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Design Analysis of a Modified Jatropha Oil Mechanical Extraction Machine

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ABSTRACT

An existing jatropha oil extraction machine was modified. The main components of the modified machine are; The frame, hopper, feed rate controller, worm shaft, speed step down shaft, extraction barrel, barrel clamp plates, choke, choke regulator wheel, choke lock knob, bed bearings, oil outlet, cake outlet, V- groove pulleys, V-belts, belt guard, belt guard cover, prime mover seating, and prime mover. The components of the modified machine were designed. The designs made include, the determination of the following; power required to drive the machine, load that can be lifted by the screw, pressure to be developed by the screw thread, machine capacity, hopper capacity, belt selection, pulley selection, the frame and others. The determined design power of the machine was 2.64 kW. The load that can be lifted by the screw was 5.82 N. The pressure to be developed by the screw thread was 172.97 N/m2 and the machine capacity was 78.95 kg/hr. In the design of the components considerations were made to select materials for the fabrication of an effective, efficient and affordable machine which was constructed from materials readily available in the local market. The targeted users of the machine are Small and Medium Enterprises (SMEs) because they require minimum investment to set up. This could be a possible solution to the unemployment problem plaguing many Nigerians, by providing a source of self-employment with income.

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INTRODUCTION

The drive to reduce dependence on fossil fuels has been great and this is attributed to gradual depletion of world petroleum reserves and the impact of increasing exhaust emissions on environment and global warming. In view of this, there is an urgent need to develop alternative energy resources such as bio-fuel (Emil et al., 2009). Jatropha curcas seeds (JCS) have been identified to have appreciable amounts of oil that can be extracted and processed to bio-fuel (Gubitz et al., 1999). The calorific value and Cetane number of Jatropha curcas oil are similar to diesel (Narayana and Ramesh, 2006). The oil is safe for use in diesel engines, offer the same performance and engine durability as petroleum-based diesel fuel. It is non-flammable and is characterized with reduced tail-pipe emissions, reduced visible smoke, non noxious fumes and odours (Du et al., 2004). Jatropha curcas seeds oil can be mixed with petroleum-based diesel in any proportion (Khan et al., 2000). Its oil blends can be used in most compression-ignition engines with little or no modifications. These features make the oil an outstanding substitute for fossil fuel and also a counter measure to greenhouse gas accumulation in the atmosphere.

Currently, processing of this crop is largely done manually with hydraulic press. This operation is energy sapping, time consuming and less efficient leading to low output per unit time. In order to mechanize jatropha curcas seed

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processing, there is an urgent need to develop a cost effective jatropha curcas seeds oil extraction machine for cottage industry production capacity, making use of locally sourced materials with indigenous technology which can be sustained and within the technological level of Nigerians in terms of local manufacture and maintenance. The aim of this research is therefore, to design the components of a modified jatropha oil mechanical extraction machine, for Small and Medium Scale processing of the seeds.

Mechanical Extraction

Nde and Foncha (2020) reported that mechanical extraction is one of the oldest methods used for oil extraction. In principle, the seeds are placed between barriers where the volume available to the seed is reduced by pressing, thereby, forcing oil out of the seeds (Elhassan, 2009). Subroto et al. (2014) said that the extraction of the oil from the seed is done in different ways. Methods used are solvent extraction, mechanical extraction, enzymatic extraction and aqueous extraction. For application in rural area, mechanical extraction is considered to be the best option because it has a lower initial investment cost and does not require highly trained personnel to operate the machines. Advantages of mechanical extraction include the production of good quality oil and the possibility of the use of its cake compared to when extraction is done in solvents. The advantages of the screw press over the hydraulic press are its slightly improved yields and its ability to be adapted for continuous processing. Screw press is therefore, adopted for the extraction based on the above reasons.

METHODOLOGY

Design Considerations

Selection of construction materials and costing

The criteria for materials selection for the various components of the machine was based on the type of force that would be acting on them, the work they were expected to perform, the environmental condition in which they would function, their useful physical and mechanical properties, the cost, toxicity of materials and their availability in the local market or the environment (Cornish, 1991). Therefore, the selection of proper materials for the construction of various components of the mechanical oil extraction machine was highly considered. Standard, common sizes, sections as well as semi-finished and finished items which were available in local market were also considered when selecting materials. In addition, selection of the machine components was made, keeping in view the need for effectiveness and efficiency. Consideration was also made to ensure that parts of the machine that come in contact with crushed seeds or oil are stainless steel or PVC, so as to guard against corrosion and rust, of the parts. Also considered was the final cost and quality of work of the constructed machine which depended on several factors, among which were; the cost of materials, the accuracy of the finished parts and the quality of workmanship.

Machine Components Design Analysis

Details of components of the machine designed are given below;

Determination of power required to drive the machine

The power required to drive the machine, P_{Total} was determined using three different formulae for power. The three forms of power were then altogether added to get the total power required to drive the machine. The three forms of power are;

1. Power required for continuous moving and rotating the worm shaft
2. Power required for transporting JCS in the extraction barrel
3. Power required for crushing JCS in the extraction barrel

The power required for continuous moving and rotating of worm shaft can be estimated using equation 1, while the torque developed can be obtained using equation 2 (Khurmi and Gupta, 2007)

Power required for worm shaft movement and rotation

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$P_{SSMR} = \frac{2\,\pi NT}{60}$………………………………….…… (1)

$T = Fr$……………………………...…………… (2)

Where, F= Weight of worm shaft = 170.89 N (by measurement)

T = Torque in Nm

N = worm shaft speed = Average worm shaft speed = $\frac{1}{3}$ (50 + 60 + 70) = 60 rpm

r = Radius of worm shaft = ½ outside diameter of worm shaft

From equation 2, T = 170.89 x $\frac{0.09}{2}$ = 7.69 N m

- Power required for worm shaft movement and rotation

$P_{SSMR} = \frac{2\,\pi\,NT}{60} = \frac{2\pi \text{ x } 60 \text{ x } 7.69}{60}$

= 48.32 Watts

- Power required for transporting JCS in extraction barrel, $P_{Transport}$

1000 mass unit of JCS = 601. 83 g (already determined by measurement)

A unit of JLS = 0.60 g

Therefore, weight of 1 JCS = $\frac{0.60}{1000}$ x 9.81 = 5.89×10^{-3} N

Based on design throughput capacity of 60kg/hr, $\frac{60{,}000}{3600}$ of JCS should be processed per second (i.e 16.67g/sec). This implies that $\frac{16.67}{0.6}$ seeds are transported per second (that is 28 seeds)

Number of seeds transported per hour = 28 x 3,600 = 100,800 seeds

T = Fr

T = $5.89\text{x}10^{-3}$ x $\frac{0.09}{2}$ = 2.65×10^{-4} Nm

P = $\frac{2\pi\ X\ 60\ X\ 2.65\ X\ 10^{-4}}{60}$ = 1.67 x 10^{-3} Watts

Considering coefficient of static friction of 0.50 (stainless steel AISI 304)

P = 1 + 0.50 (1.67 × 10^{-3}) = 2.51 x 10^{-3} Watts

Since 28 seeds are transported per second

$P_{Transport}$ = 28 x 2.51 x 10^{-3}

= 70.28 x 10^{-3}

= 0.07 Watts

- Power required for crushing JCS in the extracting barrel

The power required for JCS crushing was estimated using the power formula (Khurmi and Gupta, 2007) given in equation 3;

$P = \frac{FD}{t}$

Where;

F, D and t are force in N, distance moved in m and time in seconds

$P_{Crushing} = \frac{F_R D_g}{t}$ ………………………………… (3)

Where;

D = Dg is average geometric mean diameter of JCS = 10.97mm, = 0.01097m, already determined using the physical dimensions of jatropha seed.

F = F_R = Average rupture force of JCS at four moisture levels (102.46 N)

The average rupture force of JCS at four varied MC levels, F was determined in the laboratory, thus;

$F = \frac{\Sigma R_f}{4}$…………………………………………… (4)

ΣR_f = summation of rupture force [N]

F = Average rupture force of JCS

F = $\frac{409.83}{4}$ = 102.46 N

t = unit time, sec

$P_{crushing} = \frac{102.46 \text{ x } 0.01097}{1}$ = 1.124

1.124 x 24 = 26.98 Watts for one seed transported in the extraction barrel.

But, 28 seeds are transported per second in the extraction barrel (as determined earlier). Therefore, power required for crushing 28 seeds in the extraction barrel is

$P_{Crushing}$ = 1.124 x 28 = 31.47 Watts

$P_{Total} = P_{SSMR} + P_{Transport} + P_{Crushing}$

= 48.32 + 0.07 + 31.47

= 79.86 watts

Considering the elemental inner area of external barrel, ΣA_{EB} for JCS Pressing and elemental inner area of choke plate, ΣA_{CP}

Elemental inner area of barrel, $\Sigma A_{EB} = \pi\,(R_2^2 - R_1^2)$ ……………………………………..……(5)

Where R_1 = Internal radius of extraction barrel [mm]

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R_2 = External radius of worm shaft [mm]
$\Sigma A_{EB} = \pi\ (0.050^2 - 0.045^2) = 1.49\times10^{-3}\ mm^2$

Elemental inner area of choke plate, $\Sigma A_{CP} = (R_3^2 - R_4^2)$………………………………………………. (6)
Where;
R_3= Radius of choke gap [mm]
R_4=Radius of shaft inside the choke gap [mm]
$\Sigma A_{CP} = \pi\ (0.022^2 - 0.02^2)$
$= 2.64\times10^{-4}\ m^2$
Reduction in size = Elemental inner area of extraction barrel - Elemental inner area of choke plate
$= \Sigma A_{EB} - \Sigma A_{CP}$………………………………………………… (7)
$= 1.49\times10^{-3} - 2.64\times10^{-4} = 1.23\times10^{-3}\ m^2$

Percentage reduction in size
$= \frac{\text{Reduction in size}}{\Sigma A_{EB}}$
$= \frac{1.23\ \text{x}10^{-3}}{1.49\ \text{x}10^{-3}} \times 100$
= 82.55%

Restriction at the end of extraction barrel offered by the choke plate reduces the flow rate by 82.55%. Hence the total power obtained should be increased to achieve the design throughput. The total power, P_{Total} should be increased by multiplying it with 0.4 of percentage reduction in choke Plate (Gates Industrial, V-Belt Drive Design Manual, 2004)
Design Power, P_{Design} = 0.4 x 82.55 x P_{Total}……………….………………. (8)
$P_{Design} = 0.4 \times 82.55 \times 79.86 = 2636.98$ W
= 2.64 kW

Design of screw thread

To design the screw thread, the following design requirements and calculations were established;

1. Diameter of extraction barrel, D_{EB} = 100 mm
2. Length of extraction barrel, L_{EB} = 401 mm
3. Thread (screw winding breath), = 7 mm

The geometric mean diameter, Dg of JCS ranges between 9.67 – 12.12 mm. The thread pitch should be able to accommodate all the ranges of Dg of JCS;

1. Thread pitch, Tp = 15 mm
2. Considering tapered worm shaft with outside diameter = 90 mm
3. Root diameter of screw shaft at intake: 70 mm
4. Root diameter of screw shaft at exit = 80 mm
5. Root diameter of screw winding at midpoint, d_1 = 85 mm
6. Depth of screw winding at intake, d_i = 10 mm
7. Depth of screw winding at exit, d_e = 5 mm
8. Average depth of screw winding, $d_a = \frac{di + de}{2}$ = 7.5 mm
9. Length of shaft covered with screw, L_1 = 380 mm

Total number of screw windings on the screw shaft
$\frac{L_1}{\text{B+T}_\text{p}}$
$= \frac{380}{7+15} = 17.3$
~ 18 windings (should be whole number)

The perimeter (or length) of a screw winding was determined using the formula for calculating the circumference of a circle.
Circumference of a circle = $2\pi r$
Where, r is the radius of the circle
Perimeter or length of a screw winding
L = 2πr = πd
= 2x π x85
= 267.04 mm

To determine the volume of a screw winding the formula for calculating the volume of a cuboid was used
Volume of a cuboid = L x B x h
L, B and h are length, breadth and height or depth of the screw winding, respectively.
Volume of a screw winding Vsw = L x B x h
= 267.04 × 7 × 7.5
= 14019.36 mm³

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Volume of 18 screw windings, $18V_{sw}$ = 18 (L x B x h)
= 18×14019.36
= 252,348.43 mm^3

Determination of load that can be lifted by the screw

In determining the volume of screw shaft (V_{ss}) and the inner volume of extraction barrel the formula for calculating the volume of a cylinder was used

Volume of a cylinder: $\pi r^2 h$………………………………………………. (9)

Where;

r and h are the radius and height of the cylinder respectively.

V_{ss}, considering root diameter = $\pi r_1^2 L_1$

Where;

r_1 and L_1 are the root radius of screw winding at midpoint and the length of shaft covered with screw respectively.

V_{ss}, Considering root diameter……………… (10)

= π x $(\frac{85}{2})^2$ x 380

= 2,156,310.658

= 2.156 x 10^6 mm^3

Inner volume of the extraction barrel

= $\pi\ r_2^2\ l_2$………………………………………… (11)

Where, r_2 and l_2 are the radius and length of extraction barrel respectively.

= π x50^2 x 401

= 3,149,446.635

= 3.149×10^6 mm³

Inner volume of extraction barrel occupied by screw shaft = Volume of 18 screw windings + Volume of screw shaft considering the root diameter………………...…………. (12)

= 2.524 x 10^5 + 2.156 x 10^6 mm^3

= 2.408 x 10^6 mm^3

Inner volume of extraction barrel for JCS pressing = Inner volume of extraction barrel - Inner Volume of barrel occupied by screw shaft………………………………………… (13)

= 3.149 x 10^6 – 2.408 x 10^6

= 7.41 x 10^5 mm^3 [Volume for seed pressing]

= 741 cm^3 [Volume for seed pressing]

Bulk density of JCS = 0.40g/cm³………………………………. (14)

Assuming the volume of JCS can be reduced by half (½) by crushing and pressing its bulk density after pressing, ρ_{ba} will be twice of its bulk density before pressing, ρ_{bb}. Therefore, the mass of JCS that the extraction barrel can accommodate, that is the mass of JCS that can be lifted by the screw (which can be expressed as M = ρ_{ba} x inner volume of barrel for JCS pressing) will be equal to (0.8 x 741) = 592.8 g = 0.593 Kg

Load to be lifted by screw $= M\ X\ g$

= 0.593 x 9.81

= 5.82 N

Determination of the pressure to be developed by the screw thread

To determine the pressure to be developed by the screw thread, the relationship below was used;

$P = \frac{F\ or\ L}{A}$…………………………………………….(15)

Where;

P = Pressure, in N/m^2

F = Force or load to be lifted by screw, N

A = Area of a screw, mm^2

= L X B

= 267.04 x 7

= 1869.28 mm^2

Total area of 18 screw windings,

= 1869.28 x 18

= 33647.04 mm^2 = 33647.04 x 10^{-6} m^2

$$P = \frac{5.82}{33.647\ x\ 10^{-3}}$$

= 172.97 N/m^2

Determination of machine capacity

The capacity of the machine was determined using the relationship given by Samar et al. (2019), below;

$M_C = \frac{M_S}{t}$………………………………………. (16)

Where;

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M_C is the machine capacity, kg/hr, M_s is the weight of input sample, kg and t is time required to empty the hopper, hr.

M_s =1.25 kg, t = 57secs = $\frac{57}{3600}$ hr. Therefore, M_C = $\frac{1.25\ X 36}{57}$ = 78.95 kg/hr

Determination of hopper design

The total volume of hopper V_H was calculated by considering the three separate sections of the hopper, then calculating the volume of the individual sections, thereby adding the three volumes to get the total volume, as can the as can be seen in Fig 2.1 (A & B)

The first section is a cylinder, the second section was a conical frustum and the last section was also a cylinder.

From fig. 3.3 B,

$\frac{x}{37} = \frac{x+233}{177}$

177x = 37x +8621

140x = 8621

x= $\frac{8621}{140}$ = 61.58mm

Vol. of frustum OPRSO = Volume of big cone, OPQO, - Volume of small cone, SRQS

= $\frac{1}{3}\pi$ (177^2 x 294.58 - 37^2 x 61.58)

Vol$_2$= 9, 580,050. 648mm^3

= 0.009580m^3

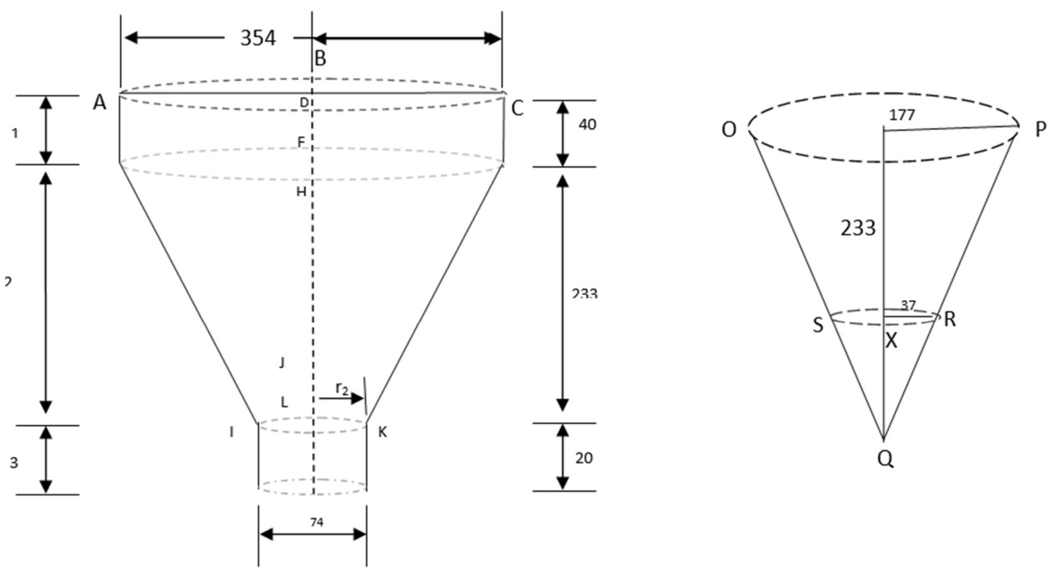
Vol$_3$, which is also a cylinder (hopper neck)

= π x 37^2 x 20

= 86,051.429 mm^3

= 0.00008605 m^3

Total volume of hopper, V = Vol$_1$ + Vol$_2$ + Vol$_3$



(A) (B)

Fig. 2.1 (A): Complete shape of the hopper and (B), representation of middle section of the hopper.

Vol$_1$ which is cylinder ABCGHE = π r$_1$ h$_1$

= π x 177^{2}x40

= 3,938,502.857mm^3

= 0.0039385m^3

Vol$_2$ which is conical frustum

= (3,938,502.857 + 9,580,050.648 + 86,051.429)

= 13,604,604.93 mm³

= 0.0136 m^3

Weight of the hopper

Mass of hopper, M = ρ x V

(Density of stainless steel x Vol$_3$) + (density of mild steel x Vol$_1$) + (density of mild steel x vol$_2$)

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ρ_{ss}= Density of stainless steel, AISI 304 = 7930Kg/m³ (Amardeep steel Centre, Amardeepsteel.com)
ρ_{ml}= Density of mild steel, = 7860 kg/m³ (Sharma, htts://www.Scribed.com
Total mass of hopper = 30.9566 + 75.2988 + 0.6824 = 106.9378 kg
Weight of hopper = 106.9378 x 9.81 =1049.06 N

Belt selection

- Determination of belt tension

The belt tension was determined in accordance with Khurmi and Gupta (2007)
P = (T_1-T_2) V…………………………………. (17)
Where P = Design Power = 2640W
T_1 and T_3 = Belt tension at tight side [N]
T_2 and T_4 = Belt tension at slack side [N]
V = speed of belt [m/s]

- Determination of speed ratio

$$N_1 D_1 = N_2 D_2$$
1440 × 0.09 = N_2 x 0.3
$N_2 = \frac{1440 \times 0.09}{0.3}$ = 432rpm
$$N_2 D_2 = N_3 D_3$$
432 × 0.15 = N_3 x 0.3
$N_3 = \frac{432\ X\ 0.15}{0.3}$ = 216rpm

Where N_1, N_2 and N_3 are speeds at prime mover, speed reduction shaft and screw shaft, respectively, in rpm
D_1, D_2 and D_3 are diameter of pulleys at prime mover, speed reduction shaft and at screw shaft, respectively, in m.

- Determination of belt speed

Considering the prime mover pulley (main driver)
Where; N = 1440rpm (Speed at prime mover)
Diameter = 0.09 m (Prime mover pulley diameter)
V = $\frac{\pi dN}{60}$ = $\frac{\pi \times 0.09 \times 1440}{60}$ = 6.79m/s

Considering the speed reduction pulley (auxiliary driver)
Where; N = 432 rpm (speed at speed reduction shaft pulley)
Diameter = 0.15m (speed reduction shaft pulley diameter)
V = $\frac{\pi dN}{60}$ = $\frac{\pi \times 0.15 \times 432}{60}$ = 3.39 m/s

Pulley selection

- Determination of angle of wrap

Angle of wrap is the angle subtended by the length of belt which is wrapped around the pulley. It was calculated using equation 18 as given by Khurmi and Gupta (2007).
$\theta = [180 - 2\sin^{-1}(\frac{D-d}{2c})]\frac{\pi}{180}$........... (18)

Where;
θ= Angle of wrap [rad]
D = Diameter of the driven pulley [mm]
d = Diameter of the driver pulley [mm]
C: Centre distance between driver pulley and the driven pulley [mm]

Depending on available space and the required compactness, the Centre distance, C can be selected using equation 19 and 20 (Khurmi and Gupta, 2007).
D < C < 3 (D + d)………………….……….. (19)
C > $\frac{D+d}{2}$ + D ………………………………… (20)

Considering speed reduction shaft,
From equation 19,
300 < C < 3 (300 + 90)
300 < C < 1170
From equation 20,
C > $\frac{300+90}{2}$ + 300
C > 495

Let C = 560mm [space constraint]
From equation 18,
θ = [180 - 2sin^{-1} ($\frac{300-90}{2 \times 600}$)]$\frac{\pi}{180}$
= [180 – 2 sin $^{-1}$ (0.175)] $\frac{\pi}{180}$
= [159.8]$\frac{\pi}{180}$
= 2.79 rad.

Considering worm shaft
From equation 19,
300 < C < 3 (300 +150)
300 < C < 1,350
From equation 20,
C > $\frac{300 + 150}{2}$ + 300
C > 525

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Let C = 540mm [Space constraint]
From equation 18,

θ = [180 – 2 sin $^{-1}$ ($\frac{300-150}{2 \times 540}$)] $\frac{\pi}{180}$

= [180 – 2 sin $^{-1}$ ($\frac{150}{1080}$)] $\frac{\pi}{180}$

= [180 – 2 sin $^{-1}$ 0.1389] $\frac{\pi}{180}$

θ = [164.03] $\frac{\pi}{180}$

θ = 2.87 rad

To determine the belt tension recall that
Design Power, P = 2640 W
Considering the horizontal drive, prime mover pulley governs the design [Main driver]

P = (T_1 – T_2) 6.79

2640 = (T_1 - T_2) 6.79

T_1-T_2 = $\frac{2640}{6.79}$ = 388.81

T_1 - T_2 = 388.81 N .... (21)

The relation between T_1 and T_2 for the V-belt drive is

2.3log ($\frac{T_1}{T_2}$) = $\mu\ \theta\ cosec\ \beta$..................... (22)

Where μ = Frictional coefficient of belt and pulley = 0.4 (khurmi and Gupta, 2007)

θ = Angle of wrap = 2.79 rad

β = $\frac{1}{2}$ Groove angle = 17.5°

From equation 22,

Log ($\frac{T_1}{T_2}$) = $\frac{\mu\ \theta\ cosec\ \beta}{2.3}$

$\frac{T_1}{T_2}$ = antilog ($\frac{\mu\ \theta\ cosec\ \beta}{2.3}$)

$\frac{T_1}{T_2}$ = antilog ($\frac{0.4 \times 2.79 \times cosec\ 17.5}{2.3}$)

$\frac{T_1}{T_2}$ = antilog $\frac{0.4 \times 2.79 \times 3.326}{2.3}$

$\frac{T_1}{T_2}$ = antilog (1.6138)

$\frac{T_1}{T_2}$ = 41.10 N

T_1 = 41.10 T_2.................................... (23)

Substitute T_1 = 41.10 T_2 in equation 21

41.10 T_2 – T_2 = 388.81

40.10 T_2 = 388.81 N

T_2 = $\frac{388.81}{40.10}$ = 9.70 N

From Equation 3.49, T_1 = 41.10 T_2
Therefore,

T_1 = 41.10 x 9.70

= 398.67 N

When the effect of Centrifugal force on belt is taken into account, then;

T = T_1 + C_f.................................... (24)

C_f = M V^2.. (25)

M = $\frac{1}{2}$ (W_1+W_2) t x ρ......................... (26)

Where;
T = Maximum tension [N]
C_f = Centrifugal force in belt [N]
M = Belt mass per meter [kg/m]
Using standard V-Belt of "type" the following sizes are obtained:
W_1 = Top width of belt = 17mm
W_2 = Bottom width of belt = 9.5mm
t = Belt thickness = 11 mm
ρ = Mass density of Belt: 1250 kg/m^3 [khurmi and Gupta, 2007]

From equation 26,

M = $\frac{1}{2}$ (17 x 10^{-3} + 9.5 x 10^{-3}) 11 x 10^{-3} x 1250

= 0.1822 kg/m

From equation 25,

C_f = 0.1812 x 6.79^2

= 8.40 N

From equation 24,

T = 398.67 + 8.40

= 407.07 N

Considering Vertical drive, speed reduction pulley governs the design [auxillary driver]

P = (T_3 - T_4) 3.39

2640 = (T_3 - T_4) 3.39

T_3 - T_4 = $\frac{2640}{3.39}$ = 778.76 N

T_3 - T_4 = 778.76 N... (27)

Also, the relation between T_3 and T_4 for the V-belt drive is;

2.3log ($\frac{T_3}{T_4}$) = $\mu\ \theta\ cosec\ \beta$.................................. (28)

Log ($\frac{T_3}{T_4}$) = $\frac{\mu\ \theta\ cosec\ \beta}{2.3}$

$\frac{T_3}{T_4}$ = antilog $\frac{\mu\ \theta\ cosec\ \beta}{2.3}$

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$\frac{T_3}{T_4}$ = antilog $\frac{0.4\ x\ 2.87 \times cosec\ 17.5}{2.3}$

$\frac{T_3}{T_4}$ = 45.69 N

T_3 = 45.69 T_4.................................... (29)

When the equation T_3 = 45.69 T_4 is substituted in equation 27, then;

45.69 T_4 - T_4 = 778.76

44.69 T_4 = 778.76

$T_4 = \frac{778.76}{44.69}$

=17.43 N

From equation 29,

T_3 = 45.69 x 17.43

= 796.38 N

T = T_3 + C_f.. (30)

$C_f = MV^2$

M = $\frac{1}{2}$ (W_1+ W_2) t x ρ

= 0.1822kg/m (As obtained from equation 26)

C_f = 0.1822 x 3.39^2

= 2.09 N

T = 796.38 + 2.09

= 798.47 N

Determination of belt length

The length of the belt was obtained from equation 31 as given by Khurmi and Gupta (2005);

$L_b = \frac{\pi}{2}(d_1 + d_2) + 2x + \frac{(d_1-d_2)^2}{4x}$ (mm) .... (31)

Where;

L_b = Total length of the belt,

d_1 = Diameter of larger pulley,

d_2 = Diameter of smaller pulley,

x = Distance between the centres of two pulleys, where; 560 mm was adopted as the centre distance from the prime mover pulley diameter (main driver) and speed reduction pulley (auxiliary driver).

$L_b = \frac{\pi}{2}(150 + 90) + 2 \times 560 + \frac{(150-90)^2}{4\ x\ 560}$

= 1.571(240) + 1120 + 1.607

= 377.04 + 1120 + 1.607

= 1498.647 mm, = 1499 mm

Design of frame d_{is}

Surface area of the frame, ABCD

Area, $A = L\ X\ B$

$= (425 + 195)\ X\ 240$

$= 148{,}800\ mm^2 = 148.80\ x\ 10^{-6}\ m^2$

Load the frame can withstand

Permissible compressive stress was calculated from equation 32 as given by Khurmi and Gupta (2005);

$s_c = \frac{P}{A}$ .... (32)

Where;

s_c is permissible compressive stress in N/m^2

P = Load in N

A = Area in m^2

P = s_c x A

= 75 x 10^6 x 148.80 x 10^{-6}

= 11.16 k N

Weight of machine acting on the frame = 1.86 k N (as measured)

Surface area of the frame = 148.80 x 10^{-6} m^2

Compressive stress acting on the surface of the frame;

$s_c = \frac{P}{A}$

$= \frac{1860}{148.80\ x\ 10-6}$

= 12.5 x 10^6 N/mm^2

So, the frame will not collapse due to the stress exerted on it, since the stress exerted on it is within the permissible stress.

Volume of the frame

Volume of frame = Volume of frustum ABCDEFGH. It is a pyramidal frustum.

Volume of frustum = Volume of big pyramid (AICD) – Volume of small pyramid (EIGH)

Volume of pyramid = $\frac{1}{3}\ x\ base\ area\ x\ h$

Volume of frustum = $\frac{1}{3}\ x\ base\ area\ x\ H - \frac{1}{3}\ x\ base\ area\ x\ h$

Volume of frustum = $\frac{1}{3}\ x\ (620\ x\ 480) x\ 1224.84 - \frac{1}{3}\ x\ (240\ x\ 320)\ x\ 624.84$

Volume of frustum = $121{,}504{,}128 - 15{,}995{,}904$

Volume of frustum = 105,508,224 mm^3

= 0.106m^3

Working Principle of the Machine

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The working principle of the jatropha oil mechanical extraction machine is explained in accordance with the assembled view of the machine (Fig. 1). The jatropha seeds ready to be processed are introduced into the hopper (6) while the feed rate controller (4) is already closed. The operator will then kick start the diesel engine (prime mover), (2) and allow the engine performance to stabilize. The operator will then set the desired pressure of operation by regulating the choke using the choke regulator wheel (11) and will then lock the wheel in a fixed point using the choke nut/knob (10).

Now, all the necessary settings are made and the power engine is stable, the operator will then pull out the feed rate controller (4) to the desired aperture so as to allow a regulated quantity of jatropha seeds get into the extraction barrel (7). The barrel which is the housing in which crushing and squeezing of seeds for oil extraction takes place is incorporated with an uneven screw type tapered shaft. The shaft has decreasing pitch away from the hopper and an increasing diameter away from the hopper (inlet point of the jatropha seeds). This unique property of the shaft allows it to pack much material (processed jatropha seeds) and gradually confine it in its subsequent continuously reducing compartment, thereby, increasing compression and pressure hence, expelling the oil present in the material (jatropha seeds). The oil that is extracted drains out through the perforations beneath the barrel and be collected by the oil collector and finally, delivered through the oil outlet (12). The extreme end of the barrel is closed by a thick barrel clamp plate which provides an aperture for the ejection of the press cake. This aperture at the barrel clamp plate is regulated with the aid of the choke present on the shaft. The wider the aperture the faster the rate of press cake ejection and the lower the compression and pressure that is inside the barrel and vice versa. The press cake is ejected through the cake outlet. All components of the jatropha oil mechanical extraction machine were logically arranged on the frame (13) while, the prime mover was mounted on the prime mover seating (1). The power generated from the prime mover was transmitted using belt and pulley system (3) which was properly guarded, for operator's safety and against harsh weather conditions, using a belt guard (5).

RESULTS AND DISCUSSION

Figure 1 below shows the assembled view of the designed machine. The Jatropha oil mechanical extraction machine was designed to cater for the needs of Small and Medium Scale entrepreneurs for domestic use, or use by independent entrepreneurs, on a daily hire bases for commercial purpose. The machine can be driven from one location to another when mounted on a moving truck. It is also of low cost when compared with imported oil extraction machines The main components of the modified machine are; The frame, hopper, feed rate controller, screw shaft, speed step down shaft, extraction barrel, barrel clamp plates, choke, choke regulator, choke regulator wheel, choke regulator cover, choke lock knob, bed bearings, oil outlet, cake outlet, sole, V-groove pulleys, V-belts, belt guard, belt guard cover, prime mover seating, and prime mover. The source of power is generator engine.

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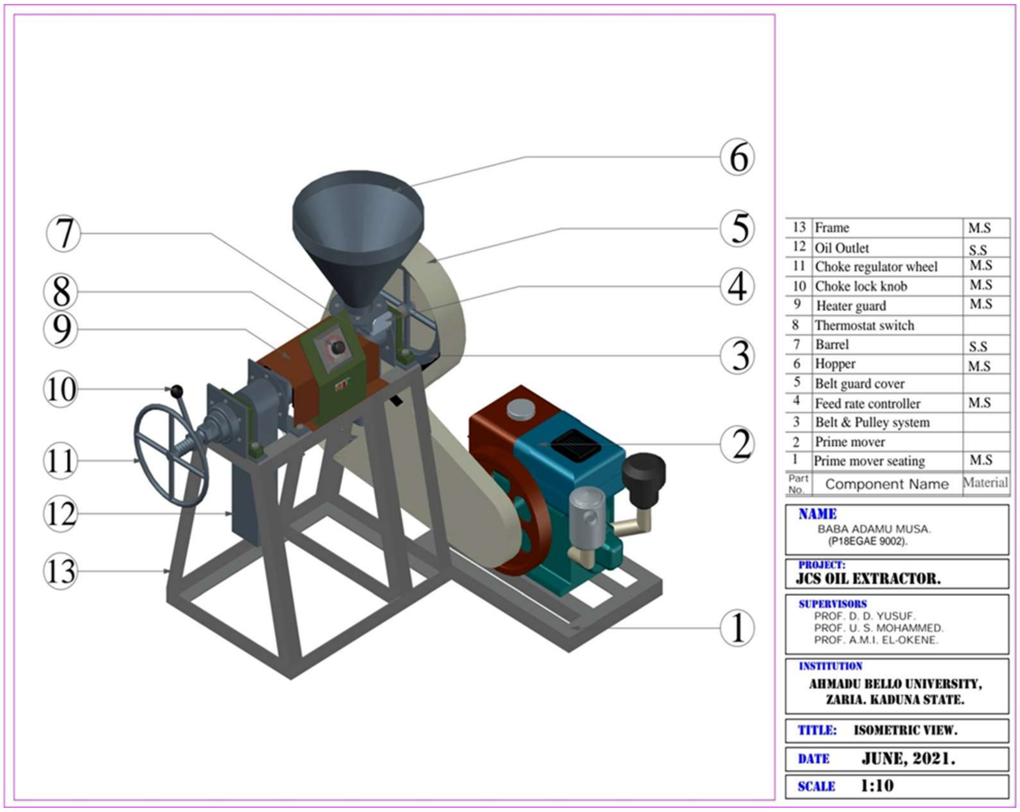

Fig. 1: Assembled view of the jatropha oil mechanical extraction machine

CONCLUSION

A jatropha oil mechanical extraction machine was successfully designed for Small and Medium Scale processing of jatropha seeds. Some of the designs carried out include the determination of the following; design power, design of screw thread, load that can be lifted by the screw, pressure to be developed by the screw thread, machine capacity, hopper design, weight of hopper, belt selection, pulley selection, belt length, load the frame can withstand, volume of the frame. The design of the components was conducted in accordance with engineering codes and standards.

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