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Development of a Double Row Tractor Operated Yam Minisett Planter

A. K. Arkoh, E. Y. H. Bobobee, A. Addo

Abstract: Drudgery in manual yam minisett planting was identified as a major constraint facing yam cultivation in Ghana. The main objective of the study was to develop a double row yam minisett (DRYM) planter using Finite Element Analysis (FEA). A potato planter was adopted and modified to suit the design of DRYM planter. The main task was to perform 3D modeling of the planter major components using FEA method. Analysis of systems (ANSYS) software was used for FEA. Minisett and soil physical properties were factored into the design processes. Total deformation and equivalent (Von-Mises) stress were 0.442 mm and 7.37 MPa for hopper; 0.01 mm and 9.18 MPa for ridger bottom and that of furrow opener were $1.8^{-0.6}$ mm and 6.27 MPa, respectively. Maximum total deformation and equivalent (Von-Mises) stress were below material specification of 50 mm and 250 MPa for structural steel, and 20 mm and, 440 MPa for mild steel, respectively. The study concluded that the entire design was within the material property and permissible stress limits of the materials used. Yam planter development will enhance farmer satisfaction.

Keywords: Finite element analysis, metering mechanism, manual planting, mechanised planting.

I. INTRODUCTION

Yam (*Dioscorea spp.*), a tuber crop originated from West Africa, where about 95% of yam production obtained [1]. Ghana was the second higher yam producer next to Nigeria (64.2%) [1]. Ghana was leading by 94% of yam export from West Africa [2]. Despite several prospects in yam production, planting of yam is manual in Ghana and sub-Saharan Africa [3]. Manual planting according to [4] is extremely labour demanding, tedious and time-consuming. It also affects many practices such as plant spacing, depth and sett covering. Currently, the problem related with yam harvesting in Ghana is partly solved by Bosrotsi et al.[5] but planting of yam minisett still depends on traditional practices. To curb these challenges, there is the need to mechanize yam minisett planting. Planters have a key role to play in increasing crop productivity. To design a planter, crop physical properties were among factors considered. Buitenwerf et al. [6] considered shape of potato, belt speeds, and number of cups for planter assessment. Again, Al-Gaadi and Marey [7], concluded that ground wheel speed, different seed sizes, and shapes affect the plant spacing. The objective of the study was to design and fabricate a double row yam minisett planter using Finite Element Analysis (FEA).

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* A. K. Arkoh

A. K. Arkoh*, Automotive and Refrigeration Engineering, Takoradi Technical University, Takoradi, Ghana. Email: albert.arkoh@ttu.edu.gh

E. Y. H. Bobobee, Agricultural and Biosystems Engineering, Kwame Nkrumah University of Science and Technology, Kumasi, Ghana. Email: Emmanuel.bobobee@gmail.com

A. Addo, Agricultural and Biosystems Engineering, Kwame Nkrumah University of Science and Technology, Kumasi, Ghana. Email: ahmaddo@gmail.com

II. METHODOLOGY

A. Yam minisett physical properties measurement

To mechanize the planting of yam minisett, the potato planter was adopted and modified to suit the planting of yam minisett. The features of minisett (planting material) is a form of sector which has five (5) faces of about 80% fresh and remaining 20% skin [8], unlike potato planting material which has skin all round. Due to the nature of the sett, modification of potato planter included metering mechanism, positioning of hopper, furrow opener and coverer. Factors such as minisett and soil physical properties, angle of repose of the minisett were considered during the planter design.

Yam tubers of 82% moisture content (wet base) acquired from Crop Research Institute (CRI) Kumasi, Ghana was sliced into an average size of 50 g. **Fig. 1** shows the sett size and device developed for the determination of the angle of repose. Fifty (50) setts were randomly selected for sett physical properties determination. The measurements included the arc length (cm), thickness (cm), radius (cm), mass (g) as well as the angle of repose of the yam minisett. Arc length, thickness, and radius of the sett were taken by using venial caliper and meter rule. Electronic balance was used to measure the weight of the sett. The method of measuring the angle of repose was adopted from Awulu et al. [4].

Fig.1: Angle of repose determination

The method was modified by using a 2.5 mm mild steel plate inclined together with a graduated protractor. The yam sett was moved along the inclined material in relation to the graduated protractor and observed closely to see the angle at which the yam sett started to slide or rolls easily down the inclined structure. The angle at which set began to slide was recorded as the angle of repose for the yam sett.

B. Design Principles and Requirement of the Planter

The main features of a double-row mechanical yam minisett planter were the hopper, the metering mechanism, furrow opener, coverer, frame, and land wheels. The planter was designed to hitch to the three-point linkages of the tractor. Planter was ground-wheel driven to establish planter performance relative to tractor speed. The selection and design calculations of the planter components were based on materials specification, soil and yam physical properties. The design of yam minisett planter did not consider the orientation of sett from the hopper into the soil since results from Arkoh and Bobobee [8] confirmed that healthy sett germinates so far as soil touches the skin of the minisett. However, the following considerations were made:

1. The planter should be within the buying capacity of small-scale farmers who cultivate 1-4 acres of field.
2. The planter should be able to plant minisett sizes up to 80 g and different yam varieties.
3. Materials for the fabrication should be readily available.
4. The planter should have high planting capacity compared to manual planting
5. The planter must be able to work in sandy-loam soil

C. Experimental setup and structural analysis

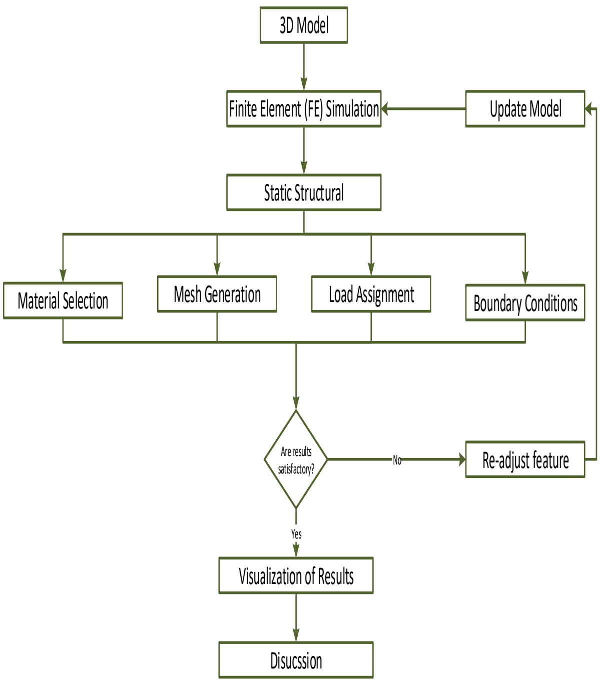

Fig 2: FEM simulation setup flow

Fig. 2 presents a FEM simulation setup flow chart for Double Row yam Minisett (DRYM) planter design processes. Data acquired from minisett, soil condition and field experiments were subjected to the planter designing. A three-dimensional geometric model of the planter assembly was created in Autodesk Inventor Professional Computer-Aided Design software (CAD) and exported into the finite element software (ANSYS). ANSYS V.18 (a commercial finite element package) was used in the study. The high productivity computing (HPC) utilized in the study using a Workstation Server Computer at the KNUST Mechanical laboratory. Finite Element Analysis (FEA) method was used to study and predict the structural analysis of the bodies under static and dynamic loading conditions. The structural prediction of the planter was done through simulation to find the concentration of total deformation and Von-Mises (Equivalent) stress.

D. Hopper design

The hopper on the planter was, formed in a trapezoidal shape to ensure a free flow of setts by the assistance of gravitational force. The type of metering mechanism selected describes the design and position of the hopper. The angle of repose of the sett informed the vertical positioned 90^0 to the horizontal and slope of the hopper was formed 68^0 to the horizontal for easy flow of setts which conforms to the angle of repose of the sett determined. The material 2.5 mm thick mild steel (AISI 1018) was used for hopper design. **equation** (1, 2 and 3) was used for hopper design. Key considerations in the design of hopper were as follows:

- Weight of yam sett in the hopper, (W_{ys})
- Bulk density of yam sett, g/cm^3, (γ_{ys})
- The inner volume of hopper, mm^3, (V_h)
- The volume of yam sett, mm^3, (V_{ys})
- Number of yam sett, (n)

$$V_h = 1.1\, V_{ys} \quad (1)$$
$$V_{ys} = W_{ys}/\gamma_{ys} \quad (2)$$
$$V_h = 1.1\, W_{ys}/\gamma_{ys} \quad (3)$$

The total mass of sett and the inner volume was multiplied by a design factor of 1.1 to give the total load exerted on the hopper and volume of hopper respectively. Hence, the maximum weight of sett was utilized in the calculation. Hopper design dimensions includes top width of the hopper 100.0 mm, bottom width 106 mm, height 800 mm, angle subtended by hopper 134.3^0, volume (Actual) (3.943 m^3) and volume (3D model 4.24 m^3).

E. Structural analysis of hopper

Material specifications of mild steel AISI 1018 for the hopper was presented in **Table 1**. The selection of material was based on material engineering properties and availability. Boundary and loading conditions values considered for hopper loading setup includes number of yam sett 80, maximum mass per sett (80 g), total mass of sett 6400 g and total load of sett exerted on the hopper 7.04 kg (69.06 N).

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Table 1. Properties and material specifications of the hopper

Mechanical Properties of Hopper Material							
Material and composition	Ultimate tensile strength (MPa)	Tensile Yield stress	Bulk Modulus	Poisson ratio	Elongation at break (in 50 mm)	Density	Young's Modulus
AISI 1018 Mild steel/Low carbon steel	440 MPa	370 MPa	140 GPa	0.290	15.0%	7870 kg/m^3	205 GPa

Source: [9]

F. Furrow opener and ridger bottom design

The type of furrow opener adopted was ridger type which gives a 'V' shape opening because of its cuts and transferring of soil sideways for easy planting [9]. The material used for the furrow opener and ridger bottom was 6 mm structural steel (A36) thick because of its strength and engineering property. Presented in **Table 2** and **3** were material specifications used on the bases of the assumed soil, furrow opener and ridger bottom specification for the design. Predicted horizontal and vertical forces on planter were determined jointly by using modified General Soil Mechanics Equation (GSME) for blades and narrow tines concept and spreadsheet prepared by [11]. According to [12], the use of spreadsheets makes the calculation of draught and vertical forces of tine or mouldboard plough simple.

Table 2: Material specifications of furrow opener and ridger bottom

Material name	Material Properties				
	Ultimate tensile strength (MPa)	Yield stress (MPa)	Bulk modulus (GPa)	Poisson ratio	Elongation at break (in 50mm)
A 36 Structural Steel	450	250	200	0.26	10%

Table 3. Static structural analysis settings utilized in the study

Static Structural analysis			
	Hopper	Furrow Opener	Ridger Bottom
Material	AISI 1018 Mild steel	A36 structural steel	A36 structural steel
Loading Type	Constant amplitude fully reversed	Constant amplitude fully reversed	Constant amplitude fully reversed
Mesh statistics	Node: 34395 Elements: 16671	Node: 3990 Elements: 1750	Node: 34395 Elements: 16671
Stress components	Von Misses Equivalent Stress (Maximum Distortion Energy criterion)	Von Misses Equivalent Stress (Maximum Distortion Energy criterion)	Von Misses Equivalent Stress (Maximum Distortion Energy criterion)
Analysis Type	Linear elastic static structural	Linear elastic static structural	Linear elastic static structural
Solver	Mechanical APDL	Mechanical APDL	Mechanical APDL
Load	69.0624	2.47 kN	-
Temperature Gravity	22°C 9.81 ms^{-2}	22°C 9.81 ms^{-2}	22°C 9.81 ms^{-2}

G. Draught and vertical forces prediction

Equation (4) and (5) were used to determine furrow opener horizontal and vertical draught forces. The inertia forces were neglected because, the planter speed was below 10 km/h, the soil was homogeneous and isotropic [13].

$$H_t = \left[\left(\gamma d^2 N_\gamma + c d N_c + q d N_q\right)\left(w + d\left(m - \frac{1}{3}(m-1)\right)\right)\right] \sin(\alpha + \delta) \quad (4)$$

Force = [(Soil factors) (Implement size)] Direction

$$V_t = -\left[\left(\gamma d^2 N_\gamma + c d N_c + q d N_q\right)\left(w + d\left(m - \frac{1}{3}(m-1)\right)\right)\right] \cos(\alpha + \delta) \quad (5)$$

where, H_t = Draught/horizontal force (kN), V_t = Vertical Force (kN), c = Cohesion between soil (kN/m^2), γ = Density/bulk unit weight of soil (kN/m^3), d = depth(m), w = width (m), α = Rake angle (degree), q = Surcharge (kN/m^2), N_γ = Gravitational factor, N_c = Cohesion factor, N_q = Surcharge factor, v^2 = speed (ms-1), δ = Interface friction angle (degree), m = rapture distance.

Determination of 'N' factors and rapture distance (m) were obtained from the 'N' factor chart, rapture distance and rake angle diagram respectively introduced by [13]. Soil properties used in the analysis (sandy loam) includes density (15 kN/m^3**)**, cohesion (10 kN/m^2), interface friction angle (22 degree) and adhesion (0) [13]**.** **T**he furrow opener parameters used in the design includes depth of opening 0.2 m, width of furrow opener 0.12 m, rake angle 45^0 and, tractor speed km/h.

I. Number of cups design

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The shape factor (S) of the yam sett was determined using **equation (6).**

$$S = 100l2wt \tag{6}$$

Where l is the length, w the width and t the thickness of the sett in mm, with $t<w<l$

Meanwhile, the number of cups were calculated using **equation** (7, 8, 9) adopted from [15]. The distance between cup to cup was assumed to be 34 cm because the metering mechanism was designed to be driven by the ground wheel which in tend was designed base on the machine's height [16].

$$I = \frac{(c \times t)}{a} \tag{7}$$

$$n = \frac{Ds}{ds} \tag{8}$$

$$c = \pi \times d \tag{9}$$

where, I = Number of cups; cm; a = cup to cup spacing, speed ratio (n), Ds = (No. of teeth on the driver sprocket ds = No. of teeth on driven sprocket, c = circumference of drive wheel, and d is diameter of the wheel.

H. Driving shaft design

Mild steel AISI 1018 material was used for shaft design because of strength and torsional resistance. Torsional torque transmitted and diameter of a shaft was calculated by using the American Society of Mechanical Engineering (ASME) code in **equation** (10 and 11) [14]. The Keyway dimension (40 x 4 x 4 mm) was provided to fit the sprocket to the shaft to facilitate the movement of the sprocket on the shaft.

The following assumptions were made for the shaft design:

1. For both tension and compression, 84 MPa was used for shafts with a keyway for maximum permissible working stresses.
2. Average shaft speed was 600 rev/min^{-1}, power transmitted by the shaft (p) = 0.02 kW
3. Shaft subjected to combined twisting and bending,
4. Ductile materials such as mild steel was used
5. Material is linearly elastic, or Hook's law is valid
6. There are no internal stresses before loading
7. Load is static
8. The safety factor was 2

$$T = \frac{60 \times P}{2\pi N} \tag{10}$$

$$d = \frac{3\sqrt{16 \ X \ T}}{\tau \ \pi} \tag{11}$$

Where, d is shaft diameter, P shaft transmitted power, τ = maximum allowable working stress (tensile or compressive) induced due to bending moment, T = Torque transmitted by the shaft and N = Speed of the shaft in revolution per minute (r.p.m).

J. Bearing and chain selection

The selection of bearing was done by comparing tolerance classifications of national standards. Chain length was calculated using **equation 12** adopted from Ranjan [17].

$$m = \frac{2c}{p} + \frac{Z_1 + Z_2}{2} + (Z_2 - Z_1)^2 / 2\pi p \tag{12}$$

Where, m = number of chain links, C = Centre to centre distance between two sprocket, m, Z_1 = Number of teeth in driver pulley, Z_2 = Number of teeth in the driven pulley, P = chain pitch (15 mm).

Assumptions

- Low power transmission
- Planting is assumed to be a low-speed operation
- Sprocket of the same size was used for the yam sett conveyor and metering shaft.

K. Bending moment determination

Shearing forces of a shaft was determined by taking moment at reactions (R_1 and R_2) and resolve using **equation** (13) and (14) for resolving reactions [15].

$$R1 \ or \ R2 = \frac{W}{2} x \ clockwis \tag{13}$$

$$R = \frac{W}{2} x - wx + \frac{w}{2} L \tag{14}$$

Where (w) is downward load acting on the sprockets, (L) is shaft total length, (x) is distance from fulcr

L. Mainframe

Weight and strength were two design factors considered in the determination of the material for the frame. Galvanize Mild steel square pipe of 4.0 x 4.0 cm and 4.0 mm thickness with rigidity and high strength properties were used due to its general engineering purposes [17]

M. Sett chute

The location of the sett chute was positioned on the outer part of the hopper by the side on which the cup-chain runs through. Mild steel sheet plate with 1.0 mm thickness was molded in a channel shape to guide cup-chain of metering mechanism and yam minisett during filling.

N. Ground wheel design

Wheels were designed for loose soils. The ground wheel diameter was selected based on the machine's height. The design of the width of the wheel was depend on the type of soil and wheel sinkage etc. Assumptions considered were;

Ground wheel diameter (D) (height of planter) = 0.8 m

Peripheral distance = $\pi D = 2.52 \ m$

Forward speed of planter = 7.2 km/h

The ground wheel covers 2.52 m horizontal distance in one revolution, at 7.2 km/h. Flat bar thickness of 3 mm was rolled to form a circle of 40 cm diameter to form a driving wheel. Sixteen (16 mm) iron rods were attached alternately throughout the circumference of the wheel to provide lugs for effective gripping of the ground surface. The arrangement of 16 mm iron rod across the inner diameter of the wheel served as spokes. The spokes were joined to a hub containing a bearing for a shaft to run through to withstand the torsional moment.

O. Planter power requirement

The tractor selection for pulling (DRYM) planter was based on the assumption that drawbar horse power (DBHP) of the tractor is about 70% of engine brake horse power (BHP) W of its engine when the machine is in good condition [18].

III. RESULTS AND DISCUSSION

A. Furrow opener, ridger bottom and hopper result

Figure 3 presents the effect of furrow opener rake angle on draught and vertical force. The mean draught (horizontal) and vertical forces on the furrow opener were 2.41 and -0.95 kN, respectively. The predicted forces imply that low horse power tractor such as category I (50 W) was able to pull the planter. The negative value indicates that furrow opener was able to penetrate in the soil. There was an increased rake angle with increasing draught and vertical forces. The result was in agreement with [14] that, to achieve a small draught force and penetration, implements must be designed with a small rake angle.

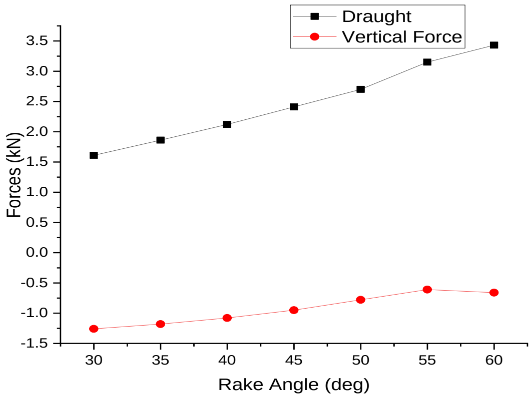

Fig. 3: Effect of rake angle on draught and vertical force

The furrow opener total deformation, equivalent (Von-Mises) stress, and maximum directional deformation (x-axis) results were 0.002 mm, 6.27 MPa and 0.0009 mm respectively. All these values were below the deformation, equivalent (Von-Mises) stress and directional deformation of 50 mm, 250 MPa and 20 mm, respectively of the selected furrow opener material (A36 structural steel). Recorded stress and deformation values were concentrated between shares and wings, and at the shares (tip) of the furrow opener respectively. This indicates that the furrow opener model was below structural steel specification used for deformation (50 mm), equivalent (Von-Mises) stress (250 MPa) and directional deformation (20 mm)

Again, ridger bottom total deformation, equivalent (Von-Mises) stress, and maximum directional deformation (x-axis) results were 0.01 mm, 9.18 MPa and 6.20 mm, respectively. The values were below the deformation, equivalent (Von-Mises) stress and directional deformation 50 mm, 250 MPa and 20 mm), respectively of the selected ridger bottom material (A36). The higher stress and deformation values (9.18 MPa) and (0.01 mm) respectively were concentrated at the shares and wings (tip) of the ridger bottom. Simulation results suggest that the ridger bottom model was within the acceptable design and permissible stress limits of structural steel (A36) used because deformation, equivalent (Von-Mises) stress and directional deformation recorded were below the properties of the structural steel used 50 mm, 250 MPa and 20 mm, respectively.

Total deformation, equivalent (Von-Mises) stress, magnitude, and force concentration on hopper, furrow opener and ridger bottom of the planter were presented in **Fig. 4, 5** and **6**. **Fig. (4a)** was the total deformation (meshed) contour plot for hopper and **(4b)** was the total deformation contour plot, **Fig. (5a)** was the total deformation (meshed) contour plot furrow opener and **Fig. (5b)** was the total deformation contour plot. Whereas **Fig. 6** was the total deformation (meshed) contour plot for ridger bottom and **Fig. 6b** was the total deformation contour plot-ridger bottom. Total deformation, equivalent (Von-Mises) stress, and maximum directional deformation (y-axis) were 0.44 mm, 7.37 MPa and 0.08 mm respectively. Stress and deformation values of 7.37 MPa and 0.44 mm respectively were concentrated at the top side of the hopper, which suggests that loading of minisett should not go beyond the point at a higher concentration for the hopper to withstand anticipated stress and deformation without deteriorating.

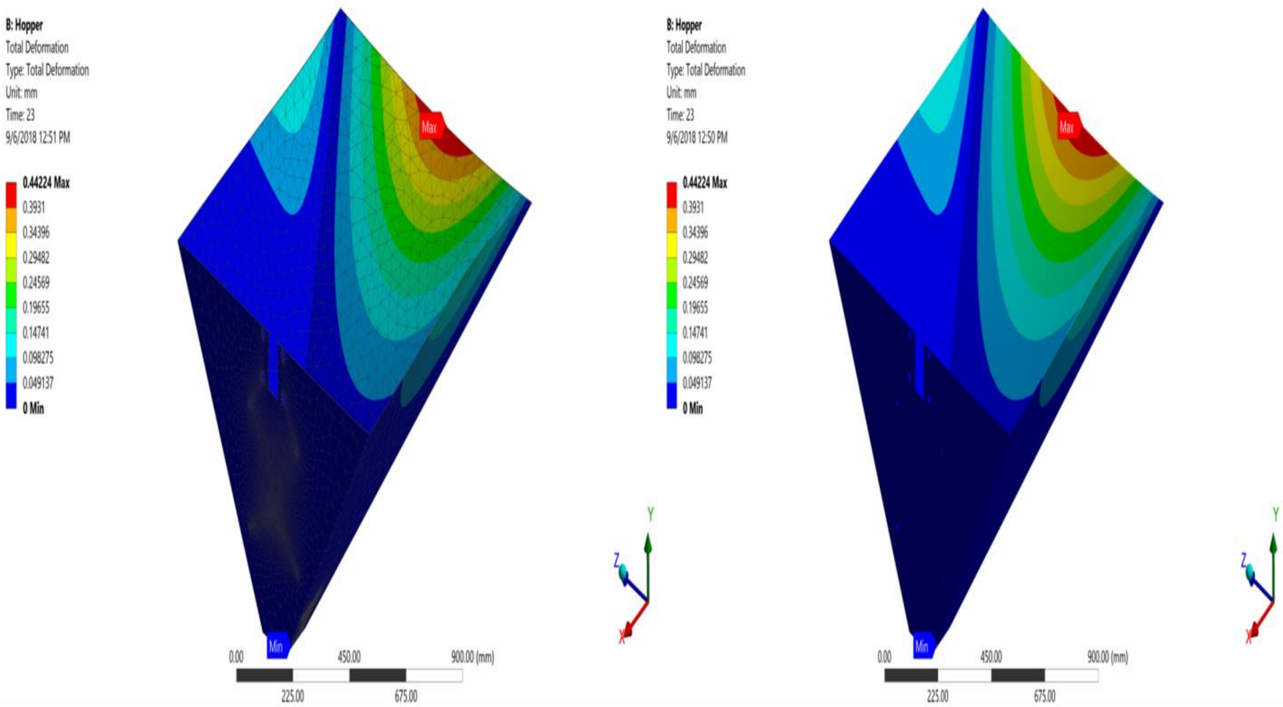

Fig. 4a. Total deformation (meshed) contour plot- hopper, b) Total deformation contour plot

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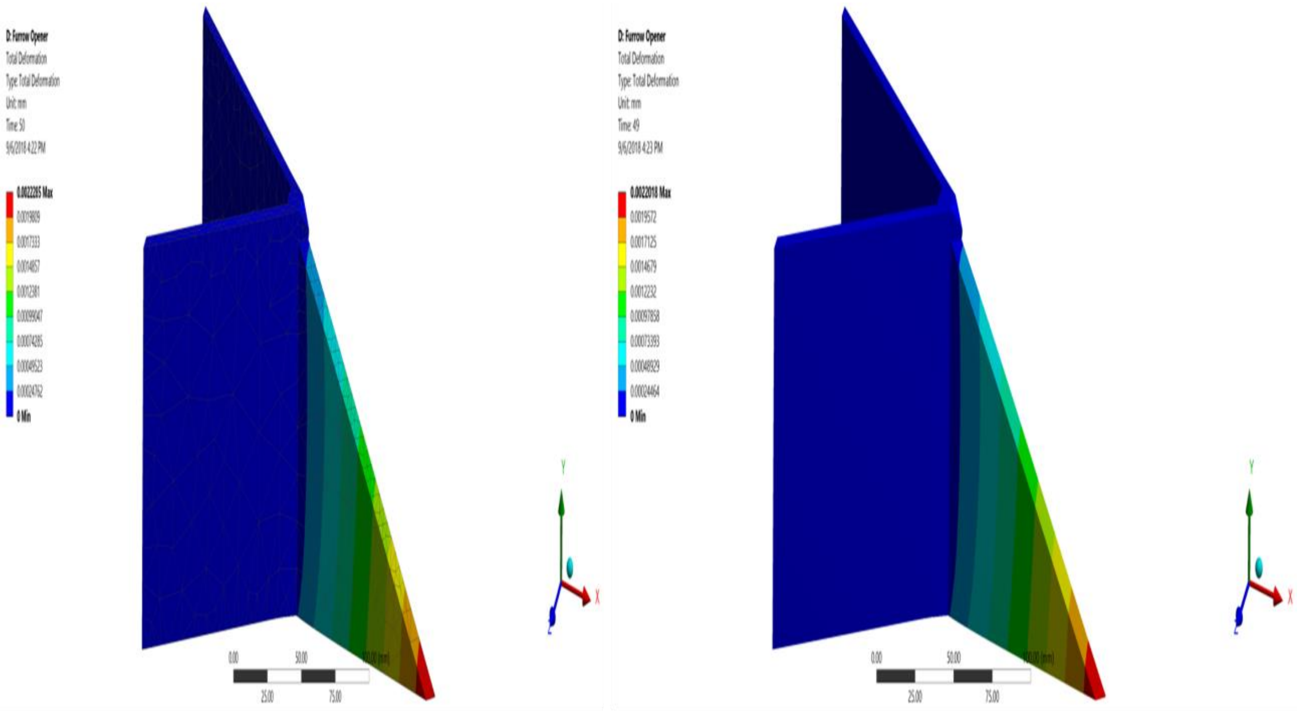

Fig. 5a. Total deformation (meshed) contour plot- furrow opener, 5b) Total deformation contour plot

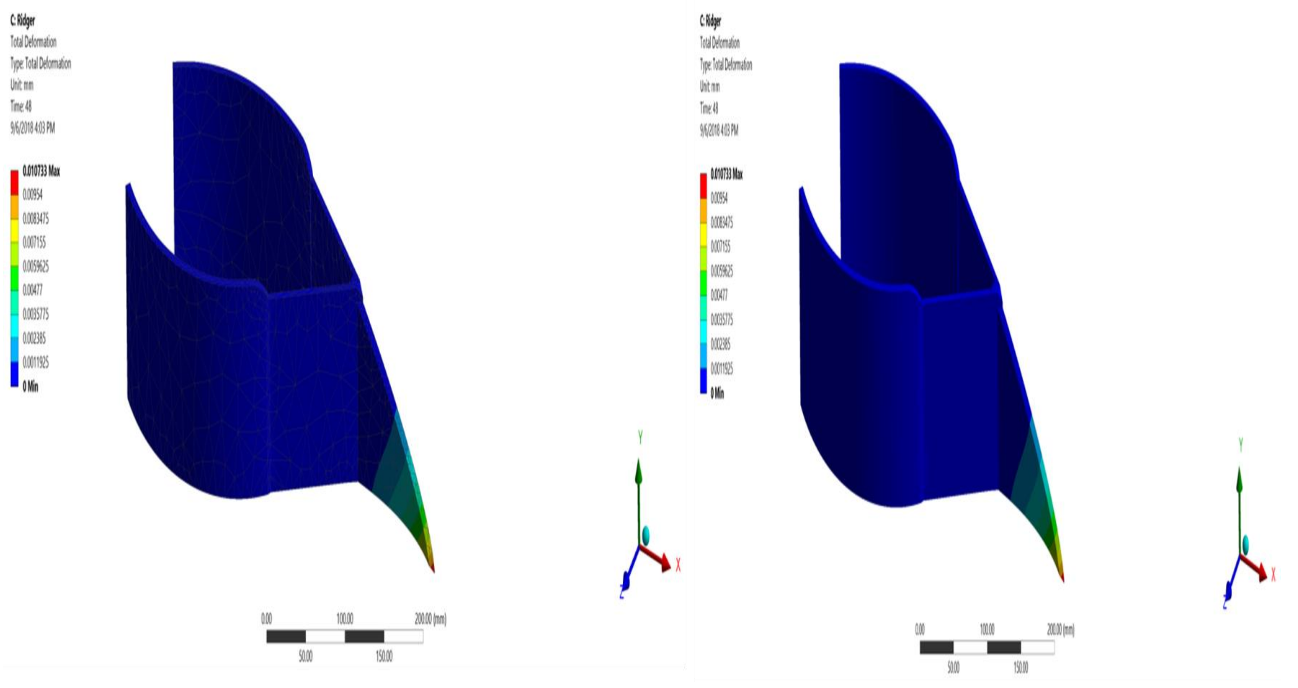

Fig. 6a. Total deformation (meshed) contour plot- ridger bottom, 6b) Total deformation contour plot

The general result suggests that the hopper model values were lower per design and material specifications for deformation (50 mm), equivalent (Von-Mises) stress (440 MP) and directional deformation (25 mm) of the selected hopper material AISI 1018. Hence the design of hopper was safe to use because [19] reported that high stress and deformation values have a shorter operational life cycle as a result of high wear and fatigue rate. Total deformation, equivalent (Von-Mises) stress, magnitude, and force concentration on the whole planter was presented in **Fig. 7 (a) and (b).** Total deformation shown in **Fig. 7a** was 15.93 mm against (50 mm) whereas equivalent (Von-Mises) stress presented in **Fig. 7b** was 75.37 MPa against (250, 440 MPa) of mild and structural steel respectively. The entire FEA result for the planter suggests that planter assembly was designed within the safe boundaries, because deformation, equivalent (Von-Mises) stress and directional deformation recorded for respective components were below the properties of the material used.

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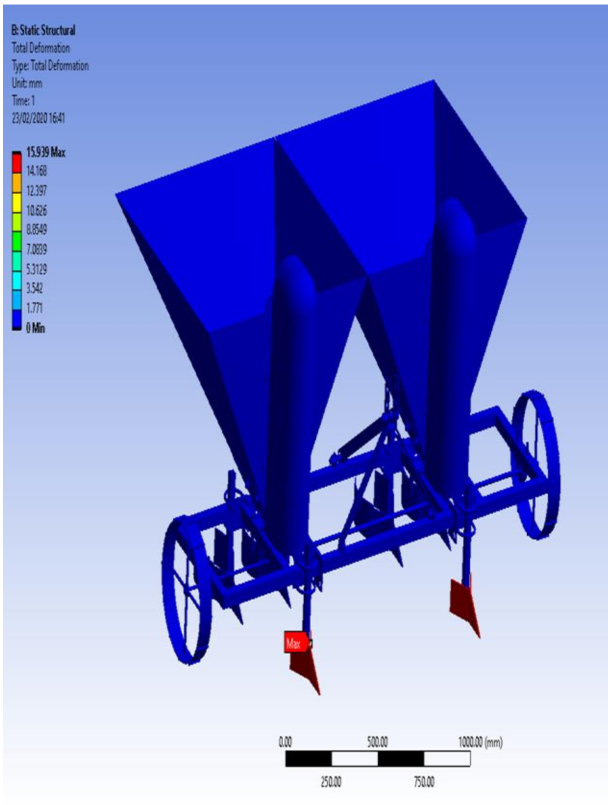

Fig. 7a. Total deformation,

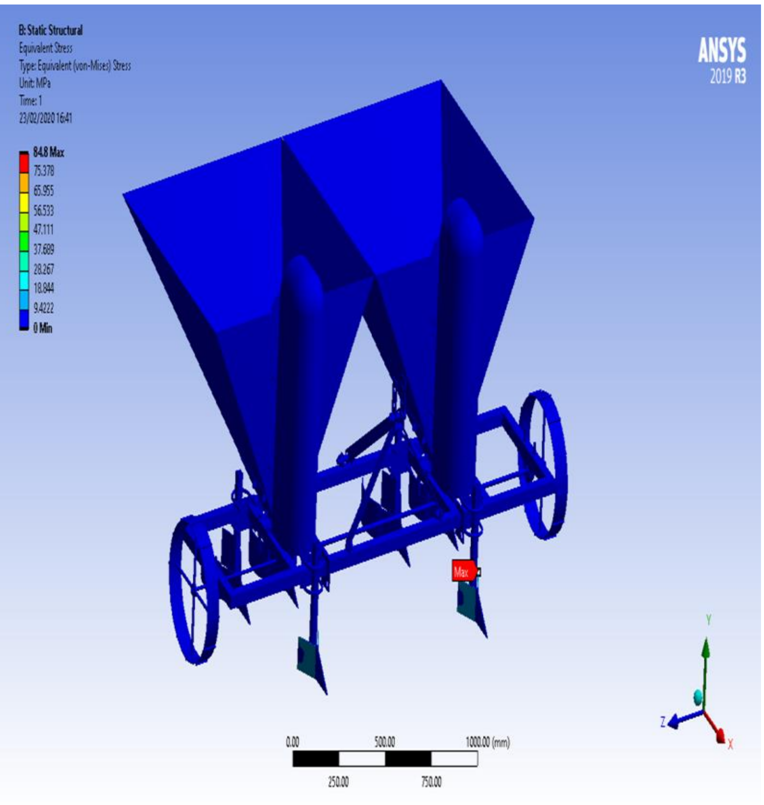

Fig. 7b) Equivalent (Von-Mises) contour plot

B. Number of cups, ground wheel and driving shaft

The cup and wheel for planter metering mechanism were presented in **Fig. 8a and 8b**. **Fig. 8**a was the cup for metering mechanism while Fig. 8b was the ground wheel for the planter. The metering mechanism consist of 5 number of cups at 35 cm interval and circumference of the drive shaft was 194.8 cm. Seed metering was cup-chain. Also, mild steel shaft of 25 mm diameter of mass 5 kg was used as transmission shaft whiles 3.18 kNm torsional torque was transmitted by the shaft. Loads supported by bearing at reactions (R_1 and R_2) were 69 kN at each supported end of the shaft. The design implies that loading beyond the calculated load may bend the shaft and fail.

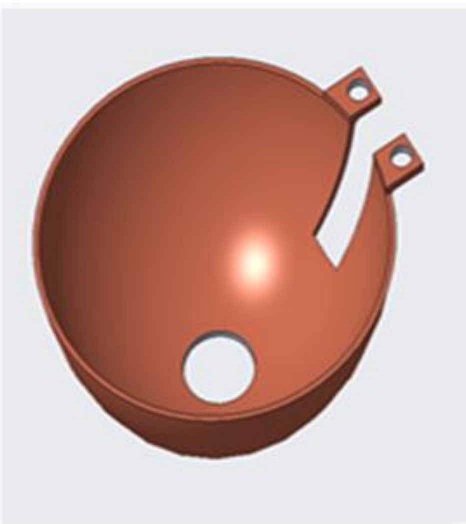

Fig. 8a. Cup for metering and **8b. Ground wheel for the planter**

C. Figure 8: Cup for metering and Ground wheel

Comparing tolerance classifications of national standards were considered 7026T1 for bearing; thrust load was 2 kN, speed: 5000 rev/min^{-1}, lubrication: grease and run time: 50 h (ISO standards and JIS B 1514). The pictorial and 3rd angle projection drawing were shown in **Fig. 9** (a) and (b), respectively.

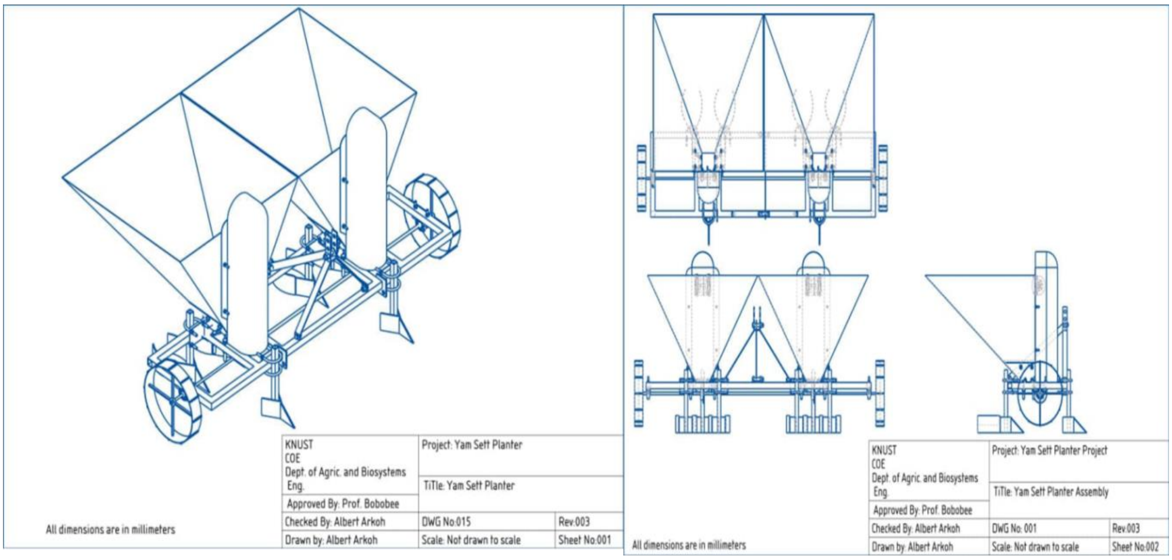

Fig. 9a: Planter Pictorial Drawing; 9b, Planter 3rd Angle Projection

D. Tractor requirement to pull DRYM planter

Drawbar Horsepower (DBHP) of tractor was 35 kW, and the size of the tractor required to pull the DRYM planter was category I of 50 hp. **Fig. 10** presets finished yam planter coupled to the tractor.

Fig. 10: DRMM planter Hitch to Category I Tractor during field trials

IV. CONCLUSION

The minisett yam planter has been developed with the following specifications: length, width, height as 177 × 54 × 130 cm and a weight of 120 kg. The predicted horizontal draught and vertical forces on the furrow opener were 2.41 and -0.95 kN, respectively. Further studies should focus on determining the field draught on the planter to validate the predicted forces to establish the real condition of loads acting on the planter.

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AUTHORS PROFILE

A. K. Arkoh (Ph. D), from Kwame Nkrumah University of Science and Technology (KNUST) Kumasi, Ghana, Department of Agricultural and Biosystems Engineering and currently lecture at Takoradi Technical University, Department of Automotive and Refrigeration Engineering.

E. Y. H. Bobobee, an Associate Professor at Kwame Nkrumah University of Science and Technology. (KNUST) Kumasi, Ghana, Department of Agricultural and Biosystems Engineering

A. Addo, A Professor at Kwame Nkrumah University of Science and Technology. (KNUST) Kumasi, Ghana, currently lecture at KNUST, Department of Agricultural and Biosystems Engineering.

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