2292 m
  = 53962848 J
  = 53.963 MJ

2.0 DESIGN OVERVIEW

GENERAL ASSUMPTIONS

- ➢ Load hoisted are $1m^3$ concrete cube
- ➢ Density of concrete is 2400 kg/m^3
- ➢ Frictional force of moving parts is negligible
- ➢ Gravitational acceleration is 9.81 m/s^2
- ➢ Materials are homogenous and does not have any manufacturing defects

DESIGNED SUBSYSTEMS

- **TRANSLATION MECHANISM**
  - ➢ Utilises 2x2 wheels connected to 5 shafts and 6 gears to move horizontally along a single I-beam, powered by a motor.
- **HOISTING MECHANISM**
  - ➢ Utilises 2 separate cables connected to a single hook on one end, and two pulley drums on another to hoist the concrete cube loads.
  - ➢ Pulley drums are connected to 2 shafts and 2 gears, with one of the shafts connected to another motor and an alternator on each end to power the hoisting mechanism.
- **HOIST CASING**
  - ➢ Custom-made casings are used to build the body of the hoist, and bolts and nuts are used to secure them in place, while bearings and circlips are used when attaching the hoist casing onto the shafts of the translation and hoisting mechanisms.

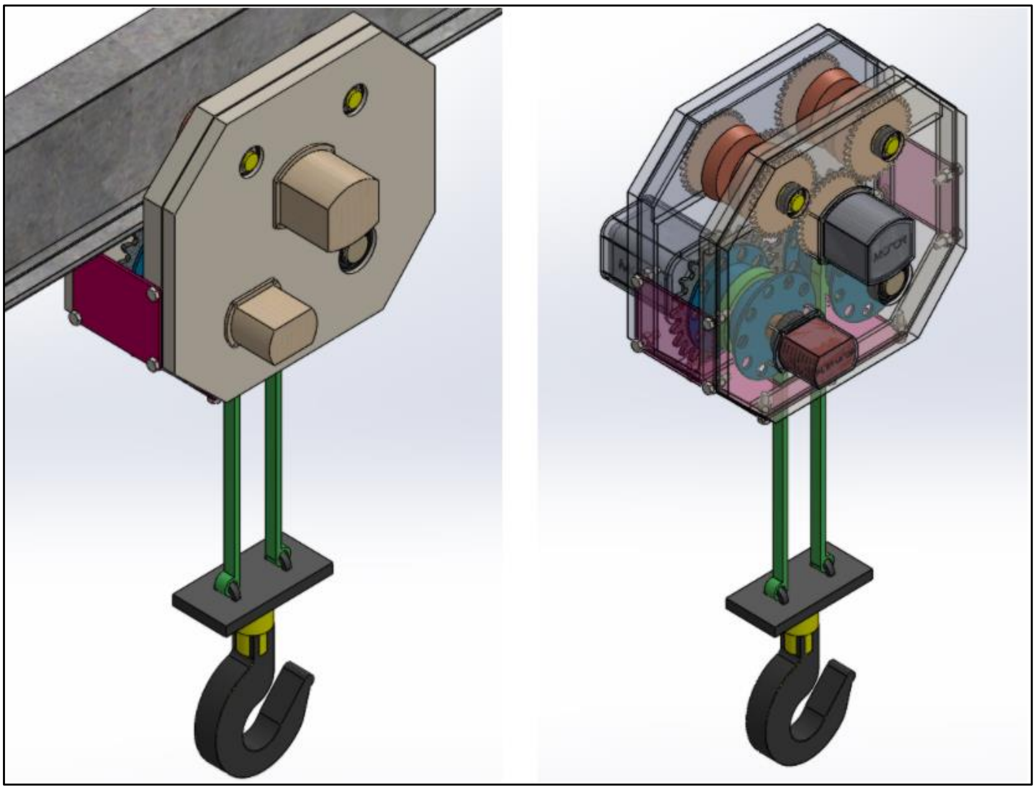


Figure 5: Isometric View of Overall Hoist Design

2.1 SUBSYSTEM 1: TRANSLATION MECHANISM

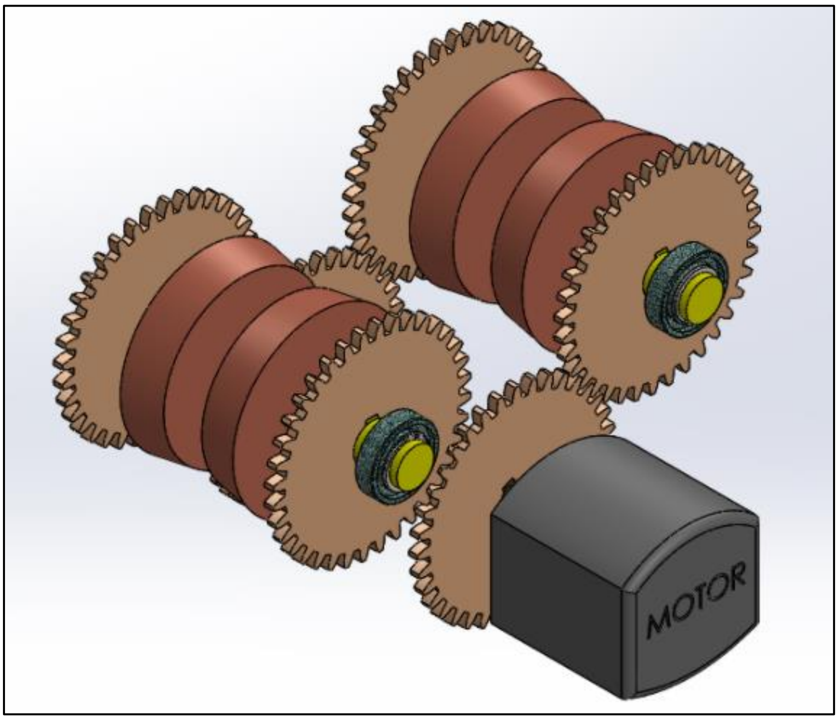


Figure 6: Isometric View of Translation Mechanism Subsystem

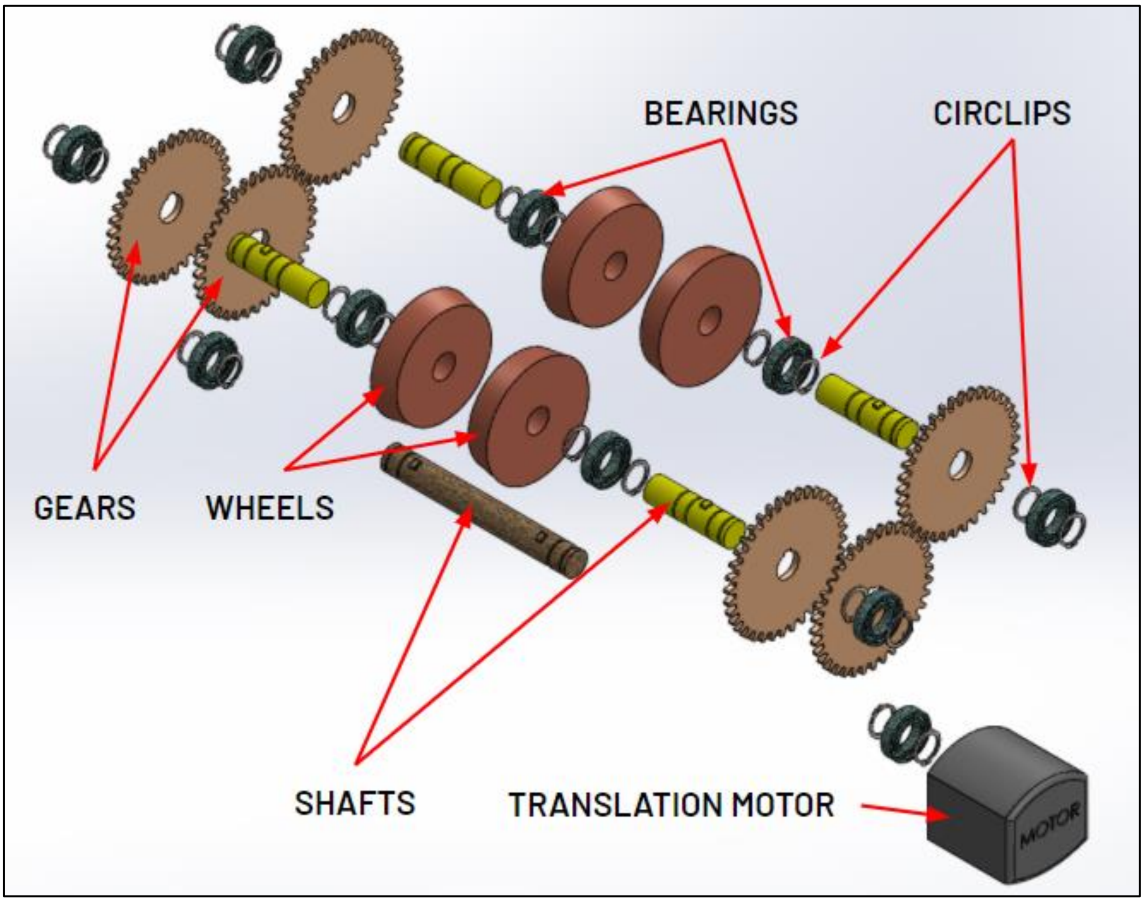


Figure 7: Exploded View of Translation Mechanism Subsystem

2.2 SUBSYSTEM 2: HOISTING MECHANISM

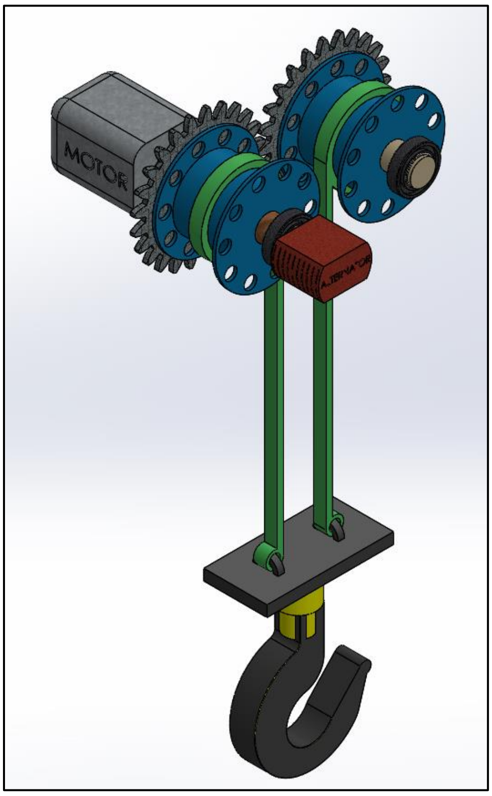


Figure 8: Isometric View of Hoisting Mechanism Subsystem

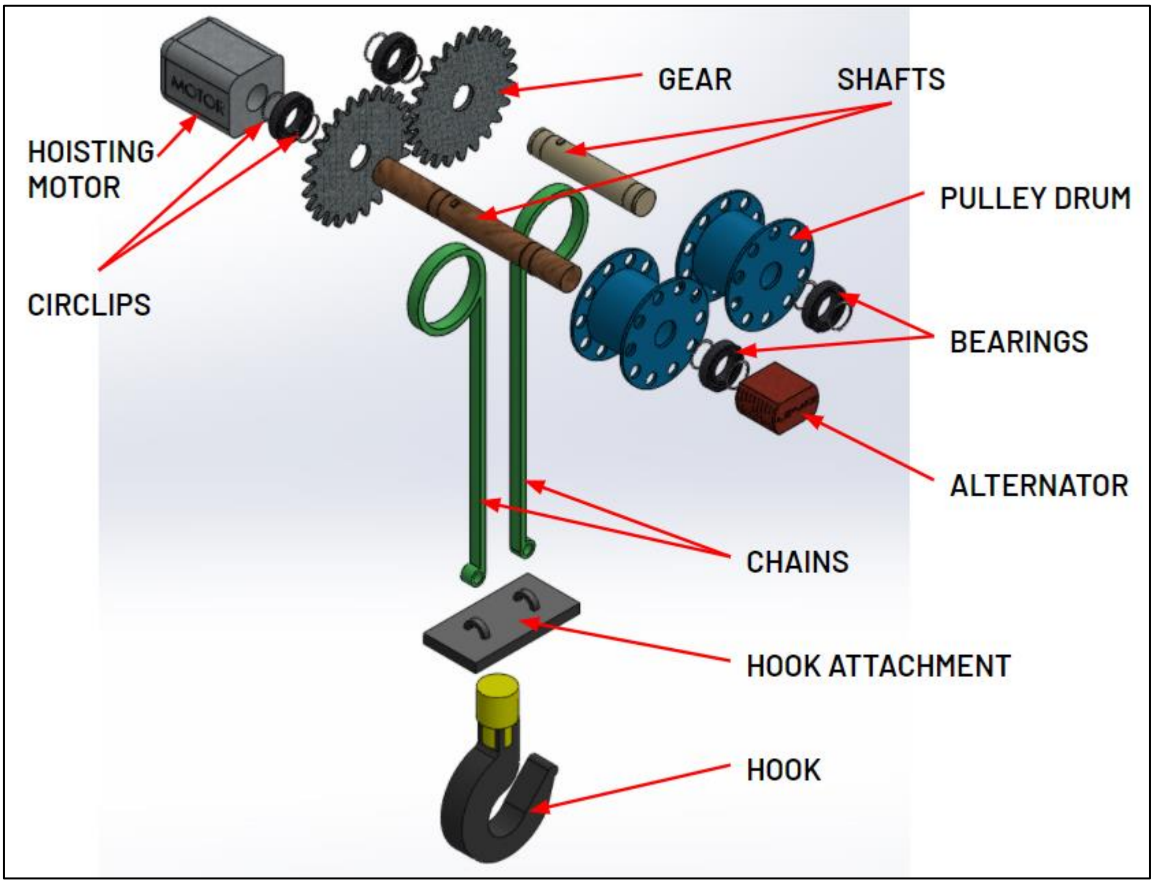


Figure 9: Exploded View of Hoisting Mechanism Subsystem

2.3 SUBSYSTEM 3: HOIST CASING

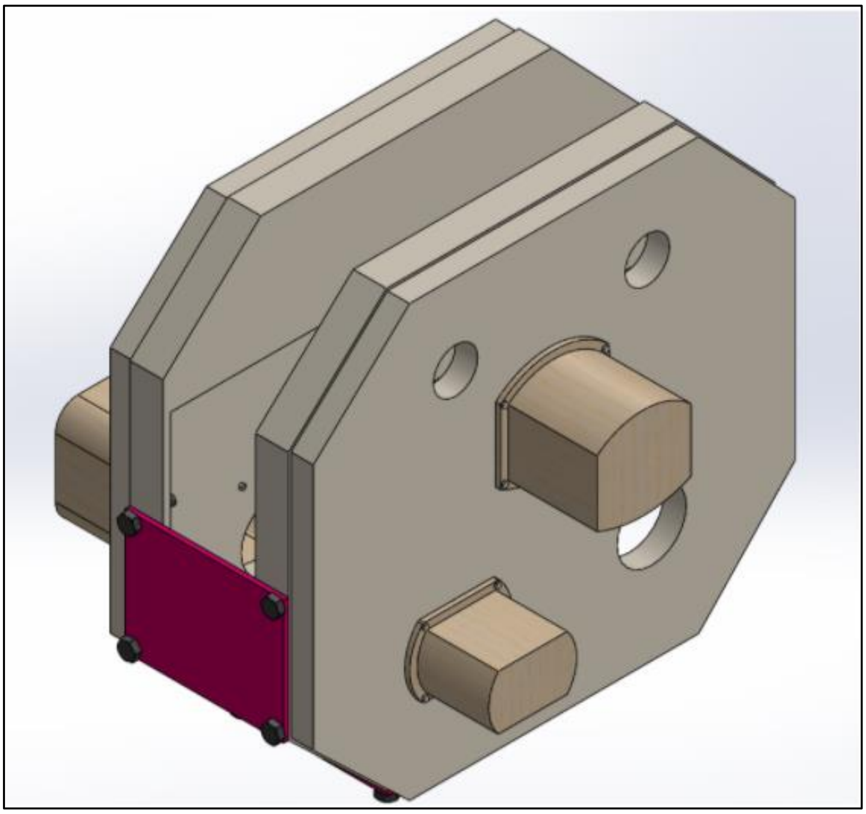


Figure 10: Isometric View of Hoist Casing Subsystem

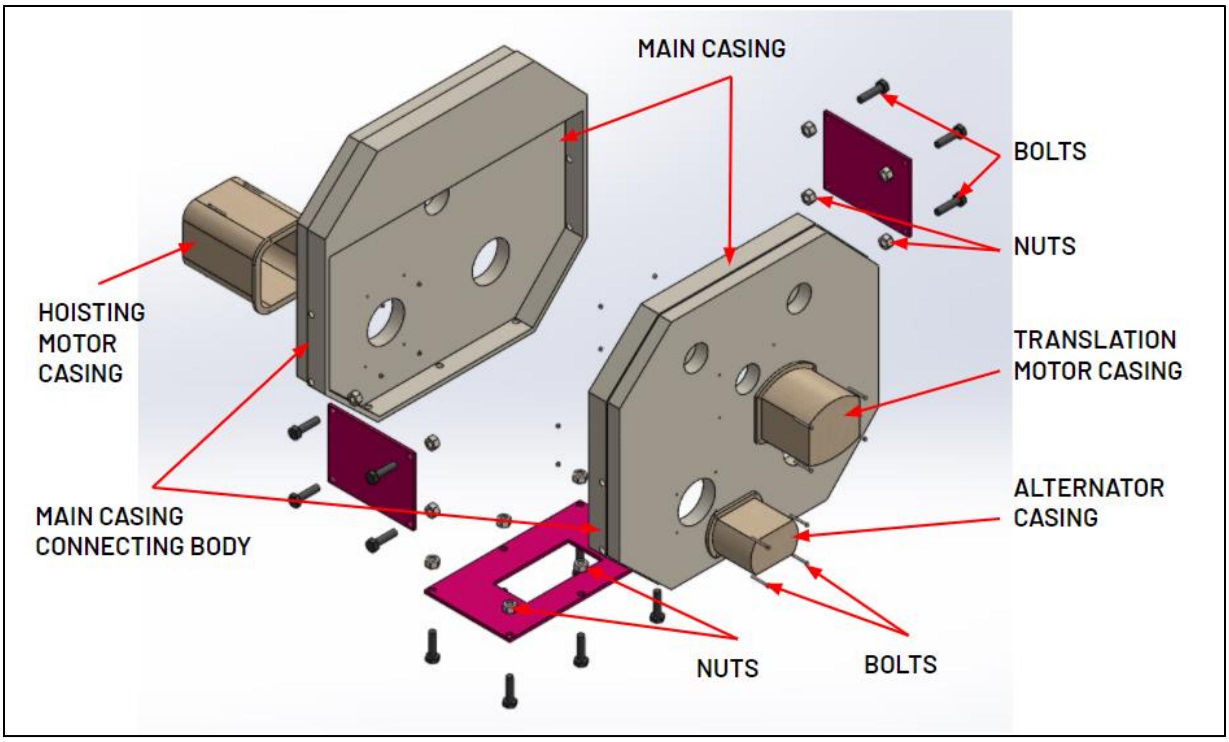


Figure 11: Exploded View of Hoist Casing Subsystem

3.0 HOOK

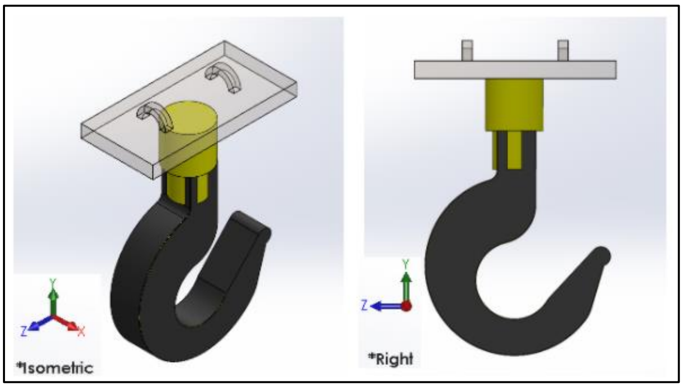


Figure 12: Optimised Hook Designed to Lift 1m³ Concrete Blocks

A hook was designed to lift $1m^3$ concrete blocks with a mass of 2400kg each. It is attached to the load before lifting up the load and released from the load once the load is placed at its intended location. The hook is made up of AISI 4340 Annealed Steel with a material density of 7850 kg/m^3. Upon optimisation, the thickness of the hook is increased from 6.85mm to 26.85mm.

3.1 <u>FREE BODY DIAGRAM (FBD) - OPTIMISED HOOK</u>

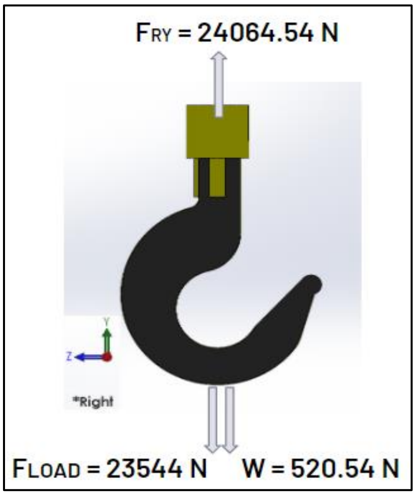


Figure 13: Free Body Diagram of Optimised Hook

(Free body diagram and corresponding calculations of the initially selected hook can be found in Appendix D)

3.1.1 FREE BODY DIAGRAM CALCULATIONS - OPTIMISED HOOK

$Assumptions$:

- $Volume\ of\ Concrete\ Block\ Load\ =\ 1\ m^3$
- $Volume\ of\ Optimised\ Hook\ =\ 0.00675955\ m^3$
- $Density\ of\ Hook\ Material\ (Annealed\ AISI\ 4340\ Steel) =\ 7850\ kg/m^3$

$$\begin{aligned} Mass\ of\ Concrete\ Block\ Load &= m_1 \\ m_1 &= \rho \times V \\ &= 2400\ kg/m^3\ x\ 1\ m^3 \\ &= 2400\ kg \end{aligned}$$

$$\begin{aligned} Mass\ of\ Initially\ Designed\ Hook &= m_2 \\ m_2 &= \rho \times V \\ &= 7850\ kg/m^3 \times 0.00675955\ m^3 \\ &= 53.062\ kg \end{aligned}$$

$$\begin{aligned} Weight\ of\ Load\ Applied\ on\ Optimised\ Hook &= F_{LOAD} \\ F_{LOAD} &= m_1 \times g \\ &= 2400\ kg \times 9.81\ m/s^2 \\ &= 23544\ N \end{aligned}$$

$$\begin{aligned} Self-Weight\ of\ Optimisef\ Hook &= W \\ W &= m_2 \times g \\ &= 53.062\ kg \times 9.81\ m/s^2 \\ &= 520.54\ N \end{aligned}$$

$$\begin{aligned} F_{RY} &= W + F_{LOAD} \\ &= 24064.54\ N \end{aligned}$$

3.2 DESIGN ANALYSIS FRAMEWORK 1 (DAF)

3.2.1 FINITE ELEMENT ANALYSIS (FEA)

Due to the complex geometrical shape of the hook, the forces on the hook were tough to be analysed using hand calculation. Thus, the Finite Element Analysis (FEA) is run onto the initially designed part. Through FEA, it is found that the maximum stress undergone by the hook is 402.0 MPa, as shown in the Von Mises Stress Plot in Figure 14 below. The corresponding minimum safety factor of the hook design is 1.169, as shown in the Safety Factor Plot of the hook, in Figure 15 below.

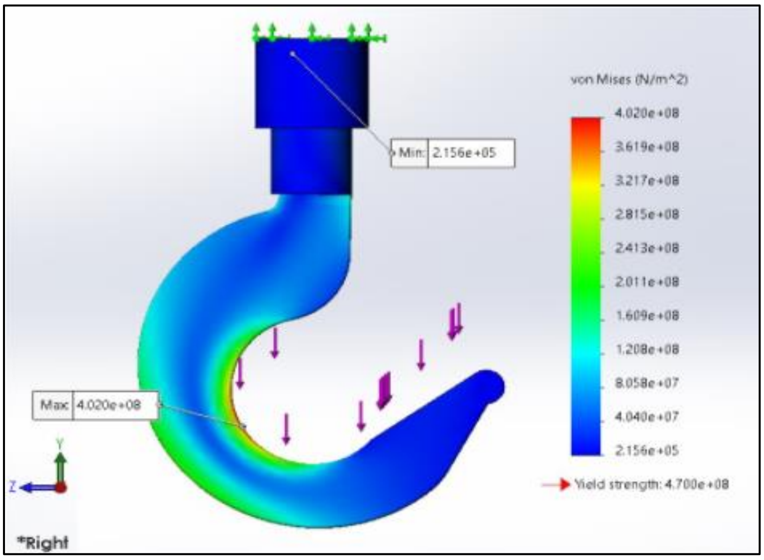


Figure 14: Von Mises Stress Plot of Initially Designed Hook

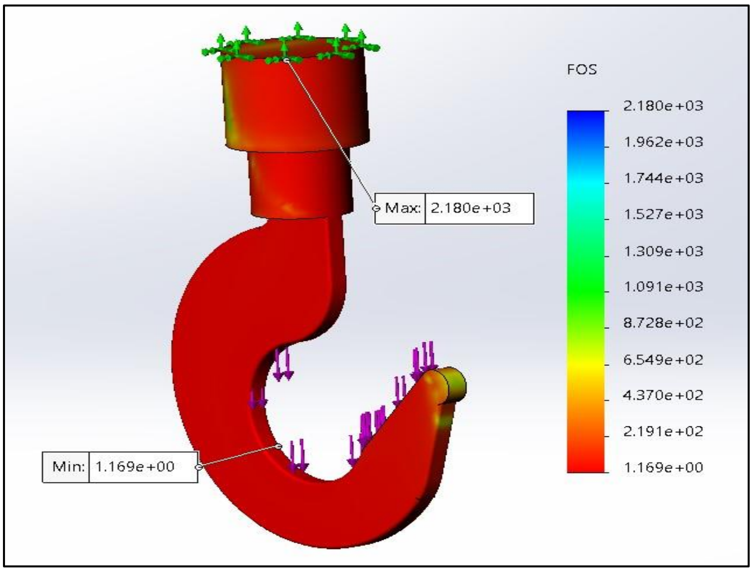


Figure 15: Safety Factor Plot of Initially Designed Hook

3.3 OPTIMISATION

From the initial design of the hook, it is found that the minimum safety factor of the hook is 1.169, which is extremely low. To achieve a safety factor of about 5, a modification is made onto the hook's design, where the thickness of the hook is increased from 6.85mm to 26.85mm, as shown in Figure 16 below.

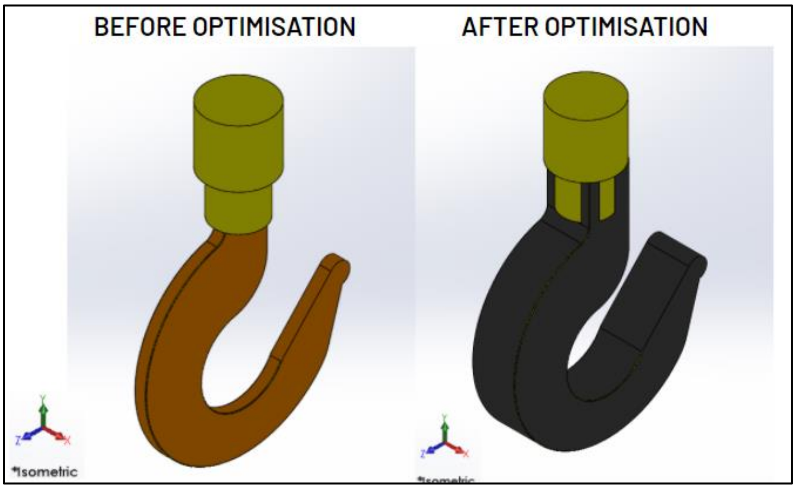


Figure 16: Comparison of Hook Size Before & After Optimisation

Once the thickness of the hook was increased from 6.85mm to 26.85mm, the Finite Element Analysis (FEA) was again run onto the part. Through FEA, it is found that the maximum stress undergone by the optimised hook is now 93.08 MPa, as shown in the Von Mises Stress Plot in Figure 17 below. The corresponding minimum safety factor of the hook design is 5.006, as shown in the Safety Factor Plot of the hook, in Figure 18 below. This satisfies the current requirement to produce a safety factor of as close as possible to 5.

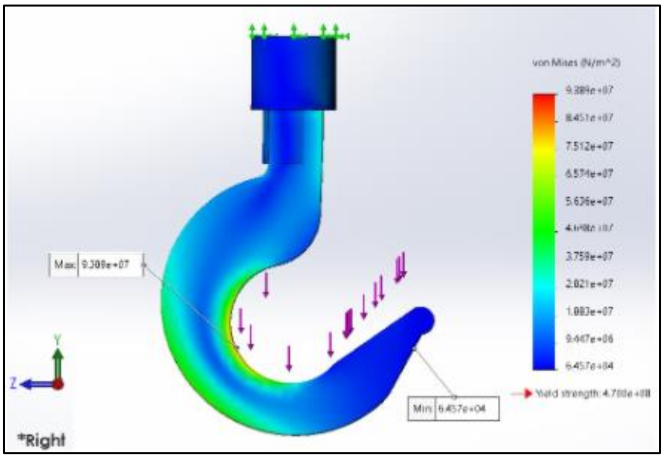


Figure 17: Updated Von Mises Stress Plot of Optimised Hook

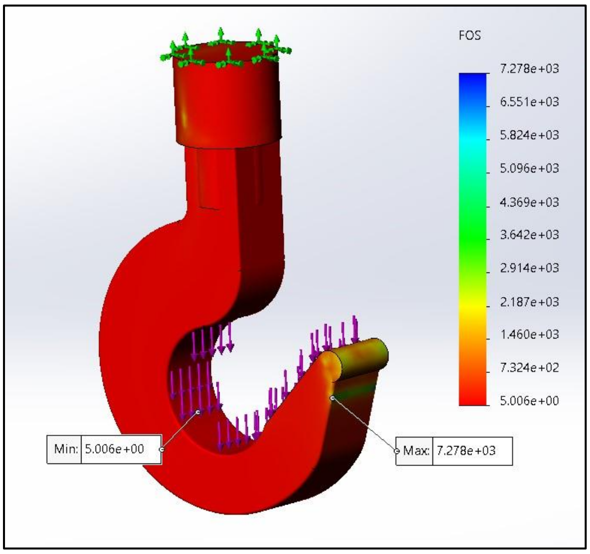


Figure 18: Updated Safety Factor Plot of Optimised Hook

3.4 <u>COMPONENT COSTING</u>

Pure AISI 4340 Steel for casting purposes delivered to Malaysia from Alibaba.com costs about RM7.90/kg including delivery charges. An estimate of RM200 is charged to design a custom mould for the casting process, and an RM100/hour rate is charged for engineering work such as melting, casting and annealing processes done to produce the hook. According to the costing analysis report obtained via the Solidworks software, the estimated price to fabricate the hook is RM891.79. This costing analysis report, along with pricing details of the material used can be found in Appendix C.

4.0 HOOK ATTACHMENT

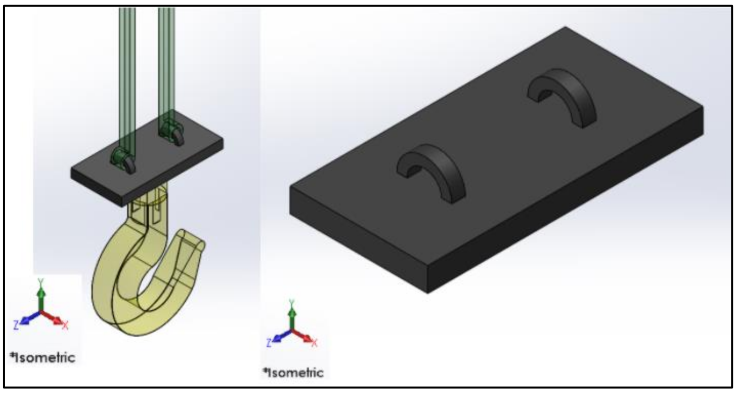


Figure 19: Optimised Hook Attachment Connected to A Hook & Two Chains

A hook attachment is designed to be welded to a hook at its bottom surface and attached to two chains on its top surface. This hook attachment helps the load supported by the hook to be propagated equally along two chains to hoist the load vertically. The hook attachment is made from AISI 4340 Annealed Steel with a material density of 7850 kg/m³. The hook attachment is subjected to tensile stress, shear stress and bending stress. Upon optimisation, a slight change is made onto the geometrical shape of the two handles present on the top surface of the hook attachment to reduce the force on the corners of the handles. The initial design of the hook attachment before optimisation is as below in Figure 20.

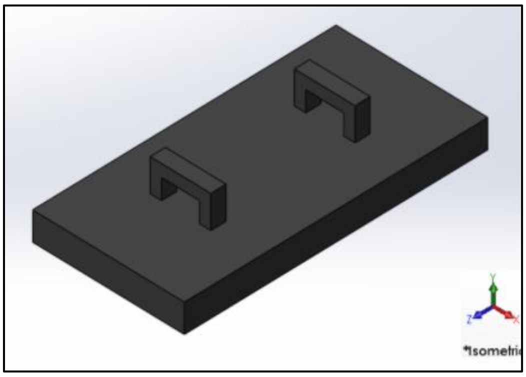


Figure 20: Initial Design of Hook Attachment

4.1 FREE BODY DIAGRAM (FBD)- OPTIMISED HOOK ATTACHMENT

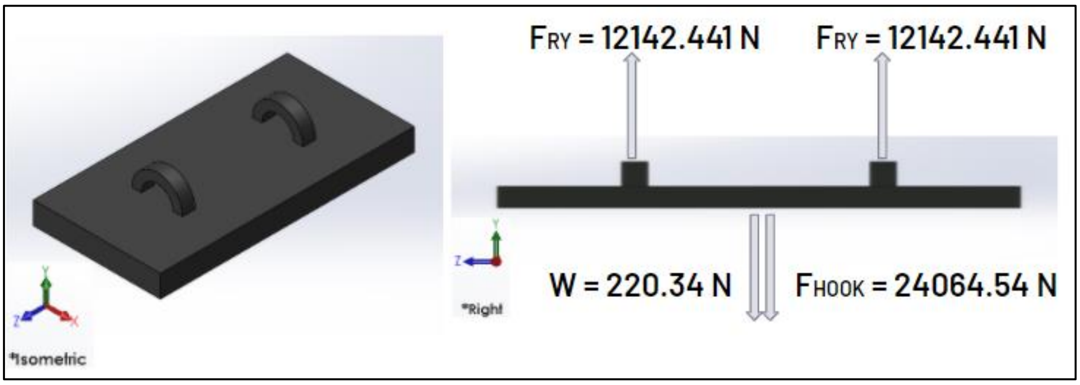


Figure 21: Free Body Diagram of Optimised Hook Attachment

(Free body diagram and corresponding calculations of the initially selected hook attachment can be found in Appendix D)

4.1.1 FREE BODY DIAGRAM CALCULATIONS - OPTIMISED HOOK ATTACHMENT

$Assumptions$:

- $Volume\ of\ Optimised\ Hook\ Attachment\ = 0.00286126\ m^3$
- $Density\ of\ Hook\ Attachment\ Material\ (Annealed\ AISI\ 4340\ Steel) =\ 7850\ kg/m^3$

$$\begin{aligned} Mass\ of\ Initially\ Designed\ Hook\ Attachment\ &=\ m \\ m\ &=\ \rho\ \times\ V \\ &=\ 7850\ kg/m^3\ \times\ 0.00286126\ m^3 \\ &=\ 22.461\ kg \end{aligned}$$

$$\begin{aligned} Self - Weight\ of\ Initially\ Designed\ Hook\ Attachment &=\ W \\ W\ &=\ m\ \times\ g \\ &=\ 22.461\ kg\ \times\ 9.81\ m/s^2 \\ &=\ 220.341\ N \end{aligned}$$

$$F_{HOOK}\ =\ 24064.540\ N$$

$$W +\ F_{HOOK}\ =\ 24284.881\ N$$

$$\begin{aligned} F_{RY}\ &= (W +\ F_{HOOK})\ /\ 2 \\ &=\ 24284.881\ N\ /\ 2 \\ &=\ 12142.441\ N \end{aligned}$$

4.2 DESIGN ANALYSIS FRAMEWORK 2 (DAF)

4.2.1 SHEAR FORCE & BENDING MOMENT DIAGRAM

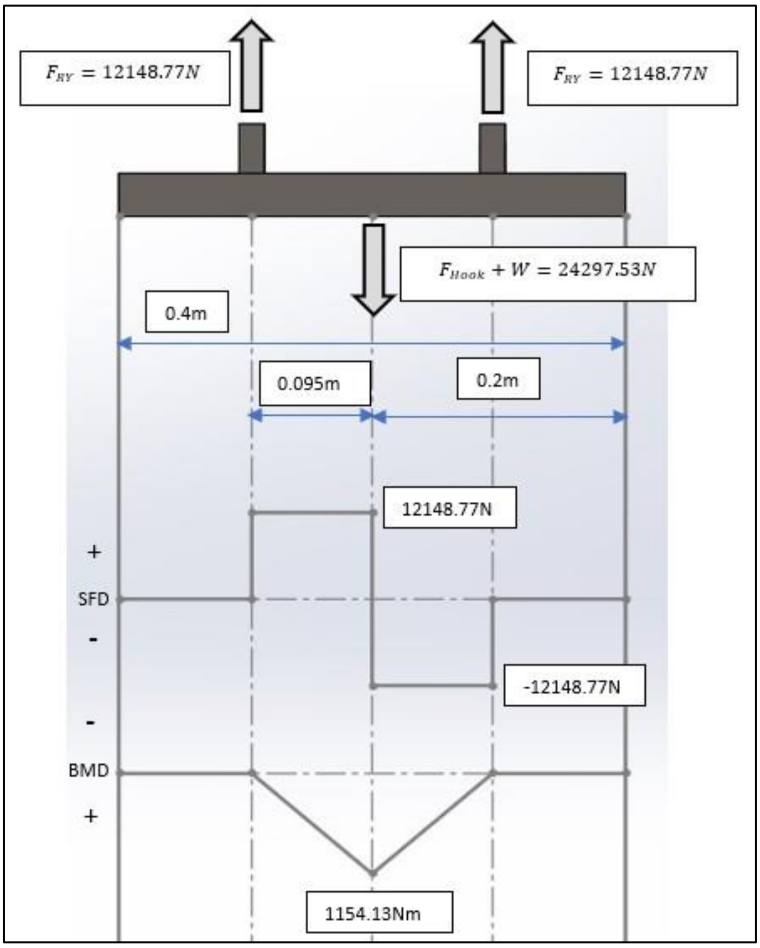


Figure 22: Shear Force & Bending Moment Diagrams of Initially Designed Hook Attachment

4.2.2 TENSILE STRESS ANALYSIS

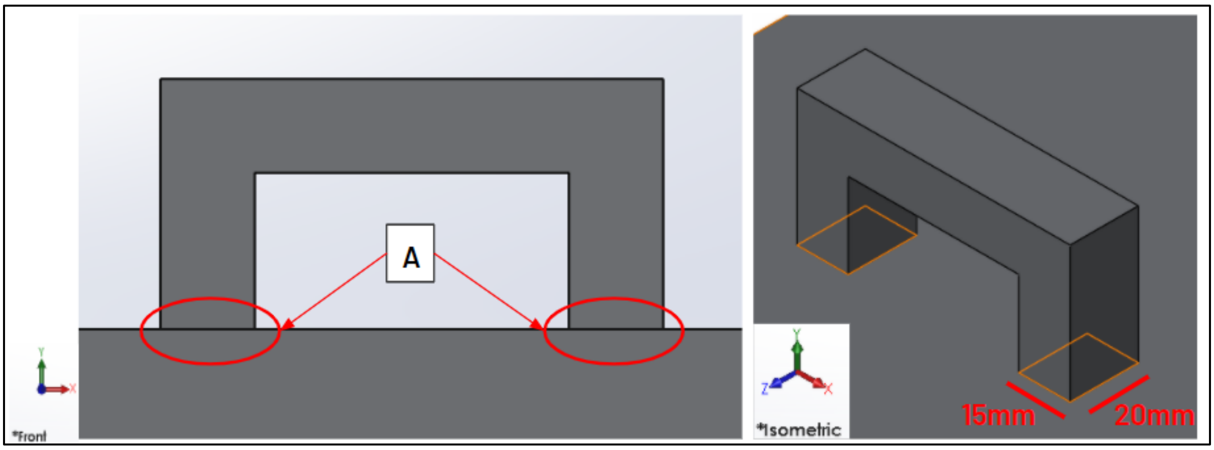


Figure 23: Surface Area on Initially Designed Hook Attachment Subjected to Tensile Stress

$$\begin{aligned} Surface\ Area &= A \\ A &= 2 \times (15\,mm \times 20\,mm) \\ &= 600\,mm^2 \end{aligned}$$

(Value of F below can be found from the Free Body Diagram of the initially designed hook attachment in Appendix D)

$$\begin{aligned} \sigma_T &= F / A \\ &= 12148.76\ N\ /\ 600\ mm^2 \\ &= 20.248\ \text{MPa} \end{aligned}$$

Since the hook attachment is made from AISI 4340 Annealed Steel, it's σ_Y = 470 MPa, the corresponding safety factor is:

$$\begin{aligned} \sigma_Y\ /\ Safety\ Factor &\leq \sigma_T \\ Safety\ Factor &\geq \sigma_Y\ /\ \sigma_T \\ &\geq (470\ MPa\ /\ 20.248\ MPa) \\ &\geq 23.212 \\ Minimum\ Safety\ Factor &= 23.212 \end{aligned}$$

4.2.3 <u>SHEAR STRESS ANALYSIS</u>

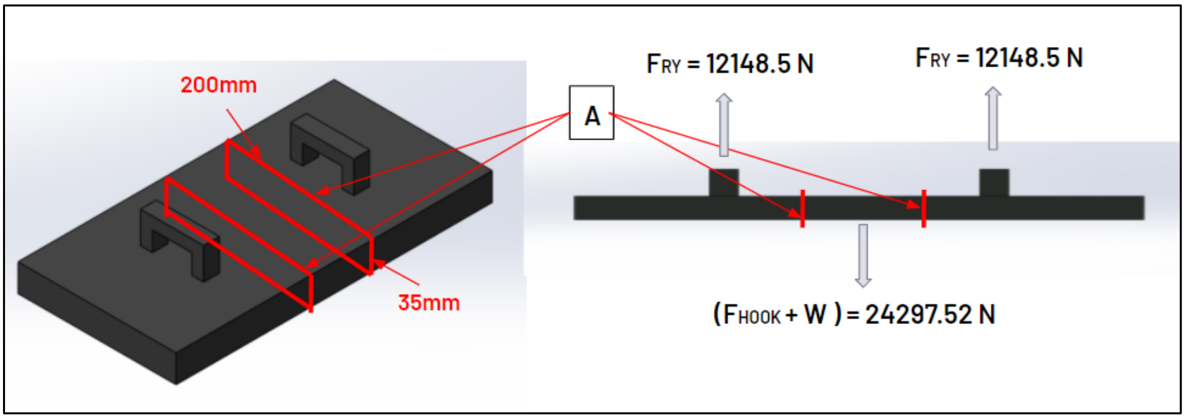


Figure 24: Surface Area on Initially Designed Hook Attachment Subjected to Shear Stress

$$\begin{aligned} Surface\ Area &= A \\ A &= 200\ mm \times 35\ mm \\ &= 7000\ mm^2 \end{aligned}$$

$$\begin{aligned} Maximum\ Shear\ Force &= F \\ F &= 24297.52\ N \end{aligned}$$

$$\begin{aligned} \tau &= F\ /\ 2A \\ &= 24297.52\ \text{N}\ /\ (2 \times 7000\ \text{mm}^2) \\ &= 1.736\ MPa \end{aligned}$$

$$\begin{aligned} \sigma_Y\ /\ Safety\ Factor &\leq \tau \\ Safety\ Factor &\geq 0.58\ \sigma_Y / \tau \\ &\geq (0.58\ x\ 470\ MPa\ /\ 1.736\ MPa) \\ &\geq 157.07 \\ Minimum\ Safety\ Factor &= 157.07 \end{aligned}$$

4.2.4 BENDING MOMENT ANALYSIS

$$\begin{aligned} Second\ Moment\ of\ Area &= I \\ I &= (1/12) \times (b \times h^3) \\ &= (1/12) \times (200\ mm \times 35\ mm^3) \\ &= 714583.33\ mm^4 \end{aligned}$$

$$\begin{aligned} Vertical\ distance\ away\ from\ the\ neutral\ axis &= y \\ y &= (35\ mm\ /\ 2) \\ &= 17.5\ mm \end{aligned}$$

$$\begin{aligned} \sigma_B &= (\mathrm{M} \times \mathrm{y})\ /\ \mathrm{I} \\ &= 1154132.2\ Nmm \times 17.5\ mm\ /\ (714583.33\ mm^4) \\ &= 28.264\ MPa \end{aligned}$$

$$\begin{aligned} \sigma_Y\ /\ Safety\ Factor &\leq \sigma_B \\ Safety\ Factor &\geq \sigma_Y / \sigma_B \\ &\geq (\ 470\ MPa\ /\ 28.264\ MPa) \\ &\geq 16.629 \\ Minimum\ Safety\ Factor &= 16.629 \end{aligned}$$

4.3 OPTIMISATION

Since the initially designed hook attachment is most likely to fail under bending stress, where the minimum safety factor found on the initial design of the hook attachment is 16.629, which is above a value of 5, the design is considered safe. Though the design is safe, a small optimisation is done onto the hook attachment handles' geometrical shape to reduce the stresses at the corners of the handles. This way, the stress on the hook attachment's handles will be slightly relieved. A comparison between the hook attachment before and after optimisation can be seen in Figure 25 below.

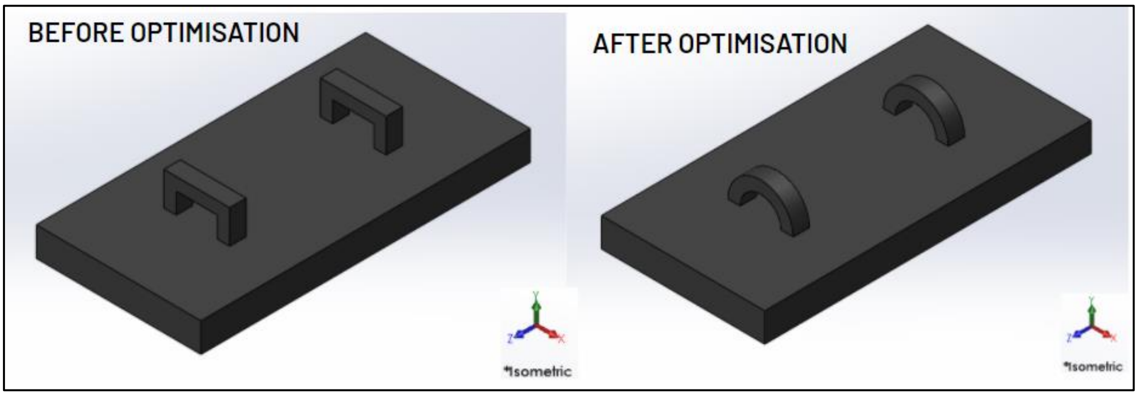


Figure 25: Comparison of Hook Attachment Before & After Optimisation

4.4 COMPONENT COSTING

Pure AISI 4340 Steel for machining purposes delivered to Malaysia from Alibaba.com costs about RM7.90/kg including delivery charges. An estimate of RM100/hour rate is charged for machining and annealing processes done to produce the hook attachment. According to the costing analysis report obtained via the Solidworks software, the estimated price to fabricate the hook attachment is RM 1500.50. This costing analysis report, along with pricing details of the material used can be found in Appendix C.

5.0 CHAIN

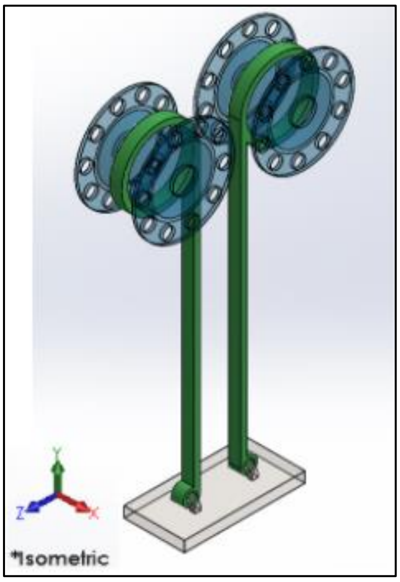


Figure 26: Chains Coiled Around the Pulley Drums and Connected to The Hook Attachment

Two long chains are coiled around the pulley drums and connected to the hook attachment on the other end. The rotational movement of the pulley drums cause the chains to coil or uncoil themselves from the drums, causing vertical movement of the hook attachment, and the load supported by it. The chains are subjected to tensile stress. Though the chains are represented by plain solid rods in the computer-aided designing (CAD) process as shown in Figure 26 above, in reality, the chains used are two long overhead lifting chains made up of connected common studless links as seen below in Figure 27. These chains are initially selected to be G80 Alloy Chains, with a trade size of 8mm. But upon optimisation, the initially selected chains are replaced with G80 Alloy Chains, with a trade size of 16mm.

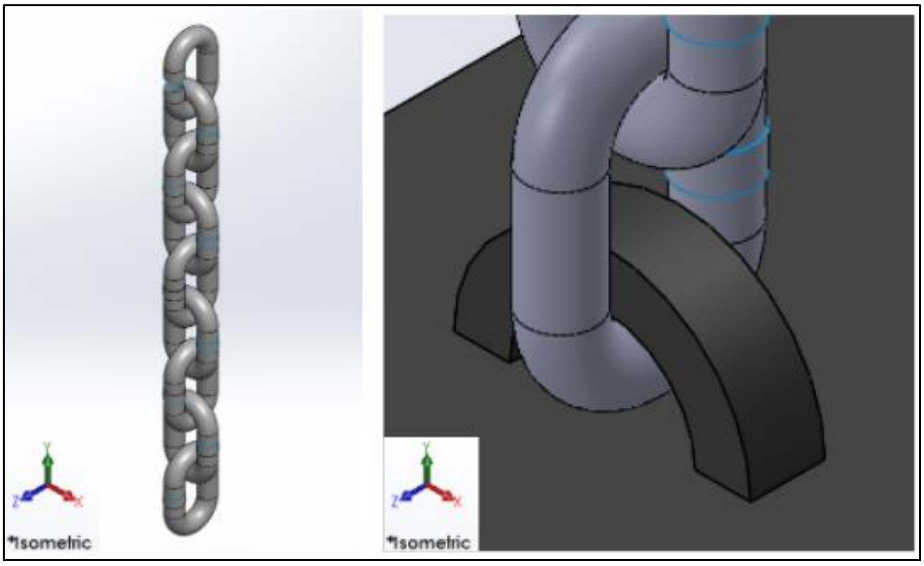


Figure 27: Actual Representation of The Overhead Lifting Chains Used

5.1 FREE BODY DIAGRAM (FBD) - OPTIMISED CHAINS

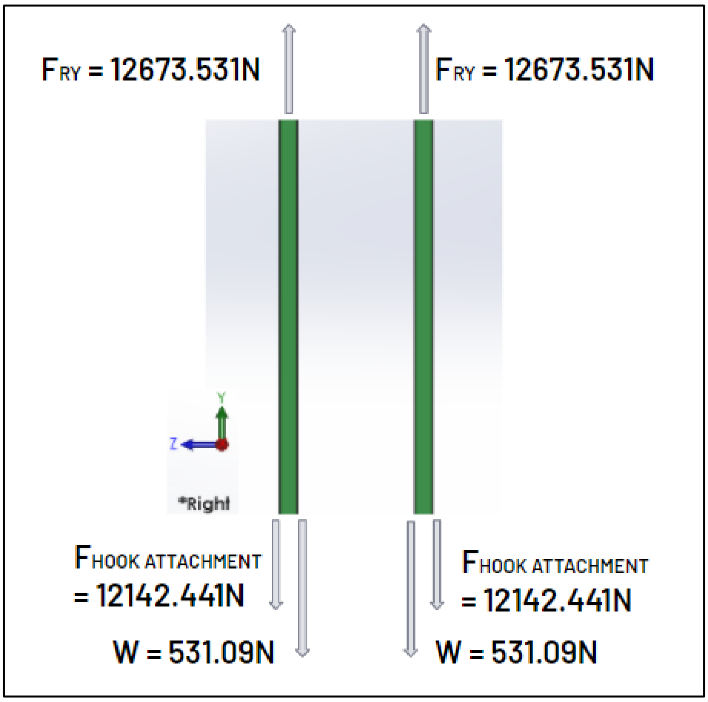


Figure 28: Free Body Diagram of Optimised Chains

5.1.1 FREE BODY DIAGRAM CALCULATIONS - OPTIMISED CHAINS

Though the free body diagram of the chains are represented using the CAD model, the calculations done to obtain the forces present on the free body diagram must utilise an actual chain link from the selected G80 Alloy Chains, as shown in Figure 29 below.

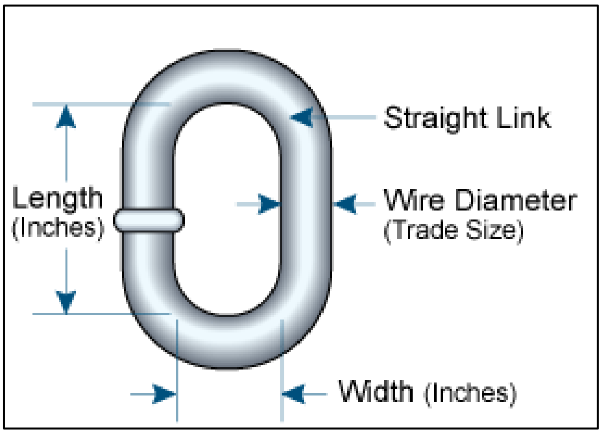


Figure 29: Basic Dimensions On The Actual Optimised Chain Link Used

$Information\ Provided\ in\ The\ Chain\ Catalogue\ from\ Appendix\ B$:

- $Trade\ Size\ of\ a\ G80\ Alloy\ Chain\ Link\ =\ 16\ mm$
- $Wire\ Diameter\ of\ a\ G80\ Alloy\ Chain\ Link\ =\ 0.630''\ \times\ 25.4\ mm\ =\ 16.002\ mm$
- $Length\ of\ a\ G80\ Alloy\ Chain\ Link\ =\ 1.890''\ \times\ 25.4\ mm\ =\ 48.006\ mm$
- $Total\ Mass\ of\ 500\ feet\ of\ G80\ Alloy\ Chain\ =\ 725\ kg$

$$\begin{aligned} Mass\ of\ Chain\ Per\ Meter\ Of\ Length\ &=\ (\ 725\ kg\ /\ 200\ feets\ ) \times (\ 1\ feet\ /\ 3.281\ m) \\ &=\ 1.105\ kg/m \end{aligned}$$

$Assuming\ the\ chains\ are\ hanging\ freely\ from\ the\ pulley\ drums\ at\ a\ maximum\ height\ of\ 49\ m:$

$$\begin{aligned} Mass\ of\ 49m\ of\ Freely\ Hanging\ Chain\ &=\ m \\ m\ &=\ 1.105\ kg/m\ \times 49\ m \\ &=\ 54.137\ kg \end{aligned}$$

$$\begin{aligned} Self\ Weight\ of\ Each\ Chain\ &=\ W \\ W\ &=\ m \times g \\ &=\ 54.137\ kg \times 9.81\ m/s^2 \\ &=\ 531.09\ N \end{aligned}$$

$$F_{HOOK\ ATTACHMENT}\ =\ 13317.81\ N$$

$$\begin{aligned} F_{RY}\ &=\ F_{HOOK\ ATTACHMENT}\ +\ W \\ &=\ 12142.441\ N\ +\ 531.09\ N \\ &=\ 12673.531\ N \end{aligned}$$

5.2 DESIGN ANALYSIS FRAMEWORK 3 (DAF)

5.2.1 TENSILE STRESS ANALYSIS

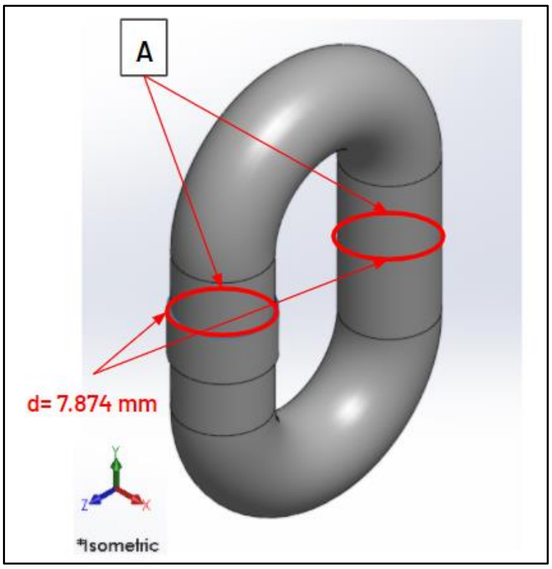


Figure 30: Surface Area on Chain Subjected to Tensile Stress

$$\begin{aligned} Surface\ Area\ &=\ A \\ A\ &=\ \pi\ \times\ r^2 \\ &=\ \pi\ \times\ (d/2)^2 \\ &=\ \pi\ \times\ (7.874\ mm/2)^2 \\ &=\ 48.694\ mm^2 \end{aligned}$$

$$\begin{aligned} \sigma_T\ &=\ F/\ 2A \\ &=\ F_{HOOK\ ATTACHMENT}\ /\ 2A \\ &=\ 12142.441\ N\ /\ (\ 2\ \times\ 48.694\ mm^2) \\ &=\ 124.681\ MPa \end{aligned}$$

According to the information provided in the chain catalogue from Appendix B, the Working Load Limit (WLL) of the initially chosen G80 Alloy Chains with a trade size of 8mm and a safety factor of 1 is 2041kg. Thus, the Working Load Limit (WLL) of the chains in Newton (N), and the corresponding safety factor is:

$$\begin{aligned} WLL &= 2041\,kg \times 9.81\,m/s^2 \\ &= 20022.21\,N \end{aligned}$$

(Value of $F_{HOOK\ ATTACHMENT}$ & W below can be found from the Free Body Diagram of the initially chosen chains in Appendix D)

$$\begin{aligned} WLL\,/\,Safety\ Factor &\geq (F_{HOOK\ ATTACHMENT} + W) \\ Safety\ Factor &\geq WLL\,/\,(F_{HOOK\ ATTACHMENT} + W) \\ &\geq (20022.21\,N\,/\,12276.551\,N) \\ &\geq 1.631 \\ Minimum\ Safety\ Factor &= 1.631 \end{aligned}$$

5.3 OPTIMISATION

From the initially selected G80 Alloy Chains with a trade size of 8mm, it was found that we had achieved a minimum safety factor of only 1.631, which is very low. To achieve a minimum safety factor of 5, a new selection of chains was done. G80 Alloy Chains with a trade size of 16mm were used to replace the initially selected chains.

According to the information provided in the chain catalogue from Appendix B, the Working Load Limit (WLL) of the G80 Alloy Chains with a trade size of 8mm and a safety factor of 1 is 8210kg. Thus, the Working Load Limit (WLL) of the chains in Newton (N), and the corresponding optimised safety factor is:

$$\begin{aligned} WLL &= 8210\,kg \times 9.81\,m/s^2 \\ &= 80540.1\,N \end{aligned}$$

$$\begin{aligned} WLL\,/\,Safety\ Factor &\geq (F_{HOOK\ ATTACHMENT} + W) \\ Safety\ Factor &\geq WLL\,/\,(F_{HOOK\ ATTACHMENT} + W) \\ &\geq (80540.1\,N\,/\,12276.551\,N) \\ &\geq 6.560 \\ Minimum\ Safety\ Factor &= 6.560 \end{aligned}$$

From the calculations above, the minimum safety factor found for the optimised chains is 6.560. This satisfies the current requirement to produce a safety factor of as close as possible to 5. Thus, the final optimised chains selected are G80 Alloy Chains with a trade size of 16mm.

5.4 COMPONENT COSTING

G80 Alloy Chains with a trade size of 16 mm cost RM 5.61/kg when delivered to Malaysia from Alibaba.com, including delivery charges. As calculated before, one set of chains used on one of the pulley drums in the hoist system is 54.137 kg. Since two sets of chains are used in the entire hoist, the total mass of chains used is 108.274 kg, which equates to a cost of RM 607.42. Catalogues and pricing details of the G80 Alloy Chains can be found in Appendix B.

6.0 PULLEY DRUM

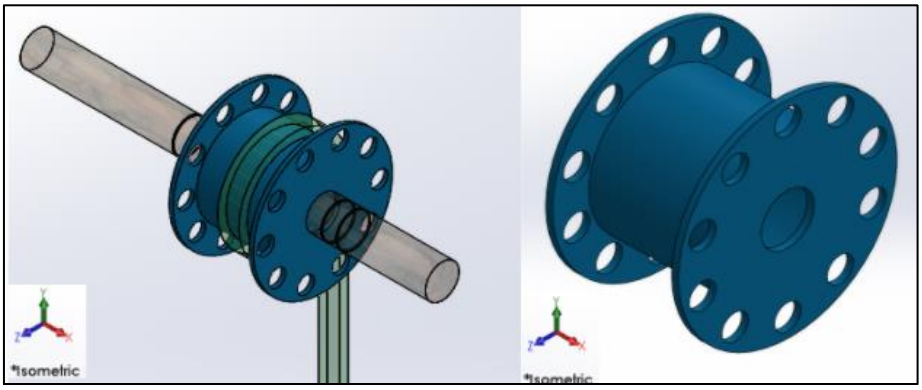


Figure 31: Pulley Drum Supported by A Shaft While A Chain Is Coiled Around It

A pulley drum is supported by a shaft, while acting as a spool that the chain is coiled around when hoisting loads vertically. The pulley drum is made from AISI 4340 Annealed Steel with a material density of 7850 kg/m³. Upon optimisation, a modification is done onto the pulley drum where its inner volume was hollowed out, and equispaced holes were made onto the sides of the drum. The initial design of the hook attachment before optimisation is as below in Figure 32.

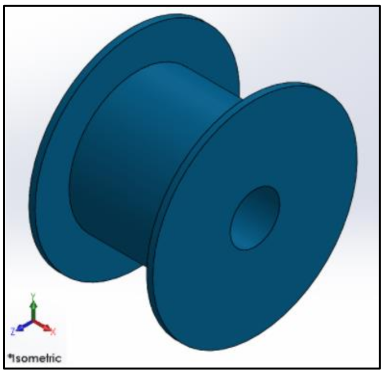


Figure 32: Initial Design of Pulley Drum

6.1 FREE BODY DIAGRAM (FBD) - OPTIMISED PULLEY DRUM

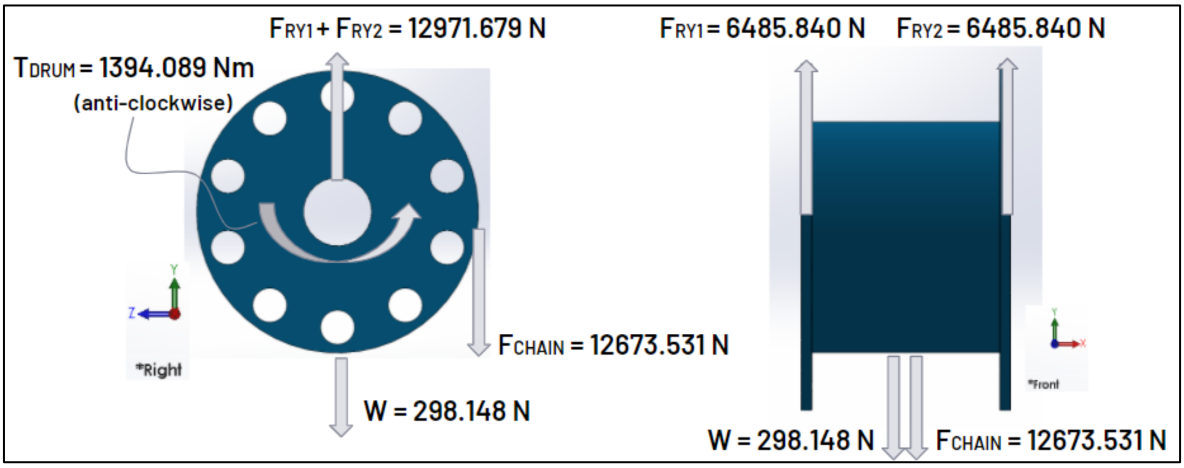


Figure 33: Free Body Diagram of Optimised Pulley Drum

(Free body diagram and corresponding calculations of the initially selected pulley drum can be found in Appendix D)

6.1.1 FREE BODY DIAGRAM CALCULATIONS - OPTIMISED PULLEY DRUM

$Assumptions$:

- $Volume\ of\ Optimised\ Pulley\ Drum\ =\ 0.00387162\ m^3$
- $Density\ of\ Pulley\ Drum\ Material\ (Annealed\ AISI\ 4340\ Steel)\ =\ 7850\ \ kg/m^3$
- $Radius\ of\ Pulley\ Drum\ =\ 110\ mm\ /\ 0.110\ m$

$$\begin{aligned}
Mass\ of\ Initially\ Designed\ Shaft\ &=\ m \\
m\ &=\ \rho\ \times\ V \\
&=\ 7850\ kg/m^3\ \times\ 0.00387162\ \mathrm{m}^3 \\
&=\ 30.392\ kg
\end{aligned}$$

$$\begin{aligned}
Self-Weight\ of\ Initially\ Designed\ Shaft &=\ W \\
W\ &=\ m\ \times\ g \\
&=\ 30.392\ kg\ \times\ 9.81\ m/s^2 \\
&=\ 298.148\ N
\end{aligned}$$

$$F_{CHAIN}\ =\ 12673.531\ N$$

$$\begin{aligned}
\mathrm{F_{RY1}}\ +\ \mathrm{F_{RY2}}\ &=\ \mathrm{F_{CHAIN}}\ +\ W \\
&=\ 12971.679\ N
\end{aligned}$$

$$\mathrm{F_{RY1}}\ +\ \mathrm{F_{RY2}}\ =\ 6485.840\ \mathrm{N}$$

$$\begin{aligned}
Torque\ On\ Pulley\ Drum\ &=\ T_{DRUM} \\
T_{DRUM}\ &=\ F_{CHAIN}\ \times\ radius \\
&=\ 12673.531\ N\ \times\ 0.110\ m \\
&=\ 1394.089\ Nm
\end{aligned}$$

6.2 DESIGN ANALYSIS FRAMEWORK 4 (DAF)

6.2.1 FINITE ELEMENT ANALYSIS (FEA)

Due to the complex geometrical shape of the pulley drum, the forces on the pulley drum were tough to be analysed using hand calculation. Thus, the Finite Element Analysis (FEA) was run onto the initially designed part. Since the chain is coiled around the pulley drum, there is a uniform external radial pressure formed around the pulley drum, and following is the relevant calculations to find it's value:

$Known\ Information$:

- $F_{CHAIN}\ =\ 12673.531\ N$
- $Mean\ Radius\ of\ Turns\ of\ Chain\ on\ Drum\ =\ r$

$$\begin{aligned}
r\ &=\ Radius\ of\ Drum\ +\ Smallest\ Radius\ of\ Chain \\
&=\ 110\ mm\ +\ 8\ mm \\
&=\ 118\ mm
\end{aligned}$$

- $Smallest\ Diameter\ of\ Chain\ =\ b$

$$b\ =\ 16mm$$

$$\begin{aligned}
Uniform\ External\ Radial\ Pressure\ &=\ p \\
p\ &=\ F_{CHAIN}\ /\ (r\ \times\ b) \\
&=\ 12673.531\ N\ /\ (118\ mm\ \times\ 16\ mm) \\
&=\ 6.713\ MPa
\end{aligned}$$

Through FEA, it is found that the maximum stress undergone by the initially designed pulley drum is 15.22 MPa, as shown in the Von Mises Stress Plot in Figure 34 below. The corresponding minimum safety factor of the pulley drum design is 40.76, as shown in the Safety Factor Plot of the pulley drum, in Figure 35 below.

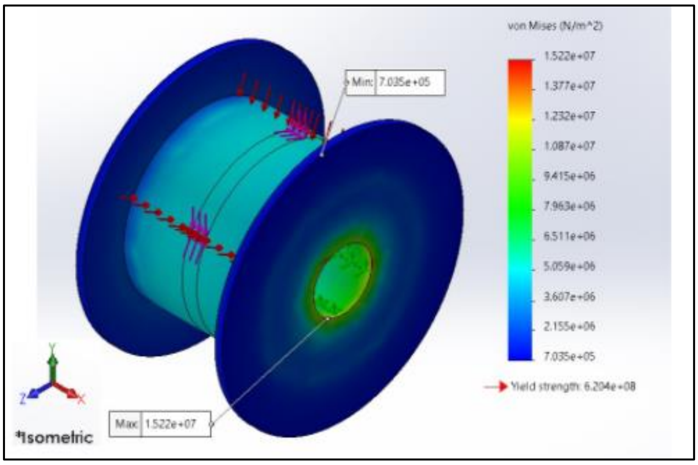


Figure 34: Von Mises Stress Plot of Pulley Drum

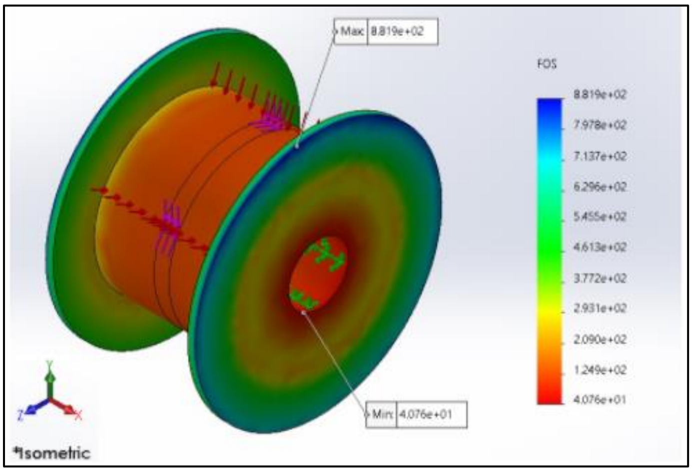


Figure 35: Safety Factor Plot of Pulley Drum

6.3 OPTIMISATION

From the original design of the hook, it is found that the minimum safety factor of the initially designed pulley drum is 40.76, which is very high. To achieve a safety factor of about 5, a modification is made onto the pulley drum's design, where the inner volume of the pulley drum is made hollow, and equispaced holes are made onto the sides of the pulley drum. This significantly reduces the self-weight of the pulley drum. Comparison of the pulley drum before and after optimisation can be seen in Figure 36 and Figure 37 below.

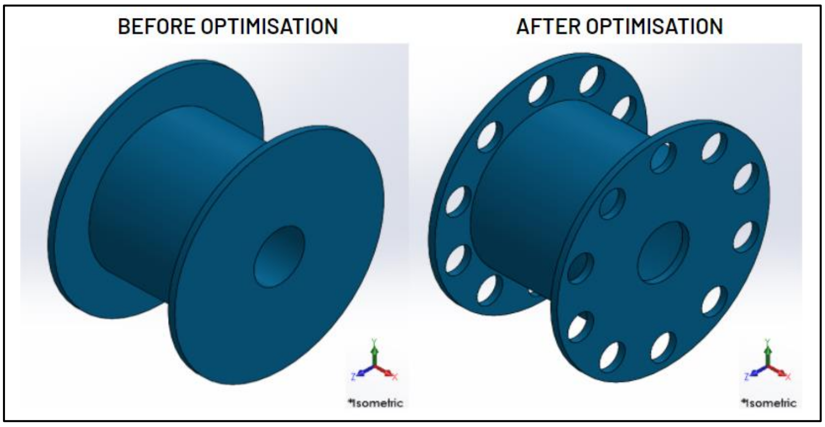


Figure 36: Comparison of Pulley Drum's Outer Surfaces Before & After Optimisation

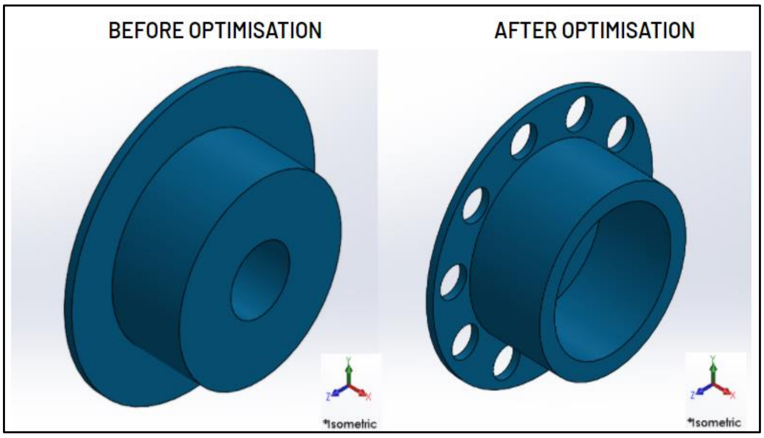


Figure 37: Comparison of Pulley Drum's Inner Surfaces Before & After Optimisation

Once the optimisation was done onto the pulley drum, the Finite Element Analysis (FEA) was again run onto the part. Through FEA, it is found that the maximum stress undergone by the optimised pulley drum is now 123.5 MPa, as shown in the Von Mises Stress Plot in Figure 38 below. The corresponding minimum safety factor of the pulley drum design is 5.025, as shown in the Safety Factor Plot of the pulley drum in Figure 39 below. This satisfies the current requirement to produce a safety factor of as close as possible to 5, while significantly reducing the self-weight of the pulley drum.

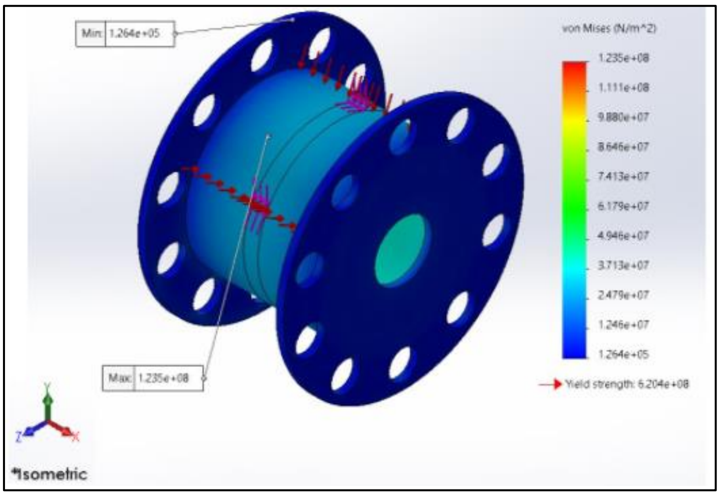


Figure 38: Updated Von Mises Stress Plot of Optimised Pulley Drum

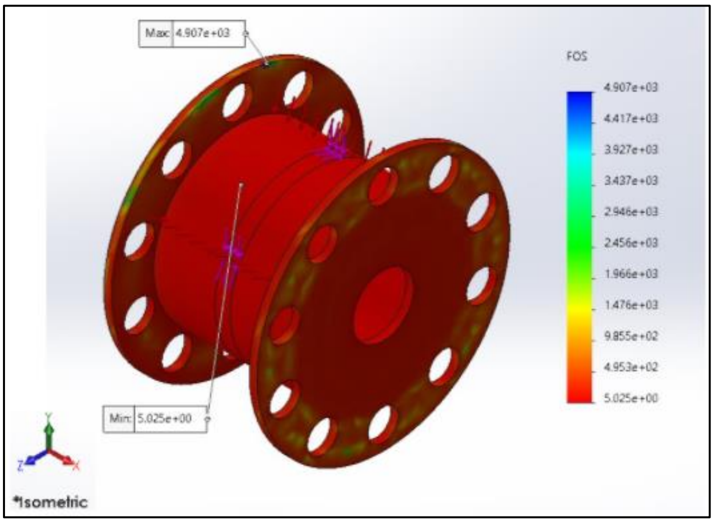


Figure 39: Updated Safety Factor Plot of Optimised Pulley Drum

6.4 COMPONENT COSTING

Pure AISI 4340 Steel delivered to Malaysia from Alibaba.com costs about RM7.90/kg including delivery charges. An estimate of RM300 is charged to design a custom mould for the casting process, and an RM100/hour rate is charged for melting, casting, machining, and annealing processes done to produce the pulley drum. According to the costing analysis report obtained via the Solidworks software, the estimated price to fabricate a single pulley drum is RM 689.80. This costing analysis report, along with pricing details of the material used can be found in Appendix C.

7.0 PULLEY DRIVING SHAFT

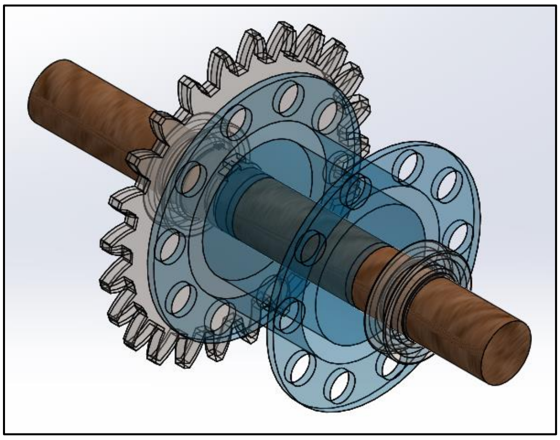


Figure 40: Pulley Driving Shaft Supporting a Bearing, a Gear, Two Bearings, Four Circlips On It

The pulley driving shaft, as shown above in Figure 40, holds a pulley drum and a gear, and is supported by two bearings and four circlips. Since the self-weight of the bearings and circlips are very insignificant compared to the other forces acting on the shaft, the self-weights are neglected. The torque from the hoisting mechanism's motor is channelled to the shaft to allow the hoisting mechanism to hoist loads vertically. The pulley driving shaft is made out of AISI 4340 Annealed Steel with a material density of 7800 kg/m³. The initially selected shaft has a diameter of 50mm with the diameter of the grooves for the circlips to sit on the shaft being 45.9mm. Upon optimisation, the shaft's diameter was increased to be 80mm, with the diameter of the grooves for the circlips to sit on the shaft being 75.9mm.

7.1 FREE BODY DIAGRAM (FBD) - OPTIMISED PULLEY DRIVING SHAFT

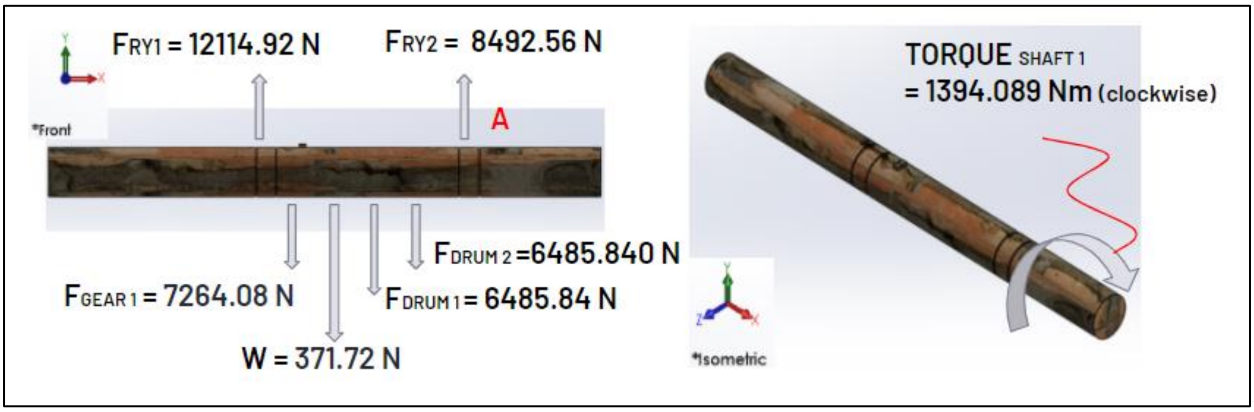


Figure 41: Free Body Diagram of Optimised Pulley Driving Shaft

7.1.1 FREE BODY DIAGRAM CALCULATIONS - OPTIMISED PULLEY DRIVING SHAFT

$Assumptions$:

- $Volume\ of\ Optimised\ Shaft\ =\ 0.00482699\ m^3$
- $Density\ of\ Pulley\ Drum\ Material\ (Annealed\ AISI\ 4340\ Steel)\ =\ 7850\ \ kg/m^3$

$$
\begin{aligned}
Mass\ of\ Optimised\ Shaft\ &=\ m \\
m\ &=\ \rho\ \times\ V \\
&=\ 7850\ kg/m^3\ \times\ 0.00482699\ \mathrm{m}^3 \\
&=\ 37.892\ kg
\end{aligned}
$$

$$
\begin{aligned}
Self-Weight\ of\ Optimised\ Shaft &=\ W \\
W\ &=\ m\ \times\ g \\
&=\ 37.892\ kg\ \times\ 9.81\ m/s^2 \\
&=\ 371.72\ N
\end{aligned}
$$

$$
\begin{aligned}
\Sigma M@A\ &=\ 0 \\
0\ &=\ (F_{RY1}\ \times\ 0.3529\ m)\ -\ (F_{GEAR1}\ \times\ 0.290\ m)\ -\ (W\ \times\ 0.251\ m)\ - \\
&\quad\ (F_{DRUM2}\ \times\ 0.250\ m)\quad -\ (F_{DRUM1}\ \times\ 0.070\ m) \\
0\ &=\ (F_{RY1}\ \times\ 0.3529\ m) -\ (7264.08\ N\ \times\ 0.290\ m) -\ (371.72\ N \times 0.251\ m) \\
&\qquad -(6485.84\ N\ \times\ 0.250\ m)\ -\ (6485.84\ N\ \times\ 0.070\ m) \\
\mathrm{F_{RY1}}\ &=\ 12114.92\ \mathrm{N}
\end{aligned}
$$

$$
\begin{aligned}
\Sigma F_Y\ &=\ 0 \\
0\ &=\ (F_{RY1}\ +\ F_{RY2}) - (7264.08\ N + 371.72\ N +\ 6485.84\ N +\ \ 6485.84\ N) \\
F_{RY2}\ &=\ 8492.56\ N
\end{aligned}
$$

(Free body diagram and corresponding calculations of the initially selected shaft can be found in Appendix D)

7.2 DESIGN ANALYSIS FRAMEWORK 5 (DAF)

7.2.1 SHEAR FORCE & BENDING MOMENT DIAGRAM

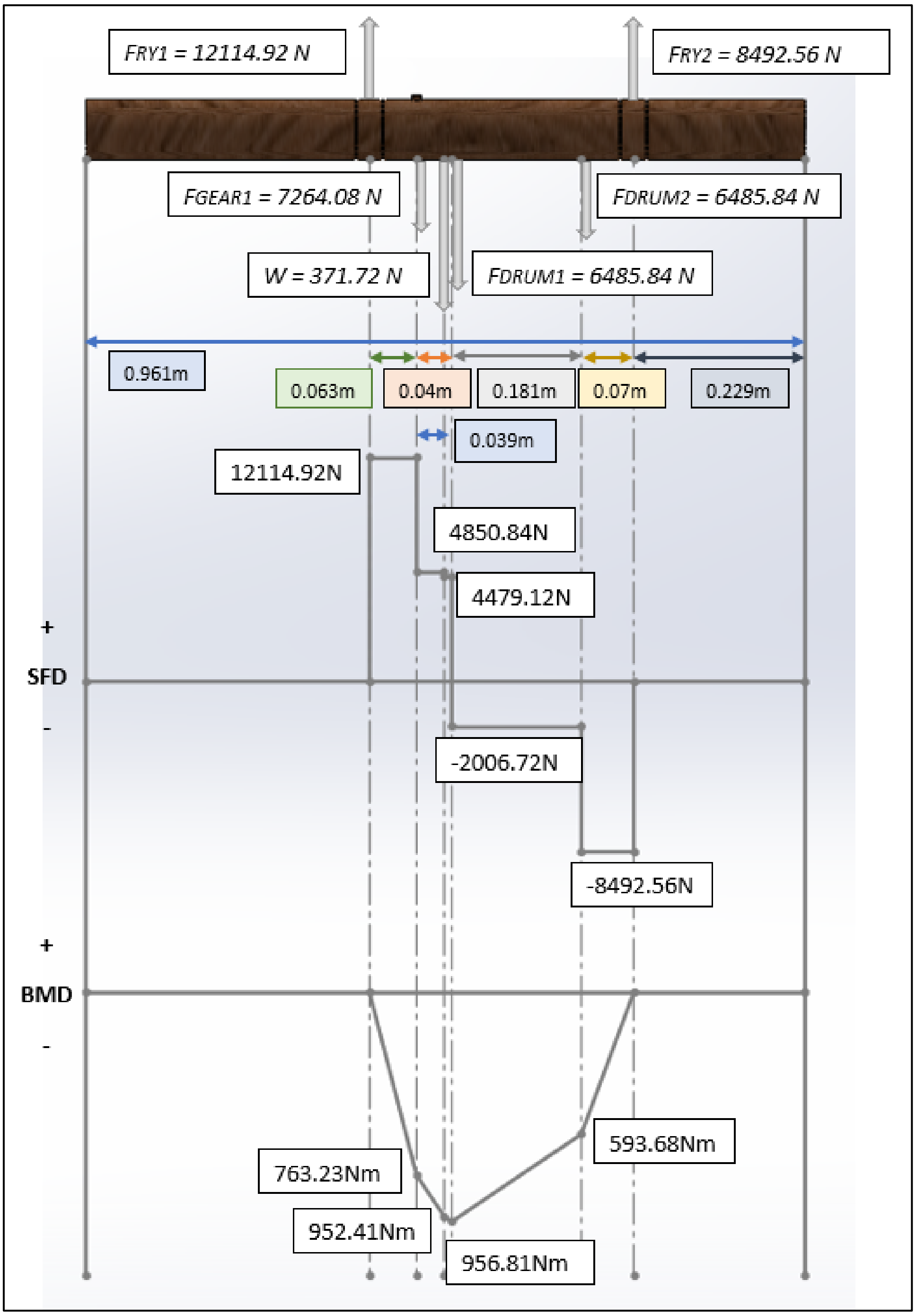


Figure 42: Shear Force & Bending Moment Diagrams of Initially Designed Pulley Driving Shaft

(Shear Force & Bending Moment Diagrams of the initially selected shaft can be found in Appendix D)

(Values below can be found from the data of the initially designed pulley driving shaft in Appendix D)

$$\begin{aligned} Diameter\ of\ Shaft &= 50mm \\ Diameter\ of\ Grooves\ on\ Shaft &= 45.9mm \end{aligned}$$

For shaft analysis, the smallest diameter of the pulley driving shaft is utilised, which refers to the diameter of grooves on the shaft:

$$\begin{aligned} d &= 45.9mm \\ &= 0.0459m \end{aligned}$$

$Torque\ on\ Shaft\ (T) = 1394.089\ Nm$
$Maximum\ Shear\ Force\ on\ Shaft\ (V) = 11953.05\ N$
$Maximum\ Bending\ Moment\ on\ Shaft\ (M) = 940.46\ Nm$
$Ultimate\ Tensile\ Strength\ of\ Shaft\ Material\ (AISI\ 4340\ Annealed\ Steel) - \sigma_U = 745\ MPa$
$Yield\ Strength\ of\ Shaft\ Material\ (AISI\ 4340\ Annealed\ Steel) - \sigma_Y = 470\ MPa$

7.2.2 TORSIONAL STRESS ANALYSIS

$$\begin{aligned} Torsional\ Stress\ on\ Shaft &= \tau \\ \tau &= (16 \times T) / (\pi \times d^3) \\ &= (16 \times 1394.089\ Nm) / (\pi \times (0.0459m)^3) \\ &= 73.421\ MPa \end{aligned}$$

7.2.3 BENDING STRESS ANALYSIS

$$\begin{aligned} Bending\ Stress\ on\ Shaft &= \sigma_B \\ \sigma_B &= (32 \times M) / (\pi \times d^3) \\ &= (32 \times 940.46\ Nm) / (\pi \times (0.0459m)^3) \\ &= 99.061\ MPa \end{aligned}$$

7.2.4 ASME CODE CALCULATION

By comparing the torsional stress and bending stress on the shaft, it is known that the shaft is more likely to fail due to bending stress since the bending stress magnitude is greater than the torsional stress magnitude. Hence, the ASME Code calculation is introduced to find the corresponding minimum safety factor of the shaft that will not fail due to bending stress. To perform the ASME Code Calculations, the shaft's endurance limit must first be calculated.

ENDURANCE LIMIT

$$For\ bending\ load: S_n = S_n' \times C_L \times C_G \times C_S \times C_T \times C_R$$

$$\begin{aligned} Where\ S_n' &= 0.5 \times S_U \\ &= 0.5 \times 745\ MPa \\ &= 372.5\ Mpa \end{aligned}$$

$$\begin{aligned} Load\ Factor\ for\ Bending\ Load, C_L &= 1.0 \\ Gradient\ Factor\ for\ d = 45.9mm,\ C_G &= 0.9 \\ Surface\ Factor\ for\ A\ Machined\ Shaft, C_S &= 0.76 \end{aligned}$$
(As shown below in Figure 43 below) [2]

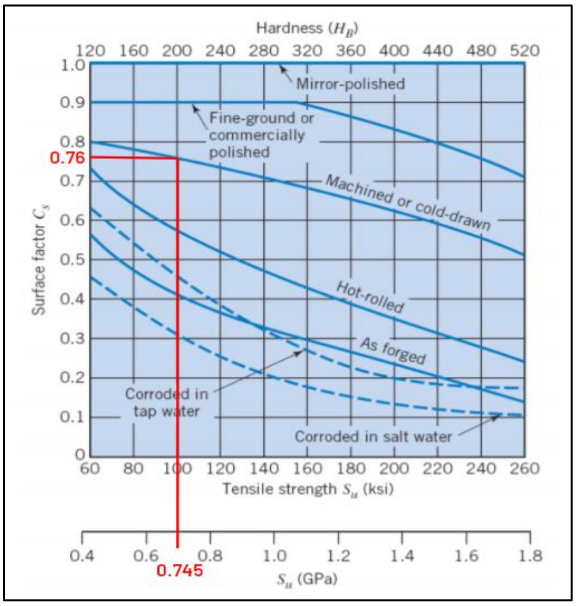


Figure 43: Surface Factor for A Machined Shaft

$$Temperature\ Factor\ for\ T\ \leq\ 840°F\ , C_T\ =\ 1.0$$
$$Reliability\ Factor\ of\ 50\%, C_S\ =\ 1.0$$

$$\begin{aligned} For\ bending\ load{:}\ S_n\ &=\ S_n{}'\ \times\ C_L\ \times\ C_G\ \times\ C_S\ \times\ C_T\ \times\ C_R \\ &=\ (372.5\ MPa) \times\ 1.0\ \times 0.9\ \times\ 0.76\ \times\ 1.0\ \times 1.0 \\ &=\ 254.79\ MPa \end{aligned}$$

ASME CODE

$$d\ =\ [\ (32\ \times\ n_s\ /\ \pi)\ \times \sqrt{\ (M\ /\ \sigma_e)^2\ +\ ¾\ (T\ /\ \sigma_y)^2\ )}\ ]^{1/3}$$

$$\begin{aligned} where\ \sigma_e\ &=\ Endurance\ Limit\ Stress\ \approx\ S_n \\ &=\ 254.79\ MPa \end{aligned}$$

Rearranging the above equation in terms of n_s:

$$n_s\ =\ (\pi\ \times d^3)\ /\ (32\ \times \sqrt{\ (M\ /\ \sigma_e)^2\ +\ ¾\ (T\ /\ \sigma_y)^2\ )}\ )$$

$$n_s\ =\ \ (\pi\ \times (0.0459m)^3)\ /\ (32\ x\ \surd(\ (940.46\ Nm\ /\ 254.79\ \times\ 10^6\ Pa)^2\ +\ ¾\ (1394.089\ Nm\ /\ 470\ \times\ 10^6\ \ Pa)^2\ )$$

$$Minimum\ Safety\ Factor\ =\ n_s\ =\ 2.111$$

$$Torsional\ Stress, \tau = \frac{16T}{\pi d^3}$$

Operating Torque, T (Nm)	1394.089
Diameter, d (m)	0.0459
Torsional Stress, τ (MPa)	73.421

$$Bending\ Stress, \sigma = \frac{32M}{\pi d^3}$$

Max Bending moment, M (Nm)	940.46
Diameter, d (m)	0.0459
Bending Stress, σ (MPa)	99.061

$$Material\ Property$$

Material Used	AISI 4340 steel, annealed
UTS (Pa)	745000000
Yield Strength, σy (Pa)	470000000

$$ASME\ Code\ Calculation, S.F. = \frac{d^3\pi}{32\sqrt{(\frac{M}{\sigma_e})^2 + \frac{3}{4}(\frac{T}{\sigma_y})^2}}$$

Diameter, d (m)	0.0459
Max Bending moment, M (Nm)	940.46
Operating Torque, T (Nm)	1394.089
Yield strength of shaft material, σy (Pa)	4.7E+08
Endurance limit stress, σe (Pa)	2.55E+08
Safety Factor, S.F.	$\boxed{2.111}$

$$Endurance\ limit\ stress, \sigma_e$$

S'n (0.5 x UTS) (Pa)	372500000
CL, Load factor	1
CG, Gradient Factor	0.9
CS, Surface Factor	0.76
CT, Temperature Factor	1
CR, Reliability Factor	1
Endurance limit stress, σe (Pa)	254790000

Figure 44: Design Analysis Framework Calculations of Pulley Driving Shaft Verified on Microsoft Excel Software

7.3 OPTIMISATION

Diameter of the shaft (mm)	CG, Gradient Factor	Safety Factor
45.9	0.9	2.111
55.9	0.8	3.512
65.9	0.8	5.754
75.9	0.8	8.791

Table 1: Iterations Done on Microsoft Excel Using Different Diameters of Grooves on Shaft

From the original design of the pulley driving shaft, it is found that the minimum safety factor of the initially designed shaft is 2.111, which is extremely low. Even though a minimum safety factor of 5 is tried to be achieved when designing or selecting other components in the hoist, it is important that the shaft has a greater value of safety factor as the shaft is the most crucial component in the hoist system. If the shaft fails, all other components that the shaft carries are in big risk of failing and getting damaged too. Thus, the target safety factor for the designed shaft is a minimum of 8. To achieve a safety factor of about 8, a Microsoft Excel file is used to iterate the previous calculations done onto the shaft, while increasing the smallest diameter of the shaft until a satisfying minimum safety factor is achieved. In this iteration process, the self-weight of the shaft and the corresponding maximum bending moment on the shaft is kept unchanged, since this information produces very insignificant variations in the results. When the diameter of the shaft is increased up to 80mm, where the diameter of the grooves on the shaft is 75.9mm, the minimum safety factor of the shaft is 8.791, as shown in Table 1 above. Since this value is close to our target safety factor value of 8, the calculations done onto the shaft in the Design Analysis Framework above is repeated again, but this time the variation in the self-weight of the shaft and the corresponding maximum bending moment on the shaft is also taken into account.

$$Diameter\ of\ Shaft = 80mm$$
$$Diameter\ of\ Grooves\ on\ Shaft = 75.9mm$$

For shaft analysis, the smallest diameter of the pulley driving shaft is utilised, which refers to the diameter of grooves on the shaft:

$$d = 75.9mm$$
$$= 0.0759m$$

$Torque\ on\ Shaft\ (T) = 1394.089\ Nm$

$Maximum\ Shear\ Force\ on\ Shaft\ (V) = 11953.05\ N$

$Maximum\ Bending\ Moment\ on\ Shaft\ (M) = 956.81\ Nm$

$Ultimate\ Tensile\ Strength\ of\ Shaft\ Material\ (AISI\ 4340\ Annealed\ Steel) - \sigma_U = 745\ MPa$

$Yield\ Strength\ of\ Shaft\ Material\ (AISI\ 4340\ Annealed\ Steel) - \sigma_Y = 470\ MPa$

$$Torsional\ Stress\ on\ Shaft = \tau$$
$$\tau = (16 \times T) / (\pi \times d^3)$$
$$= (16 \times 1394.089\ Nm) / (\pi \times (0.0759m)^3)$$
$$= 16.238\ MPa$$

$$Bending\ Stress\ on\ Shaft = \sigma_B$$
$$\sigma_B = (32 \times M) / (\pi \times d^3)$$
$$= (32 \times 956.81\ Nm) / (\pi \times (0.0759m)^3)$$
$$= 22.290\ MPa$$

ENDURANCE LIMIT

$$For\ bending\ load: S_n = S_n' \times C_L \times C_G \times C_S \times C_T \times C_R$$

$$\begin{aligned} Where\ S_n' &= 0.5 \times S_U \\ &= 0.5 \times 745\ MPa \\ &= 372.5\ Mpa \end{aligned}$$

$$\begin{aligned} Load\ Factor\ for\ Bending\ Load, C_L &= 1.0 \\ Gradient\ Factor\ for\ d = 75.9mm,\ C_G &= 0.8 \\ Surface\ Factor\ for\ A\ Machined\ Shaft, C_S &= 0.76 \\ Temperature\ Factor\ for\ T \leq 840°F\ , C_T &= 1.0 \\ R\quad eliability\ Factor\ of\ 50\%, C_S &= 1.0 \end{aligned}$$

$$\begin{aligned} For\ bending\ load: S_n &= S_n' \times C_L \times C_G \times C_S \times C_T \times C_R \\ &= (372.5\ MPa) \times 1.0 \times 0.8 \times 0.76 \times 1.0 \times 1.0 \\ &= 226.48\ MPa \end{aligned}$$

ASME CODE

$$d = [\,(32\ n_s / \pi) \times \sqrt{(\ (M / \sigma_e)^2 + ¾\,(T / \sigma_y)^2\,)}\ ]^{1/3}$$

$$\begin{aligned} where\ \sigma_e &= Endurance\ Limit\ Stress \approx S_n \\ &= 226.48\ MPa \end{aligned}$$

Rearranging the above equation in terms of n_s:

$$n_s = (\pi \times d^3) / (32 \times \sqrt{(\,(M/ \sigma_e)^2 + ¾\,(T / \sigma_y)^2\,)}$$

$$n_s = (\pi \times (0.0759m)^3) / (32 \times \sqrt{(}\,(956.81\ Nm / 226.48 \times 10^6\ Pa)^2 + ¾\,(1394.089\ Nm / 470 \times 10^6\ Pa)^2\,)$$

$$Minimum\ Safety\ Factor = n_s = 8.682$$

From the calculations above, it can be seen that the minimum safety factor found for the optimised pulley driving shaft is 8.682. This satisfies the current requirement to produce a safety factor of as close as possible to 8. This result is also verified using the Microsoft Excel file to run the relevant calculations, as shown below in Figure 45. Thus, the final optimised shaft has a diameter of 80mm, while its grooves have a diameter of 75.9mm.

$Torsional\ Stress, \tau = \frac{16T}{\pi d^3}$

Operating Torque, T (Nm)	1394.089
Diameter, d (m)	0.0759
Torsional Stress, τ (MPa)	16.238

$Bending\ Stress, \sigma = \frac{32M}{\pi d^3}$

Max Bending moment, M (Nm)	956.81
Diameter, d (m)	0.0759
Bending Stress, σ (MPa)	22.290

Material Property

Material Used	AISI 4340 steel, annealed
UTS (Pa)	745000000
Yield Strength, σy (Pa)	470000000

$ASME\ Code\ Calculation, S.F. = \frac{d^3\pi}{32\sqrt{(\frac{M}{\sigma_e})^2 + \frac{3}{4}(\frac{T}{\sigma_y})^2}}$

Diameter, d (m)	0.0759
Max Bending moment, M (Nm)	956.81
Operating Torque, T (Nm)	1394.089
Yield strength of shaft material, σy (Pa)	4.7E+08
Endurance limit stress, σe (Pa)	2.26E+08
Safety Factor, S.F.	8.682

$Endurance\ limit\ stress, \sigma_e$

S'n (0.5 x UTS) (Pa)	372500000
CL, Load factor	1
CG, Gradient Factor	0.8
CS, Surface Factor	0.76
CT, Temperature Factor	1
CR, Reliability Factor	1
Endurance limit stress, σe (Pa)	226480000

Figure 45: Design Analysis Framework Calculations of Optimised Pulley Driving Shaft Done on Microsoft Excel Software

7.4 SHAFT TOLERANCE

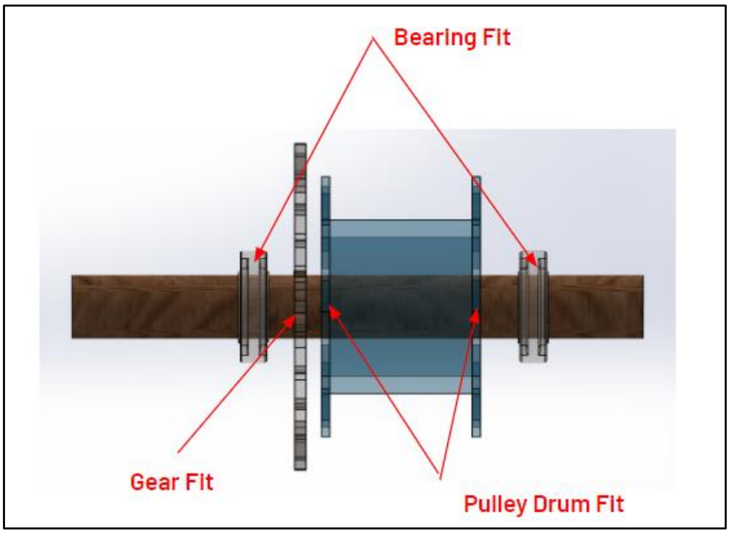


Figure 46: Types of Fits on Pulley Driving Shaft

The pulley driving shaft holds three major components in the hoist, which includes the bearings, the gear, and the pulley drum. To fit all the components well as intended on the shaft, tolerancing is crucially necessary. The pulley driving shaft's tolerance depends on the components it is attached to, and the functionality of those individual components.

7.4.1 SHAFT TOLERANCE WITH BEARING

Figure 47: Tolerancing for the Shaft with The Inner Diameter of The Attached Bearings

H7n6 N7h6	"interference transition fit"	•used for tight assembly of components where vibration is not tolerated. •generally result in a push fit. •keyed assembly of parts such as high speed fans and motor parts.

Figure 48: Selection of ISO Tolerance Fit for Bearings

With reference to Figure 48 above, it can be justified that the main reason for selecting a tolerance fit of "H7n6" is because the inner surface of the roller bearings is connected to the pulley driving shaft, and the inner diameter of the bearing must be tightly assembled over the shaft to reduce the chances of vibration being produced in the system. Besides that, a push fit is required in this application, which is one of the reasons this tolerance fit has been selected. Since the bearing is a purchased component, and the shaft is a fabricated component, the hole basis chart of tolerance fits is used to determine a suitable fit where the diameter of the shaft varies but the diameter of the hole remains unchanged.

TABLE A.2 SELECTED FITS FOR GENERAL USE (HOLE BASIS)

Basic size mm		Clearance fits												Transition fits				Interference fits			
		Upper and lower deviations for tolerance class (Units μm)																			
		H11	c11	H9	d10	H9	e9	H8	f7	H7	g6	H7	h6	H7	k6	H7	n6	H7	p6	H7	s6
Above	Up to and incl	+	−	+	−	+	−	+	−	+	−	+	−	+	+	+	+	+	+	+	+
0	3	60 0	60 120	25 0	20 60	25 0	14 39	14 0	6 16	10 0	2 8	10 0	0 6	10 0	6 0	10 0	10 4	10 0	12 6	10 0	20 14
3	6	75 0	70 145	30 0	30 78	30 0	20 50	18 0	10 22	12 0	4 12	12 0	0 8	12 0	9 1	12 0	16 8	12 0	20 12	12 0	27 19
6	10	90 0	80 170	36 0	40 98	36 0	25 61	22 0	13 28	15 0	5 14	15 0	0 9	15 0	10 1	15 0	19 10	15 0	24 15	15 0	32 23
10	18	110 0	95 205	43 0	50 120	43 0	32 75	27 0	16 34	18 0	6 17	18 0	0 11	18 0	12 1	18 0	23 12	18 0	29 18	18 0	39 28
18	30	130 0	110 240	52 0	65 149	52 0	40 92	33 0	20 41	21 0	7 20	21 0	0 13	21 0	15 2	21 0	28 15	21 0	35 22	21 0	48 35
30	40	160 0	120 280	62 0	80 180	62 0	50 112	39 0	25 50	25 0	9 25	25 0	0 16	25 0	18 2	25 0	33 17	25 0	42 26	25 0	59 43
40	50	160 0	130 290																		
50	65	190 0	140 330	74 0	100 220	74 0	60 134	46 0	30 60	30 0	10 29	30 0	0 19	30 0	21 2	30 0	39 20	30 0	51 32	30 0	72 53
65	80	190 0	150 340																	30 0	78 59
80	100	220 0	170 390	87 0	120 260	87 0	72 159	54 0	36 71	35 0	12 34	35 0	0 22	35 0	25 3	35 0	45 23	35 0	59 37	35 0	93 71
100	120	220 0	180 400																	35 0	101 79
120	140	250 0	200 450	100 0	145 305	100 0	84 185	63 0	43 83	40 0	14 39	40 0	0 25	40 0	28 3	40 0	52 27	40 0	68 43	40 0	117 92
140	160	250 0	210 460																	40 0	125 100
160	180	250 0	230 480																	40 0	133 108
180	200	290 0	240 530	115 0	170 355	115 0	100 215	72 0	50 96	46 0	15 44	46 0	0 29	46 0	33 4	46 0	60 31	46 0	79 50	46 0	151 122
200	225	290 0	260 550																	46 0	159 130
225	250	290 0	280 570																	46 0	169 140
250	280	320 0	300 620	130 0	190 400	130 0	110 240	81 0	56 108	52 0	17 49	52 0	0 32	52 0	36 4	52 0	66 34	52 0	88 56	52 0	190 158
280	315	320 0	330 650																	52 0	202 170
315	355	360 0	360 720	140 0	210 440	140 0	125 265	89 0	62 119	57 0	18 54	57 0	0 36	57 0	40 4	57 0	73 37	57 0	98 62	57 0	226 190
355	400	360 0	400 760																	57 0	244 208
400	450	400 0	440 840	155 0	230 480	155 0	135 290	97 0	68 131	63 0	20 60	63 0	0 40	63 0	45 5	63 0	80 40	63 0	108 68	63 0	272 232
450	500	400 0	480 880																	63 0	292 252

Table 2: Selected ISO Tolerance Fit - Hole basis

The purchased bearing has an inner diameter of 80 mm. Therefore, with reference to Table 2 above, the suitable tolerance fit for the pulley driving shaft running through the bearing is found, as following:

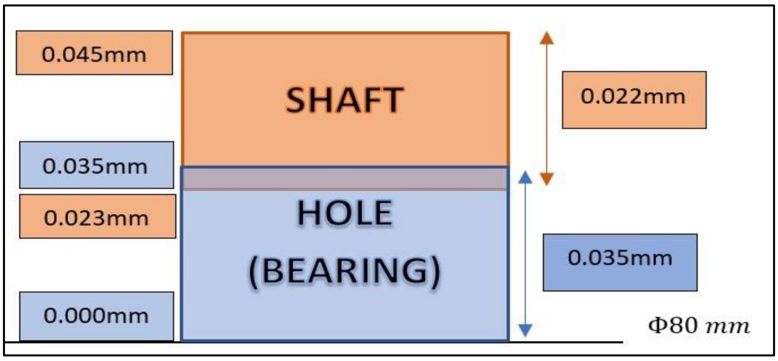


Figure 49: Selected Tolerance Fit Assembly for Bearing & Main Casing

<u>TOLERANCE:</u>

$\boldsymbol{Hole\ (Bearing)}$

$Maximum\ diameter\ of\ hole\ (Bearing) = 80 + 0.035 \quad = 80.035\ mm$

$Minimum\ diameter\ of\ hole\ (Bearing) = 80 + 0 \quad = 80.000\ mm$

$Difference\ between\ the\ LH\ and\ SH = 80.035 - 80.000 \quad = 0.035\ mm$

$\boldsymbol{MMC\ format\ for\ hole\ size\ (Housing)} \quad = \boldsymbol{\emptyset 80.0^{+0.035}_{-0}\ mm}$

$\boldsymbol{Shaft}$

$Maximum\ diameter\ of\ shaft\ (Bearing) = 80 + 0.045 \quad = 80.045mm$

$Minimum\ diameter\ of\ shaft\ (Bearing) = 80 + 0.023 \quad = 80.023mm$

$Difference\ between\ the\ LS\ and\ SS = 80.045 - 80.023 \quad = 0.022mm$

$\boldsymbol{MMC\ format\ for\ shaft\ size\ (Bearing)} \quad = \boldsymbol{\emptyset 80.045^{+0}_{-0.022}\ mm}$

INTEFERENCE FITS CHECK:

$$
\begin{aligned}
Maximum\ Interference &= Smallest\ Hole\ (SH) - \ Largest\ Shaft\ (LS) \\
&= 80.000\ -\ 80.045 \\
&= -\ 0.045\ mm
\end{aligned}
$$

$$
\begin{aligned}
Maximum\ Interference &= Largest\ Hole\ (LH)\ -\ Smallest\ Shaft\ (SS) \\
&= 80.035 - 80.023 \\
&= 0.012\ mm
\end{aligned}
$$

7.4.2 SHAFT TOLERANCE WITH PULLEY DRUM

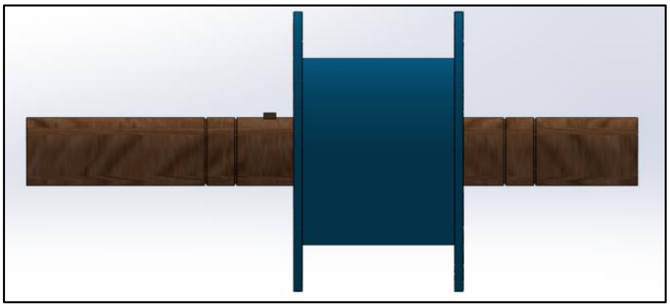


Figure 50: Tolerancing for the Shaft with The Inner Diameter of The Pulley Drum

H7n6 N7h6	"interference transition fit"	•used for tight assembly of components where vibration is not tolerated. •generally result in a push fit. •keyed assembly of parts such as high speed fans and motor parts.

Figure 51: Selection of ISO Tolerance Fit for Pulley Drum

With reference to Figure 51 above, it can be justified that the main reason for selecting the "N7h6" tolerance fit for the pulley drum connected to the pulley driving shaft " is because the inner surface of the roller bearings is connected to the pulley driving shaft, and the inner diameter of the bearing must be tightly assembled over the shaft to reduce the chances of vibration being produced in the system. Vibration in the system can drastically danger the hoist system and reduce its efficiency too. Besides that, a push fit is required in this application, which is one of the reasons this tolerance fit has been selected. Since the tolerance of the shaft is determined in the previous section 7.4.1, the shaft basis tolerance fit chart is used to determine the tolerance of the inner hole of the pulley drum.

TABLE A.3 SELECTED FITS FOR GENERAL USE (SHAFT BASIS)

Basic size mm		Clearance fits												Transition fits				Interference fits			
		Upper and lower deviations for tolerance class (Units μm)																			
		C11	h11	D10	h9	E9	h9	F8	h7	G7	h6	H7	h6	K7	h6	N7	h6	P7	h6	S7	h6
Above	Up to and incl	+	−	+	−	+	−	+	−	+	−	+	−	+ & −	−	−	−	−	−	−	−
0	3	120 60	0 60	60 20	0 25	39 14	0 25	20 6	0 10	12 2	0 6	10 0	0 6	0 −10	0 6	4 14	0 6	6 16	0 6	14 24	0 6
3	6	145 70	0 75	78 30	0 30	50 20	0 30	28 10	0 12	16 4	0 8	12 0	0 8	+3 −9	0 8	4 16	0 8	8 20	0 8	15 27	0 8
6	10	170 80	0 90	98 40	0 36	61 25	0 36	35 13	0 15	20 5	0 9	15 0	0 9	+5 −10	0 9	4 19	0 9	9 24	0 9	17 32	0 9
10	18	205 95	0 110	120 50	0 43	75 32	0 43	43 16	0 18	24 6	0 11	18 0	0 11	+6 −12	0 11	5 23	0 11	11 29	0 11	21 39	0 11
18	30	240 110	0 130	149 65	0 52	92 40	0 52	53 20	0 21	28 7	0 13	21 0	0 13	+6 −15	0 13	7 28	0 13	14 35	0 13	27 48	0 13
30	40	280 120	0 160	180 80	0 62	112 50	0 62	64 25	0 25	34 9	0 16	25 0	0 16	+7 −18	0 16	8 33	0 16	17 42	0 16	34 59	0 16
40	50	290 130	0 160																		
50	65	330 140	0 190	220 100	0 74	134 60	0 74	76 30	0 30	40 10	0 19	30 0	0 19	+9 −21	0 19	9 39	0 19	21 51	0 19	42 72	0 19
65	80	340 150	0 190																	48 78	0 19
80	100	390 170	0 220	260 120	0 87	159 72	0 87	90 36	0 35	47 12	0 22	35 0	0 22	+10 −25	0 22	10 45	0 22	24 59	0 22	58 93	0 22
100	120	400 180	0 220																	66 101	0 22
120	140	450 200	0 250	305 145	0 100	185 85	0 100	106 43	0 40	54 14	0 25	40 0	0 25	+12 −28	0 25	12 52	0 25	28 68	0 25	77 117	0 25
140	160	460 210	0 250																	85 125	0 25
160	180	480 230	0 250																	93 133	0 25
180	200	530 240	0 290	355 170	0 115	215 100	0 115	122 50	0 46	61 15	0 29	46 0	0 29	+13 −33	0 29	14 60	0 29	33 79	0 29	105 151	0 29
200	225	550 260	0 290																	113 159	0 29
225	250	570 280	0 290																	123 169	0 29
250	280	620 300	0 320	400 190	0 130	240 110	0 130	137 56	0 52	62 17	0 32	52 0	0 32	+16 −36	0 32	14 66	0 32	36 88	0 32	138 190	0 32
280	315	650 330	0 320																	150 202	0 32
315	355	720 360	0 360	440 210	0 140	265 125	0 140	151 62	0 57	75 18	0 36	57 0	0 36	+17 −40	0 36	16 73	0 36	41 98	0 36	169 226	0 36
355	400	760 400	0 360																	187 244	0 36
400	450	840 440	0 400	485 230	0 155	290 135	0 155	165 68	0 63	83 20	0 40	63 0	0 40	+18 −45	0 40	17 80	0 40	45 108	0 40	209 272	0 40
450	500	880 480	0 400																	229 292	0 40

Table 3: Selected ISO Tolerance Fit – Shaft Basis

With reference to Table 3 above, the suitable tolerance fit for the pulley drum attached to the pulley driving shaft is found as following:

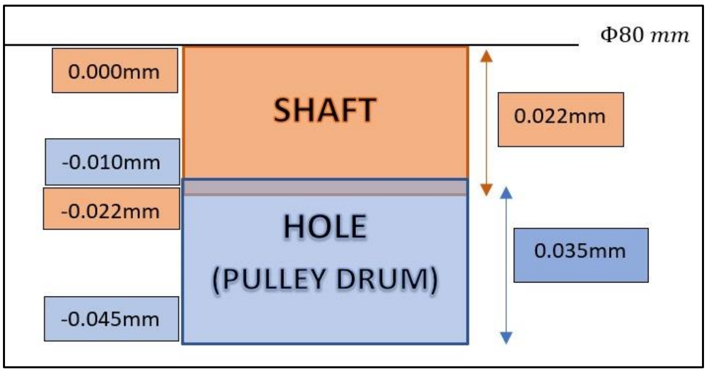


Figure 52: Selective Assembly of the shaft associated with pulley drum

TOLERANCE:

Since the optimised pulley driving shaft is manufactured to a diameter of 80.045 mm with a tolerance of -0.022mm, modified fit techniques need to be applied to determine the hole size of the pulley drum when the shaft tolerance remains the same.

MODIFIED FIT TECHNIQUES

$$Equation\ 1: (LH - SS)_{from\ EDH} = (LH - SS\ )_{component}$$
$$Equation\ 2: (SH - LS)_{from\ EDH} = (SH - LS\ )_{component}$$

(values of SS (Smallest Shaft) = 80.023 mm & LS (Largest Shaft) = 80.045 mm are obtained from basic tolerance shaft in the bearing fit)

$\underline{Equation\ 1:}$

$$\big(-0.010\ -\ (-0.022)\big) = \ (LH\ -\ 80.023)$$
$$LH\ =\ 80.035mm$$

$\underline{Equation\ 2:}$

$$(-0.045 - (0)) = (SH - 80.045)$$
$$LH = 80.00mm$$

$Difference\ between\ the\ LH\ and\ SH = 80.035 - 80.000 \quad = 0.035mm$

$\boldsymbol{Final\ MMC\ format\ for\ hole\ size\ (Pulley\ Drum)} \quad = \boldsymbol{\emptyset 80.000^{+0.035}_{-0}\ mm}$

$\boldsymbol{Final\ MMC\ format\ for\ shaft\ size} \quad = \boldsymbol{\emptyset 80.045^{+0}_{-0.022}\ mm}$

<u>INTEFERENCE FITS CHECK:</u>

$$\begin{aligned} Maximum\ Interference &= Smallest\ Hole\ (SH) - Largest\ Shaft\ (LS) \\ &= 80.000 - 80.045 \\ &= -0.045\ mm \end{aligned}$$

$$\begin{aligned} Maximum\ Interference &= Largest\ Hole\ (LH) - Smallest\ Shaft\ (SS) \\ &= 80.035 - 80.023 \\ &= 0.012\ mm \end{aligned}$$

7.4.3 <u>SHAFT TOLERANCE WITH GEAR CONTROLLING PULLEY</u>

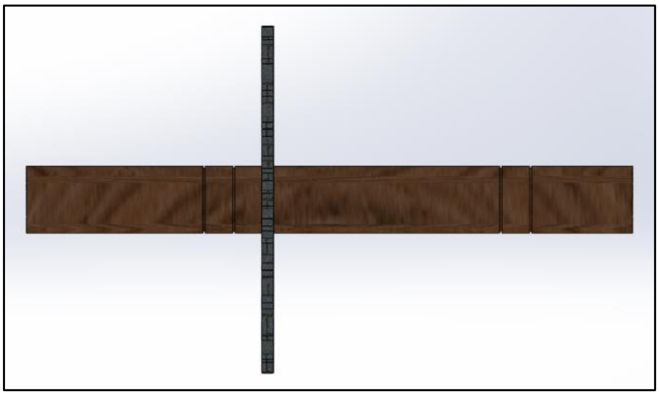


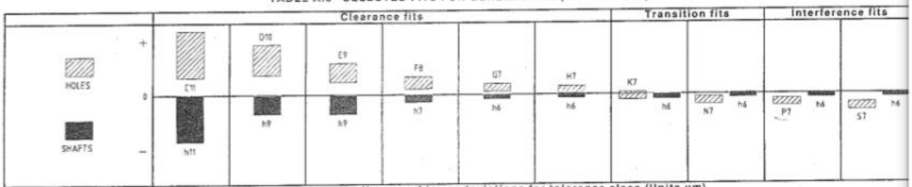
Figure 53: Tolerancing for the Shaft with the Gear Controlling Pulley

H7k6 K7h6	"true transition fit"	•zero clearance when assembled. •can be applied for parts locating (alignment) provided interference is acceptable. •relative rigidity of joint aids in vibration reduction. •parts keyed to shafts like clutch members and hand wheels. •Gudgeon pins in piston heads (red arrows in fig.).

Figure 54: Selection of ISO Tolerance Fit with main application

With reference to Figure 54 above, it can be justified that the main reason for selecting a "K7h6" tolerance fit for the gear connected to the pulley driving shaft is because the gear is keyed to the shaft through a keyhole. Besides that, this fit is selected because the assembly of the gear on the pulley driving shaft has to be very rigid in order to reduce the vibration present in the system, to avoid the gear failing or disassembling itself while being operated. Since the tolerance of the shaft is determined in the previous section 7.4.1, the shaft basis tolerance fit chart is used to determine the tolerance of the inner hole of the gear.

TABLE A.3 SELECTED FITS FOR GENERAL USE (SHAFT BASIS)



Basic size mm		Clearance fits												Transition fits				Interference fits			
		Upper and lower deviations for tolerance class (Units µm)																			
		C11	h11	D10	h9	E9	h9	F8	h7	G7	h6	H7	h6	K7	h6	N7	h6	P7	h6	S7	h6
Above	Up to and incl	+	−	+	−	+	−	+	−	+	−	+	−	+ & −	−	−	−	−	−	−	−
0	3	120 60	0 60	60 20	0 25	39 14	0 25	20 6	0 10	12 2	0 6	10 0	0 6	0 −10	0 6	4 14	0 6	6 16	0 6	14 24	0 6
3	6	145 70	0 75	78 30	0 30	50 20	0 30	28 10	0 12	16 4	0 8	12 0	0 8	+3 −9	0 8	4 16	0 8	8 20	0 8	15 27	0 8
6	10	170 80	0 90	98 40	0 36	61 25	0 36	35 13	0 15	20 5	0 9	15 0	0 9	+5 −10	0 9	4 19	0 9	9 24	0 9	17 32	0 9
10	18	205 95	0 110	120 50	0 43	75 32	0 43	43 16	0 18	24 6	0 11	18 0	0 11	+6 −12	0 11	5 23	0 11	11 29	0 11	21 39	0 11
18	30	240 110	0 130	149 65	0 52	92 40	0 52	53 20	0 21	28 7	0 13	21 0	0 13	+6 −15	0 13	7 28	0 13	14 35	0 13	27 48	0 13
30	40	280 120	0 160	180 80	0 62	112 50	0 62	64 25	0 25	34 9	0 16	25 0	0 16	+7 −18	0 16	8 33	0 16	17 42	0 16	34 59	0 16
40	50	290 130	0 160																		
50	65	330 140	0 190	220 100	0 74	134 60	0 74	76 30	0 30	40 10	0 19	30 0	0 19	+9 −21	0 19	9 39	0 19	21 51	0 19	42 72	0 19
65	80	340 150	0 190																	48 78	0 19
80	100	390 170	0 220	260 120	0 87	159 72	0 87	90 36	0 35	47 12	0 22	35 0	0 22	+10 −25	0 22	10 45	0 22	24 59	0 22	58 93	0 22
100	120	400 180	0 220																	66 101	0 22
120	140	450 200	0 250	305 145	0 100	185 85	0 100	106 43	0 40	54 14	0 25	40 0	0 25	+12 −28	0 25	12 52	0 25	28 68	0 25	77 117	0 25
140	160	460 210	0 250																	85 125	0 25
160	180	480 230	0 250																	93 133	0 25
180	200	530 240	0 290	355 170	0 115	215 100	0 115	122 50	0 46	61 15	0 29	46 0	0 29	+13 −33	0 29	14 60	0 29	33 79	0 29	105 151	0 29
200	225	550 260	0 290																	113 159	0 29
225	250	570 280	0 290																	123 169	0 29
250	280	620 300	0 320	400 190	0 130	240 110	0 130	137 56	0 52	62 17	0 32	52 0	0 32	+16 −36	0 32	14 66	0 32	36 88	0 32	138 190	0 32
280	315	650 330	0 320																	150 202	0 32
315	355	720 360	0 360	440 210	0 140	265 125	0 140	151 62	0 57	75 18	0 36	57 0	0 36	+17 −40	0 36	16 73	0 36	41 98	0 36	169 226	0 36
355	400	760 400	0 360																	187 244	0 36
400	450	840 440	0 400	485 230	0 155	290 135	0 155	165 68	0 63	83 20	0 40	63 0	0 40	+18 −45	0 40	17 80	0 40	45 108	0 40	209 272	0 40
450	500	880 480	0 400																	229 292	0 40

Table 4: Selected ISO Tolerance Fit - Shaft basis

With reference to Table 3 above, the suitable tolerance fit for the gear attached to the pulley driving shaft is found as following:

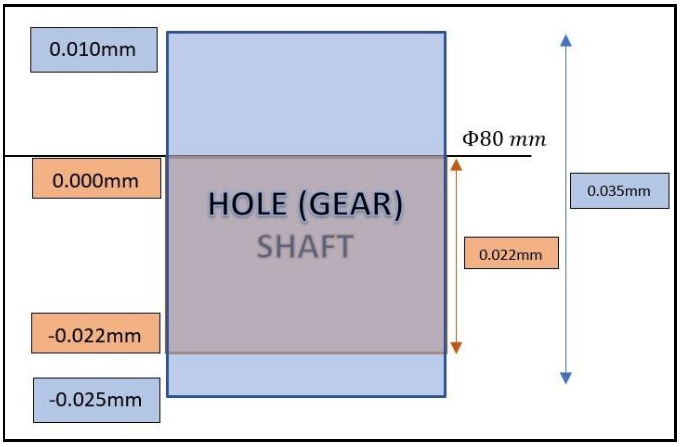


Figure 55: Selective Assembly of the Shaft Associated with Gear

TOLERANCE:

Since the final optimised shaft that has been manufactured to have a diameter of 80.045 mm with a tolerance of -0.022 mm, hence, modified fits techniques need to be applied to determine the hole size of the gear when the shaft tolerance remains the same.

INTEFERENCE FITS CHECK:

$$\begin{aligned} Maximum\ Interference &= Smallest\ Hole\ (SH) - Largest\ Shaft\ (LS) \\ &= 80.020 - 80.045 \\ &= -0.025mm \end{aligned}$$

$$\begin{aligned} Maximum\ Interference &= Largest\ Hole\ (LH) - Smallest\ Shaft\ (SS) \\ &= 80.055 - 80.023 \\ &= 0.032\ mm \end{aligned}$$

Since the value of maximum interference is negative which indicates that the smallest hole is interfering with the largest shaft while the value of minimum interference is positive which indicates that the largest hole and smallest shaft have a small clearance, hence it shows that this fit is transition fit.

7.5 COMPONENT COSTING

Pure AISI 4340 Steel delivered to Malaysia from Alibaba.com costs about RM7.90/kg including delivery charges. An estimate of RM100/hour rate is charged for machining and annealing processes done to produce the pulley driving shaft. According to the costing analysis report obtained via the Solidworks software, the estimated price to fabricate the pulley driving shaft is RM 439.88. This costing analysis report, along with pricing details of the material used can be found in Appendix C.

8.0 GEARS CONTROLLING PULLEY

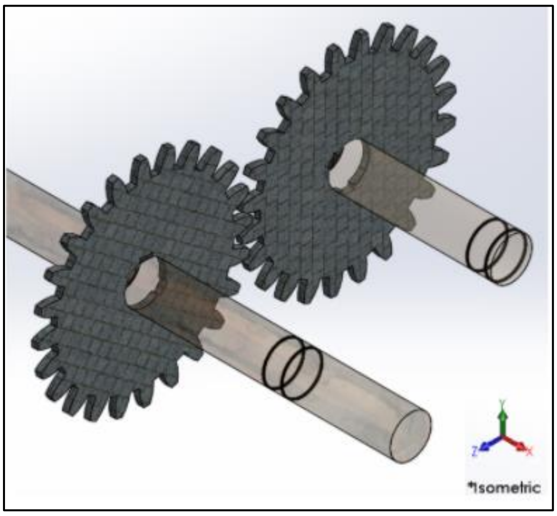


Figure 56: A Pair of Two Optimised Identical Gears Controlling the Pulley

Two meshing gears set-in place to transmit the torque from one shaft to another, with purpose to control the pulley system to hoist the load in the vertical direction. Both the selected spur gears have a 16mm module with 24 teeth on each gear, equating to a 1:1 gear ratio. The pressure angle between both gears is 20°, and the theoretical centre distance between the two gears is 395mm. The spur gears are made from Plain Carbon Steel with a material density of 7800 kg/m^3. The face width of the initially selected gears is 12mm, but upon optimisation, the face width of the gears is increased to 14mm.

8.1 FREE BODY DIAGRAM (FBD) - OPTIMISED GEARS

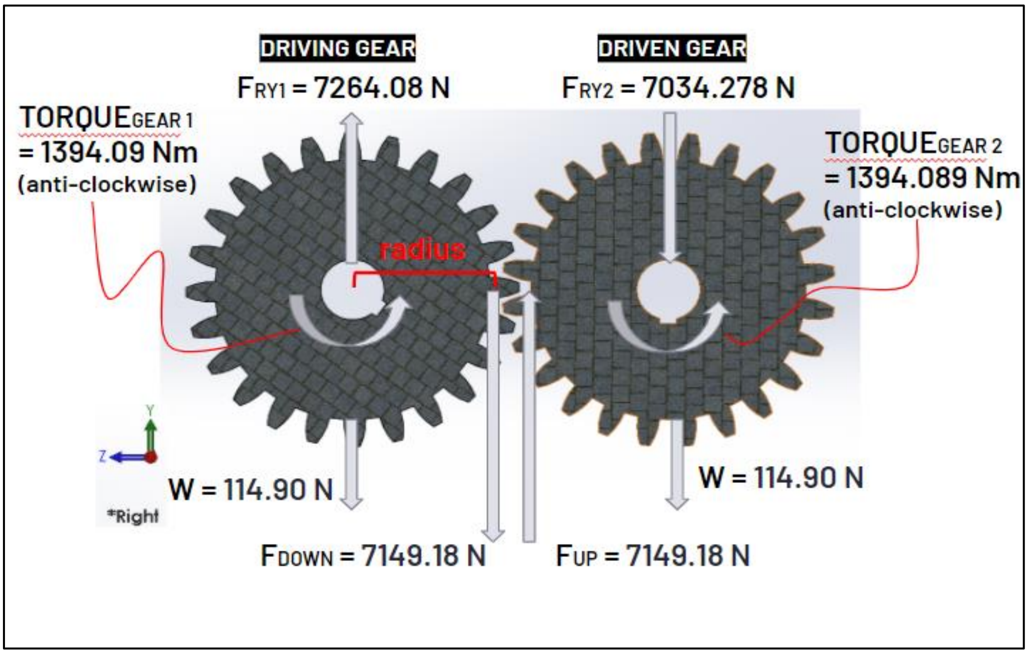


Figure 57: Free Body Diagram of Optimised Gears Controlling Pulley

(Free body diagram and corresponding calculations of the initially selected gears can be found in Appendix D)

8.1.1 FREE BODY DIAGRAM CALCULATIONS - OPTIMISED GEARS

$Assumptions$:

- $Volume\ of\ Each\ Optimised\ Gear\ =\ 0.00150164\ m^3$
- $Density\ of\ Gear\ Material\ (Plain\ Carbon\ Steel) =\ 7800\ kg/m^3$

$$\begin{aligned} Mass\ of\ Each\ Optimised\ Gear\ &=\ m \\ m\ &=\ \rho\ \times V \\ &=\ 7800\ kg/m^3\ \times\ 0.00150164\ m^3 \\ &=\ 11.713\ kg \end{aligned}$$

$$\begin{aligned} Self-Weight\ of\ Each\ Optimised\ Gear\ &=\ W \\ W\ &=\ m\ \times\ g \\ &=\ 11.713\ kg\ \times\ 9.81\ m/s^2 \\ &=\ 114.902\ N \end{aligned}$$

$$\begin{aligned} F_{DOWN}\ &=\ TORQUE_{GEAR1}/\ radius \\ &=\ 1394.089\ Nm\ /\ 0.195\ m \\ &=\ 7149.174\ N \end{aligned}$$

$$\begin{aligned} F_{RY1}\ &=\ W\ +\ F_{DOWN} \\ &=\ 114.902\ N +\ 7149.174\ N \\ &=\ 7264.081\ N \end{aligned}$$

$$F_{DOWN}\ =\ F_{UP}$$

$$\begin{aligned} F_{RY2}\ &=\ F_{UP}\ -\ W \\ &=\ 7149.174\ N\ -\ 114.902\ N \\ &=\ 7034.277\ N \end{aligned}$$

8.2 DESIGN ANALYSIS FRAMEWORK 6 (DAF)

8.2.1 INTERFERENCE CHECK

8.2.1.1 THEORETICAL METHOD

$Provided\ Information$:

$$\begin{aligned}
Gear\ Module &= m \\
m &= 16\ mm \\
Number\ of\ Teeth\ of\ Both\ Gears &= 24\ Teeth \\
Pressure\ Angle\ of\ Both\ Gears &= 20° \\
Gear\ Ratio &= 1{:}\,1 \\
Theoretical\ Centre\ Distance &= c \\
&= 395\ mm
\end{aligned}$$

$Calculated\ Information$:

$$\begin{aligned}
Diametral\ Pitch &= 25.4\ /\ m \\
&= 25.4\ /\ 16\ mm \\
&= 1.5875''
\end{aligned}$$

$$\begin{aligned}
Addendum &= 1''\ /\ P \\
&= 1''\ /\ 1.5875'' \\
&= 0.630''\ \times 25.4\ mm/inch \\
&= 16mm \\
Dedendum &= 1.25''\ /\ P \\
&= 1.25''\ /\ 1.5875'' \\
&= 0.787''\ \times\ 25.4\ mm/inch \\
&= 20\ mm
\end{aligned}$$

$$c\ =\ [\ Gear\ Pitch\ Diameter\ (d_g)\ +\ Pinion\ Pitch\ Diameter\ (d_p)\ ]\ /\ 2$$

$$Since\ the\ gears\ have\ a\ 1{:}\,1\ ratio,\ \ d_g = d_p\ :$$

$$\begin{aligned}
c &= (d_g\ +\ d_g)\ /\ 2 \\
(395\ mm) &= 2d_g\ /\ 2 \\
d_g &= 395\ mm \\
d_p &= 395\ mm
\end{aligned}$$

$$\begin{aligned}
r_g &= d_g\ /\ 2 \\
&= 197.5\ mm
\end{aligned}$$

$$\begin{aligned}
r_p &= d_p\ /\ 2 \\
&= 197.5\ mm
\end{aligned}$$

$$\begin{aligned}
Base\ circle\ radius &= r_{b,p} \\
r_{b,p} &= r_p\ \times\ cos\emptyset \\
&= 197.5\ mm\ \times\ cos(20°) \\
&= 185.60mm
\end{aligned}$$

$$Max\ possible\ addendum\ circle\ radius\ of\ pinion\ w/o\ interference\ =\ r_{a(MAX),p}$$

$$r_{a(MAX),p}\ =\ \sqrt{[\ (r_{b,p})^2\ +\ (c^2 sin^2 \emptyset)\ ]}$$
$$=\ \sqrt{[\ (185.60)^2\ +\ ((395)^2\ x\ sin^2(20°))\ ]}$$
$$=\ \sqrt{46342,49}$$
$$=\ 229.56\ mm$$

$$Addendum\ Radii\ of\ Pinion\ =\ r_{a,p}$$
$$r_{a,p} = r_a\ +\ Addendum$$
$$=\ 197.5\ mm\ + 16mm$$
$$=\ 213.5\ mm$$

$$Since\ both\ gears\ have\ a\ 1{:}1\ ratio:$$

$$r_{a(MAX),p}\ =\ r_{a(MAX),g}\ =\ r_{a(MAX)}$$
$$r_{a,p}\ =\ r_{a,g}\ =\ r_a$$

Since $r_a < r_{a(MAX)}$ there is **NO INTERFERENCE** in both the meshing pair of gears.

8.2.1.2 GRAPHICAL METHOD

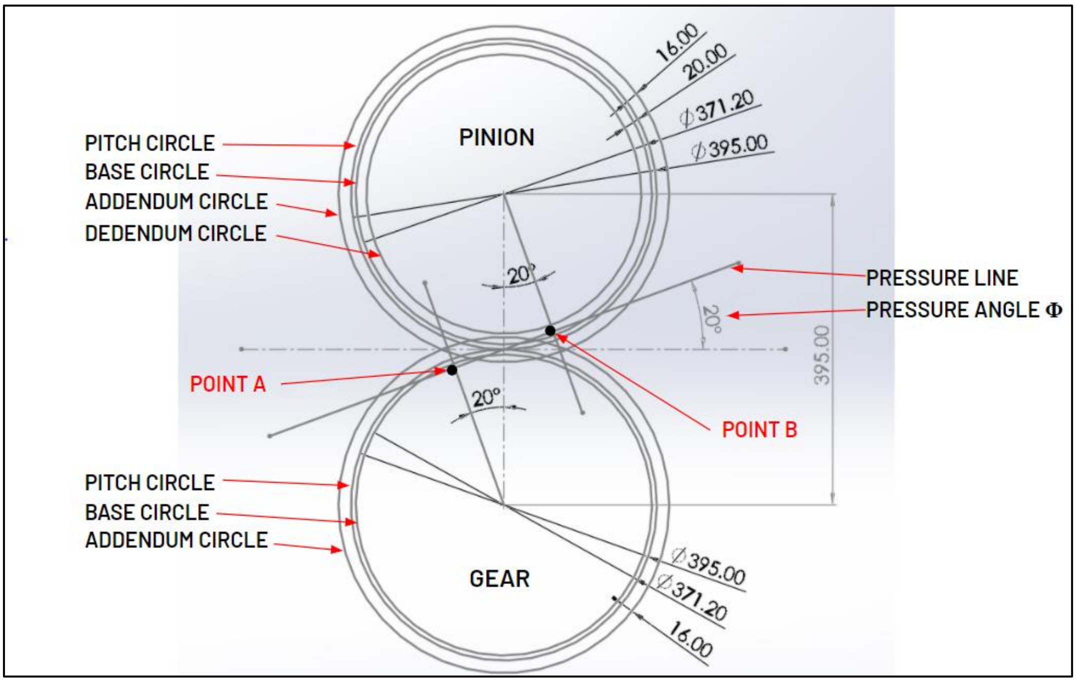


Figure 58: Graphical Method to Check Gears for Interference

Since both Point A and Point B do not cross the pinion's addendum circle and the gear's addendum circle respectively, both meshing gears will experience no interference, and there will be no prevention to the rotation of the gears.

8.2.2 STRESS ANALYSIS ON GEAR TEETH

$$TORQUE_{GEAR1} = 1394.089\ Nm$$

$$\begin{aligned} F_t &= TORQUE_{GEAR1} \ /\ \text{radius} \\ &= 1394.089\ Nm \ /\ 0.1975\ m \\ &= 7058.678\ N \end{aligned}$$

$$\begin{aligned} Module &= m \\ m &= 16\ mm \end{aligned}$$

$$\begin{aligned} Gear\ Face\ Width &= b \\ b &= 12mm \end{aligned}$$

Since both gears have 24 teeth on each of them, and they have a 20° pressure angle, this information can be used to find the Lewis Form Factor Y from the graph below [2]. The corresponding Y value found in the graph below is 0.325.

$$Y = 0.325$$

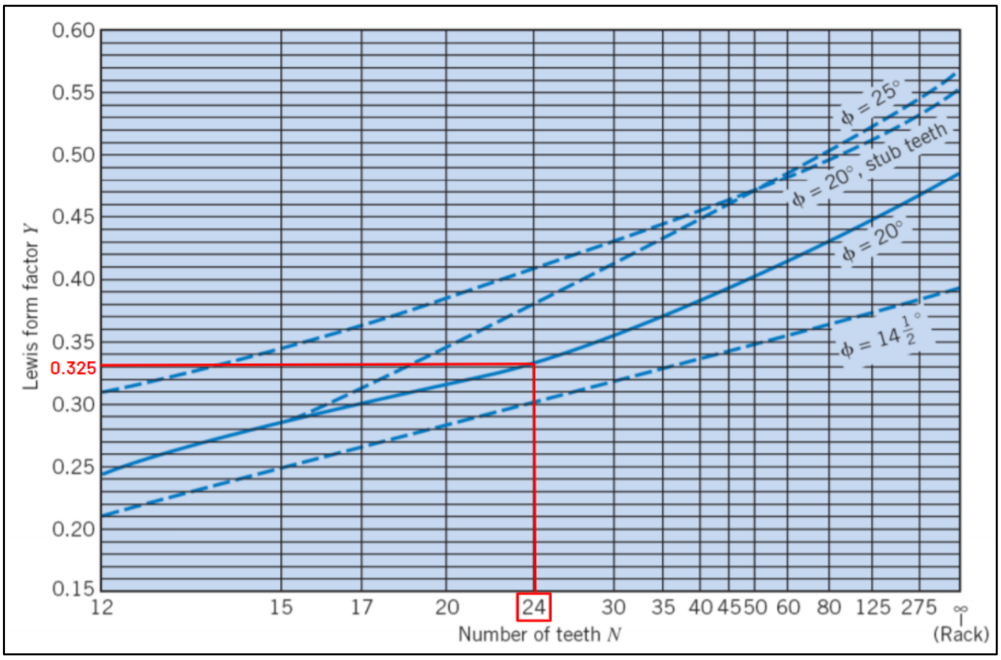


Figure 59: Graph of Lewis Form Factor Y Against Number of Teeth N

$$\begin{aligned} Maximum\ applied\ stress\ on\ gear\ tooth &= \sigma_{MAX} \\ \sigma_{MAX} &= F_t / (m \times b \times Y) \\ &= (7058.678\ N) \ /\ (16mm \times 12\ mm \times 0.325) \\ &= 113.120\ MPa \end{aligned}$$

Since the gear is made out of Plain Carbon Steel, it's σ_Y = 490MPa, the corresponding safety factor is:

$$\begin{aligned} \sigma_Y \ /\ Safety\ Factor &\leq \sigma_{MAX} \\ Safety\ Factor &\geq \sigma_Y \ /\ \sigma_{MAX} \\ &\geq (490\ MPa \ /\ 113.120\ MPa) \\ &\geq 4.332 \\ Minimum\ Safety\ Factor &= 4.332 \end{aligned}$$

8.3 OPTIMISATION

Based on the above calculations, it is found that the safety factor for the two meshing spur gears used is 3.964, which is considered to be low and can cause safety issues. To achieve a safety factor of about 5, the gear's face width, b is increased from 12mm to 14mm:

$$
\begin{aligned}
Maximum\ applied\ stress\ on\ gear\ tooth &= \sigma_{MAX} \\
\sigma_{MAX} &= F_t / (m \times b \times Y) \\
&= (7058.678\ N) / (16mm \times 14\ mm \times 0.325) \\
&= 96.960\ Mpa
\end{aligned}
$$

$$
\begin{aligned}
\sigma_Y\ /\ Safety\ Factor &\leq \sigma_{MAX} \\
Safety\ Factor &\geq \sigma_Y\ /\ \sigma_{MAX} \\
&\geq (490\ MPa / 96.960\ MPa) \\
&\geq 5.054 \\
Minimum\ Safety\ Factor &= 5.054
\end{aligned}
$$

With the changes made to the gear's face width from 12mm to 14mm, as shown in Figure 60 below, the gears' new safety factor is 5.054. This satisfies the current requirement to produce a minimum safety factor greater than 5.

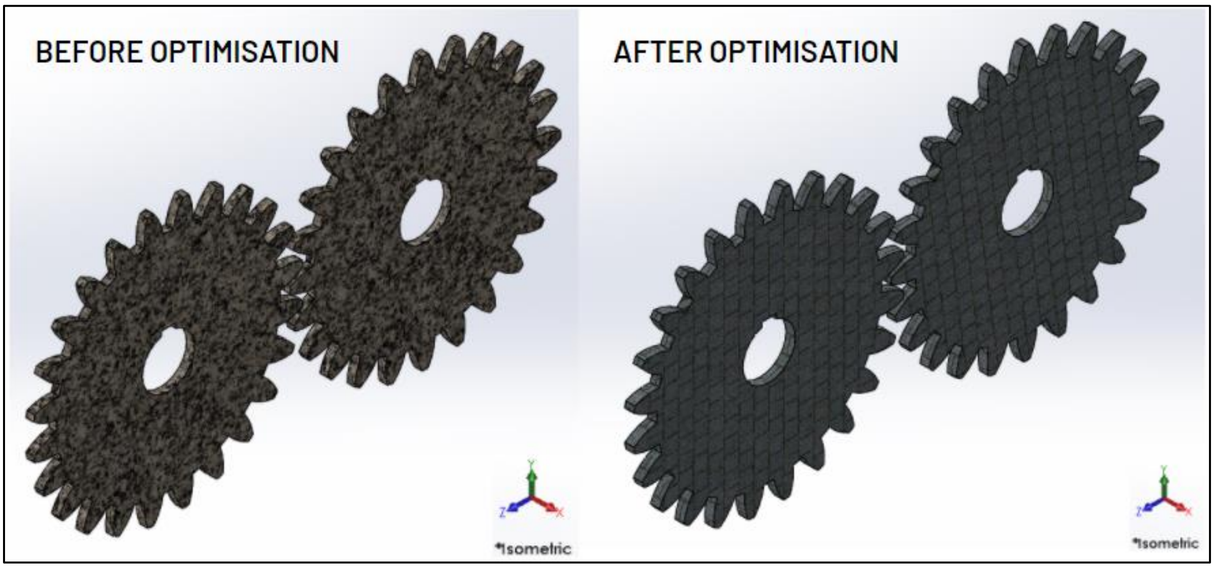


Figure 60: Comparison of Gear Face Width Before & After Optimisation

8.4 COMPONENT COSTING

Pure Carbon Steel delivered to Malaysia from Alibaba.com costs about RM2.29/kg including delivery charges. An estimate of RM100/hour rate is charged for machining processes done to produce the spur gear. According to the costing analysis report obtained via the Solidworks software, the estimated price to fabricate a single spur gear of a 16mm module with 24 teeth on each gear and pressure angle of 20° is RM 427.94. This costing analysis report, along with pricing details of the material used can be found in Appendix C.

9.0 BEARING

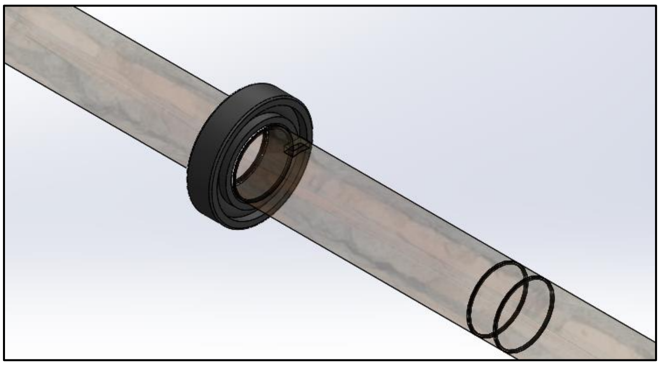


Figure 61: A Ball Bearing Placed Over the Pulley Driving Shaft

A 300 series radial ball bearing with a bearing basic number of #316 is located on the pulley driving shafts, responsible to constrain the shaft's relative motion from influencing the hoist's main casing. Since the shaft has a diameter of 80mm, the initially selected bearing also has an inner diameter of 80mm. This pre-made bearing is made from Chrome Stainless Steel. Since the self-weight of the bearing is very insignificant compared to the forces acting on it, the self-weight of the bearing was assumed to be negligible when analysing the component.

9.1 FREE BODY DIAGRAM (FBD) - OPTIMISED BEARING

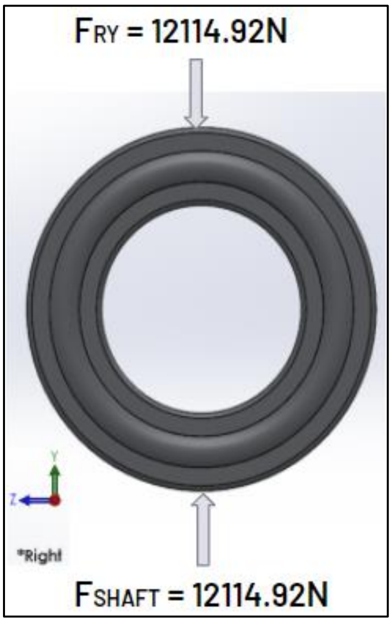


Figure 62: Free Body Diagram of Bearing

9.1.1 FREE BODY DIAGRAM CALCULATIONS - OPTIMISED BEARING

$Assumptions$:

- $Since\ the\ selfweight\ of\ the\ bearing\ is\ very\ small\ relative\ to\ the\ force\ transmitted\ to\ the\ bearing\ from\ the\ shaft, the\ self-weight\ of\ the\ bearing\ is\ negligible$

$$\begin{aligned} F_{RY} &= F_{SHAFT} \\ &= 12114.92\ N \end{aligned}$$

9.2 DESIGN ANALYSIS FRAMEWORK 7 (DAF)

9.2.1 BEARING SELECTION

Since the bearing is only subjected to a radial dynamic load, the load required for this application, C_{req} equals to:

$$C_{req} = F_r \times K_a \times (L / K_r \times L_r)^{0.3}$$
$$Where\ Fr = 13213.82\ N$$
$$L_r = 90 \times 10^6\ cycles$$

For a ball bearing experiencing heavy impact due to the constant loading and unloading of concrete blocks to the hoist, the estimated Application Factors K_a according to Table 5 below is [2]:

$$K_a = 2.5$$

TABLE 14.3 Application Factors K_a

Type of Application	Ball Bearing	Roller Bearing
Uniform load, no impact	1.0	1.0
Gearing	1.0–1.3	1.0
Light impact	1.2–1.5	1.0–1.1
Moderate impact	1.5–2.0	1.1–1.5
Heavy impact	2.0–3.0	1.5–2.0

Table 5: Application Factors K_aTable

For a hoist that is in use for 8-hours a day on every workday, the estimated Design Life L according to Table 6 below is [2]:

$$L = 30000\ hours \times (60\ minutes / 1\ hour) \times (6.174\ revolutions / min)$$
$$= 11113200\ revolutions$$

TABLE 14.4 Representative Bearing Design Lives

Type of Application	Design Life (thousands of hours)
Instruments and apparatus for infrequent use	0.1–0.5
Machines used intermittently, where service interruption is of minor importance	4–8
Machines intermittently used, where reliability is of great importance	8–14
Machines for 8-hour service, but not every day	14–20
Machines for 8-hour service, every working day	20–30
Machines for continuous 24-hour service	50–60
Machines for continuous 24-hour service where reliability is of extreme importance	100–200

Table 6: Representative Bearing Design Lives Table

For a ball bearing with a 99% reliability, the corresponding Life Adjustment Reliability Factor K_r according to Figure 63 below is [2]:

$$K_r = 0.2$$

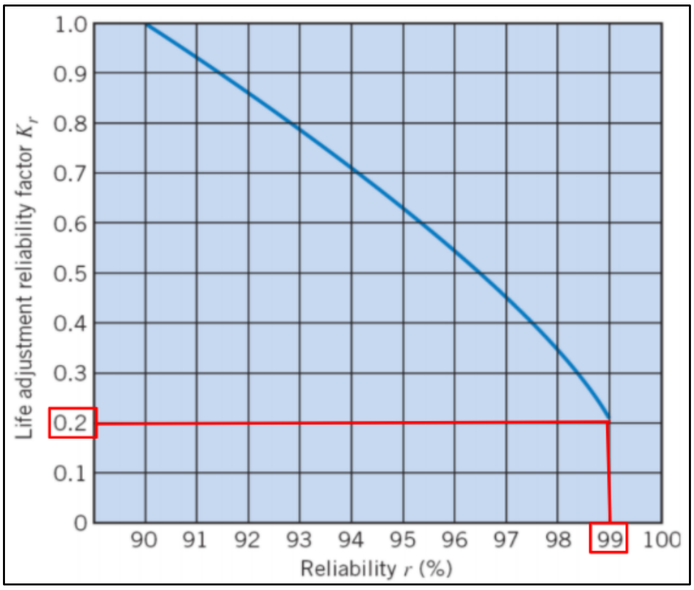


Figure 63: Graph of Life Adjustment Reliability Factor K_r vs Reliability r (%)

Thus:

$$
\begin{aligned}
C_{req} &= F_r \times K_a \times (L / K_r \times L_r)^{0.3} \\
&= (12114.92\ N) \times (2.5) \times (11113200 / 0.2 \times 90 \times 10^6\ cycles)^{0.3} \\
&= 26207.80\ N \\
&= 26.208\ kN
\end{aligned}
$$

From Table 7 below, the Bearing Rated Capacity, C of the chosen 300 series ball bearing with a bore of 80 mm is 28.0 kN [2]. This is close to, and greater than the load required for this application, C_{req} which is calculated to be 26.208. Thus, the chosen 300 series ball bearing with a bore of 80mm is suitable to be used for the hoisting application. The corresponding bearing basic number for the chosen bearing is #316, as seen in Table 8 below [2].

TABLE 14.2 Bearing Rated Capacities, C, for 90×10^6 Revolution Life with 90 Percent Reliability

	Radial Ball, $\alpha = 0°$			Angular Ball, $\alpha = 25°$			Roller		
Bore (mm)	L00 Xlt (kN)	200 lt (kN)	300 med (kN)	L00 Xlt (kN)	200 lt (kN)	300 med (kN)	1000 Xlt (kN)	1200 lt (kN)	1300 med (kN)
10	1.02	1.42	1.90	1.02	1.10	1.88			
12	1.12	1.42	2.46	1.10	1.54	2.05			
15	1.22	1.56	3.05	1.28	1.66	2.85			
17	1.32	2.70	3.75	1.36	2.20	3.55	2.12	3.80	4.90
20	2.25	3.35	5.30	2.20	3.05	5.80	3.30	4.40	6.20
25	2.45	3.65	5.90	2.65	3.25	7.20	3.70	5.50	8.50
30	3.35	5.40	8.80	3.60	6.00	8.80	2.40[a]	8.30	10.0
35	4.20	8.50	10.6	4.75	8.20	11.0	3.10[a]	9.30	13.1
40	4.50	9.40	12.6	4.95	9.90	13.2	7.20	11.1	16.5
45	5.80	9.10	14.8	6.30	10.4	16.4	7.40	12.2	20.9
50	6.10	9.70	15.8	6.60	11.0	19.2	5.10[a]	12.5	24.5
55	8.20	12.0	18.0	9.00	13.6	21.5	11.3	14.9	27.1
60	8.70	13.6	20.0	9.70	16.4	24.0	12.0	18.9	32.5
65	9.10	16.0	22.0	10.2	19.2	26.5	12.2	21.1	38.3
70	11.6	17.0	24.5	13.4	19.2	29.5		23.6	44.0
75	12.2	17.0	25.5	13.8	20.0	32.5		23.6	45.4
$\boxed{80}$	14.2	18.4	$\boxed{28.0}$	16.6	22.5	35.5	17.3	26.2	51.6

Table 7: Bearing Rated Capacity, C Table

TABLE 14.1 Bearing Dimensions (*continued*)

Bearing Basic Number	Bore (mm)	Ball Bearings OD (mm)	w (mm)	r^a (mm)	d_S (mm)	d_H (mm)	Roller Bearings OD (mm)	w (mm)	r^a (mm)	d_S (mm)	d_H (mm)
L16	80	125	22	1.02	88.1	116.3	125	22	2.03	88.4	117.6
216	80	140	26	2.03	93.2	126.7	140	26	2.54	91.2	129.3
316	80	170	39	2.03	98.6	152.9	170	39	3.18	96.0	154.4
L17	85	130	22	1.02	93.2	121.4	130	22	2.03	93.5	122.7
217	85	150	28	2.03	99.1	135.6	150	28	3.18	98.0	139.2
317	85	180	41	2.54	105.7	160.8	180	41	3.96	102.9	164.3
L18	90	140	24	1.52	99.6	129.0	140	24	Not Available		
218	90	160	30	2.03	104.4	145.5	160	30	3.18	103.1	147.6
318	90	190	43	2.54	111.3	170.2	190	43	3.96	108.2	172.7
L19	95	145	24	1.52	104.4	134.1	145	24	Not Available		
219	95	170	32	2.03	110.2	154.9	170	32	3.18	109.0	157.0
319	95	200	45	2.54	117.3	179.3	200	45	3.96	115.1	181.9
L20	100	150	24	1.52	109.5	139.2	150	24	2.54	109.5	141.7
220	100	180	34	2.03	116.1	164.1	180	34	3.96	116.1	167.1
320	100	215	47	2.54	122.9	194.1	215	47	4.75	122.4	194.6
L21	105	160	26	2.03	116.1	146.8	160	26	Not Available		
221	105	190	36	2.03	121.9	173.5	190	36	3.96	121.4	175.3
321	105	225	49	2.54	128.8	203.5	225	49	4.75	128.0	203.5
L22	110	170	28	2.03	122.7	156.5	170	28	2.54	121.9	159.3
222	110	200	38	2.03	127.8	182.6	200	38	3.96	127.3	183.9
322	110	240	50	2.54	134.4	218.2	240	50	4.75	135.9	217.2

Table 8: Bearing Dimensions Table

Since the bearing is selected to have a bearing basic number of #316, the exact Design Life, L_{exact} of the chosen bearing can now be calculated. This can be done using the below equation again:

$$C_{req} = F_r \times K_a \times (L / K_r \times L_r)^{0.3}$$

$$As\ Found\ Previously,\ Fr = 12114.92\ N$$

$$L_r = 90 \times 10^6\ cycles$$

$$K_a = 2.5$$

$$K_r = 0.2$$

Since the chosen bearing has a Bearing Rated Capacity, C of 28.0 kN:

$$C = C_{req}$$

$$C_{req} = 28.0\ kN$$

$$C_{req} = F_r \times K_a \times (L / K_r \times L_r)^{0.3}$$

$$28000\ N = (12114.92\ N \times (2.5) \times (L_{exact} / 0.2\ x\ 90\ x\ 10^6\ cycles)^{0.3}$$

$$L_{exact} = 13854709.1\ revolutions \times (1\ hour\ 60\ minutes) \times (0.16197\ min\ /revolutions)$$

$$= 37400.7\ hours$$

Comparing the initially estimated bearing Design Life, L and the newly calculated exact bearing Design Life, L_{exact}, the difference between the two design life periods equals the buffer time:

$$L_{exact} - L = Buffer\ Time$$

$$Buffer\ Time = 37400.7\ hours - 30000\ hours$$

$$= 7400.7\ hours$$

The buffer time calculated above refers to the amount of time that a maintenance engineer must replace a bearing by, after it reaches its Design Life, L, and before it actually fails. This means that after the bearing has operated for 30000 hours, it must be replaced in the subsequent 7400.7 hours before it fails.

9.3 BEARING TOLERANCE

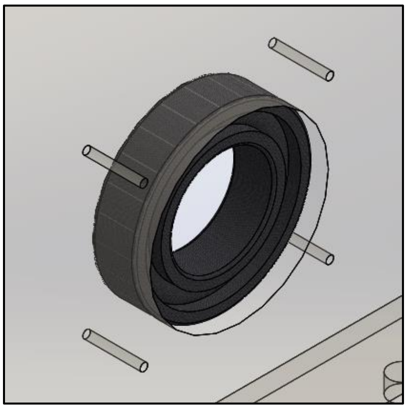


Figure 64: Bearing Held in Place in The Main Casing of The Hoist

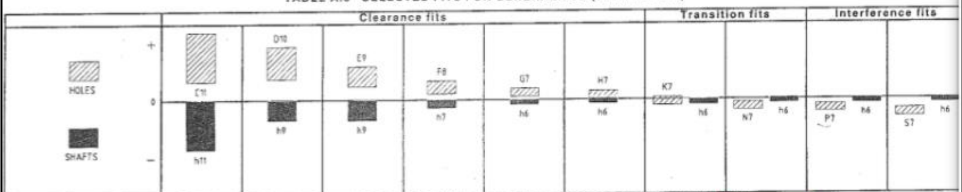
The bearings located on the pulley driving shaft are fit into custom holes made through the main casing of the hoist, that acts as the bearing's housing. The outer diameter of the specific bearing is 170mm, and a "N7h6" tolerance fit, also known as the "interference transition fit", is selected as seen below in Figure 65 [2].

H7n6 N7h6	"interference transition fit"	•used for tight assembly of components where vibration is not tolerated. •generally result in a push fit. •keyed assembly of parts such as high speed fans and motor parts.

Figure 65: Selection of ISO Tolerance Fit

One of the reasons why the "N7h6" tolerance fit was selected is because the selected bearing is required to be tightly assembled through the hole of the hoist's main casing, to reduce the vibrations in the system. Presence of vibrations in the system can cause the hoist system to be inefficient, and can even fail eventually. Besides that, the selected tolerance fit also results in a push fit, which is required in this application. Since the bearing that acts as a shaft in this case has a fixed outer diameter dimension, the size of the hole made onto the hoist's main casing is varied to determine the true interference fit.

TABLE A.3 SELECTED FITS FOR GENERAL USE (SHAFT BASIS)



Basic size mm		Clearance fits												Transition fits				Interference fits			
		Upper and lower deviations for tolerance class (Units µm)																			
		C11	h11	D10	h9	E9	h9	F8	h7	G7	h6	H7	h6	K7	h6	N7	h6	P7	h6	S7	h6
Above	Up to and incl	+	−	+	−	+	−	+	−	+	−	+	−	+ & −	−	−	−	−	−	−	−
0	3	120 60	0 60	60 20	0 25	39 14	0 25	20 6	0 10	12 2	0 6	10 0	0 6	0 −10	0 6	4 14	0 6	6 16	0 6	14 24	0 6
3	6	145 70	0 75	78 30	0 30	50 20	0 30	28 10	0 12	16 4	0 8	12 0	0 8	+3 −9	0 8	4 16	0 8	8 20	0 8	15 27	0 8
6	10	170 80	0 90	98 40	0 36	61 25	0 36	35 13	0 15	20 5	0 9	15 0	0 9	+5 −10	0 9	4 19	0 9	9 24	0 9	17 32	0 9
10	18	205 95	0 110	120 50	0 43	75 32	0 43	43 16	0 18	24 6	0 11	18 0	0 11	+6 −12	0 11	5 23	0 11	11 29	0 11	21 39	0 11
18	30	240 110	0 130	149 65	0 52	92 40	0 52	53 20	0 21	28 7	0 13	21 0	0 13	+6 −15	0 13	7 28	0 13	14 35	0 13	27 48	0 13
30	40	280 120	0 160	180 80	0 62	112 50	0 62	64 25	0 25	34 9	0 16	25 0	0 16	+7 −18	0 16	8 33	0 16	17 42	0 16	34 59	0 16
40	50	290 130	0 160																		
50	65	330 140	0 190	220 100	0 74	134 60	0 74	76 30	0 30	40 10	0 19	30 0	0 19	+9 −21	0 19	9 39	0 19	21 51	0 19	42 72	0 19
65	80	340 150	0 190																	48 78	0 19
80	100	390 170	0 220	260 120	0 87	159 72	0 87	90 36	0 35	47 12	0 22	35 0	0 22	+10 −25	0 22	10 45	0 22	24 59	0 22	58 93	0 22
100	120	400 180	0 220																	66 101	0 22
120	140	450 200	0 250	305 145	0 100	185 85	0 100	106 43	0 40	54 14	0 25	40 0	0 25	+12 −28	0 25	12 52	0 25	28 68	0 25	77 117	0 25
140	160	460 210	0 250																	85 125	0 25
160	180	480 230	0 250																	93 133	0 25
180	200	530 240	0 290	355 170	0 115	215 100	0 115	122 50	0 46	61 15	0 29	46 0	0 29	+13 −33	0 29	14 60	0 29	33 79	0 29	105 151	0 29
200	225	550 260	0 290																	113 159	0 29
225	250	570 280	0 290																	123 169	0 29
250	280	620 300	0 320	400 190	0 130	240 110	0 130	137 56	0 52	62 17	0 32	52 0	0 32	+16 −36	0 32	14 66	0 32	36 88	0 32	138 190	0 32
280	315	650 330	0 320																	150 202	0 32
315	355	720 360	0 360	440 210	0 140	265 125	0 140	151 62	0 57	75 18	0 36	57 0	0 36	+17 −40	0 36	16 73	0 36	41 98	0 36	169 226	0 36
355	400	760 400	0 360																	187 244	0 36
400	450	840 440	0 400	485 230	0 155	290 135	0 155	165 68	0 63	83 20	0 40	63 0	0 40	+18 −45	0 40	17 80	0 40	45 108	0 40	209 272	0 40
450	500	880 480	0 400																	229 292	0 40

Table 9: Selected ISO Tolerance Fit – Shaft Basis

Since the bearing has an outer diameter of 170mm, Table 9 above is used to find the correct tolerance fit for the outer diameter of the bearing corresponding to the hole of the main casing/housing holding it, as following:

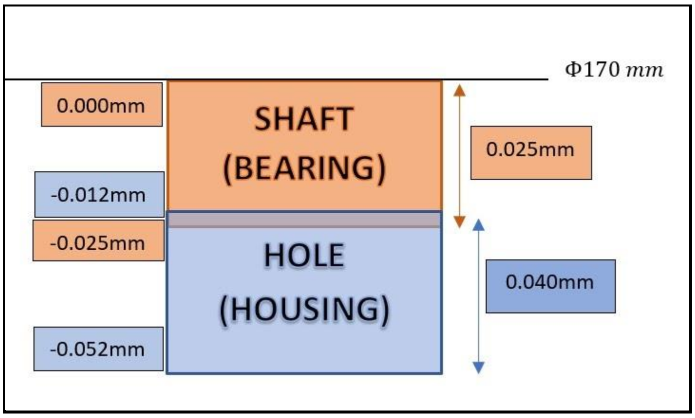


Figure 66: Selected Tolerance Fit Assembly for Bearing & Main Casing

TOLERANCE:

$\boldsymbol{Hole\ (Housing)}$

$Maximum\ diameter\ of\ hole\ (Housing) = 170 - 0.012 \quad = 169.988\ mm$

$Minimum\ diameter\ of\ hole\ (Housing) = 170 - 0.052 \quad = 169.948\ mm$

$Difference\ between\ the\ LH\ and\ SH = 169.988 - 169.948 = 0.040\ mm$

$\boldsymbol{MMC\ format\ for\ hole\ size\ (Housing)} \quad = \boldsymbol{\emptyset 169.955^{+0.040}_{-0}\ mm}$

$\boldsymbol{Shaft\ (Bearing)}$

$Maximum\ diameter\ of\ shaft\ (Bearing) = 170 - 0 \quad = 170\ mm$

$Minimum\ diameter\ of\ shaft\ (Bearing) = 170 - 0.025 \quad = 169.975\ mm$

$Difference\ between\ the\ LS\ and\ SS = 170 - 169.975 \quad = 0.025\ mm$

$\boldsymbol{MMC\ format\ for\ shaft\ size\ (Bearing)} \quad = \boldsymbol{\emptyset 170^{+0}_{-0.025}\ mm}$

INTEFERENCE FITS CHECK:

$$\begin{aligned} Maximum\ Interference &= Smallest\ Hole\ (SH) - Largest\ Shaft\ (LS) \\ &= 169.948 - 170.000 \\ &= -0.052\ mm \end{aligned}$$

$$\begin{aligned} Maximum\ Interference &= Largest\ Hole\ (LH) - Smallest\ Shaft\ (SS) \\ &= 169.988 - 169.975 \\ &= 0.013\ mm \end{aligned}$$

9.4 COMPONENT COSTING

Ball bearings with a basic number of #316 cost RM 847.95 each when delivered to Malaysia from a company called RS Pro, including delivery charges. Since four #316 ball bearings are used in the designed hoist, the total cost spent on the #316 bearings is RM 3391.80. Catalogues and pricing details of the bearings can be found in Appendix B.

10.0 FASTENER

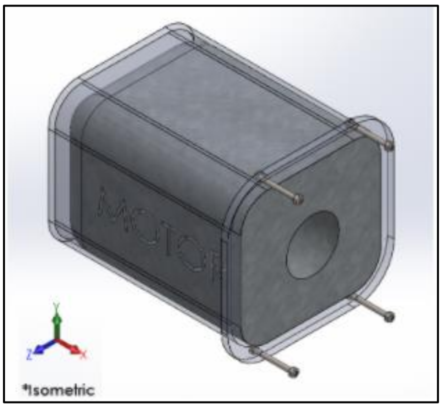


Figure 67: A Motor & Its Casing Is Supported By 4 Fasteners

Four threaded fasteners are in place to support a motor and it's casing with a combined load of 140.48 kg. Each threaded fastener is subjected to both tensile forces and shear forces. The initial fasteners selected are a Hex M8 bolt of SAE Class 4.6, but upon optimisation, the fasteners were replaced with Hex M6 bolt of SAE Class 4.6.

10.1 FREE BODY DIAGRAM (FBD) - OPTIMISED FASTENER

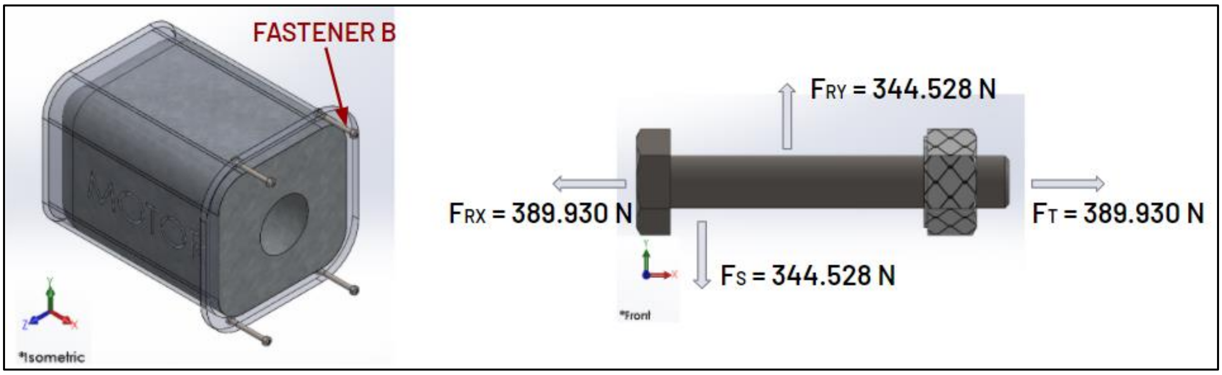


Figure 68: Free Body Diagram of Fastener B

10.1.1 FREE BODY DIAGRAM CALCULATIONS - OPTIMISED FASTENER

Assumptions:

- *Since the selfweight of the fastener is very small relative to the force transmitted to the fastener from the motor and its casing, the selfweight of the fastener is negligible*

$$Tensile\ Force\ on\ Fastener\ B: 389.930\ N \times Safety\ Factor$$
$$Shear\ Force\ on\ Fastener\ B: 344.528\ N \times Safety\ Factor$$

(Refer to Section 10.2.1 on page 54 for the values above)

To measure actual forces on Fastener B, assume Safety Factor = 1:

$$F_T = 389.930\ N$$
$$F_S = 344.528\ N$$

10.2 DESIGN ANALYSIS FRAMEWORK 8 (DAF)

Assumptions:

- *Bolt Chosen*: *M8 Hex Head Bolt with SAE Class of* 4.6
- *The strain imposed on the two bolts at the top are two times of that imposed on the two bolts at the bottom*

$$Stress\ Area, A_t\ of\ Chosen\ Bolt\ (Refer\ to\ Table\ 11\ below) = 36.6\ mm^2$$
$$Proof\ Load, S_p\ of\ Chosen\ Bolt\ (Refer\ to\ Table\ 12\ below) = 225\ MPa$$

TABLE 10.2 Basic Dimensions of ISO Metric Screw Threads

	Coarse Threads			Fine Threads		
Nominal Diameter d (mm)	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm^2)	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm
3	0.5	2.39	5.03			
3.5	0.6	2.76	6.78			
4	0.7	3.14	8.78			
5	0.8	4.02	14.2			
6	1	4.77	20.1			
7	1	5.77	28.9			
8	1.25	6.47	36.6	1	6.77	39.2
10	1.5	8.16	58.0	1.25	8.47	61.2
12	1.75	9.85	84.3	1.25	10.5	92.1
14	2	11.6	115	1.5	12.2	125
16	2	13.6	157	1.5	14.2	167
18	2.5	14.9	192	1.5	16.2	216
20	2.5	16.9	245	1.5	18.2	272
22	2.5	18.9	303	1.5	20.2	333
24	3	20.3	353	2	21.6	384
27	3	23.3	459	2	24.6	496
30	3.5	25.7	561	2	27.6	621
33	3.5	28.7	694	2	30.6	761
36	4	31.1	817	3	32.3	865
39	4	34.1	976	3	35.3	1030

Note: Metric threads are identified by diameter and pitch as "M8 × 1.25."

Table 10: Basic Dimensions of ISO Metric Screw Threads

TABLE 10.5 Specifications for Steel Used in Millimeter Series Screws and Bolts

SAE Class	Diameter d (mm)	Proof Load (Strength)[a] S_p (MPa)	Yield Strength[b] S_y (MPa)	Tensile Strength S_u (MPa)	Elongation, Minimum (%)	Reduction of Area, Minimum (%)	Core Hardness, Rockwell Min	Core Hardness, Rockwell Max
4.6	5 thru 36	225	240	400	22	35	B67	B87
4.8	1.6 thru 16	310	—	420	—	—	B71	B87
5.8	5 thru 24	380	—	520	—	—	B82	B95
8.8	17 thru 36	600	660	830	12	35	C23	C34
9.8	1.6 thru 16	650	—	900	—	—	C27	C36
10.9	6 thru 36	830	940	1040	9	35	C33	C39
12.9	1.6 thru 36	970	1100	1220	8	35	C38	C44

[a] Proof load (strength) corresponds to the axially applied load that the screw or bolt must withstand without permanent set.
[b] Yield strength corresponds to 0.2 percent offset measured on machine test specimens.
Source: Society of Automotive Engineers standard J1199 (1979).

Table 11: Specifications for Steel Used in Millimetre Series Screws and Bolts

10.2.1 <u>FORCE ANALYSIS</u>

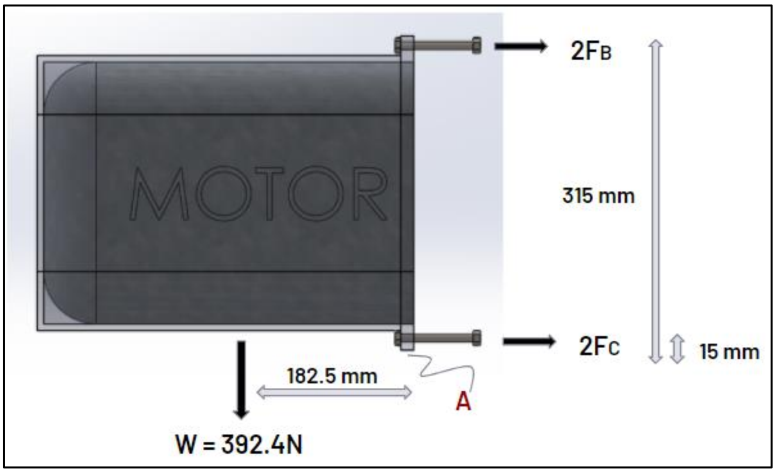


Figure 69: Force Analysis on Bolts Supporting the Motor and Its Casing

Weight of Load:

$$\begin{aligned} m &= combined\ weight\ of\ motor\ and\ housing \\ &= 130\ kg + 10.48\ kg \\ &= 140.48\ kg \end{aligned}$$

$$\begin{aligned} W &= combined\ weight\ of\ motor\ and\ housing \\ &= m \times g \\ &= 1378.11\ N \end{aligned}$$

Tensile:

$$\begin{aligned} \Sigma M@A &= 0 \\ 2\ F_B\ (315\ mm) + 2\ F_C\ (15\ mm) &= W\ (182.5\ mm) \times Safety\ Factor \end{aligned}$$

$$\begin{aligned} Since\ F_B &= 2\ F_C: \\ 2\ (2F_C)(315\ mm) + 2FC\ (15\ mm) &= W\ (182.5\ mm) \times Safety\ Factor \\ 1290\ F_C &= 251504.075\ Nmm \times Safety\ Factor \\ F_C &= 194.965 \times Safety\ Factor \end{aligned}$$

$$\begin{aligned} F_B &= 2\ x\ F_C \\ F_B &= 389.930\ N \times Safety\ Factor \end{aligned}$$

Shear:

$$\begin{aligned} Direct\ Shear &= (\ W\ /\ 4\ ) \times Safety\ Factor \\ &= (1378.11\ N\ /\ 4) \times Safety\ Factor \\ &= 344.528 \times Safety\ Factor \end{aligned}$$

10.2.2 SHEAR & TENSILE FORCE EQUIVALENT TENSILE STRESS ANALYSIS

$$\sigma_e = \sqrt{({\sigma_e}^2 + 3\tau^2)} \leq S_p$$

$$\sqrt{(389.930\ N \times Safety\ Factor\ /\ A_t)^2 + 3(\ 344.528 \times Safety\ Factor\ /\ A_t)^2} \leq S_p$$

$$Substitute\ in\ A_t = 36.6mm^2,\ S_p \leq 225\ MPa:$$

$$\sqrt{(113.504\ x\ Safety\ Factor^2) + (\ 265.833\ x\ Safety\ Factor^2)} \leq 225\ MPa$$

$$379.33\ x\ Safety\ Factor^2 \leq 50625$$

$$Safety\ Factor^2 \geq 133.457$$

$$Safety\ Factor \geq 11.552$$

$$Minimum\ Safety\ Factor = 11.552$$

10.3 OPTIMISATION

Based on the above calculations, it is found that the minimum safety factor for the M8 bolt of SAE Class 4.6 used is 11.552, which is quite high. To achieve a minimum safety factor of 5, a new selection of bolts of different sizing is done by using the following calculations:

$$Safety\ Factor = 5$$

$$\sqrt{(389.930\ N\ x\ Safety\ Factor\ /\ A_t)^2 + 3(\ 344.528\ x\ Safety\ Factor\ /\ A_t)^2} \leq S_p$$

$$\sqrt{(1949.65\ N\ /\ A_t)^2 + 3(\ 1722.64\ N\ /\ A_t)^2} \leq 225\ MPa$$

$$12703600.83\ /\ {A_t}^2 \leq 50625$$

$${A_t}^2 \geq 250.93$$

$$A_t \geq 15.84\text{m}m^2$$

TABLE 10.2 Basic Dimensions of ISO Metric Screw Threads						
	Coarse Threads			Fine Threads		
Nominal Diameter d (mm)	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm^2)	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm^2)
3	0.5	2.39	5.03			
3.5	0.6	2.76	6.78			
4	0.7	3.14	8.78			
5	0.8	4.02	14.2			
6	1	4.77	20.1			
7	1	5.77	28.9			
8	1.25	6.47	36.6	1	6.77	39.2
10	1.5	8.16	58.0	1.25	8.47	61.2
12	1.75	9.85	84.3	1.25	10.5	92.1
14	2	11.6	115	1.5	12.2	125
16	2	13.6	157	1.5	14.2	167
18	2.5	14.9	192	1.5	16.2	216
20	2.5	16.9	245	1.5	18.2	272
22	2.5	18.9	303	1.5	20.2	333
24	3	20.3	353	2	21.6	384
27	3	23.3	459	2	24.6	496
30	3.5	25.7	561	2	27.6	621
33	3.5	28.7	694	2	30.6	761
36	4	31.1	817	3	32.3	865
39	4	34.1	976	3	35.3	1030

Table 12: Basic Dimensions of ISO Metric Screw Threads

As shown in Table 12 above, it can be seen that M6 bolts have a stress area of 20.1 mm², slightly larger than the required stress area of 15.84 mm². This satisfies the current requirement to produce a safety factor of as close as possible to 5. Selecting M6 bolts over the initially utilised M8 bolts can reduce the cost of this designed hoist. The exact safety factor of the selected optimised M6 bolts is 6.344, as found below:

$$\sqrt{(389.930\ N\ x\ Safety\ Factor\ /\ A_t)^2\ +\ 3(\ 344.528\ x\ Safety\ Factor\ /\ A_t)^2}\ \leq\ S_p$$

$$A_t\ =\ 20.1\ \text{mm}^2$$

$$\sqrt{(19.40\ \times\ Safety\ Factor^2)\ +\ 3(\ 17.141\ \times\ Safety\ Factor^2)}\leq\ 225\ MPa$$

$$1257.77\ \times\ Safety\ Factor^2\leq\ 50625$$

$$Safety\ Factor^2\geq\ 40.250$$

$$Safety\ Factor\ \geq\ 6.344$$

$$Minimum\ Safety\ Factor\ =\ 6.344$$

10.4 COMPONENT COSTING

Hex M6 bolts of SAE Class 4.6 and 60mm length cost RM 0.98 each when delivered to Malaysia from a company called RS Pro, including delivery charges. Since 12 similar M6 bolts are used in the designed hoist, the total cost spent on the Hex M6 bolts of SAE Class 4.6 and 60mm length is RM 11.76. Catalogues and pricing details of the selected fasteners can be found in Appendix B.

11.0 DESIGN ANALYSIS FRAMEWORK SUMMARY

PART NO.	MAJOR COMPONENT	TYPE OF ANALYSIS DONE ONTO COMPONENTS	MATERIAL	SUBJECT TO MAXIMUM FORCE (N)	SAFETY FACTOR
1	HOOK	FINITE ELEMENT ANALYSIS	ANNEALED AISI 4340 STEEL	24064.54	5.006
2	HOOK ATTACHMENT	TENSILE STRESS ANALYSIS, SHEAR STRESS ANALYSIS & BENDING MOMENT ANALYSIS	ANNEALED AISI 4340 STEEL	24284.88	16.629
3	CHAIN	TENSILE STRESS ANALYSIS	ALLOY STEEL	12673.53	6.560
4	PULLEY DRUM	FINITE ELEMENT ANALYSIS	ANNEALED AISI 4340 STEEL	12673.53	5.025
5	SHAFT (CONTROLLING PULLEY)	TORSIONAL STRESS ANALYSIS, BENDING STRESS ANALYSIS & ASME CODE CALCULATION	ANNEALED AISI 4340 STEEL	12114.92	8.682
6	GEAR (CONTROLLING PULLEY)	INTERFERENCE CHECK, STRESS ANALYSIS ON GEAR TEETH	PLAIN CARBON STEEL	7264.08	5.054
7	BEARING	BEARING SELECTION	CHROME STAINLESS STEEL	12114.94	-
8	FASTENER	FORCE ANALYSIS, SHEAR & TENSILE FORCE EQUIVALENT TENSILE STRESS ANALYSIS	ZINC PLATED STEEL	389.93	6.334

Table 13: Summary of Design Analysis Framework

12.0 MOTOR SELECTION

12.1 HOISTING MOTOR

The hoisting motor is responsible to provide power to operate the hoisting mechanism in the designed hoist to be able to lift and stack concrete cube loads vertically. To select a suitable motor for this purpose, 3 requirements are considered during the selection process, which is the motor's rated speed, it's operating continuous torque, and its rated power. As seen below, the rotational speed of the pulley driving shaft required to hoist loads safely and efficiently is 45 rpm, where else the torque on the shaft is 1394.09 Nm. The power required to run the shaft is calculated as below:

$$Rotational\ Speed\ of\ Pulley\ Driving\ Shaft\ =\ 45\ rpm$$

$$Torque\ on\ Pulley\ Driving\ Shaft\ (T)\ =\ 1394.09\ Nm$$

$$\begin{aligned} Power &= \omega \times T \\ &= [\ 45\ \mathrm{rpm}\ \times (2\pi\ /\ 60)\ ]\ \times\ 1394.09\ \mathrm{Nm} \\ &= 4.712\ rad/s\ \times\ 1394.09\ Nm \\ &= 6569.49\ W \\ &= 6.569\ \mathrm{kW} \end{aligned}$$

Using the information above, a suitable motor is selected. As shown in Appendix B, the selected motor is an 800STK4M Model, supplied from a company named Alxion. The selected motor has a rated speed range of 30 rpm to 250 rpm, where the current value of required rotational speed of the pulley driving shaft of 45 rpm lies between. The calculated required power to be transmitted to the pulley driving shaft is 6.569 kW, and this is well between the range of the selected motor's rated power, which is 6.3 kW to 33.1 kW. The continuous torque at stall of the selected motor is 2010 Nm, which is greater than the required torque on the pulley driving shaft of 1394.09 Nm. This indicates that the motor is able to provide sufficient torque needed to lift the concrete cube loads. Based on the considerations made, the 800STK4M Model motor from Alxion is found to be the best choice to be used to power the hoisting mechanism of the designed hoist, as seen in Figure 70 below.

Figure 70: 800STK4M Model motor from Alxion

12.2 TRANSLATION MOTOR

The translation motor is responsible to provide power to operate the translation mechanism in the designed hoist to be able to travel horizontally along an I-beam fixed to the lower edge of a crane. Similar to the hoisting motor, to select a suitable motor for this purpose, 3 requirements are considered during the selection process, which is the motor's rated speed, it's operating continuous torque, and its rated power. These 3 requirements are considered based on two types of forces that are experienced by the hoist, both the rolling friction force and the acceleration force. Below are the calculations made to find the torque required to run the shaft driving the hoist's wheels:

Rolling Friction Force:

$$Mass\ of\ Entire\ Hoist\ (m_1)\ =\ 716.06\ kg$$

(Total mass of the entire hoist can be found in Appendix E)

$$Mass\ of\ Concrete\ Cube\ Load\ (m_2) =\ 2400\ kg$$

$$Total\ Weight\ Acting\ on\ The\ Hoist's\ Wheels\ =\ (m_1 +\ m_2)\ x\ g$$

$$\mathrm{W}\ =\ 3116.06 \text{ x } 9.81\ \mathrm{m/s^2}$$

$$=\ 30568.5\ \mathrm{N}$$

$$\text{Normal Force Acting Wheels (N)} =\ \mathrm{W}$$

$$N\ =\ 30568.5\ N$$

With reference to Table 14 below [3], the estimated coefficient of rolling friction of the hoist's steel wheels on the steel I-beam of the crane is:

$$\text{Hoist Wheel's Coefficient of Rolling Friction}\ =\ C_{rr}$$

$$C_{rr}\ = 0.0005\ m$$

Material	Rolling friction
Steel on Steel	0.0005m
Wood on Steel	0.0012m
Wood on Wood	0.0015m
Iron on Iron	0.00051m
Iron on Granite	0.0021m

Table 14: List of Common Coefficient of Rolling Friction

$$Radius\ of\ Wheel\ (r)\ =\ 0.115\ m$$

$$Rolling\ Friction\ Force\ on\ Each\ Hoist\ Wheel\ = (N\ \times\ C_{rr}\ /\ r)$$

$$F_R\ =\ 30568.5\ N\ \times\ 0.0005\ m\ /\ 0.115\ m$$

$$=\ 132.907\ N$$

$$Torque\ Required\ to\ Overcome\ Rolling\ Friction\ Force\ on\ Wheel\ =\ F_R\ \times\ r$$

$$=\ 132.907\ N\ \times\ 0.115\ m$$

$$=\ 15.284\ Nm$$

Acceleration Force:

$$Hoist's\ Translation\ Motion\ Velocity\ =\ 30\ rpm\ \times\ (2\pi\ /60)\ \times\ 0.115\ m$$

$$v\ =\ 0.361\ m/s$$

(Assuming the hoist takes up 3 seconds to reach a velocity of 0.361 m/s)

$$Hoist's\ Translation\ Motion\ Acceleration = v\ /\ t$$
$$a = 0.361\ m/s\ /\ 3\ seconds$$
$$= 0.120\ m/s^2$$

$$Hoist's\ Acceleration\ Force = m \times a$$
$$F_a = (716.06\ kg + 2400\ kg) \times 0.120\ m/s^2$$
$$= 373.927\ N$$

$$Radius\ of\ Wheel = 0.115\ m$$

$$Torque\ Required\ for\ Hoist's\ Acceleration\ Force = F_a \times r$$
$$= 373.927\ N \times 0.115\ m$$
$$= 43.002\ Nm$$

Using the magnitude of the normal force acting on the hoist's wheels, and an estimation of the wheel's coefficient of rolling friction, the torque required to overcome the rolling friction force on the wheels is calculated, which is 15.284 Nm. Using the magnitude of the hoist's translation acceleration force, the torque required for the hoist's acceleration force is calculated, which is 43.002 Nm. Combining both torque magnitudes calculated, the power required to run the shaft driving the wheel is calculated. As seen below, the rotational speed of the shaft driving the wheels required to move horizontally along an I-beam safely and efficiently is 30 rpm. The power required to run the shaft is calculated as below:

$$Rotational\ Speed\ of\ Wheel\ Driving\ Shaft = 30\ rpm$$

$$Torque\ on\ Wheel\ Driving\ Shaft\ (T) = 15.284\ Nm + 43.002\ Nm$$
$$= 58.286\ Nm$$

$$Power = \omega \times T$$
$$= [\ 30\ \text{rpm} \times (2\pi\ /\ 60)\ ] \times 58.286\ \text{Nm}$$
$$= 3.142\ rad/s \times 58.286\ Nm$$
$$= 183.110\ W$$
$$= 0.183\ kW$$

Using the information above, a suitable motor is selected. As shown in Appendix B, the selected motor is a 12V Worm Drive Motor, supplied from a company named Motion Dynamics. The selected motor has 2 operational speeds which is 30 rpm and 40 rpm, and this matches the current value of required rotational speed of the wheel driving shaft of 30 rpm. The calculated required power to be transmitted to the wheel driving shaft is 183 W, and this is very close to the selected motor's rated power, which is 180W. The continuous torque at stall of the selected motor is 100 Nm, which is greater than the required torque on the pulley driving shaft of 58.286 Nm. This indicates that the motor is able to provide sufficient torque needed to move the hoist horizontally at the required speed. Based on the considerations made, the 12V Worm Drive Motor from Motion Dynamics is found to be the best choice to be used to power the translation mechanism of the designed hoist, as seen in Figure 71 below.

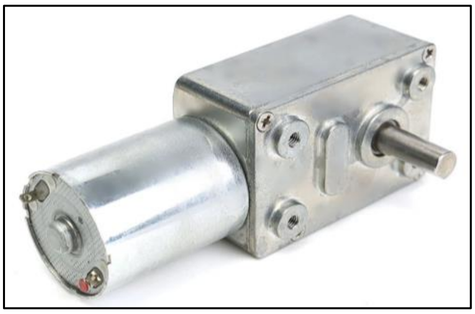


Figure 71: 12V Worm Drive motor from Motion Dynamics

13.0 ALTERNATOR SELECTION

The alternator is used to convert mechanical energy possessed by concrete cubes when they are unstacked and lowered, turning it into electrical energy. This electrical energy is then stored and released when needed, which meets the purpose of the concrete stacking strategy. When selecting a suitable alternator, it is important to consider the alternator's rated speed, it's operating torque, and its rated power. Since this alternator is utilised in the hoisting mechanism parallelly with the hoisting motor, both the alternator and the hoisting motor have the same specifications considered as below:

$$\begin{aligned} Rotational\ Speed\ of\ Pulley\ Driving\ Shaft &= 45\ rpm \\ Torque\ on\ Pulley\ Driving\ Shaft\ (T) &= 1394.09\ Nm \\ Power &= 6.569\ kW \end{aligned}$$

Using the information above, a suitable alternator is selected. As shown in Appendix B, the selected alternator is an 800STK4M Model, supplied from a company named Alxion. The selected alternator has a rated speed range of 80 rpm to 400 rpm, and when operated at half speed, the current value of required rotational speed of the pulley driving shaft of 45 rpm lies between the range. The calculated required power to be transmitted to the pulley driving shaft is 6.569 kW, and this is well between the range of the selected alternator's rated power when operated at half speed, which is 6.026 kW to 41.694 kW. The torque of the selected alternator is 2196 Nm, which is greater than the required torque on the pulley driving shaft of 1394.09 Nm. This indicates that the alternator is able to sustain the torque present on the pulley driving shaft. Based on the considerations made, the 800STK4M Model alternator from Alxion is found to be the best choice to be used to generate electricity from the concrete cube stacking strategy, as seen in Figure 72 below:

Figure 72: 800STK4M Model alternator from Alxion

14.0 OVERALL COSTING

Using a Microsoft Excel Costing Template, the cost to produce the designed hoist was estimated, as shown in Table 15 below. Columns that are highlighted in blue colour indicate that the specific components' costs are estimated based on assumptions made at the top of Costing Template in Table 15 below, or using rough information found on the internet. Columns that are highlighted in green colour indicate that the fabricated components' costs are estimated based on costing reports generated through the Solidworks software, which can be found in Appendix C. Columns that are highlighted in orange colour indicate that the purchased components' costs are actual costs found via pricing catalogues found from supplying companies, which can be found in Appendix B. From the Microsoft Excel Costing Template, it is found that the overall cost to produce the designed hoist system is RM42066.61.

Assumptions:

1) Costing is in RM		
2) Plain Carbon Steel **WHEELS** cost per kg. (RM)	**8.00**	(Note: Included fabrication process, consumables, wastage, labor, overhead, minor machining)
3) Annealed AISI 4340 Steel **SHAFTS** cost per kg. (RM)	**10.50**	(Note: Included wastage, machining, overhead and annealing)
3) Plain Carbon Steel **GEARS** cost per kg. (RM)	**9.50**	(Note: Included wastage, machining, overhead)
5) 1060 Aluminium Alloy **CASING** cost per kg. (RM)	**8.30**	(Note: Included fabrication process, consumables, wastage, labor, overhead, minor machining, painting)
6) Engineering man-hour cost (overhead considered). (RM)	**100.00**	
7) Assembly man-hour cost (overhead considered). (RM)	**60.00**	(Note: Included welding process)
8) Testing man-hour cost (overhead considered). (RM)	**80.00**	

Item	Description	Drawing No. / Part No.	Material	Qty.	Unit Weight (kg)	Purchased / Machined	Unit Cost (RM)	TOTAL COST (RM)
1.0	**Engineering Hour**	-	-	**200**	-	-	-	**20,000.00**
2.0	**Translation Mechanism Assembly**	**1000**	-	**1**	-	-	-	-
2.1	Gear	1001	Plain Carbon Steel	6	4.89	M	46.455	278.73
2.2	Wheel	1002	Plain Carbon Steel	4	2.37	M	18.94	75.76
2.3	Driving Shaft	1003	Annealed AISI 4340 Steel	1	16.37	M	171.885	171.885
2.4	Driven Shaft	1004	Annealed AISI 4340 Steel	4	4.63	M	48.5835	194.334
2.5	Bearing	Bearing Number #L16	Chrome Stainless Steel	10	0.14	P	70.64	706.4
2.6	Circlip	50mm External Circlip	Chrome Stainless Steel	20	0.02	P	17.98	359.6
2.7	Translation Motor	12V Worm Drive Motor	-	1	19.6	P	750.00	750
3.0	**Hoisting Mechanism Assembly**	**2000**	-	**1**	-	-	-	-
3.1	Hook	2001	Annealed AISI 4340 Steel	1	53.06	M	891.79	891.79
3.2	Hook Attachment	2001	Annealed AISI 4340 Steel	1	22.46	M	1500.5	1500.50
3.3	Chain	G80 Alloy Chain	Alloy Steel	2	54.14	P	303.73	607.45
3.4	Pulley Drum	2002	Annealed AISI 4340 Steel	2	30.39	M	689.8	1379.60
3.5	Driving Shaft	2003	Annealed AISI 4340 Steel	1	37.89	M	439.88	439.88
3.6	Driven Shaft	2004	Annealed AISI 4340 Steel	1	15.01	M	157.605	157.61
3.7	Spur Gear	2005	Plain Carbon Steel	2	11.71	M	427.94	855.88
3.8	Bearing	Bearing Number #316	Chrome Stainless Steel	4	0.14	P	847.95	3391.80
3.9	Circlip	80mm External Circlip	Chrome Stainless Steel	8	0.02	P	40.97	327.76
3.10	Hoisting Motor	800STK4M Model Motor	-	1	130.00	P	1200	1200.00
3.11	Alternator	800STK4M Model Alternator	-	1	130.00	P	1400	1400.00
4.0	**Hoist Casing Assembly**	**3000**	-	**1**	-	-	-	-
4.1	Main Casing	3001	1060 Aluminium Alloy	2	41.27	M	342.4995	685.00
4.2	Main Casing Connecting Body	3002	1060 Aluminium Alloy	3	2.76	M	22.9329	68.80
4.3	Hoisting Motor Casing	3003	1060 Aluminium Alloy	1	10.48	M	86.984	86.98
4.4	Translation Motor Casing	3004	1060 Aluminium Alloy	1	6.30	M	52.29	52.29
4.5	Alternator Motor Casing	3005	1060 Aluminium Alloy	1	4.02	M	33.366	33.37
4.6	Casing Bolt	M12 x 50	Zinc Plated Steel	14	0.09	P	1.97	27.58
4.7	Casing Nut	M12	Zinc Plated Steel	14	0.04	P	0.57	7.98
4.8	Connecting Body Bolt	M6 x 60	Zinc Plated Steel	12	0.19	P	0.98	11.80
4.9	Connecting Body Nut	M6	Zinc Plated Steel	12	0.08	P	0.32	3.84
5.0	**Assembly Hour**	-	-	**40**	-	-	-	**2,400.00**
6.0	**Testing & Commissioning**	-	-	**50**	-	-	-	**4,000.00**
							GRAND TOTAL	**42,066.61**

Table 15: Microsoft Excel Costing Template Used to Estimate Cost of Producing the Designed Hoist

15.0 CONCLUSION

Upon completing a very detailed engineering design process, a hoist system that is of optimal functionality, efficiency, safety, and cost effectiveness is produced. The designed hoist system costs RM42066.61 in total and has the capacity to generate a total of 53.963 Megajoules of electricity a day when operated for a span of 8 hours. Along the designing process of the hoist system, the total mass of the hoist system was also optimised to be minimal, producing a system weighing only 716.06 kg in total. Besides that, the components in the designed hoist system are optimised to have a minimum safety factor of 5, meaning the selected components are capable of withstanding 5 times the maximum amount of force it is typically subjected to. The bearings used in the hoist system are also durable and estimated to have a lifetime of up to about 3 and a half years, ensuring that the entire hoist system is very safe to use. Overall, the project requirements and expectations of producing an efficient and reliable hoist system is fulfilled with great standards.

16.0 REFERENCES

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17.0 LIST OF TABLES

18.0 LIST OF FIGURES