Design, Fabrication and Testing of A Fonio Dehusking Machine
Tokan A., Danladi Y. B., Shekarau M. B. E. and Datau S. G.

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 (Digitaria 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 B₂) and Niacin(Vitamin B₃) [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 allow 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, \( W_d \) = \( \rho_f \cdot g \cdot d \cdot \pi (d_1 - d_2)^2 \)

- Pressure on surface of drum, \( P_r \) = \( \frac{W_d}{A_s} \)

where \( A_s \) = cross-sectional area of drum

**Parameters**

- Thickness of drum, \( t_d = \frac{P_r \cdot d_2}{2 \sigma_{ail}} + c \)
- Internal diameter of drum, \( d_1 = d_o - 2t_d \)
- Weight of drum, \( W_{d2} = \rho_s \cdot g \cdot \pi (d_1 - d_2)^2 l_s \)
- Total load on drum, \( W_t = W_d + W_{d2} \)
- Torque required to move this load, \( T = \frac{W_t (d_0 - d_1)}{2} \)

**Power required to drive the shaft** \( P_t = \frac{2 \pi NT}{60} \)

**Diameter of shaft**

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

\[
d^3 = \frac{16}{\pi^2 (1 - k_f^2)} \sqrt{(k_b M)^2 + (k_c T)^2}.
\]

Where \( T \) = Torsional moment or torque; \( M = \text{Bending moment} \); \( k_f \) = allowable stress; \( k_b \) = Combined shock and fatigue factor applied to bending moment; \( k_c \) = Combined shock and fatigue factor applied to torsional moment.[1]. Based on the theory of rigidity, diameter of shaft was obtained using the relation \( d = \frac{4 [5 \times 10^6]}{\sqrt{66}} \) [17],[18]

**Welded joint**

The shank 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, \( \tau = \frac{T}{J} \cdot \frac{2T}{\pi d^2} \) [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° to the horizontal plane.

**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, \( T_p = \frac{2 \omega v}{G} \sqrt{1 + \frac{1}{2} (G - 1) / G^2} \sin^2 \phi - 1 \)

Where \( \omega v = \text{the fraction by which the standard addendum for the wheel should be multiplied; } \phi = \text{pressure angle} \); \( G = \text{Velocity ratio} \); \( T_p = \frac{D_p}{D_w} = 1.5 \).

The number of teeth on the wheel was therefore obtained from the relation, \( T_w = 1.5 T_p \)

The diameter of the wheel was therefore obtained from the relation \( D_w = 1.5 D_p \).

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, \( W_t = \frac{P}{C_s} \cdot X C_a \), Where \( P = \text{power transmitted in Watts} \); \( V = \text{pitch line velocity} = \frac{\text{DrN} \cdot m}{60 \cdot s} \); \( C_s = \text{Service Factor} \).
D = Pitch circle diameter. Applying Lewis equation,  

\[ W_T = \sigma_0 C_v b \tan \pi m y \]

\[ W_T = 4 \sigma_0 C_v \tfrac{p}{\pi} m_u = 4 \sigma_0 C_v \pi^2 m^2 y \]  
\( \text{(Taking } b = 4 p_c \)  
\( \sigma_0 = \tfrac{2m_t}{4m^2\pi^2 y N} \)  
\( \text{for unknown pitch diameter) The dynamic load } (W_d) \text{ on the tooth was found using the relation } W_d = W_T + W = \tfrac{p}{\pi} + \tfrac{21Vb}{b.c} + W_T \]  
\( \text{Neglecting the service factor [15], [20]. The static tooth load, } W_g = \sigma_1 b_1 p_c y = \sigma_1 b_1 m_u y \text{ The wear tooth load, } W_{w} = D_p h_b Q K \)  
\( \text{Where } C = \text{ deformation factor, } \sigma_0 = \text{ Allowable static stress } N/m^2, c_v = \text{ Velocity factor, } b = \text{ Tooth width, } m = \text{ module, } y = \text{ Tooth form factor, } p_c = \text{Circular pitch, } M_t = \text{Torque on smaller & weaker gear, } \sigma_e = \text{ Elastic limit stress, } W_w = \text{ Max. Load for wear, } \) 
\( D_p = \text{Pitch circle diameter, } mm, K = \text{Load stress factor (material contribution factor) N/mm}^2, b = \text{Face width of pinion, } \) 
\( \theta = \text{ Ratio factor } = \tfrac{T_0}{T_0 + 1}, K = \tfrac{\sigma_0 b_1 \sin \pi}{1.4 E_p (1 + \tfrac{1}{E_p})} = \tfrac{2\sigma_0 \sin^3}{1.4 E_p} \]  
\( \text{Where } \sigma_{es} = \text{surface endurance limit N/mm}^2, E_p = \text{Young’s Module for Pinion material N/mm}^2, \) 
\( E_G = \text{Young’s Modulus for gear material N/mm}^2, \) 
\( E_{pc} = \text{Young’s Modulus for Pinion material and gear material N/mm}^2 \)  
\( \text{Load on the bearings of the wheels due to the power transmitted were obtained as follow: } \) 

Radial load on bearings of gears,  
\( W_R = W_N \sin \theta; W_N = \text{Normal load} \)  

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 \( W_N \)  
\( W_N = \tfrac{W_T}{\cos \theta}, \text{ where } W_T = \text{Tangential load, } \theta = \text{Pressure angle} \)  

The weight of the gear is given by \( W_G = 0.00118 T_G b \text{mm}^2; (N) \)  
\( \text{Where } T_G = \text{Number of teeth on gear; } b = \text{Face with of gear; } m = \text{Module of gear}. \)  
\( \text{The resultant load on the gear wheel was found using the cosine rule thus } \) 
\( W_R = \sqrt{W_N^2 + W_G^2 + 2 W_N W_G \cos \theta} \]  

Bending moment on the shaft due to resultant load,  
\( M = W_G x \)  
\( \text{Where } x = \) 
\( \text{Overhang distance between centre of gear and bearing. } \)  
\( \text{The equivalent torque, } \) 
\( T_e = \tfrac{W_G + T_G}{\pi} = \phi \text{ } \alpha \text{ } \tau \times d^3 \]  
\( \text{Where } T = \text{Twisting moment } = W_T \times \tfrac{D_G}{2} \]  
\( \text{Therefore, } d = \tfrac{3}{16} \sqrt{M^2 + T_e^2} \)  

\[ \text{Belts are employed to transmit power from one shaft to another. The choice of belt drive was informed by some of its advantages over some other means of power transmission, such as gear and chain drives [14] The V-belt was chosen because of its numerous advantages such as compactness, quietness in operation, ease of mounting and removing, positive drive, and so on; and because the shafts between which power is being transmitted are relatively close. } \]  
\( \text{The following relations were used to determine the required parameters [11]: } \) 
\( v = \tfrac{\pi DN}{60} \)  
\( \text{The ratio of belt tension, } \frac{T_1}{T_2} = 2.3b \text{ cosec } \beta \)  
\( \text{The power transmitted, } P = \left( T_1 - T_2 \right) V \text{ Watts } \)  
\( \text{Where } \beta = \text{groove angle; } \mu = \text{coefficient of friction; } \theta = \text{angle of lap} \)  

\text{Length of the belt} \) 
\( \text{The belt driven used in is the open type, whose lengths were found using the relation } \) 
\( L = \tfrac{\pi (r_1 + r_2)}{2} + 2x + \tfrac{(r_1 r_2)^2}{x} \)  
\( \text{where } r_1 \text{ and } r_2 \text{ represent the radii of the larger and smaller pulleys, } x \text{ distance between the centres of the pulleys and } L \text{ the total length of the belt [15],[16]. } \) 

\text{Bearings} \) 
\( \text{Radial ball bearings were employed for the shaft and rollers because of their advantages; they compact in size, able to stand momentary shocks, easy to mount, and reliable in service. They are also known to have low maintenance cost and do not require starting torque. } \) 
\( \text{The following relations were used in designing the bearings: } \) 
\( \text{The Dynamic equivalent radial load, } W = XW_R + YW_A \)  
\( \text{where } W_R \text{ and } W_A \text{ are radial and axial loads respectively; } V \text{ is the rotation factor, a constant dependent on the type of bearing, while } X \text{ and } Y \text{ represent the radial and axial load factors respectively. } \) 
\( \text{The Dynamic load rating, } C = W_L \frac{L}{106} K \)  
\( \text{Where } L \text{ and } W \text{ represent the Rating life and Equivalent Dynamic Load, } \) 
\( \text{while } K \text{ is a constant } \) 
\( \text{The Bearing life, } L = 60 N \times L_H \text{ (rev.) } \) 
\( \text{Where: } N = \text{Speed in rpm}, L_H = \text{workin life(hours) } \) 

\text{separation unit} \) 
\( \text{An electrically operated fan was used to provides the air required to separate the seed from the husk. } \) 
\( \text{It was assumed that 0.01m}^3 \text{ of mixture of dehusked fonio and impurities will pass through the separation unit per unit time. Out of this quantity 5% will be sand particles. } \) 
\( \text{The following relations were used: } \)
Velocity of mixture $V_{m} = \sqrt{2gS}$

Force with which fonio falls $F_{f} = M_{f} \times V^{1}_{f}$; Force with which sand falls $F_{s} = M_{s} \times V^{1}_{s}$

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

$Q_{a} = A \mu$.

\[ \frac{2m}{A} V^{1}_{a} = \frac{A}{2} \mu \]

The separation force must be equal to the total force exerted by the mixture, $F_{i}$

\[ F_{i} = F_{s} + F_{f} = \rho_{f} Q_{a} V^{1}_{a} \Rightarrow F_{i} = \frac{\rho_{a} A d^{2} (V^{1}_{a})^{2}}{4} \]

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

\[ V_{a} = \sqrt{\frac{4F_{i}}{\rho_{a} A d^{2}}} \]

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 $= V_{m} m^{3}$; Volume of sand particles $= V_{s} m^{3}$

Mass of sand $= M_{s} kg$; Mass of fonio $= M_{f} kg$

Velocity of fonio $= V^{1}_{f} m/s$

Velocity of sand particles $= V^{1}_{s} m/s$; Velocity of mixture of fonio and impurities $= V^{1}_{m} m/s$

Diameter of circle swept by blower $= d$ m; Velocity of air $= V^{1}_{a} m/s$

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 $= Q m^{3}/s$

Efficiency of the machine

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

\[ M_{th} = \text{mass obtained by separating the totally dehusked fonio grain from those not wholly dehusked by sieving and then weighing the former in a balance.} \]

**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.

**REFERENCES**


Free patents online (2009).


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APENDIX
DESIGN ANALYSIS

<table>
<thead>
<tr>
<th>INPUT</th>
<th>CALCULATIONS</th>
<th>OUTPUT</th>
</tr>
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<tbody>
<tr>
<td>( l_d = 300 \text{ mm} )</td>
<td>Weight on drum, ( W_d ) ( \quad \begin{align*} W_f &amp;= \rho_f g l s \pi (d_f - d_o)^2 \ &amp;= 8.66 \times 9.81 \times 0.3 \times \frac{\pi (0.110 - 0.01)^2}{4} \end{align*} )</td>
<td>( W_d = 2.51N )</td>
</tr>
<tr>
<td>( d_o = 100 \text{ mm} )</td>
<td>( W_d = W_f + W_s = 200.17 \times 10^{-3} + 2.31 = 2.51N )</td>
<td></td>
</tr>
<tr>
<td>( N = 800 \text{ rpm} )</td>
<td><strong>DEHUSKING DRUM</strong> ( \begin{align*} f_1 &amp;= 10 \text{ mm} \ f_2 &amp;= 10 \text{ mm} \end{align*} )</td>
<td></td>
</tr>
<tr>
<td>( \eta_s = 7850 \text{ kg} / \text{ m}^3 )</td>
<td>( \eta_f = 8.66 \text{ kg} / \text{ m}^3 ) ( \begin{align*} \sigma_{all} &amp;= 56 \text{ MPa} \ \tau_{all} &amp;= 42 \text{ MPa} \end{align*} )</td>
<td>( \sigma_{all} = 56 \text{ MPa} ) ( \tau_{all} = 42 \text{ MPa} )</td>
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<td></td>
<td>Surface area of drum, ( A_s ) ( \begin{align*} A_s &amp;= \frac{\pi d_o^2 l_s}{4} \ &amp;= \pi \times 0.1^2 \times 0.3 \end{align*} )</td>
<td>( A_s = 2.36 \times 10^{-3} \text{ m}^2 )</td>
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<td></td>
<td>Pressure on surface of drum, ( P_r ) ( \begin{align*} P_r &amp;= \frac{W_d}{A_s} \ &amp;= \frac{2.51}{2.36 \times 10^{-3}} = 1063.56 \text{ N} / \text{ m}^2 \end{align*} )</td>
<td>( P_r = 1063.56 \text{ N} / \text{ m}^2 )</td>
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<td></td>
<td><em><em>Thickness of drum, ( t_d ) ( \begin{align</em>} t_d &amp;= \frac{P_r d_o}{2 \sigma_{all}} + c \ &amp;= \frac{1063.56 \times 0.1}{2 \times 56 \times 10^6} + 0.003 = 3\text{ mm} \ t_d &amp;= 4\text{ mm} \end{align</em>} )</td>
<td>( t_d = 4\text{ mm} )</td>
</tr>
<tr>
<td>( W_d = 2.51N )</td>
<td>Internal diameter of drum, ( d_i ) ( \begin{align*} d_i &amp;= d_o - 2t_d = 0.1 - (2 \times 0.004) = 0.092 \text{ m} = 92\text{ mm} \end{align*} )</td>
<td>( d_i = 92\text{ mm} )</td>
</tr>
<tr>
<td>( A_s = 2.36 \times 10^{-3} \text{ m}^2 )</td>
<td>Weight of drum, ( W_{d2} ) ( \begin{align*} W_{d2} &amp;= \rho g \frac{\pi (d_o - d_i)^2 l_s}{4} \ &amp;= 7850 \times 9.81 \times \frac{\pi \times (0.109 - 0.092)^2 \times 0.3}{4} \end{align*} )</td>
<td>( W_{d2} = 580.63N )</td>
</tr>
<tr>
<td></td>
<td>Total load on drum, ( W_t ) ( \begin{align*} W_t &amp;= W_f + W_{d2} = 2.51 + 580.63 = 583.14N \end{align*} )</td>
<td>( W_t = 583.14N )</td>
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<td>This load is distributed along the effective length of the shaft; thus load on shaft ( \begin{align*} \frac{583.14}{0.3} &amp;= 1943.8 \text{ N} / \text{ m} \end{align*} )</td>
<td>( \frac{583.14}{0.3} = 1943.8 \text{ N} / \text{ m} )</td>
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<tr>
<td>( W_{d2} = 580.63N )</td>
<td>( l_s = 0.3 \text{ m} ) ( \begin{align*} l_s &amp;= 36 \text{ m} \end{align*} )</td>
<td></td>
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<tr>
<td>( w = 1943.8 \text{ N} )</td>
<td><strong>Copyright © 2012 IJEIR. All right reserved</strong></td>
<td>494</td>
</tr>
</tbody>
</table>
\( \tau = 291.57 \text{N/m}^2 \)

\( N = 300 \text{rpm} \)

\( T = 879.35 \times 10^{-3} \text{Nm} \)

\( K_t = 1.5 \)

\( K_b = 1.5 \)

\( M = 21.87 \)

\( G = 80,000 \text{N/mm}^2 \)

\( \theta = 0.25^\circ \)

\( d = 25 \text{mm} \)

\( \tau = 291.57 \text{N/m}^2 \)

\( T = 879.35 \times 10^{-3} \text{N/m} \)

\( V.R = 1.5 \)

\( T_1 = 18 \)

\( \theta = 20^\circ \)

\( m = 2 \)

\( N_1 = 300 \text{rpm} \)

\( D_1 = 28 \)

\( T_1 = 27 \)

\( P = 0.5 \text{KW} \)

\( N_2 = 155.56 \text{rpm} \)

\( T_2 = 30.69 \text{Nm} \)

\( D_2 = 28 \text{mm} \)

\( W_t = 2192.14 \text{mm} \)

\( P_t = 4.89 \text{mm} \)

\( W_s = 3323.83 \text{N} \)

\( b = 15 \text{mm} \)

\( \phi = 20^\circ \)

\( \sigma_{o_t} = 350 \text{MPa} \)

\( E = \)

\( 202000 \text{N/mm}^2 \)

\( \sigma_c = 252 \text{MPa} \)

\( m = 2 \)

\( T_2 = 18 \)

\( b = 15 \text{mm} \)

\[ \tau = \frac{1943.8 \times 0.3}{2} = 291.57 \text{N/m}^2 \]

\[ M_{\text{ax}} = \frac{1943.8 \times 0.3}{8} = 21.87 \text{Nm} \]

Torque required to move this load, \( T \)

\[ T = \frac{\pi (d^4 - d^4)}{16 \times 0.1} = 879.35 \times 10^{-3} \text{Nm} \]

Power required to drive the shaft

\[ P_1 = \frac{2 \pi NT}{60} = \frac{2 \pi \times 200 \times 879.35 \times 10^{-3}}{60} = 27.63 \text{W} = 30 \text{W} \]

Diameter of shaft, \( d \)

On the basis of strength,

\[ d^3 = \frac{16 \pi}{\rho \tau} \left( MK_b^2 + TK_t^2 \right) \]

\[ = \frac{16 \pi}{\pi} \sqrt{(21.87 \times 1.5)^2 + (879.35 \times 10^{-3} \times 1.5)^2} = 8.45 \text{mm} \approx 9 \text{mm} \]

On the basis of rigidity,

\[ d^4 = \frac{584 T_1}{G \theta t} = \frac{584 \times 879.35 \times 10^{-3} \times 0.3}{80 \times 0.25} \]

\[ d = 16.7 \approx 17 \text{mm} \]

A standard diameter of 25mm was adopted

Welded joint

The shaft is connected to the hollow drum by a fillet weld.

\[ S = \frac{2.83 T}{\pi a t^2} = \frac{2.83 \times 879.35 \times 10^{-3}}{\pi \times 291.57 \times 0.025} = 4.35 \approx 5 \text{mm} \]

GEARS

\[ T_1 = 1.5 T_2 = 1.5 \times 18 = 27 \]

Pitch circle diameters of wheel and pinion, \( D_1 \) & \( D_2 \)

\[ D_1 = m T_1 = 2 \times 27 = 54 \text{mm} \]

\[ D_2 = D_1 \frac{T_2}{T_1} = \frac{54 \times 18}{27} = 28 \text{mm} \]

The speed of the wheel was assumed to be 300rpm.
| Torque, $T_P = \frac{P \times 60}{2\pi N_2} = \frac{500 \times 60}{2 \times \pi \times 155.56} = 30.69\,\text{Nm}$ |
| Tangential load, $W_T = \frac{2T_P}{D_2} = \frac{30.69 \times 2}{0.028} = 2192.14\,\text{N}$ |
| Normal load, $W_N = \frac{W_T}{\cos \phi} = \frac{2192.14}{\cos 20^\circ} = 2332.83\,\text{N}$ |
| Face width of gear teeth, $b = 3P_c = 3 \times 4.89 = 14.67 \approx 15\,\text{mm}$ |
| Radial load on bearings of gears, $W_R = W_N \sin \phi = 2332.83 \times \sin 20^\circ = 797.87\,\text{N}$ |
| Limiting load for wear, $W_w = D \cdot b \cdot Q \cdot K$ |

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

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

$W_s = \sigma_y \cdot b \cdot \pi \cdot m \cdot y$

$y = \frac{0.912}{18} = 0.05 \times 10^{-3}$

$W_s = 252 \times 15 \times \pi \times 2 \times 0.05 \times 10^{-3} = 1203.43\,\text{N}$

Dynamic load,

$W_D = W_T + \frac{21 \nu (b \cdot c + W_T)}{21 \nu + (b \cdot c + W_T)}$

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

$W_D = 204.20\,\text{N}$

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

<table>
<thead>
<tr>
<th>ROLLERS</th>
</tr>
</thead>
</table>

$W_R = 2192.14\,\text{N}$

Weight of wheel, $W_G$

$W_G = 0.00118 T_G \cdot m^2$

$W_{G1} = 0.00118 \times 27 \times 15 \times 2^3 = 1.91$

Weight of pinion, $W_{G2} = 0.0018 \times 18 \times 15 \times 2^3 = 1.27$

Resultant load on wheel, $W_{R1}$

$W_{R1} = \sqrt{2332.83^2 + 1.91^2 - 2 \times 2332.83 \times 1.91 \times \cos 20^\circ}$

$W_{R1} = 2331.04$

Resultant load on pinion, $W_{R2}$

$W_{R2} = \sqrt{2332.83^2 + 1.27^2 - 2 \times 2332.83 \times 1.27 \times \cos 20^\circ}$

$W_{R2} = 2331.56$

Bending moment on shaft of wheel, $M_1$

$M_1 = W_{R1} \times x$

$M_1 = 2331.04 \times 0.005 = 116.55$

Bending moment on shaft of wheel, $M_2$

$M_2 = W_{R2} \times x = 2331.56 \times 0.05 = 116.58$

Twisting moment on wheel shaft, $T_{s1}$

$\tau = 42\,\text{MN/} \text{mm}^2$
\[ T_{s1} = W_T \times \frac{D_2}{2} = 2192.14 \times \frac{0.084}{2} = 59.12 \text{Nm} \]

\[ T_{s2} = W_T \times \frac{D_2}{2} = 2192.14 \times \frac{0.028}{2} = 20.69 \text{Nm} \]

\[ T_{ep} = \sqrt{M_1^2 + T_{s1}^2} = \sqrt{116.55^2 + 59.12^2} = 69.12 \text{Nm} \]

\[ T_{EW} = \sqrt{M_1^2 + T_{s2}^2} = \sqrt{116.55^2 + 20.69^2} = 69.12 \text{Nm} \]

**Diameters of wheel and Pinion shafts, \( d_w, d_p \)**

\[ d_w = \frac{16T_{ew}}{\pi \tau} = \frac{16 \times 130.69}{42\pi} = 9.6 \text{mm} \]

\[ d_p = \frac{16T_{ew}}{\pi \tau} = \frac{16 \times 120.55}{42\pi} = 9.4 \text{mm} \]

**BELT DRIVE**

The power to be transmitted was assumed to be 0.5kW

\[ v = \frac{\pi DN}{60} = \frac{\pi \times 75 \times 10^{-3} \times 300}{60} = 1.18 \text{m/s} \]

**Belt tensions**

\[ T_1 = 2mv^2 = 2 \times 108.05 \times 10^{-3} \times 1.18^2 = 0.3 \text{N} \]

\[ T_2 = \frac{P - 2mv^2}{v} = \frac{0.5 - (2 \times 108.05 \times 10^{-3} \times 1.18^2)}{1.18} = 0.42 \text{N} \]

**Length of belts**

\[ L_1 = \pi (r_1 + r_2) - 2x + \frac{(r_1 + r_2)^2}{2x} \]

\[ = \pi (50 \times 10^{-3} + 37.5 \times 10^{-3}) + (2 \times 0.525) + \frac{(50 \times 10^{-3} + 37.5 \times 10^{-3})^2}{0.525} \]

\[ L_1 = 1.331 \text{m} \]

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

\[ = \pi (2 \times 37.5 \times 10^{-3}) + (2 \times 0.302) + \frac{(2 \times 37.5 \times 10^{-3})^2}{0.302} = 0.696 \text{m} \]

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

**BEARINGS**

**Bearings for rollers**

Dynamic equivalent load, \( W \)

\[ W = XYW_k + YW_A = 1 \times 1 \times 797.87 = 797.87 \text{N} \]

Life of bearing in revolutions, \( L \)

\[ L = 600L_{106} = 60 \times 300 \times 26280 = 473040000 \text{rev}. \]

Dynamic load rating, \( C \)

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

The closest standard value to this from tables was chosen: \( C = 7.65 \text{kN} \) and

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<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Weight of shaft (W&lt;sub&gt;t&lt;/sub&gt;)</td>
<td>583.14N</td>
</tr>
<tr>
<td>Weight of pulley (W&lt;sub&gt;pu&lt;/sub&gt;)</td>
<td>5N</td>
</tr>
<tr>
<td>X</td>
<td>1</td>
</tr>
<tr>
<td>W&lt;sub&gt;K&lt;/sub&gt;</td>
<td>296.57N</td>
</tr>
<tr>
<td>W&lt;sub&gt;A&lt;/sub&gt;</td>
<td>0</td>
</tr>
<tr>
<td>N</td>
<td>300rpm</td>
</tr>
<tr>
<td>L&lt;sub&gt;H&lt;/sub&gt;</td>
<td>3years = 26280hours</td>
</tr>
<tr>
<td>K</td>
<td>3</td>
</tr>
<tr>
<td>L</td>
<td>478040000rev.</td>
</tr>
<tr>
<td>W</td>
<td>296.57N</td>
</tr>
</tbody>
</table>

**Table 1: Test results**

- W<sub>t</sub> = 583.14N
- W<sub>pu</sub> = 5N
- X = 1
- W<sub>K</sub> = 296.57N
- W<sub>A</sub> = 0
- N = 300rpm
- L<sub>H</sub> = 3 years = 26280 hours
- K = 3
- L = 478040000 rev.
- W = 296.57N

**Bearing for shaft**

Radial load on bearing, W<sub>R</sub>

W<sub>R</sub> = 0.5 x Weight of shaft(W<sub>t</sub>) + weight of pulley(W<sub>pu</sub>)

= 0.5 x 583.14 + 5 = 296.57N

Dynamic equivalent load on bearing, W<sub>A</sub>

W<sub>A</sub> = 1 x 1 x 296.57 = 296.57N

Life of bearing

L = 60 x L<sub>H</sub> = 60 x 300 x 26280 = 473040000 rev.

Dynamic load rating, C

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

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 0.01 m<sup>3</sup>, out of which 5% is sand particles, then

\[ V_s = \frac{0.01}{100} \times V_s = 0.05 \times 0.01 = 0.0005m^3 \]

\[ V_f = 0.01 \times 0.95 = 0.0095m^3 \]

\[ V_{fs} = V_f \times \rho_f = 0.0005 \times 13.10 = 0.00655kg \]

\[ V_{f1} = \sqrt{2gS} = \sqrt{2 \times 9.81 \times 0.3} = 2.43m/s \]

\[ F_f = M_f \times V_{f1} = 0.08227 \times 2.43 = 0.19992N \]

\[ F_s = M_s \times V_s = 0.00655 \times 0.08227 = 0.01592N \]

\[ F_p = F_s + F_f = 0.01592 + 0.19992 = 0.259N \]

\[ \left( V_{f1}^2 \right) = \frac{4F_f}{\rho_f \pi d^2} = \frac{4 \times 0.259}{1.2 \times \pi \times 0.2^2} \]

\[ V_{f1} = 2.621m/s \]

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

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<table>
<thead>
<tr>
<th>Test No.</th>
<th>Quantity of Fonio, kg</th>
<th>Time taken to dehusk, min.</th>
<th>% of grain dehusked</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>5</td>
<td>12</td>
<td>60</td>
</tr>
<tr>
<td>2</td>
<td>5</td>
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<td>70</td>
</tr>
<tr>
<td>3</td>
<td>5</td>
<td>13</td>
<td>77</td>
</tr>
</tbody>
</table>

Fig. 1: Forces on the rollers

Fig. 2: Orthographic view of Fonio dehusking machine