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Research Article

DESIGN AND FABRICATION OF A VERTICAL PALM FRUIT DIGESTER

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A B S T R A C T

The main objective of this project is to design and fabricate a power controlled vertical palm fruit digester in order to reduce the rigours encountered by the traditional method of digestion while optimizing the production of good quality palm oil. Digestion is the process of releasing the palm oil in the fruit through the rupturing or breaking down of the oil bearing cells. The machine works on a rotary impact principle. The vertical palm fruit digester was designed, fabricated and tested. The materials for the design were locally sourced. The performance test evaluation shows that the machine has a performance capacity of 740kg/h with an efficiency of 92.31%. The digester is made up of simple components that can be easily assembled. Thus, the operation and maintenance is quite easy.

INTRODUCTION

The oil palm fruit (Elaesis guineensis) is a member of the family of (Arecaceae). The plant contains up to 400 species, each fruit is a drupe, with a fibrous and oily mesocarp and a stony endocarp or shell, the shell encloses one seed or kernel [1]. The ripe fruit is bright red except for the top which may be dark brown or black. All the fruit of an inflorescence make up a bunch. There are three naturally occurring forms of the palm oil fruit termed Dura, tenera and pisifera. The Dura form has a thick endocarp (up to ¼ inch) such that the mesocarp occupies only 35 – 65% of the fruit. The tenera form has a thin endocarp (up to 1/8 inch) and is 55 – 95% mesocarp. Yield and fruit weight are higher in dura and lower in tenera although tenera produces fruit with higher oil content but less kernel oil when compared to the dura variety [2]. The endocarp is absent in the pisifera form, they lack seed entirely. Pisifera are undesirable from a commercial stand point since they have low yield and are vigorous, however, they are extremely important in breeding. Today, many products can be made from palm oil which ranges from edible to non - edible usage as well as medicinal products. The edible usage include the production of margarine and palm oil for cooking, the non – edible usage include production of soaps, detergents, candles, cosmetics, rubber processing, lubricants and glycerol while the medicinal usage include treatment of cancer, diuretic and liniment [3].

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The major process in the oil palm process is the palm fruit digestion. According to [4], Digestion is the process of releasing the palm oil in the fruit through the rupture or breaking down of the oil bearing cells. Digestion and oil extraction are the most tedious and essential operations in traditional palm fruit processing. Therefore, early efforts concentrated on these tasks [5]. Digester comes in two categories and these are vertical and horizontal digesters [6]. In the horizontal digester, digestion is done by the beater arms which also convey the macerated oil palm fruits from the digester posterior end to the exterior end automatically due to conveyor – screw – arrangement of the beater arms [7]. The horizontal digester has been tested on several occasions and it has been established to have a thorough maceration (digestion), smooth and continuous discharging, Easy to maintain and operate, ability to break – up all the oil bearing cells, ability to digest without cracking or breaking the nuts, portability and very high efficiency. This concept is usually accompanied by low belt drive efficiency due to slippage and improper design of the pulley system and whirling of the digester main shaft due to increase in length.

Until the late 80's traditional processing techniques remained the major method used in palm oil production in Nigeria. This primitive method was found to be labour intensive time - consuming, unhygienic, drudgery, slow and very low digestion and production rates etc. the consequences of these factors was about output in terms of extraction and recovery of good quality palm oil. This, to a great extent has been responsible for the in balance in the demand and supply of oil palm products locally and internationally.

With regard to the aforementioned drawbacks, we have planned to design and fabricate a vertical palm fruit digester aimed at Optimizing the extraction and recovery of good quality palm oil and palm kernel, eliminating drudgery associated with traditional processing techniques, increasing the processing capacity of the industrial farmers, increasing foreign exchange earnings through the exportation of high quality domestic and industrial palm oil and its products, increasing the efficiency of operation and production. The Dura and Tenera species were used. After considering the existing designs, worm and wheel gear were incorporated to enable the direction of motion to be changed from horizontal plane to vertical plane. This research work was carried out in Mechanical Engineering Department, National institute of Construction technology, Uromi, Edo State, Nigeria.

Working Principle

The vertical digester consist of the hopper, digester barrel, Bearings, main shaft, Beaters arms, discharge end, worm gear, wheel gear and the prime mover. The vertical digester's barrel carries the hopper and the shaft assembly which lies in the central position of the barrel. The shaft assembly is made up of 50mm diameter x 500mm shaft with six beater arms which are arranged at specific angle and distance strongly welded to the shaft in the horizontal position. The shaft carrying the worm gear and the pulley at either ends is 50mm in diameter and 250mm long placed in the horizontal position. The worm and wheel gear arrangement is to enable the direction of motion to be changed from horizontal plane to vertical plane. The digester works on rotary impact principle. The machine shaft was designed to rotate at about 130rpm and a safe speed for the maceration of the mesocarp from the hard nut and avoiding the breaking of the nuts. There is a considerable friction between the digester drum and the macerate. On the other hand, within the macerate (nut – nut surface rubbing friction) and also as the beater arms stir up the macerate. Its existence is recognizable in the difference in extent of maceration when different quantities of fruits are put for maceration working for some time, the more the fruits the better the extent of maceration. On entry through the hopper (parboiled oil palm fruits), the digester macerate the fruits for some minutes and automatically discharge the macerate through the exit end by gravity. The power or motion is supplied by a 6.5hp petrol engine generating about 5.3KW and running at about 2600rpm. The use of gears helps to regulate the power transmitted and improves the efficiency of the machine [4].

Design Analyses and Calculation

The designs were on power determination, diameter of digester arm, and design for pulley, design for belt, selection of gear, shaft loading and shaft design consideration.

Power Determination

In accordance with the American society of Agricultural Engineers (ASAE), the rupture strength of palm fruit sterilized at 100^0c under atmospheric pressure for a period of 45 minutes is $1.082 N/mm^2$.

Using the above value, the rupture force can be determined as follows

$S_R = F_R / A_M$ (1)
Where,
S_R = rupture strength
A_M = area of palm fruit mesocarp (mm^2)
F_R = rupture force (N)
$F_R = S_R x A_M$ (2)

Assuming that the palm fruit is a sphere, the area can be determined as follows

$A_M = 4\pi(r_m)^2$ (3)
Where,
r_m = approximate radius of deformation of fruit (mm)
$A_M = 254.5mm^2$
From equation 2, we have that
$F_R = S_R \times A_M$
$F_R = 275.37N$
Torque transmitted per digester arm (T_d) = $F_R \times L_d$ (4)
Where,
L_d = length of digester arm
$T_d = 60.5814Nm$
Total torque in the digester (T) = $T_d \times n$ (5)
Where,
N = number of digester arm
N = 6
T = 363.49Nm

The angular speed of shaft (ω) is determined from the equation deduced from [8]

$F_R = m\omega^2 L_d$ (6)
$\omega = \sqrt{(F_R / mL_d)}$ (7)
$F_R = mg$ (8)
$M = F_{R/}g$ (9)

Therefore, equations (5) and (7) become

$\omega = \sqrt{(g/ L_d)}$ (10)

ω = 7rad/sec

Power required of the digester was determined with the expression by [8] which states that the power is the product of torque (T) and angular velocity (ω) as

$P_d = T\omega$ (11)
$P_d = 2544.43W$

Assuming that 12% of power is required to overcome friction, then

P_f = power required to overcome friction
$P_f = 305.33W$
Therefore, $P_m = P_d + P_f$ (12)
Where,
P_m = motor power
$P_m = 2849.75W$

Assuming that 15% of power is required to overcome electrical losses, then

$P_m = 3231.415W$

Therefore, 6.5hp diesel engine with a motor speed of 2600rpm was selected to give the required torque in the digester.

Diameter of Digester Arm

For solid shaft, which are subjected to bending and torsion, the diameter (d) is obtained from the expression given by [9] as

$S_S = 16/\pi d^3 \sqrt{(K_b M_b)^2 + (K_t M_t)^2}$ (13)
Therefore,

$$d^3 = 16/\pi\, S_{Smax} \sqrt{(K_bM_b)^2 + (K_tM_t)^2} \quad (14)$$

Where,
S_{Smax} = maximum allowable shear stress
K_b = combined shock and fatigue factor applied to bending moment
K_t = combined shock and fatigue factor applied to torsional moment

M_b = bending moment
M_t = torsional moment

For this digester, the beater arms are mainly subjected to bending moment. Therefore equation (14) reduces to

$$d^3 = 16/\pi\, S_{Smax} \sqrt{(K_b\, M_b)^2} \quad (15)$$

The bending moment on the digester arm can be determined by using the equation

$$M_b = F_R \times L_d \quad (16)$$
$M_b = 60.581\text{Nm}$

According to ASME, bending stress for shafts without keyway is 55MN/m^2 and $K_b = 2.0$

Substituting into equation (15) we have that

$d = 22.38\text{mm}$
Therefore, we use a shaft diameter of 22mm

Selection of Pulleys and Determination of their Speeds

Pulleys and belt arrangement is used to transmit power from a driving shaft to the driven shaft. Two pulleys are used for this system made of cast iron.
The relationship expression given by [8] is used to determine the transmitted speed.

$$N_1D_1 = N_2D_2 \quad (17)$$

The intended ratio of the speed of the driven pulley to that of the driver is 1:3
Where,

N_1 = speed of motor shaft
D_1 = diameter of motor pulley = 180mm
N_1 = speed of digester shaft
D_2 = diameter of digester pulley
N_1 = 850rpm
N_2 = 480rpm
D_2 = 850 x 180/480
D_2 = 320mm

Designs for Belt

This design involves the transmission of required power by a given belt. In this case the width of the belt is unknown. The power transmitted by a belt drive is a function of the belt tensions and belt speed. This is determined according to the equation giving by [9] as:

$$\text{Power } (P_m) = (T_1 - T_2)\, V \quad (18)$$

Where,
T_1 and T_2 are belt tension on the tight and slack side respectively.
V = belt speed (m/s)
Also,
$$T_1/ T_2 = e^{\theta\mu} \quad (19)$$
Where,
μ = Coefficient of friction between pulley and belt
θ = angle of wrap i.e angle of contact of belt on pulley

Determination of Angle of Wrap

The angle of wrap for an open loop belt may be determined as follows

Using the equation of provided by [13]
$$\sin\beta=(R-r)/c \quad (20)$$
$$\theta_1 = 180^0 - 2\beta = 180^0 - 2 \sin^{-1} (R - r)/c \quad (21)$$
$$\theta_2 = 180^0 + 2\beta = 180^0 + 2 \sin^{-1} (R - r)/c \quad (22)$$
Where,
C = distance between pulley centres.

This C is selected based on the assumption that

1. The smaller diameter is 1/3 of the larger pulley diameter
2. The difference between the pulley diameter

Thus, (320 – 180) mm = 140mm
For this design, let C = 400mm

Length of belt (L) is expressed as

$$L = \sqrt{4C^2 - (D - d)^2} + 1/2\, (D\theta_L + d\theta_s) \quad (23)$$

This can be simplified into
$$L = \pi\, (r_1 + r_2) + 2C + (r_1 + r_2)^2/C \quad (24)$$
$L = \pi\, (0.07 + 0.16) + (2 \times 0.40) + (0.07 + 0.16)^2/0.40$
$L = 0.7226 + 0.8 + 0.1323$
$L = 1.6549\text{m}$

Velocity of belt is obtained from

$$V = \pi D_1N_1/60 \quad (25)$$
$V = \pi \times 0.1323 \times 850 /60$
$V = 5.89\text{m/s}$

For speed up to 25m/s, vee – belt could be used. For this case vee – belts are used.

The load carrying capacity of a pair of pulley is determined by the smaller value of $e^{\theta\mu}$.

Inputting values into equation (20)

$\sin\beta = (R - r)/c$
$\sin\beta = (0.16 - 0.05)/0.4$
$\beta = 15.96^0$

From equation (21)

$\theta_1 =180^0 - 2\ (15.96)$
$\theta_1 = 148.08^0$
$\theta_1 = 2.58$ rad

From equation (22)

$\theta_2 =180^0 + 2\ (15.96)$
$\theta_2 = 211.92^0$
$\theta_2 = 3.70$ rad

To determine the pulley that governs the design

$e^{\theta\,\mu} = e^{0.3 \times 2.58} = 2.16,$

Where,
μ is coefficient of friction between pulley and belt and θ the angle of wrap.

$e^{\theta\mu} = e^{0.3 \times 3.70} = 3.03$

From equation (19)

$T_1 / T_2 = e^{\theta 1\mu}$

$T_1/ T_2 = 2.16$
$T_1 = 2.16T_2$ (26)

For this design, power to be transmitted is P_a = 2544.23 = 2.544KW.

From belt selection table, maximum power that can be transmitted by a belt at pulley ratio of 2:1 is 1.7KW

Number of belt (N_b) = 2.544/1.7

= 1.496

Number of belts required = 2

Determination of T_1 and T_2 of Belt

The maximum and minimum tension of belt can be determined using equation (18) i.e. Power (P_m) = ($T_1 – T_2$) V, ($T_1 – T_2$) 5.89 = 2544.42

Substituting equation (26) into the above, we have that

T_2 = 372.41N and T_1 = 804.40N

Determination of Belt Width

The width of belt can be obtained from the equation provided by [8]

$T_1 = F_b \times b \times t \times N_b$ (27)

Where,
F_b = tensile strength of belt
b = width of belt
t = belt thickness = 12mm (assumed)
N_b = number of belt
$F_b = 2.7MN/m^2$

Substituting values into equation (27), we have

$804.40 = 2.7 \times 10^6 \times b \times 0.012 \times 1$
b = 24.8mm
b ≈ 25mm

Gear Selection

The fundamental law of gearing which states that the common normal to the tooth profile at the point of contact must always pass through a fixed point called the pitch point in order to maintain a constant angular velocity ratio of the two gears.

According to [9], Angular velocity ratio or transmission ratio = $N_g/N_p = D_g/D_p = \omega_p /\omega_g$

Since we need a speed of between 125rpm to 150rpm in the main shaft for proper maceration,

$\omega_{pulley} /\omega_{prime} = D_{prime}/D_{pulley}$
$\omega_{pulley} = 80 \times 2600/ 350$
$= 594.28$rpm

To select the speed for the main shaft

$N_g/N_p = \omega_p /\omega_g$
$48/12 = 594.28/ \omega_g$
$\omega_g = 12 \times 594.28/48$
$\omega_g = 148.57$rpm

This speed is just enough for proper maceration with a gear well selected with gear ratio of 1: 4

Shaft Loading

The solid shaft is chosen for the digester in order to satisfy the strength and the rigidity requirements. When shaft is transmitting power under various operating and loading conditions, shafts are usually subjected to torsion, bending and axial loads [9].

a. Maximum shear stress The maximum shear stress for a circular shaft is given by
$T_{max} = 16M_t /\pi d^3$ (28)
b. Bending stress
The bending stress (S_b) is given by
$S_b = 32M_b / \pi d^3$ (29)
c. Axial loading
Tensile or compression stress is given by
$S_a = 4f_a/ \pi d^2$ (30)
d. Combined torsion and bending
The maximum shear stress theory is used for shaft subjected to twisting and bending. This is given by

$$S_{SMaX} = \sqrt{\delta_b^2 + 4\tau^2} \quad (31)$$

$$S_{SMaX} = 16/\pi d^3 \sqrt{M_b^2 + M_t^2} \quad (32)$$

When fatigue and combined shock factor is applied to bending (k_b) and torsional moment (K_t) equation (32) becomes

$$S_{SMaX} = 16/\pi d^3 \sqrt{(k_b M_b)^2 + (K_t M_t)^2} \quad (33)$$

In a belt drive machine such as palm fruit digester, the torsional moment is given by the relation

$M_t = (T_1 – T_2) R$ (34)

In designing the shaft, the following assumptions were made

i. Weight of the shaft negligible
ii. Length of the shaft = 720mm

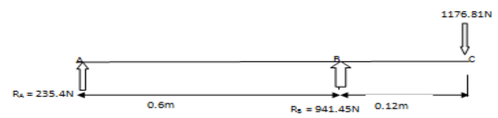

Figure 1. Shaft showing forces acting on it.

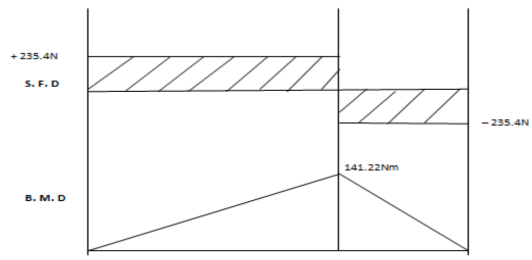

Figure 2. Shear force and bending moment diagram of the drive shaft in static position.

To determine the reactions R_A and R_B at point A and B which are the support, moment is taken about point B.

$600R_A – 1116.81 \times 120 = 0$
$R_A = 235.36N$

Sum of upward forces = sum of downward forces

$R_A + R_B = 1176.81$

$R_B = 1176.81 - R_A$
$R_B = 1176.81 - 235.36$
$R_B = 941.45N$

Maximum bending moment occur at point B, therefore

$M_b = (T_1 + T_2)\ BC$
$= 1176.81 \text{ X } 0.12$
$= 141.22Nm$

The torsional moment from equation (34) is

$M_t = (T_1 - T_2)\ R$
$M_t = (804.4 - 372.41)\ 0.16$
$M_t = 69.12Nm$

Determination of Shaft Diameter

According to ASME CODE [9], the allowable stress for shaft with keyway is $40MN/m^2$, $k_b = 1.5$ and $K_t = 1.0$

From equation (33), the diameter of the shaft can be determined as follows

$$d^3 = 16/\pi\ S_{Smax} \sqrt{(K_b\ M_b)^2 + (K_t\ M_t)^2}$$

Substituting values into the above expression, we have

$$d^3 = 16/\pi \text{ x } 40 \text{ x } 10^6 \sqrt{(1.5 \text{ x } 141.22)^2 + (1 \text{ x } 69.12)^2}$$
$$d = 0.030499m$$
$$d \approx 30mm$$

Therefore, we use a shaft diameter of 30mm.

MATERIAL AND METHODS

Description of the Oil palm Fruit Digester

The vertical digester is made up of the following components; frame stand, main shaft, Beaters arms, hopper/ digester barrel, discharge end, gear differential unit, vee belt and the prime mover (figures 1, 2, 3 and 4).

i. Frame stand: The frame stand provides rigid and skeletal support for the entire machine system. Apart from the four corner support of main frame work, it consisted of two compartments, one for the digesting chamber and the other for the prime mover (petrol engine). The frame is made of 60mm x 60mm angle iron bar. The welding provided very rigid joints. The gear support frame of 60mm x 60mm angle iron bar provided support for the gear differential unit. The frame was attached to the digesting barrel by bolts and nuts.
ii. Main shaft: The main shaft was a steel shaft of diameter 30mm and length 720mm centrally located in the digesting barrel. The shaft is carried at either ends by a sealed ball bearing of 40mm diameter.
iii. Beater arms: The main shaft has six beater arms of 22.4mm x 220mm each welded to it in the horizontal position at a vertical distance of 100mm.
iv. Hopper/Digesting barrel: The hopper was attached to the digester barrel cover. Parboiled oil palm fruits are fed into the digester barrel through the hopper. The digester barrel housed the main shaft which carries the beater arms when in operation. It was a cylindrical drum placed vertically with a hopper on the barrel cover at the top. The bottom part of the digester barrel has a discharge end.
v. Discharge end: This was located at the bottom end of the digester barrel. It directed the macerated oil palm fruit from the digester barrel to a receptacle.
vi. Gear Differential unit: This was made up of both the worm and the wheel gear. It was to enable the direction of motion to be changed from horizontal plane to vertical plane and to avoid friction between the intersecting shafts. It transmitted power to the main

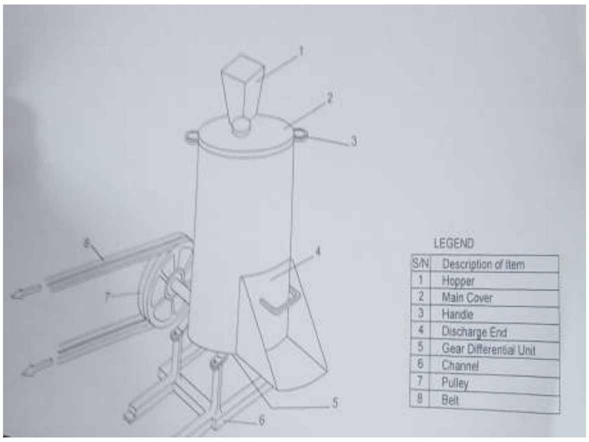

Figure 3 Isometric view of the machine

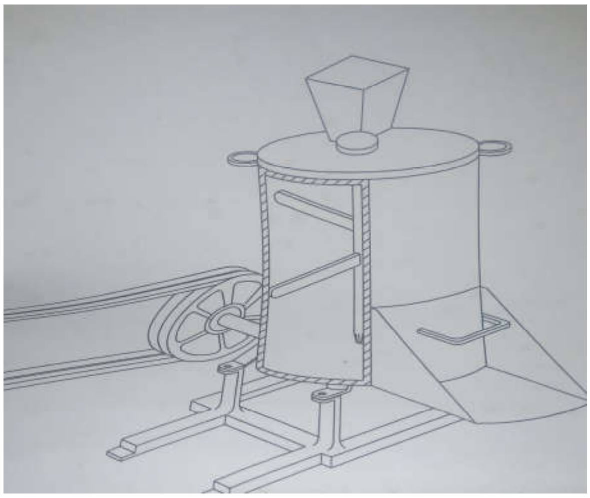

Figure 4 Sectional view of the machine

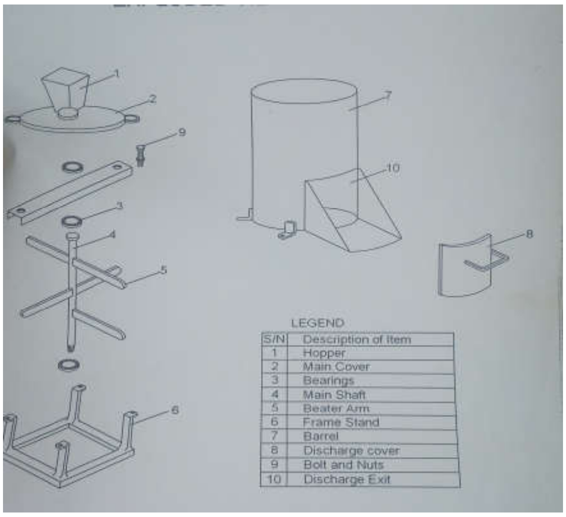

Figure 5 Exploded view of the machine

shaft carrying the beater arms at reduced speed and torque.

vii. Vee belt: The vee belt transmitted power and motion from the petrol engine to the gear differential unit through the pulley on the petrol engine and that attached to the gear differential unit.

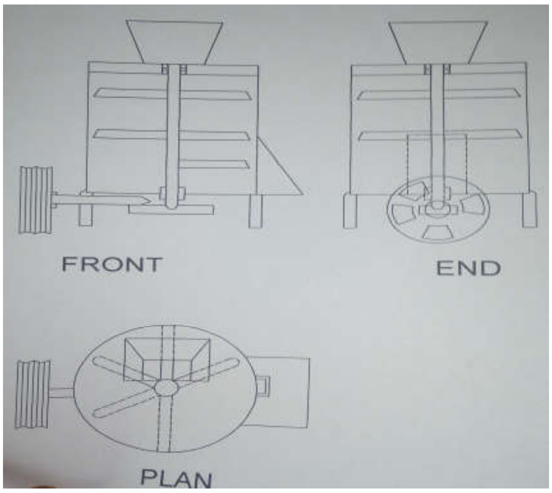

Figure 6 Orthographic view of the machine

Performance Test of the Machine

In realizing the aim of this research work, the performance should be carried out after machine parts have been assembled. The machine should be started and five samples of equal weighted mass of boiled palm fruits fed through the hopper, each time. A stop watch should be used to monitor the time taken for maceration per batch. A 6.5 – horse power diesel engine should be used as prime mover.

Testing of the machine

Five tests were carried out. A total mass of 26kg of boiled oil palm fruits will be fed through the hopper when the machine is in operation and the time for proper maceration will be recorded.

Table 1 Mass of oil palm fruit over specific time.

Test	Mass of boiled palm fruits (kg)	Time taken (s)
1	26.00	120
2	26.00	132
3	26.00	120
4	26.00	126
5	26.00	138

Efficiency of the machine

From the five tests carried out. A total mass of 26kg of parboiled oil palm fruits will be taken and fed through the hopper when the machine is in operation. The mass of palm fruits not properly macerated will be separated and weighed.

The result expected is as follows

Mass of parboiled oil palm fruits =26.00kg
Mass of oil palm fruits not properly mashed = 2.00kg
The mass output = (mass of palm fruit – mass not properly mashed) = 26.00 ─ 2.00 = 24.00kg
Efficiency = (Output/ Input) x 100% = (24.00/26.00) x 100% = 92.31%

RESULTS AND DISCUSSION

The results of the performance test show that the machine performed above 90% efficiency with an average performance of 26kg/127s. Hence, the capacity of the machine is approximately 740kg/h.

CONCLUSION

A vertical palm fruit digester was designed, fabricated and performance evaluation carried out. The test result revealed that the machine has an efficiency of 92.31% and a performance capacity of 740kg/h. The machine is made up of simple components that can be easily assembled. Local experts can maintain and operate the machine with ease. It is anticipated that the machine when commercialized will meet the need of small scale and medium scale farmers. In addition, the national economy will receive a boost since adoption of such machines will help in the production of high quality palm oil as one of the bed rock of national economy through earning of foreign exchange.

Recommendations

Having carefully observed the performance of this machine, it is expedient at this juncture to proffer some recommendations based on the findings.

1. There is the need for the inclusion of dampers in the digester to reduce to the barest minimum the problems associated with vibrations of the machine when in operation.
2. To fully realize the production of high quality oil devoid of any sort of contamination, it is necessary to use stainless steel plates for the digester barrel, shaft and beater arms.
3. The use of diesel or petrol powered engine to eliminate dependency on the epileptic electric power supply is of utmost importance.
4. The use of belt and pulleys to increase power transmission efficiency instead of gearing systems should be encouraged.

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