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Design, Fabrication and Testing of A Fonio Dehusking Machine

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***Abstract* – This paper presents the design, fabrication and testing of a Fonio dehusking machine. Though fonio is a staple food in many countries in West Africa due to its high nutrient content, it is not cultivated on a large scale because of difficulty in processing. The machine has the following units – feeding, rolling, dehusking and separating unit. the fonio passes through the rolling unit where it is abraded by two knurled shafts and is dehusked by the rotation of the dehusking drum in the dehusking unit. The fonio grain is separated from the chaff by passing the mixture through a current of air supplied by a fan. The materials used for the fabrication of the machine where selected based on the design considerations and analyses of its components. The components of the machine were selected locally and machined to specifications using machines such as the lathe, grinding machine, etc. The parts of the machine were assembled at the welding and fabrication workshop of the Federal Polytechnic, Bauchi. When tested, the machine dehusked 5kg of Fonio in 15minutes. It has an efficiency of 69%.**

***Keywords* – Fonio, Dehusking, Design.**

I. Introduction

Fonio (Digiteria exilis, ibrua), is one of the oldest cultivated cereals in Africa. The exact origin of fonio is unknown; but its use as a cereal dates back to the 14th century [1]. It is the smallest specie of millet, cultivated in many African countries. It grows well on poor soils[2]. A grain of Fonio paddy is oval in shape (i.e 1.5mm long and 0.9mm wide). The average particle size and specific gravity are 1.18mm and 1.47 respectively. Fonio has a brittle shell and can be dehusked faster when it is dry [1],[3]. It is used for the production of porridge, couscous, bread, tuwo and for brewing beer [4],[5].

Although its protein content is similar to that of other grains such as millet, it contains amino acids like Methionine, Riboflavin(Vitamin B_2) and Niacin(Vitamin B_3) [6] which are essential to human health, but deficient in other cereals. Fonio digests easily and is recommended for children, old and sick people suffering from diabetes or stomach disease. Doctors sometimes recommend it for people who want to lose weight. Though the uses of Fonio are many, its production has remained low (250 000 to 300 000 tons annually). This is because dehusking of fonio paddy is difficult [2]. In other words, only few dehusking machines are available for processing Fonio. These are some of the problems that have resulted to the non-availability of dehusked Fonio for human consumption despite its nutritional value [7],[8]. This neglect has made it to be listed among the 'lost crops of Africa'[9].

This work has the potential to encourage local famers of fonio to cultivate more of it; increase availability and competitiveness of fonio, resulting in increased consumption of same; create jobs in operation and maintenance of the machine; support sustainable land use and increased agricultural yield with the net effect of boosting food security and rural development.

Traditionally, the dehusking of Fonio is done in a mortar and the separation of the chaff from the grain is done by washing with water, or winnowing [2]. The traditional methods of processing takes 1 hour to dehusk 1kg – 2kg of Fonio paddy. Many Research institutes (National Cereals Research Institute (NCRI), Badeggi, Niger; International Plant Genetic Resource Institute (IPGRI) and National Research Institute of Mali) have supported research work on Fonio [10].

The CIRAD in collaboration with the national research institutes in Mali, Guinea, and Burkina Faso (between 1999 – 2004) developed a dehusker (GMBF dehusker) which has a production rate of 100kg/hr. It has dehusking and cleaning units. According to [2], the Fonio produced by the GMBF dehusker cooked faster than those produced by the local method. Another Fonio dehusking machine was developed in Senegal in 1993 by a mechanical engineer, Sanousi Diakite [11],[12]. [13] reports that a dehulling machine was developed in 1981 by Engr. Y. Kwa in Jos, Nigeria, but was silent on its performance. Although some of few machines exist, they are still very scarce and costly [11]. Therefore there is need for more dehusking machines to be developed for dehusking Fonio at an affordable cost, which is the main focus of this work.

The purpose of this work was to design, fabricate and test a fonio dehusking machine that is safe and economical in operation for the use of the local farmer.

II. Design Theory and Analysis

To achieve the set objectives, this machine is structured into four units: feeding, rolling, dehusking, and separation units. Each of the units performs specific function as described below:

Feeding unit

This unit is made up of a hopper, and a sieve for removing impurities such as metallic particle, stones, etc which are larger than the fonio paddy. The fonio is fed into the hopper manually. The hopper is made of steel sheets.

Rolling unit

This unit consist of a pair of knurled rollers mounted on bearings with a clearance between them, rolling at a speed ratio of 1.5:1. This enables frictional forces to be developed in the system, causing the fonio to be abraded

Dehusking unit

This unit consists of a split casing carrying two horizontal bars bolted to the lower half of the casing. Within the casing there is a threshing drum that carries conveyors and pegs. The conveyors allows the transfer of the Fonio paddy to the pegs that facilitate the dehusking. Rotation of the drum enables the dehusking of the Fonio to take place between the casing and the pegs.

Separation unit

This unit has an electrically operated blower and collectors for dehusked Fonio and the husk. The blowing was achieved through the fan. A variable resistor for adjusting the speed of the fan was incorporated.

III. Design

The components were designed based on established theories and principles, considering the loading of each member as follows.

Dehusking drum

The threshing drum is made of mild steel, consisting mainly of a hollow drum with 4 rectangular pegs welded to it. The Fonio paddy comes in between the pegs and the horizontal bars on the casing inside which the drum rotates. The loads on the drum were determined as follows:

Weight on drum, $W_d \; W_f = \rho_f . g . l_s \dfrac{\pi(d_i - d_o)^2}{4}$

$W_s = \rho_s . g . l_s . a$

$W_d = W_f + W_s$ Where l_s, ρ_f, a, W_s, W_f, d_i & d_o =length of drum, density of fonio, cross-sectional area of peg, weight ofone peg, weight of fonio between 2 pegs, internal &external diameter of drum respectively. Surface area of drum, $A_s = \dfrac{\pi d_o^2 l_s}{4}$

Pressure on surface of drum, $P_r = \dfrac{W_d}{A_s}$

where A_s = cross-sectional area of drum

Considering the drum as a pressure vessel, the following parameters were determined

Thickness of drum, $t_d = \dfrac{P_r . d_o}{2\sigma_{all}} + c$

Internal diameter of drum, $d_i = d_o - 2t_d$

Weight of drum, $W_{d2} = \rho_s . g \dfrac{\pi(d_o - d_i)^2 l_s}{4}$

Total load on drum, $W_t = W_d + W_{d2}$

Torque required to move this load, T $=W_t(\frac{d_o - d_i}{2})$

Power required to drive the shaft $P_1 = \dfrac{2\pi NT}{60}$[14]

Diameter of shaft

The shaft is subjected to both torsional and bending moments. Based on strength, the diameter of the shaft was therefore obtained theoretically using the relation $d = \sqrt[3]{\frac{16\sqrt{M^2+T^2}}{\pi\tau}}$ Where M = Bending moment, Nm; T = Torque, Nm; τ = Shear stress in shaft, N/m^2 d = Shaft diameter, m; $k = \frac{d_i}{d_o}$ [15]

The shaft diameter was also obtained based on the ASME formula for shaft subjected to both bending and torsional moments as follows

$d^3 = \frac{16}{\pi\tau(1-k^4)}\sqrt{(k_b M)^2 + (k_t T)^2}$.

Where T = Torsional moment or torque;

M = Bending moment; $K = \frac{d_i}{d_o}$,

d_o, d_i = Shaft's outside and internal diameters;

k_b = Combined shock and fatigue factor applied to bending moment

k_t = Combined shock and fatigue factor applied to torsional moment[1]. Based on the theory of rigidity, diameter of shaft was obtained using the relation $d = \sqrt[4]{\frac{584TL}{\theta G}}$ [17],[18]

Welded joint

The shaft is joined to the dehusking drum through a fillet weld, which is obviously subjected to torsion in the course of its operation.

Shear stress for the material, $\tau = \frac{Tr}{J} = \frac{2T}{\pi t d^2}$ [18]. This shear stress occurs in a horizontal plane along a leg of the fillet weld. The maximum shear occurs on the throat of the weld which is inclined at 45^o to the horizontal plane. length of throat, $t = s.\sin 45^o = 0.77s$;

Maximum shear stress, $\tau_{max} = \frac{2T}{0.77 s\pi d^2} = \frac{2.83T}{\pi s d^2}$.

Taking the allowable stress for mild steel to represent the maximum shear stress, $s = \frac{2.83T}{\pi\tau d^2}$

where d = Diameter of the solid shaft, m;

r = Radius of solid shaft, m;

T = Torque acting on the solid shaft, Nm;

s = Size (or leg)of weld;

t = Throat thickness, m;

J = Polar moment of inertia of the weld section $= \frac{\pi t d^3}{4}$ [15], [19]

Gears

The rollers, which abrade the fonio are driven by two meshing 20° full depth involute spur gears rotating with a speed ratio of 1.5 : 1. The minimum number of teeth on pinion to avoid interference, $T_p = \frac{2A_w}{G\left[\sqrt{1+\frac{1}{G}\left(\frac{1}{G}+2\right)\sin^2\phi - 1}\right]}$

Where Aw = the fraction by which the standard addendum for the wheel should be multiplied; ϕ= pressure angle; G = Velocity ratio = $\frac{T_w}{T_p} = \frac{D_w}{D_p}$ =1.5;

The number of teeth on the wheel was therefore obtained from the relation,

$T_w = 1.5T_p$.

The diameter of the wheel was therefore obtained from the relation $D_w = 1.5D_p$,

The module of gears, $m = \frac{D_w}{T_w} = \frac{D_p}{T_p} = 1.5$

Other parameters of the gears were determined using the following relations:

Tangential load, $W_T = \frac{P}{v} \times C_s$, Where

P = power transmitted in Watts,

V = pitch line velocity $= \frac{D\pi N}{60}\frac{m}{s}$, C_s = Service Factor,

D = Pitch circle diameter. Applying Lewis equation, $W_T = \sigma_o . C_v . b . \pi m . y$ $W_T = 4\sigma_o . C_v . P_c . \pi m . = 4\sigma_o . C_v . \pi^2 m^2 . y$ (Taking $b = 4p_c$) $\sigma_o = \frac{2M_t}{4m^3\pi^2 yN}$ (for unknown pitch diameter) The dynamic load (W_D) on the tooth was found using the relation $W_D = W_T + W = \frac{P}{V_r} + \frac{21V(b.c+W)_T}{21V+\sqrt{b.c.+W_T}}$, Neglecting the service factor [15],[20]. The static tooth load, $W_S = \sigma_e bP_C y = \sigma_e b\pi my$ The wear tooth load, $w_W = D_P.b.Q.K$. Where C = deformation factor. σ_o = Allowable static stress N/m^2 , c_v = Velocity factor, b = Tooth width, m = module. y = Tooth form factor, p_c =Circular pitch, M_t = Torque on smaller & weaker gear, σ_e = Elastic limit stress, w_w = Max. Load for wear, D_p = Pitch circle diameter, mm, K =Load stress factor (material contribution factor) N/mm^2. b = Face width of pinion,

$\square$ = Ratio factor $= \frac{T_G}{T_G+T_P}$; $K = \frac{\sigma_{es}^2 \sin\emptyset}{1.4}\left(\frac{1}{E_p}+\frac{1}{E_G}\right) = \frac{2\sigma_{es}^2 \sin\square}{1.4E_{PG}}$ Where σ_{es} = surface endurance limit N/mm^2 E_p = Young's Modulus for Pinion material N/mm^2,

E_G = Young's Modulus for gear material N/mm^2, E_{PG} = Young's Modulus for Pinion material and gear material N/mm^2

Load on the bearings of the wheels due to the power transmitted were obtained as follow:

Radial load on bearings of gears**,**

$W_R = W_N Sin\phi$; W_N =Normal load [15]

Rollers

The forces exerted by the fonio paddy while being abraded were assumed negligible, so the rollers were designed only as the shaft for the spur gears. The rollers, shown in fig. 1 were designed as follows:

Normal loading between the tooth surface **(**W_N)

$W_N = \frac{W_T}{\cos\square}$, where W_T= Tangential load, $\square$ = Pressure angle

The weight of the gear is given by $W_G = 0.00118 T_G bm^2$, (N)

Where T_G = Number of teeth on gear; b = Face with of gear; m = Module of gear .

The resultant load on the gear wheel was found using the cosine rule thus

$$W_R = \sqrt{W_N^2 + W_G^2 + 2W_N . W_G . \cos\square}$$

Bending moment on the shaft due to resultant load, $M = W_R . x$

Where x = Overhang, ie distance between centre of gear and bearing, m

The equivalent torque ,

$T_e = \sqrt{M^2 + T^2} = \frac{\pi}{16} \times \tau \times d^3$

Where T = Twisting moment $= W_T \times \frac{D_G}{2}$

Therefore, $d = \sqrt[3]{\frac{16\sqrt{M^2+T_e^2}}{\pi\tau}}$ [15]

Belt drive

Belts are employed to transmit power from one shaft to another. The choice of belt drive was informed by some of its advantages over some other means of power transmission, such as gear and chain drives [14] The V-belt was chosen because of its numerous advantages such as compactness, quietness in operation, ease of mounting and removing, positive drive, and so on; and because the shafts between which power is being transmitted are relatively close.

The following relations were used to determine the required parameters [11]:

The pitch line velocity of the belt,

$$v = \frac{\pi DN}{60} ....\,.........\,....(i)$$

The ratio of belt tension,

$$\frac{T_1}{T_2} = 2.3e^{\mu \ \operatorname{cosec}\beta} ....(ii)$$

The power transmitted,

$$P = (T_1 - T_2)V \text{ Watts } ....(ii\,)$$

Where β = groove angle; μ = coeicient of friction; θ = angle of lap

Length of the belt

The belt drive used in is the open type, whose lengths were found using the relation

$L = \pi(r_1 + r_2) + 2x + \frac{(r_1+r_2)^2}{x}$, where r_1 and r_2 represent the radii of the larger and smaller pulleys, x distance between the centres of the pulleys and L the total length of the belt [15],[16].

Bearings

Radial ball bearings were employed for the shaft and rollers because of their advantages; they compact in size, able to stand momentary shocks, easy to mount, and reliable in service. They are also known to have low maintenance cost and do not require starting torque.

The following relations were used in designing the bearings:

The Dynamic equivalent radial load, $W = XVW_R + YW_A$ [15].

where W_R and W_A are radial and axial loads respectively; V is the rotation factor, a constant dependent on the type of bearing, while X and Y represent the radial and axial load factors respectively.

The Dynamic load rating, $C = W\left(\frac{L}{10^6}\right)^{\frac{1}{k}}$

Where L and W represent the Rating life and Equivalent Dynamic Load,

while K is a constant

The Bearing life**,** $L = 60N.L_H$ (rev.)

Where: N = Speed in rpm, L_N = workink life(hrs)

Separation unit

An electrically operated fan was used to provides the air required to separate the seed from the husk.

It was assumed that $0.01m^3$ of mixture of dehusked fonio and impurities will pass through the separation unit per unit time. Out of this quantity 5%will be sand particles.

The following relations were used:

Velocity of mixture $V_{fs}^{1} = \sqrt{2gS}$;

Force with which fonio falls $F_f = M_f \times V_f^1$; Force with which sand falls $F_s = M_s \times V_s^1$

The force required to separate the mixture of fonio and impurities:

$$Q_a = A_a^1, \quad \text{Where: } A = \frac{\pi d^2}{4} \quad \therefore Q_a = \frac{\pi d^2 V_a^1}{4}$$

The separation force must be equal to the total force exerted by the mixture, F_t

$$\therefore F_t = F_s + F_f = \rho_f Q_a V_a^1, \quad \Rightarrow \quad F_t = \frac{\rho_a \pi d^2 (V_a^1)^2}{4}$$

This gives the velocity of air required to achieve separation, thus

$$V_a^s = \sqrt{\frac{4F_t}{\rho_a \pi d^2}}$$

Where: density of fonio = $\rho_f kg/m^3$;Density of sand particles = $\rho_s kg/m^3$; Volume of fonio = $V_f m^3$; Volume of mixture of fonio and impurities = $V_{fi} m^3$; Volume of sand particles = $V_s m^3$

Mass of sand = $M_s kg$; Mass of fonio = $M_f kg$; Velocity of fonio = $V_f^1 m/s$

Velocity of sand particles = $V_s^1 m/s$; Velocity of mixture of fonio and impurities = $V_{fs}^1 m/s$

Diameter of circle swept by blower = d m; Velocity of air = $V_a^1 m/s$,

Height from which mixture of fonio and impurities fall = S m; Force exerted by fonio = F_f N

Force exerted by sand particles = F_s N; Volumetric flow rate of air = Qm^3/s

Efficiency of the machine

The efficiency of the machine was determined by using the following relation

M_{dh} was obtained by separating the totally dehusked fonio grain from those not wholly dehusked by sieving and then weighing the former in a balance.

IV. Conclusion

The machine dehusked 5kg of Fonio paddy in 12-15 minutes. From the results, the efficiency of the machine was determined to be 69%.

For further work, the clearance between the pegs and horizontal bars in the dehusking chamber should be slightly increased to prevent crushing of some of the Fonio paddy. This will in turn increase the efficiency of the machine.

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APENDIX
DESIGN ANALYSIS

INPUT	CALCULATIONS	OUTPUT
$l_d = 300\ mm$ $d_o = 100\ mm$ $N = 800\ rpm$ $f_1 = 10\ mm$ $f_2 = 10\ mm$ $\eta_s = 7850\ kg/m^3$ $\eta_f = 8.66\ kg/m^3$ $\sigma_{all} = 56\ MPa$ $\tau_{all} = 42\ MPa$	**DEHUSKING DRUM** *Weight on drum,* W_d $W_f = \rho_f . g . l_s \frac{\pi(d_f - d_o)^2}{4} = 8.66 \times 9.81 \times 0.3 \times \frac{\pi(0.110 - 0.01)^2}{4}$ $= 200.17 \times 10^{-3}\ N$ $W_s = \rho_s . g . l_s . f_1 . f_2 = 7850 \times 9.81 \times 0.3 \times 0.01 \times 0.01 = 2.31N$ $W_d = W_f + W_s = 200.17 \times 10^{-3} + 2.31 = 2.51N$ *Surface area of drum,* A_s $A_s = \frac{\pi d_o^2 l_s}{4} = \frac{\pi \times 0.1^2 \times 0.3}{4} = 2.36 \times 10^{-3} m^2$ *Pressure on surface of drum,* P_r $P_r = \frac{W_d}{A_s} = \frac{2.51}{2.36 \times 10^{-3}} = 1063.56 N/m^2$	$W_d = 2.51N$
$W_d = 2.51N$ $A_s = 2.36x10^{-3} m^2$	*Thickness of drum,* t_d $t_d = \frac{P_r . d_o}{2\sigma_{all}} + c = \frac{1063.56 \times 0.1}{2 \times 56 \times 10^6} + 0.003 = 3mm$ t_d = 4mm was adopted for convenience of production.	$P_r = 1063.56 N/m^2$
	Internal diameter of drum, d_i $d_i = d_o - 2t_d = 0.1 - (2 \times 0.004) = 0.092m = 92mm$	$t_d = 4mm$ $d_i = 92mm$
$t_d = 4mm$	*Weight of drum,* W_{d2} $W_{d2} = \eta_s . g \frac{\pi(d_o - d_i)^2 l_s}{4} = 7850 \times 9.81 \frac{\pi \times (0.1 - 0.092)^2 \times 0.3}{4} = 580.63N$ *Total load on drum,* W_t $W_t = W_d + W_{d2} = 2.51 + 580.63 = 583.14N$ This load is distributed along the effective length of the shaft; thus load on shaft $= \frac{583.14}{0.3} = \frac{583.14}{0.3} = 1943.8 N/m$	$W_{d2} = 580.63N$ $W_t = 583.14N$
$W_{d2} = 580.63N$ $l_s = 0.3m$ $w = 1943.8N$		

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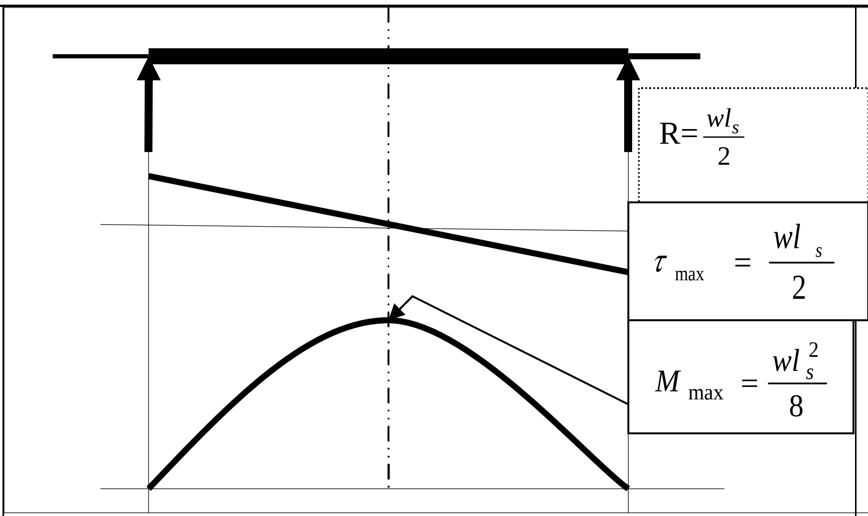
$\tau = 291.57 N/m^2$		
$N = 300rpm$ $T = 879.35x10^{-3} Nm$ $K_t = 1.5$ $K_b = 1.5$ $M = 21.87$		$\tau = 291.57 N/m^2$ M $= 21.87 Nm$
$G = 80{,}000 N/mm^2$ $\theta = 0.25^0$	$R = \tau_{\max} = \frac{1943.8 \times 0.3}{2} = 291.57 N/m^2$ $M_{ax} = \frac{1943.8 \times 0.3^2}{8} = 21.87 Nm$ *Torque required to move this load, T* $T = \frac{\pi\left(d_o^4 - d_i^4\right) \times 271.57}{16 \times 0.1} = 879.35 \times 10^{-3} Nm$	
$d = 25mm$ $\tau = 291.57 N/m^2$ $T =$ $879.35x10^{-3} N/m$	*Power required to drive the shaft* $P_1 = \frac{2\pi NT}{60} = \frac{2\pi \times 200 \times 879.35 \times 10^{-3}}{60} = 27.63W = 30W$	$T =$ 879.35×10^{-3} Nm $P_1 = 30Watts$
V.R = 1.5 $T_2 = 18$ $\theta = 20^o$ m = 2 N_1 =300rpm D_1 = 28 T_1 = 27 P = 0.5KW N_2= 155.56rpm T_P=30.69Nm D_2=28mm	*Diameter of shaft, d* On the basis of strength, $d^3 = \frac{16}{\rho\tau}\sqrt{(MK_b)^2 + (TK_t)^2}$ $= \frac{16}{\pi\tau}\sqrt{(21.87x1.5)^2 + \left(879.35 \times 10^{-3} \times 1.5\right)^2} = 8.45mm \approx 9mm$ On the basis of rigidity, $d^4 = \frac{584Tl_s}{G\theta} = \frac{584 \times 879.35 \times 10^{-3} \times 0.3}{80 \times 0.25}$ $d = 16.7 \approx 17mm$	$d = 9mm$
W_T=2192.14mm P_c=4.89mm W_N=2332.83N b=15mm $\phi = 20^o$ $\sigma_{es} = 350MPa$ $E =$ $202000 N/mm^2$ $\sigma_e = 252MPa$ $m = 2$ $T_2 = 18$ $b = 15mm$	*A standard diameter of 25mm was adopted* *Welded joint* The shaft is connected to the hollow drum by a fillet weld. $S = \frac{2.83T}{\pi\tau d^2} = \frac{2.83 \times 879.35 \times 10^{-3}}{\pi \times 291.57 \times 0.025^2} = 4.35 \approx 5mm$ **GEARS** $T_1 = 1.5T_2 = 1.5 \times 18 = 27$ *Pitch circle diameters of wheel and pinion, D_1 & D_2* $D_1 = mT_1 = 2 \times 27 = 54mm$ $D_2 = \frac{D_1T_2}{T_1} = \frac{54 \times 18}{27} = 28mm$ *The speed of the wheel was assumed to be 300rpm.*	Working diameter $d = 25mm$ $S = 3mm$ T_1=27 T_2=18

N_2=155.56rpm
e=0.0925
K=0.111
E=202000N/mm^2
W_T=2192.14N

W_N = 2332.83
T = 27
T_2 = 18
b = 15mm
m = 2
T_G = T_1, T_2

W_N = 2332.83
$\phi = 20°$

X = distance between bearing & gear = 50mm

W_T=21292.14N
D_1=0.054m
D_2=0.028m

$\tau = 42MN/mm^2$

Torque, $T_P = \dfrac{P \times 60}{2\pi N_2} = \dfrac{500 \times 60}{2 \times \pi \times 155.56} = 30.69 Nm$

Tangential load, $W_T = \dfrac{2T_P}{D_2} = \dfrac{30.69 \times 2}{0.028} = 2192.14N$

Normal load, $W_N = \dfrac{W_T}{\cos\phi} = \dfrac{2192.14}{\cos 20^o} = 2332.83N$

Face width of gear teeth, b = 3P_c = 3 ×4.89 = 14.67 $\approx 15mm$

Radial load on bearings of gears,

$$W_R = W_N Sin\phi = 2332.83 x Sin 20^o = 797.87N$$

Limiting load for wear, $W_w = D.b.Q.K$

$$W_w = 54 \times 15 \times 1.2 \times 296.30 \times 10^{-3} = 288N$$

Static tooth load (or endurance strength) of gear tooth,

$W_s = \sigma_e.b.\pi.m.y$

$$y = \frac{0.912}{T_2} = \frac{0.912}{18} = 50.67 \times 10^{-3}$$

$$W_s = 252 \times 15 \times \pi \times 2 \times 50.67 \times 10^{-3} = 1203.43N$$

Dynamic load,

$$W_D = W_T + \frac{21\upsilon(b.c + W_T)}{21\upsilon + \sqrt{b.c + W_T}}$$

$$W_D = 2192.14 + \left[\frac{21 \times 228.06 \times 10^{-3}((0.015 \times 1037.03) + 2192.14)}{21 \times 228.06 \times 10^{-3} + \sqrt{(0.015 \times 1037.03) + 2192.14}}\right]$$

$= 204.20N$

This design can be considered safe since W_D is less than both W_s and W_w.

ROLLERS

Weight of wheel, W_G

$$W_G = 0.00118 T_{Gb} m^2$$

$$W_{G1} = 0.00118 \times 27 \times 15 \times 2^2 = 1.91$$

Weight of pinion, $\mathbf{W}_{G2} = 0.0018 \times 18 \times 15 \times 2^2 = 1.27$

Resultant load on wheel, W_{R1}

$$W_{R1} = \sqrt{23322.83^2 + 1.91^2 - \left(2 \times 2332.83 \times 1.91 \times Cos 20^\circ\right)}$$

$= 2331.04$

Resultant load on pinion, W_{R2}

$$W_{R2} = \sqrt{2332.83^2 + 1.27^2 - \left(2 \times 2332.83 \times 1.27 \times Cos 20^\circ\right)}$$

=2331.56

Bending moment on shaft of wheel, M_1

$M_1 = W_{R1}$ X

$M_1 = 2331.04 \times 0.005 = 116.55$

Bending moment on shaft of wheel, M_2

$M_2 = W_{R2}$ X = 2331,56 × 0.05 = 116.58

Twisting moment on wheel shaft, T_{s1}

D_1=54mm
D_2=28mm

T_p=30.69Nm
W_T=2192.14
W_N =2332.83N

b=15mm

W_R=797.87N

W_w = 288N

W_s =1203.43N

W_D =202.20N

W_{G1} = 1.91N
W_{G2} = 1.27N

$m = 108.05\times10^{-3} kg/m$

$P = 0.5kW$

$v = 1.18 m/s$

$r_1 = 50\times10^{-3} m$

$r_2 = 37.5\times10^{-3} m$

$r_3 = 37.5\times10^{-3} m$

$x_1 = 0.525m$

$x_2 = 0.302m$

From tables,
V=1
X=1
W_A=0
W_R=779.87N
N=300rpm
L_H=3years
=26280hours
K=3
L=478040000rev.
W=797.87

$$T_{s1} = W_T \times \frac{D_2}{2} = 2192.14\times\frac{0.084}{2} = 59.12 Nm$$

Twisting moment on pinion shaft, Ts_2

$$T_{s2} = W_T \times \frac{D_2}{2} = 2192.14\times\frac{0.028}{2} = 20.69 Nm$$

Equivalent toque on wheel & pinion T_{ew}, T_{ep}

$$\mathrm{T_{ew}} = \sqrt{M_1{}^2 + T_{s1}} = \sqrt{116.55^2 + 59.12^2} = 30.69 Nm$$

$$\mathrm{T_{ep}} = \sqrt{M_1^2 + T_{s1}^2} = \sqrt{116.55^2 + 59.12^2} = 120.55\ \mathrm{Nm}$$

Diameters of wheel and Pinion shafts, d_w, d_p

$$d_w^3 = \frac{16T_{ew}}{\pi\tau} = \frac{16\times130.69}{42\pi}$$

$$d_w = 9.6mm$$

$$d_p^3 = \frac{16T_{ep}}{\pi\tau} = \frac{16\times120.55}{42\pi}$$

$$d_p = 9.4mm$$

Belt Drive

The power to be transmitted was assumed to be 0.5kW

$$v = \frac{\pi DN}{60} = \frac{\pi\times75\times10^{-3}\times300}{60} = 1.18 m/s$$

Belt tensions

$$T_1 = 2mv^2 = 2\times108.05\times10^{-3}\times1.18^2 = 0.3N$$

$$T_2 = \frac{P-2mv^2}{v} = \frac{0.5-(2\times108.05\times10^{-3}\times1.18^2}{1.18} = 0.42N$$

Length of belts

$$L_1 = \pi(r_1 + r_2) + 2x + \frac{(r_1 + r_2)^2}{x}$$

$$= \pi(50\times10^{-3} + 37.5\times10^{-3}) + (2\times0.525) + \frac{(50\times10^{-3} + 37.5\times10^{-3})^2}{0.525}$$

$$L_1 = 1.331\ m$$

$$L_2 = \pi(r_2 + r_3) + 2x + \frac{(r_2 + r_3)^2}{x}$$

$$= \pi(2\times37.5\times10^{-3}) + (2\times0.302) + \frac{(2\times37.5\times10^{-3})^2}{0.302} = 0.696\ m$$

The top widths & thicknesses of the belts were selected from tables [15]

Bearings

Bearings for rollers

Dynamic equivalent load, W

$$W = XVW_R + YW_A = 1\times1\times797.87 = 797.87N$$

Life of bearing in revolutions, L

$$L = 60NL_H = 60\times300\times26280 = 473040000 rev.$$

Dynamic load rating, C

$$C = W\left(\frac{L}{10^6}\right)^{\frac{1}{k}} = 797.87\times\left(\frac{478040000}{10^6}\right)^{\frac{1}{3}} = 6.24kN$$

The closest standard value to this from tables was chosen: C=7.65kN and

M_1=116.55Nm

M_2=116.58Nm

T_{s1}=59.12Nm

T_{s2}=30.69Nm

T_{ew}=130.69Nm

T_{ep}= 120.55Nm

$d_w = d_p \approx 10mm$

$v = 1.18m/s$

$T_1 = 0.3N$

$T_2 = 0.42N$

$L_1 = 1331mm$

$L_2 = 696mm$

Top width
=13mm
Thickness
=8mm

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W_t = 583.14N
W_{pu} = 5N
X = 1; V = 1
W_R = 296.57N
W_A = 0

N = 300rpm
L_H = 3years
=26280hours
K = 3
L = 478040000rev.
W = 296.57N

$\rho_f = 8.66 kg/m^3$

$\rho_s = 13.10 kg/m^3$

$V_{fs} = 0.01 m^3$

$S = 0.3m$

$V_f^1 = V_s^1 = V_{fs}^1$

$= 2.43 m/s$

$\rho_a = 1.2 kg/m^3$

the bearing specifications were chosen from tables based on this value. [15]

Bearing for shaft

Radial load on bearing, W_R

W_R = 0.5×Weight of shaft(W_t)+weight of pulley(Wpu)
= 0.5×583.14+5 = 296.57N

Dynamic equivalent load on bearing, W

$$W = XVW_R + YW_A = 1 \times 1 \times 296.57 = 296.57N$$

Life of bearing

$$L = 60NL_H = 60 \times 300 \times 26280 = 473040000 rev.$$

Dynamic load rating, C

$$C = W\left(\frac{L}{10^6}\right)^{\frac{1}{k}} = 296.57 \times \left(\frac{478040000}{10^6}\right)^{\frac{1}{3}} = 2.32kN$$

The standard dynamic capacity in tables closest to this value is 4kN, corresponding to bearing number 200. this bore is too small for the chosen shaft diameter. Bearing number 205 with a bore of 25mm was selected. It is of higher capacity.

Separation Unit

Assuming that the volume of dehusked fonio and impurities flowing into the separation unit per second is $0.01m^3$, out of which 5% is sand particles, then

$$Vs = \frac{5}{100} \times V_{fs} = 0.05 \times 0.01 = 0.0005m^3$$

$V_f = 0.01 \times 0.95 = 0.0095m^3$

$M_f = V_f \times \rho_f = 0.0095 \times 8.66 = 0.08227kg$

$M_s = V_s \times \rho_s = 0.0005 \times 13.10 = 0.00655kg$

$$V_{fs}^1 = \sqrt{2gS} = \sqrt{2 \times 9.81 \times 0.3} = 2.43 m/s$$

$$F_f = M_f \times V_f^1 = 0.08227 \times 2.43 = 0.19992N$$

$$F_s = M_s \times V_s = 0.00655 \times 2.43 = 0.01592N$$

$$F_t = F_s + F_f = 0.01592 + 0.19992 = 0.259N$$

$$(V_2^1)^2 = \sqrt{\frac{4F_t}{\rho_a \pi d^2}} = \sqrt{\frac{4 \times 0.259}{1.2 \times \pi \times 0.2^2}}$$

$$\therefore V_a^1 = 2.621 m/s$$

This is the velocity of air required to separate the mixture of fonio from the chaff and impurities.

W
=797.87N
L
=478040000rev.
C=7.65kN

Bearing No.
=301
Bore=12mm
Outside dia.
=37mm
Width
=12mm

W_R
= 296.57N
W = 296.57N
L = 478040000 rev.

C = 2.32kN
Bearing No.
= 205
Bore = 25mm
Outside dia.
= 52mm
Width= 15m

Vs =$0.0005m^3$
V_f = $0.0095m^3$
Mf = 0.08227kg
M_s = 0.00655kg
V_{fs}^1 =2.43m/s
F_f
= 0.19992 N
F_s
= 0.01592 N
F_t
= 0.259 N
V_a^1
= 2.621 m / s

Table 1: Test results

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Test No.	Quantity of Fonio, kg	Time taken to dehusk, min.	% of grain dehusked
1	5	12	60
2	5	15	70
3	5	13	77

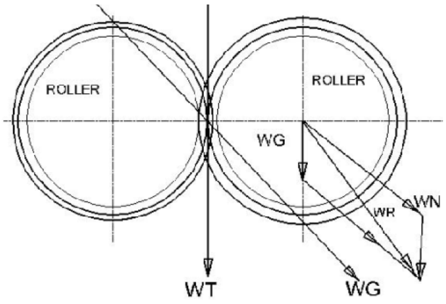

Fig.1: Forces on the rollers

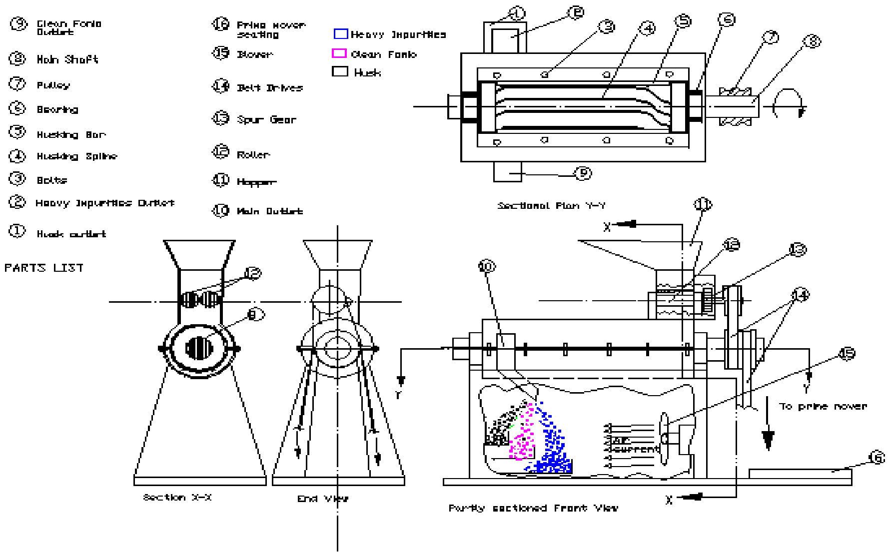

Fig. 2: Orthographic view of Fonio dehusking machine