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RESEARCH ARTICLE

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DESIGN AND CONSTRUCTION OF TEF GRAIN AND CHAFF SEPARATOR AND CLEANER

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ABSTRACT

Available evidence suggests that tef grain separating and cleaning, in the rural area of Ethiopia, is very much traditional and makes use of wind as winnower to effect separation manually operated set of flat sieve to accomplish the cleaning. This method is time and labor consuming, and often lead to contamination of the tef grain with dust, dirt, small sands or silt on the threshing ground. In an effort to alleviate the above stated problems, a small engine driven tef grain and chaff separator and cleaning machine was designed and manufactured.

Key words: belt, design, fan, moment, power, pulley shaft

1. INTRODUCTION

One of the economical cereal crops in Ethiopia is tef. It is indigenous to the country, and is a fundamental part of the culture, tradition, and food security of the people. Currently, tef is grown on approximately 2.80 million hectares of land which is 27% of the land area under cereal production. Tef accounts for about a quarter of the total cereal production and is highly economical food grain in Ethiopia (Bekabil *et al.*, 2011). The traditional methods of postharvest handling of tef usually lead into contamination of the product with stones, sticks, chaff, dirt and dust. Therefore tef grain, after threshing cannot be stored or used for consumption or as planting material due to the very fact that the presence of long straws, chaff, small fragments of spikes, leaves, dust, dirt and other foreign materials in the grain will accelerate deterioration, thus lead to poor physical condition. Tef winnowing, separation and cleaning i.e. removal of undesirable materials, is accomplished manually by tossing the grain into air and letting the wind do the separation and cleaning, removal of lightest impurities, leaves and large amount of debris with certain amount of grains. For further cleaning is usually done using sieves to remove the heavy particles and dirt larger than that of tef grain. Against all the odds, the Ethiopian farmers prefer to grow tef because it tolerance to low moisture stress, waterlogged and anoxic conditions being better than maize, wheat, and sorghum. Cattle prefer to feed on tef straw rather than any other cereal straws. Moreover, tef as grain has highest market prices than the other cereals; this includes both grain and straw, and the grain is not attacked by weevils during storage (Seyfu, 1993).

2. Materials and method

2.1. Design Considerations

The tef separator and cleaner was designed and developed based on the following considerations.

I. The availability of materials locally to reduce cost of production and maintenance.

II. The materials for the construction of the various component parts were selected on the basis of the force that would be acting on them, the work they are expected to perform and the environmental condition in which they would function.

III. Due to small size of tef grains and their low terminal velocity, separation of grains from chaff and straw and complete cleaning was thought to be practically impossible. Hence, it was decided to use combination of fan (pneumatic) and mechanical (sieves) methods to effect the separating and cleaning.

2.2. Description of the Machine

Figure 1 gives details of machine designed and constructed and used in the experiment.

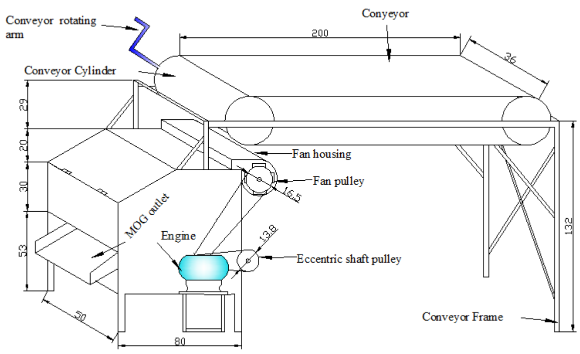


Figure 1. Details of the Experimental Tef Chaff Separating and Grain Cleaning Machine (all dimensions are in cm)

2.3. Working Principles of the Machine

The machine was designed to employ a combination of mechanical and aerodynamic principles to separate tef chaff and clean the tef grains. The separating and cleaning unit had three sieves, sieve frame, rocker arm and chaff and grain outlet pan. A mixture tef grains and chaff was uniformly spread on the belt conveyer along its entire length and fed into the separating and clean unit at predetermined rate of feeding. As the materials, tef grains and chaff were on their way into the separating unit, they were subjected to air current generated by the fan. As a result, light chaffs with low terminal velocities were blown away before they reached separating and cleaning units. Heavier materials, with higher terminal velocity, managed to land on the top sieve that permitted passage of tef grains and retained chaffs with sizes greater than the grains. The top sieve was designed and selected to retain very course materials and to convey them to the MOG outlet. The middle sieve was selected in such way that it could scalp all materials larger and heavier than grains passed through the top sieve. The purpose of the bottom sieve was to carry out further cleaning of tef grain from trash, sand, dust, dirt and broken grains.

2.4. Design Analysis

2.5. Determination of sieve slope

The sieve slopes used in the experiment were determined in the basis of the condition that the angle α be $\leq$ angle ϕ (Fouda, 2009). Where α is the inclination of the sieve with the horizontal while ϕ is the friction angle between the mixture grain mixture and the sieve surface.

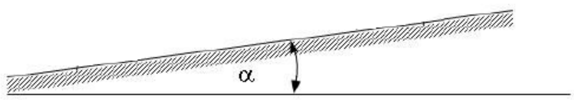


Figure 2. Inclination of sieve with the horizontal.

2.6. Design of belt conveyor

The conveyor, meant to feed the prototype machine at uniform and continuous feeding rate and avoid over loading of the sieves and/or intermittent flow of materials into the separation and cleaning unit. Conveying of materials was made possible by turning the conveyor rotating arm at constant and predetermined speed to achieve and maintain the desired feed rate (Fig. 3 and Fig. 1). According to Conveyor Belting Australia (CBA) (2009), the capacity of belt C (Eq. 1) was determined to be as 32.34 kg/s.

$$C = AV\rho \quad (1)$$

Where: A = load cross section area perpendicular to the belt computed as Width x length = 0.72 $m^{2,}$ P = density of material (1361 kg/m^3) and V = velocity of conveyor (0.033 m/s)

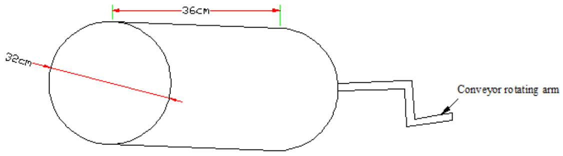


Figure 3. Material conveying drive system.

2.7. Design and fabrication of the separating and cleaning unit

The major components of separating and cleaning unit included sieves, sieve frame and rocker arm. The sieving unit consisted of three sieves for separating and cleaning simultaneously. The three sieves had length of 60 cm and width of 40 cm. The top sieve had round holes, 3 mm in diameter, the middle sieve had round holes 2 mm in diameter while the bottom sieve had 1 mm slotted holes. The vertical intervals between sieves were 10 cm (Fig. 4). The oscillating or the rocking legs were made from sheet metal having a thickness of 4 mm, a width of 4 cm and height of 50 cm. The bottom side of separating and cleaning unit was connected to connecting rod. The other end of the connecting rod was connected to a wrist pin eccentric to the wheel centre that produced a reciprocating motion on the separating and cleaning unit.

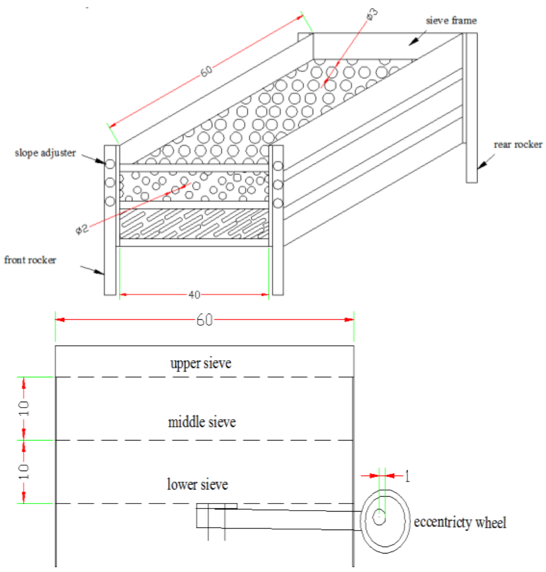


Figure 4. Schematic presentation of sieves arrangement and oscillatory motion generation system (all dimensions are in cm)

2.8. Design and construction of fan and housing

A centrifugal fan was constructed from a sheet metal of 1.50 mm thickness. It was mounted on a shaft with a diameter of 16 mm and supported on roller bearings below the feed conveyor. The fan assembly had four radial blades. The fan housing was provided with eight rectangular inlet holes, four on each side and one rectangular hole as an outlet of the air from the fan house, was provided. This hole directed air at recommended velocity towards materials falling from the conveyor into the separating and cleaning unit (Fig.5).

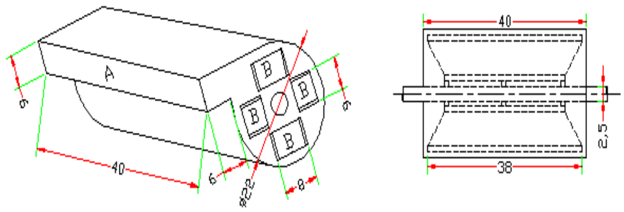


Figure 5. Fan housing and fan blade; A is outlet opening and B is inlet opening (all dimensions are in cm).

The air entrance into the fan housing was at 90^{O} to the air outlet opening. The air enters the fan housing parallel to the fan shaft and then rotated through 90^{O} by the fan blade before discharged. The fan discharged air blast under the feeding conveyor with constant velocity of 3.2 m/s (the minimum terminal velocity of tef) through the exit.

2.9. Selection of pulleys and belts

The machine required three pulleys; one pulley mounted on the crank shaft of the diesel engine as main drive, and on fan shaft and the eccentric wheel shaft one each. Two belts were used to transmit power from engine to the fan shaft and eccentric wheel shaft. The driving pulley was mounted on the crank shaft of the engine and the driven pulleys were mounted on fan shaft and eccentricity wheel shafts. Due to its availability, low cost and high performance cast iron pulleys were selected. The driving pulley was mounted on the crank shaft of the engine while the driven pulleys were mounted on fan shaft and eccentric wheel shafts. The diameter of the pulley used on the crank shaft of the engine was 120 mm. According to Sharma and Aggarwal (2006) the diameter of pulleys, center distance between pulleys, belt length and belt speeds were calculated as follows:

$$D_1N_1 = D_2N_2 \quad (2)$$

$$C = \frac{D_1 + D_2}{2} + D_1 \quad (3)$$

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

$$V = \frac{N_2 \pi D_2}{60000} \quad (5)$$

Where: D_1 and D_2 = diameters of driving and driven pulleys (mm), N_1 and N_2 = rpm of driving and driven pulleys, C = center distance between two adjacent pulleys (mm), L = length of belt (mm) and V = speed of belts (m/s).

Therefore, the diameters of pulleys used on the fan shaft and eccentric drive wheel shaft were calculated using Eq. (2) and were 165 mm and 138 mm respectively. The centre distances between the diving pulley and driven pulleys using Eq. (3) were 263 and 249 mm the engine and fan and engine and eccentric wheel shafts, respectively. The lengths of belts were calculated using Eq. (4) and found to be 1448 mm and 903 mm to connect pulleys on engine crank shaft and fan shaft and engine crank shaft and eccentric drive wheel shaft, respectively.

2.10. Determination of shaft diameter

The machine had two main shafts one for fan shaft and one for eccentric driving shaft. The diameters of the fan shaft and eccentric drive wheel shaft were determined using maximum shear stress theory. Figure 8 and 9 show forces acting on fan shaft and eccentric drive wheel shaft, respectively.

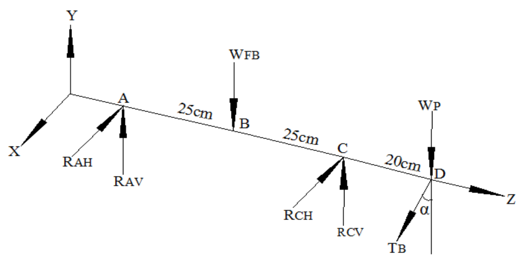


Figure 6. Forces acting on fan shaft and their locations.

Where: R_{AH} = horizontal bearing reaction force at A

R_{AV} = vertical bearing reaction force at A

W_{FB} = weight of fan blade

R_{CH} = horizontal bearing reaction force at C

R_{CV} = vertical bearing reaction force at C

T_B = total belt tension (T_i+T_j) at D

Wp = weight of fan pulley

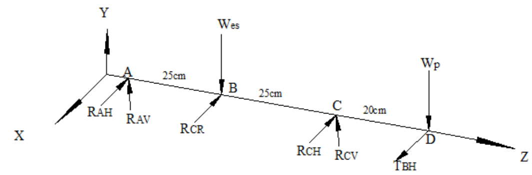


Figure7. Forces acting on eccentric drive wheel and their locations.

Where: R_{AH} = horizontal bearing reaction force at A

R_{AV} = vertical bearing reaction force at A

W_{es} = weight of eccentricity

R_{CR} = horizontal force due to connecting rod at B

R_{CH} = horizontal bearing reaction force at C

R_{CV} = vertical bearing reaction force at C

T_{BH} = horizontal tension due to belt at D (T_i+T_j)

Step1: Determining the direction belt pulls on the fan shaft (Fig. 8)

Then $\alpha = 31.33^o$

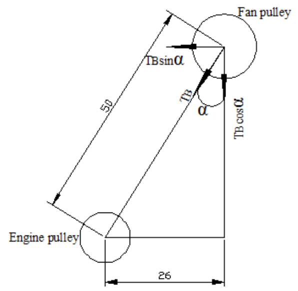


Figure 8. Direction belt pulls on the fan shaft.

Step 2: Determination of belt tensions (T_i and T_j) and torsional moment (M_t) according to sharma and Aggarwal (2006)

$$T_i = T_{max} - T_c \quad (6)$$

$$\frac{T_i - T_c}{T_j - T_c} = e^{\mu\theta \operatorname{cosec}\frac{\alpha}{2}} \quad (7)$$

$$T_{max} = \sigma a \quad (8)$$

$$T_c = mV^2 \quad (9)$$

$$\theta = 180 - 2\left[\sin^{-1}\left(\frac{D_2 - D_1}{2C}\right)\right] \quad (10)$$

$$M_t = (T_i - T_j)\frac{D_2}{2} \quad (11)$$

Where T_i, T_j, T_{max}, T_c, σ, a, m, v, μ, α and θ are the tight side, slack side, maximum centrifugal tension of a belts (N), maximum safe normal stress (N/mm^2), a is cross sectional area (mm^2), mass per unit length (kg/m) of belts, speed of belt (m/s), coefficient of friction between belt and pulley, groove angle and angle of wrap respectively.

From standard table Khurmi and Gupta (2005), Sharma and Aggarwal (2006) the value of σ, a, m, μ and α are 2.1 N/mm^2, 2.1 N/mm^2, 0.108 kg/m, 0.3 and 40º respectively.

Then according to eqn 6 to 11 T_i, T_j, T_{max}, T_c, θ and M_t are 162.04N, 19.12N, 170.1N, 8.06N, 3.05 rad and 11791 N-mm respectively

Step 3: Analysis horizontal and vertical forces on fan shafts in order to calculate bending moments

- **Forces acting on fan driving shaft - vertical (YZ) plane)**

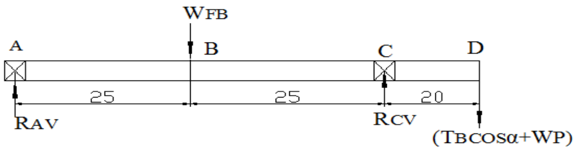


Figure 2. Free body diagram of the fan shaft on vertical (YZ) plane.

$T_B = T_i + T_j$ = 162.04 +19.12=181.16 N

W_{FB} =11measured N(measured), W_p =15 N(measured), α = 31.33^o, ($T_B \cos\alpha + W_P$) = 170 N. In order to calculate reaction forces R_{AV} and R_{CV}, it was considered that

$$\sum BM_A = 0$$

50 *R_{CV} = 25*W_{FB} + 70 ($T_B \cos\alpha + W_P$)

R_{CV} = 243.50 N

$$\sum F_v = 0$$

R_{AV} + W_{FB} + R_{CV} + ($T_B \cos\alpha + W_P$) = 0

R_{AV} = 62.50 N downward

- **Forces acting on fan driving shaft on horizontal (XZ) plane**

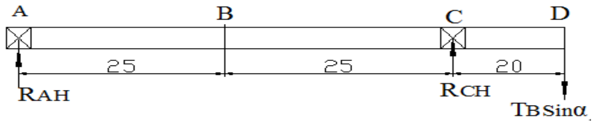


Figure 3. Free body diagram of fan shaft on horizontal (XZ) plane.

$T_B \sin\alpha$ =181.16 x sin31.33 = 94.2 N

$$\sum BM_A = 0$$

50 *R_{CH} = 70 $T_B$sin a

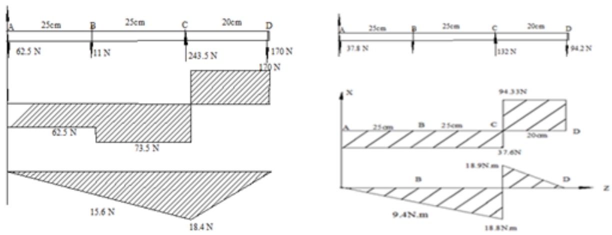
R_{CH} = 132 N

$$\sum F_h = 0$$

R_{AH} + R_{CH} + $T_B$sina = 0

R_{AH} = 37.8 N downward.

Based on the magnitude and location of all forces acting on the fan shaft shear force and bending moment on the horizontal and vertical plans containing the fan shaft were computed and plotted as follows:



a) Vertical (YZ) plane b) Horizontal (XZ) plane

Figure 4. Shear force and bending moment diagrams for fan shaft

From fig. 11 the maximum bending moment was found to be at point C of the fan shaft.

$$M_{max} = \left(M^2{}_V + M^2{}_H\right)^{1/2}$$ = (18.4^2+18.9^2)$^{1/2}$ = 26.38 N-m

Finally the diameters of the fan shaft were computed as 16mm with eqn 12.

$$d^3 = \frac{16}{\pi\tau_{max}}\sqrt{(k_b M_b)^2 + (k_t M_t)^2} \quad \textbf{(12)}$$

where: d is diameter of the shaft (m), M_t is torsional moment (Nm), M_b is maximum bending moment (Nm), K_b is combined shock and fatigue factor applied to bending moment, K_t is combined shock and fatigue factor applied to torsional moment, τ_{max} is allowable stress (55 MPa for shaft without key way and 40 MPa for shaft with key). For rotating shafts, when load is suddenly applied (minor shock): K_b = 1.2 to 2.0; K_t = 1.0 to 1.5. It must be noted that factor safety need to be considered in actual design work.

Following the same procedure the diameter of eccentric wheel drive shaft calculated as d = 20 mm.

2.11. Determination of power

The power required to operating the machine was considered to be the sum of powers required to drive the fan assembly, the eccentric drive assemble, the sieving system including the grain and chaff on the sieves, power required to oscillate the sieves and the loads on them and power required to overcome frictional resistance. The total power (P_t) required for the cleaning and separating processes was determined by using the Equation given by Nduka *et al*, (2012).

$P_t = P + 10\%P$ (10% is possible power loss due to friction drive) (12)

Where: P_t = total power required to drive the machine, P = the sum of $(T_i - T_j)$ V for fan and sieve oscillation, T_i = tight side tension of belt, 162.04 N for both fan and eccentric drive wheel belts and T_j = slack side tension of belts = 19.12 and 18.17 N for fan and eccentric drive wheel belts, respectively, and V = speed of belts = 8.64 m/s for both fan and eccentric drive wheel belt.

On the basis of the above, power required to drive the fan assembly, P_f = (162.04 N - 19.12 N) 8.64 m/s = 1234.83 W

Accordingly, the power required to drive eccentric drive wheel assembly, Pe = (162.04 N – 18.17 N) 8.64 m/s =1243.04 W

The total power to operate fan and eccentric drive assembly $P = P_f + P_w$ = 1234.43W + 1243.04W = 2477.87 W.

Overall total power P_t = P + 10% of P = 2725.66W = 3.63 hp.

CONCLUSION

The prototype machine was designed and fabricated at AAMRC on the basis of the physical and aerodynamic properties of the grain to be separated and cleaned from undesirable material(s). The physical attributes and aerodynamic properties of tef grains and the terminal velocity of the same were obtained from the data generated by the earlier research endeavor. Among range of sieve sloped, sieve oscillations and feed rates recommended by different researchers, suitable ranges based on the size of the machine that would be designed and fabricated were selected.

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