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Developing a Yam Flour Processing System

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ABSTRACT

Pounded yam is one of the staple in Africa. Unfortunately its traditional method of preparation is still time and energy consuming with low production output. A faster method of preparing instant yam flour with less energy requirement is therefore needed. A brief report on the design and development of yam flour processing system is hereby presented. The preliminary design of the system was done, followed by a detailed design to determine the specifications of the component parts. The slicing unit is made of six stainless blades which slice the yam tuber into smaller sizes of 2 mm thickness. The system has efficient drying and grinding units. The machine makes an average of 2 slices per second. An output efficiency of 82 per cent was achieved. It was observed that the size of chips affected the drying rate. Boiling of the yam chips was found to prolong drying time. This is because cooked yam dices have dense structure resulting from partial or complete gelatinization of starch granules. It is thus recommended that yam chips should not exceed 5mm in thickness and should not be cooked to avoid prolonged drying. Of course the drying chamber should be insulated.

Keywords: *Design, Yam Flour, Slicing, Drying, Grinding, Efficiency.*

1. INTRODUCTION

Yam is widely consumed in Africa and many other countries of the world. It is prepared and eaten in different forms. Nigeria is reported as the world's largest producer of yam with over 6 million metric tons annually (Osuji, 1985). It is however estimated that about 40 percent of the harvested tubers are lost due to lack of good storage facilities (Aluko, 2006). One of the interesting eating forms of yam is as pounded yam. This requires that the water content of the product be greatly reduced and the tuber converted to flour. It is interesting to note that the storage life of yam is greatly improved when in flour state. Its traditional preparation is however cumbersome. This is because it is laborious, time and energy consuming, with low production output. In other to eliminate the drudgery of pounding yam, especially at a commercial level, this project is therefore considered timely.

1.1 Statement of the Problem

In order to satisfy the local food demand of the growing population and also increase foreign earnings through exportation, Nigeria needs to drastically reduce the huge post harvest losses hitherto recorded as a result of poor storage facilities. In addressing this problem, considerations were given to the following.

1. How to achieve yam flour (pounded yam) that retains its natural colour.
2. How to achieve pounded yam with the desired visco-elastic characteristic.
3. How to eliminate the drudgery of pounding yam.
4. How to reduce the high transportation cost of yam tube

2. DESIGN CONCEPT

Figures 1.0 and 2.0 show the assembly drawing and the orthographic views of the machine, respectively. Table 1.0 shows the component parts of the machine.

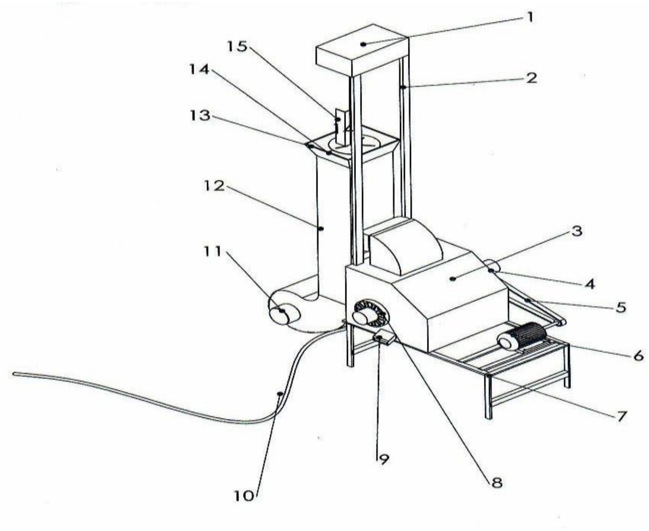


Fig 1.0 Assembly Drawing of the Machine

Table 1.0 List of Component Parts of the Machine

Identification Number	Description	Quantity
1	Motor Housing	1
2	Motor Support Frame	1
3	Grinding Unit	1
4	Drive Shaft	1
5	V-Belt	2
6	Electric Motor	2
7	Main Frame	1
8	Flywheel	1
9	Discharge Hopper	1
10	Power Cable	1
11	Blower	1
12	Drying Unit	1
13	Inlet Hopper	1
14	Slicing Disc	1
15	Inlet Channel	1

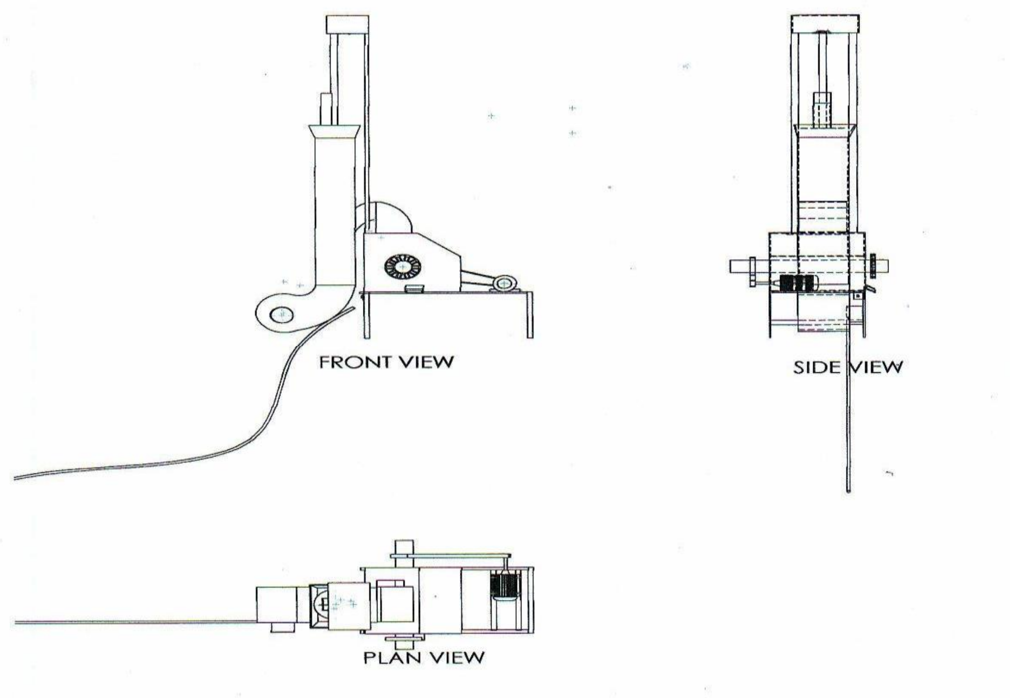


Fig 2.0 Orthographic views of the Machine

3. OPERATIONAL PRINCIPLE

Peeled yam tubers are cut into smaller sizes and charged into the hopper via the inlet column. A dead weight is introduced to push the charge into the slicing unit in order to ensure effective contact between the yam chips and the slicing disc. The blower directs heat energy generated by the heating element to the yam slices. The hot air enhances fast drying of the slices. The dried yam chips, which now has a lesser weight is gradually led to the grinding chamber by the air pressure. The dried slices are grounded to flour by the crushing blades. The mesh allows only the required grit size of flour to pass through the discharge hopper.

4. DESCRIPTION

The machine is made up of three units: the slicing, the drying, and the grinding units. The feed column makes an angle of 90" with the horizontal. This is to allow free fall of yam sizes into the slicing chamber. The slicing disc is made of 6 stainless blades which rotate about the central axis. The drying chamber has two major components; the blower and the heater. The yam chips are dried by the hot air generated in the drying chamber. The dried chips are pushed into the grinding chamber by air pressure. The grinding chamber is made of twenty four (24) crushing teeth, the drive shaft, the mesh, flywheel, and an electric motor. The dried chips are grounded to flour and are collected via the discharge hopper. The mesh prevents uncrushed yam chips from passing into the discharge hopper.

5. DESIGN ANALYSIS

5.1 Input Capacity

Wet density of yam (D. Rotundata) = 1020 kg/m^3 (Osuji, 1985)

The slicing unit is designed to slice 2 chips per second.
Average mass of yam chip = 1.784 g; Mass of 2 chips = 2 x 1.7842 = 3.5684g.
Slicing rate per hour = 12846.24g/hr = 7200chips/hr =12.846 kg/hr.

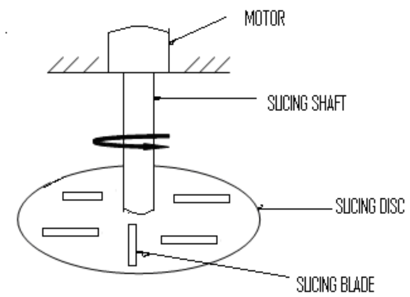
Thus, input capacity per hour = 12.846 kg/hr.
If the inlet hopper is to be charged every 30 minutes;
Input capacity = 6.5 kg; Volume of hopper = 6.5 / 1020 = 0.0064 m^3

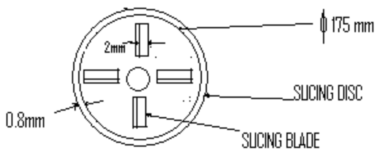
Considering 20% air clearance; Volume of hopper = 0.0064 + 0.0013 = 0.0077 m^3
Let base length of hopper, l = 0.20 m; width = 0.10 m;
height= h

Volume of hopper (trapezoid) = 3/2 (l x b x h) = 3/2 (0.20 x 0.10 x h) = 0.03h
0.0077 = 0.03h; h = 0.257 m: Use Height of Hopper, h = 0.30m

5.2 Slicing Disc Design

The slicing unit has six (6) blades of 0.8mm thick which slices the yam into 2mm thick.





Area of slicing disc = $\pi D^2/4$: where D = disc diameter = 175mm
Disc area = $(\pi \times 0.175^2)/4 = 0.024$ m^2
Perimeter (or circumference) of slicing disc = πD = 0.55 m
Thickness of slicing disc = 0.8mm
Area of slicing blade = lt: where l = length of slicing blade = 30mm
t = thickness of slicing blade = 0.8mm
Area = 0.030 x 0.0008 = 0.000024 m^2

Distance between slicing blades = $\theta\pi D$ /360 = (60 x π x 0.175)/360 = 0.09 m
where θ = 360/6 = 60°

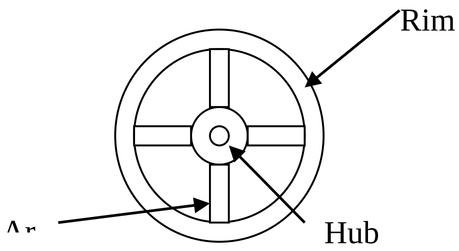
5.2 Slicing Disc Drive Shaft

Torque T = Fr where F = applied force; r = disc radius
Force acting on the shaft = weight of yam
Mass of yam = 6.5 kg; Weight of yam = 65 N
Shaft radius = 4/2 = 2mm = 2x 10^{-3}m
Torque = 65 x 2 x 10^{-3} = 0.13 N-m.
Total torque required = 0.13 x 6 = 0.78 N-m

Minimum power required, p = T ω
Where $\omega = (2\pi N)/60 = (2\pi \times 150)/60 = 15.71$ rad/sec
Power, p = 0.78 x 15.71 = 12.3 W; (Use 0.5 Horse Power motor)

Max shear stress $S_{max} = Tr/ I_0$
For a solid circular shaft, Polar moment of inertia, $I_0 = \pi D^4 /32$ (Hall et al, 191)
$I_0 = \pi \times 0.004^4/32 = 8.04$ m^4
$S_{max} = (0.78 \times 0.002)/8.04 = 1.94 \times 10^{-04}$ N/m^2

5.4 Flywheel Design



Diameter of flywheel = 245mm; Width = 60mm; Thickness = 30mm
Mean diameter = 230mm; Density of cast iron = 7260kg/m^3
Mass = $2\pi R \times A \times \rho$; where R = mean radius of rim

A = cross-sectional area of rim; ρ= density
But A = b x t; where b = width; t = thickness; A = 0.06 x 0.03 = 1.8 x $10^{-3}$$m^2$

Thus, Mass of flywheel = 2π x 0.115 x 1.8 x 10^{-3} x 7260 = 9.4kg
Tensile or Hoop stress $\sigma_t = \rho v^2$; where $v = \omega R$ (Khurmi, 2006)

But $\omega = \omega_2$ = 242.9rad/s; V = 242.9 x 0.115 = 27.9m/s
Thus, σ_t = 7260 x 27.9 = 202554N/m^2 =202.554KN/m^2

Kinetic energy of flywheel, E = ½ x I x ω^2 = ½.$MK^2\omega^2$;
Where; I = mass moment of inertia of the flywheel about the axis of rotation, kg-m^2

K = Radius of gyration of the flywheel, m
But K= R; where R= mean radius of rim; E = ½ x 9.4 x 0.115^2 x 242.9^2 = 3.6673KJ

Energy stored in the flywheel; $\Delta E = MV^2C_s$
Where M = mass of flywheel; v = linear velocity of flywheel; C_s = coefficient of fluctuation of speed
From design table (Brenda, 1999), C_s for crushing machines = 0.2

ΔE = 9.4 x 242.9^2 x 0.2 = 110.92KJ
Length (L) between fixed ends; L = $\pi D/n = 2\pi R/n$
where n = Number of arms = 4; L =(π x 230)/4 = 180.6 mm
Distributed load (W) per meter length = Centrifugal force between pair of arms
W = b.t.ρ.ω^2. R (N/m) = 0.006 x 0.003 x 7260 x 242.9^2 x 0.115 = 88667N/m

5.5 Maximum bending moment:

$M = (WL^2)/12 = (88667 \times 180.6 \times 10^{-3})/12 = 1334.4$ N-m (Shigley, 2002)
Section Module, $Z = 1/6\ (bt^2) = 1/6\ (60 \times 10^{-3} \times 30 \times 10^{-3}) = 0.0003\ m^3$
Bending Stress, $\sigma_b = M/Z = 1334.4/0.0003 = 4.448 \times 10^6\ N/m^2$
Total stress in the rim; $\sigma = ¾ (\sigma_t) + ¼ (\sigma_b)$
$\sigma = ¾ \times 202554 \times 10^3 + ¼ \times 4.448 \times 10^6 = 1.53 \times 10^2\ N/m^2$

5.6 Design of Drive Shaft

Number of crushing blade = 24; Weight of each crushing blade = 1N
Weight of driven pulley = 10N; Diameter of driven pulley = 125mm
Diameter of driving pulley = 145mm; Speed of driving pulley = 2000 rev/min

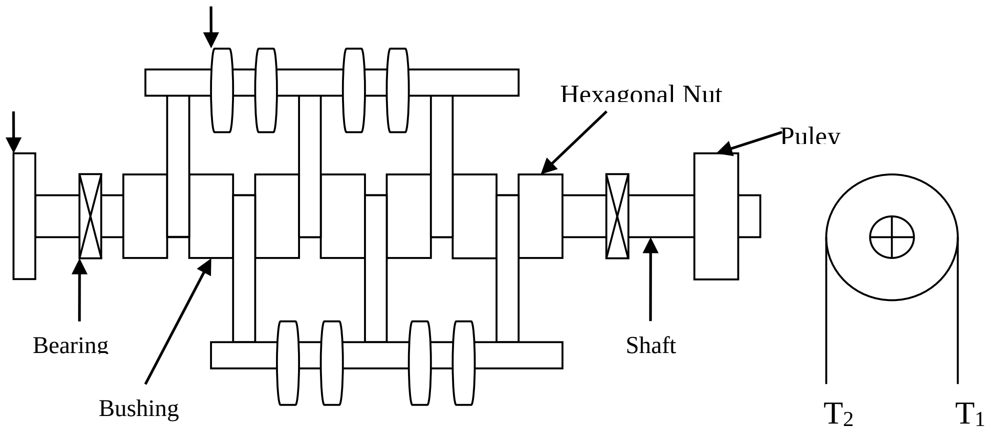


Fig.3. Force Diagram of Drive Shaft

Mean Diameter of flywheel (cast iron) = 230mm; Width of flywheel = 60mm
Thickness of flywheel = 30mm; Density of cast iron = 7260kg/m^3
Let T_1 = Tension on the tight side; T_2 = Tension on the slack side
D_1 = Diameter of driving pulley; D_2 = Diameter of driven pulley
N_1 = Rev/ Min of driving pulley; N_2 = Rev/Min of driven pulley
$N_1/N_2 = D_2/D_1$
(Shigley, 2004)

$N_2 = (2000 \times 145)/125 = 2320$ rpm
Angular velocity of driving pulley (Fig.3.0);
$\omega_1 = (2\pi N_1)/60 = (2\pi \times 2000)/60 = 209.4$rad/sec
Angular velocity of driven pulley; $\omega_2 = (2\pi N_2)/60 = (2\pi \times 2320)/60 = 242.9$rad/sec

Velocity of driving pulley $v_1 = r_1\omega_1 = (D_1\omega_1)/2 = 145/2 \times 10^{-3} \times 209.4 = 15.2$m/s
Velocity of driven pulley $v_2 = r_2\omega_2 = (D_2\omega_2)/2 = (125 \times 10^{-3} \times 242.9)/2 = 15.2$m/s
Shear force F = Power/Velocity; where Power = 2.2kw; Velocity = 15.2m/s
$F = (2.2 \times 10^3)/15.2 = 144.7$N
T = Power/Angular speed = $P/\omega_1 = (2.2 \times 10^3)/209.4 = 10.5$Nm
But, Torque $= (T_1 - T_2)r = (T_1 - T_2)D_1/2$
$T_1 - T_2 = (2 \times 10.5)/0.145 = 144.8$N
$T_1 = 144.8 + T_2$ (1)
Since velocity exceeds 10m/s, centrifugal force is considered.

Centrifugal force, $T_c = mv^2$
Where m = mass of drive belt, 0.1034kg/m; v = velocity
$T_c = 0.1034 \times (15.2)^2 = 23.9$ N
$T_{t1} = T_1 + T_C = T_1 + 23.9$ (2)

where T_{t1} = total tension on the tight side; T_{t2} = total tension on the slack side
$T_{t2} = T_2 + T_c = T_2 + 23.9$ (3)
But $T_{t1}/T_{t2} = e^{\mu\theta}$; where θ = angle of wrap, 3.1rads
μ = coefficient of friction = 0.13 (Shigley, 2004)

Thus, $T_{t1}/T_{t2} = e^{(0.13 \times 3.1)} = 1.5$; $T_1 + 23.9 = 1.5T_{t2}$
Substituting for T_1 in equation (1);$144.8 + T_2 + 23.9 = 1.5T_{t1}$;
But, $T_{t1} = T_2 + 23.9$
$144.8 + T_2 + 23.9 = 1.5(T_2 + 23.9)$
$T_2 + 168.7 = 1.5T_2 + 35.85$; $T_2 = 265.7$

From equation (1) $T_1 = 144.8\ T_2$; $T_1 = 410.5$N

$T_{t1} = 410.5 + 23.9 = 434.4$N
$T_{t2} = 265.7 + 23.9 = 289.6$N

Thus, total pull on the drive shaft due to belt tension

$T_T = T_{t1} + T_{t2}$; $T_T = 434.4\ 289.6 = 724$N

Total load on the shaft due to belt tension

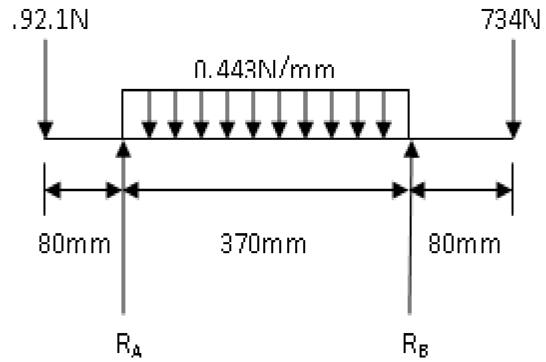
$W_p = T_T + W_p = 724 + 10 = 734$N;

where W_p = weight of pulley

Weight of flywheel $W_f = m.g$
Mass of flywheel, $M = 2\pi R \times A \times \rho$
But $A = b \times t$; where b = width of flywheel, t = thickness of flywheel
$A = 0.006 \times 0.003 = 1.8 \times 10^{-3} m^2$
$M = 2\pi \times 0.115 \times 1.8 \times 10^{-3} \times 7260 = 9.4kg$
$W_f = 9.4 \times 9.8 = 92.1N$

Since load in the crushing chamber is uniformly distributed; Total load in the crushing chamber = 20 x 2 + 20 x 5 + 1 x 24 = 164N.

5.6.1 Force Analysis



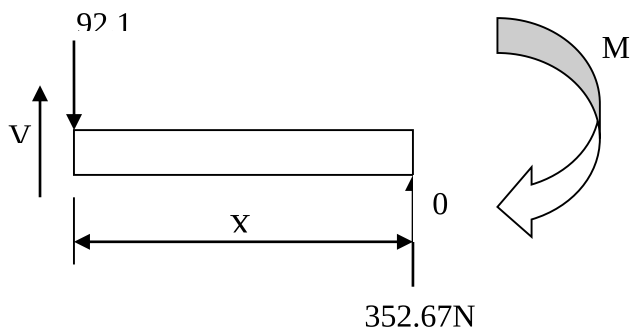
To determine the reactions R_A and R_B, at the supports, the 0.443N/mm distributed force is reduced to a concentrated force. For the distributed load; $W_d = 0.443 \times 370 = 163.91N$ (acts at the middle of the distributed load).

$\sum P_v = 0$; $-92.1 + R_A - 163.91 + R_B - 734 = 0$
$R_A + R_B = 92.1 + 163.91 + 734$; $R_A + R_B = 990.01N$

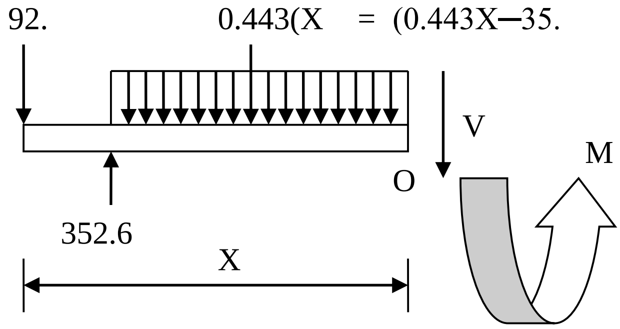
$\sum M_{RB} = 0$; $92.1(450) - R_A(370) + 163.91 (185) + 734 (80) = 0$
$370R_A = 41445 + 30323.35 + 58720$
$= 130488.35$; $R_A = 352.67N$

But $R_A + R_B = 990.01$; $R_B = 990.01 - R_A$; $R_B = 637.34N$
various sections of the shaft are taken to determine the critical points of maximum shear force, v and bending moments, M.

Taking a distance, x from the L.H.S



$\sum P_v = 0$; $-92.1 + 352.67 + v = 0$; $v = 92.1 - 352.67 = -260.57N$
Taking moment about 0: $\sum M_0 = 0$; $-92.1X$ N-mm.
Considering another section
at $80 \leq x \leq 450$



$\sum P_v = 0$; $-92.1 + 352.67 - (0.443x - 35.44) - v = 0$
$V = -92.1 + 352.67 - 0.443x + 35.44$; $V = (296.01 - 0.443x)N$.
$\sum M_0 = 0$; $M = (-0.2215x^2 + 296.01x - 29631.2)$ N-mm

Similarly: At $450 \leq x \leq 530$; $\sum P_v = 0$
$V = 92.1 - 352.67 + 163.91 - 637.34 = -734N$; $M = (734x - 271580.45)$ N-mm

5.6.2 Shear Force and Bending Moment Diagrams

Maximum bending moment (Fig.4), $M_b = 58719.55$N-mm = 58.71955N-m
Allowable shear stress, τ_a = 42 MPa (Hall et al, 1971)

Combined shock and fatigue factors for bending and torsional moments:
$K_b = 2 \sim 3$; $K_t = 1.5 \sim 3$: For safety, Let $K_b = 2$ and $K_t = 1.5$ are chosen.
$d^3 = 16/\pi\tau_a \; [(K_b M_b)^2 + (K_t T_r)^2]^{1/2}$
(Khurmi, 2006)

Where d = shaft diameter, M_b = maximum bending moment

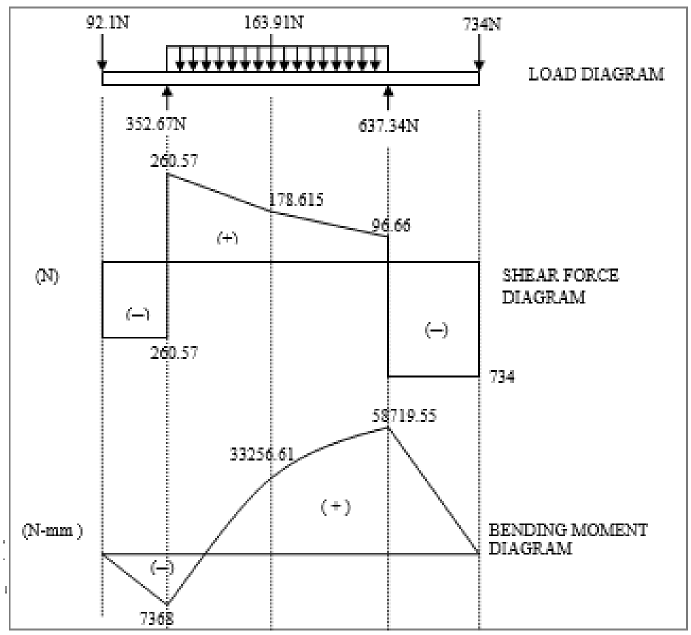


Fig.4 Shear Force and Bending Moment Diagrams

T_r = resultant torque, τ_a = allowable shear stress

$T_r = \sqrt{M_{2b}^2 + T^2} = \sqrt{58.71955^2 + 10.5^2} = 59.651N - m$

For safety, τ_a = 40MPa is chosen.

$d^3 = \frac{16}{\pi.x.40.x.10^6}\left[(2x58.71955)^2 + (1.5 \times 59.651)^2\right]^{1/2}$

d = 0.027m = 27 mm; Use d = 30mm

5.6.3 Checking for Torsional Rigidity

$\theta_o = \frac{T.L}{J.G}$ (Khurmi, 2006)

where θ_o = Angle of twist; L = Length of shaft, 0.53m
T = Torsional moment (Torque), 10.5N-m

G = Torsional modulus of elasticity = 80 x 10^9N/m^2 for steel

J = Polar moment of inertia = $\frac{\pi}{32} \times d^4 = 0.098d^4$

Thus; $\theta_o = \frac{T.L}{0.098d^4 \times G} = \frac{10.5 \times 0.53}{0.098(0.04)^4 \times 80 \times 10^9}$

=0.00028°/m

Since θ_o < 0.3°/m (permissible angle of twist); the design is safe.

5.7 Drying Time

Power= work done /Time = force x distance / Time = force x velocity = FV

F = P/v ; But V = πDN/60 where N = 1500 rev/min, D = 0.68 m, P = 2000 W
V =(π x 0.68 x 1500)/60 = 53.41 m/s; F = 2000 / 53.41 = 37.45N

But F = ma where m = mass of yam at wet state, 1180 kg
a = acceleration of hot air; a = F/m = 37.45 / 1180 = 0.03 m/s^2
Drying Time, t = Velocity/acceleration = 53.41/0.03 = 1780 sec. =30 minutes

6. DISCUSSION

Performance test result (Table 2 and Fig.5) shows that increase in grinding speed increases output quantity and reduces the discharge time. The grinding speed is directly proportional to the output and inversely proportional to the discharge time, provided the optimum speed is not exceeded. The optimum speed of 2320 rpm gives a discharge quantity of 10kg at 3600secs.From optimum speed;
Efficiency = (output/input) x 100 = 100/122 x 100 = 82%

Table 2 Performance Test Result

S/N	Grinding Speed (RPM)	Input Quantity Of Yam (kg)	Slicing Time (s)	Drying Time (mins)	Output Qty (kg)	Discharge Time (s)
1	1933	12.2	1800	50	8.5	4000
2	2000	12.2	1800	50	9.0	3900
3	2320	12.2	1800	50	10.0	3600
4	2417	12.2	1800	50	10.0	3600

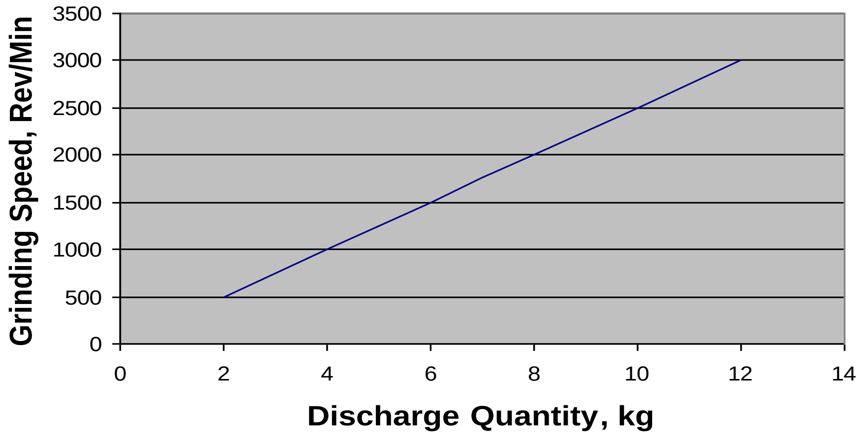


Fig.5 Performance Chart of the Yam Flour Processing Machine

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7. CONCLUSION

An output efficiency of 82 per cent was achieved. It was observed that the size of chips affects the drying rate. Boiling of yam chips prolonged drying time.

RECOMMENDATIONS

The yam chips should not exceed 5mm thick for faster drying; also it should not be cooked to avoid prolonged drying. The drying chamber should be insulated.

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