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# Design and Development of Tractor-Drawn Wheat Seed Drill

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## ABSTRACT

Cereals, pulses, and oilseeds are examples of primary agricultural products in Ethiopia that provide household income, are staple foods for the population, and contribute to the country's foreign exchange earnings. Ethiopia is the largest producer of wheat in sub-Saharan Africa. It occupies about 1.80 million hectares annually and the main wheat growing areas of Ethiopia are the highlands of the central, south-eastern and northwest parts of the country. Even though it is the major food crops for the majority of the country's population, wheat productivity per hectare is low yield as compared to other world wheat producing countries. This study was undertaken to design and develop a prototype of seed drill capable of sowing wheat seeds and applying fertilizer at predetermined row spacing and depths. The developed seed drill machine, consisting of a frame, seed hopper, seed metering devices, seed tube/spout, adjustable furrow opener, and drive wheels. The design of the drill components were based on the functional requirement and principal operation. The physical properties of wheat seed and fertilizer, agronomic requirements for wheat seed and fertilizer drill (125 kg and 150 kg per hectare), respectively and the row spacing of 20 cm were considered for design and development of the seed drill. Therefore, the suggested machine design was cost-effective to manufacture and exhibits greater design simplicity compared to alternatives.

## INTRODUCTION

In Ethiopia, primary agricultural products such as cereals, pulses, and oilseeds serve as staple foods for the population, generate household income, and contribute to national foreign exchange earnings. According to the 2015/16 Meher Season Post-harvest Crop Production Survey, approximately 12.5 million hectares of land in private peasant holdings were dedicated to grain crops, yielding over 266 million quintals. Cereals dominated this category, occupying nearly 80% of the cultivated area. Wheat alone accounted for 13.3% (1.66 million hectares) of the grain crop area, producing around 15.8% (42 million quintals) of total cereal output (CSA, 2015/2016). Despite its significance, Ethiopia's wheat productivity remains low, far below global averages (40 quintals per hectare) reported by the FAO (2009).

This underperformance stems from limited adoption of modern agricultural inputs, reliance on erratic rainfall, and outdated farming techniques. Traditional practices, characterized by inefficiency, hinder production scalability. Although agricultural extension programs have promoted improved methods, the absence of affordable and effective row-planting machinery has undermined progress. Manual planting methods are labor-intensive, leading to inconsistent seed spacing, physical strain on farmers, and restricted field capacity.

To address these challenges, imported tractor-drawn planters capable of sowing 24 rows simultaneously were introduced. However, their high cost (around half a million Ethiopian birr) renders them inaccessible to most smallholder farmers. Moreover, these machines lack separate compartments for seeds and fertilizers, mixing both in the same furrow, contrary to agronomic recommendations for optimal placement.

Institutions such as the Ethiopian Agricultural Research Institute and regional Agricultural Research centers (Melkassa, Assella, and Jimma) have focused on local initiatives to develop animal-drawn seed drills. Additionally, private companies, farmers, and individual innovators have worked on developing row-planting machines to advance wheat production in Ethiopia. Different design approaches have been used to produce various types of wheat row seeders, each with its own benefits and drawbacks.

Asella Agricultural Engineering Research Center developed the first gravity flow eight row wheat seed drill. The seeder was unable to give acceptable flow continuity and lacked control mechanism. Later, animal drawn wheat seed drill, with fluted metering mechanism for seed only and capable of planting four rows at a go, was developed. The problems observed with this wheat seed drill during field test were seed breakage, heavy weight, fertilizers were broadcasted manually and the chain miss aligned during flute exposed length adjustment.

The Melkassa Agricultural Research Center recently developed and showcased a six-row, animal-drawn wheat seed drill. This seeder is specifically designed for planting wheat at a seeding rate of 125 to 150 kg/ha but lacks a fertilizer metering mechanism, requiring fertilizers to be broadcast manually before sowing. It has a working width of 1.2 meters. Despite ongoing efforts by various researchers at different centers to design, modify, test, and evaluate wheat seed drills, there remains a shortage of reliable, farmer-friendly seed drills suitable for practical field conditions.

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Persistent challenges include uneven seeding rates, inability to simultaneously apply seeds and fertilizers, and mechanical flaws in design and manufacturing. Consequently, existing seed drills remain unsuitable for widespread farmer adoption. This research project aims to design and develop an affordable tractor-drawn wheat seed drill that ensures precise seed and fertilizer placement, addressing the gaps in current technologies.

### Objective

To design and develop tractor-drawn wheat seed drill

## MATERIALS AND METHODS

### Design Considerations

A tractor-powered wheat seed drill was developed as both a functional and experimental unit. The components of the drill were designed according to operational needs and fundamental working principles.

### Descriptions of Seed Drill Machine

The tractor-drawn wheat seed drill (Figure 1) was designed with several key components, including a frame, seed and fertilizer hoppers, furrow openers, a covering mechanism, a ground wheel, a power transmission shaft, a fluted roller metering device, seed and fertilizer delivery spouts and tubes, and a hitching system.

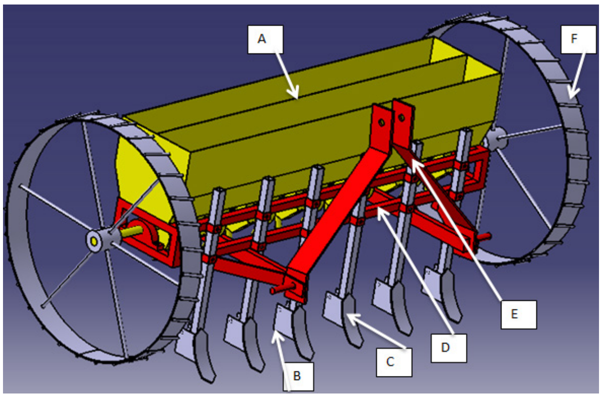


**Figure 1:** View of the prototype seed drill machine

### Determination of Working Width of Seed Drill

The working width of the machine were determined from desired area of land to be covered, speed of operation, estimated field efficiency, and available working day(s)/hour(s). Furthermore, the weight of the drill, soil resistance against the furrow openers, and rolling resistance on the wheels were estimated to decide up on the size of the drill. Thus, the width of seed drill was calculated as follows. According to Sharma and Mukesh (2010), the speed of sowing operation is from 4 to 6 km/hr. And assume that field efficiency is 80% and furrow spacing is 20cm

$$\text{desiredhectares} = (W(m) \times S(km/hr) \times \eta \times (\text{workinghours}))/10 \quad (1)$$

Let desired land to be covered is 204 ha and speed of operation is 5 km/hr.

$$\text{Width}(m) = (204(ha) \times 10)/(5(km/hr) \times 0.8 \times 425) = 1.20m \quad (2)$$

$$\text{numberoffurrow} = W(m)/\ Sp(m) \quad (3)$$

$$\text{numberoffurrow} = 1.2m/\ 0.2m$$

### Parameters

W = Working width (meters, m), S = Operating speed (kilometers per hour, km/hr), $\eta$ = Field efficiency (percentage, %), Sp = Furrow spacing (meters, m). Based on these parameters, a wheat seed drill equipped with six furrow openers was developed.

### Frame

The main frame serves as the pillar of the seed drill, supporting all other components. When selecting the frame material, weight and strength were the key design

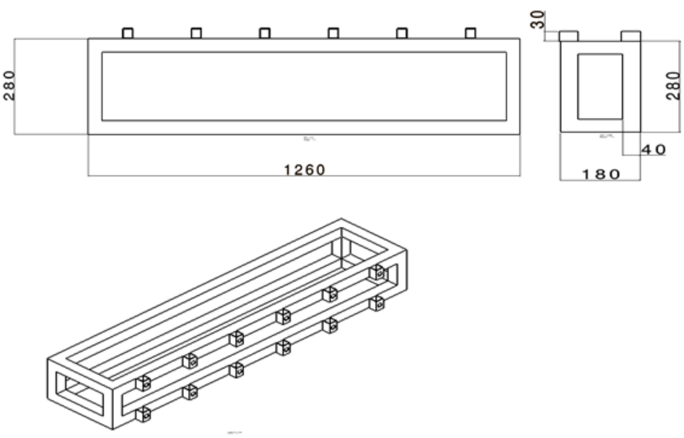


**Figure 2:** Schematic of designed frame and isometric view (all dimensions are in mm)

considerations. The frame must endure torsional and bending stresses caused by operational forces, including draft loads from the furrow openers, driving wheel unit, hitch attachment, seed hopper, and the weight of the seeds. To meet these demands, a mild steel square hollow section (40 mm × 40 mm, 1.5 mm thickness) was chosen, providing the necessary strength while maintaining structural stability.

### Hitching System Design

The drilling machine was connected to the tractor using a three-point linkage system located at the tractor's rear, which operates via hydraulic control. Key geometric measurement such as mast height, lower hitch point span, and the distance between the mast and linch pin holes were designed in compliance with the ASAE S217.12 DEC01 (ISO/DIS 730: 2007) standard.

### Seed and Fertilizer Hopper

The construction material was 1.50 mm thick mild steel sheet metal, chosen for its market availability and cost-effectiveness. The seed hopper features a lid with a top-mounted handle for easy opening. The hopper's capacity was designed based on seed and fertilizer application rates of 125 kg/ha and 150 kg/ha, respectively.

To ensure smooth flow of wheat seeds and fertilizer by gravity toward the metering mechanism, the side slopes of the hoppers were determined according to the materials' angles of repose. As a result, the hopper has a trapezoidal shape at the lower half and a rectangular shape at the upper half. It was also split into two compartments lengthwise; one for fertilizer and the other for seeds. Using the seeding rates mentioned, the theoretical hopper volume was calculated using the formula provided by Olaoye and Bolufawi (2001), as follows:

$V = S_R / (n * BD)$ (4)

Where: $S_R$ = seeding rate (kg/ha), n = number of refilling per hectare, BD = bulk density of the seeds (kg/m$^3$)

$V = 125(kg/ha) / 4 * 833.06(kg/m^3) = 0.037\ m^3$

The actual volume of a hopper was determined using the following equation (Sharma & Mukesh, 2010).

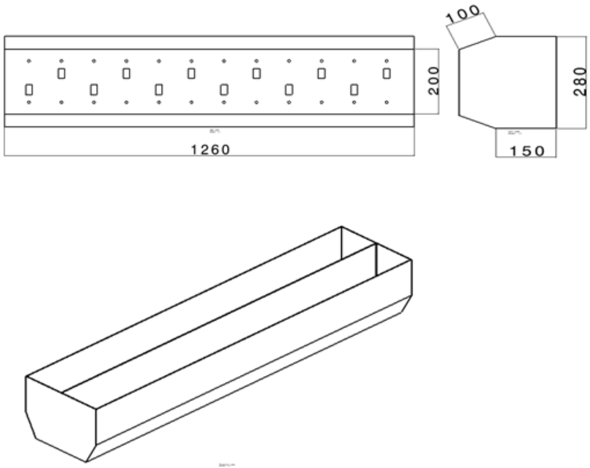


**Figure 3:** Schematic of designed hopper side, bottom and isometric view (all dimensions are in mm)

AB=27.8 cm, GH=19.8 cm, AF=15 cm, QP=10 cm, BC= 125.8 cm, EM=4 cm

$V = AB * BC * AF + 1/2(2*Gh+2*EM) * QP * HJ$ (3.13)

Actual volume of the seed hopper was determined on the basis of the assumed dimensions determined above.

$V = 125.8 * 15 * 27.8 + 1/2\ (2 * 19.8 + 2 * 4)\ 10 * 125.8 =$ 82,399 cm$^3$ or 0.082 cm$^3$

Where, b = top width, a = bottom width, h = height, l = length, VR = volume of rectangular part of the hopper, VT = volume of trapezoidal part of the hopper. The hopper is designed with two separate compartments of equal size, each with a volume of 0.041 m³, one for seeds and the other for fertilizers. The trapezoidal section of the hopper has dimensions of 1258 mm in length, 278 mm in top width, and 100 mm in height. The rectangular section measures 1258 mm in length, 278 mm in top width, and 150 mm in height. Both sections were fabricated accordingly.

### Metering Devices

The proper design of a metering device is crucial for ensuring the effective performance of a seed planter or drilling machine. The dimensions, depth, and number of flutes on the seed drill's metering roller are determined by the intended seeding rate. While various seed metering mechanisms are available, the fluted roller type was chosen due to its widespread use, versatility in handling different crop seeds (RNAM, 1991), and compatibility with local technical expertise. A straight-fluted roller design was selected to control the seeding rate. The fluted roller seed metering device was made of circular aluminum alloy. Ten slots (flutes) were designed along the length of a roller having 10 mm diameter and 5 mm depth for each flute and 5 mm spacing between the flutes in accordance to RNAM, (1991), which recommends that the number of flutes in the roller should range from 8 - 12. When ground wheel rotates, seeds in the seed hopper fill in each flutes

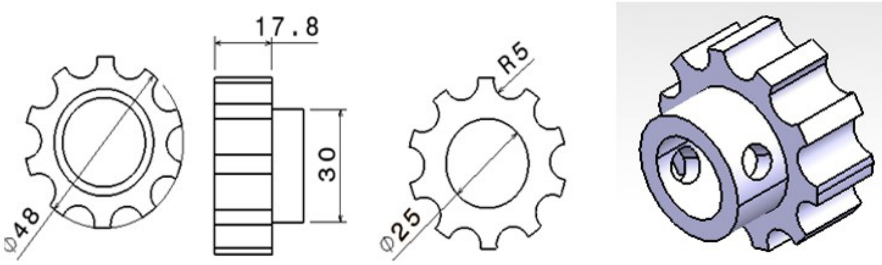


**Figure 4:** Schematic of diagram and isometric view of seed metering device (fluted roller)

turn by turn. Dimensions of fluted rollers were selected on the basis of physical properties of wheat seeds.

Design of fluted roller was calculated as follows (RNAM, 1991). The amount of seed released per meter of row length was determined by

$Vs= (s * r)/10\rho$ (5)

The recommended seeding rate for wheat ranged between 125 kg/ha and 150 kg/ha when using fertilizer (based on informal discussions with agronomists and researchers). The average seed bulk density was measured at 0.833 g/cm³.

$Vs= (125 * 0.2)/(10 * 0.833)= 3cm^3$

Where, Vs = volume of seed dropped per meter length of row, s = seed rate, kg/ha, $\rho$ = bulk density of seed in $g/cm^3$, r = row spacing, m. The cross-sectional area of a semi-circular flute was given by a following formula.

$A_f= \pi d^2 f/8$ (6)

$A_f= \pi * (1)^2/8= 0.39cm^2$

Where df = diameter of a flute, cm

The exposed length of the fluted roller was calculated by the following formula.

$l_f=(8 * s * r * d_g)/ (10 * \rho * d^2_f * N_f * i)$ (7)

The transmission ratio were a unit, as rotation of ground wheel facilitate seed metering device mounted on ground wheel shaft to rotate. Therefore, i= 1

$L_f= (8 * 125 * 0.2 * 0.7)/(10 * 0.833 * 1^2 * 10 * 1)=$ 1.68cm or 16.8mm

Where: $L_f$ = flute exposure length (cm), s = wheat seeding rate (kg/ha), $\varrho$ = seed bulk density ($g/cm^3$), r = row spacing (m), $d_f$ = flute diameter (cm), $N_f$ = number of flutes, dg = ground wheel diameter (m), i = transmission ratio. The seed volume dispensed per revolution of the fluted roller was calculated using the following equation.

$V_d= A_f N_f L_f$ (8)

$V_d= 0.39 * 10 * 1.68= 6.55cm^3$

Where: $A_f$ = area of semi-circular flute, $N_f$ = number of flutes, $L_f$ = exposed length of fluted roller, cm. The number of revolution of a fluted roller, nf were calculated by:

$n_f= i N_g$ (9)

Where, i = transmission ratio, $N_g$ = number of revolution of ground wheel $V_d$ = delivered volume per revolution of roller, $d_g$ = diameter of ground wheel, m. It was determined by the desired number of flutes and their spacing by the following formula.

$d_r= N_f(d_f+S_f)/ \pi$ (10)

$d_r= 10(10 + 5)/ \pi = 47.75 \sim 48mm$

Where, $d_f$ = diameter of the flute, mm; $s_f$ = spacing between flutes, mm; and $d_r$ = diameter of fluted roller, mm. The flow of seeds / grain through rectangular orifices was calculated according to Moysey *et al.* (1988).

$Q= -0.0342 + 770A_n \sqrt{g}D_e$ (11)

$D_e= 0.5a'b'/a'+b'$ (12)

Length (a) of rectangular orifice at the bottom of seed hopper was found to be 16.8 mm which is equal to exposed length of flute, and width of orifice (b) was selected as 20 mm to facilitate freely flow of seed. The effective seed diameter (d) was taken as 4.65 mm which was mean seed size.

$a'= a-kd$ (13)

$a'= 16.8mm - (1.4*4.65 mm)= 10.29mm$

$b'= b-kd$ (14)

$b'= 20mm - (1.4*4.65 mm)= 13.49mm$

Therefore, net effective area of orifice was calculated as the following formula

$A_n= a'b'$ (15)

$A_n= 10.29mm * 13.49mm= 138.81mm^2$ or $1.388 * 10^{-4}m^2$

Hydraulic diameter of orifice was determined by formula given below.

$De= 0.5(10.29mm * 13.49mm)/10.29mm + 13.49mm=$ 2.92 mm or $2.92 * 10^{-3}m$

The volume flow rate of seeds / grain through rectangular orifices, L/s was determined by the following formula

$Q= -0.0342 + (770* 1.388* 10^{-4}m^2(\sqrt{9.81} * 2.92 * 10^{-6}) * 1000$

Q= 0.54(i/s)

Where, Q= volume flow rate, L/s; $A_n$= net effective area of orifice, $m^2$; g= acceleration of gravity = 9.81 $m/s^2$; De= hydraulic diameter of orifice, m; d = seed effective diameter and k is a constant and have value of 1.4 for all seeds; a= length of orifice; b= width of orifice; a' and b' are effective opening of orifice.

## Design of Fertilizer Metering Devices

Standard fluted rollers were used for metering granulated fertilizer. Fertilizer flow was governed by gravity. Rollers rotate and deliver the fertilizer in a continuous stream to the drill tube. The amounts of fertilizer delivered were regulated by adjusting the length of flutes exposed to fertilizer and speed of the rollers. The gap between the roller and the base plate (flap) were adjusted according to size of granules and properties of the fertilizer. The mode of operation of the unit was similar to the fluted rollers used for grain sowing. The bulk density of NPS-fertilizer and recommended rate was found to be 0.945 g/cm3 and 150 kg/ha respectively.

The volume of fertilizer dropped per meter length of row were calculated by

$Vf= (150kg/(ha * 0.2m))/(10 * 0.945g/cm^3= 3.2\ cm^3$

The exposed length of the fluted roller was calculated by the following formula.

$l_f=(8*150kg/ha*0.2*0.7)/(10*0.945*1^2*10*1)= 1.78cm$

**Adjustable Furrow Openers**

The furrow openers were mounted on the front of the frame to create trenches for seed placement. As they cut through the soil, seeds and fertilizers are delivered through their respective tubes and deposited into the furrows via drop boots. The planter was equipped with six furrow openers, secured to the main frame using a bolt-and-nut system. This setup allowed for depth adjustment by vertically repositioning the furrow openers.

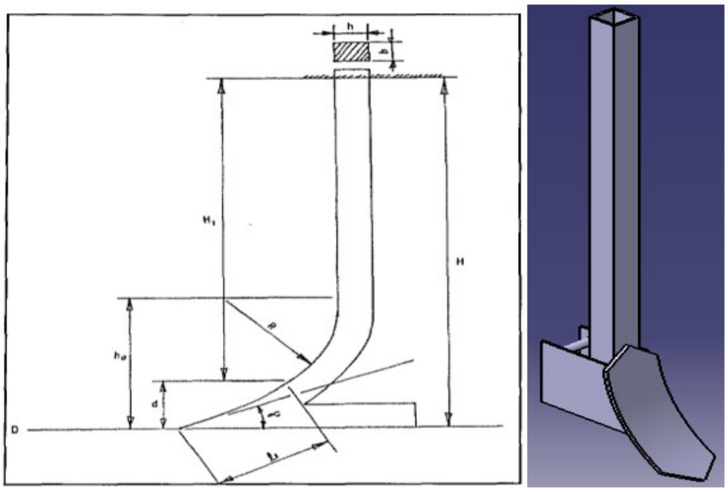

**Figure 5:** Schematic of diagram designed furrow opener

Radius of curvature of the tine was calculated by the following equation.

$R= (h_o-I_1 \sin a)/\cos a$ (16)

Where: $1_1$: Length of the shovel's breast (mm), α: Angle formed between the shovel's surface and the ground, $h_o$: Vertical distance from the shank's tip to its bent portion (mm), the tine height (H) varies based on its attachment to the frame. The frame's lower edge should maintain a minimum ground clearance ($H_1$) exceeding 200 mm. typically; the shovel slope ranges between 100 and 250 mm, with a specified curvature radius. The shank's cross-sectional dimensions are given as b × h (mm²).

Using the values $h_0$ = 140 mm, l = 110 mm, and α = 25° in Equation 16, the calculations were performed. These parameters were chosen based on Dubey's (1985) research recommendations.

$R= (140\text{-}110 \sin 25^0)/ \cos 25^0$

R= 103.2mm

$H= a_{max} + H_1 + \Delta H$ (17)

Thus, by inserting these values into Equation 17, the shank height (H) from the shovel tip to the frame was determined to be 500 mm. During the operation, an effective draught force 'D' acts at the tip of the tool that generate a bending stress (σ) due to soil resistance at the bent portion causing bending of the shank. For calculation purpose soil resistance Ko is horizontal and acts in the axis of symmetry of shovel. It was assumed to be 0.25 kg/cm² for heavy soil.

**Table 1:** Specific Soil Resistances up to a Depth of 15 cm

| **S.No.** | **Soil type** | **Specific resistance, kg/cm²** |
|---|---|---|
| 1 | Light soil | about 0.12 kg/cm² |
| 2 | Medium soil | about 0.15 kg/cm² |
| 3 | Heavy soil | about 0.20 kg/cm² |
| 4 | Very Heavy soil | about 0.25 kg/cm² |

*Source: Dubey, 2003*

Assuming, a = 3cm at the bottom and b= 5 cm at the top, the cross section of furrow is trapezoidal in shape d = 5 cm substituting the values in Equation 18. The draft force exerted on the Furrow opener was determined using the following equation (Kurtz *et al.,* 1984).

$D= Ko * n * w * d$ (18)

$D= [0.25 * 6 * (3 + 5/2) * 5] * 9081= 294.3N$

Now factor of safety was assumed to be 3. Therefore, total draught exerted on the openers was calculated as follows.

$D= 294.3 * 3= 882.9$ N

The draft force exerted on each furrow opener was determined as follows.

$D= 882.9\ N/6= 147.15$ N

Where, D=draft force, N, Ko=specific soil resistance, W= width of opener, cm; d=depth of opener, cm; n=number of furrow openers. The center of gravity of a trapezium with parallel sides "a" and "b" is at a distance of h measured from the side "b", if the height of trapezium is H.

$h= (H/3)(b+2a/b+a)$ (19)

$h= (5/3)(5 + (2*3)/5+3)= 2.3cm$ or 23mm

Assuming shank as cantilever beam, the maximum bending moment for a cantilever length of 42.3 cm (considering 5 cm furrow depth and the force is acting from the centroid of trapezoidal section, from geometry (figure 5), it acts at 2.7 cm from bottom of furrow opener). The length of shank attached to mainframe was 10 cm. Therefore, effective length of shank was determined by the following formula.

$ls= [\,50-2.7-(10/2)\,]= 42.3cm$ or 423mm (20)

Bending moment ($M_b$) was calculated as follows (Kurtz *et al.* 1984).

$M_b= D\pi * L$ (21)

$M_b= 294.3\ N * 423mm$

$M_b= 124,488.9$ Nmm or 124.5 Nm

Where: M represents the bending moment, D denotes the total draft, and L is the shank length.

The furrow opener was designed using square hollow pipes made of mild steel, with a bending stress (fb) of 56 N/mm² for the mild steel (Senger, 2002). The section modulus for each furrow opener (or Tyne) was determined using the following formula.

$Z= M_b/f_b$ (22)

$Z= (124,488.9Nmm)/(56N/mm^2)=2,223mm^3$

For design purpose, Take, b=40mm size square hollow pipe M.S and t=thickness was assumed.

Section modulus of the furrow opener, was calculated by using the formula (Seely *et al.* 1995) and (Timoshenko *et al.* 1964).

$Z= b^4-h^4/6b$ (23)

$2,223mm^3 = (40)^4 - h^4/6*40$

$h= 37.73$

Thickness of square hollow pipe mild steel used was found to be

$t= b-h/2$ (24)

$t= 40mm - 37.73mm/2= 1.13 \approx 1.5mm$

Therefore, square hollow pipe M.S furrow openers of 40mm x 40mm x 1.5 mm size was quite safe and the size was available in the market.

### Seed and Fertilizer Tube

Seed and fertilizer delivery tubes were installed to safely transport seeds and fertilizer from the metering unit to the furrow created by the opener. A 20 mm diameter circular PVC pipe was connected to the tube at the lower part of the metering flute housing (discharge spout). The discharge spouts were constructed from 1 mm thick mild steel sheet metal. Seeds and fertilizer from the hopper move through the metering mechanism into the delivery tube. The discharge tube then evenly distributes them at the desired spacing into the prepared furrow.

### Row Marker

Two adjustable row marker of tractor drawn wheat seed drill was attached on both right and left side of seed drill based on the row spacing which was 20cm apart from the end of the successively sown wheat row. During in operation, one of the row markers was raised up while the other was marking the soil for the next pass. Adjustable row marker in seed drilling machine helps the operator to maintain more accurate or constant row spacing and to avoid overlap sowing. It was made of mild steel flat iron with adjustable slot.

### Determination of the Weight of Components of the Seed Drill

To determine the loads acting on each component of the seed drilling machine, the weight of all its parts had to be calculated. Using appropriate equations, the weights of the hopper, frame, metering device, furrow openers (including covering devices and seed tubes), as well as the seeds and fertilizers inside the hopper, were estimated.

**Table 2:** Weight of each components of the seed drill

| No. | Components | quantity | Area(m²) | Volume(m³) | Density(kg/m³) | Mass (kg) | Weight(N) |
|---|---|---|---|---|---|---|---|
| 1 | Hopper | 1 | 0.65 | $6.5 \times 10^{-4}$ | 7850 | 5.1 | 50 |
| 2 | Main frame | 1 | 0.25 | $3.74 \times 10^{-4}$ | 7850 | 2.94 | 28.84 |
| 3 | Fluted roller | 12 | 0.013 | $2.1 \times 10^{-4}$ | 2700 | 0.57 | 5.6 |
| 4 | Spout | 6 | 0.29 | $2.9 \times 10^{-4}$ | 7850 | 2.28 | 22.3 |
| 5 | Furrow opener | 6 | 0.42 | $5.88 \times 10^{-4}$ | 7850 | 4.6 | 45.1 |
| 6 | Row marker | 2 | 0.12 | $2.04 \times 10^{-4}$ | 7850 | 1.6 | 15.7 |
| 7 | Hitch | 1 | 0.09 | $5.74 \times 10^{-4}$ | 7850 | 4.5 | 44.1 |
| 8 | Hopper cover | 1 | 0.35 | $3.5 \times 10^{-4}$ | 7850 | 2.75 | 26.95 |
| | **Total weight** | | | | | | **238.59** |
| | Weight of parts lay on the wheel shaft including 2% for bolts ,nuts and other | | | | | | 243.36 |
| | Grain | | | | | | 335 |
| | Fertilizer | | | | | | 379.94 |

### Determination of Force Required to Drive Drill

According to (Sharma and Mukesh, 2010)

$F_f= (C_R + i)M_w$ (25)

$F_f= ((Z/W_d)^{0.5} +i)\ M_w$ (26)

$F_f= ((2cm/70cm)^{0.5} + 0.05) * 479N= 104.9\ N$

Where, Ff = force required to pull the planter, N; $C_R$ = coefficient of rolling resistance; Z = maximum tolerable wheel sinkage, cm assume, 2 cm; Wd= wheel diameter, 70 cm; $M_w$ = machine weight on each wheel, 479N; i = gradient of the ground, let i = 5%

**Ground Wheel**

The seed drill's ground wheel, with an external diameter of 700 mm, was designed as an integral part of the seed metering mechanism, directly connecting to the seed metering device. The wheel's rim was constructed from 3 mm thick and 14 mm wide mild steel sheet metal. Each wheel featured spokes made of 12 mm diameter mild steel round bars, each 315 mm long, welded at equal intervals between the rim and the central hub, which also served as a bushing or shaft bearing. Additionally, lugs made from mild steel round bars were attached around the wheel's circumference to enhance traction and grip on the soil.

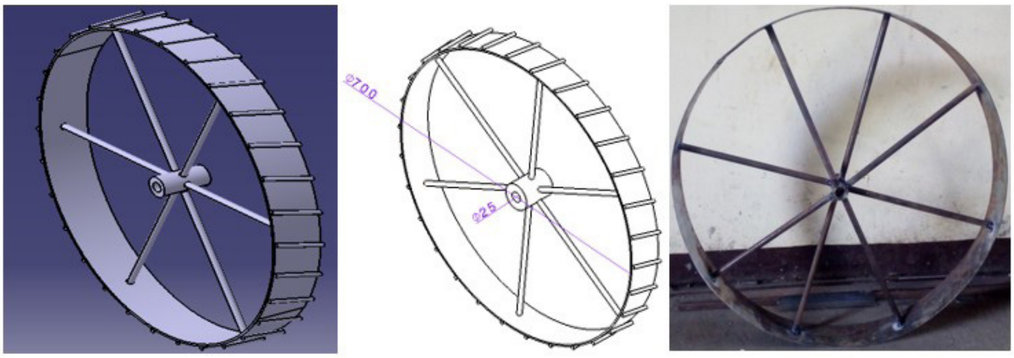


**Figure 6:** View of ground wheel

Shear stress on the wheel was calculated as dictated by Richard and J Nisbett (2011).

$\pi = T/2A_m t$ (27)

Where: π = maximum shear strength, T = the torque produced by the wheel, 88.22 Nm (as determined from the force required to pull the machine and radius of the wheel) $A_m$ = the area of the wheel calculated based on the median diameter of the wheel, tw = thickness of the wheel wall (0.003m), rm = the median radius of the wheel, r = the outer radius of the wheel (0.35m)

$F_f$= total force required to pull the seed drill machine (147.15 + 104.9) = 252.05N

$A_m = \pi(r-0.5t_s)^2$ (28)

$A_m = \pi(0.35-0.5*0.003)^2 = 0.38m^2$

$T = F_f * (W_d)/2$ (29)

T= 252.05N * (0.7)/2= 88.22 Nm

Therefore, the shear stress on the wheel:

T= 88.22/2(0.38 * 0.003)= 38.69 Kpa

Thus the calculated shear stress was much less than the maximum allowable shear stress of the mild steel sheet metal used in the construction of the wheel, 80.8 MPa, hence the wheel is safe for operation. The angle of twist produced on the rim of the wheel, by the applied torque on the wheel can be calculated using the following equations:

$\emptyset = TL/GJ$ (30)

$J = 0.5\,\pi(R_0^4 - R_i^4)$ (31)

Where: Ø= angle of twist, T = the torque produced by the wheel, 88.22 Nm,L = length or width of the wheel, 0.14m, G = modulus of rigidity, 80.8GPa for steel, J = polar moment of inertia, Ro = outer radius of the wheel, 0.35m, Ri = inner radius of the wheel, 0.32m

$J = 0.5\,\pi(0.35^4 - 0.32^4) = 7.1 * 10^{-3}\ m^4$

$\emptyset = 88.22 * 0.14/80.8 * 10^9 * 7.1 * 10^{-3} = 2.1 * 10^{-8}$ radian

Since the calculated angle of twist on the wheel is too small, it is logical to assume that the angle of twist produced by the applied torque is negligible.

**Design of Hub**

The hub is one of the most critical components of a rigid wheel, providing structural support to the spokes and the axle. The outer diameter of the hub was calculated using the formula proposed by Richard and Nisbett (2011), as follows:

D = 1.50 d+ 25.00mm (32)

D= 1.5 * 25 + 25mm= 62.5 mm= 64mm

Where: D = outside diameter of hub, mm, d = diameter of shaft, mm. The length, L, of hub was determined according to Richard and J Nisbett (2011).

$L = \pi d/2$ (33)

L= π 25mm/2= 39.3mm or 3.9cm

**Design of Spokes**

Number of spokes, n, was calculated using the equation given by Richard and J Nisbett (2011).

$n = 2.10\,\sqrt{D}/100$ (34)

$n = 210\ \sqrt{700}/100 = 5.55 \cong 6$

Where, D = diameter of wheel in mm. However, eight numbers of spokes were used to strengthen the wheel to withstand angle of twist and shear stress on the wheel.

**Design of Power Transmission Shaft**

The shaft was initially designed to be constructed from a ductile material, specifically a mild steel rod. Consequently, the design followed ductile material principles, where maximum shear stress governs strength. Assuming minimal or negligible axial loading, the shaft diameter was determined using the following equation (Theodore, 1985):

$d^3 = 16 / (\pi \times Ss) \times [(K_b \times M_b)^2 + (K_t \times T)^2]^{(1/2)}$ (35)

Where, $M_b$= bending moment, N-mm, T= torque, N-mm, d=diameter of shaft, mm

Kb= combined shock and fatigue factor applied to bending moment, Kt= combined shock and fatigue factor applied to torsional moment, Ss= allowable stress, $KN/m^2$ For rotating shafts subjected to sudden loading with minor shock, Khurmi and Gupta (2005) recommend applying combined shock and fatigue factors in the range of $K_b$ = 1.2 to 2.0 (bending) and Kt = 1.0 to 1.5 (torsion). Additionally, the allowable shear stress (Ss) should be limited to 63 MN/m² for keyway-free shafts and 42 MN/m² for shafts with keyways to ensure safe operation. The torsional moment (Mt) acting on the shaft was determined using Equations (3.45 and 3.46) as given by Ryder (1989).

$M_t$ = (P *16) /(2π *N) (36)

P= V*F (37)

P= 0.833m/s * 252.05N= 209.95 W

$M_t$ = (209.95 * 60)/(2π * 30.33)= 66Nm

Where: P = Power required to drive the machine (W), N = Shaft rotational speed (0.505 rev/s = 30.33 rpm), V = Forward speed (0.833 m/s), F = Driving force required (252.05 N). Figure 7 illustrates the load distribution acting on the shaft. The maximum bending moment was calculated using the following expressions: For vertical force equilibrium, the ground wheel shaft reactions are depicted in Figure 7.

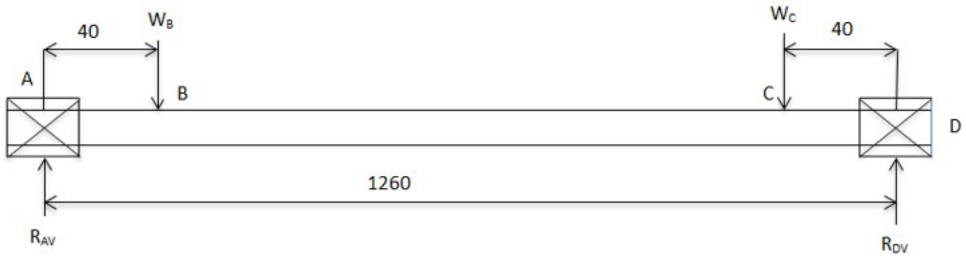


**Figure 7:** The diagram showing forces acting on shaft (all dimensions are in mm)

Where: $R_{Av}$ and $R_{DV}$ = Reactions at the support

$W_B$ = Half of total weight = 479 N

$W_C$ = Half of total weight = 479 N

The reactions, $R_{Av}$ and $R_{DV}$ were determined by taking moment about A;

$\sum M_A$= 0

$R_{DV}$ * 1260= $W_B$ * 40 + $W_C$ * 1220 (38)

$R_{DV}$ * 1260= 479 * 40 + 479 * 1220= 603, 540

$R_{DV}$= 479N

$\sum F_V$= 0 (39)

479N + $R_{AV}$= 479N + 479N= 958N

$R_{AV}$= 958N - 479N = 479N

Therefore, the maximum vertical bending moments (BM) on the shaft were determined using Figure 8 as follows;

BM at A and D = 0 Nmm

BM at B, $R_{AV}$ * 40= 479N * 40mm= 19,160Nmm

BM at C, ($R_{AV}$ * 1220) - ($W_B$ * 1180)

(479N * 1220mm) - (479N * 1180mm)= 19,160Nmm

Thus, the maximum vertical bending moment on the shaft was 19,160 Nmm = 19.16 Nm

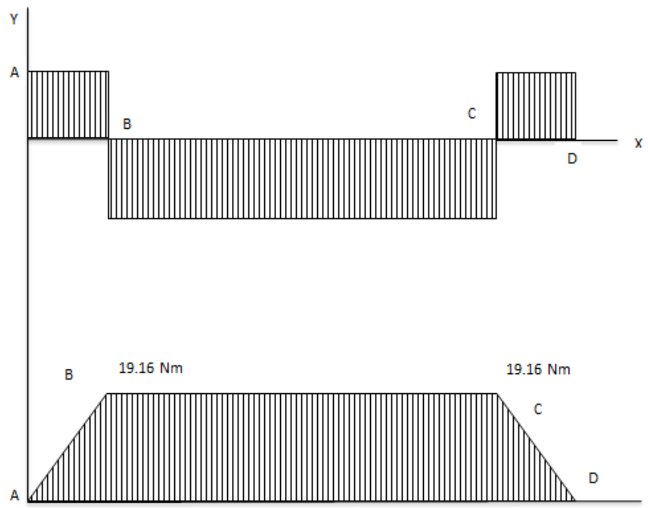


**Figure 8:** Shear and bending moment diagrams on the shaft due to vertical force

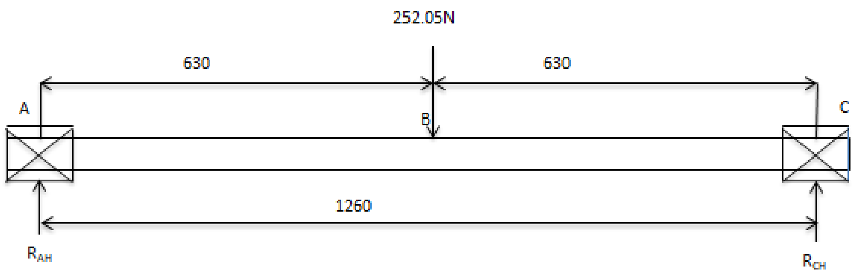


**Figure 9:** Forces on horizontal plane

The reaction forces on horizontal plane are shown on the Figure 9.

The forward draft force through ground wheel was 252.05 N horizontally and the forces were resolved as follows:

$R_{AH} + R_{CH} = 252.05N$ (40)

$R_{AH} = R_{CH} = 252.05N/2 = 126N$

Thus, bending moment on the shaft due to draft forces was calculated as follows:

Bending moment at x = 0.63 m from A,

$126 * 0.63m = 79.38Nm$

Bending moment at x = 1.26 m from A is zero.

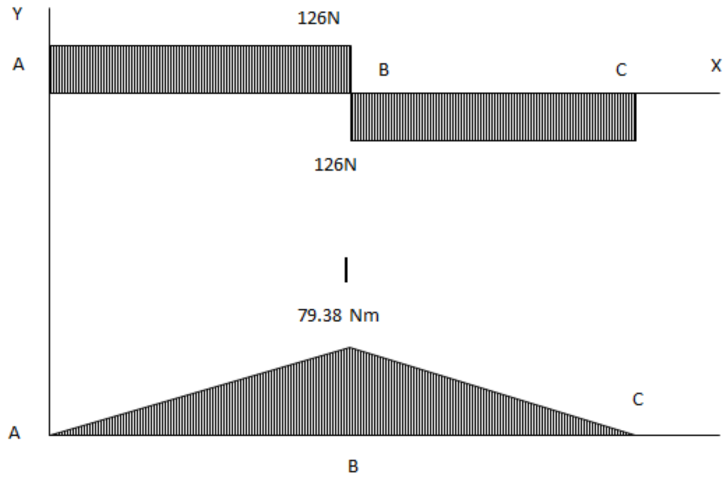


**Figure 10:** Shear and bending moment diagrams on the shaft on horizontal plane

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

$M_b = (M_V^2 + M_h^2)^{1/2}$ (41)

$M_b = ((19.16)^2 + (79.38)^2)^{1/2} = 81.66Nm$

$d^3 = 16 / (\pi \times Ss) \times [(K_b \times M_b)^2 + (K_t \times M_t)^2]^{(1/2)}$

Where: $K_b = 2$ , $K_t = 1.5$ , $Ss = 63MN/m^2$, $Mb = 81.66Nm$, $M_t = 66Nm$

$d^3 = (16/ \pi * 63*10^6 N/m^2 [(2*81.66)^2 + (1.5*66)^2]^{1/2}$

d = 0.0249m = 24.9 mm. Therefore, the standard size of 25 mm shaft diameter was used.

## CONCLUSION

The suggested machine design is cost-effective to manufacture and exhibits greater design simplicity compared to alternatives, featuring precise seed placement at the intended depth, appropriate spacing between rows, accurate fertilizer distribution, and optimal soil compaction. Implementing and producing this design will help achieve the goals and objectives of local farmers.

## REFERENCES


Central Statistical Agency (CSA). (2016). *Report on area and production of major crops: Private peasant holdings, Meher season 2015/2016*. Addis Ababa, Ethiopia.

Dubey, A. K. (2003). *Teaching material for training on computer-aided design and design methodology for agricultural machinery.* National Agricultural Technology Project on Team of Excellence on Agricultural Machinery Design and Development. Bhopal, India: CIAE.

Food and Agriculture Organization (FAO). (2009). *How to feed the world in 2050*. https://www.fao.org/fileadmin/templates/wsfs/docs/expert_paper/How_to_Feed_the_World_in_2050.pdf

Khurmi, R. S., & Gupta, J. K. (2005). *A textbook of machine design* (14th ed.). Eurasia Publishing House (Pvt.) Ltd.

Kurtz, G., Thompson, L., & Clwar, P. (1984). *Design of agricultural machinery*. John Wiley & Sons.

Morsey, E. B., Lambert, E. W., & Wang, Z. (1988). Flow rates of grains and oil seeds through sharp-edged orifices. *Transactions of the American Society of Agricultural Engineers, 31*(1), 226–231.

Olaoye, J. O., & Bolufawi, S. J. (2001). Design, fabrication and performance of a multi-purpose row planting machine. *Sustainable Environment, 3*(1), 7–20.

Regional Network for Agricultural Machinery (RNAM). (1991). Agricultural machinery design and data handbook (Seeders and Planters). *Economic and Social Commission for Asia and the Pacific,* 23–71.

Richard, G. B., & Keith, J. N. (2011). *Shigley's mechanical engineering design* (9th ed.). McGraw-Hill.

Senger, S. M. (2002). *Design and development of animal-drawn seeder for rice cum green manure crops for biasi cultivation system* (Master's thesis). Indira Gandhi Agricultural University, Raipur, Chhattisgarh, India.

Sharma, D. N., & Mukesh, S. (2010). *Farm machinery design: Principles and problems* (2nd ed.). Jain Brothers.

Theodre, B. (1985). *Mechanical engineering handbook*. McGraw-Hill.