Design and Development of an Improved Plantain Slicing Machine

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Abstract

In this work, a simple efficient and economical plantain slicing machine was designed and constructed. The entire machine is a slider-crank mechanism driven by an electric motor via a v-belt, with the driven pulley borne on a shaft supported by two bearings. The slider comprises of the hopper (which houses the plantains), guide and stanchion. The mechanism consists of two cranks, each on either side. They are linked to the slider with the aid of a connecting rod through which motion is transmitted to the hopper (slider). The to and fro motion of the slider along the bed of the guide (which houses the cutting blade) achieves the desired chip formation. The machine has average efficiency of 93%, compared with 87% and 63% of chip cutter and domestic knife respectively, at tremendous reduction of time, cost of production, labour and operating capacity of 0.1124 /s (a plantain for 9s).The efficiency of this machine is far more than the existing ones. From the results obtained during performance evaluation, the average time required slicing 30 pieces of plantain of average length of 260mm, weight of 0.3kg and diameter of 65mm using domestic knife, chip cutter and the plantain slicer were approximately 1986s, 1487s and 267s respectively. Though this plantain slicing machine produces chips of uniform thickness like chip cutter but with higher efficiency and minimal human input. This improved machine is simple, hygienic, portable, detachable, easy to operate and maintain. It is suitable for both small and large scale productions of plantain chips.

Keywords: Plantain slicing machine, chip cutter, domestic knife, efficiency.

1.0 Introduction

The need for the preservation and processing of plantain and the high demand for plantain chips necessitate the design and construction of Plantain Slicing Machine (PSM) commonly called Plantain Slicer.

Slicing of plantain have been carried out over the ages with the use of manual methods like the domestic knife, chip cutters and graters producing a non-uniform chip which is laborious, time consuming, liable to cause injury, less effective and unhygienic and could only be effective for domestic purpose. Mechanized slicer of higher efficiency was developed to overcome the shortcomings of the manual slicers. The mandolin slicers which are general fruits and vegetable slicers though small and manually operated are very efficient (Mandoline, 2004); the Knott's slicer is a leading international manufacturer of potato and plantain slicer have developed slicers ranging from low speed to high speed and automatic slicers which are quite effective, less laborious, time and cost saving (Knott, 2006). These slicers are not affordable if at all available for domestic and commercial use. Thus, the need for the development of locally made plantain slicer is relevant. Obeng (2004) developed a mechanised plantain slicer; it seeks to reduce the drudgery associated with traditional cutting of large-scale plantain into non-uniform chips. Adewunmi (2007) writes on the performance characteristics of manually operated and simple equipment for plantain chipping and revealed that plantain ripeness has significant effect on the machine efficiency whereas plantain diameter does not have effect on machine efficiency.

The main objective of the research is to develop low cost and improved higher efficient machine that can facilitate slicing of bulk plantains hygienically into chips. Plantain sliced with the improved slicer can be fried into chips that can be preserved longer for local and foreign markets.

2.0 Materials and Method

2.1 Machine Description

The entire machine is a slider-crank mechanism driven by an electric motor via a v-belt, with the driven pulley borne on a shaft supported by two bearings. The slider comprises of the hopper (which houses the plantains), guide and stanchion. The mechanism consists of two cranks, each on either side. They are linked to the slider with the aid of a connecting rod through which motion is transmitted to the hopper (slider). The to and fro motion of the slider along the bed of the guide (which houses the cutting blade) achieves the desired chip formation.

The machine comprises connecting rods and guide. The guide is a spring loaded lever system which resists the cutting force in order to keep the plantain in place. The machine is driven by a one horse power electric motor via a v-belt connected to the one of the two cranks which transfers the motion to the other crank attached to the hopper. To achieve a portable cost effective motor driven slicer, to overcome the limitation of the manual slicers, cost effective in terms of meeting local demand and simple in operation for domestic purposes, the machine which has a capacity of 30 plantains per operation is designed to cut longitudinally and can also be adapted to cut laterally producing chips of uniform thickness.

The machine is constructed using engineering materials such as mild steel, stainless steel and wood. The hopper which houses the plantain is constructed using stainless steel because of its corrosion resistance since it's the part that comes in contact with the plantains. The other metal components are made of mild steel while the crank is made of wood to reduce the overall weight of the machine. All parts are made and joined to form a unit using basic engineering manufacturing techniques such as marking out, cutting, welding and fasteners.

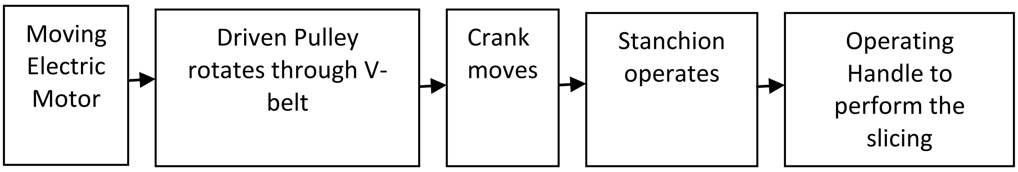

Figure 1: Schematic Illustration of the Machine.

2.2 *Design Considerations and Design of Machine Elements*

The choice of material was influenced by the cost of production, availability of the material locally, rigidity and strength of the machine, overall weight and the resistance to corrosion and rust. This was carefully chosen without compromising the efficiency of the machine and also to reduce cost so that it could be affordable for both commercial and domestic uses. Hence, the following major machine elements capacities, parameters and geometries were designed: Belt, Electric motor, Spring, and Shaft, among others.

2.2.1 Determination of belt (Cross Sectional Area): The cross sectional area of belt is expressed as:

$A = (W_1 + \frac{W_2}{2}) \times t$ (Akintunde, 2007) (1)

where, W_1 = top width of V-belt =13 mm, $W_2 = \frac{t \times W_1}{h}$ bottom width of the V-Belt, and t is nominal belt thickness (8 mm) and h is height of the groove, expressed as

$h = 0.5 \times W_1 \times \tan\beta$ (2)

where β is the arc of contact, expressed as $\beta = \frac{180-\theta}{2}$; Θ is the groove angle of sheave (between 30^0 - 40^0)

Taking Θ to be 32^0 (Khurmi, 2004), $\beta = 74^0$ and h = 22.7 mm and W_2= 4.58 mm.

Therefore, Cross Sectional Area, A of V-belt is determined to be 122.35mm

2.2.2 Determination of Tension and Torque in Belt

$T_{max} = \frac{T_1}{A}$ (Khurmi, 2004) (3)

where T_{max} is the maximum tension in belt = 2×10^6 N/m^2 (standard) (Akintunde, 2007), A is the cross sectional area of belt, and T_1 is the tension on the tight side and is estimated to be 245N

The maximum centre distance of belt (Khurmi, 2004) ,

$C = 2(d_1+d_2)$ (4)

Where d_1 = diameter of small pulley = 80mm, d_2 = diameter of large pulley = 250mm and C = 660mm,

Angle of wrap of large pulley (Khurmi, 2004),

$\alpha_2 = 180^o + 2\text{Sin}^{-1}(\frac{r_2-r_1}{C})$ (5)

$\alpha_2 = 194.8^o = 3.4$ rad/s

The tensions T_1 and T_2 is related by

$2.3\log(\frac{T_1}{T_2}) = \mu.\alpha.\cos\alpha_2$ (6)

where μ is Friction coefficient and is 0.25 for rubber, (Akintunde, 2007), T_2 is the tension in the slack side = 166N, The torque T in the belt is expressed as (Khurmi, 2004);

$T = (T_1-T_2) \times r_1$ (7)

Where r_1 is the radius of the smaller pulley; therefore, T = 3.16Nm.

2.2.3 Power and Force Transmitted by the V-Belts: The power transmitted by the belt is expressed thus

$P = (T_1-T_2) \times V_1$ (8)

Estimated to be 470W

Torque, T = Force F× radius r

P = $\frac{V_1}{r}$ × T = $\frac{V_1}{r}$ × F × r

Hence, the force transmitted by the belt is expressed as

F = $\frac{P}{V_1}$ (9)

Therefore, F = 79N.

2.2.4 Axial Load that the spring must overcome: The spring is designed such that it can withstand the force being transmitted from the electric motor to the slider. The weight of the slider and plantain was subtracted from the motor force to determine the force that the spring overcomes during chipping operation.

W_S= Weight of the slider and plantain= 252.24N; F_F= Frictional force = 252.24×0.15 = 38N; and Axial load = (79-38) N = 41N

2.2.5 Shear Stress induced in the Spring: The spring employed in this design is a helical spring. The shear stress induced in the spring due to the axial load with the neglect of friction is expressed as;

τ = the shear stress in the spring = $\frac{8KFD_o}{\pi d^3}$ (Khurmi, 2004) (10)

where, K = shear stress correction factor = (2C+1)/2C (11)

Therefore, K= (2×7.5+1) / (2×7.5) = 1.07 and τ =$\frac{8\times1.07\times79\times0.35}{\pi\times(0.004)^3}$ = 11.8 MN/m^2

2.2.6 Deflection to Axial Load

y = deflection in spring = $\frac{8FD^3n}{Gd^4}$ (Khurmi,2004) (12)

where, n = number of effective coils = (L/π D) = 3, G = modulus of rigidity for spring steel = 80 GN/m^2; therefore, y = $\frac{8\times79\times(0.035)^3\times3}{80\times10^6\times(0.004)^4}$ = 3.97m

2.2.7 Speed Ratio and Belt Speed: The speed ratio of the driving pulley and the driven pulley is expressed as:

$N_1/N_2=d_2/d_1$ (Khurmi, 2004) (13)

where N_1= speed of electric motor or driving pulley =1420 rpm, N_2= speed of the driven pulley, d_1 = diameter of the driving pulley = 80mm, and d_2 = diameter of the driven pulley =250mm

Hence, N_2= $\frac{1420\times80}{250}$ = 454.4rpm

The Belt Speed, V_1 was calculated using*:*

$V_1 = \frac{2\pi N_1 r_1}{60}$ (Khurmi, 2004) = $\frac{2\times\pi\times1420\times40}{60\times1000}$ =5.95 m/s (14)

2.2.8 *Cutting velocity*

$V_2 = \frac{2\pi N_2 r_2}{60}$ (Khurmi, 2004) = $\frac{2\times\pi\times454.4\times40}{60\times1000}$ = 0.06 m/s (15)

2.2.9 Power and Torque: The torque transmitted from the electric motor to the machine is expressed as:

$T=(T_1-T_2)\times r_1$ (V-belt Design Manual) (16)

where T is the torque transmitted by the electric motor; T_1 and T_2 are the tensions on the tight and slack sides of the belt respectively. From the relationship, the torque, T = 3.16 Nm

The power developed is therefore expressed as:

$P = \frac{T\times V_1}{1000}$ (Akintunde, 2007) = $\frac{3.16\times5.96}{1000} = 0.019$kW (17)

2.2.10 Determination of the maximum bending moment in the shaft: Weight of the slider, W_S is measured to be 252.24N; Volume, V^1 of the crank is estimated as:

$V^1 = \frac{\pi D^2 t}{4}$ (18)

where D is the diameter = 40cm and t is the thickness = 0.5cm; therefore, $V^1 = \frac{\pi D^2 t}{4} = \frac{\pi(40)^2(0.5)}{4} = 628\text{cm}^3$

Density of the crank is 0.705kg/cm^3 (standard for hard wood) (Knott, 2006); therefore, weight of the crank,w_c is 4.34N.

Volume, V^2 of the pulley is estimated as

$V^2 = \frac{\pi D^2 t}{4}$

where D is the diameter = 25cm andt is the thickness =1.6cm; therefore, $V^2 = \frac{\pi(25)^2(1.6)}{4}$ = 785cm^3

The density of the pulley is 7.84 kg/cm^3 (standard for mild steel) (Knott, 2006); therefore, the weight of the pulley, W_P is 60N.

The shaft is of length 240mm (0.24m) and carries two cranks of equal weight at the free end and a pulley at the center. The shaft is supported by two roller bearings 0.06mm from both ends.

$F_A = F_E = \frac{W_S}{2} + W_C = \frac{252.24}{2} + 4.34 = 130.46$N

$F_C = W_P = 60$N

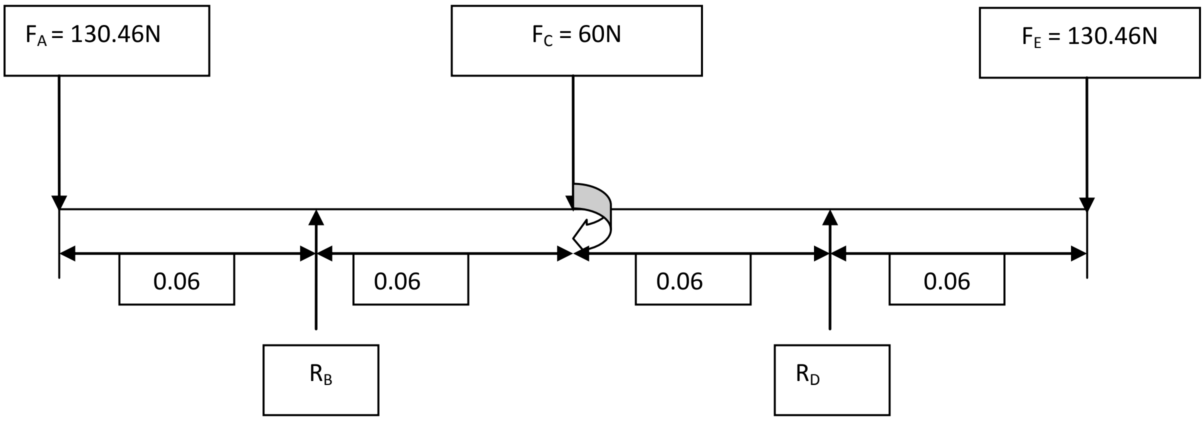

Figure 1.0: Moment of forces on the shaft

The reactions at B and D are estimated as 134N and 186.92N respectively using equilibrium principle.
The bending moment analysis is estimated thus:

At A; B.M = -130.46 × 0 = 0Nm
At B; B.M = -130.46 × 0.06 = -7.83Nm
At C; B.M = (-130.46 × 0.12) + (134 × 0.06) = -7.62Nm
At D; B.M = (-130.46 × 0.18) (134 × 0.12)- (60 × 0.06) + 3.16 = -7.84Nm
At E; B.M = (-130.46 × 0.24) + (134 × 0.18) – (60 × 0.12) + 3.16 + (186.92 × 0.06) = 0Nm

Therefore the maximum bending moment is 7.84Nm

2.2.11 Shaft Diameter: The torque transmitted to the shaft, T is estimated to be 3.16 Nm, while the maximum bending moment; M on the shaft is estimated to be 7.84Nm. From the bending moment equation, equivalent twisting moment,

$T_e = \sqrt{(M^{2+}T^2)}$ (Khurmi and Gupta, 2004) = $\sqrt{(7.84^{2+}3.16^2)} = 8.5$Nm (19)

Equivalent bending moment,

$M_e = 0.5\left(M + \sqrt{M^2 + T^2}\right)$ (Khurmi and Gupta, 2004) $= 0.5(M + T_e)$ = 8.17 Nm (20)

From the bending moment equation:

$\frac{M_E}{I} = \frac{\sigma_b}{y}$ (Rhyder, 2001) (21)

where I = moment of inertia = $\frac{\pi d^4}{64}$ (22)

σ_b = allowable bending stress = 63N/mm^2 and y = d/2. The Shaft Diameter, d is estimated to be 15mm from eqn.(22).

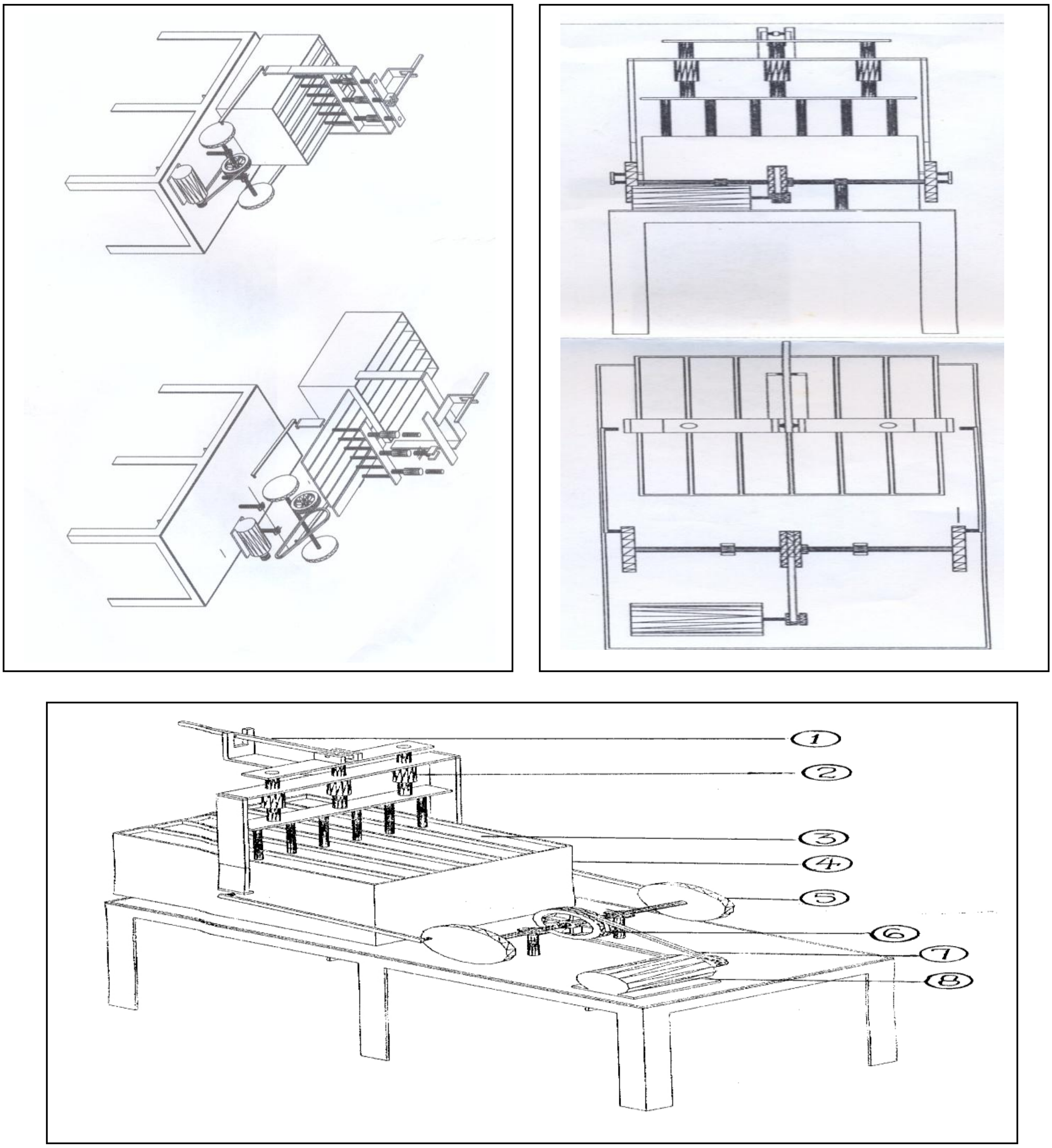

Figure 2.0: Isometric, exploded and front views of an Improved Plantain Slicer.

In the front view of the improved plantain slicer: 1. Operating Handle, 2.Stanchion, 3. Guide 4. Hopper, 5. Crank, 6. Driven Pulley, 7. V-Belt, and 8. Electric Motor

2.3 Fabrication Techniques and Machine Testing

Some engineering materials were used for the construction of the machine such as stainless, mild steels and wood. The part which accommodates the pieces of plantain is constructed using stainless steel due to its corrosion resistance as it comes in contact with the plantains. Most of the metal components are made of mild steel with the exception of the crank which is made of wood in order to reduce the overall weight of the machine. With application of basic engineering manufacturing techniques such as marking out, cutting, welding, fasteners, to mention but a few, the parts are fabricated to form a system.

3.0 Results and Discussions

3.1 Tests and Results

The machine is capable of handling 30 pieces of plantain in a single loading; domestic knife and chip cutter are also used to cut the same number of the plantains for five different times. Average time taken and efficiency were recorded. Table 1.0 below shows the results obtained.

	T (secs)	N_T	N_A	ξ	T (secs)	N_T	N_A	ξ	T (secs)	N_T	N_A	ξ	T (secs)	N_T	N_A	ξ	**Avg ξ**	**Avg.T(secs)**
Domestic Knife	2052	165	106	64	1872	198	112	57	2017	189	121	64	1982	172	115	67	**63**	**1986**
Chip Cutter	1545	479	417	87	1450	473	417	88	1473	482	423	88	1497	480	419	87	**87**	**1487**
Plantain Slicing Machine	269	452	421	93	264	450	419	93	270	455	426	94	270	451	420	93	**93**	**267**

Table 1: Experimental Results.

3.2 Functional Efficiency

The functional efficiency, ξ which is a measure of the acceptable chips to the total number of chips produced is expressed thus:

$$\xi = \text{efficiency of the operation of the slicer} = \frac{N_A}{N_T} \tag{23}$$

where, N_A = number of acceptable chips and N_T = total number of chips.
The domestic knife, chip cutter and plantain slicing machine have average efficiencies of 63%, 87% and 93% respectively.

3.3 Machine Capacity

This is a measure of the number of plantain the machine could handle per unit time and is expressed as:

$$C_m = \frac{30}{T} \tag{24}$$

In eqn. (24), T = time taken by the machine to slice 30 plantains. The machine capacity is estimated to be 0.1124/seconds (i.e. a plantain for 9 seconds) compared to the 40-80 seconds with a kitchen knife, which gives non-uniform thickness of plantain chips and mechanised slicer that takes 5-7 seconds to slice a finger of an average-size plantain but into non-uniform chips of 2-3mm in thickness.

4.0 Conclusion

This plantain slicing machine is a new design and development with improved turn rate and overall efficiency of 93% at tremendous reduction of time, cost of production and labour and operating capacity of 0.1124/seconds. The machine is simple, hygienic, portable, detachable and simple to operate. It is good for both small and large scale productions of uniform plantain chips.

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