

DESIGN AND DEVELOPMENT OF A GROUNDNUT DECORTICATING AND SEPARATING MACHINE.

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Abstract : An integrated groundnut decortivating and winnowing machine which bears a 1 Ihp, 1000 r.p.m. electric motor was designed and developed. It is a 1000mm x 400mm x300 mm machine of gross mass 33.42kg which makes it portable. Falling pods from the hopper are struck by a rotating dehusking drum of 72.2 N force, the chaffs are blown out through a window and the seeds fall down and out via a chute. All the components of this machine were sourced locally making the overall cost cheap and affordable to the people in contrast with the decorticators imported from overseas.

KEYWORDS: Groundnut, decorticator, blower, locally-sourced materials, affordable.

1.0 INTRODUCTION

Groundnut or peanut or earthnut (*Arachis hypogaea*) is one of the most important food legumes in the world (1). It has been identified as one of the leguminous species with the greatest potential for both food and industrial purposes in the tropical regions of Africa. This crop is a major source of income to the people in the major producing states of the federation and is a major source of revenue for rural and other capital development projects in such states. The production rate world-wide is over 270,000 metric tonne annually (2). Nigeria possesses a land area of 923,768 Km² of arable land i.e. about 91 million hectares of land, and vegetation ranging from the Sahel savanna in the extreme North to Swamp forest in the South. Of this, over 85 million hectares is uncultivated land, about 82 million hectares has the potentials for agriculture, and groundnut can thrive very well in all the states of Nigeria (3). Even with this little cultivated portion (6 million hectares or 55,500km² or 5.1%), Nigeria is ranked 4th in the world in the production of groundnut with over 1.55 metric ton in 2012 (2).

Nigeria, though the fourth highest producer of groundnut in the world still finds it hard in processing this oil seed in the area of dehusking. A good percentage of the farmers of this crop crack the pods using fingers and stones or by beating out the seeds with paddles and sticks. This is time wasting, boring and energy sapping. Other producers and marketers of this economically important crop who procure decortivating machines from overseas do this with tears because the stresses and costs involved in importation is enormous. Some imported large-scale plants are increasingly replacing small groundnut processing units, but due to high foreign exchange rate, the cost of such imported machines is clearly out of reach of poor farmers in Nigeria. Developing an affordable groundnut decorticator to ease the labour associated with shelling and reduce operational cost and time would enable the high potentials of the crop to be harnessed optimally.

A typical groundnut decorticator removes the seeds from the pods. In general, most decortivating machines around the world are motorized using electricity from the power grid. Some are powered using diesel or petrol generators while a few others are manually operated. In all these, huge energies involved in running the machines add to the operational costs of decortication. It is the aim of this design to develop a light decorticator fabricated from locally-sourced

materials, which is cheap and affordable to the majority of the farmers and processors of groundnut to encourage the cultivation of this important food crop.

2.0 METHODS AND MATERIALS

2.1 Description of the groundnut Sheller

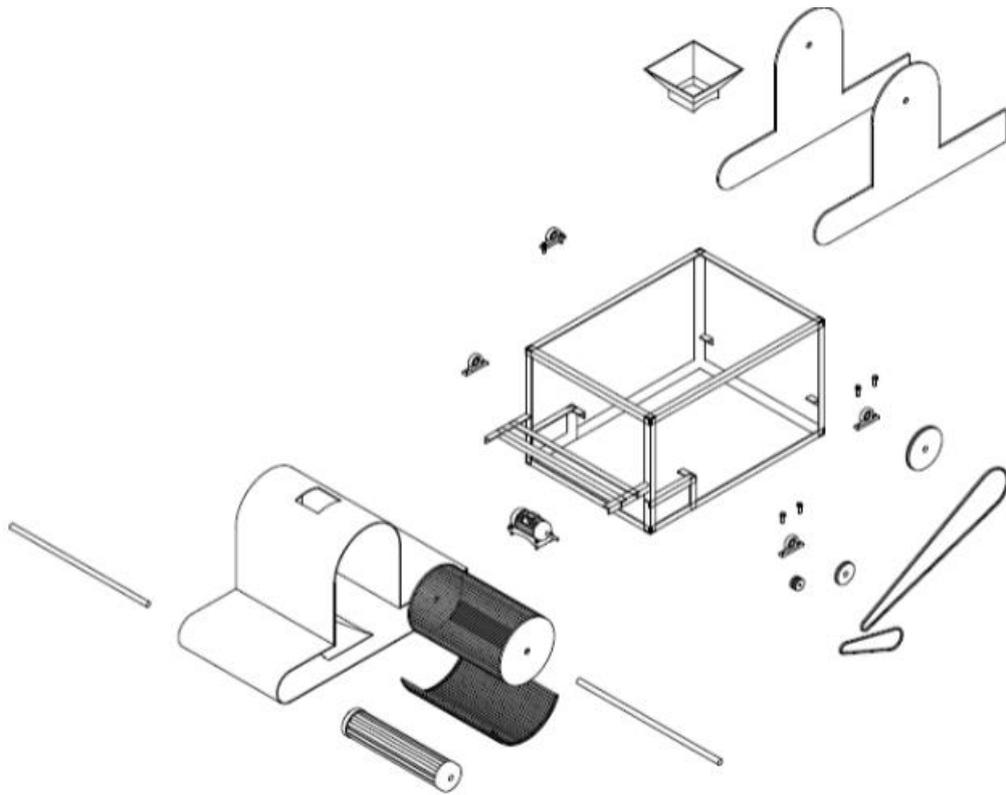
In this design, the shelling machine consists of the following basic units: the frame, hopper, shelling, winnowing and delivery units.

The frame carries the entire components of the machine. It was constructed from 50mm by 50 mm angle mild-steel iron.

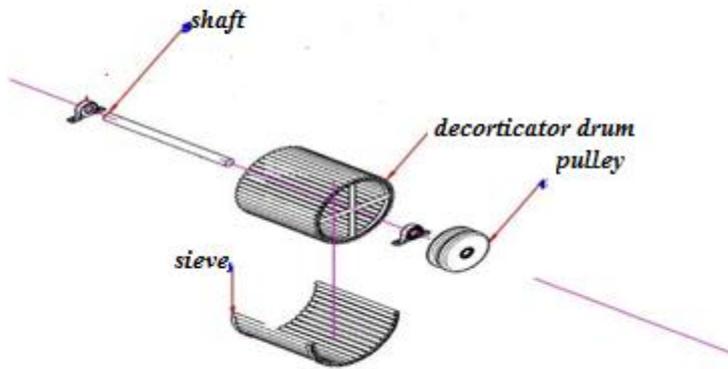
The hopper feeds the groundnut pods to be decorticated into the shelling unit, the pods falling down the shelling unit by gravity. The hopper is square shaped with the inlet side frames tilted 60° to the horizontal to prevent falling out of pods during decortication. The neck base of this hopper which leads into the shelling unit is square, 58 by 58 mm² and can only let a maximum of 86 pods into the decortication or shelling cylinder per unit time.

The decorticator drum is a cylinder formed with 6 mm diameter spiral aluminium rods with the spiral edges forming the shelling spikes. It bears two circular mild-steel plates of thickness 3.5 mm and 200 mm diameters, which were drilled through at the centre to hold the 14 mm diameter decorticator shaft in position with the end bearings.

Decortication is effected when the falling pods from the hopper hit the rotating spikes of the aluminium rods that form the cylinder drum. The nuts and chaffs by machine vibrations fall through a semi-circular sheet metal sieve of 10 mm diameter slots, fixed just under the rotating decorticator. The groundnut shells are separated from the seeds by an air stream supplied by a blower as both materials fall towards the delivery pan. The chaffs are blown out through chute 1 while the seeds fall down and out through chute 2. Undecorticated pods cannot pass through the 10 mm diameter holes of the sheet metal sieve and are swept out after decortication operations.



Figures 1- The shelling components of the machine



Figures 2 – The shelling system

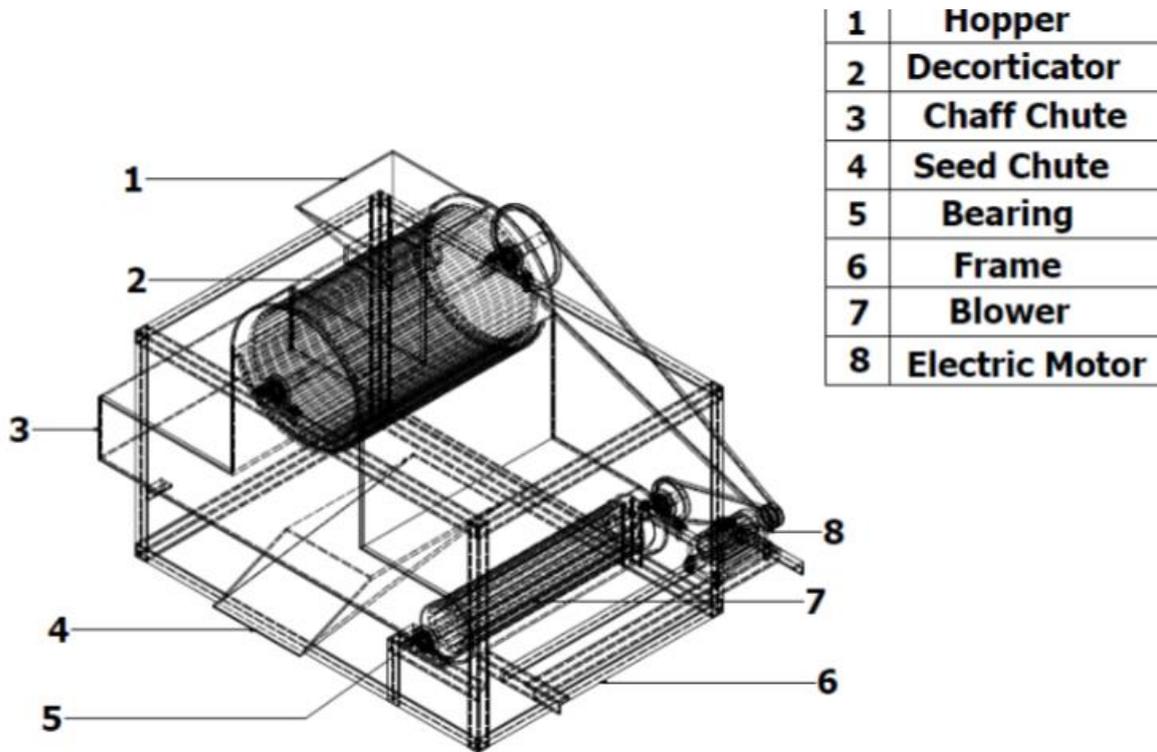


Figure 3-The wire view of the integrated machine

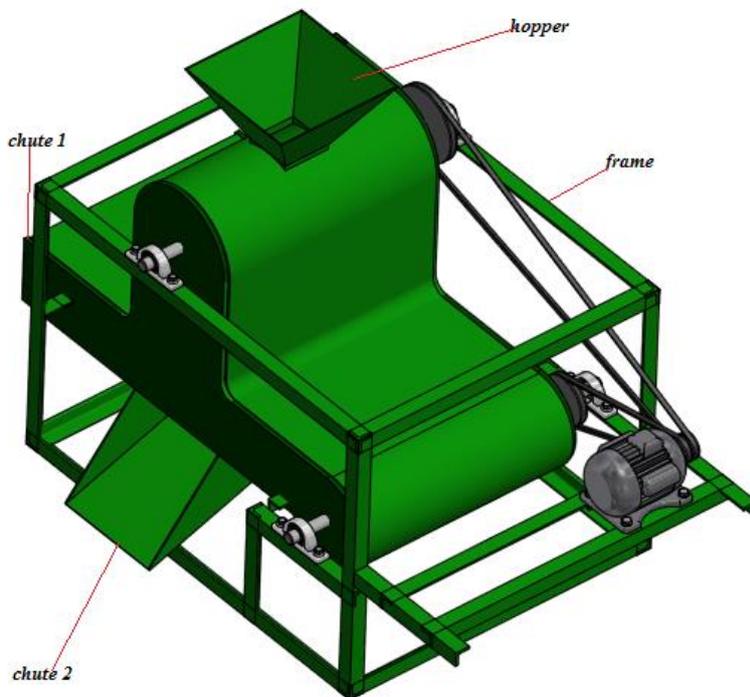


Figure 4 - The integrated decorticating and separating machine.

Using the Beamboy software, the relevant reactions and moments were obtained:

Vertical load components of the decorticator belt = $T_1 \sin 12^\circ + T_2 \sin 12^\circ$

Horizontal forces $R_{1H} + R_{2H} = T_1 \cos 12^\circ + T_2 \cos 12^\circ$

3.0 DESIGN ANALYSIS AND CALCULATIONS

Some ground nuts were stochastically collected from a basket of the dry nuts using small plate and weighed.

2.1.1 Weight of a dry undecorticated groundnut

Average mass of one dry groundnut nut was obtained using this expression, $A_m = \left(\frac{P_3 - P_1}{P_2} \right)$ (1)

where, number found in the plate = $p_2 = 67$ nuts

Mass of both plate and contents = $p_3 = 62.65$ g

Mass of empty plate = $p_1 = 13.43$ g.

2.1.2. Breaking force dry groundnut pod using a falling mass

Mass of falling body = 400g = 0.4kg

Average force required to break one pod (without crushing the seeds) may be obtained thus:

Employing Newton's law of linear motion $s = ut + \frac{1}{2} gt^2$ (2)

or $g = \frac{2s}{t^2}$ if the initial velocity $u = 0$ (3)

where t and s = time taken to hit the ground and height of fall respectively, and g = gravitational acceleration.

Thus for a mass (m) of 400g to take the time $t = 0.8$ seconds to fall freely from a height $s = 0.75$ metres to hit the nut, the

force generated by the falling body (from 2 and 3), to decorticate one pod, the force $F = mg \Rightarrow F = \frac{2ms}{t^2}$

(4)

The diameters of the motor and decorticator sheaves according to (4) are related to their revolutionary speeds as $\frac{n_1}{n_2} =$

$\frac{d_2}{d_1}$ (5)

Where n_1 = motor sheave speed = 1000rpm

n_2 = decorticator drum sheave speed = 200rpm

d_1 = motor sheave diameter = 40mm = 0.04 m; $r_1 = 0.02$ m

d_2 = decorticator drum sheave diameter.

From equation (5), $d_2 = \frac{n_1 d_1}{n_2}$ (6)

2.1.3. Hopper dimensions

Area of square hopper neck from this expression $A_h = l^2$, (7)

l being the length, 38 mm and breadth, 38 mm of hopper

Estimated surface area of one undecorticated dry groundnut pod (obtained by the expression of assuming areas of two small circular objects, each of diameter 5 mm stuck together)

$$A_g = 2\pi r^2 \quad (8)$$

∴ Estimate maximum pods that can go through the hopper neck per unit time $N_h = \frac{A_h}{A_g} =$ (9).

Also, mass of maximum pods in decorticator per unit time $m_p = N_h \times A_m$ (10)

The total force required to decorticate this maximum number of pods that may from time to time go through the hopper neck into the decorticator drum per unit time was considered as

$$F_T = F \times N_h \quad (11)$$

2.1.4 Decorticator drum

This drum was fabricated of 0.7mm spiral aluminium rods. The spiral edges form the cracking spikes.

Length of decorticator cylinder = 200 mm

Diameter of decorticator drum = 200 mm; radius = 100 mm

Mass of decorticator drum (measured without groundnut pods) = 1100 g

2.1.5. Force possessed by revolving decorticator

The centrifugal force possessed by the spiral spikes of the decorticator rods $F_c = \frac{mv^2}{r}$ (12)

where m = mass of decorticator drum + mass of maximum number of groundnut pods in it

v = linear velocity of spikes on drum since decorticator pulley speed $n_2 = 200$ rpm

and $v = \omega r$, $\Rightarrow v = \frac{2\pi n_2 r}{60}$ (13)

r = distance from axis of shaft rotation to the top edge of the 2 mm thick spiral spikes of the drum. This is radius of drum (Rs) + height of spike (Hs).

∴ Force possessed by the decorticator spikes, the centrifugal force from equation (12) = F_c

To minimize energy waste, the force possessed by the revolving decorticator should just be slightly higher than the force required to decorticate the maximum number of pods that may possibly get into the decorticator drum per unit time.

2.1.6 Torque possessed by decorticator spikes T_d

$$T_d = m\alpha r \tag{14}$$

Where mass of decorticator drum + maximum number groundnut pods = m_T

α = tangential component of the angular acceleration of the decorticator drum

$$= \frac{mv^2}{r}, \text{ (normal/centripetal components being zero).} \tag{15}$$

Radius of decorticator drum + height of spikes = r

$$\therefore \text{Torque } T_d \text{ from equation (16)} = r \times \frac{mv^2}{r} \tag{16}$$

2.1.7. Motor power requirement

This involves

(i) Power required to drive the decorticator. The power required to drive the decorticator $P_d = T_d \cdot \omega$

$$\tag{17}$$

where ω is the angular velocity of decorticator drum = $\frac{2\pi N}{60}$

$$\tag{18}$$

(ii) Power required to drive the blower

The power consumption of a blower which is the function of the power required to run it can be calculated using equation (19), according to (5), which is derived from adiabatic compression equation. This was modified by (6) as

$$P_w (kW) = \frac{Q_{air} P_1}{17.4 e_M e_B} \left[\left(\frac{P_2}{P_1} \right)^{0.283} - 1 \right] \tag{19}$$

Here, the following particulars apply for the blower chosen

- e_M : Motor efficiency (0.76)
- e_B : Blower efficiency (0.78)
- P_1 : Inlet pressure, absolute (0.3kpa)
- P_2 : Outlet pressure, absolute (0.35kPa)

P_w : Power (kW)

Q_{air} : Air flow in ambient condition ($7m^3/min$)

∴ The power requirement of the blower bears the symbol P_w as in equation (19). Therefore, the total power required by the motor to run the machine should be a little above that required by both the decorticator and blower sections ie ($P_d + P_w$) watts. An overload factor of 1.23, was used to obtain the total power $P_T = 1.23(P_d + P_w)$ watts (20)

2.1.8 Motor torque

The motor torque involved in the drive may be obtained using this expression (7)

$$T_m = P_T \times \frac{9.554}{n_1} \quad (21)$$

2.1.9 Tensions in the V-belt drive of decorticator shaft

The relationship existing between the two tensions in a V-belt arrangement according to (8) is

$$2.3 \log \left(\frac{T_1}{T_2} \right) = \mu \theta \operatorname{cosec} \beta \quad (22)$$

Where T_1 = tension on the tight side of the belt

T_2 = tension on the slack side of the belt

x = centre distance between the two shafts = 350mm = 0.350m.

r_2 = decorticator sheave diameter

r_1 = motor sheave diameter

$$\alpha = \text{angle of wrap} = x + \sin^{-1} \left(\frac{r_2 - r_1}{x} \right) \quad (23)$$

μ = coefficient of friction between the belt and groove side = 0.5123 (8)

$$\theta = \text{angle of contact on the motor sheave, and Now } \theta = 180^\circ - 2\alpha \times \left(\frac{\pi}{180} \right) \quad (24)$$

β = half of the groove angle of the sheaves, where the full angle of groove is measured as 32° . Thus $\beta = \left(\frac{32^\circ}{2} \right) = 16^\circ$

$$\text{From equation (14), } \left(\frac{T_1}{T_2} \right) = \log^{-1} \left(\frac{\mu \theta \operatorname{cosec} \beta}{2.3} \right) \quad (25)$$

Again, the power transmitted by this v-belt is related to tensions in this expression (8)

$$P_T = (T_1 - T_2)V_b \quad (26)$$

Where P_T = motor power and V_b = belt velocity which is given as $V_b = \left(\frac{\pi d_1 n_1}{60}\right)$ (27)

2.1.10 Decorticator shaft diameter

The shaft is made of steel having a yield stress of 240 Mpa. ASME code was used to design the diameter of the shaft for suddenly applied load with combined fatigue and shock factors in bending and torsion. Subsequently, the bending and twisting factors are $k_b = 2.0$ and $k_t = 1.5$ respectively. The diameter of the shaft may therefore be determined using maximum shear stress theory (4)

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

where d = shaft diameter

$$\tau_{\max} = \text{permissible shear stress} = 0.3 \text{ of the material yield stress or } 0.3 S_{yt} \quad (9)$$

$$= 0.3 \times 240 = 72 \text{ Mpa}$$

$$M_b = \text{maximum resultant bending moment} = 10.23 \text{ Nm}$$

$$M_t = \text{maximum working stress in shear} = 14.95 \text{ Nm (equation 16).}$$

$$K_b = \text{combined shock and fatigue factor for bending moment} = 2.0$$

$$K_t = \text{combined shock and fatigue factor for torsion} = 1.5$$

$$= \left(\frac{\pi}{16}\right) \cdot \tau_{\max} \cdot d^3 = \sqrt{(M_b \times k_b)^2 + (M_t \times k_t)^2}$$

$$\Rightarrow d^3 = \left(\frac{16 \sqrt{(10.23 \times 10^3 \times 2.0)^2 + (14.95 \times 10^3 \times 1.5)^2}}{\pi \times 72}\right)$$

4.0 RESULTS AND DISCUSSIONS

4.1 Results

Table 1- The preliminary parameters/calculations leading to the machine design.

s/n	Preliminary parameters	symbol	value	Unit
1	Average mass of one dry groundnut pod	A_m	0.77	G
2	Force required to decorticate one dry groundnut pod	F	0.94	N
3	Decorticator drum sheave diameter	d_2	200	Mm
4	Maximum number of pods through hopper per unit time	N_h	86	pods
5	Mass of maximum pods in decorticator per unit time	M_p	66.22	G
6	Total force required to decorticate all pods per unit time	F_T	80.63	N
7	Mass of decorticator drum (without pods)	M_d	1100	G
8	Mass of aluminum sheave attached to the decorticator shaft	M_s	183.5	G
9	Mass of belt	M_l	40	G
10	Centrifugal force of decorticating spikes	F_c	81.14	N
11	Linear velocity of spike regions	v	3.19	m/s
12	Angular velocity of spike regions	ω	20.95	rad/s
13	Torque possessed by decorticator spikes	T_d	14.95	Nm

Table 2- Belt and motor calculated parameters

s/n	Preliminary parameters	symbol	value	Unit
1	Belt length	B_l	1051	Mm
2	Tension on tight side	T_1	203.13	N
3	Tension on slack side	T_2	0.99	N
4	Centre distance between motor and decorticator shafts	X	350	mm
5	Angle of contact on motor sheave	θ	2.86	Rads
6	Angle of wrap	A	13.2	degree
7	Linear velocity of belt	v_d	1.05	m/s
7	Power transmitted to decorticator shaft	P_d	313	W
8	Power transmitted to blower shaft	P_w	33	W
9	Motor power with an overload factor of 1.23	P_m	425	W

Table 3- Total load on decorticator shaft`

s/n	Loads on decorticator shaft	Value	Unit
1	Weight of decorticator drum = (1.1 x 9.81)	10.9	N
2	Weight of maximum number of pods that can fall into the decorticator drum per unit time = (86 x 0.00077 x 9.81)	0.65	N
3	Weight of aluminum sheave attached to the decorticator shaft	1.8	N
4	Weight of belt = (9.81 x 0.04)	0.4	N
5	Weight of decorticator and pods (10.79 + 0.65) acting @ 0.20m from the left of shaft	11.44	N

Table 4-Vertical and horizontal load components on the decorticator shaft.

s/n	load components of the belt	Value	Unit
1	Vertical load components of the belt ($T_1 \sin 12^\circ + T_2 \sin 12^\circ$)	42.44	N
2	Weight of pulley and belt (1.8 + 0.4)	2.2	N
3	Vertical load effect at sheave side (42.44 + 2.2) N acting @ 0.4m from left.	44.64	N
4	Total vertical force V_1 (11.44)N + (44.64)N	56.53	N
5	Reaction supports at 0.05m for A and at 0.35 for B, both from the left.	-1.72, 57.8	N
5	Total moment arising from vertical load on shaft M_v	2.23	Nm
6	Horizontal forces $R_{1H} + R_{2H} = (T_1 \cos 12^\circ + T_2 \cos 12^\circ)$	199.6	N
7	Total moment arising from horizontal load on shaft M_h	9.98	Nm
8	The resultant maximum bending moment $M_b = \sqrt{(M_v)^2 + (M_h)^2}$ or $\sqrt{(2.23)^2 + (9.98)^2} M_b$	10.23	Nm
ss9	Maximum working stress in shear M_t	14.95s	Nm
10	Standard decorticator shaft diameter was selected	14	Mm
11	The horizontal reaction components, R_{AH} and R_{BH}	-33.3, 233	N

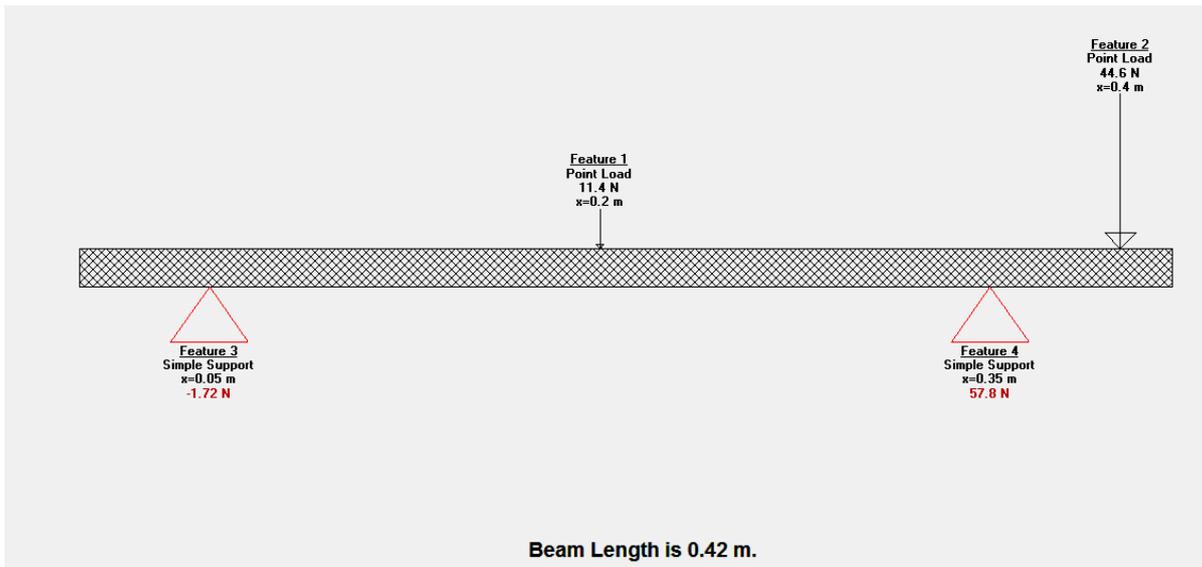


Figure 3- The decorticator free body diagram with vertical stresses

Table 5- Maximum values of vertical stresses on decorticator shaft.

Maximum Values		
	Maximum Value	Location
Bending Moment	-2.23 N-m	0.35 m
Bending Stress	22700 MPa	0.35 m
Deflection	-3060 mm	0.42 m
Slope	-1460 degrees	0.4 m

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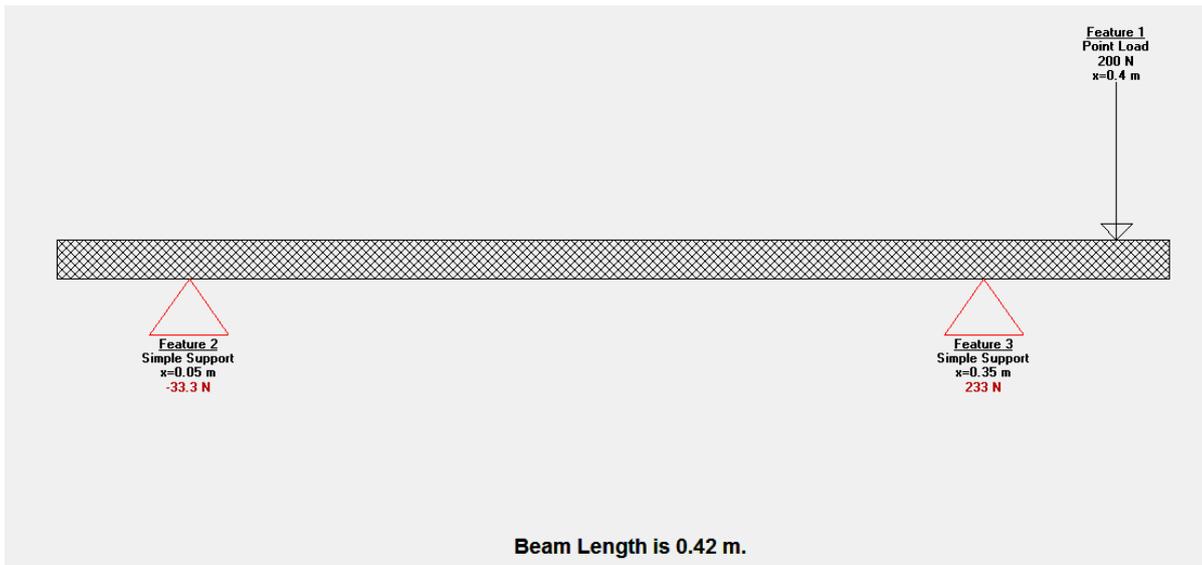


Figure 4 - The decorticator free body diagram with horizontal stresses

Table 6 - Maximum values of horizontal stresses on decorticator shaft.

Maximum Values		
	Maximum Value	Location
Bending Moment	-9.98 N-m	0.35 m
Bending Stress	102000 MPa	0.35 m
Deflection	-26000 mm	0.42 m
Slope	-9710 degrees	0.4 m

Close

4.2 Discussions

In Table 1, the preliminary parameters and calculations leading to the machine design were obtained. The total force needed to decorticate the maximum pods per unit time in the shelling unit is 80.63 N, and the force possessed by the sheller spikes is 81.14 N indicating that decortication is possible. From the motor horsepower standards of Engineering tool bar (7), a 1 horsepower, 1000 rpm motor was chosen as adequate.

Table 2 shows the calculated belt and motor parameters. The power transmitted to the decorticator and the blower are 313 and 33 W respectively, giving a total of 425 W with an overload factor of 1.23 as the total power requirement of the machine motor.

Table 3 shows the various loads on the decorticator shaft. The total weights of decorticator and pods is 11.44 N, acting at 0.20m from the left of shaft; that of the belt is 0.4 N.

Table 4 is the vertical and horizontal load components on the decorticator shaft. The total vertical force component is 56.33 N, while that of the horizontal is 199.6 N. The total vertical moment and horizontal torsional stress are 2.23 and 9.98 Nm respectively. The decorticator shaft experiences a maximum bending moment and stress in shear of 10.23 and 14.95 Nm respectively. Based on these stress values involved in the machine operation, a standard decorticator shaft diameter of 14 mm was selected.

Tables 5 and 6 show the maximum stress values on the decorticator shaft as a result of vertical and horizontal loads acting on it respectively, including the locations on the shaft where they act.

5. CONCLUSION

A simple groundnut decorticating and separating machine is designed and developed. It is portable and simple to operate. Every component of the machine was sourced locally thus giving a machine that is cheaper than the costly and complex imported heavy groundnut decorticators which are usually out of the reach of the common farmers and the local entrepreneurs.

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