American Journal of Mechanical Engineering, 2016, Vol. 4, No. 3, 108-111
Available online at http://pubs.sciepub.com/ajme/4/3/4

DOI:10.12691/ajme-4-3-4

Design, Fabrication and Testing of a Wet Legume Dehulling Machine

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Abstract The traditional method of dehulling legumes such as cowpea, soya beans and locust beans is by rubbing the seeds between the palms or by pounding in mortar but this process consumes time and energy. A motorized wet type legume dehuller was designed and fabricated to de-hull soaked seeds. It was designed to work on the principle of compression for the splitting of the soaked seed coat. The de-hulling is achieved by shear and friction between the beans coat and the wall of the de-hulling chamber. The device was designed and fabricated using locally available materials and technology. The motorized legume de-huller operates at effective de-hulling speed of 438 rpm and achieved a maximum efficiency of 95.2% for 6 minutes soaking period for cowpea, 72.2% efficiency for soya beans soaked in hot water for 30 minutes and 62.5% efficiency for 15 hours cooking period for locust beans, The de-huller was designed for feed rate of 500 seeds per second.

Keywords: *legume, dehulling, abrasive, strength, soaking period*

Cite This Article: Egbe E.A.P., and Roland B.O., "Design, Fabrication and Testing of a Wet Legume Dehulling Machine." *American Journal of Mechanical Engineering*, vol. 4, no. 3 (2016): 108-111. doi: 10.12691/ajme-4-3-4.

1. Introduction

Legume is an important food item (because of its high protein and fat contents) in Nigeria and most West African countries [1]. Dehulled cowpea is required in the preparation of some table meals (fried beans cake-akara, boiled cake-moimoi). In spite of the widespread use of beans as a staple food among Nigerians, the dehulling of beans is carried out manually. The operation generally requires soaking the cowpeas in water for 2 to 10 minutes [2] draining and rubbing between the palms, or beating with wooden pestle and mortar. Housewives and food vendors carry out these operations daily [3].

A knowledge of mechanical properties of grains, such as hardness and compressive strength are vital to engineers handling agricultural products. Hardness is defined as the ability of a material to resist indentation or abrasion. This property is required for the design of agricultural processing equipment to minimize breakage and wastage. Abrasive strength is the force requird to remove or protect the very tight coated membrane on grains such as cowpea seeds. Chukwu & Sunmonu [4] submitted that the mean compressive strength, tensile strength, abrasive strength, shear strength and torsion strength of Sampea 7 cowpea are 66.25 N, 65.53 N, 64.55 N, 65.20 N and 65.00 N, respectively and for Tvx 3236 cowpea are respectively 93.65 N, 93.55 N, 92.56 N, 93.50N and 92.75 N.

2. Materials and Method

All the materials used for different parts of the machine were locally sourced. Galvanized iron sheet was used for the hopper; due to its corrosion resisting properties. Plain carbon steel was used for the shaft due its moderate ductility and malleability which improves its workability and it is widely available and obtained at a cheaper price compared to other alloy steels. The choice of 1×1×1.5mm mild steel square pipe, for the frame, was influenced by its availability and cost of purchase compared to an angle iron. Galvanized steel pipe was used for the dehulling chamber. A thick galvanized steel sheet was used for the auger majorly because of its ability to resist rust formation on the surface which can contaminate the food substance being dehulled.

A 1.5 mm thick galvanized steel sheet was used for frame cover due to its high malleability and ductility which makes it easy to be folded and joined. Alloyed steel pulley was used due to its rigidity and its availability.

2.1. Determination of the Striping Torque of Different Cultivars of Cowpea

Two cultivars of cowpea were considered for this design, Sampea 7 and Tvx3236. The abrasive force is denoted by F_a. Therefore dehulling/striping torque required is given by:

$$T = F_a \times D_g \qquad (1)$$

where D_g = geometric mean diameter of the seed.

The approximate abrasive forces required to dehull cowpea Tvx 3236 and sampea 7, at a feed rate of 20 seeds per minute was given as 92.56N and 64.55N, by Chukwu and Sumonu, [4]. The average abrasive force for feed rate of 500 seeds per second was deduced to be 1571.1N for

Tvx3236 and sampea 7. The average diameter of seed is 6.245mm and Equation (1) yields,

$T = 1963.875 \times 0.006245 =$ 12.2644Nm for feed rate of 500 seeds per second.

2.2. Determination of the Power Required from the Motor

Effective power for dehulling, $P_D = T\omega_2$, where T = Dehulling torque for feed rate of 500 seeds per second, ω_2= 45.87rad/s (For effective dehulling speed of 438rpm [5]).

Using the relation,

$$\frac{T_m}{T} = \frac{w_2}{w_1} \quad (2)$$

Implies that,

$$T_m = \frac{Tw_2}{w_1} \quad (3)$$

where T_m= required motor torque, ω_1= 146.6rad/s =speed of rotation of the motor

.:. Power P= $T_m \times \omega_1$ = power required from the motor

Substituting T=12.2644Nm, ω_1=146.6rad/s, ω_2=45.87rad/s,

$$T_m = \frac{12.2644 \times 45.87}{146.6} = 3.83744\text{Nm}.$$

Therefore the power required from the motor becomes;

P= 3.83744×146.6 = 562.57Watts. Say one horse power, 760W (accounting for efficiency of motor).

2.3. Determination of Pulley Diameters

The electric motor for this project was supplied with a 50mm diameter pulley and runs at 1400 rpm. However the speed for optimum dehulling is 438 rpm. Thus a speed reduction was designed for.

The machine pulley diameter was determined with Equation (4)

$$N_2 D_2 = N_1 D_1 \quad (4)$$

where D_2 = diameter of the motor pulley, 50mm; N_2 = speed of the motor pulley, 1400 rpm;

N_1 = desired speed of the machine, 438rpm; D_1 = desired diameter of the pulley.

Substituting all known values yields,

$$D_1 = \frac{1400 \times 50}{438} = 159.8\text{mm}(\text{say } 160\text{mm}).$$

2.4. Determination of Tension in the Belt

Assuming mass (m) of the belt is negligible, for an A class of V-belt, Khurmi and Gupta, [6] gives the relationship between tension F_1 and F_2 (Figure 1) as,

$$2.3\log\left(\frac{F_1}{F_2}\right) = \mu\theta \operatorname{cosec} \beta \quad (5)$$

where μ =coefficient of fiction between the belt and side of groove, β = half angle of groove, T_1 = tension of tight side, T_2 = tension on slack side and θ = angle lapped by the belt on small sheave in radian. From equation (5),

$$\left(\frac{F_1}{F_2}\right) = K = \log^{-1}\left(\frac{\mu\theta\operatorname{cosec}\beta}{2.3}\right) \quad (6)$$

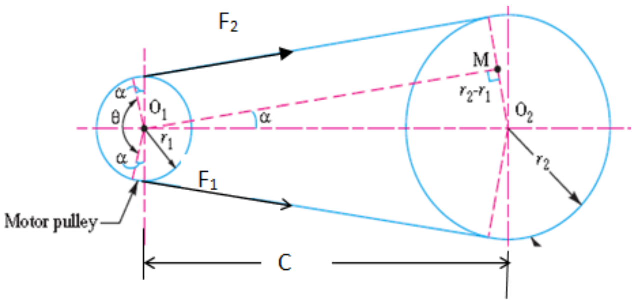

Figure 1. The Open Belt Drive

The angle of lap on the small and big pulleys are given by Equation (7) and (8) respectively,

$$\theta_s = \prod -2Sin^{-1}\left(\frac{r_2 - r_1}{C}\right) \quad (7)$$

$$\theta_L = \prod +2Sin^{-1}\left(\frac{r_2 - r_1}{C}\right) \quad (8)$$

where C is the centre distance and r_1, r_2 are the radii of the driver and driven respectively.

Torque transmitted is given by,

$$T = (F_1 - F_2)r \quad (9)$$

where r = radius of sheave.

Substituting $r_2 = 80mm$, $r_1 = 25mm$, $C = 311\text{mm}$ into Equation (7) yields;

θ_s = 2.79 radians (159.6^0)

Substituting, μ = 0.23, θ = 2.79 radians, β = 17.5, R = 0.08m and T= 12.2644 Nm,

$$\log^{-1}\left(\frac{0.23 \times 2.79\text{cosec}17.5}{2.3}\right) = 8.5.$$

From equation (6) and (9) the slack side tension is

$$F_2 = \frac{T}{r(K-1)} = \frac{12.2644}{0.08} \times \frac{1}{8.5-1} = 20.441 \text{ N}.$$

From equation (6)

$$F_1 = KF_2 = 8.5 \times 20.441 = 173.75 \text{ N}.$$

2.5. Determination of Force on the Shaft

The auger shaft and the pulley are shown in Figure 2.

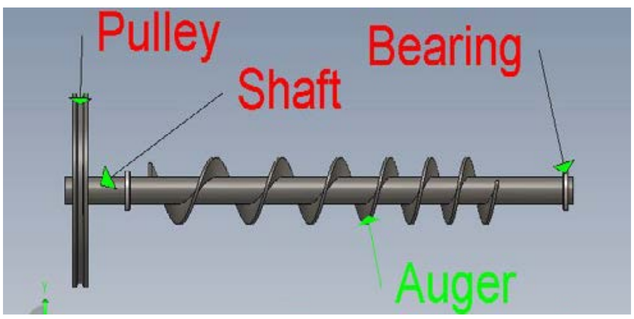

Figure 2. Components on the shaft

The actual power delivered to the shaft at point A (Figure 3) is 760 W. Thus the corrected shaft torque at this point is 16.57 Nm. The net driving force on sheave, F_N, is given by,

$$F_N = F_1 - F_2 = \frac{T_A}{r_A} = \frac{16.56}{0.08} = 207.125N.$$

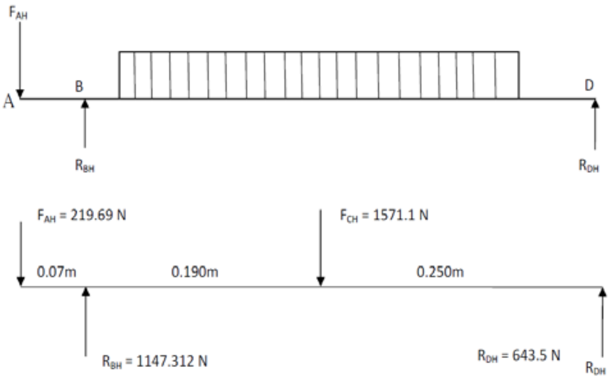

Figure 3. Horizontal Forces and Reactions

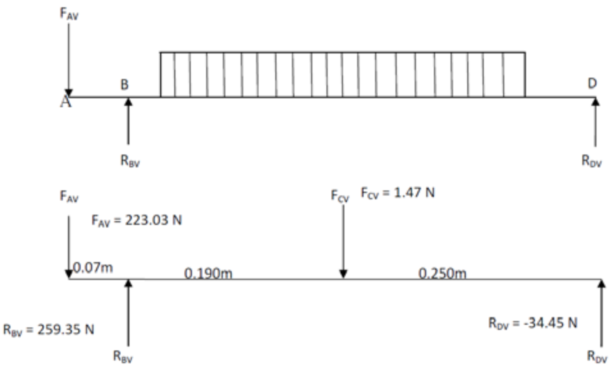

Figure 4. Vertical forces and reactions

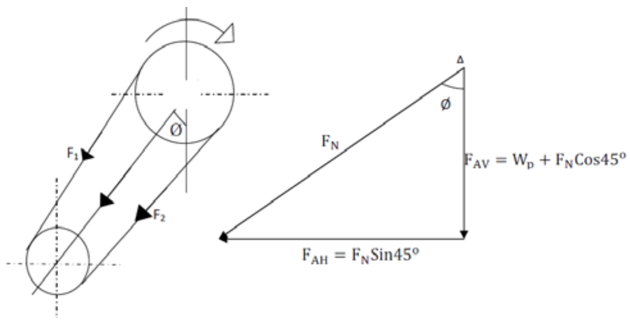

Figure 5. Forces acting on pulley at point A on the shaft

The bending force F_A resulting from the net driving force is $1.5xF_N$ [7]. Thus F_A = 310.69 N and it acts upwards at an angle of 45^o from the horizontal (Figure 5). The components of forces in the horizontal and vertical directions together with the reaction were found and the values inserted in Figure 3 and Figure 4. The maximum bending moment occur at C and found to have a value of 161.12 Nm.

2.6. Determination of Shaft Diameter

The shaft is subjected to torsion at point A and combined bending and torsion at B and C. According to Mott, [7] the diameter of shaft is given by Equation (10);

$$D = \left[\frac{32N}{\pi} \sqrt{ \left[\frac{K_t M}{S_n'} \right]^2 \oplus \frac{3}{4} \left[\frac{T}{S_y} \right]^2 } \right]^{\frac{1}{3}} \quad (10)$$

where N= design factor accounting for safety, K_t account for stress concentrations, M = bending moment, S_n’ = endurance limit, T = torque and S_y = the yield strength.

2.6.1. Diameter at A

Torque at A = 16.57 Nm, bending moment = 0 and torque can be considered as fairly steady. Applying a design factor, N = 3, S_y for the steel = 220.59MPa, S_n’ = 199.91MPa and substituting all relevant data from section 2.5, the diameter at point A becomes;

$$D_A = \left[\frac{32x3}{\pi} \sqrt{ \frac{3}{4} \left[\frac{16.57}{220.59x10^6} \right]^2 } \right]^{\frac{1}{3}} = 12.6 \text{ mm.}$$

2.6.2. Diameter at Bearing B

To the left: a relief diameter leading to the bearing with well rounded fillet radius is desired. Thus K_t = 1.5 [7], T = 16.57 Nm, M= 21.914 Nm and N = 3. Substituting into Equation (10) yields, D_{B1} = 17.55mm.

To the right: bearing seat with a shoulder fillet requires a fairly sharp fillet. Applying a K_t value of 2.5 and keeping other parameters at the values to the left, the bearing seat diameter becomes D_{B2} = 20.5 mm.

2.6.3. Diameter at C

The auger introduces high stress concentrations along the shaft within this zone. Thus K_t = 2.5 was used. The torque remains 16.57 and bending moment was found to be 161.12 Nm [Section 2.5].

Therefore:

$$D_C = \left[\frac{32x3}{\pi x10^6} \sqrt{ \left[\frac{2.5x161.12}{199.913} \right]^2 \oplus \frac{3}{4} \left[\frac{16.57}{220.59} \right]^2 } \right]^{\frac{1}{3}}$$

$$= 39.5\text{mm, Say } 40\text{mm.}$$

2.6.4. Diameter at Bearing D

There is no bending moment or torque at the second bearing but there is shear load. The resultant shear load (Figure 4 and Figure 5) V_D is 644.443 N. Applying the distortion energy criterion of failure the diameter at bearing D was found to be 5.1 mm, [say 10 mm, for ease of bearing selection].

3. Testing, Results and Discussion

Two varieties of seeds were poured into the hopper while the machine was running to evaluate the dehulling efficiency. The dehulled seeds were separated from undehulled seeds weighed separately and recorded. The dehulling efficiency was determined by the ratio of the dehulled to the overall sum total.

3.1. Efficiency of Dehulling

Efficiency of the machine was then calculated based on of dehulled and undehulled.

$$\eta = \frac{\text{Dehulled Beans}}{\text{Undehulled Beans+Dehulled Beans}} \times 100$$

The results obtained for cowpea are presented in Table 1 and that for soya beans in Table 2.

Table 1. Results for cowpea

Soaking time, min	Weight of Soaked seed(kg)	Undehulled beans (kg)	Dehulled beans (kg)	efficiency (%)
2 min	1.25	1.129	0.121	90.3
4 min	1.25	1.171	.079	93.7
6 min	1.25	1.19	0.06	95.2
8 min	1.25	1.189	0.061	95.1
10 min	1.25	1.188	0.063	95

Table 2. Soya beans (soaked in hot water)

soaking time, min	Weight of Soaked seed(kg)	Dehulled beans D.B (kg)	Undehulled beans U.B (kg)	% efficiency
20 min	2.25	1.25	1	55.6
25 min	2.25	1.5	0.75	66.7
30 min	2.25	1.625	0.625	72.2

The result obtained showed that the maximum efficiency of the machine is 95.2 % for dehulling of cowpea after 6 minutes of soaking and 72.2 % for soya beans after 30 minutes of soaking in hot water. The seeds that were not dehulled on first pass were dehulled on passing them through the machine the second time.

While carrying out the experiment, it was observed that the more the quantity of seed poured into the machine at a time the lesser the seeds that come out undehulled. Thus optimum feed rate is required to achieve maximum dehulling. The throughput capacity of the dehuller is 594kg/hr.

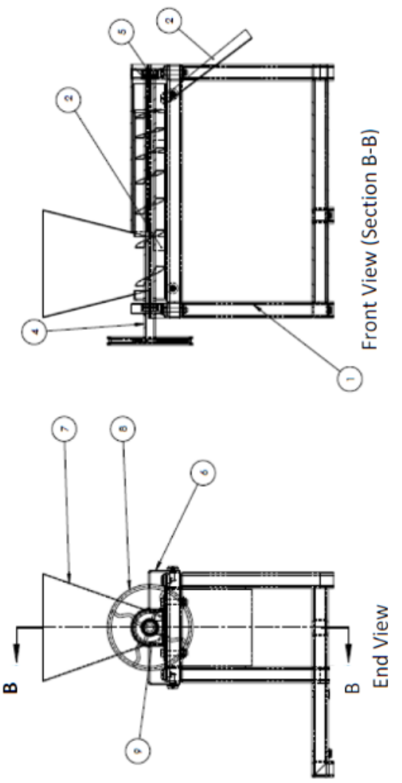

Figure 6. Sectioned End and Front Views of Legume Dehuller

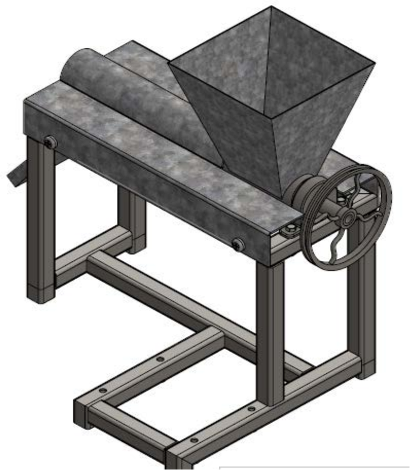

Figure 7. Isometric View of Legume Dehuller

4. Conclusion

The design, fabrication and testing of a beans dehuller with a throughput of 540kg/hr and dehulling efficiency of 95.2% was successfully carried out in this work. This work has established that development of high performance machines from locally available materials, with available technology is achievable.

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