Development and performance evaluation of palm kernel cracker

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Abstract: There is continuous increase in the demand for palm kernel oil because of its domestic and industrial applications. The lack of appropriate technology for palm fruit processing is one of the major problems militating against oil palm agro-industrial development. This study has developed and evaluated the performance of palm kernel cracker. The materials used for the fabrication was selected based on available literatures and local market availability. The machine consists of hopper, cracking chamber, horizontal shaft with beaters, discharge outlet, main frame and prime mover. Randomized block design was used for analyzing the results obtained, the treatment were three that is small, medium and large and they were replicated three times while three feed rate 500, 1000, 1500 g min^{-1} were considered as the block. The throughput capacity and cracking efficiency were determined to evaluate the performance. The optimum cracking efficiency obtained was 98.7%.This was possible with larger palm kernel sizes and feedrate of 1000 g min^{-1}.

Keywords: palm kernel, palm kernel cracker, cracking efficiency

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1 Introduction

Oil palm (*Elaeisguinensis*) is a common and widely used plant in West Africa. The wide acceptability of this plant is associated with multiple usage of different part of the plant ranging from the leaves, trunk, fruit, kernel and even the leftover after processing(John et al., 2020). Oil palm fruit and kernel are one of the plants with the highest oil recovery (oil yield). The expression of vegetable oil from oil bearing fruit and kernel has involved a wide range of processes. The common processes adopted for the expression are traditional, mechanical and chemical extraction processes (John et al., 2020). Recently it was reported that more than 42 countries were engaged in the production of palm kernel oil in the world with South-East Asia countries mainly Indonesia, Malaysia and Thailand taking the lead (Gupta and Sule, 2013). Lack of appropriate technology for palm kernel processing has been described as one of the major problems militating against Nigeria's palm kernel oil agro-industrial development (Owolarafe and Oni, 2011). Local farmers mostly from every part of southern Nigeria are still faced with the problems of how to ease the drudgery and timeliness associated with cracking the palm nuts and also separating the shell from the kernels (Ezechi and Obasi, 2006). In developing countries, small scale palm mills make use of manual labour for the

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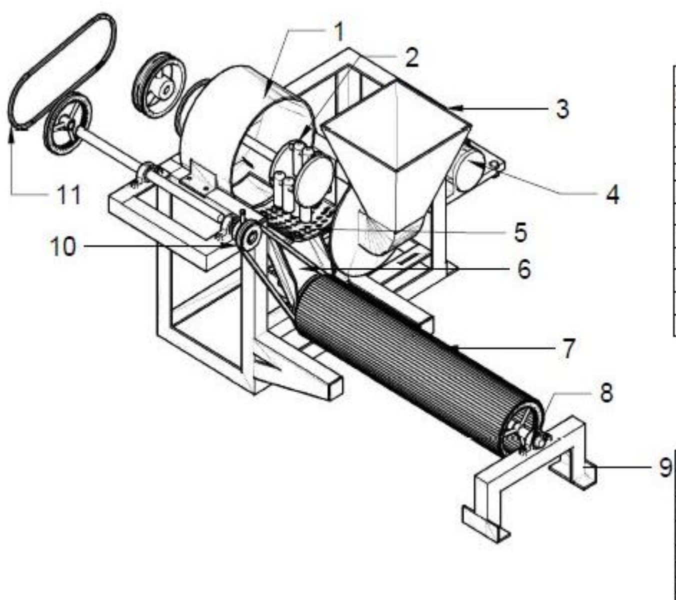
separation of the kernels. The kernels are handpicked from the mixture and at the same time the unbroken nuts are recovered and taking back to the mill for cracking. This method is slow, laborious and unsuitable for large scale mill(Akande et al., 2013).However, previous study shows that palm kernel cracking machine has been developed, but the local farmer where on able to access them because of the high cost implication and maintenance. There is need to develop a palm kernel cracking machine that is cost affordable, easily run and maintain by the local farmers. Hence, this study developed and evaluated the performance of palm kernel cracking and separating machine.

2 Materials and methods

The palm kernel nut crackerconsists of the following component: hopper, cracking chamber, horizontal shaft with beaters, discharge outlet, main frame and prime mover. The hopper was made of 4mm mild steel sheet formed into a trapezium shape with a top opening of 300mm × 300mm and a bottom opening of 100mm × 100mm with sides inclined at 60 degree to help the free flow of the palm nuts into the cracking chamber (Balogun et al., 2020). Incorporated below the hopper is a metering device to control the amount of nuts entering into the cracking chamber. The cracking chamber consists of a circular housing made from 4mm mild steel plate with side lining cracking bars and a horizontal shaft made from 40mm mild steel rod attached with 3 hammer made of 4mm mild steel flat bars arranged at intervals of 90 degrees to one another. The circular housing was 350mm diameter with an opening of 100mm ×100mm at the upper curvature where the nuts were introduced from the hopper while the lower curvature has an opening dimension of 120mm × 100mm where the cracked nuts escape into the separating chamber where the nut was separated from the chaff. The machine was powered by 1.5kW (2hp) electric motor with the aid of belts and pulley arrangement. The main frame is made from 40mm × 40mm by thickness 40mm mild steel angle iron to carry and support the machine components (Figure 1).



PART-LIST OF PALM KERNNEL CRACKER			
S/NO	COMPONENTS	MATERIAL	QTY
1	CRACKING CHAMBER	MILD STEEL	1
2	HAMMER	MILD STEEL	1
3	HOPPER	MILD STEEL	1
4	ELECTRIC MOTOR	FINISHED PRODUCT	1
5	SCREEN	MILD STEEL	1
6	MACHINE OUTLET	MILD STEEL	1
7	CRACKER SEPERATOR	MILD STEEL	1
8	BEARING	-	5
9	SEPERATOR FRAME	MILD STEEL	1
10	PULLEY	FINISHED PRODUCT	4
11	BELT	FINISHED PRODUCT	2

KWARA STATE POLYTECHNIC, ILORIN.		
DEPARTMENT OF AGRICULTURAL AND BIO-ENVIRONMENTAL ENGINEERING		
MODIFICATION AND PERFORMANCE EVALUATION OF PALM KERNEL CRACKER AND SEPERATING MACHINE		
EXPLODED VIEW OF PALM KERNEL CRACKER AND SEPERATING MACHINE		
	NAME	MATRIC NO.
DESIGNED BY:	DAVID AYODEJI ELIZABETH	HND/15/AE/FT/017
SUPERVISED BY:	ENGR. A.L BALOGUN	
APPROVED BY:	ENGR. A.L BALOGUN	
DATE	29-05-2017	SCALE: 1:30

(a) Exploded view of the developed palm kernel cracker

(b) Plate 1Pictorial view of the developed palm kernel cracker

Figure 1 The developed palm kernel cracker

2.1Design calculation and analysis

2.1.1 Hopper Design

The hopper was constructed in a trapezium shape and its volume was determined as:

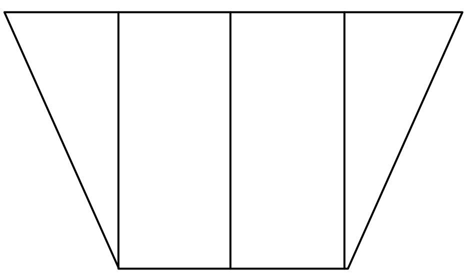


Figure 2 Shape of the hopper

Area A = $\frac{1}{2}BH^2$

Volume v = L ×B ×H (Khurmi and Gupta(2005)) (1)

Where

H= Perpendicular height = 290mm = 0.29m

B= Breath = 300mm = 0.3m, L= Length = 300mm = 0.3m, A = area?

A= $\frac{1}{2}xC = x0.3x0.3x.29^{2}$

A=$\frac{1}{2}X0.3X0.0841$A=$\frac{1}{2}X0.02523$

A= 0.5 × 0.02523 A= 0.012615 ≈ A=0.013m

Volume = V = L ×B ×H

V= 0.3 × 0.3 × 0.29 V=0.0261m^3. Hence, 0.0261m^3 is considered for the hopper.

2.1.2 Shaft design

According to maximum shear stress theory, the maximum shear stress in the shaft for round solid shaft was I max as expressed by

$\frac{1}{2}x = \sqrt{(\sigma b)^2 + 4\pi^2}$ (Khurmi and Gupta (2005)) (2)

Substituting the value of T and σb

$\tau\max\frac{1}{2}\sqrt{\frac{(32m)^2}{d^3} + 4\,\frac{(16\tau)^2}{nd^3}} = \frac{16}{nd^3}(\sqrt{M^2 + \pi^2} = \frac{\pi}{16} = X\tau\max xXd^3\ (\sqrt{M^2 + \pi^2}$

Where,

τmax = Allowable shear stress,$\bar{x}$= Actual twisting, moment,σb = Actual bending moment

M= Maximum bending moment at a point, D= Diameter of the shaft

By limiting the maximum shear stress (τmax) equal to the allowable shear streets (τ)for the material equation may be written as Khurmi and Gupta (2005):

π = Actual twisting moment, σb = Actual bending moment, M= Maximum bending moment at a point, D= Diameter of shaft

By limiting the maximum shear stress (τmax) equal to the allowable shear stress (τ) for the material, equation may be written as:

$$Te=\sqrt{m^2+T^2}=\frac{\pi}{16}\times\tau\times d^3 \quad (3)$$

To calculate for allowable shear stress (τ):

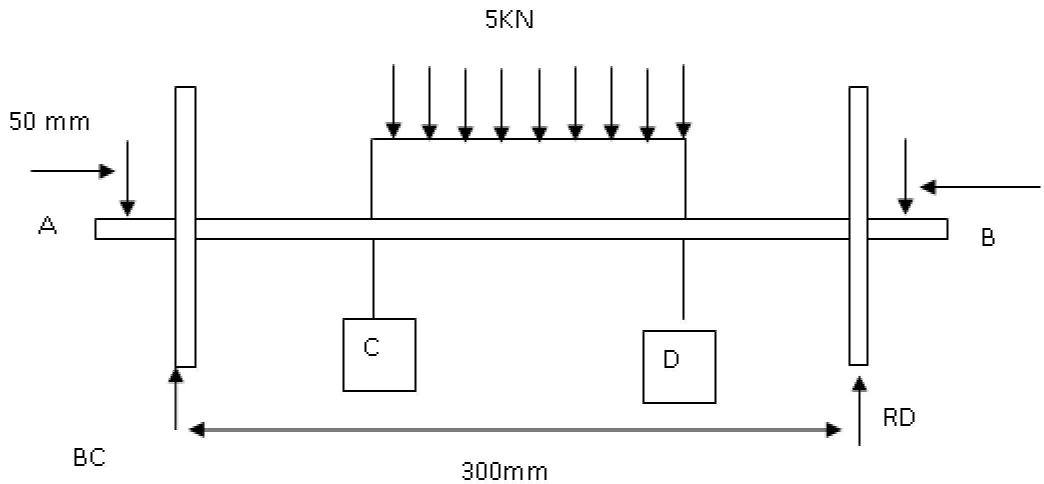


Figure 3The rotor on the shaft

Giving consideration the maximum bending moment acts on the rotor at C and D. therefore maximum bending moment (M):

M = W.L (Khurmi and Gupta (2005)) (4)

Where,

W= Weight =3KN, L = Length = 100mm

maximum bending moment (m)

M= W.L = 5 x 10^3 x 50 = 50 x 10^4 Nmm^{-1}

D = 25m

Torque (τ) = $\frac{pX\,60}{2\tau N}$ (Khurmi and Gupta(2005)) (5)

$\frac{7500\,X\,60}{2\,\tau X\,1440}=\frac{450{,}000}{9048.96}$=49.729$Nmm^{-1}$=T-$\frac{49.729}{3060}\tau$ =0.01625 Nmm^{-2} x 1000= 16.25 Mpa

2.1.3 Bending moment

Let RA and RB = Reactions of A and B respectively: - RA + RB = Total lead acting downward at C and D.

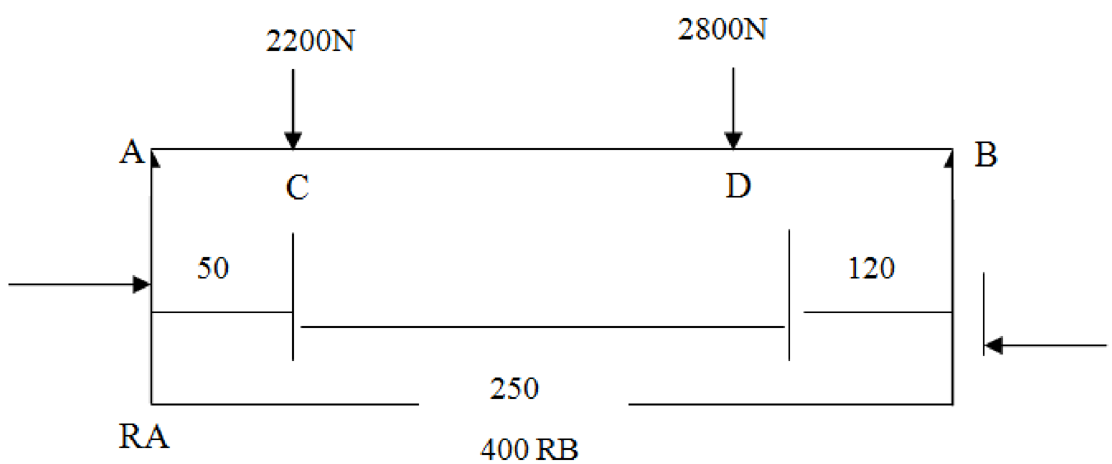


Figure 4 Bending moment of shaft

Now taking moment A,

R_B x 400 =2800 x 280 +2200 x 50894 x10^3

R_B = 894 x$\frac{10^3}{400}$ = 2235 N

R_A = 5000-2335 = 2765 N

And

Considering that the maximum bending moment was either at C or Dthe bending moment at C,

M_C = RA x 50 = 2765 x50

= 138.25 x 10^3 Nmm^{-1}

Bending moment at D

$M_D = R_B$ x 120 = 2235 x 120 = 268.2 x 10^3 Nmm^{-1}

Maximum bending moment transmitted by the shaft

M =M_D= 268.2 x 10^3 Nmm^{-1}

D = Diameter of the shaft

The equivalent twisting moment,

$$\tau e = \sqrt{(KMxM)^2 + (K\tau xT)^2} \quad \text{(Khurmi and Gupta (2005))} \quad (6)$$

Km = combined shock and fatigue factor for bending and Kt = combined shock and fatigue factor for tension

$\tau e = \sqrt{1.5\ x\ 268.2\ x10^3)^2 + (1.2\ x16.25\ x10^3}$ = $\sqrt{16184.29 + 380.25}$

=$\sqrt{162225.54}\tau e$ = 402 x 10^3 Nmm^{-1}

2.1.4 Rotor design

The rotor designed was cylindrical in shape and the volume of the rotor cup was obtained using equation of volume of the rotor cup

$$V=\pi r^2 L \quad \text{(Khurmi and Gupta (2005))} \quad (7)$$

Where,

L = Length off the Cylinder = 200mm = 0.2, R= radius = 85 mm = 0.085m

$V=\pi r^2 L$, V = 3.1428 x0.085^2x0.2, **V = 0.0045m³**

Density of mild steel materials used was given by 7840 $kgm^{-3}$with expression to determine the mass of the rotor (Khurmi and Gupta, 2005)

$$\text{Density (C)} = \frac{Mass\ (M)}{Volume\ (V)} \quad (8)$$

M = CV

M= 7840 X 0.00454, M = 35.59kg

The angular acceleration a = W^2r

$$A = \frac{(2\pi N^2}{60} r \quad \text{(Khurmi and Gupta (2005))} \quad (9)$$

A = $\frac{2X3.142\ X\ 1440^2}{60} X\ 0.085$, A= $\frac{13030502.4}{60} X\ 0.085$, = 18459.88

To convert to radian = 18459.88 x $\frac{\pi}{180}$ =322.23 $radsec^{-1}$

Therefore, centrifugal force can then be found from the expansion

F= Ma, F=Mω^2r

F= 35.59 x 322.23^2 x 0.085, F= 314107. 89 N, F= 3.14107 KN, F= 3KN

To estimate torque required to drive the rotor

$$\tau = F\ X\ r \quad (10)$$

Where, F = 314107.89

R= 0.085, T= 314107.89, T=26699.2N

Hence, Power required to drive the rotor,

$$P = T\omega \quad (11)$$

Where,

T = torque = 26699.2, W= Angular Velocity = $2\pi N$

= 2 x 3.142 x1440 = 9048.96

To convert to radius = 9048.96 x $\frac{\pi}{180} = 157.95\ rad/sec$

P = Tω=26699.2 x 157.95 =4217138.64ω

42.171kω

2.1.5 Pulley design

The pulley was designed for given the expression below Khurmi and Gupta(2005).

when the diameter of the pulley is known.

$$N_I D_I = N_2 D_2 \quad (12)$$

Where,

N1= speed of the electric motor =1490pm, N2= speed of the cracking pulley?

D1= Diameter of the cracking pulley-150mm=0.15m, D2= Diameter of the electric motor =80mm=0.08m.N1D1= N2D2 N2= 1490 x 0.08m = N_2 x 0.15m

115.2 =N_2 x 0.15m, N_2= $\frac{115.2}{0.15}$, 794.7rmp

To determine the speed of the separating drum N_3 D_3 = N_4 D_4

N_3= Speed of the electric motor = 1490 rpm, D_3= Diameter of the electric motor = 80 mm =0.08m, N_4= Speed of the separating drum? D_4= Diameter of the separating pulley= 405mm = 0.405m N_3 D_3 = N_4 $D_4$$N_3$= 1490 x 0.13 = N_4 x 0.405

N_3=$\frac{1490\ X\ 0.13}{0.405}$=$\frac{193.7}{0.405}$ = 478.3 rpm

Center to center distance of the pulley was the parameter that helped to determine the position of the cracking machine pulley from the electric motor pulley.

Computation center to center distance is as follows

2.1.6 Design for Belt

C= $\frac{D+d+d}{2}$ (Khurmi and Gupta(2005)) (13)

Where, c= diameter of larger pulley (machine)

D= diameter of smaller pulley (electric motor)

C= $\frac{178mm+130mm}{2} + 130m$, C= $\frac{308}{2} = 154 + 130$, C=$\frac{284}{1000}$ = 0.284, C= 0.3m

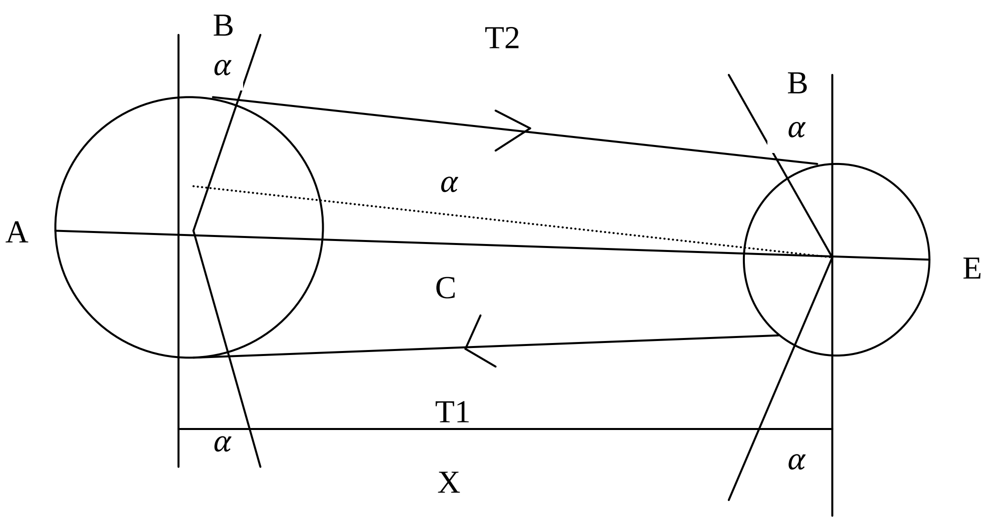


Figure 5 Design for Belt

α=$sin^{-1}\frac{r^2-r^1}{C}$

$\frac{D1}{2}$ D1 =$\frac{130mm}{2}$ = 65mm =0.06

R1=R2= $\frac{d2}{2} = \frac{178mm}{2} = 89mm = 0.089m$

αsin^{-1}($\frac{0.089-0.065}{0.2825}$) = 0.08436, α sin^{-1} 0.84336 =4.68^0

hence, Angle of content, wrap or lapse ∅

∅= 180-2 (α)

∅=180-2 (4.8 =170.4)

To concert to radium

170 X $\frac{\pi}{180}$ 170 X $\frac{3.142}{180} = 2.97\ radian$

To estimate the tension on tight side on the belt P-(T1-T2) V

Where,

Ti = Tension on the tight side of the belt

T2= Tension on the slack side of the belt

V= Velocity of the belt

V=$\frac{2\pi rN}{60}$

$\frac{2X3.142X0.065X1440}{60}$ =$\frac{588.1824}{60}$ V= 9.8ms^{-1}

Power (p) = T1-T2 = DT

DT=$\frac{P}{V}$P=$\frac{34426.93}{9.8}$ =3512.95N- T1=T2

Therefore, the co-efficient of fiction between the tight side and slack slide tension in form of co-efficient of friction and the angle of contact:

Integral of T2=T1

$\sqrt{\frac{T1}{T2}}(\frac{\propto T}{T})\ \mu \int \theta\alpha\theta$

$\frac{T1}{T2}$Log e =$\mu.\theta$or T1 T2=e$\mu.\theta$

The equation above can be expressed in term of corresponding logarithm, co-efficient of frictional (μ) between the belt and the pulley of a dry steel materials is 0.3

2.3 log =$\frac{T1}{T2}\mu\theta\frac{T1}{T2}$:- 2.3 log $\frac{T1}{T2}$Log $\frac{T1}{T2}$= 0.3 x 2.94 =0.882

- to get our T2 = P (T1-T2).T2=3912 (2.437-T2), T2=$\frac{3912}{2.457}$

Therefore T1=2.437μ andT2=1.605

2.1.7Design for belt length

To calculate for the length of the belt

L =$\pi\frac{d1+d2}{2}$ +2x+$\frac{d1+d2}{4x}$

Where, L = Length

D1= Diameter of the driver pulley (electric motor) = 130mm, D2= Diameter of the driving pulley (cracking machine)= 178mm

L=$\frac{3.142}{2}$ (130+178) +2(300) + $\frac{(130-178)^2}{4(300)}$

L= 1.571 (308) + 600 + 18.25, L=483.8 +600 +18.25,

L= 1102mm = 1.102m

L= ≈ 1m

Based on the length, its cross-section area calculated and equivalent tension (DT) obtained the standard V-belt section that will be used conveniently is the cross-section symbol C which is very close to the one calculated as shown in Table 1.

Table 1Dimension of Standard V-Belts

Type of belt	Power ranges in Kw	Minimum pitch diameter of pulley (D)mm	Top width (b)mm	Thickness (t)mm	Weight meter length in Newton (N)
A	0.7-3.5	75	13	8	1.06
B	2-15	125	17	11	1.89
C	7.5-75	200	22	14	3.43
D	20-150	355	32	19	5.96
E	30-350	500	38	23	

Source: Khurni and Gupta(2005)

Prior to the design, the B-type of belt is chosen because of the load induced by the machine on the belt and the pulley designs are also considered.

2.1.8Bearing Selection

Optimum bearing performance was achieved by selection of the appropriate bearing and shaft to suit the service application. Bearing size was generally controlled by the shaft bending and tensional stresses. In order to select a more suitable ball bearing, the basic dynamic radial load is calculated. It was then multiply by the service factor (ks) to get the design basic dynamic radial load capacity. The service factor for the ball bearings was given in the Table 2.

Table 2 Values of service factor (Ks)

S/no	Types of Service	Service Factor (Ks) For Radia Ball Bearing
1	Uniform load	1.0
2	Light shock load	1.5
3	Moderate shock load	2.0
4	Hearing shock load	2.5
5	Extreme shock load	3.0

Source: Khurmi and Gupta (2005)

The materials used for the fabrication of the machine were selected based on the following criteria's Cost of material, Durability, Size, Weight andAvailability of materials.

2.2Working principle of palm kernel cracking machine

The palm kernel seed cracking and separating machine has two distinctive parts, the cracking unit and separating unit.The both units comprises of the beater, rotor and the perforated concave cylinder. It is made of a drum 255mm diameter and 4mm thickness. The drum is covered at both ends with a mild steel sheet of 4mm thickness. The outlet and the delivery tube are connected to the cracking drum

The palm kernels are fed through the hopper into the cracking unit. The electric motor supplies the power required by the machine to crack the kernel through the belt drive. The cracking is achieved by the beaters on the cracking cylinder or hammer by beating the kernels against the concave (sieve) of 120mm x 100 mm diameter. The cracked kernel is then moved to the separating unit which is the perforated concave cylinder the cracked the separation techniques employ here separation using variance in the weight of the nut and the chaff,the nut were retained in the perorated concave cylinder while the chaff will move out from the exit perforation, the clean nut is collected at the outlet.

2.3Sample preparation

The palm kernel seeds were cleaned to free it from foreign matters. The moisture content of the palm kernel seed was determined using Tunde-Akintunde et al. (2001) recommended method for edible seed. This involved the oven drying of seed samples at 103℃ for 24hours with the moisture content of palm kernel seed is 26.9%dry basis. The palm kernel seeds were then graded into sizes ranging from 0-5 mm for the small sizes, 5-10 mm for the medium sizes while 10-15for the large sizes (Alengaramet al., 2010). Each sizes were weighed into 500g, l000g and 1500g using a digital weighing scale (see plate 2).

2.4Experimental design and layout

A Randomized block design was selected for analyzing the results obtained the treatment were three which are small, medium and large. They were replicated three times while three federate 500, 1000, 1500 $gmin^{-1}$ were considered as the block. Moisture content of 26.9% dry basis and Machine speed of 1495rpm were maintained throughout the evaluation test. The developed machine was evaluated considering the throughput capacity and cracking efficiency.

The mathematics expression used for the determination was expressed as follow

Throughput capacity

$$Ct = wt/Td \quad \text{(Rimmer et al., 2002)} \quad (14)$$

Where,

wt= total weight of the palm kernel fed into the machine ($gsec^{-1}$)

Td= total time taken by the cracked mixture to leave the separator discharge outlet (hour)

The feed rate of 500$gmin^{-1}$, 1000$gmin^{-1}$, 1500$gmin^{-1}$ at 1495rpm was used with respect to thetime.

(a)Small size particle size ranged from 0-5 mm

(b)Medium size particle size ranged from 5-10 mm

(c) Large size particle size ranged from 10-15 mm

(d) Small size (e) Medium size (f)Large size

Figure 6 Palm kernel

Table 3 Summary of the output parameter of palm kernel seed cracker

Moisture content 26.9 % db	Small	Medium	Large
Weight ($gmin^{-1}$)	500	500	500
Shaft Speed (rpm)	1495	1495	1495
Cracking Efficiency (%)	71.5	93	98.5
Kernel Breakage (%)	6.67	14.45	17.6
Through Put Capacity ($kghr^{-1}$)	37.9	30.51	23.22
Weight ($gmin^{-1}$)	1000	1000	1000
Shaft Speed (rpm)	1495	1495	1495
Cracking Efficiency (%)	50.51	85.5	98.7
Kernel Breakage (%)	7.9	14.8	12.78
Through Put Capacity ($kghr^{-1}$)	40.45	32.40	50.28
Weight ($gmin^{-1}$)	1500	1500	1500
Shaft Speed (rpm)	1495	1495	1495
Cracking Efficiency (%)	57	92	96.4
Kernel Breakage (%)	12.5	16.9	10.8
Through Put Capacity ($kghr^{-1}$)	31.58	54.77	60.95

Cracking Efficiency was given as

$$c= (wt-wn)wT \times 100 \text{(Titherington and Rimmer, 1980)} \quad (15)$$

Where,

Wt= total weight of the palm kernel seeds fed into the machine ($gmin^{-1}$)

Wn= weight of un-cracked palm kernel seed ($gmin^{-1}$)

The palm kernel fed into the machine was sorted and graded into three categories labeled as small, medium and large.

3 Results and discussion

The results obtain from the study were summarized and expressed in Table 3.

The results above presents the average values of output parameter for small, medium and large size of the palm kernel. The result showed the cracking efficiency, percentage kernel breakage and throughput capacity for the smaller sizes to be 71.5%, 6.67% and 37.9$kghr^{-1}$ respectively. This implies that the cracking and separating unit favors the feed rate of 500$gmin^{-1}$ for the smaller sizes as a result of the clearance between the hammer and the cracking chamber. For the medium sizes, the cracking efficiency, percentage kernel breakage and throughput capacity obtained were 93%, 14.45% and 30.51$kghr^{-1}$ respectively. It can be deduced that the cracking and separating machine favors the feed rate of 500$gmin^{-1}$ for this sizes as a result of the clearance between the hammer and the cracking chamber. While for the larger sizes the cracking efficiency, percentage kernel breakage and throughput capacity obtained were 98.7%, 12.78% and 50.28kghr^{-1} respectively.Ismailet al.(2015) worked on the design and development of an improved palm kernel shellingand sorting machine and reported an efficiency of 90% and throughput capacity of 59% while John et al. (2020) also worked on the development of palm kernel cracking and separating machine with efficiency of almost the same value but this efficiency was influenced by the variance in the speed of the machine. This corresponding results shows how effective and close the output of the machine develop is while the efficiency remain outstanding. It can be deduced that the cracking and separating machine favors the feed rate of 1000gmin^{-1} for this sizes as a result of the clearance between the hammer and the cracking chamber.

Table 4 presents the analysis of variance of the performance evaluation carried out on the developed machine. The ANOVA table shows that there are significant differences in the mean of the cracking efficiency of the developed palm kernel cracker at $p \geq 0.05$.

Table 4 Analysis of variance

Source of variation	Degree of freedom (df)	Sum of square (ss)	Mean of square (ms)	Observe value (f)	Required value (f)	
Total	15	25.44			5%	1%
Block	3	5.19	1.73	6.81	3.86	6.99
Treatment	3	17.69	5.90	21.07		
Error	9	2.56	0.28			

Note: since $6.81 \geq 3.86$ and $21.07 \geq 3.86$ there are significant differences in the mean cracking efficiency of the palm kernel cracker.

4 Conclusion

A palm kernel cracker was developed and performance evaluation was carried out on it to determine the cracking efficiency of the machine using different grades of palm kernel ranging from small, medium and larger palm kernel at varied feed rate of 500, 1000 and 1500 gmin^{-1}. the selected parameters were found to have significant effect on the cracking efficiency of the developed machine; the smaller size has a maximum cracking efficiency of 57% and throughput capacity of 31.58kghr^{-1} at feed rate of 1000 g, the medium size has a maximum cracking efficiency 92% and throughput capacity of 54.77kghr^{-1} at feed rate of 1000 g while the an optimum cracking efficiency of 98.7% and throughput capacity of 60.95kghr^{-1}was possible at larger palm kernel sizes and feedrate of 1000 gmin^{-1}.The operational parameters that are feedrate, and kernel size were found to be significant at $p \geq 0.05$.

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