Development of a variable size nut cracker

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Abstract: In this work, a variable size nut cracking machine has been designed, constructed and tested to improve the efficiency of shelling operation of quite a number of nuts. The machine is designed to be adaptable to the cracking requirements of some number of nuts, whose shell can be processed to brittleness. The machine performs two operations: shelling of the nut and separation of the shell from the nut. The machine comprises of a hopper, rectangular box housing the cracking drum and compression plate, and also a two stage agitated separating tray (to sort the nut from the shell), and these are all supported by a frame. The cracking drum, which is driven by a v-belt connected to an electric motor, also provides the agitation to the separating tray via a v-belt connected to a cam mounted shaft that helps push the tray against stationary springs to return the tray to its initial position upon the dwell of the cam. The continual return and compression of the tray against the spring subjects the tray to a vibration needed to enhance the travelling of the shell-nut mixture over it. The machine was tested with palm kernel. The strongest of the class of nuts it was designed to crack with a cracking efficiency of 87%. The machine operation is satisfactory with whole kernel recovery of the machine standing at a magnificent value of 87%.

Keywords: Shelling, cracking, separating tray, agitation

1   Introduction

The extracts of some kernels such as palm kernel, cashew, macadamia, almond, and walnut, have found their usefulness both in industrial processes and domestic consumption purposes. Kernel extract has increasing application in soap making, glycerin, margarine, candle, pomade, oil paint, polish and medicine while the supposed waste-the cake serve as ingredient for livestock feeds and the fibres are used in boiler as fuel (Adebayo, 2004; Emeka and Olomu, 2007).

In view of the high utility of kernel and its products, the demand for it in the world markets is increasing daily. This, therefore, calls for a more sustainable means of production and the use of modern day technological advancement to provide an easier means of production in replacement of the traditionally adopted system.

Aside from the traditional method, the nut cracking have been based on principle of hurling of the palm nuts at a fairly high speed against a stationary hard surface (Okoli, 1997). Roller crackers and centrifugal impact cracker have also been employed (Oke, 2007; Badmus, 1990). An attempt to arrive at a more efficient design made some other researchers to innovate on the use of impeller blades enclosed in a cylindrical drum (Osunde and Oladeru, 2006). Amidst all these methods, there is the absence of an effective sorting mechanism of the cracked nut from its shells. As a mixture of shells and kernels, the product needs to be separated before becoming a useful product (Oke, 2007). Moreover, majority of these inventions failed to account for variations in the sizes of the nuts, therefore leaving some unbroken nut after the cracking process, thus affecting efficiency.

The thrust of this work is to design and construct a machine which will be an improvement over the existing ones, effectively crack and separate oil seeds of varying

size that are amenable to cracking under pressure from their shells.

2 Materials and methods

2.1 Machine description

The dual purpose machine is made up of three major components, the cracking unit, support frame and sorting unit. The cracking unit comprises of a pyramidal frustum hopper and rectangular box housing the cracking drum and compression plate, while the sorting unit is an agitated two stage separating tray supported by compression spring. The frame provides support and housing to all components of the cracking and sorting unit as well as the electric motor and the two pulleys which provide the motion driving the cracking drum and the separation tray. The cracking drum, which is driven by a v-belt connected to an electric motor, also provides the agitation to the separating tray via a v-belt connected to a cam mounted shaft that helps push the tray against stationary springs which in turn return the tray to its initial position upon the dwell of the cam. The continual return

and compression of the tray against the spring subjects the tray to a vibration needed to enhance the travelling of the shell-nut mixture over it (Figure 1-2).
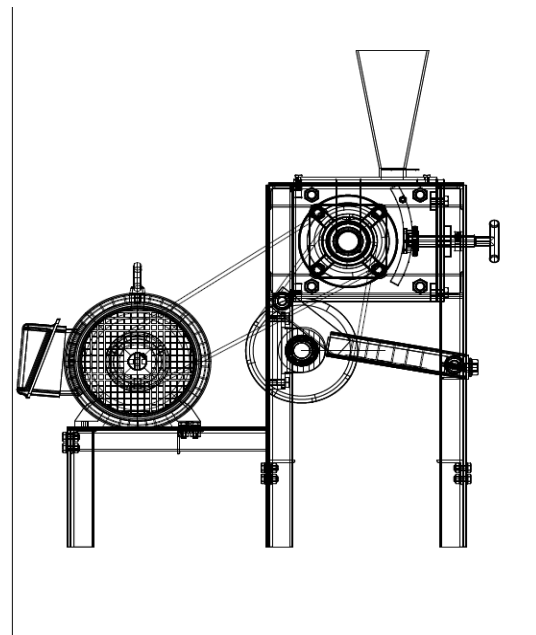

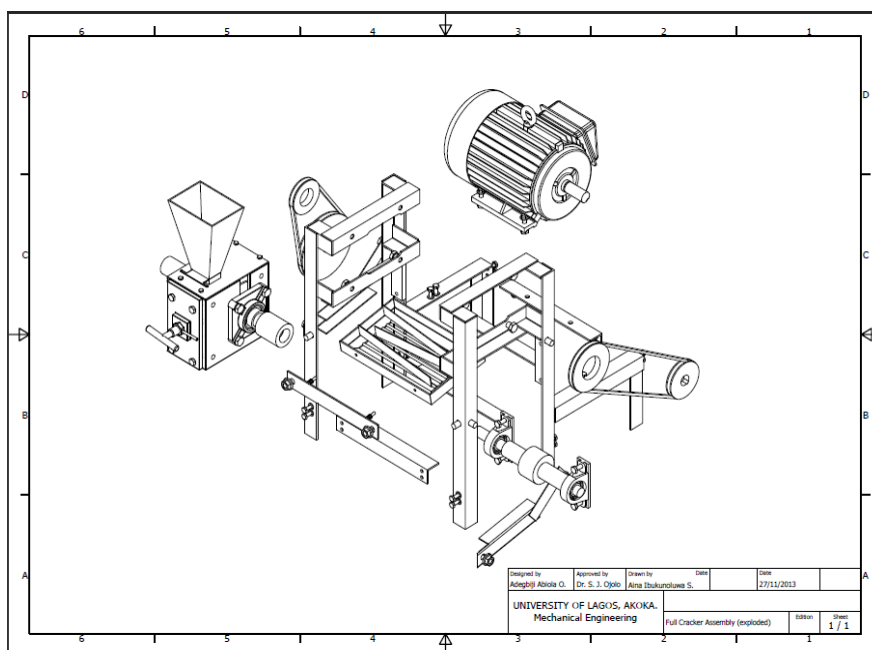Figure 1   Front orthographic view of the machine

Figure 2   Exploded isometric drawing of the machine

The machine was constructed using locally available engineering materials. All parts were made and joined to form a unit using basic engineering manufacturing techniques such as marking out, cutting, welding and fastening (Figure 3).
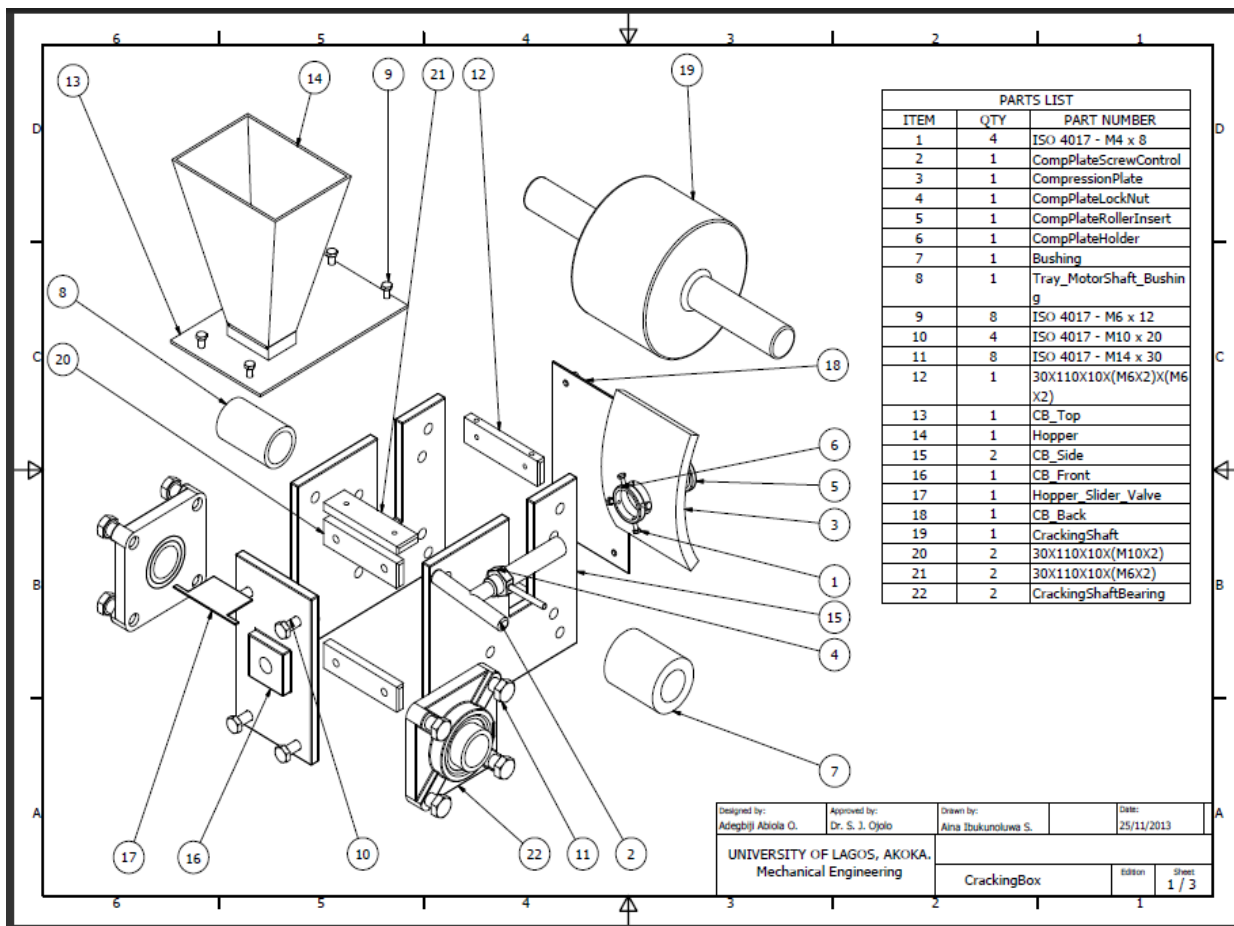
PARTS LIST		
ITEM	QTY	PART NUMBER
1	4	ISO 4017 - M4 x 8
2	1	CompPlateScrewControl
3	1	CompressionPlate
4	1	CompPlateLockNut
5	1	CompPlateRollerInsert
6	1	CompPlateHolder
7	1	Bushing
8	1	Tray_MotorShaft_Bushing
9	8	ISO 4017 - M6 x 12
10	4	ISO 4017 - M10 x 20
11	8	ISO 4017 - M14 x 30
12	1	30X110X10X(M6X2)X(M6X2)
13	1	CB_Top
14	1	Hopper
15	2	CB_Side
16	1	CB_Front
17	1	Hopper_Slider_Valve
18	1	CB_Back
19	1	CrackingShaft
20	2	30X110X10X(M10X2)
21	2	30X110X10X(M6X2)
22	2	CrackingShaftBearing

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Figure 3 Bill of materials of the machine

2.2 Design considerations and design of the machine elements

The choice of material was influenced by its availability in the local market, ease of workability with basic engineering techniques, rigidity, strength, overall weight of the machine and cost of production without compromising the efficiency, aesthetic and agronomical value. All these factors putting together make the machine affordable and adaptable for both domestic and commercial use. The following machine elements were designed using proven mathematical and engineering formulations.

2.2.1 Determination of speed of cracking drum

The impact energy / machine speed

This is the energy required for cracking.

$$Impact\ energy\ (Nm) = Kinetic\ energy = \frac{1}{2}mv^2$$

(Khurmi and Gupta, 2006)　　　　(1)

Where, m=Average mass of palm kernel nut (kg) = 0.0098 kg (Eric et al., 2009)

v = Speed of the cracking drum, m/s

Therefore, impact energy = $0.004915\,v^2$

Assuming a plastic collision between the nut and the surface of the compression shaft, then:

$$Impact\ energy = work\ of\ deformation,\ (w) = \frac{1}{2}F \times e$$

(Ojolo et al., 2010)　　　　　　　　(2)

Where F = the applied force or load (N) == $F = P \times r$

(3)

e = the deformation of the kernel, mm

P = the impact load applied to the kernel, N and

r = the ratio of the stress under impact to the direct stress or deformation under impact to the corresponding deformation.

$$r = \frac{\sigma'}{\sigma}\ \ and\ \ \sigma = \left(\frac{P}{A}\right)$$

(4)

Where, $\sigma' = 2\sigma$ (Shamma and Aggarawa, 2006)

$\therefore$ r = 2 and F = 2 P   (5)

Substituting for F in Equation 2

$$W = P.E = \frac{1}{2}mv^2 = 0.004915\,v^2 \qquad (6)$$

Where $P = 492N$ (Ezeoha et al., 2012) *and e =0.0032m* (Koya and Faborode, 2005)

Therefore, *v = 17.898m/s*

2.2.2 Determination of drum shaft diameter

$v = wr$ (Khurmi, 2005)   (7)

where r is estimated radius of cracking drum and, w = 260rad/s corresponding to angular machine speed of 2500 rpm (261.8 rad/s) employed by Eric et al. (2012).

Therefore, r = 68.84mm and d = 137.68mm

A drum diameter of 150 mm is chosen for the purpose of this design.

2.2.3 Determination of shaft power

Radius of gyration, $K = \dfrac{D}{4}$ (Khurmi, 2006)   (8)

Where D = 150 mm = 0.15 m

Therefore, K = 0.0375

$W = k \times \delta = \rho V g$ (Khurmi, 2006)   (9)

Where I_{xx} = the moment of inertia of the body, and

   m_{eq} = the equivalent mass of the body along the line of action of the tangential force.

Also, recall that for a cylindrical cross section as in the case of the cracking drums,

Moment of inertia, $I_{xx} = \dfrac{\pi d^4}{64}$ (Beers and Johnston, 2012)   (10)

Combining Equation 9 and Equation 10:

$$I_{xx} = m_{eq}K^2 = \frac{\pi d^4}{64}\,m_{eq} = 0.018\text{kg}$$

$F = m_{eq}a$ (Khurmi, 2006)   (11)

where, F = tangential force, and a = Linear acceleration

$a = w^2 r$ (Khurmi, 2005)   (12)

Substituting *a* into Equation 11

$F = m_{eq}w^2 r$ = 92.53 N

The torque required for rotation is given as:

$T = Fr$ = 6.94 Nm   (13)

The minimum power required is given:

$P_{\min} = Tw = 1.82\text{kW}$ (14)

Therefore, the minimum machine power required to crack the palm kernel nut is estimated to be 1.82 kW.

Centrifugal stress acting on the compression roller is given as:

$\sigma_\theta = \rho w^2 r^2 = \rho v^2$ (Khurmi, 2005)

ρ = density of steel = $7850\,kg/m^3$

$\therefore$ σ_θ = 2.515MPa

It is assumed that the centrifugal stress, σ_θ is equal to torsional stress, τ transmitted to the shaft, $\therefore$ τ = 2.515MPa

$\tau = \dfrac{Tr}{J}$ (Shigley, 2010)   (15)

$J = \dfrac{1}{2}\pi r^4$ = Polar moment of inertia of shaft and is given as

$\therefore \tau = \dfrac{2T}{\pi r^3}$ and $r = 12.1$mm

Therefore, $D = 2 \times r = 24.2$mm

The calculated diameter is 24.2 mm. For safety and a check against residual loads, D is taken to be 35 mm.

2.2.4 Design for spring

Estimated solid volume of tray, $V = S - s + s_r$ (16)

Where (S-s) = the solid volume of the rectangular tray

s_r = the total volume of the screening rod employed.

$S = L \times W \times H$ =0.001881 m^3 s = $l \times w \times h$ =0.001612 m^3

Volume of screening rod, $s_r = \pi r^2 h$, where r = 0.003 mm and h = 3.705 m

$s_r = 0.000105\, m^3$ and V = 0.000374 m^3

For safety, a dimensionless load factor of 2 is used for further computation on the tray.

Therefore, the solid volume of tray, $V_s = 2 \times V = 0.000748\, m^3$ and the solid mass of tray, M = 5.87kg

The desired amplitude of vibration, $\delta = 2$mm

Force in the spring, $F = k \times e$ (17)

F = W for static deflection and e = δ

$$W = k \times \delta = \rho V g \qquad (18)$$

Where, g = acceleration due to gravity = 9.81 m/s^2

Therefore spring stiffness, k = 28801.18 N/m

2.2.5 Determination of power required for agitation

Frequency of spring, $w_n = 2\sqrt{\dfrac{g}{\delta}}$ (Khurmi, 2006)

                                (19)

$w_n = 70.04$rad/s and $r = \dfrac{w}{w_n} = 3.74$

Transmissibility of amplitude:

$$\frac{x}{y} = \frac{1}{\sqrt{\left(1 - r^2\right)^2 + \left(2\zeta r\right)^2}}$$

(http://aerostudents.com/files/vibrations/solutions) (20)

Where ζ = damping ratio = 2% = 0.02 for steel, y = δ

$= 0.002$m = $m_o\left(\dfrac{er^2}{m}\right)$ and r = 3.74

Therefore, $x = 0.000266$m

Also, velocity of spring, $v = wx = 0.0695$m/s and acceleration of spring, $a = w^2 x = 18.23\, m/s^2$

Force due to the acceleration of spring, $F = ma = 101.01$N

where m = mass of the sorting tray and a, the acceleration of spring.

Agitating Torque $T = Fr = 2.718$Nm

where, r is the radius of the agitating shaft

Power required to agitate the tray, $P_V = Tw = 711.59$W

$$P_T = P_C + P_V = 2.53\text{kW} \qquad (21)$$

Where, P_T = total machine power required, P_C = power required to crack, P_V = vibration power required

2.2.6 Test for shaft suitability

With the total power obtained greater than the initial power value employed for computation in the analysis of the driving shaft, a quick check need to be carried out to ascertain the safe operation of the chosen shaft diameter of 35 mm.

Recall,

$P_T = Tw$

$\therefore\ T = 9.66$Nm

Also recall,

$r = \sqrt[3]{\dfrac{2T}{\pi \tau}} = 13.5$mm $D = 2r = 27$mm

Since 35mm > 27mm, the chosen shaft diameter is suitable for the design.

3 Results and discussion

3.1 Tests and results

     The machine was tested with palm kernel, which is the strongest of all the nuts for which it was designed. The palm kernel has a moisture content of 2.6% dry basis (Antia et al., 2014), hardness of 10.41kN/m^2 and compressive yield strength of 1.022 kN (Ezeoha and Akubuo, 2014). The test was carried out five times with forty (40) feed of palm kernel nut for each test and the number of cracked, uncracked, damaged cracked and undamaged cracked nuts were recorded from which the cracking efficiency, theoretical throughput, whole kernel recovery (WKR) and kernel breakage ratio (KBR) were estimated. Table 1 shows the test results and analysis.

$$\text{Cracking efficiency} = \frac{\text{Total number of completely cracked nuts}}{\text{Total number of feed nuts}} \times 100\% \tag{22}$$

$$\text{Whole Kernel Recovery (WKR)} = \frac{\text{Number of completely craked nuts with undamage kernel}}{\text{Total number of feed nuts}} \times 100\% \tag{23}$$

$$\text{Kernel Breakage Ratio}\,(KBR) = \frac{C_d}{C_d + C_u} \times 100 \text{ (Ndukwu and Asoegwu, 2010)} \tag{24}$$

Where, C_d = Cracked and damaged nuts, C_u = Cracked and undamaged nuts

$$\text{Sorting efficiency} = \frac{\text{Number of sorted nuts}}{\text{Total number of cracked nuts}} \times 100\% \tag{25}$$

3.2 Discussion

The results as shown in Table 1 reveal the number of cracked and uncracked nuts out of 40 feed per test. The damaged cracked nuts and undamaged cracked nuts are estimated from the cracked nuts. The average cracking efficiency is estimated to be 87% and the whole kernel recovery is estimated to be 87%. There are no damaged cracked nuts, thus the kernel breakage ratio is zero.

Table 1 Test Results and Estimation of performance criteria `

S/N	Feed (Nuts)	Cracked (Nuts)	Uncracked (Nuts)	damaged cracked (Nuts)	Undamaged cracked (Nuts)	Cracking efficiency (%)	WKR (%)	KBR
1	40	35	5	0	35	87.5	87.5	0
2	40	36	4	0	36	90	90	0
3	40	34	6	0	34	85	85	0
4	40	36	4	0	36	90	90	0
5	40	33	7	0	33	82.5	82.5	0
Mean	40	34.8	5.2	0	34.8	87	87	0

3.3 Theoretical throughput capacity of the machine (TTC)

The theoretical throughput capacity is the maximum allowable throughput capacity of the machine estimated on a theoretical basis.

$$TTC = \text{Max. surface discharge} \times \text{Angular speed of cracking drum} \tag{26}$$

x = Maximum surface discharge = 3 units

$TTC = v \times x = 7500$ units/minutes

One nut of palm kernel is equivalent to about 9.83 g

$\therefore\; TTC = 73.725$ kg/min

Therefore, if ideal operation can be maintained with adequate feed, the theoretical throughput capacity of the machine is about 73.7 kg/min.

The power requirement for cracking palm kernel nut, 1.82 kW and the theoretical throughput capacity 73.7 kg/min is relatively comparable to 1.76 kW and 94.2 kg/min obtained by Ojolo et al. (2009).

4 Conclusion

The machine performs satisfactorily when tested with palm kernel, the strongest of the class of nuts with a cracking efficiency of 87% and whole kernel recovery of 87%. The machine is adaptable to the cracking

requirements of some number of nuts, whose shell can be processed to brittleness and will improve the efficiency of shelling operation of quite a number of nuts. The machine can be modified for better efficiency by optimizing the parameters of design. Large scale commercial production of the machine could actually reduce the unit cost, thus making the use of the machine economical.

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