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Design and Fabrication of Teff Threshing Machine

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Abstract: Teff has the largest value in terms of both production and consumption in Ethiopia and the value of the commercial surplus of teff is second next to coffee. The traditional way of threshing and winnowing, leads to contamination of grains with foreign matter (pebbles, dirt, and cow dung) and loss of grains due to the foreign matter dropping in to grains and the wind blowing grains away with chaff. This design helps to find out the typical dimensions of various components of the threshing machine with great accuracy in small time and also gives fine representation of engine drive teff threshing machine. The machine was fabricated by using the low cost material available in local market. Main components of this machine are: frame, sieve, and threshing unit, concave, blower, feeding table, sucker, and grain delivery chute and engine seat. A teff threshing machine was designed, constructed and evaluated for its performance. Mild steel was used majorly for the fabrication of component parts for ease of machining, assembling, maintenance and affordability. The machine was evaluated on farm observed to acceptable threshing capacity and efficiency and the operation cost and quality loss due to traditional threshing were minimized by far.

Keywords: Teff, Threshing Machine, Design, Power Transmission and Blower

1. Introduction

Teff [*Eragrostis teff* (Zucc.)] is grain like millet, with dimensions ranging from 0.9 to 1.7 mm and 0.7 to 1 mm length and width respectively [21]. In Ethiopia next to coffee teff has shared second place in terms of production and consumption and valued commercial surplus commodity crop. Teff is noticed to have a sound amount of amino acid composition and lysine levels higher as compared to wheat and barley, and also consists of considerable amount calcium, phosphorous, iron, copper, aluminium, barium, and thiamine [15].

The Central Statistics Authority report [9] even if the yield of teff is lower as compared with other cereal crops, teff shares for about 16% coverage of the gross grain potential production of all the cereal crops cultivated in country accounts about 2.7 million hectares of farm land. Since teff is originated in Ethiopia and cultivated as a cereal crop, for

longer time and its cultivation as a cereal crops for food is not well known and recognized as food crop internationally and its threshing and harvesting process have been carried out traditionally due to lack of teff mechanization concerning teff technologies for so many years; meanwhile indigenous processing to make *budena*, pancake-like fluffy soft bread, which is a staple food for most Ethiopians.

Because of its significance in most parts Ethiopians nutrition and its rigorous production in the country, lately there is an increasing concentration on teff research to explore its potential for food and commercial advantages such as starch production [8, 5, 22].

Teff is exceedingly nutritive and is an essential chunk of Ethiopia's cultural inheritance and national identity for preparing *budena* [17]. Being categorized as one of the newest great nutrients of the 21st century, teff's worldwide popularity is quickly rising by exporting processed teff to world communities [11]. This presents an increasing

commercial opportunity for Ethiopia and particularly its farmers.

According to [5] a study of 1,800 people with celiac disease reports that regular consumption of teff significantly reduced their symptoms [6]. With an increasing number of health-conscious consumers across the world, teff has started generating a similar phenomenon with quinoa, the nutritious grains native to South America for global prominence [1].

The lesser size of teff crop incurs numerous difficulties during planting, and incidentally at the time of weeding and threshing. The quantity of teff misplaced in the harvesting and threshing progression due to its size is very high. [19, 4] Was reported that teff yields are relatively low (1.2 t/ha) and great loss rates (25-30% both before and after harvest). It diminishes the amount of grain obtainable to consumers by up to 50%. Losses during threshing may occur due to; incomplete threshing (grain remains on the straw); spillage and scattering during the process and consumption of grains by animals used for trampling purposes.

In Ethiopia, Teff research to date has concentrated typically on breeding and refining agronomic practices, Mechanization have not been widely investigated; habitually teff is threshed on level ground so-called 'Obdi' that is regularly splashed by cattle manure. Grain with straw is scattering over 'Obdi' and cattle/pack animals have been driven over to detach the grain from the straw. In other ways, threshing can be done by humans by beating the harvested teff with a stick [16].

Even though the multi crop thresher adapted and modified by different research institutions; the machine does not thresh and clean the grain as desired. These leads our farmers for extra cleaning process using winnowing machines for cleaning and invite further labour cost for out-of-date method of cleaning process of threshed teff which is made by waiting natural air, tossing the separated grain to the air to isolate the grain from the chaff and the dust [16].

The traditional way of threshing and winnowing, leads to contamination of grains with foreign matter (pebbles, dirt, and cow dung) and loss of grains due to the foreign matter dropping in to grains and the wind blowing grains away with chaff. Therefore, qualitative losses due to grain contamination with soil, animal droppings, and urine are equally as important as quantitative losses. Hence to alleviate the problem associate with tedious and time intensive process of teff threshing and winnowing a complete teff thresher involving threshing unit and winnowing assisted with aerodynamic and mechanical aspirator was designed and manufactured.

2. Materials and Methods

2.1. Design Consideration and Selection

The design of the component-units of the machine was carried out putting into consideration, the specific functions of each component units. The main units are: the threshing unit, cleaning, power requirement and the operating capacity.

2.2. Design of Threshing Machine

The threshing machine is expected to have a reasonable threshing capacity, high efficiency, and production of quality teff grain with minimum threshing and cleaning loss. Accordingly, a machine should have highly acceptable performance.

2.2.1. Frame

A frame used for supporting threshing assembly and power transmission unit. To ease operation and provide comfort to an operator, the height of the machine was decided on basis of anthropometric measurements made on the potential operators. The standard minimum ratio of top and bottom lengths of a trapezoidal frame is given as $L_1/L_2 = 0.5$. It has a trapezium shape having the size of 950 by 1100 mm, made of 40 mm by 40 mm x 4 mm angle iron. The supporting a detachable engine seat and has 680 mm width at top end. Same size of mild steel angle iron was welded across all the members of frame to make it as rigid frame. The frame was extended to accommodate the blower assembly.

2.2.2. Feeding Table

Feeding table was fastened temporary at the inlet opening of the threshing unit. It was equal in length with the threshing drum since the teff straw was stored for frequent feeding and feeding of the straw was made manually by deciding the amount of straw to be fed in to the threshing cylinder. The feeding table was made of mild steel sheet metal 1.5mm thick, has 900 mm by 800 mm dimension as shown in figure 1.
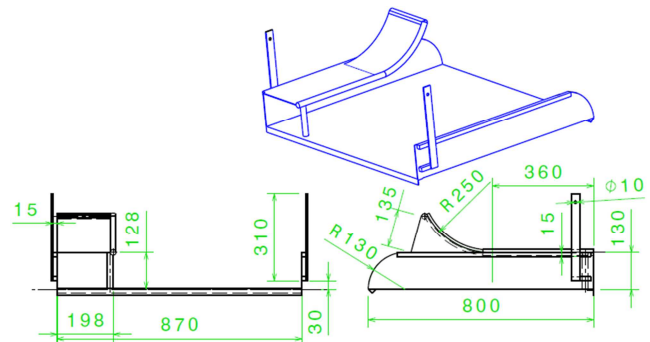

Figure 1. *View feeding table (all dimensions are in mm).*

2.2.3. Sieve Unit

Sieves were made of steel sheet metal of 1.5 mm thickness. They consist of upper, middle and lower sieves and hang on four links (2 adjustable fixed rods in the front to alter the sieve tilt angle and 2 in the rear). The upper sieve has perforated 3 mm holes diameter for separating chaff and straw from seed. The middle sieve is mounted 100 mm below and parallel to the upper screen. The middle screen has 2 mm diameter holes to remove broken straw and chaff, and impurities. The cleaning unit has one discharge chute and is oscillated by multiple linkage system. The sieve unit has a link mechanism which performs a horizontal oscillating motion as shown in figure 2 below.
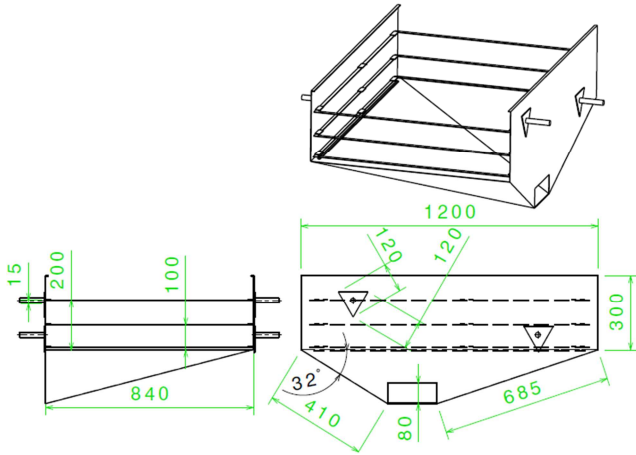
Figure 2. View of sieves and grain delivery (all dimensions are in mm).

2.2.4. Force Required for Threshing

To thresh teff crop a force 'F' is needed, which is a function of cylinder linear speed as well as the friction coefficients between the crop-crop and crop-metal.

$$F = F_C + F_r \qquad (1)$$

where:- F=the force needed for threshing crop (N); F_C=impact force of the cylinder (N); F_r=friction force (N); The impact force 'Fc' may be calculated using Eqn. (2).

$$F_C = Q(V_2 - V_1) \qquad (2)$$

where:- V_2=speed of the threshed crop as it exits the cylinder (m/s); V_1=speed of the crop as it enters the cylinder (m/s); V_2 is proportional to the linear speed of the cylinder (V); Q=feeding rate (kg/sec).

$$V_2 = aV, \quad V = \frac{R \times 2\pi \times N}{60} \qquad (3)$$

The friction force is dependent upon many factors such as the friction coefficient, type of breakage of the straw and the intense of threshing. It is however proportional to the total force necessary to thresh the crop and was calculated using Eqn. (4).

$$F_r = f \times F \qquad (4)$$

where:- R=threshing cylinder radius (m); N=cylinder (rpm); f=the coefficient f, for spike tooth type equal to 0.7 - 0.8.

2.2.5. Power Required Threshing

Required power for threshing, P_1 in watts, was obtained using Eqn. (5). For a thresher of 0.8 m having drum length, crop moisture content of 15 - 25%, feed rate of 3.5 kg/s, the coefficient a, has been determined equal to 0.70 - 0.85. The linear speed of the cylinder has also been determined equal to 15 - 37 m/s. The linear speed of the cylinder has also been determined equal to 15 - 37 m/s [13].

$$P_1 = FV = \left[\frac{Q(aV - V_1)}{1 - f} \right] V \qquad (5)$$

The feeding mechanism was manual hence the feeding speed of teff straw in to threshing unit was considered to be 12m/s since feeding speed for stationery thresher range to 15.20m/s. Therefore required power for threshing teff was calculated using Eqn. (5) and found to be 2,256.8 watts. Power is also needed to overcome the air resistance against the rotation of the cylinder and the friction force in bearings. This power was calculated using Eqn. (6)

$$P_2 = AV + BV^3 \qquad (6)$$

where:- A and B are coefficient due to friction and air resistance respectively and was determined 5 - 5.5 N and 0.045 Ns²/m² for spike tooth type. The total required power to drive the thresher unit was determined to be 3,184.22 watts using Eqn. (7) below.

$$P = P_1 + P_2 = \left[\frac{Q(aV - V_1)}{1 - f} \right] V + AV + BV^3 \qquad (7)$$

2.2.6. Power Transmission Design and Selection

The diameters of the driving and driven pulleys are unequal; the belt will slip first on the pulley having the smaller angle of lap, i.e. on the smaller pulley.

$$\text{The power transmitted} = (T_1 - T_2)v = \left[(T_1 - T_c) - (T_2 - T_C) \right] v \qquad (8)$$

But $T_2 - T_c = \dfrac{T_1 - T_c}{e^{\theta\mu}}$, hence, power, P, can be written as:

$$P = \left[(T_1 - T_2)\left(1 - \frac{1}{e^{\mu\theta}} \right) \right]$$

where:- T_1=tension on the tight side (N); T_2=tension on the slack side (N); μ=coefficient friction between pulley and belt; θ=wrap angle on the smaller pulley or driving pulley (radian); T_c=centrifugal force on the belt (N); V=peripheral velocity of the threshing drum beater (N).

The tension in the tight side of a power transmission belt

can be calculated using Eqn. (8) as suggested by [12] and was estimated to be 199.21N. The centrifugal tension of the belt (Tc) was determined using Eqn. (9) and was estimated to be 26.72N.

$$T_C = mv^2 \qquad (9)$$

The mass of the belt per unit length was calculated using Eqn. (10) and equaled to 0.09 kg/m.

$$m = \rho A \qquad (10)$$

$$A = \left(\frac{b-x}{2}\right)t + xt \qquad (11)$$

The velocity of belt passing over the follower pulley can be calculated using Eqn. (12) [12] and found to be 17.23 m/s.

$$V = \frac{\pi d_1 N_1}{60}\left(1 - \frac{S}{100}\right) \qquad (12)$$

where:- T_c and T_{max}=the centrifugal and maximum tension of the belt (N); A=cross sectional area of a belt (mm^2); m=mass per unit length of a belt (kg/m); v=speed of belt (mm/s); b=top width of the belt (mm); x=bottom width of the belt (mm); d_1=diameter of driving pulley (mm); N_1=speed of the driving pulley (rpm); S=slip (%); σ=Maximum allowable stress of belt (MPa); t=thickness of the belt (mm); ρ=density of belt material (Rubber) (kg/m^3).

Belt tension in the slunk side for v-belt was calculated using Eqn. (13) [2] and the value were equaled to be 28.99N.

$$\frac{T_1 - mv^2}{T_2 - mv^2} = e^{\frac{\mu\alpha_1}{\sin\frac{\beta}{2}}} \qquad (13)$$

where:- T_1=tension in tight side (N); T_2=tension in slunk side (N); μ =coefficient of friction between belt and pulley; α_1=angle of wrap on smaller pulley (rad); β=belt wedge angle (degree).

The coefficient of friction between pulleys and belt can be determined using Eqn. (14) [14].

$$\mu = 0.54 - \frac{0.7}{2.4 + V} \qquad (14)$$

where, V=velocity of belt (m/s)

(i). Design of Pulleys

Based on the required speed, the diameter of driving and driven pulley can be estimated based on Eqn. (15) (Aaron, 1975). Restriction of choice of belt section was raised from the view point of minimum pulley diameter. The minimum pulley diameter of 120 mm was selected for driving the driven pulley of the drum at maximum speed of 800 rpm. Therefore the diameter of the sheave on the drum was estimated using Eqn. (15) and found to be 450 mm.

$$N_1 D_1 = N_2 D_2 \qquad (15)$$

The width of the pulley (face width) 'B' is usually considered to be 25% greater than the width of belt (b) and Eqn. (16) (Richard and Nisbett, 2011) can be used to estimate the width of pulley. Since the selected V-belt as power transmission was 'B' section and its width was 17 mm, the width of the sheave was determined to be 21.25 mm using Eqn. (16).

$$B = b + 0.25b = 1.25b \qquad (16)$$

The thickness (*t*) of a pulley rim for single v-belt can be calculated using Eqn. (17) (Richard and Nisbett, 2011).

$$t = \frac{D}{200} + 3mm \qquad (17)$$

where:- N_1=maximum angular speed of driven pulley (rpm); N_2=maximum angular speed of driving pulley, (rpm); D_1, D_2=diameters of driven and driving pulley (mm); B=width of pulley (mm); b=width of belt (mm); t=thickness of the driven pulley (mm).

(ii). Estimation of Belt Length

Length of belt required to transmit power was calculated using Eqns. (18 and 19) and was determined to be 1,986.43 mm. hence from standard belt length designation 1900 mm belt 'B' was selected [12],

$$L = 2C + 1.57(D_1 + D_2) + \frac{(D_2 - D_1)^2}{4C} \qquad (18)$$

The Centre distance (C) between the two pulleys must not be less than twice the diameter of the larger pulley (Richard and Nisbett, 2011).

$$C = 2D_2 \qquad (19)$$

where: - D_1 and D_2=diameters of driving and driven pulleys respectively (mm); C=center distance (mm); L=belt length (mm).

Wrap angle on the smaller pulley is determined using Eqn. (20) [2] and was calculated as 2.51 radian.

$$\alpha_1 = 180 - 2\sin^{-1}\left(\frac{D_2 - D_1}{2C}\right) \qquad (20)$$

where: - α_1=angle of wrap on smaller pulley (degree); α_2=angle of wrap on larger pulley (degree); C=center to center distance between pulleys (mm).

According to [12] torque on a drum shaft was calculated using Eqn. (21) and found to be 35.75 Nm.

$$T = (T_1 - T_2)\frac{D_2}{2} \qquad (21)$$

where:- T_1=tension on the tight side (N); T_2=tension on the slack side (N); D_2=diameter of the largest driven pulley (m).

Using the torque applied to the drum shaft, tension forces in the tight and slunk side of the belt transmitting power from drum shaft to the blower was determined to be 372.36 N and 74.47 N respectively.

2.2.7. Drum Shaft Design

Shafts are usually subjected to torsion, bending, and axial loads. Design of shaft, from ductile materials, based on strength, is governed by the maximum shear stress theory; while shafts of brittle materials would be designed by the maximum normal stress theory. The diameter of the solid shaft shall be calculated using the equation given by the American Society of Mechanical Engineering [3] as shown below. The total distributed load rest on the shaft was

calculated by measuring the definite volume of the material used and multiplying it with known density of mild steel material. Hence the total distributed load was estimated to be 400.44N after multiplied by gravity, neglecting weight of

welding electrodes. The v-belt was laid at 65° from horizontal to connect the sheave on the engine and drum to the right and at 45° from horizontal for blower and drum pulleys as shown in figure 3.
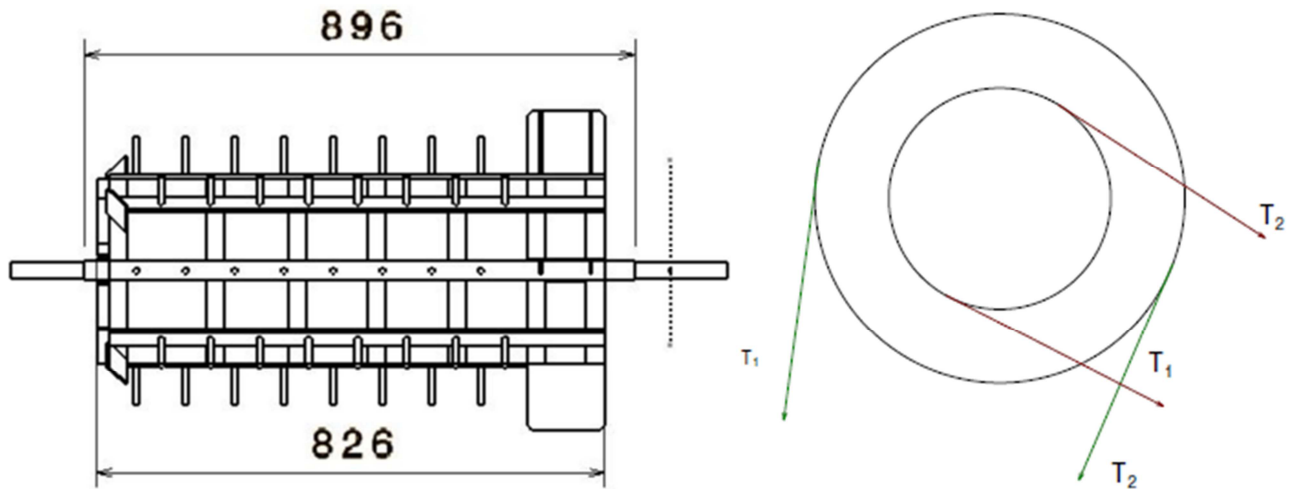

Figure 3. Layout of drum shaft (all dimensions are in mm).

The force components and reaction forces of the drum shaft were resolved as shown in figure 4 below.

Figure 4. Forces of horizontal and vertical planes (all dimensions are in mm).

The reaction forces and force components due to belt tension was extracted from figure above and indicated as shown in figure 5.
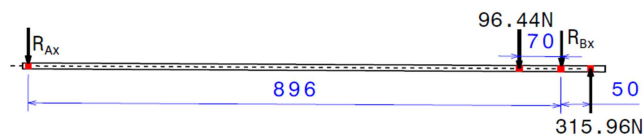

Figure 5. Free body diagram of forces on horizontal plane (all dimensions are in mm).

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

$$\sum M_B = 0$$

$$0.896m \, x R_{Ax} + 0.07m \, x 96.44N + 0.05 \times 315.96N = 0$$

$$R_{Ax} = 25.17 \text{ N up ward}$$

Taking the summation of forces in the horizontal direction the reaction force at point B was calculated to be 244.69N. Hence, shear force and bending moment were determined at critical sections as can be seen in figure 6.
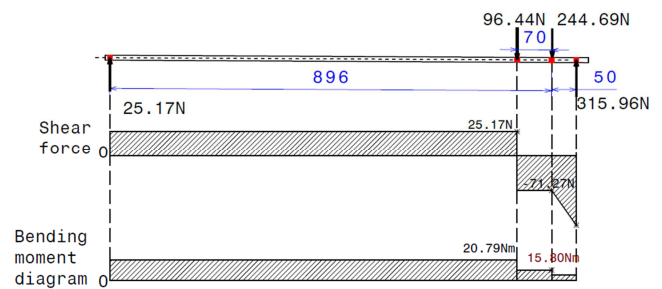

Figure 6. Shear force and bending moment diagram in horizontal plane.

The reaction and belt tension components in the vertical plane resolved and shown as indicated in figure 7 below.
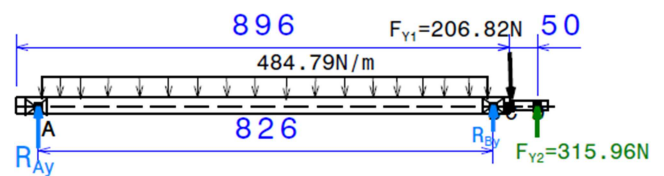

Figure 7. Free body diagram of reaction and belt tension components in vertical plane (all dimensions are in mm).

In the same fashion as in the above section, taking summation of moment at preferred point the reaction forces at point A and B shall be calculated. Hence consider the sum of moment at point B is zero.

$$\sum M_B = 0$$

$$0.826m \times R_{Ay} + 0.07m \times 206.82N$$

$$= 0.826m \times 484.79N / m \times 0.413m + 0.12m * 315.96N$$

$$R_{Ay} = 228.59N$$

Considering the sum of all forces in vertical direction to be

zero, the reaction force at bearing B was calculated to be 62.71N. Therefore by substituting the values of the computed reaction forces the shear and bending moments were determined at critical section as shown in figure 8.
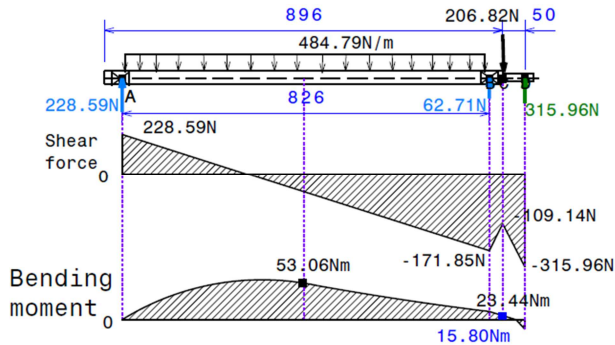

Figure 8. *Shear force and bending moment diagram in vertical plane.*

The resultant maximum bending moment on the shaft was calculated using Eqn. (22).

$$M_b = \sqrt{M_V^2 + M_h^2} \tag{22}$$

The resultant bending moment with the highest values of 53.06 Nm was occurred at the midpoint of the drum shaft due to the distributed load of the cylinder. Therefore the value was selected for design analysis purpose. The size of the shaft was determined using Eqn. (23).

$$d^3 = \frac{16}{\pi S_a} \sqrt{(C_b k_b M_b)^2 + (C_t k_t T)^2} \tag{23}$$

As recommended by Richard and Nisbett [18], K_b value of 1.20 and K_t value of 1.0 were used for sudden applied load and minor shock. The value of C_b and C_t were considered to be 2.15 and 1.98 respectively. The allowable shear stress (S_a) of 40MPa for shaft with key way was used. As a result, the minimum diameter of the shaft computed was 26.98mm. Since this value is not found in the shaft standard, the shaft size was selected to be 30mm from the catalogue.
where:- T=torsional moment of the shaft (Nm); d=diameter of the shaft (m); M_b=bending moment (Nm); S_a=shear strength (N/m^2); C_b and C_t=Stress concentration factors due to bending and torsion respectively.

2.2.8. Design of Blower

The blower was centrifugal and was constructed from a sheet metal of 1.50 mm thickness provided housing at top and bottom. It was mounted on a shaft with a diameter of 25 mm and on roller bearings was used to support. The blower assembly had four radial blades with curved periphery. The blower housing was provided with circular holes, on both side. For agricultural applications, fan speeds are recommended to be between 450 and 1000 rpm. [6] Indicated that the initial data required for the design of a fan are the mean velocity of air stream flow at its exit (Cmean), the volume flow rate (V). The flow rate of desired air stream is determined form the concentration of the material entrained by the air, μ given as ratio of *Gm to Ga* and has a value between or equal to 0.20 and 0.30. The volume flow rate of air required making separation between grains and MOG can be estimated from the equation given below.

$$V = \frac{G_a}{\rho} = \frac{G_m}{\mu \rho} \tag{24}$$

Where: - V=volume flow rate (m^3/s); Ga=the mass flow rate of air (kg/s); Gm=the quantity of material removed by the air stream per unit time (kg/s); μ=coefficient of concentration (unit less); ρ=density of air (kg/m^3).
Since the blower housing was provided with circular on its both sides the air flow rate to the blower was calculated using Eqn. (25).

$$volume \; of \; air \; in = \pi r_i^2(\omega r_i) \tag{25}$$

The blower driving pulley has diameter of 160 mm and have maximum speed the same with drum which is 800 rpm, where the driven pulley on the blower have the diameter of 250 mm; therefore the rotational speed of the blower was determined by considering pulley ratio and was found to be 512 rpm. Assuming that the fan housing has no linkage, the air flow rate at the iterance and exit was equal the radius of the opening at the air iterance was estimated using Eqn. (26).
Abduljelil Mohammed and Zewdu Abdi [1, 22] found the terminal velocity of tef grain to increasing from 3.24 m/s to 4.04 m/s with increasing moisture content from 11.94% to 27.10% (wb), the drag coefficient was decreasing from 0.76 to 0.66 with increasing in moisture content from 11.94% to 27.10% and from 3.08 to 3.96 m/s with increasing in moisture content from 6.50% to 30.10% (wb) while the drag coefficient of tef grain decreased from 0.83 to 0.65 with increasing in moisture content from 6.5 to 30.1% wet basis (wb) respectively.
Considering the terminal velocity of the teff grain which is the minimum of the range to be 3.24 m/s the radius of the air inlet was determined using the equation below and was calculated to be 55.61 mm.

$$\begin{aligned} Q_{in} &= Q_{out} \\ \pi r_i^2(\omega r_i) &= l \times h \times v \end{aligned} \tag{26}$$

The critical velocity sometimes determined from the dynamic head (N/m^2) of the air stream. Since the dynamic head of air (N/m^2) stream equals to the kinetic energy of a unit volume of air. Therefore the dynamic pressure head was calculated using Eqn. (27). Hence the dynamic head of the air at the exit of the blower was determined as 6.3 Pa.

$$h_d = \frac{mV_a}{2} = \frac{\rho_a V_a^2}{2} \tag{27}$$

where:- m=ρ_a, mass of 1/m^3 of air assume, at temperature of 20°C and 10.3x10^{4}N/m^2 tmospheric pressure, the air density ρa=1.2 kg/m^3,
The static pressure depends upon the resistance of the air

current, which is characterized the coefficient was calculated using Eqn. (21). If (G=gravitational force, N)=(R=upward force due to air stream, N) that is U=0, then Va=Vcr=critical velocity, m/s, the particle is suspended in air. [1] was determined some of the physical properties of the teff grain

and presented as shown in the following table 1 and hence the weight of the grain was considered to be the average of the two consecutive minimum moisture content which is 0.306 gram.

Table 1. *Physical properties of teff grain at various moisture contents.*

Moisture content (%, wb)	TGM (gram)	Coef. of friction, μ (on mild steel)	particle Density (kg/m³)
11.94	0.2920	0.41	1361.82
15.10	0.3201	0.50	1358.22
21.10	0.3609	0.66	1314.85
24.20	0.3918	0.58	1283.67
27.10	0.4207	0.52	1252.91

$$G = k \rho_a A V_a^2 \qquad (28)$$

Therefore, the coefficient 'k' was determined from above Eqn. (28) and was found to be

$$k = \frac{G}{\rho_a A V_a^2} = \frac{0.30 \times 10^{-3}\, kg \times 9.81}{1.2 kg/m^3 \times 0.28 \times 0.85 m^2 \times 3.24^2\, m^2/s^2} = 0.98$$

Static pressure head was determined by using Eqn. (29) and was found to be 0.26 Pa.

$$h_s = \frac{(1-k^2)h_d}{k^2} \qquad (29)$$

The total pressure, h=h_s + h_d and was determined as 6.56 Pa. The theoretical pressure head developed was determined by using Eqn. (30) and it value was calculated as 9.37 Pa.

$$H_T = \frac{h}{\eta} \qquad (30)$$

where: HT=theoretical total pressure head developed, Pa
 Η=is fan efficiency assumed to be 0.7.

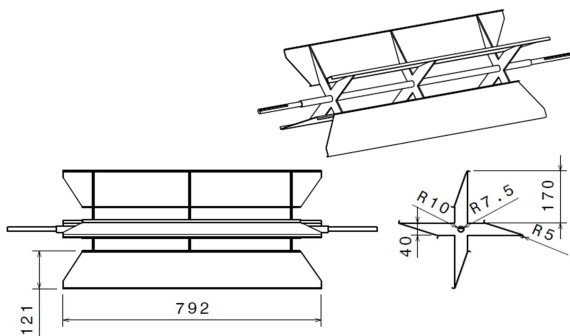### 2.2.9. Design of Impeller Components of Sucker

Figure 9. *Views of blower (all dimensions are in mm).*

The size of the teff grain is the smallest of all cereal which makes the cleaning of it very difficult by mechanical sieving and air blowing alone. Nonetheless the sucker was incorporate with in thresher and the inlet of the sucker was attached at the grain outlet to further clean the fine dust from the grain. The crown plate, base plate and vanes are the main

components of an impeller of the sucker. The individual components were designed and then fabricated. The components design is important to have a smooth entry and exit of the dust into and out of the impeller.

2.2.10. Design of Vanes and Their Angles

The vanes guide the dusts to the point of impact. The inlet vane angles, exit vane angles guide the dusts to the point of impact and they were designed for further fabrication of the impeller.

(i). Inlet Vane Angle

The dusts enter the impeller tangentially at an angle of 90°. Dusts travelling straight from the housing, takes a 90° turn at the point of entry in the impeller. The entering dusts make an angle at the point of entry and it can be calculated using equations 31 and 32 given by [20]. The velocity at the eye of the impeller was assumed to be less than terminal velocity of the teff grain that is 3.0 m/s since the tangential velocity of the impeller was determined to 1.05 m/s. Usually the dusts enter the vane area at a velocity (V_{m1}) (which is slightly higher than the velocity (V_o) at the entry point. So the velocity was fixed as 3.10 m/s. The tangential velocity (U_1) and entry velocity (V_{m1}) (are important for calculation of angle of entry of the dusts). The value of U_1 can be found out using Eqn. (31) and Eqn. (32) the angle at which the dust enters [10]. Where the diameter of the impeller at enters is given as shown in the figure 10 below and the speed of the impeller was determined to be 2000 rpm. Therefore the value tangential velocity and angle of entry were calculated and found to be 8.22 m/s and 20.81° respectively.
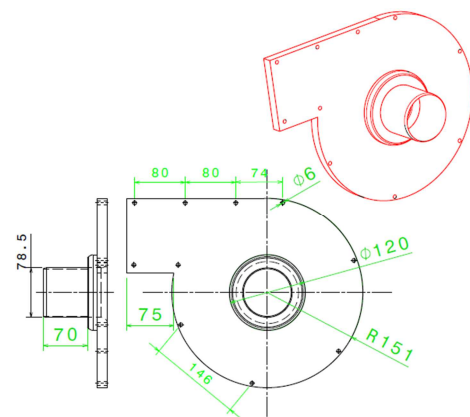

Figure 10. *Views of dust sucker (all dimensions are in mm).*

$$U_1 = \frac{\pi D_1 N}{60} \qquad (31)$$

$$\tan \beta_1 = \frac{V_{m1}}{U_1} \qquad (32)$$

The recommendation says that the inlet vane angles between 10° and 25° will give high efficiencies. So as per the recommendation the inlet vane angle was fixed as 20° [10].
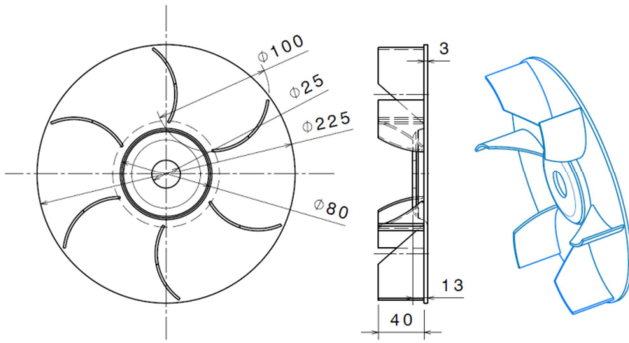

Figure 11. *Views of sucker impeller (all dimensions are in mm).*

(ii). Number of Vanes

The recommendations for number of vanes for handling bulk materials like small size solids, sewage, sludge and slurry is 3 to 6 vanes and radius of curvature is 5.0 cm [7]. So, five impellers were selected and constructed with the designed dimensions as shown in figure 11. All the calculations are made for the maximum peripheral speed, so the machine will work safely for the other peripheral speeds as well.
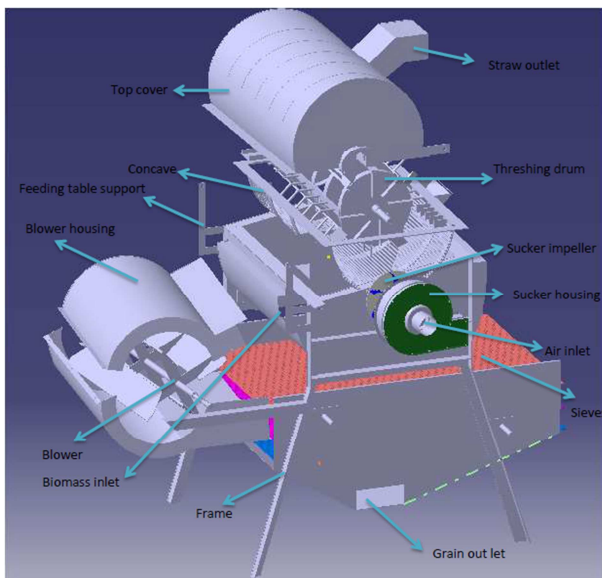
2.3. Machine Description and Operations

Figure 12. *Component parts of the thresher.*

The teff threshing machine comprises of the threshing unit, blower fan, feeding table, mechanical separators (sieves), dust suckers and the transmission unit. The threshing unit or beater is positioned above the concave where it rotates and threshes the teff by impact action. The figure 12 below some of the component parts of the threshing machine.

The blower fan is relatively positioned below the threshing unit to blow out the dust particles from the grain. The wide feeding table is provided at the opening of the threshing unit to carry teff straw to be threshed. The mechanical cleaner consists of three layers of sieves with 3, 2 and 1 mm hole diameters to separate chaff and grain. Besides to these components parts the machine consists of sucker at grain deliver unit of the thresher to further clean the grain by sucking the fine dusts remained to obtain clean teff grain. The transmission units are the pulleys and v-belts used to transmit motion and power from the prime mover to drum, blower, sucker and mechanical sieve.

2.4. Fabrication

The manufacturing process used in the fabrication of teff threshing machine is such that the total cost of fabrication is expected to be low and also one that can make use of the available materials. The manufacturing process involved in this work includes, joining of metal parts by welding, cutting using hacksaw, lathe machine, milling machine and grinding machine. Each component of the machine was fabricated separately before they were joined or welded together as the case may be.

3. Conclusion

The design and fabrication of a teff threshing machine has been successfully carried out by this work at Bako Agricultural Engineering Research Center. The machine is capable of threshing, separating stalk from grains, cleaning grains from chaff and dust, reduce the quality loss by large amount which is magnified in traditional threshing, decrease the time and cost required for threshing teff crop which is the most challenge full in it post-harvest processing than cereal crops. The contamination of foreign materials with grain; therefore, the threshing machine give a better method of threshing than the traditional methods. All the materials used were locally sourced and available on local market. The performance of the machine was evaluated at farm level and achieved good threshing capacity, threshing efficiency and cleaning efficiency with minimum loss as used to be. Generally the development of teff threshing machine create new chapter teff production particularly in reducing labor intensive teff threshing and sound full quality and quantity loss of teff due to traditional threshing.

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