Development and Assessment of Cracking and Sorting Processes of Palm Kernel Nut Machine

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Abstract — This paper presents a research on the Development and Assessment of the cracking and sorting processes of palm kernels nuts in a mechanical cracking machine. The palm kernel oil is an important agricultural produce, it has a wide range of usage aside from cooking. In attempt to reduce energy exploitation, in conjunction with the regular high rate of demand for products been extracted from palm kernel nut, the need to improve on the manual method of cracking and sorting became eminent. The mechanical cracker was made up of two units: the cracking and sorting unit. Experimental studies of properties of machine components were considered with the purpose of avoiding fatigue failure and fracture on the palm nuts during the processing duration. The mechanical cracker was designed, fabricated and performance evaluation carried out using locally sourced materials. It was developed with a 5hp electric motor to drive the machine coupled with belts and pulleys. The average data assessment shows a 2.01% of un-cracked nuts, 2.36% of partially cracked nuts, 93.58% of un-cracked nut and 2.05% broken nuts. The cracking-sorting efficiency was estimated at 94% and throughput capacity was determined to be 80kg/h while the overall palm kernel recovery efficiency was 85%. These suggest that the machine is very suitable for separating the palm kernel nuts from the cracked shells clearly and completely.

Index Terms — Palm kernel, Cracking, Sorting, Efficiency, Feed rate.

I. INTRODUCTION

The palm kernel seed is the edible seed of the oil palm tree (Elaeis Guineensis) which originated from West Africa, it dates as far back as 5000 years and it can be used to obtain palm kernel oil which can be fractionated into liquid (olein), solid (stearin), and an intermediate fraction known as shortening. Palm fruit oil can be further broken down into diverse products such as vegetable oil, candles, soaps, ice cream, and as well as pharmaceuticals [1]. The three main varieties of palm tree are the Dura, Pisifera, and Tenera. The distinguishing feature of Dura is defined with thin epicarp, large nut and shell thickness of 2 to 5 mm. The Pisifera is in some cases referred to as shell-less variety while the Tenera is a combined form of Pisifera and Dura. The Tenera has medium size nuts, thick epicarp and shell thickness of 1.0-2.5 mm [2], [3]. Palm nut consists of the shell and the kernel; and are of economic importance [4], [5]. For some number of years, extraction of oil from oil seeds require a broad range of traditional and modern (mechanical and chemical) processes [14]. Separation of the oil from the palm kernel is a significant aspect of palm kernel processing. Although, the palm oil extraction process had undergone a lot of mechanical development, the palm kernel oil extraction process which starts with the separation of the palm nut from the fiber-base is still significantly less mechanized. Cracking palm nuts to release the kernel is therefore a critical step that affects the quality of oil [6]. The traditional cracking medium takes place in two forms, either by stone arrangements or by the use of mortar and pestle. This method of cracking and separating palm kernel nuts is time-consuming, low efficiency, labour intensive to meet high market demand and can cause serious and severe injuries to human beings. There are two mechanical actions used to crack palm kernel; shock caused by an impact against a hard object and the direct mechanical energy to crack, cut or shear through the shell [7]. The cracked nut mixture comprises of whole/broken kernels, shell particles, un-cracked nuts, fine particles and dust. The modern medium consists of the use of machines and equipment like the hammer mill, roller mill, centrifugal cracker, laser beam cracker, and many more that has a better advantage over the tradition medium.

Generally, there are two types of mechanical cracking machines; the simple nutcrackers and centrifugal impact crackers. In rollers type of centrifugal crackes, the nuts are cracked in between fluted rollers revolving in opposite directions. The clearance between the rollers is invariable but the nuts are of the different sizes, which makes the machine operates at reduced efficiency [8]. Due to the high level of demand for the products gotten from processed palm kernel nuts and the energy consumption rate, the need to improve on the local method of cracking gives way to the advance mechanism to ensure easy and fast access to the nut. The main purpose of this study is to produce an efficient palm kernel cracking and separating machine, then further carry out its performance evaluation.

II. METHODOLOGY

The palm kernel machine is made up of two units: the cracking and sorting unit. The essential design parameters considered include the size and strength of materials of individual machine components with the purpose of avoiding yielding and fatigue failure throughout the required duration of use of the machine.
A. The Design Analysis of the Components

1. Belt and Pulley Drive Mechanism

The palm kernel cracking and sorting machine requires two pulleys for operation: the first pulley is attached to the electric motor (driver) and the second pulley is attached to the shaft (driven) coupled with the hopper.

The transmitted speed was evaluated using the relationship given by [9]:

\[ N_1D_1 = N_2D_2 \]  \hspace{1cm} (1)

where,

- \( N_1 \) – the speed of the electric motor = 1725 rpm.
- \( D_1 \) – diameter of the pulley connected to electric motor = 58 mm.
- \( N_2 \) – the speed of the driven pulley = 1725 rpm.

The speed of the driven pulley is determined to be:

\[ N_2 = \frac{N_1D_1}{D_2} = 1136.9 \text{ rpm} \]

2. Power requirement to drive the machine

The power required to drive the machine can be determined from equation (2):

\[ P = (T_1 - T_2)V \]  \hspace{1cm} (2)

where

- \( V \) – velocity of the belt in (m/s).
- \( T_1 \) – tension in the slack side of the belt in (N).
- \( T_2 \) – tension in the tight side of the belt in (N).
- \( P \) – power needed to drive the machine in (kW).

The speed required to drive the pulley by the driver is calculated from:

\[ V = \frac{\pi D_1 N_1}{60} \]  \hspace{1cm} (3)

\[ V = \frac{3.142 \times 0.088 \times 1136.9}{60} = 5.239 \text{ m/s} \]

For 5hp (3.73 kW) electric motor,

\[ T_1 - T_2 = 711.97N \]  \hspace{1cm} (3a)

3. Length of Belt Required for Power Transmission

The length of the belt was determined from equation (4) [10]:

\[ L_{belt} = 2C + 1.571(D_1 + D_2) + \frac{(D_2 - D_1)^2}{4C} \]  \hspace{1cm} (4)

where

- \( L_{belt} \) – length of the belt required in (mm).
- \( D_1 \) – diameter of the smaller pulley (electric motor) = 58 mm.
- \( D_2 \) – diameter of the larger pulley (driven) = 88 mm.

C – center spacing of the machine and motor pulley =270 mm (0.27 m) has suggested by [15]:

\[ L_{belt} = 2 \times 270 + 1.571(58 + 88) + \frac{(88 - 58)^2}{4 \times 270} = 770.2 \text{ mm} \]

4. Angle of Contact between the Belt and Pulleys

The contact angle \( \alpha_1 \) and \( \alpha_2 \) for small and large pulley was estimated as follows:

a. For the electric motor (driver)

\[ \alpha_1 = 180 + 2\theta \]

\[ \theta = \sin^{-1}\left(\frac{D_2 - D_1}{2C}\right) \]

\[ \alpha_1 = 180 + 2\sin^{-1}\left(\frac{D_2 - D_1}{2C}\right) = 186.37^\circ = 1.0354 \text{ rad} \]

b. For the driven

\[ \alpha_2 = 180 - 2\theta \]

\[ \alpha_2 = 180 - 2\sin^{-1}\left(\frac{D_2 - D_1}{2C}\right) = 173.63^\circ = 0.9646 \text{ rad} \]

5. Belt Tension

The tension on the belt was determined using equation (5) [11].

\[ 2.3\log(T_1/T_2) = \mu \theta \]

When two pulleys of unequal diameters are linked by an uncrossed belt, the contact angles of the small pulley are taken into consideration for estimating tension \( T_1 \) and \( T_2 \). In addition, for rubberized V-belt, the coefficient of friction between the belt and the pulley is 30 (\( \mu = 0.30 \)).

Thus,

\[ 2.3\log(T_1/T_2) = 0.3\sin\left(\frac{88 - 58}{2 \times 270}\right) = 0.9554 \]

\[ T_1 = 2.603T_2 \]  \hspace{1cm} (6)

Solving equation (3a) and equation (6):

\[ T_2 = 444.15N \text{ and } T_1 = 1156.12N \]

6. Determination of the Torque in the Driven Pulley

The torque required in the driven pulley is determined from equation (7).
\[ T = (T_1 - T_2) \times \frac{D^2}{2} \quad (7) \]

\[ T = T_1 - T_2 = (1156.12 - 444.15) \times \frac{88}{2000} = 31.33 \text{ Nm} \]

The total load to be supported by the shaft is.

\[ T_{load} = T_1 + T_2 = 1600.27 \text{ N} \]

7. Determination of the hopper’s Capacity

The hopper is another significant component of the palm kernel crushing and sorting machine as it is the region that houses the feed nut before crushing takes place. The shape, dimension and location of the hopper were chosen to ascertain the mass discharge of the nut.

The capacity of the hopper is calculated from equation 8:

\[ V_h = \frac{H}{3} \left( a'b - x'y \right) \quad (8) \]

where

- \( a \) – length of large end of top section =0.29 m.
- \( b \) – width of large end of top section=0.29 m.
- \( x \) – length of small end of bottom section=0.15 m.
- \( y \) – width of small end of bottom section =0.12 m.
- \( H \) – height of the hopper =0.38 m.
- \( V_h \) – volume of the hopper in m³.

The volume of the hopper is estimated to be 0.0196 = 0.02 m³.

8. Centrifugal Tension in the Belt and Equivalent Twisting Moment

The centrifugal tension was determined using equation 9:

\[ T_{Centrifugal} = mV_{\text{max}}^2 \quad (9) \]

The selected rubberized class B type of V-belt has the following specification [11]:

- Density of the belt, \( \rho_{belt} = 1000 \text{ kg/m}^3 \).
- Maximum shear stress, \( \sigma_{\text{max}} = 2 \text{ MPa} \).
- Pitch length of the belt, \( l_{\text{pitch}} = 2.075 \text{ m} \).
- Belt dimensions, \( t = 0.015 \text{ m} \) and \( b = 0.025 \text{ m} \).
- The mass of the belt is determined from equation (10):

\[ m = \sigma_{\text{belt}} \times b \times t \times l_{\text{pitch}} = 0.78 \text{ kg} \quad (10) \]

The maximum tension of the belt is

\[ T_{\text{max}} = \sigma_{\text{belt}} \times b \times t = 750 \text{ N} \quad (11) \]

The belt maximum speed is evaluated from

\[ V_{\text{max}} = \sqrt{\frac{T_{\text{max}}}{3m}} = 18 \text{ m/s} \quad (12) \]

Hence, \( T_{\text{Centrifugal}} = 252.72 \text{ N} \).

The equivalent twisting moment is determined from:

\[ T_{\text{equiv, cent}} = \left\{ \left( m_j \right)^2 + \left( T \right)^2 \right\}^{\frac{1}{2}} \quad (13) \]

\[ T_{\text{equiv, cent}} = \left\{ \left( 24.36 \right)^2 + \left( 31.33 \right)^2 \right\}^{\frac{1}{2}} = 39.69 \text{ Nm} \]

The diameter of the shaft is determined using equation (14).

\[ d_s = \left( \frac{16T_{\text{equiv, cent}}}{\pi f_{\text{allowable, shear stress}}} \right)^{\frac{1}{3}} \quad (14) \]

Assuming an allowable shear stress of \( f = 55 \times 10^6 \text{ N/m}^2 \).

\[ d_s = \left( \frac{16 \times 39.69}{55 \times 10^6 \times 3.142} \right)^{\frac{1}{3}} = 0.0154 \text{ m} = 15.4 \text{ mm} \]

The calculated diameter of the shaft is multiplied by a factor of four (4) to allow a variation in twisting moment and the effect of other strain actions [11].

\[ d_s = 4 \times 0.0154 = 0.0616 \text{ m} \]

9. The Design for Stiffness and Rigidity of a Shaft

The torsional deflection of a shaft was determined using equation (15).

\[ \theta = \frac{TL}{GJ} \quad (15) \]

where \( J \) is the second polar moment of inertial shaft and is determined to be 1.4138\times10^6 \text{ m}^4 using equation (16).

\[ J = \frac{\pi d^4}{32} \quad (16) \]

Therefore, modulus of rigidity (\( G \)) can be determined as:

\[ G = \frac{31.33 \times 0.52}{3.185 \times 1.4138 \times 10^{-6}} = 3.62 \text{ MN/m}^2 \]

Strain energy in the shaft due to torsion is presented in equation (17).

\[ E_{\text{strain}} = \frac{1}{2}T^2 \theta \quad (17) \]

\[ E_{\text{strain}} = \frac{1}{2} \times 31.33 \times 3.185 = 49.89J \]

10. Design of the Impeller

The optimum speed needed for crushing the nut is given by equation (18) as.

\[ E_{\text{deformation}} = 0.5mV^2 \quad (18) \]
where the mass of the Dura and Tenera nut on average were 0.0069 and 0.0083 kg.

The energy of deformation \( E_{\text{deformation}} \) of the Dura and Tenera nut were determined experimentally as 901.2 and 2002 kNm [13].

Therefore, the linear speed values of both Dura and Tenera nut is evaluated to be 16.16 and 21.96 m/s.

The average speed needed to operate the machine is 19.06 m/s.

Similarly, with the impeller radius of (88 mm), the angular speed for both nuts is obtained using equation (19) to be 184 and 250 rad/s.

\[
\omega = \frac{v}{r} \quad (19)
\]

The average angular speed needed to operate the machine is 217 rad/s.

Lastly, the average rotational speed \( N_{av} \) required to operate the machine is determined from:

\[
N_{av} = \frac{60}{4\pi} \{N_{\text{dura}} + N_{\text{tenera}}\} \quad (20)
\]

where

\[
\omega = \frac{2\pi N_{av}}{60} \quad (21)
\]

\[
N_{av} = \frac{60}{4 \times 3.142} \{184 + 250\} = 2071.2 \text{ rpm}
\]

11. Determination of the Impact Force

The cracking force is determined using equation (22)

\[
F = m\omega^2 r \quad (22)
\]

where

\[
m = \frac{I_{xx}}{k^2} \quad (23)
\]

Here, \( I_{xx} = \frac{bh^3}{12} \) and \( k = 0.289 \) h as documented in [11].

Now, \( m = \frac{bh^2}{3.468} = 0.0121 \) kg.

According to equation (22), the crushing force is now determined to be 47.86 N.

12. Throughput Capacity (TC) and Efficiency

The throughput capacity is the quantity of material moved or produced per unit time. That is, it is the measured processing speed of a machine expressed as total output in a unit period usually in hour under normal operating conditions. It can be measured as gravimetric or volumetric.

The throughput and efficiency is estimated by (24) to (27) [12].

Thus,

\[
\text{Gravimetric throughput capacity (kg/h)} = \frac{\text{Mass of cracked nut}}{\text{Cracking time}} \quad (24)
\]

13. Cracking and Sorting Efficiency

\[
\text{Cracking – Sorting Efficiency} = \frac{\text{No. of Completely shelled nuts}}{\text{No. of nuts feed}} \times 100 \quad (25)
\]

\[
\text{Kernel Nuts Recovery Efficiency} = \frac{\text{Kernel Nuts recover}}{\text{Total Kernel feed}} \times 100 \quad (26)
\]

\[
\text{Deformed Kernel Nuts Efficiency} = \frac{\text{Deformed Nuts}}{\text{Total Kernel feed}} \times 100 \quad (27)
\]

(a) Exploded drawing

(b) Assembled drawing

Fig. 1. Isometric View of the cracking and sorting machine.

<table>
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<th>TABLE 1: PART LIST NOMENCLATURE</th>
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III. RESULTS AND DISCUSSIONS

After fabrication, the results of the test analysis conducted on the palm kernel cracking and separating machine were presented in figure 1-5. The un-cracked nuts, partially cracked nuts, unbroken nuts, broken nuts and the throughput of the nuts are in the following ranges of time: 7, 11, 21, 30, 39 and 47 sec. for cracking 29, 58, 85, 112, 145 and 170g respectively. This trend is notified for the two types of nuts considered here namely; Dura and Tenera nut respectively. The cracking efficiency was estimated according to the number of absolutely peeled and sorted nuts per group of 100 nuts in accordance with the variation in feed rate and kernel size.

Fig. 4 illustrates the percentage rate of the un-cracked nuts on the transient nuts per batch fed into the cracking chamber. In this case, it is observed that the un-cracked nut passes from a minimum value at the very beginning of the transient state towards a maximum value. Furthermore, the un-cracked nuts increase with the increases of the feed rate. The variation of the unbroken nuts with the number of nut fed into the cracking chamber per batch is shown in Fig. 5; the effect of increasing the nut into the cracking chamber decreases the rate of unbroken nuts. However, for high value of nuts fed into the cracking chamber, the low values of unbroken nut are being discharged from the outlet of the machine.

Fig. 6 shows the behaviour of percentage rate of broken nuts on the nuts per batch fed into the cracking chamber at different time. In this case, it can be noticed from figure 6 that as the nuts fed into the chamber increases in value at the initial state results to increases of the broken nuts value.

Thus, partially cracked nuts increase with the decreasing of the nut fed into the cracking chamber and vice versa in Fig. 7. The graph for the un-cracked nuts, unbroken nut, partially broken nuts, and broken nut for different values of number of feed rates parameters with time variation are illustrated in Fig. 8. The graph shows the performance evaluation of the distribution data on an average of 2.01%
of un-cracked nuts, 2.36% of partially cracked nuts, 93.58% of un-cracked nut and 2.05% broken nuts.

Fig. 6. Broken nut versus number of nut per batch for different cracking time.

Fig. 7. Partially cracked nut versus number of nut per batch for different cracking time.

Fig. 8. Comparison of un-cracked nut, unbroken nut, broken nut and partially cracked nut per batch.

Fig. 9. Through put versus cracking time for different variation of nut per batch.

Fig. 10. Through put versus nut per batch for different variation of cracking time.

IV. CONCLUSION

The assessment of the developed cracking and sorting processor was successfully carried out with an overall palm kernel recovery efficiency of 85% and throughput of 80kg/h while the un-cracked nut increases with the increases of the feed rate. These metamorphoses in the production of palm kernel crackers with improved standard are a major plus to the agricultural production field of study. It was designed and fabricated using locally sourced, low cost and readily available materials which translates to affordability of machineries for the farmers. The results realised from the evaluation significantly shows that the machines are satisfactory for onward acceptance in the processing industries. In addition, it was developed considering maintainability; this makes it require little or no training for its operation and maintenance.

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