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Design and Fabrication of Variable Chip Size Plantain Slicing Machine

[1]*Ojariafe, G.A.* & [2]*Inegbedion, F.*

[1]Mechanical Engineering Department University of Benin, P.M.B 1154, Benin City, Edo State. Nigeria.
[2]Production Engineering Department University of Benin, P.M.B 1154, Benin City, Edo State. Nigeria.
Email: ga.ojariafe@uniben.edu, francis.inegbedion@uniben.edu

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Abstract

This paper describes the design and fabrication of a variable chip-size plantain slicing machine. Traditional and conventional plantain chip slicing with the use of hands and knives causes injury, is time-consuming, is unhygienic, and results in less product output; hence the design and fabrication of a variable-size chip plantain slicing machine that has the advantage of flexible adjustability for slicing different plantain slices. The design and fabrication of this machine were done using locally available materials and technology. The slicing machine thus produced from this research has the following parameters: 0.34 kW of electric power, and the motor speed was 1400 rpm. The shaft deflection was 0.0112, which has no significant effect on the cutting dimensional accuracy of the machine. It takes the fabricated plantain slicing machine an average of 5 seconds to slice. A factor of safety of 1.5 was considered. The plantain slicing machine was successfully designed, fabricated, and tested for its efficiency. A maximum efficiency of 80.95% was achieved with the operation during testing after the machine had been fabricated. This shows that the machine was effective.

1. Introduction

Plantain known also by its scientific name as Musa spp. is a fruit commonly consumed by many people around the world especially in Nigeria where it is consumed in various ways as boiled lumps, eaten whole, roasted (locally known as bole), fried chips (locally known as kpekere or dodo plants yielding fruits that remain, such as food flour for making thick food paste (locally known as elubo). Africa is thought to produce more than half of the world's plantains [1]. The region extending from the lowlands of Guinea and Liberia to the central basin of the Democratic Republic of the Congo produces the bulk (82%) of the plantains consumed in Africa. Contributions from West and Central Africa are 61 and 21%, respectively [2]. According to estimates, more than 25% of the inhabitants in West and Central Africa, or around 70 million people, rely on plantains for their carbohydrate intake and considering the over 200 million human population the food crop remains a staple food with high consumption rate and export value which can generate income and create employment. The postharvest processing of plantain often results in the production of a long fruit measuring between 30mm to as big as 60mm diameter [1] which can be cut into radial or longitudinal pieces for further processing for consumption or industrial purposes. The plantain chips cutting or slicing is traditionally or conventionally done using the hand and knife which could result to injury. The conventional method of slicing plantain is also time consuming, unhygienic and have less product output. Considering this large scale consumption of plantain and the attendant disadvantages associated with the cutting of plantain, mechanized methods that utilized machineries with various configurations and efficiencies have been proposed by many researchers. the current work is focused on designing and producing a plantain slicing machine that has an added advantage of flexible adjustability of cutting different plantain slices.

2. Materials and Method

The motorized circular blade plantain slicer of a small cylindrical hopper that allows the vertical or inclined insertion of a plantain fruit that drops by gravity and have contact with a rotating blade. The rotating blade is driven by an electric motor and they are both located underneath the frame of the machine. The blade is fixed in such a way that it can be adjusted along a fixed vertical rotating shaft so that the cutting edge can cut the plantain fruit at different pre-set distance along the length of the fruit as it drops from the hoper to the receiving plate.
To consider the most viable for production among three concepts of plantain slicing machine, an evaluation was done on the basis of significant weighted design criteria using a decision matrix, as shown in table 1. From the decision matrix Table, the concept 3 with the highest weighted total was selected.
Table 1: Decision matrix table

Weighted Criteria	Weighting	Concepts					
		Concept 1		Concept 2		Concept 3	
		Score	Total	Score	Total	Score	Total
Flexibility of slice	5	2	10	2	10	5	25
Continuous slicing operation	4	1	4	3	12	5	20
Manufacturability	3	5	15	3	9	4	12
Simplicity	2	5	10	1	2	2	4
Low cost of production	1	5	5	1	1	2	2
Total			44		33		63

From the decision matrix table with scale weight of 1 to 5, the significant criteria considered favoured the concept 3 with a weighted criteria average of 63 points compared to the concept 1 with 44 points and the concept 2 with 33 points.

2.1. Detail Design

Input parameters for the design of the plantain slicer include
Diameter of plantain (max) = 35 to 40mm
Length of plantain (max) = 150-200mm
Average weight of single plantain = 201.43g
Cutting load per unit width = 30N/40mm = 0.8N/mm
Nature of blade; stainless steel strings
Diameter of string blades; 0.8mm
Average specific gravity (dimensionless) = 1.005 [3]
Force required to cut plantain (max) = 30N [3]
Working load of string blade; 15-25% of tensile strength of the steel material (360Mpa)
Nature of loading = batch loading
Number of plantains per loading (max) = 10
Shape profile of slice = radial and longitudinal
Thickness of sliced plantain = variable depending on slicing plate adjustment

2.2 Power Required to cut a Single Plantain

One plantain fruit gives an average of 12 chips. (measured but may vary with thickness)
Energy required to cut a slice of plantain (measured value) = 0.0736 kgm [3]
Torque (T) required to cut a slice of plantain = Energy x Gravitational Acceleration

Torque, T = 0.0736 x 9.81 = 0.72 N-m
Considering a factor of safety of 1.5;
Required Torque, T = 0.72 x 1.5 = 1.08 N-m
But speed of motor is 1400m/s
Power needed to cut a plantain is given as;
P = Tx speed of cut (1)
= 1.08 x 1400 = 1512Watts
Power required to slice the plantain = 1.5kW.

2.3. Bearing Selection

The bearings are attached to the slicing tray and it bears the load of the tray and the plantain while also allowing for the less restrictive movement of the slicing tray along the rack.
Life of bearing for machines used intermittently of short period of time e.g. agro machineries = 4000 – 8000 hours [4]
Life L in revolution of the bearing = $60N.L_H$ (2)
Where;
N = number of revolution per minute of the bearing in operation,
L_H = life in working hours of the bearing
L = 60 x 1400 rpm (of motor) x 6000hours = 504000000 revolutions
But L = $\left(\frac{C}{W}\right)^k \times 10^6$ (3)
Where;
C = basic dynamic loading, W equivalent dynamic load, k = 3 for all ball bearings and 10/3 for roller bearings
For the nature of machine where the load on the bearing acts perpendicular to the direction of motion of the element, radial bearing is selected.
The bearing bore diameter equivalent to shaft diameter = 10mm corresponding to bearing no 200 with outside diameter of 30mm and width of 9mm
$W = X.V.W_R + Y.W_A$ (4)
Where; X = radial load factor, Y = axial load factor, V = rotational factor =1
W_A = axial load = weight of plantain + weight of plate + weight of tray = 2+ 30 +30 = 62N, W_R = radial load= 62N (assumed upper limit since it could have ordinarily been lesser than W_A but W_A/W_R and W_A/Co are required for the bearing selection. Since W_A/Co is unknown, 0.5 is selected and from tables, the corresponding values for X = 0.56 and Y = 1
Therefore; W = 0.56 x 1 x 62 + 1 x 62 = 96.72N
From equation (2),
C = 2152 $\left(\frac{504}{(10\text{^}6)} \times 10^6\right)^{1/3}$ = 770N = 0.770kN
From tables, bearing with basic capacities of Co = 2240N and C = 4000N is selected
Then W_A/Co = 62/2240 = 0.03
From tables, X = 1 and Y = 0
Substituting values into equation (4)
W = 1 x 1 x 62 + 0 = 62N
Therefore; basic dynamic load rating C = 62 $\left(\frac{504}{(10\text{^}6)} \times 10^6\right)^{1/3}$ = 493.52N = 0.5kN
From tables, the bearing number 200 having C = 4kN may be selected.

2.4. Area of Sheet metal for production of machine casing

From observation of existing machines of the likes of the proposed concept, measurement was selected for the various dimensions of the machine casing as follows
Length of casing = L= 460mm
Breadth of casing = B = 270mm

Height of casing=h = 280mm
Diameter of circular cut-out from top surface of casing= d mm = 220mm
Surface area of material of casing=Total Surface area of material of casing – Area of cut out materials
The cut-out materials from the casing are the top circular and rectangular shaped surfaces cut out from the material for final shaping of the casing.
Total surface area of the rectangular casing assuming it is a complete hollow box material = $L_1B_1+L_2B_2+L_3B_3+$……………………..L_nB_n (5)
Where n = the nth term number of surface of the rectangular casing in arithmetic progression.
Area of cut-out circular material= $\frac{\pi d^2}{4}$
where π=3.142,
Area of rectangular cut-out material=l x b
Total surface area of cut-out materials = $\Sigma(L \times B)_n - (\frac{\pi d^2}{4} + l.b)$ (6)
Hence surface area of the casing material = $[L_1B_1+L_2B_2+.L_{n+1}Bn_{+1}-\{(l \times b\} + \frac{\pi d^2}{4})\}$ (7)
For any cut-out shape profile from the material of casing, the final surface area of material of the casing= Total surface area of all solid casing-(Summation of all Areas of cut out shapes of the material).
I.e. SA= TSA-(a_1+………a_{n+1}), where SA=surface area, TSA= total surface area, a=area of cut out shape of material, a_n= nth term area of cut out shapes.
$L_1B_1=L_2B_2$= 0.28X0.27= 0.0756m^2
$L_3B_3=L_4B_4$=0.46X0.28= 0.1288m^2
$L_5B_5=L_6B_6$=0.46X0.27= 0.1242m^2
Therefore, TSA= 0.3286m^2

2.5. Pulley design
The pulley system schematic is shown in Figure 1, where c is the center to center distance

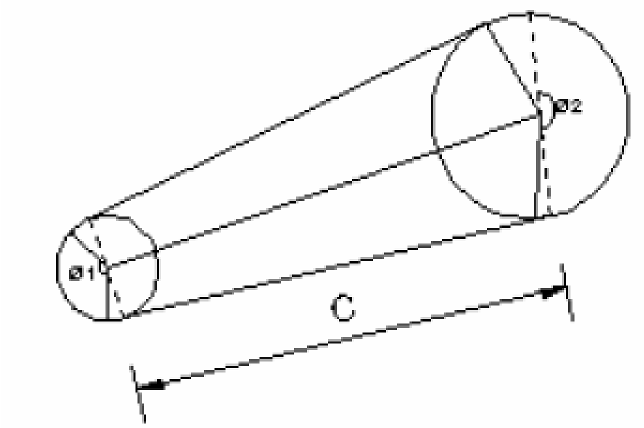

Figure 2: Pulley system

(i) The ratio of speed transmission to be x:y= 3:1 for adequate speed reduction. This is necessitated for proper sizing of the driven pulley and avoiding unnecessary speed of the rotor
(ii) Coefficient of friction between belt (leather tanned) and pulley (Cast iron) is μ = 0.35. The combination of the material for the belt and the pulley is necessitated for efficient function
(iii) Angle grooving of the pulley, is θ° = 40°, for the best performance of belt.
(iv) Diameter of small pulley = D_s = 50mm (attached to electric motor as supplied)
(vi) Diameter of big pulley = D_1
From the relationship, the center distance, c between the two pulleys is taken as the larger of the value between

$\frac{\mathbf{3Ds}}{\mathbf{2}}+\frac{\mathbf{DL}}{\mathbf{2}}$ And c = D_L, [5]

Therefore c = max ($\frac{\mathbf{3Ds}}{\mathbf{2}}+\frac{\mathbf{Dl}}{\mathbf{2}}$ and D_L) (8)

From Figure 1,

$\theta_1 = 180° - 2\sin^{-1} = (\frac{\mathbf{DL-Ds}}{(\frac{2}{c})})$ (9)

$\theta_2 = 180° + 2\sin^{-1} (\frac{\mathbf{DL-Ds)}}{(\frac{2}{c})})$ (10)

From the relationship,

D_L= 3Ds

Therefore, D_L = 3×50=150mm

Where D_L =150mm is the diameter of the large pulley, and Ds is the diameter of the smaller pulley.

The center distance, C between the two pulleys is taken as the larger of the value between $\frac{\mathbf{3Ds}}{\mathbf{2}}+\frac{\mathbf{DL}}{\mathbf{2}}$ and C = D_L,

Therefore C = max ($\frac{\mathbf{3Ds}}{\mathbf{2}}+\frac{\mathbf{D1}}{\mathbf{2}}$ and D_L) (11)

That is c = ($\frac{\mathbf{3(50)}}{\mathbf{2}}+\frac{\mathbf{150}}{\mathbf{2}}$ or 150),

Therefore, c = (150 or 150) = 150 mm.

From Figure 1 we also have,

$\theta_1 = 180° - 2\sin^{-1} = (\frac{\mathbf{DL-Ds)}}{(\frac{2}{c})}) = 180^0\ 2\sin^{-1} 0.3333 = 141^o$

$\theta_2 = 180° + 2\sin^{-1} (\frac{\mathbf{DL-Ds)}}{(\frac{2}{c})}) = 180^0 + 2\sin^{-1} 0.3333 = 219^o$

2.6. Shaft design

i. Shear stress on the shaft:

Shearing stresses are induced in the shaft due to the fact that it is subject to a torque or twisting moment. The shear stress produced in the shaft is given as:

$$\tau = \frac{Tr}{J} \quad (12)$$

Where

τ = shear stress (MPa)

T = twisting moment (Nm)

r = distance from center to stressed surface of the shaft in (mm)

J = "polar moment of inertia" of cross section (mm^4)

The maximum moment on the Shaft

The maximum moment in the circular shaft can be expressed as:

$$T_{max} = \frac{\delta j}{R} \quad (13)$$

Where

T_{max} = maximum twisting moment (Nm)

τ_{max} = maximum shear stress (MPa)

R = radius of shaft (mm)

J = the polar moment of inertia on the shaft can be expressed as

$= \frac{\pi R^4}{2} = \frac{\pi D^4}{32}$ for round solid shaft or $\frac{\pi(do^4-di^4)}{32}$ for hollow shaft (14)

d_o and d_i are the outer and internal diameter of the hollow shaft respectively

Substituting for J in equation (13), we have

$$T_{max} = \frac{\pi R^4 \tau_{max}}{2R} = \frac{\pi R^3 \tau_{max}}{2} = \frac{\pi D^3 \tau_{max}}{16} \quad (15)$$

But for a hollow solid shaft, equation (14) and (15) are expressed in terms of the outside and internal diameter of the shaft as follows, J = $\frac{\pi(R^4 - r^4)}{2} = \frac{\pi(do^4 - di^4)}{32}$ (16)

and

T = $\frac{\pi}{16}\ x\ \frac{\tau_{max}[do^4 - di^4]}{d}$ (17)

R = $d_o/2$, and r = di/2

Note: D = diameter of shaft and it is given as

$(\frac{T_{max}}{\tau_{max}})^{1/3}$ (18)

But recall, T = 1.47Nm as calculated (refer to chapter two; torque calculation).

If an allowable shear stress is taken (31 to 47MPa for alloy cast steel and iron)

We take 45MPa,

Then inputting this value of T and τ into equation (17), we have

D = $1.72(\frac{1.47\ x\ 1000}{31})^{1/3}$ = 6.2mm say 1.2cm.

ii. Torsional deflection of the shaft:

The angular deflection of a torsion solid shaft can be expressed as

$\theta = \frac{584LT}{GD^4}$ (19)

Where;

θ = angular shaft deflection (degrees)

L = length of shaft

T = torque transmitted by shaft in

G = modulus of rigidity (MPa)

D = diameter of shaft = (5 x 8 x 4 x 10 x 1.47)/ 210 x 10^3 x 10^4 = $0.0112(1)^4$ = 0.0112^0

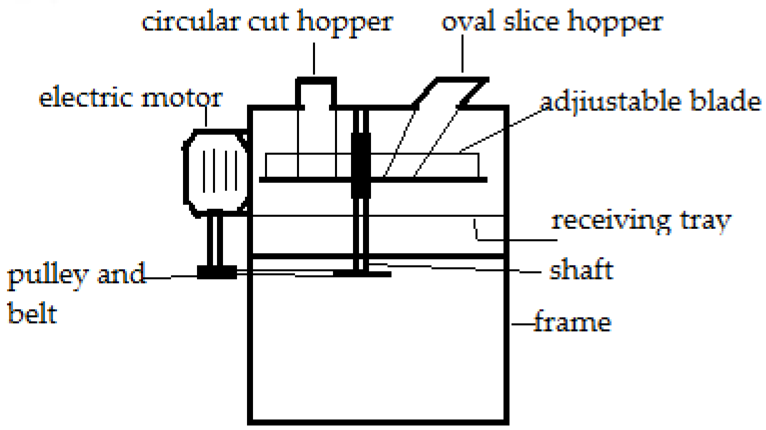

Figure 3a. Concept 3 motorized circular blade plantain slicer

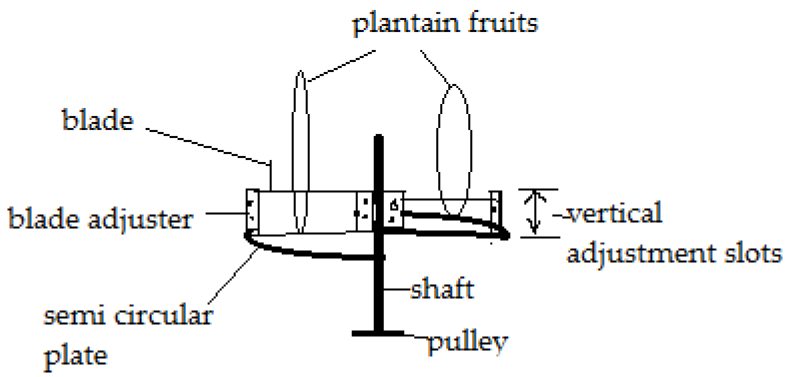

Figure 3b. Cross section of slicing blade assembly of concept 3

3.0. Results and Discussion

The results obtained from the use of the fabricated plantain slicing machine is shown in Tables 2.

Table 2: Results of sliced ripen plantain

Trials	No of slices	No of uniform slices	No of defective slice	Time to slice (sec)
1st trial	13		2	5
2ndtrial	12		2	4
3rd trial	13		2	5
4th trial	12		2	6
5th trial	13		3	5
Total	63		11	25

From Table 2 it can be observed that for the five trials for an average of 51/5 = 10.2 good plantain slices gotten, an average of 11/5 = 2.2 slices was bad for every trial.

From the Table 2, the machine was able to slice plantains with high precision and few errors in the slice resulting in the non-uniform slices. From the five trials slicing operation carried out, the good sliced plantains which conformed to the slicing blade and plate spacing were more than the non-uniform slices.

The Efficiency (e) of the machine in slicing the ripe plantain = $\frac{51}{63}$ x 100 = 80.95%.

It also took the fabricated plantain slicing machine an average time of $\frac{25}{5}$ = 5 seconds to slice the ripe an average sized plantain.

Production Rate = 12 plantain slices in 5 seconds

This connotes that the machine slices 13 pieces of plantains in 3 seconds.

Average weight of Plantain slice (measured) = 0.0101kg

Weight of 12 Plantain = 0.0101 x 13 = 0.13kg

Capacity in kg/sec = 0.13/3 = 0.043

Capacity in kg/hour = 0.043 x 60 x 60 = 156kg/hr

The machine was tested for various sizes of plantain and result showed that the nature of the plantain (ripe and unripe) affected its slicing efficiency. This was because too much ripe plantains stoked to the slicing blade and plate and it caused a slowing down of the cut through. However, the smearing of the blades was minimized by the guiding loading cylindrical hopper.

4.0. Conclusion

The plantain slicing machine was successfully designed, fabricated and tested for its performance efficiency. The output throughout of the machine was calculated to be 156kg per hour. This was an improvement on some reviewed plantain slicing machines. The aim of the research which is the design and fabrication of variable chip size plantain slicing machine and the objectives were achieved.

References

[1] Food and Agriculture Organisation of the United Nations (FAO), (1990)
[2] Food and Agriculture Organisation of the United Nations (FAO), (1986)
[3] Sonawane, S.P., Sharma, G.P., and Pandya, A.C., (2011): Design and development of power operated banana slicer for small scale food processing industries. Res. Agr. Eng., 57: 144–152.
[4] Khurmi R. S., and Gupta J. K., (2005): A Textbook of Machine Design, 14th Edition, Eurasia
[5] Kartika, S.B, Arahanth, 2013. Design and development of a potato slicer. IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE) ISSN: 2278-1684, PP: 21-26
[6] Akinyemi, S. O., Aiyelaagbe, I. O., & Akyeampong, E. (2010): Plantain (Musa spp.)

Cultivation in Nigeria: a Review of Its Production, Marketing and Research in the Last Two Decades. Acta Horticulturae, 1, 879, 211-218.

[7] Azeez, A.A 2012.Development of a motorized plantain slicer. A project submitted to the Department of Agricultural Engineering, College of Engineering. Federal University of Agriculture, Abeokuta, Ogun state Nigeria in partial fulfilment of the requirement of the award of bachelors of Engineering

[8] Ezugwu, C.A, Osueke, A. O. Onokwai, R. S. Diarah, T. M. Olayanju, S. O. Braimoh, (2019). Design and Fabrication of a motorized/ powered operated plantain slicer for optimum chips production Food and Agriculture Organisation of the United Nations (FAO), (1987): Root and Tuber Crops, Plantains and Bananas in Developing Countries: Challenges and Opportunities. Rome, Italy; Plant Production and Protection Paper No. 87.83

[9] International Institute of Tropical Agriculture (IITA), (1998): Plantain and Banana Improvement Program-Annual Report for 1997. International Institute of Tropical Agriculture, Onne, Nigeria.

[10] International Institute of Tropical Agriculture (IITA), (2015): Banana and Plantain, retrieved from<http://www.iita.org/banana-and-plantain> on 26th March, 2015. Publishing House (Pvt.) Ltd., Ram Nagar, New Delhi.

[11] Marriott J. and P.A. Lancaster (1983): Bananas and Plantains, In: Chan, H.T. (ed.):
Handbook 6 of Tropical Foods. Marcel Dekker, New York, USA. pp. 85-143.

[12] Mishra, A.A. 2013. Design and fabrication of twisted potato crisps maker. 2nd international conference and exhibition on food processing and technology, October 28-30, 2013 Kansas City USA

[13] Okafor, B.E., and Okafor, V.C., (2013):Design of a Plantain Chips Slicing Machine.
International Journal of Engineering and Technology Volume 3 No. 10

[14] Okonkwo, J.C., Nwokocha, H.N., and Ifenkwe, O.P. (1986) Survey Report on Irish Potato Production and Storage in Local Government Areas of Plateau State, Annual Report, National Root Crops Research Institute, Umudike, Umuahia, Nigeria

[15] Olawale, O, F.C. NnajiInternational Journal of Mechanical Engineering and Technology
(IJMET) Volume 10, Issue 01, 2019, pp. 1485–1495, Article ID: IJMET_10_01_151

[16] Oyedele, O. A., Adeyanju O. O., and Owolabi, H. B. (2018).design, fabrication and testing of a dual powered plantain slicer.World Journal of Engineering Research and Technology. Vol. 4, Issue 2, 116-125

[17] Tchango, T. J., Bikoi, A., Achard, R., Escalant, J.V., and Ngalani, J.A. , (1999): Plantain:Post-Harvest Operation, Food and Agriculture Organisation, Cameroon.

[18] Zhou D., Claffee M., Lee K. and McMurray G., (2006): Cutting, ‘by Pressing and Slicing. Applied to the Robotic Cut of Bio-materials, Part I: Modeling of Stress Distribution, in the Proceedings of the IEEE International Conference on Robotics and Automation (ICRA06)