KWAME NKRUMAH UNIVERSITY OF SCIENCE AND TECHNOLOGY

WEIGHT OPTIMIZATION OF SMALL TRICYCLE TRUCK WITH TIPPING MECHANISM

PRESENTED BY

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DECLARATION

I hereby declare that this thesis is the result of my own original research work undertaken under the supervision of the undersigned, that all works consulted have been referenced and that no part of the thesis has been presented for another degree in this university or elsewhere.

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(Candidate)	Signature	Date

CERTIFICATION

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Dr. S. M. Sackey	………………………	……………………….
(Head of Department)	Signature	Date

ABSTRACT

Weight Optimization of Small Tricycle Truck with Tipping Mechanism

Vitus Mwinteribo Tabie

This thesis presents a weight optimization of small tricycle truck with tipping mechanism. The use of tricycle trucks for waste collection is common in many developing countries. This is because of the heavy investment needed in the purchase of heavy waste collection trucks. The operation of these tricycle trucks in Ghana was very much limited since it could only tip its content on the ground. A tipping mechanism was developed to improved the operation of the equipment. Nonetheless the addition of the tipping mechanism has increased the weight considerably.

In order to reduce the weight of the Tipping Mechanism, finite element analysis was performed on the bin, linkage mechanism, power screw and nut. The exercise involved validating the design changes made in the stress analysis environment. The work flow was repeated until the weight of the designs was optimized against the design criteria. Siemens Solid Edge ST3 software package, NX Nastran (7) solver was used in the optimization process. With optimum weight/ size, the components were produced, assembled and the tricycle tested for functionality. The overall weight of the tricycle has been reduced from 143 kg to 127.6 kg which represents 11% reduction in weight. The test of the prototype results shows the torque requirement of between 20 Nm and 45 Nm to ride on a horizontal plane, an improvement over the existing tricycle (25 Nm – 60 Nm).

DEDICATION

For my handsome, intelligent, and humorous Son Mendel. In your three short years, you have given me the strength and focus to accomplish my goals. You have been my beacon of light when things seemed darkest, and my biggest fan when others vanished.

ACKNOWLEDGMENTS

Merriam-Webster describes a journey as something suggesting travel or passage from one place to another. Two years ago a journey began, and now ends with the completion of this thesis. Many people helped me reach the end of my passage successfully. First, and foremost, I want to thank God, for He made a way when all roads seemed blocked. I want to thank my wife and son for inspiring me to aim higher, work harder, and reach for the stars. Many thanks due to my mother and father for their words of encouragement and inspiration. My immediate and extended families deserve many thanks, for they helped me to keep my eyes on the prize.

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CHAPTER 1

INTRODUCTION

1.1 BACKGROUND

The Tricycle Waste Collector Concept was adopted by Zoomlion Ghana Limited in 2006 when the company started its operation in Ghana. The tricycles come with a container at the rear and can be used effectively in collecting refuse from residential areas that are not accessible to heavy duty waste management vehicles, and discharged into bigger containers. The development and use of the tricycle waste collector has greatly improved sanitation in the country. Currently, Zoomlion has been able to handle 70 percent of the solid waste generated in the towns and cities of Ghana through the use of this concept (Agyepong, 2011). However the operation of the truck is such that it is not capable of delivering its content into disposal containers. The bed of the tricycle is 0.4 m from the ground. The content are therefore dumped onto the ground and later shoveled into the disposal containers which are about 1.0 m high. The illustration is as in figure 1.1.

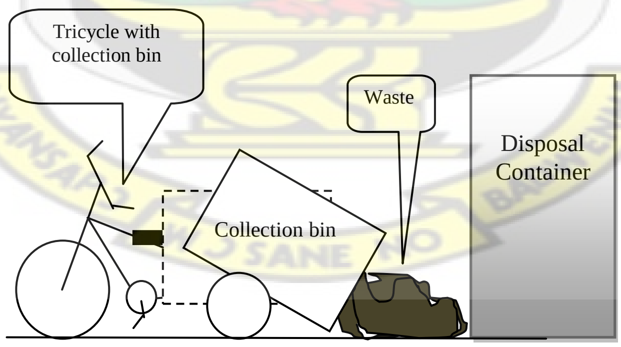


Figure 1-1: Illustration of current solid waste tricycle operation (Source: Fiagbe et al 2011)

A lift-tipping mechanism was developed by Fiagbe et al (2011) to enable the truck directly deposit the waste into the disposal containers. The addition of the tipping mechanism has, however, increased the weight of the tricycle from 127 kg to about 143 kg. Assuming an operator's weight of 70 kg, the total weight would be about 213 kg when empty. The test of the prototype results showed the torque requirement of between 25 Nm and 60 Nm to ride on a horizontal plane. This project seeks to reduce the weight of the tricycle for effective operation.

1.2 OBJECTIVES

The main objective of this research is to optimize the weight of small tricycle truck with tipping mechanism.

The specific objectives are to:

1. Develop optimization criterion (objective function) and constraint functions for each component of the mechanism.
2. Create simulations of various designs of selected components that focus on reducing the mass of the current designs.
3. Redesign and manufacture the optimized components.
4. Assemble and test the tricycle for functionality.

1.3 JUSTIFICATION

The lift-tipping mechanism for tricycles has improved the operation of the equipment Fiagbe et al (2011). Exposure of operator to insanitary condition has been reduced. However, the weight of the equipment has been increased due to the addition of the mechanism. On level ground, a 70 kg person requires about 30 watts to ride at 15 km/h using a bicycle and energy expenditure in terms of kcal/(kg·km) is 1.62 kJ/(km·kg) or 0.28 kcal/(mi·lb) for cycling. However the energy expenditure in using the tricycle with a gross weight of 213 kg (i.e. a 70 kg person riding the tricycle) for 9 km is 0.399 kcal, (Whitt et al, 1982) which is far above energy expenditure for riding a tricycle for 15 km (0.28 kcal). The gross weight of a tricycle is between 150 kg – 200 kg, (Porter, 2002). The gross weight of the tricycle with tipping mechanism is 213 kg which is above the recommended range of weight for tricycle trucks. Reduction of the overall mass is thus much desirable. Weight optimization of the tipping mechanism will help reduce the weight. A minimum weight optimization of the mechanism was done on various elements of the system. The optimum values obtained will be used to redesign the system. This is aimed at reducing the overall weight of the tricycle and hence making operation of the tricycle easier.

1.4 METHODOLOGY

Literature was reviewed with respect to optimization techniques, operation of the tricycle with tipping mechanism and other related areas in order to get acquainted with the relevant works that have been done. An optimization criterion and constraint functions were developed for selected components of the system using basic geometric and stress formulas. Finite element analysis was performed on the bin, linkage mechanism, power screw and nut. The exercise involves validating the design changes made in the Stress Analysis environment. The work flow is repeated until the mass of the designs are optimized against the design criteria.

With optimum weight/ size, the components were produced, assembled and the tricycle tested.

CHAPTER 2

LITERATURE REVIEW

2.1 HISTORY OF TRICYCLE TRUCK

The first tricycle was built in 1680 by a German paraplegic named Stephan Farffler in 1689 who lived near Nuremburg. He was a watch-maker and the tricycle had gears and hand cranks, (Retro Pedal Cars, 2011). A brief history of tricycle is given in Table 2.1 below

Table 2-1: A Brief History of the Tricycle

Date	Activity	Inventor/associated person(s)
1680	first tricycle was built	Stephan Farffler
1789	Tricycle invention was published in Journal de Paris	Blanchard and Maguier
1818	Tricycle patented in England	Denis Johnson
1876	Coventry Lever Tricycle introduced	James Starley
1877	Coventry Rotary Tricycle introduced	James Starley
1879	Twenty types of tricycles were produced in Coventry	
1884	Over 120 different models produced	20 manufacturers
1885	Second generation of tricycles (two rear wheels with front wheel bisecting their track)	Robert Cripps
1892	Third generation of tricycles (All wheels are of equal size)	Starley Psycho
1895	Largest tricycles was manufactured	Waltham Manufacturing Co.
1900	pneumatic tired safety bicycle was developed which took away most of the tricycle business since it provided an adequate amount of stability for most riders	

1998	Largest tricycle record broken. It was built for 24 people and used steering, wheels, an axle, springs, and shock absorbers from trucks. All 24 people pedal. The top speed was 25 mph, and it cruises at about 15 mph. It had three gears, and was 7.7 meters long and 2.04 meters wide. Without passengers, it weighed 0.707 tonnes (1,559 lbs.)	Fumair flyer

2.2 TRICYCLE TRUCK WITH TIPPING MECHANISM

The use of tricycle trucks for waste collection was adopted by Zoomlion Ghana Limited in 2006 when the company started its operation in Ghana. The technology had greatly improved sanitation in the country (Agyepong, 2011). The tricycles are of two main types; manual and motorized tricycle trucks. The method employed in the disposal was unhealthy to the operator. A tipping mechanism was designed by Fiagbe et al (2011) to enable the truck directly deposit the waste into the disposal containers. Figure 2.1 shows the tricycle with tipping mechanism at rest and at full tipping positions.

(a) (b)

Figure 2-1: Tricycle with Lift-tipping mechanism (a) at rest and (b) full tipping positions.

The tipping mechanism is made of linkage bars, and worm and wheel gear set, power screw and nut, and a reinforced welded cross bar structure. The linkage is a five bar mechanism (Figure 2.2). The components are: member 1 – frame; member 2 – lifting bar; member 3 – lifting support bar; member 4 – tipping arm and member 5 – bin. Member 6 is a slider. The lifting bar and lifting support bar are connected at the point ‘C’. The tipping arm, member 4 is connected to the lifting support bar, member 3 at point ‘D’. The tipping arm and the lifting bar are connected to the bin at point ‘E’ and ‘F’ respectively. The lifting support bar is also connected to the frame or ground at point ‘B’. The lifting bar is also connected to the frame at point ‘A’ and is allowed to move or slide horizontally. All joint are pin joints except joint ‘A’ which is sliding along member 1. In operation, reverse paddling is employed in lifting and tipping of the tricycle.

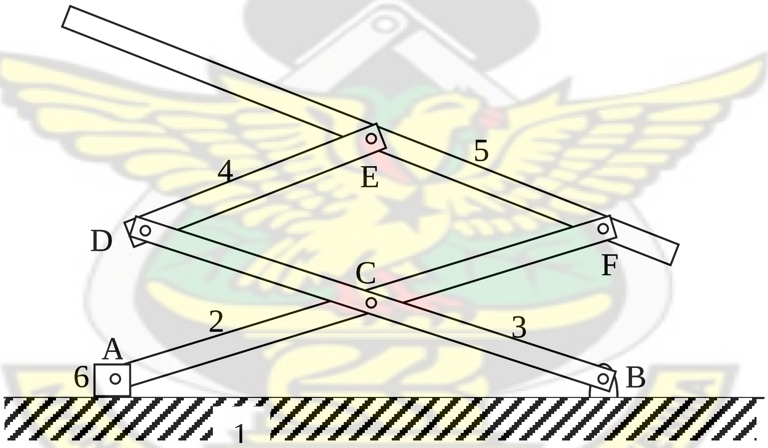


Figure 2-2: Schematic of kinematic linkage chain, (Source: Fiagbe et al, 2011)

2.2.1 Linkage bars

The linkage bars consist of 3 cm×6 cm rectangular bars of length 120 cm for lifting the lifting and its support bars to ensure the tipping arm is of length 35 cm and able to attain the tipping angle of 45^0 needed to achieve full tipping at the disposal site.

2.2.2 Worm and wheel gear set mounting

The worm and wheel gear set arrangement is mounted on the fork assembly of the manual tricycle chassis. The worm shaft has a sprocket which connects with a chain to another sprocket with the pedal. This arrangement as seen from Figure 2.3 is located on the left side of the tricycle. Actuation of the tipping mechanism is achieved by employing power screw with worm and wheel connection. The cycling paddle was used in operating the system in such a way that to lift-tip the system, reverse paddling is used whilst forward paddling is used to lower the system as well as move the tricycle in the forward direction. The power transmission is by chain connection from the paddle sprocket through the worm to the power screw, Figure 2.3. An innovative clutch-coupling device was developed and used to enable isolation of the lift-tipping mechanism from riding of the tricycle. The clutch-coupling device helps to engage the lift-tipping system for actuation when needed so that the same paddle is used, in riding the tricycle. In the case of the motorized tricycle, the same engine is used to actuate the lift-tipping mechanism.

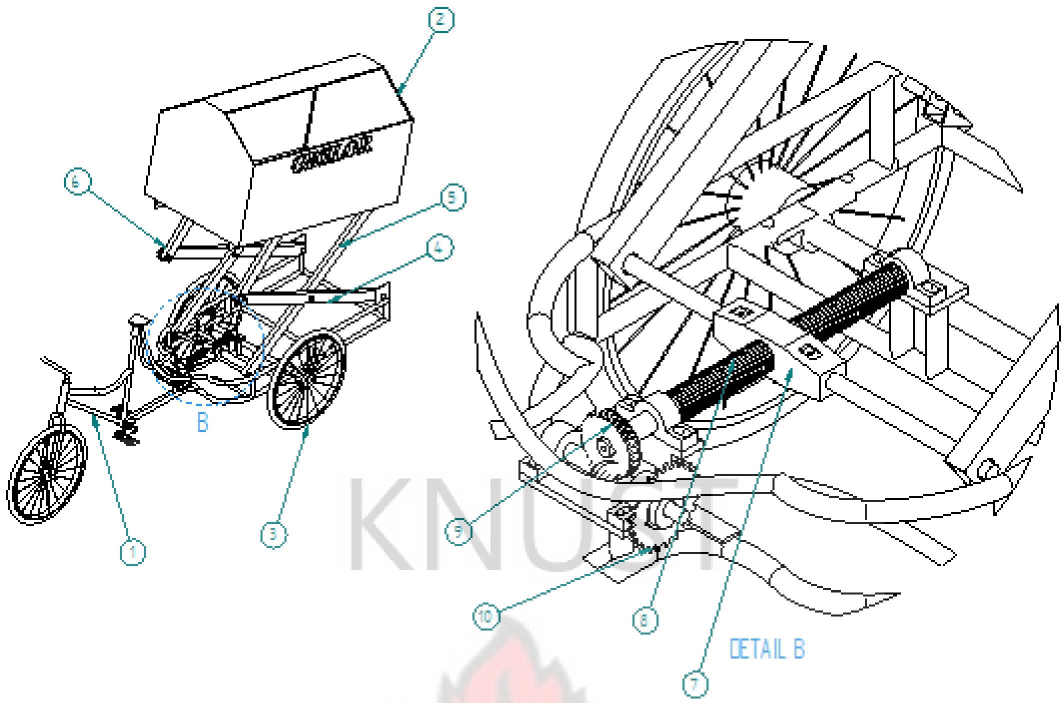


Figure 2-3: Tricycle drawing
1. Tricycle frame; 2. Bin; 3. Wheel; 4. Lifting support bar; 5. Lifting bar; 6. Tipping bar; 7. Power nut; 8. Power screw; 9. Worm wheel; 10. Sprocket
Fiagbe et al (2011)

2.3 OPTIMIZATION TECHNIQUES

Optimization is described as any process which seeks to find the best possible solution to a problem. Mechanism optimization is the repeated analysis of randomly determined mechanisms to find the best design (Scardina, 1996). The best solution will effectively satisfy the design constraints and produce the minimum value for the objective function. When multiple or conflicting constraints are present in the problem, the process of finding the best solution becomes more difficult. A weighting procedure may be used in considering conflicting design constraints. The relative importance of each constraint may be specified in the objective function. Depending on the weighting process, the designer can tailor the final solutions. Optimization yields mathematically correct solutions, but these solutions may possess mechanical defects, thus the designer sorts through all the solutions to find the best possible solution. For most design optimization problems, there are five steps formulation procedure, (Jasbir, 2004). These are:

Step 1: Project/problem statement.

This is a descriptive statement which states the overall objectives of the problem and the requirements to be met.

Step 2: Data and information collection.

This involves gathering information on material properties, performance requirements, resource limits, cost of raw materials, and other relevant information. In addition, analysis procedures and analysis tools are identified at this stage.

Step 3: Identification/definition of design variables.

The next step in the formulation process is to identify a set of variables that describe the system, called design variables (optimization variables). They are regarded as free because any value can be assigned to them. Different values for the variables produce different designs. The design variables should be independent of each other as far as possible. If they are dependent, then their values cannot be specified independently.

The number of independent design variables specifies the design degrees of freedom for the problem. For some problems, different sets of variables can be identified to describe the same system. The problem formulation will depend on the selected set. Once the design variables are assigned numerical values, the system is said to be designed.

Step 4: Identification of a criterion to be optimized.

There can be many feasible designs for a system, and some may be better than others. To compare different designs, there must be a criterion. The criterion must be a scalar function whose numerical value can be obtained once a design is specified. Such a criterion is usually called an objective function for the optimum design problem, which needs to be maximized or minimized depending on problem requirements. Some examples of objective functions include: cost (to be minimized), profit (to be maximized), weight (to be minimized), energy expenditure (to be minimized), ride quality of a vehicle (to be maximized), and so on. In this research, the objective function (weight) is to be minimized.

Step 5: Identification of constraints.

Constraints are restrictions placed on a design. The final step in the formulation process is to identify all constraints and develop expressions for them. Most realistic systems must be designed and fabricated within given resources and performance requirements. For example, structural members should not fail under normal operating loads. Resonance frequencies of a structure must be different from the operating frequency of the machine it supports; otherwise, resonance can occur causing catastrophic failure. Members must fit into available amounts of space. All these and other constraints must depend on the design variables, since only then do their values change with different trial designs; i.e., a meaningful constraint must be a function of at least one design variable.

2.4 OVERVIEW OF OPTIMIZATION METHODS/ TECHNIQUES

There are several methods/ techniques of optimization. These include calculus, Linear Programming, Dynamic Programming, Geometric Programming, Search Method and Finite Element Based Optimization.

2.4.1 Optimization using Calculus

Classical methods of optimization are based on calculus which specifically determines the optimum value of a function as indicated by the derivatives. In order to optimize using calculus, the function must be differentiable and any constraint must be equality constraint. Three steps are involved in the optimization process using calculus.

a) Definition of stationary points

b) Checking for the necessary and sufficient conditions for the relative maximum/ minimum of a function

c) Definition of global optimum in comparison to the relative or local optimum

For a continuous and differentiable function, f(x), a stationary point x* is a point at which the derivative vanishes, i.e.

$$\frac{df}{dx} = 0 \text{ at } x = x^*. \qquad 2.1$$

Where x* belongs to its domain of definition. A stationary point may be a minimum, maximum or an inflexion point.

Necessary condition for a single variable function f(x) defined for $x \in [a,b]$ which has a relative maximum at x = x*, x* $\in [a,b]$ if the derivative $\frac{df(x)}{dx}$ exists as a finite number at x = x* then

$$\frac{df(x^*)}{dx} = 0. \qquad 2.2$$

The above theorem holds good for relative minimum as well and it only considers a domain where the function is continuous and derivative.

For the same function, $f(x)$ let

$$f'(x^*) = \frac{df(x^*)}{dx} = f(n-1)(x^*) = 0, \qquad 2.3$$

but $f(n)(x^*) \neq 0$, then it can be said that $f(x^*)$ is

(a) a minimum value of $f(x)$ if $f(n)(x^*) > 0$ and n is even

(b) a maximum value of $f(x)$ if $f(n)(x^*) < 0$ and n is even

(c) neither a maximum or a minimum if n is odd.

This is sufficient condition for optimality of a single variable function.

In case of multivariable functions a necessary condition for a stationary point of the function $f(\mathbf{X})$ is that each partial derivative is equal to zero. In other words, each element of the gradient vector $\Delta_x f$ defined in equation 2.4 must be equal to zero.

$$\Delta_x f = \begin{bmatrix} \frac{\partial f}{\partial x_1}\left(X^*\right) \\ \frac{\partial f}{\partial x_2}\left(X^*\right) \\ \vdots \\ \vdots \\ \frac{\partial f}{\partial dx}\left(X^*\right) \end{bmatrix} = 0 \qquad 2.4$$

Sufficient condition

For a stationary point $\mathbf{X}^*$ to be an extreme point, the matrix of second partial derivatives (Hessian matrix) of $f(\mathbf{X})$ evaluated at $\mathbf{X}^*$ must be positive definite when $\mathbf{X}^*$ is a point of relative minimum, and negative definite when $\mathbf{X}^*$ is a relative maximum point.

2.4.2 Linear Programming

Linear programming is an optimization procedure applicable where both the objective function and the constraints can be expressed as linear combinations of the variables. The constraints equations may be equalities or inequalities. The most popular method used for the solution of *Linear Programming Problems* (LPP) is the ***simplex method***. The Simplex method was developed by Dantzig in 1947. (Some commercial codes now use an alternative method, called the Interior Point method, developed by Karmarkar in 1984). The general procedure of simplex method is as follows:

1. General form of given LPP is transformed to its *canonical form*.
2. A basic feasible solution of the LPP is found from the canonical form (there should exist at least one).
3. This initial solution is moved to an adjacent basic feasible solution which is closest to the optimal solution among all other adjacent basic feasible solutions.
4. The procedure is repeated until the optimum solution is achieved.

2.4.3 Dynamic Programming

Dynamic programming is a method for solving complex problems by breaking them down into simpler subproblems. It is applicable to problems exhibiting the properties of overlapping subproblems which are only slightly smaller and optimal substructure. When applicable, the method takes far less time than naive methods. The key idea behind dynamic programming is quite simple. In general, to solve a given problem, we need to solve different parts of the problem (subproblems), and then combine the solutions of the subproblems to reach an overall solution. The dynamic programming is a paradigm of algorithm design in which an optimization problem is solved by a combination of caching subproblem solutions and appealing to the "principle of optimality." (Rashid, 2010)

There are three basic elements that characterize a dynamic programming algorithm:

(i) **Substructure**

Decompose the given problem into smaller (and hopefully simpler) subproblems. Express the solution of the original problem in terms of solutions for smaller problems. It is not usually sufficient to consider one decomposition, but many different ones.

(ii) **Table-Structure**

After solving the subproblems, store the answers (results) to the subproblems in a table. This is done because (typically) subproblem solutions are reused many times.

(iii) **Bottom-up Computation**

Combine solutions of smaller subproblems to solve larger subproblems, and eventually arrive at a solution to the complete problem. The idea of bottom-up computation is as follow:

Bottom-up means

i. Start with the smallest subproblems.
ii. Combining theirs solutions obtain the solutions to subproblems of increasing size.
iii. Until arrive at the solution of the original problem.

There are two important elements that a problem must have in order for dynamic programming technique to be applicable

a) **Optimal Substructure**

A problem is said to have **optimal substructure** if an optimal solution can be constructed efficiently from optimal solutions to its subproblems

b) **Polynomially many (Overlapping) Subproblems**

An important aspect to the efficiency of dynamic programming is that the total number of distinct sub-problems to be solved should be at most a polynomial number. Overlapping subproblems occur when recursive algorithm revisits the same problem over and over.

2.4.4 Geometric Programming

Geometric programming is a relatively new method of solving a class of nonlinear programming problems. It is used to minimize functions that are in the form of polynomials subject to constraints of the same type. It differs from other optimization techniques in the emphasis it places on the relative magnitudes of the terms of the objective function rather than the variables.

Instead of finding optimal values of the design variables first, geometric programming first finds the optimal value of the objective function. This feature is especially advantageous in situations where the optimal value of the objective function may be all that is of interest. In such cases, calculation of the optimum design vectors can be omitted. Another advantage of geometric programming is that it often reduces a complicated optimization problem to one involving a set of simultaneous linear algebraic equations. The major disadvantage of the method is that it requires the objective function and the constraints in the form of polynomials. (Rao 2009)

Degree of Difficulty is used to determine whether geometric programming is suitable for a particular problem or not. The quantity N-n-1 is termed a *degree of difficulty* in geometric programming. In the case of a constrained geometric programming problem, N denotes the total number of terms in all the polynomials and n represents the number of design variables (Rao 2009). If N-n-1 = 0, the problem is said to have a zero degree of difficulty and the solution is directly obtained by solving the system of constraints. For a degree of difficulty greater than zero, geometric programming will work, but the method involves the solution of nonlinear equations, which will probably be more time consuming than in some other methods. It is to be noted that geometric programming is not applicable to problems with negative degree of difficulty

2.4.5 Search Method

In optimization by means of search method, values of the objective function are determined and conclusions are drawn from the values of the function of various combinations of independent variable. There are a number of variations of the search method.

Fibonacci Method

Fibonacci method can be used to find the minimum of a function of one variable even if the function is not continuous. This method, like many other elimination methods, has the following limitations (Rao, 2009):

1. The initial interval of uncertainty, in which the optimum lies, has to be known.
2. The function being optimized has to be unimodal in the initial interval of uncertainty.
3. The exact optimum cannot be located in this method. Only an interval known as the *final interval of uncertainty* will be known. The final interval of uncertainty can be made as small as desired by using more computations.
4. The number of function evaluations to be used in the search or the resolution required has to be specified beforehand.

Exhaustive Search

The exhaustive search method can be used to solve problems where the interval in which the optimum is known to lie is finite. Let x_s and x_f denote, respectively, the starting and final points of the interval of uncertainty. The *exhaustive search method* consists of evaluating the objective function at a predetermined number of equally spaced points in the interval (x_s, x_f), and reducing the interval of uncertainty using the assumption of unimodality. Suppose that a function is defined on the interval (x_s, x_f) and let it be evaluated at eight equally spaced interior points x_1 to x_8. Assuming that the function values appear as shown in Figure 2.5, the minimum point must lie, according to the assumption of unimodality, between points x_5 and x_7. Thus the interval (x_5, x_7) can be considered as the final interval of uncertainty.

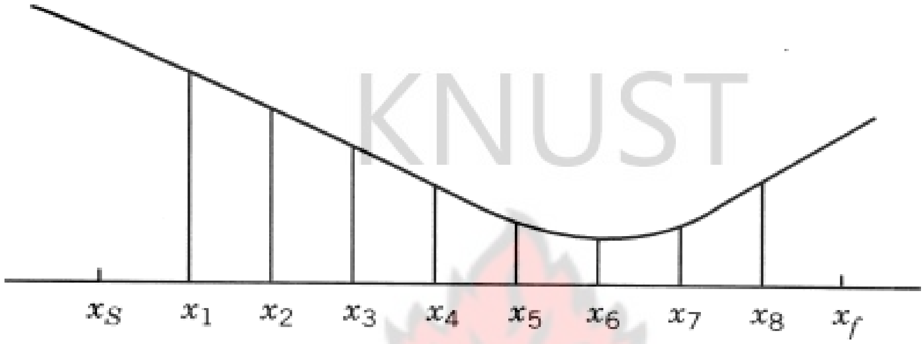


Figure 2-4: Exhaustive search

Dichotomous Search

The exhaustive search method is a simultaneous search method in which all the experiments are conducted before any judgment is made regarding the location of the optimum point. The *dichotomous search method*, as well as the Fibonacci is sequential search methods in which the result of any experiment influences the location of the subsequent experiment. In the dichotomous search, two experiments are placed as close as possible at the center of the interval of uncertainty. Based on the relative values of the objective function at the two points, almost half of the interval of uncertainty is eliminated.

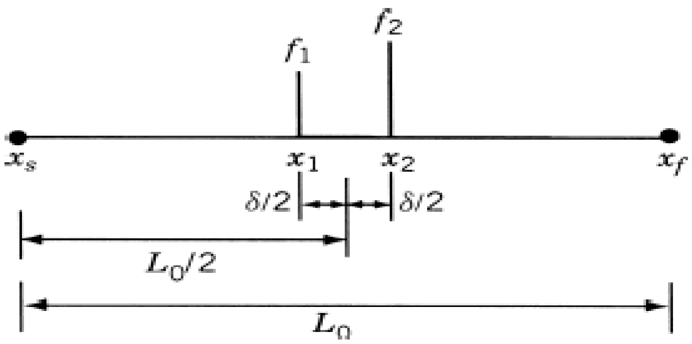


Figure 2-5: Dichotomous Search

From figure 2.6, x_1 and x_2 are positions of two experiments and δ is a small positive number.

2.4.6 Finite Element Based Optimization

Although the mathematical techniques described above can be used to solve most engineering optimization problems, the use of engineering judgment and approximations help in reducing the computational effort involved. One of such types of approximation techniques that can speed up the analysis time without introducing too much error is finite element analysis-based optimization. Finite element analysis optimization involves simulations of various designs so as to reduce the mass of the existing designs. The exercise involves validating the design changes made in the Stress Analysis environment. The work flow is repeated until the mass of the design is optimized against the design criteria.

FEA Software List

There are a lot of finite element analysis softwares. Notable among them are; Abaqus, ALGOR, ANSYS, FEMAP, HyperMesh, Nastran, NEiNastran, Solidworks Simulation and STRAND7. The high level of analysis functionality exposed by Femap and supported by the user interface provides great value to full-time analysts and occasional-use multi-disciplinary engineers alike and cannot be matched by other solution offerings in the industry. This combined with the level of advanced analysis solutions offered by NX Nastran makes the Femap with NX Nastran combination able to solve more complex engineering problems in a straightforward manner.

Nastran is traditionally a very capable solver for dynamic response analysis, and additional dynamics solutions that are supported by Femap. Femap also offers comprehensive analysis functionality that is easy to use and quick to learn. This brings about a reduced training overhead and allows engineers to maintain maximum productivity even when Femap is used on an occasional basis. This among other reasons makes NX Nastran a better choice to other FEA softwares.

NASA structural analysis is a finite element analysis (FEA) program which was originally developed for NASA in the late 1960s under United States government funding for the Aerospace industry. NX Nastran is a general purpose Finite Element Analysis solver capable of simulating a broad range of engineering problems in many different industries. The NX Nastran software is composed of a large number of building blocks called modules. Each module is a collection of subroutines which is designed to perform a specific task, such as processing model geometry, assembling matrices, applying constraints, solving matrix problems, and calculating output quantities.

NX Nastran analysis capabilities available for Femap

- **Basic** includes linear static, normal modes, buckling, model checkout, spot weld, steady-state and transient heat transfer, basic non-linear, design sensitivity and unlimited problem size capabilities.
- **Dynamic response** enables product performance to be evaluated in both the time and frequency domains.
- **Superelements** enables very large and complex finite element models to be solved as smaller substructures called superelements.
- **DMAP** is a programming capability that allows customers to expand NX Nastran and support custom applications.
- **Optimisation** automates the iterative process of improving product performance.
- **Rotor dynamics** predicts the dynamic response of rotating systems such as shafts, turbines, and propellers to determine critical shaft speeds.

- Advanced non-linear with large deformation; non-linear materials; time-dependent loads; deformable and rigid contact. Explicit non-linear time integration for impact analysis.
- **Structural analysis toolkit** saves post-processing time through organization of results data and calculation of additional results quantities

KEY TERMS

Assembly

Two or more components (parts or subassemblies) considered as a single model. An assembly typically includes multiple components positioned absolutely and relatively (as required) with constraints that define both size and position. Assembly components may include features defined in place in the assembly. Mass and material properties may be inherited from individual part files.

Stress analysis

An analysis showing that the model is statically and dynamically stable and free from divergence on application of external loads and frequencies.

In this optimization, stress analysis is used to ensure that the material and geometry of the components can handle the loads without deforming and failing.

Simulation

The term Simulation has grown to be an equivalent term to analysis. Stress Analysis is used to analyze the material at the point of maximum load on the components.

Von Mises Stress

Three-dimensional stresses and strains build up in many directions. A common way to express these multidirectional stresses is to summarize them into an Equivalent stress, also known as the von-Mises stress. A three-dimensional solid has six stress components. Sometimes a uniaxial stress test finds material properties experimentally. In that case, the combination of the six stress components to a single equivalent stress relates the real stress system.

2.5 JUSTIFICATION FOR CHOICE METHOD

In order to optimize using calculus, the objective function must be differentiable and any constraint must be an equality constraint. In linear programming both the objective function and the constraints need to be expressed as linear combinations of the variables. In dynamic programming, different parts of the problem (subproblems) are first solved, and then combine the solutions of the subproblems to reach an overall solution. Geometric programming is used to minimize functions that are in the form of polynomials subject to constraints of the same type. In optimization by means of search method, values of the objective function are determined and conclusions are drawn from the values of the function of various combinations of independent variable.

The mathematical techniques (calculus, linear programming, dynamic programming, geometric programing and search methods) described above can be used to solve the optimization problem. However, the use of engineering judgment and approximations help in reducing the computational effort. An approximation technique that can speed up the analysis time without introducing much error is finite element analysis-based optimization. Femap also offers comprehensive analysis functionality that is easy to use. This among other reasons makes finite element analysis-based optimization a better choice to other optimization techniques. This method much appropriate to the optimization of tricycle with tipping mechanism based on above mentioned advantages; precision, comprehensive analysis base and time requirement for completion of the thesis.

CHAPTER 3

DESIGN PROBLEM FORMULATION

3.1 INTRODUCTION

This chapter presents the design problem formulations of the bin, Linkage mechanism, power screw and power screw nut. The first step develops a design objective or problem statement of each component. The problems stated are then translated into a mathematical statement for optimization using the five-step process. Finally, Nastran Simulation and Optimization is performed.

3.2 THE BIN

Project/Problem Statement

The existing bin is as shown in figure 3.1. The objective is to optimize the bin for the tricycle to carry solid waste. The design objective is to minimize the total mass of bin. It must satisfy stress, size and manufacturing constraints.

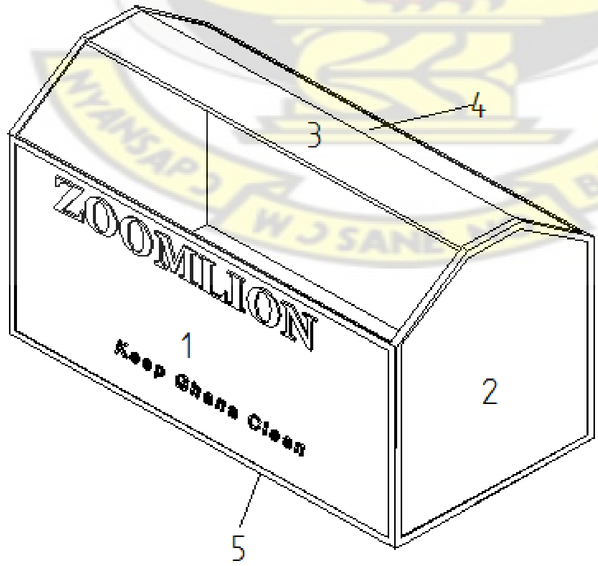


Figure 3-1: Schematic of the Bin: 1. Side; 2. End; 3. Top. ; 4. Cover; 5. Bottom;

Data and Information Gathering

The dimension of the bin is as shown on figure 3.2. The overall weight of the bin is 50 Kg. The thickness of the metal plate used is 1.25 mm and it is reinforced at the edges with $30 \times 30 \times 1.5$ mm angle bar.

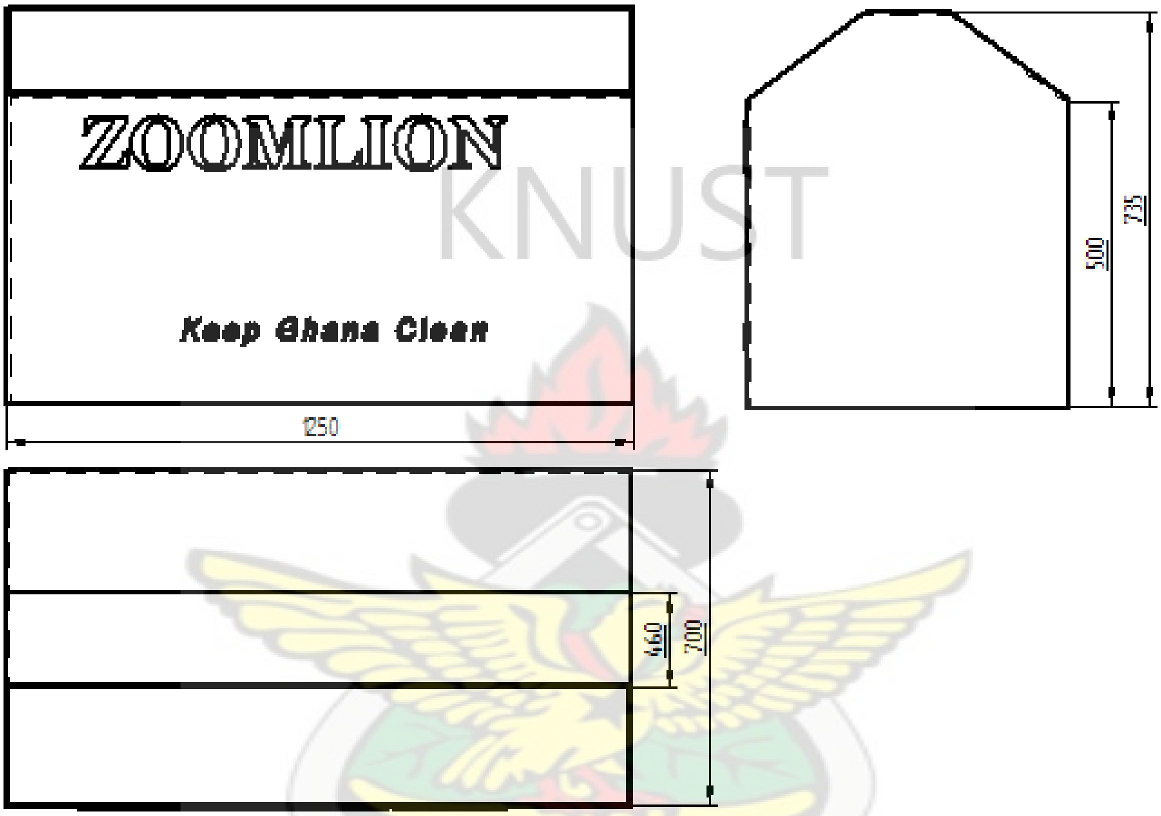


Figure 3-2: Orthographic views of bin

Definition of Design Variables

The thickness (t) of the material was chosen as the design variable by merit. The existing design is made of 1.25 mm thick steel plates that are welded together and braced with an angle iron.

Identification of a criterion to be optimized

The bins' mass is identified as the objective function in the problem statement. Since it is to be minimized, it is called the cost function for the problem. An expression for the mass is determined by the cross-sectional area of the plates and associated design variables.

Mass of bin is calculated by multiplying the surface area by the density and thickness of the metal plates used. Thus;

$$Mass\ of\ Bin = 5.1\rho t \ \text{m}^2 \qquad 3.1$$

Where $\rho = 7850$ kg/m^3 for mild steel.

Identification of constraints

The base of the bin is bolted to the bin support arms (GD). Thus the bin base has no degree of freedom. It makes sense to apply fixed constraints to the nodes.

3.3 LINKAGE MECHANISM

Project/Problem Statement

The objective is to optimize the weight of the linkage mechanism shown in Figure 3.3 to support a load W without structural failure. The design objective is to minimize the total mass of the mechanism. It must satisfy stress, size and manufacturing constraints.

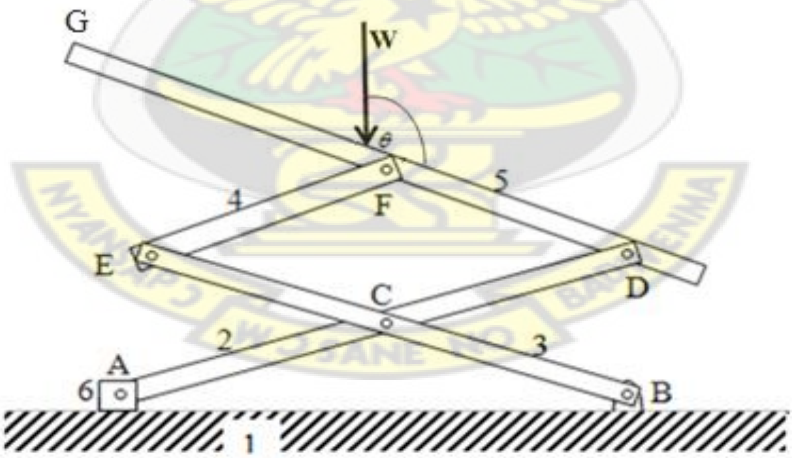


Figure 3-3: Kinematic illustration of the linkage

Data and Information Gathering

The maximum load that the tricycle is recommended to carry is 50 kg (490.5 N). When the mechanism is in operation, the load has a tendency to be between the angles 0^0-38^0. The free body diagram is shown in Figure 3.4. Its components are: AD is the lifting bar; BE is the lifting support bar; EF is tipping arm and GD is the bin support and A is a slider.

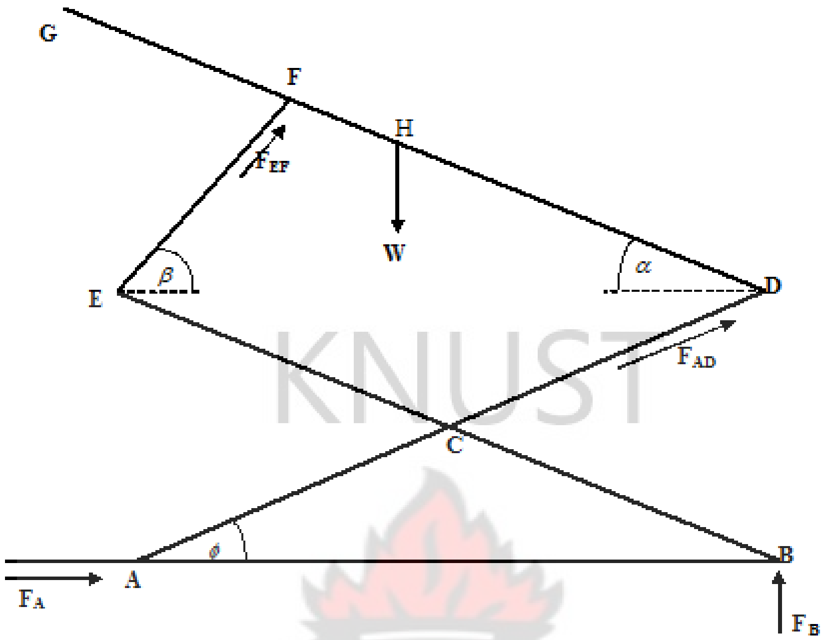


Figure 3-4: FBD of tipping Mechanism

Considering member GD in Figure 3-4

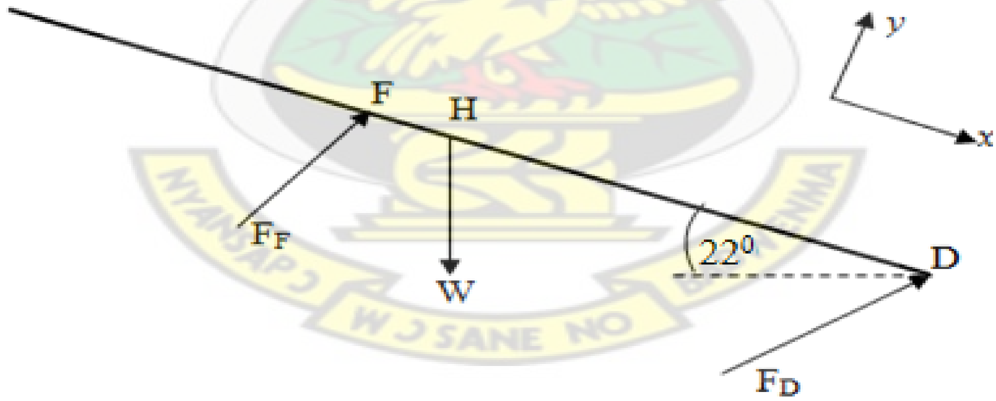


Figure 3-5: FBD of member GD

Using the dimensions of the existing system, the lifting bar is 1.20 m and tipping arm is 0.35 m.

FD| = 0.85 m, from trigonometry, $\beta = 66^0$ (see figure 3.4) and F_F makes an angle of 92^0 with the member GD.

|GD| = 1.2 m

W = Maximum load of solid waste in bin + weight of member GD + weight of bin

$= (50 \times 9.81) + (7850 \times 1.96 \times 10^{-4} \times 1.2 \times 9.81) + (50 \times 9.81)$

= 999.1 N

Taking moment about D

$$(W \times \cos 66^0 \times 0.6) - (F_F \sin 92^0 \times 0.85) = 0$$

F_F = 287.02 N

Summing forces in the Y direction

$$F_{Dy} - (W \times \cos 22^0) + (F_F \sin 92^0) = 0$$

$$F_{Dy} = 639.5N$$

Summing forces in the X direction

$$F_{Dx} + (W \times \sin 22^0) - (F_F \cos 92^0) = 0$$

$$F_{Dx} = 384.3N$$

$$F_D = \sqrt{639.5^2 + 384.3^2} = 746.1N$$

Consider member EF in figure 3-4

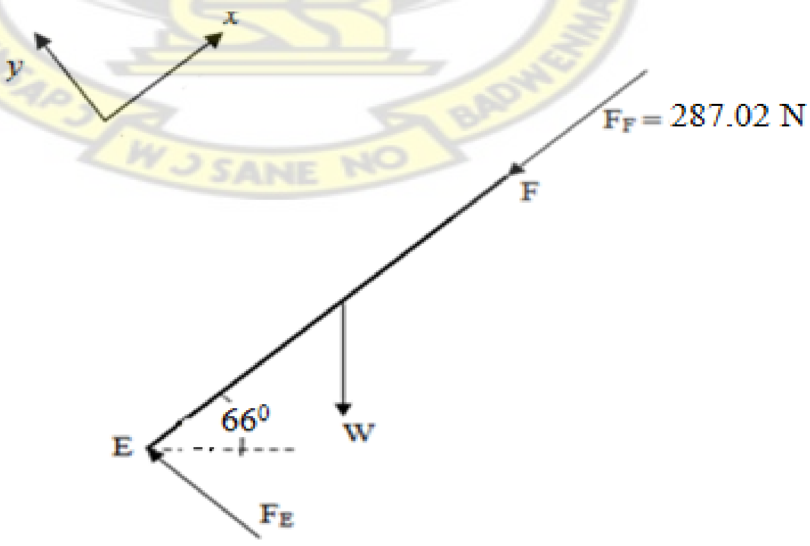


Figure 3-6: FBD of member EF

Weight of member EF, $w = 7850 \times 3.44 \times 10^{-4} \times 0.35 \times 9.81$

= 9.3 N

Summing forces in the Y direction

$F_{Ey} = (w \times \cos 66^0) = 9.3N$

Summing forces in the X direction

$F_{Ex} - (W \times \cos 66^0) + 641.7 = 0$

$F_{Ex} = 637.9N$

Hence

$F_E = 637.9N$

Consider member AD in figure 3-4

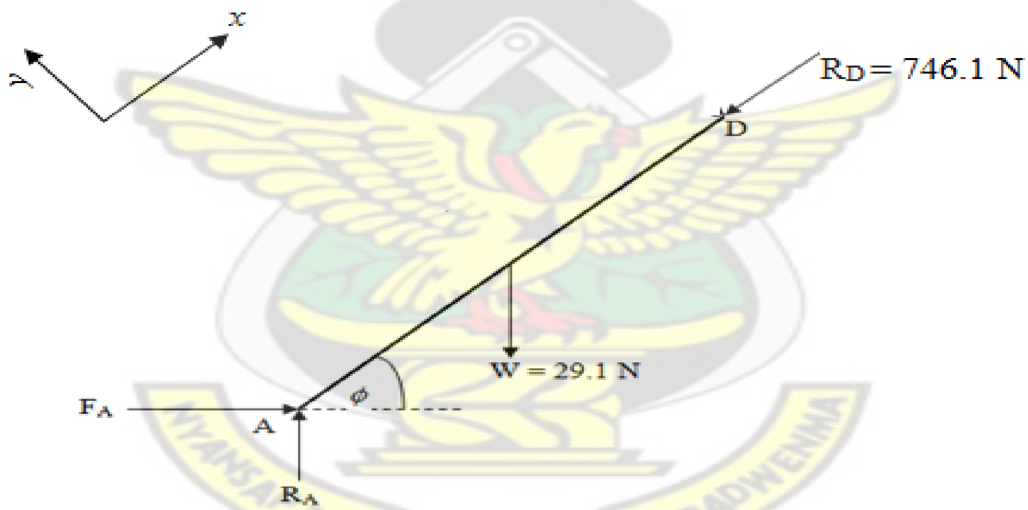


Figure 3-7: FBD of member AD

At full tipping position, lifting height of 0.75 m is achieved

|BD| = 0.75

Hence $\phi = \sin^{-1}\left(\frac{0.75}{1.20}\right) = 38.7^0$

Summing forces in the Y direction

$-F_A \cos 51.3^0 + R_A \cos 38.7 - w\cos 38.7^0 = 0$

$0.780\ R_A - 0.625\ F_A = 22.7$

Summing forces in the X direction

$0.625\ R_A + 0.780\ F_A = 392.7$

Solving for R_A and F_A gives

$R_A = 264.4N$

$F_A = 292.4N$

Consider member EB in figure 3-4

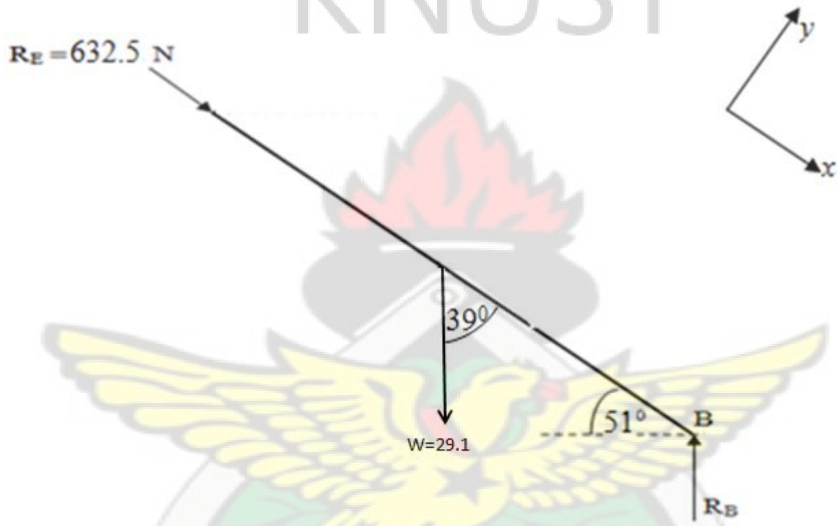


Figure 3-8: FBD of member BE

Summing forces in the Y direction

$R_B \sin 39^0 - w\cos 39^0 = 0$

$R_B = 36.2N$

Definition of Design Variables

The design variables depend on the size of the bars. Three different sizes of rectangular pipes on the market were considered. They are

$40\text{mm} \times 20\text{mm} \times 1.5$ mm – option 1

$50\text{mm} \times 25\text{mm} \times 1.5$ mm – option 2

$60\text{mm} \times 30\text{mm} \times 1.5$ mm – option 3 (Existing design)

Identification of a criterion to be optimized

The mechanism's mass is identified as the objective function in the problem statement. An expression for the mass is determined by the cross-sectional shape of the bars and associated design variables.

Mass = Density $\times$ volume

$= \rho \times A \times L$ (for each component of the mechanism)

Note $\rho = 7850$ kg/m^3

Identification of constraints

It is important to include all constraints in the problem formulation because the final solution depends on them. As installed on the tricycle, the lifting support bar pivots about a bolt through to the tricycle chassis. Furthermore, the lifting bar is also supported on a pin that is held by the power screw nut. The pin is held in roller bearing which provides motion along the rails. Thus the only remaining degree of freedom at the bearing is a rotation about the pin. It makes sense to apply known loads and pin constraints at these nodes. The system therefore has a one degree-of-freedom that is in the direction along AB (refer to figure 3.4).

3.4 POWER SCREW

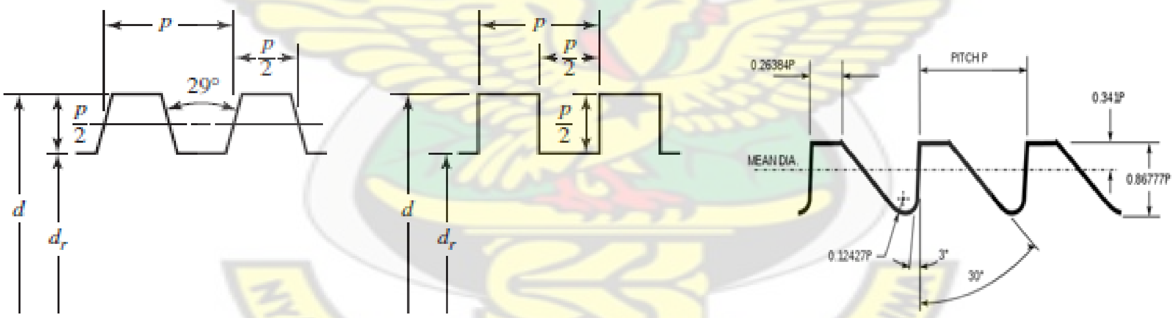
The power screws (also known as ***translation screws***) are used to convert rotary motion into translatory motion. Power screws are used in vices, lead screw of lathe, screw jack, testing machines, presses, etc. In most of the power screws, the nut has axial motion against the resisting axial force while the screw rotates in its bearings. In some screws, the screw rotates and moves axially against the resisting force while the nut is stationary and in others the nut rotates while the screw moves axially with no rotation.

Types of Power Screw

Power screws are classified by the geometry of their thread. V-threads are less suitable for Power screws than others such as Acme because they have more friction between the threads. Their threads are designed to induce this friction to keep the fastener from loosening. Power screws, on the other hand, are designed to minimize friction. Therefore, in most commercial and industrial use, V-threads are avoided for Power screw use. Nevertheless, V-threads are sometimes successfully used as Power screws, for example on micro-lathes and micro-mills.

Square thread Type

Square threads are named after their square geometry. They are the most efficient, having the least friction, so they are often used for screws that carry high power. But they are also the most difficult to machine, and are thus the most expensive.



A B C

Figure 3-9: Types of screw threads
A) Acme thread B) Square thread C) Buttress thread

Acme thread Type

Acme threads have a 29° thread angle, which is easier to machine than square threads. They are not as efficient as square threads, due to the increased friction induced by the thread angle.

Buttress thread Type

Buttress threads are of a triangular shape. These are used where the load force on the screw is only applied in one direction. They are as efficient as square threads in these applications, but are easier to manufacture.

Problem/ Project Statement

The purpose of this project is to design a minimum mass square thread power screw to carry loads along the rails, F_A as shown in Figure 3-10.

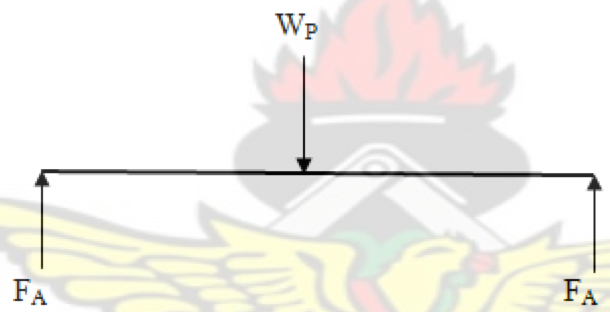


Figure 3-10: Load on Power Screw

Data and Information

From figure 3.10,

Load on Power Screw $=W_P$

Assumption: Power Screw is situated just at the middle of the cross link shown in figure 3-10.

$$W_P = 2F_A$$

$$= 2 \times 292.4$$

$$= 584.8N$$

With a factor of safety of 2, the design load is calculated as

Design Load = 1.2 W_P

$= 1169.6$ N.

Let root diameter = D_r

Pitch diameter = D_p (mean diameter; d_m)

The tensile area $A_t = \frac{\pi}{4}\left[\frac{D_r + D_p}{2}\right]^2$

Lead angle, $\lambda = \tan^{-1}\left[\frac{p}{\pi D_P}\right]$

Consider figure 3-11

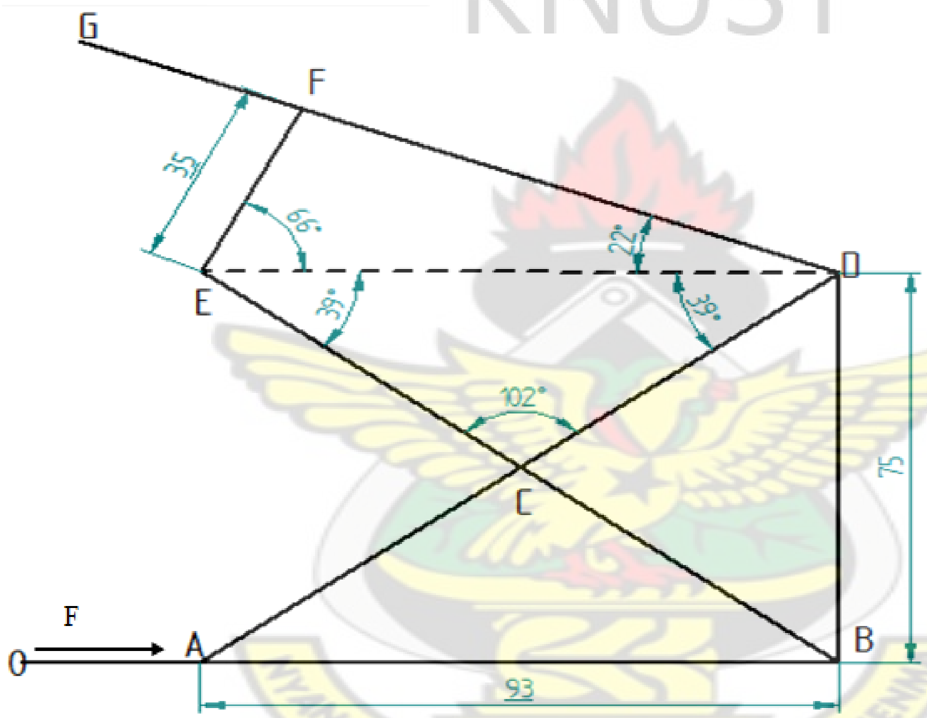


Figure 3-11: Schematics of linkage bars

BE = AD, AC =BC

When the bucket is in the full tipping position at tipping angle of 22^0

|OA| = 120 – 93 = 27 cm

Hence the minimum threaded length of the power screw is 270 mm

Assume a threaded length of 400 mm.

Expected normal tensile stress $\sigma_e = \frac{F}{A_t} = \frac{F}{\frac{\pi}{4}\left[\frac{D_r + D_p}{2}\right]^2}$

The torque required to move a load up the thread is given as

$$T_u = \frac{FD_p}{2}\left[\frac{L + \pi f D_p}{\pi D_p - fL}\right]$$

Assume for well lubricated steel screw acting in steel nut $f = 0.15$

The torque required to lower the load

$$T_d = \frac{FD_p}{2}\left[\frac{\pi f D_p - L}{\pi D_p + fL}\right]$$

Efficiency = $\frac{FL}{2\pi T_u}$

Linear velocity, V = 0.0033 m/s

$$\omega = \frac{2 \times 0.0033}{D_p}$$

$$P = T\omega$$

Definition of Design Variables

The design variables depend on the size of the power screw. Three different pitch diameters were selected to include that of the existing one. They are

36 mm – option 1 (Existing design)

25 mm – option 2

20 mm – option 3

Identification of Criterion to Be Optimized

The criterion to be optimized is the mass of the power screw. An expression for the mass is determined as follows.

Mathematical Formulation:-

Objective Function = f(x) = To minimize weight

Primary Design Equation (PDE):- (Weight of screw is the criterion of optimization)

Let, ρ = Mass Density = 7850 kg/m^3

A = Area in mm^2

L = Length in mm

Mass of screw = mass x acceleration due to gravity

= density x volume

= ρ x A x L

Objective Function = f(x) = minimize mass

= ρ x A x L

= ρ x (π/4) x dm^2 x L

Identification of Constraints

From the existing design, the power screw is supported in two ball bearings. One end of the power screw is fixed to the worm wheel which it rotates with. Two constraints were applied thus to the power screw model; the fixed constraints at the input to the screw and a cylindrical constraints at the other end.

CHAPTER 4

FINITE ELEMENT BASED OPTIMIZATION

4.1 INTRODUCTION

Finite element analysis optimization involves simulations of various designs so as to reduce the mass of an existing design. The exercise involves validating the design changes made in the Stress Analysis environment. The work flow is repeated until the mass of the design is optimized against the design criteria.

NX Nastran offers high level Femap solutions to complex engineering problems in a straightforward manner. Nastran is traditionally a very capable solver for dynamic response analysis, and additional dynamics solutions that are supported by Femap. Femap also offers comprehensive analysis functionality that is easy to use and quick to learn. This brings about a reduced training overhead and allows engineers to maintain maximum productivity even when Femap is used on an occasional basis. This among other reasons makes NX Nastran a better choice to other FEA softwares.

The components were first modeled using solid part and sheet metal parts tools of solid Edge. To model parts in Solid Edge, the following basic workflow was used in the solid part or sheet metal part environment:

i. Draw a sketch for the first feature.
ii. Add dimensions to the sketch.
iii. Extrude or revolve the sketch into a solid feature.
iv. Add more features.
v. Edit the model dimensions and solid geometry to complete the part.

vi. Create a drawing.

The processes involved in modeling a Power Screw nut are illustrated below

Modeling a Power Screw Nut

The startup screen contains shortcuts to create new files based on common templates, in the section at left solid edge environment labeled Create. The ISO Part shortcut was chosen to create a new synchronous part file.

Step 1: model the base feature

A rectangle was drawn to initiate the construction of the base feature of the model. To draw the rectangle on the XZ principal plane, on the command menu, at the top of the Solid Edge application, choose Home tab>Draw group>Rectangle by 3 Points. Position the cursor over the origin point of the base coordinate system and move the cursor to draw a rectangle as shown in figure 4.1 below. A Width, Height and Angle boxes on the Rectangle command bar was updated to 156 mm 40 mm and 180^0 respectively.

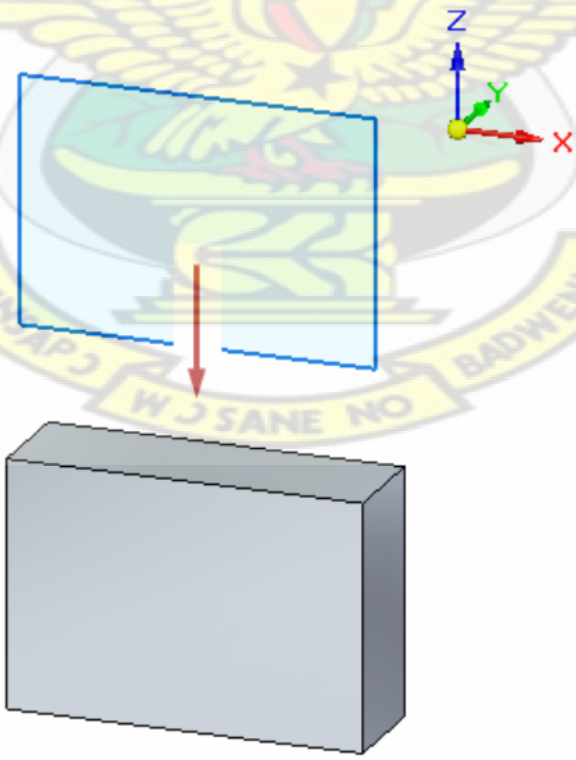


Figure 4-1: Model of Base Feature

Choose Home tab>Solid group>Extrude to model the base feature. 40 mm was entered for the width of the solid.

Choose Home tab>Solid group>cut to model the feature. From the Home tab>Draw group, the feature was draw after selecting the XZ plane as shown in figure 4.2 below. The sketch was closed and the model cut out.

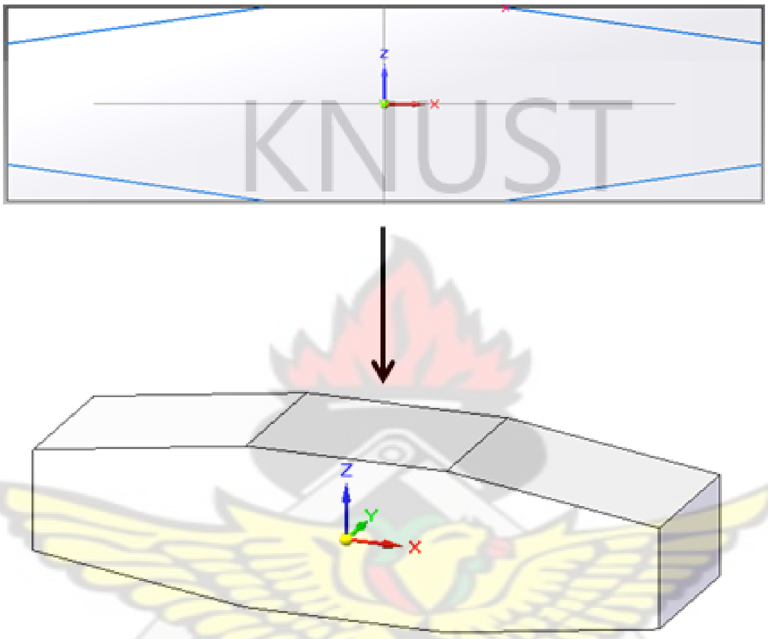


Figure 4-2: Modeling the Power Screw Nut

Creating holes in the model

Choose Home tab>Solid group>Hole to create hole. From the Home tab>Draw group, the circle was draw after selecting the YZ plane. 20 mm was entered as the diameter of the circle. The sketch was closed and the hole created with a depth of 45. The process was repeated to create the rest of the holes as and model is as shown in figure 4.3 below.

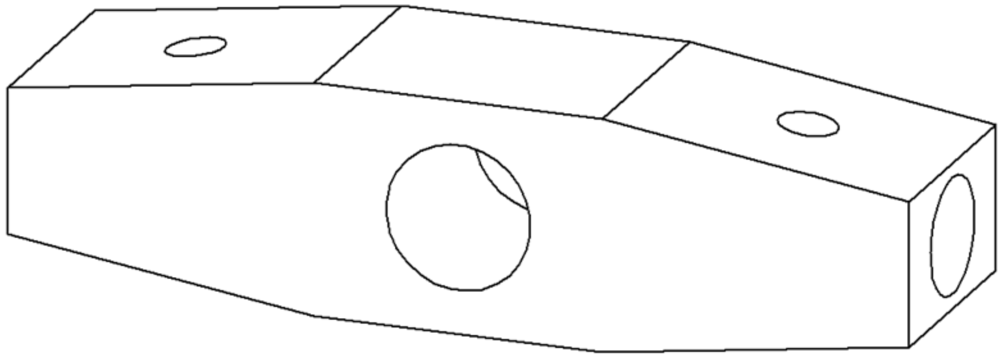


Figure 4-3: Model of Power Screw Nut

They were then imported into the assemble part environment for assembling and/or simulation. The process of simulation is of seven steps:

i. Creating a new study
ii. Geometry specification
iii. Defining the loads
iv. Defining constraints
v. Meshing
vi. Executing the calculation (solve)
vii. Expose/explore results and manipulate them.

Creating a new study

The second group in the ribbon on the solid edge simulation window is the Study Group. The general condition of the study was defined at this step. In the Create Study dialog box, the Study type was set to Linear Static and the Mesh type to Tetrahedral.

A Linear Static analysis is appropriate to calculate displacements, strains, stresses, and reaction forces under the effect of applied loads.

Tetrahedral meshing breaks the part into a number of smaller volumes for analysis. For most models, tetrahedral meshing is appropriate.

Geometry specification

At this step, the solid(s) that will be part of the study were defined.

Definition of the Loads

The following type of load can be applied to a model in solid edge simulation environment;

i. Typical Force
ii. Pressure
iii. Gravity
iv. Centrifugal
v. Temperature
vi. Displacement

Typical load type was selected and applied to the model. The direction of the load was set on a QuickBar and the value of the force set in the Value dialog box, which appeared when the load was applied.

Definition of constraints

The define constraints step allow user to define the degree of liberty the study will have. The following types of constraints were used in solid edge simulation

i. Fixed: it fixes a point or part of a model to a specified location and orientation relative to a reference point.
ii. Pin1: A pin constraint is used to connect the model to the ground. It is not used to connect two parts of the model together, such as a pin through two links of a

mechanism. Pin constraint is applied to cylindrical surfaces to prevent the surfaces from moving or deforming in combinations of radial, axial, or tangential directions.

iii. Cylindrical: this type of constraint has only two degree of freedom ;one translation and one rotation but with the joints being circular

Meshing

A mesh is a system of grid points that overlay the model geometry. The Tetrahedral Mesh dialog box has many options to redefine the mesh according to a region.

Calculation

The calculation process is initiated on the solve tab. The system processes the information that was defined and solves the study.

Results

When the solve is complete, results are displayed on the model, and the Simulation Results ribbon was displayed.

4.2 NASTRAN SIMULATION AND OPTIMIZATION OF BIN

Modeling and Assembling

Siemens Solid Edge ST3 software package, NX Nastran 7 solver was used as the finite element meshing utility for the optimization study. The base of the bin was considered for the analysis since it carries load. The components were modeled using solid part and sheet metal parts and assembled as in figure 4-1.

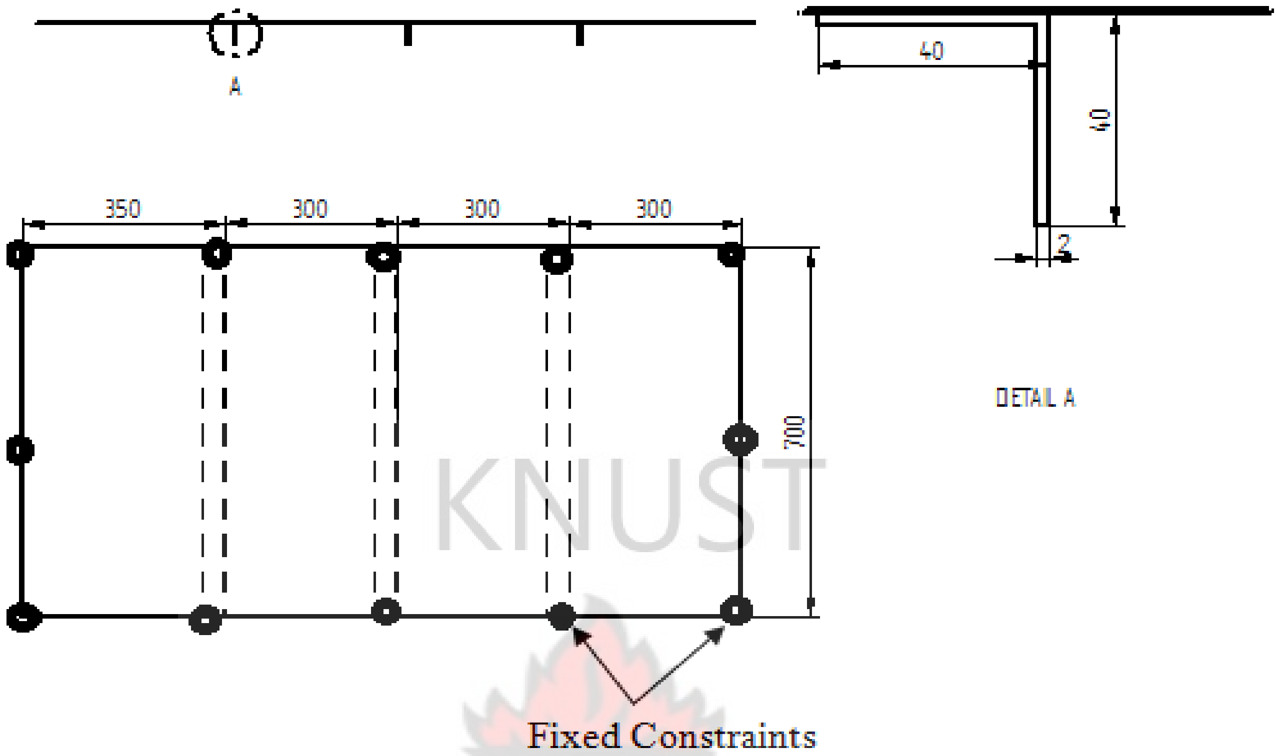


Figure 4-4: Detail drawing of base of bin

Tetra-Meshing

Once the components were modeled and assembled, a tetrahedral mesh was applied to the model to enable Solid Edge Simulation and finite element analysis (FEA).

Connectors

Glued connectors were used at welded joints. A glued interface is useful in areas where large transitions in mesh refinement are required. It is best to create the glued interface at a reasonable distance from a detailed area where precise stresses are required, as there will be some degree of stress discontinuity across an interface which has widely varying mesh densities. A glue interface can be set when defining a contact property. A glued interface is also useful across interfaces where peeling of the interface is known to be minimal or non-existent, as glued interfaces analyses faster than contact interfaces.

Application of Loads and Constraints

The base carried a load of 50 kg (490.5 N). This load was assumed to be uniformly distributed. Fixed constraints were applied to the nodes and edges to produce a zero degree-of-freedom.

Material Definition

The last input parameter required before implementation of the optimization study is the material identification. The material was specified as 1020 HR steel with mass density of 7850 kg/m^3 and ultimate tensile strength of 358.527 MPa and Poisson's ratio 0.33.

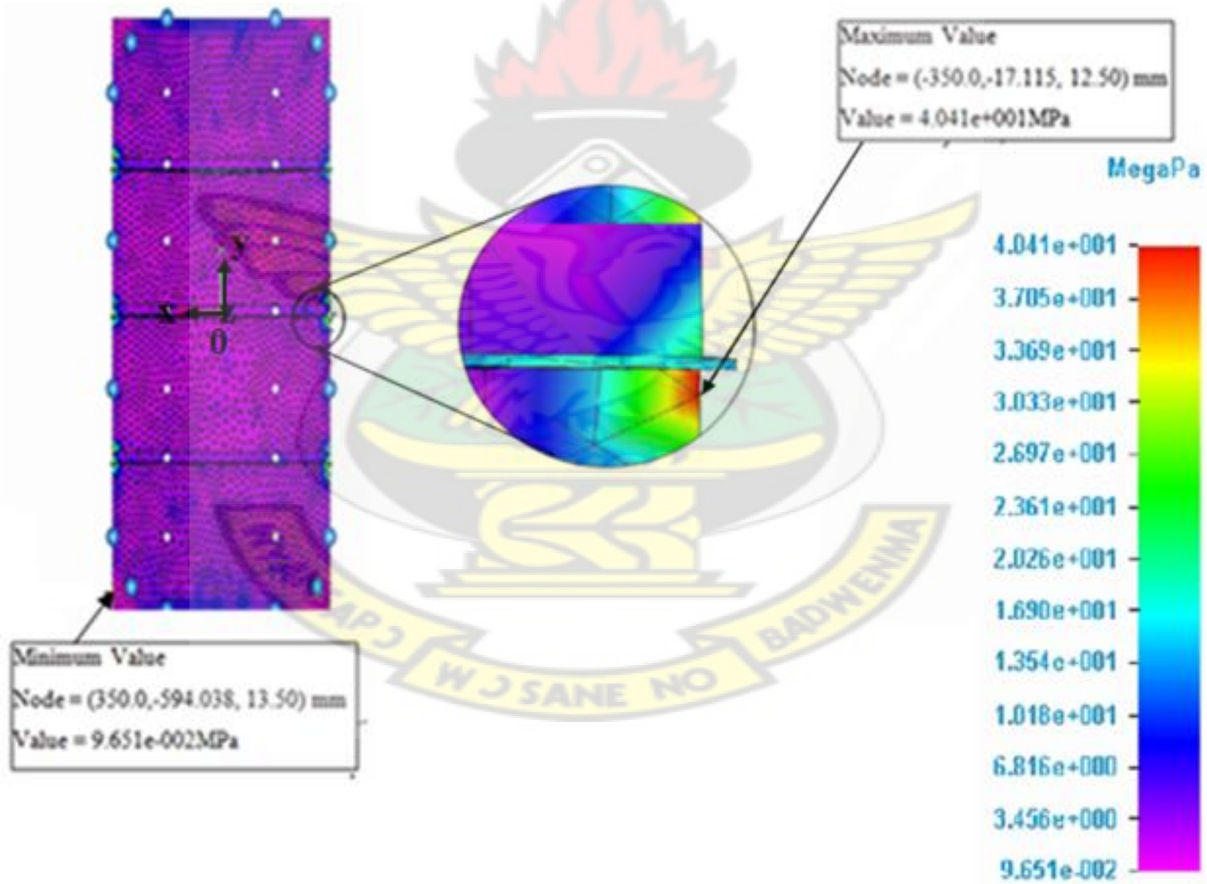


Figure 4-5: Simulation of bin

From Figures 4-2, the minimum and maximum stresses on the model are 0.9651 MPa and 40.4 MPa respectively.

4.3 NASTRAN SIMULATION AND OPTIMIZATION OF POWER SCREW

The Solid Edge software was again used as the finite element meshing utility for the optimization study. The component was modeled using solid part. Tetra-Mesh was applied on the model. Two cylindrical constraints were applied at the bearing supports and fixed constraints at the ends of the power screw as shown on figure 4-5.

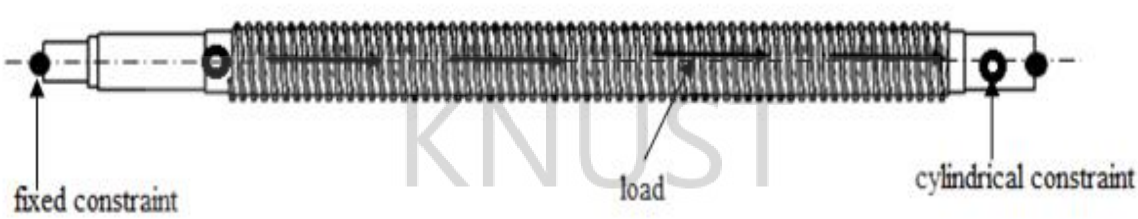


Figure 4-6: Schematic drawing of power screw

Loads

Using the previously determined load cases from the FBD models, the model is updated to include a load of 549 N along the axis of the power screw.

Material Definition

The last input parameter required before implementation of the optimization study is the material identification. In the case of the power screw, the material was specified as 1020 HR steel with mass density of 7850 kg/m^3 and ultimate tensile strength of 359 MPa and Poisson's ratio 0.33.

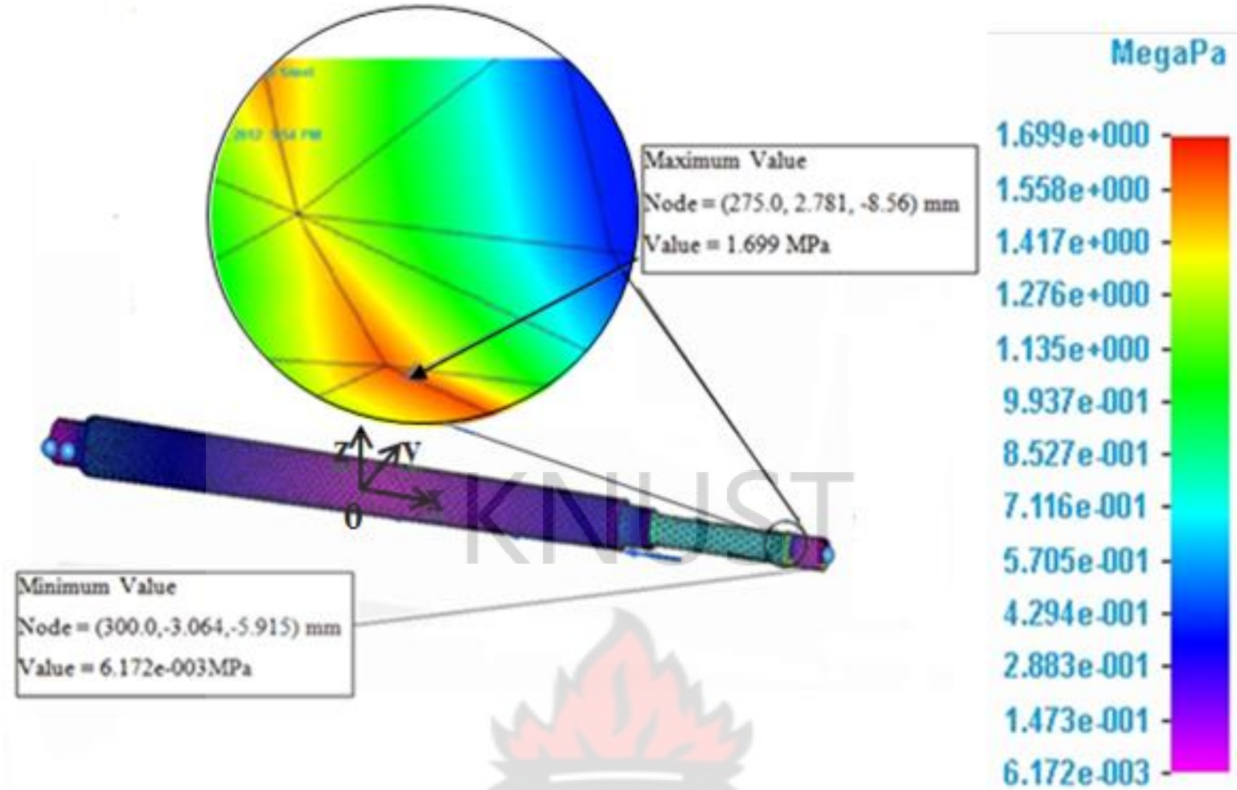


Figure 4-7: Simulation of Power Screw

Figure 4-4 represents the stress analysis results for option 1. The minimum stress on the power screw is 617 KPa and the maximum stress is 17 MPa.

4.4 NASTRAN SIMULATION AND OPTIMIZATION OF LINKAGE

The components were modeled using solid part and assemble part functions of the software. Tetra-Meshing was applied to the model to enable Solid Edge Simulation and finite element analysis (FEA) to be carried out.

Treatment of Bolt Holes

As in any finite element analysis, proper boundary conditions (BC's) are crucial if the results are to be of any significance, and this is especially true for topology optimization studies since these BC's will be the basis for the resulting distribution of material. An example of this lies in the treatment of the boundary condition around the region of a bolted or pinned joint. Bolts and pins can only transfer compressive load (unless, of course, they are bonded in place), thus they can only push on another surface that is in contact. In a finite element model, however, this phenomenon is somewhat difficult to capture as it requires the use of non-linear gap elements which have a very low stiffness in tension to simulate the lack of connectivity (Fornace, 2006). Bolt connectors were used at the bolt holes.

Constraints

As installed on the tricycle, the lifting support bar pivots about a pin through to the tricycle chassis. Furthermore, the lifting bar is also supported on a pin that is held by the power screw nut. The pin is supported on roller bearing which provides the intended motion. The linkage has one degree-of-freedom which is along the power screw axis. Pin constraints were applied at the nodes of the mechanism.

Loads

Using the previously determined load cases from the FBD models, the assembly model is updated to include the load vectors. Table 4.1 shows a summary of the four load cases used in the optimization of the linkage mechanism. Note that there are no applied moments to the system, and all forces are shown in units of Newtons.

Table 4-1: Summary of Linkage Mechanism load cases

Load Name	*Load Type*	*Load Value*	*Load Direction*	*Load Direction Option*
Force 1	Force	499.55 N	(0.00, 0.00, -1.00)	Along a vector
Force 2	Force	Fx: 146.20 N, Fy: 0 N, Fz: 0 N		Components
Force 3	Force	Fx: 0 N, Fy: 0 N, Fz: 132.20 N		Components
Force 4	Force	Fx: 0 N, Fy: 0 N, Fz: 18.10 N		Components

Material Definition

The last input parameter required before implementation of the optimization study is the material identification. In the case of the linkage mechanism, the material was specified as

1020 HR steel with mass density of 7850 kg/m^3 and ultimate tensile strength of 358.527 MPa and Poisson's ratio 0.33.

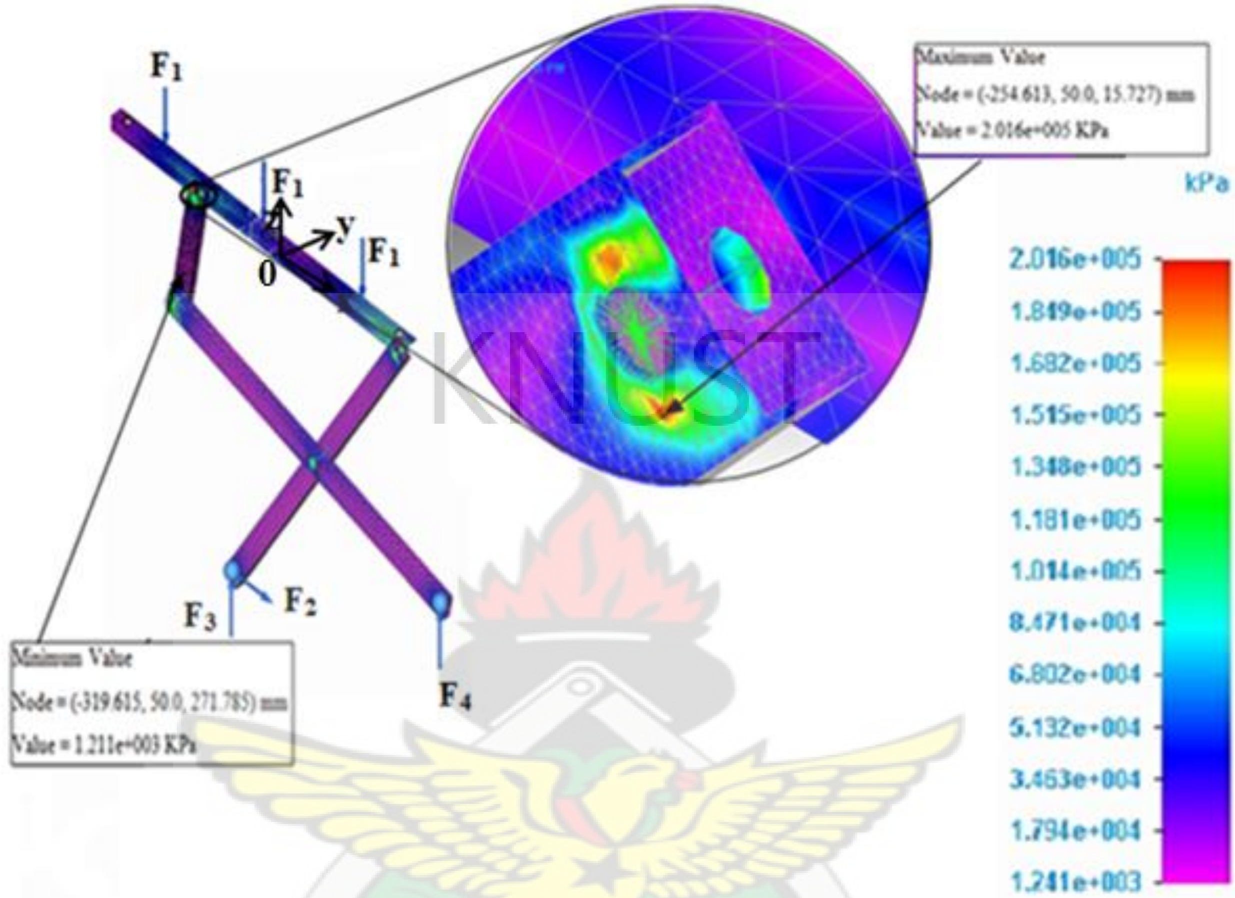


Figure 4-8: Simulation of Linkage Mechanism

Figure 4-5 represents the stress analysis results for option 1.

4.5 POWER SCREW NUT

Three different models were developed to include the existing design (option 1). The second model (option 2) has some materials of option 1 cutout. The dimensions of option 1 were also reduced to form option 3. Figure 4-6 shows a schematic drawing of the power screw nut with loading and constraints.

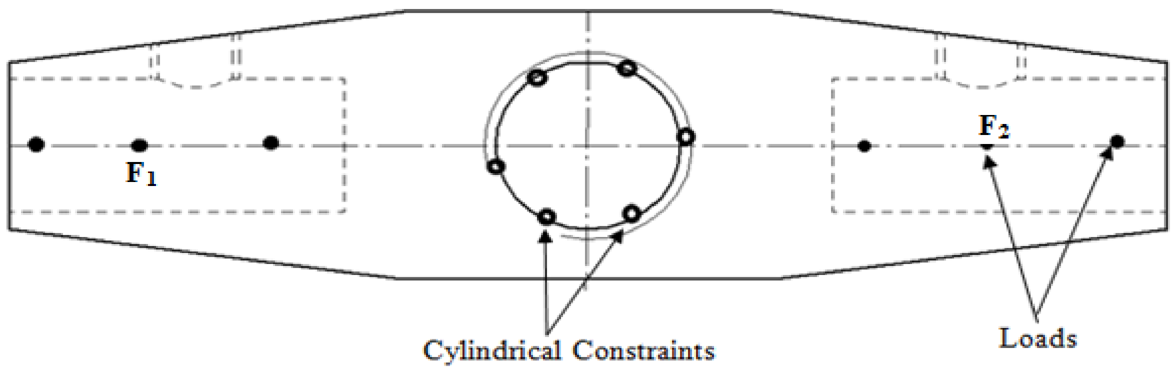


Figure 4-9: Schematic drawing of Power Screw Nut

Table 4-2: Loads

Load Name	Load Type	Load Value	Load Direction	Load Direction Option
Force 1	Force	Fx: 292.40 N, Fy: 0 N, Fz: 0 N		Components
Force 2	Force	Fx: 292.40 N, Fy: 0 N, Fz: 0 N		Components

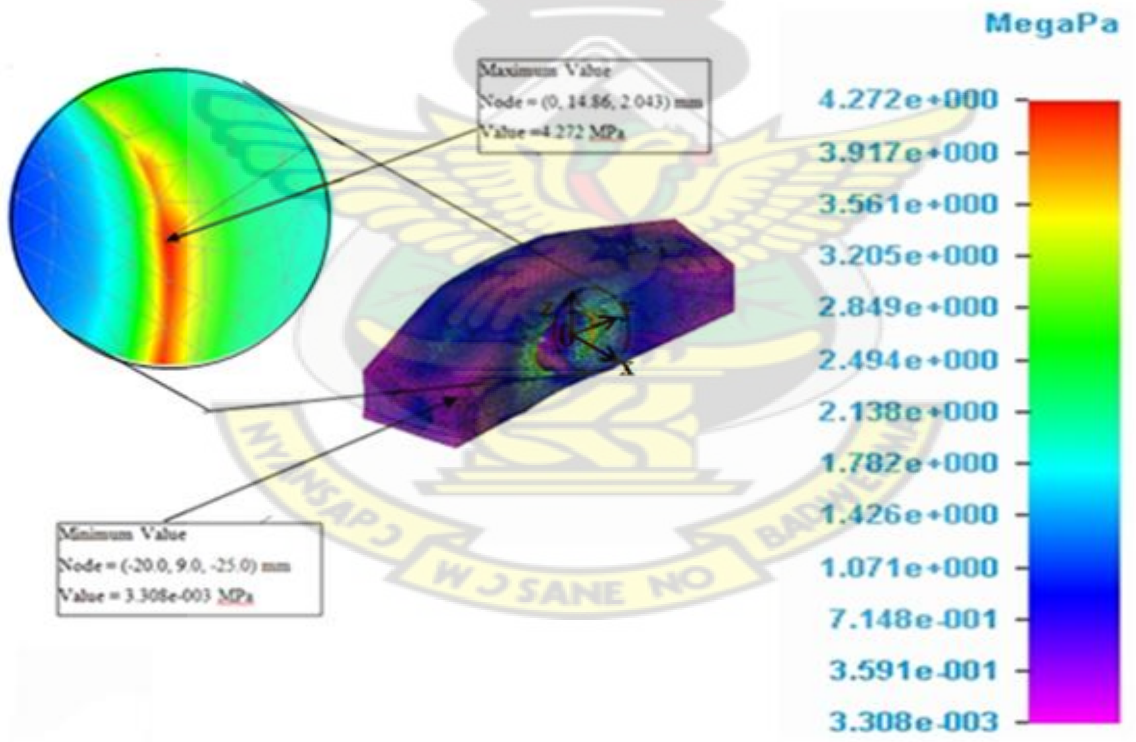


Figure 4-10: Simulation of Power Screw Nut

Figure 4-7 represents the stress analysis results for option 2. The minimum stress from the three options is 0.3308 MPa and the maximum stress is 42.72 MP

CHAPTER 5

RESULTS AND DISCUSSIONS

5.1 BIN

From table 5-1 and 5-2, the maximum displacement and maximum stress is 0.224 mm 40.41 MPa respectively. With material yield strength of 262 MPA, there are no severe stress concentrations that would indicate a faulty design. FEA results indicate a factor of safety of roughly 6.5. The maximum displacement at the node is however 0.224 mm and a safety factor of 2 is recommended. Mild steel is selected on merit for the bin design and the production drawing is as shown in appendix E.

Table 5.1 and 5-2 presents the FEMAP displacement and stress results when base plate of the bin is designed with a 1 mm thick mild steel and galvanized metal sheets.

Table 5-1: Displacement Results

Result component: Total Translation				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000 mm	350.000 mm	194.423 mm	12.500 mm
Maximum	0.224 mm	23.813 mm	448.843 mm	13.500 mm

Table 5-2: Stress Results

Result component: Von Mises				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.9651 MPa	350.000 mm	-594.038 mm	13.500 mm
Maximum	40.41 MPa	-350.000 mm	-17.115 mm	12.500 mm

5.2 LINKAGE MECHANISM

Summary of the results for three concepts of the linkage mechanism obtained from FEMAP are presented in table 5.3.

From table 5-3, the minimum stress and displacement occurs in option 3; 138.9 MPa and 7.689 mm respectively. Option 1 has the highest stress and displacement conditions; 201.6

MPa and11.647 mm. However, there are no severe stress concentrations that would indicate a faulty design, and the highest stress levels appear to be on the order of 138.9 – 201.6 MPa. FEA results indicate a factor of safety of roughly 2. Option 1 is selected for linkage mechanism design and production drawing is in appendix E.

Table 5-3: Summary of Stresses and displacement results

STUDY	Maximum displacement	Maximum stress
option 1	11.647 mm (-625, 20, 20)	201.6 MPa (-254.613, -50, 15.727)
option 2	7.875 mm (-625, 19.768, 20.296)	167.2 MPa (779.102, -68.890, -533.833)
option 3	7.689 mm (-625, 19.768, 20.296)	138.9 MPa (-231.363, -40.23, -5.741)

5.3 POWER SCREW

Summary of the results for three concepts of the power screw obtained from FEMAP are presented in table 5.4.

Table 5-4: summary of FEMAP stress and displacement results

STUDY	Maximum displacement	Maximum stress
option 1	3.551e-004 mm	16.99 MPa
option 2	4.917e-004 mm	23.66 MPa
option 3	7.399e-004 mm	39.1 MPa

From table 5-4, the minimum stress and displacement occurs in option 1; 16.99 MPa and 3.551e-004 mm respectively. Option 3 has the highest stress and displacement conditions; 39.1 MPa 7.399e-004 mm respectively. However there are no severe stress concentrations that would indicate a faulty design and the highest stress levels appear to be on the order of 17 – 39 MPa. FEA results indicate a factor of safety of 2. Option 2 is chosen based on manufacturing and assembling constraints. The production drawing for the power screw design is as shown in appendix E.

5.4 POWER SCREW NUT

Summary of the results for three options of the power screw nut obtained from FEMAP are presented in table 5.5

Table 5-5: summary of FEMAP stress and displacement results

STUDY	Maximum displacement	Maximum stress	Mass
Option 1	0.001004 mm	42.7 MPa	1.572 kg
Option 2	0.002047 mm	60.84 MPa	1.363 kg
Option 3	0.001462 mm	61.06 MPa	1.129 kg

From table 5-5, the minimum stress and displacement occurs in option 1; 42.7 Mpa and 0.001004 mm respectively. Option 3 has the highest stress conditions; 61.06 Mpa and option 2, the highest displacement 0.002047 mm. However, there are no severe stress concentrations that would indicate a faulty design, and the highest stress levels appear to be on the order of 42 – 61 Mpa. FEA results indicate a factor of safety of 2. Appendix E contains the production drawing for the power screw nut and option 3 is selected.

CHAPTER 6

CONCLUSION AND RECOMMENDATION

6.1 CONCLUSION

This research was developed to optimize tipping mechanism for small tricycle trucks for solid waste collection. The objective of the design of minimum weight of the tricycle truck is achieved through this work. The optimized components of the tricycle had been designed. The optimization of four major components of the tricycle; the bin, linkage mechanism, power screw and nut, was done using Siemens Solid Edge ST3 software package, NX Nastran (7) solver. The overall weight of the tricycle has been reduced from 143 kg to 127.6 kg which represents 10.77% reduction in weight. The gross weight (i.e. 70 kg mass person using tricycle) is 197 kg which is within the gross weight category of tricycle (i.e. 150 – 200 kg). Table 5-6 shows the weight of optimized components and the existing components. The test of the prototype results shows the torque requirement of between 20 Nm and 45 Nm to ride on a horizontal plane; an improvement over the existing tricycle (25 Nm – 60 Nm). The optimization process has improved the overall performance of the tricycle in terms of weight and torque requirement.

Table 6-1: weights of optimized components and existing components

Part	Weight (Kg)		% Reduction
	Old	New	
Bin	50	40	20.0
Linkage Mechanism	11.6	7.5	35.4
Power Screw	2.6	1.8	30.8
Power Screw Nut	1.6	1.1	31.3
Chassis and frame extensions	77.2	77.2	0.0
Total	**143**	**127.6**	**10.77**

6.2 RECOMMENDATION

There is unlimited scope of future work in design optimization of the tipping mechanism, by increasing number of constrains for the components optimized, by considering different materials, by considering different geometrical and design aspect losses so that the design can be minimize more efficiently. Other components such as the chassis and frame extensions may be optimized as well.

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APPENDICES

APPENDIX A: Nastran Report on Bin Base Simulation and Analysis

Bin Base Option 1

Material Properties

Mild Steel

Property	Value
Density	7850.000 kg/m^3
Coef. of Thermal Exp.	0.0000 /C
Thermal Conductivity	0.032 kW/m-C
Specific Heat	481.000 J/kg-C
Modulus of Elasticity	199947.953 MPa
Poisson's Ratio	0.290
Yield Stress	262.001 MPa
Ultimate Stress	358.527 MPa
Elongation %	0.000

Loads

Load Name	Load Type	Load Value	Load Direction	Load Direction Option
Force 1	Force	490.50 N	(0.00, 0.00, -1.00)	Along a vector

Constraints

Constraint Name	Constraint Type	Degrees of Freedom
Fixed 1	Fixed	FREE DOF: None
Fixed 2	Fixed	FREE DOF: None

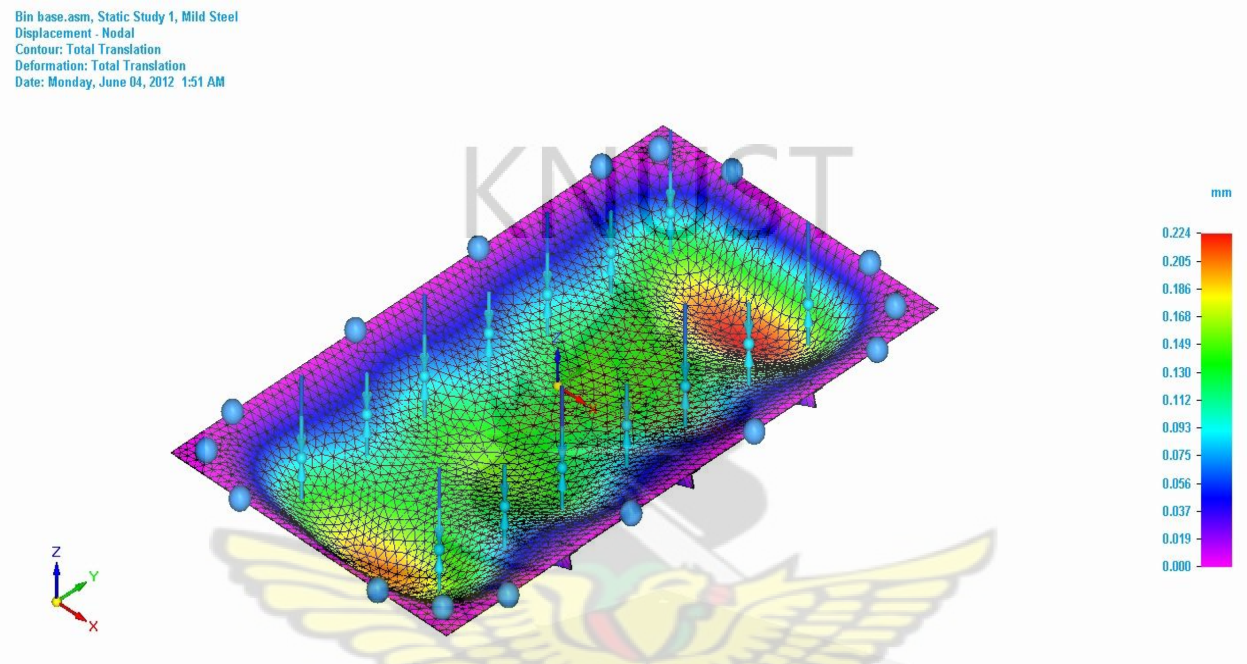
Connector

Connector Name	Connector Type	Search Distance	Minimum Search Distance	Coefficient Of Static Friction	Penalty Value
Connector 1	Glue	1.00 mm			100.00
Connector 2	Glue	1.00 mm			100.00
Connector 3	Glue	1.00 mm			100.00

Results

Displacement Results

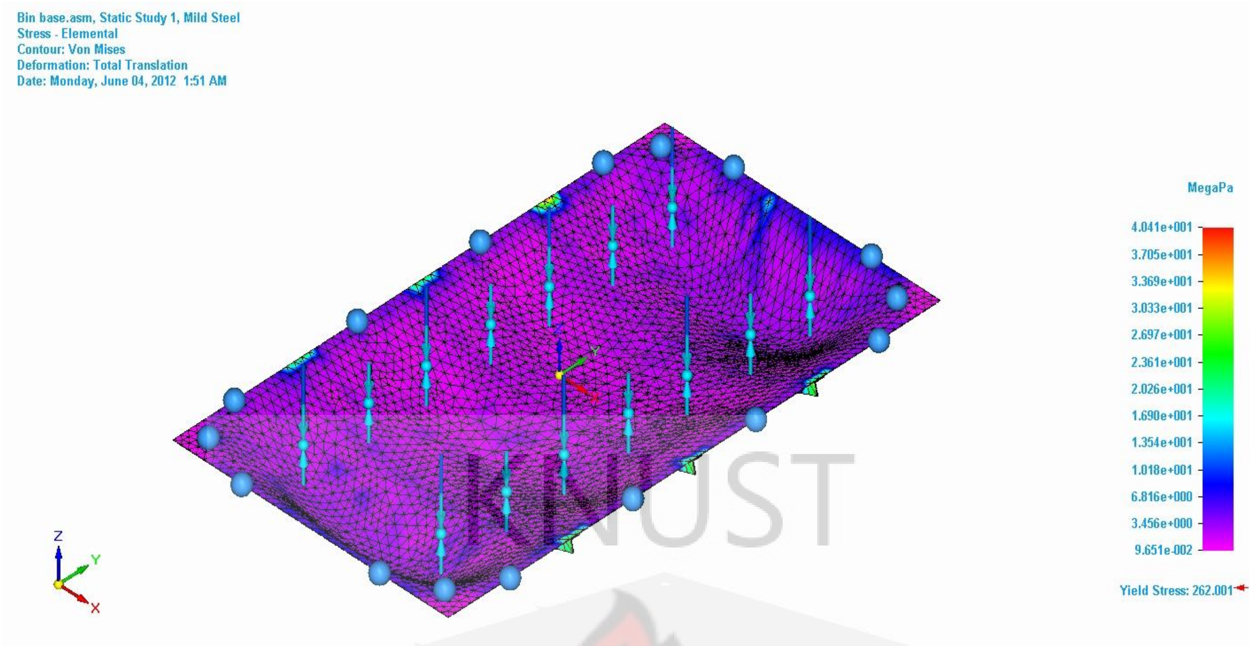
Result component: Total Translation				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000 mm	350.000 mm	194.423 mm	12.500 mm
Maximum	0.224 mm	23.813 mm	448.843 mm	13.500 mm



Total Translation

Stress Results

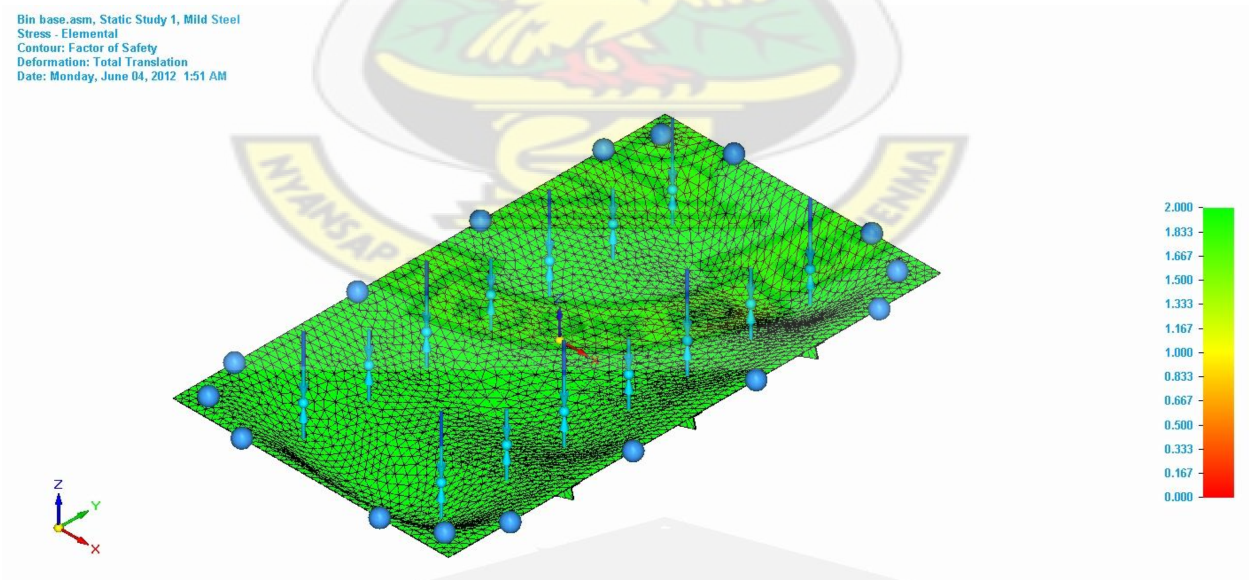
Result component: Von Mises				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.9651 MPa	350.000 mm	-594.038 mm	13.500 mm
Maximum	40.41 MPa	-350.000 mm	-17.115 mm	12.500 mm



Von Mises

Factor of Safety Results

Result Component: Factor of Safety				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000	-350.000 mm	-17.115 mm	12.500 mm
Maximum	2.000	350.000 mm	-594.038 mm	13.500 mm



Factor of Safety

Bin Base Option 2

Material Properties

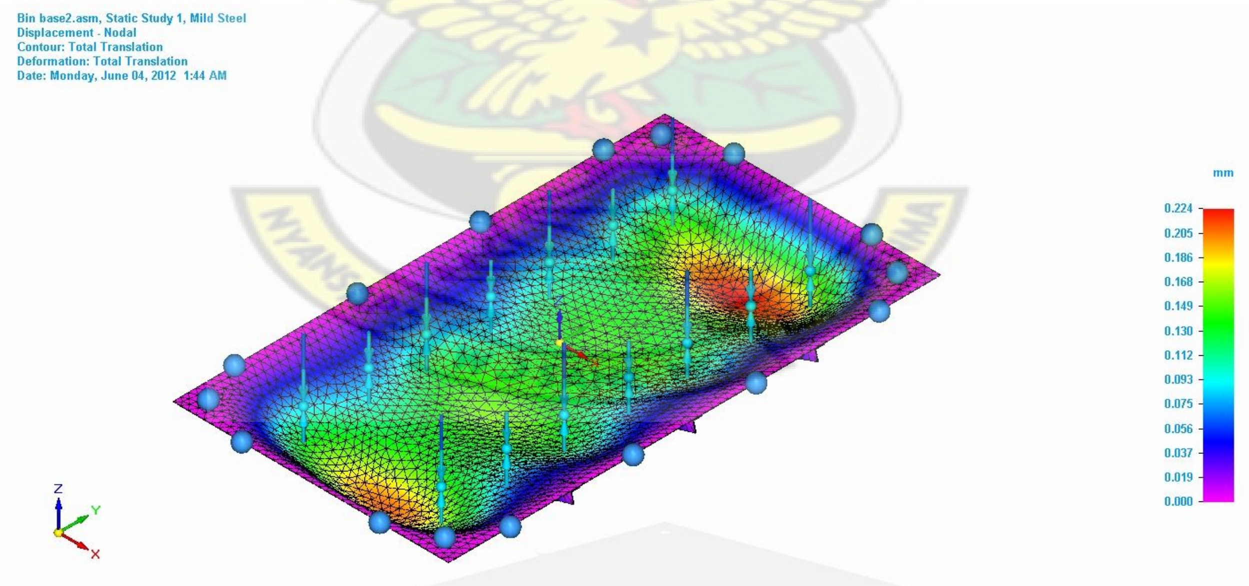
Galvanized steel

Property	Value
Density	7833.000 kg/m^3
Coef. of Thermal Exp.	0.0000 /C
Thermal Conductivity	0.032 kW/m-C
Specific Heat	481.000 J/kg-C
Modulus of Elasticity	199947.953 MPa
Poisson's Ratio	0.290
Yield Stress	262.001 MPa
Ultimate Stress	358.527 MPa
Elongation %	0.000

Results

Displacement Results

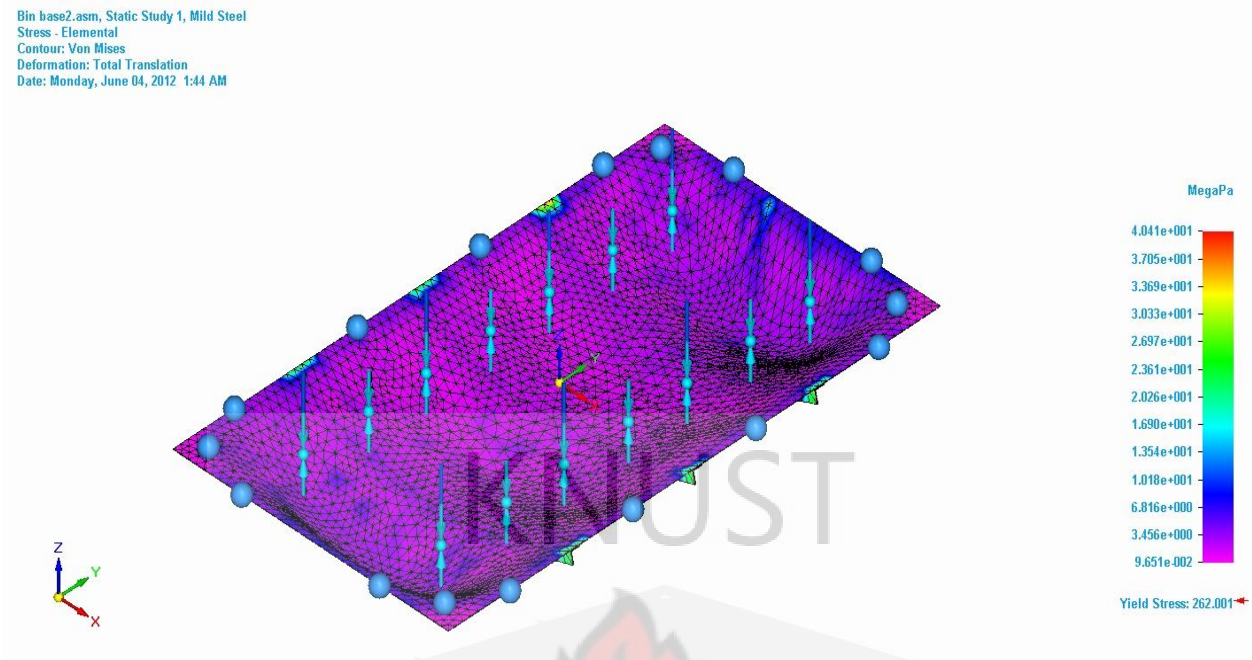
Result component: Total Translation				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000 mm	350.000 mm	194.423 mm	12.500 mm
Maximum	0.224 mm	23.813 mm	448.843 mm	13.500 mm



Total Translation

Stress Results

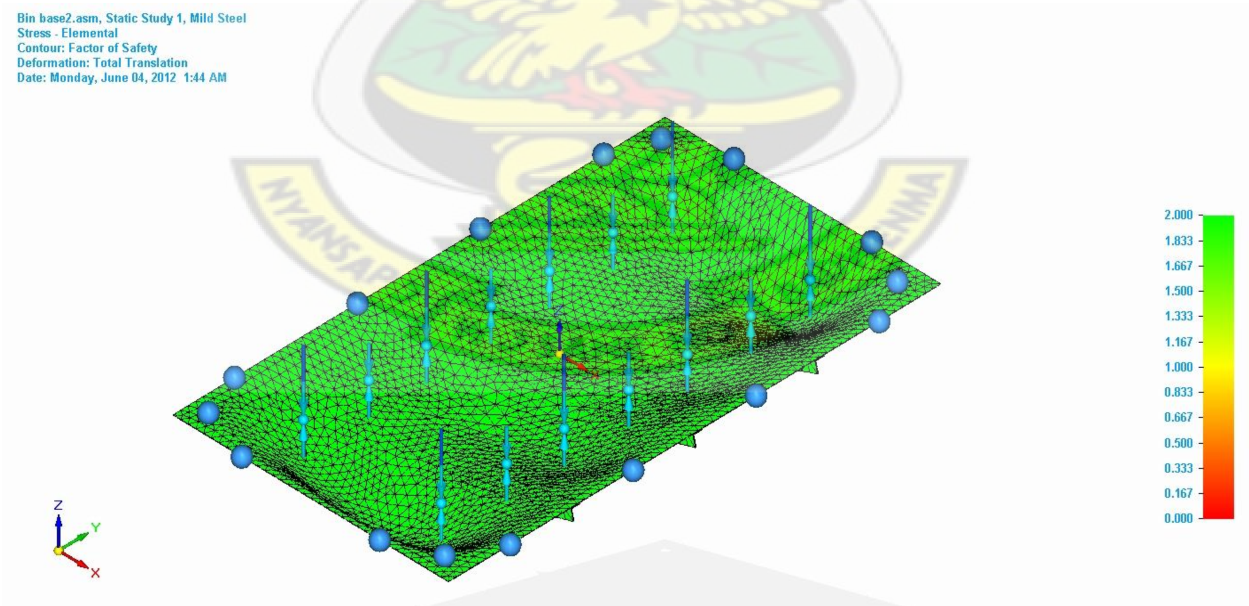
Result component: Von Mises				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.9651 MPa	350.000 mm	-594.038 mm	13.500 mm
Maximum	40.41 MPa	-350.000 mm	-17.115 mm	12.500 mm



Von Mises

Factor of Safety Results

Result Component: Factor of Safety				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000	-350.000 mm	-17.115 mm	12.500 mm
Maximum	2.000	350.000 mm	-594.038 mm	13.500 mm



Factor of Safety

APPENDIX B: Nastran Report on Linkage Mechanism Simulation and Analysis

Linkage Mechanism Option 1

Material Properties

Mild Steel

Property	Value
Density	7850.000 kg/m^3
Coef. of Thermal Exp.	0.0000 /C
Thermal Conductivity	0.032 kW/m-C
Specific Heat	481.000 J/kg-C
Modulus of Elasticity	199947.953 MegaPa
Poisson's Ratio	0.290
Yield Stress	262.001 MegaPa
Ultimate Stress	358.527 MegaPa
Elongation %	0.000

Loads

Load Name	Load Type	Load Value	Load Direction	Load Direction Option
Force 1	Force	441.45 N	(0.00, 0.00, -1.00)	Along a vector
Force 2	Force	Fx: 146.20 N, Fy: 0 N, Fz: 0 N		Components
Force 3	Force	Fx: 0 N, Fy: 0 N, Fz: 132.20 N		Components
Force 4	Force	Fx: 0 N, Fy: 0 N, Fz: 18.10 N		Components

Constraints

Constraint Name	Constraint Type	Degrees of Freedom
Pinned 1	Pinned	FREE DOF: None

Connector

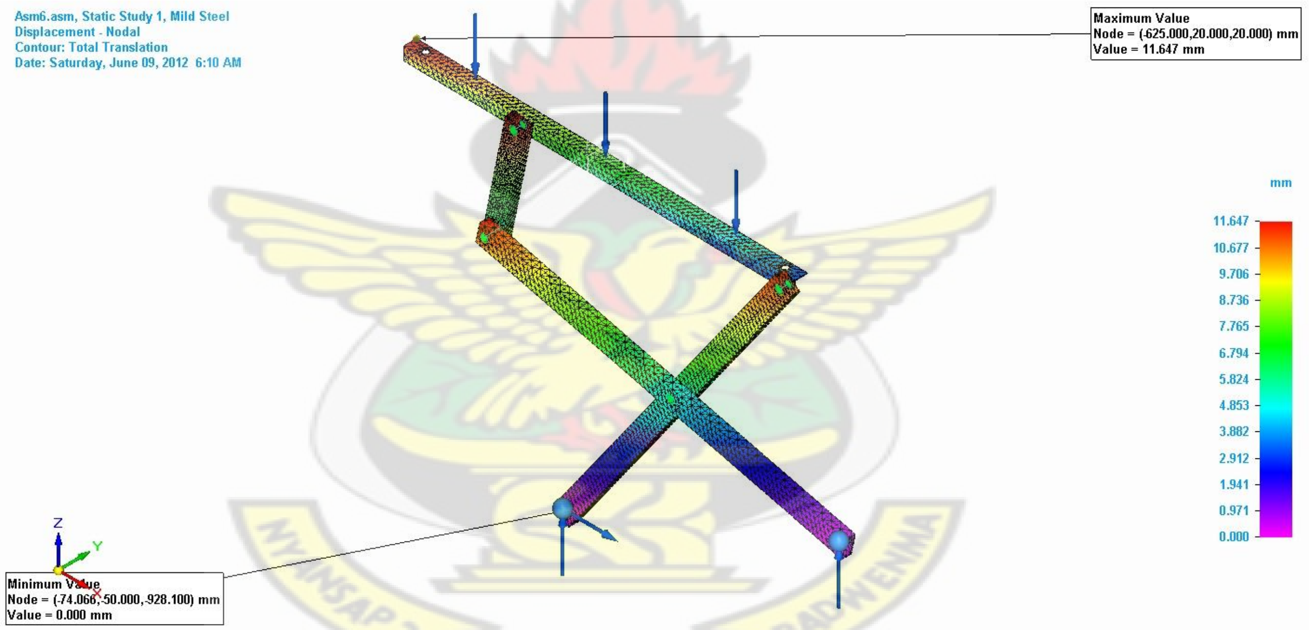
Connector Name	Connector Type	Search Distance	Minimum Search Distance	Coefficient Of Static Friction	Penalty Value
Bolt Connection 1	Bolt Connection				

Bolt Connection 2	Bolt Connection
Bolt Connection 3	Bolt Connection
Bolt Connection 4	Bolt Connection

Results

Displacement Results

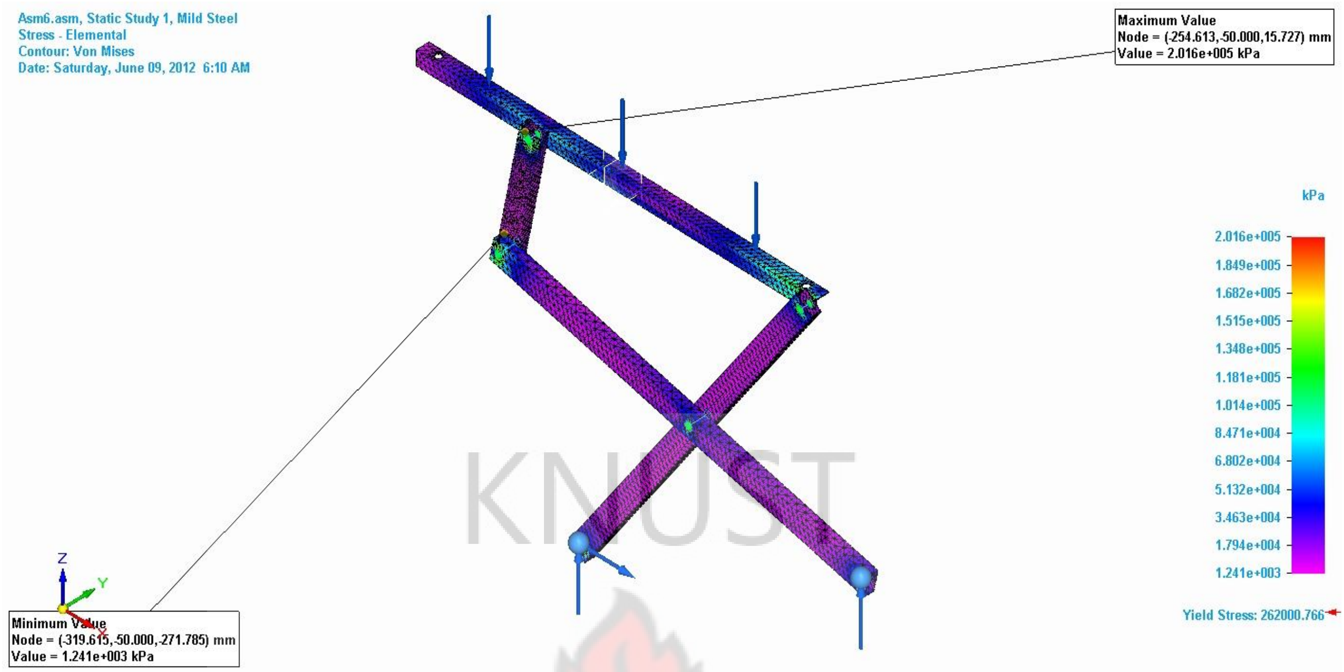
Result component: Total Translation				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000 mm	-74.066 mm	-50.000 mm	-928.100 mm
Maximum	11.647 mm	-625.000 mm	20.000 mm	20.000 mm



Total Translation

Stress Results

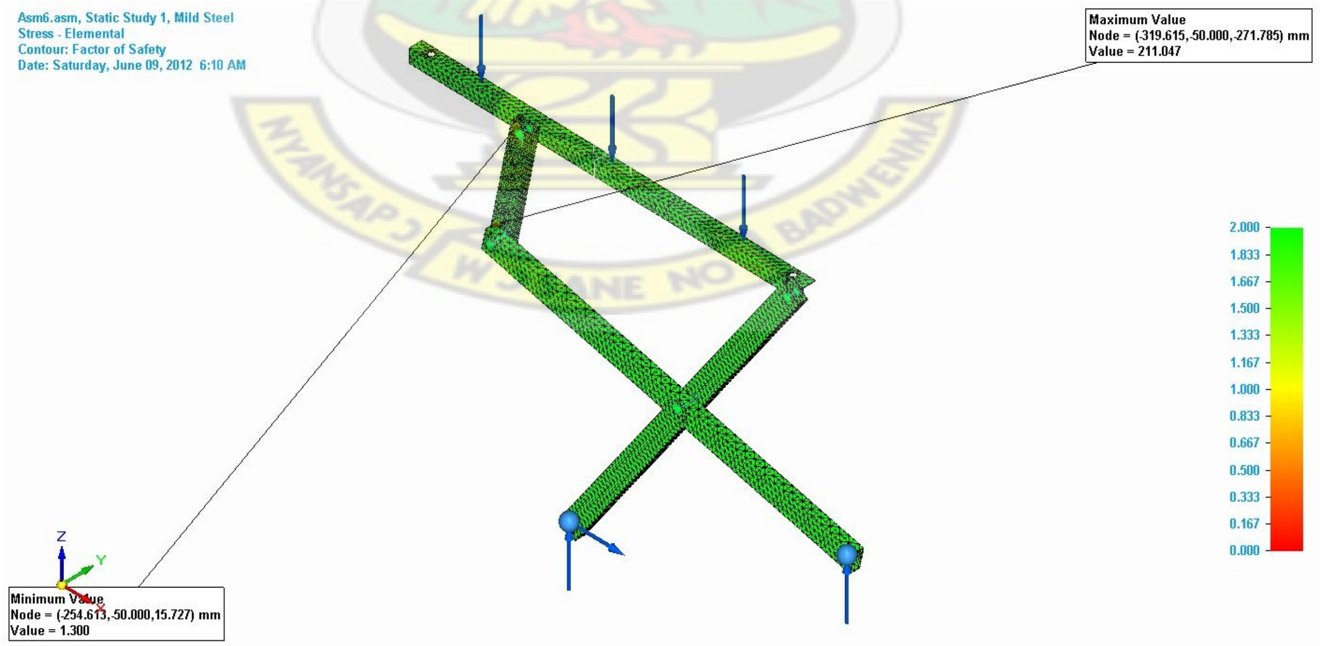
Result component: Von Mises				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	1.241 MPa	-319.615 mm	-50.000 mm	-271.785 mm
Maximum	201.6 MPa	-254.613 mm	-50.000 mm	15.727 mm



Von Mises

Factor of Safety Results

Result Component: Factor of Safety				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000	-254.613 mm	-50.000 mm	15.727 mm
Maximum	2.000	-319.615 mm	-50.000 mm	-271.785 mm

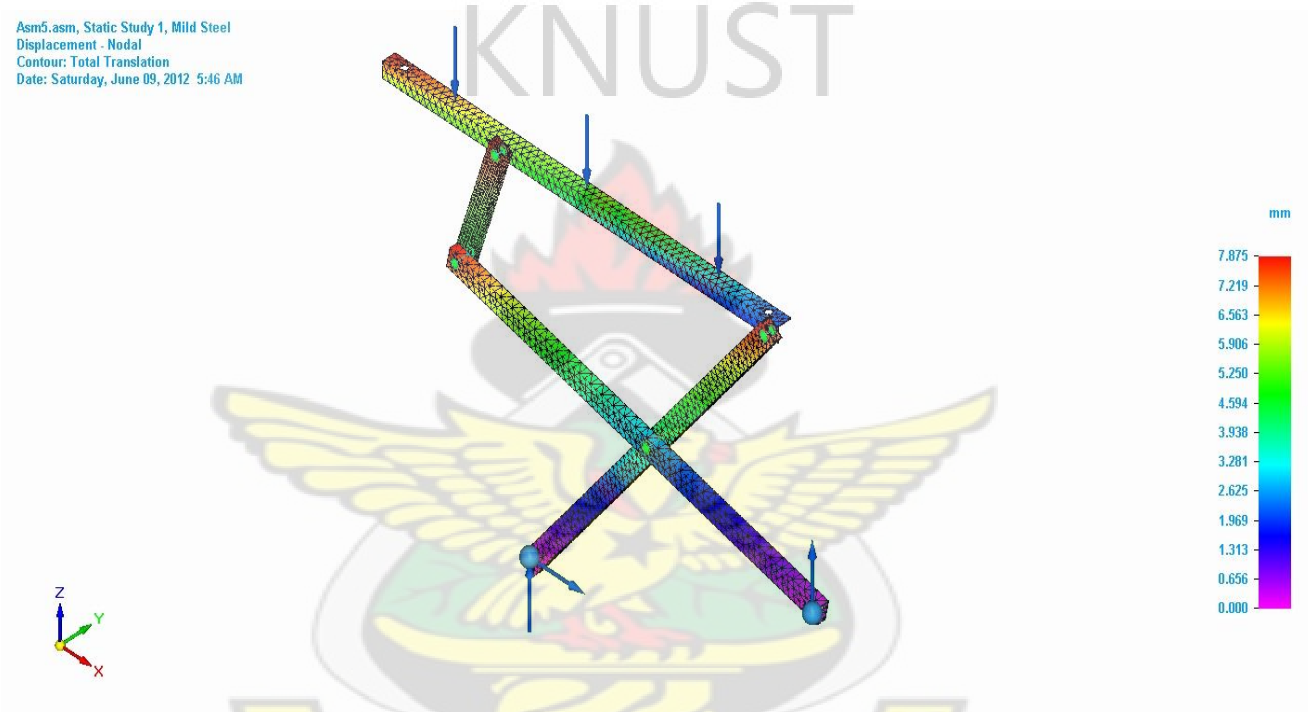


Factor of Safety

Linkage Mechanism Option 2

Displacement Results

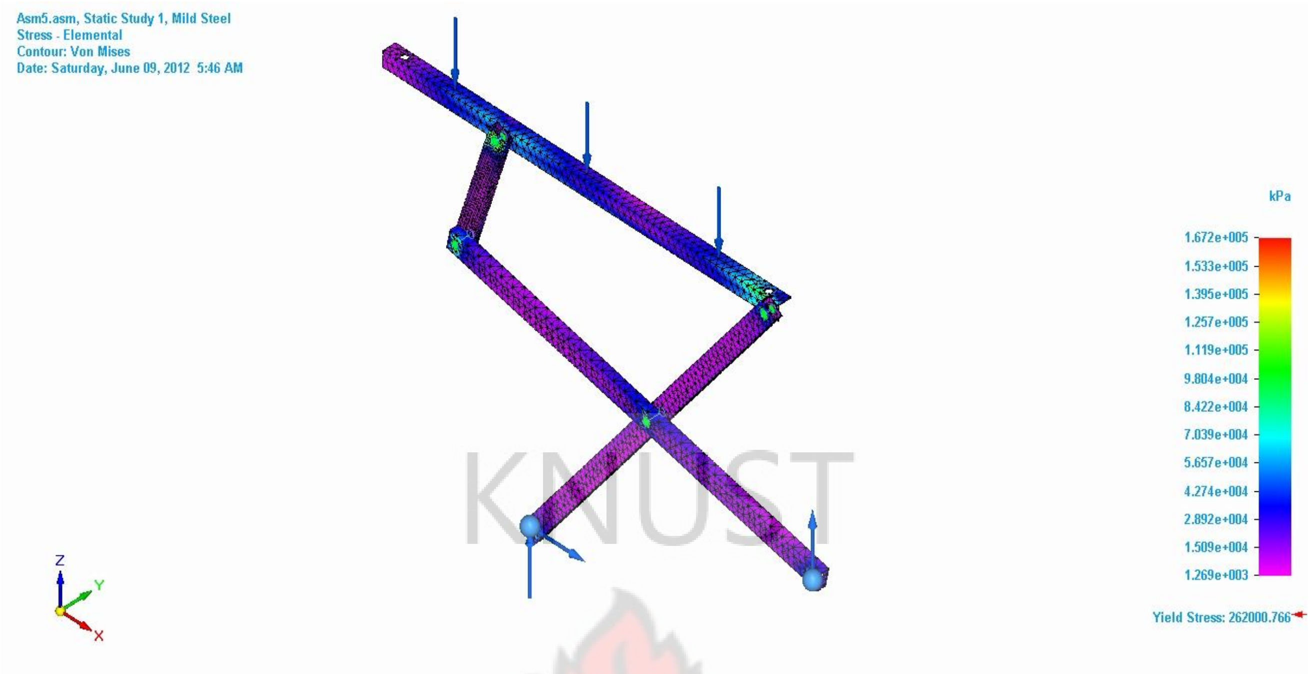
Result component: Total Translation				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000 mm	-118.600 mm	-45.232 mm	-897.311 mm
Maximum	7.875 mm	-625.000 mm	19.768 mm	20.296 mm



Total Translation

Stress Results

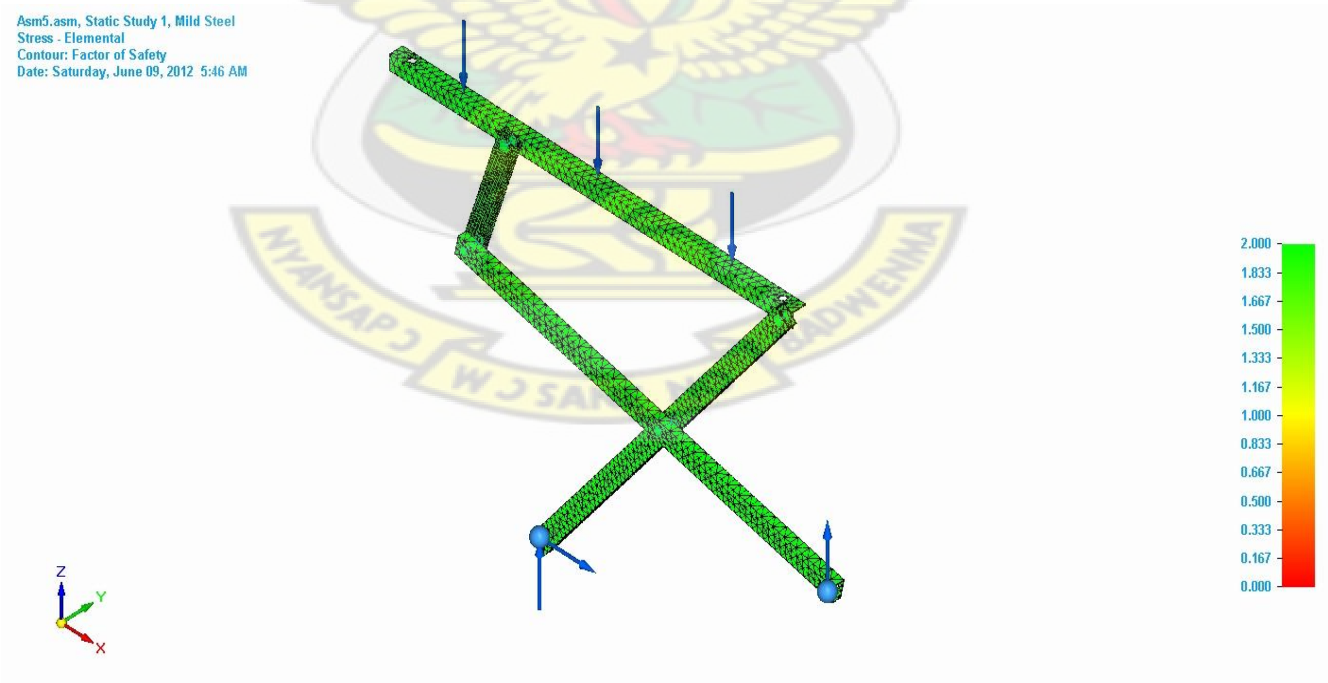
Result component: Von Mises				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	1.269 MPa	-351.534 mm	-20.232 mm	-301.450 mm
Maximum	167.2 MPa	779.102 mm	-68.890 mm	-533.833 mm



Von Mises

Factor of Safety Results

Result Component: Factor of Safety				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000	779.102 mm	-68.890 mm	-533.833 mm
Maximum	2.000	-351.534 mm	-20.232 mm	-301.450 mm

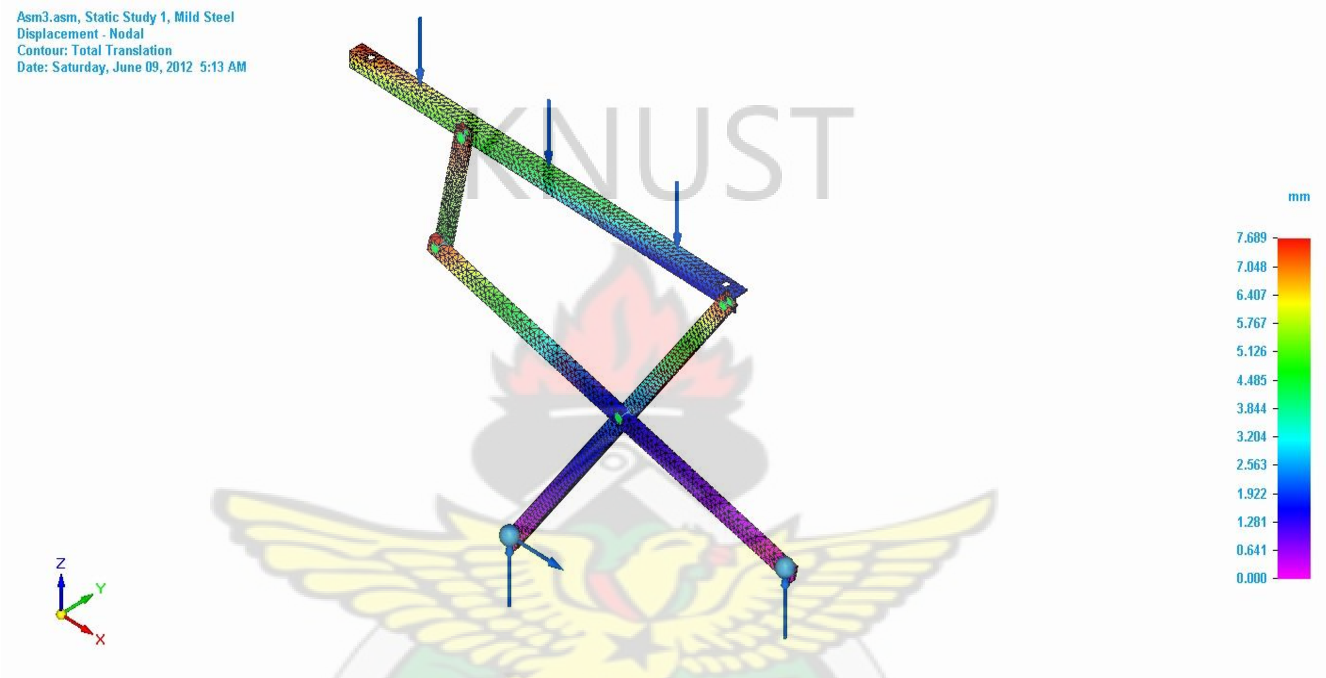


Factor of Safety

Linkage Mechanism Option 3

Displacement Results

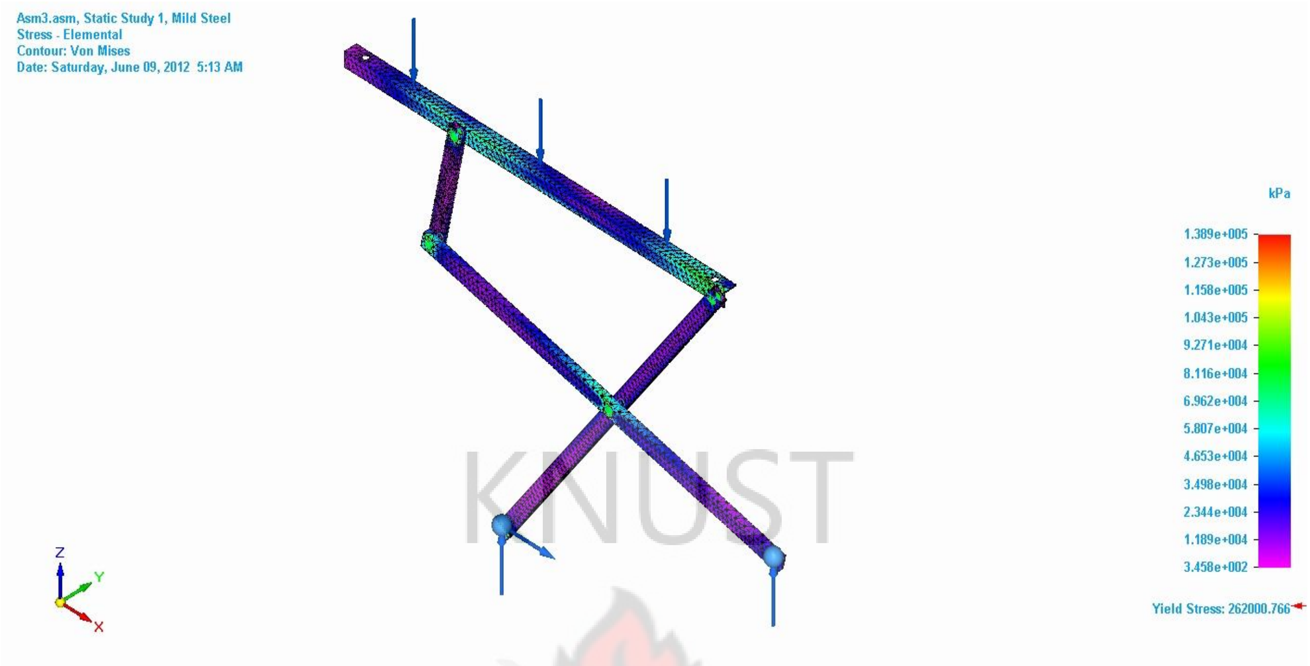
Result component: Total Translation				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000 mm	-78.440 mm	-40.232 mm	-945.756 mm
Maximum	7.689 mm	-625.000 mm	19.768 mm	20.296 mm



Total Translation

Stress Results

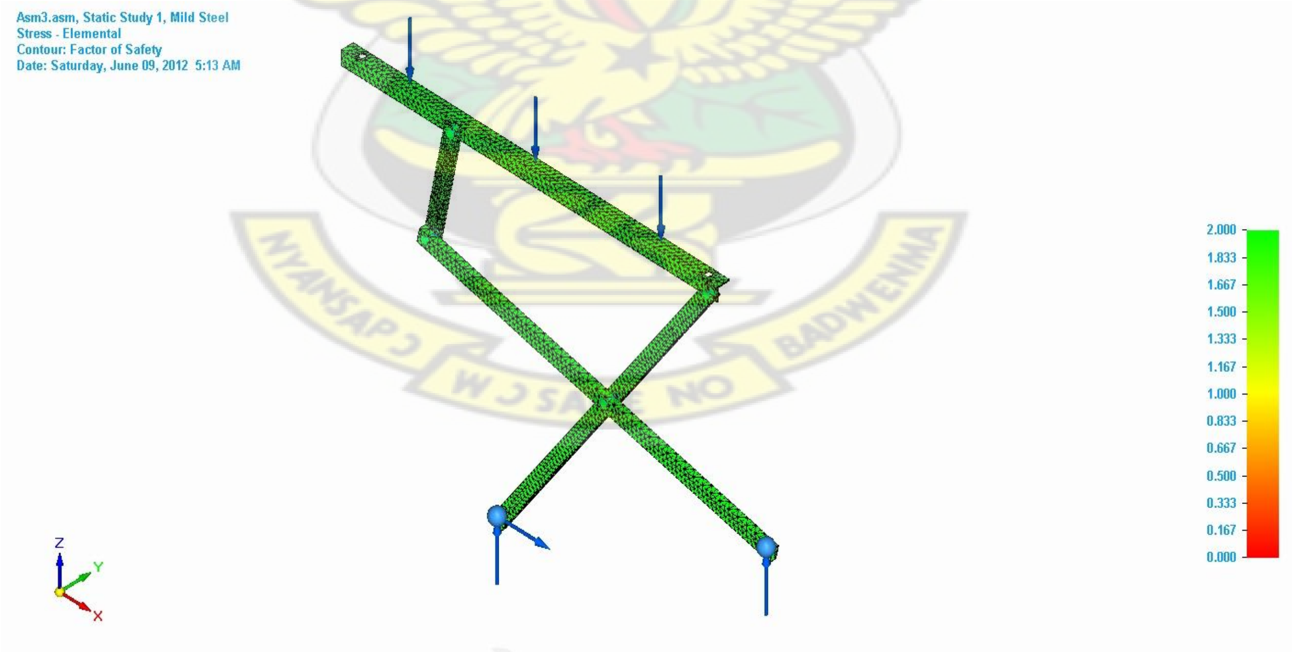
Result component: Von Mises				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.3458 MPa	-308.241 mm	-20.232 mm	-312.908 mm
Maximum	138.9 MPa	-231.363 mm	-40.232 mm	-5.741 mm



Von Mises

Factor of Safety Results

Result Component: Factor of Safety				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000	-231.363 mm	-40.232 mm	-5.741 mm
Maximum	2.000	-308.241 mm	-20.232 mm	-312.908 mm



Factor of Safety

APPENDIX C: Nastran Report on Power Screw Simulation and Analysis

Power Screw Option 1

Solids

Solid Name	Material	Mass	Volume	Weight
Power Screw 1.par	Mild Steel	2.570 kg	327375.422 mm^3	25.184991 N

Material Properties

Mild Steel

Property	Value
Density	7850.000 kg/m^3
Coef. of Thermal Exp.	0.0000 /C
Thermal Conductivity	0.032 kW/m-C
Specific Heat	481.000 J/kg-C
Modulus of Elasticity	199947.953 MPa
Poisson's Ratio	0.290
Yield Stress	262.001 MPa
Ultimate Stress	358.527 MPa
Elongation %	0.000

Loads

Load Name	Load Type	Load Value	Load Direction	Load Direction Option
Force 1	Force	584.80 N	(-1.00, 0.00, 0.00)	Along a vector

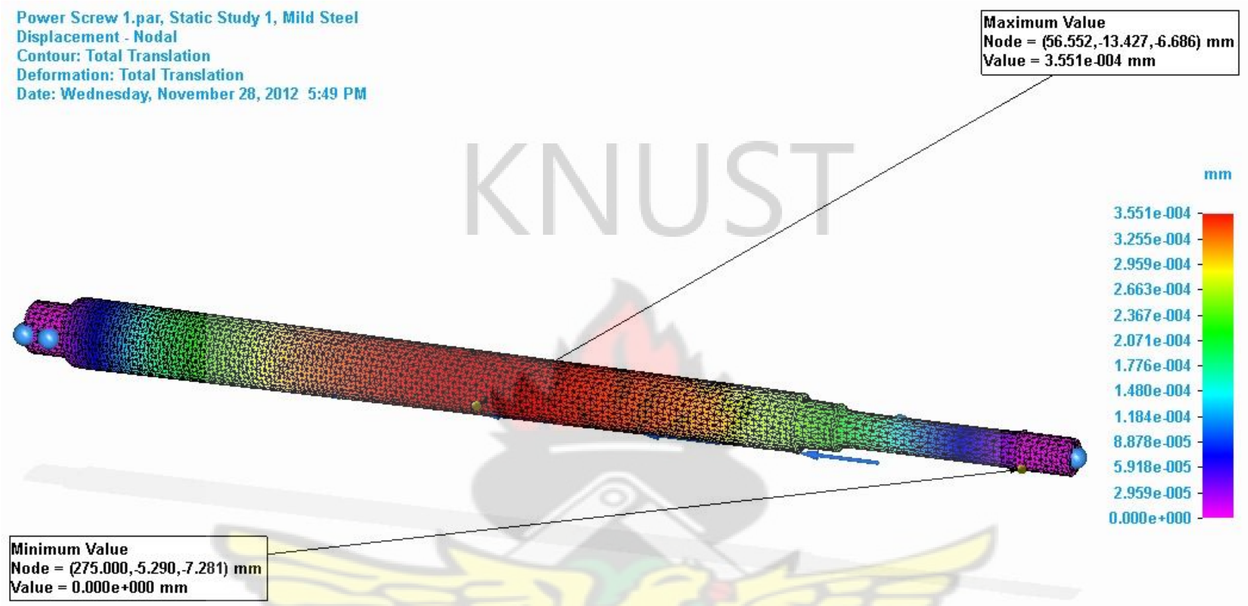
Constraints

Constraint Name	Constraint Type	Degrees of Freedom
Fixed 1	Fixed	FREE DOF: None
Cylindrical 1	Cylindrical	FREE DOF: None

Results

Displacement Results

Result component: Total Translation				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000e+000 mm	275.000 mm	-5.290 mm	-7.281 mm
Maximum	3.551e-004 mm	56.552 mm	-13.427 mm	-6.686 mm



Total Translation

Stress Results

Result component: Von Mises				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.6172 MPa	300.000 mm	-3.064 mm	-5.915 mm
Maximum	16.99 MPa	275.000 mm	2.781 mm	-8.560 mm

Von Mises

Factor of Safety Results

Result Component: Factor of Safety				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000	275.000 mm	2.781 mm	-8.560 mm
Maximum	2.000	300.000 mm	-3.064 mm	-5.915 mm

Factor of Safety

Power Screw Option 2

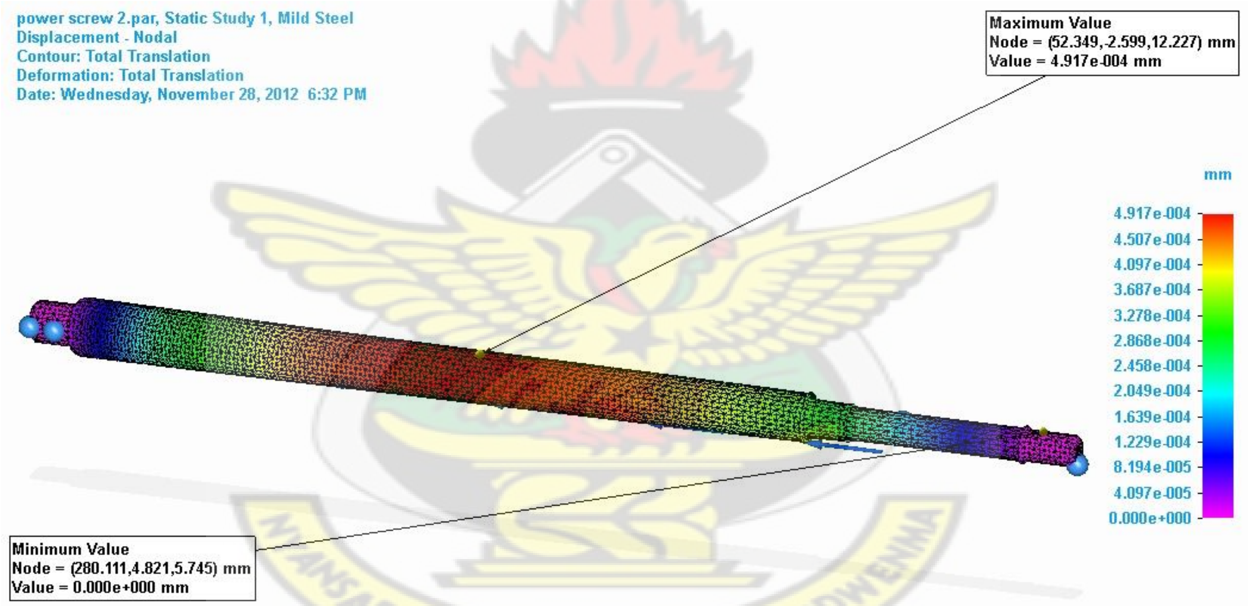
Solids

Solid Name	Material	Mass	Volume	Weight
power screw 2.par	Mild Steel	1.793 kg	228398.760 mm^3	17.570717 N

Results

Displacement Results

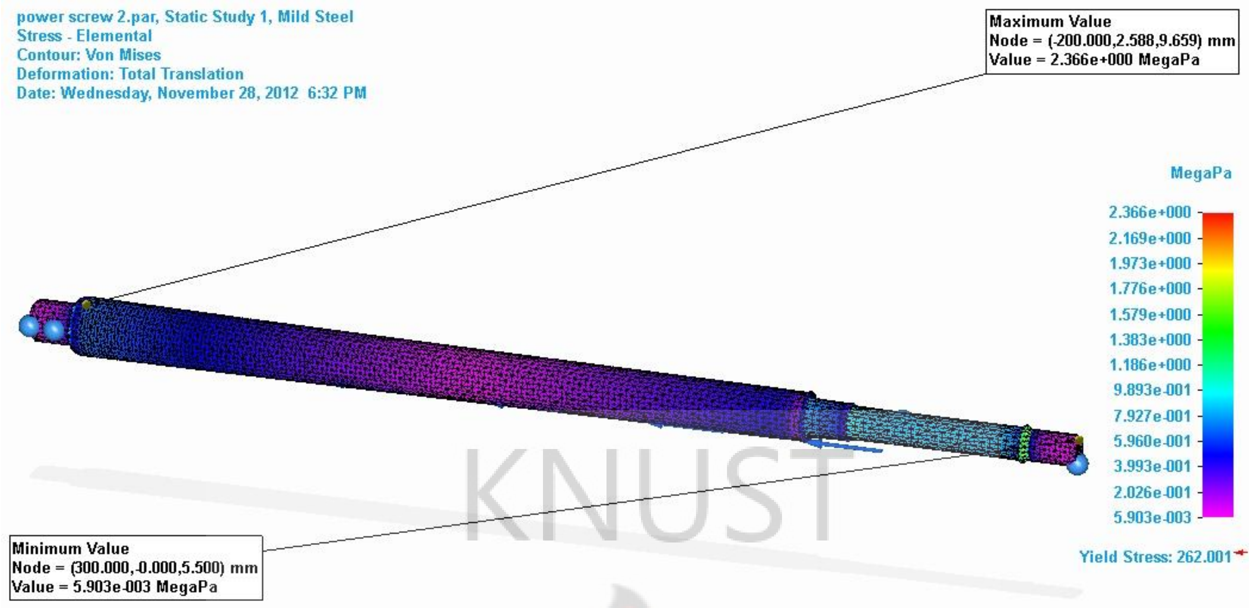
Result component: Total Translation				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000e+000 mm	280.111 mm	4.821 mm	5.745 mm
Maximum	4.917e-004 mm	52.349 mm	-2.599 mm	12.227 mm



Total Translation

Stress Results

Result component: Von Mises				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.5903 MPa	300.000 mm	-0.000 mm	5.500 mm
Maximum	23.66 MPa	-200.000 mm	2.588 mm	9.659 mm



Von Mises

Factor of Safety Results

Result Component: Factor of Safety				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000	-200.000 mm	2.588 mm	9.659 mm
Maximum	2.000	300.000 mm	-0.000 mm	5.500 mm

Factor of Safety

Power Screw Option 3

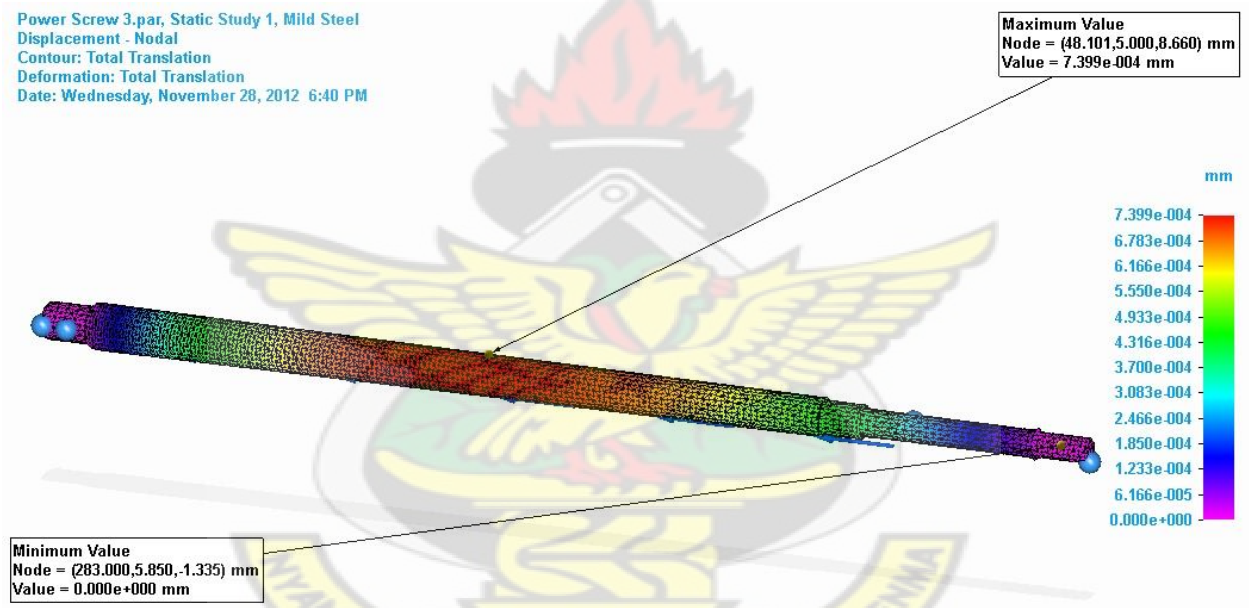
Solids

Solid Name	Material	Mass	Volume	Weight
Power Screw 3.par	Mild Steel	1.171 kg	149150.253 mm^3	11.474129 N

Results

Displacement Results

Result component: Total Translation				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000e+000 mm	283.000 mm	5.850 mm	-1.335 mm
Maximum	7.399e-004 mm	48.101 mm	5.000 mm	8.660 mm



Total Translation

Stress Results

Result component: Von Mises				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.1843 MPa	300.000 mm	3.909 mm	3.117 mm
Maximum	39.13 MPa	-200.000 mm	5.894 mm	-6.802 mm

Von Mises

Factor of Safety Results

Result Component: Factor of Safety				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000	-200.000 mm	5.894 mm	-6.802 mm
Maximum	2.000	300.000 mm	3.909 mm	3.117 mm

Factor of Safety

APPENDIX D: Nastran Report on Power Screw Nut Simulation and Analysis

Nut Option 1

Solids

Solid Name	Material	Mass	Volume	Weight
NUT.par	Mild Steel	1.572 kg	200231.155 mm^3	15.403783 N

Material Properties

Mild Steel

Property	Value
Density	7850.000 kg/m^3
Coef. of Thermal Exp.	0.0000 /C
Thermal Conductivity	0.032 kW/m-C
Specific Heat	481.000 J/kg-C
Modulus of Elasticity	199947.953 MPa
Poisson's Ratio	0.290
Yield Stress	262.001 MPa
Ultimate Stress	358.527 MPa
Elongation %	0.000

Loads

Load Name	Load Type	Load Value	Load Direction	Load Direction Option
Force 1	Force	Fx: 292.40 N, Fy: 0 N, Fz: 0 N		Components
Force 2	Force	Fx: 292.40 N, Fy: 0 N, Fz: 0 N		Components

Constraints

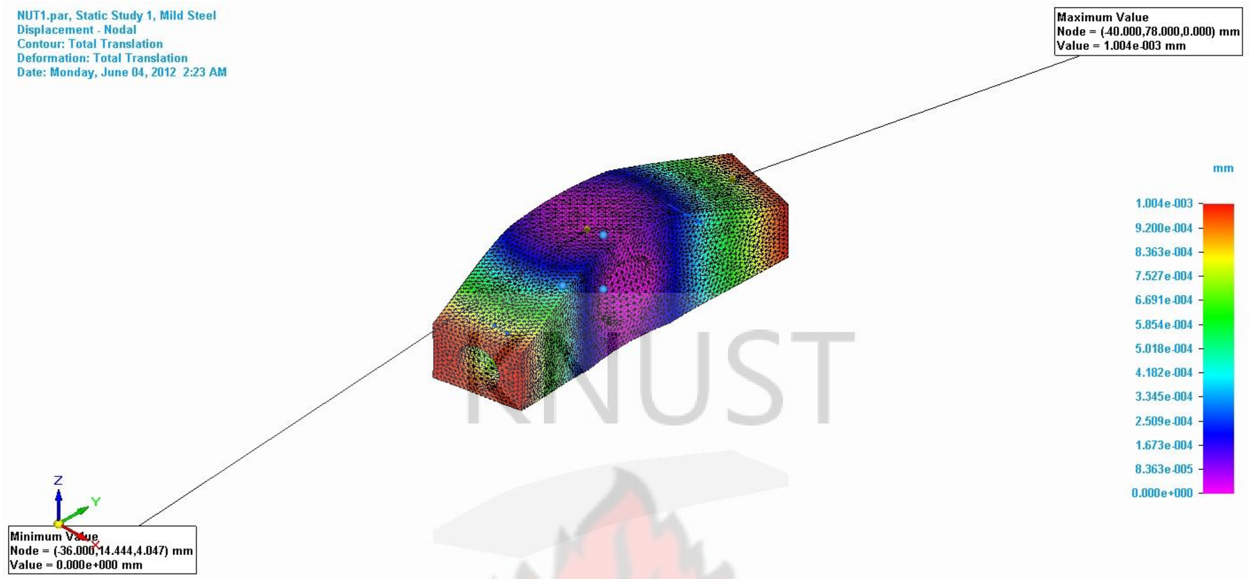
Constraint Name	Constraint Type	Degrees of Freedom
Cylindrical 1	Cylindrical	FREE DOF: None

Results

Displacement Results

Result component: Total Translation				
Extent	Value	X	Y	Z

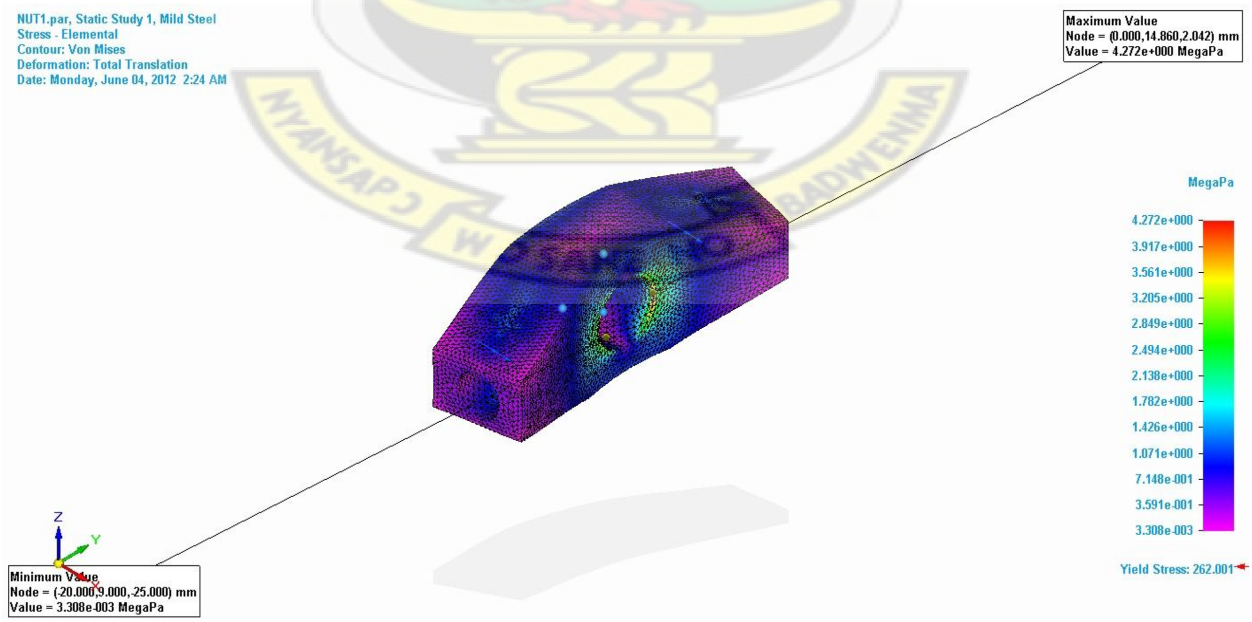
Minimum	0.000e+000 mm	-36.000 mm	14.444 mm	4.047 mm
Maximum	1.004e-003 mm	-40.000 mm	78.000 mm	0.000 mm



Total Translation

Stress Results

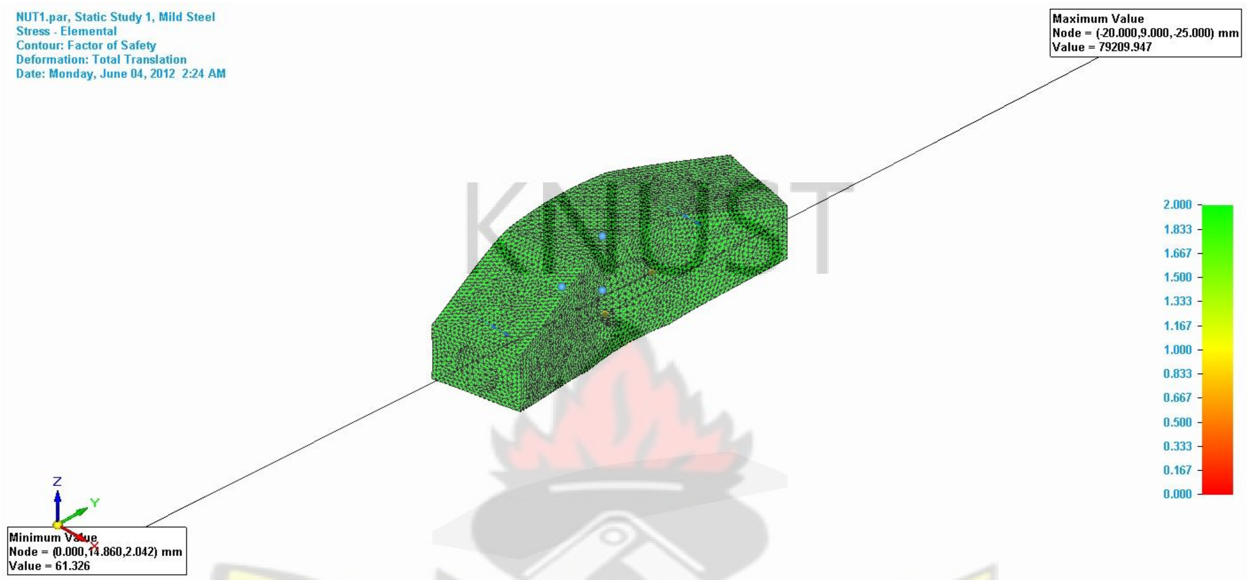
Result component: Von Mises				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.3308 MPa	-20.000 mm	9.000 mm	-25.000 mm
Maximum	42.72 MPa	0.000 mm	14.860 mm	2.042 mm



Von Mises

Factor of Safety Results

Result Component: Factor of Safety				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000	0.000 mm	14.860 mm	2.042 mm
Maximum	2.000	-20.000 mm	9.000 mm	-25.000 mm



Factor of Safety

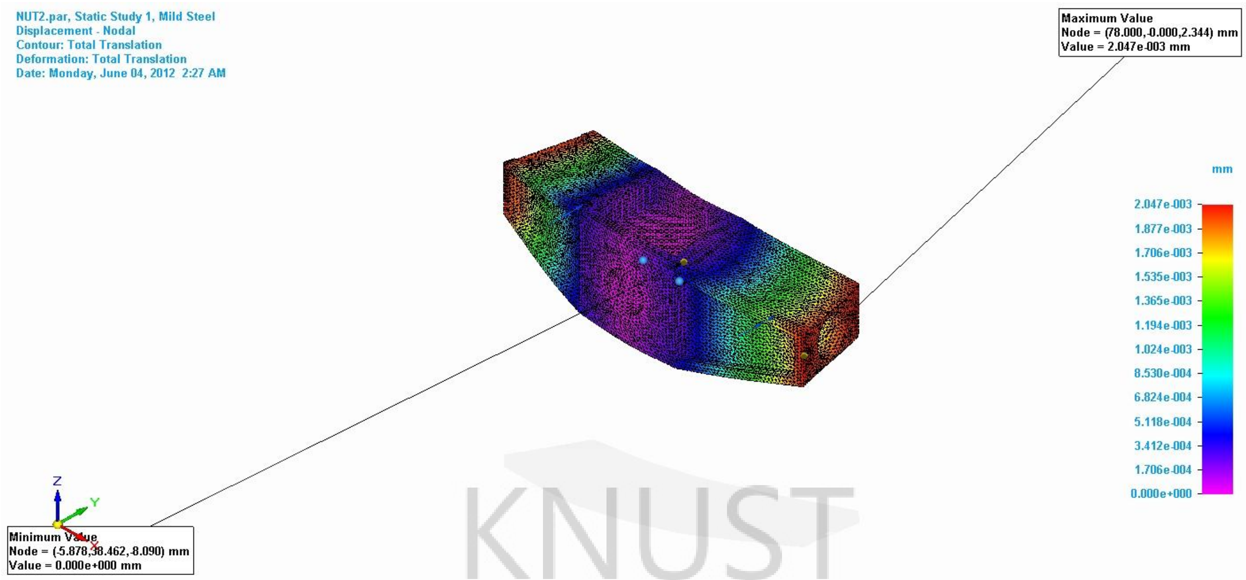
Nut Option 2

Solid Name	Material	Mass	Volume	Weight
NUT2.par	Mild Steel	1.363 kg	173572.885 mm^3	13.352962 N

Results

Displacement Results

Result component: Total Translation				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000e+000 mm	-5.878 mm	38.462 mm	-8.090 mm
Maximum	2.047e-003 mm	78.000 mm	-0.000 mm	2.344 mm



Total Translation

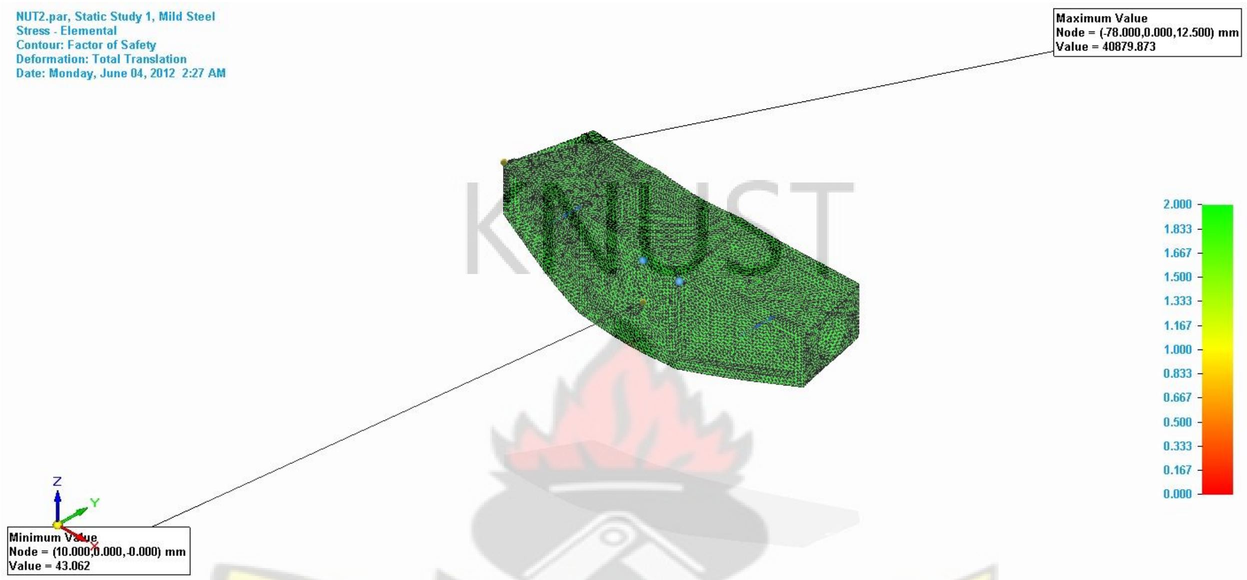
Stress Results

Result component: Von Mises				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.6409 MPa	-78.000 mm	0.000 mm	12.500 mm
Maximum	60.84 MPa	10.000 mm	0.000 mm	-0.000 mm

Von Mises

Factor of Safety Results

Result Component: Factor of Safety				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000	10.000 mm	0.000 mm	-0.000 mm
Maximum	2.000	-78.000 mm	0.000 mm	12.500 mm



Factor of Safety

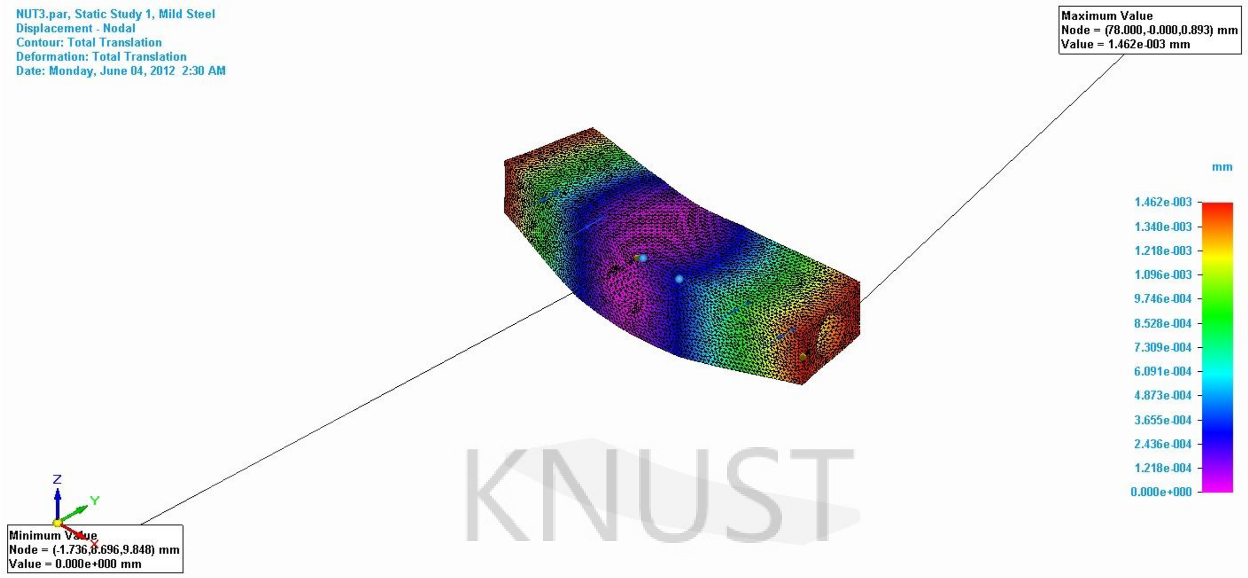
Nut Option 3

Solid Name	Material	Mass	Volume	Weight
NUT3.par	Mild Steel	1.129 kg	143870.338 mm^3	11.067945 N

Results

Displacement Results

Result component: Total Translation				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000e+000 mm	-1.736 mm	8.696 mm	9.848 mm
Maximum	1.462e-003 mm	78.000 mm	-0.000 mm	0.893 mm



Total Translation

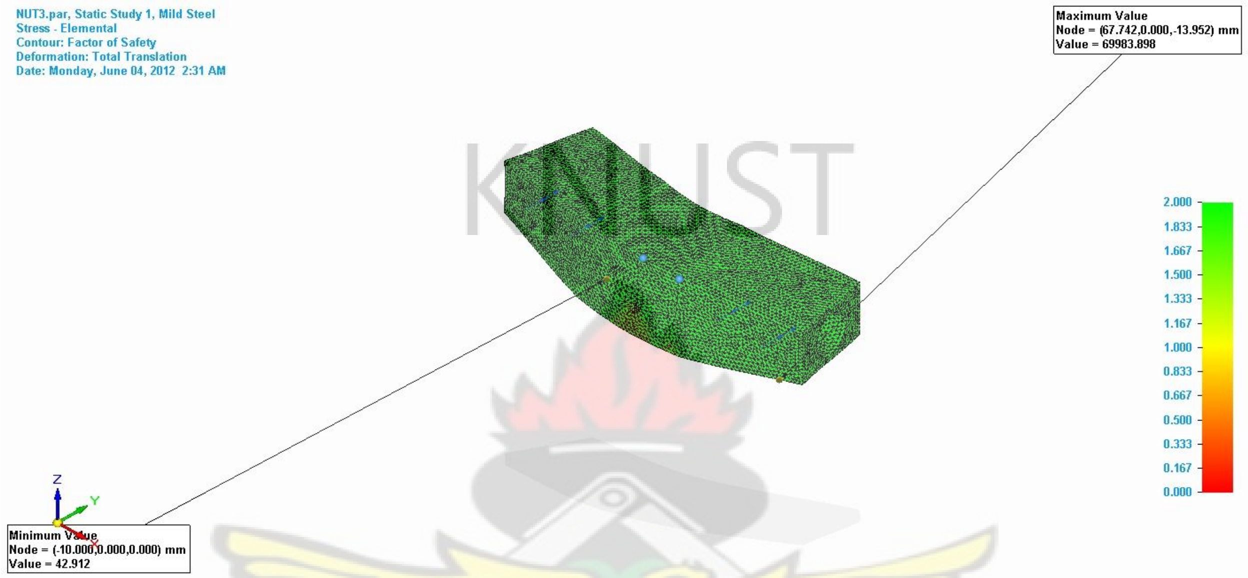
Stress Results

Result component: Von Mises				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.3744 MPa	67.742 mm	0.000 mm	-13.952 mm
Maximum	61.06 MPa	-10.000 mm	0.000 mm	0.000 mm

Von Mises

Factor of Safety Results

Result Component: Factor of Safety				
Extent	**Value**	**X**	**Y**	**Z**
Minimum	0.000	-10.000 mm	0.000 mm	0.000 mm
Maximum	2.000	67.742 mm	0.000 mm	-13.952 mm



Factor of Safety

APPENDIX E: Working Drawings of Design

Appendix E presents the production drawings of the optimum designs. This includes working drawings of the linkage, power screw, power screw nut and bin.