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## THE DESIGN, CONSTRUCTION AND TESTING OF A VERTICAL SQUEEZE CASSAVA PULP DEWATERING MACHINE

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Accepted Date: 15 July 2009

### Abstract

*Gari is a popular staple food in Nigeria. It is got from the dry frying of dewatered cassava pulp. Cassava is a plant which originated from South America but is now grown in most of the African countries. The Nigerian variety grows to a height of 0.6 to 2m. The root is the most useful part of the plant. It is about 35cm in length with a diameter of about 10cm in the mid part of the root and tapers to the end of the root. The roots which are brown are peeled. The inside which are white are soaked in water and allowed to ferment for about 3days. They were then grated and water added to about 80% moisture content to form the pulp. The cassava pulp was put into jute sack and tied for two days. Instead of the local method of stacking the pulp in sacks and putting heavy objects on top to dewater for two days, a motorized vertical squeeze dewatering machine was designed and fabricated to do the job of dewatering 200kg pulp in 33.72 minutes. This machine was 7 times quicker than the IITA multi-purpose press and 40 times quicker than the local method of dewatering. This was made possible by dewatering screws working with a total torque of 1182KNmm, a maximum principal stress of  $100\text{N/mm}^2$ , and a maximum shear stress of  $60\text{N/mm}^2$ . The pulp platform were transported up and down by action of screw followers. They operated with induced sheer stress in the screw and followers of 17.76 and  $14.69\text{N/mm}^2$  which were adequately below the bearing pressure of  $18\text{N/mm}^2$ . The bevel gears were designed to carry static, dynamic and maximum tangential tooth load of 19.8, 1.07 and 9.05KN respectively. The rapid rate of dewatering will enhance the production of gari and thus assist development.*

**Keywords:** pulp, dewater toxic, squeeze, cyanide, grate.

### 1. Introduction

Cassava which is known biologically as “*manihot esculenta crantz*” is a crop which has many varieties. It is also known as manioc or yucca in some countries. The common variety seen in the southern part of Nigeria and at Ilorin when fully grown do have height of between 0.6 to 2m .The root, which is the most useful part of the crop is about 35cm in length .It has a diameter of about 10cm at the mid part of the root, but taper to about 4mm at the tail end of the root. Cassava was first grown in South America where it was cultivated for over 5000 years before it was introduced to Nigeria by the Portuguese in the 17<sup>th</sup> Century. The cultivation of the cassava crop is by the propagation of stem cuttings. The crop takes about 9,12,18 or 24 months to mature for harvesting depending on species. The roots are dug up from the soil, removed from the plant and washed before being processed, Pierres [1].

About 40% of African farmers grow cassava. The fresh root contains 50 to 75% of water, and less than 1% protein. The remaining 99% is mostly starch. The leaves and other parts contain a glyceride, linamarin from which hydrogen cyanide is released by enzymic action Davidsons [2]. Hydrogen cyanide is toxic to human health. The root is very useful for industrial and domestic applications. In the industrial use of cassava, detoxification of HCN takes place in the chemical processes when sliced raw roots are sometimes fermented for carbohydrates which are used for beer brewing and distillation. Cassava flour has found useful application in the baking industry. It is as industrial starch, used for the production of gasoline and acetone. The dried cassava chips are processed into animal feed, glues and manufacturing additive.

About 60% of cassava is used today for industries application and 40% for domestic consumption. The domestic product of cassava in West Africa consist of a fermented, semi-dextrimized meal called gari .The cassava root is peeled, grated into pulp or mesh, tied in sack and allowed to ferment for a few days, then dewatered. The dewatered mesh is partially gelatinized, dry fried sometimes with palm oil to percentage driness of about 12% and then milled. The gari is eaten as a paste called eba with gravy, or soaked in water and eaten with fish, meat, groundnuts or “kulikuli”. Gari is a very popular meal because of its relative cheapness when compared to other meals. This is because cassava will grow on any type of soil, and it does not require fertilizer.

As far back as 1962, researchers had observed the need for a mechanical system of dewatering of cassava mesh in the gari processing method. Akindele [3] observed the need for a neater mechanical way of dewatering gari pulp to change the unhygienic local method of water removed by compressing mesh sacks with heavy stones and metal objects. Egharevba [4] classified the screw-type press used for dewatering of cassava mesh into those where the mesh socks are place on a platform and the top press bar is screwed to compress the mesh sack, and those that have the mesh sack on a platform which is screwed upward to be compressed by a press bar that is stationary at the top. The IITA put forward a double screw press in 2007 which has a loading capacity of 200 to 350 kg applied to 5 bags per batch of cassava pulp. The rate of dewatering is such that a pulp with initial moisture content of 70-80% is reduced to 40-50% in 4hours. The motorized double screw compressive traverse cassava pulp dewatering machine has a capacity of 200kg. The pulp was reduced from 80% wetness to 25% in 2hours.

## **2. Description of the double screw vertical compression cassava pulp dewatering machine.**

The machine is shown on Fig. 1. The two power screws 55mm diameter are the main features of the double screw vertical compression cassava pulp dewatering machine. The power screws are threaded with acme threads right handed at top half and made to be left handed at the bottom of the screws. That way it was possible to make the top and bottom platforms to move vertically towards each other in compression. A power shaft which drives the two power screws is connected to the screws by means of an input bevel gears and an intermediate gear.



Fig. 1. Cassava Pulp Dewatering Machine

A 7.5hp single phase motor which runs at a speed of 1440rpm drives the power shaft. The intermediate gear effects speed reductions of 320, 77.6 and 22.2 rpm which are achieved with speed ratios of 1:4.125 and 1:35 respectively. The output powers to the two screws are 1.5hp each. The top and bottom cassava pulp platforms are attached to the power screws through internally threaded screw followers which cause the platforms to be transported either towards or away from each other. The cassava pulp is thus squeezed and dewatered when the platforms move towards each other in compression. During dewatering which takes 34minutes, the motor is switched off. After dewatering the motor is switched on, thus causing the cassava pulp platform to move away from each other. The cassava pulp moisture of 80% is reduced to 25% in 34 minutes. The machine takes 200 kg or four bag of pulp per batch. All fabricated components are made from mild steel, except the sleeve which is made from phosphor bronze material to reduce friction and minimize wear. The machine has a dimension of 1320x1400x525mm.

### 3. Theoretical Analysis

The machine was designed for the purpose of dewatering four bags of cassava pulp with the total weight of 220kN. The load taken by each compound screw was 110kN. The design of the compound thread was right handed for upper half and left handed for the bottom half of thread.

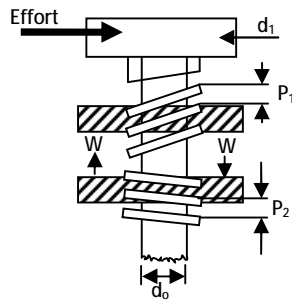


Fig. 2. Compound screw arrangement

Since load is transmitted under compression, screw was made from medium carbon steel with ultimate crushing stress of  $320\text{N/mm}^2$ , yield stress ( $\sigma_c$ ) of  $200\text{N/mm}^2$ , and shear ( $\tau$ ) of  $120\text{N/mm}^2$ , Raymond [5]. In compression, the screw was subjected to the equation given by,

$$\text{Load}(W) = \frac{A_s \times S_y}{F S} \quad (1)$$

Where  $W$  is load,  $A_s$  is area of screw,  $\tau_y$  is yield stress and  $FS$  is the factor of safety. Substituting in equation (1),

$$110 \times 10^3 = \frac{1}{4} \times p \times d_s^2 \times \frac{200}{2} \text{ at F. S. } = 2$$

$$\therefore d_s = 37.42\text{mm}$$

From Acme Tread the values that were chosen are shown below which were safe, when the allowable stresses of the screw material was considered.

Outer diameter of screw  $d_o = 55\text{mm}$   
 Core diameter of screw  $d_c = 45.5\text{mm}$   
 Pitch of thread  $p = 9\text{mm}$

Other dimensions of the Acme thread were

$$a = \frac{P}{2.7} = \frac{9}{2.7} = 3.333\text{mm}$$

$$c = a - 0.0052 = 3.328\text{mm}$$

$$h = 0.5p + 0.01 = 0.5(9) + 0.01 = 4.51\text{mm}$$

$$\text{Screw mean diameter, } d = \frac{d_o + d_c}{2} = \frac{55 + 45.5}{2} = 50.25\text{mm}$$

Where  $a$  is the width of top of thread,  $c$  is the width of the bottom of thread,  $h$  is height of thread,  $p$  is the pitch of thread, see Fig 3, and  $d$  is the screw mean diameter with  $d_o$  and  $d_c$  as stated above.

$$\tan a = \frac{P}{p d} \quad (2)$$

$$= \frac{9}{p \times 50.25}$$

$$\alpha = 3.26^\circ$$

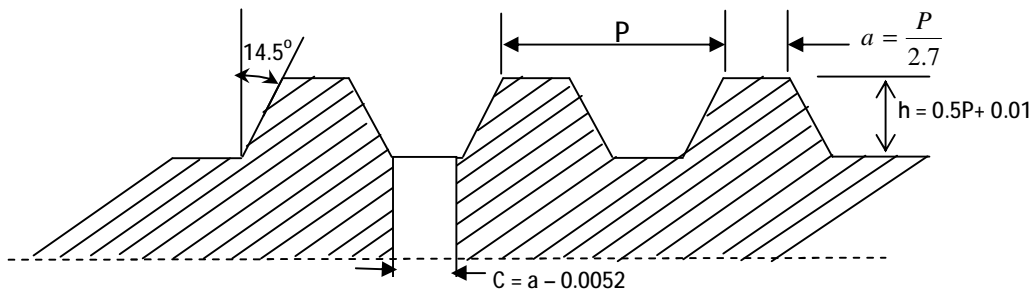


Fig. 3. Acme Thread specifications

Total torque to overcome friction at the thread surfaces of the screw is given by Lewis, W. (1984) as,

$$T = Wd \left[ \frac{\tan a + \tan f}{1 - \tan a \tan f} \right] \quad (3)$$

$$\text{Where } \tan f = \frac{m}{\cos b} \quad (4)$$

At  $\beta = 14.5^\circ$  and  $\mu = 0.15$  taken from Table for tread friction, Khurmi [6],

$$\tan f = \frac{0.15}{\cos 14.5} = 0.155$$

$$\therefore T = 110 \times 10^3 \times 50.25 \times \left[ \frac{0.057 + 0.155}{1 - (0.057 \times 0.155)} \right]$$

$$= 11.02 \times 10^5 \text{ Nmm}$$

Torque due to upper screw and the lower screw is

$$T_1 = T_2 = \frac{11.82 \times 10^5}{2} = 5.91 \times 10^5 \text{ Nmm}$$

Compressive stress  $\tau_c$  due to axial load is

$$s_c = \frac{W}{A_c} \quad (5)$$

$$= \frac{110 \times 10^3}{\frac{1}{4} \times \pi \times d_c^2} = \frac{110 \times 10^3}{\frac{1}{4} \times \pi \times 45.5^2} = 67.65 \text{ N/mm}^2$$

Shear stress on the screw due to torque is

$$t = \frac{16T_1}{\pi d_c^3} \quad (6)$$

$$= \frac{16 \times 5.91 \times 10^5}{\pi \times 45.5^3} = 31.95 \text{ N/mm}^2$$

Maximum principal stress in tension and compression is

$$d_c = \frac{1}{2} \left[ d_c + \sqrt{d_c^2 + 4t^2} \right] = \frac{1}{2} \left[ 67.65 + \sqrt{67.65^2 + 4 \times 31.95^2} \right] = 80.35 \text{ N/mm}^2$$

The value of stress used was

$$d_d = \frac{d_y}{FS} = \frac{200}{2} = 100 \text{ N/mm}^2$$

Therefore the designed screw will not fail as  $d_c$  is below the working stress

Maximum shear stress is

$$\hat{t} = \frac{1}{2} \left( \sqrt{t_c^2 + 4t^2} \right) \quad (7)$$

$$= \frac{1}{2} \left( \sqrt{67.65^2 + 4 \times 31.95^2} \right) = 46.52 \text{ N/mm}^2$$

The value of shear stress use in the design was

$$\hat{t}_d = \frac{t}{FS} = \frac{120}{2} = 60 \text{ N/mm}^2$$

Since these maximum stresses in compression, tension and in shear were within working limits for the material hence, the design for the screws was safe.

#### 4. Design of the screw follower

The screw follower has the function of moving up and down the thread of the screw depending on the motion of the screw. The screw follower was threaded on the inside and it was attached to the pulp platform on the outside. This way it was made to convey the cassava pulp upwards for compression or downward to release the dewatered pulp. Assuming the cassava pulp was uniformly distributed over the cross sectional area of the screw follower, the bearing pressure

$$SF_b = \frac{W}{\frac{D}{4} \times (d_o^2 - d_c^2) n} \quad (8)$$

Where n is number of threads in the screw follower, and the bearing pressure  $SF_b$  from Table of Limiting Values of bearing pressure is  $18\text{N/mm}^2$ . The continuous rubbing, friction and heat generation of the screw follower when it moved against the screw, made the choice of phosphor bronze material to be suitable for the construction of the screw follower. From equation (8)

$$n = \frac{110 \times 10^3}{p \times 18 \times \frac{(55^2 - 45.5^2)}{4}} = 8.15$$

Using F.S value of 1.5,  $n = 8.15 \times 1.5 \approx 13$  thread  
 Height of screw follower  $r = n \times p = 13 \times 9 = 117\text{mm}$   
 Design choice was for  $r = 120\text{mm}$   
 Shear stress induced in screw was,

$$t_s = \frac{W}{p \times n \times d_c \times a} \quad (9)$$

$$= \frac{110 \times 10^3}{p \times 13 \times 45.5 \times 3.333} = 17.76 \text{ N/mm}^2$$

Shear stress induced in the follower was

$$t_{SF} = \frac{110 \times 10^3}{p \times 13 \times 55 \times 3.333} = 14.69 \text{ N/mm}^2$$

Since  $\tau_s$  and  $\tau_{SF}$  were below the bearing pressure of  $18 \text{ N/mm}^2$  therefore the design for the screw follower was safe. The tearing strength of the screw follower was

$$d_t = \frac{d_y}{F.S} = \frac{W}{p/4(D_1^2 - D_0^2)} \quad (10)$$

$$\frac{100}{2} = \frac{110 \times 10^3}{p/4(D_1^2 - 55^2)}$$

$$\therefore D_1 = \sqrt{582.13} = 76.33 \text{ mm}$$

The design choice of 100mm is robust and safe.

### 5. Design of length of screw

The design for screw was at the critical load  $P_c$  when compressive axial load carried by the screw was just sufficient to initiate buckling. The Euler's equation of column end buckling is

$$P_s = \frac{p^2 EI}{(L_s)^2} \quad (11)$$

$$= \frac{p^2 EI}{(2L_s)^2}$$

$P_s = \text{design load} + \text{load variation} = 220 + 20 = 240 \text{ KN}$

$I = \text{second moment of area of the screw}$

$$\frac{pd_m^4}{64} = \frac{p \times 50.25^4}{64} = 3.13 \times 10^5 \text{ mm}^4$$

$$\therefore L_s = \sqrt{\frac{p^2 \times 200 \times 10^3 \times 3.13 \times 10^5}{4 \times 240 \times 10^3}} = 802.33 \text{ mm}$$

Therefore two lengths of screws chosen were each 800mm in length.

#### 5.1. The design of beam strength of the bevel gears

By the application of Lewis equation (12), obtained from Khnrm, [6],

$$W_T = (d_o C_v) b p m y \left( \frac{L - b}{L} \right) \quad (12)$$

where  $\delta_o$  is allowable static stress,



$C_v = \frac{6}{6 + V}$  is the velocity factor, V is peripheral velocity, b is face width, m is module  $\frac{D}{T}$ , D is gear diameter,  $T_n$  is number of teeth, y is tooth form factor, L is cone distance given in equation (13),  $D_o$  is pitch diameter of gear and  $D_p$  is pitch diameter of pinion.

$$L = \sqrt{\left(\frac{D_o}{2}\right)^2 + \left(\frac{D_p}{2}\right)^2} = \frac{m}{2} \sqrt{T_G^2 + T_P^2} \quad (13)$$

$$V = \frac{p D N}{60} \quad (14)$$

Substituting for  $y = 0.121$  and  $0.04$ ,  $\delta_o = 210\text{N/mm}^2$ ;  $V = 0.16\text{m/s}$  and  $12.5\text{m/s}$ ;

$$C_v = \frac{4.5}{4.5 + 0.16} = 0.966;$$

∴ From equation 12;

$$W_T = 210 \times 10^6 \times 0.966 \times 0.02 \times 3.142 \times 0.005 \times 0.121 \times \left(\frac{0.0728 - 0.02}{0.0728}\right) = 5592.86\text{N}$$

The dynamic load ( $W_D$ ) transmitted is,

$$W_D = \frac{T_a}{m T_n} + \frac{21V(b \times c + W_1)}{21V \sqrt{(b \times c + W_1)}} \quad (15)$$

$$= \frac{1378000}{5 \times 283} + \frac{21(0.16)(0.02 \times 474 + 973.8)}{21(0.16) + \sqrt{(0.02 \times 474 + 973.8)}} = 973.4 + 95.16 = 1068.96\text{N}$$

$$\hat{W}_T = 2 \times W_T - 2 \times W_D = 2 \times 5592.86 - 2 \times 1068.96 = 9048\text{N}$$

Table 1. Value of allowable static stress ( $\delta_o$ )

| Material                            | Allowable static stress ( $\delta_o$ ), (N/mm <sup>2</sup> ) |
|-------------------------------------|--|
| Forged Carbon Steel (Casehardening) | 126  |
| Forged Carbon Steel (Untreated)     | 140 – 210  |
| Forged Carbon Steel (Heat treated)  | 210 – 245  |

and  $C_v =$  Velocity factor

$$= \frac{3}{3 + v} \text{ (for ordinary cut gear at V up to 12.5m/s)}$$

$$= \frac{4.5}{4.5 + v} \text{ (for carefully cut gear at V up to 12.5m/s)}$$

$$= \frac{6}{6 + v} \text{ (for very accurate cut gear at } V \text{ up to 20 m/s)}$$

$$= \frac{0.75}{0.75 + \sqrt{v}} \text{ (for precision cut gear at } V \text{ up to 20m/s)}$$

### 5.2. Static Tooth Load ( $W_s$ )

This is also known as the endurance strength of tooth. From the Lewis Equation,

$$W_s = d_c \times b \times p \times m \times y \quad (16)$$

At Brinell Hardness Number (BHN) of 300, the flexural endurance limit ( $\delta_c$ ) was 520MN/m<sup>2</sup>.

$$\therefore W_s = 520 \times 10^6 \times 0.02 \times 3.142 \times 0.005 \times 0.121 = 19769.5N$$

For pulsating load system like this dewatering machine,

$$W_s \geq 1.35W_D \quad (17)$$

Therefore the design was safe.

### 5.3. Wear Tooth Load ( $W_w$ )

The governing equation that gear tooth can carry is given in the equation;

$$W_w = D_p \times b \times Q \times K \quad (18)$$

where  $W_w$  is the maximum/limiting load for wear,  $D_p$  is the pitch diameter of the pinion,  $b$  is the face width of the pinion,  $Q$  is ratio factor for external gear, and  $K$  is material combination factor N/mm<sup>2</sup>.

$$Q = \frac{2T_D}{T_o + T_p} \quad (19)$$

$$K = \frac{(d_{se})^2 \text{Sin} q}{1.4} \left( \frac{1}{E_p} + \frac{1}{E_G} \right) \quad (20)$$

where  $\delta_{se}$  is surface endurance limit and is 770N/mm<sup>2</sup> at BHN of 300,  $E_p$  is the Young's Modulus of pinion material which is 210GN/m<sup>2</sup>,  $E_G$  is the Young's Modulus of gear which is 84GN/m<sup>2</sup>, and  $\theta$  is the pressure angle of gear taken as 20°. From equation (19);

$$Q = \frac{2 \times 28}{28 + 8} = 1.56$$

From equation (20);

$$K = \frac{770^2 \sin 20}{1.4} \left( \frac{1}{210 \times 10^9} + \frac{1}{84 \times 10^9} \right) = 2.41 \text{ N/mm}^2$$

Using equation (18) gives,

$$W_w = 40 \times 20 \times 1.56 \times 2.61 = 3257.3 \text{ N}$$

Since  $3257.3 \text{ N} > 1068.96 \text{ N}$

or  $W_w \geq W_D \Rightarrow$  the design is safe.

#### 5.4.Reduction gear ratio

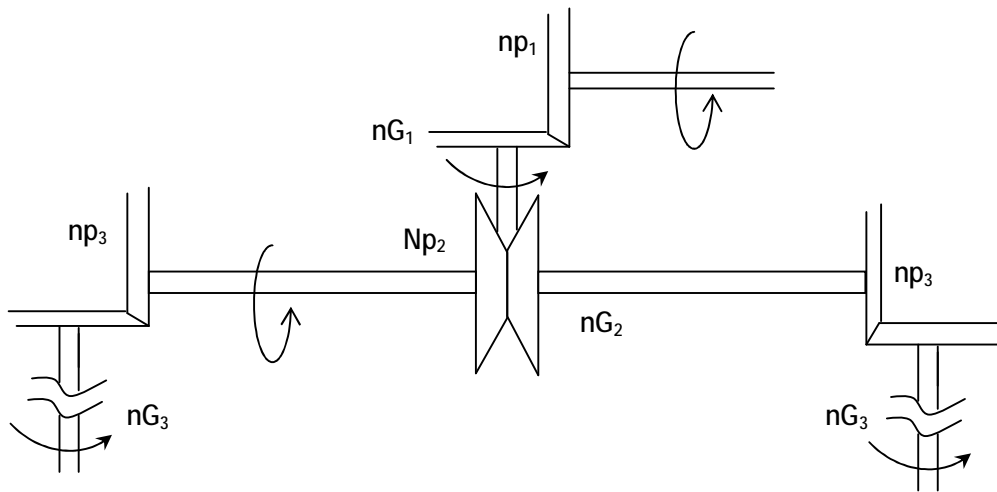


Fig. 4. Reduction gear ratio

There were 3 stages of gear reduction.

Stage 1

$$T_{p1} = 10 (\text{number of teeth}) \quad T_{G1} = 45 (\text{number of teeth}) \quad n_{p1} = 1440 \text{ rpm}$$

$$n_{G1} = \frac{n_{p1} \times T_{p1}}{T_{G1}} = \frac{10 \times 1440}{45} = 320 \text{ rpm}$$

Stage 2

$$T_{p2} = 8 \quad T_{G2} = 33 \quad n_{p2} = n_{p1} = 320 \text{ rpm}$$
$$n_{G2} = \frac{n_{p2} \times T_{p2}}{T_{G2}} = \frac{8 \times 320}{33} = 77.6 \text{ rpm}$$

Stage 3

$$T_{p3} = 8 \quad T_{G3} = 23 \quad n_{p3} = n_{G2} = 77.6 \text{ rpm}$$
$$n_{G3} = \frac{n_{p3} \times T_{p3}}{T_{G3}} = \frac{77.6 \times 8}{28} = 22.2 \text{ rpm}$$

### 5.5. Peripheral velocity and module for gears

These were determined to be

|         |                 |
|---------|-----------------|
| Stage 1 | 2.93m/s and 4mm |
| Stage 2 | 0.67m/s and 5mm |
| Stage 3 | 0.16m/s and 5mm |

### 5.6. Efficiency of gears

Power output from gear is

$$P_o = \frac{W_T \times V}{C_s} \quad (21)$$

where V is pitch circle velocity, C<sub>s</sub> is service Factor; P<sub>o</sub> and W<sub>T</sub> are as given above. Substitute in (21)

$$P_o = \frac{2 \times 5592.86 \times 0.16}{0.8} = 2237.14 \text{ W}$$

Input power is P<sub>1</sub> given in

$$P_1 = 7.5 \text{ hp} = 7.5 \times 746 \text{ W} = 5595 \text{ W}$$
$$\therefore \text{Efficiency} = \frac{\text{Output power}}{\text{Input power}} = \frac{2237.14}{5595} \times 100 = 40\%$$

Gear efficiency was 40%

## 6. Construction material selection and procedure for fabrication and assembly

Four pieces of 101.6 x 101.6mm angle bars, 900mm in length were used for the construction of the main upright frame. They were placed in pairs and welded to give 101.6 x 203.2 x 101.6mm U-channel frame upright, which formed each side of the machine. Similar angle bar was used to fabricate the base frame but to a length of 1400 mm which was welded to the base of the upright. Two holes of 35mm diameter were drilled at the base frame centerline and at distance 800mm apart, to accommodate the tail of the screw thread and serve as guide for the roller ball bearings. Two 55mm nuts were used to hold the screw at the base frame.

The screws, power shaft, bevel gears, spur gear and cassava pulp platform came under direct load, torque, compressive stress, tensile and shear stress which did not exceed the strength of medium carbon steel (mcs). The mcs chosen was 0.6% carbon and 70% perlite. Two mcs shafts of 55mm diameter and 1150mm length were cut. They were plane turned, recessed and step turned to diameters of 49mm for bearing ends and nut allowances. Acme threads were cut with an outer diameter of 55mm and inner diameter of 47mm at a pitch of 9mm. The screw, completed with left and right threads starting from end points of the shafts and meeting at the middle of the shaft, formed a compound screw over 400mm length each way, which made up the dewatering screw of length 800mm.

The bevel gears of PCD 175, 165 and 140mm for the 1<sup>st</sup>, 2<sup>nd</sup>, and 3<sup>rd</sup> stages and having 45, 33, and 28 teeth respectively were used. The bore were annealed and expanded to diameter of 35, 30 and 40mm for the 1<sup>st</sup>, 2<sup>nd</sup>, and 3<sup>rd</sup> stages respectively. Keys and keyways were cut which made the fitting of the pinions to the shafts to be rigid. The shafts of the pinions were machined on the lathe to the dimensions of 45, 40 and 35mm outer diameter for the 1<sup>st</sup>, 2<sup>nd</sup>, and 3<sup>rd</sup> stages respectively. The shafts were cut to the lengths of 250, 200, and 300 respectively to fit the input gear and the intermediate gear arrangements see Figure 2. The intermediate gear was an arrangement of two 330 by 170 and 190 by 100mm bevel gears which were fitted together to form the unit. The two main drive shafts were connected to the intermediate gear through a set of aligning roller ball bearings and flanges. The dewatering screws were connected to the drive shaft through a set of bevel gears and aligning ball bearings. The dewatering screws were held in a bottom plate by a set of aligning roll ball bearings.

Two bars of phosphor bronze material which has high resistance to wear and tear at high friction, were taken. The bars of diameter 100mm and length 120mm were turned on the lathe to give a plane surface which was bored. They were internally threaded to 13 Acme threads as required to the length of 120mm. The inner and outer diameters were 55 and 80mm respectively, and this was the screw follower which transported the cassava pulp platform up and down the screws. Threaded holes were made on the screw followers through which bolts were used to secure the cassava pulp platform.

Though the output power of the machine by design was about 3 hp, it was found necessary to use a 7.5 hp single phase motor to drive the dewatering machine. This followed trial tests with 3.5, 5 and 7.5 hp motors which were available, that showed the 7.5 hp motor to have dewatered the cassava pulp most efficiently and at shortest time.

The same 101.6 x 203.2 x 101.6mm U-channel of length 500mm was used to make the cassava pulp platform. The pulp platforms were bolted to the screw follower with the flat surface of the two channels facing each other. The vertical double squeeze cassava pulp dewatering machine is shown in Fig. 1.

## 7. Machine test and performance evaluation

The samples of cassava pulp that were tested were taken from those in popular usage in Nigeria. There are two types, the “fine gari” and the “Kpokpo gari” or “coarse gari” varieties. The fine gari cassava pulp, 85% finess, and the coarse or Kpokpo gari cassava pulp, 65% finess were tested on the dewatering machine. Fine cassava pulp was put into a jute sack which was tied at the neck and weighed. A weight of 13.40kg was recorded. The pulp was placed on the machine platform. The motor was activated and the top and bottom platforms moved towards each other. The two platforms met and dewatering took place for 2.35 minutes. The residual mass was weighed, and water loss noted. The partially dewatered pulp was placed back on the platform and the machine was reactivated for 1.37minutes. Again the residual mass was weighed after the toxic water was partially driven off. The dewatering experiment was then carried out at interval of 5 minutes and records were taken as entered on Table 2. The percentages of water was calculated. The experiment was repeated for the coarse cassava pulp of 65% finess. The record was entered on Table 3. The average of 5 sample tests are shown for each of the fine and coarse pulps.

Table 2. Result of dewatering fine cassava pulp at 80% moisture

| Mass of Cassava pulp | Dewatering period (minutes) | Final dewatered pulp (kg) | Toxic water loss (kg) | Percentage Toxic water loss (%) |
|----------------------|-----------------------------|---------------------------|-----------------------|---------------------------------|
| 13.40                | 2.35                        | 11.50                     | 1.90                  | 14.18                           |
| 11.50                | 1.37                        | 10.00                     | 1.50                  | 13.04                           |
| 10.00                | 5.00                        | 8.50                      | 1.50                  | 15.00                           |
| 8.50                 | 5.00                        | 7.40                      | 1.1                   | 12.94                           |
| 7.40                 | 5.00                        | 6.40                      | 1.0                   | 13.51                           |
| 6.49                 | 5.00                        | 5.40                      | 1.0                   | 15.63                           |
| 5.40                 | 5.00                        | 4.60                      | 0.8                   | 14.81                           |
| 4.60                 | 5.00                        | 4.0                       | 0.6                   | 13.04                           |
| TOTAL                |                             |                           |                       | 70.15%                          |

$$\text{Final water loss (\%)} = \frac{\text{initial mass} - \text{final mass}}{\text{initial mass}} \times 100 = \frac{13.40 - 4.0}{13.40} \times 100 = 70.15\%$$

## 8. Results and discussion

By design, values of the compressive stress and shear stress for each of the dewatering screws were 67.65 and 31.95N/mm<sup>2</sup> respectively. These gave a maximum principal stress in tension and compression for screw of 80.35N/mm<sup>2</sup>. To ensure that screw did not fail, 100N/mm<sup>2</sup> was used in the design as the maximum principal stress. The maximum shear stress by calculation was 46.52N/mm<sup>2</sup>. The choice of 60N/mm<sup>2</sup> was made for the maximum shear stress to ensure the safe operation of the screw in shear. The total torque of each dewatering screw as designed was 1182kNmm. The screw follower, which transported the cassava pulp for dewatering, was designed to have 13 threads.

Table 3. Result of dewatering coarse cassava pulp at 80% moisture

| Mass of Cassava pulp | Dewatering period (minutes) | Final dewatered pulp (kg) | Toxic water loss (kg) | Percentage Toxic water loss (%) |
|----------------------|-----------------------------|---------------------------|-----------------------|---------------------------------|
| 13.40                | 2.35                        | 11.70                     | 1.70                  | 12.69                           |
| 11.70                | 1.37                        | 10.20                     | 1.50                  | 12.83                           |
| 10.20                | 5.00                        | 8.80                      | 1.40                  | 13.73                           |
| 8.80                 | 5.00                        | 7.60                      | 1.20                  | 13.64                           |
| 7.60                 | 5.00                        | 6.70                      | 0.90                  | 11.84                           |
| 6.70                 | 5.00                        | 5.70                      | 1.00                  | 14.93                           |
| 5.70                 | 5.00                        | 5.00                      | 0.70                  | 12.28                           |
| 5.00                 | 5.00                        | 4.50                      | 0.50                  | 10.00                           |
| <b>TOTAL</b>         |                             |                           |                       | <b>66.40 %</b>                  |

The shear stress induced in screw and that induced in the follower were 17.76 and 14.69N/mm<sup>2</sup> respectively. Therefore the bearing pressure was adequate at 18N/mm<sup>2</sup>. The tearing strength of 50N/mm<sup>2</sup> was used to design for a screw follower external diameter of 100mm which was robust and safe. The use of the design load and variation of 240kN was used to derive a screw length of 800mm as shown in Figure 2.

The bevel gears that were used were selected to carry a static load, dynamic load and maximum tangential tooth load of 19.8, 1.07 and 9.05kN respectively. The static load was greater than 1.35 times the dynamic load and the wear load was also greater than the dynamic load for design to be safe. The efficiency of the gears was designed to be low at 40%. This was because an input single phase motor 7.5 hp was used at more than twice the output power, 3 hp, to ensure rapid dewatering and durability. This machine dewatered 200kg cassava pulp of 80% moisture to 25% moisture in about 34 minutes. The machine frame was designed to the load of cassava pulp, pulp platform, dewatering screws, power shafts, bevel gears, intermediate gear, input shaft and single phase motor. When the sack full sample of fine cassava pulp 85% finess was squeeze by the dewatering machine, toxic water, milky brown in colour flowed out. The flow of toxic water slowed down after a while to trickling rate, then dripped and the drops were slower in forming.

The sack containing the sample was removed from the machine and weighed. The dewatered pulp weighed 11.50kg which gave a percentage water loss of 14.18%, see Table 1.

When the partially dewatered pulp was again dewatered, the drips were slower in forming, and after 1.37 minute they appeared to have ceased. It was found to weigh 10kg, which placed the percentage water loss at 13.04%. The subsequent changes in the dewatering tests were done at 5 minutes interval. The final sample weight at the end of 33.72minutes was 4kg, which gave a water loss of 70.15%. The coarse pulp sample, 65% finess, was carried out through the same test as above, see results on Table 2. The toxic water dewatering rate was found to be slower than that of the fine gari by about 3.75%. This was because the coarse pulp stores more toxic water than the fine gari because of its grain size, and took longer time to give up its water.

## 9. Conclusion

The vertical double squeeze cassava pulp dewatering machine was designed for a cassava pulp capacity of 200kg or 4 bags. The dewatering screws operated at a maximum principal stress of  $100\text{N/mm}^2$ , maximum shear stress of  $60\text{N/mm}^2$  and at a total torque of 1182kNmm. The screws which were made with acme thread, had external and internal diameters of 55 and 45.5mm respectively and were 800mm in length..

The screw followers which transported the pulp platform up and down the screw were designed with induced shear stress in the screw and in the follower of 17.76 and  $14.69\text{N/mm}^2$  respectively. Bevel gears were used which carry a static, dynamic and maximum tangential tooth load of 19.8, 1.07 and 9.05kN respectively. The efficiency of the gears was compromised for speed in dewatering. It was 40% in operation because a relatively large single phase 7.5hp motor was used. When the average of five samples of fine cassava pulp of 80% moisture content were dewatered to the moisture level of 29.85% it took 33.72minutes, see Table 1. The average of five samples of coarse pulp, 65% moisture content, were dewatered in the same time of 33.72 minutes 80% moisture content to 33.6%. The coarse cassava pulp dewatering rate was found to be slower than that of the fine gari by 3.75%.

The closest in performance to the vertical squeeze cassava pulp dewatering machine was the IITA Multi-Purpose batch type dewatering machine. The new motorized vertical squeeze dewatering machine was 7 times faster in dewatering than the manual IITA machine. It also removed 10 to 20% more toxic water from cassava pulp than the IITA machine. The rate of gari production was 7 times more with new machine than with the IITA machine, and was about 40 times faster than the local method. The new vertical squeeze dewatering machine will cause more Nigerians to be fed, as more will be produced. Through this, development will be enhanced.

## Acknowledgment

The authors acknowledge Mr. Raji and Mr. S. Arinde for their help during the fabrication of the machine. Also acknowledged are O. Medaiyese, M. Badrudeen, T. Ojo and E. Oladimeji for their assistance in the test exercise.

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