Design and Development of an Engine-Operated Weeding Machine for Wheat Farm

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Abstract
Weeds constitute a serious problem to wheat crops and cause a great loss to the yield. Manual weeding is labor-intensive and time-consuming. Chemical weed control has a negative impact on both the environment and humans. To overcome these problems, an engine-operated weeder was designed and developed at Asella Agricultural Engineering Research Center (AAERC). The developed weeder was designed on the basis of agronomic and machine parameters. The developed prototype weeder consists of the mainframe, weeder blade, ground wheel, and power transmission system. The rated engine speed of 2800 rpm was reduced to 46 rpm of the ground wheels by using bevel gear, chain, and sprocket mechanism in three stages. The overall dimension of prototype weeder was 1650 mm in length, 800 mm in width and 1050 mm in height. The total production cost of the engine-operated weeder was 11,409.92 ETB. This paper is focused on machine design analysis and fabrication of prototype. Performance evaluation of prototype would be addressed in the next coming paper.
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1. INTRODUCTION

Wheat (*Triticum aestivum L.)* is one of the most important food crops of the world and a part of the family Poaceae that includes major cereal crops of the world such as maize, wheat, and rice. It is the staple food of the diet of several Ethiopians and provides about 15% of the caloric intake of the population of more than 90 million countries (FAO,2015). Wheat is one of the most important crops in Ethiopia, ranking fourth in total cereal production after maize, sorghum, and teff which contribute 10-12% each (Minot *et al.*, 2015). More than 4.7 million households are involved in wheat production each year, producing about 3.9 million tons of wheat on 1.6 million hectares of land, with a mean yield of 2.6 tons/ha (CSA,2013).

After South Africa, Ethiopia is the second-largest wheat producer in sub-Saharan Africa (FAO, 2015). Wheat is mainly grown in the highlands of Ethiopia, with latitudes 6 up to 16° N, longitude 35 to 42°E, at altitudes 1500-2800 meters above sea level, and an average minimum temperature of 6^0C to 11^0C (MOA, 2012). In Ethiopia, wheat covered an area of 1,696,082.59 ha, with average productivity of 2.6 t /ha during the main cropping season of Meher and a total production of 45,378,523.39 quintals (CSA, 2016). According to (CSA, 2014) reported that in the Oromia region, wheat covered an area of 875,641.45 hectares and total production was 24,703,210.41 quintals, and in Arsi, 208,308.22 hectares which produce 6,484,360.05 quintals. Out of the total grain crop area, 522,857.64 hectares were under cereals.

Despite its importance in Ethiopia, the national average wheat yield is 2.6 tons/ha, which is 12% below the average wheat yield in Africa and 24% below the average wheat yield in the world (CSA, 2016). Factors that reduce wheat yields are soil fertility decline, weeds, diseases, and insects. Weeds are one of the major constraints of wheat production and weed control is an important factor in increasing yields. There are many reasons for low wheat yields, but weed infestation is a fundamental and major factor in low yields in the crop production system (Shehzad et al., 2012). Weed is a plant that grows where humans don't want it and it can be another crop plant or a wild species (Gavali and Kulkarni, 2014).

Weed control is one of the most difficult tasks in agricultural production. Weed losses exceed those caused by any other agricultural pest. In Ethiopia, crop yield losses due to weeds vary from crop to crop and from region to region, due to different biotic and abiotic factors, it has been estimated that weeds cause a yield reduction due to delaying weeding by 15 percent to 62 percent (Kebede, 2000). The weed controls are mainly done by manual, chemical, and mechanical methods. In manual weeding, weeds are removed by using an indigenous tool, which is more effective but it is expensive, labor-intensive as well as time-consuming. In addition, the labor requirement for weeding depends on the weed flora, weed intensity, weeding time, and soil moisture content at the time of weeding. Nowadays, the use of herbicides is increasing day by day. It is preferred as a quick and effective weed control method without damaging the plants. But, it has adverse effects on human health and the environment. Today, the agricultural sector requires weed control without using chemicals to ensure food safety. Consumers demand high-quality food products and are particularly concerned about food safety. However, mechanical weeder is expected to encourage subsistence farmers leading to increased production and hence reducing poverty (Olukunle & Oguntunde, 2006). Mechanical weed control is effective not only in controlling weeds but also in loosening the soil between rows and increasing air and water retention (Hegazy et al., 2014). But this method of weed control has received much less scientific attention compared to the other weeding method in Ethiopia. In Ethiopia, weed control is done by manual weeding and chemicals by using herbicides. Weeding by manual methods requires extra labor force for a farmer. Cutlass and hoe are handy tools used for this purpose. Manual weed control is the most widely used weed control method but is labor-intensive, time-consuming, involves a lot of drudgeries, and causes health problems for a long time. Chemical weed control affects animals and human beings. It has consequences like cancer disease, environmental air pollution, increased acidity, and salinity of the soil. In most of the highlands, crops are planted at the same time and weeding operations are also performed at the same time. This results in shortages of labor during the peak seasons of weeding. The weeding labor bottleneck is especially problematic because some varieties are liable to weeding time and delay in weeding decreases crop yields due to competition for light, water, and nutrition. The use of a mechanical weeder is reducing drudgery, ensures ease of operation during weeding, and resultantly increases production. Therefore, to increase agricultural production and reduce the time and cost of weeding operations, there should be an urgent need to design and develop an engine-operated weeding technology. Hence, the main objective of this study was to design and develop an engine-operated weeding machine and to carry out the cost analysis of the developed weeding machine.

2. Materials and Methods

The design of the prototype engine-operated weeder consisted of the following main components; mainframe, weeder tine, depth control wheel, ground wheel, handle, engaging and disengaging unit, power transmission system (bevel gear mechanism, and chain and sprocket mechanisms).

2.1. Design Principles and Considerations

Engine-operated weeder was designed and developed by considering agronomic and machine parameters. The agronomic parameters like crop variety, row-to-row spacing, and other parameters like weeding time interval and physical properties of soil. Crop variety is an important parameter, which influences the mechanical weeding operation of the growth factor and the power requirement to operate the machine under field conditions. Row spacing helps in allowing the weeding tool to its effective operation. The row spacing of wheat varies from 180 to 200 mm. Because of this row spacing overall width of the machine was taken as 800 mm. The ground clearance of engine operated weeder was chosen as 300 mm. A critical weed control period is defined as a period in a crop's life cycle when it must be kept weed-free to avoid yield loss. The soil properties relevant to the design of the tool for weeding were identified as soil type, moisture, and bulk density. The moisture content of soil affects the draft required for the weeding tool of the weeder and the slip of the ground wheel. Soil having more moisture content gives more slip. Optimum soil moisture is needed at the time of weeding to minimize the field losses and energy input. Bulk density of soil is the measure of compaction of soil condition which influences the draft required for weeding. Based on crop and weed parameters, it was proposed to develop engine operated weeder for a 20 cm row spacing crop. Considering the draft limitations of the weeder and ensuring good maneuverability, an-engine operated weeder was designed and developed.

2.2. Functional Requirements

Different components of the weeder are designed for its functional and structural requirements.
The following functional requirements were considered while designing the weeder

- ☑ The weeder should cut, destroy and uproot weeds present in a furrow in a single run.
- ☑ The weeder should aerate the soil for the proper growth of crops
- ☑ Damage to crops during weeding operations should be as low as possible

2.3. Force required to drive the weeder

A rolling resistance force (F), which is assumed to act horizontally at the wheel and ground interface and wheel contact patch, therefore, the force required to drive the weeder was determined according to (Reece, 2002).

$$F_f = \left(\left(\frac{z}{Wd}\right)^{0.5} + i\right) \times W \qquad (1)$$

$$= \left(\left(\frac{2.5}{50}\right)^{0.5} + 0.05\right) \times 172\ \text{N} = 47.06\ N$$

Where: F_F = Force required to drive the weeder, N
C_R = Coefficient of rolling resistance
Z = maximum wheel sinkage depth (on a soft surface z ≈ 0.05Wd= 2.5 cm),
W_d = Wheel diameter, 50 cm, W = Weight on each wheel, 344 /2 = 172 N

i = Gradient of the ground, let i=5%

2.4. Power requirement of the weeder

Soil resistance has a considerable effect on the power requirement of the weeder. Also, the width of the cut and speed of operation influences the power requirement of the weeder. For calculating the power requirement of the weeder, maximum soil resistance should be taken (0.43 – 0.75 kg/cm^2) (Wilkinson and Braunback,1977). The speed of operation of the weeder 0.56 to 1 m/sec should be considered. The total width coverage of cutting tine was 60cm. The depth of operation was considered as 2 up to 5 cm and transmission efficiency was also determined.

$$Pd = \frac{SR \times d \times w \times v}{75} \quad (2)$$

$$= \frac{0.75 \times 5 \times 36 \times 1}{75} = 1.8 \text{ hp}$$

Where: Pd= power requirement, hp
SR = soil resistance, (0.75 kg/cm^2)
d = depth of cut, (5 cm)
w = effective width of cut, (36 cm)
v = speed of operation, m/s

Hence, the total power requirement for the weeder is estimated by:

$$\text{pt} = \frac{\text{pd}}{\eta} \quad (3)$$

$$= \frac{1.8}{0.82} = 2.195 \text{ hp or } 1.64 \text{ kW}$$

Where: Pd = Power required to dig the soil, hp
η =Transmission efficiency (82- 90%) (Reece, 2002). Hence, according to the power requirement, the available 5hp, (Honda GK-140) petrol engine was selected as the source of power. The engine was mounted in front of the ground wheel shaft.

2.5. Torque on the ground wheel

Torque on the ground wheel was determined according to Sharma and Mukesh, (2010).

$$\text{T} = \text{F}_\text{f} \times \left(\frac{\text{Wd}}{2}\right) \quad (4)$$

$$= 47.06 \times \left(\frac{0.5}{2}\right) = 11.765\ Nm$$

2.6. Selection of chain

Locally available chains and sprockets were selected for transmitting power from an engine to the ground wheel shaft by chain and sprocket mechanism. The chains are mostly used to transmit motion and power from one shaft to another. The power required to operate the weeder was transmitted from the engine and drive wheel through a chain drive. Since the power transmitted in the weeder is very low, the smallest size available chain, i.e. bicycle chain was used for engine operated weeder. For power transmission, a smaller teeth size sprocket was fitted on the engine shaft. Another sprocket with larger teeth size was fitted on the driving shaft of the ground wheels.

2.7. Determination of chain length

The chain length is given as suggested by Sharma and Mukesh (2010). The length of the chain was calculated as the number of links x chain pitch i.e.

$$\text{L} = \text{M} \times \text{p} \quad (5)$$

Where: m = number of chain links
p = chain pitch, mm,

But, the number of chain links

$$\text{M} = \frac{2\text{C}_\text{d}}{\text{p}} + \frac{(\text{N}_1 + \text{N}_2)}{2} + \frac{(\text{N}_2 - \text{N}_1)^2}{4\pi^2\text{cd}} \quad (6)$$

Where: C_d = Centre to center distance, mm
P = Chain pitch, (08 B =12.7 mm)
N_1 = Number of teeth on the smaller sprocket.
N_2 = Number of teeth on the larger sprocket

Putting the values in eqn.5

$$\text{M} = \frac{2 \times 257 \text{ mm}}{12.7\text{mm}} + \frac{(14 + 36)}{2} + \frac{12.7 \text{ mm } (36 - 14)^2}{4\pi^2 \times 257\text{mm}}$$

$$= 66.76 \text{ pitches}$$

The nearest even number of pitches for the chain is 66 pitches. Therefore, the corrected length of the chain was calculated according to Eqn (5)

$$L = M \times p = 66 \times 12.7 \text{ mm} = 838.2 \text{ mm}$$

The exact center-to-center distance between the sprockets is expressed by Sharma and Mukesh (2010)

$$C = (e + \sqrt{e^2 - 8m}\ ) \times \frac{P}{4} \qquad (7)$$

But,

$$e = Lp - \frac{(N_1 + N_2)}{2} \qquad (8)$$

$$= 66 - \frac{(14+36)}{2} = 41$$

$$m = \left(\frac{N_2 - N_1}{2\pi}\right)^2 \qquad (9)$$

$$= \left(\frac{36 - 14}{2\pi}\right)^2 = 12.26$$

Therefore, the corrected center-to-center distance between the sprockets was:

$$C = (41 + \sqrt{(41)^2 - 8 \times (12.26\ )} \times \frac{12.7\ mm}{4}$$

C =256.495 mm

$C \approx 257$ mm

2.8. Chain velocity

Average chain velocity was calculated by Sharma and Mukesh (2010).

$$Vav = \frac{N_e \times P \times Rpm}{376} \qquad (10)$$

Where:

Ne = the number of teeth on the driving sprocket, 14

Rpm = maximum revolution of the driving sprocket, (46 rpm)

P= commercially available chain pitch, (0.25 in)

Vav = Average chain velocity, m/s

Therefore, the estimated average chain velocity was 0.428 m/sec

2.9. Determination of load in chain

The total load (force) on the driving side of the chain is given by Sharma and Mukesh, (2010).

$$FT = F + Fc + Ff \qquad (11)$$

Where: FT= the total force, N

F = the force due to power transmission, N

F_f = Frictional force, N

Fc = Centrifugal force on the chain, N

$$F = \frac{P}{Vav} \qquad (12)$$

But, $P = T \times \omega$

$$= 11.765 \text{ Nmm} \times \frac{2\pi \times 46 \text{ rpm}}{60} = 56.67\text{W}$$

$$F = \frac{P}{Vav} = \frac{56.56 \text{ W}}{0.428 \text{ m/s}} = 132.41 \text{ N}$$

Where:

p = power at the weeder wheel, w

Vav = Average chain velocity, m/s

$$Fc = \frac{W \times (Vav)^2}{g} \qquad (13)$$

$$Fc = \frac{6.6 \times (0.428)^2}{9.81} = 0.123 \text{ N}$$

$$Ff = W \times kf \times Cc \qquad (14)$$

$$= 6.60\ N/m \times 2 \times 0.257m = 3.392\ N$$

Where:

Cc = Nominal center to center distance between the sprockets, 257 mm

P= Power at the weeder wheel/ power to be transmitted, hp

Kf = Friction factor = 4 for the horizontal drive, 2 for the inclined drive, and 1 for the vertical drive since the sprockets are inclined aligned, kf = 2 (Norton, 2005).

W = Weight per meter of the chain, 6.60 N/m

g = Gravitational acceleration, m/s^2

Hence, the calculated total load on the chain was 136 N

According to the American National Standard Institute (ANSI) standard, the minimum tensile strength of the chain is 3470N. To avoid breakage or failure of the chain, the safety factor should be more than one. So, Checking the safety factor, S_f (Sharma and Mukesh, 2010).

$$Sf = \frac{\text{tensile strength}}{\text{FT}} \tag{15}$$

$Sf = \frac{3470\ \text{N}}{136\ \text{N}} = 25$, The value is greater than unity. Therefore, the chain selected is safe.

2.10. Bearing selection

The size of a bearing to be used depends on the size of the shaft required and the available space. In addition, a bearing must have a high enough load rating to provide an acceptable combination of life and reliability (Khurmi and Gupta, 2005). Bearing size can be determined using the maximum resultant force acting on a bearing and the desired maximum lifespan.

The bearing pressure Pb on the edge of the shaft is given by:

$$P_b = \frac{R}{A} \tag{16}$$

$$P_b = \frac{167}{0.314} = 531.8\ \text{N/mm}^2$$

$$R = \sqrt{RA_V{}^2 + RA_H{}^2} = \sqrt{(165.2)^2 + (23.53)^2}$$

$$= 167\text{N}$$

R =Maximum resultant force acting on a bearing, N

A = Area of contact, $\left(\pi \frac{d^2}{4} = 0.314\ mm^2\right)$

RA_V = maximum resultant force acting on a bearing at point A on the vertical plane (N),

RA_H = maximum resultant force acting on a bearing at point A on the horizontal plane (N),

The allowable bearing capacity P_{bp} is given by (ACME, 1987)

$$P_{bp} = P_0 \times C_1 \times C_2 \tag{17}$$

$$= 2820 \times 0.3 \times 0.72 = 609.12\ \text{N/mm}^2$$

Where:

P_0 = Allowable bearing pressure (2820 N/mm^2)

C_1 = Correction factor (0.3) (ACME, 1987)

C_2 = Correction factor (0.72) (ACME, 1987)

The condition to be satisfied for selecting the bearing is that, $P_b < P_{bp}$

Therefore, the bearing is selected as the condition for selecting the bearing is satisfied

$$P_b\,(531.8) < P_{bp}(609.12)$$

Dynamic load rating can be determined using Eqn. (18), (Khurmi and Gupta, 2005)

$$C = \left(\left(L\alpha fR\frac{N \times Hm}{10^6}\right)^{\frac{1}{k}}\right) \tag{18}$$

$$C = \left(\left(3 \times 0.167\ \text{KN}\ \frac{46\ \text{rpm} \times 20{,}000\text{hr}}{10^6}\right)^{\frac{1}{3}}\right)$$

$$= 0.613\ \text{KN}$$

Where:

C = basic dynamic load rating, (KN)

N = revolution of the shaft in rpm

Hm=desired maximum lifespan of the bearings (hr)

Lαf =Load application factor or service factor

k = exponent for the life equation (3.0 for ball bearings, 10/3 for roller bearings)

The bearing size was selected using the maximum resultant reaction force (0.167 KN) acting on the bearing and the dynamic load rating (0.613 KN) was determined using Eqns. (16) and (18). The application factor serves as a factor of safety; increases the design load to take into account overload and dynamic loading. For machinery applications with moderate impact, the application load factor (Lαf) ranges from 1.5 to 3.0. The maximum value of application factor 3.0 was used for bearing selection. The desired maximum life of a bearing was selected based on the application of the bearing. For machines used for 8 hr per day; hence, the desired maximum lifespan of the bearings ranges from 12000-20,000hr. Therefore the maximum lifespan value of 20,000 hr was selected to determine the basic dynamic load rating. Bearing number 204 with a bore of 20 mm and width of 14 mm was selected since the minimum shaft diameter has been determined to be 20 mm.

2.11. Design of power transmission system

The power transmission system was designed to reduce engine output shaft speed from 2800 rpm to 46 rpm on the ground wheel shaft. The power reduction was designed in 3 steps. Three-speed reduction steps are given below:

a) First step speed reduction:

In the first step, speed reduction was obtained using a bevel gear mechanism. The bevel gear is used to change the direction of motion by 90˚ and reduce the speed. Assume the number of teeth of the pinion gear mounted on the countershaft as 10 with a speed ratio of 2:1. The number of larger teeth on the other shaft was calculated by using the formula Khurmi and Gupta (2005);

$$\text{Speed ratio} = \frac{T_2}{T_1} \tag{19}$$

Where: T_1= Number of teeth on pinion gear shaft;
T_2 = Number of teeth on larger gear / the other side of the shaft. From eq. (19)

$$\frac{T_2}{T_1} = 2, \quad \frac{T_2}{10} = 2, \qquad T_2 = 20;$$

So, 20 teeth on the other side of the gear were selected. The speed of the shaft in the power transmission system was calculated using the formula by Khurmi and Gupta (2005).

$$N_1 T_1 = N_2 T_2 \tag{20}$$

Where: N1 = speed engine of the output shaft, rpm;
N_2 = speed on the other side of the bevel gear shaft, rpm,
T_1 = No. of teeth on pinion shaft;
T_2 = No. of teeth on the other side of the bevel gear shaft, N_1 = 2800 rpm, T_1= 10 teeth, T_2=20 teeth; from eq. (19)

$$N_2 = \frac{N_1 \times T_1}{T_2} \quad \text{so,} \quad N_2 = \frac{2800 \times 10}{20} = 1400 rpm$$

N_2 = 1400 rpm, in first-speed reduction step engine speed, reduced from 2800 to 1400 rpm.

b) Second step speed reduction:

In the second step speed reduction, a chain and sprocket have been selected. For the power of 5 hp, 10B-ISO chain number was selected as per IS 2403-1991. Assume the number of teeth on the sprocket which is mounted on the bevel gear shaft as 14 with a speed ratio of 4:1. The number of teeth of a sprocket on the middle shaft was calculated by using the formula Khurmi and Gupta (2005):

$$\text{Speed ratio} = \frac{T_4}{T_3} \tag{21}$$

Where, T_3 = Number of teeth on a sprocket mounted on a bevel gear shaft;
T_4 = Number of teeth on the sprocket which is mounted on the middle shaft eq. (21)

$$\frac{T_4}{T_3} = 4 = \frac{T_4}{14} = 4 = T_4 = 56$$

An available sprocket with 56 teeth was selected. The speed of the middle shaft in a power transmission system is calculated using the formula by Khurmi and Gupta (2005)

$$N_3 T_3 = N_4 T_4 \tag{22}$$

Where:
N_3= Speed of bevel gear shaft (rpm),
N_4= Speed of countershaft (rpm);
T_3 = Number of teeth on a sprocket mounted on a bevel gear shaft;
T_4= Number of teeth on the sprocket which is mounted on the middle shaft.
$N_2 = N_3$ = 1400 rpm, T_3 = 14 teeth, T_4 = 56 teeth. So, From eq. (22)

$$N_4 = \frac{N_3 \times T_3}{T_4} \quad \text{so,} \quad N_4 = \frac{1400 \times 14}{56} = 350 \text{ rpm}$$

So, N_4 = 350 rpm, thus in the second step the speed was reduced from 1400 rpm to 350 rpm.

c) Third step speed reduction:

Third, the step speed was reduced using a set of chain and sprocket mechanisms. Assume the number of teeth of a small sprocket mounted on the other end middle shaft as 12 teeth and a speed ratio of 7.5:1. The number of teeth on larger sprockets mounted on the ground wheel shaft was calculated using the formula Khurmi and Gupta (2005).

$$\text{Speed ratio} = \frac{T_6}{T_5} \qquad (23)$$

Where:
T_5 = Number of teeth on smaller sprocket mounted on the middle shaft;
T_6 = Number of teeth on larger sprocket mounted ground wheel shaft. From eq. (23)

$$\frac{T_6}{T_5} = 7.5, \quad \frac{T_6}{12} = 7.5, \qquad T_6 = 90;$$

Thus, 90 teeth on a larger sprocket were selected. The speed of the ground wheel shaft in the power transmission system was calculated using the formula by Khurmi and Gupta (2005).

$$N_5T_5 = N_6T_6 \qquad (24)$$

Where:
N_5 = Speed of smaller sprocket mounted on the middle shaft, rpm; N_6 = Speed of larger sprocket mounted on ground wheel shaft, rpm, T_5 = No. of teeth on a smaller sprocket mounted on the middle shaft, T_6 = No. of teeth on a larger sprocket mounted on the ground wheel shaft,

$N_4 = N_5 = 350$ rpm, $T_5 = 12$ teeth, $T_6 = 90$ teeth, from eq. (24)

$$N_6 = \frac{N_5 \times T_5}{T_6} \quad \text{so,} \quad N_6 = \frac{350 \times 12}{90} = 46 \text{ rpm}$$

The speed was reduced from 350 rpm to 46 rpm in the third step of the speed reduction process. As a result, for the weeding operation, engine speeds were dropped from 2800 rpm to 46 rpm in power transmission.
Generally, engine power was transmitted to ground wheels through a power transmission system (bevel gear, chain, and sprocket mechanism). The power transmission system consisted of speed reduction bevel gear, chain, and sprocket. The power should be transmitted from the engine to the intermediate shaft which is connected to the bevel gear and from the bevel gear shaft to the chain and sprocket then the ground wheel starts forward direction and the weeder is start operations

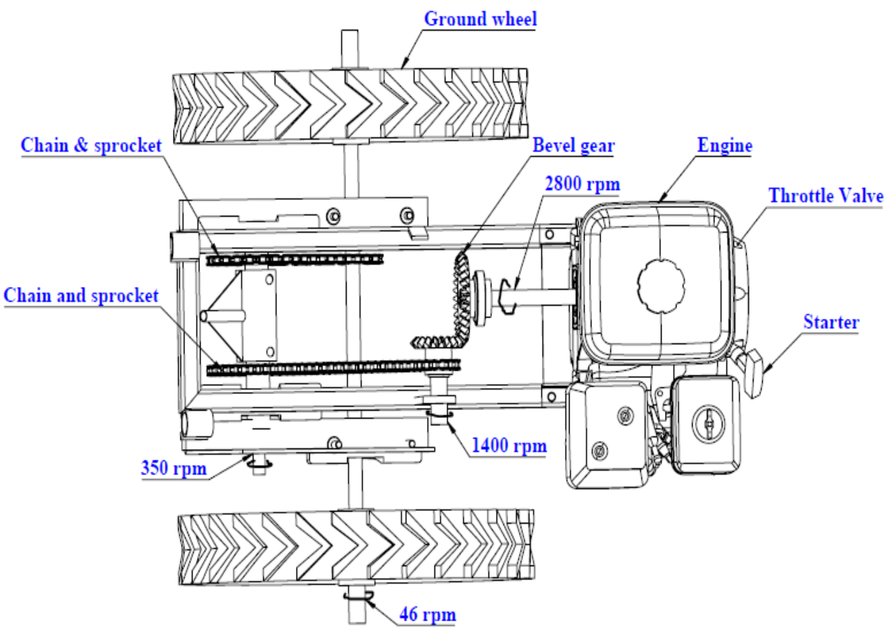

Figure 1: Diagram showing the power transmission system

2.12. Design of weeding blade

The tyne of the weeder should be used to uproot the weed at the desired depth without damage to the crops. The depth of weeding at which weed should be uprooted in the soil depends on the crop variety and the soil moisture level. For uprooting the weed, a sweep-type cutting blade was used as a uniform depth of weeding should be required. A sweep-type blade was selected for the fixed on engine operated weeder frame. The performance of the sweep blade was better than the straight and curved blade with minimum draft force per unit working width and having the highest performance index reported by Biswas and Yadav (2004). While designing the sweep blade following assumptions were taken into consideration. Wheat row to row spacing =18 cm to 20 cm, the depth of weeding varies from 2 cm to 5 cm. crop protection zone 4 cm. The angle of internal friction (Ø) ranges (10^0 to 30^0) depending on soil type.

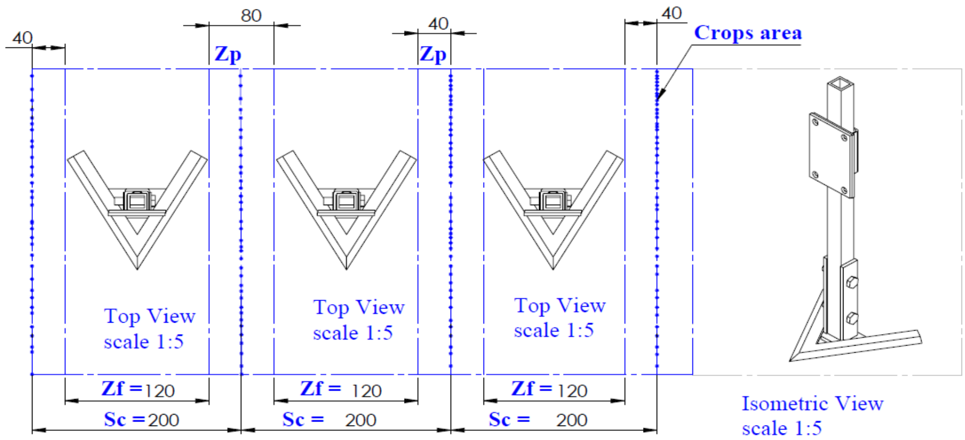

Figure 2: Diagram showing the weeding blade (all dimensions are in mm).

The cutting width of the sweep type tyne was found by using the formula Sharma and Mukesh, (2010).

$$S_c = Z_f + Z_P \quad (25)$$

Where, S_c = row spacing, cm.

Z_f = effective soil failure zone, cm

Z_p = crop protection zone, cm

$20\text{ cm} = Z_f + 4 \times 2\text{ cm}$

$Z_f = 20\text{ cm} - 8\text{ cm} = 12\text{cm}$

The protection zone is multiplied by two since the protection zone has to be provided on both sides of the crop. The effective soil failure zone was calculated by using the formula by Sharma and Mukesh (2010).

$$Z_f = [\,W + 2d\tan\varnothing_s] \quad (26)$$

$12 = W + 2 \times 5\tan 15^0$

$W = 12 - 10\tan 15^0 \approx 10\text{ cm}$

So, the width of the sweep was taken 100 mm

While designing the sweep, the apex angle, and condition for easy undercutting of the weeds by the sweep blade were taken into account. The sweeps were attached to the shank with the help of a nut and bolt.

2.13. Design of the shank

The shank was designed to have proper fixing on the tool frame of the engine-operated weeder. Three shanks of the weeder were fitted on the main frame of the weeder with the help of bolt and nut arrangement with provisions to adjust the depth by moving the shank vertically.

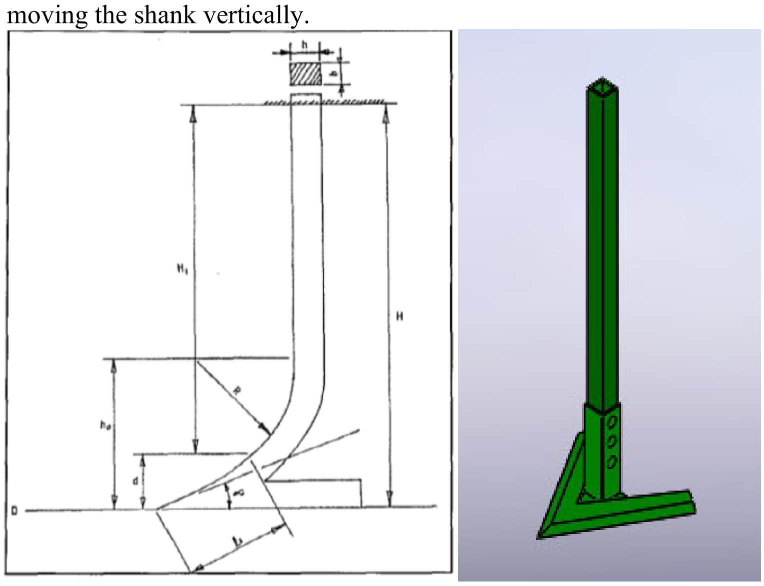
Figure 3: Schematic of diagram designed shank of the weeder

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

$$R_C = \frac{h_o - l_1\sin\alpha}{\cos\alpha} \quad (27)$$

Where: l_1= the length of breast of the shovel, α = angle between surface of shovel and ground,

h_o = Height of the shank from its tip to the bent portion, mm

The height H of the tine depends on the manner of its fastening to the frame. The minimum clearance H_1 between the ground and the lower edge of the frame should be > 200 mm. The slope of the tine is most frequently adopted in the range from 100 to 250 mm and the radius of curvature $R \leq 120$ mm. Let, b x h = bare cross-section area (cm^2) L = length of breast of tyne Substituting the values, ho = 140 mm, 1 = 110 mm, and α = 25° in Equation (4). The value of" ho", "l" and "α" were selected in accordance with research findings by Varshney *et al.,* 2005. Now radius of curvature was determined as;

$$R_C = \frac{h_o - l_1 \sin\alpha}{\cos\alpha}$$

$$R_C = \frac{140\text{ mm} - 110\sin 25^0}{\cos 25^0} = 103.2\ mm$$

$$H_T = a_{max} + H_I + \Delta H$$

Where, ΔH = the length of the upper part of the tine serving for tine fastening, it was assumed as 100 mm and H_1= 353.5 mm

$$a_{max} = l\sin 25^0 = 110\sin 25^0 = 46.50\ mm$$

Therefore, substituting those values in equation H_T, Height (H) of shank from the tip of tine to the frame was calculated as follows

$$H_T = a_{max} + H_I + \Delta H$$
$$= 46.50\text{ mm} + 353.5\text{ mm} + 100\text{ mm}$$
$$= 500\text{ mm}$$

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 the design of the shank soil type of the study area was a sandy loam soil which can be considered as a heavy soil thus the unit draft was taken as 0.25 kg/cm^2, Assuming, a = 3 cm at the bottom and b = 5 cm at the top, the cross section of furrow is trapezoidal in shape. The draft force exerted on the cutting blade was determined using the following equation

$$D = K_O \times n \times w \times d \qquad (28)$$
$$= \left[0.25 \times 3\text{ cm} \times \left(\frac{3\text{ cm}+5\text{ cm}}{2}\right) \times 5\text{cm}\right] \times 9.81$$
$$= 0.25\text{ kg/cm}^2 \times 3 \times 10\text{ cm} \times 5\text{ cm}$$
$$= 147.15\text{ N}$$

Now factor of safety was assumed to be 3. Therefore, total draught exerted on the weeder tine and each draft of the weeder tines was determined by the following equation.

$$D_t = D \times SF$$
$$= 147.15\text{ N} \times 3$$
$$= 441.45\text{ N}$$

$$\text{The maximum draft force exerted on sweep type tyne} = \frac{\text{Total draft}}{\text{No. of tynes}}$$

$$\text{Maximum draft} = \frac{\text{total draft}}{\text{No. of tynes}} = \frac{441.45}{3} = 147.15\ \text{N (for each tyne of the weeder)}$$

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

$$h = \frac{H}{3}\left(\frac{b+2a}{b+a}\right) = \frac{500}{3}\left(\frac{50 + (2 \times 30)}{50+30}\right) = 23\ mm$$

Assuming shank as cantilever beam, the maximum bending moment for a cantilever length of 42.3 cm (considering 5 cm tine depth and the force is acting from the centroid of trapezoidal section, from geometry (fig.3), it acts at 2.7 cm from bottom of tine). The length of shank attached to mainframe was 10 cm. Therefore, effective length of shank was determined by the following formula

$$l_s = \left(50 - 2.7 - \frac{10}{2}\right) = 423\ mm$$

The maximum bending moment (M_b) calculated as follows

$$M_b = D \times l_s$$
$$= 147.15\ N \times 423\ mm$$
$$= 62{,}244.45\ Nmm$$

Bending stress is given in the formula (Sharma and Mukesh, 2010). Assuming bending stress (σb = 56 N/mm^2), section modulus of the shank (Z) was determined as the following formula.

$$Z = \frac{M_b}{\sigma b} \qquad (29)$$

$$Z = \frac{62{,}244}{56} = 1{,}111\ mm^3$$

Where:

Mb = maximum bending moment, Nmm

σb = bending stress (56 N/mm^2 for mild steel)

Z = section modulus of the shank, mm^3

b = width size of mild steel, mm

For design purpose, Take b=30 mm size square hollow pipe M.S and t= thickness was assumed. Section modulus of the shank of the weeder was calculated by using the formula by Khurmi and Gupta (2005)

$$Z = \frac{b^4 - h^4}{6b} \quad (30)$$

$$Z \times 6 \times 30 = 30^4 - h^4 = 1{,}111\ mm^3 \times 6 \times 30\ mm$$

$$h^4 = 610{,}020\ mm^4$$

$$h = \sqrt[4]{610{,}020} = 27.95 \approx 30\ \text{mm}$$

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

$$t = \frac{b - h}{2} = \frac{30 - 27.95}{2} = 1.025\ mm \approx 1.5\ mm$$

Therefore, shank was made from MS square pipe material $30\ mm \times 30\ mm \times 1.5\ mm$ size was quite safe and the size was available in the market. The projected end of the shank was fitted to the sweep with nut and bolt and the other end to the main frame of the engine-operated weeder.

2.14. Design of handle

The handle of the weeder should be designed to be adjustable for the different heights of the male/female which can adjust according to own height which reduces drudgery. The adjustable handle should help the operator of the weeder at the time of operation. Two handles were provided at the rear of the machine which should be attached to the mainframe. The length of the handle was calculated based on the average standing height of the person operator. Therefore angle of inclination (θh) with the horizontal was calculated by Sharma and Mukesh (2010).

$$\tan \theta h = \frac{a_1}{a_2} \quad (31)$$

Where,

a_1 = height of the center of wheel to the elbow, 80 cm

a_2 = horizontal distance of wheel center from the operator in operating condition normal to the elbow line, 115 cm

$$\tan \theta h = \frac{80\ \text{cm}}{115\ \text{cm}} = 0.696 = \theta h = \tan^{-1}(0.696) = 35^{\circ}$$

$$\sin(\theta h) = \frac{900\ \text{mm}}{\text{Lh}}$$

$$\text{Therefore,} \quad \text{Lh} = \frac{900\ \text{mm}}{\sin(35^{\circ})} = 140\ \text{cm}$$

Therefore, the Length of the designed handle was 140 cm from the standing height of the operator.

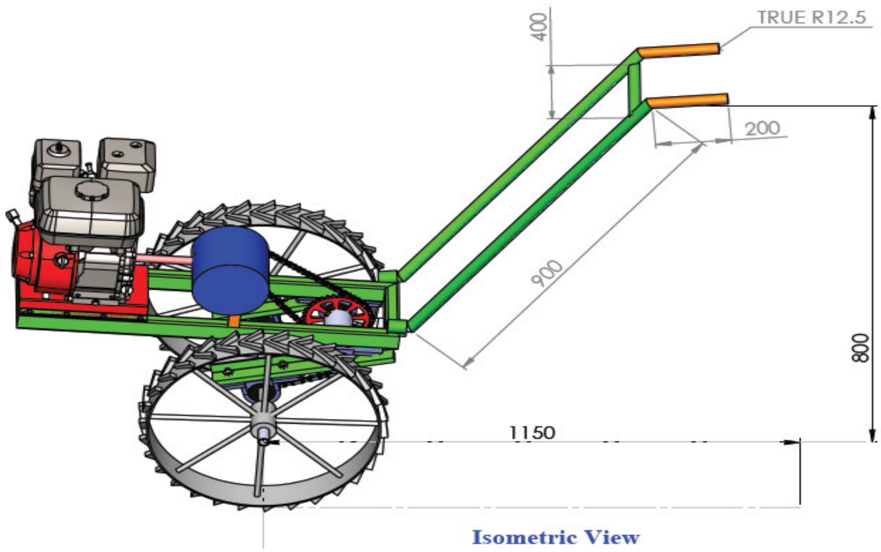

Figure 4: Diagram showing the dimensions, in mm, of the handle

2.15. Drive wheel design

The function of the wheel is to provide traction and supply power to the cutting unit of the weeder. The type of drive wheel should be used depending on the ground conditions. Verma (1986) suggested that the diameter of ground wheels was 22.5 to 40 cm for animal operation and 40 to 60 cm for power operation. To design the diameter of the ground wheel it was taken 50 cm for this work. The wheel of the weeder was made from 3 mm thick and 80 mm wide sheet metal. The maximum shear strength (τ_{max}) of sheet metal is 80 MPa. Each wheel requires eight spokes made from mild steel rods with a diameter of 12 mm and length of 200 mm welded to the rim and hub at the center of the wheel that served as bushing or shaft bearing, at equal intervals.

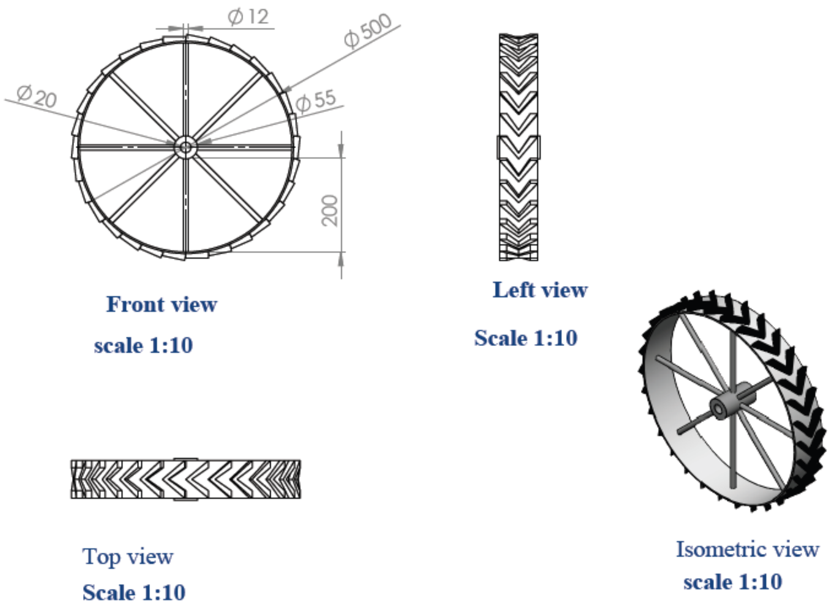

Figure 5: Details of the drive wheel

According to Nisbett and Richard (2011), the shear stress on the wheel was estimated as below.

$$\tau = \frac{T}{2\,A_m \times t_w} = \frac{T}{2\pi t_w (r_0 - 0.5 t_w)^2} \quad (32)$$

$$T = F_f \times \left(\frac{Wd}{2}\right)$$

$$T = 47.06\ \text{N} \times \left(\frac{0.5\ \text{m}}{2}\right) = 11.765\ \text{Nm}$$

$$\tau = \frac{11.765\ \text{Nm}}{2 \times \pi \times 0.003 \times (0.25 - 0.5 \times 0.003)^2}$$

$$\tau = 10{,}107.37\ \text{N/mm}^2$$

$$\tau = 10.11\ \text{kPa}$$

Where: T = the torque produced by the wheel, Nm
r_o = Outer radius of the wheel, 0.25 m
t_w = Thickness of the wall/wheel, 0.003 m
τ = Shear stress on the wheel, kPa
F_f = Force required driving weeder, (47.06 N)
W_d = Ground wheel diameter, (50 cm)

The shear stress of the wheel was compared with the maximum allowable shear stress of the metals, 10.11 kPa $\ll \ll \tau_{Max}$, 80 MPa. Hence, the wheel was safe from failure for weeding the crops.

2.16. Design of hub

The hub of a rigid wheel is one of the most significant components. The spokes and shaft are supported by it. The hub's diameter was estimated using the formula below. The exterior diameter of the hub is given by Nisbett and Richard (2011)

$$D = 1.5d + 25.00\ mm \tag{33}$$
$$= 1.50 \times 20\ mm + 25.00\ mm$$
$$= 55\ mm$$

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

The length, L of the hub is given by Nisbett and Richard (2011)

$$L = \frac{\pi \times d}{2} \tag{34}$$
$$= \frac{\pi \times 20\ \text{mm}}{2} = 31.42\ mm$$

Design of lugs

To ensure proper traction, lugs are provided around the circumference of the ground wheel. The lugs are welded to the ground wheels outside circumference. The soil acceleration force was calculated using an equation as given by Srivastava (2003).

$$F_{s1} = \frac{\rho_{\times} g}{g}\, b \times d \times V_o^{\,2}\, \frac{\sin\theta}{\sin(\theta + \alpha)} \tag{35}$$

Where: F_{S1} = soil acceleration force, N
b = width of penetration lugs, m
d = depth at the penetration of lugs, m
V_0 = forward speed of weeder, m/s
θ = tool lift angle, degrees
α = angle of forward failure surface, degree
ρ = bulk density of soil, $\frac{kg}{m^3}$ and, g = gravitational force, m/s^2

The sizes of lugs on the ground wheel were selected as 25 mm in width and 10 mm in thickness. The projection of lugs is considered from the tip of the circumference of the ground wheel as 18 mm, the depth of lugs penetrated in the soil. Lugs are welded perpendicular to the ground wheel at 90º to the soil surface. The bulk density of soil was 1400 kg/m³ and a maximum forward speed, of 2.5 km/hr was determined. It is assumed the internal angle of friction was 36º, Angle of the forward failure surface was calculated using the formula (Kankal *et al*., 2014)

$$\alpha = \frac{1}{2}(90 - \varphi) \tag{36}$$

Where: φ = angle of internal friction;

$$\alpha = \frac{1}{2}(90 - 36) = 27^0$$

Hence, $F_{s1} = \frac{1400 \times 9.81}{9.81} \times 0.025 \times 0.018 \times (0.69)^2\, \frac{\sin 90}{\sin(90+27)} = 0.337\ \text{N}$

Considering three lugs are in contact with soil, the total soil acceleration force is given by

$$A_0 = 3 \times F_{s1} \tag{37}$$

$A_0 = 3 \times 0.337 \text{ N} = 1.0098 \text{ N}$
Where: A_0 = total soil acceleration force on the ground wheel, N, F_{S1} = total soil acceleration on each lug, N
The total soil acceleration at the center of the projected length of lugs and hence the maximum bending moment is given at this point. The maximum bending moment is given by
$M = A_0 \times L$
M = 1.0098 N × 9 mm = 9.09 Nmm
Where: M = maximum bending moment, Nmm,
A_0 = total soil acceleration force on the ground wheel, N
L = distance between the point of action of soil and top edge of the ground wheel, mm
The bending stress induced in the material of the lugs was calculated following the formula given by (Khurmi and Gupta, 2005)

$$\sigma_b = \frac{M}{Z} \quad (38)$$

Where, M = maximum bending moment on the lugs, Nmm σ_b = stress induces on the material of lugs, N/mm², Z = section modulus of the ground wheel, mm³; section modulus for the rectangular section is given by Varshney *et al.*, (2005).

$$Z = \frac{1}{6} bt^2 \quad (39)$$

Where, Z = section modulus, mm³
t = thickness of lugs, mm
b = width of lugs, mm;
Putting value in equation (37);

$$Z = \frac{1}{6} \times 25 \times 5^2 = 104.16 \text{ mm}^3$$

Putting values of Z and M in equation (38)

$$\sigma_b = \frac{M}{Z} = \frac{9.09 \text{ Nmm}}{104.16 \text{ mm}^3} = 0.087 \text{ N/mm}^2$$

Comparing bending stress calculated in equation (38) with allowable bending stress. The bending stress calculated (0.087 N/mm²) was less than the allowable bending stress of 70 N/mm². Hence, the design is safe.
Considering the spacing between lugs as 48 mm and the numbers of lugs is obtained as the following equation.

$$N = \frac{\pi D_g}{S} \quad (40)$$

Where, Dg = diameter of the ground wheel, mm.
S = spacing between lugs, mm
N = numbers of lugs

$$N = \frac{\pi D_g}{S} = \frac{\pi \times 500}{48} = 33$$

Hence, 33 lugs have been provided on the ground wheel. Lugs calculated in number welded on the outer periphery of the drive wheel at 48 mm equal intervals to facilitate easy traction in soil. The lugs 18 mm in height welded at an angle of 30 degrees with the axis of rotation to reduce the slip.

2.17. Determination of shaft diameter
A shaft is a rotating machine element that transmits power (Khurmi and Gupta, 2005). The design of the shaft includes the determination of the correct shaft diameter to ensure satisfactory strength and rigidity while transmitting power under various operating and loading conditions. The design of the shaft is based on the maximum shear stress theory. Shafts are usually subjected to torsion, bending, and axial loads. For a solid shaft having little or no axial loading, the diameter of the shaft was calculated using the equation given by the ASME code (ASME, 1995)

$$d^3 = \frac{16}{\pi \times \tau_{max}} \sqrt{(K_b \times M_b)^2 + (K_t \times M_t)^2} \quad (41)$$

Where,
d = Diameter of the shaft; mm
Mt = Tensional moment; Nm
Mb = Bending moment; Nm
τ_{max}= Allowable stress; MN/mm²
Kb = Combined shock and fatigue factor applied to bending moment; Kt = Combined shock and fatigue factor applied to tensional moment;
For rotating shafts, when the load is suddenly applied with minor shock, Kurmi and Gupta (2005) recommended that values of Kb = 1.2 to 2.0 and Kt = 1.0 to 1.50 be used. Furthermore, it described that for the shaft without a

keyway, the allowable stress (τ) must be 55 MN/mm^2 and for the shaft with a keyway the allowable stress (τ) should not exceed 40 MN/mm^2

2.18. Determination of torque transmitted by the shaft

The torsional moment transmitted through the shaft was calculated using the following formula(Ryder, 1989)

$$M_t = \frac{P \times 60 \times 10^3}{2 \times \pi \times N} \quad (42)$$

Where, P = Power, kW
T = Torque transmitted by the shaft, Nm
N = Speed of the shaft, rpm

$$P = V \times F \quad (43)$$

But, $V = \omega r = \frac{2\pi \times 46\, rpm}{60} \times 0.25\text{ m} = 1.2\text{ m/s}$

$P = 1.2\text{ m/s} \times 47.06\text{ N} = 56.67\text{ W}$

Where: P = Power required to drive a machine, W
F = Force require to drive the weeder, N
V = Forward speed of operation, m/s

Therefore, the torsional moment transmitted through the shaft was calculated as follows

$$M_t = \frac{56.67 \times 60}{2 \times \pi \times 46} = 11.765\text{ Nm}$$

The maximum resultant bending moment on the shaft was determined from the following expressions (Nisbett *et al.*, 2011).

$$M_b = \sqrt{(Mv)^2 + (Mh)^2} \quad (44)$$

Where,
Mb = Maximum resultant bending moment, Nm
Mv = Vertical bending moment, Nm
Mh = Horizontal bending moment, Nm

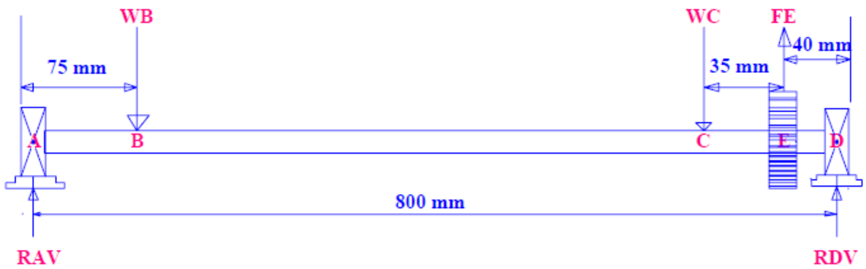

Figure 6: Vertical load distribution on the shaft. (All dimensions are in mm)

Where:
RAV =Vertical reaction at wheel A
RDV=Vertical reactions at wheel D,
WB = Half of the total weight acting at bearing B (172 N), WC = Half of the total weight acting at bearing C (172 N),
FE = Vertical chain force/ load (136 N)

To know the unknown forces of RAV and RDV, we use equilibrium equation methods. The reactions RAV and RDV were determined by taking moments about A;

$\sum M_A = 0$

$RDV \times 800 + FE \times 760 - WC \times 725 - WB \times 75 = 0$

$RDV \times 800 = 172 \times 725 + 172 \times 75 - 136 \times 760$

$800RDV = 172 \times 725 + 172 \times 75 - 136 \times 760$

$800RDV = 34{,}240$

$$RDV = \frac{34{,}240}{800} = 42.8\text{ N}$$

Using equilibrium equations methods the summation of all forces along the y-axis gives the unknown value.

$\sum F_Y = 0$

$RDV + FE + RAV = WC + WB$

$42.8 + 136 + RAV = 344$

$RAV = 344 - 42.8 - 136$

$RAV = 165.2\text{ N}$

To draw a shear force diagram, shear force at all segment points should be calculated as follows.

For A, Shear force = 165.2 N
For AB, Shear force = 165.2 N − WB = 6.8 N downward
For BC, shear force = −6.8 N − WC = 178.8 N downward
For CE, Shear force = −178.8 N + FE = 42.8 N downward
For ED, Shear force = 42.8 N downward
For D, Shear force = −42.8 N + 48.2 N = 0 N
Thus, the maximum vertical bending moment (BM) on the shaft was computed using Figure 13 as follows:-
BM at A and D = 0 Nmm
BM at B, RAV × 75 mm = 165.2N × 75 mm = 12,390 Nmm = 12.39 Nm
BM at C, RAV × 725 mm − WB × 650 mm

165.2N × 725 mm − 172N × 650mm = 7,970 Nmm = 7.97 Nm

BM at E, RAV × 760 mm − WB × 685 mm − WC × 35 mm 165.2N × 760 mm − 172 × 685 mm − 172 × 35 mm = 1,712 Nmm = 1.712 Nm
Thus, the maximum vertical bending moment on the shaft is 12,390 Nmm = **12.39 Nm**

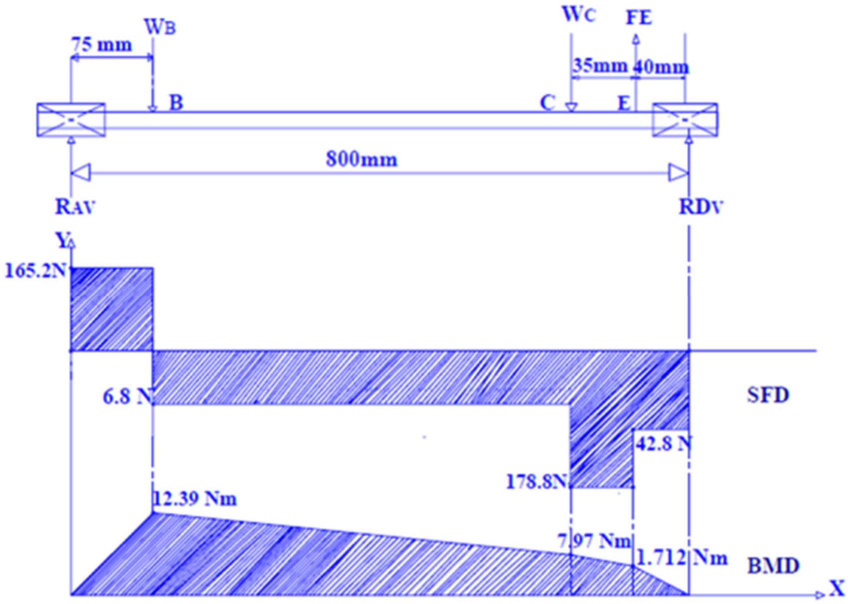

Figure 7: Shear forces and vertical bending moment diagrams
The reaction forces distributions on the horizontal plane are shown below

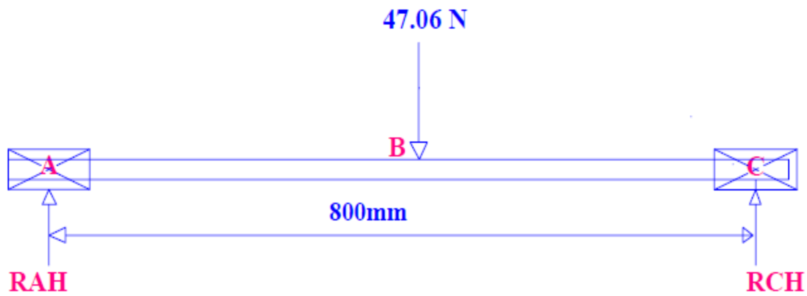

Figure 2: The reaction forces distribution on the horizontal plane
The forward driving force of the weeder through the wheel is 47.06 N horizontally and resolving the forces horizontally:

$\sum M_A = 0$
RCH × 800 − 47.06N × 400 mm = 0
800 RCH = 47.06N × 400 mm
RCH = 23.53 N
$\sum FY = 0$
RAH + RCH = 47.06 N
RAH + 23.53 N = 47.06 N
RAH = 47.06 N − 23.53 N
RAH = 23.53 N

Hence, the bending moment on the shaft due to horizontal forces was calculated as follows:
BM at x = 400 mm from point A
BM at A and C= 0 Nmm

$$\text{BM at B}, R_{AH} \times 400 \ = 23.53\ \text{N} \times 400 \ = \mathbf{9.412\ Nm}$$

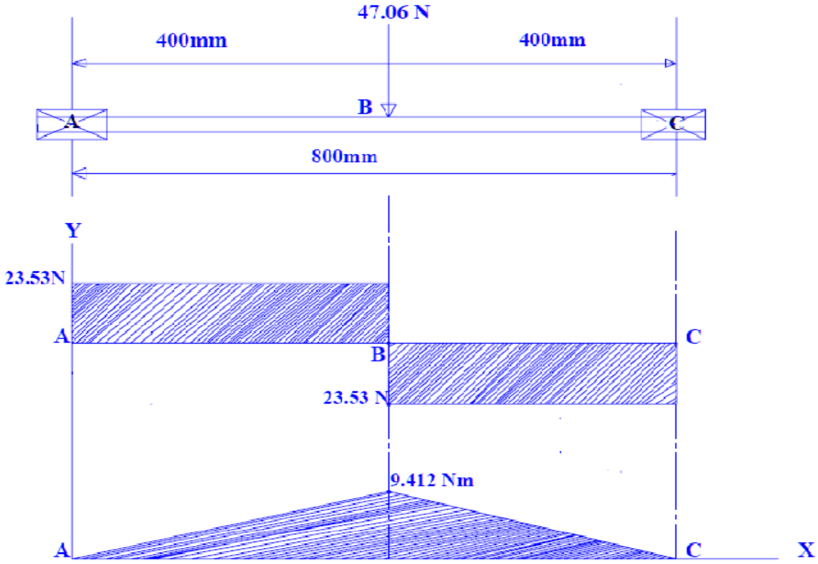

Figure 8: Shear forces and horizontal bending moment diagrams on the shaft

The maximum horizontal bending moment on the shaft is 9.412 Nm at 400 mm from A. The total resultant components of horizontal and vertical bending moments on the shaft were obtained as follows:

$M_b \ = \sqrt{(Mv)^2 + (Mh)^2}$

$M_b \ = \sqrt{(12.39\ \text{Nm})^2 + (9.412\ \text{Nm})^2}$

$M_b \ = 15.56\ \text{Nm}$

$d^{\,3} = \dfrac{16}{\pi \times \tau_{max}} \sqrt{(\ K_b \ \times M_b \ )^2 + (K_t \ \times M_t\,)^2}$

Where, $M_t = 11.765\ \text{Nm}$

$M_b \ = 15.56\ \text{Nm}$

$K_t = 1.5$

$K_b = 2$ and $\tau_{max} = 40\ \text{MPa}$

$d^{\,3} = \dfrac{16}{\pi \times \tau_{max}} \sqrt{(\ K_b \ \times M_b \ )^2 + (K_t \ \times M_t\,)^2}$

$d^{\,3} = 0.0000046\ \text{m}$

$d = 0.01663\text{m} \approx 16.63\ \text{mm},$

Therefore, the standard size of 20 mm shaft diameter has been used for the weeder machine.

3. Results and Discussion

The experimental findings obtained from the present study have been discussed in the following heads:

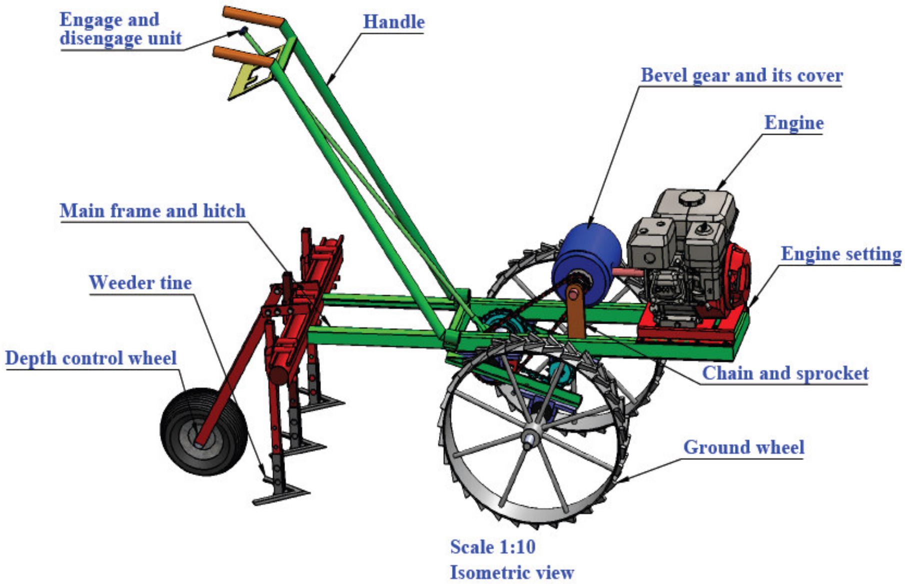

Fig 9 Isomeric view of a designed machine

3.1. Development of an engine-operated weeder

The prototype an-engine operated weeder was fabricated based on dimensions obtained from the design. The prototype weeder consists of a mainframe, handle, power transmission system, ground wheel, sweep blade, shank, and engage and disengage clutch. The isometric view of the engine-operated weeder is shown in Fig.9. Components of the prototype weeder are presented in Table 1. The specifications of the prototype machine are presented in Table 1 also. The overall dimension of the prototype weeder was 1650 mm in length, 800 mm in width, and 1050 mm in height.

Table1: List of materials and costs for production of an engine-operated weeder prototype

S/№	Components	Type of raw materials	Standard Size (mm)	Unit price (EBirr)	Used material size	Total price (EBirr)
1	Main frame	MS rectangular	30 × 50 × 6000	678.00	2,140 mm	241.82
2	Blade	Sheet metal	3×1000 × 2000	1500.00	36,000 mm^2	27.00
3	Shank	MS square pipe	2.5 × 25 × 6000	550.00	1,650 mm	151.25
4	Handle	water pipe	Ø 25 × 6000	485.00	2,200 mm	177.83
5	Axle shaft	MS rod	Ø 20 × 6000	598.00	800 mm	79.73
6	Chain	Cast Iron	12.7 mm pitch	600.00	2 psc	1200.00
7	Sprocket	Cast Iron	14 teeth	400.00	2psc	800.00
8	Sprocket	Cast Iron	36 teeth	530.00	2psc	1060.00
9	UCP bearing	Cast steel	204	380.00	4pcs	1520.00
10	Clutches	Mild Steel	Ø30 × 6000	750.00	800 mm	100.00
11	Engine setting	MS angle iron	4 × 40 × 6000	760.00	560 mm	79.93
12	Ground wheel	Sheet metal	2×1500×2000	1200.00	251,200mm^2	100.48
13	Bevel Gear	Cast steel		1500.00	1 psc	1500.00
14		Bolt and nut	M10 ×30mm	5.33	20 psc	106.60
15		Bolt and nut	M8 × 30mm	4.50	12 psc	54.00
16	Spokes	Round bar	Ø 12 × 6000	598.00	3,200 mm	318.93
17	Hub	Mild Steel shaft	Ø 55 × 6000	850.00	62.84 mm	8.90
18	Lugs	Round bar	Ø 12 × 6000	598.00	4,480 mm	446.51
Sub-total						**8,122.98**

Table 2: Specifications of the engine-operated weeder

Sr. No.	Particulars	Details
1	Name of machine	Engine operated weeder
2	Make of machine	AAERC
3	Overall dimension of the machine (L x W x H)	1650 x 800 x 1050 mm
4	Weight of machine	34.4 kg
5	Power source	5 hp petrol start diesel run engine
6	Fuel used	diesel
7	Fuel tank capacity	3.9 lit
8	Engine details	4 stroke, 1 cylinder
9	Speed at engine	2800 rpm
10	Displacement	197 cm^3
11	PTO shaft rotation	Counter-clockwise from drive end
12	Weight of engine	14 kg
13	Gear type	Bevel
14	Chain drive	ISO 10 B bush roller chain
15	Clutch	Dog clutch
16	Axle	20 mm in diameter
17	Ground wheel	500 mm in diameter
18	Lug	33 no. 25 x 25 mm in size lugs welded at the periphery of the ground wheel
19	Details of weeding components	
	Frame dimension (L x B) mm	960 x 240 mm
	Type of blade	Sweep type
	No of blade	3
	Distance between blade	Adjustable
20	Shank	25 mm x 25 mm x 2.5 mm in dia. and 500 in length

Main frame

The rectangular main frame of size 960 x 600 x 240 mm was fabricated. The frame of the weeder was made from mild steel (M.S) rectangular pipe shape of 50 × 30 sizes (mm) × 4 mm thickness cross-section which is the standard (4.46 rectangle pipe per kg/meter) type. All components of the weeder were assembled and fitted on the frame.

Ground wheel

The ground wheel provides good traction and gives power to the cutting unit of the weeder. The ground wheels were used to get traction in field conditions. Two mild steel lugged ground wheels of 500 mm diameter were mounted on the opposite end of the ground wheel shaft both ends of the central shaft connected to the transmission box. The spacing between two wheels can be adjusted based on the row spacing of the crop. MS rods of 12 mm diameter 8 in numbers were welded as spokes on the central hub of the ground wheel. The 55 mm long hub was made to fit on the 20 mm diameter ground wheel shaft.

Handle

Two handles were provided at the rear of the machine which should be attached to the mainframe. The handle was made of a 25 mm diameter water pipe with a plastic grip at the ends. The overall length of the handle was 1200 mm with two bends from the point of attachment by bolt and nut and a height of 900 mm from ground level, also convenient for the operators.

4.Conclusions

Solid work drawing (fig.9) was used for the design and fabrication of an engine-operated weeder. The different component of the weeder mentioned in the previously described subheadings was fabricated and assembled in the workshop of AAERC. The power transmission system was designed to reduce engine output shaft speed from 2800 rpm to 46 rpm on the ground wheel shaft. Each of the major components was designed, fabricated, and then assembled properly. This paper is focused on machine design analysis and fabrication of prototype. Performance evaluation of propototype would be addressed in the next coming paper. The total production cost of the engine-operated weeder was 11,409.92 birr.

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