Journal of Applied Sciences Research, 5(10): 1285-1297, 2009

The Design, Constrution and Testing of a Vertical Squeeze Cassava Pulp Dewatering Machine

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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. Instead of the local method of stacking the pulp in sacks and puting 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 100N/mm^2, and a maximum shear stress of 60N/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 14.69N/mm^2 which were adequately below the bearing pressure of 18N/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.

Key words: pulp, dewater toxic, squeeze, cyanide, grate.

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 17th 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 [5].

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 [6]. 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.

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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 [1] 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. [8] He proposed the idea of putting the sacks of cassava mesh in a box, and using a pulp compressor to dewater all the mesh in the box. As the box restricted the outward flow of the water from the pulp, engineers introduced the open screw press which proved satisfactory when compared with the other systems, except for the cost of manufacture and maintenance, Ajibola [9]. In 1990 Eghareuvba classified the screw-type press used for dewatering of cassava mesh into those where the mesh sacks 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.

The machine is shown on Figure 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 Figure 2. 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 see Figure 3. 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 is 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 of moisture of 50% 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.

Theoretical Analysis: The machine was designed for the purpose of dewatering four bags of cassava pulp with in extra 10% load of Pulep for Safety which makes a Total weight of 220kg. The load taken by each compound screw was 110kN. The design of the compound thread was right handed for upper half and left handrd for the bottom half of thread.

Since load is transmitted under compression, screw was made from medium carbon steel with ultimate crushing stress of 320N/mm^2, yield stress (σ_c) of 200N/mm, and shear (τ) of 120N/mm^2, Raymond, F. ()

In compression, the screw was subjected to the equation given by,

$$Load\,(W) = \frac{A_s \times \sigma_y}{FS} \tag{1}$$

Where W is load, A_s is area of screw, τ_y is yield stress and FS is the factor of safety.

Substituting in equation (1),

$$110\times10^3 = \frac{1}{4} \times \pi\, d_s^{\,2} \times \frac{200}{2} \text{ at F. S. =2}$$

\ d_s = 37.42mm

From Acme Tread Table (Appendix 1) 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= 55mm

Core diameter of screw d_c =45.5mm

Pitch of thread p =9mm

Other dimensions of the Acme thread were

$$a = \frac{p}{2.7} = \frac{9}{2.7} = 3.333mm$$

c = a – 0.0052 = 3.328mm

h =0.5p +0.01 = 0.5 (9) +0.01 = 4.51mm

Screw mean diameter,

$$d = \frac{d_o + d_c}{2} = \frac{55 + 45.5}{2} = 50.25mm$$

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 5, and d is the screw mean diameter with d_o and d_c as stated above.

$$Tan\,\alpha = \frac{P}{\pi d} \qquad (2)$$

$$= \frac{9}{\pi \times 50.25} \qquad a = 3.26^0$$

Total torque to overcome friction at the thread surfaces of the screw is given by Lewis, W. [4] as,

$$T = Wd\left[\frac{\tan\alpha + \tan\phi}{1 - \tan\alpha\tan\phi}\right] \qquad (3)$$

Where $\tan\phi = \dfrac{\mu}{\cos\beta}$ (4)

At b = 14.5^0 and m =0.15 taken from Table for tread friction, Khurmi, R. (2005),

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

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

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

Torque due to upper screw and the lower screw is

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

Compressive stress t_c due to axial load is

$$\sigma_c = \frac{W}{A_c}$$

$$= \frac{110 X 10^3}{\frac{1}{4} X \pi X d_c^2} \frac{110 \text{ X } 10^3}{\frac{1}{4} \text{ X } \pi X 45.5^2} 65 \text{N/mm}^2$$

Shear stress on the screw due to torque is

$$\tau = \frac{16 T_1}{\pi d_c^3} \qquad (6)$$

$$= \frac{16 \text{ X } 5.91 \text{ X} 10^5}{\pi \text{ X } 45.5^3} 1.95 \text{N/mm}^2$$

Maximum principal stress in tension and compression is

$$\delta_c = \frac{1}{2}\left[\delta_c + \sqrt{\delta_c^2 + 4\tau^2}\right]$$

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

The value of stress used was

$$\delta_d = \frac{\delta_y}{FS} = \frac{200}{2} = 100 N/mm^2$$

Therefore the designed screw will not fail as δ_c is below the working stress Maximum shear stress is

$$\hat{\tau} = \frac{1}{2}\left(\sqrt{\tau_c^2 + 4\tau^2}\right) \qquad (7)$$

$$= \frac{1}{2}\left(\sqrt{67.65^2 + 4 \text{ X } 31.95^2}\right)$$

= 46.52 N/mm

The value of shear stress used in the design was

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

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.

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 is,

$$SF_b = \frac{W}{\frac{\pi}{4} X (d_0^2 - d_c^2) n} \qquad (8)$$

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 18N/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[5]

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

Using F.S value of 1.5,
n = 8.15 X 1.5 ≈ 13 thread
Height of screw follower r = n X p = 13 X 9 = 117mm
Design choice was for r = 120mm
Shear stress induced in screw was,

$$\tau_s = \frac{W}{\pi \times n \times d_c a} \qquad (9)$$

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

Shear stress induced in the follower was

$$\tau_{SF} = \frac{110\times10^3}{\pi\times13\times55\times3.333} = 14.69\,N/mm^2$$

Since t_{s} and t_{SF} were below the bearing pressure of 18 N/mm^2 therefore the design for the screw follower was safe.

The tearing strength of the screw follower was

$$\delta_t = \frac{\delta_y}{F.S} = \frac{W}{\pi/4\left(D_1^2 - D_0^2\right)} \quad (10)$$

$$\frac{100}{2} = \frac{110\times10^3}{\pi/4\left(D_1^2 - 55^2\right)}$$

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

The design choice of 100mm is robust and safe.

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

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

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

P_s = design load + load variation = 220 +20 = 240KN
I = second moment of area of the screw

$$= \frac{\pi d_m^4}{64} = \frac{\pi\times50.25^4}{64} = 3.13\times10^5 mm^4$$

$$\therefore L_s = \sqrt{\frac{\pi^2\times200\times10^3\times3.13\times10^5}{4\times240\times10^3}} = 802.33mm$$

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

The Design of Beam Strength of the Bevel Gears: From Fig. 3 and with the application of Lewis equation (12), obtained from Khnrmi, R. [3],

$$W_T = (\delta_o C_v)\, b\pi m y\left(\frac{L-b}{L}\right) \quad (12)$$

where δ_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.

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

$$V = \frac{\pi DN}{60} \quad (14)$$

Substituting for y = 0.121 and 0.04, δ_o = 210N/mm2; V = 0.16m/s and 12.5m/s; $C_v = \frac{4.5}{4.5+0.16} = 0.966$;
\ From equation 12;

$$W_T = 210\times10^6 \times 0.966\times0.02\times3.142\times0.005\times0.121\times\left(\frac{0.0728-0.02}{0.0728}\right) = 5592.86N$$

The dynamic load (W_D) transmitted is,

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

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

$\hat{W}_T = 2 \times W_T - 2 \times W_D = 2 \times 5592.86 - 2 \times 1068.96 = 9048N$ and C_v = Velocity factor

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

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

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

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

Static Tooth Load (W_s): This is also known as the endurance strength of tooth. From the Lewis Equation,

$$W_s = \delta_c \; x \; b \; x \; \pi \; x \; m \; x \; y \tag{16}$$

At Brinell Hardness Number (BHN) of 300, the flexural endurance limit (δ_c) was 520MN/m^2.

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

For pulsating load system like this dewatering machine,

$$W_s \geq 1.35 W_D \tag{17}$$

Therefore the design was safe.

Wear Tooth Load (W_w): The governing equation that gear tooth can carry is given in the equation;

$$W_w = D_p \; x \; b \; x \; Q \; x \; K \tag{18}$$

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^2.

$$Q = \frac{2T_D}{T_o + T_p} \tag{19}$$

$$K = \frac{(\delta_{se})^2 \, Sin\theta}{1.4}\left(\frac{1}{E_p} + \frac{1}{E_G}\right) \tag{20}$$

where δ_{se} is surface endurance limit and is 770N/mm^2 at BHN of 300, E_p is the Young's Modulus of pinion material which is 210GN/m^2, E_G is the Young's Modulus of gear which is 84GN/m^2, and q is the pressure angle of gear taken as 20^o. From equation (19);

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

From equation (20);

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

Using equation (18) gives,

$$W_w = 40 \; x \; 20 \; x \; 1.56 \; x \; 2.61 = 3257.3N$$

Since 3257.3N > 1068.96'
or W_w ³ W_D $\Rightarrow$ the design is safe.

Reduction Gear Ratio: There were 3 stages of gear reduction.

Stage 1

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

$$n_{G1} = \frac{n_{p1}\ x\ T_{p1}}{T_{G1}} = \frac{10\ x\ 1440}{45} = 320\,rpm$$

Stage 2

$T_{p2} = 8 \qquad T_{G2} = 33 \qquad n_{p2} = n_{p1} = 320\,rpm$

$$n_{G2} = \frac{n_{p2}\ x\ T_{p2}}{T_{G2}} = \frac{8\ x\ 320}{33} = 77.6\,rpm$$

Stage 3

$T_{p3} = 8 \qquad T_{G3} = 23 \qquad n_{p3} = n_{G2} = 77.6\,rpm$

$$n_{G3} = \frac{n_{p3}\ x\ T_{p3}}{T_{G3}} = \frac{77.6\ x\ 8}{28} = 22.2\,rpm$$

Peripheral Velocity and Module for Gears: These were determined to be

Stage 12.93m/s and 4mm
Stage 20.67m/s and 5mm
Stage 30.16m/s and 5mm

Efficiency of gears: Power output from gear is

$$P_o = \frac{W_T\ x\,V}{C_s} \tag{21}$$

where V is pitch circle velocity, C_s is service Factor; P_o and W_T are as given above.
Substitute in (21)

$$P_o = \frac{2\ x\ 5592.86\ x\ 0.16}{0.8} = 2237.14\,W$$

Input power is P_1 given in

$$P_1 = 7.5hp = 7.5\ x\,746W = 5595W$$

$$\therefore Efficiency = \frac{Output\ power}{Input\ power} = \frac{2237.14}{5595}\,x\,100 = 40\%$$

Gear efficiency was 40%

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 frame as shown in Figure 2............ 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 1st, 2nd, and 3rd 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 1st, 2nd, and 3rd 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 1st, 2nd, and 3rd 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 platform 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 Figure 1.

Fig. 1: Cassava Pulp Dewatering Machine

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 1. The percentages of water was calculated. The experiment was repeated for the coarse cassava pulp of 65% finess. The record was entered on Table 2. The average of 5 sample tests are shown for each of the fine and coarse pulps.

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^2 respectively. These gave a maximum principal stress in tension and compression for screw of 80.35N/mm^2. To ensure that screw did not fail, 100N/mm^2 was used in the design as the maximum principal stress. The maximum shear stress by calculation was 46.52N/mm^2. The choice of 60N/mm^2 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. The shear stress induced in screw and that induced in the follower were 17.76 and 14.69N/mm^2 respectively. Therefore the bearing pressure was adequate at 18N/mm^2. The tearing strength of 50N/mm^2 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 was 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.

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 100N/mm^2, maximum shear stress of 60N/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 14.69N/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.

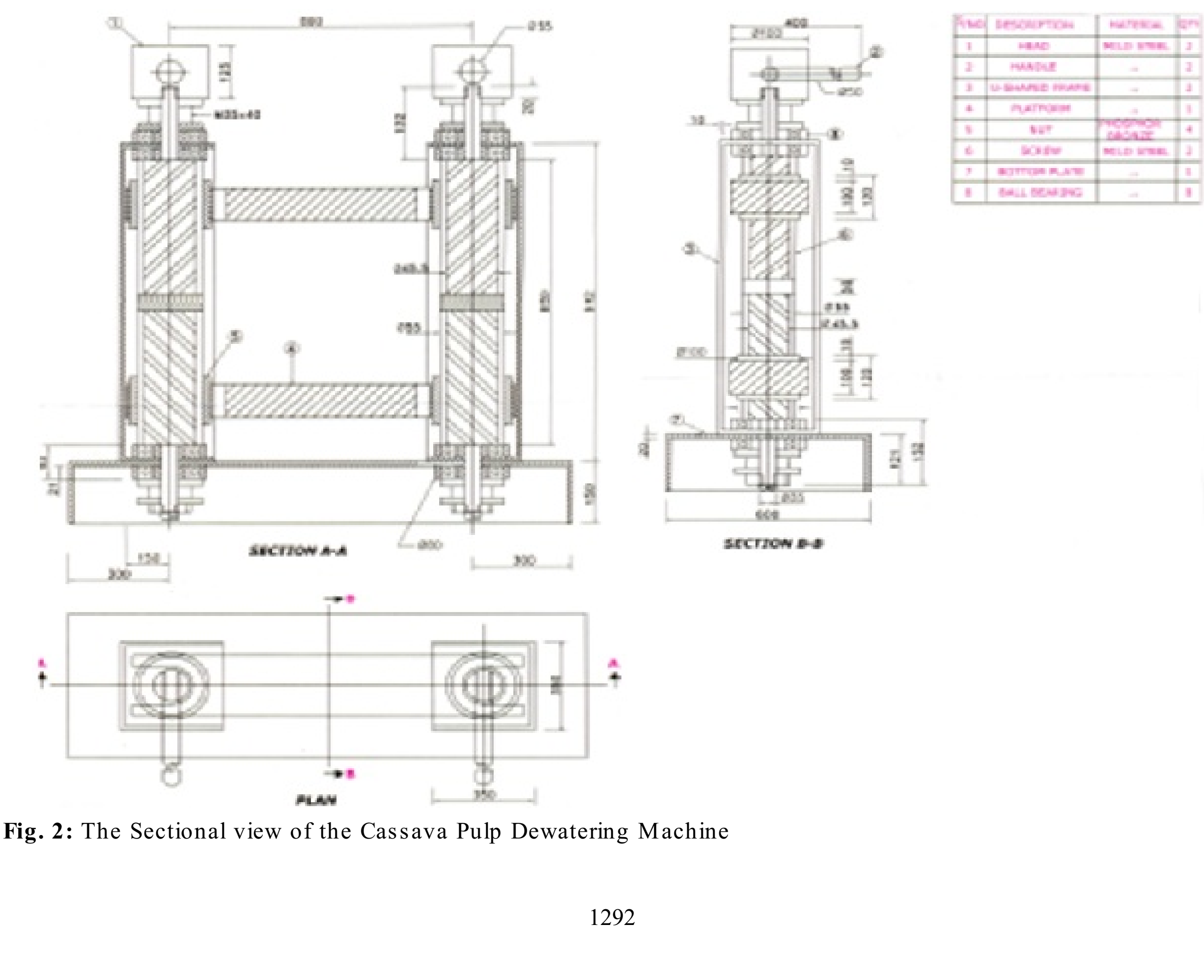

Fig. 2: The Sectional view of the Cassava Pulp Dewatering Machine

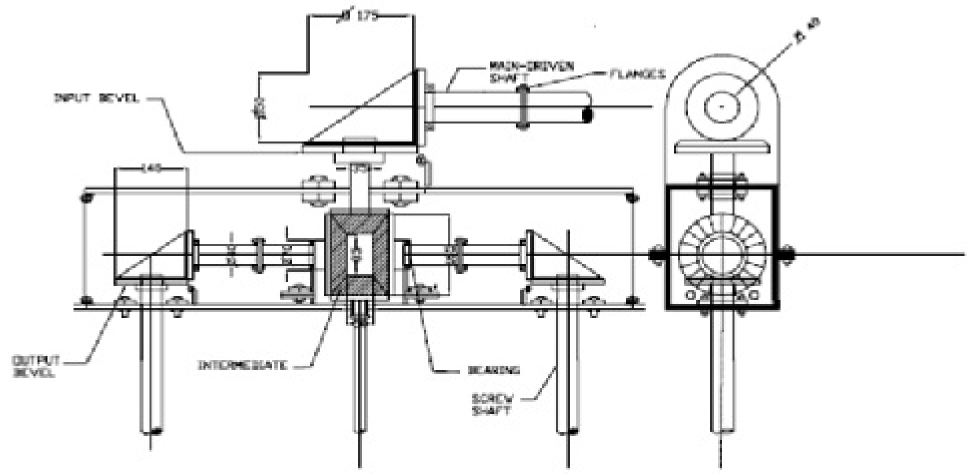

Fig. 3: The gear arrangement

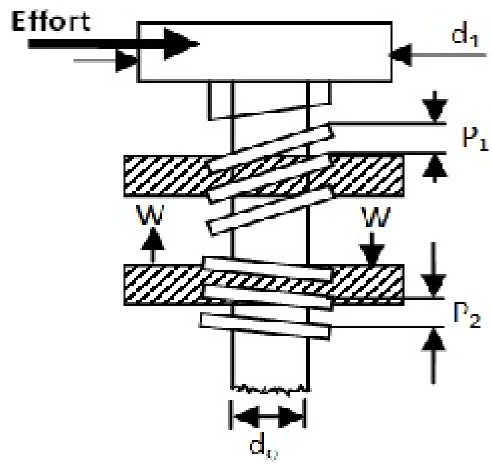

Fig. 4: Compound screw arrangement

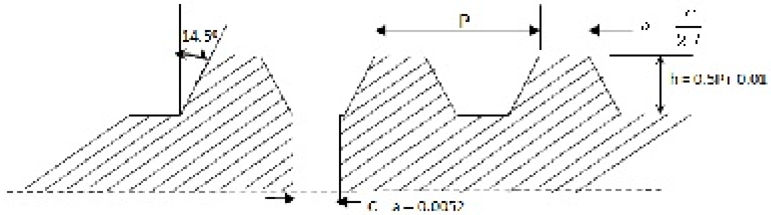

Fig. 5: Acme Thread specifications

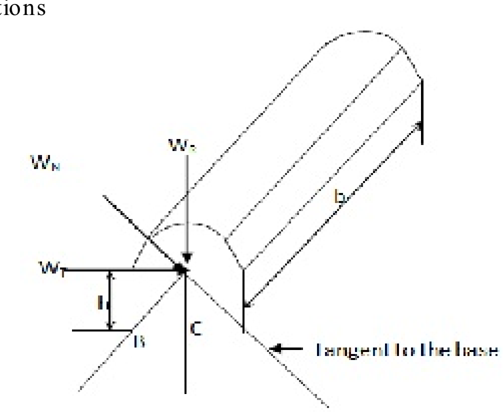

Fig. 6: Showing line of action of forces in a gear tooth

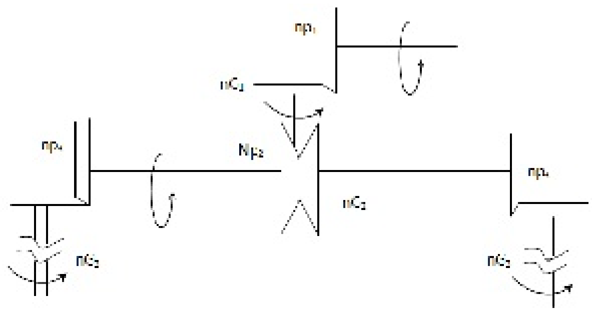

Fig. 7: Reduction gear ratio

Table 1a: Value of allowable static stress (δ_o)

Material	Allowable static stress (δ_o), (N/mm^2)
Forged Carbon Steel (Casehardening)	126
Forged Carbon Steel (Untreated)	140 – 210
Forged Carbon Steel (Heat treated)	210 – 245

Table 1b: 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%

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

Table 2: 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
TOTAL				66.40 %

Nomenclature

Symbol	*Meaning*	*Unit*
IITA	International Institute for Tropical Agriculture	
W	Load	kN
A_s	Area of screw	mm^2
σ_y	Yield stress	kN/mm^2
F.S	Factor of safety	
d_s	Diameter of screw	mm
d_o	Outer diameter of screw	mm
d_c	Core diameter of screw	mm
P	Pitch of thread	mm
a	Width of top of thread	mm
c	Width of the bottom of thread	mm
h	Height of thread	mm
d	Screw mean diamete	rmm
α	Helix angle	°
φ	Friction angle	°
β	Thread angle	°
μ	Coefficient of friction	
σ_c	Compressive stress	N/mm^2
τ	Shear stress on the screw	N/mm^2
T	Torque on screw	Nmm
$\hat{\sigma}_c$	Principal stress in compression	N/mm^2
σ_d	Value of stress	N/mm^2
τ_d	Value of shear stress used	N/mm^2
$\hat{\tau}$	Maximum shear stress	N/mm^2
Sfb	Screw follower bearing pressure	N/mm^2
n	Number of threads in the screw follower	
τ_s	Shear stress induced in screw	N/mm^2
τ_{sf}	Shear stress induced in the follower	N/mm^2
τ_t	Tearing stress	N/mm^2
D_1	Calculated outer diameter of follower	mm
P_s	Design load plus variationk	N
I	Second moment of area of the screw	mm^4
L_s	Length of screw	mm

Continue

δ_o	Allowable static stress	N/mm^2
C_v	Velocity factor	
V	Peripheral velocity	mm/s
b	Face width of tooth	
m	Module	mm
D	Gear diamete	rmm
T_n	Number of teeth	
y	Tooth form factor	
L	Cone distance	mm
D_o	Pitch diameter of gea	rmm
D_P	Pitch diameter of pinion	mm
W_D	Dynamic load	N
T_a	Actual torque at the 3rd stage	Nmm
W_T	Tangential tooth load	N
W_s	Static tooth load	N
BHN	Brinell Hardness Number	
δ_c	Flexural endurance limit	
W_w	Wear load	N
Q	Ratio factor for external gear	
K	Material combination factor	N/mm^2
δ_{se}	Surface endurance limit	N/mm^2
E_p	Young's Modulus of pinion material	GN/m^2
E_G	Young's Modulus of gear material	GN/m^2
θ	Pressure angle of gear	^{o}
P_o	Power output from gear	W
P_i	Power input	W
PCD	Pitch circle diamete	rmm

Appendix I

Nominal or Major diameter (d_o)	Minor or Core diameter (d_c)	Pitch (P), mm	Area of Core (A_c), mm^2
30	23.5		434
32	25.5		511
34	27.5	6	594
36	29.5		683
38	30.5		731
40	32.5		830

Continue

42	34.5	7	935
44	36.5		1046
46	37.5		1104
48	39.5		1225
50	41.5	8	1353
52	43.5		1486
55	45.5		1626
58	48.5		1847
60	50.5	9	2003
62	52.5		2165

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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