World Journal of Advanced Engineering Technology and Sciences
eISSN: 2582-8266
Cross Ref DOI: 10.30574/wjaets
Journal homepage: https://wjaets.com/

(Research Article)

Check for updates

Design parameters of an acha thresher

Theresa Kangyang Kaankuka [1,*], Isaac Nathaniel Itodo [1] and Ambo Mamai Ezekiel [2]

[1] *Department of Agricultural and Environmental Engineering, Joseph Sarwuan Tarka University (Formerly University of Agriculture), P. M. B. 2373, Makurdi, Nigeria.*
[2] *Department of Agricultural Engineering, Federal University, Wukari, Nigeria.*

World Journal of Advanced Engineering Technology and Sciences, 2025, 15(02), 930-942

Publication history: Received on 26 March 2025; revised on 05 May 2025; accepted on 07 May 2025

Article DOI: https://doi.org/10.30574/wjaets.2025.15.2.0556

Abstract

Acha (Digitaria exilis) also known as fonio, is a crop with significant food and nutritional value primarily grown in Nigeria and other West Africa countries, where it's threshing and cleaning processes remains problematic. In an effort to alleviate the problems associated with the traditional method of acha threshing and cleaning which are uneconomical, time-consuming and labour-intensive, an acha threshing machine was designed. The thresher consists of five principal units: feeding, threshing, vibrating/oscillating, cleaning, and power units. The feeding unit is shaped as a trapezoidal hopper with dimensions 400×210 mm (larger rectangle) and 338×210 mm (smaller rectangle), designed to hold 5 kg of acha panicles. The threshing unit features a drum with a diameter of 435 mm and a length of 740 mm, equipped with 44 threshing spikes (20 mm diameter, 92.5 mm length) suitable for acha's fragile grains to minimize damage. The separator has dimensions of 725 mm by 595 mm and uses a sieve with 4 mm holes. The power required for the thresher is 4.0 kW which operates the threshing drum at 10.212 m/s and drives a reciprocating sieve for cleaning the threshed acha. The design is suitable for small-scale farmers involved in acha production in Nigeria, offering a significant increase in labour productivity (8.4 times that of manual threshing). Evaluating the developed acha thresher demonstrated its ability to efficiently thresh acha and separate the chaff/straw from it, with a reduction in grain breakage compared to traditional methods which presents a feasible solution for mechanizing acha threshing.

Keywords: Acha Threshing; Separation; Cleaning; Feeding Unit; Threshing Unit; Separator

1. Introduction

Acha is produced in large quantity in Nigeria and has a great market potential for domestic and industrial uses. In the year 2002, a total of 347, 330 hectares of land were devoted to acha production in Africa with Nigeria alone providing almost half of the area (Ayo *et al.*, 2010). In Nigeria, acha is grown in commercial quantities in some States such as Bauchi, Kaduna, Kebbi, Plateau, Nassarawa, Niger, Gombe and FCT with Plateau State being the highest producer with an estimated production of 20,000 ton per annum (Kaankuka *et al.*, 2014).

Acha can produce yield in areas where other crops may fail, because of its draught resistance qualities. It is the crop of choice for dry region and areas with unreliable rainfall. Despite the food and nutritional value of acha, it does not attract large international and domestic demand. Inspite of the popularity of acha as a food item for millions in Africa, and the availability of informative reports of its nutritional composition (Philip and Itodo, 2006; Jideani and Jideani, 2011), its threshing and cleaning have remained serious problems to the farmers. The main problem associated with the threshing and cleaning of acha in Nigeria is the use of traditional methods of seed separation from stalk which are uneconomical, time consuming, injurious to the finger and fatigue associated. Manual threshing of acha is classified as heavy work load in terms of energy expenditure. Not only are these manual threshing techniques time and human energy consuming but

* Corresponding author: Theresa Kangyang Kaankuka

also damaging to the crop's kernels. The time required for threshing depends on variety, moisture content of the grain, and the method of threshing (Sale *et al.*, 2016).

Traditionally, threshing is done by hand; this involves beating the acha sheaves with a stick which is very laborious. Also, traditional threshing on non-paved soil results in high rates of sand contamination, whereas threshing on a paved surface is somewhat better. Threshing on a canvas is considered best when threshing manually. Philip and Itodo (2006) highlighted the tedium and drudgery associated with the manual methods which are inefficient and labor-intensive. They also opined that acha production have to be stimulated by initiating research activities to address its mechanization constraints.

There is no known developed machine for threshing acha in the market. This study therefore, focuses on the design parameters of an acha thresher for its construction to address the challenges associated with its manual threshing methods, which are uneconomical, time-consuming, and labour-intensive. The acha thresher is expected to be simple to operate, maintain and affordable suitable for low-income farmers, utilizing locally available materials.

2. Materials and methods

2.1. Material Selection

Mild steel was selected for the production of machine members such as cylinder, sides, cylinder cover of cylinder, separator, chaff outlet, frame, feed hopper, grain delivery chute among others. A round mild steel rod was used for the shafts and a mild steel cylindrical rods was used for the cylinder beaters.

2.2. Design Considerations

The acha thresher was designed based on the following considerations.

- The availability of materials locally to reduce cost of production and maintenance.
- The materials for the construction of the various component parts were selected on the basis strength and durability.

2.3. Design Analysis

The acha thresher consists of five principal units, viz.: the feeding, threshing, vibrating/oscillating, cleaning and power units.

2.3.1. Feeding unit

Feed hopper

An inclined (30o) feeding hopper was provided to feed the material to the threshing unit. The feeding hopper was designed trapezoidal in shape with rectangular feeding ends, and has a capacity to contain 5 kg of acha panicle. Equation 1 was used to obtain the volume of the hopper.

$$V_h = \frac{M_{ap}}{\rho_{ap}} \qquad \text{……………..(1)}$$

Where;

- V_h = Volume of hopper (m^3),
- M_{ap} = Mass of acha panicle = 5 kg (preliminary study),
- ρ_{ap} = Density of acha panicle = 198.66 kg/m^3 (Experimental).

The dimensions of the hopper were chosen based on proportionality and aesthetics. The dimension of the larger rectangle was 400×210 mm and the smaller rectangle was 338×210 mm. The length of the hopper was calculated using Equation 2 given for the volume of a trapezoidal structure.

$$V_h = \frac{1}{2}(a + b)hL \text{ ………..(2)}$$

Where;

- V_h = Volume hopper (m^3),
- a and b = lengths of the parallel sides of the trapezoid (m),
- h = height of the trapezoid (m),
- L = height of the prism (the distance between the two trapezoidal bases) (m).

2.4. Threshing unit

The threshing drum consists of circular plates, rectangular bars, beaters and straw walker, all made of mild steel.

2.4.1. Weight of threshing drum

Weight of the circular plates

Two circular plates of diameter 230 mm and thickness 4 mm were designed to form the point of attachment of the rectangular bars onto which the threshing spikes and straw walker were attached. The circular plates were drill at the centre to give an opening of 30 mm diameter to accommodate the main threshing shaft. The weight of the circular plates was determined using Equation 3 (Hannah and Hillier, 1999).

$$W_c = 2 \times \rho_c V_c g = 2 \times \rho_c[\pi(R_c{}^2 - r_c{}^2)t]g \ \ldots\ldots\ldots\ldots\ldots \qquad (3)$$

Where;

- W_c = Weight of the circular plates (N),
- ρ_c = Density of the circular plates (mild steel = 7850 kg/m^3),
- g = Acceleration due to gravity (9.81 m/s^2),
- V_c = Volume of the circular plates material (m^3),
- R_c = Radius of the circular plates (0.115 m),
- r_c = Radius of the hole on circular plates (0.015 m),
- t = Thickness of the circular plates (0.004 m).

2.5. Weight of the rectangular bars

Four rectangular bars of mild steel of length 740 mm, width 30 mm and thickness 10 mm were designed for attachment of the spikes and straw walker. Equation 4 was used to determine the weight of the rectangular bars (Hannah and Hillier, 1999).

$$W_b = 4 \times \rho_b V_b g \qquad \ldots\ldots\ldots\ldots\ldots(4)$$

Where;

- W_b = Weight of the rectangular bars (N),
- ρ_b = Density of the rectangular bars (mild steel = 7850 kg/m^3),
- g = Acceleration due to gravity (9.81 m/s^2),
- V_b = Volume of the rectangular bars material (m^3).

2.6. Weight of the threshing spikes

The forty-four (44) threshing spikes are cylindrical rods of 20 mm diameter and 92.5 mm length. Equation 5 was used to determine the weight of the threshing spikes (Hannah and Hillier, 1999).

$$W_s = 44 \times \rho_s V_s g \qquad \ldots\ldots\ldots\ldots\ldots(5)$$

Where;

- W_s = Weight of the threshing spikes (N),
- ρ_s = Density of the threshing spikes (mild steel = 7850 kg/m^3),
- g = Acceleration due to gravity (9.81 m/s^2),
- V_s = Volume of the threshing spikes (m^3).

2.7. Weight of the straw walker

The straw walker was made of four rectangular mild steel plate of length 148 mm, width 92.5 mm and thickness 3 mm attached along its length at the end of the threshing drum to move the threshed straw out of the threshing chamber. Equation 6 was used to determine the weight of the straw walker (Hannah and Hillier, 1999).

$$W_w = 4 \times \rho_w V_w g \qquad \text{.......................(6)}$$

Where;

- W_w = Weight of the straw walker (N),
- ρ_w = Density of the straw walker (mild steel = 7850 kg/m^3),
- g = Acceleration due to gravity (9.81 m/s^2),
- V_w = Volume of the straw walker (m^3).

Therefore, the total weight of the threshing drum was;

$$W_T = W_c + W_b + W_s + W_w \quad \text{..................(7)}$$

$$W_T = 204.66 \text{ N}$$

2.8. Belt drive to threshing drum

Power is transmitted from the electric motor to the threshing drum through a V-belt placed vertically. In order to ensure firm grip between the belt and pulley, the required length of the belt was determined using Equation 8 (Khurmi and Gupta, 2008).

$$L = 2X + \frac{\pi}{2}(d_M + d_D) + \frac{(d_D - d_M)^2}{4X} \quad \text{..........................(8)}$$

Where;

- L = Length of belt, m
- X = Distance between pulley centres = 300 mm,
- d_M = Driver (prime mover) pulley diameter = 75 mm,
- d_D = Driven (drum) pulley diameter = 300 mm.

2.9. Threshing drum shaft design

The shaft was designed to carry the weight of the threshing drum and the drive pulley force. It was subjected to bending and twisting moment**s**. The shaft diameter was designed on the basis of strength using Equation 9.

$$d^3 = \frac{16}{\pi Ss}\sqrt{(K_b M_b)^2 + (K_t M_t)^2} \qquad \text{..................(9)}$$

Where;

- d = shaft diameter, mm
- S_S = Maximum permissible shear stress (N/mm^2),
- K_b = Combined shock and fatigue factor applied to bending moment,
- K_t = Combined shock and fatigue factor applied to torsional moment,
- M_b = Maximum bending moment, Nmm,
- M_t = Torsional moment, Nmm.

The torque (M_t) of threshing drum was calculated from Equation 10 (Khurmi and Gupta, 2008).

$$M_t = (T_1 - T_2) R \quad \text{..(10)}$$

Where;

- M_t = Torsional moment of threshing drum pulley, Nm,
- T_1= Tension in tight side of the belt on the threshing drum pulley, N,
- T_2 = Tension in loose side of the belt on the threshing drum pulley = 58.53 N,
- R = Radius of threshing drum = 207.5 mm = 0.2075 m.

The tension on the tight side of the belt (T_1) was calculated using Equation 11 (Khurmi and Gupta, 2008).

$$T_1 = T - T_C \text{................(11)}$$

Where;

- T = maximum tension in the belt (N),
- T_C = Centrifugal tension (N).

The maximum tension of the belt (T) was calculated using Equation 12 (Khurmi and Gupta, 2008).

$$T = \sigma a \quad = \sigma bt \qquad \text{...........................(12)}$$

Where;

- T = maximum tension (N),
- σ = allowable stress (MPa),
- a = area of belt (mm^2),
- b = width of belt (mm),
- t = thickness of belt (mm).

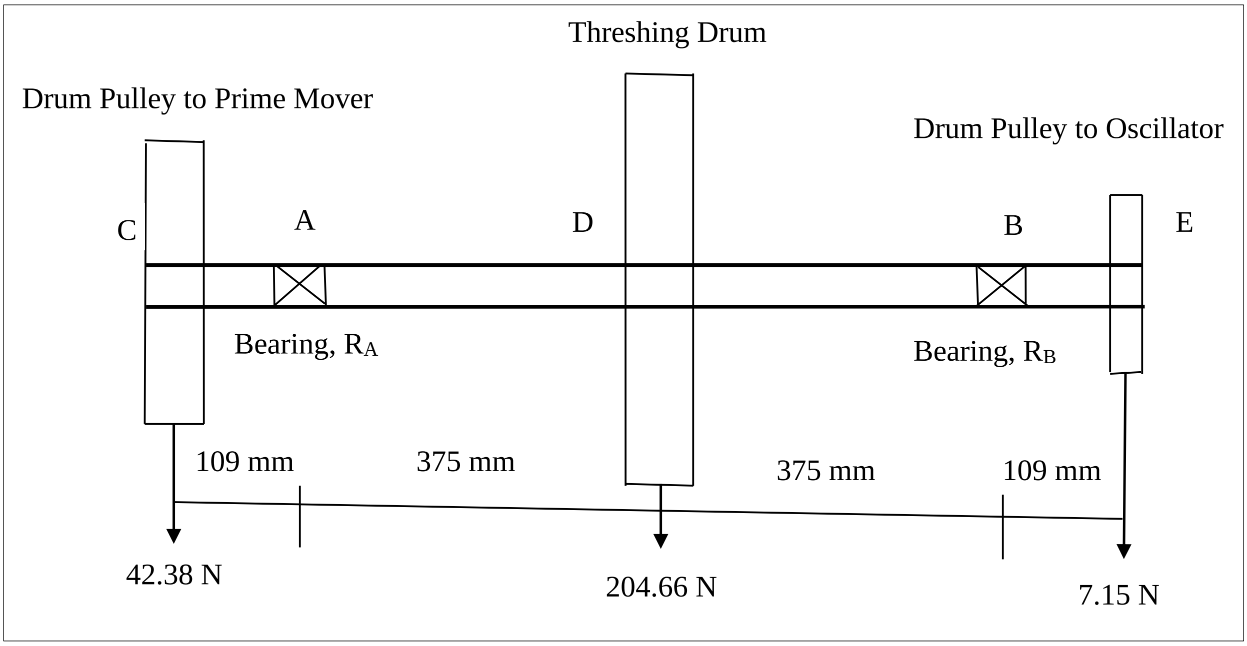

Figure 1 Bending Loads of the Threshing Drum Shaft

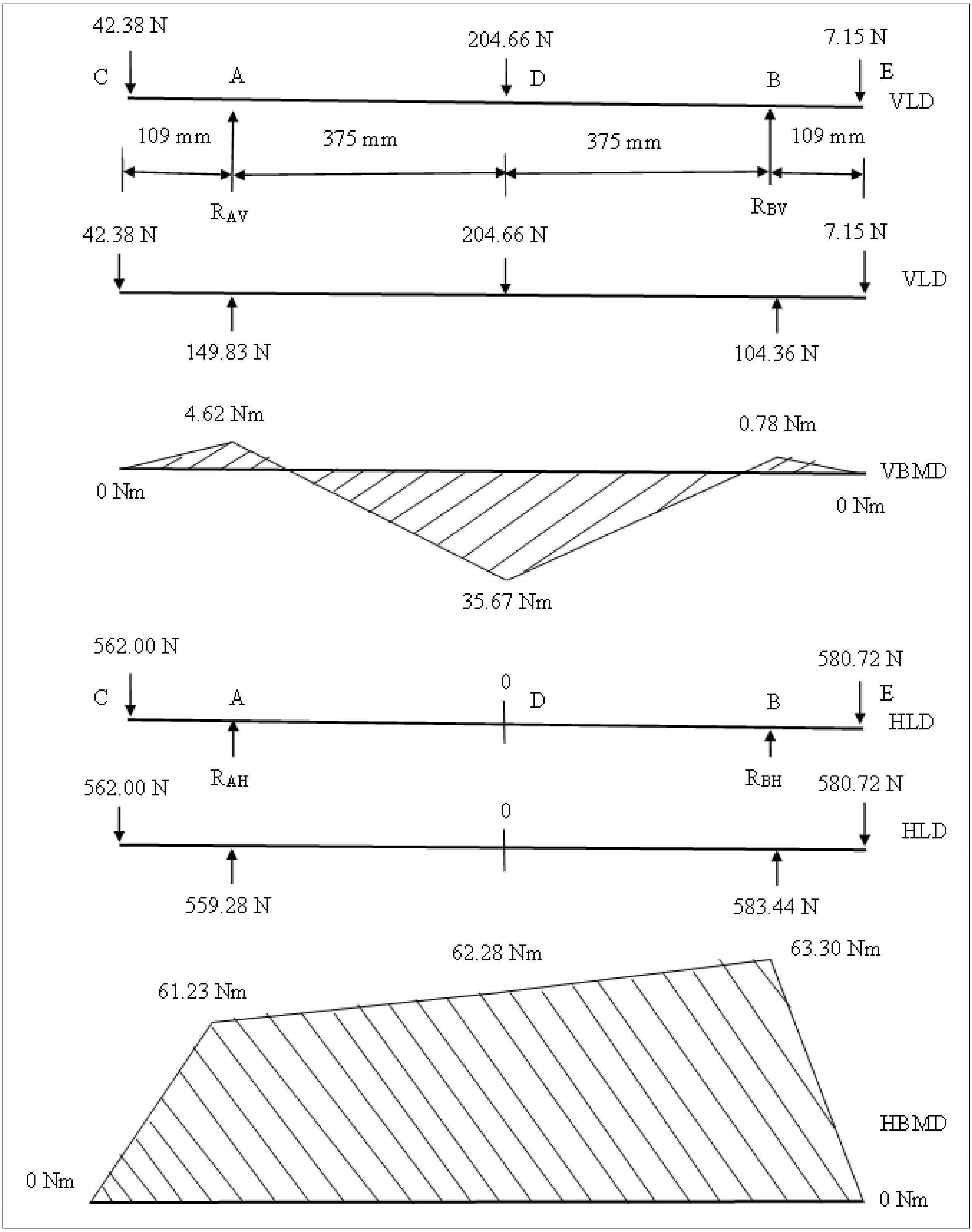

Figure 2 Bending Moment Diagrams for Vertical and Horizontal Loading of Threshing Drum Shaft

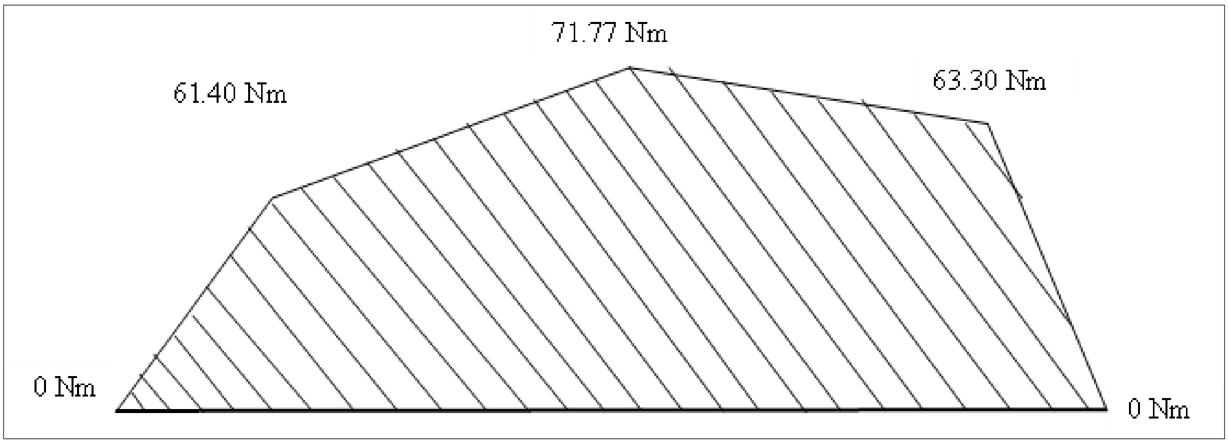

Figure 3 Combined Bending Moment Diagram for Vertical and Horizontal Loading of Threshing Drum Shaft

From the calculations, maximum bending moment will occur at D. M_{bmax} = 71.77 Nm

The Centrifugal tension (T_C) was calculated using Equation 13 (Khurmi and Gupta, 2008).

$$T_C = mv^2 \qquad \text{……………..(13)}$$

Where;

- m = Mass of belt per meter length (kg/m),
- v = Peripheral Velocity of the reciprocator pulley belt (m/s).

The Peripheral Velocity (V) of the reciprocator pulley belt was calculated using Equation 14 (Khurmi and Gupta, 2008).

$$V = \frac{\pi D_R N_R}{60} \qquad (14)$$

Where;

- V = Velocity of belt (m/s),
- D_R = Diameter of reciprocator pulley (mm),
- N_R = Speed of reciprocator (rpm).

The Tension (T_2) in loose side of the belt was calculated using Equation 15 (Khurmi and Gupta, 2008).

$$2.3 \text{ Log}\left(\frac{T_1}{T_2}\right) = \mu\theta\text{cosec}\beta \ (15)$$

Where;

- T_1 = Tension in tight side (N),
- T_2 = Tension in loose side (N),
- μ = Coefficient of friction between belt and pulley,
- θ = Angle of contact (radians),
- β = Semi-groove angle of the pulley (degrees).

2.9.1. Vibrating/oscillating unit

The reciprocator is meant to convert rotary motion to reciprocating motion of the sieve (Sharma and Aggarwal, 2006). Abdel *et al*. (2007) suggested a short stroke length for eccentricity. Therefore, the dimensions of the cam radius and thickness were chosen based on proportionality of the load to be carried and were taken as 65 mm and 40 mm respectively for short displacement.

2.10. Ratio of length of connecting rod to crank radius, n

$$n = \frac{L}{r} \quad \text{……………(16)}$$

Where;

- n = Ratio of length of connecting rod to crank radius,
- r = Crank radius (0.065 m),
- L = Length of connecting rod (0.045 m).

2.11. Maximum displacement of the crank

The maximum displacement (X_c) of the crank was calculated using Equation 17 (Khurmi and Gupta, 2008).

$$X_c = r\left(1 - \cos\theta + \frac{\sin^2\theta}{2n}\right) \text{………… (17)}$$

2.12. Maximum angle of the crank

$$a_s = \frac{dv_d}{d\theta} = 0$$

$$a_s = \omega r\left(\cos\theta + \frac{2\cos\ 2\theta}{2n}\right) = 0 \quad \text{………18)}$$

Where;

- ω = Angular velocity (rad/s),
- r = Crank radius (0.065 m),
- θ = Crank angle from dead center (degree).

2.13. Angular Velocity of crank, ω

$$\omega = \frac{2\pi N}{60} \quad \text{……………(19)}$$

Where;

N = Crank speed = 203 rpm.

2.14. Maximum velocity of crank

The maximum velocity of the crank was calculated using Equation 20 (Khurmi and Gupta, 2008).

$$V_{s(max)} = \omega r\left(\sin\theta + \frac{\sin 2\theta}{2n}\right) \quad \text{…………………..(20)}$$

2.14.1. Cleaning unit

The cleaning unit is the component where separation of acha and chaff/straw takes place. The cleaning unit consists of a reciprocating sieve. The sieve is made of 2 mm thick metal sheet and measures 725 mm by 595 mm, perforated with 4 mm drill bit covering surface area of about 45%. The sieve is connected to the reciprocator shaft carrying a cam of 30 mm travel with linkages and a bearing.

2.15. Surface area of the separator

The surface area of the separator was calculated using Equation 21.

$$S_s = L_s W_s \text{……………..(21)}$$

Where;

- S_s = Surface area of the separator (m^2),
- L_s = Length of the separator surface (725 mm = 0.725 m),

- W_s = Width of the separator surface (595 mm = 0.595 m).

2.15.1. Mass of the separator

The mass of the separator was determined from the density of the mild steel used in construction (7850 kg/m3), surface area and thickness of the material.

$$M_s = \rho S_s t \qquad \text{.....................(22)}$$

Where;

- M_s = Mass of the separator (kg),
- ρ = Density of the separator material (mild steel = 7850 kg/m^3),
- S_s = Surface area of the separator (0.431 m^2),
- t = Thickness of the separator material (2 mm = 0.002 m).

The separator has 45% open area. Therefore, the actual mass of the separator was 55% of the total mass calculated.

2.15.2. Power unit

The total power required to drive the acha thresher is the sum of the power to operate the reciprocator, power to turn the unloaded cylinder, power requirement due to air resistance and power to detach acha grains.

2.16. Power required to operate the reciprocator

The power required to operate the reciprocator was determined using Equation 23 (Khurmi and Gupta, 2008).

$$P_1 = (T_1 - T_2)\, V \qquad \text{.....................(23)}$$

Where;

- P_1 = Power required for operating the reciprocator (W),
- T_1 = Tension in tight side of the reciprocator pulley (N),
- T_2 = Tension in loose side of the reciprocator pulley (N),
- V = Velocity of the reciprocator pulley belt (m/s).

2.17. Power required to turn the unloaded cylinder

The power required to turn the unloaded cylinder was calculated using Equation 24 given by Olaoye *et al* (2011).

$$P_2 = \frac{2\pi \times N \times r \times Mc}{60 \times 75}\left(g + \frac{Vt^2}{r}\right) \qquad \text{..........(24)}$$

Where;

- P_2 = Power required to turn the unloaded cylinder (W),
- N = Speed of the threshing cylinder (650 rpm),
- r = Radius of cylinder (m),
- Mc = Mass of threshing cylinder (kg),
- g = Acceleration due to gravity (9.81 m/s^2),
- V_t = Peripheral velocity of the threshing cylinder (10.212 m/s).

2.18. Power requirement due to air resistance

This was calculated using Equation 25 (Ndrika, 1997).

$$P_3 = k_f \times F_r \times V_t^2 \qquad \text{..................(25)}$$

Where;

- P_3 = Power requirement due to air resistance (W),

- k_f = Constant which is equal to 0.06 (Ndrika, 1997),
- F_r = Feed rate of acha panicle (0.417 kg/s),
- V_t = Peripheral velocity of the threshing cylinder (10.212 m/s).

2.19. Power required to detach acha grains

The power required to detach acha grains was determined using Equation 26 (Ndirika, 1997).

$$P_4 = K_b \left[\frac{(V_t F_t)^3}{\rho_w {L_c}^2}\right]^{\frac{1}{2}} \qquad \text{…………(26)}$$

Where;

- P_4 = power required to detach acha grains (W),
- K_b = Slippage factor = 0.4 (Ndrika, 1997),
- V_t = Cylinder velocity (10.212 m/s),
- F_t = Feed rate of acha panicle (0.417 kg/s),
- ρ_w = Bulk density of acha (1112.42 kg/m^3),
- L_c = Concave length (750 mm = 0.75m).

The total power required by the acha thresher was obtained using Equation 27.

$$P = P_1 + P_2 + P_3 + P_4 \qquad \text{……………(27)}$$

Considering transmission and other losses a factor of safety of 1.2 was assumed (Olaoye *et al.*, 2011). Therefore, the design power (P_T) was obtained as;

$$P_T = P \times 1.2 \qquad (28)$$

$$P_T = 3.84702 \text{ kW}$$

A prime mover of 4 kW was selected for the drive.

2.20. Working Principle of the Thresher

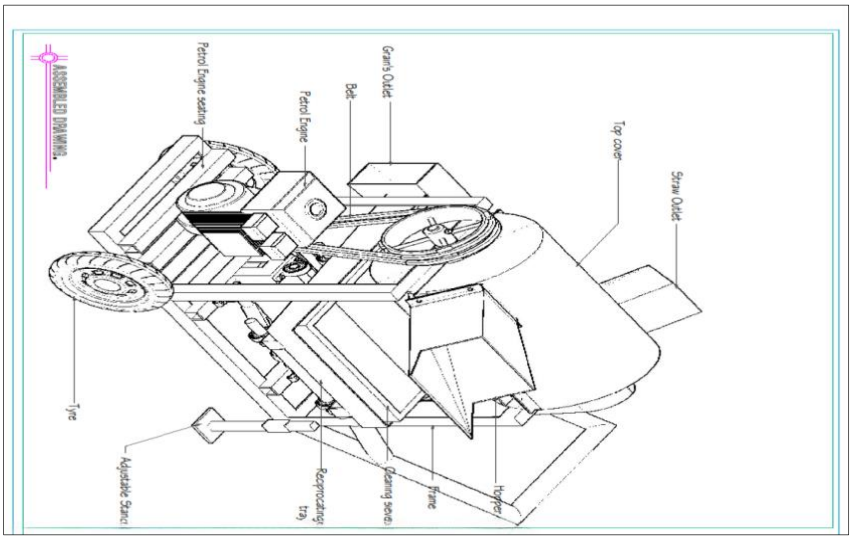

Figure 4 The Designed Acha Thresher

The acha panicle was introduced into the hopper and this flows down into the threshing chamber by gravity flow. The prime mover drives the threshing drum by a shaft connected to its pulley. As the threshing drum rotates, the spike tooth bars beat and rub the unthreshed acha panicles against the concave to affect the removal and release of the acha grains with minimum damage. Also, the cleaning unit derives its power from the threshing drum shaft through belt and pulley system, it reciprocates at the bottom of the concave sieve to separates chaffs and impurities. Figure 4 shows the isometric drawing of the designed acha thresher and Figure 5 shows the exploded drawing of the designed acha thresher.

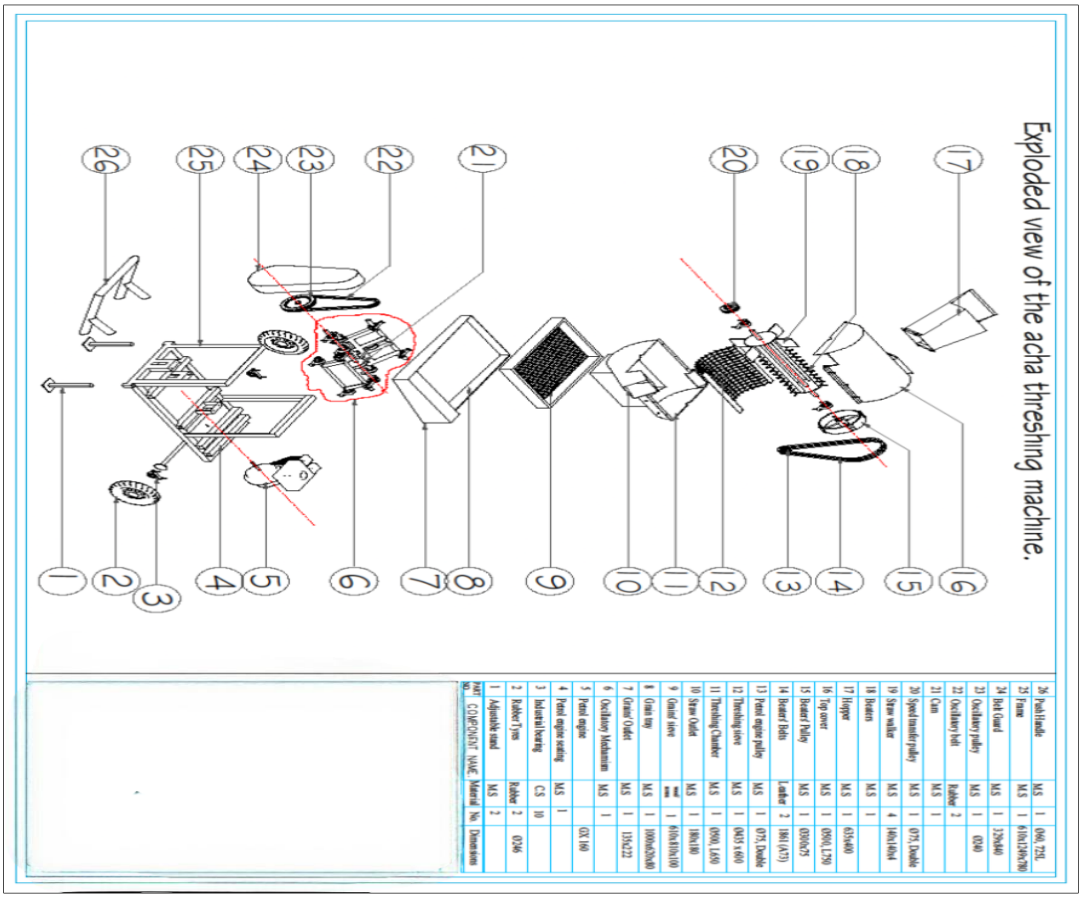

Figure 5 The Exploded Drawing of the Designed Acha Thresher

3. Results and discussion

Table 1 shows results of the designed acha thresher components and their specifications. The components were designed based on basic design formula and acceptable assumptions. The design considered crop factors, machine factors and operational factors. Crop factors were related to the acha crop characteristics on which the mechanical systems of the machine depend on.

The machine factors designed for were the threshing drum type and diameter; and spike shape, size and number. Drum diameter was identified by Huynh *et al*. (1982) as one of the parameters that influence grain separation from the stalks and passage of grains through the concave grate. The operational factors considered in the design were cylinder speed (10.212 m/s), feed rate and method of feeding (hold-on/batch type).

Table 1 results give the design parameters and the dimensions for the components of the acha thresher. These were used to construct the acha thresher which is capable of threshing acha and separating the chaffs from grains. The developed thresher was operated at a speed of 10.21 m/s by a prime mover of 4 kW power rating and this effectively threshed acha. The spike-tooth was chosen and used for the construction of the cylinder because it is typically best, due to its simple design, high threshing efficiency, and suitability for threshing small grains.

Results of specification for the acha thresher components obtained from the design calculations, which resulted in its construction showed that the mechanical process of acha threshing is a suitable method for separating acha grains from its panicles because of its effectiveness, both for small and medium scale farmers. In addition, from the output of the machine, it will be concluded that further design and installation of a commercial plant is viable.

Table 1 Some Design Values Used for the Acha Thresher

S/no	Designed parameters	Specifications
1	Volume of hopper (V_T)	25,170 mm^3
2	Weight of threshing drum pulley	42.38 N
3	Threshing drum speed	10.212 m/s
4	Threshing drum belt length	1231.313 mm
5	Weight of threshing drum	204.66 N
6	Length of threshing drum	740 mm
7	Diameter of threshing drum	435 mm
8	Threshing drum shaft diameter	30 mm
9	Threshing drum shaft length	1040 mm
10	No. of Spikes on threshing cylinder	44
11	Concave clearance	10 mm
12	Concave length	750 mm
13	Concave screen diameter	455 mm
14	Sieve hole size	4 mm
15	Cam radius	65 mm
16	Cam thickness	40 mm
17	Separator length×width	725×595 mm,
18	Machine length×width×height	1864×1439×1427 mm
19	Power requirement	4.0 kW

4. Conclusion

Design parameters of the principal units for the development of the acha thresher have been provided to facilitate its construction. These principal units are; feeding, threshing, vibrating/oscillating, cleaning, and power units. The acha thresher of is a way forward to encourage acha producers increase their production, it will also have the advantages of being simple to operate, easy to main and affordable to the peasant farmer. Threshing with the machine will increase quantity and quality of threshed acha, save time and energy over the normal manual way of threshing.

Compliance with ethical standards

Disclosure of conflict of interest

No conflict of interest to be disclosed.

References

[1] Abdel, T.M., Baiomy, M.A. and El-Khawaga, S. (2007). Design and Fabrication of Local Rice Separating Machine. *Journal of Agricultural Science Mansoura University*, 32(7): 5355-5368.

[2] Ayo, J.A., Ikuomola D.S., Sanni, T.A., Esan, Y.O., Ayo, V.A. and Ajayi, G. (2010). Evaluation of Nutritional Quality of Soybean-Acha Composite Biscuits. *Nigerian Food Journal*, 28(2): 132-138.

[3] Hannah, J. and Hillier, M.J. (1999). Applied Mechanics, Third Edition. Pearson Education Limited, Harlow, England. 445 pp.

[4] Huynh, V. M., Powell, T. and Siddal, J. N. (1982). Threshing and separating process – A mathematical model. *Transaction, American Society of Agricultural Engineers,* 25(1):62-73

[5] Jideani, I.A. and Jideani, V.A. (2011). Developments on the Cereal Grains *Digitaria exilis* (acha) and *Digitaria iburua* (iburu). *Journal of Food Science Technology*, 48(3): 251-259.

[6] Kaankuka, T.K., Itodo, I.N. and Obetta, S.E. (2014). Effect of Cylinder Type and Speed on Acha (*Digitaria exilis*) Dehulling*. International Journal of Applied Engineering and Technology*, 4(2): 125-131.

[7] Khurmi, R.S. and Gupta, J.K. (2008). A Textbook of Machine Design, 14th Edition. New Delhi: Eurasia Publishing House (PVT) Limited, Ram Nagar, India, 1230 pp.

[8] Ndirika, V.I.O. (1997). Modeling the Performance of Selected Stationery Grain Threshers. Unpublished PhD Thesis, Department of Agricultural Engineering, Ahmadu Bello University, Zaria, Nigeria. pp 27 - 91.

[9] Olaoye, J.O., Oni, K.C. and Olaoye, M.O. (2011). Computer Applications for Selecting Operating Parameters in a Stationary Grain Crop Thresher. *Journal of Agricultural Technology*, 7(1): 39-56.

[10] Philip T.K. and Itodo, I.N. (2006). Acha (*Digitalia spp.*) a "Rediscovered" Indigenous Crop of West Africa. *Agricultural Engineering International*: *the CIGR Ejournal.* Invited Overview Number 23, Volume VIII, pp 1-9.

[11] Sale, N.A., Muhammed, U.S., Dalha, I.B. and Idris, S.I. (2016). An Improved IAR Sorghum Thresher. *Agricultural Engineering International: CIGR Journal*, 18(3): 119-126.

[12] Sharma, P.C. and Aggarwal, D.K. (2006). Machine Design. S.K. Kataria and Sons, Nai Sarak Dechi, India.