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DESIGN AND DEVELOPMENT OF PLANTAIN FIBRE EXTRACTION MACHINE

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ABSTRACT
The traditional retting technique of extracting plantain fibres is faced with various constraints such as longer extraction time and poor fibre production rate. This study is targeted at overcoming the limitations associated with traditional methods of extracting plantain fibres through the development of electrically powered machine capable of extracting plantain fibres. The method employed involves the selection of appropriate materials, design, fabrication and assembly of the various components of the machine parts. From the analysis a 2 horsepower electric motor is required to drive the machine. The length of flat-belt required to drive the pulley was 1.47 m at an angle of lap on the smaller pulley of 2.87 rad. A resultant load of 174.87 N act on 11.6 mm diameter shaft with maximum bending moment of 10.33 Nm. The total weight of shaft, pulping drum and bearing on frame was 67.51 N. Also, a force of 150 N, 200 N and 250 N could pulp a thickness of plantain ribs of approximately, 6 mm, 6.5 mm and 7 mm, respectively. The test result showed that the machine could extract a sliced pseudo stem thickness between 4.0 mm and 10 mm at 27 and 42 seconds, respectively. It is expected that the plantain fibre extraction machine fabricated would save plantain fibre production time.

Keywords: *Plantain fibres, Pseudo stem, extraction machine, retting technique*

1. INTRODUCTION

In recent times, attention have been given to the use of plantain fibres as reinforcement in thermoplastic composites because of its low density, lightweight; recyclability, biodegradable and appreciable mechanical properties and structural application [1-5]. To better understand the properties of natural fibre reinforced composite materials, it is imperative to know the physical, mechanical and chemical properties. The strength characteristics of fibre depend on the properties, fibrillary structure, lamellae matrix, method of processing and chemical modification [6-8]. Investigation on the eco-method of extraction of natural fibre, lifecycle and final disposal had been studied [9-10]. The traditional process such as the retting method to extract plantain fibres involved the anaerobic and biological organism (such as bacteria and fungi) presence in the medium where the pseudo stems were stored in order to decompose the lignin, pectin and other substances. However, the traditional methods of plantain fibre extraction (retting) require about 2–6 weeks decomposing the non-fibrous constituent of the pseudo stem before further modification could be done to obtain the fibres [11-13]. This method of fibre extraction is time consuming. Recently, attention has been given to the mechanical techniques; processing and automated mechanisms involved in plantain fibre extraction in order to eliminate the manual methods and to increase the quality and improve the processing time [14-15]. Hence, the purpose of this study is to develop and fabricate an electrically powered plantain fibre extraction machine. This extraction machine would help to reduce plantain fibre extraction rate and human labour associated with retting method.

2. MATERIALS AND METHODS

The study was carried out at the Federal University of Petroleum Resources Effurun, Delta State, Nigeria. All raw materials for this work were locally sourced within the University community. The following steps were considered; Design considerations, component design/calculations, component assembly and description of machine.

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2.1 Design Considerations

In the selection of the various materials employed in the fabrication of the plantain fibre extraction machine the following considerations among others were given attention;

i. Strength: Each component was selected based on its ability to withstand the loads/stresses, torques and frictional forces, it would experience during pulp operation. Steel was one of main materials selected.

ii. Corrosion resistance: Since the pseudo stems that would be extracted were moist and wet, corrosion resistance materials were considered. Other important considerations are, wear resistance, use of standard part for ease of replacement, cost, safety of operator.

2.2 Component Design/Calculations

2.2.1 Length of Open Belt Drive

From [16], the expression for length of belt drive is obtained as;

$$L = \pi(r_2 + r_1) + 2x + \frac{(r_2 - r_1)^2}{x} \qquad (1)$$

Where, $r_1$and r_2 = Radius of the larger and smaller pulleys, respectively

x =Distance between the centres of two pulleys required

2.2.2 Velocity Ratio of Belt Drive

The velocity ratio of belt drive could be expressed as;

$$\frac{N_2}{N_1} = \frac{d_1}{d_2} \qquad (2)$$

Where: d_1 is the Diameter of the driver; d_2 *is the* Diameter of the follower,$N_1$is the Speed of the driver in rpm and $N_2$is the Speed of the follower in revolution per minute (rpm)

2.2.3 Expression of the Force/Torque Needed to Pulp the Plantain Rib

The modulus of elasticity E could be expressed as [1; 16]

$$E = \frac{\mathrm{f}l^3}{48\mathrm{y}I} \qquad (3)$$

Where: E is the modulus of elasticity; y is the Deflection in mm; f is the pulping force/load in N; I is the moment of inertia and l is the length of material/ plantain rib

Rearranging, we have

$$EI = \frac{\mathrm{f}l^3}{48\mathrm{y}} \qquad (4)$$

where, EI is the flexural rigidity of the plantain rib

Also,

$$I = \frac{\mathrm{B}H^3}{12} \qquad (5)$$

Where: I is the moment of inertia; B is the breadth of plantain rib in mm; H is the height/thickness of plantain rib in mm.

2.2.4 Torque needed to Pulp Plantain Rib

The torque required to pulp the rib is expressed as [16]

$$\mathrm{T} = \mathrm{f} \times \mathrm{r} \qquad (6)$$

where, f is the force needed to pulp the plantain rib; r is the radius of the rolling drum

2.2.5 Length of an Arc

The length of arc could be expressed as;

$$L = \frac{\theta}{360} \times 2\pi r \qquad (7)$$

But:

$$n = \frac{360}{\theta} \qquad (8)$$

where, n is the number of blades

2.2.6 Power Required to Drive Drum

The power required to drive the drum is expressed as [16];

$$\mathrm{P} = \frac{2\pi\ \mathrm{TN}}{60} \qquad (9)$$

where, P is the power transmitted to shaft; T is the torque in Nm and N is the speed of the shaft in revolution per minute (rpm).

2.2.7 Power Transmitted by Belt

The required power transmitted by belt could be expressed as [16],

$$\mathrm{P} = (\mathrm{T}_1 - \mathrm{T}_2)\mathrm{v} \text{ in watts} \qquad (10)$$

Where: $T_1$is the Tension in the tight side of belt in newton, N, T_2 is the Tension in the slack side of belt in newton, N and v is the Velocity of the belt in metre per second, m/s. The ratio of driving tensions for flat belt drive is expressed as

$$23log\left(\frac{T_1}{T_2}\right) = \mu.\theta \qquad (11)$$

Where: θ is the Angle of contact in radians and $\boldsymbol{\mu}$ is the Coefficient of friction between the belt and pulley.

2.2.8 Angle of Contact or Lap

The angle of contact is expressed in radian as [16],

$$\theta = (180 - 2a)\frac{\pi}{180} \qquad (12)$$

And the centrifugal tension T_C T_c is,

$$T_c = m.v^2 \qquad (13)$$

where, m = Mass of belt per unit length in kg; v = Linear velocity of belt in $\frac{\mathrm{m}}{\mathrm{s}}$

2.2.9 Maximum Tension in the Belt

The maximum tension T in the belt could be computed using the expression [16],

$$T = \sigma . b . t \tag{14}$$

where,
σ is the Maximum safe stress; b is the Width of the belt, and t is the Thickness of the belt Tension on the tight side, T_1

$$T_1 = T - T_c \tag{15}$$

where, T=Maximum tension; T_c = Centrifugal tension.

2.2.10 Design for the Shaft

The belt is inclined at 30^0 to the horizontal. The forces acting on the pulley comprises of vertical and horizontal forces [16],

(a) Forces exerted by Pulley
(i) vertical load acting on shaft at position of pulley, F_{PV}.

$$F_{PV} = W_P + (T_1 + T_2) \times \sin 30^0 \tag{16}$$

where, W_P =Weight of pulley
(ii) Horizontal load acting on shaft at position of pulley, F_{PH}.

$$F_{PH} = (T_1 + T_2) \times \sin 30^0 \tag{17}$$

(b) Vertical load on shaft exerted by Roller drum assembly, F_R

$$F_R = W_R + F_{RT} \tag{18}$$

Where: W_R is the Weight of Roller drum, and F_{RT} is the Tangential force exerted by roller drum as a result of rotation

$$F_{RT} = \frac{T}{R} \tag{19}$$

2.2.11 Diameter of the Shaft Determination

Using equivalent twisting moment T_e [16], we have that

$$T_e = \sqrt{(K_m \times M_B)^2 + (K_t \times T)^2} = \frac{\pi}{16} \times \tau \times d^3 \tag{20}$$

Using equivalent bending momentM_e, we have,

$$M_e = \frac{1}{2}\left[K_m \times M + \sqrt{(K_m \times M_B)^2 + (K_t \times T)^2}\right]$$
$$= \frac{1}{2}[K_m \times M_B + T_e] = \frac{\pi}{32} \times \sigma_b \times d^3 \tag{21}$$

Where: k_m is the Combined shock and fatigue factor for bending, M is the Bending moment, T is the Twisting moment or torque acting upon the shaft; T_e is the equivalent twisting moment,τ is the Shear stress induced due to twisting moment, σ_b is the Bending stress induced due to bending moment and d is the Diameter of the shaft.

2.2.12 Bearing Selection

For the bearing selection, the SKF bearing catalogue was adopted. Resultant force on the Bearing 1,

$$B_1 = [R_{1V}^2 + R_{1H}^2]^{0.5} \tag{22}$$

Resultant force on the Bearing 2,

$$B_2 = [R_{2V}^2 + R_{2H}^2]^{0.5} \tag{23}$$

Where: R_{1V} is the reaction 1 on vertical loading diagram of the shaft, R_{2V} is the reaction 2 on vertical loading diagram of the shaft, R_{1H} is the reaction 1 on horizontal loading diagram of the shaft and R_{2H} is the reaction 2 on horizontal loading diagram of the shaft

$$\text{Bearing Life} = \frac{60 \times \text{N} \times \text{operating time}}{10^6} \tag{24}$$

Where: N is the Number of revolution
Load carrying capacity on Bearing 1,

$$B_1 = [Bearing\ life]^{\frac{1}{k}} \times Load\ factor \tag{25}$$

Load carrying capacity on Bearing 2,

$$B_2 = [Bearing\ life]^{\frac{1}{k}} \times Load\ factor \tag{26}$$

Where: k is the exponent of the life equation, 3 (for ball bearing)

2.3 Design Calculations

The following specifications for the plantain fibre machine are considered; Plantain width =150 mm maximum; plantain maximum thickness (sliced trunk thickness) =10mm; driving means = electric motor; motor operating voltage = 220V; type of motor = single phase induction motor; feeding method = manual; coupling method = belt drive; belt type = flat rubber reinforced belt.

2.3.1 Determination of the Force/Torque Needed to Pulp the Plantain Rib

The following plantain rib parameter for ease of computation was adopted
For plantain rib at: L = 400mm; B = 100mm and Thickness H= 8 mm, y = 4 mm
Using the Modulus of Elasticity E_p of plantain fiber expressed by Indira et al. [17] as,

$$E_p = 29 \text{ GPa} = 29 \text{ x } 10^3 \text{ N/mm}^2 \tag{27}$$

From equation (5),

$$I = \frac{BH^3}{12} = \frac{100 \times 8^3}{12} = 4267mm^4 \tag{28}$$

And from equations (4), (27) and (28), we have
Flexural Rigidity, EI =

$$EI = 29 \text{ x } 10^3 \times 4267 = 124 \text{ x } 10^6 \text{ N}/mm^2 \tag{29}$$

Deflection, y = 4 mm. We have $f = \frac{EI \times 48 \times y}{l^3} = 384\text{N}$

2.3.2 Analysis of Thickness/Pulping Force

Combination of equations 4 and 5 give

$$\frac{EBH^3}{12} = \frac{fl^3}{48y} \tag{30}$$

$$12fl^3 = 48EBH^3y \tag{31}$$

$$f = \frac{48EByH^3}{12l^3} = \frac{4EByH^3}{l^3} \tag{32}$$

where, E, B, Y and L are constant for a given rib sample hence,

$$f = KH^3 \quad (33)$$

$$K = \frac{4BEy}{l^3} \quad (34)$$

$$f = 0.725H^3 \quad (35)$$

Hence, a force of 150 N, 200 N, 250 N would give a thickness H of rib as $H \simeq 6$ mm, $H \simeq 6.5$ mm, $H \simeq 7$ mm, respectively.

2.3.3 Torque Required from Rolling Drum to Pulp Rib

Figure1 is a schematic view of a rolling drum structure showing the radius of the drum and it beating blade.
The rolling drum carries the beating blades. According to Ray *et al.* [18], a typical rolling drum carries 17 – 27 blades.
Let, r is the radius of drum
where,
L is the distance between two blades = length of arc 1 and 2
Number of blades for this design = 17
Distance between two blade, L = 40 mm
Drum radius = 108mm.
Drum diameter = 2 × radius = 216 mm
Drum thickness = 4 mm
Blade length = distance between the rolling plates.
The value of maximum plantain rib width used was = 300 mm
Blade thickness = 3 mm flat bar
The 3mm thickness flat bar has a breadth of 25 mm.
Hence,
Torque = force×radius

$$T = Fr \quad (36)$$

where, F = 150N; r = 108 mm = 0.108 m, hence Torque = 16.2 Nm

2.3.4 Velocity Ratio of Belt Drive

The following specifications were obtained from the selected belt

$d_1 = 0.08\,\mathrm{m}$; $d_2 = 0.2\,\mathrm{m}$

From computation we have $N_1 = 1420$rpm
Hence, equation (2) gives, $N_2 = 568$ rpm

2.3.5 Power Required to Drive Drum

The drum is driven by an electric motor, with motor power transmitted by belt. Hence,

$$P = \frac{2\pi TN}{60} \quad (37)$$

where, P is the power required in watts (W), T is the torque in Nm, N is the speed of shaft in rpm, T =16.2Nm and N = 568 rpm. Substituting the values into (37) yields, P = 0.964 kW

Power required in Horse Power

1hp = 745.7 W, therefore, 0.964 kW = 1.3 hp

Hence, required motor rating selected to drive the machine is 2 hp.

2.3.6 Required Tension on Belt

Figure 2 is a schematic diagram for the belt drive mechanism indicating the tight and slack side.
Let T_1 and T_2 = Tensions in Newton on the tight and slack side of the belt respectively
r_1 and r_1 = Radius of the driver and follower, respectively.

(i) *Design and Analysis of Belt Drive*

Total length of flat-belt, L

$r_1 = 40\,\mathrm{mm} = 0.04\,\mathrm{m}$. $r_2 = 100\,\mathrm{mm} = 0.1\,\mathrm{m}$

$x = 450\,\mathrm{mm} = 0.45\,\mathrm{m}$

From equation (1), L will become 1.47m

(ii) Angle of Lap on the Smaller Pulley (θ)

$d_1 = 0.08\ m$; $d_2 = 0.2\ m$
From equation (12), θ = 2.87 rad

(iii) Mass of the Belt per Metre Length (Rubber belt)

The density of rubber belt selected was 1140 kg/m^3 [16].

$$\text{Hence, m = Area x length x density} \quad (38)$$

where, b = 12.5mm; t = 5mm; area, a = 62.5mm^2 ; $\rho = 1140\ \mathrm{kg/m^3}$. Hence m = 0.07125kg/m

(iv) Velocity of the Belt

$$v = \frac{\pi d_1 N}{60} \quad (39)$$

where, $d_1 = 0.08\ m$; $N_1 = 1420$ rpm. Hence from (39) v = 5.949 m/s

(v) Tension in tight side, T_1 and slack side, T_2

From equations 11 to 15, applying the coefficient of friction between belt and pulley of 0.3 [16], we have: $T_1 = 226.96$ N and $T_2 = 95.74$ N

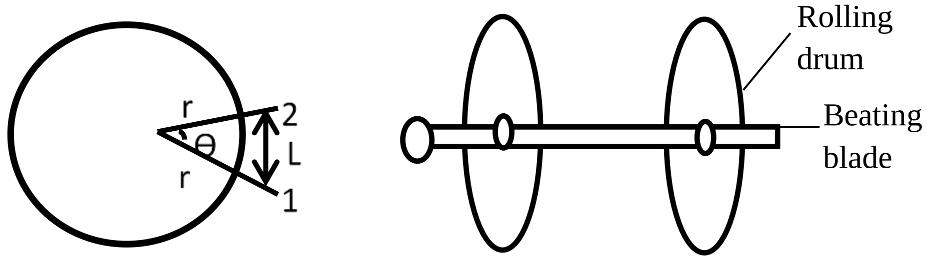

Figure 1: Rolling Drum Structure and Radius of Drum and Beating Blade

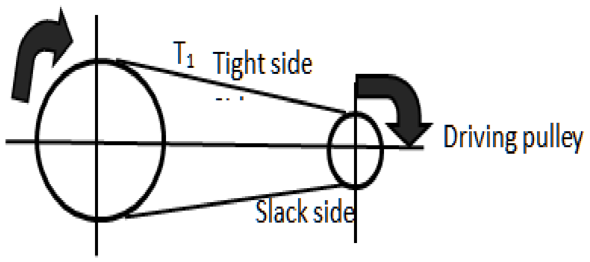

Figure 2: Belt Drive Mechanism

2.3.7 Rolling or Pulping Drum Shaft Design

The main loads on the shaft were the bearing reactions, the weights of the pulley, belt tension, weight of gear, weight of roller assembly and tangential forces on the gears and roller assemblies as a result of torque, cutting drum and belt tensions. The shaft will be subjected to fluctuating torque and bending moments, and therefore combined shock and fatigue factors were taken into account.

Since the feeding of the plantain sliced trunk is gradual and steady, thus, $k_m = 1.5$ and $K_t = 1.0$ were applied as in [16].

(i) Loading on the Shaft

Substituting T_1 and T_2 into equations (16) and (17) where, $W_P = 2kg = 19.62$ N

(a) vertical load acting on shaft at position of pulley from equation (16) becomes $F_{PH} = 180.98$ N

(b) Horizontal load acting on shaft at position of pulley, from equation (17) it becomes, $F_{PH} = 279.48$ N

(c) Vertical load on shaft exerted by pulping drum assembly,

F_R was obtained from equation (19) where, $W_R = 49.519$ N; $T = 12.26$ Nm; Radius of roller drum $= 0.1$ m. Therefore FRT $= 122.6$N. $F_R = 49.519 + 122.6 = 174.871$N

(ii) Shaft Diameter Calculation

Figure 3 and 4 are the schematic representation of the shaft loading in the vertical direction exerted by the roller drum while Figure 5 is the vertical bending moment diagram. Also, Figure 6 is the schematic diagram of the shaft loading on the horizontal direction exerted by the roller drum. The horizontal bending moment is shown in Figure 7.

(a) Loading diagram

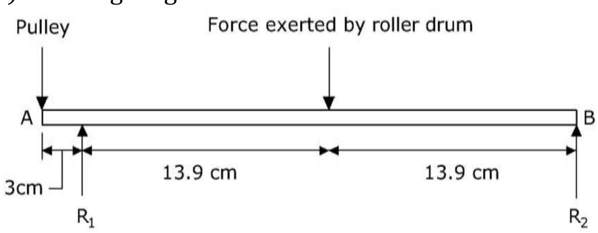

Figure 3: Shaft Loading Diagram

(b) Vertical Loading diagram

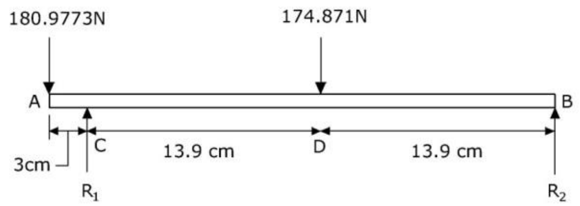

Figure 4: Vertical Loading on Shaft

Solving for reactions,

$$R_1 = R_{1V} = 287.943\text{N} \text{ and } R_2 = R_{2V} = 67.906 \text{ N}$$

(c) Bending moment (B.M) Diagram

$$\text{B. M at A and B, MA} = \text{MB} = 0$$

$$B.M\ at\ C, M_{CV} = 180.977 \times 0.03 = 5.42932Nm$$

B. M at D,

$$M_{DV} = 180.977 \times (0.03 + 0.139) - (287.943 \times 0.139) = -9.439\text{Nm}$$

(d) Vertical Bending Moment Diagram

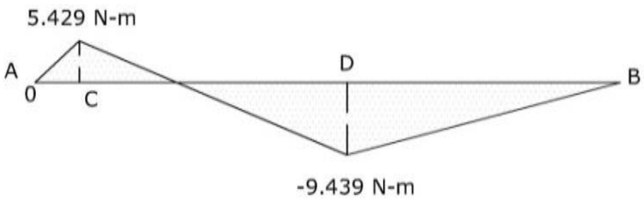

Figure 5: Vertical Bending Moment

(e) Horizontal Loading Diagram

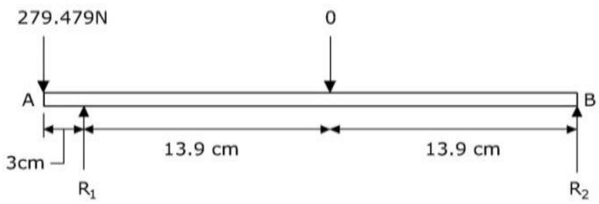

Figure 6: Horizontal Loading Diagram

Solving for reactions

$$R_1 = R_{1H} = 309.639 \text{ N } R_2 = R_{2V} = -30.16 \text{ N}$$

(f) Horizontal Bending Moment Diagram

We know that B. M at A and B, MA $=$ MB $= 0$

$$\text{B. M at C, } M_{CH} = 279.479 \times 0.03 = 8.3844\text{Nm}$$

$$\text{B. M at D, } M_{DH} = 279.479 \times (0.03 + 0.139) - (309.639 \times 0.139) = 4.1922\text{Nm}$$

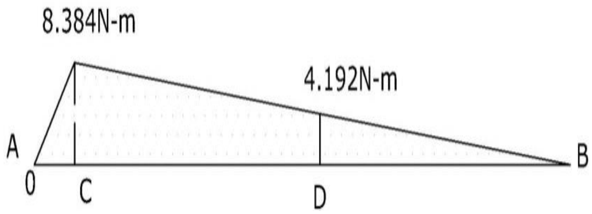

Figure 7: Horizontal Bending Moment Diagram

(g) Resultant Bending Moment Diagram

Resultant bending moment at C

$$M_C = \sqrt{(M_{CV}^2 + M_{CH}^2)} \quad (40)$$

From 40, $M_C = 9.9154$ Nm

Resultant bending moment at D,

$$M_D = \sqrt{(M_{DV}{}^2 + M_{DH}{}^2)} \quad (41)$$

From (41), MD = 10.328 Nm

The resultant bending moment diagram is shown in Figure 8.

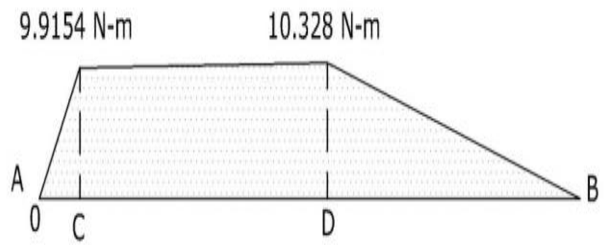

Figure 8: Resultant Bending Moment Diagram

Therefore, the maximum bending moment is 10.328 N

(h) Diameter of the Shaft

Applying the methods of equivalent twisting moment T_e and equivalent bending moment M_e to the shaft diameter, taking the larger values as the minimum shaft diameter. Using equivalent twisting moment, in equation (20); $T_e = 19.76$ Nm

$$T_e = \frac{\pi \times \tau \times d^3}{16} \quad (42)$$

$\tau = 56N/mm^2 = 56$MPa, hence, from (42), d = 12.2mm

Using equivalent bending moment, from equation (21), $M_e = 17.62$ Nm

But, equivalent bending moment (M_e) is given by:

$$M_e = \frac{\pi \times \sigma_b \times d^3}{32} \quad (43)$$

With $\sigma_e = 115$MPa; d = 11.6mm, Me = 17.62Nm

2.3.8 Resultant Forces Acting on Framed Structure

The frame structure that housed the various components should be able to withstand the forces acting on it to improve system stability. The following parameters were determined.

Mass of motor = 3.5kg, hence, weight of motor = mg = 9.81 x 3.5 = 34.34 N

Force pulling motor from slope = 34.34 sin 3017.17N

Force bolts holding motor = 34.34 cos30 = 29.74 N

$$F_b = \sqrt{(29.74)^2 + (17.17)^2} = 34.34 \text{ N}$$

Total Weight of shaft, pulping drum and bearing = 67.51 N

2.4 Fabrication Procedure

The main components fabricated/machined include; the frame stand that housed the other components, pulley, roller drum, drum compartment, guide, tissue scrubbing bar and shaft. The frame-stand is made of mild steel angle bar and squared pipe, cut to the required dimensions and assembled using Shielded Metal Arc welding (SMAW) process. The pulley, roller drum and shaft were also cut and machined to the required dimensions, sizes and shapes using the lathe machine. Each fabricated parts were assembled to form the single unit (extraction machine) as shown in figure 9.

2.5 Description of the Machine

When the electric motor is powered, the drive mechanism is activated, which creates a rotating force at the roller drum compartment. The rolling drum is the rotating component of the machine that strips the sliced plantain trunk (pseudo stem) into fibre strands..

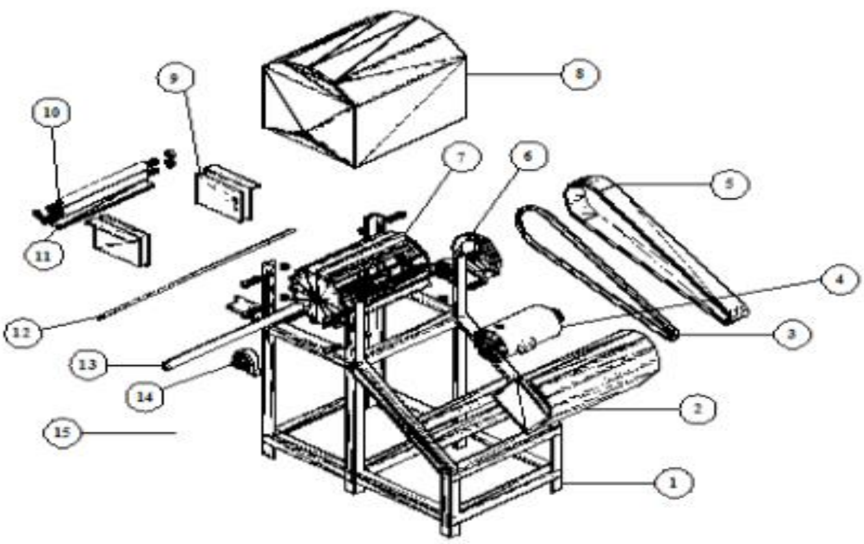

Figure 9: Exploded view of the fabricated plantain fibre extraction machine

(1) Frame, (2) Motor cover, (3) Belt, (4)Motor, (5) Belt cover, (6) Pulley, (7) Roller drum, (8) Drum compartment, (9) Guide, (10) Inlet, (11) Tissue scrubbing bar, (12) Outlet, (13) Shaft, (14) Bearing, (15) Switch

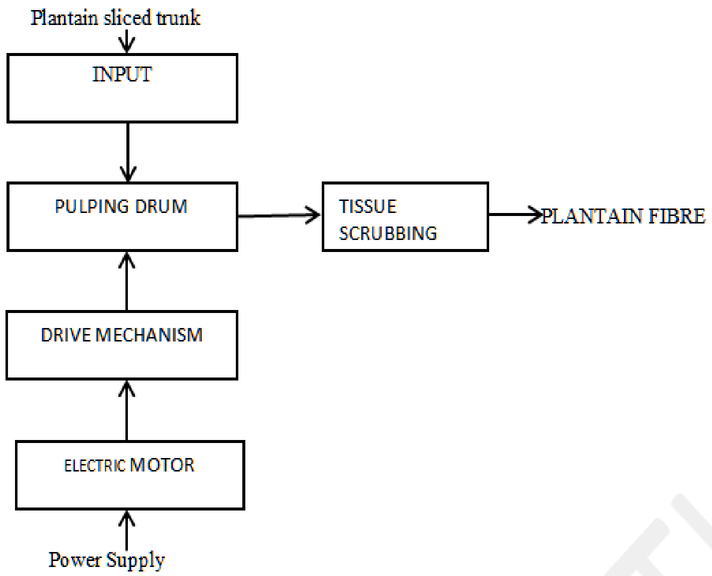

Figure 10: System Block Diagram for fibre extraction machine

The rotating force pulps the plantain ribs that were fed into the receptacle (inlet), to generate a torque greater than the plantain rig tissue strength, estimated with the expression in equation (3). The pulped plantain rib is pressed against a scrapping bar which contains scrapping blades surface to remove the crushed tissues. The pulping drum is fixed on two bearings to reduce power loss due to friction. Figure 10 is a simplified block diagram showing the extraction process adopted

2.6 AutoCAD Drawing

The first angle isometric drawings of the plantain fibre extraction machine are shown in Figures 11. All dimensions were in millimetres. Figure 12 is the pictorial view of the extraction machine.

2.7 Operational Procedure

The Plantain fibre extracting machine is an electric powered device. It needs a supply voltage of 220V to operate effectively. The guideline to operate the machine is stated as follows:

i. Connect the machine to the appropriate power source
ii. Ensure there is ample space around the machine for the operator
iii. Stripped the pseudo stem into appropriate Plantain sliced trunk
iv. Feed the Plantain sliced trunk into the pulping drum gently
v. Repeat the process to obtain the extracted fibres.

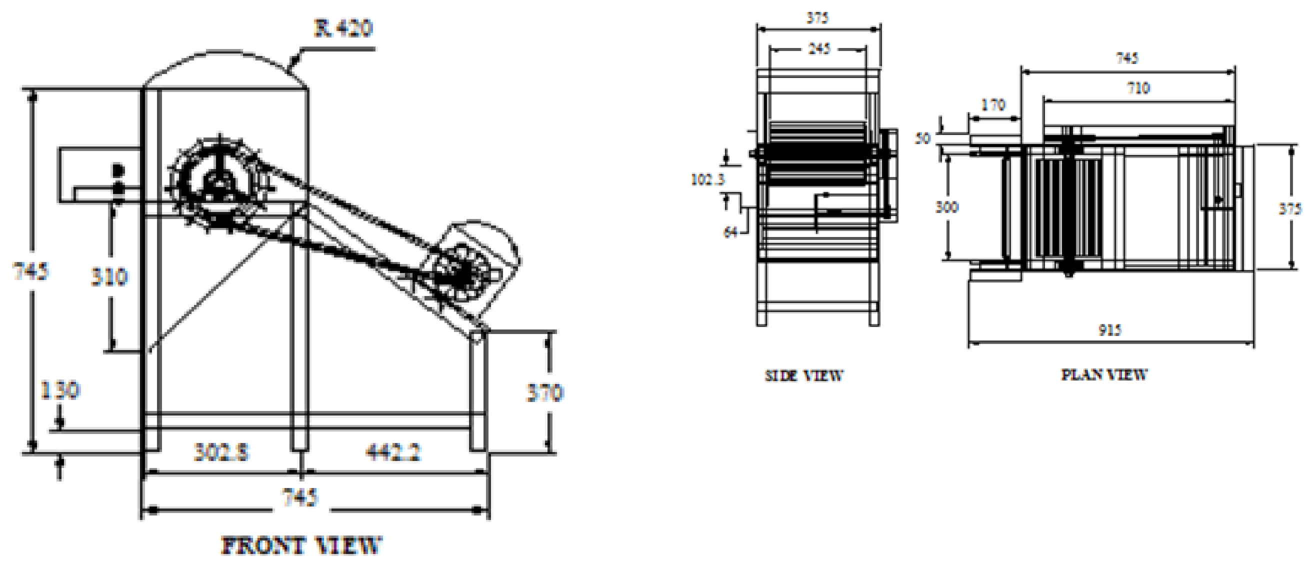

Figure 11: Isometric Drawing of Plantain Fibre Extraction Machine

Figure 12: Pictorial Front View of the Plantain Fibre Extracting

3. RESULTS AND DISCUSSION

The design calculations were employed in the fabrication of the machine. The machine was tested to ascertain the integrity and functionality of the device. At the design stage, the computation of the length, width and thickness of the plantain sliced trunk were determined. The thicknesses of plantain sliced trunk used for these test are: 4.0 mm, 4.5 mm, 5.0 mm, 6.0 mm, 6.5mm, 7.0 mm, 7.5 mm, 8.0 mm, 8.5 mm, 9.0 mm and 10 mm. The test was carried out with a 220 V single phase induction motor using a stop clock to determine the extracting time. From the test result obtained, the plantain fibre extraction machine could extract a sliced trunk of maximum width and thickness of 150 mm and 10 mm, respectively, while the length varies. The results obtained from the test are presented in the Table 1.

Table 1: Test Results

S/N	Plantain Sliced Trunk Length (mm)	Width (mm)	Thickness (mm)	Operating Voltage(V)	Fibre Extracting Time (sec)
1	150	138	8.5	220	13
2	510	30	7.5	220	17
3	480	72	5.0	220	18
4	470	40	4.0	220	18
5	562	70	7.0	220	20
6	350	130	6.0	220	26
7	310	150	4.0	220	27
8	390	139	4.5	220	28
9	490	140	8.0	220	30
10	500	145	9.0	220	31
11	520	146	6.5	220	32
12	370	150	10	220	36
13	640	150	10	220	42
		Average Extracting time			26

3.2 Simulations Result

3.2.1 Frame:

From the simulation of the frame stand in Figure 14, we observed that the frame would be subjected to a minimum stress of $5.75\text{x}10^{-7}$ N/mm^2 with 12790 nodes and a maximum stress of 0.74032 N/mm^2 with 18407 nodes at the electric motor region. The frame has a maximum stress of $5.75\text{x}10^{-7}$ N/mm^2 as against the yield strength of 620.422 N/mm^2 of the material selected (steel). From figure 14 the frame experienced more displacement at the electric motor region due to torque and reactions from the belt drive system and vibrations of the motor. The action of the motor driving the rolling drum enforces a counter reaction to balance and keep the system in equilibrium, resulting in the displacement of the horizontal members attached and which hold the motor firmly in position. The result showed that the frame could withstand the load, stress and displacement during operation.

3.2.2 Rolling Drum

Figure 15 shows the Von Mises stress analysis of the rolling drum under stress-strain conditions (with fixed geometry, torque, centrifugal force and reaction forces of fibre on rolling drum blade). The material selected was an alloy steel with yield strength of 620.422 N/mm^2, tensile strength of 723.826 N/mm^2, elastic modulus of 21,000 N/mm^2, shear modulus of 79,000 N/mm^2, a density of 7.700 g/cm^3, Poisson ratio of 0.28. The von Mises stress analysis shows a minimum stress of 1.2110 x 10^{-7} N/mm^2 with 850 nodes and a maximum stress of 0.697179 N/mm^2 with 14,947 nodes.

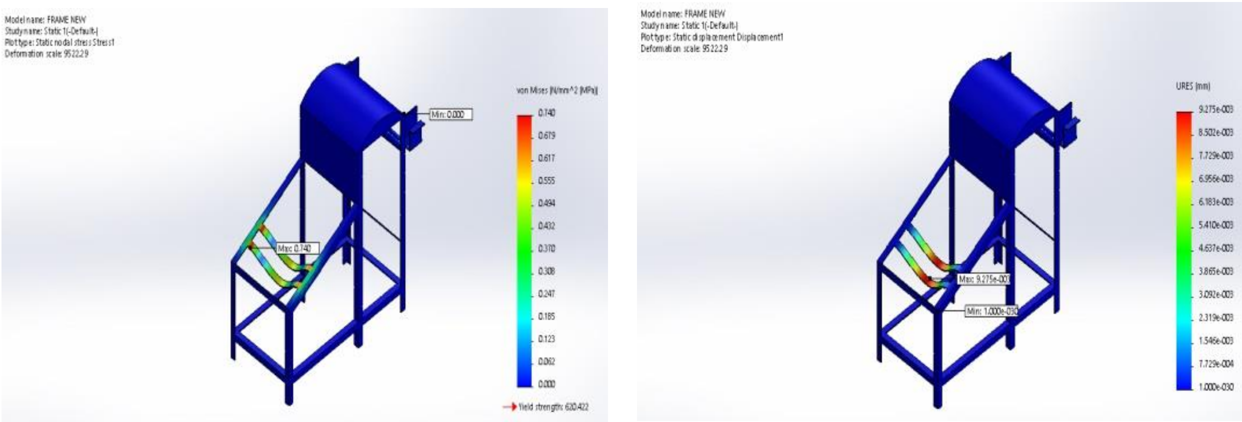

Figure 14: Von Mises Stress analysis and displacement of structure under stress-strain condition of Frame Structure

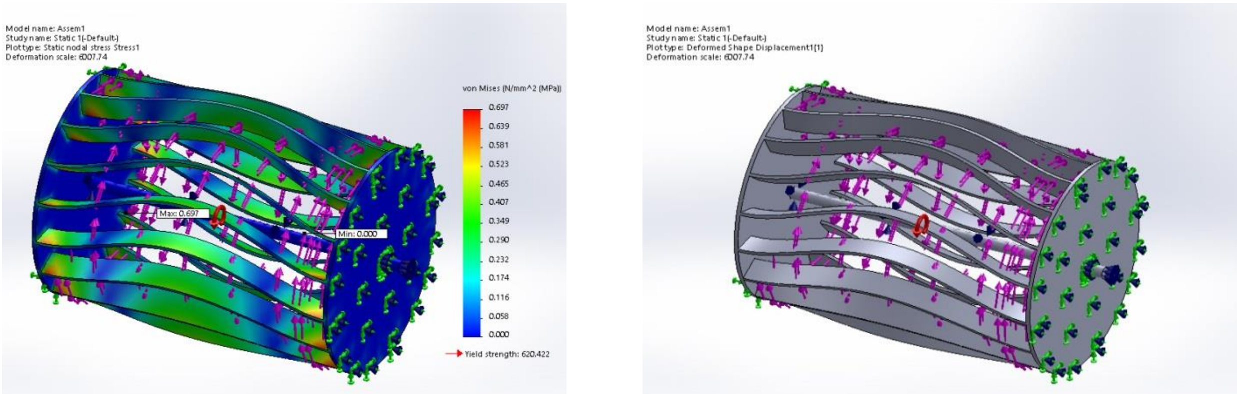

Figure 15: Von Mises Stress Analysis of Rolling Drum under stress-strain conditions

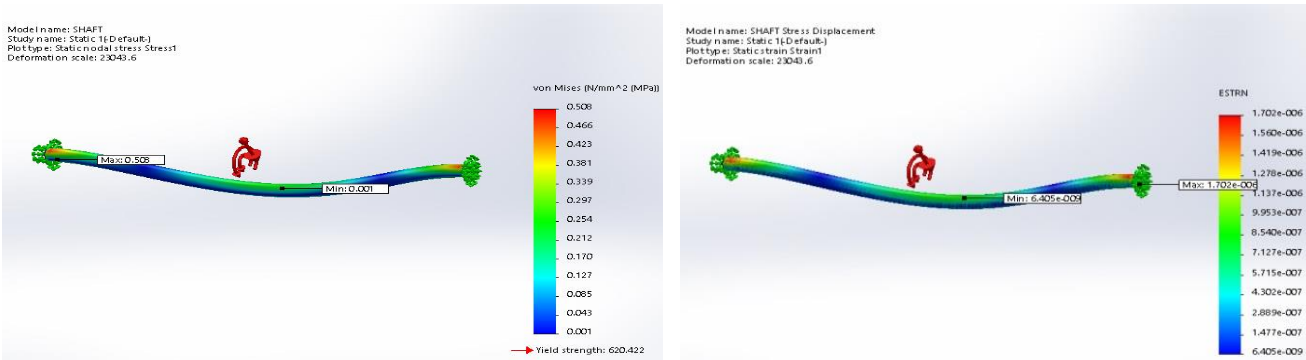

Figure 12: Von Mises Stress-Strain Analysis of Shaft

The simulation result for the rolling drum indicated that it could withstand the stresses and strains it might encounter in service, since the maximum stress of the the simulated result is less than the yield strength of steel used in the fabrication of the rolling drum.

3.2.3 Shaft

The shaft transmits drive from the motor to the shredding device via the belt drive system; through its rotation about the bearing axis. Due to its rotation, it is subject to torque, shear force and bending moments. Figure 16 shows the von Mises stress analysis of the shaft which gave a minimum stress of 0.000588227 N/mm^2 with 2,802nodes and a maximum stress of 0.507989 N/mm^2 with 1,234 nodes which is less than the yield strength of the material of 620.422 N/mm^2. From the bending moment diagram in Figure 8, the shaft would experience deformed maximum at value of 10.328Nm, an indication that the shaft could withstand the various stress and strain during operation.

4. CONCLUSION

In this work, plantain fibre extraction machine has been designed and fabricated. The design analysis obtained show that a 2 horsepower motor could drive the machine. The total length of flat-belt to drive the pulley was 1.47 m at an angle of lap on the smaller pulley of 2.87 rad. The resultant load of 174.871N act on a diameter shaft of 11.6 mm with maximum bending moment of 10.328 Nm. Total Weight of shaft, pulping drum and bearing on frame was 67.51N. Also, a force of 150 N, 200 N and 250 N could pulp a thickness of plantain ribs of approximately, 6 mm, 6.5 mm and 7 mm, respectively. The test result showed that the machine could extract a sliced pseudo stem thickness between 4.0 mm and 10 mm at 27 and 42 seconds, respectively. The plantain fibre extraction machine consists of simple components with ease of assembly and disassembly, making the maintenance of the machine easy for operators. However, it is recommended that further work be done on the efficiency and optimum performance of the plantain fibre extraction machine at low cost.

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