[1.] R.O. KUKU, [2.] A.O OPENIBO, [1.] S.B. YEKINNI, [1.] O.O. OGUNSANYA,
[1.] Q.A. AJETUNMOBI, [1.] J.O. OLADOKUN, [1.] N.A. RAJI

DESIGN AND FABRICATION OF PLANTAIN TRUNK SHREDDING MACHINE

[1.] Department of Mechanical Engineering, Faculty of Engineering, Lagos State University, Ojo, Lagos, NIGERIA
[2.] Transport and Commuter Services, Lagos State Ministry of Transportation, NIGERIA

Abstract: This study explores the design, fabrication, and performance evaluation of a plantain trunk shredding machine aimed at enhancing the efficiency and quality of fiber extraction for industrial applications. The machine was designed to be driven by a 2 horsepower engine, with a flat–belt driving system operating at an angle of lap of 2.87 rad on the smaller pulley. The resultant load acts on a shaft with a diameter of 14.6 mm, experiencing a maximum bending moment of 18.338 Nm. The total weight of the shaft, pulping drum, and bearing on the frame was 66N. Performance tests conducted revealed that the machine performs optimally at a moderate operational speed of 1420 rpm. The machine demonstrated an efficiency of 96%, effectively pulping plantain ribs of approximate thicknesses ranging from 7 mm to 9 mm with applied forces of 150 N, 200 N, and 250 N, respectively. The shredding machine achieved slicing pseudo stem thicknesses between 4.0 mm and 10 mm in 12 to 20 seconds. These results indicate that the plantain trunk shredding machine is highly efficient and suitable for industrial applications, significantly reducing processing time and labor costs while maintaining high–quality fiber production.
Keywords: trunk shredding; plantain trunk; shredder; performance evaluation; design

1. INTRODUCTION

There has been a long-standing interest in the plantain tree due to its economic and nutritional importance in tropical and subtropical regions, including Asia, the South Pacific, Africa, South America, Hawaii, Central America, the Southern United States, and Southeast Asia. Understanding the optimal conditions for growing and utilizing plantain trees is crucial for improving agricultural productivity and sustainability in these areas. The plantain tree requires a warm, humid climate to thrive, growing effectively at sea level to an altitude of 1.5 km, with an average temperature of 26.7°C and monthly rainfall of 100 mm (Sonawane et al., 2011). The plantain tree's different parts, particularly its trunk, have significant industrial and medicinal applications, making it an important focus of agricultural research. The plantain trunk has different layers; the outer layer is fibrous in nature, it is also a green sheath that is not edible and mostly tough to remove, while the next layer underneath is called the core, which is the edible part of the trunk and is white to pale green yellow with a firm dense consistency.

Research has also made known that it is the world's largest herb; all its parts have medicinal application: the flower is used in bronchitis and dysentery and on ulcers; cooked flowers are given to diabetic patients, etc. Its leaves are also useful for lining cooking pots and for wrapping food materials. Improved processes have also made it possible to utilize plantain fiber for ropes, table mats, and handbags. It also serves as food for animals such as cows, pigs, etc. It is important to know that once the plantains have been harvested, its trunk will no longer grow new bananas. If the trunk is left on the farm, it will affect the growth of young plants and later die. So instead, the trunk will be cut after the plantains are harvested. It can be cut into smaller pieces in order to mix with other materials to feed animals, shredded into strands of fiber for industrial processing, and can also be used for generating biogas and composite.

A shredding machine is designed to reduce large material objects into a smaller volume or smaller pieces. Shredding machines decrease the size of materials so that they can be utilized efficiently for their intended purpose. Shredding, like crushing, is the act of conveying a force magnified by mechanical advantage through a material made up of molecules that connect more tightly. Its deform resistance is higher than that of crushed materials (Atadious and Joel, 2018). A plantain trunk shredding machine is a device designed to reduce trunks of plantain into strands of fibers which can be effectively used for their intended purposes such as weaving table mats and handbags, in the production of currencies, bond paper which lasts for a long time, among others. Shredding of materials had been a subject of discussion over the years. Engineers had in the past

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developed machines for the purpose of shredding organic and inorganic materials. Shredding of inorganic materials includes plastic shredding and paper shredding.

Several studies have shown that the traditional methods of utilizing plantain trunks, such as manual extraction and simple mechanical processes, are often inefficient and labor-intensive. Recent advancements have focused on developing machines to automate the extraction and processing of plantain fibers.

Ganesh et al. (2017) designed and developed a natural plant extracting machine. The created equipment was perfect for producing high-quality fibers. It may also be used to remove the fibers from Agave family plants. The designed machine is a low-cost, high-output machine that is well-suited to small-scale industries. It's a basic machine with only one roller rolling on a fixed support. Horizontal stainless steel blades with blunt edges are offered with the roller. In most cases, there are 27 blades. The machine's input power is supplied by a 2-hp motor. The machine reduces labor work and increases fiber production by 20–25 times as compared to the manual process. In this process, natural plant stems/stalks/leaves are crushed between two drum rollers. Due to crushing, the pulpy part is removed and fiber is obtained.

Suhaib et al. (2016) designed and developed an automated process to extract high-quality natural fibers from banana pseudo stems. Manual extraction of banana fiber produces good quality fiber but is time-consuming. This machine will reduce manual work and is suitable for mass production. Compact structure and easy disassembling will be another advantage. This design can tackle the impurity and knot problems. Roller speed has an impact on fiber quality, whereas feed angle and clearance have an impact on fiber amount. The quality and quantity of fiber may be improved by carefully selecting these elements. It is possible to reduce time and effort by using an automated feeder and conveyor.

Mahendra et al. (2016) designed and fabricated a plantain trunk shredding equipment to develop high-quality plantain fiber from plantain pseudo stems. The machine was designed with a rotor assembly consisting of two disks on which six blunt blades are mounted, and a shaft design to drive this rotor assembly and the pulley and powering system for the machine. The machine was conceptually designed in order to prove the enzymatic treatment from banana or plantain trunk for producing fibers. Enzymatic treatment increases the thermal efficiency of fibers by the elimination of pectin and hemicelluloses while producing a slight decrease in mechanical properties, probably due to defibrillation found under SEM observations.

However, while previous studies have focused on fiber extraction from banana and plantain trunks, many have not optimized the shredding process specifically for large-scale industrial applications. Existing designs often lack efficiency in handling different trunk layers or require high maintenance due to complex mechanisms. This study introduces an improved shredding machine that enhances processing efficiency by incorporating a more robust and cost-effective design, reducing labor requirements while maximizing fiber yield and quality. By addressing key limitations in previous designs, this research contributes to the advancement of sustainable and scalable plantain fiber extraction.

2. MATERIALS AND METHODS

Design concept

The plantain trunk shredding process involves cutting the plantain trunk into fibres using blades attached to a drum connected to a horizontal shaft. The blades are made of high-quality steel, ensuring durability and longevity. The shaft is made of high-resistivity metallic materials like copper or aluminium, and bearings are attached to ensure alignment. The machine's speed depends on the connection of pulleys, with a larger pulley connected to the non-powered shaft and a smaller one to the prime mover. The machine is powered by a 2Hp electric motor, which transfers motion or energy. The design concept uses a pulley-belt mechanism to drive the shaft to shred the plantain trunk into strands. Shredding machines reduce large material objects into smaller pieces, allowing them to be used efficiently for their intended purposes. The

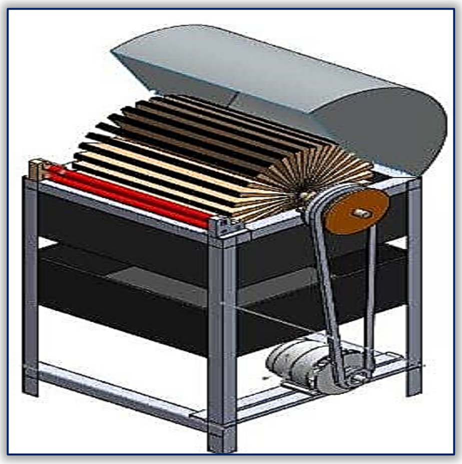

Figure 1 – Concept for Plantain Trunk Shredder

plantain trunk shredding machine is designed to produce strands of fibres suitable for various applications, such as weaving table mats, handbags, currency production, and long-lasting bond paper.

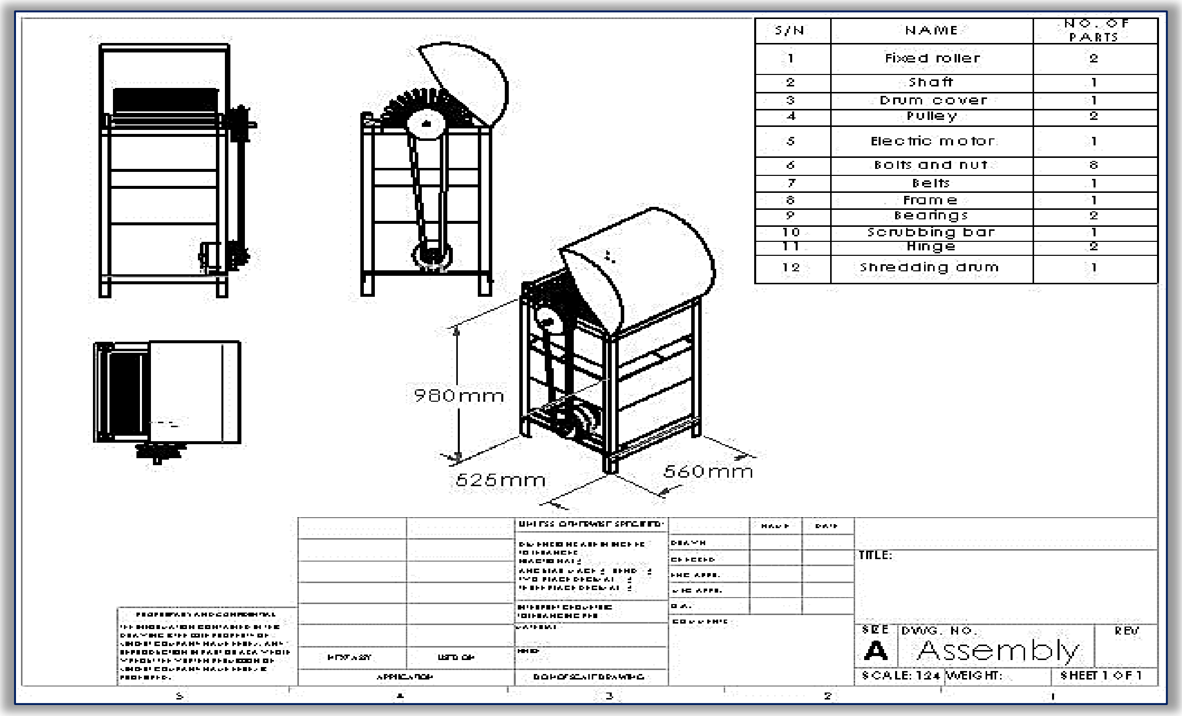

Figure 2 – The Plantain trunk shredding Machine

Design Analysis

The following major components are design in detail:

— **Determination of the force/torque needed to pulp the plantain trunk**

The following parameter of plantain trunk was adopted for easy computation:

$$L = 500\text{mm};\ \ B = 100\text{mm}\ ;\ n = 10\text{mm}\ ;\ Y = 4\text{mm}$$

According to Indira et al (2013), Modulus of elasticity Ep of plantain fibre

$$E = \ 29\ \text{GPa} = \ 29 \text{ x } 103\text{Mpa} \ = \ 29 \text{ x } 103\text{Nmm}^2$$

From Khurmi et al (2013), the modulus of elasticity E could be obtained as

$$E = \frac{Fl^3}{48yl} \tag{1}$$

where $E = $ Modulus of Elasticity ; $F = $ Pulping force or loading; $Y = $ Deflection in mm; $I = $ moment of inertia

Rearranging Equation 1 above,

$$EI = \frac{Fl^3}{48y} \tag{2}$$

where, $EI = $ Flexural rigidity of plantain rib

$$I = \frac{bh^3}{12} \tag{3}$$

Substituting the values of b and L in Equation (3)

$$I = \frac{100 \text{ x } 10^3}{12} = 8333.33\text{mm}^4$$

From Equations 1, 2 and 3 above, we have

$$EI = 29 \text{ x } 10^3 \text{ x } 8333.3 = 242 \text{ x } 10^6$$

$$L = 50\text{mm and } EI = 242 \text{ x } 10^6$$

$$F = \frac{EI \text{ x } 48 \text{ x } Y}{I3} = \frac{242 \times 106 \times 48 \times 4}{500} = 372\text{N}$$

Combining Equations 2 and 3

$$\frac{Ebh^3}{12} = \frac{Fl^3}{48y}$$

$$F = \frac{4Ebh^3y}{l^3} \tag{4}$$

Since E,B,Y and L are constants for a given sample of trunk.

Therefore, $F = 0.371h^3$

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where, $K = \frac{4Eby}{l^3} = \frac{4 \times 29 \times 10^3 \times 100 \times 4}{500^3} = 0.37$

$$F = 0.371h^3 \tag{5}$$

Hence a force of 150N, 200N and 250N would give a thickness of trunk as h = 7mm, h = 8mm, and h = 9mm respectively.

— **The required torque from rolling drum to pulp the trunk**

According to (Rayet et al, 2013) a typical shredding drum carries 17-27 beating blades.
From Figure 2 above,

$$r = \text{radius of the drum}$$
$$L = \text{distance between two blade}$$

17 blades were selected for the design and 40mm as the length between two blades.
Obtaining the angle between two blades; gives

$$n = \frac{360}{\emptyset} \tag{6}$$

$$\emptyset = \frac{360}{n} = \frac{360}{17} = 21.176°$$

Hence, the radius of the drum is obtained as follow using formula for calculating length of an arc

$$L = \frac{\emptyset}{360} \times 2\pi r \tag{7}$$

$$r = \frac{360L}{2\ \pi r} = \frac{360 \text{ x } 40}{2 \text{ x } 3.142 \text{ x } 21.176°} = 108\text{mm}$$

$$\text{Drum diameter} = d = 2r = 2 \text{ x } 108\text{mm} = 216\text{mm}$$

The following blade parameter was selected for this design

$$\text{Blade length} = 400\text{mm}$$
$$\text{Blade width} = 25\text{mm}$$
$$\text{Blade thickness} = 3\text{mm}$$

Therefore using

$$\text{Torque} = \text{force x radius} = \text{f x r} \tag{8}$$

Where: $r = 108$ mm and $F = 150$N; $T = 150\text{N x } 108\text{mm}$; $T = 16200$ Nmm $= 16.2$Nm

Since this is the part of the machine that has direct contact with the sliced plantain trunk, it is very important to select a material that has strength to pulp the trunk, therefore mild steel is then selected.

— **Velocity Ratio of belt drive**

The belt with the following specifications was selected; so that the speed of the drum shaft is 40% of the natural speed of the system.

$$d1 = 80 \text{ mm} = 0.08\text{m}$$
$$d2 = 200 \text{ mm} = 0.2\text{m}$$

While the speed of the motor selected was 1420 rpm.
Using velocity ratio of belt drive formula (Khurmi et al, 2013)

$$\frac{N_2}{N_1} = \frac{D_1}{D_2} \tag{9}$$

where: N1 = speed of the driver = 1420 rpm; D1 = diameter of the driver = 80mm; D2 = diameter of the driven = 200mm

Therefore

$$N_2 = \frac{N_1 D_1}{D_2} = \frac{1420 \times 0.08}{0.2} = 568\text{rpm}$$

— **Power needed to drive the drum**

The power needed to drive the drum is expressed as (Khurmi et al, 2013)

$$P = \frac{2\pi TN}{60} \tag{10}$$

where P = Power transmitted to shaft in Watts, T = torque in Nm , N = Speed of the shaft in rpm

Substituting the values of T and N in Equation 10

Therefore: $P = \frac{2 \text{ x } 3.142 \text{ x } 16.2 \text{ x } 568\text{rpm}}{60}$

P = 0.964KW = 964W

The power needed in horsepower

$$\text{I hp} = 745.7\text{W}$$
$$1\text{W} = 0.001341\text{hp}$$

Therefore: 964W = 1.292hp = 1.3hp

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Hence, this machine is designed to use two-horsepower (2hp) electric motor and the material selected is cast iron with windings (standard)

— **Calculating the tensions of the belt**

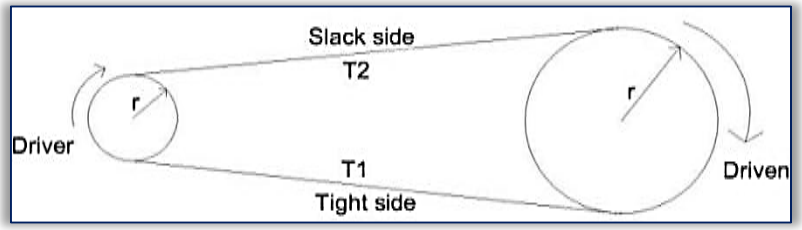

Figure 3 – Belt Drive Mechanism

In Figure 3 above,

T1 = Tensions in Newton on the tight side ; T2 = Tension in Newton on the slack side

R1 = Radius of driver; R2 = Radius of driven.

— **Calculating length of the belt**

From Khurmi et al (2013), the formula for calculating length of belt drive is obtained as

$$L = \frac{\pi\,(r_2 + r_1) + 2x + (r_2 - r_1)2}{x} \tag{11}$$

where

r1 = radius of smaller pulley = 0.04m

r2 = radius of bigger pulley = 0.1m

x = distance between the centre of the pulley = 0.45m

Substituting the values of r_1, r_2, x in Equation 11

$$L = \frac{\pi\,(0.1 + 0.04) + 2\,(0.45) + (0.1 - 0.04)2}{0.45} = 1.47\text{m}$$

According to Khurmi et al (2013), the angle of contact is expressed in radian as:

$$\emptyset = (180 - 2\alpha)\frac{\pi}{180} \tag{12}$$

where

$$\text{Sin}\alpha = \frac{r1 + r2}{x} \tag{13}$$

$$\text{Sin}\alpha = \frac{0.1 + 0.04}{0.45} = 0.3111$$

$$\alpha = \sin - 1\,(0.3111) = 18.130$$

Substitute 18.13^0 forα in Equation 12

$$\emptyset = (180 - 2 \text{ x } 18.13)\frac{\pi}{180} = 2.87\text{rad}$$

— **Calculating mass of the belt per meter length of a rubber belt**

From Khurmi et al (2013), the density of rubber belt was 1140 kg/m^2. Also, the breadth and thickness of the belt selected were 13mm and 8mm respectively.

We know that

$$\text{Density} = \frac{\text{Mass}}{\text{Volume}} \tag{14}$$

$$\text{Mass} = \text{Area x length x density} \tag{15}$$

$$\text{Mass} = 0.013 \text{ x } 0.008 \text{ x } 1140\frac{\text{kg}}{\text{m}} = 0.11856\frac{\text{kg}}{\text{m}}$$

— **Calculating Velocity ratio of belt**

From Khurmi et al (2013), the velocity ratio of a belt drive is expressed as

$$V1 = \frac{d_1 N_1}{60}\text{ms}^{-1} \tag{16}$$

Substituting the values of d1 and N1 in Equation 16

$$V1 = \frac{\pi \text{ x } 0.08 \text{ x } 1420\text{rpm}}{60} = 5.949\text{ms}^{-1}$$

The following expression are obtained from (Khurmi et al, 2013)

The ratio of driving tension for flat belt drive

$$2.3\log\frac{T1}{T2} = \mu\,\emptyset \tag{17}$$

where μ = Coefficient of friction between the belt and pulley

Coefficient of friction between rubber and dry steel is usually taken to be 0.3 (Khurmi et at, 2013).

$$\emptyset = \text{Angle of contact in radians}$$

Centrifugal Tension

$$Tc = M.V^2 \quad (18)$$

where M = mass of belt per unit length in kg ; V = linear velocity of belt in m/s

Maximum tension

$$T = \sigma.b.t \quad (19)$$

where σ = maximum safe speed; b = width of belt; t = thickness of belt

Tension on tight side

$$T1 = T - Tc \quad (20)$$

where T = Maximum tension; Tc = centrifugal tension

Substituting the value of M and V in Equation 18

$$Tc = M.V^2 = 0.11856 \text{ x } 5949^2 = 4.1959N$$

Also, maximum tension $\quad T = 2.5MPa \text{ x } 13mm \text{ x } 8mm = 260N$

Substituting value of T and Tc in Equation (20)

$$T1 = (260 - 4.1959)N = 255.8041$$

Substituting the value of N and Ø in equation (17)

$$2.3\log\frac{T1}{T2} = \mu\,\emptyset$$

$$\log\frac{T1}{T2} = \frac{\mu\,\emptyset}{2.3}$$

$$\log\frac{T1}{T2} = \frac{0.3 \text{ x } 2.87}{2.3}$$

$$\log\frac{T1}{T2} = 0.3743$$

$$\frac{T1}{T2} = e^{0.3743} = 1.4580$$

$$T2 = \frac{T_1}{1.4540}$$

$$T2 = 175.93$$

Hence the standard belt specification selected was A61 12.5 x 1600

— **Shaft Design**

The fatigue factor and combined shock was considered due to bending moment and fluctuating torque the shaft was subjected to. According to Khurmi et al (2013), $K_n = 1.5$ and $K_t = 1.0$ for gradual and steady feeding of plantain trunks.

In the design of shaft, the load to be carried by the shaft and the reaction are obtained as follow:

Weight of driven pulley $W_p = 2kg = 19.62N$

The weight of the drum is calculated below:

For each blade $\quad L = 400mm \quad b = 25mm \quad \text{and } h = 3mm$

Therefore, volume of 1 blade $\quad = L \text{ x } b \text{x } h = 0.4 \text{ x } 0.25 \text{ x } 0.003 = 3 \text{ x } 10^{-5}$

$$\text{Volume of 17 blades} = 17 \text{ x } 3 \text{ x } 10^{-5} = 5.1 \text{ x } 10^{-4} m^3$$

We know that,

$$\text{Mass} = \text{volume x Density}$$

Where density of steel = $7680\frac{kg}{m^3}$

$$\text{Mass} = (5.1 \text{ x } 10 - 4 \text{ x } 7680) = 3.9168kg = 38.42N$$

For the plate of radius $= 108mm$ and $h = 5mm$

$$\text{Volume of plate} = \pi r^2 h \quad (21)$$

$$= 3.142 \text{ x } 0.108^2 \text{ x } 0.005 = 1.83 \text{ x } 10^{-2} mm^3$$

Volume of 2 plates $= 2 \text{ x } 1.83 \text{ x } 10^{-4} m^3 = 3.66 \text{ x} 10^{-4} m^3$

Mass of plate $= 3.66 \text{ x} 10^{-4} \text{ x } 7680 \text{ kgm}^{-3} = 2.81088kg$

The total weight of the drum $= (38.42 + 27.58)$ N $= 66$N

— **Calculating shaft loading:**

Vertical loading acting on shaft due to the weight of the pulley F_{pv} as expressed by (Khrumi et al, 2013) is given as:

$$F_{pv} = W_p + (T1 + T2) \text{ x } \sin 30 \quad (22)$$

Substituting the values of W_p, T_1, T_2, in Equation 22

$$F_{pv} = 19.62 + (255.804 + 175.93) \sin 30 = 234.867$$

Horizontal load acting on shaft at the position of the pulley F_{ph} as expressed by (Khurmi et al, 2013) is given as:

$$F_{ph} = (T1 + T2) \text{ x } \sin 30 \quad (23)$$
$$= (255.804 + 175.93) \sin 30 = 215.872$$

Vertical load acting on shaft due to the weight of roller drum as expressed by (Khurmi et al, 2013) is given as:

$$F_p = WR + FR \quad (24)$$

where $WR = \text{weight of roller drum}$

$FRT = \text{Tangential force on the roller drum due to rotation}$

Since the radius of roller drum = 0.1m

$$T = (12.26 \text{ x } 0.1)\text{ N} = 122.6\text{N}$$
$$FRT = 60\text{N} + 122.6\text{N} = 182.6$$

— **Calculation of Shaft diameter:**

The reactions are obtained below:

Taking moment about point $A, \sum MA = 0$

$$183.6 \text{ x } 0.235 + 0.470\, R_B - 234.867\,(0.5) = 0$$
$$0.470\, R_B = 43.146 + 117.43$$
$$0.47\, R_B = 160.576$$
$$R_B = \frac{160.576}{0.47} = 341.65$$

Obtaining $R_A, \sum Fy = 0$

$$R_A + R_B = 183.6 + 234.867$$
$$R_A = 418.67 - 341.65 = 76.813$$

Bending moment (M) at each point

$$M_A = M_D = 0$$
$$M_{CV} = 76.813\,(0.235) = 18.05\text{Nm}$$
$$M_{BV} = 76.813\,(0.47) + 183.6\,(0.235) = 7.044\text{Nm}$$

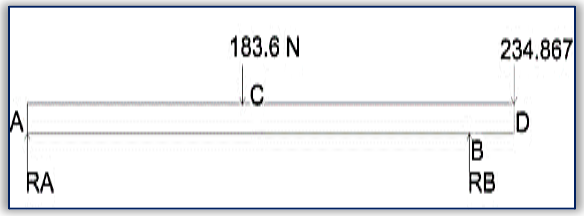

Figure 4 – Vertical loading on the shaft

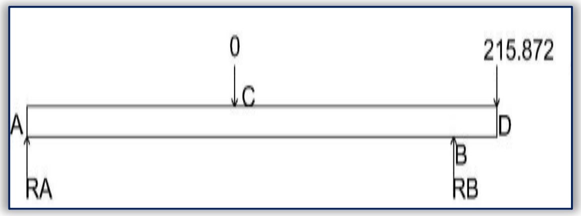

Figure 5 – Horizontal loading diagram

Taking moment about point $A, \sum MA = 0$

$$0.47\, R_B - 215.872\,(0.5) = 0$$
$$0.47\, R_B = 107.936$$
$$R_B = 229.615\text{N}$$

Obtaining $R_A, \sum Fy = 0$

$$R_A + R_B = 215.872$$
$$R_A = 215.872 - 229.651 = -13.779\text{N}$$

Bending moment (M) at each point

We know that, $M_A = M_D = 0$

Bending moment at C

$$M_{CH} = 13.779 \text{ x } 0.235 = 3.238\text{Nm}$$

Bending moment at B

$$M_{BH} = -13.779 \text{ x } 0.47) = 6.476$$

Resultant loading moment at C

$$M_C = \sqrt{M_{CV^2} + M_{CH^2}} = \sqrt{(18.05)2 + (-3.238)2} = 18.338$$

Result of bending moment at C

$$M_B = \sqrt{M_{BV^2} + M_{BH^2}} = \sqrt{(7.044)^2 + (6.476)^2} = 9.566\text{Nm}$$

With above analysis, the maximum bending moment is 18.338NM

— **Diameter of the shaft:**

According to Khurmi et al (2013), the equivalent twisting moment Te is given as:

$$Te = \sqrt{(K_m \times M_B)^2 + (K_t \times T)^2} = \sqrt{d(1.5 \text{ x } 18.338)^2 + (1 + 12.26)^2} = 23.861\text{Nm} = 23862\text{Nmm} \quad (25)$$

Recall from Equation (25),

$$Te = \frac{\pi \times t \times d^3}{16}$$

Using $T = 42\frac{N}{mm^2} = 42Mpa$

$$23862 = \frac{\pi \times 42 \times d^3}{16};\ d^3 = \frac{23862 \times 16}{3.142 \times 42} = 2893.153;\ d = 14.25mm$$

According to Khurmi et al (2013) equivalent bending moment is expressed as:

$$Me = ½\,(Km \times MB \times Te) = \frac{\pi \times \sigma_B \times d^3}{32} = 25.685Nm = 25685Nmm \quad (26)$$

Recall from Equation 26

$$Me = \frac{\pi \times \sigma_B \times d^3}{32}$$

Using $\sigma_B = 84Mpa$

Therefore: $d^3 = \frac{32 \times M_e}{\pi \times \sigma_B};\ d^3 = \frac{32 \times 25685}{3.142 \times 84} = 14.60mm$

The shaft was selected based on their ability to withstand the stresses/loads, in order to fit in with this quality, mild steel of 16mm diameter was successfully machined

— **Bearing:**

The dynamic loading (C) and resultant radial force (W) are calculated below:

$$R_A = 76.813$$
$$R_B = 341.65$$
$$W = \sqrt{(76.813)^2 + (341.65)^2}$$
$$W = 350.18N$$

Bearing Life $= 30{,}000hr$

Allowable shaft speed $n = 568rpm$

Constant for ball bearing is $k = 3$

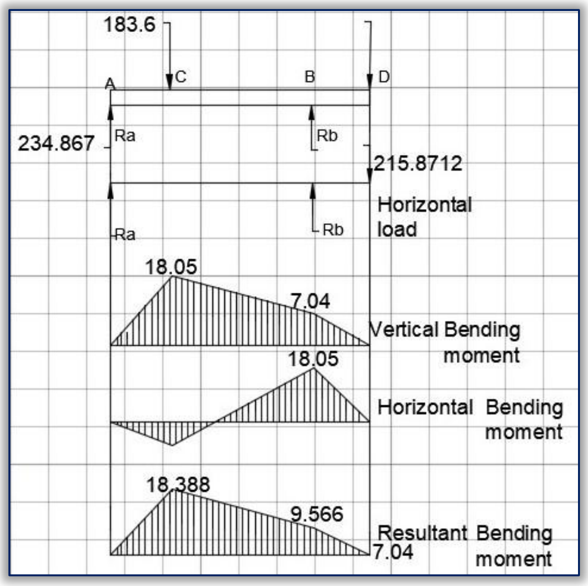

Figure 6 – Bending Moment Diagram

Therefore, the dynamic loading is given as

$$C = W\left(\frac{L_n 60n}{10^6}\right)^{\frac{1}{k}} \quad (27)$$

$$= 350.18 \times \left(\frac{30{,}000 \times 60 \times 568}{10^6}\right)^{\frac{1}{3}} = 350.18 \times 10.074 = 3527.71N = 3.528KN$$

The bearing number 200 having the dynamic loading of 4KN was selected according to (Khurmi et al, 2013), the material selected was cast iron (standard)

— **Key Design:**

A round key was selected for this design, using the shaft diameter as a guide, the size of the key which is the thickness (t) and the width (w) were determined using the empirical design mode relation for different types of keys.

To calculate the width of the key w

$$W = nd \quad (28)$$

where n = ratio of the key width to the shaft diameter (0.25 was recommended)

$$d = \text{shaft diameter} = 14.6mm$$
$$w = 0.25 \times 14.6 = 3.65mm$$

To calculate the thickness of the key

$$t = mw \quad (29)$$

where m = ratio of thickness of key to the width (1.30 was recommended)

$$t = 1.3 \times 3.65 = 4.75mm$$

Therefore, the key of 4mm and thickness of 5mm was selected.

3. RESULT AND DISCUSSION

The machine generates fibre from plantain trunks which is generally left uncared for, thereby enabling the acquisition of wealth from organic waste. In the fabrication of this machine, the design calculations were carefully employed, this aid in computation of the thickness, width and length of the plantain sliced trunk. Also in order to confirm the functionality and integrity of the components that make up the machine, different sizes of the trunks were tested on it.

The sliced trunk fed into the machine is converted to fibre which serves as raw material for many companies (textile, furniture, etc). The sliced trunk of thickness 4.0 mm, 5.0 mm, 6.0 mm, 7.0 mm, 8.0 mm, 9.0 mm and 10 mm are used for the test. The extraction time was measured using a stop clock and a 220 V single phase induction motor. The plantain trunk shredding machine could extract a cut trunk with a width and thickness of 100 mm and 10 mm, respectively, based on the test results, while the length fluctuates. The project was fabricated and tested in the workshop of the Mechanical Engineering Department, Lagos State University, Epe Campus.

Figure 6: A Pictorial View of the Designed Plantain Trunk Shredding Machine

Table 1 –Performance Evaluation Result

S/N	Plantain Sliced Trunk Length (mm)	Width (mm)	Thickness (mm)	Operating Voltage (V)	Fiber Extracting Time (sec)
1	360	100	8.0	220	13
2	500	86	7.0	220	18
3	380	72	5.0	220	13
4	490	70	4.0	220	17
5	400	70	9.0	220	15
6	350	90	6.0	220	13
7	370	75	7.0	220	12
8	390	89	10.0	220	14
9	500	96	8.0	220	20
10	480	77	6.0	220	17

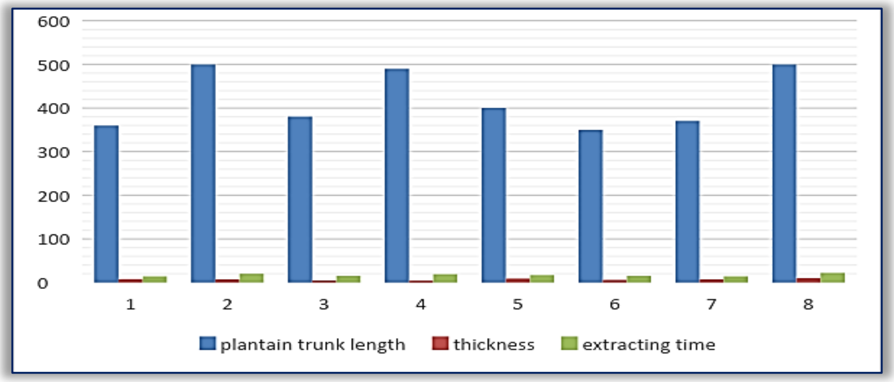

Figure 7: Performance Evaluation at 1200 rpm

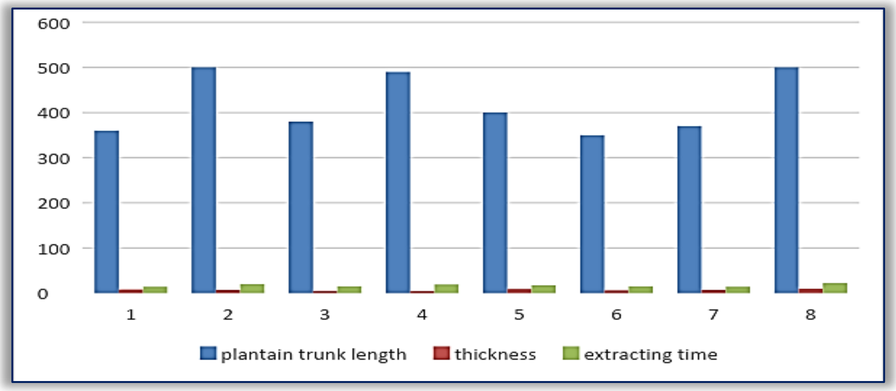

Figure 8: Performance Evaluation at 1420 rpm

The result shows that the machine performance improved at moderate speed of operation at 1420 rpm. The human-mechanical efficiency was obtained to be 96% (although this efficiency will drop due to decrease in power) at the average length of 450mm with actual thickness range between 4mm to 10mm based on the desired speed. This result is based on the spot assessment of shredding done by Yekinni Sodiq Bolanle and Ogunsanya Oladimeji Omobolanle from Lagos State University, Faculty of Engineering, Epe Campus, in the department of Mechanical Engineering Workshop. The result showed that the plantain trunk shredding machine was effective.

4. CONCLUSION

The successful design, fabrication, and performance analysis of the plantain trunk shredding machine demonstrate its effectiveness in extracting plantain fibers with high efficiency. The machine operates with a 2-horsepower engine and achieves a 96% efficiency rate, making it a reliable tool for processing plantain pseudo stems. The performance evaluation confirmed its capability to extract fibers of varying thickness within a short time frame, highlighting its practical usability.

Despite these advantages, the machine has some limitations. The dependence on a belt-driven mechanism may introduce wear and tear over time, requiring periodic maintenance. Additionally, the machine's performance may vary based on the moisture content and fiber density of the plantain trunk, which could impact shredding efficiency.

The potential applications of this machine are broad, especially in the agricultural sector, where plantain fiber can be utilized for textiles, paper production, and biodegradable packaging. By providing an efficient means of processing plantain trunks, the machine contributes to waste reduction and value addition in agricultural waste management.

Future work should focus on optimizing the design for improved durability and automation. Incorporating sensors for real-time monitoring of processing parameters and exploring alternative energy sources, such as solar power, could enhance the machine's sustainability. Additionally, further studies on different plantain varieties and their fiber characteristics could help refine the shredding mechanism for enhanced performance.

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