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STRUCTURAL ANALYSIS OF A PEDAL-POWERED MAIZE SHELLER

CAPSTONE

B.Sc. Mechanical Engineering

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ASHESI UNIVERSITY

STRUCTURAL ANALYSIS OF A PEDAL-POWERED MAIZE SHELLER

CAPSTONE PROJECT

Capstone Project submitted to the Department of Engineering, Ashesi University in partial fulfilment of the requirements for the award of Bachelor of Science degree in Mechanical Engineering.

Oheneba Atta Aggrey

2021

DECLARATION

I hereby declare that this capstone is the result of my own original work and that no part of it has been presented for another degree in this university or elsewhere.

Candidate's Signature:

……………………………………………………………………………………………

Candidate's Name:

Oheneba Atta Aggrey

Date:

May 29, 2021.

I hereby declare that preparation and presentation of this capstone were supervised in accordance with the guidelines on supervision of capstone laid down by Ashesi University College.

Supervisor's Signature:

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Supervisor's Name:

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Date:

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Abstract

In Ghana, maize is one of the highest-yielding cash crops, but the supply chain is severely hampered by the lack of access to machines for processing. This study conducts a structural analysis on a low-cost, pedal-powered maize sheller, designed and fabricated by the author and found to operate with a high shelling rate and low kernel damage. To inform an improved iteration of the design, structural analysis was carried out along the lines of reliability of the bearing that was used, factor of safety for both static and fatigue loads, weight optimization, and trend forecast for shelling performances. Using a reliability of 90%, a 180409 Instrument Precision ball bearing was calculated to have a life cycle of 3468.44 hrs which fell in the range needed for agricultural equipment. Moreover, the static and fatigue analysis revealed yield strength values that were higher than the applied stresses. As such, they were deemed safe, except for the bicycle cleats, which needed fillets to reduce stress concentrations. From the weight optimization, it was determined by the author that one-fourth of the cylinder stand's weight can be removed to save on material without hampering the assembly's structural integrity. Lastly, using a forecast of shelling trends, a trade-off was discovered between kernel damage and shelling rate.

Keywords: Structural analysis, factor of safety, reliability analysis, topology optimization.

Table of Contents

DECLARATION ii
Acknowledgements iii
Abstract iv
Table of Contents 1
List of Abbreviations 3
List of Symbols 4
List of Tables 5
List of Figures 6
Chapter 1: Introduction 7
1.1 Problem Statement 7
1.2 Project Objectives 8
1.3 Theoretical Framework 8
1.3.1 Failure Theories 9
1.3.2 Reliability Analysis 9
1.3.3 Topology Optimization 9
1.4 Motivation of project 10
1.5 Scope of Work 10
Chapter 2: Literature Review 11
2.1 Overview 11
2.2 Agricultural Machinery 11
2.3 Grass-roots farming in Ghana 12
2.4 Pedal Powered Devices 12
Chapter 3: Design 13
3.1 Overview 13
3.2 Design Criteria and Decision Matrix 13
3.3 Design Objectives 15
3.4 Design Proposal 16
3.5 Material Characteristics of existing configuration 18
Chapter 4: Analysis 19
4.1 Overview 19
4.2 Reliability of Bearing 19
4.3 Safety 23
4.3.1 Static and Fatigue Analysis for Rotating Shaft in the Sheller 23
4.3.2 Static and Fatigue Analysis for Bicycle Pedal Cleats 27

4.4 Optimization of Cylinder Stand's Weight 29
4.5 Trend Forecast for Shelling Performance 31
Chapter 5: Discussion 34
5.1 Overview 34
5.2 Discussion of Analysis for Bearing 34
5.3 Discussion on Static and Fatigue Analysis 36
5.3.1 Analysis for rotating shaft 36
5.3.2 Discussion of Safety Analysis for bicycle cleats 37
5.4 Discussion of Analysis from Weight Optimization. 37
5.5 Discussion of Shelling Performance Trend Forecast 38
Chapter 6: Challenges, Recommendations & Conclusion 40
6.1 Overview 40
6.2 Challenges Encountered & Limitations 40
6.3 Recommendations for Future Works 40
6.4 Conclusion 41
References 42
Appendix A: MATLAB Code to Compute Needed Analytical Values 46
Appendix B: Drawings of Essential Parts of Maize Sheller 48
Glossary 50

List of Abbreviations

SDGs …………………………………….. Sustainable Development Goals

TO ……………………………………. Topology Optimization

MDET …………………………………….. Maximum Distortion Energy Theory

ASME ……………………………………. American Society of Mechanical Engineers

RPM ………………………………… Revolutions Per Minute

MATLAB …………………………….. Matrix Laboratory

CAD …………………………………….. Computer-aided Design

List of Symbols

$\boldsymbol{\omega}$ = Angular velocity

$\boldsymbol{r}$ = Pitch radius of the chain wheel.

T = Torque

τ = Torsion

Z = Section Modulus

Tm = Mean Torque

Ta = Amplitude Torque

Sy = Yield strength

Se = Design Endurance Limit

Ka = Surface Factor

Kb = Size Factor

Kc = Load Factor

Kd = Temperature Factor

Ke = Reliability Factor

Kf = Miscellaneous Factor

Se' = Specimen Endurance Limit

r/min = revolutions per minute

List of Tables

Table 3. 1: Pugh Chart of Shelling Mechanism Designs [4]..14
Table 3.2: Updated Design Objectives 16
Table 3. 3: Material Properties of Galvanized Steel [5]..18
Table 3. 4: Material Properties of Alloy Steel [5]..18
Table 4. 1: Experimental Results [4]...33
Table 5. 1: Updated Experimental Results [4]...39

List of Figures

Figure 3. 1: Shelling Mechanism Prototypes [4] 13
Figure 3. 2: The fabricated bicycled-power maize sheller [4] 15
Figure 3. 3: Right view of designed bicycle-powered maize sheller 17
Figure 3. 4: Front view of designed bicycle-powered maize sheller 17
Figure 4. 1: 3D Description of How the Bearing Rotates. 19
Figure 4. 2: Input Torque Supplied by Bicycle Pedal 20
Figure 4. 3: Output Torque Obtained by Amplifying Input Torque 20
Figure 4. 4: SolidWorks Reliability Analysis on 170407 Ball Bearing 21
Figure 4. 5: SolidWorks Error Message 21
Figure 4. 6: SolidWorks Reliability Analysis on 180409 Ball Bearing 22
Figure 4. 7: Factor of Safety of a Static Event 24
Figure 4. 8: 2D Torque Analysis of Rotating Shaft 24
Figure 4. 9: ASME-Elliptic Criterion 26
Figure 4. 10: Total Life of a Rotating Shaft Under Fatigue Loading 27
Figure 4. 11: Load Diagram for Forces Normal to Bicycle Cleats 28
Figure 4. 12: Static Load Analysis on Bicycle Cleats 28
Figure 4. 13: Fatigue Load Analysis on Bicycle Cleats 29
Figure 4. 14: Load Distribution on Cylinder Stand 30
Figure 4. 15: Weight Optimization Analysis on Structural Member 31
Figure 4. 16: Graph of Experimental Trials With a 1:2 Gear Ratio 32
Figure 5. 1: Labelled Image of a Ball Bearing [5] 34
Figure 5. 2: Bearing Design Life Value for Different Applications[23] 35
Figure 5. 3: Fatigue Analysis for Rotating Shaft 36
Figure 5. 4: S-N Curve for Steel 37
Figure A. 1: Endurance limits versus tensile strengths [24] 47
Figure B. 1: Bill of Material for the Whole Assembly 48
Figure B. 2: Bill of Material for Bicycle Subassembly 48
Figure B. 3: Drawing for Sheller's Cylinder 49
Figure B. 4: Drawing for Cylinder Stand 49

Chapter 1: Introduction

1.1 Problem Statement

Maize, also known as corn, is a cereal grain originally domesticated by the indigenes of South Mexico about 10,000 years ago [1]. Maize has become a staple food in most parts of the world, and in Ghana, it is one of the highest yielding cash crops. Agriculture makes up 60% of the job percentage in Ghana, yet most farmers are still poor because they cling to traditional farming practices [2]. Meanwhile, in the United States, less than two per cent of the population is involved in agriculture [3]. Nonetheless, they can still feed the entire U.S population and even have a surplus for export. Machines increase productivity in agriculture, and for a high-yielding cash crop like maize, involving mechanization would have quite a significant effect. This motivation led the author to design and fabricate a low-cost, pedal-powered maize sheller to increase farmers' work rate during the post-harvest season and ultimately increase their revenue after harvest [4]. The problem statement then evolved into, 'Employing structural analysis to inform design improvements on the pedal-powered maize sheller that will ensure the machine is cost-effective, safe for usage, achieves higher performance, and its life cycle can be predicted”.

Strength is a property or characteristic of a mechanical element [5]. This material identity is incidental to the geometry and loading on an element that resultantly controls the element's behaviour at its critical location. The importance of this critical location cannot be overstated since failure outsets from that point which can ultimately endanger human life. As such, it is embedded at the forefront of engineering that, above everything else, safety should always be the most paramount. Through the time-lapse of history, with incidents such as the Tacoma Narrows Bridge, it is evident that mistakes and shortcuts taken during the engineering process can indeed prove catastrophic [6]. Therefore, this paper focuses on providing a structural analysis on the shelling machine to ensure a catastrophic failure does not happen in the future.

1.2 Project Objectives

With regards to the Sustainable Development Goals (SDGs) set by the United Nations, this project focuses on using clean energy to alleviate the poverty of subsistence maize farmers thus, spotlighting goals 1 and 7 of the SDGs, which are no poverty and affordable clean energy, respectively [7]. However, to be specific to the main project objectives, they are listed below as:

- Using reliability analysis as a design parameter to predict the life cycle of a bearing mechanical element.
- Using static and fatigue analysis to investigate if critical components such as shafts and bicycle cleats are safe for usage.
- Performing design topology analysis to keep the assembly low-cost by reducing its overall weight, which would, in turn, lower cost.
- Forecasting the role of gear optimization to predict shelling performance values.

1.3 Theoretical Framework

A static load is a force or couple that is unchanging in magnitude and direction that can produce axial compression or tension, shear, torsional, bending loads, or any combination of these [5]. The relation between the strength of a material and static loads interplays in making decisions concerning material geometry and requirements such as safety, reliability, and functionality. Hence below are some theories used by engineers to understand the effect of loads on any designed machine.

1.3.1 Failure Theories

Unfortunately, there is no single theory that governs the failure of structural elements. In some cases, depending on the type of material, be it brittle or ductile, this provides the clue to which specific theory to use. However, tests should be carried out first because sometimes a material considered to be ductile can fail in a brittle manner under special circumstances. To check this, a strain factor of 0.05 is used for ductile materials [5]. After the type of material has been decided, a static loading design analysis is done to affirm if the element would fail by yielding or fracturing. The results from the static loading analysis would then serve as the baseline for which a fatigue analysis can later be carried out. Fatigue analysis is essential because, whereas a static analysis accounts for one event, fatigue accounts for multiple instances.

1.3.2 Reliability Analysis

For mechanical elements such as bearings, their life cycle cannot be described using a deterministic form [5]. Instead, an invariant, statistical distribution known as the Weibull distribution is used to describe their life cycles. To employ the distribution, some useful deterministic equations addressing load versus life at constant reliability is introduced. The combination of both statistical and deterministic relationships is what helps in calculating the life cycle of the chosen mechanical element.

1.3.3 Topology Optimization

Topology optimization (TO) is a mathematical approach that optimizes the material layout within a given design under a set of boundary conditions, loads, and constraints with the aim of meeting a prescribed set of targets to improve overall system performance [8]. In the SolidWorks environment, the default optimization technique is the Best Stiffness to Weight Ratio, which gives an optimized shape of the component [9]. Conventional TO is different from shape and sizing optimizations since the design can attain any shape, while in the latter, the final shape needs to fit a pure shape [10].

1.4 Motivation of project

A bicycle-powered maize sheller is a machine designed to remove kernels from maize cobs using pedal power. As with any machine, breakdowns can always occur. As such, this project aims at conducting a structural analysis to predict structural compromises that may arise. This would provide the needed assurance as opposed to risking the machine operator's life. Also, there are cost-saving benefits since the failure of any machine component results in an incurred cost.

1.5 Scope of Work

Due to the limitations imposed by the COVID-19 pandemic, the scope of the project was limited to a virtual environment. Thus, progress on the project was heavily focused on computer simulations and analytical calculations. For computer simulations, SolidWorks was used to design and run simulations on selected parts of the assembly, while MATLAB was used to compute analytical values needed to make the project a success. There was no physical fabrication due to the health risk associated with procuring the necessary parts for the assembly.

Chapter 2: Literature Review

2.1 Overview

This chapter would seek to discuss the failures and successes of literature pertinent to agricultural machinery, grass-roots farming in Ghana, and pedal-powered technology while identifying the existing gaps evident in the existing literature.

2.2 Agricultural Machinery

Before the industrial revolution, working animals such as mules and manual labour were heavily relied on for many farm jobs. However, once the age of industrialization took over, farming methods took a great leap forward [11]. With agriculture having several sectors ranging from seed technology, irrigation to several other processes such as storage and refrigeration, the categorization is unbounded. This plethora of sectors heavily interplays with what type of equipment is needed for an agricultural task since it is not a situation of one size fits all. Current mechanized agriculture ranges from several tools such as hand tools, tractors, and farm implements such as helicopters for aerial application. The process can become more complicated with the emergence of precision agriculture that combines computers and satellite imagery to increase yield[12]. With the abundance of several agricultural types of machinery, a significant gap identified in one of the reviewed papers was how farmers are unhappy with their inability to repair their farm equipment [13]. Due to most companies using intellectual property laws and patents, it has become increasingly difficult for farmers to access information that would assist them in fixing their equipment [14]. As such, open-source agriculture has become a trending topic in the agricultural space, with several institutions collaborating to create an ecosystem of open-source technologies for farmers [15]. Consequently, this is an initiative that this project aims to contribute.

2.3 Grass-roots farming in Ghana

Agriculture remains a dominant economic sector in many developing countries such as Ghana. However, the industry is dominated by smallholder farmers, who are usually classified as subsistence farmers [16]. In Ghana, subsistence farmers typically live on their farmland, eating what they grow and selling the surplus in the local market [17]. Subsistence farmers form most of the poorest people in Ghana [18], and with agriculture making up 60% of Ghana's job percentage, this is a worrying statistic [2]. Since most farmers in Ghana are resource-poor, ensuring smallholder farmers make efficient use of the scarce resource is very important to help raise the level of productivity at such grass root levels. A significant finding identified in a reviewed paper about rice farmers in Northern Ghana revealed that mechanized systems such as irrigation helped improve the farmers' productivity immensely [16]. However, the significant hitch that hampers grass-root farmers' productivity in Ghana is the high cost of mechanized systems. As such, this project seeks to implore a low-cost form of energy, pedal power, to bridge the existing gap.

2.4 Pedal Powered Devices

In the past, for most engineers, the usage of internal combustion engines for the motion of mechanical moving parts is not uncanny. However, the production of noise and fumes from these engines is usually harmful to the device operator's health. As such, this has propelled the engineering community to veer towards green technology [19]. Using pedal power as a form of clean energy is not a new-fangled concept since simple transport units like bicycles have been using pedal power for years. From reviewed literature, significant shortcomings for most pedal technology ranging from pedal-powered sailboats to pedal-powered water pumps all have the considerable drawback of not being able to produce large amounts of energy within a relatively short amount of time [20].

Chapter 3: Design

3.1 Overview

This chapter establishes the entire proposed system and technical specifications needed for the completion of the project. These specifications were based on user and system requirements obtained from secondary research to achieve the desired design goal [4]. Though a functioning configuration exists to shell the needed maize, this project is focused on migrating the physical prototype to a software environment with the aim of performing structural analysis to inform a re-design.

3.2 Design Criteria and Decision Matrix

The first stage of the design process was the brainstorming stage which led to the emergence of three CAD shelling candidates, as seen in Figure 3.1.

(a) Grater (b) Pointed edges (c) Chains

Figure 3. 1: Shelling Mechanism Prototypes [4]

To test the efficacy of the designs, each of the above was prototyped by the author. After, they were mounted on a test stand while maize cobs were fed through to evaluate the three designs against a set of criteria [4]. The first prototype, figure 3.1a, is a 30*cm*-diameter grater with more than 50% corn kernel damage ratio that shells corn by rotating at a fixed position

[4]. Figure 3.1b, which has a kernel damage of almost 35%, is a 160*mm*-diameter cylinder with sharp ends shells maize by press-fitting the cob into the hollow cylinder and rotating it [4]. Lastly is figure 3.1c, which is a shaft with chains welded to it that rotates to beat kernels off the maize cob. The observations from the prototypes were used to create a Pugh Chart, as seen in table 3.1, to analogously rank them against the set of criteria [4].

Table 3. 1: Pugh Chart of Shelling Mechanism Designs [4]

Criteria (-1, 0, +1)		**Prototypes**		
	Baseline (By Hand)	*#1 (Grater)*	*#2 (Pointed edges)*	*#3 (Chains)*
Ease of Cleaning	+1	+1	0	+1
Exposed edges of design (Safety)	-1	-1	-1	+1
Energy efficiency	-1	0	+1	-1
Low kernel breakage	+1	0	-1	+1
Total	0	0	-1	+2

As seen from table 3.1, the third prototype had the highest score, and that shelling mechanism was what was chosen to move on to the next stage of fabrication. Upon selecting the top shelling mechanism, a full-scale assembly was fabricated, as seen in figure 3.2. The shelling mechanism consisted of a *1.5in*-diameter rotating shaft with welded chain links that were 5*in* long. Two bearings were press-fitted at the ends of the rotating shaft to reduce friction [4]. The mechanism was placed in a 13*in*-diameter old barrel that was recycled to serve as a cylinder. Moreover, a perforated foil was placed at the base of the cylinder to serve as an outlet for the kernels. A frame was built for the cylinder to keep it above ground. This was attached to a modified, used bicycle which would provide rotary power for the shelling mechanism. To prevent choking, a hatch was provided to allow for easy removal of shelled cobs. For a further detailed description of the step-by-step fabrication process, the entire fabrication process description can be found in the author's previous work [4].

Figure 3. 2: The fabricated bicycled-power maize sheller [4]

In terms of maintenance, occasional oiling to prevent rusting is required of the operator to keep the mechanism functioning.

3.3 Design Objectives

The objectives of the existing prototype were to fabricate a maize sheller which [4];

- Was low-cost (< $200).
- Can be operated through pedalling.
- Can be fabricated within the user's locality.
- Achieves high (> 80%) shelling efficiency, throughput (15 *gram/s*), and low breakage rate (< 5%).

However, after the existing configuration was fabricated and the objectives were met, the project objectives were modified to accommodate the shift in the project since the present project is focused on migrating the current prototype into a software environment. On this platform, simulations would be run to inform a re-design if necessary. Thus, the new design objectives are specified as seen in Table 3.1.

Table 3.2: Updated Design Objectives

Customer Need	What is measured	Units	Desired value
Weight	Lightweight	Kilograms	Less than the present value of 200kg
Safety	Factor of safety	None	Greater than 1.5
Reliability	Life Cycle	Years	Greater than one million cycles

3.4 Design Proposal

The objective of this project is to conduct an analysis on the structural integrity of a bicycle-powered maize sheller and, to suggest recommendations. The reason for this analysis is beholden to little or no structural analysis that was done on the initial prototype [4]. To achieve this, the bicycle-powered maize sheller, with its calculated dimensions, was designed in SolidWorks, as seen in figure 3.3 and.3.4. For the initial working prototype, a gear ratio of 3.2 was used. This was found by dividing the number of teeth of the input sprocket by the output. The gear ratio and other parameters, such as the length of the bicycle chain, were based on the availability of materials instead of being informed through analysis. After the design, analysis such as static and fatigue analysis, design topology, and other simulations were run on selected components to ensure the project objectives were satisfied.

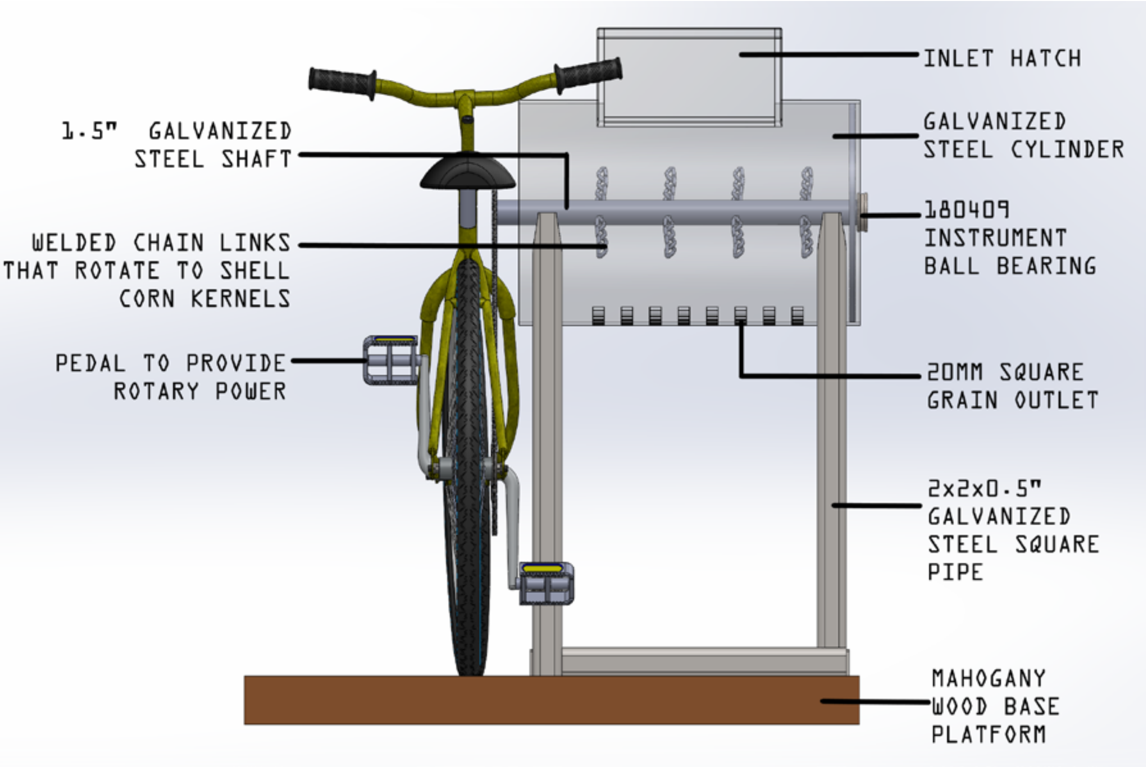

Figure 3. 3: Right view of designed bicycle-powered maize sheller

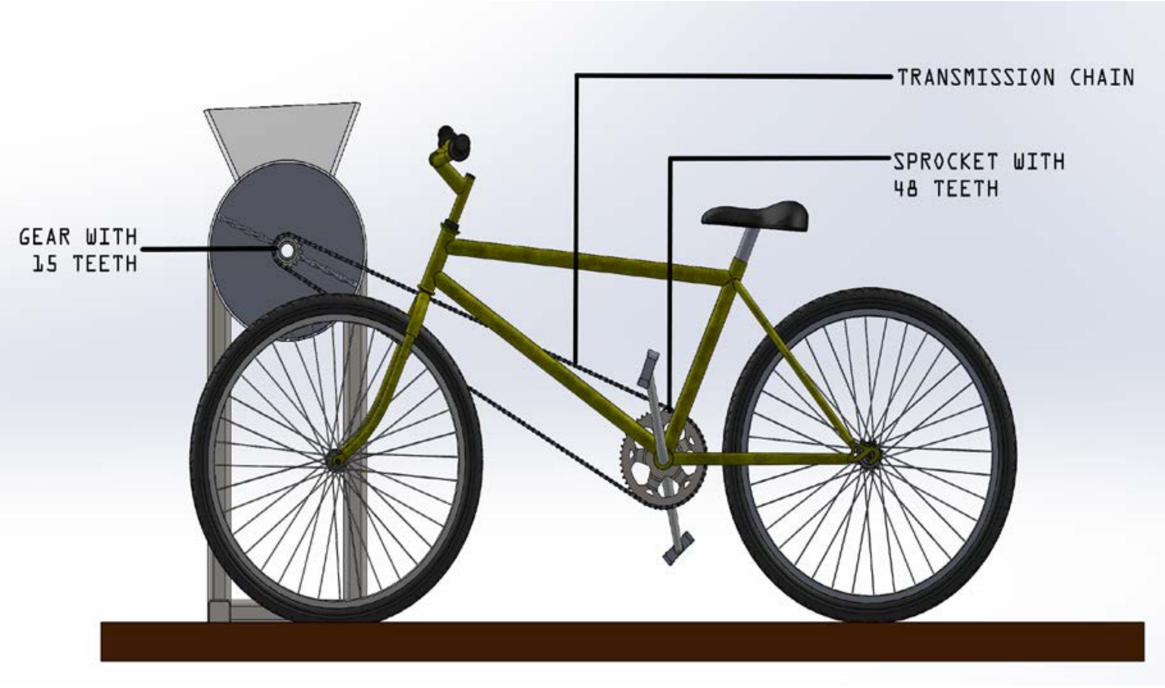

Figure 3. 4: Front view of designed bicycle-powered maize sheller

3.5 Material Characteristics of existing configuration

For the required analysis to be done, the material properties of the essential structural members would have to be known to produce the needed numerical and analytical calculations. Predominantly for the shelling assembly, galvanized steel, which is a ductile material, was used for most of the structure. Hence Table 3.4 gives the material properties that would be needed to do a numerical analysis.

Table 3. 3: Material Properties of Galvanized Steel [5]

Name	Value	Units
Elastic Modulus	2e+11	N/m^2
Poisson's Ratio	0.29	N/A
Mass Density	7.87e+3	kg/m^3
Tensile Strength	3.569e+8	N/m^2
Yield Strength	2.039e+8	N/m^2

The second list of material properties was for alloy steel, a ductile material that forms the predominant part of the bicycle's cleats. Like what was done previously, Table 3.5 provides the needed material properties to facilitate the required analysis.

Table 3. 4: Material Properties of Alloy Steel [5]

Name	Value	Units
Elastic Modulus	2.1e+11	N/m^2
Poisson's Ratio	0.28	N/A
Mass Density	7.7e+3	kg/m^3
Tensile Strength	7.238e+8	N/m^2
Yield Strength	6.204e+8	N/m^2

Chapter 4: Analysis

4.1 Overview

This chapter describes the various calculations that were set up to investigate the project objectives. To recap, the project objectives are:

- Using reliability analysis as a design parameter to predict the life cycle of a bearing mechanical element.
- Using static and fatigue analysis to investigate if critical components such as shafts and bicycle cleats are safe for usage.
- Performing design topology analysis to keep the assembly low-cost by reducing its overall weight, which would, in turn, lower cost.
- Forecasting the role of gear optimization to predict shelling performance values.

Thus, this chapter would be explaining the analysis done for each project objective.

4.2 Reliability of Bearing

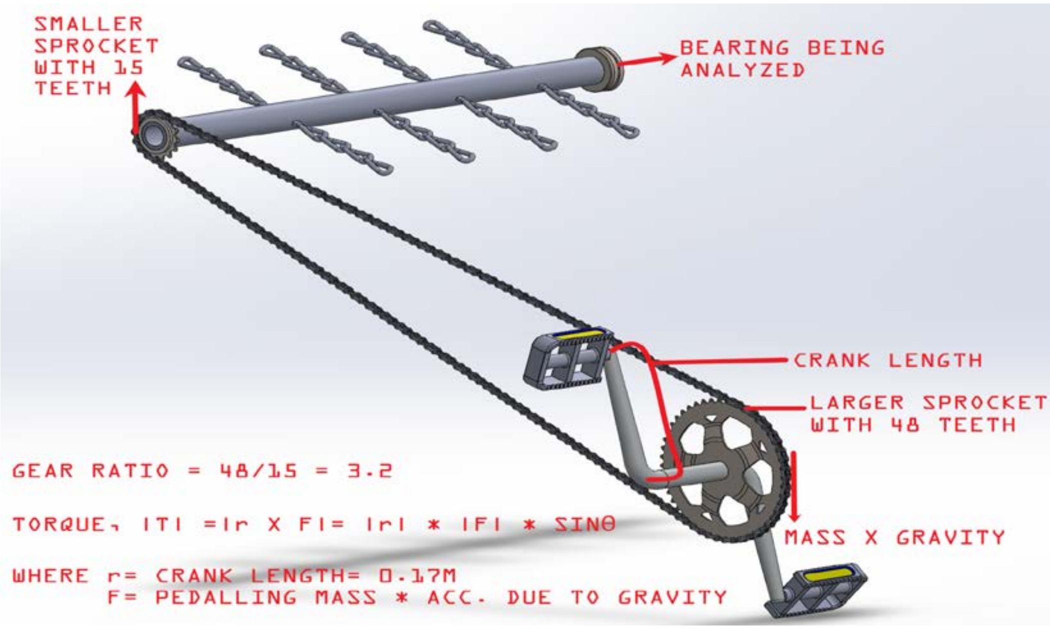

Figure 4. 1: 3D Description of How the Bearing Rotates.

Generally, engineers use bearings to reduce friction in machinery operations, and that is purposely what the bearing in figure 4.1 aims to achieve. For the sub-assembly shown above in figure 4.1, the sprocket-chain configuration is constrained to the Y-Z plane. Once the bicycle pedal starts rotating, the rotational power is transferred to the shaft, which has a gear ratio of 3.2, assuming there are no energy losses during transmission. Using an average weight of 60 kg for the operator [21], the normal force is calculated and multiplied by a crank length of 170*mm* to produce a maximum torque of 100*m*N*.

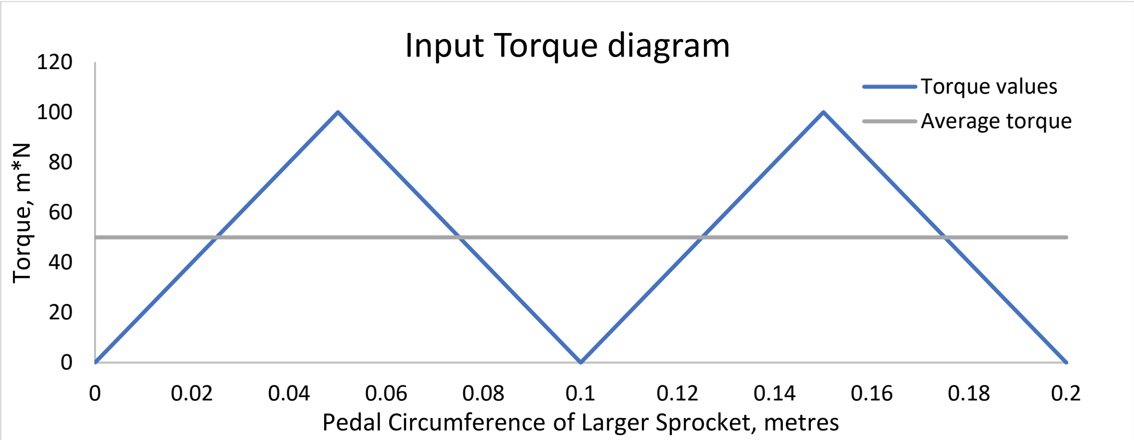

Figure 4. 2: Input Torque Supplied by Bicycle Pedal

From figure 4.3, using a gear ratio of 3.2, the graph for the output torque is like that of the input torque, only that is multiplied by a 3.2 scale factor.

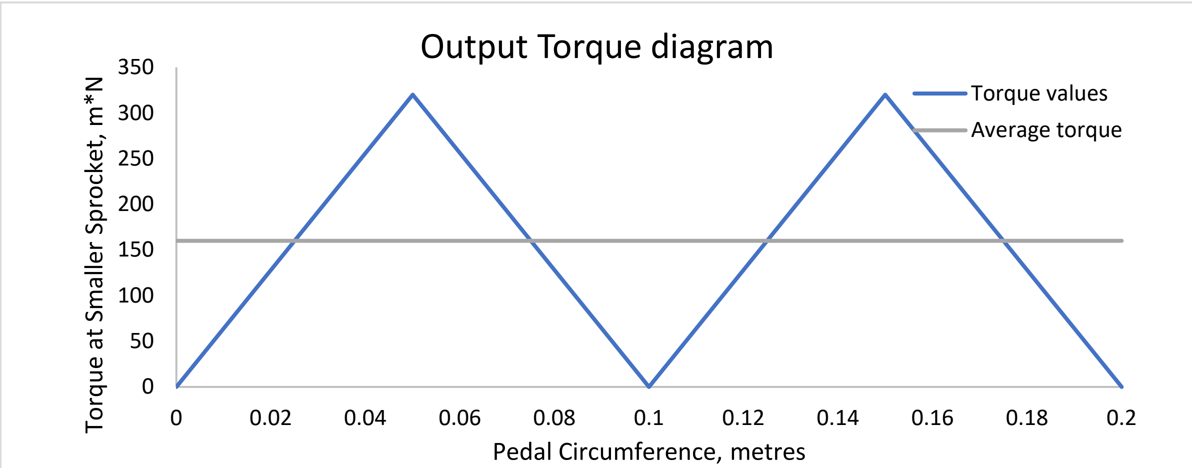

Figure 4. 3: Output Torque Obtained by Amplifying Input Torque

Incorporating the data above to the SolidWorks Bearing Calculator environment and using a revolution of 60rpm [4], the interface is shown below to obtain the basic life of a 170407-ball bearing using a reliability of 90%.

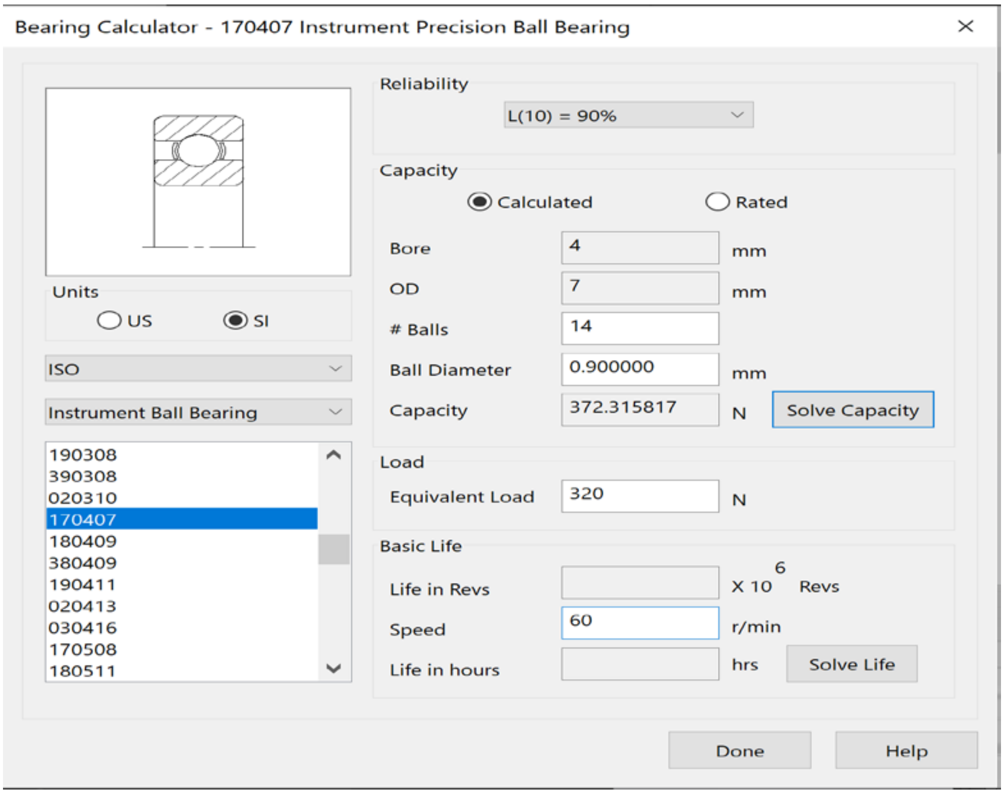

Figure 4. 4: SolidWorks Reliability Analysis on 170407 Ball Bearing

At first, a bearing of a lower grade than what was used in the physical prototype was chosen to optimize cost. However, the error message in figure 4.5 was obtained in the SolidWorks environment.

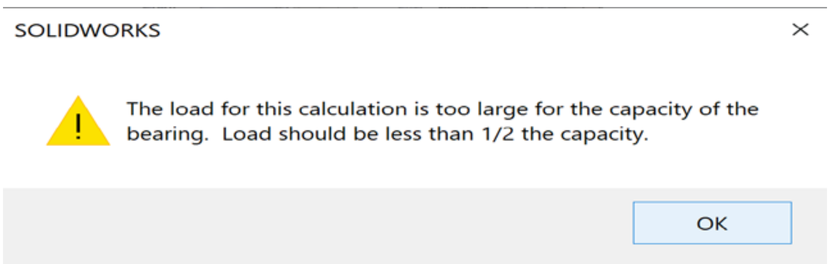

Figure 4. 5: SolidWorks Error Message

Hence, there was no other option than to use the appropriate bearing to carry out another set of bearing calculation with the same reliability, load, and speed parameters as seen in figure 4.6.

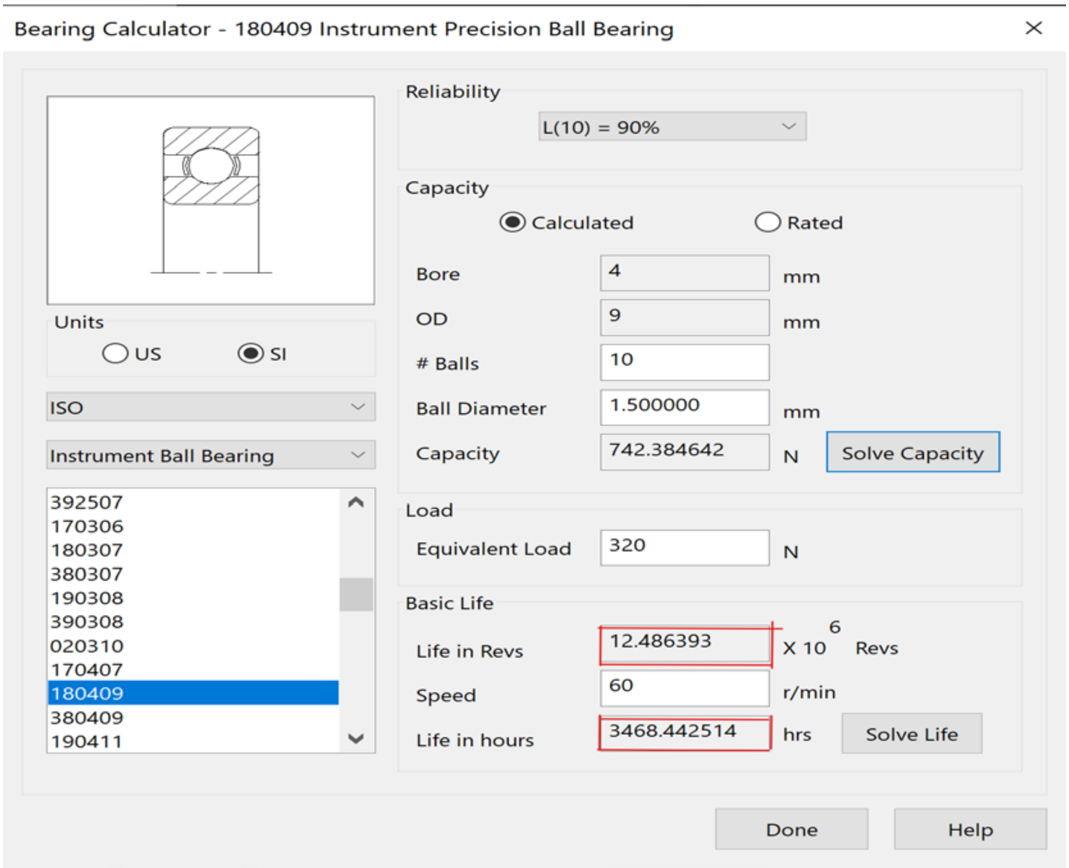

Figure 4. 6: SolidWorks Reliability Analysis on 180409 Ball Bearing

After the SolidWorks analysis, the 180409-ball bearing gave a basic life of 12.48 $*10^6$ revs and 3468.44 hrs. However, since SolidWorks is proprietary software, in the event where one does not have access to the software environment, the Basic Life of the bearing can be computed analytically using the catalogue load rating, C_{10} equation shown below:

$$C_{10} = F_R = F_D * \left(\frac{L_D * n_D * 60}{L_R * n_R * 60}\right)^{1/a} \qquad (4.1)$$

Where F is the radial loading, L_R and L_D are life in hours, n_R and n_D are speeds in rev/min and subscripts R and D are for Rated and Desired. The analytical procedure can be found in Appendix A to have a value of 3435.1 hours which is less than a one per cent error margin of the simulated reliability analysis.

4.3 Safety

A purely static loading is when the failure occurs after one cycle of loading. To calculate for a single static event, a design theory must be chosen based on the type of material, either brittle or ductile. Since the material used for the fabrication of the existing prototype was galvanized steel, which is a ductile material, a ductile design theory was chosen. There are several ductile design theories, but Maximum Distortion Energy Theory (MDET), which is the next equation shown below, was chosen as the static failure theory since that is the same failure theory used by SolidWorks.

$$\textit{Factor of safety, } n = \frac{\boldsymbol{s_y}}{\boldsymbol{\sigma_e}} \tag{4.2}$$

$$\sigma_e = \frac{1}{\sqrt{2}}\left[\left(\sigma_x - \sigma_y\right)^2 + \left(\sigma_y - \sigma_2\right)^2 + (\sigma_2 - \sigma_x)^2 + 6\left(T_{xy}^2 + T_{yz}^2 + T_{zx}^2\right)\right]^{1/2} \tag{4.3}$$

4.3.1 Static and Fatigue Analysis for Rotating Shaft in the Sheller

For the first static event, the MDET equation above was used to analytically compute a safety factor of 2.7, as seen in Appendix A. To cross-check this value, static analysis was run in SolidWorks, which gave a factor of safety of 2.4, as shown below in figure 4.7. For this computer analysis, galvanized steel was first chosen as the material. Three fixtures were defined: a bearing support at one end, a roller/slider for the shaft's cylindrical face, and a translational zero constraint in the x-direction. These boundary conditions ensure that only rotational motion in the Y-Z direction is possible for the rotating shaft. Lastly, a maximum torque of 320N.m was applied to the face opposite to the bearing fixture. After a default mesh was created and a simulation was run to generate the result in figure 4.7.

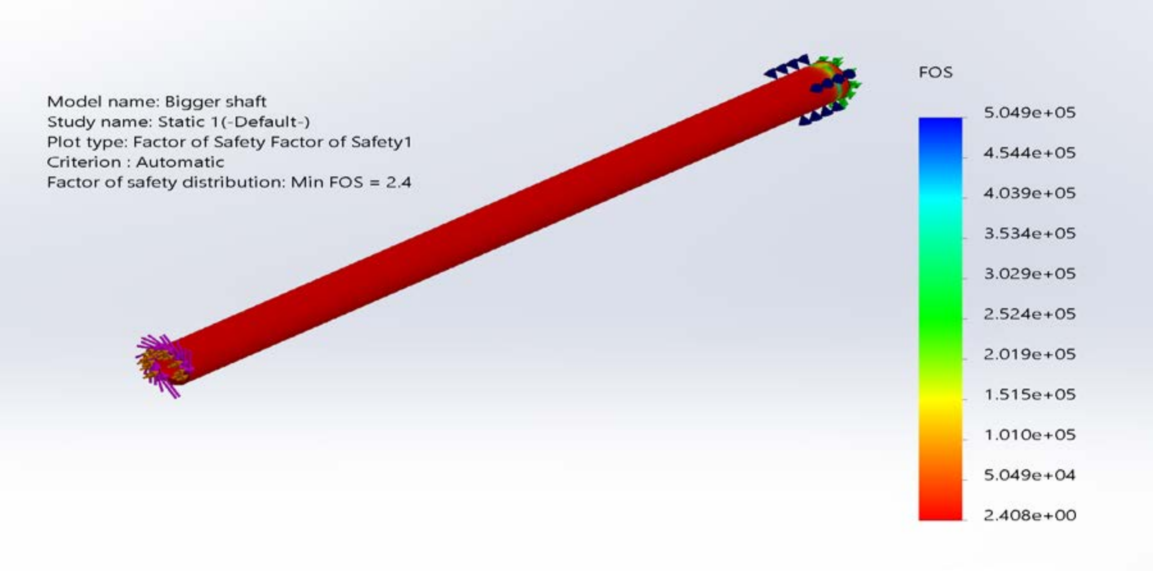

Figure 4. 7: Factor of Safety of a Static Event

From both values of 2.4 and 2.7, since yield strength (Sy) was twice more than the applied stress in the first event of loading, the analysis was continued to investigate if the rotating member would fail by yielding or fatigue.

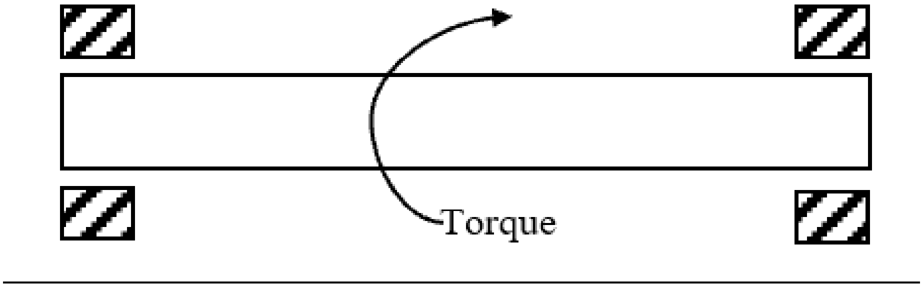

Figure 4. 8: 2D Torque Analysis of Rotating Shaft

A: Load Analysis- Repeated torque loading (T) ~ (0, T-max)

$$Amplitude,\ Ta = 0.5^*\ [T\text{-}max + T\text{-}min] \tag{4.4}$$

$$Mean,\ Tm = 0.5^*\ [T\text{-}max - T\text{-}min] \tag{4.5}$$

For a shaft rotating at a constant speed and transmitting a constant torque, it is considered to have a steady torque, and in this specific case of the rotating shaft inside the sheller:

$$Tm = Ta \tag{4.6}$$

B1: Stress Analysis-Torsion (τ)

$$Torsion,\ \tau = \frac{T}{Z} \tag{4.7}$$

$$Section\ Modulus\ for\ a\ hollow\ shaft,\ Z= \frac{\pi * [D^4 - d^4]}{16D} \tag{4.8}$$

B2: Equivalent Von-Misses Stresses [σ_m'; σ_a']

Since mean torque is equal to amplitude torque,

$$\sigma_m' = \sigma_a' = [\sigma_a^2 + 3Ta^2]^{1/2} \tag{4.9}$$

C: Material Properties

Knowing the tensile and yield strengths of galvanized steel as 356Mpa and 203Mpa, respectively, the design endurance limit, S_e must be specified before any failure theory can be used.

$$S_e = (Surface\ factor,\ K_a) *(Size\ factor,\ K_b) *(Load\ factor,\ K_c) *(Temperature\ factor,\ K_d) * (Reliability\ factor,\ K_e) *(Miscellaneous\ factor,\ K_f) * S_e' \tag{4.10}$$

Where the specimen endurance limit, S_e' was equal to 193.053 MPa, as seen in Appendix A. After computing the parameter modifications, which can also be found in Appendix A, the resulting design endurance limit, Se, was 64.43 MPa.

D: Design Theory

Similar to the static loading, there are several design theories at this stage, but the ASME-elliptic criterion would be used since it best agrees with experimental data and is thus recommended for failure analysis [5]. The first step for the criterion is to check if the operating point of the load lies in the fatigue or yielding region. This is done by investigating using the equations below:

$$s_m = \frac{(s_y - s_e)s_{ut}}{s_{ut} - s_e} \tag{4.11}$$

$$S_a = S_y - S_m \tag{4.12}$$

After the variables S_m and S_a have been computed, figure 4.9 helps determine whether the load region is in the fatigue region or the yield region.

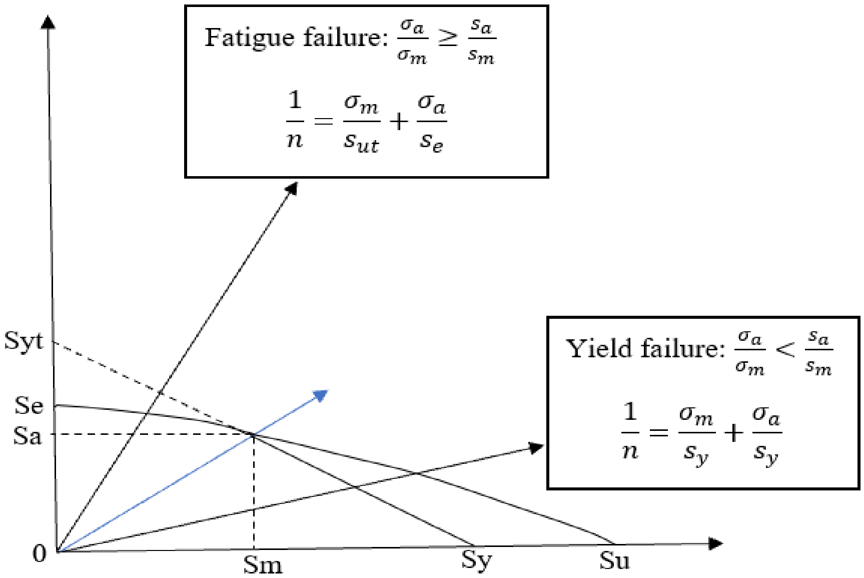

Figure 4. 9: ASME-Elliptic Criterion

For the case of the rotating shaft, it was determined that the failure of the loading lies in the fatigue region by comparing the design endurance limit of 64.43 MPa from Appendix A with the yield strength of the material, which is 203.94 MPa. Since the yield strength was greater than the design endurance limit, it was decided that the loading laid in the fatigue region. Hence, a fatigue loading simulation was done in figure 4.10 to investigate how the rotating shaft would perform for a life cycle of $10^{6,}$ which is considered infinite life for the galvanized steel material chosen [5].

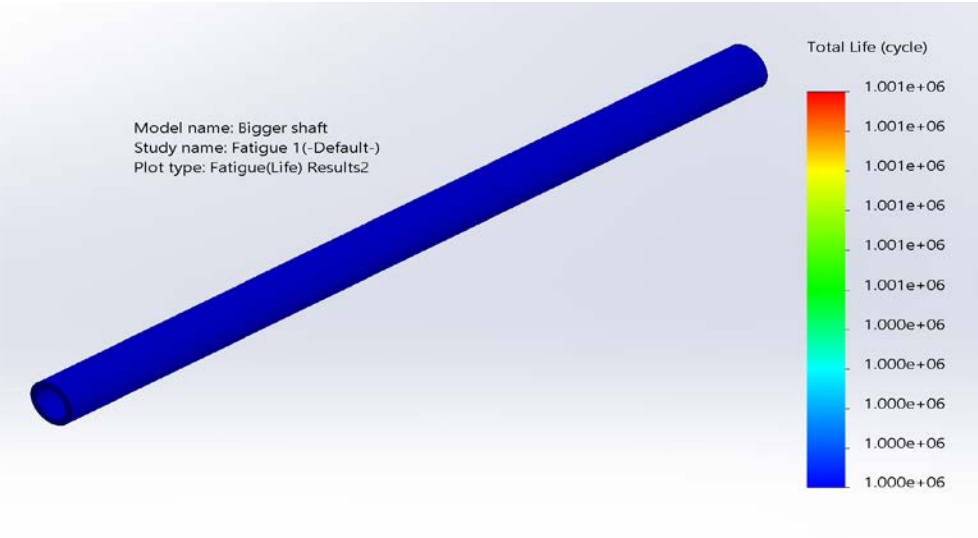

Figure 4. 10: Total Life of a Rotating Shaft Under Fatigue Loading

4.3.2 Static and Fatigue Analysis for Bicycle Pedal Cleats

Bicycle cleats, which can be either metallic or plastic, are the culprit for most pedal problems [22]. Thus, since pedal power was the input energy for the shelling configuration, it was paramount for analysis to be done on the current pedal cleats. As done in section 4.2.1, the analytical stress formulas are the same for this section. Hence, to avoid monotony, the steps would not be repeated since the MATLAB code found in Appendix A can always be used for re-calculation. For the set-up, as seen in figure 4.11, a total of 132lbs is applied to both cleats to begin the static event analysis.

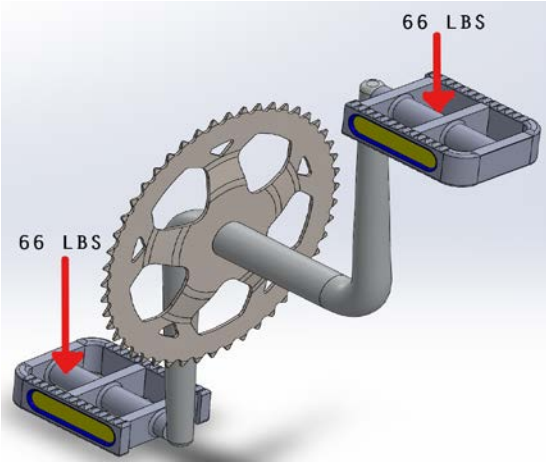

Figure 4. 11: Load Diagram for Forces Normal to Bicycle Cleats

As stated earlier, for pedal cleats, they can either be plastic or metallic, but a metallic cleat was used in the existing configuration. To be specific, an alloy steel cleat was used in fabrication, and this is the same material used to run the static simulation, as seen below in figure 4.12. After a translational zero constraint in the z-direction was created as a boundary condition. In the opposite z-direction, a force of 294.3N was applied normal to the face of the cleat. A default mesh was then created to produce the results evident in figure 4.12.

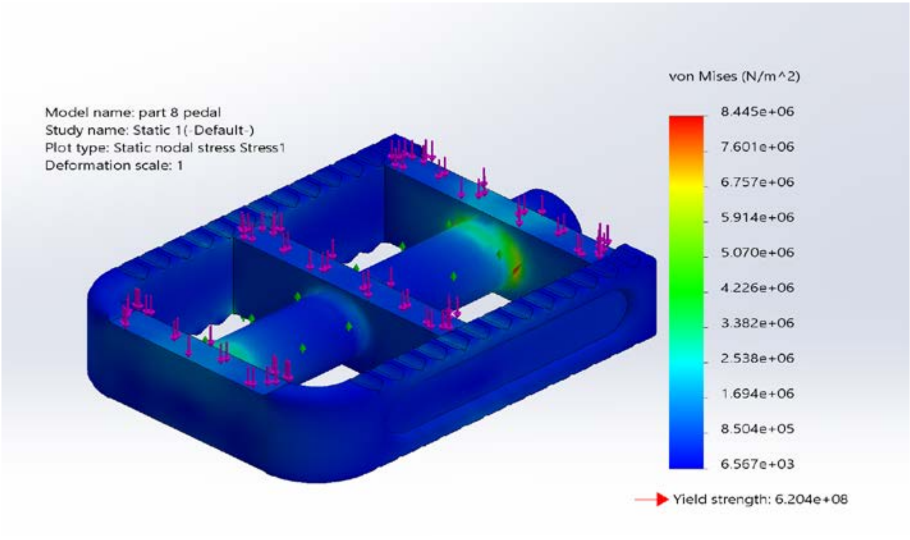

Figure 4. 12: Static Load Analysis on Bicycle Cleats

As seen in figure 4.12, the corners stresses of the cleat shown in a mixture of red and green were investigated to be found as hotspots for the first event loading when a normal force was applied. To mitigate this situation, a re-design was attempted by introducing fillets to reduce the stress concentration at those corners. Since a steel material was chosen for the bicycle cleat, a life cycle of 10^6 cycles was selected as the simulation run time since that is the number for infinite life cycles for steel [5].

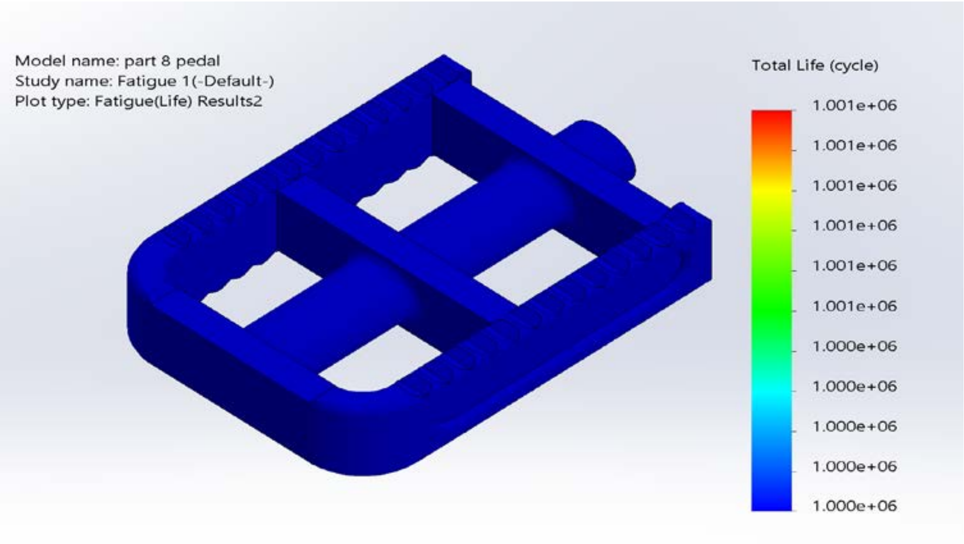

Figure 4. 13: Fatigue Load Analysis on Bicycle Cleats

4.4 Optimization of Cylinder Stand's Weight

After secondary feedback from users of the existing prototype, it was determined that the shelling assembly was heavier than it needed to be [4]. Thus, a topology optimization approach was chosen to investigate a way to make the assembly lightweight. The subassembly which was focused on was the stand of the cylinder since its analysis could easily be conducted. First, the cylinder stand was modelled in the SolidWorks software environment, and a simulation study was decided by first selecting a criterion that would maximize the stiffness to weight ratio. This criterion was chosen to preserve the cylinder's structural integrity so it can still carry out its task of supporting the cylinder stand. After the load forces exerted by the shelling cylinder were defined normal to the cylinder stand as seen in figure 4.14.

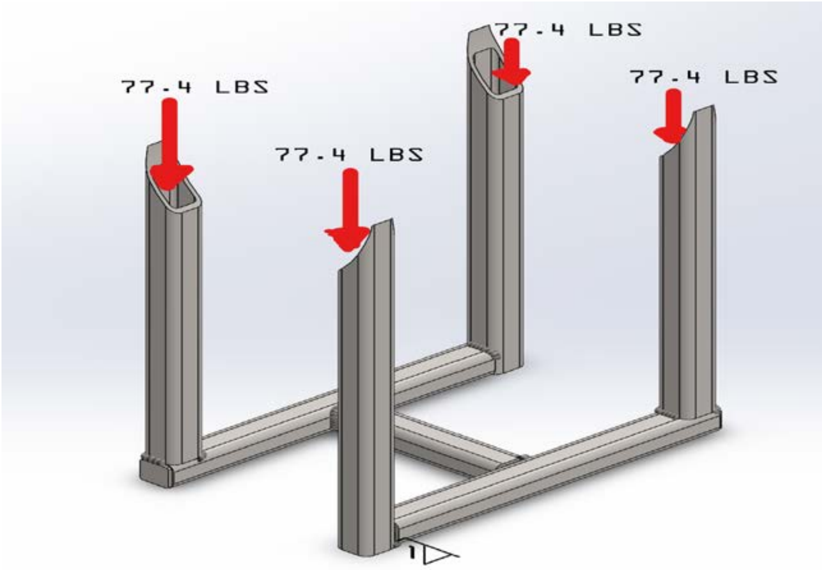

Figure 4. 14: Load Distribution on Cylinder Stand

Due to insufficient computational power from the computer system being used to run the simulation, analysis was run on only one 77.4lbs normal force. Since each vertical support received the same normal force, it was assumed that an analysis on only one vertical structural member, as seen in figure 4.15, would be a close approximation of what would happen if a simulation were run on the whole cylinder subassembly. Thus, as seen in figure 4.15, fixtures perpendicular to the right plane were defined upwards to the base of the member to prevent excessive displacements to the left or right during simulation. Considering the one-quarter model in figure 4. 15, it is expected that there would be a slight variance between the results obtained and that of the complete stand subassembly since assumptions such as uniform material properties are being made.

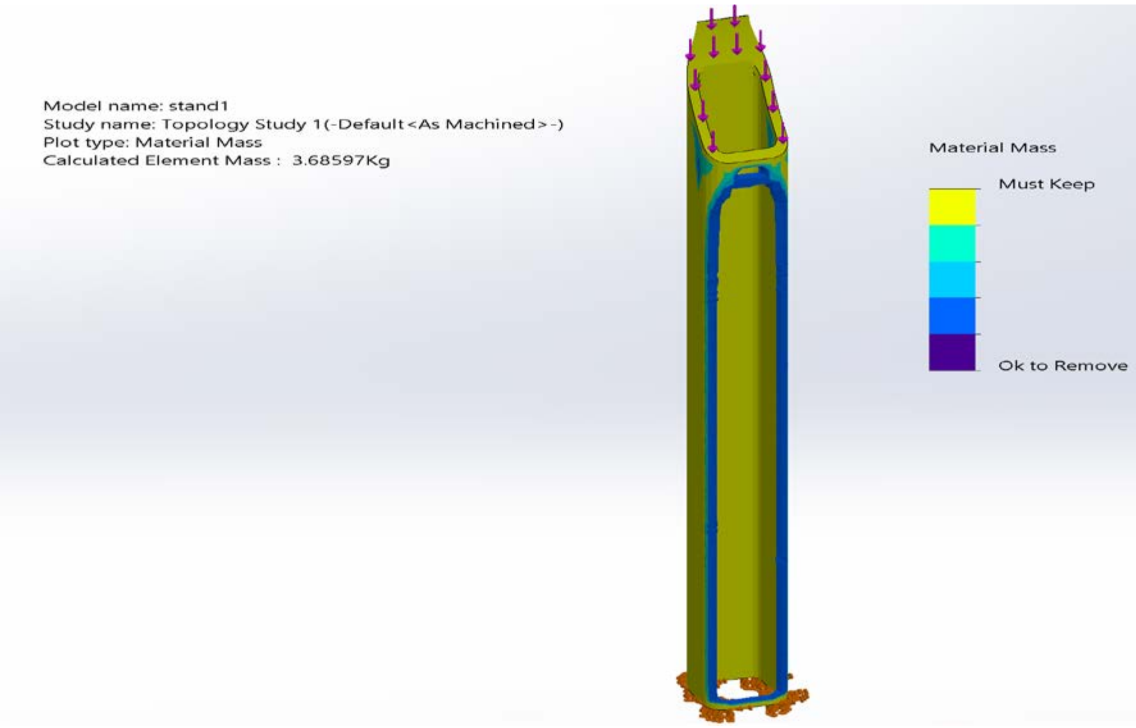

Figure 4. 15: Weight Optimization Analysis on Structural Member

4.5 Trend Forecast for Shelling Performance

The core purpose of the bicycle-powered assembly is to shell maize. Hence this sub-section focuses on how the changes made in a re-design could potentially affect the core aim of the assembly, which is shelling maize. The machine's shelling performance is hugely affected by the speed of the rotating sheller shaft, which can increase or decrease the shelling rate (gram/s). Also, the rotational shaft is constrained by the life cycle of the bearing since a higher rotational speed would reduce the life in hours of the bearing.

Using a benchmark value of above 3000 hours [23] as a safe operation for the bearing, a goal-seek function was employed to investigate the rotational speed that would yield a life cycle of 3000 hours. The bearing calculator interface, as seen in figure 4.6, was used to calculate the bearing life in hours for varied rotational speeds, which produced the plot in figure 4.16. A data-fitting analysis was then carried on the varied data results to calculate a line fitting equation of, $y = -49.25x + 6912$ which can be used to predict data values that fall outside the range of the displayed graph. This prediction would be close to accurate since the coefficient of determination, R^2, provides a strong 0.9982 linear correlation between the independent and dependent variable.

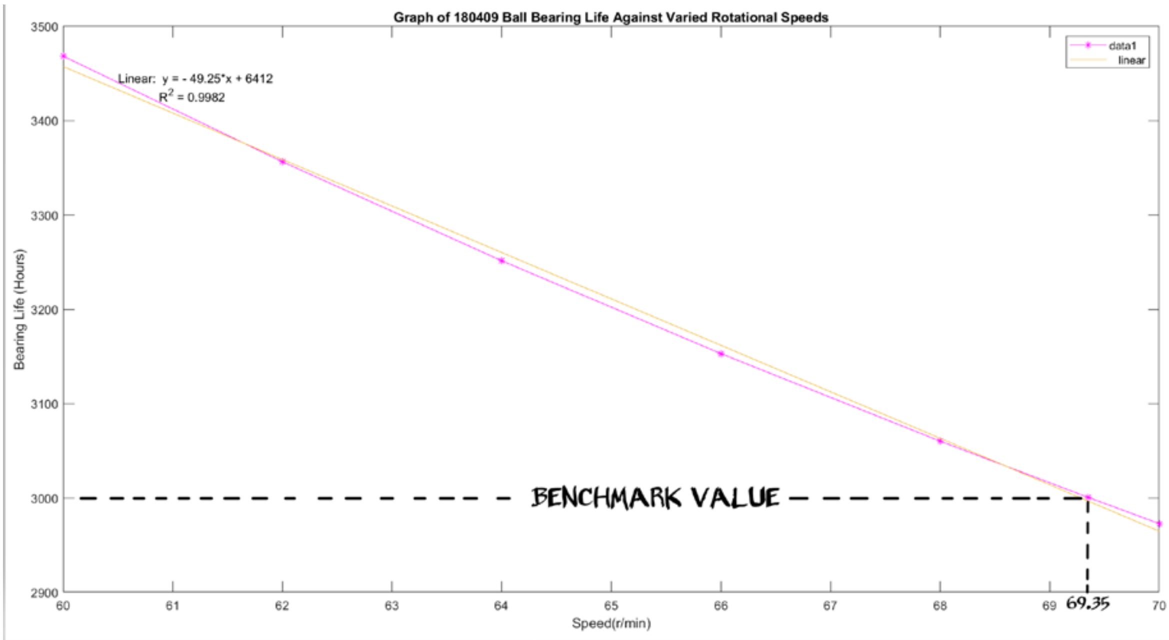

Figure 4. 16: Graph of Bearing Life Against Varied Rotation Speeds

From figure 4.16, it can be identified that any rotational speeds above 69.35 r/min would negatively affect the bearing's design life. Hence, 69.35 r/min can be set as the threshold rotational speed for the assembly.

Table 4.1 shows the experimental results obtained when the existing physical prototype was used for shelling [4]. For the values obtained in Table 4.1, a 60 rpm and a gear ratio of 1:2 was used to produce the experimental values [4].

Table 4. 1: Experimental Results [4]

Trial	**Performance Table**			
	Shelling rate (gram/s)	***% Unshelled***	***Shelling ability***	***Kernel damage (%)***
1	36.6	1%	99%	4.25%
2	42.3	5%	95%	5.72%
3	38.7	3%	97%	3.32%
Average	39.2	3%	97%	4.43%

Since a threshold value of 69.35rpm has been set in figure 4.16, to make predictions about shelling performance, the same method used to obtain the results in Table 4.1 must be followed for consistency. Thus, if a gear ratio of 1:2 gives 60rpm, then for 69.35rpm, you would need a 1:2.13 gear ratio. It is predicted that this difference in gear ratio would provide experimental values different from what is in Table 4.1. Thus, section 5.5 would be dedicated to explaining the trade-offs and implications of a gear ratio on experimental values. However, before that, the following assumptions are to be made:

- There is a linear relationship between shelling rate and gear ratio.
- There is an inverse relationship between kernel damage and gear ratio.

From the above assumptions, a factor of 1.13 would be multiplied to the shelling rate, and 0.87 would be multiplied to the kernel damage to obtain the forecasted values. Though the factors are based on assumptions, they are derived from the difference between the two gear ratios since those are the only new variables in the experiment.

Chapter 5: Discussion

5.1 Overview

This chapter explains the implications of the data obtained in the previous chapter to ensure the assembly fulfils its core objectives. Using a sequential approach, data from the four objectives, namely reliability, safety, weight, and trend forecast, would be discussed.

5.2 Discussion of Analysis for Bearing

After the bearing analysis, the ideal one that was selected was the 180409 Instrument Precision Ball Bearing. Concerning the numbering attached to the name of the bearing, the third and fourth numbers refer to the bore diameter of the bearing, while the fifth and sixth numbers refer to the outer diameter. Ideally, bearings can be chosen based on the type of contact, whether it is a sliding contact or a rolling contact. Since the power transmission from the chain drive provides a radial loading, a ball bearing was chosen since it is a type of rolling bearing that is manufactured to take pure radial loads [5]. Other types of rolling bearings could have been selected, but since the starting friction of the whole assembly was not relatively large, that was why a ball bearing was chosen. From figure 5.1, just as the name goes, ball bearings have got balls in them, which would help to explain the implication of the 90% reliability analysis done in the previous chapter.

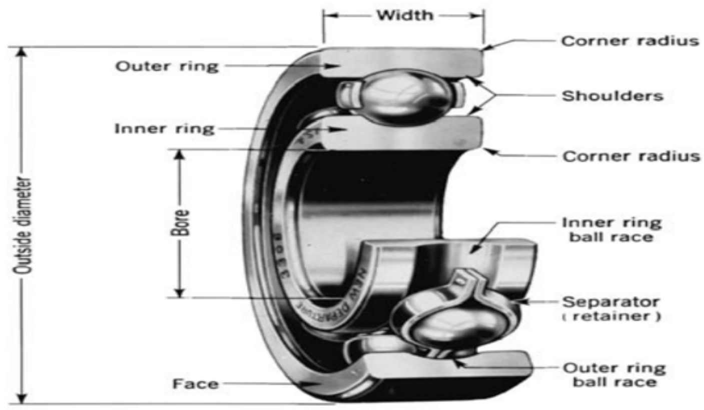

Figure 5. 1: Labelled Image of a Ball Bearing [5]

For a ball bearing with a 90 per cent reliability, also known as L_{10} life or B_{10} life, it is defined as the number of revolutions (or hours at a constant speed) that 10 per cent of the balls within the ball bearing would begin to develop evidence of failure. Different reliability percentages would give different life cycle values, however comparing the 3468.44 hours obtained after the analysis of the bearing to figure 5.2, it can be seen the values fall within the design life for agricultural equipment. The implication of the design life means it would take 3468.44 hours for 10 per cent of the balls within the ball bearing to begin developing evidence of failure.

Application	Design life L_{10}, h
Domestic appliances, instruments, medical apparatus	1000–2000
Aircraft engines	1000–4000
Automotive	1500–5000
Agricultural equipment, hoists, construction machines	3000–6000
Elevators, industrial fans, multipurpose gearing, rotary crushers, cranes	8000–15 000
Electric motors, industrial blowers, general industrial machines, conveyors	20 000–30 000
Pumps and compressors, textile machinery, rolling mill drives	40 000–60 000
Critical equipment in continuous, 24-h operation; power plants, ship drives	100 000–200 000

Figure 5. 2: Bearing Design Life Value for Different Applications[23]

Though the analysis gave a calculated design life, this value can be reduced if maintenance such as lubrication is not applied to the bearing. The most common form of lubrication is hydrodynamic lubrication, where the load-carrying surfaces of the bearing are separated by a relatively thick film of lubricant to prevent metal-to-metal contact [5].

5.3 Discussion on Static and Fatigue Analysis

5.3.1 Analysis for rotating shaft

For the static event, a 2.7 and 2.4 factor of safety values were obtained from both analytical and numerical calculations, respectively, accounting for a 12.5 error margin. This difference in values is because, for the numerical instance, a bearing fixture was defined for the shaft while a bearing was not included in the analytical to investigate the range of factor of safety values that can be obtained by the shaft. Nonetheless, once the shaft was deemed safe for a static event, an analysis was conducted to probe if the shaft would be able to endure multiple cycles of loading. Since the yield strength of 203.94MPa for galvanized steel was higher than the design endurance limit of 64MPa, it was determined that failure was most likely to happen by fatigue than yielding.

Hence, once a steel S-N curve was selected in the software environment, and a load cycle of 10^6 was chosen, it was discovered that the stresses for a million cycles as too low to be concerned about fatigue, as seen in figure 5.3. A cycle time of 10^6 cycles was chosen because, based on the steel S-N curve, infinite life usually starts after a million cycles.

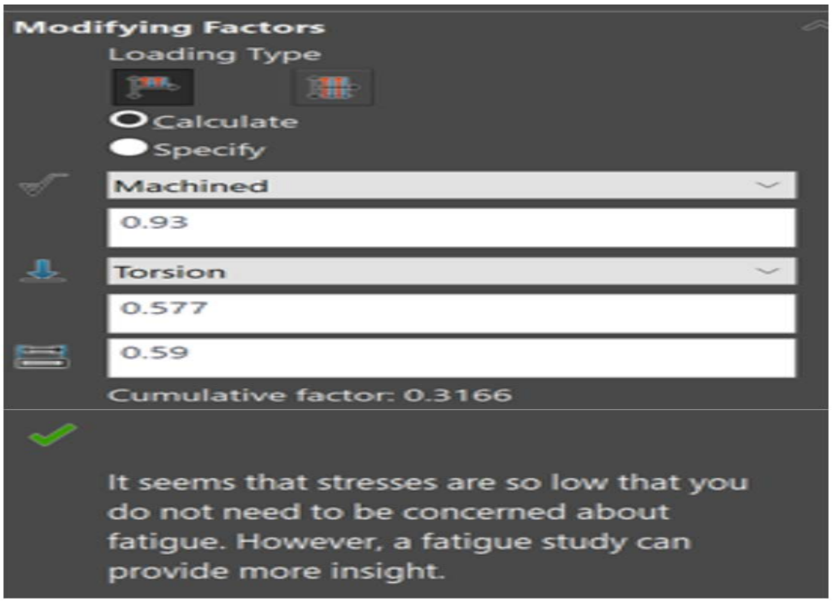

Figure 5. 3: Fatigue Analysis for Rotating Shaft

5.3.2 Discussion of Safety Analysis for bicycle cleats

After the simulation on the bicycle cleats, a re-design must be informed for the cleats since stress hotspots were detected at the corners. As stated previously, the addition of fillets would help to alleviate the stress at those hotspots. For the fatigue analysis, the S-N curve for steel, as seen in figure 5.4, was used to determine the number of cycles needed for infinite life. This was because the required S-N curve for alloy steel was not available in the software environment; hence that of steel was chosen as a close approximation.

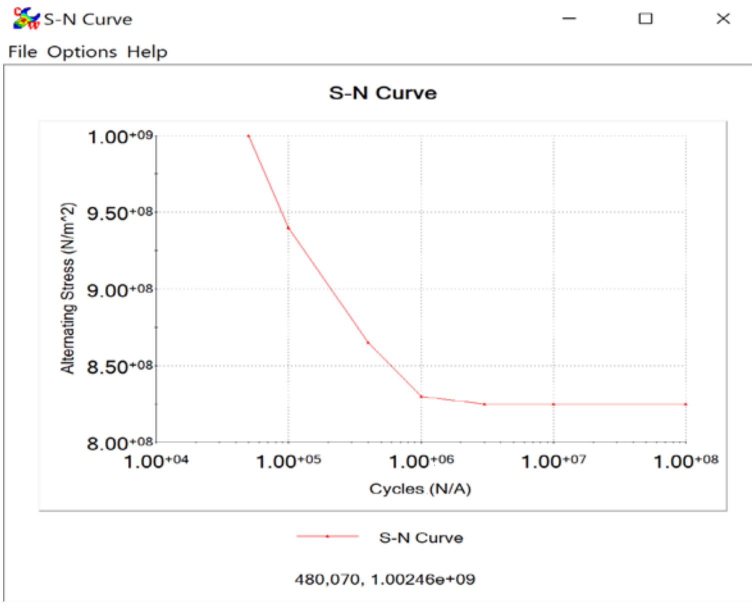

Figure 5. 4: S-N Curve for Steel

5.4 Discussion of Analysis from Weight Optimization.

Using a goal constraint of reducing the cylinder stand's mass by 30 per cent from the weight optimization section, the analysis provides a calculated elemental mass of 3.68597*kg* at the top left side of figure 4.15. With each vertical structural member having a mass of 5.2*kg* each, that means 1.51403*kg* of mass can be removed from each structural member. In other words, since there are four vertical members, 6.05612*kg* material than necessary was used for the fabrication of the cylinder stand. Hence in a re-design attempt to make the cylinder stand lightweight, an angle grinder can be used to remove the excess mass from the cylinder stand. Nonetheless, a reasonable margin of error should be allowed since the simulation was not run on the whole cylinder stand subassembly. Another alternative would be to use a cheaper material than what was used to fabricate the cylinder stand to save on cost.

5.5 Discussion of Shelling Performance Trend Forecast

For a higher gear ratio, there would be a higher rotational speed which would increase the shelling rate. However, it should be noted that there would be a trade-off for kernel damage since an increased speed would provide an excess of rotational power which can damage the maize kernels. Ideally, this optimization analysis could be better analyzed by carrying out experimental trials to predict if a higher or lower gear ratio would have a better shelling performance. Nonetheless, that does not prevent a forecast of trends for shelling performance. The main trends include:

- An increased rotational speed from a higher gear ratio would increase the shelling rate. However, it cannot surpass 69.35 r/min.
- A higher gear ratio would lower the design life for the same selected bearing since higher gear ratios provide higher rotational speeds.
- A decreased rotational speed would provide a low kernel damage rate, preventing the corn kernels from being easily attacked by pests.

Based on the assumptions made in section 4.5, the predicted experimental results would be as seen in Table 5.1, where the shelling rate is multiplied by 1.13 and kernel damage is multiplied by 0.87. These factors result from the difference between the two gear ratios since those are the only new variables in the experiment.

Table 5. 2: Updated Experimental Results [4]

Trial	**Performance Table**			
	Old Shelling rate (gram/s)	***Predicted Shelling rate (gram/s)***	***Old Kernel Damage (%)***	***Predicted kernel damage (%)***
1	36.6	41.4	4.25%	3.70
2	42.3	47.8	5.72%	4.98
3	38.7	43.7	3.32%	2.89
Average	39.2	44.3	4.43%	3.86%

From Table 5.1, a higher gear ratio would be recommended since, from the comparison of values, the increase in values from the shelling rate is relatively greater than the decrease in kernel damage.

Chapter 6: Challenges, Recommendations & Conclusion

6.1 Overview

For this chapter, key takeaway results derived from the analysis would be re-stated. In addition, this chapter will explain the challenges and limitations encountered during the virtual execution of this project. Recommendations will also be suggested for the future work of the project.

6.2 Challenges Encountered & Limitations

In this project, the major limitation was the restrictions imposed by the COVID-19 pandemic, which constrained the project to be carried from home. Though an existing prototype has been fabricated, trials were not able to be conducted on the assembly. The next challenge was the SolidWorks educational package limitation that did not allow access to certain S-N curves needed for fatigue analysis. Hence, approximations had to be made to provide a rough estimate of how the required S-N curve would look like. Lastly is the slow computer system processor that impedes the amount of simulation that can be performed. Thus, in the case of weight optimization, a quarter model had to be used to make an approximation for the whole subassembly.

6.3 Recommendations for Future Works

Due to the significant challenge of being physically restricted, specific experimental trials were not able executed. Hence, future work recommendations include conducting tests on the physical prototype to validate the forecasted trends that were projected in section 5.5. In addition, future work would include an introduction of the analysed re-design ideas into the physical prototype to investigate the extent to which the analysis matches experimental results.

6.4 Conclusion

As stated in the beginning, this project is focused on migrating a physical prototype to a software environment with the aim of performing structural analysis to inform a re-design. At the end of the analysis, the key take-away points were:

- A 180409-ball bearing is the appropriate bearing to use for the assembly since it provided a design life that can be benchmarked [23] as safe for usage.
- Since most of the stresses were below that of the material’s yield and tensile strengths, for both static and fatigue analysis, the critical parts of the assembly are safe for usage for infinite life based on the loads used for its analysis.
- A weight optimization analysis proved that 6*kg*, which is 30 per cent of the stand sub-assembly, can be eliminated through subtractive manufacturing without affecting the assembly's structural integrity. As a percentage of the weight of the entire assembly, this weight reduction would represent a 3 per cent reduction of the weight of the assembly.
- Lastly, trends were forecasted to make evident the trade-offs between choosing a high or low gear ratio. To compare the forecasted trends with actual data, physical experiments have to be performed, but for now, a higher gear ratio is recommended.

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Appendix A: MATLAB Code to Compute Needed Analytical Values

Calculation for Bearing Life

```
Fr=740; Fd=320 ; nd=60; a=3;
syms Ld
double(solve(Fr==Fd*(Ld*nd*60/1e6)^(1/a),Ld))
```

```
ans = 3.4351e+03
```

Calculations for Static and Fatigue Analysis

1. Constant Torque.
2. An ideal case of no transmission losses via chain mechanism.
3. Material is homogeneous throughout each structural member.

```
T=320e3; %Torque in mm*N

Sy=203.94; %Yield Strength in Mpa

Sut=356.9 ;%Tensile Strength in Mpa

D=38.1; %1.5 inch in millimetres

d=28.194; %1.11 inch in millimetres

Zp=(pi*(D^(4)-d^(4)))/(16*D) %Section Modulus for hollow shaft
```

```
Zp = 7.6030e+03
```

```
Torsion= T/Zp  %Mpa
```

```
Torsion = 42.0885
```

Calculating static F.O.S. of shaft in sheller

```
Oe= (1/sqrt(2))*((6*(Torsion).^2))^.5
```

```
Oe = 72.8995
```

```
MDET_FOS=Sy/Oe
```

```
MDET_FOS = 2.7976
```

Calculating design endurance limit,Se

```
a=4.51;b=-0.265 %Marin surface factors[5]
```

```
b = -0.2650
```

```
Ka= a*(Sut^b) %Surface factor
```

```
Ka = 0.9501
```

```
Kb= 0.879*(D^-0.107)  %Size factor for rotating round diameter shaft
```

```
Kb = 0.5954
```

```
Kc= 0.59 %Load factor under torsion
```

```
Kc = 0.5900
```

```
Kd= 1.0  %Temperature factor for less than 200C
```

```
Kd = 1
```

```
Ke= 1.0  %50% Reliability
```

```
Ke = 1
```

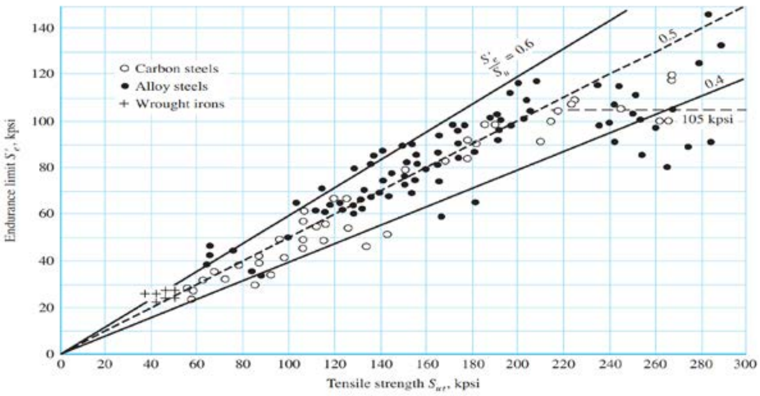


Figure A. 1: Endurance limits versus tensile strengths [24]

For Sut=356.9MPa=51.763969kpsi, tracing from the above graph, the specimen endurance limit, Se' = 193.053 MPa.

```
Se_prime=193.053;
Design_Se= Ka*Kb*Kc*Kd*Ke*Se_prime
```

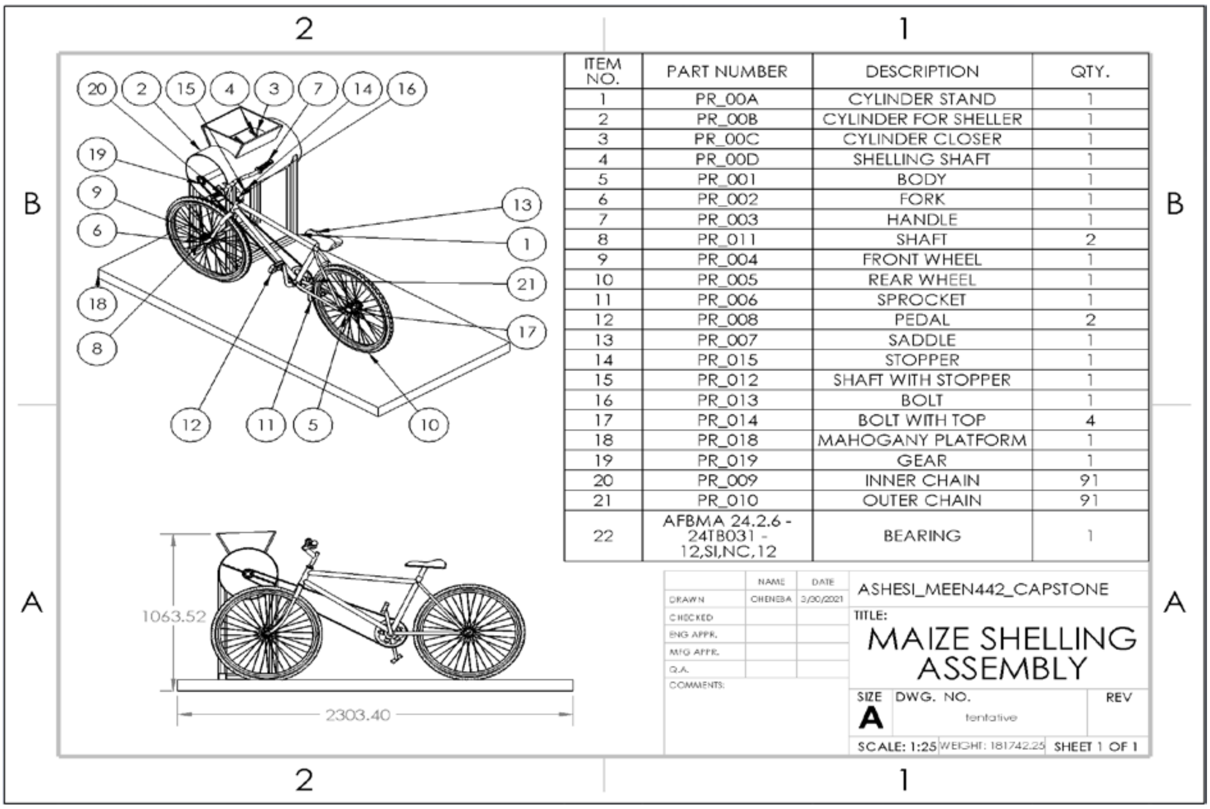
```
Design_Se = 64.4333
```

Comparing the 64.433MPa design value obtained with the yield strength of galvanized steel which is 203.94MPa, it is evident that the material would fail by fatigue loading before yielding; hence a fatigue criterion would be chosen.

<u>Plot of bearing life cycle.</u>

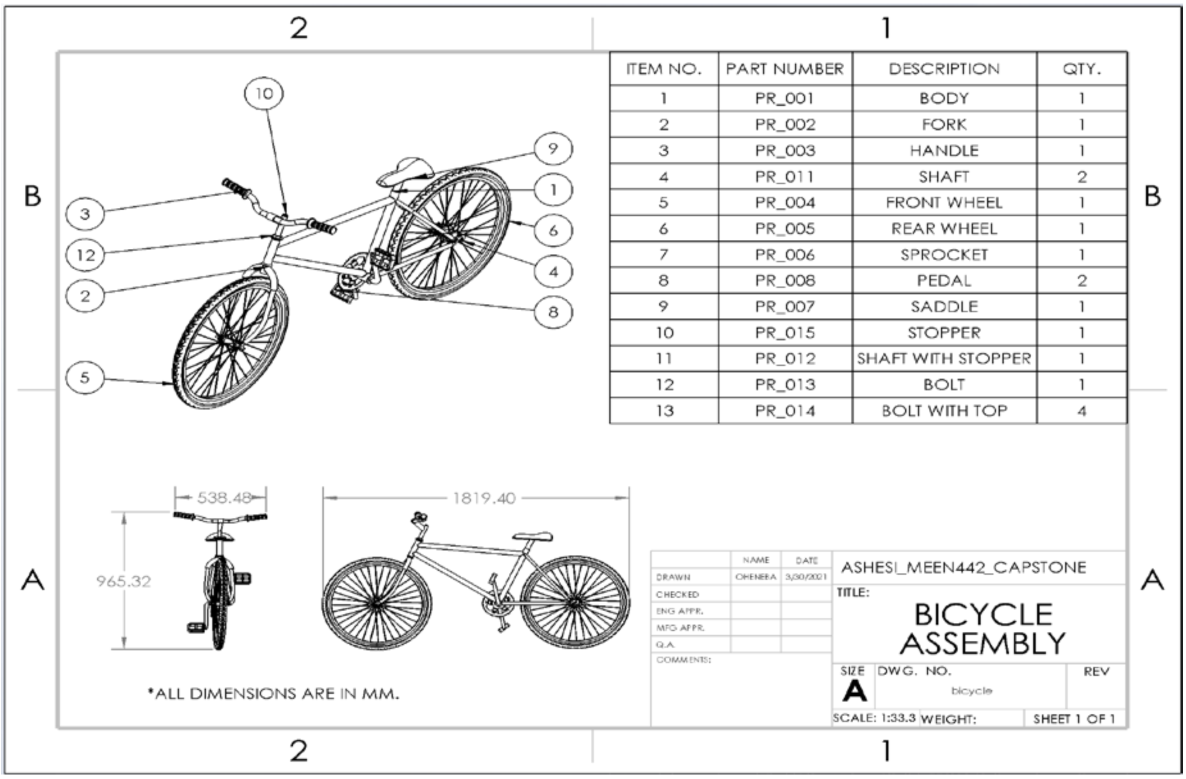
```
speed=[60;62;64;66;68;69.35;70];
hours=[3468.442514;3356.557272;3251.664857;3153.129558;3060.390454;3000.815
44;2972.950727];
plot(speed, hours,"*-m");
```

Appendix B: Drawings of Essential Parts of Maize Sheller



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	PR_00A	CYLINDER STAND	1
2	PR_00B	CYLINDER FOR SHELLER	1
3	PR_00C	CYLINDER CLOSER	1
4	PR_00D	SHELLING SHAFT	1
5	PR_001	BODY	1
6	PR_002	FORK	1
7	PR_003	HANDLE	1
8	PR_011	SHAFT	2
9	PR_004	FRONT WHEEL	1
10	PR_005	REAR WHEEL	1
11	PR_006	SPROCKET	1
12	PR_008	PEDAL	2
13	PR_007	SADDLE	1
14	PR_015	STOPPER	1
15	PR_012	SHAFT WITH STOPPER	1
16	PR_013	BOLT	1
17	PR_014	BOLT WITH TOP	4
18	PR_018	MAHOGANY PLATFORM	1
19	PR_019	GEAR	1
20	PR_009	INNER CHAIN	91
21	PR_010	OUTER CHAIN	91
22	AFBMA 24.2.6 - 24TB031 - 12,SI,NC,12	BEARING	1

Figure B. 1: Bill of Material for the Whole Assembly



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	PR_001	BODY	1
2	PR_002	FORK	1
3	PR_003	HANDLE	1
4	PR_011	SHAFT	2
5	PR_004	FRONT WHEEL	1
6	PR_005	REAR WHEEL	1
7	PR_006	SPROCKET	1
8	PR_008	PEDAL	2
9	PR_007	SADDLE	1
10	PR_015	STOPPER	1
11	PR_012	SHAFT WITH STOPPER	1
12	PR_013	BOLT	1
13	PR_014	BOLT WITH TOP	4

Figure B. 2: Bill of Material for Bicycle Subassembly

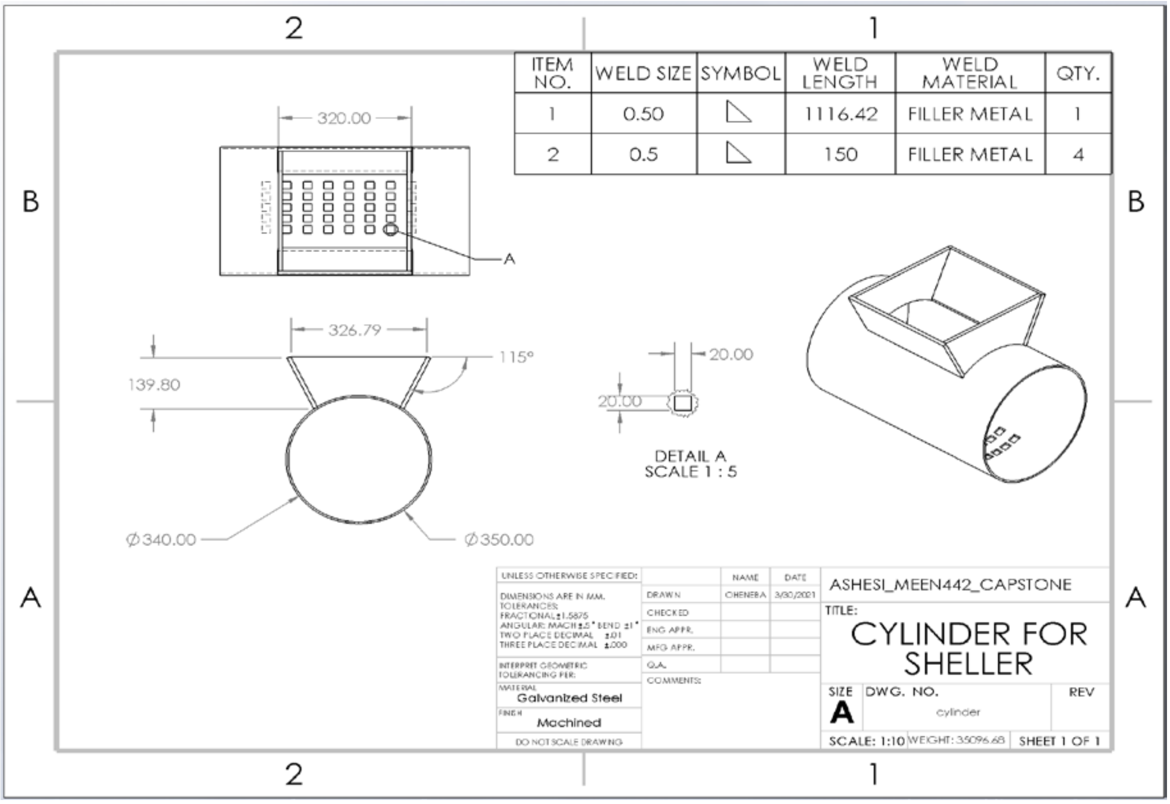


Figure B. 3: Drawing for Sheller's Cylinder

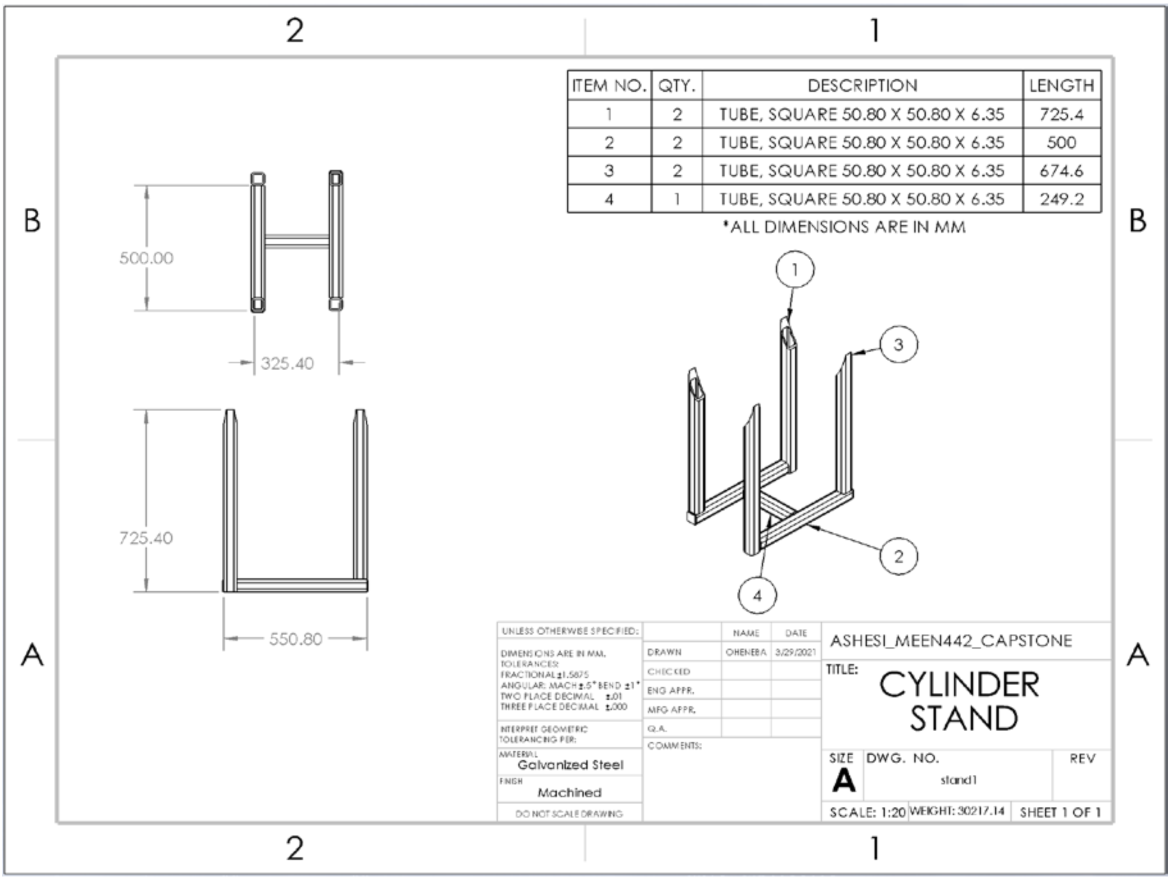


Figure B. 4: Drawing for Cylinder Stand

Glossary

Coefficient of determination, R^2: Similar to the correlation coefficient, the coefficient of determination gives a percentage of how many data points fall within the results of the line formed by the regression equation.

Elastic Modulus: It is the measure of a material's resistance to being deformed non-permanently when a stress is applied to it, and it is defined by the slope of the stress-strain graph in the elastic deformation region.

Factor of safety: This expresses a ratio of how much stronger a material is than it needs to be for an applied stress, either for a single-loading (static) or multiple-loading(fatigue) events.

Poisson's ratio: This is the measure of the expansion or contraction of a material in directions perpendicular to the direction of loading. It relates the lateral and longitudinal elastic deformation, which occur simultaneously because of compressive or tensile stresses.

Roller fixture: This is a SolidWorks simulation fixture that allows the face of a body to move freely in its plane but prevents it from moving in a direction normal to its plane. The face can shrink or expand under loading [26].

S-N Curve: Usually obtained from completely reversed stress cycles, the S-N curve is a plot of the magnitude of an alternating stress versus the number of cycles to failure for a given material [5].

Tensile strength: It is the measure of the force require to pull a material to the point where it breaks [25].

Yield strength: Yield strength is a material property that corresponds to the level of stress at which the material surpasses elastic deformation and begins to deform "plastically" or permanently. This point is known as the yield point, which is indicated on a stress-strain curve.