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Design, Fabrication and Performance Evaluation of a Multipurpose Cassava Grater

Bruno Robert Mose[1*], Victoria Achieng' Odhiambo[1], Nafula Maryvelma[1], Benard Ouma Alunda[2,3]

[1]Department of Mechanical Engineering, Jomo Kenyatta University of Agriculture & Technology, Nairobi, Kenya
[2]Centre for Mining, Environmental Engineering and Natural Resource Management (CEMEREM), Voi, Kenya
[3]Department of Mining and Mineral Processing Engineering, Taita Taveta University, Voi, Kenya
Email: *mbruno@eng.jkuat.ac.ke

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Abstract

Food security has been an issue of global concern and this has attracted a lot of research interest. Cassava is an extremely popular crop and is becoming the cornerstone for addressing food security in many parts of the world. The competing needs for cassava cuts across both human and animal consumption. It serves as a raw material in textile industry and is now one of the preferred materials for making biofuels. As the world's population continues to grow, the demand for drought resistant crops such as cassava is increasing. The high demand for various forms of processed cassava will continue to increase cassava prices making it an attractive business venture. Several small-scale cassava farmers are making a fortune and additional income through this business. Preliminary investigations show that the profitable way for a farmer to market his cassava is to add value to it. Unfortunately, cassava undergoes post-harvest physiological deterioration (PPD) after three days of harvest. In order to make cassava farming even more profitable, there is a need to process it within the shelf life of 2 - 3 days after harvesting. One way to preserve cassava is by grating it into pulp and drying it into pellets or chips. In this study, an electrically powered multi-purpose cassava grating machine with grater blades inclined at two different tooth angles, 25˚ and 30˚ was designed, fabricated and its performance characteristics investigated. The results showed that the plate with a tooth angle of 30˚ resulted in higher grating efficiency. This was attributed to better grip on the cassava when perforations on the plate inclined at 30˚. The grating capacity was also significantly improved as very small amounts of cassava slipped out un-grated.

Keywords

Food Security, Physiological Deterioration, Tooth Angle, Grating Efficiency

1. Introduction

Cassava was introduced to Africa by the Portuguese more than 300 years ago and is a major source of carbohydrate in sub-Saharan Africa. Compared with other carbohydrate sources such as rice and maize, cassava has the advantage of high productivity under marginal climatic conditions. Cassava has the capacity to yield under marginal soil conditions and is tolerant to drought [1]. The crop, therefore, plays a vital role in the food security of the world [2]-[11].

Over the past 25 years, significant market opportunities for cassava have opened up in the animal feeding industry. Moreover, cassava starch is an important ingredient in the manufacture of dyes in textile industry, drugs, chemicals, carpets and in coagulation of rubber latex. This is in addition to the capacity of cassava to feed the rapidly increasing population.

Despite the numerous advantages as a source of food for humans, livestock feed and for industrial applications, cassava undergoes post-harvest physiological deterioration (PPD) after three days of harvest due to its high moisture content of about 70%. Therefore, there is a need to process cassava within three days after harvesting in order to obtain the multiple economic benefits from the crop. The best form of preservation and reduction of post-harvest losses is the immediate processing of cassava into a stable product such as pulp which can be easily and properly dried.

Processing cassava by traditional methods suffers low productivities and a susceptibility to food contamination. Various attempts to design and fabricate cassava-grating machines have been made with a view to overcome the negative attributes of the traditional processing techniques and promote timely, large-scale processing of the tubers in a hygienic environment. These machines, however, are usually found to suffer significant corrosion due to the acidic nature of the cassava fluid [12]. Non-food grade materials used for the fabrication of cassava processing machines have been found to react with the fluid to cause contamination.

It is, therefore, essential to come up with alternative machine designs with materials that do not easily corrode. This is because cassava is becoming a vital means of livelihood to feed the global population with multiple other economic benefits, thus, having the ability to generate revenue and income for developing countries.

In this study, an electrically powered multi-purpose cassava grating machine with grater blades inclined at two angles, 25˚ and 30˚ was designed, fabricated and its performance characteristics investigated. The developed multipurpose grating machine can be used for grating not only cassava tubers but also sweet potatoes, irish potatoes, yams, arrow roots, carrots, beetroots, bananas as well as pumpkins. Moreover, the grater can be used to process medicinal herbs such as aloe Vera, and other fruits such as avocados, raw mangoes. As a result, the proposed machine can trigger and promote processing of cassava tubers as well as other farm products into commercially acceptable products including medicinal herbs, thereby increasing income obtainable by local farmers.

2. Materials and Method

The schematic of the main components of the cassava grating machine is shown in Figure 1.

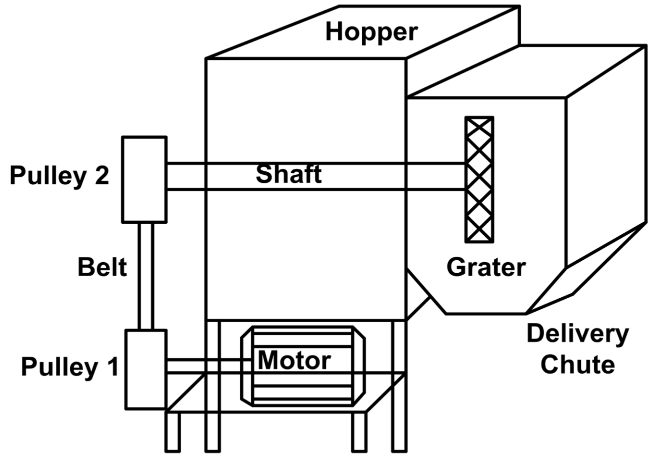


Figure 1. Main components of the grating machine.

2.1. Design Considerations

A number of considerations were put in place while designing the various components of the grater. These considerations include the following;

1) Attain high throughput of cassava grating by reducing grating time.

2) Reduce high labor requirements of traditional manually operated graters.

3) Reduce corrosion and enhance hygienic conditions by use of food grade material.

4) Enable easy disassembly, cleaning and assembly of the machine.

5) Achieve economic viability and reduce cost.

2.2. Design of Various Components

2.2.1. Hopper Design

The hopper was designed based on the quantity of cassava to be processed to meet the farmer/market demands. It was found that an average cassava farmer with acreage of 1 - 2 acres [13] required 336 kg of cassava to be grated per hour, which translates to 5.6 kg of cassava per minute. Using Equation (1), for a mass of 5.6 kg/min and an average cassava density of 670 kg/m^3, the volume of the hopper was estimated as follows;

$$\text{Volume} = \frac{\text{Mass}}{\text{Density}} \tag{1}$$

$$V = \frac{5.6\ \text{kg/min}}{670\ \text{kg/m}^3} = 8.3582 \times 10^{-3}\ \text{m}^3$$

The volume of hopper to grate 5.6 kg of cassava per minute was found to be at least 8,358,000 mm^3. The next critical item concerning the hopper was the shape. The hopper was designed to have a pyramidal shape with an angular inclination of 50 degrees to allow cassava to be fed into the grating chamber by the force of gravity so as to reduce human effort [14].

2.2.2. Belt Design

Determination of Center Distance (C)

The Rubber Manufacturing Association recommends that the distance between the center of the driving pulley and that of the driven pulley be obtained from consideration of the speed ratio,

$$\text{Speed Ratio} = \frac{D}{d} = \frac{0.2}{0.1} = 2$$

For speed ratios less than three:

$$c = \frac{D+d}{2} + d \tag{2}$$

where:

D = Diameter of the driving pulley= 0.27m;

d = Diameter of the driven pulley = 0.135m;

Center distance was computed and found as;

$$C = 0.34\ \text{m}$$

Length of Belt

A V-belt was chosen to transmit power generated by the motor to the shaft since it requires less installation work and time, absorbs shock and operates at low bearing pressure. The formula in Equation (3) was used to compute the length of the belt;

$$L = \frac{\pi(D+d)}{2} + 2C + \frac{(D-d)^2}{4C} \tag{3}$$

where:

C = Distance between the center of the two pulleys = 0.34;

L = Length of Belt Required was upon substitution found to be 1.3177 m.

A50 V belt of length 1.3 m was selected due to its low vibration, less noise, high efficiency which facilitates longer motor and V-belt life.

2.2.3. Motor Design

Capacity of the electric motor

Power transmitted by motor; P is given by Equation (4);

$$P = \frac{\pi F N_1 D_1}{60} \tag{4}$$

where:

F is the force to crush fresh cassava 105N;

N_1 is the rotational speed of motor 1140 rpm;

D_1 is the diameter of motor pulley 0.135 m.

Applying Equation (4), the power rating is obtained as 1068 Watt. A motor with a power rating of 1.5 HP = 1119-Watt, 13 Ampere, 240 volts, 1440 rpm, 50 Hz was thus selected.

2.2.4. Shaft Design

To transmit power from the shaft, various members/elements such as the pulleys, bearings and the grating unit were mounted on it. These members along with the force exerted upon them cause the shaft to bend. Therefore, the shaft is subjected to bending and torsional forces since it is utilized for transmission of torque and bending moment.

Belt Tension in Open Belt Drive

Belt tension exists when power is transmitted from one shaft to the other as indicated in Figure 2.

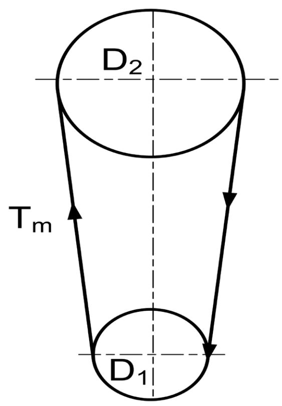


Figure 2. Open Belt Drive for power transmission.

The maximum tension on the belt was determined using Equation (5) [15]

$$T_m = \tau_{all} \times \text{cross sectional area of belt} \tag{5}$$

where; τ_{all} is the allowable shear stress on the belt.

For belt materials, allowable shear stress is 2.8 MPa. For A50 V-belt, cross sectional area is 20 mm^2. The maximum tension was found to be 56N.

Elements Mounted on Shaft

The elements mounted on the shaft along with the force exerted upon them cause bending and torsional forces.

Determination of the Bending Moment at each Point of Loading

In order to determine the bending moment at each point of loading of the shaft, it was necessary to prepare the bending moment diagram.

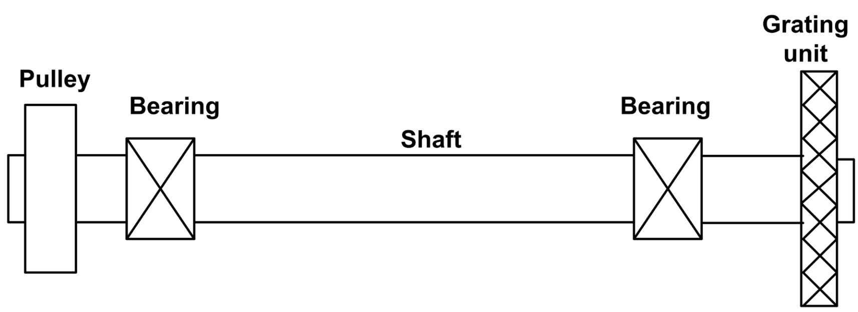


Figure 3. The main machine elements mounted on the shaft.

Force Exerted on Shaft (Vertical Force)

The main machine elements that exert forces on the shaft are the driven belt pulley and grating unit as shown in Figure 3.

The weights of pulley and grating unit were obtained by getting the product of material density and volume as follows.

$$\text{Weight of the pulley in kg, } W_p = \rho_p V_p = 2700 \times 3.5 \times 10^{-4} = 1.0 \text{ kg}$$

$$\text{Weight of the grating unit in kg, } W_g = \rho_g V_g = 2700 \times 1.1 \times 10^{-3} = 3 \text{ kg}$$

where ρ_p and ρ_g are densities of pulley and grater materials, V_p and V_g are pulley and grater volumes. Point loading, W_d on shaft due to hopper and cassava was determined,

$$W_d = g\left[\left(V_c \times \rho_c\right) + \left(\rho_H \times V_H\right)\right] = 404.94 \text{ N}$$

$$W_d = 9.81\left[\left(4.5 \times 10^{-3}\right) \times 675 + 7280 \times \left(4.5 \times 10^{-3}\right)\right] = 351.2 \text{ N}$$

where; ρ_H, ρ_c = Density of hopper material and density of cassava respectively.

For cassava, $\rho_c = 675$ kg/m³ while for hopper material, $\rho_H = 7280$ kg/m³.

Reactions at the Bearings due to Vertical Loading

In Figure 4, the expected free body diagram of the vertical forces acting on the shaft is shown. The forces acting on the shaft due to the pulley, F_p and due to the grater F_g were obtained by multiplying the weights, W_p and W_g by the acceleration due to gravity, 9.81 m/s². Note that RBV and RDV denote the reactions in the vertical direction at point B and D respectively.

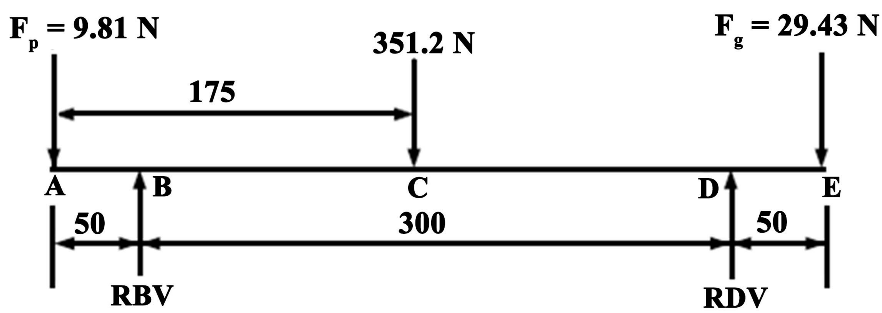


Figure 4. Free body diagram of the main forces and reactions acting on the shaft.

To determine the reactions at each bearing, moments about the two expected bearing points (B and D) were determined. Taking moments about point B;

$$RDV \times 0.3 + 9.81 \times 0.05 = 351.2 \times 0.15 + 29.43 \times 0.35$$
$$RCV = 208.3 \text{ N}$$

Taking Moments about D we get

$$RBV \times 0.3 + 29.43 \times 0.05 = 351.2 \times 0.15 + 9.81 \times 0.35$$
$$RBV = 182.14 \text{ N}$$

Shear Force Diagram

The shear force diagram was drawn as shown in Figure 5.

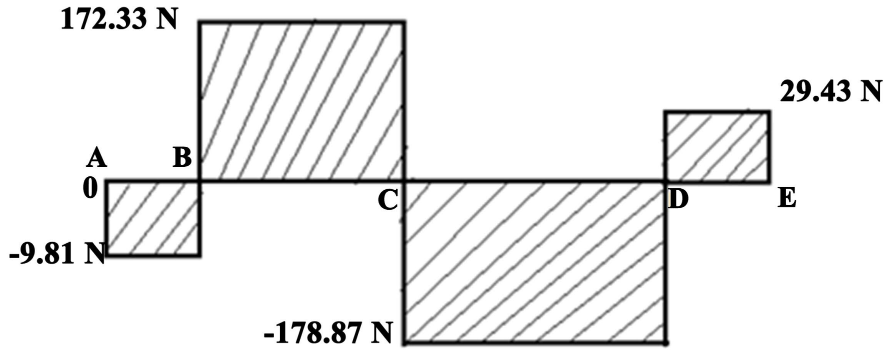


Figure 5. Shear force diagram.

Bending Moment Diagram

From the evaluation of the forces and determination of the bearing reactions, the maximum bending moment (M_{max}) for the shaft was evaluated. The following equations were used for bending moment diagram.

Section A-B: $M = -9.81x$;

Section B-C: $M = 172.33x - 9.107$;

Section C-D: $M = -178.87x + 61.133$;

Section D-E: $M = 29.43x - 11.772$;

where x is the distance from point **A** in meters.

The bending moment diagram (BMD) was drawn as shown in Figure 6. As noted, the maximum bending moment was found to be 25.36 Nm.

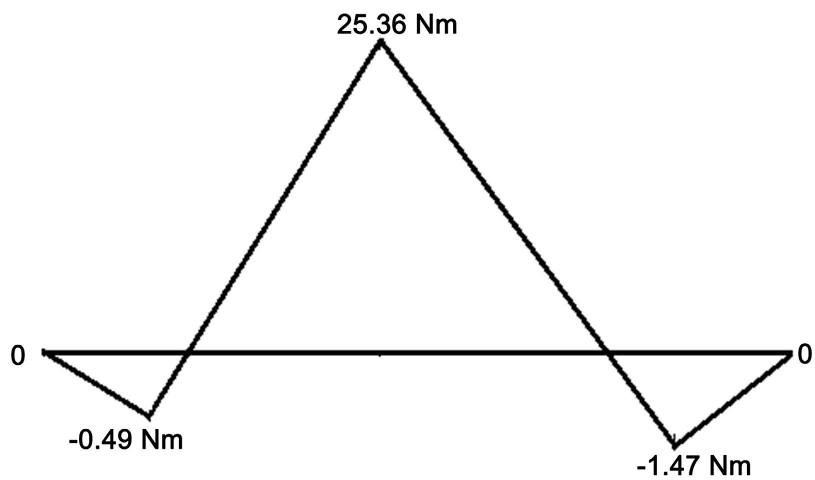


Figure 6. Bending moment diagram.

For a solid shaft having little or no axial load, the diameter of the shaft was determined from Equation (6) [16].

$$D_s^3 = \frac{16}{\pi S_s}\left[\left(K_b M_b\right)^2 + \left(K_t M_t\right)^2\right]^{\frac{1}{2}} \tag{6}$$

where; D_s is diameter of the shaft, M_t is torsional moment, M_b is bending moment,

K_b is the combined shock and fatigue applied to bending moment, K_t is the combined shock & fatigue applied to torsional moment and S_s = allowable stress.

For a shaft transmitting power (W) at a given rotational speed (rpm), the torque is given by Equation (7a).

$$T = \frac{P}{\omega} \tag{7a}$$

where; ω is the angular velocity and is given by;

$$\omega = \frac{2\pi N}{60}\,\text{rad/s}$$

From motor design, it was found that power rating of motor is 1.5 HP = 1119-Watt, 13 amperes, 240 volts, 1440 rpm, 50 Hz and with a speed reduction of 1:2 to give the driven speed of 720 rpm, torsional moment, M_t is given by Equation (7b);

$$M_t = \text{Power/Speed} = \frac{1119 \times 60}{2\pi \times 720} = 14.84\ \text{N} \cdot \text{m} \tag{7b}$$

Given that $S_s = 20 \times 10^6\ \text{N/m}^2$, $K_b = 1.5$, $K_t = 1$, $M_b = 25.36$ Nm, $M_t = 14.84$ Nm and $\pi = 3.142$; and upon substituting these values in Equation (6) we get;

$$D_s^3 = 10388.77\ \text{mm}^3 \text{ from which } D_s = 21.82\ \text{mm}$$

From table of standard shaft sizes, the 25 mm diameter was selected [15]. The shaft was joined to the driving motor by means of keys, and aligned using ball bearings. The selection of this diameter also takes into account of any frictional forces at the points where bearings are located.

2.2.5. Selection of Ball Bearings for Pillow Block

Deep Groove Ball bearings were used since they can withstand the thrust action in the machine and are easy to repair and assemble. To determine the equivalent load when only a radial load, R, is applied Equation (8) was used.

$$F_{eq} = VR \tag{8}$$

where V is rotation factor and $V = 1.0$ if the inner race of the bearing rotates, or $V = 1.2$ if the outer race rotates. Noting that this is a pure radial load, the inner race is to be pressed onto the shaft and rotate with it. Therefore, the factor $V = 1.0$ in Equation (8) was applied.

The deep-groove ball bearing is to carry 100 kg of pure radial load from a shaft that rotates at 720 rpm. The design load is obtained as 100 kg. The bearing is to be mounted on a shaft with a diameter of 25 mm. Based on these parameters, bearing number 6205 was selected. The summary of data for the selected bearing: bearing number: 6205, single-row, deep-groove ball bearing, bore: $d = 25$ mm, outside diameter: $D = 52$ mm, Width: $B = 15$ mm.

2.2.6. Perforated Plate

The perforated plate, which is a critical component in cassava grating machines was made from food grade stainless steel. The perforated plate was fixed on the grating unit which was in turn mounted on the shaft. The force acting on the plate is given by Equation (9).

$$F = V_p \rho g \tag{9}$$

where g is acceleration due to gravity (9.81 m/s^2), V_P is volume of the plate and ρ is the material density, F is the force required to crush cassava. Rearranging Equation (9), we can get the diameter of the perforated plate using Equation (10).

$$\frac{\pi D^2}{4} = \frac{F}{\rho g t} \tag{10}$$

Substituting the values, diameter of perforated plate was found to be 306 mm.

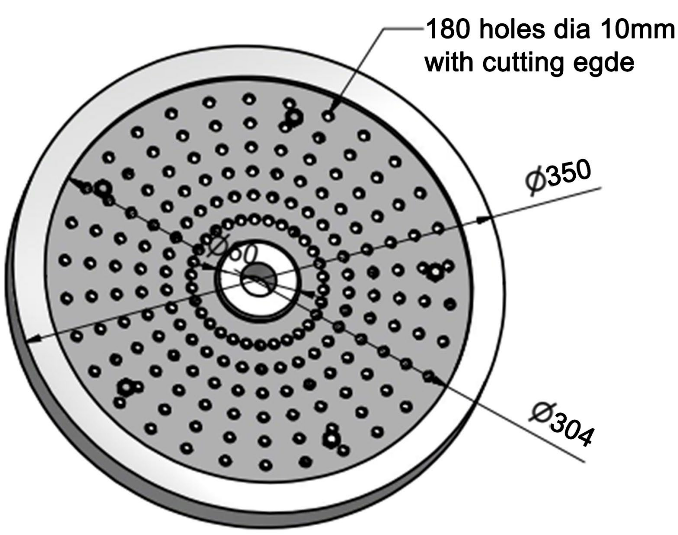
Tooth Angle of the Perforations

The angle of inclination of the perforations was calculated based on the radius of the perforations and the height of the inclination to be made. For this case a radius of 4 mm and a height of 8.5 mm were used. Applying the formula $\tan\theta = \frac{r}{h}$ an angle of 25˚ was obtained. 60˚ staggered round holes selected for the perforation pattern because they are the most economical, strongest and most versatile as compared to rectangular, square and slotted holes.

3. Results and Discussions

After designing and fabricating the motorized cassava grater, experimental work was carried out to investigate the performance characteristics to determine the Grating efficiency, Grating Capacity and Grating Losses of an electrically powered multi-purpose cassava grating machine with grater blades inclined at tooth angles of 25˚ and 30˚.

In Figure 7(a) the perforated plate with inclined blades used for experimental work is shown. On the other hand, Figure 7(b) shows grating operation of the cassava grater during performance testing. In Figure 7(c) weighing scale measuring the mass of grated cassava is illustrated.



(a)

(b)

(c)

Figure 7. (a) Perforated plate, (b) performance testing of cassava grater, (c) mass measurement.

For comparison, performance characteristics of a grating plate whose blades were not inclined at any special angle was investigated. The results are shown in Table 1.

Table 1. Grating time for various masses of cassava and losses for grater blades inclined haphazardly.

Loading Number	Initial mass of un-grated Cassava (kg)	Time taken to grate (sec)	Mass of grated cassava (kg)	Losses (kg)
1	1.25	24.59	1.064	0.186
2	1.20	30.86	1.041	0.209
3	1.05	27.03	0.960	0.090
4	1.17	23.41	0.854	0.316
Total	**4.67**	**105.89**	**3.919**	**0.801**

Average mass of initial cassava

$$=\frac{\text{Total Mass of initial Cassava}}{\text{Number of Loadings}}=\frac{4.67}{4}=1.1675\text{ kg}$$

Average mass of grated cassava

$$=\frac{\text{Total Mass of Grated Cassava}}{\text{Number of Loadings}}=\frac{3.919}{4}=0.97975\text{ kg}$$

Average Time Taken to Grate 1 kg

$$=\frac{\text{Total Time Taken to Grate}(\text{sec})}{\text{Number of Loadings}}=\frac{105.89}{3.919}=27.020\text{ seconds}$$

Grating efficiency, η_g

$$=\frac{\text{Mass of well grated cassava}}{\text{Initial mass of cassava tuber}}\times 100\%=\frac{0.97975}{1.1675}\times 100\%=83.92\%$$

Grating Capacity

$$=\frac{\text{Mass of total grated cassava}}{\text{Grating time}}\frac{\text{kg}}{\text{hr}}=\frac{0.97975}{0.007353}\frac{\text{kg}}{\text{hr}}=133.2\text{ kg/hr}$$

Grating Losses

$$=\frac{\text{Initial mass of cassava tuber}-\text{Mass of total grated cassava}}{\text{Initial mass of cassava tuber}}\times 100\%$$

$$=\frac{1.1675-0.97975}{1.1675}\times 100\%=16.05\%$$

For the case when the blades were not inclined at a specific angle, the results show that it requires an average of 27.02 seconds to grate 1.0 kg of cassava. The grating efficiency when blades are haphazardly inclined was found to be 83.92% and the grating capacity was 133.2 kg/hr.

When the blades were inclined at an angle of 30˚, the measurements of grating time for different masses of cassava as well as losses were recorded in **Table 2**. It was shown that average grating time was 14.183 seconds for 1.0 kg of cassava when blades are inclined at 30˚ angle. The grating efficiency increased from 83.92% when blades are haphazardly inclined to 93.54%. In addition, it was observed that inclining the grating blades at 30˚ increased the grating capacity from was 133.2 kg/hr to 398.33 kg/hr.

Table 2. Grating time for various masses of cassava and losses for grater blades inclined at 30˚.

Loading Number	Initial mass of un-grated Cassava (kg)	Time taken to grate (sec)	Mass of grated cassava (kg)	Losses (kg)
1	2.00	27.07	1.980	0.020
2	2.09	28.14	1.812	0.278
3	2.05	26.87	1.898	0.152
4	2.03	27.22	1.920	0.110
5	2.05	26.29	1.950	0.100
Total	**10.22**	**135.59**	**9.560**	**0.660**

Average mass of initial cassava

$$= \frac{\text{Total Mass of Ungrated Cassava}}{\text{Number of Loadings}} = \frac{10.22}{5} = 2.044\ \text{kg}$$

Average mass of grated cassava

$$= \frac{\text{Total Mass of Grated Cassava}}{\text{Number of Loadings}} = \frac{9.560}{5} = 1.912\ \text{kg}$$

Average Time Taken to Grate 1 kg

$$= \frac{\text{Total Time Taken to Grate}(\text{sec})}{\text{Number of Loadings}} = \frac{135.59}{9.56} = 14.183\ \text{seconds}$$

Grating efficiency, η_g

$$= \frac{\text{Mass of well grated cassava}}{\text{Initial mass of cassava tuber}} \times 100\% = \frac{1.912}{2.044} \times 100\% = 93.54\%$$

Grating Capacity

$$= \frac{\text{Mass of total grated cassava}}{\text{Grating time}} \frac{\text{kg}}{\text{hr}} = \frac{1.912}{0.0048} \frac{\text{kg}}{\text{hr}} = 398.33\ \text{kg/hr}$$

Grating Losses

$$= \frac{\text{Initial mass of cassava tuber} - \text{Mass of total grated cassava}}{\text{Initial mass of cassava tuber}} \times 100\%$$

$$= \frac{2.044 - 1.912}{2.044} \times 100\% = 6.458\%$$

When the blades were inclined at an angle of 25˚, the grating time for various masses of cassava and losses were recorded in **Table 3**.

Table 3. Grating time for various masses of cassava and losses for grater blades inclined at 25˚.

Loading Number	Initial mass of un-grated Cassava (kg)	Time taken to grate (sec)	Mass of grated cassava (kg)	Losses (kg)
1	2.02	27.25	1.947	0.073
2	1.98	26.98	1.775	0.205
3	2.13	28.15	1.869	0.261
4	2.05	27.07	1.863	0.187
5	2.01	26.58	1.924	0.086
Total	**10.19**	**136.03**	**9.378**	**0.812**

Average mass of un-grated cassava

$$= \frac{\text{Total Mass of initial Ungrated Cassava}}{\text{Number of Loadings}} = \frac{10.19}{5} = 2.038\ \text{kg}$$

Average mass of grated cassava

$$= \frac{\text{Total Mass of Grated Cassava}}{\text{Number of Loadings}} = \frac{9.378}{5} = 1.8756\ \text{kg}$$

Average Time Taken to Grate 1 kg

$$=\frac{\text{Total Time Taken to Grate}\left(\text{sec}\right)}{\text{Number of Loadings}}=\frac{136.03}{9.378}=14.052\text{ seconds}$$

Grating efficiency, η_g

$$\eta_g=\frac{\text{Mass of well grated cassava}}{\text{Initial mass of cassava tuber}}\times 100\%=\frac{1.8756}{2.038}\times 100\%=92.03\%$$

Grating Capacity

$$=\frac{\text{Mass of total grated cassava}}{\text{Grating time}}\frac{\text{kg}}{\text{hr}}=\frac{1.8756}{0.004735}\frac{\text{kg}}{\text{hr}}=396.12\text{ kg/hr}$$

Grating Losses

$$=\frac{\text{Initial mass of cassava tuber}-\text{Mass of total grated cassava}}{\text{Initial mass of cassava tuber}}\times 100\%$$

$$=\frac{2.038-1.8756}{2.038}\times 100\%=7.949\%$$

At 25˚ inclination of the grating blades, it was possible to grate 1.0 kg of cassava within an average period of 14.052 seconds. The efficiency increased to 92.03% from 83.92% when no inclination was included on the grating plate. The losses also decreased from 17.15% to 7.949% as a result of inclining the perforations at 25˚. The results obtained from this study are summarized in **Table 4**.

Table 4. Summary of results.

TOOTH ANGLE	25˚	30˚	Haphazard Angle
Average Mass of Ungrated Cassava (kg)	2.038	2.044	1.1675
Average Mass of Grated Cassava (kg)	1.8756	1.912	0.97975
Average Grating Time (s) for 1 kg	14.052	14.183	27.020
Grating Capacity (kg/hr.)	248.19	253.82	133.2
Grating Efficiency (%)	92.03	93.54	83.92
Grating Losses (%)	7.949	6.458	17.15

From the results in **Table 4**, it was noted that inclining the perforations on the grating plate at angles of 25˚ and 30˚ has a significant effect on reducing the grating time required to process 1 kg of cassava. This leads to increased grating capacity and efficiency. These findings show that design of cassava grating machine with higher performance characteristics such as grating efficiency, grating capacity and grating Losses, has the potential to accelerate cassava production as a vital means of livelihood for feeding the global population and spur economic development.

4. Conclusions

In this study, an electrically powered multi-purpose cassava grating machine with grater blades inclined at two angles, 25˚ and 30˚ was designed, fabricated and its performance characteristics investigated. From the results of the experiment, the following conclusions were made.

1) The plate with the tooth angle of 30˚ gave better results than that with a tooth angle of 25˚ because the former provided a better grip on the cassava ensuring higher grating efficiency and a reduction in losses.

2) The cassava grating machine will encourage farmers who have not ventured into cassava farming to do so and those practicing in small-scale to expand their acreage due to low post-harvest losses.

3) The machine is also an avenue for job creation for people in and around cassava growing areas for post-harvest processes which include sorting, peeling, cleaning, grating, drying and packaging.

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Conflicts of Interest

The authors declare no conflict of interest.

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