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DEVELOPMENT OF A RECIPROCATING MOTION CASSAVA SLICING MACHINE

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Abstract
The development of cassava reciprocating slicing machine was achieved using locally sourced materials that is affordable and effective. Although hand slicing is the cheapest form of slicing operation, but it has posed to be labour intensive, time-wasting, and hazardous. This necessitated the design and development of a cassava slicing machine. The machine's capacity for boiled and unboiled cassava root was calculated as 22.8kg/hr, with an average slicing time of 0.005hr for boiled and 0.00455hr for unboiled cassava root. The machine has low labour requirements and power consumption. The cassava reciprocating slicing machine use electric motor of 0.75kw (1hp) rating, with a speed of 99rpm. The machine is made with stainless steel for the slicing section and other components with mild steel and has an overall efficiency of 91.05%. The machine reduces drudgery and also enhances mass production of cassava chips, implying more profit.

Keywords: Cassava, slicer, reciprocating motion, slicing efficiency, throughput capacity

Introduction
Cassava (Manihot esculenta Crantz), originating from South America, is one of the world's main root crops and constitutes the most important staple of rural and urban households in Sub-Saharan Africa (Spencer and Ezedinma, 2017; Petsakos *et al*., 2019; Ndjouenkeu *et al*., 2020).Nigeria is the world's largest producer of cassava (FAO, 2021), hosting a diverse array of cassava farmers and processors, with the large majority being small-scale operators (Forsythe *et al*., 2016). The tuberous root and its products feed more than 500 million African households with an average annual consumption of 100kg of roots per person, with Nigeria's annual production of 59.47 million metric tons, 65% of which is consumed locally (FAO, 2021).

Cassava roots contain more than 60% water. However, their dry matter contents are seen to be very rich of about 250 to 300kg for every ton of fresh roots. It also has the cheapest source of calories available, and its roots contain a significant amount of vitamin c, thiamine, riboflavin and niacin (Save and Grow, 2017). Cassava is seen as the most essential grown food in Africa. The crop plays a significant role in ending famine in Africa because it has high starch content, tolerance to poor soil conditions, availability over the years, and suitability for diverse smallholder farming systems and technology (Folefac *et al.,* 2017).

Depending on preference and local customs, cassava can be used up either in a fresh form or in various industrially or traditionally processed forms. Cassava flour is one of these products, and chips, starch and tapioca. Although it has low nutritional value, it remains an essential food in several parts of the world as it is less expensive than wheat and can be used to produce other varieties of food products (Gomes, 2017). Apart from its traditional use in Africa, it is widely used in the production of livestock and fish feed production in most developed countries of the world (Ugoamadi, 2012). Cassava chips processing is an important stage that influences the quality of the final product i.e., cassava flour. It serves as a source of cash income for rural growers and processors, majority of whom are women. Demand for cassava is expected to increase as consumer preference changes with the development of new cassava products (Alacho *et al*., 2013). Traditionally, cassava is processed into chips by peeling, slicing into chunks and drying on the floor or by the roadside.

Current usage of cassava chips suffers from the poor quality, high post-harvest losses, and poor safety. Processors fail to deliver regular supplies of high quality flour at a competitive price despite the premium prices the industrial users pay for quality (Kleih *et al*., 2012). Slicing is a size reduction process (Etoamaihe and Iwe, 2014). Therefore, slicing can improve cassava products quality, even though the process is not extensively used in cassava processing. Reduction in the size of the cassava roots to be processed into a food product that

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Plate 1: A photograph of the developed cassava slicing machine

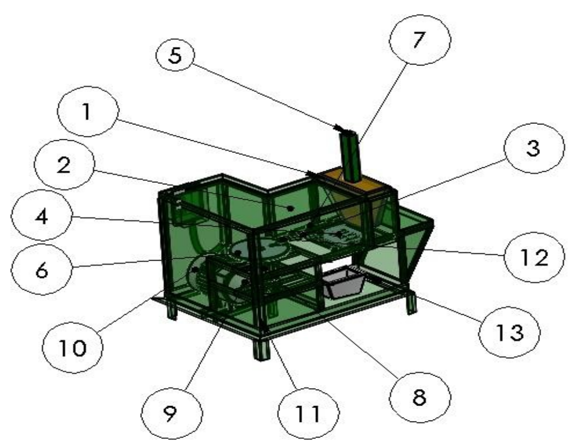


Figure 1: Parts of the developed cassava slicing machine

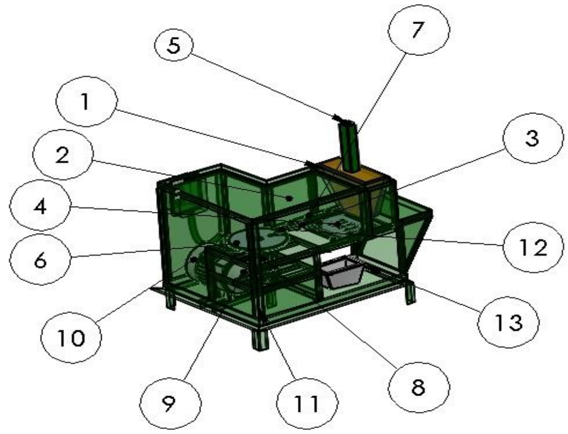


Plate 1: A photograph of the developed cassava slicing machine

Figure 1: Parts of the developed cassava slicing machine

Table 1: The cassava slicing machine parts description

S/no	Descrption of the Machine Component Parts	Materials used for Machine
1	Hopper	2mm Stainless steel
2	Tray	2mm Stainless steel
3	Push rod	Mild steel
4	Slicer	2mm Stainless steel
5	Steady	Wood
6	Disc	Cast iron plate
7	Bearing	25mm diameter
8	Disc shaft	Steel 25mm diameter
9	Bevel gears	Cast iron
10	Geared motor	1hp
11	Frame	Mild steel angular rod (50mm×50mm×5mm)
12	Stopper	Cast iron
13	Bowel	Stainless steel

Selection of bevel gear and determination of speed

Since two cast iron bevel gears of diameters 60mm each were used for the centrifuge power transmission. These diameters were obtained using equation 1 thus;

$$\text{V.R= velocity ratio}=\frac{D_G}{D.P.}=\frac{T_G}{T_P}=\frac{N_P}{N_G} \text{ ……….. (1)}$$

Where, T_P = no of teeth on the pinion=17 teeth; T_G = no teeth on the gear=28.17 teeth; D_G = pitch diameter of the gear=60mm; D_P = pitch diameter of the pinion=60mm; N_P = no speed on the pinion = 99rpm; H_g = no of speed on the gear = 99rpm. According to Nwankwojike (2012), to avoid interference, the value of the number of teeth on the pinion (T_p) = 17 was determined using equation 2 thus;

$$T_P=\frac{2A_W}{G[\sqrt{1+\frac{1}{G}(\frac{1}{G}+2)\sin^{2}{}_{\theta-1}}]} \text{ …………………. (2)}$$

Where = fraction by which the standard addendum for the wheel should be multiplied=1; G=gear ratio or velocity ratio=1:13; φ = pressure angle or angle of obliquity= 20°.

Using the tooth profile of the 20o hobs may cut full depth involute system. The increase of pressure angle from 14½ to 20° result in a more vital tooth because the tooth acting as a beam is wider at the base. For the pitch angles and the formative number of the bevel gear, since the shafts are at right angles, equations 3 and 4 by (Akinnuli *et al*., 2015) were used to determine the pitch angles for the gear and pinion θ_{p1} and θ_2 as 4.4° and 86° respectively, thus;

$$\theta_{p1}=\tan^{-1}\left[\frac{1}{V.R}\right] \text{ …………………….. (3)}$$

$$\theta_{p2}=90-\theta_{p1} \text{ ……………………….. (4)}$$

Where; V.R = velocity ratio of the gear=1:13

Due to accuracy on the bevel gear tooth profile, according to Akinnuli *et al.* (2015), the formative or equivalent number (T_{EP} and T_{EG}) for bevel gear was calculated as 17 and 244 teeth respectively using equation 5 thus;

$$T_E=\frac{2R_B}{m}$$

$$=\frac{2\,R\sec\theta_p}{m}=T\sec\theta_p \text{ ……………….. (5)}$$

Where; T = Actual number of teeth on the gear=17 teeth;

θ_p = pitch angle or half of the cone's angle= 4.4^0 According to Akinnuli *et al.* (2015), The Lewis form factor (y'p) and (y'G) for 20^0 full depth involute system for the machine was determined using equation 6 as 0.100 and 0.150 respectively, for 20^0 full depth involute system;

$$y = 0.154 - \frac{0.912}{T} \quad \text{(6)}$$

Where; T = actual number of teeth on the gear=17

Since the allowable stress (σo) for both the pinion and the gear is the same (56mpa or N/MM^2), i.e., the standard value of cast iron ordinary gear and (y'p) is less than (y'G), therefore the pinion is weaker. Thus the design should be based upon the pinion.

In that case, we know that the torque (T) on the pinion was calculated using equation 7 as 72330Nmm thus;

$$T = \frac{P \times 60}{2\pi N_p} \quad \text{(7)}$$

Where, P= power in kilowatt=0.75kw; N_P= number of speed on pinion=99rpm

The tangential Load (W_T) on the pinion is also calculated as 2127.4N where our module (m) = 4 using equations 8 to10 (Ukpabi, 2014) thus;

$$W_T = \frac{2T}{D_p} \quad \text{(8)}$$

$$\text{But: } T_P = \frac{D_P}{m} \quad \text{(9)}$$

$$m = \frac{D_P}{T_P} \quad \text{(10)}$$

Where; m = module=4; Dp = pitch diameter on the pinion =60mm=0.06; Tp = number of teeth on the pinion=17.

Also our pitch line velocity (V) is calculated as 0.311m/s using equation (11) thus;

$$V = \frac{\pi D_P N_P}{60} \quad \text{(11)}$$

For satisfactory operation of the bevel gears, the face width of the gear(b), the length of the pitch cone element or slant height of the pitch cone(L), was calculated as 35mm using mathematical relations in equation 12 by Igudu (2009) thus;

$$R_m = [L - \tfrac{b}{2}] \sin\theta_{P1} = [L - \tfrac{b}{2}] \frac{D_P}{2L} \quad \text{(12)}$$

Where,

$$\sin\theta_{P1} = \frac{D_P/2}{L} \quad \text{(13)}$$

Making L subject of formula in equation 13, we have equation 14 thus;

$$L = \frac{m.T_G}{2 \sin\theta_{P2}} \quad \text{(14)}$$

Assuming the face width (b), at 1/3rd of the slant height of the pitch cone (L), therefore (b) = 12mm using equation 15 thus;

$$b = \frac{L}{3} \quad \text{(15)}$$

For satisfactory operation of the bevel gears, since the face width should be from 6.3m to 9.5m and where m is the module (m=4). Therefore (b) was calculated to be 27 using equation 16 thus;

$$b = 6.3m - 9.5m \quad \text{(16)}$$

To check for gear wears in this case by Khurmi and Gupta (2008), the load stress factor depends upon the maximum fatigue limit of compressive stress, the pressure angle and the modulus of elasticity of the materials the gears. so, therefore, according to Buckingham, the load stress factor (K) 2.31 using equation 17 thus;

$$K = \frac{(\sigma_{es})^2 \sin\varphi}{1.4}\left(\frac{1}{E_P} + \frac{1}{E_G}\right) \quad \text{(17)}$$

Where; σ_{es} surface endurance limit as 630Mpa or N/mm^2; φ = pressure angle=20^o since the tooth profile of the gear are of 20^o full depth involutes; E_P = Young modulus for the material of the pinion in N/mm^2 and E_G = Young modulus for the material of the gear in N/mm^2 or the flexural endurance limits(σe) is obtained as 84 X 10^3Gpa by Akinnuli *et al.,* (2015) according to standards. Also, based on the formative or equivalent number of teeth, we have that the Ratio factor(Q) = 1.87 which is determined using equation 18 thus;

$$Q = \frac{2T_{EG}}{T_{EG} + T_{EP}} \quad \text{(18)}$$

Where; T_{EP} and T_{EG}t The formative number of teeth on the gear and pinion for the machine was determined as 17 and 244 teeth, respectively.

The Maximum or limiting Load for the wear (W_w) for satisfactory purpose of the design was calculated as 7018.6N using equation 19 thus;

$$w_w = \frac{D_p.b.Q.k}{\cos\theta_{p1}} \quad \text{(19)}$$

Where; w_w= 7018.6N; D_p=60mm=0.06m; b= 27, Q= 1.87

Since $W_W > W_T$ (7018.6N > 2127.4).We say the design is satisfactory from the standpoint of the wear.

For Inaccuracies of tooth spacing, irregularities in tooth profiles and deflections of teeth under the Load. The dynamic tooth load (W_D) was calculated as 2691.4N using Buckingham's equation obtained by (Akinnuli *et al.,* 2015) given by,

$$W_D = W_T + \frac{21v(b.c + W_T)}{21v + \sqrt{b.c + W_T}} \quad \text{(20)}$$

Where, W_T= Steady transmitted load =2127.4N; V= pitch line velocity =0.311m/s; c= face width of the gears =27mm; c= deformation or dynamic factor=238N/mm.

The value of the dynamic factor (c),which is calculated as 238N/mm, was determined using this relation;

$$c = \frac{k.e}{\frac{1}{E_P}+\frac{1}{E_G}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (21)$$

Where k = form tooth factor for 20° full depth involutes teeth gear =0.111; e = tooth error factor for 20° by Akinnuli *et al.,* (2015) first-class commercial gear at 4mm module = 0.051; also E_P and E_G Young's modulus for the material of the gear is 84Gpa, respectively.
Checking for the static tooth load (W_S) for good design, the value of the static tooth load was calculated as 4275.6N using equation 22 thus;

$$W_S = \sigma_e.b.p_c.y = \sigma_e.b.\pi m.y \quad \ldots\ldots\ldots\ldots\ldots\ldots (22)$$

Where σ_e= flexural endurance limits=84Gpa.B.H.N = 160 for cast iron; b = face width = 27mm; m = module=4mm; y = tooth form factor =0.100mm and 0.150mm each.
Since $W_S > W_D$ (4275.6 > 2691.4), we say the design is satisfactory from the standpoint of the Dynamic Load.

Determination of shafts diameters
The diameter of the shaft (d), which was designed based on rigidity and stiffness because the vibration of the shaft would be hazardous, was determined as 20.2mm, but based on a factor of safety, a standard 25mm shaft was used for the drive (Shingly and Mischke, 2001) using equation 23 thus;

$$T_e = \frac{\pi}{16} \times \tau(d_p)^3 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (23)$$

Where, Te = equivalent twisting torque=90540N-mm; dp = diameter of the pinion shaft; τ =shear stress for the material of the pinion=42mpa standard.
The maximum bending moment, MD = on the shaft was determined in which the bearing reactions were calculated first. The total axial and radial forces (i.e. $W_{R.H.}$ and W_{RV}) acting on the pinion shaft was calculated as 69.6N and 905N using equations 24 and 25 thus

$$W_{RH} = W_{Ttan\varphi sin\theta_{p1}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (24)$$

$$W_{RV} = W_{Ttan\varphi cos\theta_{p1}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (25)$$

Where, The bending moment due to $W_{R.H}$ and W_{RV} is obtained as RA= 124.76N-mm and RB=53.84N-mm
The resulting bending moment was calculated as 36306.9 by resolving horizontally and vertically
For Horizontal loading, the bending moment at M_A and M_D = 0

M_{CH}= 4360N-mm, and $M_{D.H.}$=2784N-mm
For vertical loading, $M_{CV} = 0$ and
M_{DV}=36200N-mm

The bending moment at M_C and M_D were determined using equations 26 and 27 thus;

$$M_C = \sqrt{(M_{CV})^2 + (M_{CH})^2} = 4360\text{N-mm} \ldots\ldots (26)$$

And the resultant bending moment at D,

$$M_D = \sqrt{(M_{DV})^2 + (M_{D.H.})^2} = 36309.9\text{N-mm} \ldots\ldots (27)$$

Therefore, the resultant bending moment is at maximum at D
Maximum bending moment M - M_D = 36309.9N -mm and d Diameter
Since the shaft is subjected to twisting moment (T) and the resulting bending moment (M), therefore, the equivalent twisting moment (Tp) is calculated as 90540N-mm having our K_m = 1.5 and K_t = 1.0 as a recommended value for gradually applied Load for stationary shafts using equation 28 thus;

$$T_e = \sqrt{(K_{m\times M})^2 + (K_{t\times T})^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (28)$$

where, M = max bending moment=36309.9N-mm; T = torque=72330N-mm; Km and Kt = combined shock and fatigue factor for bending and torsion determined as 1.5 and 1.0, respectively.

The shape of the hopper is that of a frustum of a pyramid at an inclined angle of 30°. The volume of the machine hopper was designed to enable the length of 60mm and a height of 80mm shredded cassava to be cut. Therefore, the volume of the hopper was calculated as 231104 mm^2 using equation 29 thus;

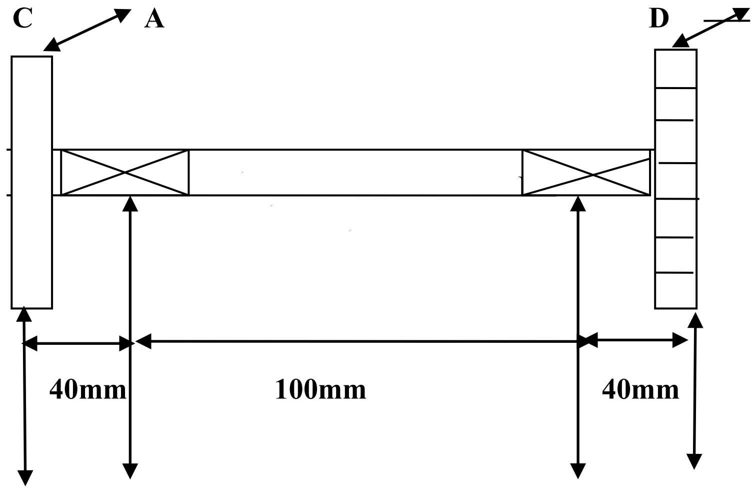
$$v = \frac{1}{2}h(A_1 + A_2 + \sqrt{A_1 A_2}) \quad \ldots\ldots\ldots\ldots (29)$$

Where, V=volume of the hopper =231104 m^3 and A_1 and A_2= Area of the top and base
The shredding plate is placed at an angle of 45°, and the Area of the shredding plate was calculated as 36550mm2.

Area of the shredding plate = length of the shredding plate × the width of the plate ….(30)

Where, Length=205mm and width =170mm

(W_d) = Load due to the Rotating disc. Vertical Load due to bevel gear. (W_G) = from the electric motor.



RA= 124.76 **RB**=53.84

Fig 2: Showing the arrangement of the forces acting on the shaft

Where W_d = weight of the rotating disc=109N; and W_G = weight of the bevel gear=2127.4N

Capacity of the prime mover (Electric motor)

The capacity of the prime mover required to drive the machine was determined based on the twisting torque on the shaft the optimum known speed of the shaft. According to Ugoamadi (2012), the power (P) required to drive this slicing machine is expressed in equation 31

$$p = \frac{(T_1 - T_2)v}{1000}, kw \ldots\ldots\ldots\ldots\ldots\ldots\ldots (31)$$

Where N = speed of the shaft=99rpm; Torque on the shaft= 72330Nmm. Thus P=0.75kw (≈1hp).

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Performance tests

In the analytical test of the performance of the slicing machine, five experiment tests run for both cooked and uncooked cassava root was made after the machine was produced. The cassava root was weighed, peeled, washed and was fed through the hopper for slicing. During the five experiment test run for the performance, the optimal time for slicing each cassava root was determined. In addition, the weight of the cassava roots was also determined after cutting at the same speed of the shaft, 99rpm. During the experiment, the performance of the machine was evaluated by determining the machine capacity, Pc = (kg/h) and Efficiency, η(%) of the engine and Average slicing time for both the boiled and unboiled cassava root were developed as in equations 32 and 33, respectively,

$$P_c = \left(\frac{W}{h}\right) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (32)$$

Where, W = Average weight of cassava after slicing for either cooked or Uncooked;

Average slicing time

$$\text{Efficiency } (\eta) = \frac{\text{Weight of cassava sliced(Kg)}}{\text{total weight of cassava(Kg)}} \times 100\% \ldots\ldots (33)$$

Table 2 shows the five experimental test runs for the boiled cassava roots.

Table 2: Five test run for boiled cassava roots

The original weight of cassava before slicing(g)	The average weight of cassava after cutting (g)	Average slicing time (sec)
94.3	84.7	15
164	151.9	25
105	96.1	19
98.6	88.3	17
122.7	110.8	20
Total: 586	**531.8**	**96**

Table 3 shows the five experimental test runs for the unboiled cassava roots.

Table 3: Five test run for unboiled cassava roots

The original weight of cassava before slicing(g)	The average weight of cassava after cutting (g)	Average slicing time (sec)
95	83.2	10
110	101.3	18
101	92	12
167.5	156.8	22
119.2	107.8	20
Total: 592.2	**541.1**	**82**

The Average slicing time for the five experimental test runs for unboiled and boiled cassava roots is calculated using equation 34 thus;

$$\text{Average slicing time} = \frac{\text{Total average time of either cooked cassava root}}{\text{Number of experiments taken}} \text{.......}(34)$$

But since we tested for both the cooked and uncooked cassava root, the overall Efficiency, capacity and overall average slicing time of the machine was calculated using equation 35 thus;

$$\text{Overall efficiency} = \frac{A+B}{2} \text{............} (35)$$

Where, A= Efficiency for cooked cassava root and B= Efficiency for uncooked cassava root.

And the overall Capacity (Throughput) for the uncooked and cooked cassava root for the machine using the relation above is calculated using equation 36 thus;

$$\text{Overall Capacity} = \frac{C+D}{2} \text{..............}(36)$$

Where, C= capacity (throughput) of the cooked cassava root and D= Capacity (throughput) of the uncooked cassava root

Discussion of Results

The results of evaluation for the cassava slicing machine according to Table 2 and 3 shows that the optimum Efficiency of the boiled and unboiled cassava root is 90.7% and 91.4%. Still, the overall average efficiency of the machine for both cooked and uncooked cassava root is obtained to be 89.5%. Similarly, the capacity (Throughput) of the machine for the cooked and uncooked is calculated as 22.1kg/hr and 23.5kg/hr. Still, the overall average capacity of the machine for both was obtained as 22.8kg/hr with an average slicing time of the machine acquired to be 17.8secs for both the cooked and uncooked at a speed of 99rpm. The developed cassava root slicing machine was found to perform satisfactorily with all the mass of the preconditioned cassava root fed into it at a constant operational speed of 99rpm when tested.

Cost estimation of the cassava slicing machine

The materials used in the fabrication of various components of this machine and the labour costs involved were quantified and presented in Table 4. The total cost of producing the slicing machine is sixty-seven thousand and two hundred naira only (N67,200).

Table 4: Bill of engineering materials and evaluation for the cassava slicing machine

S/no	Material Description	Quantity	Unit price ₦	Amount ₦
1	$1\frac{1}{2}$ inch angle iron	1length	2800	2800
2	1inch angle iron	1lengths	1150	1150
3	$1\frac{1}{2}$mm thick stainless-steel plate	1 piece	6000	6000
4	$1\frac{1}{2}$ mm thick mild steel plate	$\frac{1}{2}$ sheets	13000	6500
5	0.75kw (1hp) geared motor	1units	18000	18000
6	φ20mm x 500mm shaft	1 piece	2000	2000
7	Bevel gears	2 piece	2500	5000
8	Bearings PL 204	2 piece	2500	5000
9	φ180mm slotted flat pulley	1	6000	6000
10	Small ball bearings	12 pieces	200	2400
11	Electric cable (Workshop flux)	4 meters	350	1400
12	3 phase electric switch	1	1500	1500
13	φ12mm x 300mm rod	1 piece	250	250
14	Bolts and nuts	Lot	Lot	800
15	Spray paint (metallic)	2 tins	1200	2400
16	Transportation/ contingencies	-	-	6000
	Total			**67,200**

Conclusion

The design and development of a reciprocating motion cassava slicing machine are developed using locally available cheap materials and technology, affordable to all categories of farmers, processors and producers because of the low cost of production (N67,200.00). According to the analytical test run of the project, it was noted and observed that the machine is very efficient and has the overall capacity (Throughput) of cassava slicing machine for cooked and uncooked calculated as 22.8kg/hr, at a specified speed of 99rpm. The thickness of the range of 1mm-3mm, and most of the materials are made of stainless steel to avoid food contaminations due to corrosion and rust. The use of the machine is encouraged since it is of high output, affordable and cheap, requires low labour input, Low maintenance is needed and is very effective and efficient. Therefore, farmers and producers should adapt to boost their products. Petrol/diesel-powered engines can also be used to operate this cassava root slicing machine in areas of constant outage of power supply. Therefore, it is recommended that manufacturers go into mass production of this machine further to reduce its cost through bulk purchase of raw materials. Also, automation of the cassava slicing machine should be recommended to determine the optimum performance.

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