Design and Production of a Vertical Mobile Palm Fruit Digester

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ABSTRACT

This work designs and produces a portable palm fruit digester which is effective, efficient, affordable and can be transported from one place to another. The aim of the mobile palm fruit digester is to alleviate the problem of transporting harvested palm fruits from source to point of digestion which in most cases cost a fortune. Fresh palm fruits were bought from the market; these fruits are boiled and allowed to cool. The samples are weighed and then feed into the machine; the time for digestion is recorded. The digester mass are then poured on a basin, and the undigested with the partially digested fruits are selected and weighed. The machine is found to have an average digestion capacity of 1090.47kg/hr and average digestion efficiency of 89.82%.The machine is fabricated from locally sourced material and this makes it affordable to many oil palm farmers.

Keywords: *Palm fruit, digester, mobile, machine, design and production*

INTRODUCTION

The need for improvement in fabricated machines to enhance processing of food crops in the field of Engineering brought about the design and construction of the vertical mobile palm fruit digester. The oil palm tree is a tropical plant commonly found in warm climates at altitudes of less than 1600 feet above sea level (Hartley, 1998). It is one of the most economic trees in the world today as virtually every part of the tree is of economic importance to man (Hartley, 1998). The tree produces one of the edible oil for cooking known as palm oil (Asoiro and Udo, 2013). Red palm oil, which is the most important product of the oil palm, is also used for soap and table fats such as

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margarine and for the production of cosmetics, it is a raw material in pharmaceutical industries for the manufacture of drugs (Asoiro and Udo, 2013). The meal obtained after oil extraction from the palm kernel nut is used as animal feed (Asoiro and Udo, 2013). High quality palm oil is needed for domestic and industrial applications (Asoiro and Udo, 2013). The palm oil fruits are drupe prolate and spheroid in shape and the length which varies between 25 – 55mm and having a diameter of 25mm (Hartley, 1998). This consists of a thin outer exo-carp and meso-carp from which palm oil is extracted (FAO, 2002). The ripen fruit is reddish-orange in colour except for the top which may be dark brown based on their relative thickness, they can be classified as either "Dura" or "Pisifera" (FAO, 2002).

With the uses of palm oil being over emphasized, it is necessary to make palm oil available for both domestic and industrial uses. This can only be achieved by digestion, which is the process of releasing the palm oil in the fruit through the rupture or breaking down of the oil – bearing cells (Agbonkhese, Omoikholo and Okojie, 2018). It has been proved that the key process in the oil palm production process is digestion (fruit pulping) (Asoiro and Udo, 2013). This is true in the sense that, it is only when the fruits have been thoroughly pulped that the various oil palm products like edible oil, soap, confectioneries e.t.c. can be derived (Asoiro and Udo, 2013).

Many methods have been applied for the digestion of palm fruit ranging from manual to mechanized methods, but most of these methods are fixed permanently and cannot be moved from one location to another which is a great disadvantage. The main objective of this work is to produce a portable digester which is mobile efficient and can be transported from one place to another.

MATERIALS AND METHOD

The materials used for the fabrication of the mobile digester were mainly mild steel. The design was targeted toward low cost construction, high digestion efficiency and means of mobility. The various parts of the machine were produced using fabrication, assembly and machining processes.

Components of The Vertical Mobile Palm Fruit Digester

i. *Hopper:* The boiled palm fruits are introduced into the machine through the hopper.
ii. *Cylindrical Casing (Digestion chamber):* This is where digestion of the palm fruits takes place.

iii. *Bevel Gear:* The rotating vertical shaft is attached to the Bevel gear, the function of the Bevel gear is to convert horizontal motion to vertical motion.
iv. *Shaft:* The shaft is vertically positioned and the Hammers (beaters) are welded to it.
v. *Hammers:* The Hammers are attached to the vertical shaft to act as bitters. The hammers provide the size reduction effect on the palm fruit.
vi. *Frame:* Angle bars were use to make the frame. The frame provides rigid and skeletal support for the entire machine. Apart from the far corner support of the main frame work, there is handling at one end and design to accommodate tyres for easy movement.
vii. *Discharge:* This is mainly constructed from 3mm thin mild steel plate. These digested mashes are removed from the machine through the discharge.

Mode of Operation of the Machine

A 5 H.P Internal Combustion (I.C) engine was used as the prime mover for the palm fruit digester, a V – belt was used to transfer motion from the I.C engine to pinion shaft of the bevel gear with the help of the bevel gear arrangement, horizontal motion was converted to vertical motion. The hammers (bitters arms) are welded to the vertical shaft. The hammers undertake the digestion process. The digester is set into operation by the help of the I.C engine, it is then allowed to run for some seconds, boiled palm fruits are fed into the digester through the hopper, as the fruits come in contact with the hammers (bitter arms) inside the digester, the fruit are pounded thereby rupturing the oil bearing cells in the fruits (i.e the mexocarp). The digester fruits now form a pulp and the digested mass are collected from the discharge.

Determination of The Crushing Force

To determine the force necessary to digest, an experiment was carried out with cooked palm fruits. This test was carried out to determine the force at which the meso-carp will fail when known masses are dropped from given heights. From the experiment, the average force of failure was determined to be 144.9N, with a factor of safety of 1.5. This crushing force is equivalent to the centrifugal force F_C acting at the hammers (bitters) of the digester aid given as (Khurmi and Gupta, 2008)

$F_C = m\grave{u}^{2-}r$ ……………………………..…... (1)

Where

F_C = centrifugal force (N) = 144.9(N)
m = Mass of blade (kg) = 0.8kg
ù = Angular speed in rad/secs?
r = Length of blade (m) 0.17m

Actual Power Selection

Torque transmitted per digester hammer is given as $T_H = F_C \times r$ ………… (2)
Where,

T_H = Torque transmitted per digester hammer (Nm)

From eq (2) T_H was calculated as 24.64 Nm.
Total torque transmitted = $T = n \times T_H$ …………………. (3)
Where,

n = number of hammers = $4, T =$ Total torque transmitted by hammers.

Substituting values into equal (3), we have T = 98.53Nm

Power Required of The Digester

The power required of the digester is given as

$P_d = TW$ ……………………………………. (4)

Where P_d = power required of the digester.
Substituting into equal (4) P_d was calculated as 3216 W which is equal to 4.311 hp.

Pulley Selection

Pulley and belt arrangement is used to transmit power from driving shaft to the driven shaft (Agbonkhese, Omoikholo and Okojie, 2018).

Torque of electric motor (driving shaft) can be obtained as

Motor power (P_m) = $T_m W_m$ ………………………….. (5)
Where,

P_m = motor power (w),
T_m = motor torque and
W_m = angular speed of motor.

Motor torque $T_m = \frac{P_m}{\omega_m}$ ………………………….. (6)
A 5 h.p electric motor was selected, speed 1200 rev/min.
Motor power P_m = 5 h.p = 3730 watts.

Angular velocity ω_m can be deduced using motor rating as, N = 1200 r pm.

$W_m = \frac{2\pi N}{60} = 125.68$ rad/s

Motor torque (T_m) = 29.68Nm.

To prevent a belt drive due to loading from complete shipping, the motor pulley diameter D_i is given as

$$D_1 = 57\ (T_m)\ ^1/_3 \qquad \text{………………………….. (7)}$$

Where

D_1 = diameter of motor pulley (mm)

D_1 was calculated to be 176mm.

Also, the diameter of the driven pulley was calculated from

$$D_1 = D_1 \times \frac{\omega_1}{\omega_2}(1-\in) \text{ (Hall, Holowenko and Laughlin 2002) ….. (8)}$$

Where, D_2 = diameter of driven pulley (mm),

ù$_1$, = Angular speed of motor shaft (rad/sec)

ù$_2$ = Angular speed of driven shaft (rad/sec),

^{a} = slipping co – efficient = 0.02

Assuming a velocity ratio of 2.0 in order to reduce the speed of 1200 rpm from the 3730 watts into driven shaft we have.

$$\frac{T_1}{T_2} = e^{\mu\theta}$$

$$\frac{\omega_1}{\omega_2} = 2.0 \qquad \text{……………………………. (9)}$$

Therefore D_2 = 344mm

Design for Belt

Determination of belt tension T_1, T_2 involves the transmission of required power by a given belt. The power transmitted by a belt drive is a function of the belt tension and belt speed. This is given as

$$P_m = (T_1 - T_2)V \qquad \text{……………………….. (10)}$$

Where T_1 and T_2 are the belt tensions on the tight side and slack side respectively, V = belt speed (m/s)

Also

……………………….. (11)

Where μ = co–efficient of friction between pulley and belt = 0.3 (for rubberized belt), θ = angle of wrap (contact) of belt on pulley.

Determination of Angle of Wrap

Angle of wrap for a v – belt is determined by

………………….……….. (12)

$$\theta_2 = 180^o + 2\beta = 180^o + 2\sin^{-1}\left(\frac{R-r}{CS}\right)$$ ………………….……….. (13)

$$Sin\beta = \left(\frac{R-r}{CS}\right)$$ ………………..…………(14)

Determination of center distance (CS)

The center distance for a V-belt is obtained as

$$C_s = 0.55\ (D_1 + D_2) + T_B$$ ....….. (15)

Where T_B = thickness of belt. For this design C_S was taken to be 300mm.

Determination of belt length (L_b)

For a V-belt, the length of belt between motor pulley and driven pulley in determined as

$$L_b = 2C + 1.57(D_1 + D_2) + \left(\frac{D_1 - D_2}{4C}\right)$$ ………………….. (16)

Where L_b = Length of belt.

Length of belt was calculated to be 1439.92mm velocity of belt is obtained from

$$V = \frac{T_1 D_1 N_1}{60}$$ ………………………….. (17)

Where

V = velocity of belt m/s

N_1= speed motor (rev/mm),

V was calculated as 11.05m/s

Inputting values into equation (14), we have

$$\beta = 16.26^o$$

inputting values into equations (12) and (13) respectively we have

θ_1 = Angle of wrap for driving pulley = 147.48^0 = 2.58 rad.

= Angle of wrap for driven pulley = 212.52^0 = 3.72 rad.

The load carrying capacity of a pair of pulley is determined by the smaller value of value of (Agbonkhese, Omoikholo and Okojie, 2018).

Substituting values into equation $e^{\mu\theta}$, we have

For driven pulley $e^{\mu\theta} = e^{2.58 \times 0.3} = 2.16$

For driving pulley $e^{\mu\theta} = e^{3.72 \times 0.3} = 3.05$

From above, it shows that the driven pulley governs the design.

Note from equation (11)

$$\frac{T_1}{T_2} = 2.16 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (18)$$

$$T_1 = 2.16\, T_2$$

Substituting values into equation (10), we have T_1 = 291 N and T_2 = 628.56N.

Design for Shaft

A shaft is the rotating machine element which transmits power from one member to the other (Nwaigwe, Nzediegwu and Ugwuoke, 2012). Solid shaft was chosen for this design to satisfy the strength and rigidity requirement. The shaft for the digester is subjected to twisting moment only. For a shaft subjected to twisting moment only, the diameter of the shaft is calculated as.

$$M_t = \frac{T_1}{16} \times \zeta_{max} \times d^3 \quad \ldots\ldots\ldots\ldots\ldots\ldots (19)$$

Where

M_t = twisting moment (Nm)

ζ = torsional shear stress $(N/m)^2$,

d = Diameter of shaft(m)

But $M_t = (T_1 - T_2)\, R$ …………………………… (20)

Substituting values into equal (20)

Take $_{max}$ = 40 MN/m^2

Substituting values into equation (19), we have diameter of shaft (d) = 15.58mm using a factor of safety of 3 we have 15.58 x 3 = 46.74mm

Diameter of shaft = 47mm

Gear Selection

A bevel gear was selected for this design. The bevel gears are used for transmitting power at a constant velocity ratio between two shaft whose axis intersect at a certain angle (Khurmi and Gupta, 2008).

Determination of the Number of Teeth on Pinion and Gear

When two gears are in mesh, the smaller gear is called the pinion and the larger gear is called gear.

The number of teeth on the pinion is given as

(Khurmi and Gupta, 2008) ……………………….. (21)

And the number of teeth on the gear is given as $T_G = \frac{D_G}{m}$ …………… (22)

Where

T_p = number of teeth on pinion,
T_G = number of teeth on gear
D_G = pitch diameter of the gear,
D_p = pitch diameter of the pinion
m = module (mm)

Mitre type of bevel gear was used for this design where the shaft angle is 90^o.

Determination of Induced Bending Stresses in Bevel Gears

Determination of the induced bending stresses in bevel gears comes from the American Gear manufactures association (AGMA) and is given as

$\sigma = W_t \frac{P}{F}\frac{I}{J}$ (Shigley and mischke, 2001) …………………………….. (23)

Where:

σ = max bending stress tooth,
W_t = Transmitted tangential load,
J = Geometry factor [obtained from chart based on the number of teeth on gear and pinion]

F = face width (length of one tooth)mm

Determination of Force Acting At the Mean Radius (Pitch Radius)

To get the tangential force (W_t) acting at the pitch circle, we divide the torque on the gear by the pitch radius as:

$$W_t = \frac{T}{R_m} \text{ (Shigley and mischke, 2001) } \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (24)$$

Where

T = torque on gear,

R_m = mean radius (pitch radius).

Determination of Velocity Ratio (Transmission Ratio)

Velocity ratio for bevel gear is given as:

$$V \cdot R = \frac{D_G}{D_p} = \frac{T_G}{T_p} = \frac{N_P}{N_p} \text{ (Khurmi and Gupta, 2008) } \ldots\ldots\ldots\ldots (25)$$

Where

N_p = speed of pinion (rev/min),

N_G = speed of gear (rev/min),

V.R = velocity ratio.

For this design, the velocity ratio is 1.

Determination of the Pitch Angle (Cone Angle for Bevel Gears)

The pitch angle for bevel gear can be determined from

$$\delta_p = \tan^{-1}\left[\frac{1}{V.R}\right] = \tan^{-1}\left[\frac{D_p}{D_G}\right] = \tan^{-1}\left[\frac{T_p}{T_G}\right] = \tan^{-1}\left[\frac{N_G}{N_p}\right] \quad \ldots\ldots\ldots\ldots (26)$$

According to Khurmi and Gupta (2008),

$$\delta_p = \tan^{-1}[V.R] = \tan^{-1}\left[\frac{D_G}{D_G}\right] = \tan^{-1}\left[\frac{T_G}{T_p}\right] = \tan^{-1}\left[\frac{N_p}{N_G}\right] \quad \ldots\ldots\ldots\ldots (27)$$

Where δ_p = pitch angle for pinion, δ_G = pitch angle for gear.

For this design, $T_p = 29$, $T_G = 29$, $\delta_p = 45^0$, $\delta_G = 45^0$, V.R = 1

Performance Evaluation

Fresh palm fruits were bought from the market. These fruits were boiled and allowed to cool. The samples were weighed and then fed into the machine. The time for digestion was recorded. The digested mass was then poured on a basin, and the undigested with the partially digested fruits selected and weighed. This test was carried out five times, the performance of the digester was evaluated by determining the rate of digestion D_d(kg/hr) and efficiency of digestion $ç_d$(%) (Asoiro and Udo, 2013) as:

……………..…………….....…….. (28)

And $\eta_d = \frac{W_d}{W} 100$ ....…………..………..…….... (29)

Where

η_d = efficiency of digestion (%),
W = initial weight of palm fruit fed into machine(kg),
W_d = Weight of digested fruit(kg)
D_d =rate of digestion (kg/hr) and
T_d = digestion time (mm). The result is tabulated below.

Table 1: Determination of digestion capacity and efficiency

Initial Weight of palm fruit(kg) (W)	Weight of digested fruit (Wd)	Weight of undigested / partially digested fruit (Wp) (kg)	Average digestion time (Td) (min)	Average digestion time (Td)(hr)	Digestion capacity Dd (kg / hr)	Digestion efficiency ηd (%)
5	4.22	0.78	3.6	0.06	83.33	84.4
10	9.02	0.98	4.2	0.07	142.86	90.2
15	13.97	1.03	8.4	0.14	107.14	93.1
20	17.89	2.11	11.6	0.19	105.3	89.5
25	22.97	2.03	13.8	0.23	108.7	91.9

Source: Experimentation (2018)

From the performance results shown in table 1, it can be observed that digestion time increase with increase in mass of the palm fruit. But for the case of the digestion capacity and digestion efficiency, increase in mass can either lead to an increase or decrease in digestion capacity and efficiency respectively. The average digestion capacity was found to be 109.42kg/hr and average digestion efficiency was found to be 89.82%. This means that the machine can digest an average of 109.42kg of palm fruits in one hour.

CONCLUSION

A mobile palm fruit digestion was designed, fabricated and tested to determine its performance. The machine was found to have an average digestion capacity of 109.47kg/hr and average digestion efficiency of 89.82%. This shows that the machine was efficient in digestion of palm fruit. The machine is simple to operate, compact, easy to maintain and being mobile, it can be transported to any location for digestion. The machine cost N50,000 to produce; this makes it affordable to oil palm farmers.

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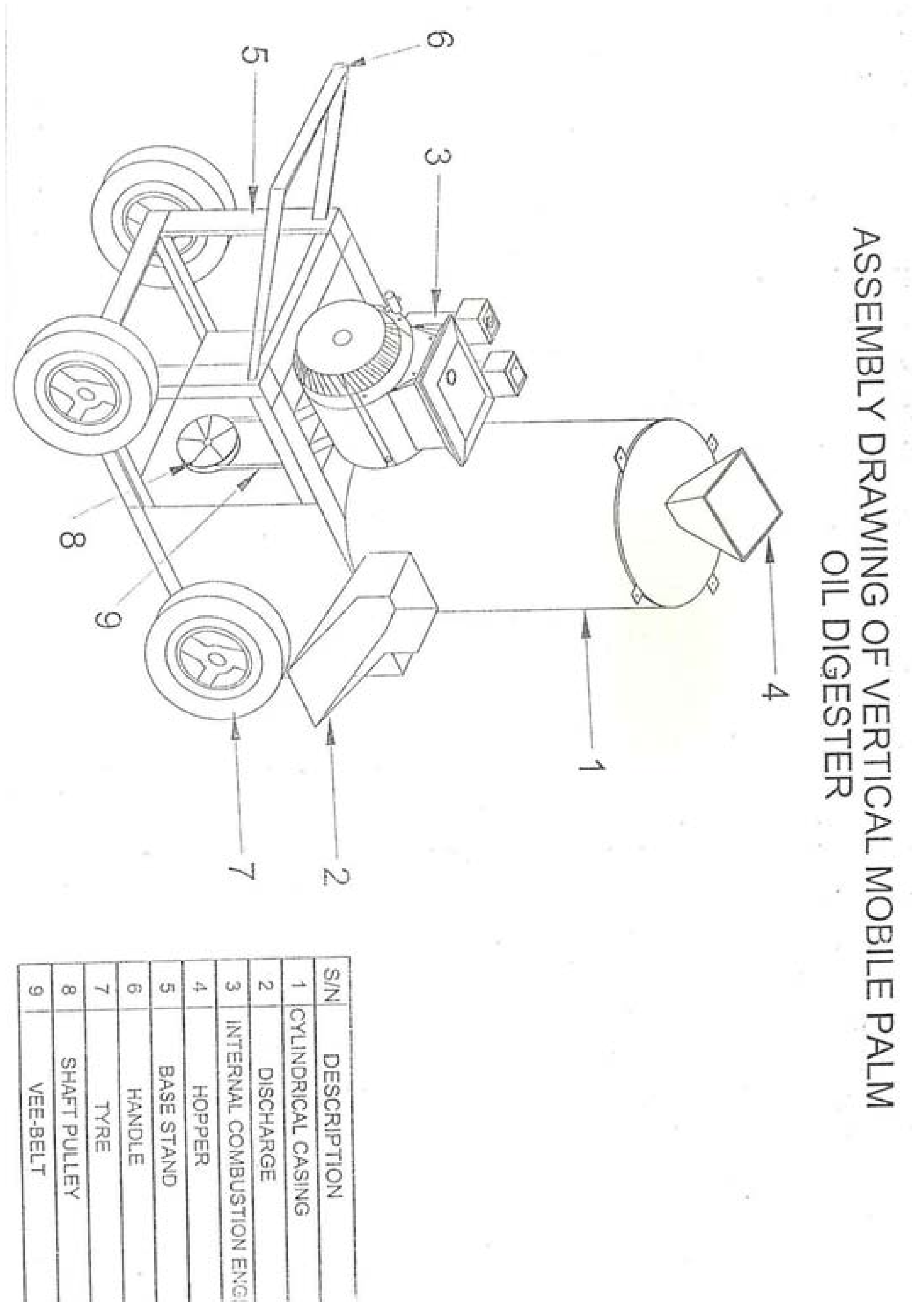
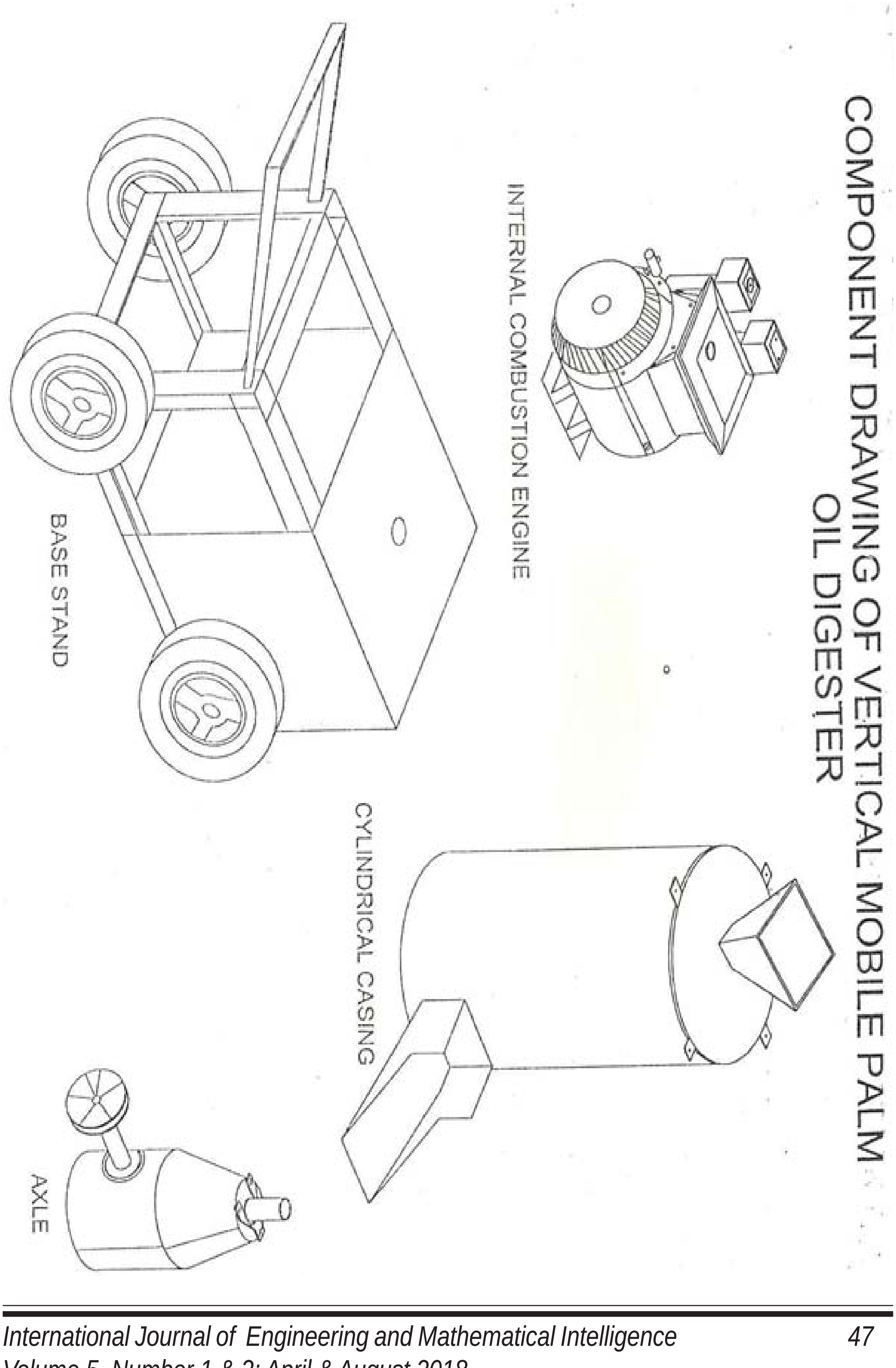

ASSEMBLY DRAWING OF VERTICAL MOBILE PALM OIL DIGESTER
1
2
3
4
5
6
7
8
9
S/N DESCRIPTION
1 CYLINDRICAL CASING
2 DISCHARGE
3 INTERNAL COMBUSTION ENG
4 HOPPER
5 BASE STAND
6 HANDLE
7 TYRE
8 SHAFT PULLEY
9 VEE-BELT


COMPONENT DRAWING OF VERTICAL MOBILE PALM
OIL DIGESTER
INTERNAL COMBUSTION ENGINE
BASE STAND
CYLINDRICAL CASING
AXLE