Design and Construction of Dried Cassava Pellets Grinding Machine

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ABSTRACT : The use of grinding machine is one of the simplest methods of processing agricultural raw materials alternative to the traditional methods of using stone, mortar and pestle. Grinding process reduces the size of solid materials by mechanical action, and it achieves this by dividing them into smaller particles. Grinding of agricultural products is one of the oldest cultural techniques of humanity. In this research work, design and construction of dried cassava pellets grinding machine was carried out. The dried cassava grinding machine is made up of the following component parts; electric motor, main frame, pulley, transmission belt (V-belt), shaft, bearing, and vibratory tray sieve. The summary of results obtained from design calculation shown that; velocity of 5.54m/s, power of 1177.1W, torque of 10.16Nm were required for the operation of the machine. Also, the maximum deflection (0.00647mm) obtained is negligible relative to the length of belt (225mm) and the diameter of the shaft (25mm). This implies that the shaft will retain its ability to function optimally under the applied total transverse load (338.77N). Moreover, static failure analysis was carried out on the machine using SolidWorks CAD modeling. The results obtained show that the Von Mises stress is less than the stress required to cause yielding. Therefore, the design is safe.

KEYWORDS - Design, construction, dried cassava, pellets, machine, torque, power

I. INTRODUCTION

Cassava, (Manihotesesculenta, Crantz) is a tuberous starchy root crop of the family Euphorbiaceae [1]. It is a food crop, known worldwide for drought tolerance and for thriving well on marginal soils [2]. Nigeria is presently the largest producer of cassava in the world with an annual output of over 34 million tonnes of tuberous roots [3]. It is majorly classified as sweet or bitter (manihotutilissimaomanihot palmate) cassava respectively [4]. According to Olukunle [5], cassavaproduction is needed in several areas; for enhanced food security, means of foreign exchange and tool for rapidindustrialization. However, the drudgery in processing cassava can be minimized or eliminated through adequate mechanical processing [6].

The use of grinding machine is one of the simplest methods of processing agricultural raw materials alternative to the traditional methods of using stone, mortar and pestle [7]. Grinding process reduces the size of solid materials by mechanical action, and it achieves this by dividing them into smaller particles [8]. Grinding of agricultural products is one of the oldest cultural techniques of humanity. As a result of size reduction, processing, and storage, farmers were forced to develop technology for grinding their produce. The most extensive application of grinding in the food industry is in the milling of the cassava pellets to make flour, but it is equally used in many other processes, such as in the grinding of corn, for the manufacture of corn starch, grinding of millet, grinding of millet. There are usually two different methods used in effecting size reduction of dried cassava pellets. The grinding carried out by pounding via mortar and pestle, and the grinding done by crushing between two stones via grinding stone). The method of the pestle and mortar is widely used in the West-African country. However, the traditional method of grinding stone, pestle and mortar is time consuming.
and tasking. The traditional method is very laborious, and it is hard work for anyone to grind more useful quantity in a short period of time. To solve the problem of grinding dried cassava pellets traditionally, a mechanical method via the use of grinding machine was used in this research work. Grinding machines are machines that use the principles of abrasion, compression, attrition/shearing, impact or friction forces to effect size reduction in Agricultural raw materials. The basic principle behind most of our local grinding machines is friction. In order to effect size reduction, the two frictional surfaces of the grinding machines have to come together to crush the material between them [9].

II. MATERIAL AND METHOD

The dried cassava grinding machine is made up of the following component parts, which includes:
   (a) Electric motor
   (b) Main frame
   (c) Pulley
   (d) Transmission belt (V-belt)
   (e) The shaft
   (f) The bearing
   (g) Vibratory tray sieve

2.1 Design Calculation

2.1.1 Speed Ratio of Belt Drive

Velocity ratio for belt drive is the ratio between the velocity of the driver and the driven. It may be expressed mathematically as:

\[ \frac{N_2}{N_1} = \frac{D_1}{D_2} \] (1)

Where:
- \( D_1 \) = diameter of the driver = 75mm
- \( D_2 \) = diameter of the follower = 150mm
- \( N_1 \) = speed of the driver = 1440rpm
- \( N_2 \) = speed of the follower = ?

Therefore;

\[ N_2 = \frac{(1440 \times 75)}{150} = 705 \text{rpm} \]

2.1.2 Velocity of Belt

\[ v = \frac{d_1}{2} \times \frac{2\pi N_2}{60} \] (2)

\( d_1 = 0.075 \text{m} \)
\( N_2 = 1410 \text{rpm} \)
\[ v = \frac{0.15}{2} \times \frac{2 \times \pi \times 705}{60} \]
\[ v = 5.54 \text{m/s} \]

2.1.3 Length of V-belt

\[ L = \pi (r_2 + r_1) + 2x + \frac{(r_2 - r_1)^2}{x} \] (3)

\[ L = \pi (0.0375 + 0.075) + 2(0.225) + \frac{(0.0375 - 0.075)^2}{x} \]
\[ L = 0.8 \text{m} \]

2.1.4 Center to Center Distance between Pulleys

The centre to centre distance is given by Equation (4)

\[ C = D_1 + D_2 \] (4)

Taking center to center as
- \( C = 150 + 75 \)
- \( C = 225 \text{mm} \)
2.1.5 Angle of Lap or Contact on Smaller Pulley
\[
\theta_1 = \pi - \left( \frac{D_2 - D_1}{C} \right) = 2.81 \text{rad} = 160.98 \text{degree}
\] (5)

2.1.6 Angle of Lap or contact on Large Pulley
\[
\theta_1 = \pi + \left( \frac{D_2 - D_1}{C} \right) = 3.47 \text{rad} = 198.79 \text{degree}
\] (6)

2.1.7 Cross-sectional Area of Belt
\[
A = \frac{1}{2} (X + Y)H
\] (7)

X = 13mm
H = 8mm
Y = ?

Q = \tan 15 \times 8 = 2.14
Y = X - 2(Q)

A = \frac{1}{2} (13 + 8.72)8
A = 86.88 \text{ mm}^2

2.1.8 Torque Transmitted by Shaft
\[
T = \frac{P \times 60}{2 \pi \times N}
\] (8)
P = 1500W
N_1 = 1410rpm
T = \frac{1136.59 \times 60}{2 \times \pi \times 1410}
T = 10.16 \text{ Nm}

2.1.9 Centrifugal Force
\[
F = \frac{T}{r}
\] (9)

T = 10.16 \text{ Nm}
R = 37.5mm = 0.0375
F = \frac{10.16}{0.0375}
F = 269.87 \text{ N}

2.1.10 Stress acting on belt
\[
\sigma = \frac{F}{A}
\] (10)
F = 269.87 \text{ N}
A = 86.88 \text{ mm}^2
\[
\sigma = \frac{269.87}{86.88} = 3.15 \text{ N/mm}^2
\]

2.1.11 Maximum Tension
\[
T = \sigma \times a
a = 81.963 \times 10^{-6}
\sigma = 3.15 \text{N/ mm}^2
T = 81.963 \times 10^{-6} \times
T = 229.5 \text{ N}
\]

2.1.12 Tension in Tight Side, T_1
\[ T_1 = T - T_c \]  
Centrifugal tension is neglected  
Therefore \( T_1 = 229.5 \text{N} \)

### 2.1.13 Belt Tension Ratio

\[
\frac{T_1}{T_2} = e^{\mu \theta_1 \csc \frac{\alpha}{2}} \tag{2.13}
\]

Where:
- \( \alpha = 32^0 \)
- \( \mu = \text{Coefficient of friction} = 0.22 \)
- \( T_1 = \text{Tension in the tight side} \)
- \( T_2 = \text{Tension in the slack side} \)
- \( \theta_1 = 2.81 \text{rad} \)

\[ T_1 = e^{0.22 \times 2.81 \times \csc \left( \frac{32}{2} \right)} \]

\[ \frac{T_1}{T_2} = e^{2.24} \]

\[ \frac{T_1}{T_2} = 9.43 \text{N} \]

### 2.1.14 Tension of Slack Side, \( T_2 \)

\[ T_1 = 9.43 \text{N} \]
\[ T_2 = 229.5 \text{N} \]
\[ \frac{T_2}{T_2} = 9.43 \]
\[ T_2 = 24.34 \text{N} \]

### 2.1.15 Power Transmitted by v-belt

\[ P = (T_1 - T_2) V \] \tag{2.14}

\[ V = 5.54 \text{m/s} \]
\[ T_1 = 229.5 \text{N} \]
\[ T_2 = 24.34 \text{N} \]
\[ P = (229.5 - 24.34) \times 5.54 \]
\[ P = 1136.59 \text{W} \]

### 2.1.16 Shaft Design

A shaft is a rotating device use to transmitting power and motion from one point to another. A solid shaft is used base on the following:

i. To obtain the maximum tensional rigidity possible for the minimal diameter

ii. Ensure that the shaft would withstand the applied transverse and axial loads without risk of failure

To this end, a mild steel solid shaft was selected.

#### 2.1.16.1 Minimum Diameter of the Shaft

\[ \frac{T}{J} = \frac{G \theta}{L} = \frac{T}{r} \] \tag{2.15}

Length = 300mm = 0.3m

\( G = \text{(Modulus of rigidity) for mild steel } = 78 \text{GN/m}^2 [10] \)

\( \theta = \text{Taking a minimum angle of twist of 0.05} \)

\( T = \text{Torque} \)

\( J = \text{second polar moment} = \frac{\pi D^4}{32} \)

Ultimate tensile strength UTS = 440MPa
Torque (T) = 59.68Nm
Mathematically
\[
\frac{T}{\pi D^4/32} = \frac{G \theta}{L}
\]
(2.16)
Also,
\[
d = \sqrt[4]{\frac{32TL}{\pi GD^4}}
\]
(2.17)
d = 0.025m = 25mm
Hence a diameter of 25mm was selected for the solid shaft in order to allow for extraneous torsional loads on the shaft and to obtain torsional shiftiness.

2.1.16.2 Shaft Volume
\[
V = \frac{\pi D^2}{4} \times L
\]
(2.18)
\[
V = \frac{3.142 \times 0.025^2}{4} \times 0.3
\]
V = 1.473 × 10^{-4} m^3

2.1.16.3 The weight of the shaft
\[
W_s = \text{Density} \times \text{Volume} \times g
\]
(2.19)
Where density of mild steel = 7850 Kg/m^3 [10]
7850 × 3.865 × 10^{-4} × 9.81 = 29.76

2.1.16.5 The Weight of the Grinding Plate Mounted on the Shaft
Grinding plate dimensions
Diameter (Ø) = 150 mm
Thickness (t) = 20 mm
Volume of the grinding plates = \[
\frac{\pi D^2}{4} \times t \times n
\]
\[
= \frac{3.142 \times 150^2}{4} \times 20 \times 2
\]
= 706950 mm^3
\[
= 7.0695 \times 10^{-4} \ m^3
\]
∴ Weight of the grinding plate = Density × Volume × 9.81
= 54.44N
Hence the total transverse load on the shaft = 254.57 + 29.76 + 54.44
= 338.77N
Assuming the total load is distributed evenly across the length of the shaft since the greatest transverse load is due to the weight of the loaded cassava to be ground
\[
\frac{5 \times W^2}{384EI} \ [12]
\]
Where:
E = 200 GN/m^2
I = Moment of inertia
\[
\text{Maximum Deflection} = \frac{5 \times 338.77 \times (0.3)^2}{384 \times 200 \times 10^7 \times I}
\]
\[
= 1.985 \times 10^{-12}
\]
But I = \[
\frac{\pi d^4}{64}
\]
\[
= \frac{3.142 \times 0.05^4}{64} = 3.068 \times 10^{-7}
\]
∴ Maximum deflection = \[
1.985 \times 10^{-12}
\]
\[
= 6.47 \times 10^{-6} \text{ m} = 0.00647 \text{ mm}
\]
2.1.17 Maximum Bending Moment of the Shaft (Load on bearing)

Distributed load per unit length $= \frac{255}{0.3} = 850 N/m$

$R_A + R_B = 255 + 30 + 54 = 339$

$R_A + R_B - EM_B = 0$

$(R_A + 300) - (30 \times 150) - (255 \times 150) = 0$

$R_B = \frac{(30 \times 150) - (255 \times 150)}{300} = 142.5 N$

$R_A = (339 - 142.5) = 196.5 N$

2.1.18 Volume of Hopper

$$V_h = \frac{1}{3} \left[ (A_{base} \times H) - (a_{base} \times h) \right]$$

$$V_h = \frac{300}{400 + x} = \frac{100}{x}$$

$300x = (400+x) 100$

$300x = 40000 + 100x$

$200x = 40000$

$x = \frac{40000}{200} = 200 mm$

Hence Volume $= \frac{1}{3} \left[ (300^2 \times (400 + 200)) - (300^2 \times 200) \right]$

$= 17333.333 mm^3$

$= 0.0173 m^3$

Density of dried cassava to be load ranges from 1239 to 1500 Kg/m$^3$ [11]

$\therefore maximum mass to be loaded in the hopper = density \times volume$

$m = \text{Density} \times \text{Volume}$

$m = 1500 \times 0.0173 = 25.95 Kg$

Max Weight $= mg = 25.95 \times 9.81 = 254.57 N$

2.1.19 Fabrication of the Machine

**Frame**

The stand of the machine was fabricated with angular mild steel bar of cross section 305mm x 760mm. The angular mild steel bar was chosen because of its rigidity, availability and relatively cheap. A mild steel sheet of 5mm x 305mm x 760mm was brushed. The angular bar was cut into four pieces to form the four Legs of the stand and welded to a frame to form a table stand. Two rollers were screwed to the legs for easy mobility of the machine.

**Shaft**

A mild steel bar of 25mm x 300mm was mounted on the lathe chuck. Both ends were faced and then the rod turned into the designed diameter.

**Hopper**

The hopper is a truncated pyramid in shape. The truncated pyramid was constructed on drawing paper to hopper’s specification. The shape was cut out and pasted on the galvanized steel sheet. Scribe was used to trace the shape on the steel sheet and snipers used to cut out the marked out shape. The cut out sheet was folded to the required shape and the lapping edges were welded to form a hopper.

**Vibratory Filtration Components**

The basement of the filter was developed from 400mm x 400mm mild steel plate and was suspended on a spring which is vibrated by a shaft on load.

Figure 1 shows the isometric model view of the dried cassava pellets grinding machine.
III. RESULTS AND DISCUSSION

The summary of results obtained from design calculation is shown below.

- Belt length = 225mm
- Angle of lap on small pulley = 2.81rad
- Angle of map on large pulley = 3.47rad
- Tension on tight side = 273.6N
- Tension of slack side = 25.2N
- Peripheral Velocity = 5.54m/s
- Power required by the machine = 1177.1W
- Torque = 10.16Nm
- Shaft diameter = 25mm
- Shaft volume = 0.0001473m$^3$
- Density of shaft = 29.76kg/m$^3$
- Weight of grinding plate = 54.44N
- Total transverse load on shaft = 338.77N
- Maximum deflection = 0.00647mm
- Volume of hopper = 0.0173m$^3$

The maximum deflection (0.00647mm) obtained is negligible relative to the length of belt (225mm) and the diameter of the shaft (25mm). This implies that the shaft will retain its ability to function optimally under the applied total transverse load (338.77N). The shaft is fixed between the bearings on both ends which are in turn fixed to the frame. Hence the axial loading on the shaft may be considered negligible. Thus, the shaft dimensions and material (50 mm solid mild steel shaft) was selected to provide optimum function under the expected axial, transverse and torsional loading condition. The performance test results obtained with the machine is shown in Table 3.1. The machine throughput capacity is calculated from equation (1) [13].

$$MTC = \frac{M_1}{T}$$  \hspace{1cm} (1)

The efficiency is given by equation (2)

$$Eff. = \frac{M_2}{M_1} \times 100$$  \hspace{1cm} (2)

The average efficiency is calculated from equation (3).

$$Ave. = \frac{\sum}{n}$$  \hspace{1cm} (3)
Table 1: Performance Test Results

<table>
<thead>
<tr>
<th>S/N</th>
<th>M1 (Kg)</th>
<th>M2 (kg)</th>
<th>T (min.)</th>
<th>MTC(kg/min)</th>
<th>Eff. (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>30.00</td>
<td>25.01</td>
<td>8.00</td>
<td>3.75</td>
<td>83.37</td>
</tr>
<tr>
<td>2</td>
<td>28.45</td>
<td>20.85</td>
<td>6.45</td>
<td>4.41</td>
<td>73.29</td>
</tr>
<tr>
<td>3</td>
<td>25.05</td>
<td>15.45</td>
<td>5.05</td>
<td>4.96</td>
<td>61.68</td>
</tr>
<tr>
<td>4</td>
<td>23.20</td>
<td>14.20</td>
<td>4.45</td>
<td>5.21</td>
<td>61.20</td>
</tr>
<tr>
<td>5</td>
<td>20.45</td>
<td>12.25</td>
<td>4.00</td>
<td>5.11</td>
<td>59.90</td>
</tr>
<tr>
<td>Σ</td>
<td>127.15</td>
<td>87.76</td>
<td>27.95</td>
<td>23.44</td>
<td>339.44</td>
</tr>
<tr>
<td>Ave.</td>
<td>25.43</td>
<td>17.55</td>
<td>5.59</td>
<td>4.69</td>
<td>67.89</td>
</tr>
</tbody>
</table>

*M1 = Mass of dried cassava, M2 = Mass of properly grind cassava pellets

As shows in Table 1, performance test with the grinding machine was carried out five times with different masses of dried cassava that vary in weight. The average of mass of dried cassava pellets fed into the grinding machine and the mass of properly grind dried cassava to require sizes were calculated and it was used to determine the efficiency of the machine. An average efficiency of 67.89% was obtained and this shows that the machine is good and its performance was satisfactory. Figure 2 shows the graph of mass of dried cassava pellets, ground cassava pellets and efficiency. Figure 3 shows the graph of machine throughput capacity (MTC) and time of grinding. From the graph, the higher the mass, the longer the time of grinding. Also, the mass of cassava pellets is a function of the machine throughput capacity.
Static failure analysis was carried out using SolidWorks CAD modelling. Figure 4 shows the model information.

![Model Information](image)

The material properties are shown in Table 2.

<table>
<thead>
<tr>
<th>Model Reference</th>
<th>Properties</th>
<th>Components</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boss-Extrude2</td>
<td>Solid Body</td>
<td>SolidBody</td>
</tr>
<tr>
<td>Revolve2</td>
<td>Solid Body</td>
<td>SolidBody</td>
</tr>
</tbody>
</table>

![Material Properties Table](image)

Figure 5 shows the static failure analysis using Von Mises criteria. The Von Mises stress is at maximum towards the fixed end of the shaft and hopper and the value obtained is lower than the yielding stress of the material. Therefore, the design is safe.
IV. CONCLUSION

Nigeria is presently the largest producer of cassava in the world with an annual output of over 34 million tonnes of tuberous roots. Processing of cassava for storage is usually done traditionally by average Nigerians. This research work focused on the design and construction of dried cassava pellets grinding machine. The results obtained from the test performance analysis carried out on the machine design for domestic and commercial use in Nigeria show that the grinding machine was efficient and can be used across Nigeria towns and cities for processing of cassava tubers. This machine can replace the traditional method currently adopted by average Nigerian.

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