AMERICAN JOURNAL OF SMART TECHNOLOGY AND SOLUTIONS (AJSTS)

ISSN: 2837-0295 (ONLINE)

VOLUME 3 ISSUE 2 (2024)

PUBLISHED BY
E-PALLI PUBLISHERS, DELAWARE, USA

American Journal of Smart Technology and Solutions (AJSTS)
Volume 3 Issue 2, Year 2024
ISSN: 2837-0295 (Online)
DOI: https://doi.org/10.54536/ajsts.v3i2.3337
https://journals.e-palli.com/home/index.php/ajsts

Design and Development of a Tractor-Drawn Cultivator with a Pulverizing Attachment

Husen Bona Wako[1*], Adesoji Matthew Olaniyan[2]

Article Information

Received: August 02, 2024

Accepted: August 30, 2024

Published: September 03, 2024

Keywords

Clods, Cultivator, Draft, Pulverizing Roller, Soil Manipulation, Tillage

ABSTRACT

Soil tillage is one of the most critical processes in the agricultural production system, requiring the greatest energy and time of any operation. A cultivator with pulverizing attachment was designed and developed to reduce secondary tillage operations to a single pass to ensure timeliness in seedbed preparation. It consists of a pulverizing roller as an active unit and cultivator tynes as a passive unit. The implement has the following major components; a frame, cultivator tines, and a pulverizing roller consisting of central and peripheral shafts on which discs are arranged. The cultivator tynes were fitted on the main frame while the pulverizing roller was attached at the rear end of the frame with the help of bearings. The pulverizing blades were welded on discs in such a way as to crush the soil by impact force. In this way, the force exerted by the pulverizer on the clod is distributed uniformly in the shaft which is used to hold the pulverizing roller. The pulverizer was designed to break big clods formed during tillage operation and the implement was attached to the tractor by three point hitch. In operation, the cultivator tynes open the furrow, and the roller cuts and pulverizes the soil at optimum conditions for tillage. With a production cost of ETB 17,587.19, the implement is within the economic reach of an average smallholder farmer in Ethiopia.

INTRODUCTION

Tillage is the process of mechanically manipulating soil to make it more conducive to crop cultivation. Soil tillage entails breaking the earth's compact surface to a particular depth and loosening the soil mass to allow crop roots to penetrate and disseminate into the soil (Zhou *et al.*, 2020). It is the mechanical manipulation of soil and plant debris to prepare a seedbed for the planting of seeds that will produce grain for human use. Tillage also breaks up the soil, improves the release of soil nutrients for crop growth, eliminates weeds, and improves water and air circulation in the soil (Reicosky & Allmaras, 2003).

Tillage is considered one of the most important processes in agriculture, as it gives suitable conditions for root growth, which in turn supports the growth of plants, as it reduces soil resistance, increases the ventilation process, and eliminates weeds (Al-Shamiry *et al.*, 2020). Tillage in the traditional sense is one of the least fuel-efficient procedures. According to (Digman, 2012), just 20% of diesel fuel energy is available at the tractor's drawbar, but only 4% of that energy is transformed into soil-breaking energy.

Tillage implements works based on two working motions, sliding type, and rotating type. Devices like moldboard plows and cultivators use sliding action to cut the soil. By functioning in a rotational motion, disc plows, disc harrows, clod crushers, and rollers cut and pulverize the soil. Because of the soil frictional force and the contact area of the implement, sliding-type implements consume more drafts than rotating-type implements. A negative draft was produced by rotary-type implements. As a result, the concept of combining sliding and rotary implements saves a lot of power, time, and money (Parmar & Gupta, 2001).

Tillage operations can be performed simultaneously in a single pass by a combination tillage tool mounted on a tractor. According to (Manian & Kathirvel, 2001), combined tillage is a method of manipulating the soil by using two or more different tillage implements at the same time to reduce the number and time of field operations. According to (Prem *et al.*, 2016), combined tillage is the use of two or more implements at the same time to alter the soil. Combined tillage, in a broad sense, refers to the integrated management of resources such as time, energy, fuel, labor, soil, and water conservation while also boosting output and better utilizing natural resources. It also helps to maintain agricultural production.

Combined tillage implement reduced larger size clods in the soil, improved aeration and moisture holding capacity, and obtained medium uniformity of soil and finer pulverization modulus. The combination tool achieved maximal soil loosening as evidenced by the low soil bulk density range of 1.15 gcm^{-3} as opposed to the normal 1.4 gcm^{-3} encountered in conventional tools operated field. The use of combined tillage for seedbed preparation can save 44 to 55% in cost and 50 to 55% in time (Kailappan *et al.*, 2001).

(Maheshwari *et al.*, 2005) reported that soil aggregates of size 12 to 14 mm in the final seedbed were acceptable for sowing crops. Clod formation after plowing or disking is a significant issue. Clods block the penetration of seed drill furrow openers and prevent close contact between

[1] Agricultural Machinery Engineering Department, Oromia Agricultural Research Institute, Jimma Agricultural Engineering Research Center, Jimma, Ethiopia
[2] Department of Agricultural Engineering, Haramaya Institute of Technology, Haramaya University, P. O. Box 138, Dire Dawa, Ethiopia
[*] Corresponding author's e-mail: husenifnaan@gmail.com

seeds and soil. To avoid the aforementioned issues clods must be pulverized. Because of the time, efficiency, and cost savings, combining implements and minimizing the number of passes is becoming more common (Alkhafaji, 2020).

In Ethiopia, farmers are using the conventional tillage system which disturbs the soil more; increases soil compaction increases erosion capacity by wind and water. The combined tillage used was also imported from other countries which is too expensive. The implementation did not consider small-scale mechanization, affordable power, high land agriculture, costly, and not scale appropriately.

The overall dimensions and weight of the tillage tool were heavy, not easily dismantled, or repaired and it was complicated. Therefore, the objective of this study was to design and develop a tractor-drawn cultivator with a pulverizing attachment that would be operated within the techno-economic status of smallholder farmers in Ethiopia.

MATERIALS AND METHODS

Determination of Physical Properties of Soil

Moisture Content of Soil

Before field activities, soil samples were taken from 0 to 20 cm below the soil surface to determine moisture content. Three soil samples were randomly taken from test plots and the weight of each sample was determined using an electronic balance. The samples were then maintained in a hot air oven for 24 hours at a temperature of 105°C. The moisture content (dry basis) was calculated using the formula below (Javadi & Hajiahmad, 2006).

$$M_C = (W_W - W_d)/W_d \times 100 \qquad (1)$$

Where, (i) M_C = Moisture content of the soil, % db; (ii) W_W = Weight of wet soil, g; and (iii) W_d = Weight of oven-dry soil, g.

Determination of Bulk Density of Soil

The oven dry mass per unit volume of the soil is known as bulk density. It was determined by taking soil samples from various sites throughout the field with a core sampler. The formula was used to compute bulk density (Javadi & Hajiahmad, 2006).

$$\gamma_d = M/V \qquad (2)$$

Where, (i) γ_d = dry bulk density of soil, g/cm^3; (ii) M = oven-dry mass of soil, g; (iii) V = volume of the core sampler, cm^3.

Overall Structure and Descriptions of the Implement

A cultivator with pulverizing attachment was designed to reduce the number of secondary tillage operations to single pass and enhance timeliness in seedbed preparation. As shown in Figure 1, it consists of the following main components; frame with cultivator tynes, pulverizing attachment having pulverizing blades, disc, and three-point linkage unit.

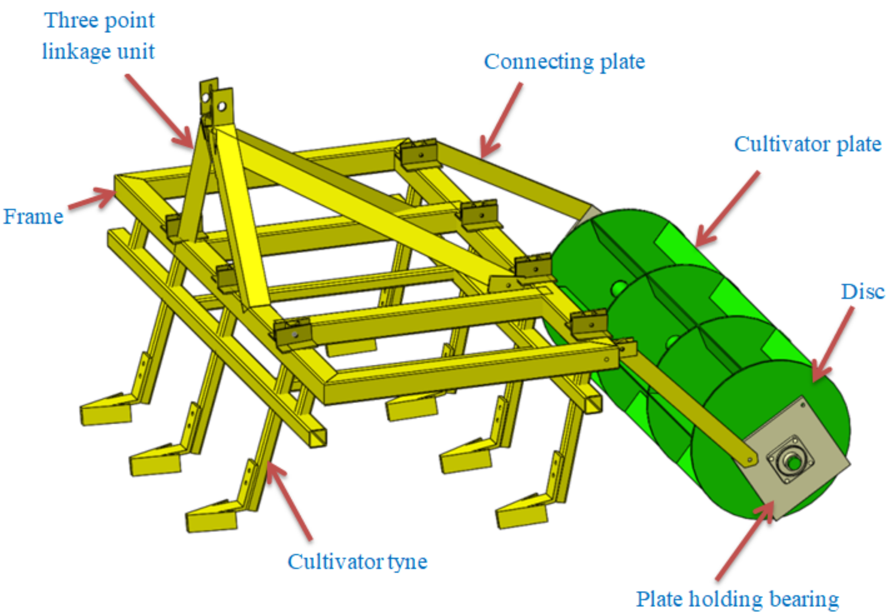

Figure 1: Cultivator with Pulverizing Attachment

Design of Major Components

Components of the cultivator with pulverizing attachment were designed, developed, and fabricated based on parameters like functional requirements, operational requirements, and strength. While fabricating the tractor-drawn cultivator with a pulverizing attachment the basic emphasis was the simplicity of fabrication, the use of locally available materials, the minimum cost of fabrication, and the ease of assembling and dismantling for repair were considered. The mechanical design details were also given due attention so that it was given adequate functional rigidity for the design of the machine.

The implement was designed having in mind that the frame should have sufficient strength to hold the tines and also withstand failure due to developed stresses and bending moment in mind that it will work on heavy soils. The assumptions made in the design of a cultivator with pulverizing attachment were as follows (Parmar & Gupta, 2001); (i) No draft was included for pulverizing attachment because it is a rotating unit; (ii) The average speed of operations of the tractor in the field was kept at = 3 Km/h; (iii) Maximum soil resistance was considered as = 0.75 Kg/cm^2; (iv) Co-efficient of friction in unploughed soil was taken= 0.85; (v) A seven-tyne cultivator having a working depth of 30 cm was considered.

Design of Frame

The frame members were welded together at the ends since a fixed-type end connection was selected for the frame. Euler's theory for the crippling and buckling load (p_{cr}) under various end conditions is given by the equation below (Khurmi & Gupta, 2005).

$$p_{cr} = (\pi^2 EI)/(L_e/r)^2 \quad (3)$$

Where, (i) E = modulus of elasticity for the mild steel material (E= 210 GPa); (ii) A = cross-section area for a hollow rectangular shape, cm^2; (iii) p_{cr} = Euler's critical load, N; (iv) L_e= effective length of the frame, cm; (v) r = radius of gyration of the cross-section; (vi) I = polar moment of the cross-section.

The frame was considered as a beam at the tyne and hitch support portion and all components were loaded vertically so it is considered as a column. Assuming, σ_y = 250 MPa and comparing the critical load of the frame with yield strength, whether the frame is saved or not. From the available data, let us determine the dimension of the frame. The actual length of the frame, L = 160 cm, equivalent length, Le = L=160 cm for the frame one end fixed and another end free.

$L_e/r = \pi \times\sqrt{(E/\sigma_y)} = \sqrt{((210\ Gpa)/(250\ Mpa))} = 91.05$

The crushing stress is given by

$\sigma_{cr} = (\pi^2E)/(L_e/r)^2 = (\pi^2\times210\ Gpa)/91.05^2 = 250\ Mpa$

$L_e/r = 91.5$, L_e = 160 cm then $r^2 = L_e/91.5$

= (160 cm)/91.5 = 1.749 cm, and r^2 = 3.06 cm^2

Radius of gyration $r^2 = I/A = (bh^3)/12hb = h^2/12$

Then $h^2 = 12\times r^2 = 12 \times 3.06\ cm^2 = 3\ 6.71\ cm^2$

h=6.59 cm, say h=6 cm

The dimensions of the frame were made from mild steel rectangular pipe which had h = 6 cm and thickness t = 0.4 cm and width = 7.5 cm.

The critical load is found as:

$P_{cr} = (\pi^2EI)/L_e^{\ 2} = (\pi^2\times210(h^3b)/12)/L_e^{\ 2}$

$= (\pi^2\times210\times(60^3\times75)/12)/160^2 = 1{,}091.92\times10^8$ kN

Hence, critical stress

$\sigma_{cr} = P_{cr}/A = P_{cr}/(b\times h)$

$= (1{,}091.92\times10^8\ KN)/(45\ cm^2) = 242.65$ Mpa

Comparing the critical stress wit h the yield strength of the material, critical stress is less than the yield strength of the material ($\sigma_{cr} \ll \sigma_y$). According to Euler's theory of buckling, for slender columns, the critical buckling stress is usually lower than the yield stress. Hence, the designed frame was saved from buckling. The frame of the cultivator was made from mild steel (M.S) rectangular pipe shape of 6 × 7.5 sizes (cm) × 0.4 cm thickness cross-section. All components of the cultivator were assembled and fitted on the frame.

Design of Cultivator Tines

A cultivator was used to open the furrow at the desired depth of operation. The tine-type cutting blade was used as uniform depth of operation or plowing was required. A cutting unit of tine was fitted at one end of the share and the other end was attached with a frame by bolts and nuts for adjusting depth. The thickness, width, and length of the tyne were decided on the assumption given by (Sharma & Mukesh, 2010).

Let, b x h = bare cross-section area, cm^2

L= length of breast of tine

The length of the inclined part of the tyne generally ranges from 10 to 20 cm, and the radius of curvature R < 12 cm. The minimum clearance length of tine (H_I) between the land surface and the lower edge of the frame was 20 cm. The height of the tine was calculated as the formula given below.

$$H_T = a_{max} + H_I + \Delta H \quad (4)$$

= 10+35+15 = 60 cm

Where, (i) a_{max} = depth of tool, cm; (ii) H_I = length of tyne, cm; (iii) ΔH = length of tine used for fastening with frame, cm.

Load angle was determined by;

$$\tan\alpha = 10/30 = 0.33 \quad (5)$$

$\alpha = \tan^{-1}(0.73) = 36.12°$

Now radius of curvature was determined as;

$$R_C = (H_a - l_1 \sin\alpha)/\cos\alpha \quad (6)$$

= (20-18sin36.12)/cos36.12 = 11.62 cm

Where, (i) l_1 = length of breast of the share; (ii) H_a = height of the first curvature point of the tyne from the point of share.

The cutting blade of the tine was exposed first to bending due to soil resistance. The soil resistance, F_X is horizontal and acts in the axis of symmetry of the tyne. Average soil resistance was obtained by the formula given below (Kumar *et al.*, 2017) and Table 1.

$$F_X = d \times w \times K_O \quad (7)$$

= 30×20×0.2 = 120×9.81 = 1,777.2 N

Where, (i) d = Effective working depth of tine (cm); (ii) w = Working width of the tine (cm); (iii) K_O = Specific soil resistance (kg/cm^2).

Table 1: Specific Soil Resistance at a depth of 15 cm

S/N	Soil Type	Specific resistance, kg/cm^2
1	Light	0.12
2	Medium	0.15
3	Heavy	0.20
4	Very heavy	0.25

Source: (Dubey, 2003)

The draft force exerted on the cutting blade was determined using the following equation;

$D = K_O \times n \times w \times d$ (8)
$= 0.8 \times 7 \times 20 \times 30 = 3{,}360$ kgf $= 32{,}691.6$ N

Where, (i)D = draft force, kgf; (ii) K_O= soil resistance, kg/cm^2; (iii) w = width of tine, cm; (iv) d = depth of tine, cm; (v) n = number of tine.

Finally, the total draft required for operation and each draft of the tines was calculated by,

$D_t = D \times FOS \times g$ (9)
$D_t = 3{,}360 \times 1.5 \times 9.81 = 49{,}442.4$ N

Where, (i) D_t = total draft, N; (ii) D = draft, N; (iii) FOS = factor of safety; (iv) g = force due to gravity, m/s^2.

D_i = (Draft (N))/(number of rows) (10)
$= 32{,}961.6/7 = 4{,}708.8$ N

It was assumed that the draft force on the tine was determined and acting at a height of h/3 from the bottom of the tine. The distance of the draft application on furrow opener tine (d) was calculated by the formula given below (Sharma & Mukesh, 2010).

$d = h/3$ (11)
$= 60/3 = 20$ cm

Moment arm length = (h-d) = 60-20 = 40 cm (12)

The maximum bending moment was determined according to the following formula.

Bending moment of tyne (BM)
= Draft × moment arm length (13)
= 4,708.8 N×40 = 188,352 N m

Therefore, the maximum bending moment of tine was determined by multiplying the bending moment of tine with a factor of safety. (take, FOS=1.5)

Maximum bending moment
= bending moment × FOS (14)
$M_b = 188{,}352$ Nm$\times 1.5 = 282{,}528$ Nm

Mild steel was used for the design of tine and assuming bending stress, σ_b = 56 N/mm^2. Section modules of each cutting blade (Z) were determined as the following formula.

$Z = (Mb)/\sigma_b$ (15)
$= (282{,}528)/56 = 5.05$ cm^3

Where, (i) Mb = maximum bending moment; (ii) σb = bending stress; (iii) Z = section modulus of tine; (iv) b = width of mild steel.

According to (Khurmi & Gupta, 2005), the Section modulus of tine of a cultivator with pulverizing attachment was calculated by using the formula,

$Z = 1/6 \times b^3$ (16)
5.05 cm^3 = $1/6 \times b^3$
$b^3 = 5.05$ cm$^3 \times 6 = 30.27$ cm^3
$b = \sqrt[3]{(30{,}270.84)} = 3.12$ cm≈ 4 cm

So, the 4 cm ×4 cm size of the share was developed from a mild steel material.

Bending Stress, σ_b causing the tine to bend was calculated by the expression(Kumar *et al.*, 2017).

$\sigma_b = (6D_t\,(H_1+a))/(tb^2)$ (17)
$= (6 \times 49{,}442.4(35+20))/(0.6 \times 40^2) = 16{,}995.82$ N/cm^2

Where, (i) D_t= draft at the tip of tine, N; (ii) H_I = length of tine, cm; (iii) a = Effective working depth of tine, cm; (iv) t = thickness of the tine, cm and (v) b = width of tine, cm.

Torsional stress acting on the tine when turning the openers inside the soil at headland is given by (Kumar *et al.*, 2017).

$\tau = (9D_t\,(W_w/4))/(2b^2)$ (18)
$= (9 \times 49{,}442.4(40/4))/(2(40)^2)$
$= 4{,}449{,}816/3200 = 1{,}390.57$ N/cm^2

Where, (i) W_w = Effective working width of tine, cm, and (ii) b = width of tine, cm

Then, the reduced stress is calculated as;

$\delta = \sqrt{(\sigma b^2+4\tau^2)}$ (19)
$= \sqrt{((16{,}995.82)^2+(1{,}3902.57)^2)}$
$= 17{,}052.61$ N/cm^2 = 170.53 N/mm^2

Where, (i) δ = Reduced stress, kg/cm^2; (ii) σ_b = bending Stress, N/cm^2 and (iii) τ = Torsional shear stress, N/cm^2. Since the calculated stress (170.53 N/ mm^2) was less than the allowable stress of mild steel (200 N/mm^2), the design was safe and the tine can perform well.

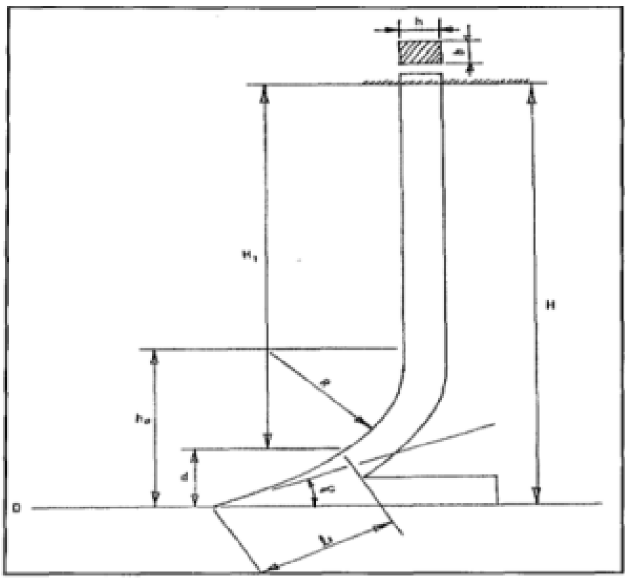

Figure 2: Schematic Diagram of Cultivator Tine

Design of Pulverizing Roller

Basic Design Parameters and Specifications

Pulverizing roller attachment to the cultivator pulverizes the soil to a greater degree. The pulverizing roller consists of discs, a central shaft, and pulverizing members. The pulverizing roller was designed for cutting, mixing, and clod breaking which ultimately pulverizes the soil by impact force. The cutting and clod-breaking action of this unit provides better soil preparation.

Roller diameter = 45 cm

Length of roller = 130 cm

Circumference = $\pi \times D$ (20)

= 3.14×45=141.3 cm

Spacing between pulverizing blades was calculated by the following formula (Prem *et al.*, 2016).

$S_b = (\pi \times D)/N_b$ (21)

= 143.3/6 = 23.55 cm (taking, N_b=6)

Where, (i) S_b = Spacing between the pulverizing blade; (ii) N_b = number of pulverizing blade

Depending upon the type of soil and moisture content in the soil, the torque required to break big clods varies between 0.5 Nm. Therefore, the power required to operate the pulverizer generally ranges between 1-2 hp at an operating speed of 1.1 m/s and 10% slippage (Prem *et al.*, 2016).

Speed = πDN (22)

1.1 = 3.14×45×N

N = 1.1/141.3 = 0.008≈1

We know that,

$P = 2\pi NT/60$ (23)

Where, (i) p = power required, W; (ii) N = rpm of the roller; (iii) T = torque applied, N-m

$T = (p\times60)/(2\pi\times N)$ (24)

2 = (2×3.14×1.×T)/4500

T = (2×4500)/(2×3.14×1)=9000/6.28=1433.12 Nm

When the pulverizer was operated on the ground it was subjected to a sudden load, therefore while designing, the torque was taken as 2 times the rated torque.

$T_{max} = 2\times T$ (25)

T_{max} = 2×1433.12=2866.24 Nm

Design of Pulverizing Blades

Pulverizing blades were fabricated and fixed into the disc at 900 in such a way that it has continuously come in contact with soil. Pulverizing blades were designed as follows.

$A_T = p_L \times T_b$ (Parmar & Gupta,2001) (26)

= 130 cm×0.4 cm=52 cm^2

Where, (i) A_T = total area of pulverizing blade striking on soil, cm2;(ii) p_L = pulverizing length, cm; (iii) T_b = thickness of blade, cm.

The maximum soil resistance = Total area of pulverizing blade × the specific soil resistance

$M_{sr} = A_T \times S_{sr}$ (27)

= 52 cm^2×0.75 Kg/cm^2 =39 Kgf

Where, (i) M_{sr}= maximum soil resistance, Kgf; (ii) A_T= total area of the blade, cm^2; S_{sr} = specific soil resistance Kg/cm^2.

The maximum bending moment in the blade was calculated by the following formula.

$M_b = S_r \times R_d$ (28)

= 39 Kg×22.5 cm=877.5 Kg cm

Where, (i)M_b= maximum bending moment, Kg cm; (ii) S_r= soil resistance, Kg/cm^2 ; (iii)R_d= radial distance, cm.

The maximum bending stress for mild steel is 700 MPa, calculating actual bending stress as per the following formula (Sharma & Mukesh, 2010).

$\sigma_b = (Mb \times y)/I$ (29)

700 = (877.5×12×w×2)/(2×0.4×w^3)

w = √(21060/560)=6.14 cm≈7 cm=7 cm

Where, (i) M_b= maximum bending moment; (ii)σ_b= actual bending stress, N/cm^2 (iii) y = distance height, = d/2; (iv) I = moment of inertia,= $bt^3/12$; (v) w = width of pulverizing blades.

Hence design is safe. Mild steel flat with 7 cm width, 130 cm length, and 0.4 cm thick was selected for pulverizing blades.

Design of the Pulverizing Roller Drive Shaft

The design of the shaft must involve the determination of the minimum diameter of the shaft material to ensure satisfactory strength and rigidity when the shaft is transmitting power under various operating and loading conditions. 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 the maximum shear stress. For a shaft having little or no axial loading, the diameter of the shaft was obtained using the ASME code Equation (39), (ASME, 1995) given as:

$d^3 = 16/\pi S_s\, [(k_b m_b)^2+(k_t m_t)^2]^{(1/2)}$ (30)

Where, (i) d = diameter of the shaft, mm; (ii) M_t = torsional moment, Nm; (iii) M_b = bending moment, Nm; (iv) K_b = combined shock and fatigue factor applied to bending moment; (v) K_t = combined shock and fatigue factor applied to torsional moment; (vi) S_s = allowable stress, MN/m^2.

For rotating shafts when the load is suddenly applied with minor shock (Khurmi & Gupta, 2005), recommended that values of K_b = 1.2 to 2.0 and K_t = 1.0 to 1.50 be used. Furthermore, it was noted that for the shaft without the keyway, the allowable stress (S_s) must be 55 MN/m2, and for the shaft, with the keyway, the allowable stress (S_s) should not exceed 40 MN/m^2.

The torsional moment (M_t) on the shaft was calculated using Eqn. below (Ryder, 1989).

$M_t = (p\times60)/(2\pi\times N)$ (31)

$p = V\times F$ (32)

Where, (i) P = power required to drive the pulverizer; (ii) N = speed of the shaft, (1 rev/sec.); (iii) V = forward speed (0.833 m/s); (iv) F = force required to drive the machine (2.4 N).

Figure 6 shows the load distribution on the shaft. The maximum bending moment on the shaft was determined from the following expressions.

$M_b = (M_v{}^2+M_h{}^2)$ (33)

Where, (i) M_v = vertical bending momentum, Nm; (ii) M_h= horizontal bending momentum, Nm

The reaction of forces on the driven shaft is as shown below for vertical forces

The total resultant components of horizontal and vertical bending moments on the shaft can be obtained as follows.

$M_b = (M_v^2+M_h^2)^{1/2}$ =62,655,333.955 Nmm

$M_t = (p\times60)/(2\pi\times N)$

$= (2\times60)/(2\times3.14\times1)$=51,858.25 Nmm

$d^3 = 16/(3.14\times55)\ [(2\times62{,}655.33)^2+(1.5\times51.86)^2]^{1/2}$

d^3 = 6,768,148.8/172.7= 39,304 mm

$d = \sqrt[3]{39{,}304}$= 34 mm

M_t = torsional moment; Nm

M_b = bending moment; Nm

Kb = 2; Kt = 1.5, and Ss = 55 MN/m^2

Therefore, the standard size of 50 mm shaft diameter was used.

Since the actual diameter was greater than the calculated diameter. Therefore, the shaft has proper strength for operation.

Selection of Bearing

The size of a bearing to be used depends on the size of the shaft required and the available space. In addition, the bearing must have a high enough load rating to provide an acceptable combination of life and reliability.

The bearing size was determined using the maximum load acting on it and the desired maximum lifespan. The dynamic equivalent radial load (W) is calculated by the equation given below.

$W = X.V.W_R+Y.W_A$ (34)

$W = 0.4 \times W_R+ 1.6 \times W_R= 2 \times 20 \times W_R$= 80 kN

W_R = 40 KN

Where, (i) W_A= dynamic axial load; (ii) W_R= dynamic radial load; (iii) X= radial load factor; (iv) Y= axial load factor; (v) V = A rotation factor, 1, for all types of bearings when the inner race is rotating. Dynamic load rating can be determined using Eqn. 35.

$C= W\ (L/10^6)^{1/k}$ (35)

$= 40\times (20{,}000/10^6)^{1/3}$=11 kN

Where, (i) L = Rating life; (ii) C = Basic dynamic load rating; (iii) W = Equivalent dynamic load, and k = 3, for ball bearings, = 10/3, for roller bearings.

Therefore the maximum lifespan value of 20,000 hr was selected to determine the basic dynamic load rating. Bearing number 207 with a bore of 35 mm was selected since the minimum shaft diameter has been determined to be 35 mm. Hence, two bearings used for the shaft of the cultivator with pulverizing attachment were selected based on the general criteria for bearing selection.

Determination of Weight and Power Requirements

Weight of the Implement

To estimate the loads on every part of the cultivator with pulverizing attachment, it should be necessary to estimate the weight of all parts. Accordingly, the weights of the cultivator tyne, frame, pulverizing blade, share (cutting unit of the cultivator), and disc were estimated (ITSI-SU, 2011). The total weight of the cultivator with a pulverizing attachment including the weights of the frame, pulverizer blade, three-point linkage unit, cultivator tine, and share of the cultivator was 1331.48 N. Taking 2% margins for weights of welding, bolts, nuts, etc. Finally, the total weight of the cultivator with pulverizing attachment was found 1358.11 N.

Draft Requirement

The draft requirement of the tractor-operated cultivator with a pulverizing attachment was estimated using factors related to implement and the type of soil. The specific soil resistance of medium black soil of the area was 0.75 kgcm^{-2} (Parmar & Gupta, 2001).

Total working width of the cultivator

= number of tine × tine spacing (36)

= 7 × 22.86 cm =160 cm or 1.6 m

Cross section area of furrows = Total working width of the cultivator x depth of furrow (37)

= 160 cm × 30 cm = 4800 cm^2.

The maximum draft required to drive the cultivator is calculated as follows.

$D = C_{AC}\times S_R\times g\times FOS$ (38)

= 4800 cm^2 × 0.75 Kg/cm^2 × 9.81 m/s^2 × 1.5

= 52,974 N

Where, (i) D = Maximum draft; (ii) C_{AC}= Cross-section area of the furrows; (iii) S_R = maximum soil resistance; (iv) g = force due to gravity; (v) FOS= factor of safety

Power Requirement

The power required to pull the designed implement was estimated as follows.

$P_d = (D\times S)/1000=(52{,}974\times0.833)/1000$=44.13kW (39)

Where, (i) P_d = power required to drive the implement; (ii) D = draft required to drive a cultivator; (iii) S = speed of operation.

The power required to operate the cultivator with pulverizing attachment was calculated as follows;

P= (Power required to drive the implement)/(Coefficient of friction) (40)

P= 44.13/0.85=51.82≈52 kW

Where, (i) P = power required to operate the implement; (ii) D = draft requirement of the implement; (iii) S = forward speed of the tractor.

Laboratory Testing of Prototype

Preliminary performance tests were carried out to obtain actual data on overall implement performance and work capacity. Performance evaluation (in terms of soil tilling quality and machine parameters) was determined. The parameters that were taken during the preliminary test were; the mean weight diameter of the soil, wheel slippage, soil inversion, draft, the width of cut, and volume of soil handled.


Figure 3: Prototype Cultivator with Pulverizing Attachment after Fabrication and during Testing

Mean Weight Diameter of the Soil

Soil samples were collected randomly from the tilled plots, with three replications, using an auger at 0-30 cm depth after the tillage process to estimate the clod mean weight diameter. At room temperature, the moist soil samples were allowed to air. The air-dried soil samples were sieved with a set of sieves. Clod's mean weight diameter was calculated using the following formula.

$$M_{WD} = \sum_{i=1}^{n} \frac{W_i}{w} D_i \quad (41)$$

Where, w_i=the weight of soil on each special sieve (kg)
w = the total weight of experimented soil (kg)
Di = Net diameter of each sieve (mm)

Wheel Slippage

The distances the tractor traveled a head at every 10 revolutions under load and no load on the same surface were measured after a mark was created on the tractor drive wheel with colorful tapes. The speed reduction can be calculated as follows.

Travel reduction = $(M_2-M_1)/M_2 \times 100$ (42)

Where, M_2 = Distance covered at every 10 revolutions of a tractor drive wheel with no load (m)
M_1 =Distance covered at every 10 revolutions of tractor drive wheel with load (m).

Width of Cut

The width of the tillage implement's cut was taken by measuring the width of the furrow using a measuring tape every 5 meters along its length. The width of the cut was determined by averaging ten readings.

Draft

The following equation was used to calculate the draft of the implement (ASME, 1995).

$D=F_i\,[A+B(S)+C(S)^2\,WT]$ (43)

Where, D = drawbar pull, N
F= a dimensionless soil texture adjustment parameter
i = soil factor = (1 for fine, 2 for medium, and 3 for coarse texture soils)
A, B & C = machine – specific parameters
S = forward speed, km/h
T = tillage depth (cm) for major tools
W = machine working width, m.

Soil Inversion

The number of weeds or residues of the previous crop remaining on the soil surface after the operation was compared to the number before the operation. For counting weeds or stubbles, a square frame with sides of 100 cm was employed.

F(%) = (B-A)/B×100 (44)

Where, F= indicator for soil inversion,%
B= number of weeds or crop residue before operation per unit area,
A = number of weeds or crop residue exposed on the surface after operation.

Volume of Soil Handled

As shown in the equation below, the amount of soil handled was estimated by multiplying the field capacity with depth of cut.

$V(m^3/h) = 10000E_{FC}\,D_C$ (45)

Where, V = Volume of soil handled (m^3 /ha)
E_{FC} = Effective field capacity (ha/h)
D_C = Depth of cut (m)

RESULTS AND DISCUSSION

Physical Properties of Soil

Moisture Content of Soil

The mean data on soil moisture content before tillage operations at 0-20 cm depth was recorded and presented in Table 2. The moisture content of the soil varied from 13.9 to 14.54% with an average value of 14.14%.

Table 2: Soil moisture content (dry basis)

Sample No.	Mass of wet soil (gm)	Mass of dry soil (gm)	Soil Moisture content (%)	Average (%)
1	256.68	220.8	13.98	14.14
2	265.26	226.69	14.54	
3	244.52	210.5	13.91	

Bulk Density of Soil

Values of bulk density before tillage operations at 0-20 cm depth were recorded and presented in Table 3. The bulk density of the soil was found to be in the range of 1.43 g/cm^3 to 1.54 g/cm^3 with an average value of 1.49 g/cm^3.

Table 3: Bulk density (gm/cc)

Sample No.	Mass of wet soil (gm)	Mass of dry soil (gm)	The volume of the core sampler (cc)	Bulk density (gm/cc)	Average
1	256.68	220.8	147.2	1.5	1.49
2	265.26	226.69	147.2	1.54	
3	244.52	210.5	147.2	1.43	

Laboratory Testing Results

Mean Weight Diameter of the Soil

The range of the clod means weight diameter was observed between 10.21 to 11.77 mm. Maximum clod means weight diameter of was at a tractor forward speed of 2.5 km/hr. The minimum value of clod means weight diameter was at a tractor forward speed of 3.5 km/hr. It is seen from the result that a smaller clod percentage was produced as the tractor's forward speed increased. The bigger clod percentage decreased and the smaller clod percentage increased.

The increase in speed that causes a decrease in bigger clod percentage might be a result of the vibrating effect of the implement associated with an increase in the operating speed of the tractor. More clods break into smaller ones as the implement with higher speeds throws the soil. The improved soil aggregation was attainable at higher tractor forward speed.

Wheel Slippage

Maximum wheel slippage (12.85%) was obtained at 3.5 km/hr tractor forward speed. Minimum wheel slippage (13.44%) was obtained at 2.5 km/hr tractor forward speed. Wheel slippage increased negatively as the tractor's forward speed increased for the tested implement. The negative increase in wheel slippage might be a result of the rolling resistance of the tractor tires and the rolling action of the implement that oppose the rotational movement for the increased tractor forward speed. A negative increase in slippage percentage was observed with an increase in tractor forward speed.

Width of Cut

The width of cuts varied between 140.00 to 143.83 cm. The width of cuts was found maximum (143.83 cm) at 3.5 km/hr tractor forward speed while the minimum width of cuts (140.00) was obtained at 2.5 km/hr tractor forward speed. The width of cuts increased with the tractor's forward speed. The reason might be due to the high soil thrown by the implement at a higher tractor forward speed than a lower tractor forward speed.

Draft

The draft of the implement ranged between 8.19 to 8.59 KN. The maximum draft was obtained at 3.5 km/hr tractor forward speed. The minimum draft was observed at 2.5 km/hr tractor forward speed. The draft requirement of the implement increased as the tractor's forward speed increased. The reason might be due to higher force requirement at a higher speed than lower speed.

Soil Inversion

The maximum soil inversion (89.17%) was found at 3.5 km/hr tractor forward speed. Minimum soil inversion (82.41%) was found at 2.5 km/hr tractor forward speed. Soil inversion increased as the tractor's forward speed increased. There was an increase in the percentage of soil inverted as the tractor's forward speed increased. It might be a result of the implement's ability to turn and throw the soil when at a higher speed than at a lower speed.

Volume of Soil Handled

The amount of soil handled varied from 671.07 to 672.09 m^3/ha. The maximum volume of soil handled was at 3.5 km/hr tractor forward speed. The minimum volume of soil handled was observed at 2.5 km/hr tractor forward speed. The volume of soil handled increased as the tractor's forward speed increased. This might be due to the increment of effective field capacity as the tractor's forward speed increased.

CONCLUSION

A cultivator with pulverizing attachment was designed and developed to reduce secondary tillage operations to a single pass to ensure timeliness in seedbed preparation. The cultivator tines were fitted on the frame and at the back side of the frame pulverizing, the roller was attached with help of bearings.

The pulverizing blades were welded on the discs, in such a way that was crushed the soil by impact force. In this, the force exerted by the pulverizer on the clod is distributed uniformly in the shaft which is used to hold the pulverizing roller.

The roller was attached to a frame that holds cultivator tines and a three-point linkage unit. The pulverizer was designed to break big clods formed during tillage operation. The developed implement was attached to the tractor by three point hitch. Cultivator tines open furrow and operated roller cut and pulverize soil at optimum condition for tillage.

The preliminary test of the cultivator with pulverizing attachment in terms of the width of cuts, draft requirement, wheel slippage, soil inversion, the volume of soil handled, and clod mean weight diameter was evaluated. Maximum clod means weight diameter of was observed at a tractor forward speed of 2.5 km/hr. The minimum value of clod means weight diameter was observed at a tractor forward speed of 3.5 km/hr. Maximum wheel slippage (12.85%) was obtained at 3.5 km/hr tractor forward speed. Minimum wheel slippage (13.44%) was obtained at 2.5 km/hr tractor forward speed.

The width of cuts was found maximum (143.83 cm) at 3.5 km/hr tractor forward speed while the minimum width of cuts (140.00) was obtained at 2.5 km/hr tractor forward speed. The maximum draft was obtained at 3.5 km/hr tractor forward speed while the minimum draft was observed at 2.5 km/hr tractor forward speed.

The maximum soil inversion (89.17%) was found at 3.5 km/hr tractor forward speed while minimum soil inversion (82.41%) was found at 2.5 km/hr tractor forward speed. The amount of soil handled varied from 671.07 to 672.09 m^3/ha. The maximum volume of soil handled was at 3.5 km/hr tractor forward speed. The minimum volume of soil handled was observed at 2.5 km/hr tractor forward speed.

REFERENCES

Alkhafaji, A. J. (2020). Designing and testing of triple combination tillage implement. *Plant Arch, 20*(1), 2363-2366.

Al-Shamiry, F. M. S., Al-Qarni, A. M. and Munassar, A. S. (2020). Effect of Tillage Depth and Tractor Forward Speed on Some Technical Indicators of the Moldboard Plow. *International Journal of Progressive Sciences and Technologies, 23*(2), 28-37.

Askari, M. & Khalifahamzehghasem, S. (2013). Draft force inputs for primary and secondary tillage implements in clay loam soil. *World Applied Sciences Journal, 21*(12), 1789-1794.

ASME. (1995). Design of transition shafting. American Society of Mechanical Engineering, New York, USA.

Digman, M. (2012). Saving time and fuel during tillage. *2011 Wisconsin Crop Production Association Distinguished Service Awards*, 95.

Dubey, A. K. (2003). Design of sowing machine. Teaching material for training on computer-aided design and design methodology for agricultural machinery. *Technical bulletin no CIAE/AMD/2003/296*, 92-102.

ITSI-SU, (2011). Mass, volume, and density of regularly and irregularly shaped objects. Internet: <http://www.pdesas.org/module/content/resources/14756/view.ashx>.

Javadi, A. R. Z. H. A. N. G. & Hajiahmad, A. (2006). Effect of a new combined implement for reducing secondary tillage operation. *International Journal of Agriculture and Biology, 8*(6), 724-727.

Kailappan, R., Vijayaraghavan, N. C., Manian, R., Duraisamy, G. & Amuthan, G. (2001). Combination tillage tool-I Design and development of a combination tillage tool. *Agricultural Mechanization in Asia Africa and Latin America, 32*(3), 19-22.

Khurmi, R. S. & Gupta, J. K. (2005). Theory of Machines. Eurasia publishing house Ltd. New Delhi, 8th Edition.

Krutz, G. and Thompson, L. (1984). *Design of agricultural machinery* (No. 631.3 K7).

Kumar, A. A., Kumar, A. A., Vidhyadhar, V., Mohan, K., Suresh, C., Rao, A. S. & Ramana, M. V. (2017). Design and development of groundnut planter for power weeder. *AMA, Agricultural Mechanization in Asia, Africa, and Latin America,* 48(3), 25-30.

Maheshwari, T. K., Thakur, T. C. & Varshney, B. P. (2005). Spiked clod crusher and planker performance under different soil conditions. *Agricultural Engineering Today, 29*(3and4), 6-11.

Manian, R. & Kathirvel, K. (2001). Development and evaluation of an active-passive tillage machine. *Agricultural mechanization in Asia Africa and Latin America, 32*(1), 9-18.

Parmar, R. P. & Gupta, R. A. (2016). Development and performance evaluation of a powered pulverizing unit with a cultivator. *Agricultural Engineering Today, 40*(2), 3-9.

Prem, M., Swarnkar, R., Kantilal, V. D. K., Jeetsinh, P. S. K. & Chitharbhai, K. B. (2016). Combined tillage tools-a review. *Current Agriculture Research Journal, 4*(2), 179.

Reicosky, D. C. & Allmaras, R. R. (2003). Advances in tillage research in North American cropping systems. *Journal of Crop Production, 8*(1-2), 75-125.

Ryder G. H. (1989). Strength of Materials, 3rd ed. Macmillan, Nigeria.

Sharma, D. N., & Mukesh, S. (2010). Farm Machinery Design: Principal and Problems. Jain Brothers. New Delhi. 196-199.

Zhou, H., Zhang, C., Zhang, W., Yang, Q., Li, D., Liu, Z. & Xia, J. (2020). Evaluation of straw spatial distribution after straw incorporation into the soil for different tillage tools. *Soil and Tillage Research, 196,* 104440.