

RESEARCH PAPER
DESIGN ANALYSIS OF AN EXISTING ACHA (*DIGITARIA EXILIS*) DEHULLING MACHINE

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ABSTRACT

An existing acha dehulling machine was improved upon by re-designing. The performance of the existing dehulling machine was first evaluated to determine the required modification. Improvements were undertaken in the cylinder speed, cylinder-concave clearance, hopper slope and opening, cylinder type and fan speed. The physical properties, terminal velocities of acha and the acha chaff, velocity of air required for the cleaning were determined and used to re-design the machine. Major component parts were designed using relevant engineering design principles. The modified acha dehuller had cylinder speed, cylinder-concave clearance, hopper slope, hopper hole opening and fan speed of 2800 rpm, 10 mm, 57°, 10 mm and 2800 rpm respectively as against the existing dehuller values of 934 rpm, 20mm, 67°, 20 mm and 934 rpm respectively. Terminal velocities of acha and the acha chaff, velocity of air required for the cleaning determined were 3.96 m/s, 1.9 m/s and 2.5 m/s respectively.

Keywords: *acha, dehulling machine, fonio, performance, terminal velocity*

INTRODUCTION

Acha (*Digitaria exilis* (Kippist) Stapf) also known as fonio is a minor cereal in many countries of West Africa where it is the staple food crop for several millions of tribal people (Vietmeyer *et al.*, 1996), and it is considered to be the tastiest of all cereals (Cruz, 2004). The challenge today is to produce enough acha to meet the growing demands for its products (Philip and Itodo, 2006).

Processing acha is a difficult and time-consuming task because of the extremely small size of the grain. One gram of acha contains nearly 2000 grains and each egg-shaped grain is only about 1 - 1.5 mm long (Cruz, 2004). Traditionally, dehulling paddy acha is accomplished by pounding using a pestle and mortar. The productivity of this work is very low, accompanied with dirt, sand and other foreign materials that have to be removed. It takes

nearly one hour to pound just one or two kilograms of acha paddy. Thus, mechanizing the processing and cleaning of acha is essential both to reduce the painstaking work for women and to improve the quality and availability of the product in the market.

Few acha dehulling machines exist in West Africa, so they are scarce and costly. Tokan *et al.*, (2012) developed an acha dehulling machine which dehulled 5kg of acha in 15 minutes. Using "Sanoussi" principle, Marouzé *et al.* (2008), reported a low output of about 25 kg/hr with a high percentage of broken grains. In Nigeria an indigenous artisan fabricated an acha dehulling machine which had no engineering design. Also, no performance evaluation was carried out on the machine to ascertain its level of efficiency. There was therefore the need to provide this relevant information as a way forward in developing acha dehuller on a commercial basis. This was carried out at the University of Agriculture, Makurdi, Nigeria.

The purpose of this work therefore, is to provide relevant engineering design of an existing acha dehuller using conventional engineering design principles and equations to modify it.

MATERIALS AND METHODS

Description of the prototype acha dehuller

The prototype acha dehuller (Plate 1) which was designed and improved upon is electrically driven. It is made basically of a hopper, the dehulling unit, the fan unit, the cleaning chamber and the outlets. The hopper has a trapezoidal shape and welded onto the dehulling unit. It has the capacity to hold 9 kg of acha, with a circular opening of 20 mm at the bottom to discharge the unde-hulled acha into the dehulling unit. The dehulling unit comprises the cylinder and concave. The cylinder was constructed from mild steel rods of 5.6 mm in diameter welded together at a space of 12 mm from each other to form an open rasp bar cylinder type. The concave which encloses the cylinder was constructed from mild steel rods of 5.6 mm diameter but closely welded (no spacing).

The concave-cylinder clearance is 20mm and this is where dehulling takes place by impact and rubbing actions. Fig. 1 is the detailed drawing of the acha dehuller.

Design analysis of machine

Design considerations

The main considerations in the design and improvement of the acha dehuller were that:

- The dehulling efficiency of the machine will be improved by changing the cylinder material.
- The cleaning efficiency of the dehuller will be improved upon.
- The flow of acha from the hopper to the dehulling chamber will be reduced by reducing the orifice opening.
- The cylinder speed will be increased to improve the peripheral speed so as to achieve more impact and rubbing actions.

Design of machine components

Design of hopper

The hopper is the housing where acha paddy to be dehulled are stored and fed into the machine at a determined rate. The hopper has a shape which facilitates loading, maximum volume utilization, a reliable and complete discharge of material by gravity.

Hopper flow characteristics

A. Coefficient of mobility

The coefficient of mobility is the ease with which the grain moves. This has to do with the internal friction and friction of material on surfaces. A coefficient of mobility > 0.3 is an indication of a high movement and flow of properties (Irtwange, 2000). Using the formula (Eq. 1) given by Stepanoff (1969),

$$m_i = 1 + 2\mu_i^2 - 2\mu_i(1 + \mu_i^2)^{1/2} \quad (1)$$

where

m_i = coefficient of mobility



a. Side view

b. Front view

Plate 1: Acha dehulling machine

A. Hopper B. Electric motor C. Dehulling unit cover D. Electric motor/dehuller belt E. Frame F. Dehulling unit G. Fan unit H. Dehulled acha chamber I. Air chamber J. Cleaning chamber

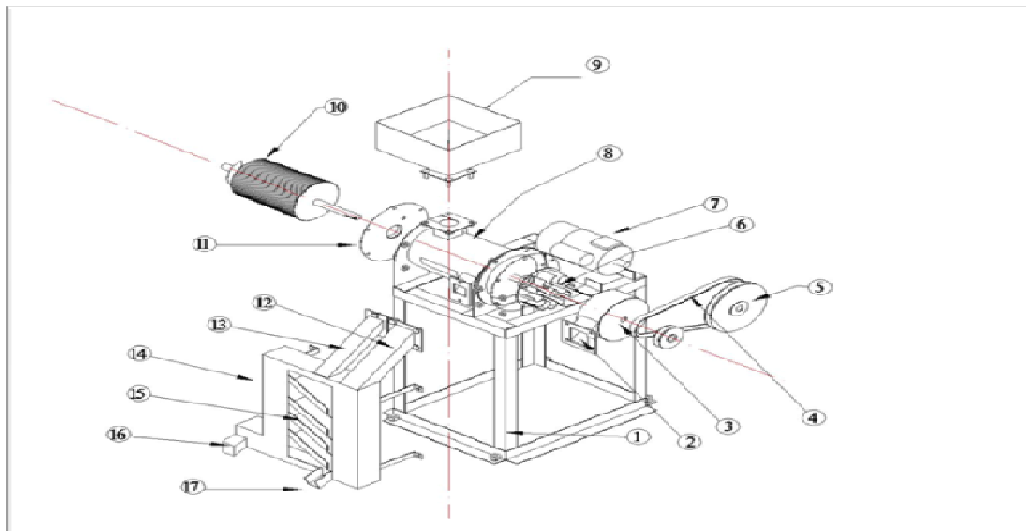


Fig. 1: Acha dehulling machine

1. Frame 2. Air out 3. Fan housing 4. V-belt 5. Electric motor pulley 6. Fan 7. Electric motor 8. Concave housing 9. Hopper 10. Dehulling cylinder 11. Dehulling chamber cover 12. Air chamber 13. Dehulled 14. Cleaning chamber 15. Cleaning fins 16. Chaff Outlet 17. Grain Outlet

$\mu_i = \tan \Phi_i =$ coefficient of internal friction = 0.456 (was determined experimentally)

$\Phi_i =$ angle of internal friction of acha

The method as explained by Stepanoff (1969) and cited by Irtwange (2000) was used to determine the coefficient of internal friction.

B. Size of opening (minimum non-arching dimension)

The size of controlling dimension of the discharge opening should be greater than minimum non-arching dimension for the acha grains. This is expressed in equation 2 as reported by Stepanoff (1969).

$$s = \frac{2\tau_o(1 + \sin \Phi_i)}{k\rho} \quad (2)$$

Where,

$s =$ type of opening size (cm)

$\rho =$ bulk density of the material (g/cm^3)

$\tau_o =$ initial shear stress of the sample (g/cm^2)

$\Phi_i =$ angle of internal friction of acha

$k =$ constant depending upon the type of opening

C. The hopper side slope

The slope angle of the side wall of the hopper should be greater than the angle of internal friction for easy flow of material and was calculated using equation 3 as provided by Stepanoff (1969).

$$\beta \leq 45^\circ + \frac{\Phi_i}{2} \quad (3)$$

Where,

$\beta =$ slope angle of hopper side wall ($^\circ$)

$\Phi_i =$ angle of internal friction of acha

Design of belt

Velocity ratio of the V-belt drive was calculated using equation 4 as provided by Khurmi and Gupta (2005).

$$\frac{N_c}{N_m} = \frac{D_m}{D_c} \quad (4)$$

Where,

$N_c =$ speed of cylinder (rpm)

$N_m =$ speed of the electric motor (rpm)

$D_c =$ diameter of cylinder pulley (mm)

$D_m =$ diameter of electric motor pulley (mm)

Velocity of belt

The peripheral velocity of the belt on the electric motor is given by equation 5 as provided by Khurmi and Gupta (2005).

$$V_m = \frac{\pi D_m N_m}{60} \text{ m/s} \quad (5)$$

Determination of centre distance of electric motor and cylinder pulleys

The centre distance (x) was obtained from equation 6.

$$x = \text{maximum } (2r_1; 3r_2 + r_1) \quad (6)$$

Where,

$x =$ centre distance between pulleys (mm)

$r_1 =$ radius of big pulley (mm)

$r_2 =$ radius of small pulley (mm)

From equation 6, two centre distances were obtained, but the larger was chosen.

The centre distance should not be greater than three (3) times the sum of the sheave diameters or less than the diameter of the larger pulley (Khurmi and Gupta, 2005). The calculated centre distance (x) satisfies these two statements.

Length of the open belt drive for the electric motor and cylinder (Eq. 7) given by Khurmi and Gupta (2005).

$$L = \frac{\pi}{2}(D_1 + D_2) + 2x + \frac{(D_1 - D_2)^2}{4x} \quad (7)$$

Where,

L = total length of belt (mm)
 x = centre distance between pulleys (mm)
 D₁ = diameter of larger pulley (mm)
 D₂ = diameter of smaller pulley mm
 Determination of angle of rap or contact for the electric motor and cylinder
 For an open V-belt drive system as given by Khurmi and Gupta (2005) is

$$\sin \alpha = \frac{r_1 - r_2}{x} \quad (8)$$

Angle of contact of cylinder (smaller) pulley

$$\theta^\circ = 180^\circ - 2\alpha \quad (9)$$

Ratio of driving tensions for V-belt drive was calculated using equation 10 by Khurmi and Gupta (2005),

$$2.3 \log \frac{T_1}{T_2} = \mu \theta \operatorname{cosec} \beta \quad (10)$$

Where,

T₁ = tension in the tight side of the belt (N)
 T₂ = tension in the slack side of the belt (N)
 μ = coefficient of friction between the belt and sides of pulley groove
 θ = angle of contact (°)
 β = half the groove angle of pulley (°)

For a belt speed of more than 10 m/s, the effect of centrifugal tension was considered in the design (Khurmi and Gupta, 2005). The calculated belt speed was 12.5m/s which implied that centrifugal tension was considered for the design. Centrifugal tension is given by equation 11 as,

$$T_c = mv^2 \quad (11)$$

Where,

T_c = centrifugal tension (N)
 m = mass of belt per unit length (kg/m)
 v = linear velocity of belt (m/s)

Power for dehulling acha

Three main powers needed by the major components were considered as the total power required by the dehulling machine. The powers considered are given in equation 12 as,

$$P_T = P_C + P_f + P_D \quad (12)$$

Where,

P_T = total power required (W)
 P_C = power to drive the cylinder (W)
 P_f = power to drive the fan (W)
 P_D = power to cause acha to dehull (W)

A. Power to drive the cylinder (P_C) as given by Khurmi and Gupta (2005)

$$P_C = T_{qc} \omega_c \quad (13)$$

$$T_{qc} = W_c R_c \quad (14)$$

Where,

T_{qc} = torque of cylinder (Nm)
 W_c = weight of cylinder (N)
 ω_c = angular velocity of cylinder (rad/s)
 D_c = diameter of cylinder (m)
 R_c = radius of cylinder (m)

$$\omega_c = \frac{2\pi N_c}{60} \quad (15)$$

N_c = speed of the cylinder (rpm)

B. Power to drive fan (P_f)

Power required to drive fan depends on the conveying air volumetric flow rate and total system pressure drop. The power requirement was computed from equation 16 given by Srivastava *et al.*, (1996) as

$$P_f = \frac{\Delta p Q_v}{\eta_b} \quad (16)$$

Where,

P_f = fan power (W)
 Δp = total system pressure drop (Pa)
 Q_v = Q_A = volumetric flow rate of air (m³/s)

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η_b = fan efficiency (0.5 to 0.7)

Q_v = area of cleaning chamber X velocity of air (m^3/s)

V_A = velocity of air, 2.5 m/s (chosen) this value is in between the terminal velocity of dehulled acha and chaff for effective separation.

Airflow resistance given by Shedd (1953) and cited by ASAE (2007) in equation 17 is

$$\frac{\Delta p}{L} = \frac{aQ^2}{\log_e(1+bQ)} \quad (17)$$

Where,

Δp = pressure drop (Pa); L = depth (m); a = constant for particular grain ($Pa.s^2/m^3$); Q = airflow ($m^3/s-m^2$); b = constant for particular grain ($m^2.s/m^3$)

C. Power to dehull acha as given by Khurmi and Gupta (2005)

$$P_D = T_D \omega_c \quad (18)$$

$$T_D = \frac{\pi(D_c)^3 \tau}{16} \quad (19)$$

Where,

P_D = power required to dehull acha (W)

T_D = torque of cylinder in relation with the shear stress of acha (Nm)

ω_c = angular velocity of cylinder (rad/s)

D_c = diameter of cylinder (m)

τ = shear stress of acha (Nm^{-2})

Design of fan

The separation of the acha grains from chaff is possible because of the differences in the aerodynamic properties of the two materials particularly their terminal velocities. After leaving the dehulling chamber, the particles' (grains and chaff) motion in air begins for cleaning. The fundamental forces involved as particles moved in the air are the weight of the particles

and the aerodynamic drag. The aerodynamic drag force is a function of the relative velocity of the particle with the air (V_r), the density of air (ρ) and the size of the particle as expressed by its frontal area (A_f). The drag force is related to properties of the particles and of the fluid by equation 20 (Mohsenin, 1986).

$$F_D = M.g = \frac{1}{2} C_d \rho_A A_f V_r^2 \quad (20)$$

Where,

F_D = drag force (N)

M = mass of object (kg)

g = gravitational acceleration (m/s^2)

C_d = drag coefficient

ρ_A = density of air (Kg/m^3)

A_f = frontal area (m^2)

V_r = relative velocity of air with seed or terminal velocity of seed (m/s)

The projected (frontal) area of acha was calculated by the formula given by Bilanski (1962) and reported by Mohsenin (1986) for seed grains of ellipse shape. The dimensions of the acha seeds were measured in three directions using a digital vernier caliper gauge (± 0.01 mm). The major diameter (M_d) was the length of the seed, the intermediate diameter (I_d) was the width, and the minor diameter (N_d) was the thickness of the seed.

$$A_f = \frac{\pi M_d I_d}{4} \quad (21)$$

Where,

A_f = projected (frontal) area (mm^2)

M_d = major diameter of acha (mm)

I_d = intermediate diameter of acha (mm)

The drag coefficient (C_d) was calculated from the expression in equation 20

Air volume

The requirement for air discharge through a fan can easily be estimated on the basis of velocity of air required for cleaning (V_A), depth (D) and width (W) of air stream in the cleaning chamber which is required by the air. Using equation 22 to calculate the actual airflow,

$$Q_A = V_A DW \quad (22) \quad \text{limiting value.}$$

Where,
 Q_A = Actual airflow (m^3/s)

$$\frac{T_m}{J} = \frac{G \theta}{L} \quad (24)$$

According to Joshi (1981), the efficiency of the fan is considered to be 30% above the calculated value to account for losses.

Where,
 θ = Torsional deflection or angle of twist (radians)

T_m = Twisting moment or torque on the shaft (N-mm)

J = Polar moment of inertia of the cross-sectional area about the axis of rotation

Design of shaft for cylinder and fan

The main shaft transmits power from the electric motor to the cylinder and fan. Therefore, the shaft was designed based on strength and rigidity. According to the American Society of Mechanical Engineers (ASME) code for the design of transmission shafts, the maximum permissible working stresses (σ_u) in tension or compression may be taken as 84 MPa for shafts with allowance for keyways and maximum permissible shear stress (τ_u) may be taken as 42 MPa (N/mm^2) for shafts with allowance for keyways.

$$= \frac{\pi}{32} d^4 (mm^4)$$

G = Modulus of rigidity for the shaft (GPa)

L = Length of shaft (mm)

A. Strength criterion

Since mild steel was used for the shaft, maximum shear stress theory was used for the design of the shaft diameter and it is given in equation 23 by Khurmi and Gupta (2005) as,

Design of bearing

The normal procedure of selecting a bearing for a particular application is to determine the equivalent dynamic load (W), and the fatigue or rating life of the bearing (L) and calculate the basic load rating or dynamic carrying capacity (C), which is used in choosing the bearing. Khurmi and Gupta (2005) gave the following as the relationship (Eq. 25),

$$d^3 = \frac{16}{\pi \tau} \sqrt{(K_m M)^2 + (K_t T)^2} \quad (23)$$

$$L = \left(\frac{C}{W}\right)^k 10^6 \text{ revolutions} \quad (25)$$

Where,
 τ = Allowable shear stress (MPa)
 K_m = Combined shock and fatigue factor for bending
 K_t = Combined shock and fatigue factor for torsion
 M = Bending moment (N-mm)
 T = Twisting moment (N-mm)
 d = diameter of solid shaft (mm)

Where,
 L = fatigue or rating life of the bearing (hr)
 W = equivalent dynamic load (kN)
 C = basic load rating or dynamic carrying capacity (kN)
 $k = 3$, for ball bearing

B. Design for torsional rigidity

The design of shaft for torsional rigidity was based on the permissible angle of twist. For line shafts or transmission shafts, deflections of 2.5 to 3 degrees per meter length may be used as

Design of dehuller frame

The frame supports the total weight of the dehuller components. These weights include: Weights of hopper, concave, cylinder, electric motor, shaft, fan and fan casing, fan/cylinder and electric motor pulleys and cleaning chamber.

Rigidity and strength were the most important criteria considered for the frame design. The frame design involved knowing the kind of loading that it would be subjected to, selecting the correct steel sections for the frame construction and analyzing all possible forms of failure that could occur on the frame to ensure safety of the design.

The frame was designed based on Euler's column assumptions. According to Euler's theory, the crippling or buckling load (W_{cr}) under various end conditions is represented by a general equation (26),

$$W_{cr} = \frac{C\pi^2 EI}{l^2} = \frac{C\pi^2 E Ak^2}{l^2} \quad (26)$$

Where,

W_{cr} = Crippling or buckling load (N)

C = Constant, representing the end conditions of the column or end fixity coefficient

E = Modulus of elasticity or Young's modulus for the material of the column (N/mm^2)

I = $A.k^2$ = Moment of inertia (mm^4)

A = Area of cross-section (mm^2)

k = Least radius of gyration of the cross-section (mm)

l = length of the column (mm)

RESULTS AND DISCUSSION

Some basic parameters used for the design of the acha dehuller based on design considerations, preliminary work and analysis of information are presented in Table 1. Hopper flow characteristics determined are presented in Table 2.

The values for the hopper flow characteristics are presented in Table 2. Flow properties have relevance in the design and management of gravity and forced flow equipment. The angle of repose was determined as 24.50° . The angle of repose was used to determine hopper inclination. The value for the coefficient of internal friction for acha was calculated as 0.456 and this agreed with Stepanoff (1969) who reported that values for coefficient of internal friction should lie between 0.4 – 1.2 for indication of flowability.

The coefficient of mobility (m_i) calculated from the coefficient of internal friction of the acha is shown in Table 2. The coefficient of mobility obtained was 0.414. The coefficient of mobility represents fluidity or freedom of motion of a substance and should be >0.3 . Since m_i for acha is greater than 0.3, this means that acha is free flowing grain.

Table 1: Some parameters considered for the design of the Acha dehuller

S/No.	Parameters	Values for machines	
		Existing	Improved
1	length of threshing cylinder (lc) (mm)	220	220
2	cylinder speed (nc) (rpm)	934	2800
3	fan speed (rpm)	934	2800
4	concave clearance (mm)	20	10
5	concave length (mm)	260	260
6	hopper side slope	67°	57°
7	Cylinder type	Rod Material	Abrasive Material
8	terminal velocity of acha (v_i) (ms^{-1})	-	3.96
9	terminal velocity of acha chaff (ms^{-1})	-	1.9
10	velocity of air require for cleaning (v_a) (ms^{-1})	-	2.5

Table 2: Hopper flow characteristics of Acha

S/No.	Parameter	Value
1	Angle of repose	24.5°
2	Coefficient of internal friction μ_i	0.456
3	Coefficient of mobility m_i	0.414
4	Size of hopper opening	10 mm
5	Hopper side wall slope	57°

From Table 2, the calculated value of opening dimension for free flow of acha through hopper (non-arching) was 10 mm. The term ‘arching’ is used to designate the cause of stoppage of flow of bulk foods from bunker silos and hopper. The value for the hopper side slope was calculated as 57°. In handling gravity flow of granular materials through chutes, the relationship between the volume of flow, stream thickness, and chute inclination angle is an important design criterion. Undesirable build up (clogging) will also be avoided if the chute is designed with correct inclination angle to handle a given flow rate.

CONCLUSION

An improvement on an existing acha dehulling machine was carried out. The existing acha dehuller was first tested to determine its performance and observe its limitation, which also served as the control. Major component parts were re-designed using relevant engineering design principles to develop an improved acha dehuller. The acha dehuller comprised basically of the hopper, dehulling unit, fan unit, cleaning unit and the outlets.

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