Design and development of a tractor-drawn multi-row garlic planter

Obsa Girma[1]*, Simie Tola[2], Adesoji Olaniyan[1]

(1. Department of Agricultural Engineering, Haramaya Institute of Technology, Haramaya University, Dire Dawa, 138,Dire dawa, Ethiopia;
2. Program of Agricultural Machinery Engineering, Department of Mechanical Systems and Vehicle Engineering School of Mechanical, Chemical, and Materials Engineering, Adama Science and Technological University, Adama, 1888, Adama, Ethiopia)

Abstract: Garlic is a very significant crop that is grown all over the world, especially in Ethiopia. Garlic is widely used as a clove, for consumption and medicinal purposes. Despite wide use, many problems are not solved in the production system, among which planting technique seems the first. Due to the lack of a garlic planter system farmers still use traditional methods for planting garlic cloves this traditional method is inefficient, tiresome, labour-intensive, and time-consuming. So by providing planting machines the above problems was eliminated and improve planting efficiency and garlic productivity. The main aim of this study was to design and development of a tractor-drawn multi row garlic planter capable of planting garlic clove. The following method was used, studying physical properties of garlic clove, design all of machine component construction process, laboratory test and cost analysis. The developed tractor drawn garlic planter consists of hopper, feed cup, clove metering mechanism, furrow opener, ground wheel, three-point hitch and furrow covering device. The clove rate for seeds at different hopper filling and they were recorded were 122.86, 116.01 and 111.23 kg respectively. The meter clove were observed average mechanical damage at different hopper filling level (¼ fill, ½ fill and full fill) were 0.803%, 0.91% and 1.124% in of garlic clove respectively. The production cost of garlic planter is $ 854.46. Therefore garlic planting machine can assist the farmers to save time, reduce planting operation cost and prevent drudgery of labour. Creating a good platform for the farmers to adopt and use the garlic planter effectively and efficiently should be considered by the policy makers.

Keywords: design, development, garlic planter, cost analysis.

Citation: Girma, O., S. Tola, and A. Olaniyan. 2024. Design and development of a tractor-drawn multi-row garlic planter. Agricultural Engineering International: CIGR Journal, 26(2): 34-56.

1 Introduction

Garlic (Allium sativum L.) belongs to the alliaceous family and is a shallow-rooted vegetable crop (Gomes Viana et al., 2021). Garlic is a very significant crop that is grown all over the world, especially in Ethiopia. During the main cropping season of 2020/2021, Ethiopia produces 1.14 million quintals of garlic (Central Statistical Agency [CSA], 2021), the most of which is marketed in markets other than home consumption. Increased productivity and yield per unit area would allow farmers to earn considerable returns as a cash crop in many sections of the country. During the 2019 main crop season, Ethiopian garlic growers harvested 89.98 quintals per hectare (CSA, 2021). Ahmad et al. (2022), designed and develop seven-row tractor rear-mounted planter for garlic. By taking row width of 17.5 cm and clove to clove spacing of 10 cm, the hypothetical seed rate per hectare was 444,444 cloves per hectare. The

Received date: 2023-04-28 **Accepted date:** 2024-03-19

***Corresponding author: Obsa Girma,** Department of Agricultural Engineering, Haramaya Institute of Technology, Haramaya University, Dire Dawa, Ethiopia. Email: obsagirma9@gmail.com.

theoretical seeding mass rate was calculated to be 3.82 q ha^{-1} for Desi variety and 15.6 q ha^{-1} for Chinese variety that was near to suggested seed rate (4-7 q ha^{-1} for desi and 15-17 q ha^{-1} for Chinese). Kushwaha et al. (2020), design, develop, and assess the effectiveness of a push-style manually controlled garlic planter. The result for Hill to hill spacing, seed placement depth, seed density, number of seeds per hill, soil cover over the seed, missing hills, operating speed, and field capacity were 7.36 cm, 4.98 cm, 1.1%, 4.98%, 13.46%, 3.31 km h^{-1}, and 0.0367 ha h^{-1}, respectively. Zilpilwar and Yadav (2020) develop tractor-driven garlic clove planters with miss index, multiple index, quality of feed index, mechanical clove damage, the planters' effective field capacity and field efficiency are 3.64%, 5.64%, 90.72%, 5.70%, 0.32 ha h^{-1}, and 79.02%, respectively. The lack of a garlic clove planter in Ethiopia, on the other hand, has an impact on garlic production because traditional methods for planting garlic seeds are inefficient, tiresome, labour-intensive, and time-consuming, causing serious backaches for farmers, high production costs, and manual planting cannot achieve planting uniformity, resulting in non-uniform growth. As a result, employing a garlic planter machine, production maximization and labour drudgery are achieved by minimizing these issues. To overcome this problem, it is vital to design, construct a garlic planter machine that's feasible and straightforward to function. The aim this study is to design and development of a tractor-drawn multi row garlic planter machine that's accessible to provincial ranchers who develop garlic.

2 Materials and methods

2.1 Description of study area

The design, fabrication and performance evaluation of the tractor-drawn multi-row garlic planter was conducted at Awash Melkassa Agricultural Research Centre (AMARC) in 2021 GC, which is located at 117 Km from Addis Ababa city. The geographical location of the research centre is an altitude of 1466 m above sea level and lies on the geographical coordinates of 8°24' 0" N, 39°20' 0" E Latitude and Longitude point, respectively. The mean annual rainfall ranges between 500 mm to 800 mm and the average annual minimum and maximum temperatures are 12°C and 36°C respectively.

2.2 Overall structure of the garlic planter machine

As shown in Figure 1, the main components of the garlic planting machine include frame, clove hopper, clove metering mechanism, drive mechanism, furrow opening, clove covering devices and three point hitch.

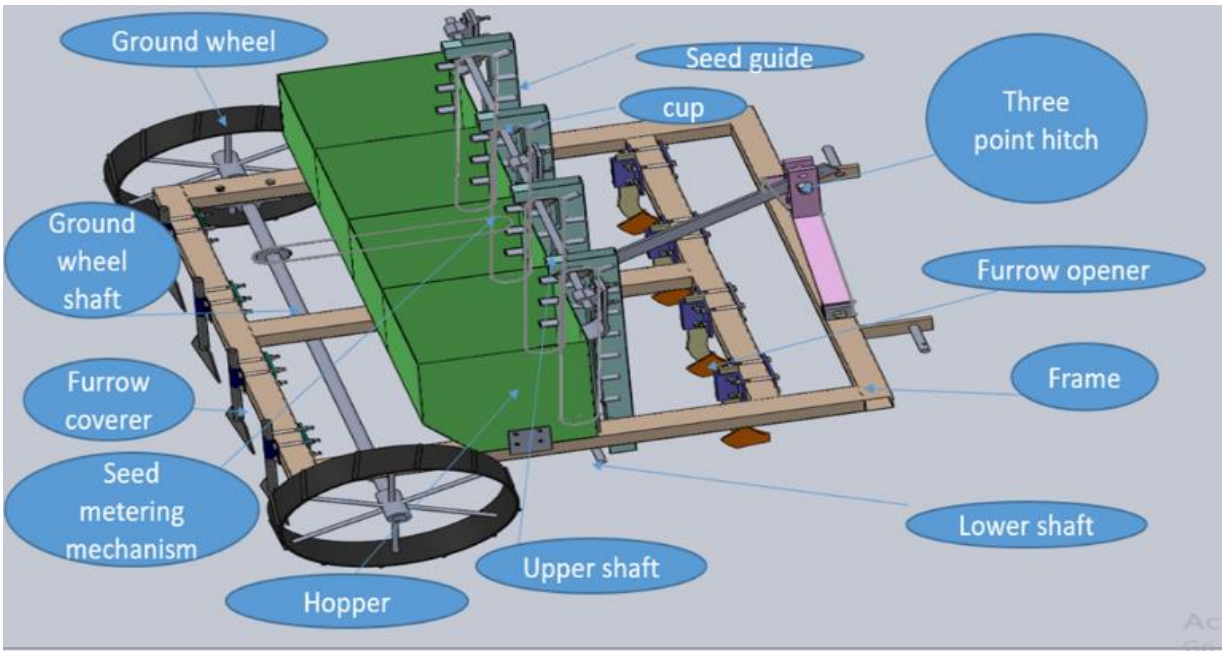


Figure 1 Overall structure of garlic planting machine

2.3 Garlic clove physical and mechanical properties

The physical properties of garlic clove are an important factor for the design of garlic planter. Gojjam and minjar varieties were selected for the study. The physical properties of garlic namely: geometric mean diameter, surface area, shape index, bulk density and angle of repose.

The geometric mean diameter of the garlic cloves was determined as follows (Garnayak et al., 2008).

$$Dg = \sqrt[3]{LWT} \tag{1}$$

Where: L = length (mm); W = width (mm); T = thickness (mm).

The surface area of the cloves was determined as the area of a sphere of the same geometric mean diameter as follows (Bakhtiari et al., 2011).

$$A_s = \pi \times Dg^2 \tag{2}$$

Where: A_s= surface area, (mm^2); D_g=geometric diameter, (mm).

Shape index is used to evaluate the shape of garlic cloves and it was calculated according to the following Equation 3 (Abd-Alla, 1993).

$$Shape\ Index = \frac{W}{\sqrt{LT}} \tag{3}$$

The garlic clove is considered as oval if the shape index>1.5, on the other hand, it is considered spherical if the shape index < 1.5.

The bulk density (ρb) of the cloves was determined using the standard test weight procedure by filling a container of 30 cm height and 10 cm diameter with the garlic cloves from a height of 15 cm from the top surface of the container at a constant rate and then weighing the cloves (Özgüven and Vursavuş, 2005).

$$\rho b = \frac{Ms}{Vc},\ \text{Kg/m}^3 \tag{4}$$

Where: Ms =total mass of cloves in the container, (Kg); Vc = volume of the container, (m^3).

The angle of repose of garlic clove was measured by the emptying method, to determine the dynamic angle of repose by using an open-ended cylinder of 15 cm diameter and 50 cm height. The cylinder was placed at the centre of a circular plate having a diameter of 70 cm and filled with a clove. The cylinder was raised slowly until it formed a cone on the circular plate. The angle of repose was calculated using equation (Karababa, 2006):

$$\theta = tan^{-1}\frac{2H}{d} \tag{5}$$

Where: θ= angle of repose, empty or filling (deg.), H= height of the cone, (cm); d=diameter of the cone , (cm).

2.4 Design computations for the machine components

2.4.1 Drawbar horsepower

To design the planter first of all, working width of the implement was calculated according to Sharma and Mukesh (2019). Draw bar horse power (DBHP) is the 60 % of brake horse power (BHP). The available horsepower of a 4WT is 25 hp, Therefore, We know 4WD tractor draw bar horse power (DBHP) = 60% of BHP. Therefore, DBHP = 25 hp × 0.6 = 15 hp.

The tractor draw bar horse power (*DBHP*), draught available (D_f) was calculated by equation (Sharma and Mukesh, 2019).

$$DBHP = \frac{D_f * V}{270} \tag{6}$$

Where: D_f = draft force, kg; V = speed, km hr^{-1}.

Generally the speed of garlic planting was done on the range 2 -5 km hr^{-1} (Knapp et al., 2011).

Let the forward speed of planter be 3 km hr^{-1}.

Putting the values in the Equation 6, Draught available for planting= $\frac{15\,hp * 270}{3\,km\,hr^{-1}}$ = 1350 N.

2.4.2 Draft on furrow openers

Draft of one furrow opener was determined as follows (Sharma and Mukesh, 2019).

De=unit draft × cross sectional area of furrow opener, N (7)

But to find the cross section area of furrow opener

$$t_o = 2a \times tan\theta + B_o + \triangle t \tag{8}$$

Where: t_o = the distance between the furrow openers, cm; a = depth of furrow, cm; B_o = width of shank, cm; $\triangle t$ = clearance between two furrows, cm.

to=2× 6 × tan30 °+4+ 19.07 30 cm

Width of furrow opener = $2a_{max} + B_o$ (9)

Width of furrow opener = 2×6 + 4 = 16cm

Cross sectional area of furrow opener =

$$\frac{depth \times width}{2} = \frac{6cm \times 16cm}{2} = 48\ cm^2 \qquad (10)$$

Then, draft on one furrow opener was determined by Equation 10 above.

D_o= *unit draft × cross sectional area of furrow opener*

D_o= 0.3 kg cm^{-2} × 48 cm^2 × factor of safety = 0.3 kg cm^{-2} × 48 cm^2 × *2 = 28.8 kgf = 28,8 N*

Number of furrow openers = $\frac{\text{Total draft}}{draft\ \ on\ one\ tine}$ = $\frac{1350}{28.8}$ = 46.8

But designing of 46 furrow opener (46 rows) of garlic planter will have a buckling effect and will also create problem in transportation and handling. Therefore, it is better to design a garlic planter with 4 rows seed and 30cm spacing between furrow openers.

Working width of garlic planter = 4 furrow opener × 30 cm spacing =120 cm

The designed machine has four rows garlic planter with different components.

2.4.3 Design of main frame

The mainframe was constructed from mild steel square pipe. Design was based on torsion and bending due to induced draft. Material of the main frame was selected based on weight and required strength. The frame is the skeletal structure of the planter which forms the platform on which other components are mounted. It was selected for its light weight but strong enough to withstand the imposed loading.

The total draft was given by Equation 11 (Bosoi et al., 1987).

$$Total\ draft = k \times n \times a \times b \qquad (11)$$

Total draft = (0.3 × 4 × 6 × 16) = 115.2

Total draft × factory of safety = 115.2 × 2 (factory of safety for M.S. =2) = 230.4 kg

Where: k= unit draft (kg cm^{-2}); n= number of furrow opener; a=depth of furrow (cm); and b= width of furrow (cm).

Torque produced on the square bar will be estimated by:

Torque on the square bar = draft × ground clearance. But, Ground clearance = 50 cm, then

T = 230.4 kg × 50 cm = 11520 kg cm

In addition to the torque, the bending moment would also be produced. The bar was considered as a simply supported beam on the frame in between the furrow openers. To estimate the maximum bending moment will be given by Equation 12 (Bosoi et al., 1987).

$$M_{max} = W \times L \qquad (12)$$

M_{max}= 230.4 kg × 150cm = 34560 kg cm

Where: M_{max} = maximum bending moment, W= total draft (total weight on frame), L= total length of the frame.

Equivalent torque due to torsion and bending moment is also estimated by Equation 13 (Khurmi and Gupta, 2005).

$$Te = (M\max^2 + T^2)^{1/2} \qquad (13)$$

Te = $(34560^2 +\ 11520\ ^2)^{\frac{1}{2}}$= 48875.22 kg-cm

Where: Te= equivalent torque, (kg cm), T= torque on the bar, (kg cm), M_{max} = Maximum bending moment (kg cm).The maximum shear stress and polar moment of inertia developed at the centre of the tool frame was obtained by Equations 14 and 15 (Khurmi and Gupta, 2005).

$$\frac{fs * R}{4d} = \frac{Te}{I} \qquad (14)$$

Where: f_s = shear stress at any section, R = distance of the section from neutral axis (assume = 10 cm), Te = torque produced, I = polar moment of inertia, t = 9.6.

Assume that, maximum working stress of 1120 $\frac{\text{kg}}{cm^2}$ was occurred at the centre of the frame. For square section having each side measuring d.

$$I = Polar\ moment\ of\ inertia = \frac{d^4}{9.6} \qquad (15)$$

The factor of safety was taken as 4, Fs is 1,120 kg cm^{-1}.

$$\frac{1}{4}\frac{1120*10}{d} = \frac{48875.22}{\frac{d^4}{9.6}} = \frac{11200}{d*4} = \frac{48875.22*\ 9.6}{d^4} = d^3 = \frac{1876808.448}{11200} = 5.5\text{ cm} = 6\text{ cm} = 60\text{ mm}$$

Therefore, considering availability of next higher section the square bar section 60 mm×60 mm×3 mm was selected to make the frame of garlic planter.

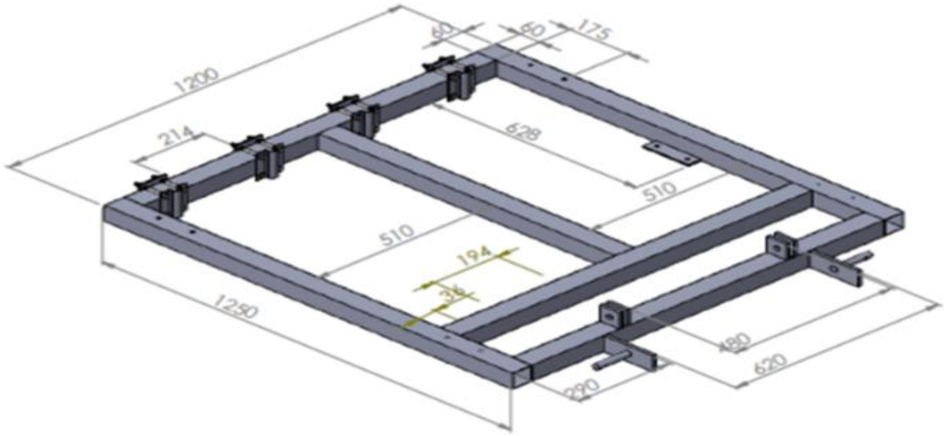


Figure 2 Garlic planter frame components

2.4.4 Design of clove hopper

On the basis of reviews and studies of physical properties of Gojjam and Minjar varieties garlic clove, the average bulk density of garlic clove was 535.7 kg m^{-3} and the angle of repose was $26°$ had taken for designing the clove hopper. Trapezoidal shape of cloves box is used in the machine for free flow of clove. The capacity of the hopper was determined by considering the following factors (Sharma and Mukesh, 2019).

Agronomic requirement garlic variety:

- Row to Row distance: 30 cm

Clove rate (assumed): 100 -140 kg ha^{-1}

- Bulk density of garlic clove: 535.7 kg m^{-3}

The working width of garlic planter

$$W_w = N \times S \quad (16)$$

Where: W_w = working width of the planter, (m), N = number of furrow openers use, S = distance between two consecutive furrow opener, (m)

W_w = 4 × 30 cm = 120 cm = 1.2 m

$W_p = W_w + 2b = 1.2 + 2 \times 0.15 = 1.5$

Then length of clove box (L_b) = $W_p - 2b$

Where: L_b = the length of clove box, (m); W_p = width of the planter, (m); b = distance between the side box wall and ground wheel, (m).

L_b = 1.5 m – 2 × 0.15 = 1.5 – 0.3 = 1.2 m

Let, take field efficiency of garlic planter = 75% = 0.75

The, actual field capacity of garlic planter.

$$AFC(\frac{ha}{hr}) = \frac{speed(\frac{km}{hr}) \times working\ width\ of\ planter(m) \times field\ efficiency\ (\%)}{10} \quad (17)$$

$$AFC = \frac{3\frac{km}{hr} \times 1.2m \times 0.75}{10} = 0.27$$

Clove mass = clove rate (kg hr^{-1}) × area covered (ha hr^{-1}) × time (hr)

Clove mass = 140 kg hr^{-1} × 0.27 ha hr^{-1} × 2 hr = 75.6 kg

Garlic clove weight are 75.6 kg and the angle of repose of garlic clove was 25.5°, therefore the angle of the inclination of the side wall of a hopper should be more than the angle of repose of the clove for easy flowing of the clove 30°. The volume of hopper was calculated by Equation 18 (Sharma and Mukesh, 2019).

$$Volume\ of\ hopper = \frac{Weight\ of\ garlic}{bulk\ density\ of\ garlic} \quad (18)$$

$$Volume\ of\ hopper = \frac{75.6\ kg}{535.7\ kgm^{-3}} = 0.14m^3$$

Consider spillage losses of 10%. Therefore total volume of hopper.

Volume of hopper = 0.14 m^3+ 0.014= 0.154 m^3

$$V_h = \left(\frac{(a+b)}{2} \times h\right) \times lb \quad (19)$$

Where: V_h = volume of the seed hopper having trapezoidal section, (m^3); a = bottom width of the hopper, (m); b = top width of the hopper, (m); h = height of seed hopper, (m); lb = length of the seed hopper or box, (m), θ = angle inclination of hopper.

Let, take trapezoidal bottom width is = 25 cm =0.25 m.

$b = a + 2l$

$$V_h = \left(\frac{a+a+2l}{2}\right) \times h \times lb$$

But, $\frac{h}{l} = tan\theta$, $l = hcot\theta$

$$V_h = \left(\frac{2a + 2h\cot\theta}{2}\right) \times h \times lb = (a + h\cot\theta) \times h \times lb$$

$0.154\ m^3 = (0.25\ m + h\ cot30) \times h \times 1.2\ m$

$h = 0.4\ m$

Then, $V_h = \left(\frac{(a+b)}{2} \times\ h\right) \times lb$

$0.154\ m^3 = (\frac{(0.25\text{m} + \text{b})}{2} \times 0.4) \times 1.2$ m

b = 0.513 m

Specification of the hopper are: V_h = volume of the seed hopper having trapezoidal section = $0.165m^3$, a = bottom width of the hopper = 0.25 m, b = top width of the hopper = 0.513, h = height of seed hopper = 0.4 m, L_b = length of the seed hopper or box =1.2 m, θ = angle inclination of hopper= 30°, Thickness of seed hopper = for mild steel sheet 2 mm.

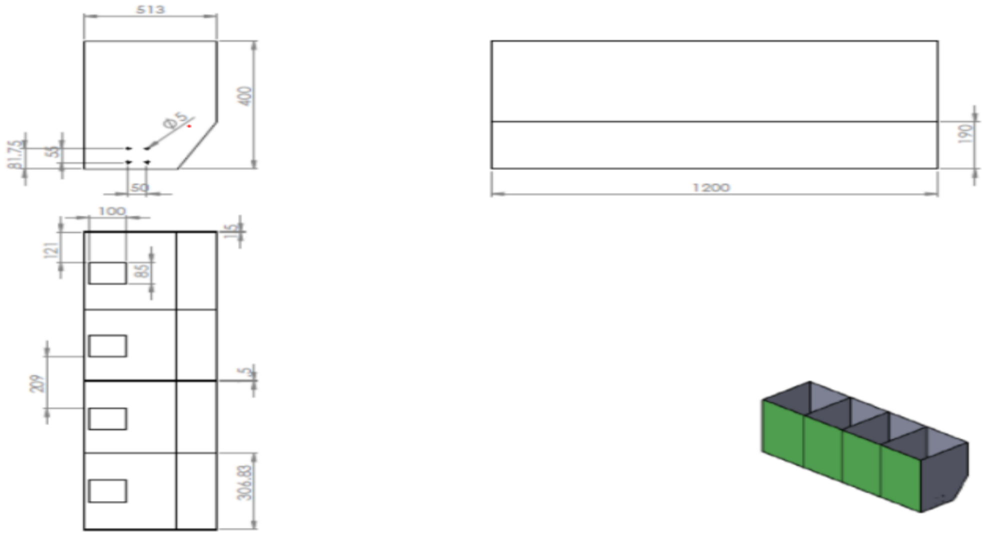


Figure 3 Clove hopper

2.4.5 Design of furrow openers

To design the furrow opener available draft from the tractor that exerted on the tip of furrow openers on the length of furrow openers was determined by Equation 20 (Bosoi et al., 1988).

$$Total\ draft = k \times n \times a\ x*b \quad (20)$$

$Total\ draft = 0.3 \times 4 \times 6 \times 16 = 115.2\ kg$

$$Each\ furrow\ opener = \frac{115.2}{4} = 28.8\ kg$$

Bending moment = draft x length of the shank

= 115.2 kg × 50 cm = 5760 kg cm

Bending stress was determined by Equation 21 (Bosoi et al., 1988).

$$f = \frac{M \times C}{I} \quad (21)$$

Where: f = bending stress, (kg m^{-2}); I = moment of inertia, (m^4); M = bending moment, (kg.cm); C = distance from the neutral axis to the point at which stress is determined, (cm).

The section modulus axis was computed by Equation 22 (Bosoi et al., 1988).

$$Z = \frac{M}{f} \quad (22)$$

Assuming bending stress is equal to 1120 kg cm^{-2}.

$$Z = \frac{5760\ kg \cdot cm}{1120\ kg \cdot cm^{-2}} = 5.14\ cm^3$$

Where: Z= section modulus of the furrow (cm^3), M= bending moment, (kg cm)

F= bending stress, (kg cm^{-2})

The width of the shank was determined by Equation 23 (Bosoi et al., 1988).

$$Z = \frac{bh^2}{6} \quad (23)$$

Where: Z= section modulus of the furrow, (m^3), b = width of shank, (cm), h = height of shank, (cm)

Width of shank was considered as 1:4, i.e. (h: 4b) (Bosoi et al., 1988).

$$Z = \frac{b(4b)^2}{6} = \frac{16b^3}{6} = 5.14\ \text{cm}^3$$

b= 1.24 cm = 12.44 mm

Considering the factor of safety and availability of material is standard size.

The thickness of shank furrow opener was selected = 8 mm

Therefore, the width of the shank = 12 mm

Height of shank = 4 ×12= 48 mm

Therefore, a Mild steel flat Tyne of 48 × 12 mm size was selected.

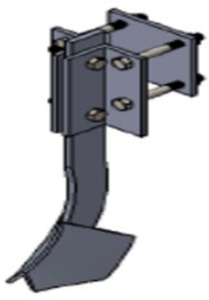


Figure 4 Furrow opener

2.4.6 Design of clove metering mechanism

In designing the clove metering cups, the feed cup type clove metering was selected for the machine for sowing of garlic clove The dimensions of each feed cup were decided by maximum size of garlic clove from Gojjam and Minjar varieties can be put in the feed cups considering the mean geometric diameter of the garlic clove and shape cup was determined based on shape index of garlic clove.

The number of cups on the metering cup (n) was determined by Equation 24 (Sharma and Mukesh, 2019).

$$n = \frac{\pi \times D}{Nr \times S} \quad (24)$$

Where: n = number of cups on the metering chain, D = diameter of transmission wheel, (cm).

Nr = Speed ratio, transmission wheel to cup, S = spacing required between two cloves (cm), But, the speed ratio of ground wheel to metering mechanism is,

$$Nr = \frac{\pi D}{Ns * S} \quad (25)$$

Take ground wheel diameter = 52 cm

Circumference of the ground wheel = πD= $\pi \times 0.52m = 1.63\ m$

Number seed required in one revolution of ground wheel (Ns) = $\frac{\pi D}{S}$ =$\frac{1.63m}{0.1\ m}$ = 16.3.7 = 16 seeds

$$Nr = \frac{\pi D}{Ns \times S} = \frac{1.63m}{16 \times 0.1\ m} = \frac{1.63m}{1.6} = 1.0.$$

The speed ratio = 1

Then, number of cups will be $n = \frac{\pi \times D}{Nr \times S} = \frac{1.63m}{1 \times 0.1\ m}$ = $\frac{1.63m}{0.1m}$ = 16.3= 16

Therefore, sixteen cups were provided on the periphery of each metering chain. The metering chain has 16 numbers of cups on its periphery.

The feed cup seed metering mechanism was selected for sowing garlic cloves, The diameter of clove metering was determined by Equation 26 (Sharma and Mukesh, 2019).

$$dr = \frac{Vr}{\pi Nr} \quad (26)$$

Where: dr = diameter of seed metering, (m), Vr = peripheral velocity, (m min^{-1}), Nr = rpm of metering mechanism

In one revolution = 1.63 meter is covered

The speed of tractor = 3 km hr^{-1} = 50 m min^{-1} = 0.833 m s^{-1}

Then, $\frac{50\ m/min}{1.63\ m/revelotion}$ = 30.67 rpm = 30 rpm

$$dr = \frac{50m/min}{\pi * 30\frac{revelotion}{min}} = 0.5 \text{ m} = 0.5 \text{ m}$$

The spacing between consecutive cups was calculated by Equation 27 (Sharma and Mukesh, 2019).

$$t = \frac{\pi dr}{n} \quad (27)$$

$$t = \frac{\pi * 0.5}{16} = 0.1 \text{ m}$$

Spacing between consecutive cups is = 0.1 m

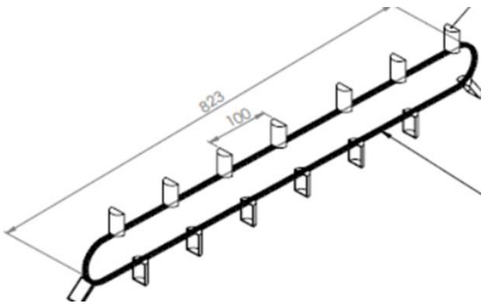


Figure 5 Clove metring mechanism

The gear ratio was determined using Equation 28 (Sharma and Mukesh, 2019).

$$\text{Gear ratio} = \frac{number\ of\ teeth\ on\ sprocketof\ seed\ metering\ shaft}{number\ of\ teeth\ on\ sprocket\ of\ ground\ wheel} \quad (28)$$

$$Gear\ ratio = \frac{14}{14} = 1:1$$

Based on Knapp et al. (2011) recommendation operation speed of tractor drawn garlic planter was 2 up to 5 km h^{-1}.

- Speed selected for design 3 km h^{-1}.
- Number of teeth on sprocket of ground wheel was 14.
- Number of teeth on sprocket of seed metering shaft was 14.

The dimension of each feed cup was selected based on the average size of garlic clove from varieties that can be put in the feed cups considering the mean geometric diameter of garlic and the shaped cup will be determined based on the shape index of garlic clove. Diameter cup 44 mm and thickness 3 mm it made from galvanized metal sheet.

2.4.7 Determination of planter components weight

2.4.7.1 Planter component weight

The weight of each part designs was calculated using Solid work software 2013. The input of the software was part design equipped with dimensions and material type selection. The software has its own material types with their densities. Once the part design was feed with dimensions and material type, the Solid work 2013 gave the output of the selected component area, volume and mass.

Table 1 A solid work software weight determination analysis for garlic planter

Sr.No	components	Quantity	Area(m^2)	Volumem^3)	Density ($\frac{kg}{m^3}$)	Unit mass(kg)	Total mass
1	frame	1	0.291	0.00231927	7870	18.25	18.25
2	hopper	1	0.20225	0.00030375	8000	2.39	2.39
3	Three-point linkage	1	0.248	0.002976	7870	23.42	23.42
4	Main shaft	1	0.0007068	0.00092	7870	7.24	7.24
5	Metering shaft	2	0.0002545	0.00033	7870	5.2	5.2
6	Furrow opener	4	0.0248	0.000372	7870	11.71	11.71
7	Furrow coverer	4	0.0159	0.0002385	7870	7.5	7.5
8	Seed tube	4	0.028	0.000195	7870	1.53	1.53
9	Bearing	7	0.0194	0.00051	7870	4.01	4.01
10	Ground wheel	2	0.163	0.00163	7870	12.35	12.35
11	Metering device	4	0.0163	0.000272	7870	2.14	2.14
12	Total weight of component parts						95.74
13	Taking 2% margins for the weights of welding bolts, nuts, etc.						1.914
14	Total weight						97.65

Total mass of all components which lay on the wheel shaft was 97.65 kg.

The weight of planter was calculated by using Equation 29 (Sharma and Mukesh, 2019).

$$W = m \times g \tag{29}$$

Where: W= weight, (N), m=total mass of the parts, (kg), g= acceleration due to gravity, (9.81 m s^{-2}),

W = 97.65 × 9.81 = 957.94 N

2.4.7.2 Weight of garlic clove

The weight of the clove in the hoppers was estimated from the volume of the hopper determined earlier above and the mass of clove was estimated using the following Equations 30 (ITSI-SU, 2011).

$$W_{clove} = M_{clove} \times g \tag{30}$$

W_{clove}= 75.6 kg × 9.81 m s^{-2} = 741.63 N

Total weight of a **4WD tractor-drawn** garlic planter was becoming: W_T = Weight of parts + weight of garlic clove

Total weight = 957.94 N + 741.63 N = 1699.57 N

2.4.7.3 Total weight of the prototype planter

The total weight of the prototype planter including the weights of clove and clove hopper, frame, clove metering shaft, furrow opener, furrow coverer, clove tube, and three point linkage was estimated to 1699.57 N.

2.4. 8 Determination of force, torque and power required to derive the planter

2.4.8.1 Determination of force required to derive the planter

A rolling resistance force (F), which was assumed to act horizontally at the wheel and ground interface or ground and wheel contact patch, was estimated using the Equations 31 and 32 (Macmillan, 2002)

$$F = (C_R + i) \times N \quad (31)$$

$$F = ((\frac{Z}{d})^{1/2} + i) \times N \quad (32)$$

But Z = on soft surface = 0.05 d = 0.05 × 50 cm = 2.5 cm

N= weight f planter on each wheel = $\frac{1699.57}{2}$ = 849.78 N

i = gradient of the ground let, i= 5% = 0.05

$F = \left((\frac{2.5}{52})^{\frac{1}{2}} + 0.05\right) * 849.78$ N = 228.8 N

F= 228.8 N

Where: F= rolling resistance force, (N); CR= coefficient of rolling resistance, d = wheel diameter, (cm);

Z = maximum wheel sinkage depth, (cm);

N = weight of planter on each wheel, (N);

i = gradient of the ground.

2.4.8.2 Determination of torque on the ground wheel

Torque on the ground wheel is estimated by using Equation 33 (Sharma and Mukesh, 2019).

$$T = F(\frac{d}{2}) \quad (33)$$

$T = 228.8\text{N}(\frac{0.5 \text{ m}}{2})$ = 57.2 N m

T= 57.2 N m

Where: T = torque on the ground wheel (N m); F = rolling resistance force (N); d = diameter of the ground wheel (m) .

2.4.8.3 Determination of power requirement of the planter

The power requirement of the planter was determined by Equation 34 (Sharma and Mukesh, 2019).

$$P = T \times Nw \quad (34)$$

Where: P = power requirement of the planter, (kW), T = torque, (N m), Nw = wheel revolution, (rpm)

P = 57.2 × (30 rpm ×$\frac{2\pi}{60}$) = 179.69 W = 0.17969 KW

P = 0.17969 KW

2.4.9 Design of chain and sprocket

Chain and sprocket that was available in Ethiopia was selected for transmitting power from ground wheel shaft to shaft of metering mechanism. The chains are mostly used to transmit motion and power from one shaft to another.

The power required to operate the seed metering mechanism was transmitted from the drive wheel through chain drive. Since the power transmitted in the garlic planter is very low, the smallest size available chain, i.e. motor bicycle chain was used for tractor drawn garlic planter. For power transmission, 14 teeth size sprocket was fitted on drive wheel. Another sprocket of 14 teeth size was used for driving shaft so that the transmission ratio of approximately 1:1 was maintained.

2.4.9.1 Calculation of chain length

The length of the chain was calculated by using Equations 35, 36 and 37 (Sadhu, 1988).

$$L = m \times p \quad (35)$$

Where: L= length of chain, (cm); m = number of chain links, P = chain pitch, (mm)

But, the number of chain links

$$M = 2Cd + \frac{N1+N2}{2} + (\frac{N1-N2}{2\pi Cd})^2 \quad (36)$$

$$Cd = \frac{cc}{p} \quad (37)$$

Where: Cd = centre distance chain, (mm), P = commercial available chain pitch, 6.35 mm ; $N1$= number of teeth of the pinion, 14, $N2$ = number of teeth of the sprocket, 14; Cc = approximate centre to centre distance between sprockets, 540 mm.

Solving for *Cd*

$Cd = \frac{cc}{p} = \frac{540\ \text{mm}}{6.35\ mm} = 85.039$ mm

Solving for *M*

M = 2 ×85.039 + $\frac{14+14}{2} + (\frac{14-14}{2\pi Cd})^2$ = 184.07

M = 184 pitches

Solving for *L*

L = *M* ×*P* = 184 ×6.35 = 1168.4 mm

L=1.168 m

Now, since the chain length in pitches is changed from 184.07 to 184 pitches, the exact centre to centre distance of the sprockets has to be corrected (Sadhu, 1988).

$$C = (e + (e^2 - 8m)^{\frac{1}{2}}) \times \frac{p}{4} \quad (38)$$

But, e and m are determined by equations 39 and 40 (Sadhu, 1988).

$$e = LP - (\frac{N1+N2}{2}) \quad (39)$$

$$e = 184 - \left(\frac{14+14}{2}\right) = 170$$

$$m = \left(\frac{N2-N1}{2\pi}\right)^2 \quad (40)$$

$$m = (\frac{14-14}{2\pi})^2 = 0$$

Therefore, the corrected centre to centre distance between the sprockets is

$C = (170 + (170^2 - 8 \times 0)^{1/2}) * \frac{6.35}{4}$

C = 539.75 mm

2.4.9.2 Determination of chain velocity

Average chain velocity was estimated by Equation 41 (Sadhu, 1988)

$$Vav = \frac{n * p * rpm}{376} \quad (41)$$

$Vav = \frac{14 * 0.375 * 30}{376} = 0.418\ \frac{m}{s}$

Where: Vav = average chain velocity, (m s^{-1}), *n* = the driven sprocket no. of teeth, *m*aximum revolution of the driving sprocket = 30 rpm, *P* = chain pitch, 6.35 mm = 0.375 inch

2.4.9.3 Determination chain force

The total load (force) on the driving side of the chain was calculated by using Equations 42, 43, 44 and 45 (Sharma and Mukesh, 2019).

$$FT = F + Fc + Ff \quad (42)$$

Where: *FT* = the total force, (N); *F* = the force due to power transmission, (N); *Ff* = frictional force, (N); *Fc* = centrifugal force on the chain, (N).

$$F = \frac{P}{Vav} \quad (43)$$

$$F = \frac{185.29}{0.418} = 429.88 \text{ N}$$

Where: *F* = the force due to power transmission, (N), *P* = power at garlic planter wheel or power to be transmitted, (Watt), *Vav* = average chain velocity, (m.s^{-1}).

$$Fc = \frac{W \times \text{Vav}^2}{g} \quad (44)$$

$$F = \frac{1.5 \times 0.418^{\,2}}{9.81} = 0.0267$$

Where: *Fc* = centrifugal force on the chain, (N), *W* = weight per meter of the chain (1.5 N m^{-1}); *g* = gravitational acceleration, (9.81 m s^{-2}).

$$Ff = W \times Kf \times Cc \quad (45)$$

Ef= 1.5×4 ×0.54 = 3.24 N

Where: *Ff* = frictional force, (N); *Cc* = nominal centre to centre distance between the sprockets, (0.54 m), *Kf* = friction factor = 4 for horizontal drive, 2 for inclined drive and 1 for vertical drive (Norton, 2005).

Then, *FT* = 429.88 + 0.0267 + 3.24= 433.147 N

According to American National Standard institute (ANSI) standard, the minimum tensile strength of the chain was 3470 N (Did-Daido Co, 2015). To avoid breakage and failure of the chain, the safety factor should be more than one. Checking safety factor, Sf was calculated by using Equation 46 (Sharma and Mukesh, 2019).

$$S_f = \frac{Tensile\ strength}{FT} \quad (46)$$

Sf = $\frac{3470}{433.147}$ = 8, the value is greater than unity.

Therefore, we can conclude that the chain is safe from breakage.

Where: Sf = safety factor, FT = Total force on driving chain, (N)

2.4.10 Design of ground wheel shaft

The ground wheel shaft was made of mild steel rod. The length of the bar was kept as 1500 mm to facilitate the hub on the both ends and sprocket at the one end of shaft. The shaft was rested on two hubs which was fitted on the frame.

The shaft was initially decided to be fabricated from ductile material (mild steel rod). Hence, the design was based on ductile material whose strength is controlled by maximum shear stress. For a shaft having little or no axial loading, the diameter of the shaft was obtained using the ASME code Equation 47, (ASME, 1995) given as:

$$d^3 = \frac{16}{\pi * Ss}\sqrt{(Kb \times Mb)^2 + (Kt \times Mt)^2} \quad (47)$$

Where: d = diameter of the shaft, (mm), Mt = torsional moment, (N m), Mb = bending moment, (N m), Kb = combined shock and fatigue factor applied to bending moment, Kt = combined shock and fatigue factor applied to torsional moment, S_S = allowable stress, (MN m^{-2})

For rotating shafts, when a load was suddenly applied with minor shock (Khurmi, 2005) (recommended that values of Kb = 1.2 to 2.0 and Kt = 1.0 to 1.5 to be used. Furthermore, it was noted that for shaft without a keyway, the allowable stress (Ss) must be 55 MN M^{-2}, and for shaft with key way, the allowable stress (Ss) should not exceed 40 MN M^{-2}.

2.4.10.1 Determination of vertical forces acting on the ground wheel shaft

The vertical load diagram on the ground wheel shaft was showed below.

Figure 6 Vertical force distribution on the shaft

Where: RA = vertical reaction at wheel A, RB = vertical reaction at wheel B, Wc = half of the total weight acting at bearing C, (849.78 N), Wd = half of the total weight acting at bearing D, (849.78 N), Ff = vertical chain force, (446.5367 N)

Total weight of the potato planter which lay on the shaft was 1699.57 N refer above section.

WC=Wd = 849.78 N

WC is half of total weights.

To know the unknown force of RA and RB, use equilibrium equation method.

$\sum MA=0$

$RB \times 1.4$ m + $Ff \times 1.207$ m – $Wc \times 0.0965$ m – $Wd \times 1.3035$ m =0

$RB \times 1.4$ m + 433.147 N $\times$ 1.207 m – 849.78 N $\times$ 0.0965 m – 849.78 N $\times$ 1.3035 m = 0

$RB \times 1.4$ m + 522.8 N m – 82 N m – 1107.68 N m = 0

RB = 476.34 N

The summation of all force gave RA

$RA + RB + Ff – Wc – Wd = 0$

RA + 476.34 N + 433.147 N – 849.78 N – 849.78 N = 0

RA = 790.07 N

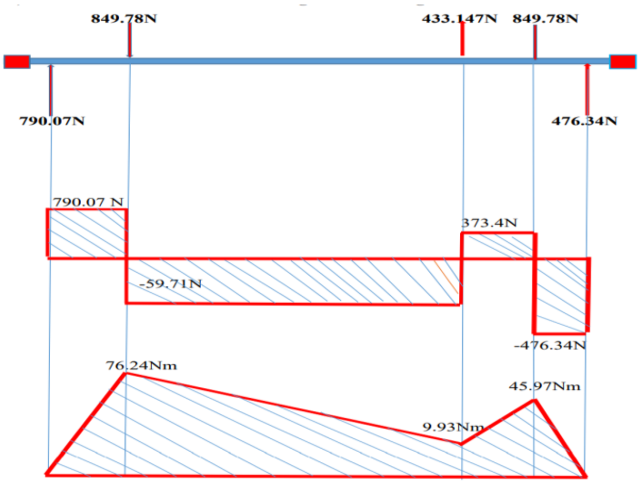


Figure 7 Vertical shear force and bending moment diagram

2.4.10.2 Determination of horizontal force acting on the ground wheel shaft

The forward driving force is 228.8 N and applied horizontally at point A and B.

$RAH + RBH = 228.8$ N

$RAH = RBH = \frac{228.8\text{N}}{2} = 114.4$ N

Where: RAH = horizontal reaction at wheel A, (N); RBH = Horizontal reactions at wheel B; FH = horizontal force (N).

Torsional moment (Mt) on the shaft were calculated using Equations 48 and 49 (Ryder, 1989)

$$Mt = \frac{P\times 60}{2\pi\times N} \quad (48)$$

$$P = F\times V \quad (49)$$

P = 228.8N × 0.833 m s^{-1} =190.59 W

$Mt = \frac{P\times 60}{2\pi\times N} = \frac{190.59\times 60}{2\pi\times 30} = 60.66$ N m

Where: Mt = torsional moment, (N m); P = power required to drive the machine, (kW); N = speed of the shaft, (rpm); P = power required to drive the machine, (kW);V= Forward speed (m.s^{-1}); F= force required to drive the machine, (N).

The maximum bending moment on the shaft was determined from the following expressions given by Equation 50 (Richard, 2011).

$$Mb = \sqrt{(Mv)^2 + (Mh)^2} \quad (50)$$

$Mb = \sqrt{(76.24)^2 + (62.92)^2}$ = 98.85 N m

Where: Mb = maximum bending moment, (N m) ; Mv = vertical bending momentum= 76.24 N m from bending moment diagram ; Mh = horizontal bending momentum = 62.92 N m from bending moment diagram

Using Equation 47 above

$d^3 = \frac{16}{\pi\times Ss}\sqrt{(Kb\times Mb)^2 + (Kt \times Mt)^2} = \frac{16}{\pi\times 55\times 10^6}\sqrt{(2\times 98.85)^2 + (1.5\times 60.66)^2}$

d = 27.2 mm

Therefore, the standard size of 30 mm shaft diameter has been used.

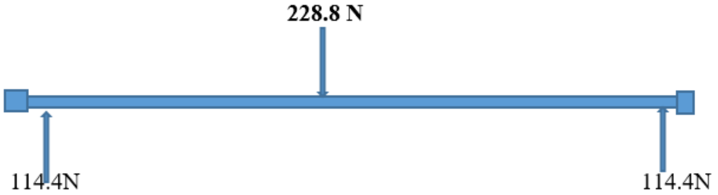


Figure 8 Horizontal force distribution on the shaft

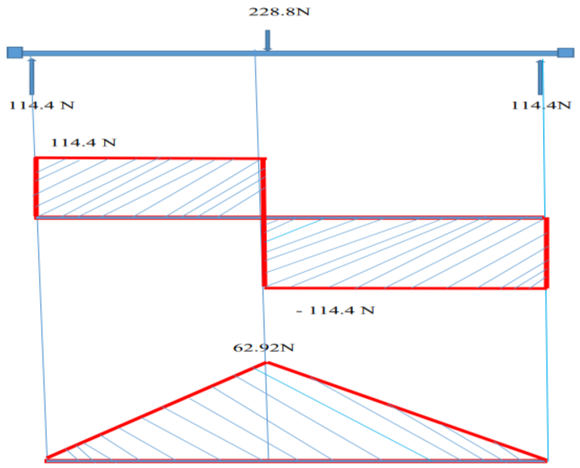


Figure 9 Horizontal shear force and bending moment

2.4.11 Design of clove metering shafts

The vertical load diagram on the driven wheel shaft was as showed below.

Weight of cloves on 16 cups + weight of chain + weight of 16 cups

Weight = 0.0024 kg + 1.5 kg + 0.2 kg = 1.7024kg * 9.81 m s^{-2} =16.7 N

Let RA, RB and RC = Reactions at A, B and C respectively

∴ RA + RB + RC = Total load acting downwards at C, D, E and F

RA + RB + RC = 16.7 N + 16.7 N + 16.7 N + 16.7 N = 66.8 N

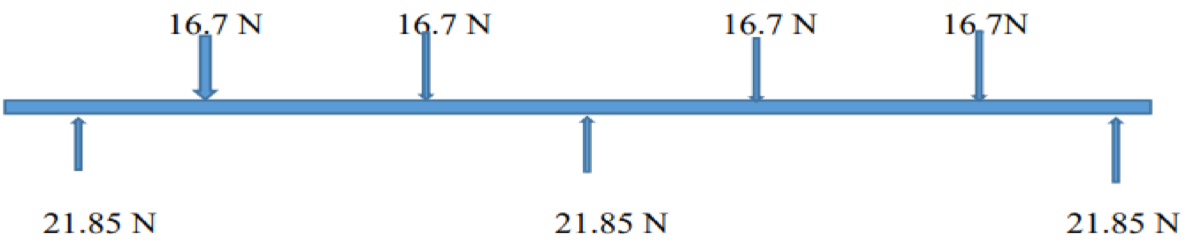


Figure 10 Force distribution on the metering shaft

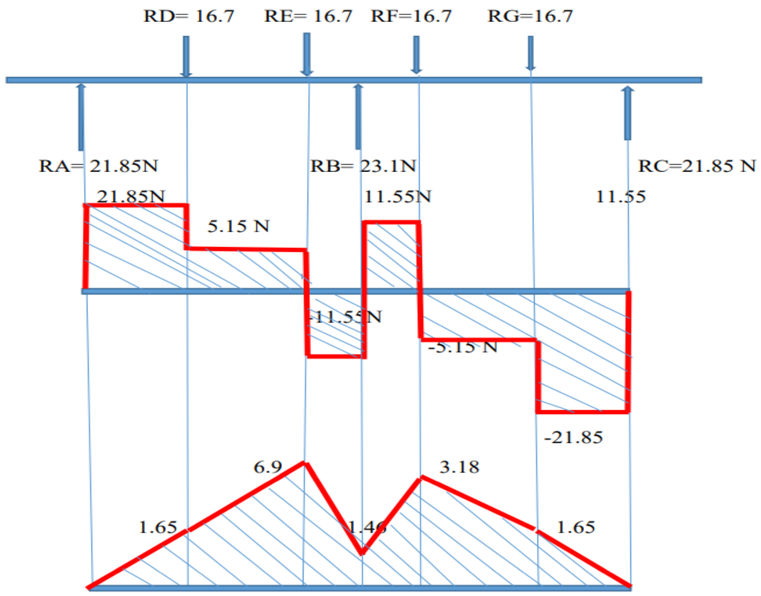


Figure 11 Metering shaft shear force and bending moment diagrams

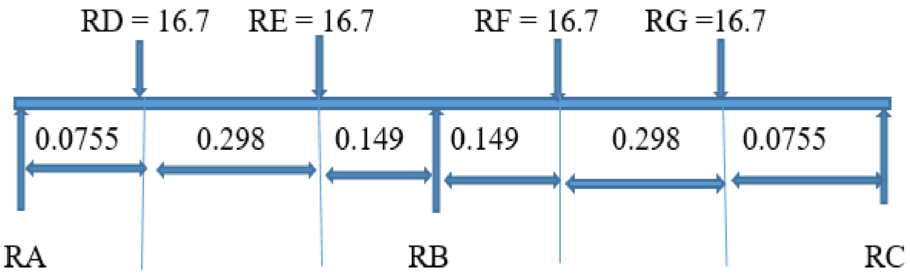


Figure 12 Force distribution on the two span of metering shaft

Considering each span simply supported.

Free bending moment,

For span AB, bending moment at D = $\frac{W*a*b}{L}$ = $\frac{16.7*0.0755*0.447}{0.5225}$ = 1.078 N m

Where: w = weight of chain, seeds and cups at the point D, a= distance of point D from reaction A (RA) b= distance of point D from reaction B (RB), L= distance of span AB or distance reaction A to reaction B.

For span AB, bending moment at E = $\frac{W*a*b}{L}$ = $\frac{16.7*0.3735*0.149}{0.5225}$ = 1.778 N m

Where: w = weight of chain, seeds and cups at the point E

a= distance of point E from reaction A (RA).

b= distance of point E from reaction B (RB).

L= distance of span AB or distance of reaction A to reaction B.

For span BC, bending moment at F = $\frac{W*a*b}{L}$ = $\frac{16.7*0.149*0.3735}{0.5225}$ = 1.778 N m

Where: w = weight of chain, seeds and cups at the point F

a= distance of point F from reaction B (RB).

b= distance of point F from reaction C (RC).

L= distance of span BC or distance of reaction B to reaction C.

For span BC, bending moment at G = $\frac{W*a*b}{L}$ = $\frac{16.7*0.447*0.0755}{0.5225}$ = 1.078 N m

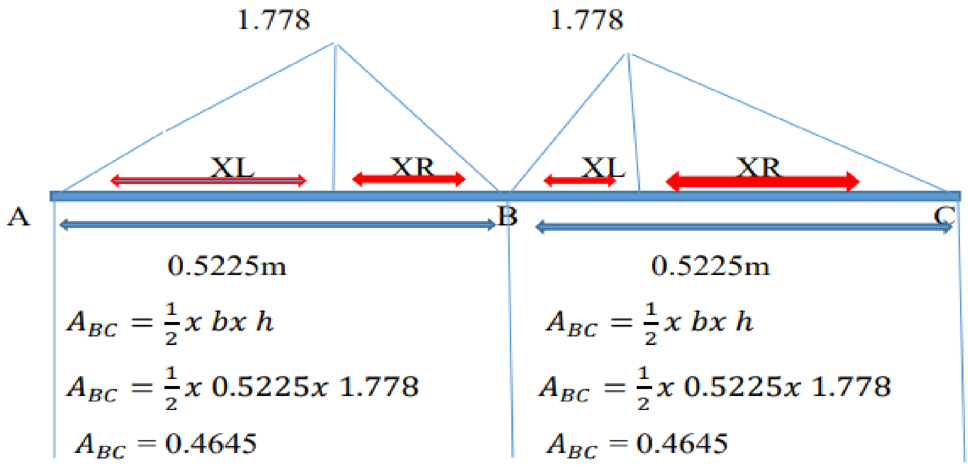
Where: w = weight of chain, seeds and cups at the point G

a = distance of point G from reaction B (RB).

b= distance of point G from reaction C (RC).

L= distance of span BC or distance of reaction B to reaction C.

Free bending moment diagram for span AB and BC, then calculating the area for span AB and BC.



X1= XL = $\frac{a+L}{3} = \frac{0.3735+0.5225}{3} = 0.299$

X1= XL = $\frac{a+L}{3} = \frac{0.149+0.5225}{3} = 0.2238$

X2 = XR = $\frac{b+L}{3} = \frac{0.149+0.5225}{3} = 0.2238$

X2 = XR = $\frac{b+L}{3} = \frac{0.3735+0.5225}{3} = 0.299$

By spplying three moment theorem for A-B-C,

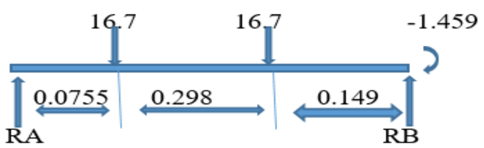
MA (L1) + MB (L1+L2) + MC ((L2) + $\frac{6\ A1\ X1}{L1}$ + $\frac{6\ A2\ X2}{L2}$ = 0

MA (L1) + 2MB (0.5225+0.5225) + MC ((L2) + $\frac{6 * 0.4645 * 0.299}{0.5225} + \frac{6 * 0.4645 * 0.299}{0.5225} = 0$

MB = -1.459

Know reaction force at RA, RB and RC can be calculated

For span AB



∑MA=0 (+ve)

16.7 *0.0755 +16.7 * 0.3735-RB *0.5225 -1.459 = 0

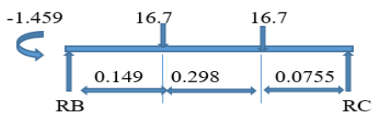
RB = 11.55 N

∑FY=0 (+ve)

RA + RB -16.7-16.7 = 0

RA = 21.85 N

For span BC



∑MA=0 (+ve)

16.7 * 0.149 +16.7 * 0.447 - RC*0.5225 + 1.459 = 0

RC = 21.85

∑FY=0 (+ve)

RB + RC -16.7-16.7 = 0

RB = 11.55

RB = 11.55 +11.5 = 23.1, from both span

Torsional moment (Mt) on the shaft was calculated using Equations 51 and 52 (Ryder, 1989)

$$Mt = \frac{P * 60}{2\pi * N} \quad (51)$$

$$P = F * V \quad (52)$$

= 16.7 N * 0.833 m s^{-1} = 13.91W

Then, $Mt = \frac{13.91 * 60}{2\pi * 30}$ = 4.43 N m

The equivalent twisting moment on the shaft was determined from the following expressions given by Equation 53 (Khurmi, 2005)

$$Te = \sqrt{(Km * M)^2 + (Kt * Mt)^2} \quad (53)$$

$$Te = \sqrt{(1.5 * 6.9)^2 + (1.2 * 4.43)^2}$$

Te = 11.635 N m

Where: M = maximum bending moment, 6.9 N m from bending moment diagram (N m), Mt =torsional moment moment, (N m), Te = Twisting moment (or torque) acting up on the shaft (N m)

From above section using Equation 47.

$$d^3 = \frac{16}{\pi * Ss}\sqrt{(Kb * Mb)^2 + (Kt\ * Mt)^{\ 2}}$$

$$d^3 = \frac{16}{\pi * 55 * \ 10^6}\sqrt{(2 * 3.69)^2 + (1.5\ *\ 11.635)^{\ 2}}$$

d = 11.9 mm

Therefore, the standard size of 18 mm clove metering shafts diameter has been used.

2.4.12 Design of ground wheel drive

Two ground engaging wheels, with external diameters of 50 cm, were designed as an integral part of the clove metering mechanism. They are connected to the clove metering device through chain and sprocket where produces the necessary force to drive the metering cup. The rim of wheel was made from mild steel flat iron 6 mm thick and 100 mm wide. Each wheel had six spokes made from mild steel rods with diameter of 12 mm and length of 225 mm, and were welded to the rim and hub at the centre of the wheel that served as bushing or shaft bearing, at equal interval

Estimation of the force required to drive the planter using Equation 54 (Sharma and Mukesh, 2019).

$$Ff = \left(\sqrt{\frac{Z}{Wd}} + i\right) *\ Mwe \tag{54}$$

Where: Ff = Force required to drive the planter, (N), Z = Maximum tolerable wheel sinkage depth, (let Z= 3 cm), Wd = Wheel diameter, (50 cm), Mwe = Machine weight on each wheel, (849.78 N), i = Gradient of the ground, (let i=5%, Macmillan, 2002)

$$\text{Ff} = \left(\sqrt{\frac{3cm}{50cm}} + 0.05\right) *\ 849.78\ N = 250.64\ \text{N}$$

Ff = 250.64 N

The torque of the ground wheel was determined by Equation 55 (Sharma and Mukesh, 2019).

$$T = Ff * (\frac{Wd}{2}) \tag{55}$$

Where: T= torque produced by the ground wheel, (N m), Ff = Force required to drive the planter, (261.55 N), Wd = Wheel diameter, (50 cm)

$$T = 250.64 * (\frac{0.5}{2}) = 62.66\ \text{N}$$

T = 62.66 N

Shear stress on the ground wheel was determined by Equation 56 (Sharma and Mukesh, 2019).

$$\tau = \frac{T}{2 * Am * tw} \tag{56}$$

Where: τ = shear stress on the ground wheel, (kPa); T= torque produced by the ground wheel, (N m); Am= the area of the wheel calculated based on the median diameter of the wheel, (m^2).

tw = thickness of the wheel wall, (0.006 m); r = the outer radius of the wheel, (0.3 m); rm= the median radius of the wheel, (m).

The area of the ground wheel was determined by Equation 57 (Sharma and Mukesh, 2019).

$$Am = \pi * (rm)^2 = \ \pi(0.25 - 0.5 *\ tw)^2 \tag{57}$$

$$Am = \ \pi(0.3 - 0.5 *\ 0.006)^2 = 0.19\, m^2$$

Then, $\tau = \frac{62.66}{2 * 0.19 * 0.006} = \frac{62.66}{0.002299} = 27.48$ KPa

Comparing this shear stress with the maximum allowable shear stress of the mild steel sheet metal << max, 80 MPa, which tells us the wheel was safe for operation.

The angle of twist can be estimated using the Equation 58 (Richard, 2011)

$$\theta = \theta_1 *\ L \tag{58}$$

$$\theta = \left(\frac{TL_m}{4GA_m{}^2 * tw}\right) *\ L\ = \left(\frac{62.66 * 2\pi(0.5 - 0.5 * 0.006)}{4 * 80 * 0.19^2 * 0.006}\right) * 0.1 = 0.0000000102 \text{ degree}$$

θ = 0.0000000102 degree (negligible)

Where: L= length or width of the wheel, 0.1m, T= torque produced by the wheel, 62.66 N m.

Lm=the length of the median line of the wheel, θ1=the angle of twist, G= modulus of rigidity, 80GPa for stainless steel.

The angle of twist produced by the wheel is negligible since the torque is small.

2.4.12.1 Design of hub

The hub of a wheel is one of the most important components of rigid wheel. It gives support to the spokes and the shaft. The diameter of the hub is calculated using following formula. The outside of the hub is given by Richard (2011).

D =1.50 d + 25.00 mm

D= 1.50 ×30 mm +25 mm= 70 mm

Where: D = outside diameter of hub, mm, d = diameter of shaft, mm

Considering the availability of material 50 mm diameter of hub is used

The length, L, of hub is given by Richard (2011).

$$L = \frac{\pi * d}{2} \quad (59)$$

$$L = \frac{\pi * 30}{2} = 47.1 \text{ mm}$$

The inner diameter of hub was taken equal to the diameter of the shaft i.e. 30 mm.

2.4.12.2 Wheel width

When two cylindrical objects of length L and diameters d_1 and d_2 contact, a narrow rectangle of width 2b and length L is created.

The contact stress between two cylinders expressed by Zhupanska (2011).

$$b = \sqrt{\left(\frac{2F}{\pi L}\right) * \left(\frac{(1-(V1)^2)}{E1} + \frac{(1-(V1)^2)}{E2} / \left(\frac{1}{d1} + \frac{1}{d2}\right)\right.} \quad (60)$$

$$Pmax = \frac{2F}{\pi bL} \quad (61)$$

The stress created has an elliptical distribution across the contact zone (Figure 13).

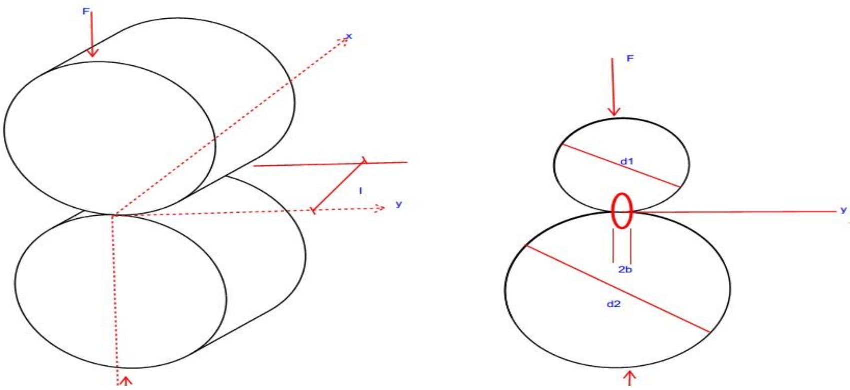


Figure 13 The contact stress between two cylinders

The above equations apply to a cylinder and a plane surface (d= ∞) contact (Richard, 2011).

Poisson's ratio is a measure of the Poisson effect, the phenomenon in which a material tends to expand in directions perpendicular to the direction of compression. Conversely, if the material is stretched rather than compressed, it usually tends to contract in the directions transverse to the direction of stretching. It is a common observation when a rubber band is stretched, it becomes noticeably thinner. Again, the Poisson ratio will be the ratio of relative contraction to relative expansion and will have the same value as above.

Where: d_1= Diameter of the wheel, 0.50m, d_2= Diameter of the contacting surface, ∞

F = the force exerted by the wheel, 849.78 N, L = Contact length/width of the wheel with the ground, 0.1 m, P_{min} = the minimum penetration resistance/pressure of the ground (at 0-10 cm depth) at optimum moisture content, cone, 550 KPa (Tsegaye, 2015)

v_1= Poisson ratio of steel, 0.3 (Norton, 2005), v_2= Poisson ratio of sandy loam soil, 0.25-0.4 (Norton, 2005), E_1 = Young's modulus of elasticity of steel, 200 GPa (Norton, 2005). E_2=Young's modulus of elasticity of sandy loam soil, 10-50 MPa (Norton, 2005). Four-wheel tractor drawn planter wheel width in the design was 10 cm and this had to be checked whether the width selected would allow the wheels to rotate on the surface of the ground without sinkage or not. If $P_{max} < P_{min}$, it is assumed that the width of the wheel is safe. If not, the width of the wheel has to be increased to some level to avoid wheel sinkage in operation. Calculating width (b)

$$b = \sqrt{\left(\frac{2x849.78N}{\pi x\ 0.10m}\right) * \left(\frac{(1-(0.3)^2)}{200\ x 10^9 pa} + \frac{(1-(0.4)^2)}{50\ x\ 10^6 pa} / \left(\frac{1}{0.5m} + \frac{1}{\infty}\right)\right.}$$

$$b = 52*10^{-3}$$

Calculating maximum pressure (P_{max}),

$$P\mathbf{max} = \frac{2 * 849.78N}{\pi b * 0.1} = 104.035*10^3$$

P_{max} = 104.035 k**P**a, <<<<<<<< P_{min} =550 k**P**a,

$P_{max} < P_{min}$,

Therefore, the size of the width of the garlic planter wheels is adequate and the wheel would rotate on the surface of the ground without sinking into the soil.

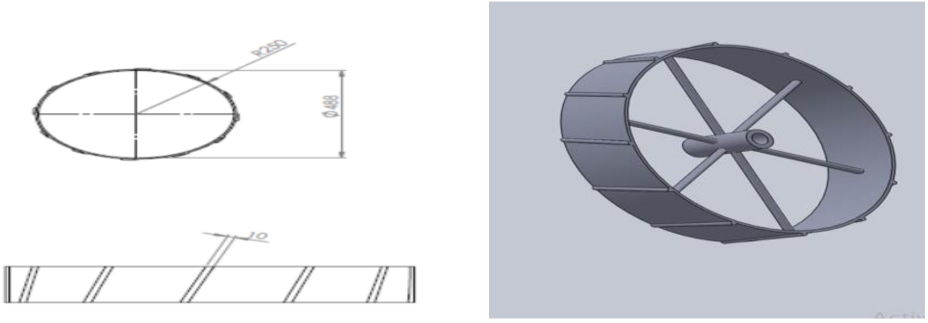


Figure 14 Ground wheel

2.4.12.3 Spokes

Each wheel had six spokes made from mild steel rods with diameter of 12 mm and length of 225 mm, and were welded to the rim and hub at the centre of the wheel that served as bushing or shaft bearing, at equal interval

2.4.13 Clove tube

Clove tube was the main unit that conveys the clove from the hopper-through metering device to the furrow opener. It was fitted to connect the feed cup outlet to the furrow opener. The clove tube carries the garlic clove discharge from metering device to the furrow openers. It is easy to fixed and remove the clove tubes as it was made from galvanized sheet metal. The uniformity of clove distribution, along and across the rows, was influenced by the clove tube. Keeping the above consideration, a sheet metal tube with an inner diameter of 100 mm was selected. The length of clove tube was kept 974 mm.

2.4.14 Furrow coverer

The furrow coverer was one of the important parts of the planter, it covers the garlic clove for proper growth of crop.

2.4.15 Three point linkage

Three-point linkages were provided to connect the frame of garlic planter to the tractor. Three- point linkages consist of two lower arms and one upper arm. To attach the implement with tractor lower and upper arms was made on the frame.

Table 2 Three-point hitch specification

Category	Hitch pin size		Lower hitch spacing	Tractor drawbar power
	Upper link	Lower link		
0	17 mm	17 mm	500 mm	< 20 hp
1	19 mm	22.4 mm	718 mm	20 -45 hp
2	25.5 mm	28.7 mm	870 mm	40 -100 hp
3	31.75 mm	37.4 mm	1010 mm	80 – 225 hp
4	45 mm	51 mm	1220 mm	180 – 400 hp

Source: American Society of Agricultural Engineers ASAE (2010)

2.5 Laboratory test

2.5.1 Calibration of the planter

After fabrication of the planter, calibration of the prototype was carried out to determine the seeding rate obtainable at different hopper capacity and the variation among furrow openers when the machine was stationary. The procedure for calibration were as follows: (1) The width of coverage of planter is equal to the product of the number of furrow openers and the spacing between two consecutive openers; (2) Area covered in 10 revolution of ground wheel was determined; (3) The planter was jacked up so that the ground wheel runs freely. A mark was made on the drive wheel and at some convenient place on the body of planter so as to count the revolution of the drive wheel easily; (4) The cloves were filled in the hopper and plastic bag were placed under the furrow openers; (5) The quantity of cloves dropped from furrow openers for 10 revolution were collected and weighed; (6) The quantity of cloves dropped was converted on hectare basis i.e.kg/ha; and (7) Three replications were taken for 2 km hr^{-1}, 2.5 km hr^{-1} and 3 km hr^{-1} speed for garlic cloves were observed.

2.5.2 Cloves damage test

The test was conducted to find out the percentage of damage of garlic cloves that takes place during actual operation. The percentage of damage of clove during calibration was found out by the rotating ground wheel. The number of cloves collected was weighed. Out of this, the damaged cloves was separated. The metered clove were identified and treated as damaged cloves was weighed separately and percentage damage was calculated as follows.

$$Damage\ percentage \frac{weight\ of\ damaged\ clove}{total\ weight\ of\ sample} \times 100 \quad (62)$$

2.5.3 Clove germination test

Garlic clove samples (100) cloves was taken randomly and kept in the clove germinator and water was sprinkled at optimum water requirement of garlic clove. The temperature of the clove germinator was maintained at optimum. The sprouts of clove was observed after seventh-day and number of sprouts was count daily. The counting of sprouts was stopped when the number remains constant. Since the sample is 100 cloves, therefore, the number of sprouts straight away gave the germination percentage of cloves. The germination percentage of garlic cloves was determined before and after the laboratory test of metering systems (Belcher, 2013).

2.5.4 Soil moisture content

The sample were collected from five randomly selected sites across the field. The moisture content was determined in the laboratory by the oven-dry method. The moisture content (dry basis) was determined by Equation 63 (Belcher, 2013)

$$W = \frac{Ww-Wd}{Wd} \times 100\% \quad (63)$$

Where: W= Moisture content, (% db.), Ww= Wet mass of soil, g, Wd = Dry mass of soil, g

2.5.5 Soil bulk density

The bulk density of the soil was determined by the core cutter method. The core sampler of the soil of known volume was collected and weighed. The ratio of the dry weight of soil to the volume gives the bulk density. The soil bulk density was determined by using Equation 64 (Rangapara, 2014).

$$\text{Bulk density of soil}(\text{g } cm^{-3}) = \frac{weight\ of\ dry\ soil\ sample\ (g)}{volume\ of core\ sampler\ (cm^3)} \quad (64)$$

3 Result and discussion

3.1 Physical properties of garlic clove

In order to get some of physical properties of the garlic clove, 40 sample cloves were randomly taken from Gojjam and Minjar variety of garlic clove and their length, width and thickness were measured using Vernier calliper. The geometrical dimension of the principal axes (L is length, D is the width and T is the thickness) of randomly selected garlic seeds were measured using Vernier calliper. The mean length (L), width (W), thickness (T), geometric mean diameter, surface area and shape index of two different garlic was found in the ranges from 28.55 to 29.33 mm,13.17 to 13.23 mm, 10.44 to 10.57 mm, 15.53 to 15.89 mm,769.18 to 807.88 mm^2 and 0.755 to 0.78 respectively. Angle of repose garlic was maximum 26 °, minimum 25 °and average is about 25.5 °.

The average values obtained for geometric mean diameter, surface area and shape index of the selected garlic cloves was became 15.71mm, 788.53mm^2 and 0.767mm respectively. The results in Table 3 show that garlic seeds were more or less spherical in shape. As a result, metering devices with circular cups were designed to accommodate it.

Table 3 Physical properties of two different garlic varieties

Physical properties	Units	Two different varieties of garlic seed		
		Gojjam	Minjar	Average
Mean Length	Mm	28.55	29.33	28.94
Mean Width	Mm	13.17	13.23	13.2
Meant thickness	Mm	10.44	10.57	10.5

Mean geometric diameter	mm	15.53	15.89	15.71
Mean Surface area	mm²	769.18	807.88	788.53
Mean Shape index		0.78	0.755	0.767
Angle of repose	°	26	25	25.5

The obtained shape index was compared with the recommended limits and classified into different classes (The garlic bulb is considered as oval if the shape index > 1.5, on the other hand, it is considered spherical if the shape index < 1.5). Shape index of garlic bulbs was calculated by equation below Equation 65 (Abd-Alla, 1993).

$$Shape\ Index = W/\sqrt{LT} \tag{65}$$

Gojjam, shape index = 0.78 spherical

Minjar, shape index = 0.755 spherical

3.2 Laboratory performance of garlic planter

The newly developed tractor drawn garlic planter was tested in the laboratory to evaluate its performance. The results are discussed in the following paragraphs.

3.2.1 Effect of operating speed on seed damage at different hopper fills

Three replication is carried out for calibrating the garlic planter in the laboratory. Figure 15 indicates the effect of three different speed at three hopper level on clove damage for different observation varied with operating speed 2 km hr^{-1}, 2.5 km hr^{-1} and 3 km hr^{-1}.

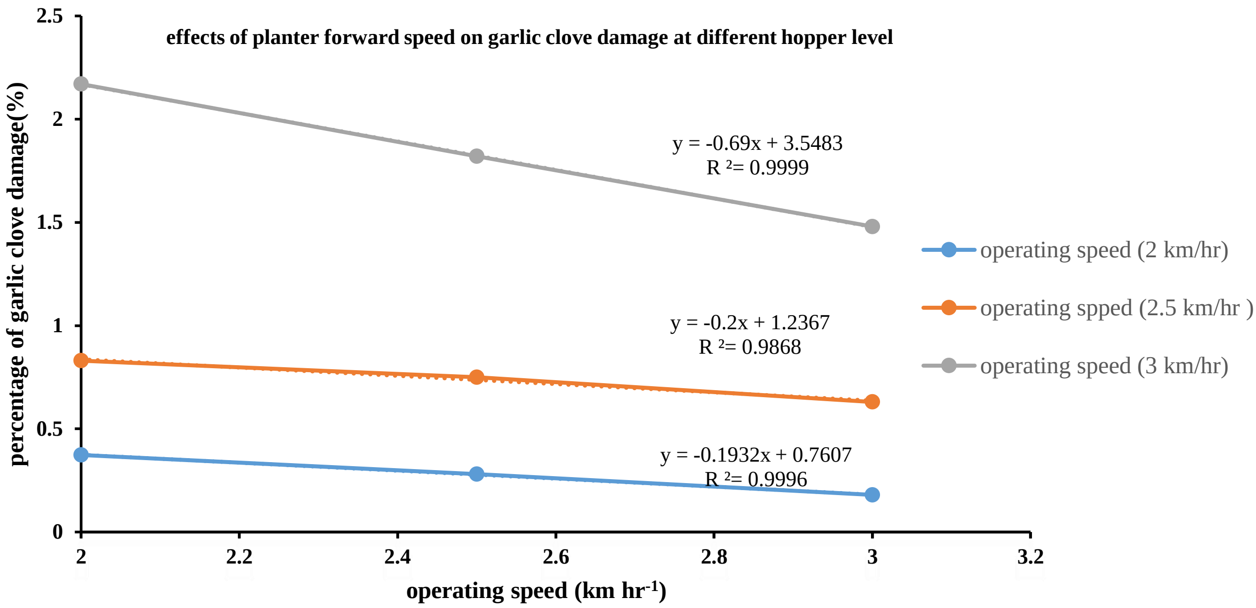


Figure 15 Effects of planter forward speed on garlic clove damage at different hopper fill levels

From Figure 15, it is shown that as the speed increased, the mechanical damage of garlic clove increased. It is due to higher rotational speed of metering mechanism at higher forward speed. At higher rotational speed the cup strikes the cloves with great impact, resulting in mechanical damage.

The correlation between seed damage at different operating speed and level of hopper fill was showed in Figure 15. The data obtained from the testing were analysed, linear regression showed good correlation between seed damage and operating speed in all trials, high coefficient of determination (R^2=0.9996, R^2=0.9868 and R^2=0.9999) were observed with complete hopper fill, ½ hopper fill and ¼ hopper fill respectively.

3.2.2 Hopper filling effect on the clove rate

Clove rate increased with the increase in hopper fill and operating speed decrease as shown in Figure 18. The increase in clove rate with the increase in hopper fill and decrease in operating speed might be due to decrease in exposure time to pick the clove and due to decrease slip of clove from the cup due to lower speed of metering device.

This may be due to the variability of garlic clove in size and shape, vibration of metering chain when hopper fill level is low and the friction between the metering chain and garlic clove at the picking zone due operating speed.

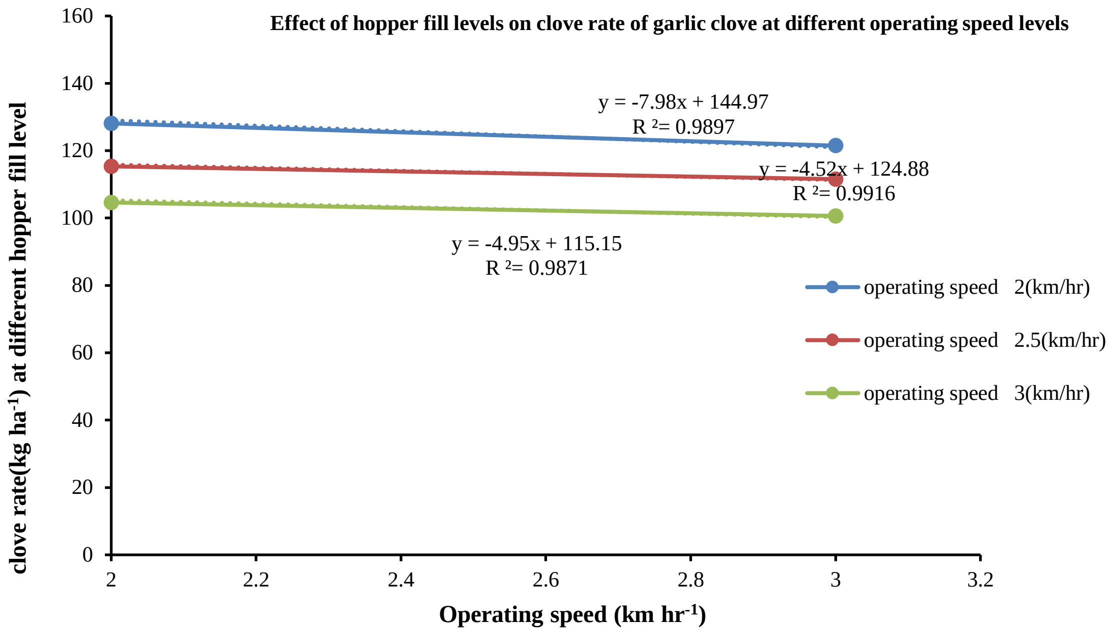


Figure 16 Effect of hopper fill levels on clove rate of garlic clove at different operating speed levels

Figure 16 indicates the effect of hopper filling on clove rate (kg hr^{-1}) of garlic clove for different observation varied with the hopper filling (full, ½ fill and ¼ fill). The clove rate of the planter cloves was tested at three levels of hoppers fills for three different operating speed.

The correlation between clove rate at different level of hopper fill and operating speed was showed in Figure 16 The data obtained from the testing were analysed, linear regression showed higher correlation between clove rate and hopper fill level (complete hopper fill, ½ hopper fill and ¼ hopper) in all trials, high coefficient of determination ($R^2 = 0.989, R^2 = 0.991\ and\ R^2 = 0.987$), were observed with 2 km hr^{-1}, 2.5 km hr^{-1} and 3 km hr^{-1} respectively.

3.3 Physical properties of soil

3.3.1 Moisture content and bulk density of soil

During carrying out the experiments, the soil conditions of the experimental field were studied and different parameters were calculated (Table 4). The soil of the field was sand–loam soil. Moisture content of soil was measured by oven dry method. Five soil samples were taken randomly at 5 different locations in the plot using core sampler of 5cm diameter and 5cm height.

The mean data on soil moisture content after tillage operations at 0 to15 cm are recorded and presented. Average value of moisture content (Dry basis) and bulk density of experimented plot was found 11.78% and 3.224 gm cm^{-3} respectively.

Table 4 Moisture content and bulk density of soil

Sample No	Weight of soil(gm)	Weight of soil after oven dried(gm)	Soil moisture content (Db) (%)	Volume of core sampler (cm^3)	Bulk density(gm cm^{-3})
1	358.86	321.86	11.49	98.17	3.278
2	341.3	303.33	12.51	98.17	3.089
3	351.13	316.8	10.83	98.17	3.22
4	364.06	322.23	12.98	98.17	3.28
5	355.16	319.66	11.1	98.17	3.256
Average	347.13	316.77	11.78	98.17	3.224

3.4 Cost analysis

Table 5 Materials for construction of the planter and its cost

Sr.No	Type of material	Specification	Material used	Cost (USD)
1	Hollow square pipe	60 mm × 60 mm × 3 mm	6 m	$ 17.46
2	Sheet metal	3 mm × 1000 mm × 3000 m	4 m	$ 7.27
3	Galvanized metal sheet	2 mm × 1000 mm × 2000	2 m	$ 6.55
4	Angel iron	5 × 40 × 40	3	$ 6.18
5	Round bar	Ø20 × 6000 mm	2.4 m	$ 5.09
6	Round bar	Ø 30 × 6000 mm	1.5 m	$ 4.91
7	Metal bolt and nut	M-6 × 60	8 pcs	$ 0.72
8	Metal bolt and nut	M-8 × 50	10 pcs	$ 1.27
9	Metal bolt and nut	M-10 × 100	16 pcs	$ 2.91
10	Metal bolt and nut	M-10 × 120	32 pcs	$ 6.4
11	Motor cycle Chain	6.35 mm pitch	6pcs	$ 30.55
12	sprockets	14 Number of teeth	9 pcs	$ 8.18
13	Bearing	P206	2 pcs	$ 10.91
14	Bearing	P203	4 pcs	$ 21.82
15	Flat iron	10 × 60 × 6000	6m	$ 4.91
16	Electrode	Ø 2.5	2 pack	$ 5.45
17	Paint	2 litre	2 litre	$ 2.73
	Total cost(USD)		$ 143.31	

Table 6 Machine and labour cost

Type of machine	Machine cost/hr	Working hour	cost	Labour cost/hr	Working hour	Total price
Universal metal cutting	$ 12	8	$ 96	$ 0.7	10	$ 7
Welding machine	$ 8	8	$ 64	$ 1	12	$ 12
Power hack saw	$ 6	2	$ 12	$ 0.6	3	$ 1.8
Lath machine	$ 14	12	$ 168	$ 2	12	$ 24
Rolling machine manual	$ 5	5	$ 25	$ 0.6	8	$ 4.8
Radial drill machine	$ 8	4	$ 32	$ 1	10	$ 10
Grinding machine	$ 6	5	$ 30	$ 0.6	10	$ 6
Bending machine	$ 4	2	$ 8	$ 0.7	4	$ 2.8
Sub total			$ 435			$ 68.4

Total fabrication cost of garlic planter.

Table 7 Cost summary

No	Cost variables	Summary
A	Raw material cost	$ 143.31
B	Material wastage 2.5%	$ 3.58
C	Machine cost	$ 435
D	Labour cost	$ 68.4
E	Overhead cost 5% (C+D)	$ 25.17
F	Profit 10% (A+B+C+D+E)	$ 67.55
G	Sells tax 15% (A+B+C+D+E+F)	$ 111.45
H	Selling price	$ 854.46

The result showed that the developed tractor drawn garlic planter worked satisfactory functionally and therefore proposed its use for the planting of the garlic using 18.65 KW tractor. However, tractor drawn garlic planter which is estimated to be $ 854.46 to be high for a single farmer to own the machine but by renting farmers can save time, reduce production cost (planting operation cost) and prevent drudgery of labour.

4 Conclusion

The following conclusion could be made from the laboratory test of tractor drawn garlic planter:

- The tractor drawn garlic planter works satisfactory and saves time.
- The clove rate for seeds at different hopper filling and they were recorded were 122.86, 116.01 and 111.23 kg respectively.

• The meter clove were observed average mechanical damage at different hopper filling level (¼ fill, ½ fill and full fill) were 0.803%, 0.91% and 1.124% in of garlic clove respectively.

• The developed tractor drawn garlic planter worked satisfactory functionally, can save time, reduce planting operation cost, prevent drudgery of labour, and therefore proposed its use for the planting of the garlic seed.

References

Abd-Alla, H. E. S. 1993. Effect of coating process on seeds viability and some physio mechanical properties of Egyptian cotton. *Journal of Agricultural Sciences, Mansoura (Egypt)*, 18(8): 2384-2396.

Ahmad, Shahzad, Zarina Yasmeen, Muhammad Ashraf, Abdul Ghafoor, Manzoor Ahmad, Abdul Aleem, Ghulam Saddique, Muhammad Iqbal Sabar, Maqsood Ahmad, and Iftikhar Haider. Performance Assessment of Tractor Mounted Garlic Planter using different Varieties of Garlic. *Journal of Agricultural Research.6*0, no. 3 (2022): 251-257.

ASAE Standards. Three-Point Free Link Attachment for Hitching Implements to Agricultural Wheel Tractors, 2010.

ASME. 1995. Design of Transmission Shafting. New York, USA; American Society of Mechanical Engineering.

Bakhtiari, M. R., D. Ahmad, J. Othman, and N. Ismail. 2011. Physical and mechanical properties of kenaf seed. *Applied Engineering in Agriculture*, 27(2): 263-268.

Belcher, Donald Jenks. *The measurement of soil moisture and density by neutron and gamma-ray scattering*. No. 127. Civil Aeronautics Administration, Technical Development and Evaluation Center, 1950.

Bosoi, Evgenii Sergeevich, O. V. Verniaev, I. I. Smirnov, and E. G. Sultan-Shakh. 1987. Theory, construction and calculations of agricultural machines.

Central Statistical Agency. 2021. Federal democratic Republic of Ethiopia, central statistical agency, agricultural sample survey 2020/2021, Vol. II, Report on “Crop and Livestock Product Utilization (Private Peasant Holdings, Meher Season)”. Addis Ababa, Ethiopia. CSA.

Did-Daido Co. 2015. DLCatalog. Available at: https://www.did-daido.co.jp/documents/en/catalog/dlcatalog_e.pdf. Accessed 15 02 2015.

Garnayak, D. K., R. C. Pradhan, S. N. Naik, and N. Bhatnagar. 2008. Moisture-dependent physical properties of jatropha seed (Jatropha curcas L.). *Industrial Crops and Products*, 27(1): 123-129.

Gomes Viana, J. P., C. de Jesús Pires, M. M. Bajay, S. E. dos Santos Valente, J. B. Pinheiro, M. I. Zucchi, A. C. de Almeida Lopes, and R. L. Ferreira Gomes. 2021. Do the importations of crop products affect the genetic diversity from landraces? A study case in garlic (*Allium sativum* L.). *Genetic Resources and Crop Evolution*, 68: 1199-1211.

ITSI-SU. 2011. *Mass, volume and density of regularly and irregularly shaped object*. Available at: *http://www. pdesas. Org/module/content/resources/14756/view. ashx*.

Karababa, E. 2006. Physical properties of popcorn kernels. *Journal of Food Engineering*, 72(1): 100-107.

Khurmi, R. S., and J. K. Gupta. 2005. *A Textbook of Machine Design*. S. Chand publishing, *Eurasia publishing house (pvt) ltd. Ram Nagar,New Delhi-110055.*

Knapp, W. M., G. R. Deboy, W. E. Field, G. W. Krutz, and C. L. Corum. 2011. Establishing the need for an engineering standard for agricultural hitch pins. ASABE Paper No. 1111027. St. Joseph, Mich.: American Society of Agricultural and Biological Engineers.

Kushwaha, D. K., U. B. Singh, and C. P. Singh. 2020. Performance evaluation of a push-type manually operated garlic planter. *International Journal of Current Microbiology and Applied Sciences*, 9(8): 2348-2356.

Macmillan, Ross Heatley. *The mechanics of tractor-implement performance: theory and worked examples*. Melbourne University Custom Book Centre, 2002.

Norton, M. An introduction to the synthesis and analysis of mechanisms and machines. In M. Norton, *Design of farm machinery.* New York, USA: R.H. Macmillan, Melbourne, Australia, 2005.

Özgüven, F., and K. Vursavuş. 2005. Some physical, mechanical and aerodynamic properties of pine (*Pinus pinea*) nuts. *Journal of Food Engineering*, 68(2): 191-196.

Rangapara, D. J. 2014. Development of mini tractor operated picking type pneumatic planter. Ph.D. diss., Anand Agricultural University.

Richard, G.B. a. *Shipley’s Mechanical Engineering Design.* New York, USA: McGraw-Hill, 2011.

Ryder, G. H. 1989. *Strength of Materials*. 3rd ed. Nigeria: Macmillan.

Sadhu, S. A. 1988. *Machine Design.* New delhi, India: Khanna Publishers.

Sharma, D. N., and S. Mukesh. *Farm machinery design:*

Principles and problems. Jain brothers, 2019.

Tsegaye, A. 2015. Development of animal drawn multi-crop planter. Ph.D. diss., Haramaya University.

Zhupanska, O. I. 2011. Contact problem for elastic spheres: applicability of the Hertz theory to non-small contact areas. *International Journal of Engineering Science,* 49(7): 576-588.

Zilpilwar, S., and R. Yadav. 2020. Study of garlic (*Allium sativum* L.) clove planters. *Journal of Pharmacognosy and Phytochemistry*, 9(6S): 496-501.