DEVELOPMENT OF A RECIPROCATING WEEDING MACHINE FOR COMMODITY CROP PRODUCTION

Folasayo Fayose1*, Adesoji. Olaniyan1, Ayomide Osetoba, Kayode Ogunleye, Oluwaseun Ilesanmi
Federal University Oye Ekiti, Nigeria
*Corresponding author: folasayo.fayose@fuoye.edu.ng

ABSTRACT

A lot of efforts have been made to develop different types of weeding machines in Nigeria with different drive mechanisms, features, designs, and steering mechanisms. Up to this point, no model or design have been commercialized but they remain in the prototype stage. So weeding is still a manual operation in Nigeria and most African countries. Although chemical weeding is being adopted in many places, its side effects has been a major concern. Therefore, a motorized single row, reciprocating weeder was designed, fabricated and tested. The result indicated a quality performance efficiency of 82.7% and field capacity of $0.03m^2/s$ as against $0.01m^2/s$ with manual weeding. The material for the construction was sourced from readily available materials.

Keywords: Reciprocating mechanism, Weeding, Machine design, efficiency, field capacity

1.INTRODUCTION

Weeds are one of the major causes of loss of agricultural produce. Weeds compete with crops for essential nutrients. In agriculture, it's a very difficult task to weed out unwanted plants manually as well as using bullock operated equipment which may further lead to damage of main crops. Weed infestation on Nigerian soils is quite high particularly during the raining seasons when soil moisture is high and plant growth conditions are optimum. This higher competitive nature of weeds compared to crops poses serious threat to farming including decrease in crop yield, impairment of crop quality, harbouring of plant pests and diseases, increase in irrigation costs, injury to livestock and decrease in land values. Oni (1990) [1] reported that 50 to 70% of yield reduction is caused by poor weed control. Therefore, in order to improve crop production, weed control is a necessity.

The methods employed in controlling weeds include biological, chemical, manual and mechanical weeding. Manual weeding is most practiced in Africa particularly in Nigeria where about 75% of the population is engaged in farming [2]. This method is labour intensive and its one of the major problems of farming in Nigeria [2, 3]. Mechanical weed control is very effective as it helps to reduce drudgery involved in manual weeding, it kills the weeds and also keeps the soil surface loose ensuring soil aeration and water intake capacity. Mechanical weeding can help to retain removed weeds for manure/mulch purposes unlike chemical weeding which burns up the weeds. Tractor mounted weeders are very effective but they're not affordable by local farmers. In developed countries, chemical weeding is more prominent than mechanical weeding . However, in the recent times, the problem of environmental degradation, adverse effect on farmers' health and pollution is making the world to have a re-think on the adoption of mechanical weeders.

Although a lot of efforts have been made in the past to develop weeders in Nigeria, many of the efforts have not been adopted by the end users i.e. the farmers [4, 5,6]. Moreover, the Department of Agricultural and Bioresources Engineering, Federal University, Oye Ekiti have developed reciprocating weeders [7]. As part of the efforts to develop an effective mechanical weeder, the development of a low profile weeder whose cutting blade is actuated by the ground wheel traction through a reciprocating mechanism is described in this paper.

2.DESCRIPTION AND DESIGN CONSIDERATIONS

2.1 DESCRIPTION

The major components of the weeder include the main frame, speed reducing gear, stanchion, wheels, reciprocating arms, blades, shafts, IC Engine, Chain and sprocket, coupler. When the IC Engine is started, its speed is stepped down by ratio 1 to 15. The reciprocation prevents the need to move the blades back and front as this action is automatically achieved in one process. While designing and in material selection, consideration was given to techno-economic status of the small-scale farmers in the rural communities who are the intended users of the machine. It is simple to operate unlike flaming which require high skills, it does not constitute environmental hazards like chemical weeding. Instead of burning up the weeds like in chemical weeding, this mechanical weeder can harvest the weeded material and retain them for manure/mulch purposes. It maintains the seed bed in a good tilth and aeration during the growth of crop by loosening the soil between rows, thus increasing air and water intake capacity. The design considers a crop grown on an at least (25 cm × 25 cm) row spacing. The blades were made to have overlap of cut of 1cm per row. Therefore the total effective width of cut by the blades is 28cm.

2.2 DESIGN CONSIDERATIONS

From the design point of view- cutting blades, crank shaft, reciprocating mechanism and sprocket and chain were the important components to be designed for the mechanical weeder. The following deductions were made and adapted for the effective design of this mechanical weeder.

1.The Prime mover

The weeder is to be moved via the prime mover.

2.The draft requirement of the machine

This factor is influenced by the following four main groups of factors according to Gordon spoor [5]

a.Soil/Soil parameters, soil/metal parameters, implement shape and forward speed.

(a) From it is apparent that minimizing the normal load in frictional soils and the area in cohesive soils will minimize the soil shear strength and hence, the draft force. Vibrations, can reduce the apparent normal load in frictional soils [5]. Hence a reciprocating mechanism is used in this project to generate vibration.

(b) Soil/Metal parameters: Removing the rust from a tine by grinding can reduce angle of soil/interface friction.

(c)The blade's shape: The tine rake angle has a very large effect on the draft [5]. The draft increases slowly from rake angles of to. Therefore for minimum draft, cutting below soil surface, good disturbance of weed, both for sparsely and intensely distributed weeds, a rake angle of was selected.

(d)Increased forward: speed increases draft with most implements, hence in this design, and from a work study, a forward speed of 1m/s was selected.

3.The control of shallow weeds can be best achieved by cutting below them and lifting them onto the surface, and for this, a wide tine type of tool would seem to be the most appropriate. [5]

4.The shapes and dimensions of the tools are such as should maximize strength, minimise weight and cost of production.

5.The soil conditions under which the machine will be adapted are cultivated medium soil to light soils. The soil is assumed to be stump free without stones and tough roots. According to Ademosun 1991, typical parameters of agricultural soils on which a metallic soil engaging implement is applied is used as follows:

$\phi = angle\ of\ shearing\ resistance\ of\ soil = 30^{0}$

$\delta = angle\ of\ soil/interface\ friction = 10^{0}$

$c = cohesion = 10 - 15\ kN/m^{2}$

$C_{a} = adhesion = 3.5\ kN/m^{2}$

$\sigma = soil\ unit\ weight = 17.3\ kg/m^{3}$

Surcharge pressure = 0

maximum working depth of machine = 2.54 cm

Designed width of cut: 28 cm

Assumed speed of operation = 1m/s

6.The frame of Machine

The frame forms the mounting support of all other units of the weeder. Hence, stability and strength were the two major points of considerations.

7.The crank mechanisms

The lengths of the links comply with the conditions that is four bar parallel crank mechanism [6]

3. DESIGN ANALYSIS

3.1 Design of Soil Cutting Blades

According to Godwin (2007), the rake angle of the cutting blade, should be greater than zero so as to achieve weed cutting below the soil surface and good disturbance of both sparsely and densely distributed weeds, but it should be as low as possible in order to minimize draught. The obtained values will be used to determine the draught, bending moment, and moment of inertia of the blade. The blade would cut the weed just immediately below the soil surface.The blade thickness was designed on the basis of drag/forces acting on it [7].

Therefore, from the expression for the actual soil factor, N, is given by

$$N = N_\delta = 0 \quad \frac{[N_\delta = \phi\ ]^{\frac{\delta}{\phi}}}{[N_\delta = 0]}$$

The actual soil weight, cohesion and adhesion factors are calculated to be 1.41, 1.86 *and* 0 respectively. Therefore, soil force per unit width of blade is given as:

$$F_U = \sigma b^2 N_\sigma + C_a b N_C$$

Where, F_U = *soil force per unit width of blade*

σ = *soil unit weight* = $17.3\ kg/m^3$

b = Working depth of blade = $2.54cm$

N_σ = *Soil weight factor* = 1.41

N_C = *Cohesion factor* = 1.86

C_a = *cohesion* = $10kN/m^2$

$$F_U = 0.488KNm^{-1}$$

Therefore, draught per unit width of blade is given as:

$$D_U = F_U \sin(\alpha + \delta) + b C_a \cot\alpha$$

$$D_U = 1.1423KNm^{-1}$$

Total Draught = $D_U = 1.1423KNm^{-1}$ × width of blades, = $1.1423KNm^{-1}$ × 0.28m= 0.64 kN

Power requirement = Force × velocity

Assuming a factor of safety 2.5, then the required power for blades = 1.6 kW

When $\delta = 0, \quad m = 3.40$

When $\delta = \phi = 30^0, \ m = 3\ 45$

Where m is the soil rupture distance ratio. Hence, applying similar analysis as applied for soil factor, the actual soil rupture distance ratio is 3.4165

Therefore, the actual weed rupture distance, f, is 8.68cm

3.2.1 Determination of bending moment for the blade

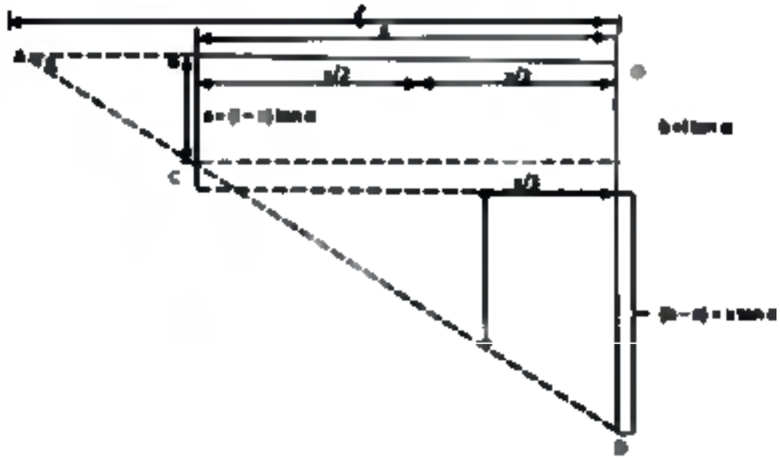

Figure 1 Analysis of the centroid of the tine

Considering the soil directly on top of the cutting blade, from point 0, the centroid of the section OBCD, $\bar{X}$ is given by:

$$\frac{x}{2}(a+b) \quad \bar{X} = ax\frac{x}{2} + {}^{1}/_{2}\,(b-a)\,x \times \frac{x}{3}$$

$$i.e. \quad \bar{X} = \frac{(2a+b)x}{3(a+b)}$$

$$\bar{X} = \frac{(2(l-x)\tan\alpha + l\tan\alpha)x}{3(l-x)\tan\alpha + l\tan\alpha)}$$

$$\bar{X} = \frac{(3l-2x)x}{3(2l-x)}$$

If the load intensities at points 0 and A are respectively w_0 and w_a, the average load intensity over the length $l = \frac{W_0}{2}$

Therefore, the total vertical force on the blade, v_t, is given by

$$V_t = \frac{W_0 L}{2}, \quad \therefore W_0 = \frac{2V_t}{L}$$

Also, the average load intensity over the length x

$$= \frac{W_0 + W_a}{2}$$

Since $\frac{W_a}{L-X} = \frac{W_0}{L}$

$$i.e. \quad W_a = \frac{W_0(L-X)}{L}$$

Therefore, the average load intensity over the length x

$$= \frac{2W_0 L - W_0 X}{2L}$$

Total load acting over the length x

$$= \frac{(2W_0 L - W_0 X)X}{2L}$$

i.e. The shearing force at point A

$$= \frac{-W_0 X(2L - X)}{2L}$$

i. The bending moment at point A

$$= \frac{-W_0 X(2L - X)(X - (3L - 2X)X)}{6L(2L - X)}$$

i.e. The bending moment at point A

$$= \frac{-fx^2(3L - X)}{6L}$$

At $X = 1$, the bending moment at point A

$$= \frac{-FL^2}{3} = \frac{2}{3} V_t L$$

i.e. The bending moment for the blade of length l is given as

$$M_b = \frac{2}{3} V_t cos\alpha L$$

The total vertical force on the blade is given as

$$V_t = W(F_u \cos(\propto + \delta) + bc_\alpha$$

$$V_t = 4914 \times 10^{-2} KN.$$

Hence, the bending moment for the blade, M_b, is

$$M_b = \frac{2}{3} V_t cos\alpha L$$

$$M_B = 9.51 \times 10^{-3} KNM^{-1}.$$

3 2.3. Determination of the blades thickness

From bending stress equation, $I = \frac{M_b h}{2\sigma} = \frac{Wh^3}{12}$

$$h = (\frac{12M_b}{2\sigma w})^{0.5}$$

Where,

w is the width of the blade.

h is the thickness of the blade.

σ is the normal stress i.e 0 constant for steel.

Therefore, the actual thickness of the blade is given as

$$h = c\left(\frac{12M_b}{2\sigma w}\right)^{0.5}$$

Where c is the factor of safety.

$h = 2.88mm.$

3 3.Design of the effective length of steering shaft

This design is based on Euler theory

- Shaft diameter (Solid) = 25mm

Required Data:

1. Weight of the front tyre (wheel) P, Mass = Volume XDensity

   Density of tyre = 650kg/m^3

Volume of tyre = $2\pi^2 Rr^2$ = 0.0037m^3

Weight of the tyre P = 23.6061

2. I_X = moment of Inertia

$$= \frac{\pi}{64} \times d^4$$

$= 19.177 \times 10^3$ mm

3. A = area
$= \frac{\pi}{4} d^4$ = 490.9372mm^2

$$\lambda = \frac{l}{r} \quad \text{......5}$$

Where;
l = maximum allowable length (effective length)
λ = slenderness ratio
r = radius of gyration

$$r = \sqrt{\frac{I_x}{A}} \quad \text{......6}$$

= 6.25mm

$$\sigma = \frac{P}{A} \quad \text{......7}$$

= 0.048N/mm^2
From permissible stress table in axial compression, corresponding to this value of σ = 0,048 N/mm^2 And Yield stress for mild steel, f_y = 250N/mm^2

λ = 39.52
from equation 5;
$l = \lambda \times r$ = 0.355m ≈0.4m
The effective length for the steering shaft to withstand load during operation and easy movement is 0.4m

3.4 Design of the crank shaft

Required Data

1. Load of sprocket on shaft Q_s = 8.13N
2. The weight of small sprocket on shaft weight $= Volume \times density \times 9.81$

The material selected for the sprocket is mild steel. Density of mild steel as previously calculated= 7821.95kg/m^3

$$Volume = \frac{\pi d^2 h}{4}$$

Where, $d = 60mm$, $h = 0.01m$
$Volume = 2.83 \times 10^{-5} m^3$.
Therefore, weight of small sprocket on shaft
$= 2.171N$.

3. weight of crank

Material for the crank is Mild steel, Hence density= 7821.95kg/m^3

a. Volume of crank= $L \times B \times H = 3 \times 10^{-4} m^3$.

Hence, weight of crank= 23.02N.

4. Load on shaft

Total load on shaft = $load\ of\ crank + force\ on\ the\ 3tines.$
To calculate maximum load on crank, $F = mw^2r$
Where,

$M = mass\ of\ the\ reciprocating\ part\ (the\ frame, blade\ and\ support)$

i. Mass of blades = $density \times volume \times number\ of\ blades$

Volume of blade = $1.95 \times 10^{-5} m^3$.

The blades are of mild steel, Hence, density = $7821.95 kg/m^3$.

Mass of blades = $0.610 kg$.

ii. Mild steel blade's handle

Diameter = $0.02m$, Height = $0.4m$

$$Volume = \frac{3.142 \times 0.02^2 \times 0.4}{4} = 1.26 \times 10^{-4} m^3.$$

Therefore, Mass of frame = $1.26 \times 10^{-4} m^3 \times 7821.95 \times 4$

Mass = $3.93 kg$.

iii. Mild steel frame made up of flat bars and channels.

To calculate the volume for portion (a) of flat bar

$Volume = (l \times b \times h) - (volume\ of\ bore)$

$$= (0.3 \times 0.05 \times 0.005) - \frac{3.142 \times 0.064^2 \times 0.005}{4}$$

$$= 7.5 \times 10^{-5} - 1.61 \times 10^{-5}$$

$$= 5.89 \times 10^{-5} m^3$$

To calculate the volume for portion (c) of flat bar

$Volume = (l \times b \times h) - (volume\ of\ bore)$

$$= (0.3 \times 0.05 \times 0.005) - \frac{3.142 \times 0.064^2 \times 0.005}{4}$$

$$= 7.5 \times 10^{-5} - 1.61 \times 10^{-5}$$

$$= 5.89 \times 10^{-5} m^3$$

To calculate the volume for portion (b) and (d) of flat bar = $2b$

$Volume = (0.3 \times 0.02 \times 0.005)\ m^3 \times 2$

$Volume = 3 \times 10^{-5} m^2 \times 2 = 6 \times 10^{-5} m^3$

$\therefore$ Total volume of frame = $volume\ of\ (a) + volume\ of (c) + 2 \times volume\ of (b)$

$Total\ volume = 17.8 \times 10^{-5} m^3 = 1.78 \times 10^{-4} m^3.$

Hence, Mass = $Volume \times density$

$Mass = 1.392 kg.$

iv. Mass of Anchor of blade to blade holder.

$Volume\ of\ anchor = 1.2 \times 10^{-4} m^3.$

$Mass = volume \times density$

$Mass = 1.2 \times 10^{-4} \times 7821.95 kg/m^3$

$Mass = 0.94 kg.$

v. Mass of the connecting Rod.

$Volume = 3 \times 10^{-5} m^3.$

$Mass = 3 \times 10^{-5} m^3 \times 7821.95 kg/m^3$

$Mass\ of\ the\ connecting\ rod = 0.235kg.$
Therefore, mass of the reciprocating part $=$

$$Mass\ of\ blades + Mass of\ blade\ handle + Mass\ of\ frame + Mass\ of\ connecting\ rod.$$

$Mass\ of\ the\ reciprocating\ part = 8.34kg.$
Calculation of the frame hind support

$$Volume = \frac{\pi d^2 h}{4}$$

Where, $d = 0.2m, \quad h = 0.05m$
$Volume = 1.57 \times 10^{-4}m.$
$Height = 1.23kg.$
Therefore, $to\ calculate\ F = MW^2 r,$
Where, $W = angular\ speed = 2\pi n \qquad (n = 200rpm.$

$$r = \frac{25}{2} = 12.5cm = 0.125m.$$

$F = 445.37N.$
Force on the tine as calculated previously $= 4.86 \times 10^{-2}kN\ Per\ blade$
$Number\ of\ blades = 2$, Therefore, total force on blade $= 9.72$N
Total load of crank on shaft $= 445.37N + 0.1944$N $= 539.7$N.
Since Number of crank on shaft $= 2$.
Therefore, force on each crank$= \frac{539.7}{2} = 269.85N$

$Volume = 1.57 \times 10^{-4}m.$
$Height = 1.57 \times 10^{-4}m \times 7821.95kg/m^3$
$Height = 1.23kg.$
Therefore, $to\ calculate\ F = MW^2 r,$
Where, $W = angular\ speed = 2\pi n \qquad (n = 200rpm.$

$$W = 2 \times \frac{22}{7} \times \frac{200}{60} = 20.9rad/s.$$

$$r = \frac{25}{2} = 12.5cm = 0.125m.$$

$F = 8.34 \times 20.9^2 \times 0.125 = 445.37N.$

Force on the tine as calculated previously $= 4.86 \times 10^{-2}kN\ Per\ blade$
$Number\ of\ blades = 2$, Therefore, total force on blade $= 9.72$N
Total load of crank on shaft $= 445.37N + 0.1944$N $= 539.7$N.
Since Number of crank on shaft $= 2$.
Therefore, force on each crank$= \frac{539.7}{2} = 269.85N$

1. ***Procedure***

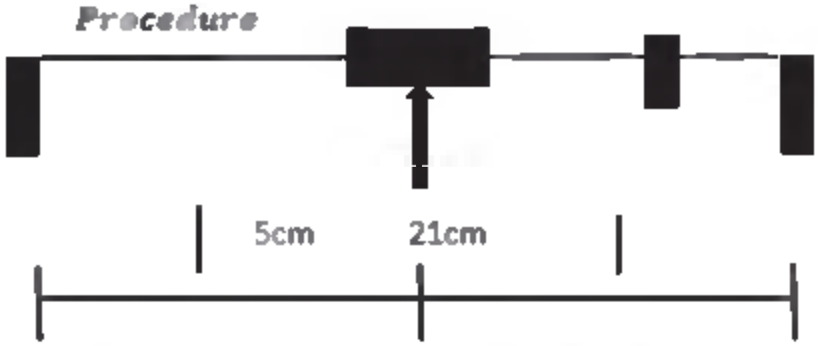

Fig. 3 2 Vertical loading

2. ***Vertical loading***

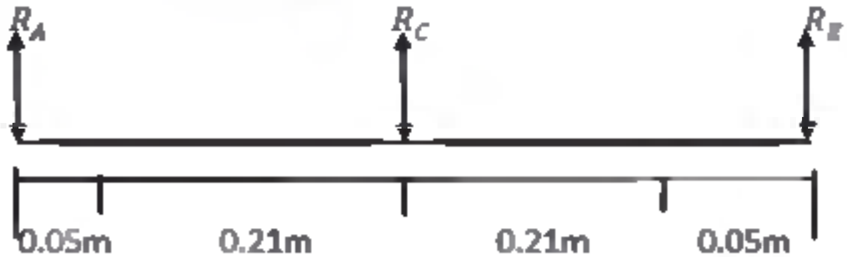

Fig. 3.1 Vertical loading

$$R_B + R_D = R_A + R_C + R_E$$

$$R_B = R_D = \frac{R_A + R_C + R_E}{2}$$

Where, $R_A = 23.02N$, $R_C = 2.171N$, $R_E = 23.02N$

$$R_B = R_D = \frac{23.02N + 2.171N + 23.02N}{2} = 24.1N$$

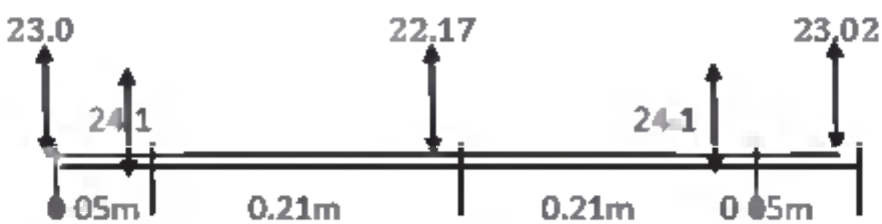

Fig 3.2 Vertical loading.

Vertical Moment Diagram

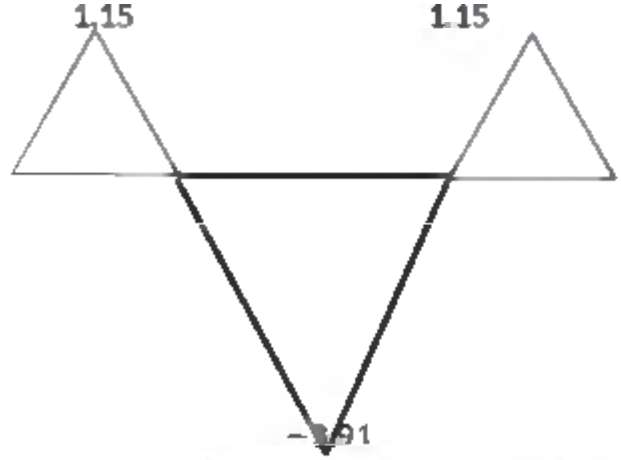

Fig 3.3 Vertical Moment Diagram.

Hence, vertical bending moment $M_V = -3.91Nm$.

1. ***Horizontal Loading***

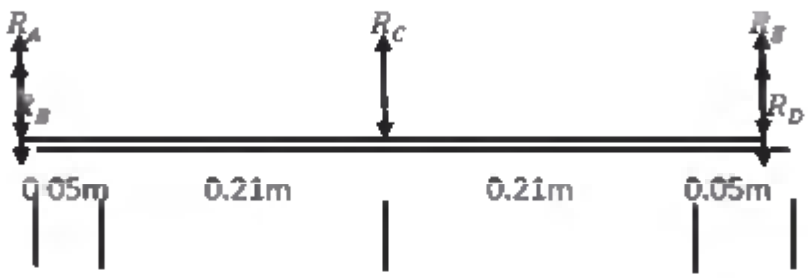

Fig 3 4 Horizontal loading.

$$R_B + R_D = R_A + R_C + R_E$$

$$R_B = R_D = \frac{R_A + R_C + R_E}{2}$$

Where. $R_A = 269.85N$. $R_C = 8.181N$. $R_E = 269.85N$

$$R_B = R_D = \frac{269.85N + 8.18N + 217.25N}{2} = 273.94N.$$

Horizontal Moment Diagram

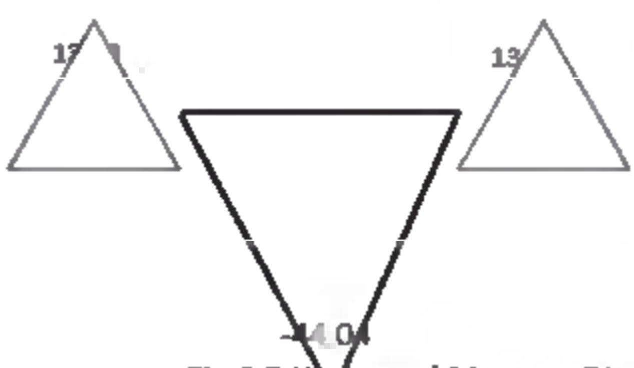

Fig 3.5_Horizontal Moment Diagram.

Hence, maximum bending moment $M_h = -44.04Nm$.

1. <u>Resultant bending moment M_b</u>

$M_b = 44.2$

2. <u>Torsional moment M_t</u>

Where, $M_t = \frac{P}{2\pi n}$

$M_t = 0.2401$

Diameter of shaft, d

$$d^3 = \frac{16}{\pi \delta s}\sqrt{(K_b M_b)^2} + \sqrt{(K_t M_t)^2}$$

Where,

$d = diameter$

$M_t = Torsional\ moment\ Nm$

$M_b = Bending\ moment\ Nm$

$K_b = Combine\ shock\ and\ fatigue\ factor\ applied\ to\ bending\ moment$

$K_t = Combine\ shock\ and\ fatigue\ factor\ applied\ to\ Torsional\ moment$

$\delta s = Allowable\ stress\ 40 \times 10^{-6}\ for\ steel.$

$d = 34mm\ shaft.$

3.5 Design of the Reciprocating mechanism

The design of the reciprocating mechanism, involves the determination of the optimum length of crank that will allow for weeding with minimum overlapping of the weeding surface and also the travel speed correlating with reciprocating motion of the cutting blade.

Diameter of the wheel = $40cm$

Speed of travel = $1m/s$.

Let the length of crank be xcm.

The distance covered by the wheel at 1 revolution = $\pi \times 40cm = 126cm$.

But engine speed = $3000rpm$.

The speed of the reducing gear $15 : 1$. Therefore, the rpm of the crank, which is attached to the shaft of the reducing gear = $200rpm$.

If the speed of rotation of crank $i.e\ nc$ is 5 times that of the wheel $i.e\ nc = 5nw$.

hence, when $\pi w = 1 revolution = 126cm$.

$$nc = 5\ revolution\ of\ blades = 10 strokes$$

$$i.e\ in\ 1\ revolution\ of\ crank = 2 strokes\ of\ the\ blade$$

Therefore, the wheel covers = $\frac{126cm}{5} = 25.2cm$.

To attain an overlap of $3cm$ of cut by the blade, the total length travel of the blade should exceed the distance covered by the wheel.

$25.2cm + 3cm = 28.2cm$.

Therefore, the length of crank to achieve this equals

$3 + (2x) = 28.2cm$

$2x = 28.2cm - 3cm$

$2x = 25.2cm$

$x = \frac{25.2cm}{2} = 12.6cm.$

3.6 Design of the Sprocket and Chain

The design is based on the design data by PSG Tech 2016 [13]

i. Chain pitch = $\frac{Sprockets\ centre\ to\ centre\ distance}{Distance\ between\ centre\ of\ sprockets\ in\ Pitch}$

$a = [30\ to\ 50]p$

Where,

$a = Optimum\ centre\ distance, mm.$

$p = Pitch\ of\ chain, mm.$

Selecting a centre distance of $210mm$,

Therefore chain pitch p = $\frac{210}{30} = 7mm.$

From the design data table, 8.00 pitch is close to the calculated value. From the table, a chain having a pitch of 8.00 has a projected Bearing area of $0.22cm^2$ and a minimum breaking load of $800kgf$ and weight per meter value of $0.33kgf$.

ii. $\frac{Z_2}{Z_1} = \frac{N_1}{N_2} = i$

From the table selected, transmission ratio $i = 5$

$rpm\ of\ Engine = 3000rpm$

$reducing\ gear\ box = 15:1$

Where,

$Z_2 = Number\ of\ teeth\ of\ large\ sprocket = 115$

$Z_1 = Number\ of\ teeth\ of\ small\ sprocket = 23$

$N_1 = Speed\ of\ rotation\ of\ small\ sprocket = 200rpm$

$N_2 = Speed\ of\ rotation\ of\ small\ sprocket = 40rpm$

Available sizes of sprockets with transmission ratio 1 : 5 $= 60mm : 300mm$

Where, $d_1 = 60mm$, $d_2 = 300mm.$

the design power transmitted on the basis of allowable bearing stress

$$N = \frac{\delta .\ A.\ V}{102k} KW$$

Where,

$\delta = allowable\ bearing\ stress\ [kgf/cm^2]$

$A = Projected\ bearing\ area\ cm^2$

$V = chain\ veleocity\ m/s$

$k_s = Service\ factor = 1.25$

δ From the table, speed of rotation of small sprocket $i.e\ n_1 = 200rpm$

$\delta = 3.15$, $A = 0.22$, $V = ?$

To get V $i.e\ v = \frac{\pi d_2 n_2}{60}$

Where, $d_2 = 300mm = 0.3m,\ n_2 = 200rpm$

$V = \frac{1320}{420} = 3.14m/s.$

iii. Design power transmitted on the basis of allowable bearing stress

$$N = \frac{\delta\ .\ A.\ V}{102k} = 0.0171kw$$

Therefore, design per strand = $\frac{Design\ power}{multiple\ strand\ factor}$

$Design\ power = Design\ per\ strand\ \times multiple\ strand\ factor$

$= 1 \times 0.0171kw = 0.0171kw.$

iv. **The load on shaft is given by**

$Q_a = K_1 \times P_1$

Where,

$Q_a = Load\ of\ shaft\ due\ to\ chain$

$K_1 = Load\ factor$

$P_1 = Tangential\ force\ due\ to\ power\ transmission\ kgf$

$$P_1 = \frac{102N}{V}\ if\ N\ is\ kw$$

$P_1 = 0.5555kgf$

Load factor $K_1 = 1.5$

$Q_a = 1.5 \times 0.5555$

$Q_a = 0.8333kgf$

$Q_a = 0.8333 \times 9.81$

$Q_a = 8.175N.$

v. Length of chain

$$= 2a_p + \frac{Z_{1+Z_2}}{2} + \frac{\left(\frac{Z_{2-Z_1}}{2\pi}\right)^2}{a_p}$$

Where, $a_p = \frac{a_o}{p}$

$a_p = Approximate\ centre\ distance\ in\ multiple\ of\ pitches$

$a_o = Initially\ assumed\ centre\ distance, mm$

$p = Pitch, mm$

$a_p = \frac{210}{7} = 30mm$

Therefore, length of chain:

= 136.14cm

3.7 AUTOCAD DETAILED DRAWING

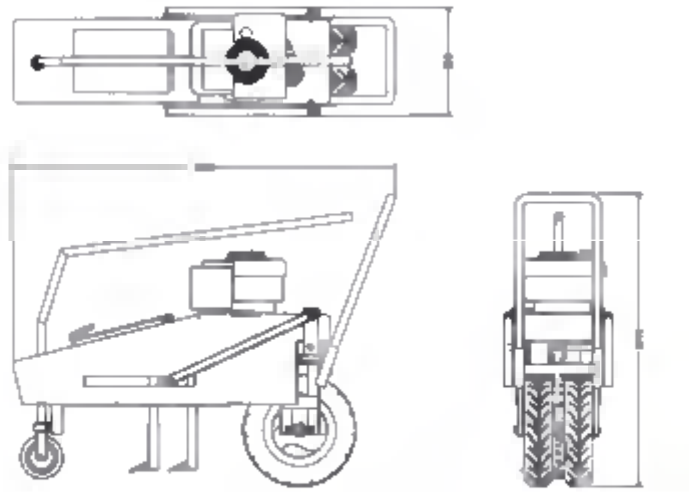

Fig. 2. Orthographic drawing of the machine

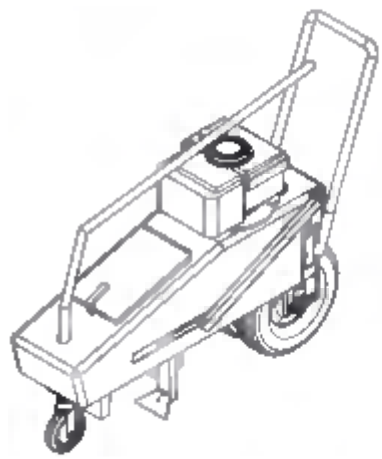

Fig. 3. Isometric drawing of the weeding machine

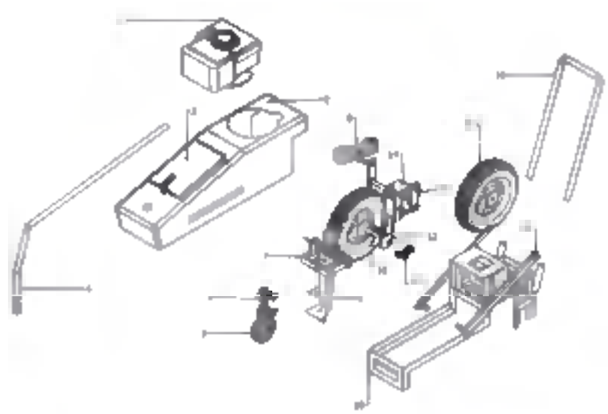

Fig. 5. Detailed/exploded view of the weeding machine

3.8 Machine Testing

The following efficiencies were determined when testing the machine on the farm according to Terari . Each experiment was replicated thrice

A plot 10m x 10m in size was selected. The weeds on plot was weighed and the weeding was done manually and with use of the machine. Time taken to complete the weeding operations was noted.

1. The Weeding efficiency (functional efficiency).

$$functional\ efficiency = \frac{Weight\ of\ weeds\ removed\ on\ the\ farm}{Actual\ weight\ of\ weeds} \times 100$$

2. The Field capacity = Area weeded/time taken

Fig. 6. The machine showing the internal parts.

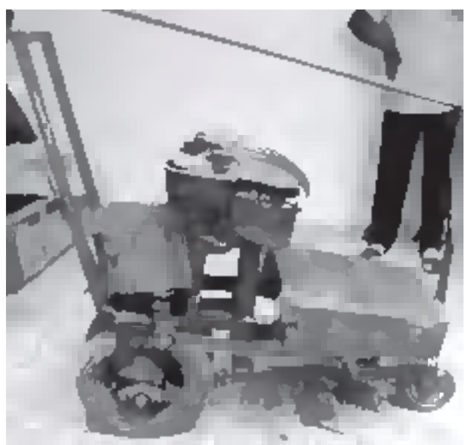

Fig. 7. Picture of the machine complete with housing

4. RESULT AND DISCUSSON

The following efficiencies were determined when testing the machine on the farm.

The Weeding efficiency (functional efficiency).

$$functional\ efficiency = \frac{Weight\ of\ weeds\ removed\ on\ the\ farm}{Actual\ weight\ of\ weeds} \times 100$$

Functional efficiency = 82 70 %

The Field capacity. For the machine = $0.03m^2/s$ while the manual weeding was $0.01m^2/s$.

The following were the reason for the above performance.

Occasionally slips, function of speed, rough and uneven ground terrains

4 CONCLUSION

A reciprocating weeder was designed and fabricated for commodity crops. Preliminary test result indicated a weeding efficiency of 82.7%, and field capacity of 0.03m^2 /s as against 0.01m^2/s with manual weeding. Powered by a 5 kW compression-ignition internal combustion engine (CI-ICE), the weeding machine has an effective cutting width of 28 cm and a production cost of NGN 175, 000.00 (USD 583). The performance of the weeder can be improved by replacing mild steel with aluminum alloy materials on various identified components of the machine where loads are not concentrated.

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