

# International Journal of Innovative Research in Science, Engineering and Technology

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## Design of a Portable Cassava Chopping Machine

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**ABSTRACT:** Cassava is a root and tuber crop grown across the globe mainly for consumption and industrial purposes. It takes about two to three days to deteriorate after harvest, hence the urgent need to speed up its postharvest handlings. Cassava tubers are mostly chopped into different sizes, dried, and stored for more days; this has led to the development of different chipping/chopping machines which produce fixed chopped-sizes of tuber as the desired results. This study is to design a portable cassava chopping machine that is capable of producing different chopped-sizes of tubers. The machine uses shear strength caused by the rotary motions of the blades to chop the tubers. After theoretical evaluation, the machine has 814.4-1227.4 kg/hr capacity,  $\geq 95\%$  efficiency, and no/minimal flesh losses/mechanical damage. This machine is simple to operate and it is recommended for small, medium, and large-scale cassava processors.

**KEYWORDS:** Cassava tubers, chopping machine, chopping blades, and sieve.

### I.INTRODUCTION

Cassava originates from South America, Western Brazil, and it's a perennial and consumable root and tuber crop. In the Amazon Region, it's one of the consumed food, and it's grown across the globe especially in the tropical and sub-tropical regions [1]. It is typically grown on small-scale by traditional means, for this reason, less attention has been given to its processing and production input. Cassava has moved from a poor man's food to an export commodity due to the current demand for cassava chips and pellets from countries like China[2], [3]. It is the most perishable crop amongst the root and tuber crops, which deteriorates two to three days after harvest [4]. For consumptive purposes, some of the basic cassava processing operations are peeling, sieving, drying, frying and extrusion, grating, and milling [5].

Many machines have been developed including the ones designed by [6]–[8]. These machines have a single cutting blade and a sieve. Henceforth, the machines have fixed/constant chipped-sizes of tuber as the desired output, hence if different desired-sizes are needed, two or more machines will be required to achieve that goal. Also, these chipped tubers are very tiny in sizes, which makes their handlings difficult.

Cassava tubers are mostly chopped, dried, and stored for some time, because reducing the sizes via chopping operations reduces the surface area which facilitates the drying process. Hence, the objective of this study is to design a portable cassava chopping machine that is capable of producing different chopped-sizes of tuber for small, medium, and large-scale cassava processors.

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## II. RELATED WORK

Different types of cassava chipping/chopping machines have been developed, by some individual researchers and some research institutions like NCAM. A cassava chipping machine that has a disc plate as its chipping tool was developed [6]. The machine is supported on a metallic frame, and it runs smoothly at a speed of 400-500 rpm. The machine has maximum efficiency and chipping capacity to be 92.6% at 300rpm and 245 kg/hr at 500rpm respectively, and minimum efficiency of 86.5% and 240 kg/hr capacity at 400rpm, with a constant/fixed chipped size of tubers. [8] developed a cassava chipping machine that has a similar design structure and features as the previous design, as well as its operational mechanisms. The machine was evaluated at 300-450 rpm operating speed and the chipping capacity ranged from 56.05-500.94 kg/hr and overall efficiency of 74.91%. [7] improved on the machine developed by [6], this machine has a trapezoidal hopper, and at 1400 rpm motor speed, the theoretical efficiency was calculated to be 94.56%. the machine was fabricated at The Federal University of Technology, Akure, Nigeria.

## III. MATERIALS AND METHODS

### Design Consideration

The engineering properties and availability were the basis of the selection of materials for designing the machine. The following were put under consideration:

- The availability of materials for construction.
- The power, speed, and force required to chop the tubers.
- Machine portability and ease of operation.
- Machine Capacity and efficiency
- Easy maintenance of the machine
- Chopping sizes of tubers
- Cost affordability

### Machine Description

The machine consists of a supporting frame on which the entire components are mounted. Power is transmitted from the electric motor (placed on a wooden board) to the blade shaft via belt and pulleys system. The chopping blades, mounted on the blade/rotating shaft, is being fixed inside the chopping unit perpendicularly, and the chopping unit is cylindrical, where the actual chopping operation takes place. The hopper is fixed at the top corner of the chopping unit, and the discharge unit is fixed at the opened-bottom of the chopping unit. The sieve is placed between the bottom-open end of the chopping unit the discharge unit. The machine has other components like bearing to enhance smooth rotation of the shaft; and bolts & nuts that fasten other components together. Fig.1 shows the interior and the exterior components of the machine. The components are assembled. The assembled machine must be oiled and started for about 5 minutes before the cassava tubers are fed into it every day. The chopping blade or the sieve can be changed based on the operator's preference.

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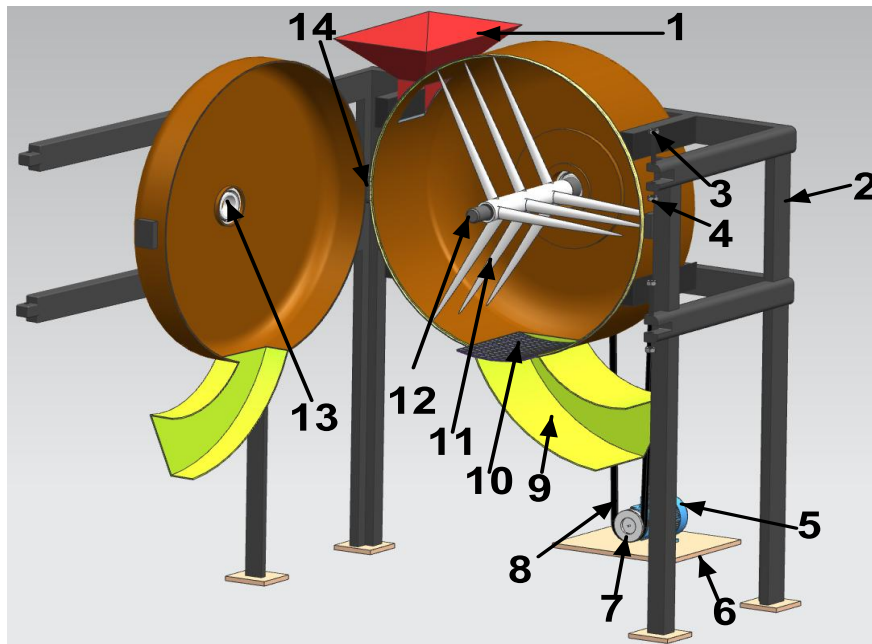


Fig.1 Isometric view of the interior and the exterior components of the modeled machine

1-hopper, 2-supporting frame, 3&4-bolt & nut, 5-electric motor, 6-wooden board, 7-pulley, 8-V-belt, 9-discharge unit, 10-sieve, 11-chopping blades, 12-rotating shaft, 13-bearing, 14-hinge.

## IV. DESIGN ANALYSIS AND SPECIFICATIONS

### Chopping Blades

The chopping blades are fixed on a hollow shaft, which is driven by the shaft of the large pulley. Two chopping blades were designed, which have the same length and a blade thickness of 1.5 mm. Fig. 2 shows the Type-A and Type-B chopping blades, they look the same except the number of single blades and the blade spacings. The Type-A has 9 chopping blades at 0.6 m spacing, and the other has 15 blades with 0.3 m spacing.

The clearance distance between the tip of the blades and the walls of the chopping unit is 5 mm, and this enhances continuous contact between the blades and the tubers.

$$F_c = f_r + f_s \quad (1.0)$$

$F_c$ =the force required to chop the tubers (N),  $f_r = \mu * f_s$ =frictional force between the blades and the tubers (N),  $f_s = A * S_f$ = the required chopping shear force (N),  $S_f$ =the shear strength of the tuber ( $N/m^2$ ),  $A = W * L$ =area of contact between the blades and the tubers ( $m^2$ ),  $W$  and  $L$  are width and length of the contact surface between the blade and tuber (m). Fig. 3 shows the force diagram of the chopping blade during operation.

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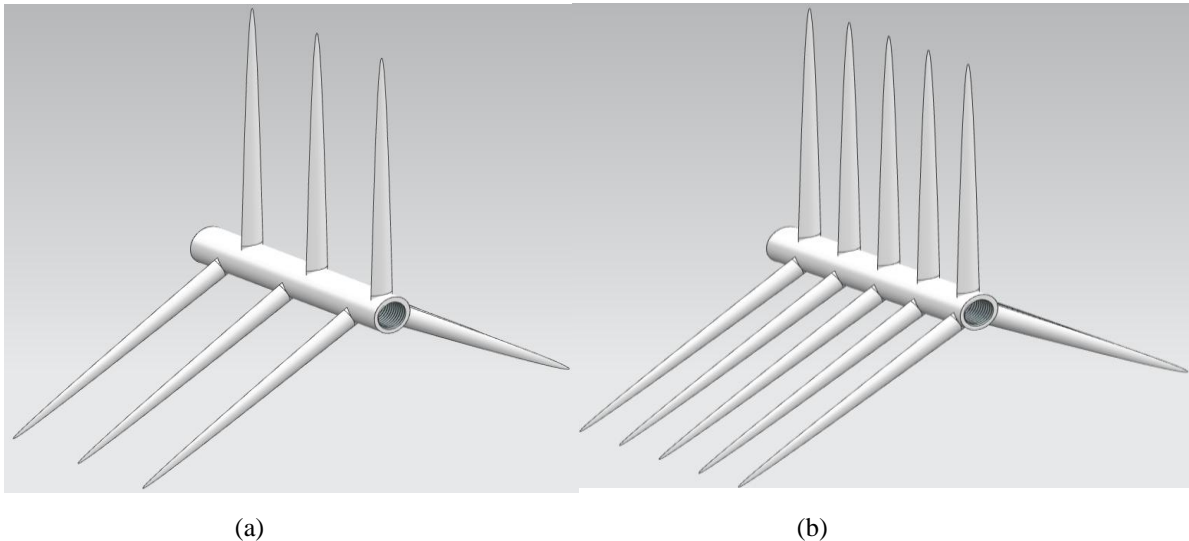


Fig.2 Types of the chopping blades (a) Type-A has 9 blades fixed a shaft (b) Type-B has 15 cutting blades fixed on a shaft. Each blade has a 1.5 mm in width. The Type-A is purposely designed for sieve size-3, where the Type-B was designed for the sieve size-2 and size-3.

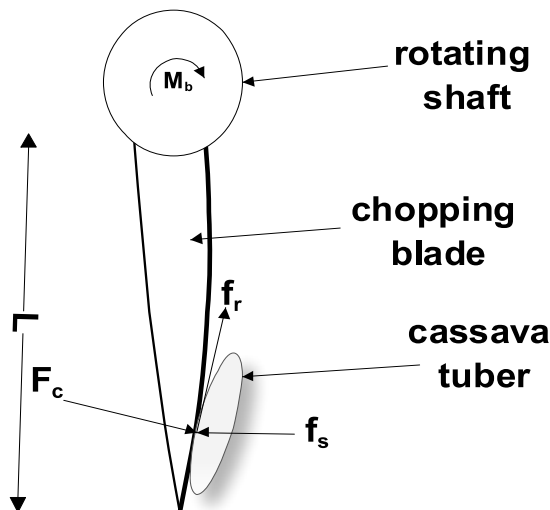


Fig.3 Force diagram of the chopping blade

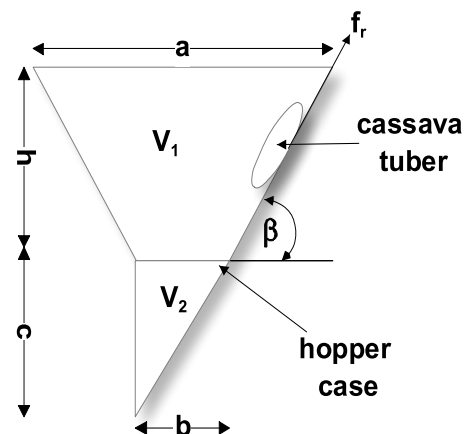


Fig.4 Side view of the hopper

Cassava tubers dropped into the machine through the hopper as shown in Fig. 4, they slight gently and fall by gravity into the chopping unit. Chopping takes place as the blade cut through the tubers as shown in Fig. 3, the forces required to complete the operation were calculated and factored into the design considerations.

## Hopper Design

The angle of inclination and the volume were the considered parameters for the hopper design as shown in Fig. 4, the expressions of the parameters are:

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The volume: the hopper has an opened-trapezoidal top and an opened-triangular based fixed at the top of the chopping unit.

$$\text{Trapezoidal-top volume (m}^3\text{), } V_1 = 0.5(a + b) * h \quad (2.0)$$

$$\text{Triangular-base Volume(m}^3\text{), } V_2 = 0.5 * b * c \quad (2.1)$$

$$\text{Therefore, hopper total volume (m}^3\text{), } V_T = V_1 + V_2 \quad (2.2)$$

$$\text{The angle of inclination, } \theta = \tan^{-1} \mu \quad (3.0)$$

$\mu$ =coefficient of friction between the hopper and the tubers.

### Power Required

A 5 hp (3.729 kW) electric motor was used, and the design power was calculated as

$$P_d = P_r + P_c \quad (4.0)$$

$$P_r = W_b * L * V_b \quad (4.1)$$

$$P_c = F_c * L * V_b \quad (4.2)$$

$P_d$ = Design power (kW),  $P_r$ =Power to rotate the blade (kW),  $P_c$ =Power required to chop the tuber (kW),  $V_b$ =the rotating speed of the blade (m/s),  $L$ = length of the rotating blade.

The transmission power required was calculated as [9]

$$P_t = \frac{P_d}{K_s} \quad (4.3)$$

$P_t$ = Transmission power (kW),  $K_s$ = Service factor, the transmission power calculated was used to select the best electric motor (rated power in horsepower).

$$K_s = K_i + K_o + K_e \quad (4.4)$$

$K_o$ = Service correction factor,  $K_e$ = Environment correction factor,  $K_i$ = Idler correction factor=0, no idler was used. The designed power calculated to be 3.543 kW.

### Shaft Design

Two shafts were used, one transmits power from the motor to the machine through the belt and the other transmits power and causes rotary motion of the blades to chop the tubers in the chopping unit. The shafts were designed according to these calculations [10]:

$$\tau = \frac{P * 60}{2\pi N} \quad (5.0)$$

$\tau$ = torsional moment (Nm),  $N$ =shaft speed (rpm),  $P$ =power required to drive the shaft

$$d^3 = \frac{16}{\pi \sigma_a} \sqrt{(K_b M_b)^2 + (K_t M_t)^2} \quad (5.1)$$

$\sigma_a$ =allowable shear stress (N/m<sup>2</sup>),  $d$ =diameter of the shaft (m),  $M_b$ =bending moment (Nm),  $K_b$ = combined shock and fatigue due to bending moment,  $K_t$ = combined shock and fatigue due to moment of torsion.

The bending moment:

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$$M_b = \frac{\pi \sigma_b d^3}{32} \quad (5.2)$$

$\sigma_b$ =Bending stress induced due to bending moment (N/m<sup>2</sup>).

$$\tau_{max} = 0.5\sqrt{(\sigma_b)^2 + 4(\tau)^2} \quad (5.3)$$

$\tau_{max}$ =maximum shear stress (N/m<sup>2</sup>),  $\tau$ =The equivalent twisting moment (Nm).

### Pulley Design

In designing the pulleys, shown in Fig. 5, references were made from [9] to choose the appropriate dimensions. Two pulleys were required for the machine, and the diameter of each was derived from this equation:

$$DN = dn \quad (6.0)$$

Where D=diameter of the blade pulley (m), d=diameter of the motor pulley (m), N and n are their respective speed (rpm).

### Belt Selection

For light drives, agricultural purposes, 10m/s maximum belt speed was taken [11]. Regarding the belt specifications [9], a classical V-belt (A-Section) was chosen from with designed dimensions as shown in Fig. 5 and the following calculations were used:

#### 3.6.1 The belt length

$$L_b = 2C_i + \frac{\pi}{2}(D + d) + \frac{(D-d)^2}{4C} \quad (7.0)$$

$L_b$ =belt length (m) and  $C_i$ =assumed center distance (m)

#### Pulley size

The actual pulleys center distance,  $C_a$

$$C_a = \frac{b + \sqrt{b^2 - 8(D-d)^2}}{8} \quad (7.1)$$

$$b = 2L_b - \pi(D + d) \quad (7.2)$$

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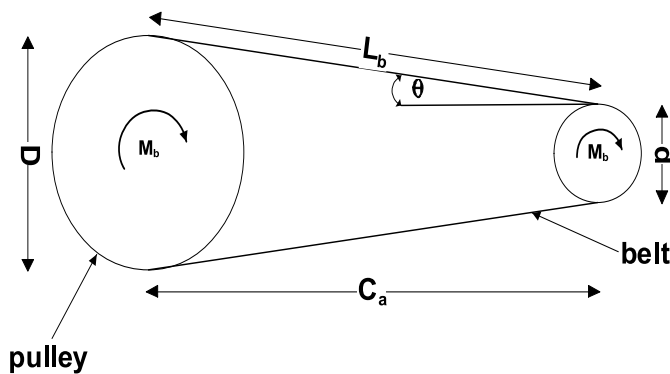


Fig. 5 Pulleys and belt arrangement

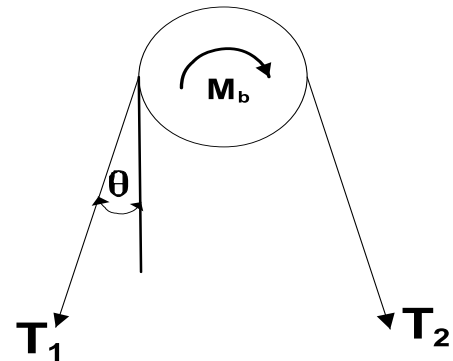


Fig.6 Force of Tension exerted in the belt on both sides

The smaller pulley is mounted on the electric motor shaft, and the large pulley is mounted on the shaft that runs the blades. The two pulleys are connected via a V-belt, and power is transmitted from the motor to power the machine via the belt. The pulley size, angle of wrap, belt length, and center length of the pulleys were calculated and factored into the designed considerations.

### Number of belts

The number of belts required:

$$n_b = \frac{P_d}{P_f} \quad (7.3)$$

$$\text{Where } P_f = (P_s + P_a) * K_c, \text{ and } K_c = K_\theta * K_l \quad (7.4)$$

$P_s$ =basic power rating for small pulley datum diameter(kW),  $P_a$ =additional power rating for the speed ratio (kW),  $K_\theta$ =arc correction factor,  $K_l$ =belt length correction factor. Equation (7.4) was substituted back into equation (7.3).

During the belt installation, the following measures were taken,

$$\text{The minimum center distance, } C_{amin} = C_a - I_a \quad (7.5)$$

$$\text{Maximum center distance, } C_{amax} = C_a - T_a \quad (7.6)$$

$I_a$ =Belt Installation allowance,  $T_a$ =Take-up allowance

### Belt tension

The tensional forces of the belt were calculated according to [9], [11] as shown in Fig. 6, since the pulleys are fixed, their weight on the belt were neglected:

$$\frac{T_1}{T_2} = e^{\mu\theta} \quad (7.7)$$

$$\tau = (T_1 - T_2) * R_p \quad (7.8)$$

Where  $T_1$  and  $T_2$  are the tensional forces of the slack and tight sides of the belt respectively (N),  $R_p$ = radius of the pulley (m),  $\tau$ =torsional moment (Nm),  $\mu$ =coefficient of friction between the belt and the pulley, and  $\theta$ = angle of wrap/contact



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made between the belt and the pulley ( $^{\circ}$ ). Equation (7.7) and equation (7.8) were solved simultaneously to get  $T_1$  and  $T_2$ .

$$\alpha = \sin^{-1}\left(\frac{D-d}{2c_a}\right) \quad (7.9)$$

$\theta$  for the small pulley =  $180^{\circ}-2\alpha$ , and  $\theta$  for the large pulley =  $180^{\circ}+2\alpha$

$$V_t = \frac{\pi D_p N_p}{60} \quad (7.91)$$

$V_t$ =belt speed (m/s),  $D_p$ =pulley diameter (m),  $N_p$ =speed of speed (rpm)

### Sieves

Three types of metallic sieves of different sieve sizes were designed, shown in Fig. 7, the selection was based on the operator's preferred chopped-sizes of the tubers. The hole sizes of the sieves are 0.2m\*0.2m, 0.3m\*0.3m, and 0.4m\*0.4m, of which they are easily changeable, no/fewer skills are required to the change sieves. It is placed between the chopping unit and the discharge unit of the machine.

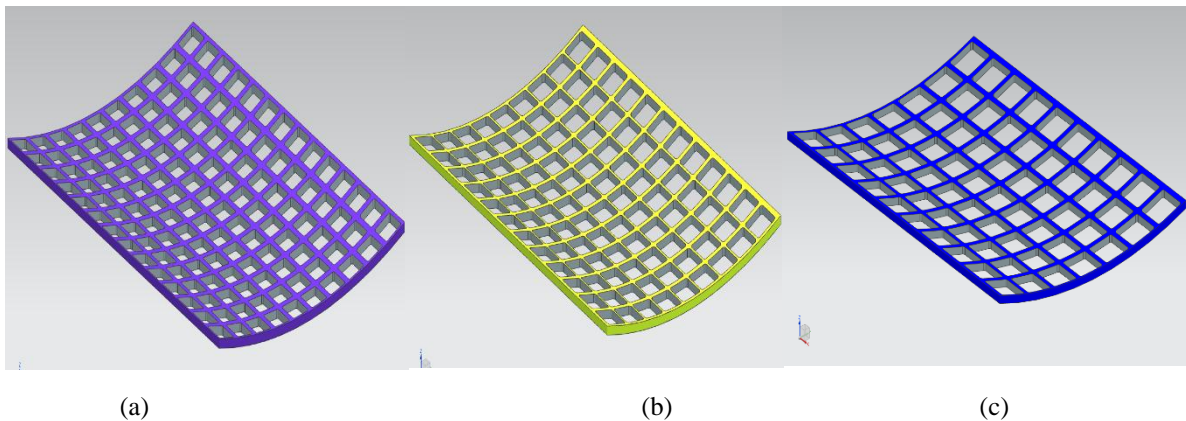


Fig.7 Model of the types of sieve sizes and their dimensions (a) Size-1: 0.2m\*0.2m (b) Size-2: 0.3m\*0.3m (c) Size-3: 0.4m\*0.4m. Three different sieve sizes, the sieve size is chosen based on the desired chopping size. The sieve is paired with the correct chopping blade type to achieve the desired output.

### Chipping units

The chopping chamber is cylindrical that houses mainly the cutting blades and the sieve. The following parameters of the chamber were calculated:

Volume:

$$V = \pi r^2 h \quad (8.0)$$

The volume was calculated to be  $0.393 \text{ m}^3$ . The machine was designed to efficiently operate by not exceeding 40% drum fill. Hence, the feeding rate should not be more than 40% of the volume ( $0.157 \text{ m}^3$ ).

Mass:

$$M_c = \rho V \quad (8.1)$$

Where h, r, and  $\rho$  are the height (m), radius (m), the density of the material used ( $\text{kg/m}^3$ ) of the unit respectively.



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## V.DESIGN AND PERFORMANCE EVALUATION

The chopped-sizes of tubers was the main focus during the evaluation. At constant operating speed, the Type-A and the Type-B chopping blades are being paired with the right sieve size to achieve the desired results.

The machine designed was theoretically evaluated using the design parameters. The throughput capacity, chopping efficiency, and mechanical damages were calculated. Taking the density of cassava tuber to  $1300 \text{ kg/m}^3$  [12], and neglecting the energy loss through heat and friction:

*The machine capacity:*

According to [13], the machine capacity (kg/hr) calculated as:

$$M_{cp} = \frac{0.55 \cdot n}{t} \quad (8.2)$$

Where 0.55 is the average weight of a cassava tuber (kg), n= the number of tubers fed into the machine, t=the time taken to complete the chopping operation (hr).

Therefore, the average volume of a tuber,

$$V_{CT} = 0.55 \text{ kg} / 1300 \text{ kg/m}^3$$

The average maximum number of tubers to occupy the 40% drum fill,

$$n_{max} = \frac{40\% \text{ drum fill}}{V_{CT}} \quad (8.3)$$

Therefore,  $n_{max} \approx 372$  pieces of cassava in the drum at a time.

The Type-A chopping blade is designed to averagely chop 372 pieces of tubers into the required sizes in less than 15 minutes (0.25 hr), whereas the Type-B chopping blade completes the same operation in less than 10 minutes (0.1667 hr). Therefore, the machine capacity for the two types are:

Type-A

$$M_{cp} = \frac{0.55 \text{ kg} \cdot 372 \text{ maximum pieces of tubers}}{0.25 \text{ hr}} = 818.4 \text{ kg/hr} \quad (8.4)$$

Type-B

$$M_{cp} = \frac{0.55 \text{ kg} \cdot 372 \text{ maximum pieces of tubers}}{0.1667 \text{ hr}} = 1227.4 \text{ kg/hr} \quad (8.5)$$

After a complete chopping operation, the efficiency is calculated to be:

$$M_{cp} = \frac{\text{Total mass of cassava tubers chopped (kg)}}{\text{Time taken to chop the tubers (hr)}} \quad (8.6)$$

*The machine efficiency*

A 5 hp (3.729 kW) electric motor was selected to power the machine whose minimum designed power to effectively drive the chopping blades is 3.543 kW, hence, the theoretical efficiency was calculated to be as:

$$\eta_t = \frac{\text{power output (kW)}}{\text{power input (kW)}} * 100 = \frac{\text{Designed power (kW)}}{\text{Transmission power (kW)}} * 100 \quad (8.7)$$

$$\eta_t = \frac{3.543 \text{ kW}}{3.729 \text{ kW}} * 100 = 95\%$$

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Therefore, the theoretical efficiency of the machine,  $\eta \geq 95\%$

After completing the chopping operation, the efficiency is calculated to be:

$$\eta_a = \frac{\text{Total mass of cassava tubers chopped (kg)}}{\text{Total mass of the tubers fed into the machine (kg)}} * 100 \quad (8.8)$$

With the required speed, chopping blades, and sieve selected, no/minimal flesh losses/mechanical damages are recorded, unless the maximum drum fill (40%) is exceeded.

$$f_l = \frac{\text{Total mass of tubers lost/broken during operation (kg)}}{\text{Total mass of the tubers fed into the machine (kg)}} * 100 \quad (8.9)$$

## VI.CONCLUSION

A portable cassava chopping machine, shown in Fig. 8, that has 3 types of sieve sizes (size-1, size-2, and size-3) and chopping blades (Type-A and Type-B), has been designed. The machine uses shear strength caused by the rotary motions of the blades to chop the tubers. The performance of the machine, after theoretical evaluation, shows a major improvement compared to previous designs. The ability of the machine to produce different chopped-sizes reduces production cost on purchasing different machines to achieve that same goal. The theoretical evaluation shows that the machine has 814.-1227.4 kg/hrthroughput capacity, more than 95% efficiency, and no/minimal flesh losses/mechanical damages. The simplicity of the machine, performance, portability, and easy maintenance make it suitable and adaptable to small, medium, or large-scale cassava tuber processors.

## VII.RECOMMENDATIONS

This machine is recommended for small, medium, and large-scale cassava processors.

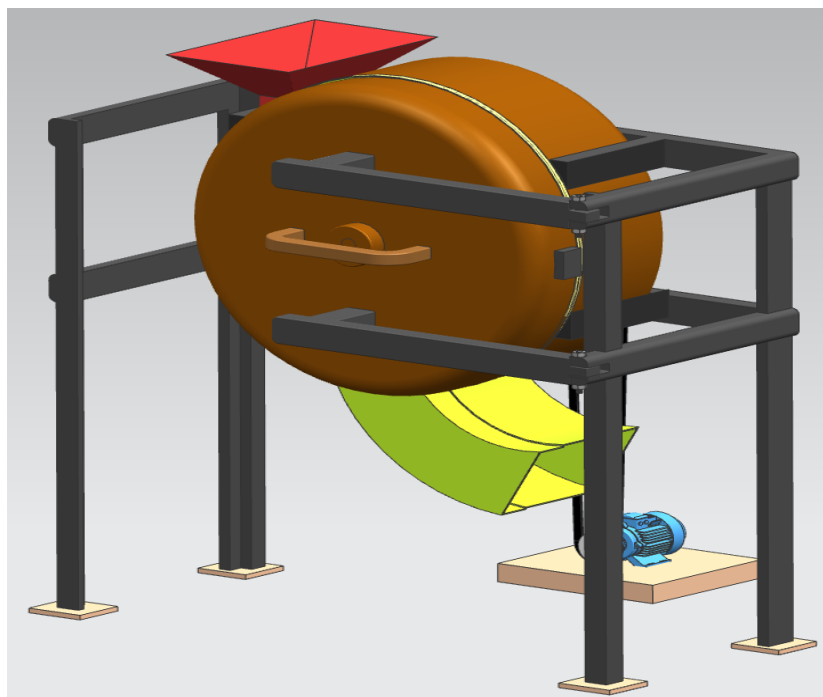


Fig.8 Model of the portable cassava chopping machine

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## DEDICATION

This work is dedicated to my mother and all my family members.