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DESIGN AND FABRICATION OF A MOVABLE PALM KERNEL CRACKING MACHINE FOR RURAL APPLICATION

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Abstract: **This developed and produced palm kernel cracking and separating machine. This is geared towards eliminating the drudges involved in cracking and separating palm kernel by the rural farmers. The machine was modeled with solidworks version 2010. Mathematical design was carried out to help select materials that will be adequate to withstand the stress during operation. The machine was produced by following the standard procedure for fabrication processes. The produced machine was surfaced finished to improve on the aesthetics. The machine was produced with locally sourced materials for cost effectiveness. The cost evaluation gave a total cost of production of five hundred and seventeen thousand, five hundred and eighty five naira (N 517,585) only. The performance evaluation carried on the machine gave processing capacity (PC) of 60 Kg/hr, throughput efficiency of 83%, shelling efficiency of 75.25%, separation efficiency of 89.75% and a breakage efficiency of 12.38%. Hence the machine is suitable for use by farmers in rural areas.**

Keywords: **Palm kernel cracking machine, cracking chamber, agricultural prime mover engine.**

1. INTRODUCTION

Palm kernel and other products from palm fruit processing are very important wealth creating products in Nigeria. The palm kernel seed is rich in palm kernel oil which can be extracted for different applications, while the cake are mostly used for animal feed production.

The process of extracting the kernel seed involves the cracking of the shell to expose the seed. Palm kernel cracking is one of the unit operations of palm oil processing. It is the shelling of the outer surface (shell) of palm kernel for the recovery of palm kernel seed. Cracking can be achieved through the application of forces on materials in various forms, such as impact, compression or attrition (Odewole and Ajibade, 2015).

The methods of cracking palm kernel can be broadly classified into two; manual cracking method and cracking with machines. The manual method involves the use of stone to exert impact. This method is full of drudgery (hard and boring), and the output is low with the accompanying risks. While the modern method involves the use of machines to provide impact on rotation. This method offset the disadvantages of the manual method (Ezechi and Obasi, 2006).

In most rural areas of Southern Nigeria where these resources (palm plantation) exists in large quantities, the local farmers still uses the manual method of cracking and separating palm kernel seed from its shell, due to the unavailability of machines for cracking and separation of palm kernels, and the limited resources to acquire the readily available ones. Hence these farmers are faced with the problem of how to easily and quickly crack their palm kernels and as well separate the shells from the seeds without exerting much energy and time at relatively low cost. In attempt to solve these problems of cracking and separation of palm kernels, this project becomes imperative. The project therefore seeks to provide assistance to the rural dwellers that engages in palm oil and kernel processing business for a suitable, convenient and cheap method of cracking their palm nuts.

The palm kernel cracking machine is a machine that evolved as a result of research work done in the processing of palm kernel, and the application of its products. Many researchers have worked in the field of palm kernels processing and the application of their by-products. However, research work is still ongoing because of the economic importance of palm kernels. Some researchers have been mentioned here to show their contributions to the processing of palm kernels.

Asibeluo and Abu (2015), worked on the Design and Construction of a palm kernel cracker and separator. They found out that the conventional palm kernel cracker which uses long rotating hammer for cracking nuts was less efficient since it misses any nut that is not in line of cracking action. Also they added that the conventional cracker is only capable of cracking the palm kernels without separating the shells from the seeds. These constraints were overcome by them in designing a cracker with a cracking rectangular channel welded to a cracking flywheel, with a centralized hole through which every nut must pass to make contact with the flywheel. Also they incorporated two different sets of separators for separating the kernel seeds from the cracked shells.

Ismail et.al. (2015) designed and developed an improved palm kernel shelling and sorting machine. They evaluated the physical and mechanical properties of the palm kernel nuts so as to determine the critical load required to cause fracture on the palm kernel nuts without damage to the nut meat. The necessary evaluation and designing of the cracking and sorting units were achieved through these design criteria: load estimation, kernel size, moisture content of shells and motion resistance of kernel seeds and shells. They employed these design parameters in calculating the momentum necessary to achieve the needed force of cracking (impact force) and effective sorting approach. They employed evaluation of velocity in the design of the optimal configuration of the impeller and kernel shell characteristics for the sorting technique.

Andoh et.al. (2010) worked on the development of a model for selecting a suitable sieve size that can be used to crack palm kernel nut in order to increase productivity and also improve the quality of the palm kernel oil. From their findings, it was established that separating Dura and Tenera varieties of palm kernel nuts before cracking increases the cracking efficiency, and hence productivity.

Asha et.al. (2016) investigated the mathematical analysis of deformation of kernel shell, crack growth and fracture of the shell as a result of its spontaneous collision with the stony wall of the shelling drum, and shattering of the shell and the kernel under this impact load. They found out that many factors like moisture content of palm kernel, speed of the rotor, thickness of the shell, magnitude of the impact force, size and weight of kernel are responsible for effective cracking.

Ndukwu and Asoegwu (2011) developed a mathematical model for predicting the cracking efficiency of vertical-shaft centrifugal palm nut cracker using Buckingham's pi (π) theorem. The result of the prediction was in agreement with the experimental one.

Jimoh and Olukunle (2012) investigated the effect of heat treatment during mechanical cracking using varieties of palm nuts. Their findings show that highest throughput, functional efficiency and quality performance efficiency were achieved with reduced mechanical damage.

Manuwa (2007) investigated the properties of size, moisture content and shell thickness of Dura palm nut variety. From his findings the regression equations established and validated show close agreement between measured and predicted forces.

Nasir (2005) designed and constructed from locally available materials hammer mill for grain particles, such as maize, millet, guinea corn and other coarse material of cassava tuber, yam tuber, beans etc. The grinding process was achieved by the use of hammer to beat the material fed into fine particles. He found out that the main shaft speed of 700rpm transmitted by a belt drive from one horse – power electric motor was suitable to mill effectively.

Most of the reviewed works in palm kernel cracking machine does not incorporate the separating unit, thereby resulting to the drudges of separating the nut from the cracked shell, which is energy and time consuming. Hence the need for this work which incorporates the separating unit with the cracking unit.

2. MATERIALS AND METHOD

2.1 Materials

The machine has several components, which include cracking chamber, impellers and impeller shaft, hopper, frame with tyres, drive shaft and sheaves pulleys, belt and bearings, separator, and the prime mover. The hopper has two frustums of pyramid. The bottom part of the bigger frustum is attached to the top of the smaller frustum. The bottom end of the smaller frustum is connected to the cracking chamber. The cracking chamber incorporates an opening at the bottom which serves as an outlet for cracked nuts to pass onto the separator. Inside the cracking chamber is the impeller shaft to which three pieces of impeller blades were attached at 120° to one another. The prime mover is made of NIKO engine model R175, which powers the machine.

2.2 Design Considerations

The following factors were considered in the design of this palm kernel cracking and separating machine; operating environment, choice of power source, and the capacity of the machine. For the operating environment, the machine will be produced for rural farmers where palm kernels are available in large quantity. Hence a diesel engine was considered as the prime mover due to unavailability of stable electric power supply in most rural areas of Nigeria where this machine will be mostly used. Also a through put capacity of 900kg/day was considered since the machine is for SMEs.

2.3 Mathematical Design

2.3.1 Design of Sheaves

Sheaves are machine parts which work in conjunction with belts to provide the required transmission. Its rotational speed is the same as the speed of the prime mover. Here the considered prime mover is a NIKO diesel engine with its basic technical data presented in Table 1.

Table 1: Basic technical data of Agricultural prime mover, NIKO engine.

$Engine\ model$	$R175\ Diesel$
$-Maximun\ power\ output$	$4.85\ Kw$
$-\ Rated\ speed,\ n$	$2{,}600\ rpm$
$-\ Net\ weight$	$60\ Kg$

The diameter of the sheave attached to a flywheel with flange is 100 mm. The flywheel was used to smooth out changes in the speed of shaft caused by torque fluctuations.

The torque developed by the engine was determined using equation (1) (Ugural, 2004)

$$P = \frac{Tn}{9.55} \tag{1}$$

Where the power P is in Kw, torque, T is in Nm and the speed of the shaft n in rpm are obtained from table 1.

Also to determine the diameter of the pulley, speed ratio equation (2) was employed, since it is desired to reduce the speed of the engine to one quarter.

$$\frac{N_2}{N_1} = \frac{D_1}{D_2} \tag{2}$$

Where N_2 is the rotational speed of the shaft attached to the impeller

N_1 is the rotational speed of the shaft attached to the engine

D_1 is the diameter of the sheave attached to the engine and

D_2 is the diameter of the sheave attached to the impeller shaft.

2.3.2 Determination of Belt Speed.

Belt speed was determined using equation (3)

$$v = \omega r_1 \tag{3}$$

Where v is the belt speed in ms^{-1}, r_1 is the radius of the sheave attached to the prime mover, and ω is the angular velocity of the sheave as expressed in equation (4)

$$\omega = \frac{\pi n}{30} \ rads^{-1} \tag{4}$$

2.3.3 Determination of Centre Distance of the Sheaves and Belt Length.

The centre distance (C) of the sheaves was calculated using equation (5) (Sadhu, 2004)

$$C = 0.55(d_1 + d_2) + t \tag{5}$$

Where d_1 and d_2 are the diameters of the smaller sheave and larger sheave respectively, and t is the belt thickness, which is equivalent to 10.5

Also the length of the belt (L) was obtained using equation (6) (Sadhu, 2004).

$$L = 2C + 1.57(d_1 + d_2) + \frac{(d_2 - d_1)^2}{4C} \tag{6}$$

2.3.4 Determination of Angle of Contact.

The angle of contact θ can be calculated from the formula as seen in equation (7).

$$\theta = \pi \pm 2\sin^{-1}\left(\frac{d_2 - d_1}{2C}\right) \tag{7}$$

Where the +ve sign is used for the bigger sheave and –ve sign for the smaller sheave.

2.3.5 Determination of Belt Tension.

The tension in the tight side, Te_1 and slack side, Te_2 of the drive belt were obtained from the equation (8) for impending motion according to Shigley (1989), while neglecting centrifugal force.

$$\frac{T_{e1}}{T_{e2}} = e^{\mu\theta} \tag{8}$$

Where μ is the coefficient of friction between leather belt and cast iron, and is given as 0.35 (Shigley, 1989).

2.3.6 Determination of the Power and Torque Transmitted.

The power transmitted to the shaft according to Shigley (1989) is given by:

$$P = \frac{T_{e1} - T_{e2}}{v} \tag{9}$$

While the torque at the impeller shaft (main shaft) is given by Ugural (2004) as;

$$T = (T_{e1} - T_{e2})R \tag{10}$$

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2.3.7 Design for the SHAFT.

The shaft is a rotating member of the machine that transmit the power from the prime mover. According to ASME code for the design of transmitting shaft is based on the maximum shear stress theory. The maximum shear stress τ max corresponds to the allowable shear stress and takes into account the effect of shock and fatigue by introducing two constants into the equation of maximum shear stress as presented in equation (11) (Sadhu, 2004).

$$\tau_{\max} = \frac{5.1}{d^3}\sqrt{(c_m M)^2 + (c_t T)^2} \qquad (11)$$

The constants C_m and C_t are numerical values for combined shock and fatigue factors to be applied to the bending and twisting moments respectively. While M and T are the bending moment and torsional moment respectively, and are calculated by drawing the load diagram of the driven shaft as seen in figure 1.

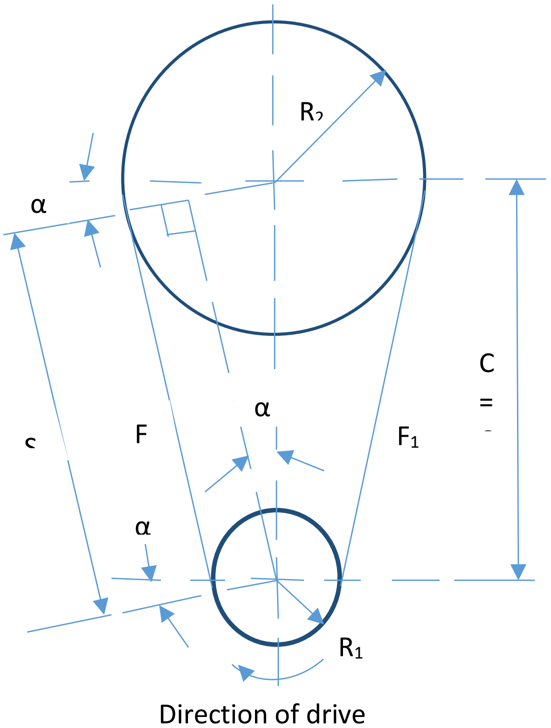

Figure 2: Shows the geometry of open belt drive.

From the geometry of the drive the angle α was found according to Shigley (1989), to be given by equation (12).

$$Sin\ \alpha = \frac{R_2 - R_1}{C} \qquad (12)$$

While the resultant force on the shaft F is given by equation (13)

$$F = F_1 \cos 4^o + F_2 \cos 4^o \qquad (13)$$

Where F_1 and F_2 are the forces on the tight and slack sides of the belt, which corresponds to Te_1 and Te_2 respectively.

2.3.8 Determination of Speeds Required at the Impellers

The mass and energy at breaking point obtained from experimental results of Morakinyo and Bamgboye (2019), was used to determine the velocity (v) required for cracking using equation (14).

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$$\frac{0.03\,Kg}{2}v^2 = 1.65 \tag{14}$$

While the angular speed was determined using equation (15) as stated by Khurmi and Gupta.

$$w = \frac{v}{r} \tag{15}$$

2.3.9 Determination of the Tangential Force and Torque at the Impellers

The tangential fore F = mα (Hannah and Stephens) (16)

Where m is the mass of the impeller, which depends on the moment of inertia I_{AA}, and radius of gyration K as given by equation (17) (Surendra).

$$I_{AA} = mK^2 \tag{17}$$

Where $I_{AA} = \frac{bh^3}{3}$ (18)

And radius of gyration, K = $\sqrt{\frac{I_{AA}}{A}}$ (19)

Also the angular acceleration $\alpha = \omega^2 r$ (20)

Hence F = $m\omega^2 r$ (21)

Therefore the Torque T = Fr (22)

The diameter of the impellers, was obtained using equation (23) according to Hannah and Stephens (1991)

$$\tau_{max} = \frac{5.1}{d^3}\sqrt{(k_m M)^2 + (k_t T)^2} \tag{23}$$

Where M is the bending moment on the impellers as a result of the reactions on the impellers as seen in figure 2. Also the resulting bending moment diagram is seen in figure 3.

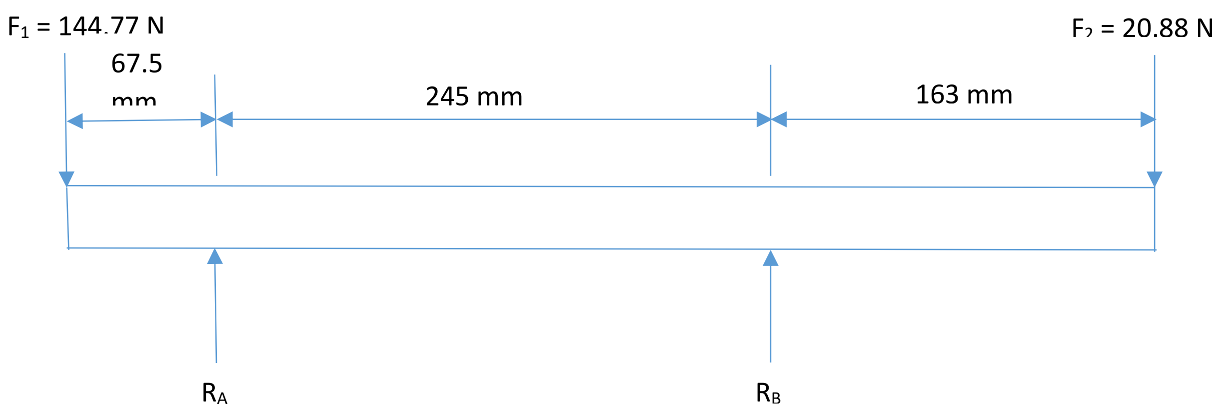

Figure 2: Shows the system of load diagram of the impeller shaft.

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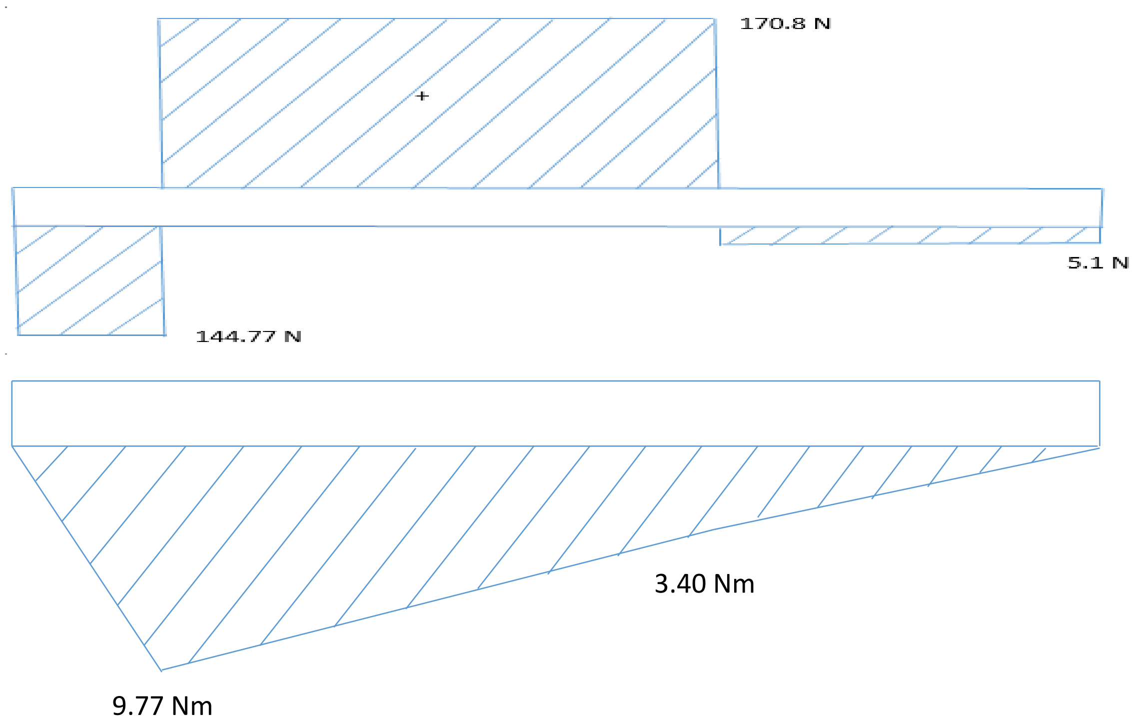

Figure 3: Show the system of shearing forces and the bending moments on shaft

From Figure 2, R_A and R_B are the reaction forces on the bearings. To be able to determine the bending moment, the reactions are to be determined.

$$144.77 \text{ N x } 0.0675 + R_B \text{ x } 0.245 = 20.88 \text{ N x } (0.245 + 0.163) \quad (24)$$

$\therefore$ $R_B = -5.114020408 \approx -5.11$ N.

Taking moment at point B, gives equation (25), for determination of R_A

$$M_B: 144.77 \text{ x } (0.675 \text{ m} + 0.245 \text{ m}) = 0.245R_A + 20.88\ (0.163 \text{ m}) \quad (25)$$

$\therefore$ $R_A = 170.7640204$ Nm.

Also Taking moment of at point A, gives $M_B = 3.40$ Nm and $M_y = 0$.

Hence $k_m = 1.5$ and $k_t = 1.0$ for gradually applied or steady load on a rotating shaft, and $\tau_{max} = 55$ N/mm^2 (Shigley, 1989).

2.3.10 Determination of shaft Stiffness or Rigidity.

The stiffness or rigidity of shaft is very important in a situation where the torsional deformation of the shaft and lateral deflection is limited as demanded by the design. Stiffness relates to the ability of a part to resist deflection or deformation. Hence for transmission shafts the torsional deformation should be limited to 1^o in 20 diameters. The angular twist of machine shafts should be limited to $\frac{1}{3}^o$ per metre for ordinary service, 0.25^o per metre for variable loads and $\frac{1}{6}^o$ per metre for suddenly reversed loads and long feed shafts (Singh).

The angle of twist (torsional deflection) of a circular shaft is expressed as:

$$\theta = \frac{TL}{GJ} \quad (26)$$

Where T is the torque or torsional stress on the shaft = 2.71 Nm, L is the length of shaft = 475.5 mm, G is the modulus of rigidity of the shaft = 79GPa for structural steel, ASTM-A36 (Ugural, 2004), and J is the polar moment of inertia (second moment of area) of the shaft expressed as:

$$J = \frac{\pi d^4}{32} \text{(Shigley, 1989)} \tag{27}$$

Therefore the torsional deflection per unit length can be obtained using equation (28)

$$\frac{\theta}{L} = \frac{T}{GJ} \tag{28}$$

Which gives $\theta = 0.011714981^{o} \approx 0.01171^{o} < 1^{o}$ in 20 mm diameter shaft.

2.4 Materials selection, Fabrication and Costing.

The materials used for the components of the palm kernel cracking machine were selected based on availability, cost and usability. The selected materials were sourced from a local market in Awka, Anambra Nigeria. Most of the selected materials were of junk low and high carbon steel for cost effectiveness.

The materials selected for the machine production were sourced and cut into the required dimensions according to the design specifications. The cut materials were welded together using electric arc welding machine to form the machines various component, such as the hopper, shaft with impellers, cracking chamber, and supporting frame. The fabricated components where grinded, to remove excess weld deposits, sand-papered to remove rust and dirt, washed, dried and painted to improve its aesthetics. The fabricated components were assembled using bolt and nuts, alongside the selected pulleys and belts.

The costing was also done to determine the cost of production of the machine. This was carried out by itemizing the materials used, its quantity and the unit cost, before totaling the cost, including the cost of labour.

2.5 Performance Evaluation

Performance evaluation was carried out on the machine to determine its performance in the areas of it production capacity, throughput efficiency, shelling efficiency, separation efficiency, and breakage efficiency.

2.5.1 Production Capacity

The production capacity (PC) expressed in Kg/hr is quantity of palm kernel that the machine was able to crack and separate within an hour, expressed mathematically as seen in equation (29).

$$PC = \frac{W_{fk}}{t} \text{(Kg/hr)} \tag{29}$$

Where W_{fk} is the measured weight of palm kernel that was fed into the machine (Kg), and t is the time taken to crack and separate the nut from the shell.

2.5.2 Throughput Efficiency

Throughput efficiency (ε_T) is the ratio of quantity of material s that came out after processing to the quantity of materials that was feed in for processing, expressed in percentage (%). This is given mathematically by equation (30).

$$\varepsilon_T = \frac{W_s + W_n}{W_{fk}} \times 100 \tag{30}$$

Where W_s is the weight of shell that came out of the machine after processing.

2.5.3 Shelling Efficiency

The shelling efficiency (ε_{sh}) is the ratio of properly shelled palm kernel gotten from a measured quantity of the palm kernel after processing to the measured quantity of the palm kernel after processing. This is expressed mathematically as seen in equation (31).

$$\varepsilon_{sh} = \frac{q_{sn}}{q_n} \times 100 \tag{31}$$

Where q_{sn} is the number of properly shelled nut from the measured quantity of palm kernel after processing, while q_n is the number of both shelled and unshelled from the measured quantity after processing.

2.5.4 Separation Efficiency

Separation efficiency ε_{sp} is the effectiveness of the machine to separate the shell from the nut. This is obtained by picking the quantity of shell picked from the measured quantity of the palm kernel after processing and applying it in equation (32) to obtain the separation efficiency.

$$\varepsilon_{sp} = (1 - \frac{q_s}{q_n}) \times 100 \qquad (32)$$

Where q_s is the number of shell picked from the measured quantity of the palm kernel after processing.

2.5.5 Breakage Efficiency

The breakage efficiency (ε_b) is the ratio of the number of broken nuts in a measured quantity of the palm kernel after processing to the total number of nut in the measured sample after processing, expressed mathematically in equation (33).

$$\varepsilon_b = \frac{q_b}{q_n} \times 100 \qquad (33)$$

3. RESULTS AND DISCUSSIONS

3.1 Developed Palm Kernel Cracking Machine

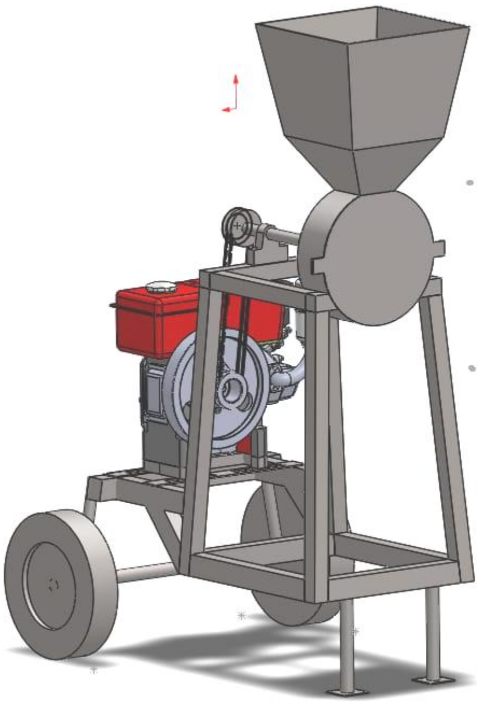

Figure 4: The modeled assembled drawing of the fabricated palm kernel cracking machine.

The developed palm kernel cracking machine as seen in figure 4 was modeled with solidworks 2010 version. It is composed of various components such as the hopper, cracking chamber, driving shaft with impellers, supporting frame, pulleys and belt, and the prime mover.

The hopper is the channel through which the materials are fed into the machine. It was fabricated in two portions of frustum of pyramid. The bigger frustum has parallel sides of top dimension 400 mm square and base dimension 300 mm square, and height of 295 mm. The smaller frustum has parallel sides of top dimension 300 mm square and base dimension of 96 mm square, with a height of 142 mm. the hopper was designed in this form to prevent the run off of nuts, which is capable of causing injury to the operator.

The cracking chamber is where the actual cracking takes place. It was fabricated with 1.5 mm thick sheet metal that was rolled into a cylinder of 350 mm diameter and 96 mm width. The sides of the cylinder were covered with circular sheets of metal with provision made on one side for to allow for the rotation of the impeller shaft. The bottom part of the cracking chamber was cut out circumferentially to the dimension of 130 mm, and crossed at six places at an intervals of 21 mm to serve as sieve that separate the shell from the nut.

The shaft was made medium carbon steel of dimension machined to 30 mm diameter and length of 730 mm. A length of 230 mm was stepped down to 25 mm diameter from one end for the assembling of sheave thereby leaving the length of 500 mm with the diameter of 30 mm, to which the impeller blade was attached to. The impellers which serves as the beater that cracks the nut was produced from a high carbon steel material. The material was cut to the dimension of 130 mm long and 70 mm width and welded at angles of 120^o to one another on the shaft.

The supporting frame is the skeletal fame of the machine that carry's other components of the machine. It was fabricated with a steel angle bars of dimension 60 x 60 x 4 mm thick, and 50 x 50 x 4 mm thick. These materials were cut and joined with welds to form the frame in figure 5, which supports and carry other components.

The prime mover is made of agricultural diesel engine of model NIKO R175, with a maximum power output of 4.85KW and a rated speed of 2,600 rpm which was reduced to the required speed by means of pulley and belt. The engine was use to power the developed machine.

3.2 Production Cost Evaluation

Table 2, shows the production cost of the newly developed palm kernel cracking machine. The total cost of production including the labour cost is five hundred and seventeen thousand, five hundred and eighty five naira (N 517,585), which is equivalent to 660 USD at an exchange rate of 785 naira/USD.

Table 2: Bill of material quantity and cost for the production of the develop machine

S / No.	*Material*	*Unit*	*Quantity*	*Unit Cost (#)*	*Total Cost (#)*
1	*Mild steel plate* (1,219 *x* 609.6 *x* 1.5)	*mm*	2 *pieces*	30,000.00	60,000.00
2	*Elegant rug*	*square yard*	5.9	1,350.00	7,965
3	*Niko Agric. Engine*	*R* 175 *model*	1	125,000.00	125,000.00
4	*Pillow–block bearing*	*UCP size* 205	6	2,500.00	15,000.00
5	*Pillow–block bearing*	*UCP size* 206	2	3,000.00	6,000.00
6	*Angle iron* (50 *x* 50 *x* 4)	*mm*	4	5,300.00	21,200.00
7	*Angle iron* (60 *x* 60 *x* 4)	*mm*	2	7,500.00	15,000.00
8	*Galvanized rod* (ϕ50)	*mm*	1 *length*	8,000.00	8,000.00
9	*Carbon steel electrode*	*packets*	2	3,500.00	7,000.00
10	*Cutting disc*	*pieces*	5	600	3,000.00
11	*Filing disc*	*pieces*	5	600	3,000.00
12	*Shaft* (ϕ30 *x* 762)	*mm*	1	5,000.00	5,000.00
13	*Fuel* (*Transport for vehicle*)	*litres*	30	216.67	6,500.00
14	*Shaft* (ϕ25 *x* 1,219)	*mm*	1	6,000.00	6,000.00
15	*Mild steel pipe* (ϕ50)	*mm*	1 *length*	10,000.00	10,000.00
16	*High carbon* (*spring*) *material*		1	6,000.00	6,000.00
17	*Stainless electrode*	*packets*	1	5,000.00	5,000.00
18	*Sheave* (ϕ25 *x* 355.6)	*mm*	1	25,000.00	25,000.00
19	*Wheel tyre*		2	8,000.00	8,000.00
20	*Diesel for generator operation*	*litres*	40	850	34,000.00
24	*Mild steel electrode*	*packet*	0.5	2,000.00	2,000.00
25	*Angle iron* (25 *x* 25 *x* 1)	*mm*	2	1,800.00	3,600.00
26	*Angle iron* (38 *x* 38 *x* 4)	*mm*	1	2,800.00	2,800.00
27	*Mild steel plate* (1,219 *x* 609.6 *x* 1)	*mm*	1	16,000.00	16,000.00
28	*Labour charge*				42,600.00
29	*Transportation*				20,500.00
30	*Miscellaneous*				10,300.00
	Total				517,585

3.3 Performance Evaluation Result

Table 3: Performance evaluation result

Parameters	Ist Run	2nd Run	3rd Run	average
W_{fk} (Kg)	20	20	20	20
W_s (Kg)	8.8	9.2	9.4	9.13
W_n (Kg)	7.7	7.2	7.5	7.47
t (hr)	0.34	0.32	0.34	0.33
q_n	172	168	177	172.33
q_{sn}	129	134	126	129.67
q_s	18	16	19	17.67
q_b	23	20	21	21.33
PC (Kg/hr)	60			
ε_T (%)	83			
ε_{sh} (%)	75.25			
ε_{sp} (%)	89.75			
ε_b (%)	12.38			

Table 3, shows the result obtained from testing the newly developed palm kernel cracking machine. From the performance evaluation as seen in table 3, the machine gave a processing capacity (PC) of 60 Kg/hr, throughput efficiency of 83%, shelling efficiency of 75.25%, separation efficiency of 89.75% and a breakage efficiency of 12.38%. This shows that the machine is suitable for cracking and separation of palm kernel for rural farmers. These efficiencies were compared with a bar chart as shown in figure 5.

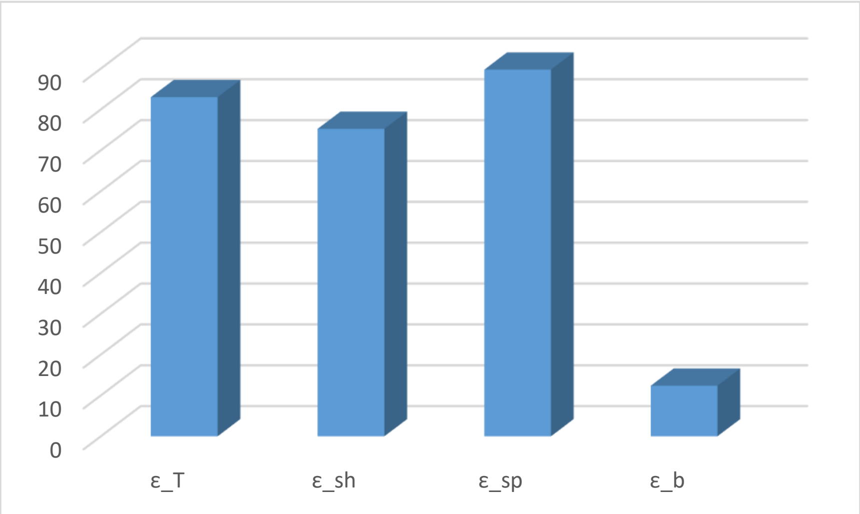

Figure 5: Efficiencies of the produced machine

Figure 5, compares of the efficiencies of the produced machine. It can be seen from the figure that separation efficiency has higher value followed by throughput efficiency, which shows that the performed utmost during separation. However high shelling efficiency was also recorded which shows that the machine performed well during shelling. Very low breakage efficiency was recorded as required, which shows proper performance.

4. CONCLUSION

The design and fabrication of a movable palm kernel cracking machine was successfully carried out. The performance evaluation test gave processing capacity (PC) of 60 Kg/hr, throughput efficiency of 83%, shelling efficiency of 75.25%, separation efficiency of 89.75% and a breakage efficiency of 12.38%. The obtained efficiencies were far higher than the traditional cracking machine. It also eliminates the stress of operating the traditional one, thereby satisfying the aim of embarking on the project. And it is appropriate to say that the cost of production can be greatly reduced when mass produced.

Therefore the produced machine is recommended for use in rural areas for the purpose of cracking and separation of nut from its shell for SMEs.

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