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Development and Performance Evaluation of a Palm Kernel Oil Extraction Machine

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ABSTRACT

Oil palm is a valuable crop and of considerable commercial importance because of its major products, which are red palm oil and palm kernel oil. The processing of the palm fruit locally is crude and tedious for the local processors in Nigeria. The drudgery and time spent in extracting palm oil discourage the youth and affect the productivity of palm oil in rural communities. The primary objective of this work is to design and fabricate an efficient palm kernel oil crushing machine. This machine, featuring high torque and low rotational speed, incorporates an adjustable choke mechanism. This mechanism allows precise regulation of back pressure, controlling the thickness and dryness of the pressed cake as it passes between the barrel and the crushing shaft. Key components, including the hopper, crushing chamber, shaft, pulleys, and belts, were designed and then fabricated using standard techniques such as cutting, welding, drilling, bending, and casting. The innovative design processes cold palm kernel seeds without pre-treatment; the crushing shaft within the barrel performs both breaking, cooking, and palm oil extraction processes. Driven by a 10hp electric motor, a processing capacity of 100 kg per hour gives an oil extraction efficiency of 87.10%. The developed machine has a high oil extraction efficiency and material discharge efficiency. Machine processing of palm oil extraction will alleviate the rigors of manual processing and enhance product quality.

INTRODUCTION

Palm oil is a globally significant component of the vegetable fruit items. Nigeria currently holds the position of the third-largest palm oil producer globally, trailing behind Indonesia and Malaysia. Both in the pre-independence era and the post-independence period, the palm oil extraction process and other agricultural-related products have remained a major source of employment for skilled and unskilled workers, significantly contributing to improved living standards among the Nigerian population (Alam and Fakhru, 2013).

Nigeria's population is conservatively estimated at 190 million, and the local demand for palm oil continues to rise due to rapid population growth within the country. Furthermore, on a global scale, the consumption of palm oil has seen a significant increase, surging from 14.6 million tons in 1995 to 61.1 million tons in 2015, establishing it as the most widely consumed oil worldwide. Key consumers of palm oil include China, India, Indonesia, and the European Union. Notably, India, China, and the EU heavily rely on imports to meet their palm oil demands as they do not produce crude palm oil domestically. In 2015, these three regions accounted for a substantial 47.9 percent of global imports (Owolarafe and Oni, 2011; Orhorhoro *et al.,*2016).

Processing this item becomes imperative to prevent wastage. Traditional methods of palm oil extraction offer lower efficiency and risk of contamination from dirt, microorganisms, and other impurities. This study, therefore, aimed to develop a machine that will extract palm kernel oil using locally available material (Alam and Fakhru, 2013). Extensive research and development efforts in various engineering disciplines have led to the evolution of a well-designed sequence of processing steps for extracting high-yield, quality palm oil suitable for the international edible oil trade. The oil extraction process involves receiving fresh fruit bunches from the plantations, sterilizing and threshing them to release the palm fruit, followed by mashing the fruit and pressing out the crude palm oil. The crude oil undergoes further treatment to purify and dry it, making it ready for storage and export (FAO, 2005).

Modern methods of palm oil extraction offer significant advantages over traditional methods, leading to increased efficiency, improved quality, and greater sustainability. Modern extraction methods, such as screw presses and solvent extraction, can recover a much higher percentage of oil from the palm fruit compared to traditional methods. This translates to increased productivity and profitability. The process significantly reduces the time required for each stage of the extraction process, from sterilization and stripping to pressing and purification. This allows for larger volumes of palm oil to be processed in a shorter time (Patil *et al.,* 2017; Glavič *et al.,* 2020).

The palm oil extraction process begins when fresh palm fruit bunches are fed into the hopper, where they are mechanically conveyed into the crushing chamber. Inside the crushing chamber, the palm fruits are subjected to pressure and shear forces from a rotating shaft, which separates the oil from the solid parts. The oil extract flows through a mesh filter to remove any remaining solid impurities (Laurance *et al.,* 2010). Large-scale plants capable of producing palm oil to meet international standards generally handle between 3 to 60 tonnes of fresh fruit bunches per hour. These installations are equipped with mechanical handling systems such as bucket and screw conveyors, pumps, and pipelines, allowing for continuous operation, dependent on the availability of fresh fruit bunches (Nzeka, 2014). However, due to the growing shift towards mechanized palm oil extraction in Nigeria, this research aims to increase the productivity of palm oil, improve product quality, reduce labour dependency, and enhance the overall sustainability of the palm oil industry through the fabrication of a palm oil extraction using locally available materials.

METHODOLOGY

Materials

The materials used include: mild steel plate; medium carbon steel shaft; mild steel U-channel; angle bar; electric motor; speed reducer, roller bearing, bolt, and nuts. Factors considered when selecting materials for fabrication of the machine include: availability of the materials, formability, mechanical strength, good corrosion resistance, and cost effectiveness.

Methods

Design of machine components

Drive Power: The machine drive torque, T and power, P, were computed as 1392.75Nm and 9.90Hp, respectively, using Equation (1):

$$T = F \times L \times n \qquad (1a)$$

$$P = F \times \omega \times \eta \qquad (1b)$$

where: F = compressive yield load for a mixture of the palm kernel = 619N (Ezeoha *et al*., 2012); L = length of crushing camber = 400mm; n = number of crusher pitches = 5, ω = angular velocity of screw shaft $\frac{2\pi N}{60}$, N = screw shaft speed = 40rpm and η = electric motor efficiency = 0.79. A 10hp electric motor with a speed of 1440 rpm is selected for this machine.

Electric motor/gearbox speed ratio: The input speed of the gearbox pulley (N_2) was determined as 600rpm and the screw press shaft speed (N_3) was computed as 40rpm from Equation (2):

$$N_2 = \frac{d_1}{d_2} \times N_1 \quad (2a)$$

$$N_3 = \frac{1}{g_r} \times N_2 \quad (2b)$$

where: N_1 = electric motor speed = 1440rpm; d_1 = electric motor pulley diameter = 125mm and d_2 = gearbox pulley diameter = 300mm, N_3= gearbox pulley speed = 600rpmand g_r = gearbox speed ratio = 15.

Tensions in the belt: Following the work of Olunlade *et al.* (2018), the tension T_1acting on the tight side of the belt and tension T_2 acting on the slack side of the belt were computed as 989.41N and 197.88N using Equation 3.

$$T_1 - T_1 = \frac{60\ P\ F_s}{\pi\ d_1 N_1} \quad (3)$$

$$T_1 + T_1 = \frac{60\ P\ F_s}{\pi\ d_1 N_1} \quad (4)$$

where: P = motor power = 7460W; d_1 = motor pulley diameter = 0.125m; N_1= speed of the motor = 1440rpm and F_s = service factor = 1.5 (heavy-duty machine).

Power transmitted per belt: Power transmitted per belt was computed as 7459.99W using Equation (5) as:

$$P = (T - T_2)V_b \quad (5)$$

Where T = tension acting on the tight side of the belt = 989.41N; T_2= tension acting on the slack side of the belt = 197.88N;d_1 = motor pulley diameter = 0.125 m; N_1 = speed of the motor = 11440rpm and $V_b = Velocity\ of\ the\ belt = \frac{\pi\ 179.9}{60} = 9.42\frac{m}{s}$.

Number of belts required for the drive: The number of belts required for the drive was computed as 2 using Equation (6):

$$No\ of\ the\ belts = \frac{Designed\ power}{Power\ transmitted\ per\ belt} \quad (6)$$

where: Designed power = 7460W and power transmitted per belt =7459.99W.

Centre distance, C: The center distance between motor and gearbox shafts was computed as 337.5mm using Equation (7), following the experimental work of Olunlade et *al.* (2018)

$$C = 3r_1 + r_2 \quad (7)$$

Where: r_1 = radius of motor pulley = 62.5mm and r_2 = radius of gearbox pulley = 150mm.

Length of belt, L_bThe length of the belt was computed as 1365.27mm using Equation (8) as:

$$L_b = 2C + \pi(r_2 + r_1) + \frac{(r_2 + r_1)}{C} \quad (8)$$

where: r_1 = radius of motor pulley = 62.5mm, r_2 = radius of gearbox pulley = 150mm and C = center distance between motor and gearbox shafts = 337.5mm.

Belt specifications: The power transmitted per belt is 7459.99W, and hence we select the B series V-belts (power range of B V-belt range: 2 – 15kW per belt). The length of the V-belt selected was 1415mm, which is nearer to L_b = 1365.27mm calculated (Khurmi and Gupta, 2009).

Pulley specifications: The pulley face width was calculated as 44mm using Equation (9), according to Khurmi and Gupta (2009)

$$B = (n - 1)e + 2f \quad (9)$$

Where: n = number of V-belt grooves on pulley, e = 19mm and f = 12.5mm (Khurmi and Gupta, 2009).

Calculation of bearing pressure σ_{bear} on the electric motor pulley hub: The torque being transmitted from the 125mm diameter electric motor pulley to the gearbox input shaft via the 300mm diameter pulley is vibratory and heavy. Moreover, the diameters of the electric motor and the gearbox input shafts are both greater than 22mm; hence, the rectangular taper keys were selected for design, in accordance with BS 4235: Part 1:1972. And, since the rectangular taper key is wider than it is deep, the key will fail in compression before it will fail in shear (Melk *et al*., 2018). Hence, the bearing pressure on the key between the motor shaft and pulley was computed as 4.12N/mm^2 using Equation (9) as:

$$\delta_{bear} = \frac{2\,M_t}{(h-t)l_d} \qquad (9)$$

where: M_t = torque transmitted = 49470 Nmm, b = 16mm,l_d = length of key =6b = 96mm, h = 10mm, d_{sm} = motor shaft diameter = 50mm and t_1 = 5mm.

Since, $\delta_{bear} \leq [\delta_{bear}]_{all}$ = 70N/mm^2, the design is satisfactory from the standpoint of bearing pressure.

Calculation of bearing pressure σ_{bear} on gearbox input shaft pulley hub: The bearing pressure on the key between the gearbox shaft and pulley was computed as 9.89 N/mm^2 using Equation (10) as:

$$\delta_{bear} = \frac{2M_t}{(h - t_1)l_d d_{sg}} \qquad (10)$$

where: M_t = torque transmitted = 118730 Nmm, b = 16mm, 1_d = length of key =6b = 96mm, h = 10mm, d_{sg} = motor shaft diameter = 50mm and t_1 = 5mm. Since, $\delta_{bear} \leq [\delta_{bear}]_{all}$ = 70N/mm^2, the design is satisfactory from the standpoint of bearing pressure.

Calculation of Shaft Diameter: The shaft was designed from ductile materials. The design of the shaft of ductile materials based on strength is governed by the maximum shear theory of failure. Hence, applying the ASME code equation for the design of transmitting a solid shaft under torsion and bending, the shaft diameter was computed as 53.29mm using Equation (11) as:

$$d^3 = \frac{16}{\pi\tau_{all}}\sqrt{(K_b M_b)^2 + (K_t M_t)^2} \qquad (11)$$

Where$\tau_{all} = \frac{0.5\,\alpha_y}{sf} = 60\ MN/m^2$; σ_y= yield strength for shaft material = 60MN/m^2;sf= factor of safety 4; K_t = combined shock and fatigue factor applied to torsional moment = 1.0; K_b = combined shock and fatigue factor applied to bending moment = 1.5;M_t = Torque on shaft = 1780.94Nm and M_b = bending moment on shaft = 52.5Nm.

Calculation of Angular Deflection of Shaft: The design of the shaft for rigidity was based on the permissible angle of twist α. The permissible angle of twist, α, varies between 0.3°/m ≤ α < 3°/m for line shafting. And the permissible angle of twist was computed as 1.0°/m using Equation (12) as:

$$\alpha = \frac{584\,M_t}{Gd^4} \qquad (12)$$

Where: M_t = Torque on shaft = 1780.94Nm; l = length of shaft =1000mm; G = modulus of rigidity = 80 x 10^{9}N/mm^{2}and d = shaft diameter = 60mm.

Calculation of lateral deflection of shaft: The deflection at the mid-point of the simply supported shaft of length l = 1000mm in which W = 300N is at a distance 350mm from each bearing support was computed as 2.33 x 10^{-5}mm/m using Equation (13) as:

$$\delta = \frac{W\,l^3}{48\,E\,I} \qquad (13)$$

Where: w = load on screw shaft = 300N; l = length of shaft between bearings 7000mm; E = Modulus of elasticity = 207 x 10^{9}N/m, and $I = \frac{\pi\,d^4}{64} = 6.36 \times 10^{-7}\text{m}^4$

Since $\delta_{max} < \delta_{all}$, then the design is satisfactory based on of lateral deflection for machining shafting.

Calculation of critical speed of shaft: Using the Rayleigh–Ritz equation, the critical speed of the shaft was computed as 6196.24rpm using Equation (14) as:

$$W_c = \frac{60}{2\pi}\sqrt{\frac{g}{\delta}} \quad (14)$$

Where: δ = deflection at the mid-point of the simply supported shaft = 2.33 x 10^{-5}mm/m and g = acceleration due to gravity = 9.81m/s^2. Since, W_c = 6186.24rpm is far greater than 40rpm, then the design is satisfactory based on critical speed analysis.

Design of Screw on Shaft: The shaft is required to carry a screw on its surface to enable it to perform the crushing and conveying operations of the kernel seeds. The screw adopted for this design was the square thread made from a 12.5 x 12.5mm mild steel rod. This was chosen over having the screw machine on high-carbon steel because of the availability of the rod and the cost of machining. The major drawback is that the screw has to be replaced from time to time on the shaft because of wear. Tests show that the screw can last for 12-18 months.

Determination of the Pitch of the Screw: Since the screw size, h = 12.5mm^2 and shaft diameter, d_s=60mm, then we have the following specifications: screw diameter, d_m = d_s + 2h = 85mm ;internal diameter of barrel, d_i = d_s + 2h +5 = 90mm; and pitch of the screw, p = 85mm.

Reaction on the Screw Shaft: The reaction on the screw thread, R mainly due to the torque being transmitted to the screw shaft and was computed as 41904.47N from Equation (15) as:

$$R = \frac{2\,M_t}{d_m} \quad (15)$$

where: d_m = screw diameter = 85mm and Mt = torque transmitted = 1392.47Nm.

Maximum pressure in the Barrel: The maximum pressure in the press barrelσ_{max}, was computed using Equation (16) as:

$$\sigma_{max} = \frac{12.55\ N/mm^2}{\pi\, n\, d_m h} \quad (16)$$

Where: d_m = screw diameter = 85mm; h = the screw height= 12.5mm; n = number of screw pitches and R = reaction on the screw thread = 41904.47N.

Design of hopper: The volume of the hopper through which palm kernels are fed into the press barrel for palm kernel oil extraction was computed as 0.017m^3 from Equation (17) as:

$$V_b = \frac{1}{3}(a^2 + ab + b^2)H \quad (17)$$

Where: a = top length of hopper = 300mm, b =base length of hopper = 100mm, and H = height of hopper = 400mm.

Design of Barrel: The barrel interior was designed in such a way as to increase the friction between the palm kernel seeds and the barrel walls as the shaft rotates. This is done in order to reduce the degree of slippage and rotation of the pressed cake with the shaft. The barrel is divided into two halves for easy, regular maintenance.

Design of Barrel Thickness: The thickness of the barrel was computed as 6.81mm by using the modified thin wall formula (Khurmi and Gupta, 2009), as:

$$t = \frac{P \times d_m}{2 \times \sigma_{all}} + C \quad (18)$$

where: σ_{max}=12.55N/mm^2, σ_{all} =140N/mm^2 and C= 3, for steel pipe, barrel thickness = 10mm and external barrel diameter, d_o = d_i + 2t = 90 + 2(10) = 110mm.

Design of crushing arms thickness and width: Since the crushing is split into two halves, bolts are then needed to lock them together. Fig.1 shows a cross-section of the barrel, the barrel arms the formwork. The crushing arm thickness specifications are made of a 20 mm width; lower crushing arm width = 10mm; crushing arm length = 120mm; bolt sizes – M10 x 1.5 x 45 and number of bolts = 8.

Machine Capacity: The theoretical capacity of the expeller was determined as 117.93 kg/hr using a modified form of the equation given by (Adetola *et al.,*2014) as:

$$Q = 60 \frac{\pi}{4} (d_m^2 - d_s^2) P\, N\, \varphi\, \rho \qquad (19)$$

Where, d = diameter of the screw thread = 85mm, d_s = base diameter of the screw shaft = 60mm, p = screw pitch = 85mm, N = rotational speed of the screw shaft = 40rpm, φ = filling factor = 0.8, and ρ = bulk density of palm kernel seed = 604kg/m^3.

Design of Flange Coupling: The shaft speed is low and there is an accurate, rigid axial alignment; hence, the rigid flange coupling was adopted. The flange coupling is welded to a hub, which is then held to the screw shaft by the fusion of bolts and nuts to allow for maintenance. The design of the flange diameter and the web thickness for geometry and strength was done by assuming that: bolts are finger tight and the load is transferred from one-half of the coupling to the other by a uniform shear stress in the shank of the bolt; the shaft is subjected to shock and fatigue and the bolts are uniformly arranged in the flange on bolt circle diameter. The results obtained are: pitch circle diameter = 125mm; flange diameter = 200mm; number of bolts = 6; web thickness = 15mm and bolt specification is: M12 x 1.5 x 60.

Design of Flange Hub: The diameter of the locking bolts, the number of bolts, as well as the length of the hub were calculated by assuming that the allowable bending stress between the shaft and bolts is less than or equal to the allowable bending stress in the shaft/hub materials. They are: hub diameter = 90mm; hub length = 105mm; hub thickness = 20mm and bolts specification: M16 x 2.5 x 120.

Selection of Bearings: Bearings are selected based on load type and size. Due to the nature of the load on the machine, a tapered-roller bearing was selected in accordance with the SKF General Bearings Catalogue (1987). From the reference catalogue, series 32310 tampered roller bearings of bore 50mm were selected. Overall radial load; due to the loads on the shaft and the cantilever load imposed on the shaft by the coupling, was determined. Also, the rated life of the more heavily loaded bearing at the support near the choke mechanism was determined. The rated life for the series 32310 tapered-roller bearing at this support was found to be: L_h = 15,129.33hours. Since the basic rating life for a crushing machine working for 10 hours or less is between 10,000 and 25,000 hours, the bearing selected for use is very adequate.

Design of Foundation Bolts: The diameter of the foundation bolts used to hold the machine to the foundation platform was calculated by assuming that the bolts will fail in shear under the action of the axial force in the barrel, which tends to move the machine formed along the foundation platform and a frictional force exists between the machine framework and the foundation platform. Equating the resistance offered by n bolts (n = 4) with the axial forces less the frictional force between the machine framework and the foundation platform, the foundation bolt diameter is 10mm.

Design of Foundation Block: When the machine is vibrating, the mass of the foundation should be three to five times the mass of the machine (Serinivasulu and Vaidyanathan, 1990). Hence, the weight of the foundation block is taken as equal to 4 times the weight of the machine. Given: weight of the machine = 276kg and assuming the density of concrete to be 1500kg/m^3, length of foundation block = 1800mm and width of foundation block = 1200mm; then the depth of foundation = 426mm.

Description of the Machine: The detailed drawings of the palm kernel oil expeller machine are shown in and 5. The machine consists of different parts such as: the hopper; the frame; the barrel, the electric motor, the pulleys, the gearbox or speed reducer, the screw shaft, the adjustable cone, the bearings, the oil and press cake receptacles. The Palm kernel oil expeller machine consists of: the hopper shaped as a truncated frustum of a pyramid, with top and bottom having rectangular forms, is mounted on the barrel with shut off slide; the framework of outline dimension 1320mm long x 800mm high x 800 mm wide; bolted to the foundation platform; two pulleys (125mm diameter at the electric motor shaft and 300mm diameter at the gearbox of input shaft); a gearbox of speed ratio 15:1; a rigid flange coupling between the gearbox and the screw shaft; a screw shaft of length 1000mm and diameter 60mm, coupling to the gearbox; a barrel with two arms (80mm and 130mm wide respectively) and length 450mm and diameter (90mm internal and 110mm external); the choke mechanism screwed to the screw-shaft at the end of the barrel; two, 32310 series tapered roller bearings, located at the two ends of the shaft; a hopper with shut off slides, mounted on the barrel at its beginning; and two trays underneath the barrel, one to collect palm kernel oil and the other to collect the pressed cake.

Working principles of the Machine: The kernels are fed to the screw press via a hopper mounted on the barrel with a shut-off slide. As the screw shaft rotates inside the cylindrical barrel, kernels are crushed against the barrel walls as well as conveyed toward the far end of the screw-shaft, where a steel adjustable choke mechanism presses the crushed kernels against the wall of the barrel to squeeze out the kernel oil. The kernel oil then flows out of the barrel through small drainage slots at its bottom into an oil tray, while the “pressed cake” discharges from the cone section into another tray. By regulating the clearance between the barrel and the adjustable cone (i.e., choke mechanism), the thickness and dryness of the pressed cake are regulated with this choke mechanism.

RESULTS AND DISCUSSION

Design of Palm Oil Extractor Machine

The design of the palm oil extractor machine was based on mechanical crushing, which can extract every possible drop of the palm kernel oil. This is to ensure a better and more effective performance of the palm oil expeller. The major components of the developed palm oil extraction machine include: the base stand, motor, shaft, pulleys, belt, bolts, nuts, hopper, and stirrer. Figures 1 and 2 show the autographic and isometric views of the developed palm oil extraction machine. The palm kernel fruits are pushed through the chamber, crushed, and the oil is separated from the solid parts, i.e., fibers and kernels. The extracted oil flows through a mesh filter to remove any remaining solid impurities.

The leftover cake, made up of palm kernels and fibers, exits from the rear of the press. The generator powers the shaft, enabling consistent rotational speed and compression that are optimized to maximize oil yield while maintaining energy efficiency. Figure 3 shows the pictorial view of the fabricated palm oil extraction machine. Figures 4 and 5 show the orthographic and isometric views of the palm kernel oil expeller machine fabricated. The palm kernel oil expeller machine is composed of the following parts: pulleys, gearbox, belt, keys, spline hub, bearings, bolts/nuts, and an electric motor.

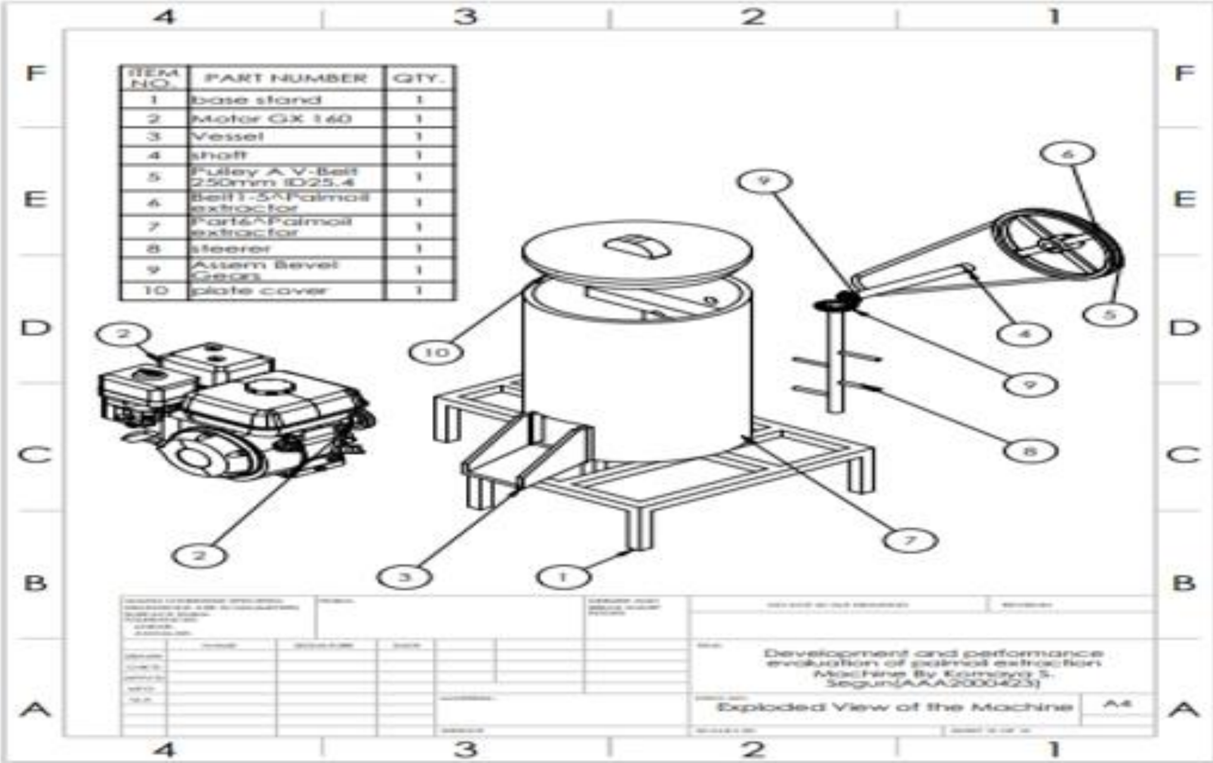

Figure 1: The Exploded View of the Palm Developed Oil Extraction Machine

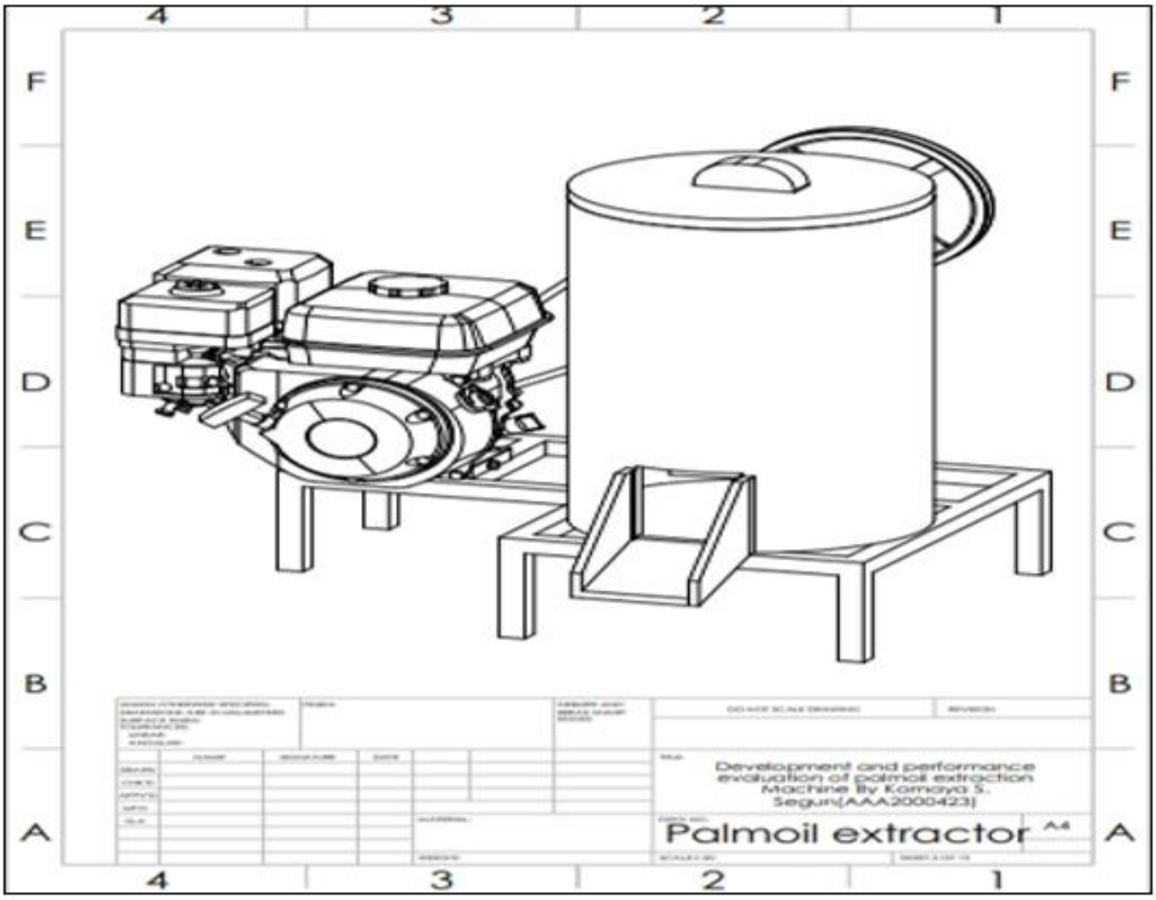

Figure 2: The Isometric View of the Palm Oil Extraction Machine

Figure 3: Fabricated Palm Oil Extraction Machine

The machine parts list is shown in Table 1. Construction of the various components was achieved using various fabrication techniques such as cutting, welding, drilling, bending, and casting.

Performance Parameters of the Palm Oil Machine

The performance of the palm oil extraction machine was assessed based on several parameters, including the machine's capacity and efficiency. The volume of the machine cylinder was found to be 1.69 m^3, and it can accommodate a maximum of 100 kg of mesocarp at a time. This ensures that the machine is capable of processing substantial quantities of palm fruits in a single cycle, thereby increasing its overall productivity.

Table 1 shows the density of crude palm oil at room temperature. The density values vary from 0.6530 g/ml^3 to 0.6364 g/ml^3. The as-received palm kernel for 5%, 10%, and 15% moisture content is tabulated and presented in Table 1a. The result shows that 5% conditioned moisture content has the highest oil yield of 11.56% followed by 10% conditioned moisture content with 12.07%, 15% conditioned moisture content has the least weight content of 12.08%. Table 1b shows the throughput efficiency (kg/hr) for 5% conditioned moisture content. A maximum throughput efficiency of 89.50 kg/hr was obtained at 47.6 rpm for 0.45 hours. A throughput efficiency of 87.10 kg/hr was obtained for 10% conditioned moisture content for 0.39 hours (Table 1c).

Similarly, Table 1d shows a 15% conditioned moisture content sample. A maximum throughput efficiency of 93.4% was obtained for 0.48 hours. It was observed that there is no significant difference between the time taken for each sample to reach the maximum throughput efficiency.

Oil Yield (%): It represents the weight or volume of essential oil obtained from a given weight of the plant source material

Extraction efficiency (%): Extraction efficiency depicts the level of effectiveness of the developed machine, comparing the volume of oil extracted to the volume of extractible oil in the processed seeds.

Throughput capacity (kg/hr): Throughput capacity quantifies the machine's capability in terms of groundnut seeds it can process per unit time.

Table 1a. Moisture content (g) of palm kernel as received from the market

Conditioned Moisture	Content (%)	Initial Weight	Final Weight	Initial -Final Weight
51	5.10	13.54	1.56	11.56
101	5.13	13.65	1.58	12.07
151	5.12	13.60	1.52	12.08

Calculated average weight content = 11.90 g.

Table 1b. Parameter for palm kernel seed (5% moisture content)

Shaft speed (rpm)	Cake Mass(kg)	Lost Mass(kg)	Run Mass(kg)	Time (hrs)	Throughput Efficiency (kg/hr)
47.6	0.40	0.80	0.30	0.45	89.50
87.0	0.30	0.80	0.40	0.38	67.10
92.0	0.35	0.70	0.45	0.39	87.10

Table 1c. Parameter for palm kernel seed (10% moisture content)

Shaft speed (rpm)	Cake Mass(kg)	Lost Mass(kg)	Run Mass(kg)	Time (hrs)	Throughput Efficiency(kg/hr)
47.6	0.30	1.00	0.20	0.44	73.80
87.0	0.25	0.90	0.35	0.36	61.50
92.0	0.35	0.70	0.45	0.39	87.10

CONCLUSION

A palm oil extraction machine was designed and fabricated using locally available materials. Testing was conducted on the developed palm oil extraction machine. The machine efficiently extracted palm oil with minimal energy consumption and within a shorter processing time compared to the traditional method.

i. The performance evaluation showed that the machine achieved an oil extraction efficiency of 87.10%, demonstrating its ability to recover a significant portion of the oil content in the palm fruits. The machine demonstrated a processing capacity of 100 kg per hour, making it suitable for small-scale operations.

ii. The material discharge efficiency was recorded at 75 percent, indicating that most of the processed material was effectively expelled. These results demonstrate that the machine provides a practical and affordable solution for small-scale palm oil extraction, offering a viable alternative to conventional methods

iii. The developed palm oil extraction machine enhances the efficiency of the fruits and processing time, maximizes the palm oil yield and minimizes waste. This contributes to a circular economy and reduces environmental impact.

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