Weight Optimization Of A Lift-Tipping Mechanism For Small Solid Waste Collection Truck

Vitus M. Tabie, Yesuenyeagbe A. K. Fiagbe

Abstract: This paper deals with Optimization of a lift-tipping mechanism for a small solid waste collection truck. Finite element analysis was performed on a linkage mechanism that operates the tipping mechanism. 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. The weight of the linkage mechanism has been reduced from 11.6 kg to 7.5 kg which represents 35.4% reduction in weight.

Key Terms: Optimization, Finite element modelling, simulation

1 INTRODUCTION

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 [1]. 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 [2] to enable the truck directly deposit the waste into the disposal containers. Figure 1 shows the tricycle with tipping mechanism at full tipping positions.

Figure 1: Tricycle with tipping mechanism at full tipping position

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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). 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.

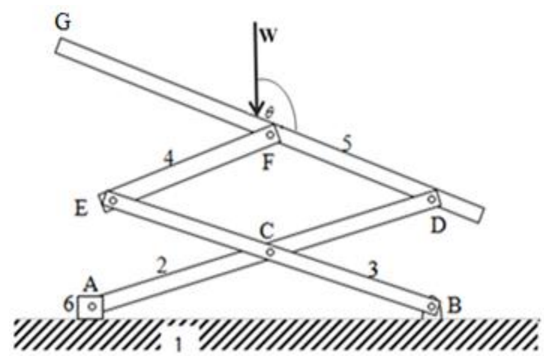

Figure 2: Schematic of kinematic linkage chain, (Source: [2])

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 380 needed to achieve full tipping at the disposal site. The weight of the lifting bar was found to be 11.6 kg.

1.1 Problem Statement

The lift-tipping mechanism for tricycles has improved the operation of the equipment [2]. 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, [3] 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, [4]. 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 linkage mechanism will help reduce the overall weight of the tricycle

2 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 the linkage mechanism using basic geometric and stress formulas and finite element analysis performed on it using Siemens Solid Edge ST3 software package, NX Nastran (7) solver. 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.

2.1 Problem Formulation

The objective is to optimize the weight of the linkage mechanism shown in Figure 2 to support a load W without structural failure.

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.

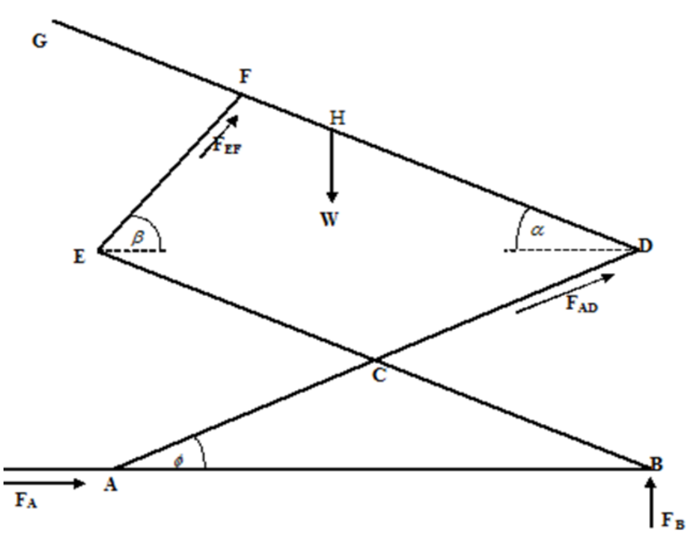

Figure 3: FBD of tipping Mechanism

Considering member GD in Figure 3

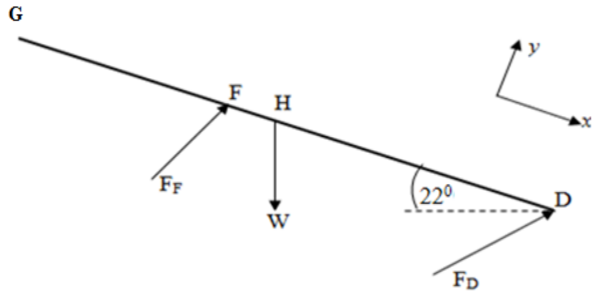

Figure 4: FBD of member GD

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, $\beta = 66^0$ (see figure 3) 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

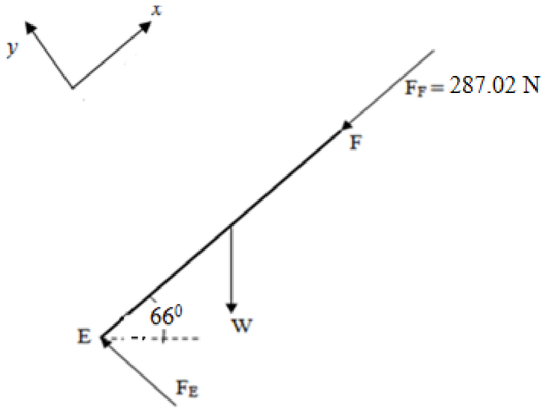

Figure 5: FBD of member EF

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

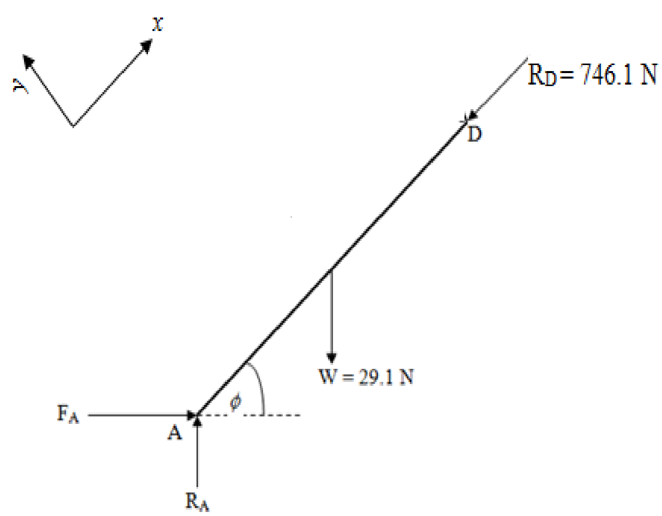

Figure 6: FBD of member AD

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

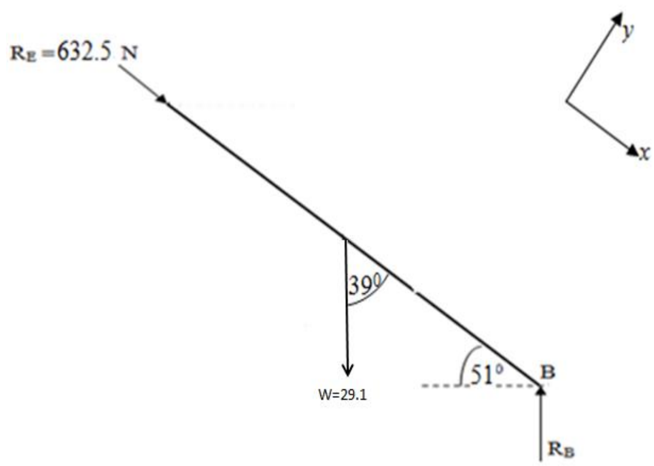

Figure 7: FBD of member BE

Summing forces in the Y direction

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

$$R_B = 36.2N$$

2.2 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

40mm × 20mm × 1.5 mm – option 1

50mm × 25mm × 1.5 mm – option 2

60mm × 30mm × 1.5 mm – option 3 (Existing design)

2.3 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 4).

3 NASTRAN SIMULATION AND OPTIMIZATION OF LINKAGE
The components were modelled 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.

3.1 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 [5]. Bolt connectors were used at the bolt holes.

3.2 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.

3.3 Loads
Using the previously determined load cases from the FBD models, the assembly model is updated to include the load vectors. Table 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 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

3.4 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 mild steel.

4 RESULTS
Summary of the results for three concepts of the linkage mechanism obtained from FEMAP are presented in table 2. From table 2, 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 was selected for linkage mechanism design

Table2: 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)

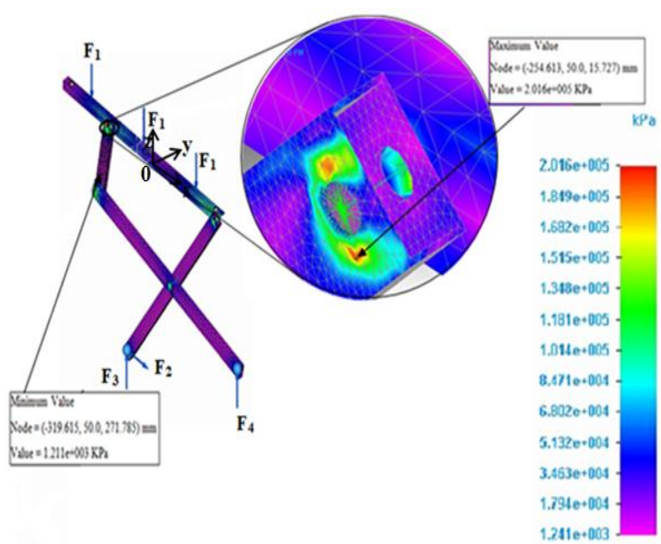

Figure 8: Simulation of Linkage Mechanism

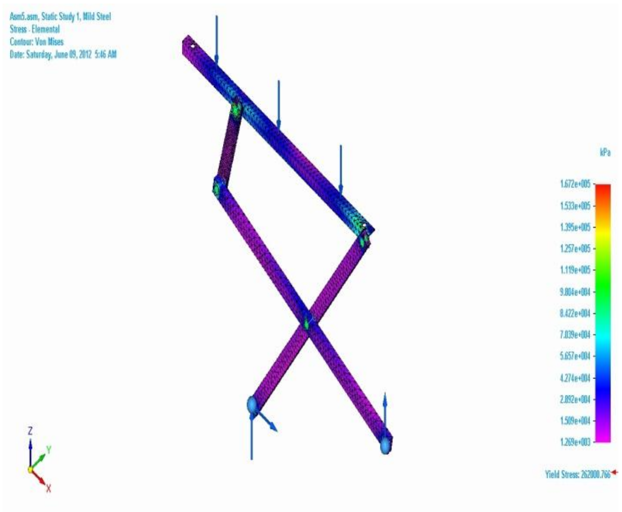

Figure 9: Simulation of Linkage Mechanism

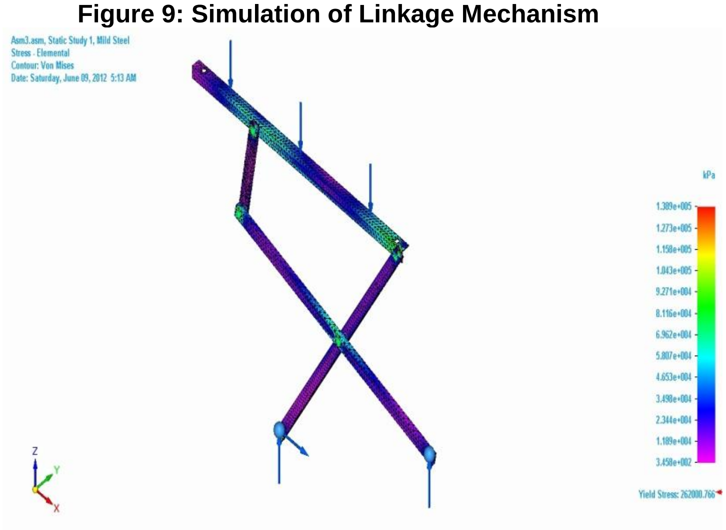

Figure 10: Simulation of Linkage Mechanism

5 CONCLUSION AND RECOMENDATION

This research was developed to optimize tipping mechanism for small tricycle trucks for solid waste collection. The weight of the linkage mechanism was reduced from 11.6 kg to 7.5 kg which represents 35.4% reduction in weight.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). 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 truck in terms of weight and torque requirement. There is however 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 sothat the design can be minimize more efficiently. Other components such as the chassis and frame extensions,the bin, power screw and nut may be optimized as well.

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