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Development and Performance Evaluation of a Variable-Pitch Tapered-Shaft Screw Press for Palm Oil Extraction

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Abstract

Palm fruit oil extraction is a difficult task to undertake. Screw press technology offers a solution to its extraction. However, most screw press available requires high power input for effective operation, thus affecting cost. Therefore, there is a need for improvement in order to limit cost and increase extraction efficiency. This work focuses on development of a variable-pitch tapered-shaft (VATS) screw pressfor enhancing palm fruit oil extraction efficiency. The VATS screw press machine consists of three main components including the hopper, the steamer, and the pressing unit, and was operated at temperature of 90, 110 and 130°C, shaft speed of 30, 45, and 60 rpm and heating time of 10, 15, 20 minutes. The performance evaluation carried on the machine includes oil yield, extraction efficiency and extraction loss. The results showed that the average oil yield, extraction efficiency and extraction loss were 83.72, 97.73 and 2.37% respectively. Whereas, a higher machine efficiency of 94.45% was obtained at a temperature of 130°C, shaft speed of 60 rpm, and heating time of 15 minutes. It is hoped that the information on the design concept for VATS screw press for palm fruit oil extraction will be useful for the vegetable oil industry.

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1. INTRODUCTION

Production of vegetable oil is generally determined by the extraction methods that are based on the four main principles visa: traditional wet rendering process, hydraulic press, solvent extraction, and screw press (Savoire et al., 2013; Amiolemhen & Eseigbe, 2019; Alam et al., 2020; Chukwunwike& Ojimelukwe, 2021). Among all, a screw press is mostly used due to its simplicity and conveniency. One major limitation of a screw press is pressure build up during operation, causing shaft rolling and operational failure. In the past decades, efforts have been made to improve the efficiency of a screw press. Typically, pressure modulation, and adjustable choke mechanism that allows the adjustment of back pressure have been developed and optimized (Adetola et al., 2014). However, factors such as applied pressure, rotational speed, temperature and moisture content considered in optimization process requires significant power input for effective operation (Tagoe et al., 2012; Aremu & Ogunlade, 2016; Atoyan et al., 2000, Amiolemhen & Eseigbe, 2019). The most two important parameters viz; input power and throughput (efficiency) capacity, mostly considered in the design of a screw press for oil seeds, depend on the rate of energy consumption within the pressing zones. To obtain high efficiency, higher power is needed where the highest pressures arise in the material to be pressed (Okafor, 2015; Atoyan et al., 2000). For instance, a powered driven small-sized palm oil expeller with a uniform pitch shaft has extraction efficiency of 79.5%, with maximum extraction capacity of 532 kg/h, requires up to 5 hp electric motor (Adetola et al., 2014). Generally speaking, the available screw press requires higher power to operate. On the other hand, a screw press with uniform pitch shaft uses high speed/high power motor to effectively extract oil from oil-based seed materials, thus affecting cost (Olayanju

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et al., 2016). In the performance evaluation of some screw press, Firdaus et al. (2017) and Muhammad et al. (2021) reported ineffective press (in terms of high oil loss and nut breakage), and shaft failure, respectively.

Hence, there is a need to further improve extraction efficiency of a screw press for palm oil extraction. However, this work aim at development of a low torque and high efficiency palm fruit oil extraction machine using variable-pitch tampered-shaft (VATS) method that reduces time of cycle requires for screw press to complete a revolution.

2. MATERIALS AND METHOD

2.1. The Concept of the Machine and Working Principle

The VATS mechanism was proposed to achieve a higher extraction capacity and lower extraction loss than that of the uniform shaft. Figure 1 shows the typical diagram for the proposed concept and the prototype VATS, whereas Figure 2 shows the machine. The high-pitched shaft is extreme (up to 50 mm) in the extraction unit and began at 18 mm. When the oil seeds are fed through the hopper, the rotating shaft picks the materials at low pitch level and pushes them along the pitch until it reaches the extreme. Moreover, the whole extracting unit does not have to be oversized in other to cope with infrequent intensive extraction requirements, because the pressure required for extraction reaches its peak value in fewer revolutions and shorter time when compared with the same sized uniform pitch screw (Ndirika & Onwualu, 2016), thus making VATS screw press machine a novel extraction method.

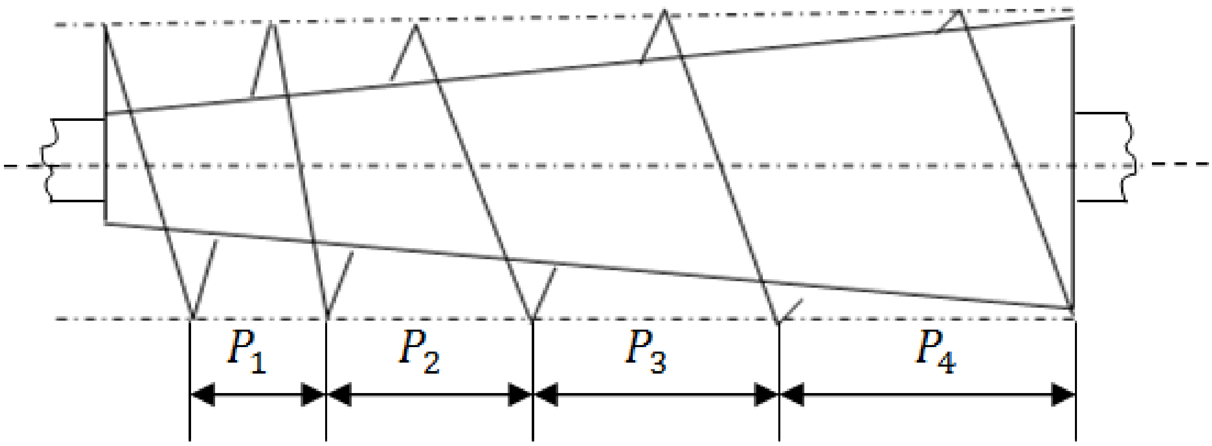

Figure 1. *Typical diagram for the proposed concept and the prototype VATS screw press*

2.2. Design Calculation

The exploded view showing the component parts and the pictorial view of the palm oil extraction machine were presented in Figure 2 and Figure 3 respectively. Themachine designed with Autodesk Inventor software 2018 based on calculations. The design consists of five (5) different sections which are the steamer chamber, the hopper, the VATS pressing unit, capacity and power requirement of the machine (Figure 2). In addition, the functional parameters like motor speed, and machine torque were adequately considered.

2.2.1. Design of the Steamer Chamber

The steamer is made up of a pulverized hole of 10 mm diameter and it is placed on top of a boiler with a controlled temperature in order to steam the palm fruit at a specified temperature. It has a gate of distance 98 mm. However, the steam chamber was designed according to Adetola et al. (2014).

Volume of the steamer (V_s)

$$V_s = \pi r^2 h \tag{1}$$

$$V_s = \pi \times 0.12^2 \times 0.05 = 0.00226 \text{ m}^3$$

The palm fruit has a configuration of length (0.025m), breath (0.015m) and width (0.015m). Therefore, the volume was determined using the Equation 2.

Volume of one palm fruit, (V_{pf})

$$V_{pf} = lbh \tag{2}$$

$$V_{pf} = 0.025 \times .015 \times 0.015 = 5.6 \times 10^{-6} \text{m}^3$$

The amount of palm fruit contained in one volume of steamer container ranges from 8 to 10 kg

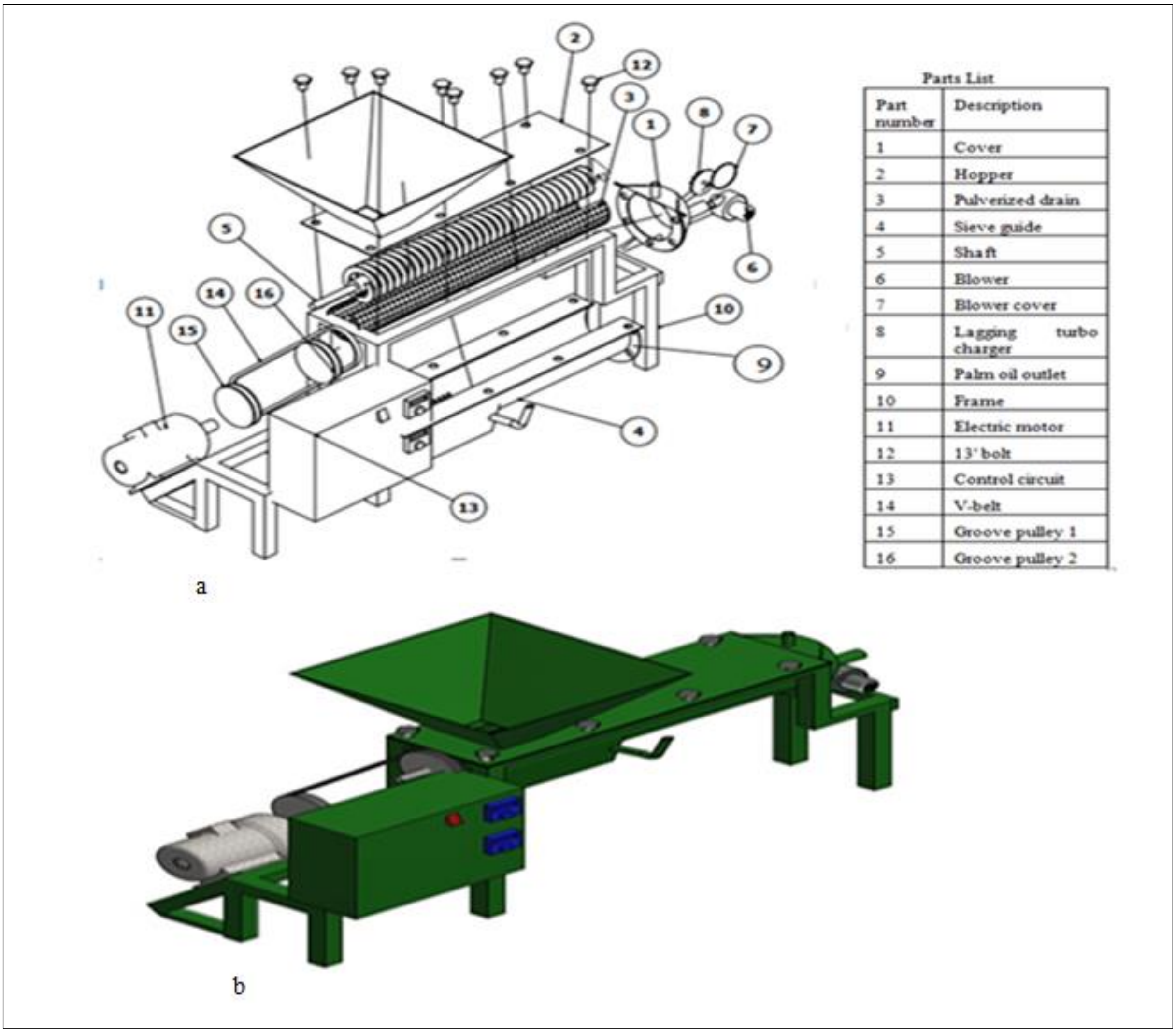

Figure 2. *The machine showing components part;* ***a)*** *exploded view,* ***b)*** *3D view*

Figure 3. *Pictorial view of the machine*

2.2.2. Design of the Hopper

The hopper, as shown in Figure 4, has its volume (V_h) designed to accommodate less volume of palm fruit as produced from steamer container in order to avoid bottle neck operation by using Equation 3.

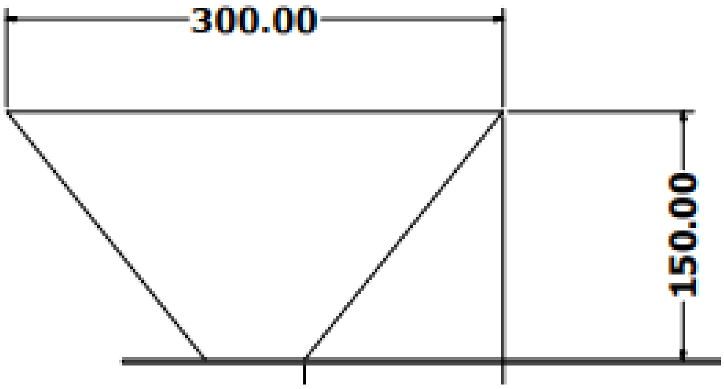

Figure 4. *The hopper*

$$V_h = \frac{1}{2}(a+b)hz \tag{3}$$

$$V_h = \frac{1}{2}(.3+0.05)0.15 \times .3 = 3.3 \times 10^{-5}\ \mathrm{m}^3$$

2.2.3. Design of the VATS Pressing Unit

By considering the most critical point of the palm fruit loading the hopper at full load, the volume of the pressing chamber V_{pc} is given as,

$$V_{pc} = \pi r^2 l \tag{4}$$

Density of steamed and bulk density of digested palm fruit is 373.47 and 1060 *kg/m*^{3}respectively. The mass of steamed palm fruit (M_{sf}) at full load = Density x Volume.

$$M_{sf} = \rho_{sf} \times V_{sf} \tag{5}$$

$$V_{pc} = \pi \times 0.032^2 \times 0.537 = 0.001727 \text{ m}^3$$

$$M_{sf} = \rho_{sf} \times V_{sf}$$

$$M_{sf} = 1060 \times 0.001727 = \ 1.83 \text{ kg}$$

Weight, $W = mg$

$$W = 1.83 \ \times 9.81 = 17.96 \text{ N}$$

Figure 5 shows VATS pressing screw wounded around a shaft capable to withstand the torsional and bending stress. However, ASME code equation of shaft was used to compute the shaft diameter Amiolemhen and Eseigbe (2019).

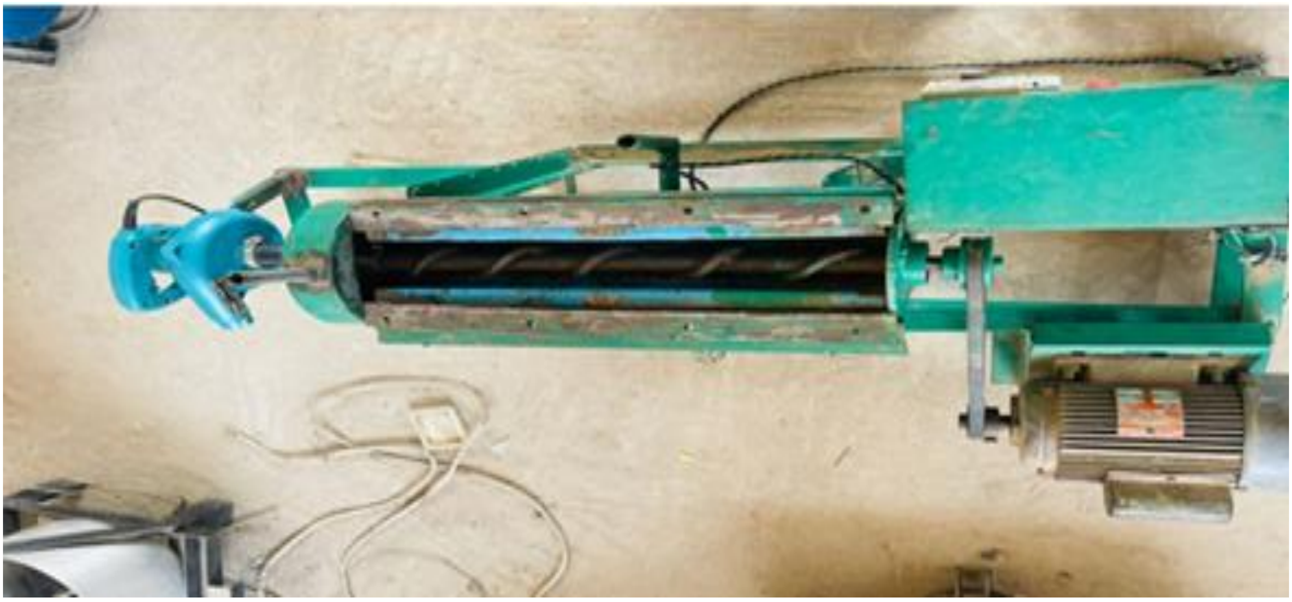

Figure 5. *The VATS pressing screw*

$$d^3 = \frac{16}{\pi\tau_{all}}\sqrt{(K_b M_b)^2 + (K_t M_t)^2} \tag{6}$$

$$\tau_{all} = \frac{0.5\sigma y}{sf} = 60 \text{ MN/m}^2$$

σ_y is the yield strength for shaft material = 60 MN/m^2 (taking factor of safety to be 2)

where K_t is the combination of shock and fatigue factor applied under torsion moment, $K_b$is the combination of shock and fatigue factor applied under bending moment, M_t is the torque on shaft, M_b is the bending moment on shaft. The values for K_t, K_b, M_t, and M_b are 1.0, 1.5 98.01 Nm and 50.3 Nm respectively.

Circular rotational motion of the shaft,

$$v = \omega r \tag{7}$$

or

$$v = \frac{\pi N r}{30} \tag{8}$$

Base on the shaft analysis, the machine is considered to operate at low speed (i.e. 90 rpm)

$$v = \frac{\pi \times 90 \times 0.0275}{30} = 0{,}259 \text{ m/s}$$

Centrifugal force F_c pressing the palm fruit,

$$F_c = Ma = M\omega^2 r \tag{9}$$

And angular velocity is

$$\omega = \frac{2\pi N}{60} \tag{10}$$

Mass of shaft,

$$m_s = \rho \times V_s \tag{11}$$

$$m_s = 7850 \times 0.001449 = 11.37 \text{ kg}$$

Total mass at the compression chamber M_t,

M_t = Mass of shaft + mass of palm fruit

$$M_t = 11.37 + 5 = 16.37 \text{ kg}$$

$$F_c = M\left(\frac{2\pi N}{60}\right)^2 r$$

$$F_c = 16.37\left(\frac{2\pi \times 90}{60}\right)^2 \times 0.0275$$

Torque developed,

$$T = F_c r \tag{12}$$

$$T = 40 \times 0.0275 = 1.1 \text{ Nm}$$

2.2.4. Belt Design

The belt design is a crucial part that determines the operation of an entire drive system of the machine. Open belt, as shown in Figure 6, was considered because of its flexibility. Typically, the length of belt, power transmitted, angle of contact or lap, and load carried by the screw were designed according to Firdaus et al. (2017) and Muhammad et al. (2021) reported ineffective press (in terms of high oil loss and nut breakage), and shaft failure during testing of constant pitch screw press machine.

2.2.4.1. Length of Belt

The length of the belt is given as thus:

In terms of pulley radii

$$L = \pi(r_1 + r_2) + 2x + \frac{(r_1 - r_2)^2}{x} \tag{13}$$

In terms of pulley diameter

$$L = \pi(d_1 + d_2) + 2x + \frac{(d_1 - d_2)^2}{4x} \tag{14}$$

where r_1 and r_2 is the radii of the larger and smaller pulleys, respectively, x is the distance between the centers of the two pulleys,and L is total length of belt.

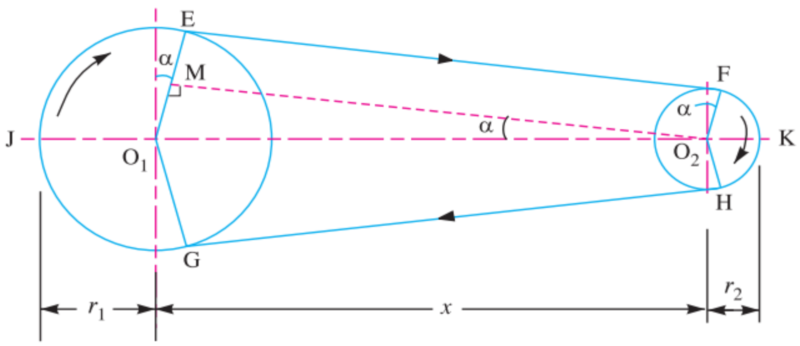

Figure 6. *Schematic diagram of an open belt drive*

2.2.4.2. Power Transmitted by the Belt

Figure 7 illustrates the power transmitted through the belt driven by the two pulleys. Considering T_1 and T_2 (in Newton), and r_1 and r_2 (in metres) and v (m/s). The effective force to drive the pulleys is T_{Eff}.

$$T_{Eff} = T_1 - T_2 \tag{15}$$

The work done per second on the belt is given as:

$$W = T_{Eff}V \ (Nm/s) \tag{16}$$

Power transmitted is

$$P = T_{Eff}W \ (Nm/s) \tag{17}$$

However, torque exerted on the driving pulley is given as

$$T_{Eff} r_1 \tag{18}$$

and the torque exerted on the driven pulley is

$$T_{Eff} r_2 \tag{19}$$

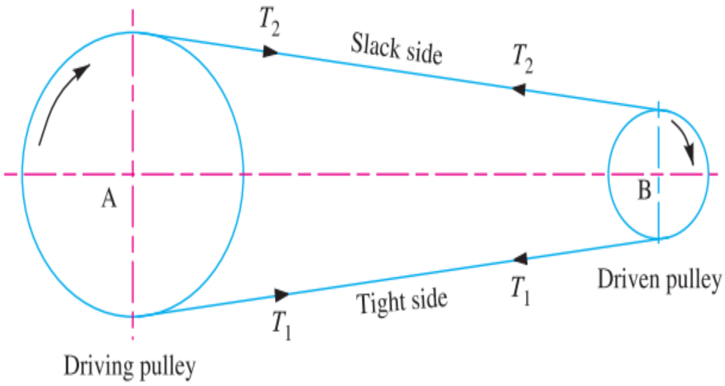

Figure 7. *Power transmitted through the belt*

2.2.4.3. Angle of Contact or Lap

For open belt, angle of contact is given as:

$$\frac{T_1}{T_2} = e^{\frac{\mu\theta}{sin\beta}} \tag{20}$$

$$sin\alpha = \frac{r_1 - r_2}{x}$$

where x is distance between the center of two pulleys, α is angle of wrap, μ is coefficient of friction between the belt and pulley, θ is angle of contact between the belt and pulley, β is Half the wedge angle.

2.2.5. Design of the Load to be Carried by the Screw

The load on the screw (W_e) was determined according to Adetola et al. (2014) as thus:

$$(W_e) = \mathrm{T}\left(\frac{D_{m/2}\tan\theta + \mu/\cos\alpha}{1 - \mu\tan\theta\cos\alpha}\right) \tag{21}$$

$$\alpha = tan^{-1}(\tan\theta_n cos\alpha)$$

where T is the torque transmitted by the screw shaft, D_m is mean thread diameter, μ is coefficient of friction, θ_n is thread (lift) angle, α = Helix angle.

$$\alpha = \tan^{-1}(\tan(45)\cos(20) = 17.24^o$$

$$(W_e) = 1.11\left(\frac{\frac{0.018}{2}\tan(45) + 0.3/\cos(20)}{1 - 0.3\tan(45)\cos(20)}\right) = 1.0340\ \mathrm{kg}$$

2.2.6. Design of the Worm Shaft

The design of the worm shaft was according to Adetola et al. (2014) with slight modifications, as given below:

$$P_s = \pi \tan(\varphi) d_{wm} \tag{22}$$

where φ is lead angle, d_{wm} is mean shaft diameter and P_s value is $P_s = \pi \tan\,(0.7) \times 0.018 = 0.0476$ m

2.2.7. Design of the Capacity of the Machine

The theoretical capacity of the expeller was determined using the modified form of the equation given by Adetola et al. (2014),

$$Q = 60 \times \frac{\pi}{4} \times (D^2 - d^2) P_s \rho N \emptyset \tag{23}$$

where Q is theoretical capacity, D is shaft diameter, d is screw diameter, ρ is density of palm kernel,P_s is screw pitch, N is shaft speed and $\emptyset$ is filling factor.

$$Q = 60 \times \frac{\pi}{4} \times (0.05^2 - 0.018^2)\ 0.0476 \times 373.47\ \times 90 \times 1 = 164.0609\text{m}^3/\text{min}$$

Theoretically, Q = 2.73 m^3/h

2.2.8. Design of the Pressure to be Developed by the Screw Thread

Pressure developed by the screw thread in oil expeller can be determined using the given equation;

$$P_r = \frac{W_e}{A_p} \tag{24}$$

$$A_p = \pi D_m n h$$

where A_p is the pressing area, n is the number of threaded turns subject to load, h is thread depth

$$A_p = \pi \times 0.018 \times 11 \times 0.001 = 6.2\ 10 \times 10^{-4}\text{m}^3$$

2.2.9. Design of the Power Requirement of the Machine

The power required for oil extraction can be given as:

$$P = Q L_s \rho g F \tag{25}$$

where P is the power required for extraction, Q is the volumetric capacity, L_s is the length of screw shaft, ρ is the density of palm kernel, g is the acceleration due to gravity and F is material factor.

Substituting Q = 2.73 m^3/h, Ls = 0.61m, ρ = 373.47 kg/m^3, g = 9.81 m/s^2, F = 0.3 into Equation 25, Hence P = 1.5 kw

The power of the electric motor was determined according to (Adetola et al., 2014) as given below:

$$P_m = \frac{P_e}{\eta} \tag{26}$$

where P_m is the power of the electric motor, and η is the drive efficiency

Given the η = 75% or 0.75, P_m = 2.0hp. Therefore, a 2hp three-phase electric motor was used to drive the machine.

2.3. Material Selection and Fabrication of Machine Components

The orthoganal view of the machine components are presented in Figure 8 in the appendix. The machine hopper was formed using 4 pieces of standard mild steel of dimension 300x150x200 mm. The shaft (of 18 mm base diameter) was fabricated from the mild steel rod of diameter 25 mm and length 210 mm. The screw thread was machined at a variable pitch from 50 mm to 18 mm to form a tapered screw conveyor of ten screw turns. A mild metal plate of 10 mm, 2 x 2 feet was cut, machined and used to form the barrel. However, small openings were made on the lower side of the barrel to allow the passage of the extracted oil. The frame was fabricated by using angle iron of dimension 40x40x30 mm. All the fabrication processes including marking out, machining, cutting, joining, drilling and fitting were performed at the farm power and machinery laboratory, Department of Food and Agricultural Engineering, Kwara State University, Malete, Nigeria. The workshop machines and equipments including grinding machine, lathe machine, welding machine, scriber, steel rule, and compass, centre punch, oxy-acetylene gas, saw frame and cutting blade were used. The specification of the materials used is presented in Table 1.

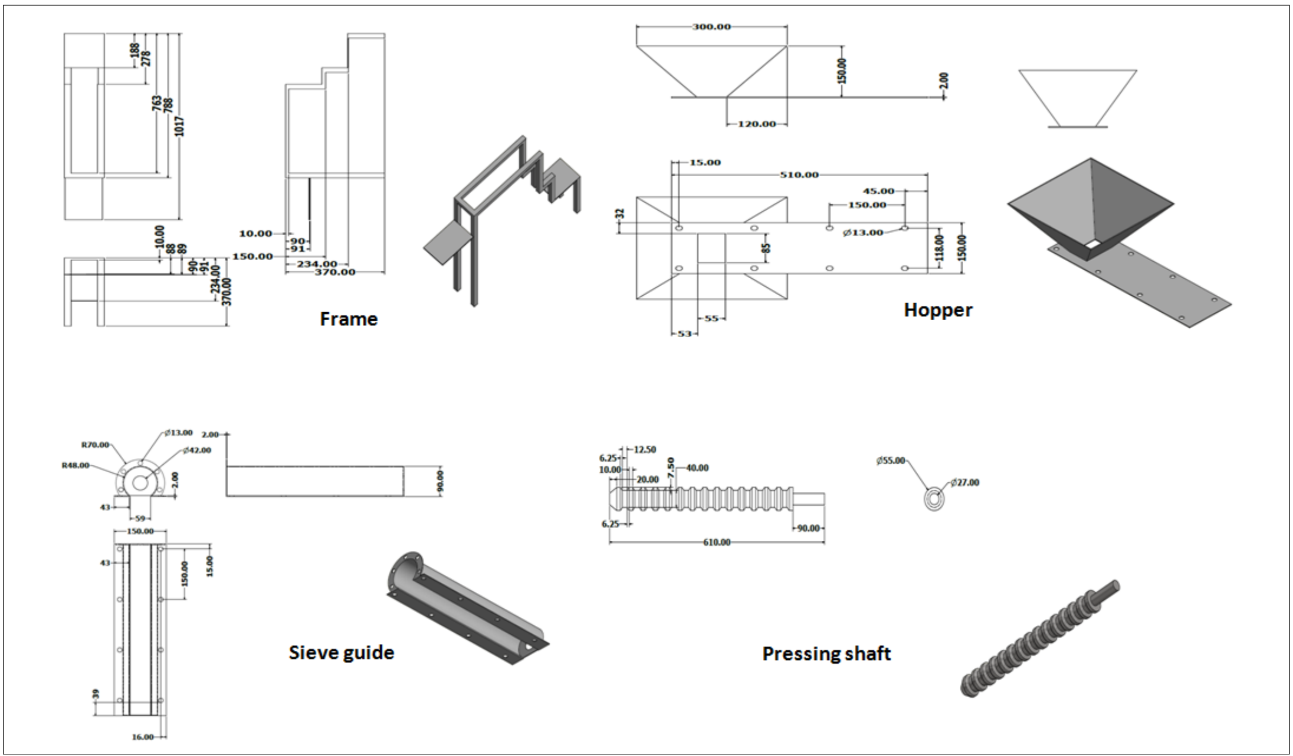

Figure 8. *Orthogonal view of some component parts*

2.4. Machine Operation and Testing

Palm fruit bunches were obtained from local farmers at Kwara State University Malete (KWASU), Nigeria (51° 27' 49.144" N 19° 13' 4.303" E). The fresh palm fruits were taken to the Farm power and machinery laboratory, Department of Food and Agricultural Engineering, KWASU, while dirt were removed and the fruits were weighed and prepared for oil extraction. The prepared palm fruits were charged into the machine that was operated at temperatures of 90, 110, 130°C, heating time of 10, 15, 20 minutes, and shaft speed of 30, 45, and 60 rpm. The shaft of the machine was designed with two segments, with counter motion orientation to delay the palm fruit in the steaming chamber. In order to extract the oil, the steamed palm fruits were conveyed, squeezed and pressed by the variable-pitch tapered-shaft, while the collecting turbo impeller collect the oil from the pressed seeds and pushes it to the outlet chute. The variable-pitch tapered shaft rotates counter clockwise, pushing out the drained chaff and the kernel seed out of the outlet chute of the pressing chamber. The fresh palm fruits, the oil extracted, and the residual cake (kernel and fibre mixture) were collected and evaluated. The values evaluated are oil yield, machine extraction efficiency and extraction losses based on Aremu and Ogunlade (2016), as given below:

$$\text{Oil yield (\%)} = \frac{W_{OE}}{W_{TS}} \times 100 \tag{27}$$

$$\text{Extractionefficiency} = \frac{W_{OE}}{\alpha W_{TS}} \times 100 \tag{28}$$

$$\text{ExtractionLoss(\%)} = \frac{W_{TS} - (W_{OE} + W_{CK})}{W_{TS}} \times 100 \tag{29}$$

where;

α : Oil content of the palm fruit

W_{TS} : Total weight of sample (g)

W_{OE} : Weight of oil extracted (g)

W_{CK} : Weight of cake (g)

Table 1. *Specification of construction material*

s/n	Material	Description	Quantity
1	Metal plate	2 mm (0.08") mild steel	1 full sheet
2	Fly bar	3 mm (0.1") mild steel	1 full length
3	Motor	2 hp	1
4	Metal plate	8 mm (0.3")	2 x 2 feet
5	Ball bearing	Angular contact bearing	2
6	Fastener Bolt	13"	16 pieces
7	Motor speedcontroller	PWM (Pulse Width Modulation)	1
8	Temperatureregulator		1
9	Shaft	2.5 m (98") mild steel	2
10	Oil pump	Centrifugal pump	1
11	Pipes	1", 0.5" and 0.25"	
12	Pulleys		2
13	Belt		1

3. RESULTS AND DISCUSSION

Table 2 shows the average oil yield (%), extraction efficiency (%) and extraction loss (%) for different processing conditions. The average oil yield, extraction efficiency and extraction loss were 83.72, 97.73, and 2.37% respectively. The extraction efficiency (97.73%) is high when compared to the findings of Adetola et al. (2014) that has extraction efficiency value of 79.5%. The highest oil yield (84.90%) was obtained when the machine was operated at a temperature of 110°C, shaft speed of 45 rpm and 15 minutes heating time. The results obtained show that the machine operates smoothly without jamming and effectively extract the crude oil from the palm fruits.

Table 2. *Average* oil yield (%), extraction efficiency (%) and extraction loss (%) for different processing conditions*

Temperature (°C)	Speed (rpm)	Heating Time (minutes)	Oil Yield (%)	ExtractionEfficiency (%)	ExtractionLoss (%)
90	30	10	82.51	97.23	2.48
		15	84.35	98.08	2.01
		20	84.01	98.01	2.82
110	45	10	83.45	97.54	2.23
		15	84.90	98.21	1.98
		20	83.52	98.00	2.53
130	60	10	83.06	97.25	2.45
		15	84.52	98.45	2.02
		20	83.19	96.78	2.78

**Standard error = 0.21*

4. CONCLUSION

In an attempt to improve the efficiency of a screw press extraction machine, a variable-pitch tampered-shaft (VATS) screw press was designed, constructed and evaluated for palm fruit oil extraction. The machine was simple in term of the design and easy to operate and maintained. Powered by 2hp three phase electric motor, the VATS screw press machine has average oil yield, extraction efficiency and extraction loss of 83.72, 97.73, and 2.37% respectively from the palm fruits. In conclusion, theVATS screw press machine allows higher efficiency with low extraction loss and can be applied for small and medium scale palm fruit oil extraction operation. Though, the technology can be further optimized to increase production rate of palm screw press extraction machine and providemore employment opportunitiesin the vegetable oil industry, future work should focus on the effect of extraction variables (such as time and temperature) on the quality of the extract oil.

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CONFLICT OF INTEREST

There is no conflict of interest.

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