

Design and development of diesel engine driven mechanical sugarcane harvester

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Abstract: Sugar production in Ethiopia started in 1954/55 when the Wonji Sugar Factory was commissioned and produced 15,843 tons of white sugar in the first campaign. Sugarcane harvesting is the most costly operation in sugarcane farming. Although more sophisticated self-propelled sugarcane harvesters have been used in developed countries, manual harvesting is still practiced in most undeveloped countries like Ethiopia. A simple, efficient, and less complicated mechanical harvesting machine aid was desirable in areas where manual sugarcane harvesting. Manual sugarcane harvesting requires intensive labor, quite a lot of money, and time-consuming tasks that involve considerable human drudgery. So, this study focused on designing and developing sugarcane harvesting machines that are cheap, easily affordable by the farmers and sugar factories, easy to maintain, and less laborious to use. The developed sugarcane harvesting machine consists of a frame, discs with a cutting unit, shaft, safety frame, wheels, and 3.75 kW power diesel engines. Hence this project work overcomes these problems and aims to develop a small-scale sugarcane harvesting machine. The materials used for construction and manufacturing were by considering the different parameters during operation and harvesting.

Keywords: bevel gear, discs, sugarcane, sugarcane harvesting machine, sugar cane mechanization

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1 Introduction

Sugarcane harvest is a complex logistical operation that involves the cutting and loading of cane in the fields, the harvesting operation must be carefully planned and coordinated to avoid any waste of valuable resources. Harvesting of sugarcane at the proper time i.e., peak maturity, by adopting the right technique is necessary to realize the maximum weight of the mill (Chen et al., 2021). Cane harvesting is the most costly operation in sugar cane farming. Variable sizes and designs of semi-mechanization sugarcane harvesting machinery are available. Full mechanization systems may be whole stalk harvesting

systems or chopper harvesting systems. The important issues that have to be addressed are improving harvesting rates and reducing extraneous matter levels of the cane delivered to the factory. In Ethiopia sugar cane harvesting system is manual. It requires several laborers to perform the work. During this operation, there is a chance of damage by insects, snakes, injuries, muscle pain, and some other parts of pain (Abdel Mawla et al., 2014; Ahmed and Alam-Eldin, 2015). Different types of harvesting machines are available in the market namely paddy harvester, tea harvester, potato harvester, and wheat harvester are available on a small scale except the sugarcane harvesting machine (Adarsh et al., 2013; Jamadar et al., 2017). The

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mechanization of sugarcane harvesting is an essential input to modern agriculture, as it enhances better productivity, besides reducing human drudgery and the cost of cultivation. The cost of sugarcane harvesting by the mechanized process is almost one-third of the complete manual process (Abdel Mawla et al., 2014). The main aim of this study was to design and fabricate a diesel engine motor-driven sugarcane harvesting machines which were made from Ethiopian available materials, cheap, but strong enough to perform those tasks.

2 Materials and methods

2.1 Study site description

The sugarcane harvesting machine prototype fabrication and its performance test were conducted in the Wonji Shoa sugar factory Dodota sub-branch workshop and farm field respectively. It was one of the sugar factories in Ethiopia which was found at the

Oromia region near Adama City at 25.3 km and from Addis Ababa at 110 km. The factory's locations are approximately latitude $8^{\circ}22'30''$ to $8^{\circ}25'0''$ N and longitude $39^{\circ}12'30''$ to $39^{\circ}17'30''$ E. Wonji Sugar Factory was the oldest and the pioneer in the history of Ethiopia's sugar industry and started production in 1954. Accordingly, the newly built and modern Wonji Shoa Sugar Factory has currently a design capacity of crushing 6,250 tons of cane a day and producing 174,946 tons of sugar per annum which with further expansion work reach up to 12,500 productions to 220,700 tons of sugar a year. Currently cane agricultural expansion project is being carried out around the areas known as Wakie Tiyo, Welenchiti, and North Dodota areas where cane cutting had been operated by manpower.

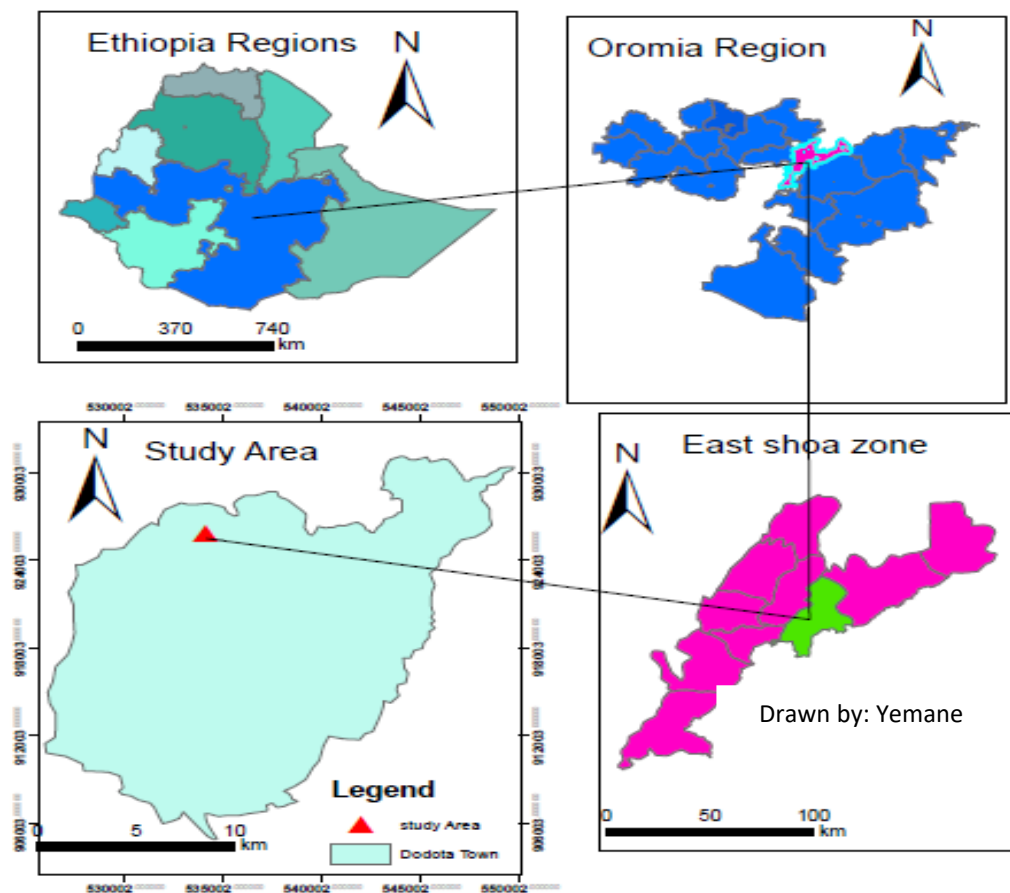


Figure 1 Location map of the study area

2.2 Material selection

From nature, the stem of the sugarcane was harder than other parts of the sugarcane. So, the material

selection could be related to the strength of the sugarcane stem to minimize and control the bending and breaking-off part of the machine during harvesting.

During construction, different materials were selected from the store and other related were bought from the market to fulfill the requirements. The materials selected from the store and markets were based on durability, strength, endurance, availability, and cost.

The material designed for sugarcane harvester's machine parts like disc and cutter must be resistant to corrosion and it was made from non-corrosive

materials like stainless steel, but some parts like the frame, and safety guard were constructed from high thickness flat sheet metal. The shaft was selected from the available store and machined to the optimum required size. The horizontal shaft and wheel shaft are 50 mm in diameter and the vertical shaft is 30 mm in diameter.

Table 1 Tools and equipment used for harvesting machine development

No	Machine name	Purpose	No	Machine name	Purpose
1	Drill machine	Making hole	7	Hammer	To strike an object
2	Lathe machine	Threading /cutting	8	Screwdriver	Tighten screw
3	Grinding machine	Cutting tool	9	Spanner	Tighten nut and bolt
4	Welding machine	Welding	10	Snip	Cutting sheet metal
5	Steel tape	Measure linear distance	11	File	Smooth rough edge
6	Vernier calipers	Measure outer and inner diameter			

Table 2 Sugarcane harvesting machine specifications

No	Components	Quantity	Material and specification
	IC engine	1	3.75 kW diesel engine
	Discs	2	Made from a steel alloy with a bottom and topper cutter having 300 mm in diameter and thickness of 6 mm and 3 mm respectively.
	Cutter	6 (for bottom) 4 (for topper)	Made from stainless steel which was connected to each disc The bottom cutter had a dimension 90 mm × 80 mm × 50 mm and 3 mm of length, bottom width, top width, and thickness respectively and The Topper cutter had a dimension of 120 mm × 70 mm × 50 mm and 3 mm in length, bottom width, and top width and thickness respectively.
	Frame	-	Made from 6 mm thickness flat metal of 800 mm length, 600 mm width, 200 mm height.
	Shaft	2	Machined from the mild steel. The horizontal shaft has a dimension of 50 mm in diameter and a length of 1100 mm. The vertical shaft has a diameter of 30 mm and a length of 1450 mm.
	Gear	2	Used to transfer horizontal rotation motion to the vertical shaft (Gearwheel and pinion have 17 and 17 teeth number respectively). Made from a steel alloy.
	Bearing	7	Made from chromium alloy steel which was attached to the rotating part to transfer rotation
	Gearbox	1	Machined from sheet metal used to cover the gears
	Pulley	2	Pulley made from the cast with larger Pulley $\varnothing = 138$ mm, smaller pulley $\varnothing = 50$ mm
	Belt	1	Made from strong plastic material with a total length of 1095.16 mm. Selected from store B-49.
	Handle	2	Made from round hollow bar steel used to push the machine
	Wheels	3	Made from flat metal with a thickness of 4 mm.

2.3 Section of the power

The power required for cutting sugarcane was done by considering the different forces or shear strength. The shear force required to cut the wood was found to be 450 N and the shear force required to cut the sugarcane was 106.57 N (Zode et al., 2015). So, the shear force required to cut sugarcane was less than the wood cutting. The torque could be determined by Equation 1 by considering the radius of the disc with the cutter and the force required to cut the sugarcane.

$$T = F \times r \quad (1)$$

Where:

T is the torque (N m);

F is the force required to cut the sugarcane (N) ;

r is the radius of the discs (m).

$$T = 106.57 \text{ N} \times 195 \times 10^{-3} \text{ m} = 20.78 \text{ N m}$$

To minimize the power consumption of sugarcane harvesting the different authors recommended the velocity of the cutter. The optimal rpm to cut the sugarcane was 13.8 m s^{-1} , with an oblique angle of 35° and a disc angle of 27° (Gupta and Oduori, 1992). Whereas for 0° oblique angle and 8° tilt angle of the cutter edge, the recommended optimal velocity of the cutter to cut the sugarcane was 22 m s^{-1} (Lui et al.,

2007). From the previous research, the recommended velocity for my project was 22 m s⁻¹.

$$\text{Optimal rpm required to cut sugarcane} = \frac{\text{blade cutting velocity (m s}^{-1}\text{)} \times 60 \text{ (s)}}{\text{radius (m)} \times 2\pi} \tag{2}$$

So, the optimum rpm was = $\frac{22(\text{m s}^{-1}) \times 60 \text{ s}}{0.195\text{m} \times 2\pi} = 1085$ rpm.

The power required to cut the sugarcane was by Equation 3.

$$P = \frac{2\pi NT}{60} \tag{3}$$

$$P = \frac{2\pi \times 1085 \times 20.78}{60} = 2.359 \text{ kW } 3.145 \text{ hP} \approx$$

4.9 hP the engine was needed for operation.

Where:

P is power required to cut the sugarcane, kW;

N is speed of cutter, rpm;

T is torque of the cutter, N m.

2.4 Design of the frame

A strong and less-weight frame with the desired thickness and according to the row spacing of sugarcane was fabricated to hold the power unit, power transmission, and safety frame. As per the design the parts were cut by a cutting machine and welded with a welding machine for suitable strength. It was constructed by joining the flat sheet metal of size 800 mm×600 mm×100 mm in length, width, and height respectively with a thickness of 6 mm.

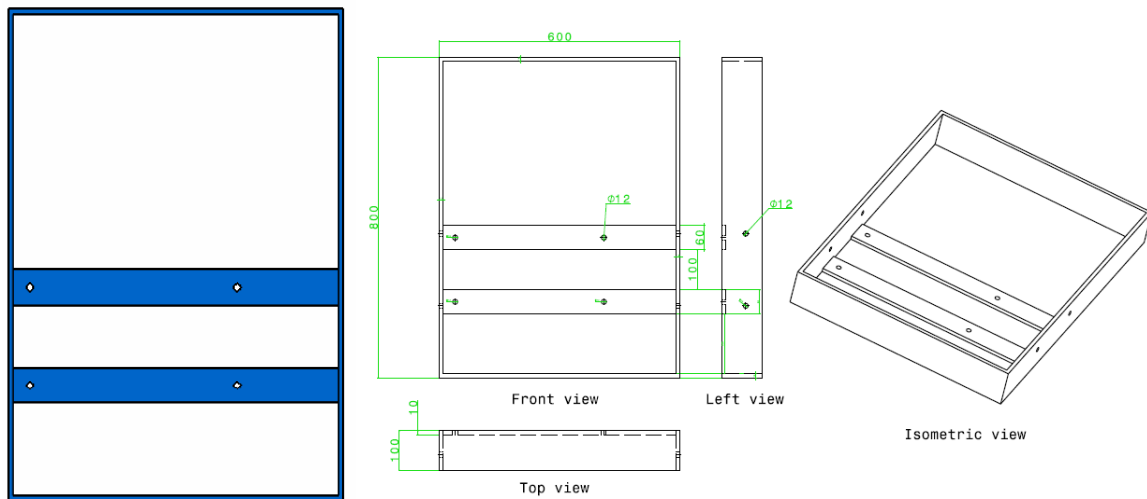


Figure 2 Frame design

2.5 Design of motor box

The motor box was a rectangular-shaped structure that was constructed from 2 mm thick sheet metal with

a dimension of 400 mm length, 400 mm width, and height of 500 mm. It was attached by a nut to the mainframe.

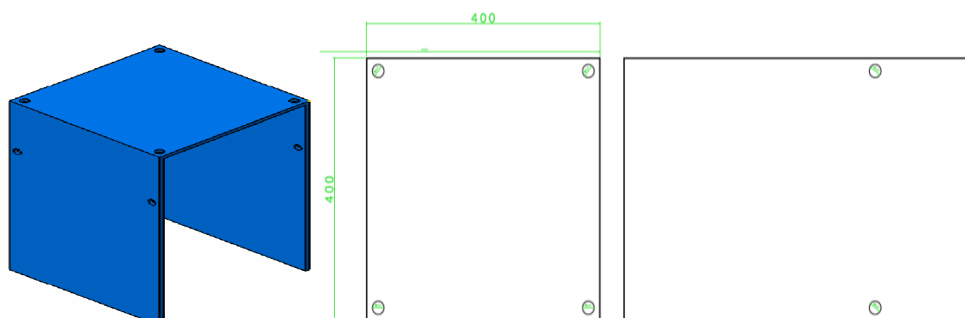


Figure 3 Motor box

2.5 Design of the discs with cutter and its kinematic consideration

Cutters commonly used on agricultural field machinery typically perform either, a combination of reciprocating and translating motions, or a

combination of rotary and translator motions. In either case, the translator motion, which should be referred to as advance motion, was necessary to transport the implement to its work (Gupta and Oduori, 1992). The rotational motion from the horizontal shaft to the

vertical shaft was transferred by the action of the gear in the gearbox. The vertical shaft was machined with access to pass through the hole of the gear to transfer the rotational motion to the cutting part of the machine. The equal rotational motion was transferred to the cutter for the bottom and top. As the machine travels to harvest the sugarcane, the discs rotate uniformly about its fixed axis from the motor to the horizontal shaft and then to the vertical shaft. During this, the sugarcane was beaten firstly by the top cutter which has a longer as compared to the bottom one, and then followed by the bottom cutting disc. Gradually the beaten sugarcane was inclined due to the centripetal force which attracts the bottom part of the sugarcane and drops it to the ground to the right side of the operator.

The six cutting blades were attached and distributed uniformly to the bottom disc with a space of 60° and each cutter was constructed with a thickness of 4 mm. The cutting edge was machined with a

grinder machine at an angle of 8° . The cutters must be strong enough to overcome the force and minimize the bending during operation. The disc was connected to the vertical shaft by the action of a bolt and nut. The bolt and nut facilitate that if the cutter or disc is bent and damaged during operation it is easy to remove the disc and cutter for maintenance.

The top cutter was also attached and distributed on the 300 mm diameter disc at the angle of 90 degrees and it has a length of the 120 mm having a 3 mm thickness. The top cutter was less in weight to minimize the load on the body. When compared with a bottom cutter the top cutter needs less energy minimizing the number of cutters and the thickness of the disc. They were longer than the bottom cutter to remove the top part of the sugarcane first and minimize the load applied due to it. Each cutter was sharp, longer, and light in weight to minimize the load and it must be strong enough to overcome the force during harvesting.

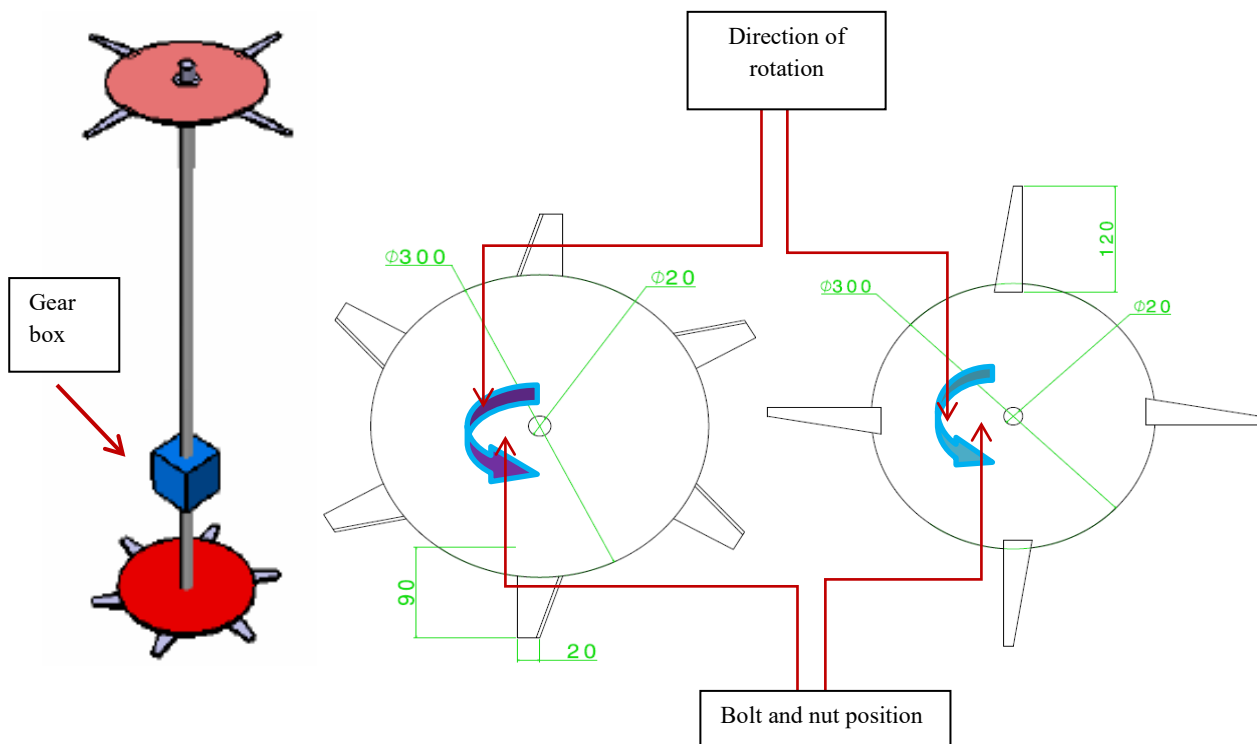


Figure 4 Disc with cutters

Figure 4 explains the configuration of cutting systems and rotation of the cutter, at an arbitrary time, which should be the initial moment of configuration. At this moment, the origin in which the bolt and nut

position, \circ , coincides with the center of rotation of the cutter.

The following assumptions were made in the cutting analysis of the cutter:

(1) The cutting edge was fixed orientation relative to the rotating shaft.

(2) The blades were uniformly distributed around the periphery of the disc. Thus, if there are n cutting blades in the total circumference of the disc, then the angle was equal to $\frac{2\pi}{n}$ radians or $\frac{360}{n}$ degrees.

(3) The cutter rotates in the horizontal plane about its centroid axis with a constant rotational velocity ω (rad s⁻¹).

The angular momentum is the tendency of the rotating object to continue with the same angular speed in the same direction and the angular momentum developed at the tip of the cutter was Equation 4.

$$M = m \times \omega \times r \quad (4)$$

Where: M is angular momentum, kg m² s⁻¹; m is the mass of discs with cutters, kg; ω is the angular velocity of cutter, rad s⁻¹ and r is the radius of discs with cutter, m.

$$M = (5.45 \text{ kg} + 0.0276 \text{ kg}) \times 22 \text{ m s}^{-1} \times 0.195 \text{ m} \\ = 23.49 \text{ kg m}^2 \text{ s}^{-1}$$

Centripetal force is a force that causes an object to move in the circle at a constant speed because the centripetal acceleration is pointed toward the center of the circle, then from Newton's second law in vector form was calculated as Equation 5 (Abdulkadir et al., 2009).

$$F_{cp} = \frac{m \times V^2}{r} \quad (5)$$

Where: F_{cp} is centripetal force, N; m is the mass of disc, kg; V is the linear velocity of the disc, m s⁻¹ and r is the radius of the disc, m.

$$F_{cp} = \frac{m \times V^2}{r} = \frac{5.4776 \text{ kg} \times (22 \text{ m s}^{-1})^2}{0.195 \text{ m}} = 13595.68 \text{ N}$$

2.6 Selection of pulley

Pulleys were one of the most important power transmission elements, but they require careful design. A pulley reduces these effects with rolling contact between the cable and the machine, but it must be of sufficient size, typically 20 times the cable diameter, to prevent fatiguing of the cable strands. If the power ranges from 1.5 to 15 kW, the diameter was preferred to use a pulley having up to 200 mm diameter and a B-belt section (Adarsh et al., 2013).

Table 3 Belt selection criteria

Section belt	Power ranges in KW	Top width, mm	Bottom width, mm	Thickness, mm	Weight per meter length in N
B-section	1.5-15	17	9	11	1.98

The pulley diameter was 138 mm for the driven and 50 mm driver, with bevel gear teeth number 17 for both the driver gear and driven pinion respectively. The coefficient friction and groove angles were 0.3 and 20° respectively. If the power (3.36 kW or 4.9 hp) and rpm (3000) of the engine are known from the manufacturer's manual then calculate the torque on the shaft of the engine.

The torque on the engine shaft due to a belt is calculated as Equation 6.

$$T = \frac{P \times 60}{2\pi N} \quad (6)$$

$$T = \frac{3.36 \times 60}{2\pi \times 3000} = 10.7 \text{ N m for motor shaft.}$$

The velocity of the belt was calculated as Equation 7 (Khurmi and Gupta, 2005).

$$V = \frac{\pi \times d \times N}{60} \quad (7)$$

Where: V is the velocity of the belt, m s⁻¹; d is the diameter of the pulley at the engine, m and N is the speed of the motor pulley, rpm.

$$V = \frac{\pi \times d \times N}{60} = \frac{3.14 \times 0.05 \times 3000}{60} \quad \text{or} \quad = \frac{3.14 \times 0.138 \times 1085}{60} = 7.85 \text{ ms}^{-1}$$

From the speed ratio of the motor shaft to the horizontal shaft, the speed of the horizontal shaft was obtained by the following Equation 8 (Khurmi and Gupta, 2005).

$$N_1 \times D_1 = N_2 \times D_2 \quad (8)$$

Where: N_1 is angular speed of motor shaft, rpm, D_1 = diameter of a driving pulley, m, N_2 = angular speed of the horizontal shaft, rpm, D_2 = diameter of the driven pulley, m.

From Equation 8, the speed of the horizontal shaft was 1085 rpm. This rpm was required to cut the sugarcane which was equal with the optimum rpm of the sugarcane cutting speed.

The belt was selected from the store and it has its length. The selected belt was B-49. The length of the belt was determined by the following relation (Khurmi and Gupta, 2005).

$$L = 2X + \pi \frac{D_1 + D_2}{2} \tag{9}$$

Where:

L = Length of the belt, mm;

D_1 = Diameter of driver pulley, 50 mm;

Length of the belt:

D_2 = Diameter of driven pulley, 138 mm;

X = Centre to center distance of belt, 400 mm;

$$L = 2 \times 400 + \pi \frac{138 + 50}{2} = 1095.16 \text{ mm.}$$

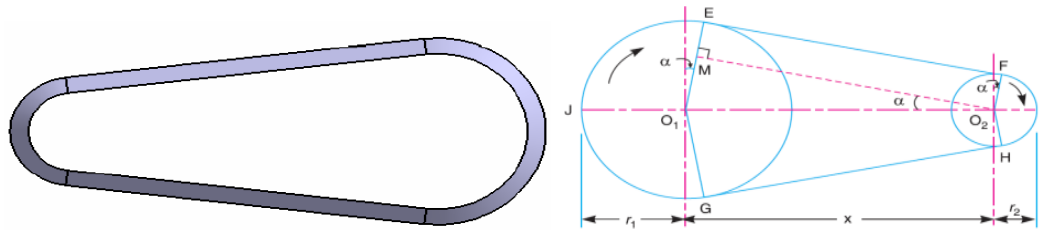


Figure 5 Belt expression

Centrifugal tension: Since the belt continuously runs over the pulleys, therefore, some centrifugal force is caused, whose effect is to increase the tension on both, the tight as well as the slack sides. At lower belt speeds (less than 10 m s^{-1}), the centrifugal tension was very small, but at higher belt speeds (more than 10 m s^{-1}), its effect was considerable and thus should be taken into account (Khurmi and Gupta, 2005). The calculated belt velocity was 7.85 m s^{-1} ; so it was less than 10 m s^{-1} .

$$F_{cf} = T_c = mv^2 \tag{10}$$

$$F_{cf} = T_c = 5.5246 \times 7.85^2 = 340.43 \text{ N}$$

Where:

T_c centrifugal force, N;

m is the mass of the disc, kg;

V is the speed/velocity of the belt, m s^{-1} .

The input power from the diesel engine motor was $4.9 \text{ hp} = 3675 \text{ Watt}$ and the maximum twisting moment developed on the shaft was determined from the input power as Equation 11 (Bashir et al., 2013).

$$P_{in} = \frac{2\pi N_1 T}{60} \tag{11}$$

$$P_{in} = \frac{2\pi \times 3000 \times 10.7}{60} = 3675 \text{ W or } 4.9 \text{ hp}$$

Where:

P_{in} is input power from motor, W;

N_1 is the angular speed of the motor shaft, rpm

T is the twisting moment of the motor shaft, N m.

The output power transmitted through the belt T_1 and T_2 of the harvester machine was determined as Equation 12 (Bashir et al., 2013).

$$P_d = \frac{2\pi N_2 T}{60} \tag{12}$$

Where:

P_d = output power required to cut the sugarcane, W;

T = twisting moment developed on the horizontal shaft, N m;

N_2 = angular speed of the horizontal shaft, 1085 rpm.

$$P_d = \frac{2\pi N_2 T}{60} = 1215 \text{ W}$$

The belt efficiency to transmit power from the power source to the horizontal rotating shaft was determined as Equation 13 (Bashir et al., 2013).

$$\text{Power transmitted over the belt (\%)} = \frac{\text{Output power}}{\text{input power}} \times 100 \tag{13}$$

$$\text{Power transmitted over the belt (\%)} = \frac{1215}{3675} \times 100 = 33.05\%.$$

This value shows the amount of power transmitted over the belt from available power.

The total driving force of the harvester was determined by the Equations 14-16 (Khurmi and Gupta, 2005).

$$F = T_1 - T_2 \tag{14}$$

$$T_1 = 3 \times T_c \tag{15}$$

$$F = 3 \times T_c - T_2 \tag{16}$$

Where:

F is the force required to drive the machine (N);

T_1 is tension on the right side of the belt (N);

T_2 is Tension on the slack side of the belt (N);

T_c is centrifugal force (N) and T_1 is $3 \times T_c$.

However, the value of the driving force was determined from the output force as shown in Equation 17 (Bashir et al., 2013).

$$P = Fv \tag{17}$$

Where:

p = output (transmitted) power through belt T_1

and T_2, W ;

F = force required to drive, N ;

v = velocity of the belt, ms^{-1} .

$$F = \frac{P}{v} = \frac{1215}{7.85} = 154.76 \text{ N.}$$

$$T_c = mv^2 = N \tag{18}$$

$$F = (3T_c - T_2) ;$$

$$T_2 = 3T_c - F = (3 \times 340.43 \text{ N}) - 154.76 \text{ N} = 866.53$$

N ;

$$T_1 = 3T_c \tag{19};$$

$$T_1 = 1021.29 \text{ N.}$$

Power transmitted per belt:

$$P = (T_1 - T_2) \times V \tag{20} = (1021.29 \text{ N} - 866.53 \text{ N}) \times$$

$$7.85 \text{ m s}^{-1} = 1214.86 \text{ W}$$

The number of belts required =

$$\frac{\text{Total power transmitted (kW)}}{\text{Power transmitted per belt (kW)}} \tag{21}$$

$$\text{The number of belts required} = \frac{1214.9}{1214.86} = 1$$

Hence the 1 belt was required to transfer the power during operation.

2.7 Shaft Selection

The failure of the shaft was likely to be in shear stress and the design considered this fact. Loading, the diameter was obtained using the ASME code Equation 22 (Khurmi and Gupta, 2005).

$$d^3 = \frac{16}{S_s} \sqrt{(K_b \times M_b)^2 + (K_t \times M_t)^2} \tag{22}$$

Where: d is the diameter of the shaft, mm; M_t is the torsion moment, N m; M_b is the bending moment, N m, K_b is the combined shock and fatigue factor applied to the bend moment, K_t is the combined shock and fatigue factor applied to the torsion moment and S_s is allowable stress; $MN m^{-2}$.

Table 4 SME code for shaft design

Type of load	Stationery shaft		Rotating shaft	
	K_b	K_t	K_b	K_t
Gradually applied load	1	1	1.5	1
Suddenly applied load, minor shock	1.5-2	1.5-2	1.5-2	1-1.5
Suddenly applied load, heavy shock	-	-	2-3	1.5-3

Source: Khurmi and Gupta (2005).

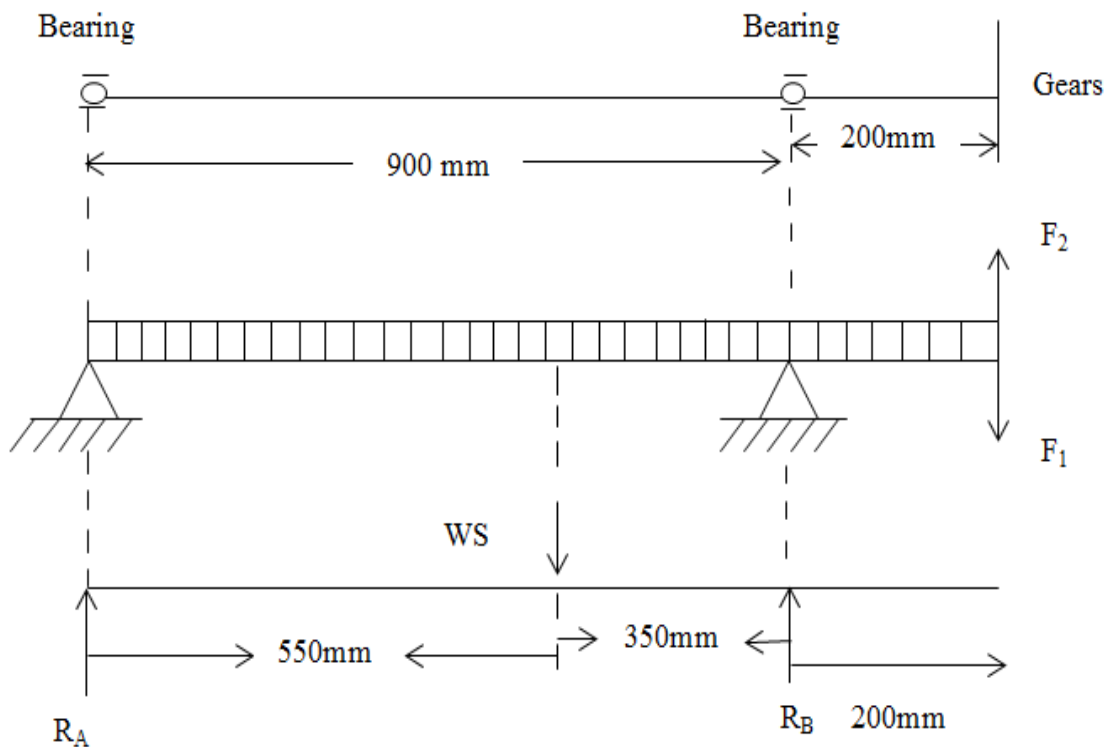


Figure 6 Vertical force distribution on the shaft

Note: F_1 = forces on the bottom disc and cutter, N and F_2 = forces on the top disc and cutter, N.

Vertical forces distribution on the shaft: The vertical load diagram on the driven pulley shaft is shown below. The total weight on the horizontal shaft was calculated by summing all the weight that was imposed on it. The weight on the shaft was a pulley, bottom cutter, bottom disc, top cutter, top disc, gears, and vertical shaft weights.

The forces distributions at two force support R_A and R_B were not equal which means 1/3 for R_A and 2/3 for R_B because the forces at the R_B were greater and 2/3 length of the shaft was supported by pivot R_B . The total weight on the shaft, W_S is 185 N.

The weight for each support pivot was described as follows.

W_A = weight of the pulley + half of the horizontal shaft;

$$W_A = 66.27 \text{ kg m s}^{-2} = 66.27 \text{ N};$$

W_B = half weight of the horizontal shaft + vertical shaft weight + gears weight + top cutter weight + top disc + bottom disc + bottom cutter;

$$W_B = 118.73 \text{ kg m s}^{-2} = 118.73 \text{ N}.$$

To calculate unknown variables R_A and R_B let me calculate the moment at each point by using equilibrium equations. The clockwise direction was negative and the counter-clockwise direction was positive.

$$\begin{aligned} \sum MA &= 0 & (23) \\ (-W_S \times 0.55\text{m}) - (F_1 \times 1.1\text{m}) + (R_B \times 0.9) + (F_2 \times 1.1\text{m}) &= 0 \\ (-185 \text{ N} \times 0.55\text{m}) - (28.56\text{N} \times 1.1\text{m}) + (R_B \times 0.9) &+ (54.17\text{N} \times 1.1\text{m}) = 0 \\ -115.5\text{Nm} - 31.416\text{Nm} + 0.9 R_B\text{m} + 59.587\text{Nm} &= 0 \\ R_B &= 97 \text{ N} \end{aligned}$$

$$\begin{aligned} \sum MB &= 0 & (24) \\ (-R_A \times 0.9\text{m}) + (W_S \times 0.35) + (F_2 \times 0.2) - (F_1 \times 0.2\text{m}) &= 0 \\ (-0.9R_A) + (185\text{N} \times 0.35\text{m}) + (54.17\text{N} \times 0.2\text{m}) &- (28.56\text{N} \times 0.2\text{m}) = 0 \\ -0.9R_A + 73.5\text{Nm} + 10.834\text{Nm} - 5.712\text{Nm} &= 0 \\ R_A &= 88 \text{ N} \end{aligned}$$

2.8 Bevel gear design

A bevel gear is shaped like a section of a cone and is primarily used to transfer power between intersecting shafts at right angles. The teeth of a bevel gear may be straight or spiral.

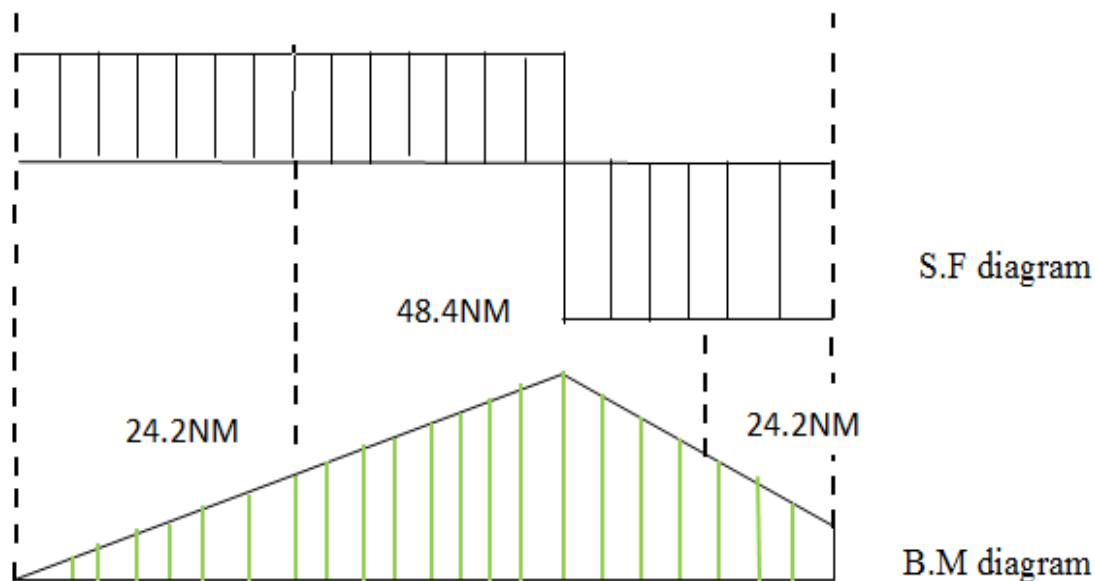


Figure 7 Shear force and bending moment diagram

Table 5 The minimum number of teeth to prevent undercut

Pressure angle	Combination of number of teeth					$\frac{z_1}{z_2}$
14.5°	29/over 29	28/over 29	27/over 31	26/over 35	25/over 40	24/over 54
20°	16/over 16	15/over 17	14/over 20	13/over 30	-	-
25°	13/over 13					

Source: Element of metric gear technology, Kohara Gear Company of Japan

The 4.9 HP engine running at 3000 rpm was used as the prime mover. The horizontal and vertical shaft has 1085 rpm.

Where: Z_1 = bevel gear, driver pinion =17; Z_2 = driven gear, driven pinion = 17; N_1 = speed at horizontal shaft (1085), rpm; N_2 = speed at vertical shaft and cutter (1085), rpm.

The speed transmitted to N_2 was calculated by the following relation

$$\frac{Z_1}{Z_2} = \frac{N_2}{N_1} \tag{25}$$

Hence the N_2 has 1085 rpm and the speed of the cutter which was connected to the vertical shaft would be 1085 rpm.

The velocity ratio or input gear ratio

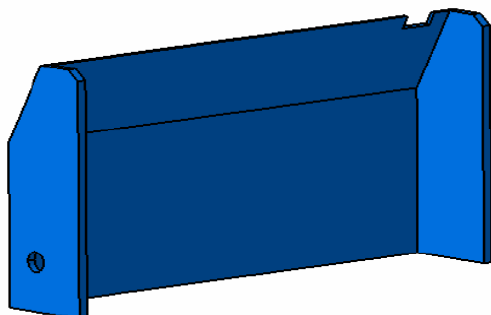
$$I = \frac{Z_2}{Z_1} = \frac{17}{17} = 1 \tag{26}$$

The module for my design was 3 because of the pitch diameter of the gear and the number of teeth. The module is unit less to be determined from the AGMA standard table for tooth form. The module m is used as metric standard input and relates the number of teeth with pitch diameter for pinion and gear by the following formula. Bevel gear teeth were located along with the elements of the cone and dependent on the ratio of several teeth in the two mating gears (Mott, 2004).

The following table indicates the parameters used to design the bevel gears. Each parameter was considered by using gear design specifications.

Table 6 Gear design elements and values

Parameter	Values
Number of teeth	$Z_1 = 17$
	$Z_2 = 17$
Module (m)	3
Pitch diameter d_1 and d_2	$d_1 = m \times Z_1 = 51$ $d_2 = m \times Z_2 = 51$
Pitch cone angle for pinion and gear [degree]	$\delta_1 = \tan^{-1} \left(\frac{\sin \Sigma}{Z_1 \cos \alpha} \right) = 45^\circ$ $\delta_2 = \Sigma - \delta_1 = 45^\circ$
Outer cone distance (R_c) [mm]	$\frac{D_2}{\sin \delta_2} = 36.06$
Face width or tooth flank width, b [mm]	$\frac{R_c}{2}$ or 10m = 12.02
Mean cone distance (R_m) [mm]	$R_m = R_c - 0.5 b = 30.05$
Addendum (h_a) [mm]	$h_a = 1.00m = 3$
Dedendum (h_f) [mm]	$h_f = 1.188m = 3.564$ mm
Working depth (h_w) [mm]	$2m = 6$
Whole depth (h_t) [mm]	$h_t = 2.188m + 0.05 = 6.614$
Clearance (c) [mm]	$C = 0.188m + 0.05 = 0.614$
Dedendum angle (θ_f) [degree]	$\theta_f = \tan^{-1} \left(\frac{h_f}{R_m} \right) = 5.64^\circ$
Addendum angle (θ_a) [degree]	$\theta_a = \tan^{-1} \left(\frac{h_a}{R_m} \right) = 4.755^\circ$
Outer cone angle for pinion and gear (δa_1 and δa_2) [degree]	$\delta a_1 = \delta_1 + \theta_a = 49.755^\circ$ $\delta a_2 = \delta_2 + \theta_a = 49.755^\circ$
Root cone angle for pinion and gear (δf_1 and δf_2) [degree]	$\delta f_1 = \delta_1 - \theta_f = 39.36^\circ$ $\delta f_2 = \delta_2 - \theta_f = 39.36^\circ$
Outside diameter for pinion and gear ($d a_1$ and $d a_2$) [mm]	$d a_1 = d_1 + 2h_a \cos \delta_1 = 55.24$ $d a_2 = d_2 + 2h_a \cos \delta_2 = 55.24$



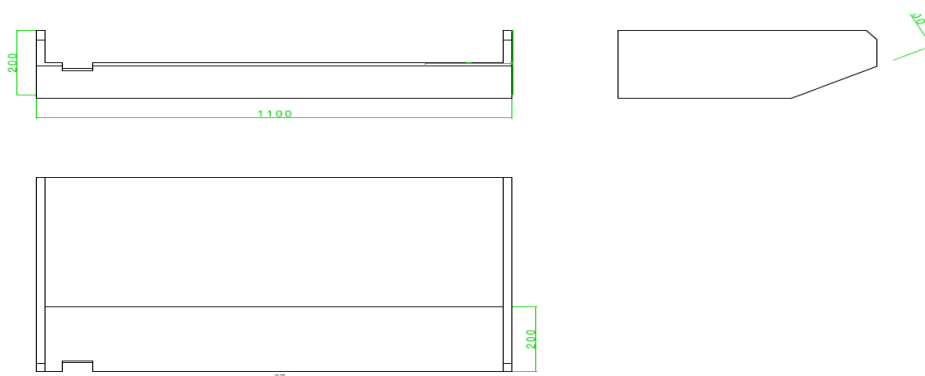


Figure 8 Safety frame

2.8 Design of safety frame

To prevent the possibility of shaft damage, and contact with other external materials most probably the sugarcane while harvesting or operating the machine, the safety frame was designed to be horizontal shaft length.

2.9 Design of wheels

Wheels are the circular block of hard metal which was durable and can carry the load that was imposed over it. It has a circular hole that was bored in which the shaft rotates when a force was applied by the operators to the wheels about its axis. Metal welded

wheels were moved faster than other types of material due to their weight and material properties (Rajya et al., 2019). The ground wheel diameter varies from 200 to 600 mm; based on which the suitability diameter was selected. The two back wheels were 600 mm in diameter, 12 mm in width, and 4 mm in thickness and the shaft was 820 mm in length with a diameter of 50 mm. The support wheel was constructed for the case of balancing and minimizing the direct contact of the blade with the ground; the gap between the wheel and the cutter was 150 mm.

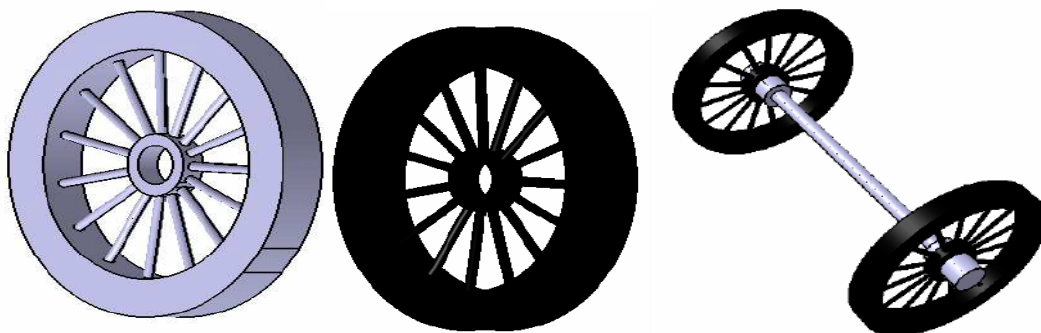


Figure 9 Back wheels

In front of the machine, there was a small wheel constructed in front of the right side of the wheel to support the load and balancing system. It was 250 mm in diameter with a 4 mm thickness.

2.10 Design of handle

During handle design, there were different parameters; size, shape, surface, security, stiffness, sitting, surroundings, the skill of the operator, and sensing features should be considered (Patkin, 2001). The other factor was the height of the handle hold; which would not be the same for all people. A height that is appropriate for a male may not be a comfort for a female. In this case, the operator of each may bend or stoop and the female would try to reach up above her limited height. Handhold height system that offers continuous vertical handles that can be grasped

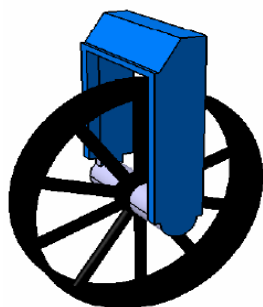


Figure 10 Front wheel

anywhere along their length or a series of handholds at different heights. Seven percent of low back injuries were associated with slipping/tripping accidents during pulling or pushing operations (Snook, 1978). For maximum power in an adult male, it should be 3 or 4 cm. in diameter (Yadav and Pund, 2007).

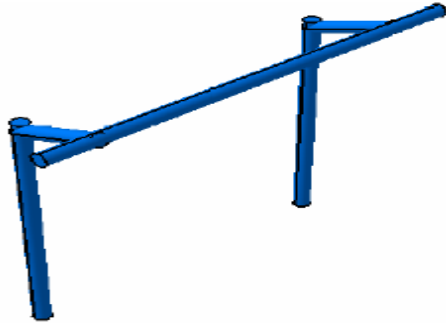


Figure 11 Handle

2.11 Weight Determination of Components of the harvester

To estimate the loads on every component of the prototype harvester, it was necessary to estimate the weight of all parts. The frame, discs, cutter, shaft, wheel, and safety frame of the harvester were calculated by using appropriate equations.

2.11.1 Weight of the frame

The high-thickness sheet metal was used in the construction of the harvester frame. The weight of the frame was calculated as follows.

$$A_{FM} = L_{FM} \times \text{Width of the frame} \quad (26)$$

The total length of the frame was

$$L_{FM} = (2 \times 800) + (2 \times 600) = 2800 \text{ mm}$$

$$\text{Area} = 100 \text{ mm} \times 6 \text{ mm} = 600 \text{ mm}^2$$

The volume of frame material can be computed using Equation 27 (ITSI-SU, 2011);

$$V_{FM} = \text{Area} \times L_{FM} \quad (27)$$

$$V_{FM} = 600 \text{ mm}^2 \times 2800 \text{ mm} = 1680,000 \text{ mm}^3 = 0.00168 \text{ m}^3$$

The mass of the frame was computed using the following Equation 28 (ITSI-SU, 2011);

$$M_{FM} = V_{FM} \times \rho_{FM} \quad (28)$$

$$M_{FM} = 0.00168 \text{ m}^3 \times 7860 \text{ kgm}^{-3} = 13.2 \text{ kg}$$

The weight of the frame material can be computed using Equation 29 (Gat, 1988).

$$W_{FM} = M_{FM} \times g \quad (29)$$

$$W_{FM} = 13.2 \text{ kg} \times 9.81 \text{ m s}^{-2} = 129.49 \text{ N}$$

Where:

L_{FM} = Length of the frame material, m;

V_{FM} = Volume of the frame material, m^3 ;

ρ_{FM} = density of the frame material, kgm^{-3} ;

M_{FM} = Mass of the frame material, kg and W_{FM} = Weight of the frame material, N.

2.11.2 Weight of the cutters for both top and bottom

The weight of the cutter was calculated by considering two parameters which were rectangular and triangular shape.

Weight of the bottom cutter can be computed using Equation 30 (Gat, 1988).

$$W_{FM} = V_{FM} \times g \times V_{BC} \quad (30)$$

The volume of the bottom cutter was

$$V_{BC} = (6 \times 90 \text{ mm} \times 20 \text{ mm} \times 3 \text{ mm}) + (6 \times \frac{1}{2} \times 90 \text{ mm} \times 20 \text{ mm} \times 3 \text{ mm})$$

$$V_{BC} = 32,400 \text{ mm}^3 + 16,200 \text{ mm}^3 = 48,600 \text{ mm}^3 = 0.000486 \text{ m}^3$$

$$W_{BC} = 0.000486 \text{ m}^3 \times 7860 \text{ kgm}^{-3} \times 9.81 \text{ ms}^{-1} = 37.47 \text{ N}$$

The weight of top cutter can be computed using the following Equation 31 (Gat, 1988).

$$W_{TC} = M_{FM} \times g \times V_{TC} \quad (31)$$

The volume top cutter was calculated as

$$V_{TC} = (4 \times 120 \text{ mm} \times 10 \text{ mm} \times 3 \text{ mm}) + (4 \times \frac{1}{2} \times 120 \text{ mm} \times 10 \text{ mm} \times 3 \text{ mm})$$

$$V_{TC} = 21,600 \text{ mm}^3 = 0.000216 \text{ m}^3$$

$$W_{TC} = 0.000216 \text{ m}^3 \times 7860 \text{ kgm}^{-3} \times 9.81 \text{ ms}^{-1} = 16.65 \text{ N}$$

Where:

W_{BC} = weight of the bottom cutter, N;

M_{BC} = Mass of the bottom cutter, kg;

V_{BC} = volume of the bottom cutter, m^3 ;

ρ_{BC} = density of the disc material for the bottom cutter, kg m^{-3} ;

W_{TC} = weight of the top cutter, N;

M_{TC} = Mass of the top cutter, kg;

V_{TC} = volume of the top cutter, m^3

ρ_{TC} = density of the disc material for the top cutter, kg m^{-3} .

2.11.3 The weight of the top and bottom disc

The bottom disc area was determined by using

Equation 32 (Math.com, 2012).

$$A_{BD} = \pi r^2 \quad (32)$$

$$A_{BD} = 3.14 \times (150 \text{ mm})^2 = 70,650 \text{ mm}^2$$

The volume of the disc was also calculated as (ITSI-SU, 2011).

$$V_{BD} = A_{BD} \times \text{Thickness of the disc} \quad (33)$$

$$V_{BD} = 70,650 \text{ mm}^2 \times 6 \text{ mm} = 423,900 \text{ mm}^3 = 0.000423 \text{ m}^3$$

The weight of the disc was calculated by using Equation 34 (Gat, 1988).

$$W_{BD} = M_{BD} \times g \quad (34)$$

The Mass of the disc was determined by the following equation (ITSI-SU, 2011).

$$M_{BD} = V_{BD} \times 7860 \frac{\text{kg}}{\text{m}^3} = 0.000423 \text{ m}^3 \times 7860 = 3.32$$

kg.

$$W_{BD} = 3.32 \text{ kg} \times 9.81 \frac{\text{m}}{\text{s}^2} = 32.56 \text{ N}$$

The top disc area was by using Equation 35 (Math.com, 2012).

$$A_{TD} = \pi r^2 \quad (35)$$

$$A_{TD} = 3.14 \times (150 \text{ mm})^2 = 70,650 \text{ mm}^2$$

The volume of the disc was also calculated as Equation 36 (ITSI-SU, 2011).

$$V_{TD} = A_{TD} \times \text{Thickness of the disc} \quad (36)$$

$$V_{TD} = 70,650 \text{ mm}^2 \times 3 \text{ mm} = 211,950 \text{ mm}^3 = 0.000211 \text{ m}^3$$

The Mass of the disc was determined by the Equation 37 (ITSI-SU, 2011).

$$M_{TD} = V_{TD} \times 7860 \frac{\text{kg}}{\text{m}^3} \quad (37)$$

$$M_{TD} = 0.000211 \text{ m}^3 \times 7860 = 1.65 \text{ kg.}$$

The weight of the disc was calculated by using Equation 38 (Gat, 1988).

$$W_{TD} = M_{TD} \times g \quad (38)$$

$$W_{TD} = 1.65 \text{ kg} \times 9.81 \frac{\text{m}}{\text{s}^2} = 16.18 \text{ N}$$

Where: A_{BD} = Area of the bottom disc, m^2 ; V_{BD} = volume of the bottom disc, m^3 ; M_{BD} = Mass of the bottom disc, kg; W_{BD} = weight of the bottom disc, N; A_{TD} = Area of the top disc, m^2 ; V_{TD} = Volume of the top disc, m^3 ; M_{TD} = Mass of the top disc, kg; W_{TD} = Weight of the top disc, kg.

2.11.4 Weight of safety frame

The safety frame was also classified to two shapes

which was rectangular and triangular shape to determine the weight.

$$V_{SF} = A_{SF} \times \text{thickness of safety frame} \quad (39)$$

$$A_{SF} = (2 \times 250 \text{ mm} \times 200 \text{ mm}) + (2 \times 130 \text{ mm} \times 200 \text{ mm}) + (70 \times 200) + 1100 \text{ mm} \times 450 \text{ mm} = 661,000 \text{ mm}^2$$

$$V_{SF} = 661000 \text{ m}^2 \times 2 \text{ mm} = 1,322,000 \text{ mm}^3 = 0.001322 \text{ m}^3$$

The Mass of the safety frame was determined by the Equation 40 (ITSI-SU, 2011).

$$M_{SF} = V_{SF} \times 7860 \frac{\text{kg}}{\text{m}^3} \quad (40)$$

$$M_{SF} = 0.001322 \text{ m}^3 \times 7860 = 10.39 \text{ kg.}$$

The weight of the disc was calculated by using Equation 41 (Gat, 1988).

$$W_{SF} = M_{SF} \times g \quad (41)$$

$$W_{SF} = 10.39 \text{ kg} \times 9.81 \frac{\text{m}}{\text{s}^2} = 101.92 \text{ N}$$

Where: A_{SF} = area of the safety frame, m^2 ; V_{SF} = volume of the safety frame, m^3 ; M_{SF} = mass of the safety frame, kg; W_{SF} = weight of the safety frame, N.

2.11.5 Weight of motor cover

The motor covering shape was rectangular-shaped sheet metal. The area of the metering shaft was estimated as cited by Math.com (2012).

The mass of the motor cover was estimated using Equation 42 (ITSI-SU, 2011).

$$M_{MC} = V_{MC} \times \rho_{MC} \quad (42)$$

$$V_{MC} = ((400 \text{ mm} \times 500 \text{ mm} \times 2 \text{ mm}) \times 2) + (400 \times 400 \times 2 \text{ mm}) = 0.00072 \text{ m}^3$$

$$M_{MC} = 0.00072 \text{ m}^3 \times 7860 \frac{\text{kg}}{\text{m}^3} = 5.65 \text{ kg}$$

The weight of the motor cover was calculated using the Equation 43 (Gat, 1988).

$$W_{MC} = M_{MC} \times g \quad (43)$$

$$= 5.65 \text{ kg} \times 9.81 \frac{\text{m}}{\text{s}^2} = 55.42 \text{ N}$$

Where: V_{MC} = Volume of the motor cover; M_{MC} = Mass of the motor cover; W_{MC} = Weight of the motor cover.

2.11.6 Weight of the shaft

The shaft used in this project was three; horizontal shaft, and vertical shaft. So, the area, mass, volume, and weight were calculated. Hence, the volume of the horizontal shaft was estimated using Equation 44

(ITSI-SU, 2011).

$$V_{sh} = A_{sh} \times L_{sh} \quad (44)$$

The area of the horizontal shaft was estimated as cited by Math.com (2012).

$$A_{sh} = \frac{\pi D^2}{4} = \frac{3.14(50 \text{ mm})^2}{4} = 1,962 \text{ mm}^2 = 0.001962 \text{ m}^2;$$

$$V_{sh} = 0.001962 \text{ m}^2 \times 1.1 \text{ m} = 0.00215 \text{ m}^3;$$

The mass of the horizontal shaft was estimated using Equation 45 (ITSI-SU, 2011).

$$M_{sh} = V_{sh} \times \rho_{sh} \quad (45)$$

$$M_{sh} = 0.00215 \text{ m}^3 \times 7860 \frac{\text{kg}}{\text{m}^3} = 16.89 \text{ kg}$$

The weight of the horizontal shaft was calculated using the Equation 46 (Gat, 1988).

$$W_{sh} = M_{sh} \times g \quad (46)$$

$$W_{sh} = 16.89 \text{ kg} \times 9.81 \frac{\text{m}}{\text{s}^2} = 165.69 \text{ N}$$

Area of the vertical shaft was also calculated as cited by Math.Com (2012).

$$A_{sh} = \frac{\pi D^2}{4} \quad (47)$$

$$A_{sh} = \frac{3.14(30 \text{ mm})^2}{4} = 706.5 \text{ mm}^2 = 0.0007065 \text{ m}^2$$

The volume of the vertical shaft was estimated using Equation 48 (ITSI-SU, 2011).

$$V_{sh} = A_{sh} \times L_{sh} \quad (48)$$

$$V_{sh} = 0.0007065 \text{ m}^2 \times 1.45 \text{ m} = 0.00102 \text{ m}^3$$

The mass of the vertical shaft was estimated using Equation 49 (ITSI-SU, 2011).

$$M_{sh} = V_{sh} \times \rho_{sh} \quad (49)$$

$$M_{sh} = 0.00102 \text{ m}^3 \times 7860 \frac{\text{kg}}{\text{m}^3} = 8.01 \text{ kg}$$

The weight of the vertical shaft was calculated as using the equation of Gat (1988).

$$W_{sh} = M_{sh} \times g \quad (50)$$

$$W_{sh} = 8.01 \text{ kg} \times 9.81 \frac{\text{m}}{\text{s}^2} = 78.57 \text{ N}$$

Where: A_{SH} = Area of the shaft, m^2 ;

V_{SH} = volume of the shaft, m^3 ;

M_{SH} = Mass of the shaft, kg;

W_{SH} = weight of the shaft, N.

2.11.7 Wight of the pulley

The area of the pulley was calculated by considering the shape and dimension during design.

The volume of the pulley on the horizontal shaft was calculated using Equation 51 (ITSI-SU, 2011).

$$V_P = ((\pi r_1 - \pi r_2) \times t) - \left(\frac{1}{2} a \times b \times h\right) + (a \times b \times h) \quad (51)$$

$$V_P = (((3.14 \times 69^2) - (3.14 \times 25^2)) \times 50 \text{ mm}) - \left(\frac{1}{2} \times 10 \text{ mm} \times 40 \text{ mm} \times 138 \text{ mm}\right) + (10 \text{ mm} \times 40 \text{ mm} \times 138 \text{ mm})$$

$$V_P = 649,350 \text{ mm}^3 - 110400 \text{ mm}^3 = 0.000538 \text{ m}^3$$

The mass of the pulley was estimated using Equation 52 (ITSI-SU, 2011).

$$M_P = V_P \times \rho_P \quad (52)$$

$$M_P = 0.000538 \text{ m}^3 \times 7860 \frac{\text{kg}}{\text{m}^3} = 4.22 \text{ kg}$$

The weight of the pulley was calculated using the equation of Gat (1988).

$$W_P = M_{sh} \times g \quad (53)$$

$$W_P = 4.22 \text{ kg} \times 9.81 \frac{\text{m}}{\text{s}^2} = 41.39 \text{ N}$$

Where: V_P = Volume of the pulley, m^3 ; M_P = Mass of the pulley, kg; ρ_P = Density of the pulley material, kg m^{-3} ; W_P = Weight of the pulley, N.

2.11.8 Weight of the handle

The area of the handle was estimated as Math.com (2012) equation.

$$A_H = \frac{\pi D^2 - d^2}{4} \quad (54)$$

$$A_H = \frac{\pi(25^2 - 20^2)}{4} = 176.625 \text{ mm}^2 = 0.000176 \text{ m}^2$$

The volume of the handle was estimated using Equation 55 (ITSI-SU, 2011).

$$V_H = A_H \times L_H \quad (55)$$

$$V_H = 0.000176 \text{ m}^2 \times (0.5 \times 2 \text{ m} + 0.8) = 0.000316 \text{ m}^3$$

The mass of the handle was estimated using the following equation of ITSI-SU (2011).

$$M_H = V_H \times \rho_H \quad (56)$$

$$M_H = 0.000316 \text{ m}^3 \times 7860 \frac{\text{kg}}{\text{m}^3} = 2.48 \text{ kg}$$

The weight of the handle was calculated using the equation of Gat (1988).

$$W_H = M_{sh} \times g \quad (57)$$

$$W_H = 2.48 \text{ kg} \times 9.81 \frac{\text{m}}{\text{s}^2} = 24.32 \text{ N}$$

Where: A_H = Area of the handle, m^2 ; V_H = Volume of the hand, m^3 ; M_H = Mass of the handle, kg; W_H = Weight of the handle, N.

2.11.9 Total weight of the sugar cane harvesting

The total weight of the prototype harvester including the weights of the main frame, safety frame,

pulley, horizontal shaft, vertical shaft, bottom, and top disc, bottom and top cutter, motor cover bevel gears and gear box was estimated to be 699.66 N. Taking 2% margins for the weights of

Welding bolts, nuts, belts, etc. the total weight was found to be 713.53 N.

2.12 Estimation of Force Required Driving the Harvester

A rolling resistance force (F), which is assumed to act horizontally at the wheel and ground interface or ground and wheel contact patch, was estimated using Equation 58 (Reece, 2002).

$$F = (CR + i) \times N \quad (58)$$

$$F = \left(\left(\frac{z}{d} \right)^{0.5} + i \right) \times N = \left(\left(\frac{3}{60} \right)^{0.5} + 0.05 \right) \times 356.82 \\ = 97.62 \text{ N}$$

Where:

CR = coefficient of rolling resistance;

d = wheel diameter, m, 0.6 m;

z = maximum wheel sinkage depth, m (on a soft surface $z = 0.05d$, let $z \approx 0.03\text{m}$);

N = weight of the planter on each wheel, N, $(713.53\text{N} / 2) = 356.76\text{N}$;

i = gradient of the ground, let $i = 5\%$.

2.13 Torque on the ground wheel

The torque on the ground wheel was calculated by

using the following formula (Sharma and Mukesh, 2010).

$$T = F \left(\frac{d}{2} \right) \quad (59)$$

$$T = 97.62 \text{ N} \left(\frac{0.6}{2} \right);$$

$$T = 29.28 \text{ N m.}$$

2.14 Fabrication of the machine

The software drawing was developed, and fabricated, and the five horsepower motors were assembled in the proper position over the frame. The different component of the harvesting machine parts mentioned in the previously described subheadings was fabricated step by step by considering their position and design and assembled in the workshop of the Wonji Shoa sugar factory which is located at latitude $8^{\circ}22'30''$ to $8^{\circ}25'0''\text{N}$ and longitude $39^{\circ}12'30''$ to $39^{\circ}17'30''\text{E}$. The bevel gear, horizontal and vertical shaft, blade and cutter, frame, safety frame, and handle were fabricated from selected materials with their operational property in the sugarcane factory. The entire supporting frame and safety frame were marked and cut, then joined by welding, and screwing according to functional requirements. The cutter and disc were selected and cut with the machine and then welded with their appropriate position.



Figure 12 Machine model and construction of machine in the shop

3 Discussion

The sugarcane harvesting machine was designed, developed, and fabricated. The designing system was performed by using CATIA-V5 software. After designing each part, the assembling and construction system was preceded. actual in field machine

performance such effective field capacity, field efficiency, harvesting cost, ergonomic parameters for the operators were evaluated. The effective field capacity of the machine at 30 minutes and 180 minutes are 0.0048 and $0.0028 \text{ ha hr}^{-1}$ respectively. The machine is efficient in operation and reduces workers

load and increases efficiency of workers by 44% than manual harvesting and total field efficiency of the machine is 86%. Harvesting costs were decreased by 57% compare to manual harvesting and ergonomically fit at average weight and height of workman.

4 Conclusion

A diesel engine-driven sugarcane harvester has been designed and fabricated with Ethiopian available materials to match the need and alleviate the difficulties during harvesting the sugarcane manually, and easily purchased by small and medium-scale farmers. It is simple and does not require any special skill to operate. With proper attention to its maintenance which may include; regular lubrication of moving parts, replacement of worn-out/ damaged parts, cleaning, and proper storage after use, this equipment can serves and adequately alleviate the sugar factories and farmers' difficulties in sugarcane harvesting. The motor driven sugarcane harvesting machine works with better efficiency in the field as compared with manual harvesting activities and the height of the cut above the ground was good. Most importantly, the prototype harvester can be efficiently, effectively, and economically used by the majority of Ethiopian sugar factories and smallholder farmers that produce sugarcane. Finally, it's recommended that the developed machine should be modified so as to make the machine suitable for all weather condition and ergonomically suitable for operators of all sex and can also be redesigned for harvesting maize and other related crops which was similar with that of sugarcane.

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Conflict of Interest

There is no computing interest in this work that could have appeared to influence the work reported in this paper.

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