Finite Element Analysis of a Washing and Preheating Unit Designed for Plantain Flour Process Plant

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Abstract- This paper presents the design and finite element analysis (FEA) of a washing, conveying and preheating unit, intended to help complete the drying of plantain within an hour at a maximum temperature of 70°C in a plant that processes unripe plantain into flour. Model was developed for it, its functionality and structural stability were evaluated using SolidWorks computer aided design application software after selection of suitable materials for and design analysis of its components. Its frame analysis showed a maximum stress of 52.52 MPa, resultant displacement of 0.9215 mm, elastic strain of 1.495×10^{-4} and a minimum factor of safety (FOS) of 4.7, when a force of 1 kN was applied. FEA of each control-gate-spring showed a maximum stress of 29.05 MPa, resultant displacement of 0.6902 mm, elastic strain of 9.547×10^{-5} and a minimum of 7.6 FOS, when a tensile force of 0.12 kN was applied. FEA of each roller showed a maximum stress of 50.16 MPa, elastic strain of 1.458×10^{-4} and a minimum of 3.4 FOS, when a torque of 1.023 kNm was applied. These results imply that the design of the unit is safe/adequate for fabrication, and able to satisfactorily serve its intended purpose, since the yield strengths of the materials selected were not exceeded as evidenced by the FOS values obtained. This unit, when fabricated and incorporated into the process plant, would help to drastically reduce plantain processing time in order to be able to meet the increasing demand for plantain flour.

Keywords: Design, Finite Element Analysis, Plantain Washing, Conveying Preheating, Flour Process Plant.

1. Introduction

Hygiene is very important in processing of farm produce into value-added and storable products. It is important for healthy living. Farm produce must be thoroughly washed after harvesting in order to remove soil, dirt, chemical residues and other foreign materials before processing them [1, 2]. If they are not thoroughly washed, it can adversely affect their quality; lead to spoilage and mold growth during storage; foodborne illness; reduced shelf life and profits [3]. Washing of farm produce is a surface treatment and a value-addition operation which can minimize microbial contamination, ensure food safety and prevent other food hazards [4]. Washing of farm produce is usually achieved through still water, moving water, water flood, water spray, rotating drum, paddle conveyor washer, belt conveyor, eccentric drum washer and hand washing methods. The mechanical action employed or used in each of these methods is very important for efficient washing to be done. There is also a need for washing operation in processing of farm produce into value-added products in order to comply with food safety and processing policies.

Hence, a lot of researchers have worked on machines, systems and equipment for washing farm produce in food processing industries.

Mendenhall *et al.* [1] designed a fruit and vegetable washer for foodservice industries by modifying a household dishwasher. The modification was achieved by introducing a new holding rack and a new water jet system into the dishwasher. To ensure intensity and uniformity, several standard nozzles were tested on a spray table. A small-scale mechanical carrot washer was developed by Moos *et al.* [2] for preparing research samples. Sample sizes (ranging from 3 to 11 kg), operator's safety, a low operating speed (to prevent bruising and breakage), low water pressures and flow rates, retention of small pieces, ease of sample loading and unloading, time and cost savings compared with manual washing systems were all considered in developing the washer. The washer is a non-immersion rotary washing system that made use of a horizontal 208 liters barrel. The barrel was equipped with a low–pressure spray wand and was supported by roller drive wheels. Ambrose and Annamalai [4] also developed a manually operated batch-

type washing machine for root vegetables with a holding capacity of 10 kg. The development of the machine was based on the anthropometric data of agricultural workers for easy operation of the machine. An average efficiency of 93.5% was obtained from performance trials of the machine.

A fruit washing machine was also developed by Oyeleke *et al.* [5] to operate in batches of 50 oranges per batch. The machine consists of feeding tray, pulley, belt, belt guard, motor seat, an electric motor, hose, frame, rollers, water pump, fruit collector, water tank, drain net, shaft, bearing, conveyor belt, pipes, washing chamber and nozzle. The machine was observed to have a washing capacity and efficiency of 16.3 kg/h and 62.5% respectively. This machine could only wash and convey fruits horizontally, but can be used for inclined operations as in plantain processing plant by incorporating flights and other accessories. Kenghe *et al.* [6] developed the prototype of a mechanical fruit washer, which has a capacity of 50 kg, for the purpose of washing potatoes. The potatoes are manually loaded/fed into the cleaning unit, which is a removable rectangular box that is made of metallic net. The potatoes are washed through water turbulence that is made available from the bottom and sides of the box. The box is manually removed and offloaded after the washing process has been completed and the process is repeated again. The highest washing capacity and performance index obtained from the performance test of the mechanical washer, when the rotor's speed was 1486 rpm, were 98.18% and 3.26 respectively. This machine is not suitable for use in a plant for processing plantain into flour due to much of manual involvement. It can be said that all the machines developed by the aforementioned studies or researchers are only for batch production and that none of them, except the one developed by Oyeleke *et al.* [5], is suitable for use in a process plant where continuous flow is a major factor.

Afolami *et al.* [7] also developed and evaluated the performance of a yam washing machine for a *poundo* yam flour processing plant. The machine consists of water sprinkling system, auger shaft, washing brushes, washing chamber, frame, water pump, water tank, water inlet and outlet channels. The auger shaft acts as conveyor while the washing brushes and water jet effect the necessary washing actions on yam tubers that are introduced into the washing chamber. The efficiency and washing capacity of the machine were estimated to be 77% and two tubers of yam per minute respectively. It was observed that the washing principle employed by the machine cannot be used for washing plantain because, if the same washing principle is employed, major part of the plantain will be lost to washing. Hence, the machine is not suitable for use in a plant that processes plantain into flour. There is still a need to specifically develop a unit for plantain processing plant, which will be able to wash, transport and pre-dry or preheat plantain pulps before particulation in order to prevent the particulates from sticking together before and during drying, and to ensure that drying of plantain takes place in one hour at a maximum temperature of 70°C.

Moreover, this need came into being as a result of the quest to manage diabetes in Nigeria; and as a result of increased demand for plantain and its products arising from increased awareness about its nutritional and medicinal values [8, 9]. Study showed that diabetes is a risk factor for coronavirus (COVID-19) [10]; and its presence is usually associated with worse outcomes as well as increased mortality in COVID-19 patients [11, 12]. Plantain is known to be very good and cheap for managing diabetes mellitus without any side effects [13]. Plantain is also said to be the fourth most important crop throughout the world after wheat, rice and corn, with annual production of 32.01 million tons and very high postharvest losses due to its perishable nature [14]. Hence, the aim of this study is to design a unit for washing, conveying and preheating plantain, which will be suitable for use in the production line of a plant that processes unripe plantain into flour.

Right from time immemorial, washing of farm produce is usually done by manual means. For mass production of flour from unripe plantain and to reduce or eliminate human involvement in plantain processing, there is a need to specially develop a machine, which will be able to perform the aforementioned functions in order to reduce plantain processing time. Hence, for the plantain flour process plant, belt conveyor was considered suitable for this operation due to its inherent advantages. Belt conveyor is a material handling system that helps in transporting materials from one location to another [15]. Its usage is characterized by the following advantages: it is safe, economical, reliable, versatile, practically unlimited in range of capacities and suitable for performing numerous processing functions in connection with its normal purpose of providing a continuous flow of material between operations. Low labor and energy requirements are fundamental with belt conveyors as compared with other means of transportation [16].

2. Materials and Methods

2.1. Design Concept of the Washing and Preheating unit

Model for the unit was developed using SolidWorks computer aided design (CAD) application software. The unit assembly, which is to handle 2000 kg of plantain pulps per day, consists of a loading hopper, two cams, two followers, two springs, a perforated conveyor belt equipped with 21 flights, an electric motor, four rollers, water tank, water pump, water pipes, two sprayers, heater blower device, support frame and a housing for the washing chamber as shown in Fig. 1. Plantain pulps are manually fed into the loading hopper, from where they are introduced into the conveyor belt through a cam and follower assembly, which opens and closes the outlet gate of the loading hopper. The outlet gate of the loading hopper is spring-loaded on both sides for its return and to control the release of plantain pulps into the belt conveyor. A cam is attached on each side of the driving shaft of the conveyor belt while a follower is attached to each side of the outlet gate of the loading hopper. As the shaft rotates, the cam and follower assembly opens the outlet gate while the springs close the gate. Washing of plantain pulps are done as they travel on

the belt conveyor. The belt conveyor is perforated to allow water to drain off the surfaces of the plantain pulps. A blower-heater device also blows hot air into the washing chamber, after the washing process, to pre-dry the surfaces of the plantain pulps before particulating operation begins in order to hasten drying process in the drying section. In-between the two sprayers in the washing chamber are two rollers, which are equipped with cam-like profiles to help turn plantain pulps for thorough washing and preheating.
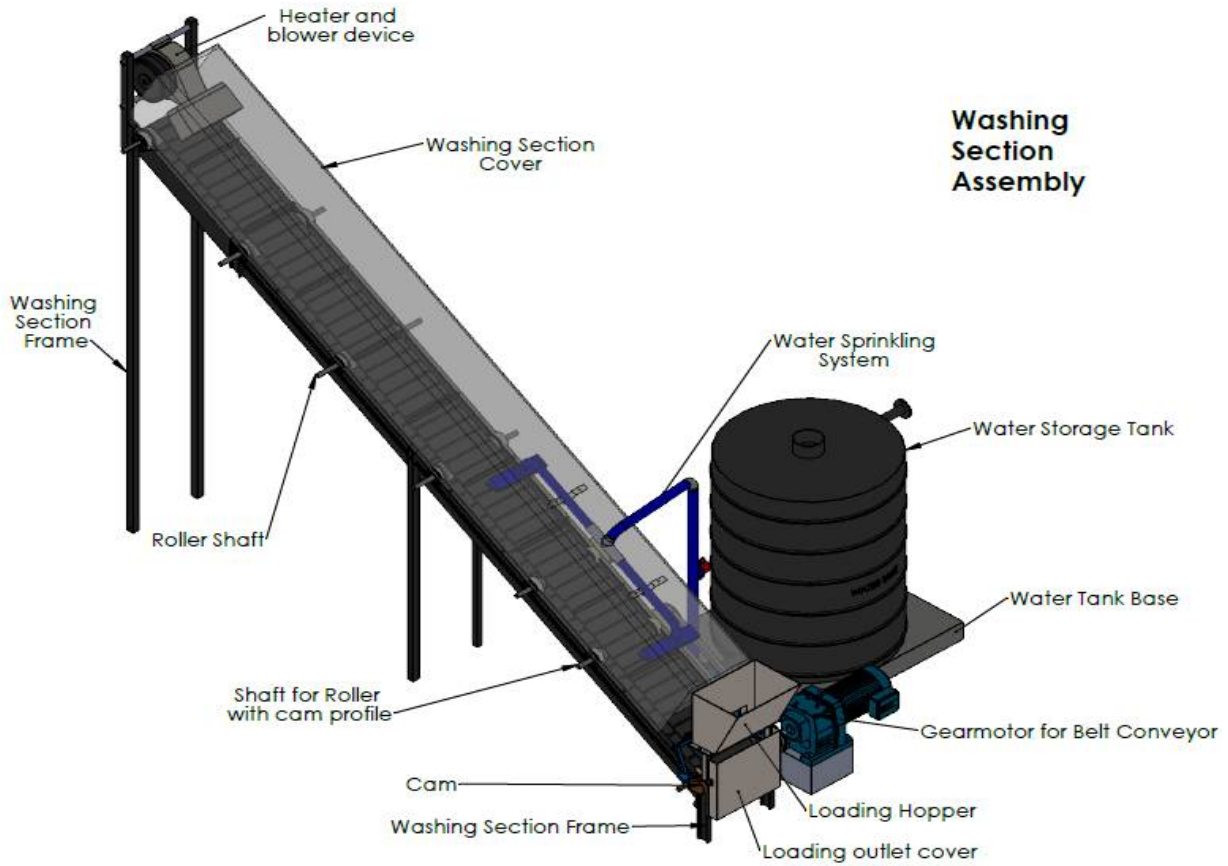
Fig. 1. Isometric view of the washing, conveying and preheating unit

2.2. Design Analysis of the Washing and Preheating Unit

2.2.1. Design of the loading hopper

The loading hopper is a stainless steel structure in form of a hollow trapezoidal prism as shown in Fig. 2. The stainless steel used for the hopper is 2 mm thick. According to Macrae *et al.* [17], the area and volume of a trapezoidal prism can be obtained using Eq. (1) and (2) respectively. Hence, the whole-prism Area, A_{wp} was calculated as 120000 mm^2 or 0.12 m^2 using Eq. (1); while the whole-prism volume V_{wp} was calculated as 0.066 m^3 or 6.6 $\times 10^7$ mm^3 using Eq. (2). The cut-away-prism volume V_{cap} was calculated as 64,323,168 mm^3 using Eq. (3), while the volume of the rectangular gate (V_{gate}) was calculated as 252027.52 mm^3 using Eq. (4).

$$A_{wp} = \frac{1}{2}(A + B) \times C \quad (1)$$

$$V_{wp} = \text{Area} \times \text{Length} = \frac{1}{2}(A + B) \times C \times \text{L} \quad (2)$$

Where: A is 400 mm, B is 200 mm, C is 400 mm, L is the length which is 550 mm, X is 400 mm, Y is 127.03 mm and Z is 100 mm.

$$V_{cap} = \frac{1}{2}((A - 4) + (B - 4)) \times (C - 2) \times (\text{L} - 4) \quad (3)$$

$$V_{gate} = Outer\ volume - Inner\ volume = (X \times Y \times Z) - ((X - 4) \times (Z - 4) \times Y) \quad (4)$$
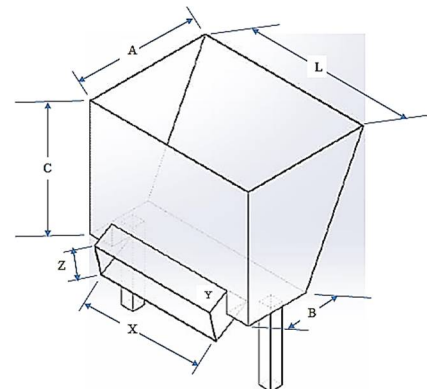

Fig. 2. Loading hopper for the unit

Volume of the cut-away-part, V_{cg} (to create space for the rectangular gate) was calculated as 105042.96 mm^3 using Eq. (5), while the net volume of loading hopper (V_{Net}), which is the real volume of hopper, was calculated as 1823816.56 mm^3 or 1.824 $\times 10^{-3}$ m^3 using Eq. (6).

$$V_{cg} = ((X-4) \times 69.54 \times thickness) + ((X-4) \times (65.09-2) \times (thickness)) \qquad (5)$$

$$V_{Net} = V_{wp} - V_{cap} + V_{gate} - V_{cg} \qquad (6)$$

Mass of the loading hopper, m_{Lh} was calculated as 14225.77 g or 14.23 kg using Eq. (7) since the density of the material used, stainless steel (ferritic), is 7800 kg/m^3. The weight of the loading hopper (W_{Lh}) was also determined to be 139.6 N using Eq. (8).

$$m_{Lh} = Material\ density \times Hopper\ volume \qquad (7)$$

$$W_{Lh} = m_{Lh} \times acceleration\ due\ to\ gavity \qquad (8)$$

2.2.2. Conveyor belt design

According to Dunlop-Enerka [18], Khurmi and Gupta [19], a belt conveyor of width 400 mm and thickness 10 mm was chosen since the length of plantain pulp ranges from 200 mm to 300 mm [20, 21, 22]. Roller length or width (L_R) of 25% greater than belt width was selected; the roller's diameter (D_R) was also chosen to be 140 mm [18, 19]. The roller length or width was determined to be 500 mm using Eq. (9). Hence, the length or width of driving as well driven roller is 500 mm. The belt length (L_{BL}) was also determined to be 9758 mm or 9.758 m using Eq. (10).

$$L_R = 1.25 \times Belt\ width = 1.25 \times B_w \qquad (9)$$

$$L_{BL} = \frac{\pi}{2}\left(D_{driving\ roller} + D_{driven\ roller}\right) + 2L_{CL} + \left(\frac{(D_{driving\ roller} - D_{driven\ roller})^2}{4L_c}\right) \qquad (10)$$

Note that $D_{driving\ roller} = D_{driven\ roller}$

Where: $D_{driving\ roller}$ is the driving roller's diameter, which is 140 mm; $D_{driven\ roller}$ is the driven roller's diameter; and L_{CL} is the distance between the centers of driving and driven rollers (also known as conveying length), which is 4659.04 mm or 4.66 m [19].

The number of flights on the conveyor belt was determined to be 21 using Eq. (11). Since the conveyor is expected to be very slow for thorough washing of plantain pulps and for water to completely drain off their surfaces, belt speed v_B was chosen to be 6 m/min. or 0.1 m/s. Hence, roller's speed N_R was determined to be 14rpm using Eq. (12).

$$n_f = \frac{Belt\ length}{Distance\ between\ two\ flights} = L_{BL}/d_f \qquad (11)$$

$$N_R = (60 \times v_B)/(\pi D_R) \qquad (12)$$

2.2.3. Conveying capacity of the belt

According to Dunlop-Enerka [18], Fenner Dunlop [23] and Daniyan et al. [24], belt's conveying capacity is a function of belt speed, cross-sectional area of Load stream and bulk density of the conveyed material. From Eq. (13) and (14), the load stream volume and mass were determined to be 17.388 m^3/h and 6.96 tons/h respectively. Therefore, the conveying capacity of the belt is 6.96 tons/h or 6960 kg/h.

$$Q_v = A \times v_B \times 3600\ m^3/h = A \times v_B\ m^3/s \qquad (13)$$

$$Q_m = Q_v \times \rho = A \times v_B \times 3600 \times \rho\ (tons/hr) \qquad (14)$$

Where: Q_v is the load stream volume in (m^3/h); A is the cross-sectional area of Load stream, which is given as 0.0483 m^2; v_B is the belt speed in (m/s), which is 0.1 m/s; Q_m is the load stream mass in (tons/h), which is the belt's conveying capacity; and ρ is the bulk density of the conveyed material (plantain pulp), which is given as 0.40 tons/m^3 or 400 kg/m^3.

2.2.4. Power requirements for the conveyor belt

As recommended by Dunlop-Enerka [18], power requirements for the belt conveyor were assessed using Eq. (15) to (18). Required motor power, P_{RM} was obtained from Eq. (19). It is to be noted that the next standard electric motor that is greater than P_{RM} value obtained from Eq. (19) was chosen as the required, sufficient motor power value [24]. The power (P_{EL}) for empty conveyor and load over the horizontal distance in (kW), conveyor elevation (H), power (P_{LF}) for lift or fall distance in (kW), and power at driving roller (P_{DR}) were determined to be 0.019 kW, 2.33 m, 0.0442 kW and 0.0632 kW from Eq. (16), (18), (17) and (15) respectively. Thus the minimum motor power needed for sizing the motor is 0.07 kW. The next larger standard electric motor is 1.5 kW; hence, the required electric motor power for the conveyor belt is 1.5 kW.

$$P_{DR} = P_{EL} + P_{LF}\ (kW) \qquad (15)$$

$$P_{EL} = \left(C_B \times v_B + Q_m/C_L \times k_f\right) \qquad (16)$$

$$P_{LF} = (H \times Q_m/367) \qquad (17)$$

$$H = \sin\theta \times L_{CL} \qquad (18)$$

$$P_{RM} = P_{DR}/\eta\ (kW) \qquad (19)$$

Where: C_B is the width factor in (kg/m), which is given as 54 kg/m; C_L is the length factor, which is given as 555 m^{-1}; k_f is the service or working conditions factor, which is given as 1.17 for slow speed; H is the conveyor elevation in (m), which is the horizontal distance between driving and driven rollers; θ is the angle of inclination, which is 30°; and η is the efficiency of the drive, which is given as 0.96 for motors or drives with pulley.

2.2.5. Peripheral and tension forces

As recommended by Dunlop-Enerka [18], the peripheral force (F_U) on the rollers was calculated as 632 N using Eq. (20). The tight side (T_1) and slack side (T_2) tension forces of the conveyor belt were estimated to be 2344.72 N and 1712.72 N from Eq. (21) and (22) respectively.

$$F_U = (P_{DR} \times 1000/V_B) = 632\ N \qquad (20)$$

$$T_1 = F_U \times C_1 \qquad (21)$$

$$T_2 = F_U \times C_2 \qquad (22)$$

$$C_2 = 1/(e^{\mu\alpha} - 1) = 2.71 \qquad (23)$$

$$C_1 = 1 + C_2 = 1 + 1/(e^{\mu\alpha} - 1) = 3.71 \qquad (24)$$

Where: C_1 is the tight side drive factor; C_2 is the slack side drive factor; and μ is the coefficient of friction between the

conveyor belt and roller, which is 0.1; and α is the angle of wrap on the rollers (also known as arc of contact or angle of contact between the conveyor belt and each roller) in radians, which is 3.142 radians since the driving and driven rollers have the same diameter.

2.2.6. Spring design

The spring used is a helical tension spring made of carbon steel wire of circular cross-section (see Fig. 3). Its modulus of rigidity is 80 kN/mm^2; its modulus of elasticity is 210 kN/mm^2; and its allowable shear stress is 420 MPa or 420×10^6 N/m^2 [19]. The total weight of the loading hopper gate was obtained as 94 N using Eq. (25). Hence, the weight of the loading hopper gate that is acting on each spring was obtained from Eq. (26) as 47 N.

$$W_{gate} = \rho_{gm} \times V_{gate} \times g \approx 94\,N \tag{25}$$

$$W = W_{gate}/2 = 47N \tag{26}$$

Where: ρ_{gm} is the density of the gate material, which is 7800 kg/m^3; V_{gate} is the volume of the loading hopper gate, which is 1.2×10^{-3} m^3 and g is acceleration due to gravity, which is 10 m/s^2.
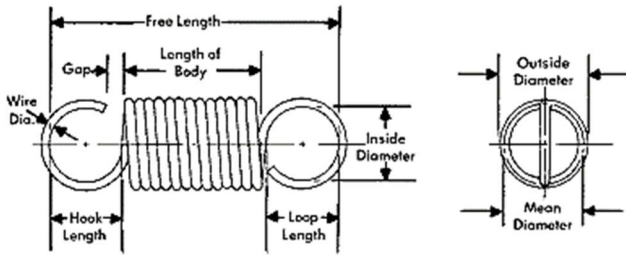
Fig. 3. Nomenclature of tension helical spring [25]

Based on the available space in the cam and follower assembly, a spring of 30mm outside diameter was chosen. A wire diameter of 3mm (0.003m) was then selected from wire diameter table [26]. According to Joerres [25], an extension helical spring with cross-center hook and loop end configuration was chosen for the operation. The spring index was therefore calculated as 9 using Eq. (29).

$$D_{sm} = (D_{so} - d_{sw}) = 27mm \tag{27}$$

$$D_{si} = (D_{sm} - d_{sw}) = 24mm \tag{28}$$

$$C = D_{sm}/d_{sw} = 9 \tag{29}$$

Where: D_{sm} is the spring's mean diameter; D_{so} is the spring's outside diameter; d_{sw} is the spring's wire diameter; D_{si} is the spring's inside diameter; and C is the spring index (see Fig. 3).

According to Childs [26], the maximum shear stress induced in the spring and Wahl's stress factor (K_w) were determined using Eq. (30a) and (30b) to be 139.1 MPa and 1.1621 respectively. This shear stress is acceptable since it is less than the allowable shear stress (420 MPa) of the spring's material.

$$\tau_{max} = K_w(8WC/\pi d_{sw}^2) \tag{30a}$$

$$K_w = \{(4C - 1)/4(C - 1)\} + \{0.615/C\} = 1.1621 \tag{30b}$$

The distance to be covered by the spring is 150mm. The free length of the spring should either be equal to or less than 150mm but must not be greater than 150mm. Hence, a spring of 120mm free length was chosen. According to Joerres [25], both L_{hook} and L_{loop} are equal to mean diameter of the spring. Hence, the number of active coils was obtained as 21 using Eq. (32).

$$L_B = L_F - (L_{hook} + L_{loop}) = 66mm \tag{31}$$

$$n_a = (L_B - d_{sw})/d_{sw} = (L_B/d_{sw}) - 1 \tag{32}$$

According to Childs [26], Budynas and Nisbett [27], the spring stiffness or rate was calculated as 1.96 N/mm using Eq. (33). Hence, the axial deflection of the spring was determined to be 24mm using Eq. (34).

$$k = Gd_{sw}/8C^3n_a = 1.96\,N/mm \tag{33}$$

$$\delta = W/k \approx 24mm \tag{34}$$

Where: L_F is the total or overall length of the spring; L_B is the body length of the spring; L_{hook} is the hook length of the spring, which is 27mm; L_{loop} is the loop length of the spring, which is 27mm; n_a is the number of active coils; k is the spring stiffness or rate; G is modulus of rigidity of the spring material, which is 80 kN/mm^2; and δ is the axial deflection of the spring.

2.2.7. Design of the water system

The water system consists of a water tank, a pump, pipes and two sprinklers. Water is pumped from the storage tank into the washing chamber. Since 1000 kg of flour will be produced per day and 50% of the weight of wet plantain pulps will be lost to drying, it was assumed that 2000 kg of plantain pulps are expected to be washed per day and the plant is expected to operate for 8 hours per day. Thus the washing capacity of the washing unit is 250 kg/h. It was also assumed that 3 liters or 0.003 m^3 of water will be needed to wash 1 kg of plantain pulp. Hence, 6000 liters or 6 m^3 of water will be used to wash 2000 kg of plantain pulps per day. Therefore, the flow rate $\dot{V}$ of water through the pipe was estimated as 0.75 m^3/h or 2.1×10^{-4} m^3/s using Eq. (35).

$$\dot{V} = \frac{Volume\ of\ water\ used\ per\ day}{Operation\ hours\ per\ day} \tag{35}$$

$$Water\ velocity\ in\ the\ pipe, v = \dot{V}/A \approx 0.7\,m/s \tag{36}$$

$$A = \pi D_{ip}^2/4 = 3.142 \times 10^{-4}m^2 \tag{37}$$

Where: A is the pipe' cross sectional area; and D_{ip} is the inside pipe diameter, which is 0.02 m.

According to Miller [28], Rajput [29], Cengel and Cimbala [30], the operating pressure or the total system head/energy H$_{Total}$ was determined to be 3.5 m by using Eq. (38).

$$H_{Total} = H_S + \frac{v_{wp}^2}{2g}\left(\frac{f \times L_{tlp}}{D_{ip}} + \sum K_L\right) \tag{38}$$

$$C_f = \frac{0.25}{\left[\log\left(\frac{k}{3.7D_i} + \frac{5.74}{Re^{0.9}}\right)\right]^2} = \frac{0.0791}{(Re)^{1/4}} \approx 7.78 \times 10^{-3} \tag{39}$$

$$f = 4 \times C_f = 0.03112 \tag{40}$$

$$Re = v_{wp}D_i/\vartheta = 10687 \qquad (41)$$

Where: H_S is the static head, which is 3.1 m; f is the friction factor; L_{tlp} is the total length of pipe used, which is 6 m; v_{wp} is the water velocity in the pipe (m/s); g is acceleration due to gravity; K_L is the loss coefficient of the pipe fitting components, which was calculated to be 5.4; C_f is the coefficient of friction; k is the roughness factor (m); Re is Reynolds number and ϑ is the kinematic viscosity of water, which is 1.31×10^{-6} m^2/s.

2.2.8. Pump selection

The power requirement for the system was calculated using Eq. (42). Therefore, the mechanical power (also known as brake horsepower) that needs to be supplied by the pump to the water for it to wash plantain pulps is 10 watts. Hence, a 0.746 kW (i.e. 1hp) pump was chosen for the water system.

$$P_{pump} = \rho_w g \dot{V} H_{Total}/\pmb{\eta}_{pump} = 10.301\ W \qquad (43)$$

Electrical power requirement was also calculated using Eq. (44). Hence, the electric power that to be consumed by the pump's motor for a flow rate of 2.1×10^{-4} m^3/s is 11.45 watts.

$$P_{elect} = \rho_w g \dot{V} H_{Total}/\pmb{\eta}_{pump} \times \pmb{\eta}_{motor} = 11.45\ W \qquad (44)$$

Where: ρ_w is water density, which is 1000 kg/m^3; η_{pump} and η_{motor} are pump and motor efficiencies, which are 0.7 and 0.9 respectively [30].

3. Results and Discussion

3.1. Simulation of the Washing and Preheating Unit

The functionality, structural stability/integrity and design adequacy of the washing and preheating unit were evaluated by simulating the model developed for it using SolidWorks CAD application software, since these must precede the fabrication of any machine after completion of its design analysis [9]. This was done to avert any possibility of failure of the unit during service and to ensure optimum reliability of the unit in service. The model of its frame was discretized into 16204 elements and 33048 nodes in order to generate its solid mesh for finite element analysis (FEA) as shown in Fig. 4. The FEA conducted on the frame showed a maximum stress of 52.52 MPa 5, a maximum resultant displacement of 0.9215 mm, a maximum strain of 1.495×10^{-4} and a minimum factor of safety (FOS) value of 4.7 experienced by the frame when a force of 1 kN was applied as shown by its finite element model (FEM) in Fig. 5, 6, 7 and 8 respectively. However, the maximum stress value obtained from the FEA is below the yield strength of the mild steel material selected for the frame (see Fig. 5).

The model for control gate spring was also discretized into 20999 elements and 43994 nodes in order to generate its solid mesh for FEA as shown in Fig. 9. The FEA conducted on each control gate spring showed a maximum stress of 29.05 MPa, a maximum resultant displacement of 0.6902 mm, a maximum strain of 9.547×10^{-5} and a minimum FOS value of 7.6 experienced by it when a tensile force of 0.12 kN was applied as shown in Fig. 10 to 13 respectively. However, the maximum stress value obtained from the FEA is observed to be lower than the yield strength of the plain carbon steel wire selected for the spring as shown in Fig. 10.

Moreover, the conveyor belt roller was discretized into 27697 elements and 54019 nodes for the generation of its solid mesh for FEA as shown in Fig. 14. The FEA conducted on each roller showed a maximum stress of 50.16 MPa, a maximum strain of 1.458×10^{-4} and a minimum FOS value of 3.435 when a torque of 1.023 kNm was applied as shown in Fig. 15, 16 and 17 respectively. Also, in preparation for FEA, each of the conveyor belt roller with cam-like lobes was discretized into 40855 elements and 80961 nodes for its solid mesh generation as shown in Fig. 18. The FEA conducted on each roller showed a maximum stress of 82.81 MPa, a maximum strain of 3.435×10^{-4} and a minimum FOS value of 2.081 when a torque of 1.023 kNm was applied as shown in Fig. 19 to 21 respectively. However, the maximum stress values obtained from the FEA are lower than the yield strength of the stainless steel material selected for the roller without lobes and for the roller with cam-like lobes as shown in Fig. 15 and 19 respectively.
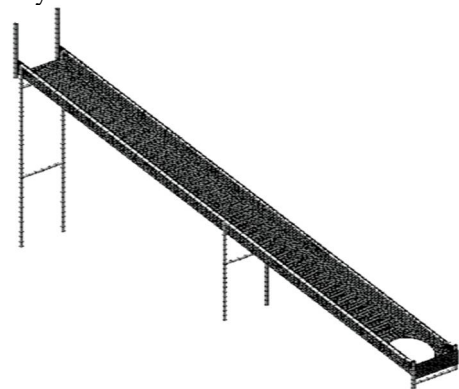

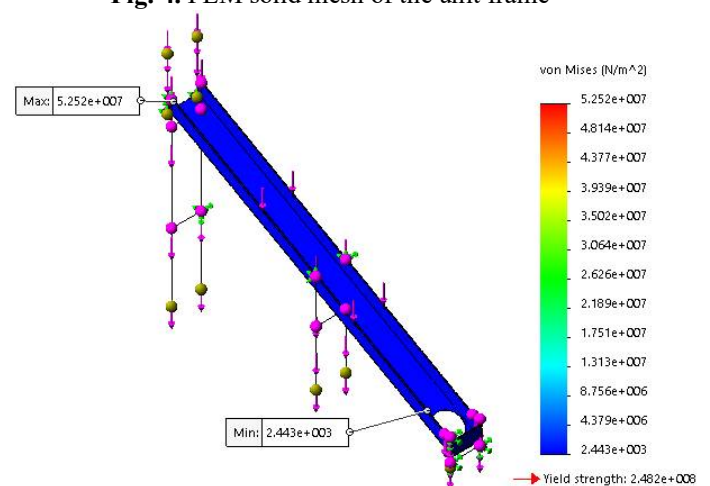**Fig. 4.** FEM solid mesh of the unit frame

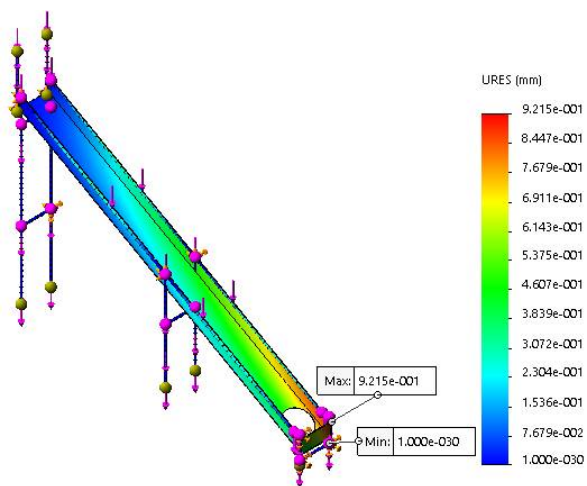**Fig. 5.** FEM of stress distribution within frame members

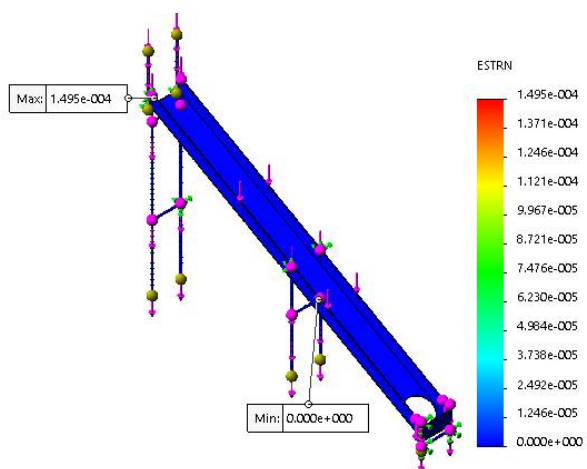**Fig. 6.** FEM of resultant displacement of frame members

Fig. 7. FEM of strain distribution within frame members
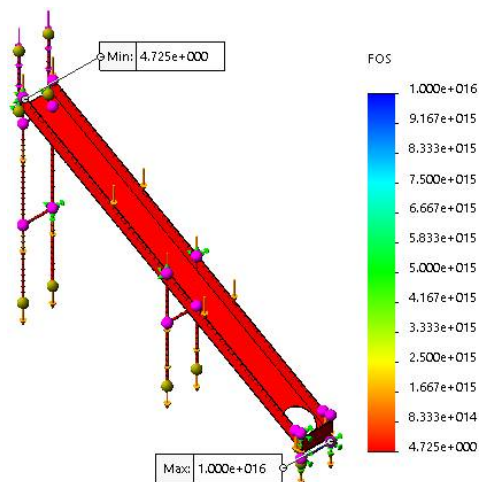

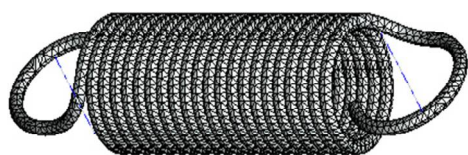**Fig. 8.** FEM of FOS distribution on frame members

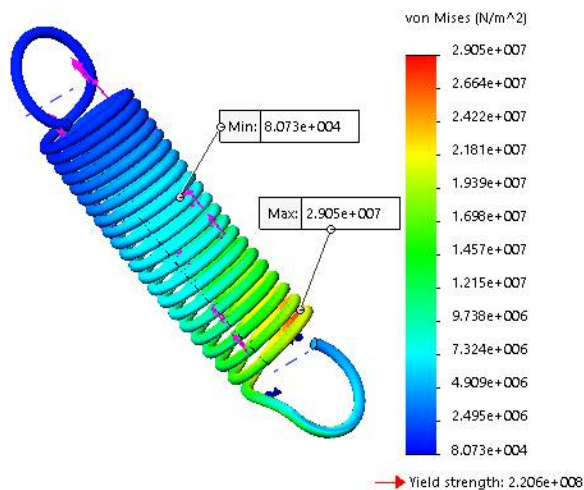**Fig. 9.** Solid mesh model of the control gate spring

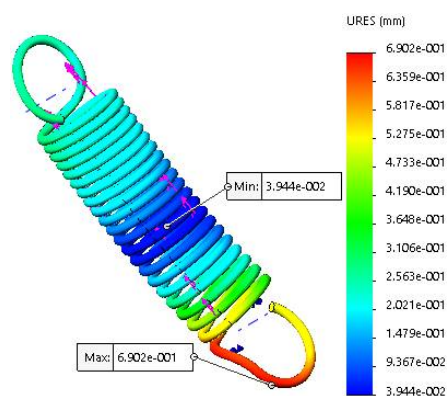**Fig. 10.** FEM of stress distribution within spring members

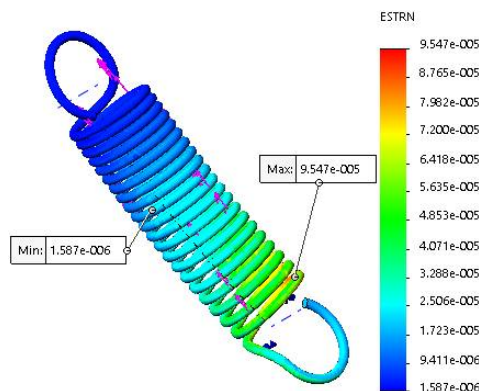**Fig. 11.** FEM of resultant displacement of spring members

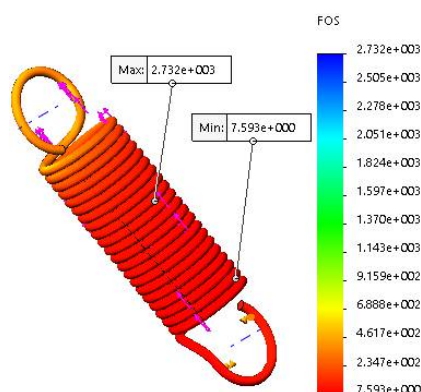**Fig. 12.** FEM of strain distribution within the control gate spring

Fig. 13. FEM of FOS distribution on spring members
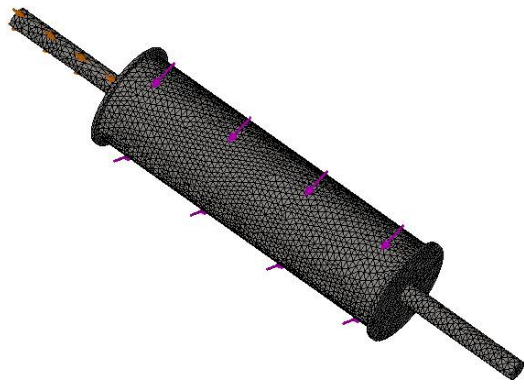
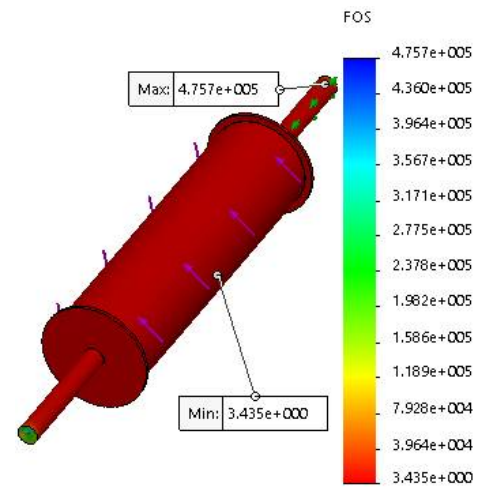**Fig. 14.** Solid mesh model of the belt conveyor roller

Fig. 15. FEM of stress distribution within the belt conveyor roller

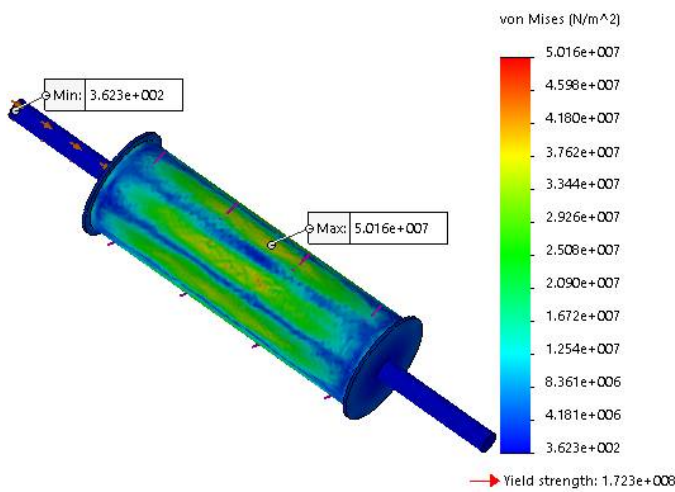**Fig. 16.** FEM of strain distribution within belt conveyor roller

Fig. 17. FEM of FOS distribution on belt conveyor roller

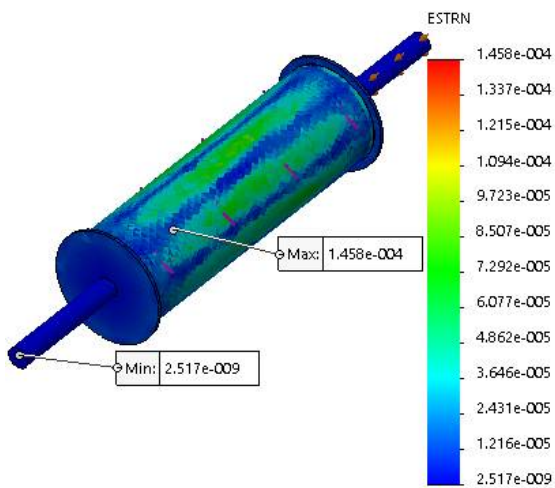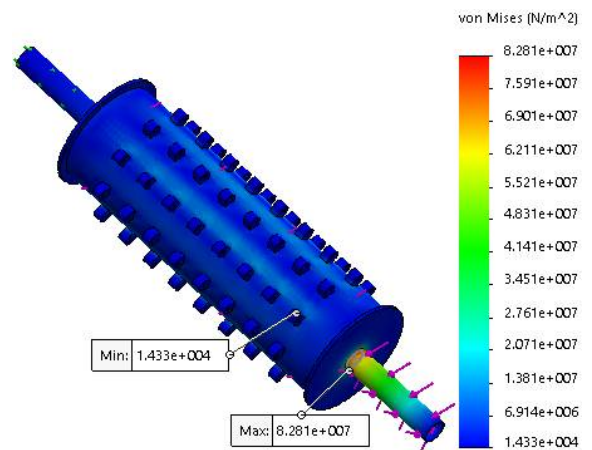**Fig. 18.** Solid mesh model of the roller with cam-like lobes

Fig. 19. FEM of stress distribution within the roller with cam-like lobes
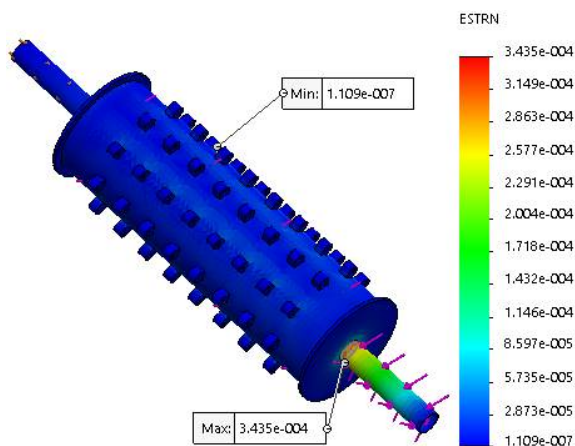
Fig. 20. FEM of strain distribution within the roller with cam-like lobes
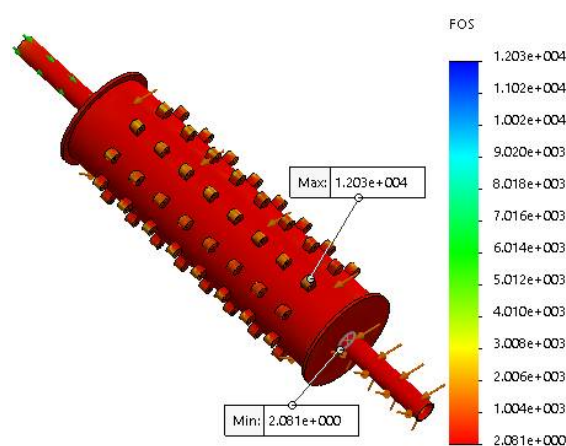

Fig. 21. FEM of FOS distribution on roller with cam-like lobes

3.2. Discussion of Simulation Results

The implication of the FEA results obtained is that the unit will be able to satisfactorily serve its intended purpose of washing, conveying and preheating plantain pulps under normal working conditions when fabricated. This is due to the fact that the maximum stress values obtained from the FEA of its component parts are far lower than the corresponding yield strength values of the materials selected for their fabrication. However, the locations that are subjected to these stress values on the unit are potential areas where failure may likely be initiated when it is fabticated after a long period of time in service. But failure may not likely occur since the FOS values obtained are high enough to prevent its component parts from experiencing structural failures under normal working conditions. It can therefore be inferred that the design of the washing, conveying and preheating unit is adequate and safe for fabrication. However, the material selected for each of its component parts can be revisited to further reduce the FOS values obtained. It to be noted that the FOS values obtained from the FEA were verified using Eq. (45) [9, 31].

$$FOS = YS/DS \qquad (45)$$

Where: YS is the yield strength of each selected material; and DS is the design or working stress.

4. CONCLUSIONS

In this research work, a unit for washing, conveying and preheating plantain pulps before particulation in a plant, that is being developed for continuous processing of unripe plantain into flour, has been designed in response to the issues raised by Olutomilola [32]. The unit is expected to help reduce human involvement in plantain processing and speedup its drying rate by preventing particulated or sliced plantain pulps from sticking together before and during drying. This will help plantain pulps to dry in one hour, at a maximum temperature of 70°C, in the drying section of the flour process plant as stated by Olutomilola [32].

The simulation conducted showed that the design of the unit is adequate and safe for fabrication having established its functionality and structural integrity as revealed by the results obtained. This is an indication that the unit will be able to satisfactorily serve its intended purpose when fabricated since the lowest FOS value obtained from the FEA of its component parts is 2.1, which is sufficient or high enough to prevent them from any form of failure in service. The component parts with high FOS values may appear to be over-designed, but the materials selected for their fabrication can be reviewed so as to further reduce the values as suggested by Farayibi [31]. As an ongoing research, the fabrication, testing or performance evaluation and automation of a prototype of the unit are already being considered. These will be followed by incorporation of the unit into the production line of the process plant.

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