DESIGN AND DEVELOPMENT OF TRACTOR DRAWN CARROT DIGGER

[1]Milkessa Alemu, [2]Solomon Abera

Oromia Agricultural Research Institute, Fedis Agricultural Research Center P.O.Box 904, Harar, Ethiopia[1], Department of Food Technology and Process Engineering, Haramaya Institute of Technology, Haramaya University, Dire Dawa, Ethiopia[2]

milkii957@gmail.com[1], aberasol22@gmail.com[2]

ABSTRACT

Carrot is one of the widely produced and consumed root vegetables in Ethiopia. Carrot harvesting was practiced by traditional method using metal hoe, which is characterized by high man-hour requirement, drudgery and considerable losses in terms of root damage and inefficient exposure. Thus, it was important to consider a small horsepower tractor drawn carrot digger for Ethiopian smallholder farmers was designed and developed by considering soil, root and machine parameters. The machine mainly consists of main frame, shank, power transmission system, V-shaped digger blade, eccentric pin and soil separator units. Some physical and mechanical properties of the carrot roots and soils, relevant to the design of the digger were also studied. The study was focused on creating a cost-effective and locally adaptable machine suitable for small- and medium-scale farmers. Engineering design principles, including force analysis, material selection, and ergonomic considerations, guided the development process. Overall, the tractor-drawn carrot harvester provides a practical mechanization solution, offering improved productivity and reduced production costs for farmers.

Keywords: Design, Development, fabrication, digger, soil separator

INTRODUCTION

Background

Carrot is an important crop and used by millions of people throughout the world. It is an important vegetable that contain carotene and a precursor of Vitamin A which keep human healthy, resistance against infectious diseases in human body. It is one of the widely produced and consumed root vegetables in Ethiopia. According to [1] in 2020 total area allocated for vegetable production in Ethiopia, about 1.1 % was covered by carrot with 101,48.23 tons of root yield was harvested.

The main reason for low productivity and quality in vegetable production is low adoption of improved cultivation practices at farmer's level. One of the constraints to increasing both areas under vegetable crops and its productivity is the low level of mechanization. In vegetable farming, agricultural operations like planting, weeding, and harvesting are more labour intensive [2].

Besides the quantum of labour, manual harvesting involves considerable drudgery and human discomfort. The labour has to stoop forward while digging/pulling carrots from the bed and also during picking up. Stooping posture results in a lot of bio-mechanical stresses in the back and has higher energy consumption as compared to other working positions [3].

Traditionally, carrots have been harvested manually using hand tools, which is labor-intensive, time-consuming, and increasingly costly due to labor shortages. These limitations have encouraged the development and design of improved carrot harvesting systems, particularly mechanized and semi-mechanized harvesters [4].

The design of carrot harvesting equipment is greatly influenced by the physical and mechanical properties of the crop. Parameters such as root length, diameter, mass, tensile strength, and the bond between the root and soil determine the force required for harvesting. Carrots are relatively fragile and susceptible to breakage; therefore, harvesting machines must be designed to minimize damage such as cracking, bruising, and surface abrasion. Mechanical damage not only lowers market value but also accelerates post-harvest deterioration [5].

Modern carrot harvesting machines generally consist of three main functional units: soil cutting or loosening, lifting or pulling, and cleaning or topping. The choice of harvesting design depends on factors such as carrot variety, planting pattern and soil type. Small-scale farmers often favor simple and low-cost harvesting designs, whereas large commercial farms prefer fully mechanized systems with higher operational capacity [6].

Recent research on carrot harvesting design emphasizes improving harvesting efficiency, reducing energy consumption, and minimizing crop damage while adapting machines to small and medium-sized farms. There is also growing attention to ergonomic design, sustainability, and the use of locally available materials to lower production costs. These design improvements aim to enhance productivity, maintain crop quality, and support sustainable agricultural mechanization [7].

From an engineering and sustainability perspective, the carrot harvesting machine improves field efficiency, energy utilization, operational precision and sustainable agricultural practices. Therefore, the adoption and development of carrot harvesting machines is justified to enhance productivity, profitability, and sustainability in carrot production systems and this project was aimed to design and develop tractor drawn carrot harvester

MATERIALS AND METHODS

Design Considerations

A tractor drawn carrot harvester was developed as both a functional and experimental unit. The components of the harvester were designed according to operational needs and fundamental working principles.

Description of the Machine

The major parts were soil cutting blade, frame, three-point hitches, shanks, soil separator, depth gauge, input shaft, eccentric pin and connecting rod (figure 1).

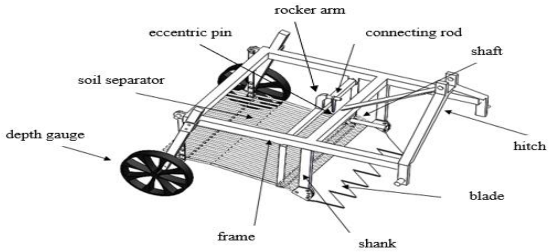

Figure 1: Description of the machine.

Design of the components

Design of frame

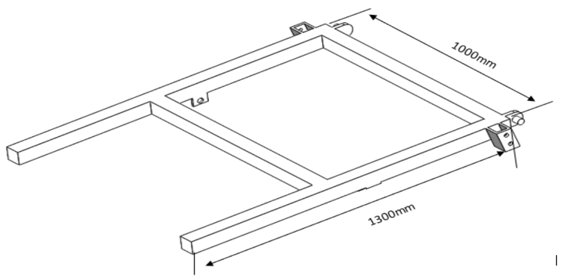

Figure 2: Frame

It is back bone of the machine and all other parts of the digger are mounted on it. The main frame was designed to bear a torsion force developed and was fabricated with mild steel rectangular section (60 × 60 × 6 mm).

Power Required for Soil Cutting

From the designer point of view, the working depth of blade is an important parameter as it affects the power requirement of digger. The working depth of digging blade was mainly dependent on the depth of carrot in the soil. Considering the probable variation in depth of carrot on different varieties of soil and to harvest them without damage, optimum depth of operation was selected as 25cm and width of 90cm which was the depth to width ratio of 0.28. Therefore, it was less than 0.5 so-called wide blades [8].

The draft of the blade was calculated using the general soil mechanics equation for a blade deforming the soil in two dimensions [9] given by Equ. (1).

$$F_s = \left(\gamma d^2 N_\gamma + q d N_q + c d N_c + c_a d N_{ca}\right) \times w \qquad (1)$$

Where:

Fs = soil resistance force, kN

γ = unit weight of soil, kg/m^3

C = apparent cohesion, kN/m^2

Ca = soil-interface adhesion, kN/m^2

d = depth of operation, m

w = width of operation, m

q = surcharge pressure on soil, kN/m^2

Nγ = soil friction cutting coefficient

Nq = soil overburden cutting coefficient

Nc = soil cohesion cutting coefficient

Nca = soil adhesion cutting coefficient

Nγ, Nc, Nq and Nca are dimensionless N- factors, which describe the shape of soil failure surface and this is a function of angle of shearing resistance of soil (Φ), angle of soil metal friction (δ) and geometry of loaded interface *i.e.* rake angle (α).

For determination of draft, the following assumptions were made [10].

- *i.* Soil is homogenous and isotropic,
- *ii.* Average bulk density of soil is 1750 kg.m^{-3},
- *iii.* Soil is in friable range of moisture content with cohesion (C) of 710 kg m^{-2}, angle of internal friction (Φ) of 25° and angle of soil metal friction (δ) of 20° for bulk density of 1750 kg m^{-3},
- *iv.* Adhesion of soil is zero *i.e.* Ca = 0, assuming soil-metal friction to be zero.
- *v.* The surcharge above soil failure zone is negligible, *i.e.* q= 0,
- *vi.* Usual variations in rake angle of the digging blade range between 15 to 25° in the experiments.

Based on the above assumptions, the Equ. (1) could be reduced as follows

$$Fs = (\gamma d^2 N_\gamma + c d N_c) w \qquad (2)$$

The relationship between the N-factor and the rake angle at different angle of internal friction for a perfectly smooth (δ=0) and perfectly rough (δ = Φ) interface was taken from graph [9]. The values of N-factor for intermediate degree of roughness of the interface could be interpolated using the following equation [11]

$$N = N_o[\frac{N_\emptyset}{N_o}]^{\frac{\delta}{\emptyset}} \qquad (3)$$

Where:

Nδ= the required value of the appropriate N factors (Nγ or N c)

Nδ= 0 and Nδ= Φ= the corresponding value of the N-factor at δ= 0 and δ= Φ, respectively obtained from appropriate chart [8]

Following values for the different parameters in the Equ (2). were used for determination of passive resistance of the blade:

$\gamma = 1750\ kg.m^{-3}$, $C = 710\ kg\ m^{-2}$, $\Phi = 25°$, $\delta = 20°$, $\alpha = 25°$, d= 0.25 m

N-factors were taken from universal soil cutting force chart from (Appendix Graph B1.3 and B1.4):

$N_\gamma = 1.25$ when $\delta = 0$, $N\gamma = 1.65$ when $\delta=\Phi$, and NC = 0.48 when δ=0, NC = 1.82 when δ= Φ

Hence, using Equ. (3), Nγ=1.52 and N_c=1.39.

Now, substituting the values of Nγ and Nc in Equ. (2), passive resistance (Fs) is given as:

$$Fs= (1750kg.m^3 \text{ x } (0.25)^2 \text{ x } 1.83 + 710kg.m^2 \text{ x } 0.25 \text{ x } 1.68) \text{ x}0.9m = 3807.92N$$

Therefore, Fs for an effective width of cut of 0.90 m of blade is 3807.92N

According to [8] the passive resistance force (Fs) is loaded at an angle of friction (δ) to the interface of a blade. Hence, the component of forces in parallel (Fbh) and perpendicular (Fbv) to the blade face was given below.

$$Fbh = F \sin \delta \qquad (4)$$

$$=3807.92N \text{ x } \sin 20^0$$

$$=1302.38N$$

$$Fbv =F \cos\delta \qquad (5)$$

$$=3807.92N \text{ x } \cos 20^o$$

$$= 3578.27N$$

The power required for soil cutting (Psc) was using Equ. (6):

$$Psc = F \text{ x} V \qquad (6)$$

Where:

Fp = soil resistance force, kN

V = operating speed, km/hr.

$$Psc = 3807.92N \text{ x}0.97m / s$$

$$= 3693.68W$$

The obtained value of F_{bh} and F_{bv} were used to determine the bending moment of the digger blade.

The draft of the blade was calculated 3807.92N using the general soil mechanics equation for a blade deforming the soil in two dimensions [9] given by Equ. (2). Usual variations in rake angle of the digging unit ranged between 15 to 25°.

Design of digging unit

The digger blade width was selected to cover all plant rows in a bed without crop damage, based on a three-row planting system. Blade thickness was determined by the acting load, and the blade was fabricated from mild steel. The cutting edge was sharpened to a 16° taper [8] with a V-shaped front cutting face as recommended by [12]. The blade was screw-mounted to the shank for easy replacement, and its rake angle was adjustable from 0–30° using a semi-circular slot for lifting and lowering.

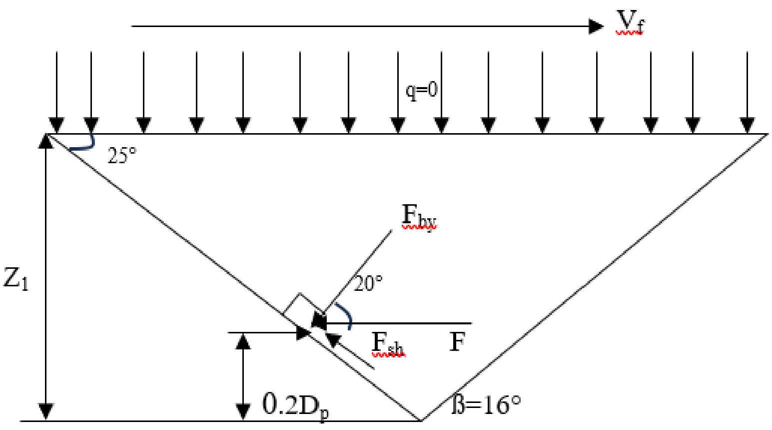

Figure 3: Diagram of the soil reactions acting on a blade.

Where:

Fs= Passive soil resistance

F_{sh} = Component of Fs parallel to the blade face

F_{by}= Component of Fs perpendicular to the blade face

D_h= Horizontal component of Fs (draft)

β= Tip angle of blade, degree

Z_1= working dept of blade

Q= Surface pressure

δ= Coefficient of friction, degree

ϕ= Coefficient of internal friction, degree

α= Rake angle of blade, degree

V_f= forward speed of travel

The perpendicular component (Fbv) of soil load can be caused bending moment, whereas the horizontal component (Fbh) induced direct stress on the blade. The force acts at the center of resistance of the blade as shown in figure 3. Average soil resistance of the blade acts at a distance of 0.2 of the depth of cut measured from the cutting edge [13]. The blade is supported on the shank at a distance of 250mm from each side of the cutting edge. Therefore, distance between the center of resistance and the point of support (Dr) could be determined by:

$$D_r = D_c - 0.2D_c \qquad (7)$$

$$D_r = 250mm - 0.2x250mm$$

$$= 200mm$$

According to [8] soil resistance force (F_s) could be resolved into components of forces perpendicular (F_{bv}) and parallel (F_{bh}) to the surface of a blade. The perpendicular component (F_{bv}) and the horizontal component (F_{bh}) of Fs cause bending and direct stress respectively. Equ. (8) gives the bending moment due to vertical force (F_{bv}) on the blade:

$$BM = F_b \times d_r \qquad (8)$$

$$BM = Fbv \text{ x} 200$$

$$= 3578.27\ N \text{ x } 200mm$$

$$=715654 Nmm$$

Bending stress on the blade (σ_b) is:

$$\sigma_b = \frac{B.M}{\frac{1}{2}bt^2} \quad (9)$$

Where:

B.M. = Bending moment, N-cm

t = Thickness of the blade, cm

b = Width of the blade at the point of mounting, cm

dr = distance from cutting edge to the center of the resistance, cm

Then,

$$\sigma_b = \frac{715654}{\frac{1}{6} * 900\ t^2}$$

$$= \frac{5933.63\text{N}}{t^2}$$

Direct stress (σ_d) due to F_{bh} could be:

$$\sigma_d = \frac{F_{bh}}{b \times t} \quad (10)$$

Then,

$$= \frac{1302.38N}{900mmxt}$$

$$= \frac{1.45N/mm}{t}$$

Total stress (σt) induced was estimated using Equ. (11):

$$\sigma_t = \sigma_b + \sigma_d \quad (11)$$

$$\sigma_t = \frac{5933.63\text{N}}{t^2} + \frac{1.45N/mm}{t}$$

Usually, the factor of safety (*fos*) for mild steel was recommended to be 1.2 and equating the total stress (σ) with safe stress 58.86 MPa of mild steel [10] the thickness of blade (t) can be determined using quadratic equations theorem.

Hence,

$$\sigma_y = \left(\frac{5933.63\text{N}}{t^2} + \frac{1.45N/mm}{t}\right) xfos$$

$$58.86Mpa = \left(\frac{5933.63\text{N}}{t^2} + \frac{\frac{1.45N}{mm}}{t}\right) x1.2$$

$$t=10mm$$

Hence, thickness of blade was kept 10 mm and the total width of blade was kept 900 mm, as per requirement for digging.

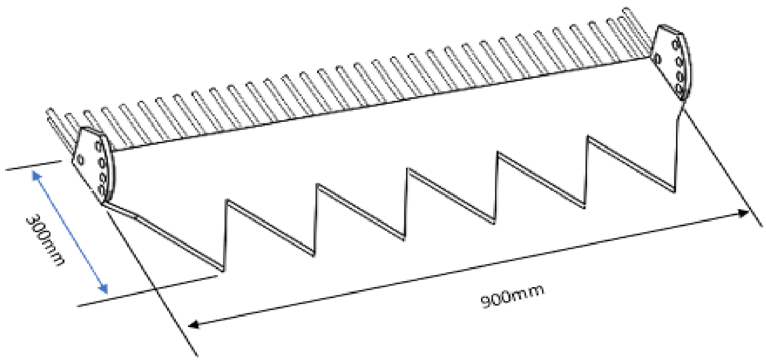

Figure 4: V-type soil cutting blade(unit)

Design of shank

The perpendicular component of the soil load caused direct stress, whereas the horizontal component (drawbar pull) induced bending stress on the shanks [8].

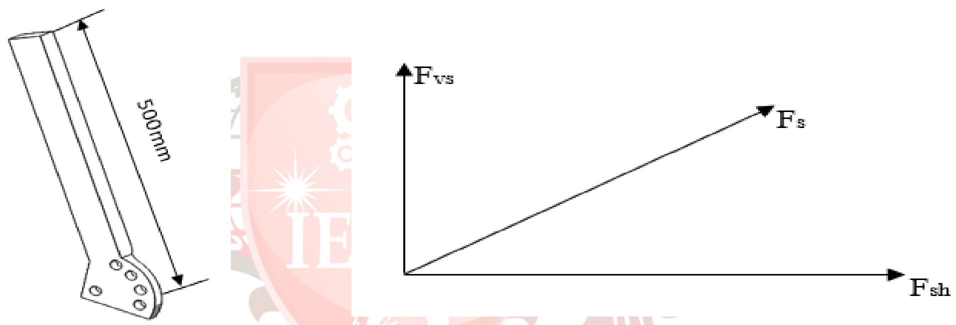

Figure 5: a) shank and b) forces on shank

The force required to move the machine was the sum of the horizontal forces (Fsh) and the vertical force (Fsv) on the blade (figure 5b). Horizontal forces (Fsh) can be found using in Equ. (12).

$$F_{sh} = F_s \sin(\alpha + \delta) + C_a dwcot\alpha \qquad (12)$$

$$=3807.92\text{N} \sin(25^o+20^o) + (0) \times 0.25\text{m} \times 0.9\text{m} \times \cot 25^o$$

$$F_{sh}=2692.6\text{N}$$

Therefore, the bending moment (BMs) on the shank due to Fsh as illustrated in Equ. (13).

$$BM_s = F_{sh} + L_s \qquad (13)$$

$$=2692.6\text{N} \times 550\text{mm}$$

$$=1{,}480{,}930\text{Nmm}$$

Using Equ. (14), bending stress on bade (σb) was:

$$\sigma_b = \frac{B.M}{\frac{1}{6}bt^2} \qquad (14)$$

$$\sigma_b = \frac{1{,}480{,}930\text{Nmm}}{\frac{1}{6} \times 200mmxt^2}$$

$$\sigma_b = \frac{44427.9}{t^2}$$

Direct stress (σ_d) caused by the vertical force of soil using Equ. (15).

$$F_{vs}=F_s \, Cos(\alpha+\delta)+C_a dw \quad (15)$$

=3807.92N cos (25+20) + 0x0.9m × 0.25m

F_{vs}=2692.6N

$$\sigma_b=\frac{F_{vs}}{b\times t}$$

$$\sigma_b=\frac{2692.6\text{N}}{200mm\times t}$$

$$\sigma_b=\frac{13.5N/\text{mm}}{t}$$

Total stress (σ) computed was using Equ. (16).

$$\sigma=\sigma_b+\sigma_d \quad (16)$$

$$58.86\text{Mpa}=\left(\frac{44427.9}{t^2}+\frac{13.5N/\text{mm}}{t}\right)1.2\,,$$

t =30mm

Analysis of drawbar power

The mechanism of the tractor (both the transmission and wheels) is converting the rotary motion of the engine to the linear motion of the drawbar. Drawbar power of a tractor is a product of drawbar force and forward speed [14].

$$P_d=P\times V \quad (17)$$

Where:

Pd = drawbar power, w

P = Drawbar pull, N

V = Travel speed, m/s

The draft is the horizontal component (F_{sh}) of soil cutting force (Fs) [11]. Hence the draft power can be estimate-using Equ. (18).

$$P_d=F_{sh}\times V \quad (18)$$

=2692.6N × 0.97m/s

=2223.8W

Soil reaction force or thrust (H) and equilibrium of the external horizontal forces acting on the tractor and rolling resistance force (R) estimated in Equ. (19) [14].

$$H=F_{sh}+R \quad (19)$$

Then, rolling resistance (R) can be computed in Equ. (20):

$$R=\mu W \quad (20)$$

$m=m_m+m_s$

=97.25kg+47.6kg

=144.85kg

$W=mg$

=114.85kg × 9.81m/s^2

=1126.68N

R=0.124 × *1126*.68N

=139.7N

H=2692.6N × 139.7N

H=2832.3N

[14] also explained comparative slip-pull performance of wheel slip of four- wheel drive tractor on stubble, uncultivated and loamy surface soil at 3138.72 N drawbar pull is about 1-5%. Thus, wheel slip (i) can estimated in Equ. (21).

$$i = \frac{V_O - V}{V_O} \qquad (21)$$

Vo = linear velocity of the ground wheel of a tractor, m/s. 4

$$0.05 = \frac{V_O - 0.97m/s}{V_O}$$

Vo=1.02m/s

Power loss (P_L) of 4WD due to wheel slip (i) and rolling friction is [14] computed in Equ. (22):

$$P_L = P_w + P_d \qquad (22)$$

Where:

Pw = Wheel power, W

Pd = Drawbar power, W

$$P_L = HV_o + PV \qquad (23)$$

=2832.3N × 1.02m/s + 2692.6N × 0.97m/s

P_L=277.13W

The efficiency of power transmission for tractors between transmission inputs to PTO is around 90-92%. The 4WD tractor transmission usually consists of a variable speed V-belt drive from the engine flywheel. A small gearbox may then be fitted, which in turn drives the wheels through chains. Power losses in the mechanical transmission system of 4WD tractors are usually small and less than 8% [14]. Hence, wheel power (Pw) estimated in Equ. (24):

$$P_w = P_{sc} + P_L \qquad (24)$$

Where;

Psc = Power required for soil cutting, (Psc = 3693.68W)

$$P_w = 3693.68W + 277.13W$$

=3970.81W

Power losses in the mechanical transmission of 4WD tractors are usually less than 8% i.e. 92% efficiency [14].

Power at a mechanical system of the tractor (Pt) was:

$$P_t = \frac{P_w}{\eta} \qquad (25)$$

$$= \frac{3970.81\text{W}}{0.9}$$

P_t= 4412.01W

Design of power transmission system

Power is transmitted to the soil-separating unit through a slider–crank mechanism consisting of a propeller shaft, crank, and connecting rod. Power flows in two stages: from the tractor PTO to the crank via the propeller shaft, and from the crank to the separating unit through the connecting rod. The propeller shaft (30 mm diameter, 300 mm length, 10 splines) connects the PTO to the crank through a bearing. The connecting rod is made of mild steel

(250 × 20 × 10 mm) and is linked to the crank by a 45 mm ball bearing, with the other end attached to the reciprocating frame. This mechanism converts rotary motion into reciprocating motion for soil separation. The crank has a diameter of 120 mm and thickness of 10 mm, and its speed depends on PTO speed. Stroke length adjustment is provided through three eccentric positions (0, 60, and 120 mm) offset from the crank center.
The crank radius was decided by the Equ. (26) [15].

$$X = r\left[(1 - cos\theta) + n - (n^2 - sin\theta)^{\frac{1}{2}}\right] \qquad (26)$$

Where:

X = stroke length, 12cm;
r = crank radius, cm;
θ = angular displacement of crank;
n = *l* / r; and
l = connecting rod = 30cm.

For maximum displacement of the unit, θ = 180° hence *cos*θ = -1 and *sin*θ = 0

$$X = r[(1 - (-1) + \left(\frac{30}{r}\right) - ((\frac{30}{r})^{\,2} - 0)^{\frac{1}{2}}$$

$$12 = r[2 + \frac{30}{r} - \frac{30}{r}]$$

r=6cm

For the displacement of 12 cm of the stroke length, the radius of crank was decided as 6 cm. This shows that the stroke length was twice the offset crank radius.

Design of Shaft

A shaft is a rotating machine element that is used to transmit power from PTO to eccentric pin. The power is delivered to the shaft by some tangential force and the resultant torque (or twisting moment) set up within the shaft permits the power to be transferred to various machines linked up to the shaft. The primary determination of the correct shaft diameter to ensure satisfactory strength and rigidity when the shaft is transmitting power under various operating and loading condition [16].

- **Determination of shaft diameter**

In harvesting shaft design case, the crank shaft subjected both twisting and bending moments at the same time. Bending moment due to the PTO, the weight of the harvester and twisting moment due to the power transmission, the diameter of the shaft was calculated as [16].

$${d_1}^3 = \frac{16}{\pi\tau_{max}}\sqrt{(K_bM_b)^2 + (K_tM_t\ )^2} \qquad (27)$$

Where:

${d_1}^3$= diameter of the shaft (mm)
M_t= torsional moment (Nm)
M_b=bending moment (Nm)
K_b= combined shock and fatigue factor applied to bending moment;
K_t = combined shock and fatigue factor applied to torsional moment;

τ_{max} = Allowable Stress (MN/m^2)

According to [17] code for the design of transmission shafts, the maximum permissible shear stress given 40 MPa for shafts with allowance for without keyways. For rotating shafts when the load is suddenly applied (Khurmi and Gupta, 2005b)K_b= 2.0 to 3.0, K_t= 1.5 to 3.0.

Let considered from the weights and forces acting on the shaft. The forces acting on the shaft were considered to be the concentrated load from the PTO shaft and the load from the eccentric pin. The length of shaft 0.30m and bearing was mounted 0.15m from both ends. Weight of the harvester was 953.7N and weight of soil separator 241.24 N.

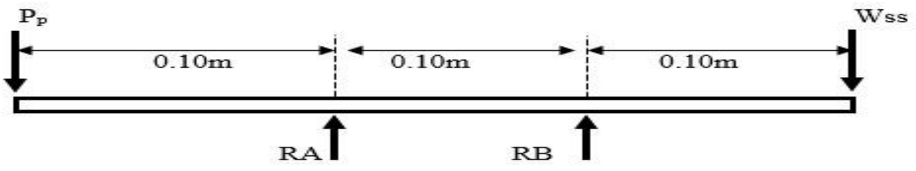

Figure 6: Free body diagram of all force acting on shaft

Where:

R_A=reactions at the support A(N)

R_B =reactions at the support B(N)

W_{ss}= weight of soil separation (N)=241.24N

P_p =PTO power (N) =15.44KN

The reaction forces values were determined by taking the summation of moment at desired location determined as flow:

$$\sum MA=0$$

$$0.1P_p + 0.1 \times R_B - 0.2W_{ss} = 0$$

$$0.1 \times 15440 + 0.1 \times R_B + 0.2 \times 241.24N=0$$

$$1544 + 0.1 \times R_B + 48.25=0$$

$$R_B=15922.5N$$

$$\sum Fy=0$$

$$R_A + R_B - P_p - W_{ss} = 0$$

$$R_A + 15922.5 - 15440 - 241.24$$

$$R_A=241.26N$$

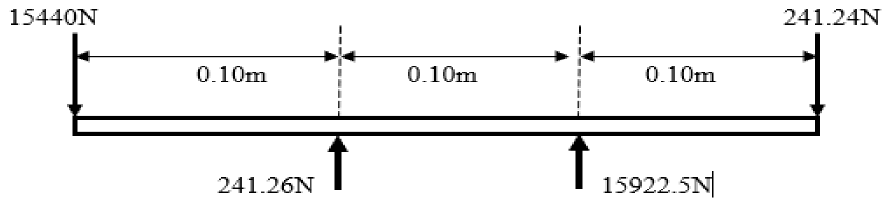

Figure 7: Diagram of all the forces acting on drum shaft

- **Shear force (SF) acting on shaft**

Shear force at point 'A' was 241.26N

Shear force between sections A and C

$$R_A + (-P_p)$$

241.26N+ (-15440N) = -15198.74N

Shear force between sections A and B

$$R_A + R_B - P_p$$

241.26+15922.5-15440 = 723.76N

Shear force between section B and D

$$R_B - W_{ss}$$

15922.5-241.24=18681.26N

Shear force at point D

$$R_A - R_B - P_p + -W_{ss}$$

241.26N+15922.5N-15440N-241.24N= 0N

- **The bending moments (BM) on the shaft were**

The bending moments C=0 Nm

The bending moments at A

$$0.1(R_B + P_p + W_{ss})$$

0.1 (-15922.5 + 15440 + 241.24)

= -24.1Nm

The bending moments at B

$$0.1\left(R_B + P_p + W_{ss}\right) + 0.1(R_A + P_p + W_{ss})$$

$$0.1(\ 15440\ N + 241.26\ N - 15922.5N) +\ 0.1(241.24\text{N} - 15440\text{N} + 15922.5)$$

= -24.1+(-72.4)

=48.3Nm

The bending moments point D= 0 Nm

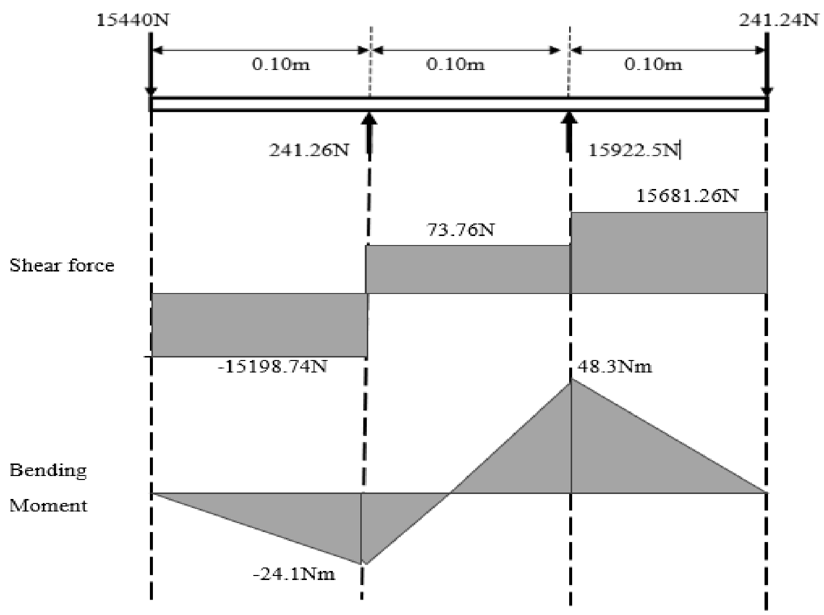

Figure 8: Diagram of Shear force and bending moments the on shaft

The torsional moment was determined as given by [16].

$$M_t = \frac{60 \times P}{2\pi N} \qquad (28)$$

Where:

M_t = torsional moment (Nm)

P = power required (W)

N= maximum PTO speed required (rpm) was taken as 540rpm [18].

Then,

$$M_t = \frac{60 \times P}{2\pi N}$$

$$M_t = \frac{60 \times 3960w}{2 \times \pi \times 540rpm}$$

$$=70.03Nm$$

Now let determine the shaft diameter by Equ. (27).

$$d_1{}^3 = \frac{16}{\pi S_s}\sqrt{(K_b M_b)^2 + (K_t M_t\ )^2}$$

$$d_1{}^3 = \frac{16}{\pi\ 40Nmm2}\sqrt{(\ 2 \times\ 48.3 \times 1000\ Nmm)^2 + (1.5 \times\ \ 70.03\ \times 1000\ Nmm)^2}$$

$$d_1{}^3 = 26.28mm$$

The designed shaft diameter of 30 mm was selected for the shaft by considering 20% safety factors and availability.

Bearing selection

The bearing selected based on its load carrying capacity, life expectancy and reliability. The selection process must consider all factors which affect bearing performance and cost. The static equivalent radial load for radial or roller bearings under combined radial and axial or thrust loads is given by the greater magnitude of those obtained by the following equations [16].

$$C = W\left[\frac{L}{10^6}\right]^{1/K} \qquad (29)$$

Where:

C = Basic dynamic load rating (KN)

L = Rating life (revolution)

W = Equivalent dynamic load (N)

k = exponent for life equation with (3, for ball bearings)

Let considering an average bearing life of 5 years, with 8 hours of operation per day and 60 days per year, the life of the bearing is estimated to be 2400 hours.

$$L_H = 5 \times 8 \times 60 = 2400\text{Hours}$$

$$L = 60\ n \times L_H$$

$$60 \times 800 \times 2400 = 115200000\text{Rev}$$

$$W = x \times\ v \times\ W_R \times\ \ K_S$$

$$W = 0.56 \times 1 \times 627.72N \times 1.5$$

$$W = 527.3N$$

Where:

W_R = Radial load

V = A rotation factor

x = Radial load factor, and

$$C = 527.3\left[\frac{115200000}{10^6}\right]^{1/3}$$

C=18kN

No. 305 bearing was selected which has needed capacities from [16].

Power-Takeoff

Is power required by the implement from the PTO shaft of the tractor or engine. Typical PTO power requirements can be determined using rotary power requirement parameters given in [19]. Implement power take-off power can be calculated as:

$$P_{pto} = a + bw + cF \qquad (30)$$

where:

P_{pto}= power-takeoff power required by the implement kW;

w = implements working width, m;

F = material feed rate, t/h wet basis;

a, b, and c are machine specific parameters [19].

Design of soil separation unit

The design of the soil separator unit was based on its functional requirements. The soil separation unit was placed just behind the blade to receive the dugout crop and soil mass. To separate the soil from carrot, a set of mild steel rods at the rear of the blade were arranged lengthwise along the line of travel of the digger.

The soil attached to the digger was removed with the help of oscillation produced by the crank mechanism.

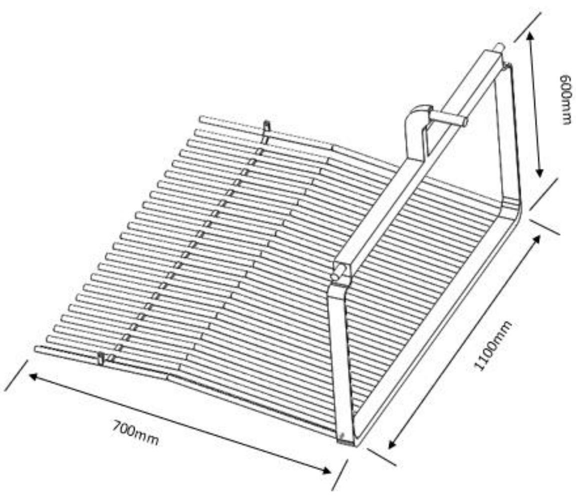

Figure 9: Soil separator unit

Determination of length of soil mass separator

For the determination of the volume of material flow on the soil separator, the volume of the soil flow was determined as follows [20].

$$Vs = AxS \quad (31)$$

Where;

Vs=Volume of soil

A=Area coverage

S=speed of travel

Average forward speed = 3.0 km/hr =0.97m/s.

Working width of machine = 0.9 m and

Working depth = 0.25 m

$$Vs = 0.9m \times 0.25m\,x \times 0.83m/s$$

$$Vs=0.18m^3/s$$

Weight of soil

$$Ws = Vsxd \quad (32)$$

Where;

Ws=Weight of soil

Vs=Volume of soil

D=density

$$Ws = 0.18m^3/s \times 1470kg/m^3$$

$$Ws= 264.6kg/s$$

Assume the weight of the crop in one meter length of bed is 4 kg.

Then,

$$W = Wc \times S \quad (33)$$

Where;

W= Weight of crop per sec

W_c= Weight of crop in one meter length of bed

S= average speed

= 4kg/m × 0.83m/s = 3.32kg/s

Total material to be handled (Q_{out}) =267.92kg/s. Now equating the volume of material flow with the volume of the separator by assuming that this material spread uniformly on the separator in 10 cm thick layer, the following was obtained as given below. As the speed of separator was depends upon the eccentricity of crank of the digger.

$$Qout = \gamma \times length \times thickness\ of\ material \times speed\ of\ separator \quad (34)$$

$$267.92kg/sec = 1470kg/m^3 \times L\,x0.1m \times 3km/h$$

L = 0.61 ≈0.7 m

Therefore, the separator of 0.7m length was fabricated using mild steel rod of 10 mm diameter and the spacing between the rods was selected 40 mm keeping in view the approximate minimum size of the carrot to be retained on the separator and gave maximum opening of the separator area for sieving of the soil. The gap between the two consecutive rods of soil separator was kept in the range such that the crop should not pass between two consecutive rods. Adjustment was provided to vary the angle of separating unit with the help of a lever attached to the frame. Therefore, the gap 40 mm were kept in actual field design to passes more soil without any damage.

Depth Gauge

The mechanism for the depth control system was fabricated to set up digging depth. The wheel was fabricated from 50 × 6mm flat iron with the wheel diameter of 400mm and attached to the sliding arm made from square hollow mild steel of 40 × 500mm attached to the frame body on both sides. The depth of the lift arm shall be designed to be adjusted varied within a range of zero to 200mm vertically [21].

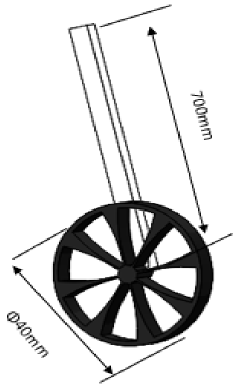

Figure 10: Depth control wheel

Hitch System

The three points hitch system was welded on the main frame. The hitch system was arranged as per the specification of IS 4468 [22]. The most height of the hitch system was kept at 600 mm from the frame for controlling the depth and providing the stability of digger. The hitching system of the carrot harvester implement was based on the standard one-hole hitch specified in [23] where the pin sleeve and hitch frame were adaptable to the hand tractor

Draft and power requirements

Draft is the horizontal force required to pull an implement. It includes soil–crop resistance and rolling resistance, with the exception of manure injection, where transport wheel resistance is added. For seeding implements and shallow tillage tools, draft mainly depends on implement width and operating speed. In deeper tillage, draft is additionally influenced by soil texture, working depth, and tool geometry.

Typical draft requirements can be calculated as by using Equ (35) [19].

$$D = Fi(A + BV + CV^2) \times wd \qquad (35)$$

Where:

D = implement draft, N;

F = a dimensionless soil texture adjustment parameter

i =1 for fine, 2 for medium and 3 for coarse textured soils

A, B and C are digger-specific parameters

v = speed, km/h.

w =width, m or number of rows or tools

d = tillage depth

Then, the drawbar power was calculated using the following equation:

$$P_{db} = \frac{D \times V}{3.6} \qquad 36$$

Where:

P_{db}= drawbar power, kw

v = travel speed km/h

Figure 11: Designed and manufactured carrot digger

CONCLUSION

The well-designed carrot harvester developed through careful calculation and engineering analysis demonstrates that proper design methodology is essential for achieving efficient, reliable, and crop-safe mechanized harvesting. By basing the design on key physical and mechanical properties of carrot roots and soil conditions, critical parameters such as digging depth, blade geometry, lifting angle, conveyor speed, and power requirement were accurately determined. These calculated design values ensured effective soil penetration and carrot loosening while minimizing root breakage and field losses.

The integration of appropriately sized components, including the digging blade, conveying and cleaning mechanisms, and power transmission system, resulted in smooth operation and optimal energy utilization. Stress, power, and capacity calculations further confirmed the structural strength and durability of the harvester under field working conditions. As a result, the machine is capable of maintaining consistent harvesting performance with reduced vibration, clogging, and mechanical failure.

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