Design and Fabrication of Bambara Groundnut (*Vigna subterranea (L) Verdc*) Combined Sorter, Sheller and Cleaning Machine

Negedu, A. M., Irtwange, S.V and Ijabo, O. J.

Department of Agricultural and Environmental Engineering,
University of Agriculture, P.M.B 2373, Makurdi, Benue State, Nigeria
Email of corresponding author: negeduarome@gmail.com

ABSTRACT

Postharvest processing of agricultural materials like Bambara Groundnut (BGN) is highly essential for its storability and consumption by humans. Machines need to be designed and fabricated to achieve this purpose. Consequently, some engineering properties of BGN relevant to its sorting, shelling and cleaning were determined. These properties governed the design and construction of Bambara groundnut Sorter, Sheller and Cleaning Machine that works on the principle of vibration, roller and pneumatic mechanism. The hopper, the drum shaft and the blower shaft were designed and constructed to the dimensions of 0.032 m^3 (volume), 40 mm (diameter) and 650 mm (length) respectively. The maximum vertical bending moment of 13.39 Nm and maximum horizontal bending moment of 79.95 Nm of the drum shaft were obtained from calculation. The vertical and horizontal bending moments occurred at 352.5 mm and 55 mm from the right end of the drum shaft respectively. The overall efficiency of the machine is 85.50%.

Keywords: Sorter, Sheller, Cleaning, Vibration, Roller, Pneumatic, Machine

1. INTRODUCTION

Bambara groundnut is an underutilized Africa legume which provides security to many farmers as it shows considerable drought resistance [1]. It is an indigenous African crop grown across the continent from Senegal to Kenya and from the Sahara to South Africa [2]. It is the third most important grain after groundnut and cowpea [3]. A survey conducted throughout the Middle-belt and South-Western zones of Nigeria on the shelling method used by most rural women in the villages is crushing the nuts in a mortar using pestle and this result to a lot of kernel breakages [4]. The shelling of pod to obtain clean seed is one of the most tedious operations in Bambara nut processing. As a result, it has constituted a bottle- neck to the large scale production and processing of this important proteinous crop [2]. Sometimes, this material is placed on the sack and is then beaten with stick. The separation of the shell from the nuts is by local winnowing, this method is quite tedious. Surveys amongst local communities in northern Côte d'Ivoire revealed that the Bambara groundnut seeds are mainly used for medical treatments as opposed to other parts of the plant. The seeds are used to treat anemia, ulcers (black Bambara groundnut variety mixed with an unidentified plant) and menorrhagia during pregnancy (hemostatic drink prepared by a mixture of Bambara groundnut flour and *Pupalia lappacea* (L.) Amaranthaceae dissolved in water) [5].

According to [6], machine vibration is simply the back and-forth movement of machines or machine components. Most times machine vibration is unintended and undesirable. Machine vibration can often be intentionally designed and so have a functional purpose because not all kinds of machine vibration are undesirable. For example, vibratory feeders, conveyors, hoppers, sieves, surface finishers and compactors are often used in industry, Combines, Tractor PTO's, Crop Processing, Crop Planting, Baling, Moving, Pulverizing and Fertilizing, Fans, Blowers, and Misc [7]. Causes of vibration are repeating forces, loosening and resonance. Repeating forces act on machine components and cause the machine to vibrate. Repeating forces in machines are mostly due to the rotation of imbalanced, misaligned, worn, or improperly driven machine components. There are various ways we can tell that something is vibrating. We can touch a vibrating object and feel the vibration. We may also see the back-and-forth movement of a vibrating object. Sometimes vibration creates sounds that we can hear or heat that we can sense [6].

Adigun and Oje [8] reported that nuts whose shells/pods cannot be easily broken by the roller cracker are commonly cracked using a centrifugal cracker. Oluwole *et al.* [9] developed and tested a sheanut cracker working on the principle of impaction and pneumatically separating the shells from the kernel.

Oluwole *et al.* [10] developed a centrifugal bambara groundnut sheller consisting of a feed hopper with a flow rate control device, shelling unit, separating unit and power system. This study was undertaken to design and construct Bambara groundnut sorter, sheller and cleaning machine. The machine comprised three chambers namely; sorting chamber, shelling chamber and cleaning chamber. Sorting chamber distinguished the machine and made it unique from those developed by other researchers. Sorting took place at the hopper zone to classify the pods into three groups of sizes namely; large, medium and small. The pods were conveyed into the corresponding compartments of the shelling zones with their different concave clearances and aperture. The essence of this was to boost effectiveness of the machine in terms of breakages of seed and unshelling of pods after machine operation. The machine is designed to perform better because of the inclusion of sorting and compartmentalization of shelling chamber. This paper presents the design, fabrication, assembly and description of the machine.

2. MATERIALS AND METHODS

2.1 Some engineering properties of Bambara groundnut

Relevant engineering properties of Bambara groundnut for design of the machine to sort, shell and clean it were measured. The average geometric mean diameters of Bambara groundnut large pod, medium pod and small pod are 16.78±0.49 mm, 15.15±0.49 mm and 13.24±0.56 mm respectively. The average geometric mean diameters of Bambara groundnut large seed, medium seed and small seed are 11.98±0.58 mm, 9.97±0.55 mm and 8.60±0.54 mm respectively. The geometric mean diameter of the pod in the different categories governed the choice of hole diameter on the sieves in the hopper and those of seeds governed the choice of hole diameter on the cylinder concave in the shelling chamber. The average cracking strength of Bambara groundnut pod was measured as 0.17±0.03 N/mm^2. This value was employed in calculating the power required to break Bambara groundnut pod. The average terminal velocity of the Bambara groundnut seeds was determined to be 20.19±2.75 m/s. The terminal velocity value governed the choice of fan speed. The angle of repose and static coefficient of friction for large pod on metal sheet averaged 29.2±0.79^0 and 0.56±0.02 respectively. These values helped to determine the angle at which the sieves were placed in the hopper.

2.2 Design of Machine Components

Some criteria were taken into consideration, which include: The type of load and stress caused by the load, motion of the parts of the machine. Selection of materials, for example factors governing the choice of materials which includes strength, durability, flexibility, weight, corrosion and machinability. Friction resistance and lubrication: careful attention was given to all the surfaces which move in contact with others whether in rotating, sliding or rolling bearings. Convenience and economic features in designing the operating features was carefully analyzed. Some other factors considered were safety of operations, cost of production, workshop facilities and final assembly

[11]. The components that were designed include the following: Hopper, Shelling chamber, Drum Shaft, Fan Shaft, Belt drive and Frame.

2.2.1 *Hopper*

The hopper was meant for feeding in Bambara groundnut pods into the device. This component is designed into three sections to sort Bambara groundnut pods into three (3) different particle sizes. This was constructed by principle of development of shapes from metallic materials based on its suitability. The hopper was made of two volumetric shapes; cylindrical and conical (see Figure 1), the first cylindrical section had two sieves fixed in it.

The volume of Section 1 was calculated as 29965254 mm^3using Equation (1). The volume of section 2 was calculated with the combination of equations 2 that is 1799318.67 mm^3, therefore the total volume of hopper was calculated to be 31764572.67 mm^3

$$v_1 = \pi r^2 H \quad (1)$$

where,
r = radius of the cylinder = 170 mm, H = height of the cylinder = 330 mm

$$V_2 = v_{a+b} - v_b \quad (2)$$

$$v_{a+b} = \frac{1}{3}\pi r^2 h \;\; = v_b$$

where,
r = radius of cone
h = height of cone

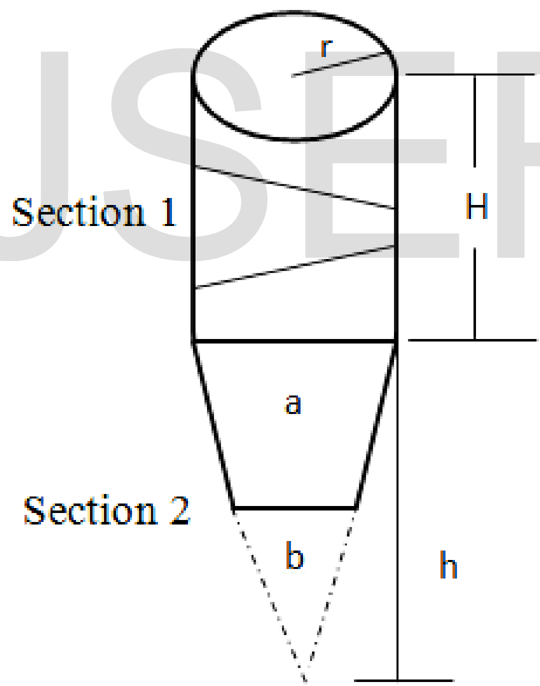

Figure 1: Schematic Representation of the Hopper

2.2.2 *Shelling Chamber*

The shelling chamber consists of the cylinder concaves and the shelling drums.

Weight of the shelling drum: According to Hannah and Hillier [12], the weight of the shelling drum can be calculated using Equation (3).

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$$W_c = \rho_c V_c \quad 3$$

$$V_c = \pi D_c L_c t_c$$

ρ_c = density of cylinder material = 7860 kg/m^3
V_c = volume of cylinder
π = acceleration due to gravity = 9.81 m/s^2
D_c =diameter of cylinder =165 mm = 0.165 m
L_c = length of cylinder = 175 mm = 0.175 m
t_c = thickness of cylinder 2 mm = 0.002 m
Therefore,
$V_c = 1.81 \times 10^{-4}$ m^3
W_{sd} = 13.99 N

Weight of the spikes on the drum: The area of a single spike on the drum was calculated using Equation (4):

$$A_s = 2\pi r(h + r) \quad 4$$

r = 5 mm
h = 25 mm
$A_s = 9.426 \times 10^{-4}$ m^2
$V_s = 9.426 \times 10^{-4} \times 0.025 = 2.357 \times 10^{-5}$ m^3
$M = \rho \times V_s$
M = 0.182 kg
W_s = 1.79 N
For 7 spikes on a single anchor and a total of 4 anchors on a drum, the total weight of spikes therefore = 1.79 × 7 × 4 = 50.05 N.
Therefore, the total weight of a single shelling drum, in one of the three compartments,
W_{sd} = 13.99 + 50.05 = 64.04 N.

Shear failure analysis of shelling drum: The shearing stress of the pressing drum is defined by Equation (5):

$$\tau = \frac{16M_t}{\pi d^3} \quad [11] \quad 5$$

For belt drive, the torque is found from

$$M_t = (T_1 - T_2)R \quad 6$$

where
τ = shearing stress =?
M_t = torsional moment
d = core diameter of pressing drum = 163 mm = 0.163 m
T_1 = tension in tight side = 1220.05 N
T_2 = tension in slack = 231.07 N
R = radius of drum pulley = 135 mm = 0.135 m
Therefore,
M_t = 133.5 Nm
τ = 162913.89 N/m^2

Power requirement of the machine: The total power requirement of the machine is the sum of the power required to drive the blower and the shelling drum and to shell the Bambara groundnut pod shown in Equation (7):

$$P = P_F + P_D + P_S \quad 7$$

(a) Power to drive the fan, P_F

The power required to drive the fan was obtained from the following expression

$$P_F = \frac{Q \times P_s}{\eta} \quad 8$$

Where
P_F = power required to drive the fan
Q = fan flow rate 0.11 m^3/s
P_s = static pressure of fan = 336 N/m^2
η = static efficiency of fan = 75% = 0.75
Therefore,
P_F = 49.28 W = 0.066 hp, using the factor of safety of 1.5 gives P_F =0.099 hp

(b) Power to drive the shelling drum, P_D

$$P_D = T_D \omega_D \quad 9$$

But $T_D = W_D R_D$
Where
W_D = weight of shelling drum = 64.04 N
R_D = radius of shelling drum = 165 mm = 0.165 m
ω_D = angular velocity of shelling drum = $\frac{2\pi N}{60}$
N = 200 rpm
ω_D = 20.95
Therefore,
T_D = 10.57 Nm
P_D = 221.36 W = the power to drive a single drum out of the three drums. Therefore, power to drive the three fixed drums = 221.36 × 3 = 664.07W =

0.89 hp. Using the factor of safety of 1.5 gives P_D =1.34 hp

(c) Power to shell Bambara groundnut pods, P_S

Following the procedure of Ikechukwu *et al.* [13] for designing power requirement to shell groundnut pods; the design proceeds as follows.
The power required in breaking the pods in the shelling chamber may be obtained from;

$$P_S = T \times \omega_D \tag{10}$$

Considering a single spike, the torque required to drive the system may be obtained from the following expression:

$$T = \eta_a \times \eta_s \times F \times r \tag{11}$$

$$\omega_D = \left(\frac{2\times\pi\times N_D}{60}\right)$$

Where,
T = torque required
η_a= number of active anchor at a time = 4
η_s= number of spike per anchor = 7
F = force per spike required to break the groundnut pod
r = distance from the axis of rotation to the point of action of the force = 0.015m
ω_D = angular velocity of drum
N_D = speed of rotation of the shelling drum = 200 rpm
The average force required to break the Bambara groundnut pod was measured to be approximately equal to 2 N.
Therefore,
$T = 0.84\ Nm$
ω_D = 20.95 m/s
P_S = 17.59 W = 0.024hp × 3 = 0.07 hp, using the factor of safety of 1.5 gives P_S = 0.11 hp
Therefore, the total power required to drive the machine is $0.099 + 1.34 + 0.11 = 1.55\ hp.$
However, 5.0hp motor was chosen because it was intended to contribute to machine vibration for effective sorting of the material.

2.2.3 Drum Shaft

A shaft is a rotating machine element used to transmit power from one place to another that is from the electric motor to the cylinder. According to the American Society of Mechanical Engineers (ASME) code for the design of transmission shafts, the maximum permissible working stresses (σ_u) in tension or compression may be taken as; (a) 112 MPa for shafts without allowance for keyways (b) 84 MPa for shafts with allowance for keyways
Maximum permissible shear stress (τ_u) may be taken as; (a) 56 MPa for shafts without allowance for keyways (b) 42 MPa for shafts with allowance for keyways.

(a) Design of Drum Shaft based on Strength

The shaft design consists of the determination of the correct shaft diameter to ensure satisfactory strength and rigidity when the shaft is transmitting power under various operating and loading conditions. Mild steel was used for the construction of shaft diameter, therefore the following equation apply.

$$d^3 = \frac{16}{\pi S_s}\sqrt{(K_b M_b)^2 + (K_t M_t)^2} \quad [11]$$

Where
S_s = maximum permissible shear stress

$$S_s = \frac{ultimate\ strength\ in\ shear}{factor\ of\ safety} \tag{12}$$

K_b = combined shock and fatigue factor applied to bending moment
K_t = combined shock and fatigue factor applied to torsional moment
M_b = maximum bending moment
M_t = torsional moment
For shaft having keyway and load gradually applied, K_b = 1.5 and K_t = 1.0 [11]. Assuming factor of safety of 1.5, the working stresses for shaft with key is as given in equations 13 and 14.

$$\sigma_w = \frac{\sigma_u}{FS} = \frac{84MPa}{1.5} = 56\text{MPa} \tag{13}$$

$$\tau_w = \frac{\tau_u}{FS} = \frac{42MPa}{1.5} = 28\text{MPa} \tag{14}$$

Vertical Loading of Shaft

$$W_T = W_{d1} + W_{d2} + W_{d3} + W_p \qquad (15)$$

Where,
W_T = total vertical load
W_{di} = weight of drum = 64.04 N
W_p = weight of drum pulley = 86.45 N
Therefore,
W_T = (3 × 64.04) + 86.45 = 278.57 N
From Figure 2 (a)
$\sum F_V = \uparrow + = 0$:
$R_{AV} + R_{BV} = W_{d1} + W_{d2} + W_{d3} + W_p = 278.57N$

$M_A \circlearrowleft + = 0$
$0.595R_{BV} = 64.04 \times 0.0975 + 64.04 \times 0.2975 + 64.04 \times 0.4975 + 86.45 \times 0.65$
$R_{BV} = 190.5$ N
$R_{AV} = 88.07$ N
The free body diagram and the shear force diagram of the shaft is shown in Fig. 6.

Vertical Bending Moment
From Figure 2 (a),
At F: $M_{FV} = 0$ Nm
At B: $M_{BV} = 86.45 \times 0.055 = 4.75$ Nm
At E:
$M_{EV} = 86.45 \times 0.1525 - 190.5 \times 0.0975 = -5.35$ Nm

At D:
$M_{DV} = 86.45 \times 0.3525 - 190.5 \times 0.2975 + 64.04 \times 0.2 = -13.39$ Nm

At C:
$M_{CV} = 86.45 \times 0.5525 - 190.5 \times 0.4975 + 64.04 \times 0.4$[illegible]$64.04 \times 0.3$[illegible]$= 8.59$ Nm

At A: M_{DV} = 0Nm

Horizontal Loading of Shaft
The total horizontal load (belt tension) action on the pulley at point D is given by;
$T_T = T_1 + T_2$ = 1451.12 N
From figure 3 (a)
$\sum F_H \uparrow = + = 0$:
$$R_{AH} + R_{BH} = T_1 + T_2 = 1451.12 \qquad (16)$$

$M_A \circlearrowleft \rightarrow + = 0$:
$0.595R_{BH} = 1451.12 \times 0.65$
$R_{BH} = 1585.26$N
$R_{AH} = -134.14$ N

Horizontal Bending Moment
At F: M_{FH} = 0Nm
At B: $M_{BH} = 1451.12 \times 0.055 = 79.81$ Nm
At E:
$M_{EH} = 1451.12 \times 0.1525 - 1585.26 \times 0.0975 = 66.73$

At D:
$M_{DH} = 1451.12 \times 0.3525 - 1585.26 \times 0.2975 = 39.90$

At C:
$M_{CH} = 1451.12 \times 0.5525 - 158.26 \times 0.4975 = 13.08$

At A: M_{DH} = 0Nm
The vertical and horizontal bending moment diagram is obvious in Fig. 3.

Resultant bending moment
$$M_b = \sqrt{M_V{}^2 + M_H{}^2} \qquad (17) \quad [11]$$

At B : $M_b = \sqrt{4.75^2 + 79.81^2}$
= 79.95 Nm
At E : $M_b = \sqrt{5.35^2 + 66.73^2}$
= 66.94 Nm
At D : $M_b = \sqrt{13.39^2 + 39.90^2}$
= 42.09 Nm
At C : $M_b = \sqrt{8.59^2 + 13.08^2}$
= 15.65 Nm
From calculation, the maximum bending moment will occur at B, that is:
M_{bmax} = 79.95 Nm
Therefore,
d = 31.96 mm

(b) Design of Drum Shaft based on Torsional Rigidity

Design of shaft for torsional rigidity is based on the permissible angle of twist. The amount of twist

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permissible depends on the particular application and varies from 0.3 deg/m for machine tool shafts to about 3 deg/m for line shafting.
This is obtained from;

$\theta = \frac{584\, M_t L}{G\, d^4}$ *(for solid shaft)*

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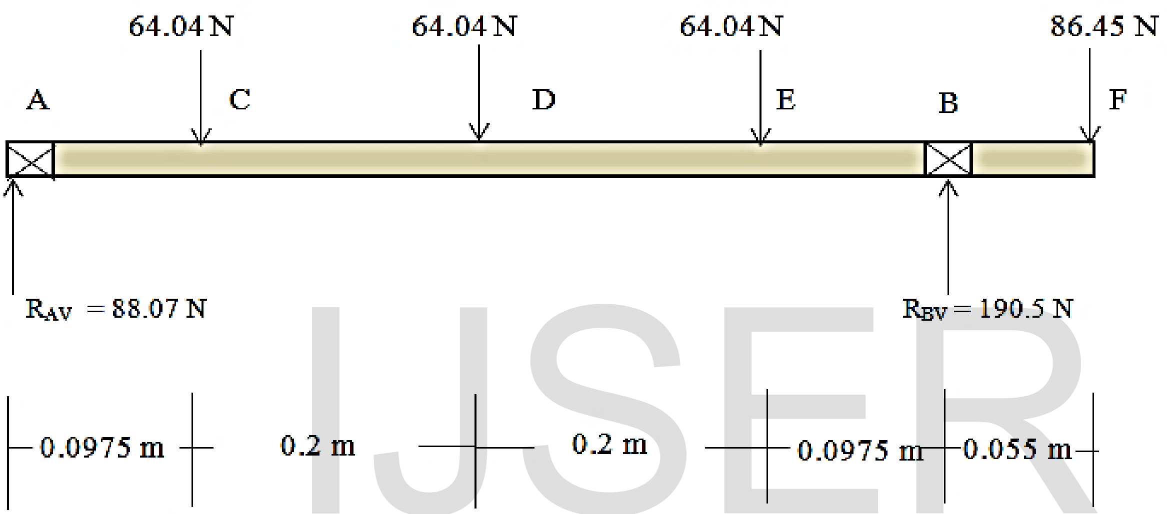
Or $d^4 = \frac{584 M_t L}{G\theta}$ [11]

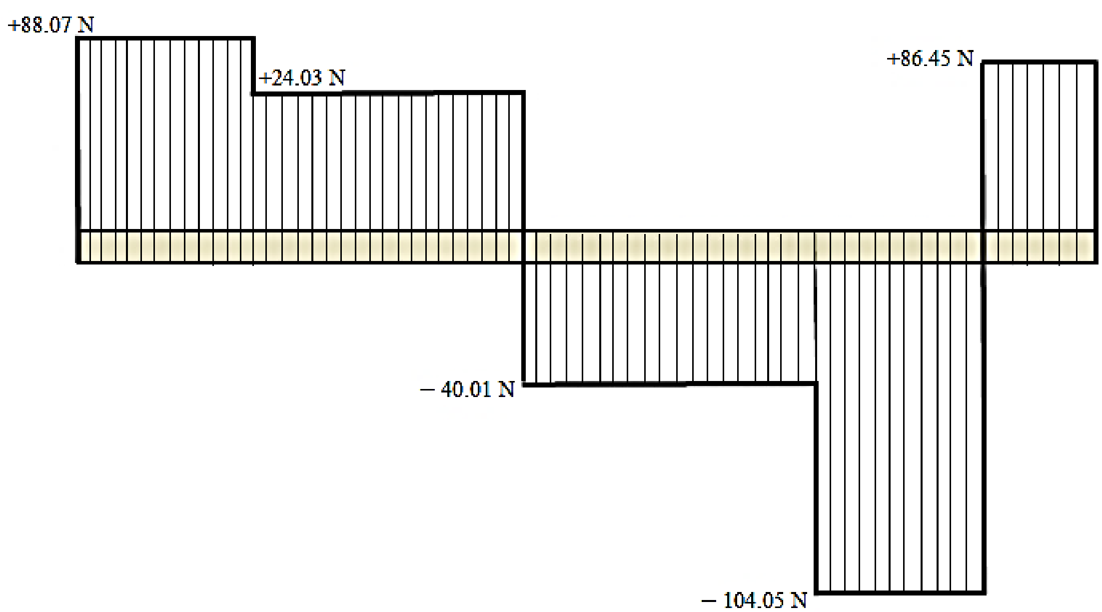
Where,
θ = angle of twist, deg.
L = length of shaft. = 650 mm = 0.65 m
M_t = torsional moment = 133.5 Nm
G = torsional modulus of elasticity = 80 GN/m^2
d = shaft diameter
Therefore,
d = 38.11 mm
The summary of shaft diameter is:

- Based on strength, *d* = 31.96 mm
- Based on torsional rigidity, *d* = 38.11 mm

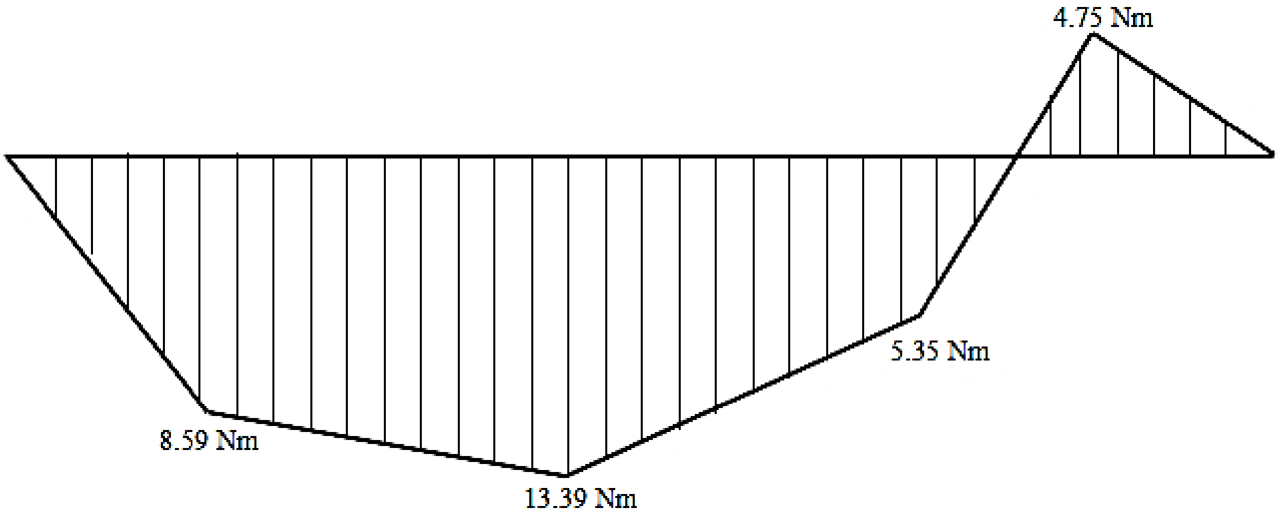
In order to satisfy all conditions of design, shaft diameter of 40 mm was chosen.



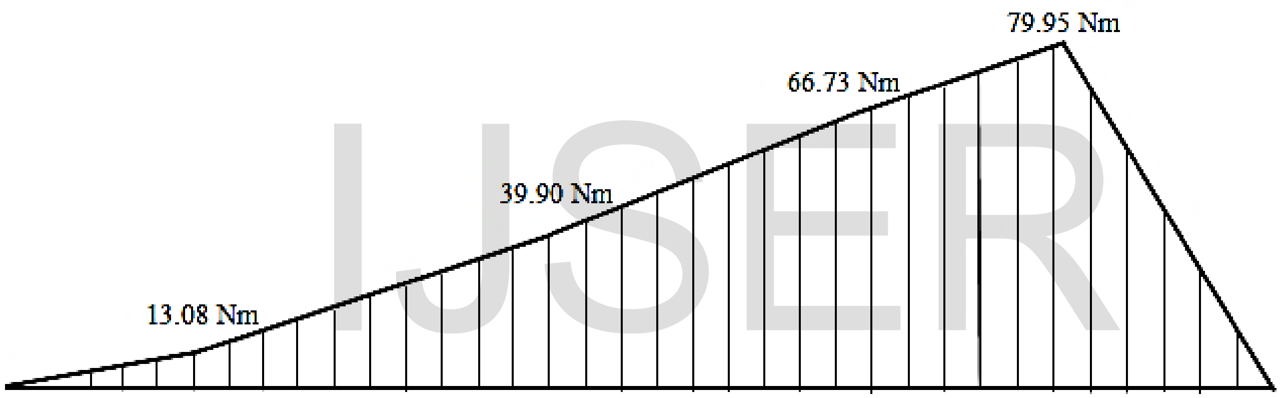
(a)



(b)
Fig. 2: (a) Free Body Diagram (b) Shear force diagram



(a)



(b)

Fig. 3: (a) Vertical Bending Moment Diagram (b) Horizontal Bending Moment Diagram

2.2.4 Fan Shaft

Axial fan is selected for this design. Weight acting on the fan are weight of the blades and weight of pulleys.

Weight of blade,

$W_b = length \times breath \times thickness \times density \times 9.81 \times no.of\ blades$

$l = 0.5\ m$

$b = 0.1\ m$

$t = 0.0012\ m$

$\rho = 7860\ k/m^3$

No of blades = 4

W_b = 18.51 N

- Weight of pulley = 7.62 N + 17.15 N = 24.77 N

Total weight acting on the fan shaft = 18.51 N + 24.77 N = 43.28 N

(a) Design based on Torsional Rigidity

Design of shaft for torsional rigidity is based on the permissible angle of twist. The amount of twist permissible depends on the particular application

and varies from 0.3 deg/m for machine tool shafts to about 3 deg/m for line shafting.
This is obtained from;

$\theta = \frac{584\, M_t L}{G\, d^4}$ *(for solid shaft)*

Or $d^4 = \frac{584 M_t L}{G\theta}$ [11]

Where,
θ = angle of twist, deg.
L = length of shaft. = 650 mm = 0.65 m
M_t = torsional moment = ?
$M_t = (T_1 - T_2)R$
R = 0.1m
T_1 = 1220.05 N
T_2 = 231.07 N
M_t = 98.90 Nm
G = torsional modulus of elasticity = 80 GN/m^2
d = shaft diameter
Therefore,
d = 35.37 mm
The diameter of fan shaft selected is 40 mm

2.2.5 **Belt Drive**

The procedure for selecting a V-belt drive is dependent on the motor horse power and the speed (rpm) rating. V-belts are rated from class A to E [14]. Figure 4 shows belt drive arrangement.

(a) Velocity Ratio of V-belt Drive

The velocity ratio of belt drive is given by;

$V_{R1} = \frac{N_B}{N_M} = \frac{D_M}{D_B}$ 19

$V_{R2} = \frac{N_D}{N_B} = \frac{D_B}{D_D}$ 20

Where
N_B = Speed of blower = ?rpm
N_M = speed of motor = 1440 rpm
N_D = speed of drum = ?rpm
D_M = diameter of motor pulley = 50 mm
D_B = diameter of blower pulley = 80 mm
D_D = diameter of drum, 360 mm

$V_{R1} = \frac{N_B}{1440} = \frac{50}{80}$

N_B = 900rpm

$V_{R2} = \frac{N_D}{900} = \frac{80}{360}$

N_D = 200 rpm
Class B V-belt was chosen for design. This is because, the power of electric motor is 3.75 kW (because, 1 hp = 745.69 W, therefore, 5 hp = 3728.45 W = 3.75 kW).

(b) Weight of Pulleys

We consider the pulley as a small cylinder with height, h and diameter, d. The pulley was cast iron with density of 7730 kg/m^3.
Therefore, weight of pulley;

$W_p = \rho_p V_p$ [12] 21

But, $V_p = \pi r_p^2 h_p$

Where,
ρ_p = density of pulley = 7730 k/m^3
V_p = volume of pulley
r_p = radius of pulley
h_p = height of pulley
π = acceleration due to gravity = 9.81 m/s^2

(c) Weight of Motor Pulley

r = 25 mm = 0.025 m
h = 20 mm = 0.02 m
V_p = 0.000039275 m^3
W_p = 2.978N

(d) Weight of Blower Pulleys

First Pulley: r = 40 mm = 0.04 m; *h* = 20 mm = 0.02 m; $V_p = 1.00544 \times 10^{-4}$ m^3; W_p = 7.62 N
Second Pulley: r = 60 mm = 0.06 m; *h* = 20 mm = 0.02 m

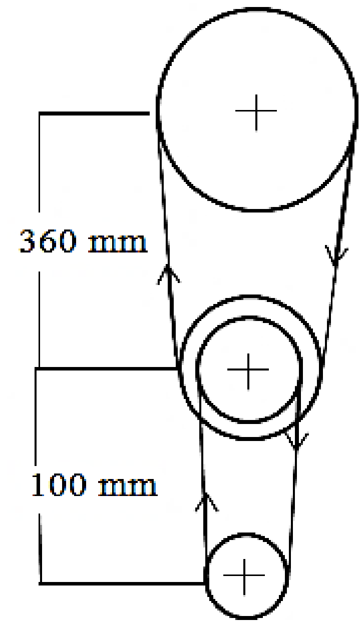


Figure 4: Diagram of the Belt Drive arrangement

2.2.6 *Design of Frame*

Rigidity and strength are the most important criteria to be considered on the frame design. The frame design involves knowing the kind of loading that it was subjected to. Selecting the correct steel sections for the frame construction and analyzing all possible forms of failure that could occur on the frame to ensure safety of the design. The main frame was made of mild steel material.

Size of Frame

The total weight acting on the frame is the of weight of blower pulleys, weight of drum pulley, weight of drum, weight of bearings, weight of hopper, weight of shaft, weight due to tension, weight of screens (sorter) and weight of screen (concave). The following assumptions was considered; (a) The frame is fixed (b) The total load acting on the frame is uniformly distributed (c) The frame is to be constructed from 35 $\times$ 35 mm angle iron.

Total Weight on Frame

Weight of hopper = 59.79 N, Weight of screens = 3.7 N, Weight of pod chutes = 9.54 + 5.58 + 3.95 = 19.07 N, Weight of drum = 64.04 × 3 = 192.12 N, Weight of pulleys = 2.98 + 7.62 + 17.13 + 86.45 = 27.73 N
Weight of bearings = 17.88 N
Weight of shaft = 61.95 N
Weight of screen (concave) = 0.09 N
Therefore, the total weight on the frame = 382.33 N
The frame design is based on Euler's column assumption [15]. The assumptions are: (a) The cross-section of the column is uniform throughout its length (b) The column is perfectly straight initially and the load applied is truly axial (c) The column material is perfectly elastic, homogeneous and isotropic, and thus obeys Hooke's law. (d) The length of the column is very high as compared to its cross-sectional dimensions. (e) The shortening of column due to direct compression, which is very small is neglected (f) The failure of column occurs due to buckling alone (g) The weight of the column itself is neglected
A standard angle iron of the following dimension was used.
Width = a = 35 mm
Thickness, t = 3 mm
The angle iron was designed based on the formula given by Ryder, [16] in Equation 22:

$$A = t(2a - t) \qquad 22$$

Where,

A = area of section
a = width = 35 mm
t = thickness = 3 mm
Therefore,
A = 201 mm^2
Distance from the neutral axis to the extended fibre, y is given as

$$y = a - \frac{a^2 + at - t^2}{2(2a - t)} \qquad 23$$

Substituting a and t gives
y = 34.22 mm
Moment of inertia, I

$$I = \frac{1}{3}[ty^3 + a(a-y)^3 - (a-t)(a-y-t)^3] \qquad 24$$

Substituting a, t and y gives
I =40194.15 mm^4
Section modulus Z is calculated from

$$Z = \frac{I}{y}$$

Therefore,
Z = 1174.58 mm^3
Radius of Gyration is calculated from

$$K = \sqrt{\frac{I}{A}} \qquad 25$$

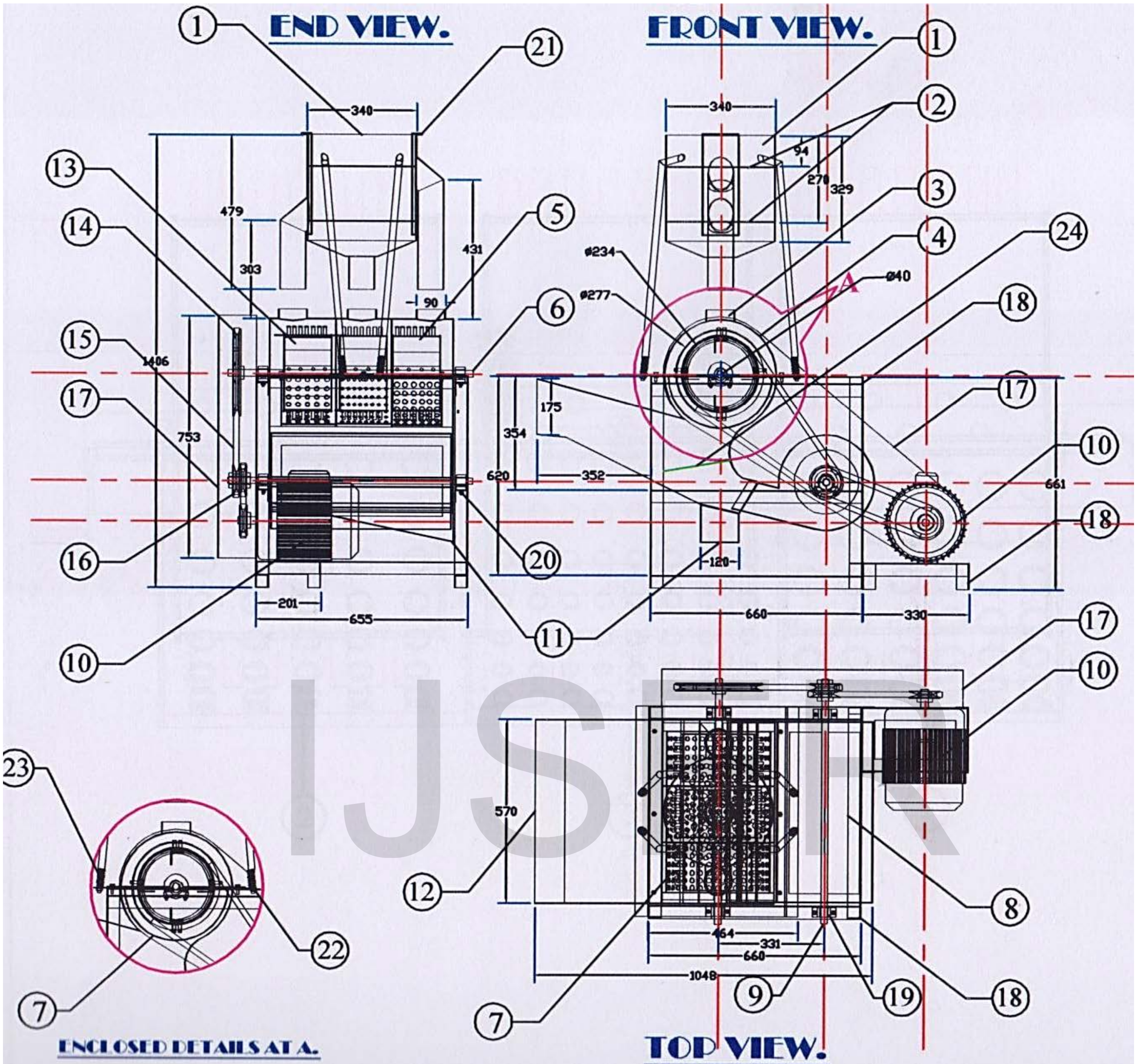
Therefore,
K = 14.14 mm
According to Euler's theory, the crippling or buckling load (W_{CR}) under various load end conditions is represented by a general equation given as

$$W_{CR} = \frac{C\pi^2 EI}{l^2} = \frac{C\pi^2 EAK^2}{l^2} \qquad 26$$

Where,
W_{CR} = crippling or buckling load
C = constant, representing the end conditions of the column or end fixity coefficient = 0.25 (one end fixed and the other end free) [14].
E = modulus of elasticity or young modulus for the material of the column = 210 GPa = 210000 N/mm^2
$I = AK^2$ = moment of inertia = 2842.14 mm^4
A = area of cross-section = 201 mm^2
K = least radius of gyration of the cross-section 14.14 mm
L = length of the column =620 mm
Therefore,
W_{CR} = 54185.5 N
This shows that the load that would cause crippling or buckling is 54185.5 N but the total load acting on the frame is 382.33 N. This indicates that the dimensions of the frame chosen are appropriate to bear the load without crippling or buckling.

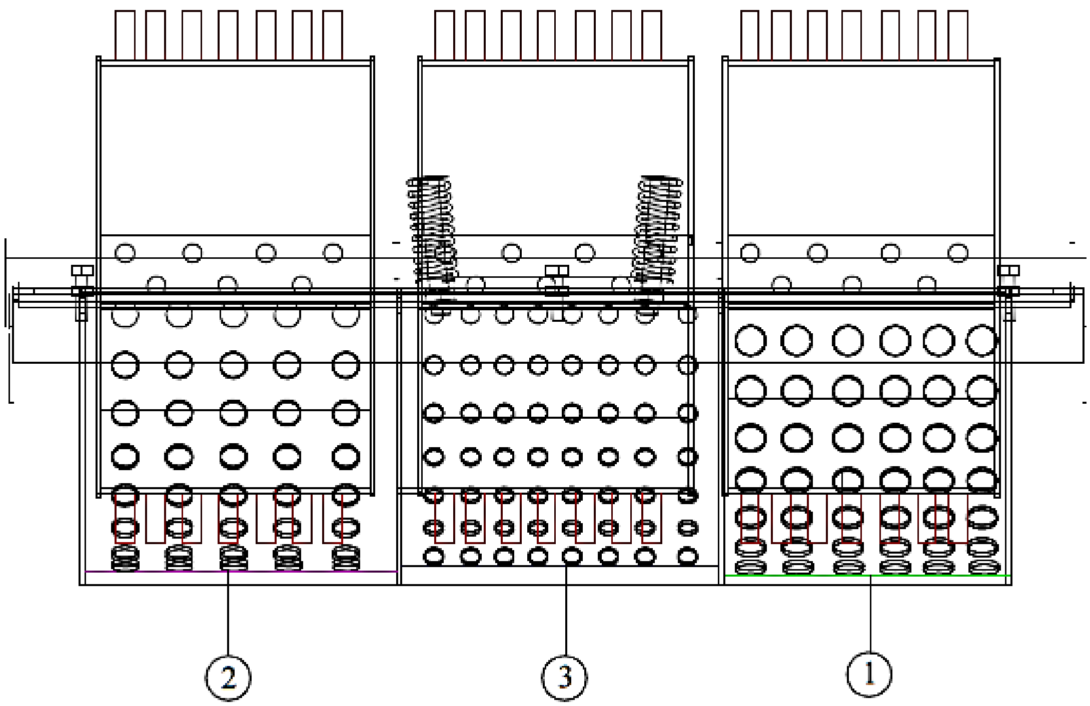
2.3 *Fabrication and Assembly of the Machine Components*

The machine was fabricated with the various types of engineering materials as shown in Table 1. The orthographic projection of the machine is obvious in Figure 5. Figure 6 shows the clearance of the machine. More so, the exploded view of the machine is obvious in Figure 7. All dimensions are in mm.

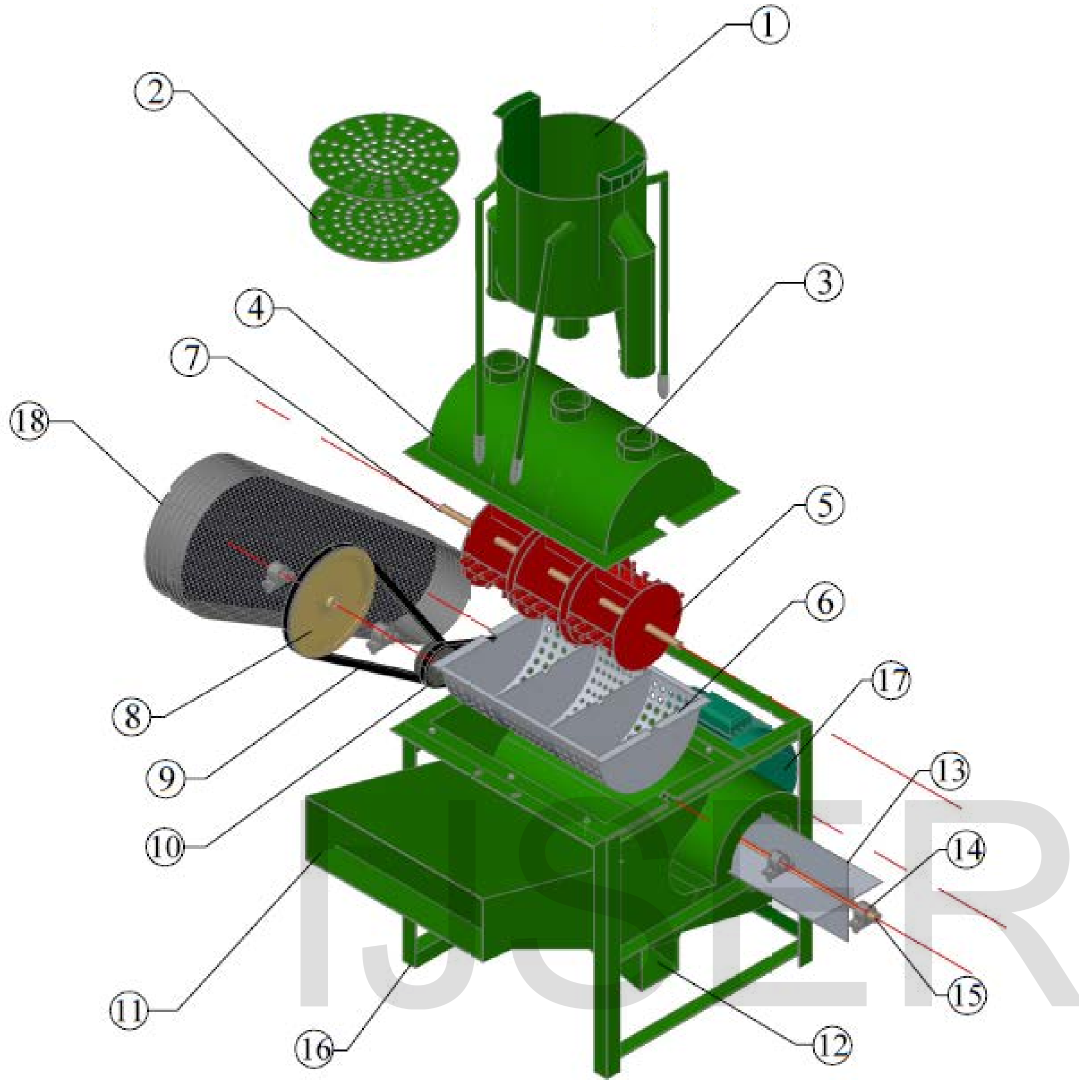


(1) Hopper, (2) Perforated screen (3) Inlet port for pods (4) Shelling Chamber (5) Spikes (6) Drum Shaft (7) Perforated Concave Screen (8) Fan Blade (9) Fan Shaft (10) Electric Motor (11) Seed Outlet (12) Chaff outlet (13) Anchor (14) Drum Pulley (15) V-Belt (16) Double groove blower pulley (17) Belt drive guard (18) Frame (19) Industrial bearing (20) M12 bolt and nut (21) Feed Control Device (22) M17 nut (23) Compressible Spring (24) Slanted screen for pods/chaff flow

Fig. 5: Orthographic Drawing of the Machine



(1) Large shelling zone (2) Medium shelling zone (3) Small shelling zone
Fig. 6: Concave Clearance of the different Shelling Zones



(1) Hopper (2) Perforated screens (3) Inlet port for pods (4) Top cover of shelling Chamber (5) Shelling drum (6) Concave frame (7) Drum shaft (8) Drum pulley (9) V-belt (10) Double groove blower pulley (11) Chaff outlet (12) Seed Outlet (13) Fan blade (14) Fan shaft (15) Industrial bearing (16) Frame (17) Electric Motor

Fig. 7: Exploded View of the Machine

2.4 Description of the Machine

According to Commtest [6], repeating forces in rotating machine component causes machine vibration. This theory governed the choice of attaching about 12.75N of metal rod to the inner side of the drum of central shelling zone which caused the drum to wobble in all directions and consequently increased the vibration of the sieves fixed in the hopper. The machine was designed to operate on the principle of vibration, roller and pneumatic mechanisms. The machine consists of a feed hopper, shelling unit, cleaning unit and power system. A cylindrical hopper of three sections was suspended on the frame over the shelling unit with the aid of four spiral springs. The two sieves fixed in the first and the second hopper sections were perforated based on the principal dimensions of the pod and inclined at the pod's angle of repose.

The first two hopper sections have flow rate control devices fixed at the side, the devices were used to obtain gate closing to allow proper sorting and gate opening for the discharge of pods from the hopper into the different shelling zones. Three chutes conveying pods were connected from the hopper to the inlet ports on the shelling unit; two chutes being from the hopper side and the third terminates at the hopper base. The end of the chutes was made free to allow the hopper to take advantage of the mechanical vibration of the machine brought about by the wobbling action of the shelling drum. This aided proper sorting of the pods into three predetermined pod sizes. The shelling system was divided into three compartments each having different rollers and three different cylinder concaves with different clearances which were perforated according to the principal dimensions of the predetermined pod sizes. The rollers were arranged on a single shaft and were driven by a belt and pulley arrangement. Each of the rotary rollers has four (4) anchor and twenty eight (28) spikes welded to it. The clearance that is less than the predetermined pod size but greater than the seed size was created with the fixed cylinder concave in the various compartments.

During machine operation, the pods from the chutes dropped between the rotary roller and the cylinder concave. The shelling bar (spikes) of the rotary roller collected the pods and compress them against the fixed cylinder concave and by that action the pods were shelled. The mixture of seed and shell fell under the influence of gravity and machine vibration through the opening at the base of the shelling unit onto a slanted metal surface placed beneath the shelling unit. The mixture of seeds and chaffs flowed on this surface down to the winnowing zone where the air pressure blew away chaffs from the pure seeds.

The winnowing system consists of the winnowing chamber (where the seed and shell are separated by air current set little below the terminal velocity of the seed) with the aid of the blower which was powered by an electric motor. When the mixture of seed and shell fell through the winnowing chamber, the horizontal air current from the blower lifted the lighter shells and carry them out through the shell outlet. The power system consists of an electric motor, belt and pulley drives by which the machine was actuated. The components was assembled and mounted on a support frame which was fabricated from 35 mm x 35 mm angle iron. Table 2 shows the details of the materials and specification.

Table 2: Materials and Specification

S/N	Items	Material/ Specification	Quantity
1.	Electric motor	5.0 hp	1
2.	Bolts and nuts	Mild steel (M18)	12
3.	Roller drum	Mild steel Ø165×175 mm	3
4.	Drum Anchor support	Angle iron 35× 35 mm	1
5.	Shaft	Mild steel Ø40×650 mm	2
6.	Ball bearings	-	4
7.	Roller drum/blower housing	Mild steel Ø240×510 mm	3
8.	Shelling bar/spike	Cylindrical metal bar	3
9.	Blower blade	Metal sheet Ø300×640 mm	4
10.	Hopper	Mild steel Ø340×330 mm	1
11.	Pods conveying chutes	Mild steel Ø80×200 mm	2
12.	Seed collection chute	Mild steel 100×50×140 mm	1
13.	Chaff collection chute	Mild steel 345×250 mm	1
14.	Wire net	Soft iron 630×150 mm	2
15.	Drive belt	Leather	3
16.	Drive pulley	Cast iron Ø50×20 mm	1
17.	Drum pulley	Cast iron Ø220 ×20 mm, Ø270×20mm, Ø320×20 mm,	3
18.	Blower pulley	Cast iron Ø80×20 mm, Ø60×20mm	2
19.	Spiral spring	Steel 10 × 40 mm	4
20.	Tool frame	35 × 35 mm angle iron	2
21.	Feed rate control devices	Mild steel 80×90 mm	3
22.	Cylinder concave	Mild steel 175×270 mm	3
23.	Abrasives	2 mm thick (smooth)	1
24.	Paint	Green paint	I tin

2.5 Design Summary and Operational Considerations

The summary result of the machine design is obvious in Table 3. For a mechanical machine like bambara groundnut combined sorter, sheller and cleaning machine to work efficiently, to meet its service demand and be durable; certain factors must be taken into consideration before, during and after its operation. Some of these factors that can boost the efficiency and durability of the machine can be highlighted in the following ways. (a) The moving parts should be lubricated to minimize friction (b) The machine should be properly coated to avoid rust (c) The storage environment of the machine should be well ventilated to minimize the rate of corrosion (d) The machine should be cleaned after operation

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Table 3: Design Summary of the Machine

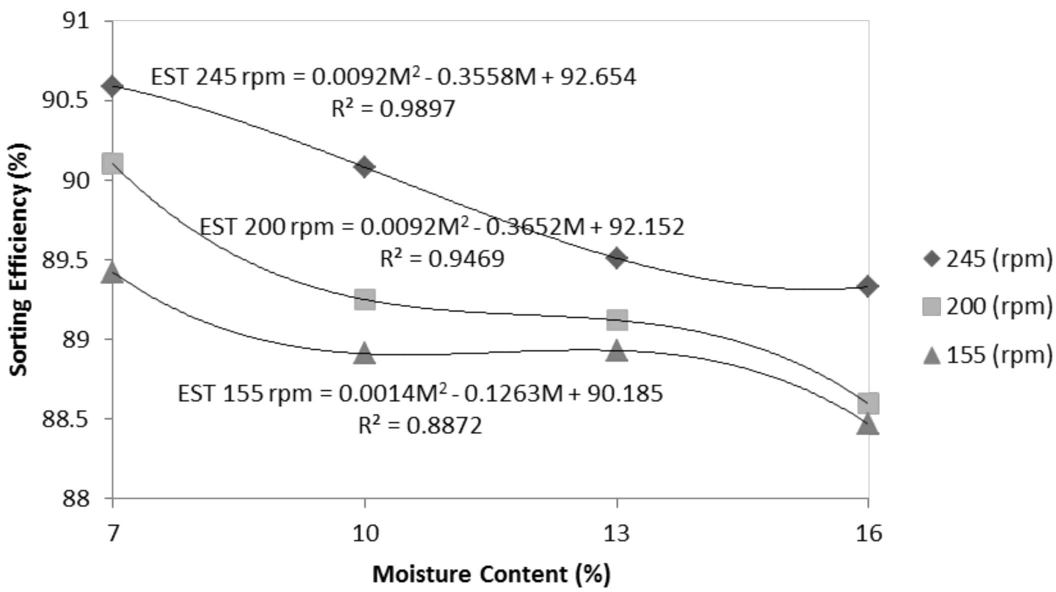
S/N	Components	Volume (m^3)	Weight (N)	Diameter (m)	Length (m)	Height (m)
1.	Hopper	0.032	78.860	0.340	-	0.479
2.	First Sorter Screen	-	3.700	0.016[a]	-	-
	Second Sorter Screen	-	3.700	0.014[a]	-	-
3.	Concave Screens L		0.090	0.014[a]	-	-
	M		0.090	0.012[a]	-	-
	S		0.090	0.010[a]	-	-
4.	Concave Clearance L	-	-	-	-	0.017
	M	-	-	-	-	0.015
	S	-	-	-	-	0.013
5.	Belt Length: M-B	-	-	-	0.406	-
	B-D	-	-	-	1.099	-
6.	Motor Pulley	-	2.980	0.050	-	0.020
7.	Blower pulleys: a.	-	7.620	0.080	-	0.020
	b.	-	17.150	0.060	-	0.020
8.	Drum pulley	-	86.450	-	-	0.020
9.	Shelling drum L	-	64.040	0.165	0.175	-
	M	-	64.040	0.165	0.175	-
	S	-	64.040	0.165	0.175	-
10.	Drum shaft	-	61.95	0.04	0.65	-
11.	Fan shaft	-	61.95	0.04	0.65	-
12.	Key	-	-	0.005	0.05	-
13.	Bearings	-	4.47	0.040	-	0.020
14.	Frame	-	-	-	0.660	0.620
15	Machine	-	573.50	-	0.990	1.406

[a] = aperture, L = Large Size Zone, M = Medium Size Zone, S = Small Size Zone, M-B = Motor Pulley to Blower Pulley and B-D = Blower Pulley to Drum Pulley.

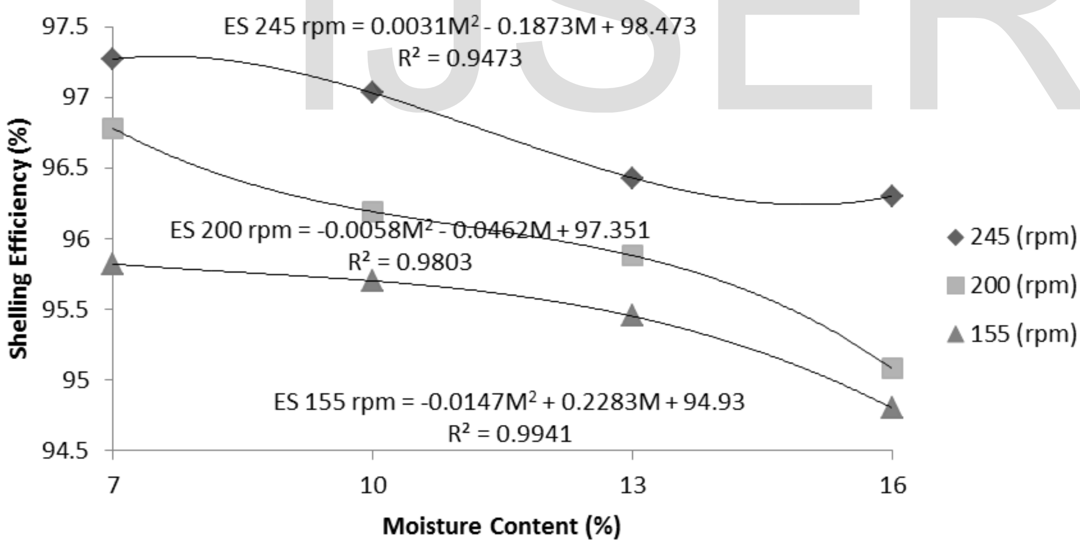
3. Result and Discussion

3.1 Result

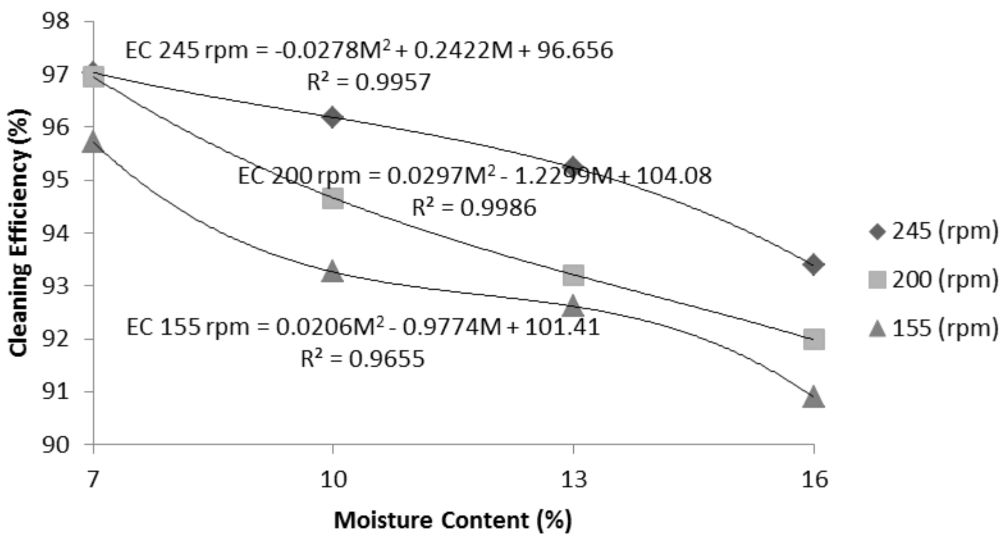
The result of the performance evaluation is shown in Figure 8.


EST 245 rpm = 0.0092M² - 0.3558M + 92.654
R² = 0.9897
EST 200 rpm = 0.0092M² - 0.3652M + 92.152
R² = 0.9469
EST 155 rpm = 0.0014M² - 0.1263M + 90.185
R² = 0.8872
Sorting Efficiency (%)
Moisture Content (%)
245 (rpm)
200 (rpm)
155 (rpm)

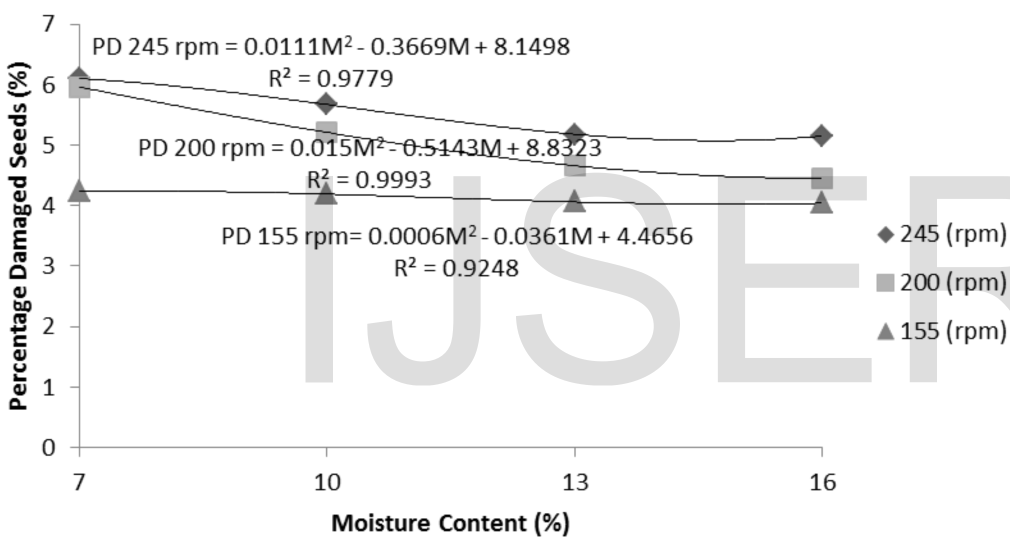
(a)


ES 245 rpm = 0.0031M² - 0.1873M + 98.473
R² = 0.9473
ES 200 rpm = -0.0058M² - 0.0462M + 97.351
R² = 0.9803
ES 155 rpm = -0.0147M² + 0.2283M + 94.93
R² = 0.9941
Shelling Efficiency (%)
Moisture Content (%)
245 (rpm)
200 (rpm)
155 (rpm)

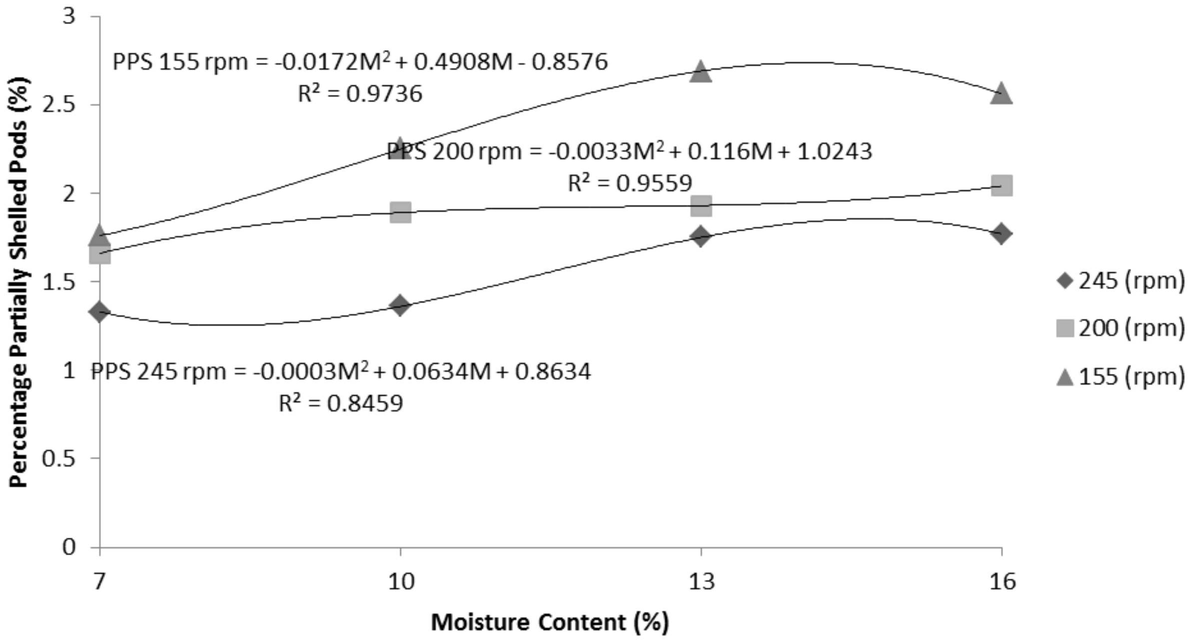
(b)


EC 245 rpm = -0.0278M2 + 0.2422M + 96.656
R² = 0.9957
EC 200 rpm = 0.0297M2 - 1.2299M + 104.08
R² = 0.9986
EC 155 rpm = 0.0206M2 - 0.9774M + 101.41
R² = 0.9655
Cleaning Efficiency (%)
Moisture Content (%)
245 (rpm)
200 (rpm)
155 (rpm)

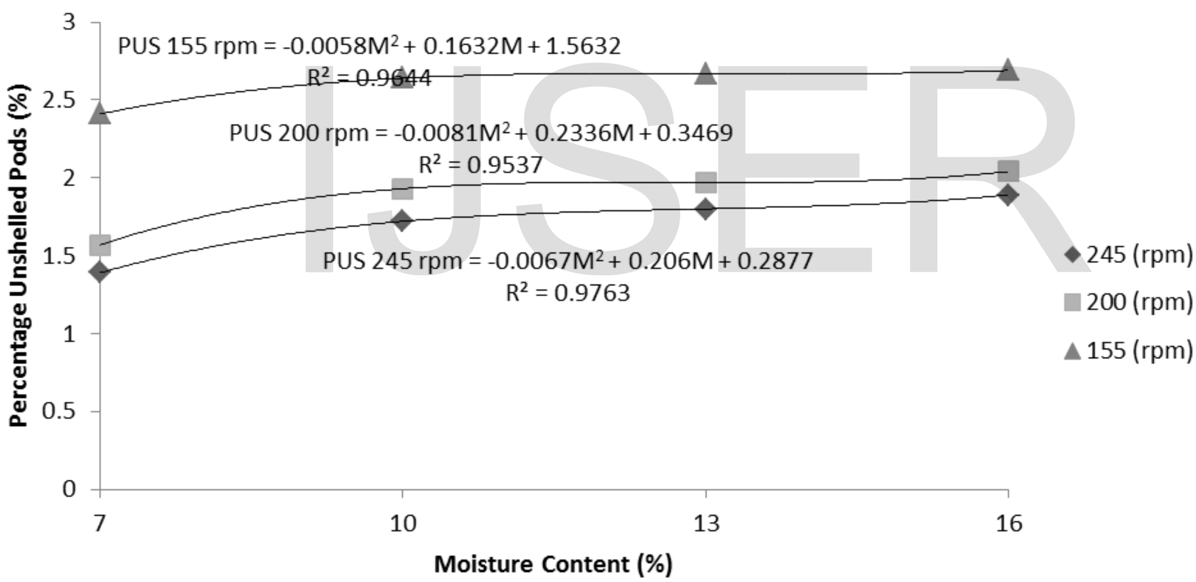
(c)


PD 245 rpm = 0.0111M2 - 0.3669M + 8.1498
R² = 0.9779
PD 200 rpm = 0.015M2 - 0.5143M + 8.8323
R² = 0.9993
PD 155 rpm= 0.0006M2 - 0.0361M + 4.4656
R² = 0.9248
Percentage Damaged Seeds (%)
Moisture Content (%)
245 (rpm)
200 (rpm)
155 (rpm)

(d)



(e)



(f)

Figure 8: Characteristic curve of the effect of moisture content and shaft speed variations on (a) sorting efficiency of the machine; (b) shelling efficiency of the machine; (c) cleaning efficiency of the machine; (d) percentage damaged seeds after machine operation; (e) percentage partially shelled pods after machine operation; and (f) percentage unshelled pods after machine operation.

3.2 Performance tests of the machine

Firstly, an unaccounted mass of Bambara groundnut pods was fed into the machine and shelled; this was done to eliminate machine material loss which could affect the evaluation of performance indicators. After this, full hopper equivalent to 2815 number of Bambara groundnut pods at specified moisture content were fed into the first section of the hopper. The mechanical vibration of the machine enhanced by addition of weight to the middle shelling drum and the spiral spring on which the hopper was suspended made the pods to sieve into the predetermined (three) categories of pod sizes.

The small size pods were allowed to fall directly into the shelling zone ahead of the large and medium size pods. This was done to optimize the power required to drive the shelling drum. After size sorting of the pods was achieved, the flow rate control devices for large and medium pods were opened and the pods were discharged freely under the influence gravity and vibration into their respective shelling chamber. The pods were shelled until the shelling chamber was emptied. The average time for batch operation of the machine at the various run was noted to be 3 min.

The number of pods that were completely shelled and unbroken (N_1), completely shelled but broken (N_2), completely shelled but partially broken (N_3), partially shelled and unbroken (N_4) and the number of unshelled pods (N_5) was determined at the end of each run. The quantity of shells that was cleaned out was collected. The quantity of shells that was not cleaned but collected with the seeds was separated. The test was conducted at four desired moisture content of the material and at three different shaft speeds. At a combination of each of the above conditions, the test was replicated three times and the performance of the machine was evaluated on the basis of the following indices and their values were obtained according to the formulae.

(a) Sorting efficiency, $E_{ST} = \left[\frac{N_T - N_{CT} - N_2 - N_4 - N_5}{N_T}\right] \times 100\%$

(b) Shelling efficiency, $E_S = \left(\frac{N_1 + N_2 + N_3}{N_T - N_{CT}}\right) \times 100\ \%$

(c) Cleaning efficiency, $E_C = \left(\frac{M_{cs}}{M_{Ts}}\right) \times 100\ \%$

(d) Percentage damage, $p_D = \left(\frac{N_2}{N_T - N_{CT}}\right) \times 100\ \%$

(e) Percentage of partially shelled pods, $p_{PS} = \left(\frac{N_4}{N_T - N_{CT}}\right) \times 100\ \%$

(f) Percentage of unshelled pods, $p_{US} = \left(\frac{N_5}{N_T - N_{CT}}\right) \times 100\ \%$

where,
M_{cs} = Mass of cleaned shells
M_{Ts} = Total mass of shells
N_{CT} = Total Number of pods clogged on the screen after machine operation.
N_T = Total Number of pods fed into the hopper of the machine.

3.3 Performance evaluation of the machine

The performance efficiency of machines for processing agricultural materials depend on various factors such moisture content of the material, the speed of the shelling drum, the fan speed and feeding rate. Four levels of moisture content (7, 10, 13 and 16%) and three speeds of the drum (155, 200 and 245 rpm) were considered in the performance evaluation of the machine. The performance parameters are sorting efficiency, shelling efficiency, cleaning efficiency, percentage damaged seeds, percentage partially shelled pods and percentage unshelled pods.

3.3.1 Sorting Efficiency

The sorting efficiency ranged from minimum of 88.47% to a maximum of 90.59%. The sorting efficiency decreased with increase in moisture content because the pod size and weight of the material tends to increase with increase in moisture content and as such, the material clogged more on the screen. Generally, increase in shaft speed resulted to increase in sorting efficiency. It was observed that about 98% of the two seeds per pods of the Bambara groundnut present in the sample sorted were retained on the first sieve and were conveyed into the large zone of the shelling chamber where they were shelled with the beating effect of spikes welded to the rotating drum. It can also be seen from the characteristic curve in Figure 8(a) that at the shaft speed of 245 rpm, the sorting efficiency of the machine decreased with increase in moisture content of the biomaterial. The trend is the same for speed of 200 rpm and 155 rpm. The reason for decrease in sorting efficiency can be blamed on the fact that the pod size and weight of the material tends to increase with increase in moisture content and as such, the material clogged more on the screen. Generally, increase in shaft speed resulted to increase in sorting efficiency. It was observed that about 98 % of the two seeds per pods of the Bambara groundnut present in the sample sorted were retained on the first sieve and were conveyed into the large zone of the shelling chamber where they were shelled.

3.3.2 Shelling Efficiency

Maximum shelling efficiency of 97.27% was obtained at 245 rpm shaft speed and 7% (wb) moisture content. This value shows an improvement over the values obtained by; Adedeji and Danladi [17] as 83.20%, Atiku *et al.* [2] as 80.00%, Bishiri *et al.* [18] as 93.15% and Simonyan, [18] as 95.00%. This is because sorting of pods was integrated into the machine to categorize the pods into their individual shelling zones and this is what caused the shelling efficiency of the machine to be higher than those of the authors cited. This value is lower than the cracking efficiency of 100% obtained by Oluwole *et al.* [9] when radially positioned vanes impeller was used to crack Sheanut. Ikechukwu *et al.* [13] obtained 95.25 % shelling efficiency for groundnut. Additionally, in Figure 8(b), the shelling efficiency of the machine increased with increase in shaft speed from 155 rpm to 245 rpm and decreased with increase in moisture content from 7 % (wb) to maximum of 16% (wb). This is in line with Atiku *et al.* [2] and Aremu *et al.* [20]. This is due to the fact that the increase in shaft speed increased the rate of crushing of the material and also increase in moisture content of the material reduced its brittleness which is a property that enhances the crackability of the pods.

3.3.3 Cleaning Efficiency

Maximum cleaning efficiency of 97.03% was obtained at 245 rpm and 7% (wb) moisture content. This value is almost exactly the same with 97% cleaning efficiency obtained by Oluwole *et al.* [9] for Sheanut cracker. But shows improvement over those reported by Yusuf and Suleiman [20], Adedeji and Danladi [17], Atiku *et al.* [2] and Bishiri *et al.* [18] as 87.00%, 78.9%, 79.50% and 85.33% respectively for Bambara groundnut. Higher cleaning efficiency of 98.00% was reported by Simonyan, [19]. This is because blower speed was considered as a factor in his machine performance test. Figure 8(c) shows that the cleaning efficiency decreased with decrease in shaft speed and increase in moisture content. The reason is because the speed of the blower was kept constant, the beater rotated at less speed and the hulls from the pods gained more weight, thereby making some shells to escape the air pressure line and followed the seed through the seed collection chute. However, the result is in line with [22] and [23].

3.3.4 Percentage Damaged Seeds

Minimum percentage damaged seed of 4.04% was obtained at moisture content of 16% (wb) and 155 rpm shaft speed. This result shows improvement over those reported by Atiku *et al.* [2], Bishiri *et al.* [18] and Simonyan, [18] as 20.00%, 4.90% and 4.70% respectively. The result is better than those of the authors cited because three different clearance zones were integrated into the machine under study to control the effect of size variations of the pods. But the results of percentage broken seeds of 0.2% reported by Adedeji and Danladi [17] for Bambara groundnut and 1.1% and 2.8% reported by Ashish and Handa [24] for electrical operation and manual operation respectively for groundnut are better. Maduako *et al.* [24] reported percentage seed damage of 12.4% when they tested their Cayor groundnut sheller. Furthermore, Figure 8(d). It is obvious that at all levels of shaft speed, percentage damaged seeds decrease with increase in moisture content from of 7 % (wb) to maximum of 16 % (wb. This is due to the fact that the increase in moisture content of the material reduced its brittleness which is the property that enhances the ease with which the seed is cracked. Generally, more seeds were broken at higher shaft speed and lower moisture content. Maximum value of 6.10 % at 245 pm shaft speed and 7 % (wb) moisture content was obtained confirming the fact that decrease in moisture content and increase in shaft speed increased the percentage damaged seeds after machine operation.

3.3.5 Percentage Partially Shelled Pods

Minimum percentage partially shelled pod of 1.33% was obtained at the moisture content of 7% (wb) and shaft speed of 245 rpm. This value shows a significant improvement over the 10% percentage partially shelled pods reported by Atiku *et al.* [2]. The percentage partially shelled pods generally decreased with increase in shaft speed and increased with increase in moisture content. The geometric mean diameter of the partially shelled pods that were collected at the end of machine operation was found to range from 9.80 to 10.50 mm. The characteristic curve of the percentage partially shelled pods is shown in Figure 8(e). It can be seen that the triumvirate shaft speed of 245 rpm, 200 rpm and 155 rpm with respect to increase in moisture content did not produce a linear curve for percentage partially shelled pods. This irregularity is traceable to the fact that some pods might have followed through a wrong channel into a wrong shelling zone, thereby causing the pods to fall through the perforation without fully partaking of the beating effect of the shelling drum. This is why it can be seen from the curve that, at low shaft speed, more pods were partially shelled. The percentage partially shelled pods generally decreased with increase in shaft speed and increased with increase in moisture content. The geometric mean diameter of the partially shelled pods that were collected at the end of machine operation was found to range from 9.80 to 10.50 mm.

3.3.6 Percentage Unshelled Pods

Minimum percentage unshelled pods of 1.39% and maximum of 2.67% were obtained in this study. This value demonstrated improvement when compared with those reported by Bishiri *et al.* [18] and Atiku *et al.* [2] as 2.7% and 7.0% respectively;. But it is not better than 0.68% obtained by Adedeji and Danladi [17]. This can be attributed to the speed of shaft (360 rpm) used in the testing of their machine. The geometric mean diameter of the unshelled pods portrayed the same range with those partially shelled as stated above. It was discovered that the unshelled pods were only one seed per pod. Furthermore, the bambara seeds either for two seeds per pod or one seed per pod fell individually through the aperture of the concave screen after shelling. Figure 8(f) presents the characteristic curve of the percentage unshelled pods. Generally, increase in moisture content increased percentage unshelled pods, also increase in shaft speed decreased percentage unshelled pods. The geometric mean diameter of the unshelled pods portrayed the same range with those partially shelled as stated above. It was discovered that the unshelled pods were only one seed per pod. Furthermore, the Bambara seeds either for two seeds per pod or one seed per pod fell individually through the aperture of the concave screen after shelling.

3.4 Overall efficiency of the machine

The overall machine efficiency (E_M) was calculated from the average maximum values of sorting efficiency, shelling efficiency and cleaning efficiency. This is as expressed below:

$$E_M = \left[\frac{sorting\ efficiency\ (\%)}{100} \times \frac{shelling\ efficiency\ (\%)}{100} \times \frac{cleaning\ effi}{100}\right.$$

$$= \left[\frac{90.59\%}{100} \times \frac{97.27\%}{100} \times \frac{97.03\%}{100}\right] \times 100$$

$= 0.854998 \times 100$

$= 85.50$ %.

3.5 Throughput Capacity

The average one thousand pod weight was measured to be 1.5 kg. That is the average weight of each pod is 0.0015 kg. Therefore, for 2815 pods fed into the machine, the average weight is 4.22 kg. The average time of machine operation was 3 minutes. Consequently, the throughput capacity = 4.22kg/3min = 84.40 kg/hr.

4. CONCLUSION

The Bambara groundnut combined sorter, sheller and cleaning machine was designed and fabricated in accordance to standard methods. Some measured engineering properties of Bambara groundnut relevant for its sorting, shelling and cleaning aided in the design. The overall efficiency of the machine was calculated to be 85.50% and the throughput capacity of 84.40 kg/hr was obtained. Although the machine is capable of reducing drudgery in processing the biomaterial; the overall efficiency and throughput capacity show that there is need for more improvement. Therefore, vibration by reciprocating mechanism and increasing the size of the shelling chamber can be considered in order to improve or perfect the efficiency of the machine.

REFERENCES

[1] Brough S.H., Azam-Ali S.N., and Taylor A.J., (1993). The potential of bambara groundnut (*Vigna subterranean*) in vegetable milk production and basic protein functionality systems. Food Chem., p. 47, pp. 277-283.

[2] Atiku, A. A., Aviara, N. A., and Haque, M. A., (2004). Performance evaluation of a Bambara groundnut sheller. *Agricultural Engineering. International: CIGR Journal of Scientific Research and Development.* Manuscript P 04002, VI Pp. 1-18.

[3] Ezeaku, C.A., (1994). Fracture Characteristics of Bambara groundnut in compressive loading. M.Sc. Eng. Thesis, Department of Agricultural Engineering, University of Nigeria, Nsukka. Pp. 9-42.

[4] Olaniyan, M. A. and Oje, K. (1999). Viscoelastic-Behaviour of Shea Nuts Under Application of Force. A Paper Presented at the 21st Annual Conference of the Nigerian Society of Agricultural Engineers (NSAE) Technical Transactions P.4 and Pp. 50 – 57.

[5] Koné. M., Paice, A.G. and Touré, Y (2011) Bambara groundnut [*Vigna subterranea* (L.) Verdc. (Fabaceae)] usage in human health. In: *Nuts and Seeds in Health and Disease Prevention* (edited by V.R. Preedy, R.R. Watson & V.B. Patel). UK: Elsevier Inc. p. 192-194.

[6] Commtest the revolution (2006). Beginner's Guide to Machine Vibration. *Commtest Instruments Ltd* 28-b Moorhouse Avenue Christchurch New Zealand. Pp 3-6.

[7] Garrett S. (2012). Agricultural Rotating Machinery. Vibration Identification and Solutions. Vibration Industrial Balancing and Equipment Services Corporation 720-999 W Broadway, BC V5Z 1K5. P 4.

[8] Adigun, Y. J. and Oje, K. (1993). Thevetia fruit processing: energy requirements for cracking the nut. *Proceeding of Nigerian Society of Agricultural Engineers*, 15: 128-139.

[9] Oluwole, F. A., Aviara N.A., and Haque M.A., (2004). Development and performance tests of a sheanut cracker. *Journal of Food Engineering.* p. 65, pp. 117-123.

[10] Oluwole, F.A., Abdulrahim A.T., and Olalere R.K., (2007). Effect of moisture content on crackability of Bambara groundnut using a centrifugal cracker. *International. Agro Physics*, 21, 179-184.

[11] Bhandari, V. B. (1994). Design of Machine Elements. *Tata McGraw-Hill Publishing Company Limited, New Delhi.* Pp. 331-334, 499-501.

[12] Hannah, J. and Hillier, M. J. (1999). Applied Mechanics. Third Edition. Pearson Education Limited, Harlow, England.

[13] Ikechukwu, C. U, Olawale, J. O. and Ibukun, B. I. (2014). Design and Fabrication of Groundnut Shelling and Separating Machine. *International Journal of Engineering Science Invention. Volume 3 Issue 4. Pp. 60-66.*

[14] PSG Tech. (1989). Design Data. Faculty of Mechanical Engineering, Poolamedu Sathankulam Govindsamy Naidu College of Technology (PSG Tech), DPV Printers, Coimbatore, India. Pp. 1-25.

[15] Khurmi, R. S. and Gupta, J. K. (2007). Mechanical Engineering-conventional Objectives. *Eurasia Publishing House Limited*, Ram Nagar, New Delhi. Pp 522-556

[16] Ryder, G. H, (1983). Strength of Materials. Third Edition. SI Units, London, English *Language Book Society and Macmillan Press Limited*. Pp. 71-256.

[17] Adedeji, M. A. and Danladi, W. Y. (2016). Modification and construction of motorized Bambara nut decorticating machine. *Global Advanced Research Journal of Engineering, Technology and Innovation* (ISSN: 2315-5124) Vol. 5(1) pp. 001-006.

[18] Bashiri, I. M., Zubairu M. and Olorunaye, E. S. (2013). Design and Construction of a Machine for decorticating Bambara nut**.** *Journal of Research in National Development*, 11(2): 596-8303.

[19] Simonyan, K. J. (2006). Development and performance evaluation of a motorized Bambara groundnut (*vigna subterranean)* sheller. *Journal of Agricultural Engineering and Technology* (JAET), 14: 46-51.
[20] Aremu, D. O., Adewumi, I. O. and Ijadunola, J.A. (2015). Design, fabrication and performance evaluation of a motorized maize shelling machine. *Journal of Biology, Agriculture and Healthcare.* Vol.5, No.5: 2224-3208
[21] Yusuf, D. D and M. L. Suleiman(2004). Design, Development and Performance Evaluation of Bambara Groundnut Sheller, *Journal of Agricultural Technology* (JAT), 8(1): 42-52.

[22] Agidi, G. Ibrahim, M. G. and Matthew S. A. (2013). Design, fabrication and testing of a millet thresher. *Net Journal of Agricultural Science.* Vol. 1(4), pp. 100-106, ISSN: 2315-9766
[23] Afify, M. K., El-Sharabasy, M. M. A. and Ali, M. M. A. (2007). Development of a local threshing machine suits for threshing black seed (*Nigella sativa*). *Misr Journal of Agricultural Engineering*, 24(4): 699-724.

[24] Ashish, S. R and Handa, C. C. (2014). Design and fabrication of Groundnut sheller Machine. *International Journal for Innovative Research in Science & Technology*, 1: 38-44.
[25] Maduako, J. N., Saidu, M., Mathias, P. and Vanke, I. (2006). Testing of an engine-powered groundnut shelling machine. *Journal of Agricultural Engineering Technology,* 14: 29-37