

Design of a Jatropha Oil Expelling Machine

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Abstract

Biodiesel is a fuel type that has already been proven suitable for use in diesel engines. The non-edible vegetable oil of *Jatropha curcas*, which grows in tropical and subtropical climates across the developing world, promises a commercially viable alternative to diesel. Jatropha oil is renewable, environmental friendly and can be easily produced in rural areas. This paper presents the design of an efficient Jatropha oil expelling machine carried out at the University of Lagos, Nigeria. The design of this Jatropha oil expelling machine will enhance the production of biodiesel from Jatropha seeds. The designs of the crushing system and pre-heating systems before pressing will enable the machine operate at efficiency greater than 90%. This machine is easy to maintain and economic for commercial uses. The Jatropha oil expelling machine has been designed to a capacity of 1000 kg/hr and can expel oil at the rate of 2.2 m³/hr. The machine is powered using electric motor of 40kW to be operated for 8 working hours. This machine is an improvement over existing machine and presupposes appropriate technology for manufacture, operation and maintenance.

Keywords: Jatropha oil, expelling, efficiency, biodiesel, machine design, crusher, pre-heating.

INTRODUCTION

Biodiesel is a fuel type that has already been proven suitable for use in diesel engines. Biodiesel consists of fatty acid methyl esters (FAMES) of seed oil and fats and is produced through transesterification (Knothe and Steidley, 2005). The non-edible vegetable oil of *Jatropha curcas*, which grows in tropical and subtropical climates across the developing world (Openshaw, 2000), promises a commercially viable alternative to diesel as it has the desired physiochemical and performance characteristics comparable to diesel to facilitate continuous operation without much change in engine design. Jatropha oil has little or no sulphur content (Ogbonnaya, 2010), hence it is a promising alternative because it is renewable, environmental friendly and can be easily produced in rural areas.

Mechanical pressing and solvent extraction are the most commonly used methods for commercial oil extraction. Screw pressing is used for oil recovery up to 90-95%, while solvent extraction is capable of extracting 99% (Shahidi, 2005). In spite of its slightly lower yield, screw pressing is the most popular oil extraction method as the process is simple, continuous, flexible and safe (Zheng et al., 2005; Singh and Bargale, 2000). The extraction efficiency of oil expellers is approximately 60% (Calle et al., 2007).

Mechanical properties such as rupture force and energy required for rupturing fruit, nut and kernel provide insights on how to adapt the pressing process to Jatropha seeds (Beerens, 2007; Openshaw, 2000;

Heller, 1996; Henning, 2004). Beerens, (2007) gives some of the required properties. The aim of this work is to design an efficient Jatropha oil expelling machine that improves on the performance of previous ones and which can be sustained by the available technology in Nigeria. The machine was designed at the University of Lagos, Nigeria, and should be able to deliver 97% efficiency and have a capacity of 1000kg/hr.

DESIGN CONSIDERATIONS AND ANALYSIS

The Jatropha oil expelling machine is designed to have the following components: hopper, crusher, screw conveyor, barrel, bearings, power transmission elements (electric motor, gear box, belts and pulleys), constriction die, oil outlet or sieve, oil container.

Design of the Crusher

The crusher is designed as a pair of meshing parallel helical gears. It has an axial thrust force which has the tendency of pushing the crushed seed in the axial direction. The parameters assumed for the design are: pressure angle in the plane of rotation, $\phi_t = 20^\circ$; helix angle, $\psi = 20^\circ$; rupture or crushing force, $F = 150 \text{ N}$; number of teeth on gear, $Z = 10$; gear ratio = 1; and the largest size of a seed, s , is taken as 25 mm. For adequate operation of the crusher, it is required that the width of the gear should approximate the size of the seed to be crushed. The relation of the size of the nut with the module in the normal plane is given as $1.571m_n \geq s$; $1.571m_n \geq 25\text{mm}$; hence, $m_n = 15.9\text{mm}$ (Singh, 2008)

A preferred module of 16 mm is chosen. The module in the plane of rotation is

$$m = \frac{m_n}{\cos \psi} = \frac{16}{\cos 20^\circ} = 17 \text{ mm} \quad (1)$$

The transverse circular pitch of the gear is $P_t = 50.3 \text{ mm}$. The minimum face width of the gear is estimated as $f_{min} = 158.9 \text{ mm}$ but a face width of 160 mm is chosen. For 10 teeth on the helical gear, the pitch circle diameter of the gear is estimated to be 170 mm. The pressure angle in the normal plane, ϕ_n , is evaluated as 18.9° . The normal base pitch, P_{bn} , is 47.2 mm. The base helix angle, ψ_b , is 18.7° . The transverse base pitch, P_{bt} , is 46.4 mm. The base circle diameter, D_b , is 159.7 mm. The axial circular pitch, P_x , is 138.2 mm. The centre distance of the meshing gears is 170 mm. The maximum value for the addendum, a , is 12.8 mm. The minimum value for the dedendum, d , is 16 mm. The minimum total depth of tooth is 28.8 mm. The minimum clearance is 3.2 mm. The addendum circle diameter, D_a , is 202 mm.

The dedendum circle diameter, D_d , is 130 mm. The lead is 1,467.3 mm. The virtual or formative number of teeth is expressed as (Shigley and Mischke, 2003; Sharma and Aggarwal, 2006):

$$Z_v = \frac{Z}{\cos^3 \psi} = \frac{10}{\cos^3 20^\circ} = 12 \text{ teeth} \quad (2)$$

The length of tooth surface in the normal plane is (Singh, 2008; Shigley and Mischke, 2003):

$$f_n = \frac{f}{\cos \psi} = \frac{160}{\cos 20^\circ} = 170.3 \text{ mm} \quad (3)$$

The tangential force to the pitch cylinder in the plane of rotation is estimated by using the modified Lewis equation for helical gears which is given as (Singh, 2008; Sharma and Aggarwal, 2006):

$$F_t = \frac{\sigma_b \cdot f \cdot m_n \cdot Y_f \cdot C_v}{C_w} \quad (4)$$

Where σ_b = safe working stress or design stress; a design stress of 56 MN/m^2 was selected for ordinary cast iron, BHN180; f = face width = 160 mm ; Y_f = Lewis form factor corresponding to T_v teeth; m_n = module in the normal plane = 0.016 m ; $Y_f = \pi y$; Where, for 20° pressure angle, full depth system, C_v = velocity or dynamic factor;

$$y = 0.154 - \frac{0.912}{Z_v} = 0.154 - \frac{0.912}{12} = 0.078 \quad (5)$$

Therefore,

$$Y_f = \pi y = \pi \times 0.078 = 0.245 \quad (6)$$

From Barth's equations (Singh, 2008):

$$C_v = \frac{4.5}{4.5 + v} \quad (7)$$

The gear assembly is assumed to operate at a speed of 120 rpm, hence

$$v = \frac{\pi D N}{60} = \frac{\pi \times 0.17 \times 120}{60} = 1.07 \text{ m/s} \quad (8)$$

Therefore, $C_v = 0.81$. C_w = wear and lubrication factor = 1.35 for indifferent lubrication. Lewis equation for helix gears is used to estimate the tangential force, F_t , to be 20.7 kN. The force normal to the pitch element of the helical tooth which is also the crushing force is (Singh, 2008; Sharma and Aggarwal, 2006):

$$F' = \frac{F_t}{\cos \psi} = \frac{20597.6}{\cos 20^\circ} = 22,025.9 \text{ N} = 22 \text{ kN} \quad (9)$$

The axial force, F_a , which is the force of propulsion of the crushed seed is 7.5 kN. The radial force, F_r , which tends to move the gears apart in the radial direction is 7.5 kN. The power required to drive the gear can be estimated using (Singh, 2008):

$$P = F_t \cdot v = 22,146.4 \text{ W} = 22.1 \text{ kW} \quad (10)$$

The required torque is (Shigley and Mischke, 2003):

$$T = F_t \cdot \frac{D}{2} = 20597.6 \times \frac{0.170}{2} = 1,759.3 \text{ Nm} = 1.76 \text{ kNm} \quad (11)$$

The crusher designed is shown in figure 1.

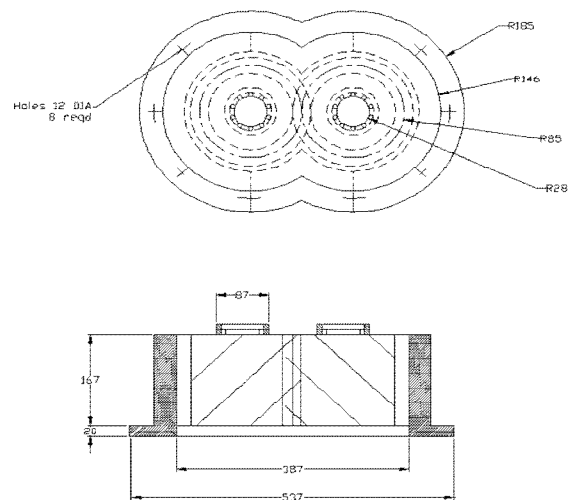


Figure 1: The crusher for the Jatropha Oil expelling machine

Design of Shaft Driving the Crusher

For the driven shaft, the length of the shaft is taken to be 310 mm and one end of the shaft is supported by a bearing while the other end hangs freely. The shafts are designed to be held to the bearings by outer rings for ease of maintenance. To design the shaft, assuming pure torsion and that for commercial steel shafting, torsional shear stress, $\tau = 55 \text{ MN/m}^2$ (Shigley and Mischke, 2003):

$$d = \left[\frac{16T}{\pi \tau} \right]^{1/3} = \left[\frac{16 \times 1759.3}{\pi \times 55 \times 10^6} \right]^{1/3} = 0.055 \text{ m} \quad (12)$$

A standard size of 56 mm is chosen. Taking length of shaft, L , as 310 mm; and modulus of rigidity in shear, G , as 84 GN/m^2 ; the torsional deflection of shaft

is (Shigley and Mischke, 2003; Sharma and Aggarwal, 2006):

$$\theta = \frac{TL}{G \pi d^4} = \frac{1759.3 \times 0.31 \times 32}{84 \times 10^9 \times \pi \times 0.056^4} = 0.0067 \text{ rad} = 0.39^\circ \quad (13)$$

Design of Key

Taking Torque, T, as 1759.3 Nm; allowable shear stress of key material, τ , as 55 MN/m^2 for commercial steel; diameter of shaft, d, as 0.056 m; length of key, l, approximately equal to face width of gear which is 0.15 m; width of key considering failure by shearing is (Singh, 2008):

$$W = \frac{2T}{d\tau l} = \frac{2 \times 1759.3}{0.056 \times 55 \times 10^6 \times 0.15} = 0.00675 \text{ m} \quad (14)$$

A standard key width of 8 mm is chosen. The standard corresponding height, t, of the key is 7mm.

Design of Crusher Housing

The housing for the crusher is a cuboid shaped component. It has two cylinders bored side by side inside it with the diameters being 4mm in excess of the addendum diameter of the crushing gears. The volume of bored cylinders is estimated as 0.0107 m^3 . The casing has a minimum thickness of 20mm. It also has an inlet point and outlet point.

Design of Screw Press

The screw press is the section of the machine which conveys the crushed seeds received from the crusher to the caking and nozzle region where pressing occurs and oil expelled. To achieve the required capacity of the machine, the minimum condition that must be satisfied is $Q \geq Q_R$.

To estimate the actual capacity of the machine, the selected parameters for the design of the screw conveyor are loading efficiency, ξ , assumed to be 0.25 for mildly abrasive material; bulk density of conveyed material, ρ , assumed to be 900 kg/m^3 ; factor of inclination to the horizontal, C, taken as 1 for 0° inclination to the horizontal; required capacity, Q_R , as 1,000 kg/h ($\cong 10,000 \text{ N/h}$); nominal size of screw, D = 100 mm; trough height from the centre of screw shaft to upper edge of the trough, a = 63 mm; width of the intermediate bearing shaft, b = 50 mm; trough width, c = 112 mm; solid shaft diameter, d_1 = 30 mm; tubular shaft diameter, d_2 = 33.7 mm; tubular shaft thickness, t = 2.5 mm; length of intermediate bearing, e = 40 mm; pitch of screw, s = 100 mm; and maximum speed of screw, n = 140 rpm. The capacity of the screw conveyor is given as (Singh, 2008):

$$Q = 150\pi D^2 s n \xi \rho C N/h = 14,844 \text{ N/h} = 1,484.4 \text{ kg/h} \quad (15)$$

The minimum speed of the screw in revolution per minute which will give the required capacity can be estimated using

$$n_{\min} = \frac{Q_R}{150\pi D^2 s \xi \rho C N/h} = 94.3 \text{ rpm} \quad (16)$$

Taking length of screw conveyor, L as 1 m (the thread is assumed to have 10 full forms); material factor, W_o , assumed to be 2.5; efficiency of the gear reducer, η , assumed to be 70%; and angle of inclination of screw shaft to horizontal, β , taken as 0° . The power required for driving the screw conveyor can be estimated using (Singh, 2008):

$$P = \frac{QL}{3600} [W_o \pm s \tan \beta] \cdot \frac{1}{\eta} \text{ kW} = 14.7 \text{ kW} \quad (17)$$

The load propulsion speed can be evaluated using (Singh, 2008):

$$v = \frac{\pi n}{60} \text{ m/s} = \frac{0.1 \times 140}{60} = 0.23 \text{ m/s} \quad (18)$$

The load per meter length of conveyor (Singh, 2008)

$$q = \frac{2}{36v} = 17,927.5 \text{ N/m} = 17.9 \text{ kN/m} \quad (19)$$

Where friction factor of material on trough, μ , is assumed to be 0.6, axial thrust on the screw is (Singh, 2008)

$$F_a = qL\mu N = 17,927.5 \times 1 \times 0.6 = 10,756.5 \text{ N} \quad (20)$$

The torque required to drive the conveyor is evaluated using

$$T = \frac{60P}{2\pi n} = \frac{60 \times 14.7 \times 10^3}{2\pi \times 140} = 1,002.7 \text{ Nm} \quad (21)$$

The length of the solid shaft which drives the conveyor is assumed to be 200mm. Therefore, the twist of the shaft is estimated using

$$\theta = \frac{TL}{GJ} = \frac{1002.7 \times 0.2 \times 32}{84 \times 10^9 \times \pi \times 0.03^4} = 0.03 \text{ rad} = 1.72^\circ \quad (22)$$

The sectional view of the screw press designed is shown in figures 2 and 3.

Design of Press Barrel

The screw conveyor press is surrounded by a barrel which is cylindrical. The clearance between the barrel and the screw conveyor is taken to be 2.5 mm. The length of the barrel, h, is approximately equal to the length of the screw conveyor, 1 m; and radius, r, of the barrel is 105 mm. Thus, the capacity or volume of the barrel is 0.035 m^3 .

The barrel carries the oil outlet sieve at a region close to the end of the screw. The region of the outlet is slightly recessed by 3 mm, by boring, to prevent the excessive back flow of oil from the caking or pressing region into the incoming feed. The overall mean thickness of the barrel is 10 mm. The barrel was also designed to be in two halves.

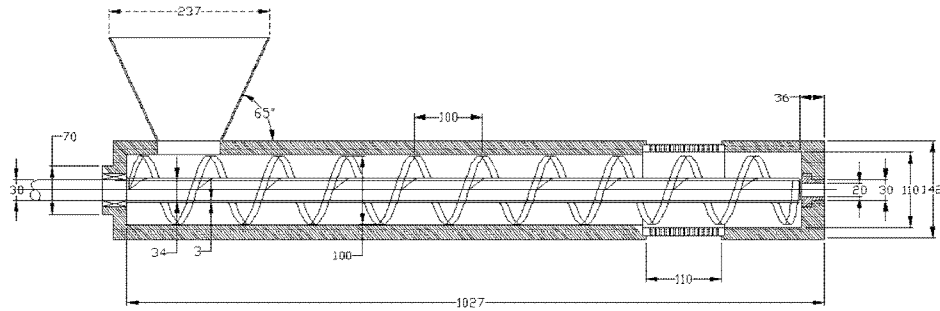


Figure 2: Sectional view of the screw press of the Jatropha oil expelling machine

Design of the Die or Constriction Nozzle

To achieve an efficiency of at least 97%, the oil recovery from the Jatropha seeds is given as

$$\text{Efficiency, } \Gamma_o = \frac{\text{Quantity of oil recovered}}{\text{Quantity of oil content}} \quad (23)$$

Oil content of Jatropha seeds in Nigeria is 33.7% (Beerens, 2007). Thus, oil content per hour for a capacity of 1000kg/h of seeds is estimated as

$$\text{Quantity of oil content} = 0.337 \times 1000 = 337 \text{ kg/h} \quad (24)$$

For an efficiency of 97%, the oil recovered is

$$\text{Quantity of oil recovered} = 0.97 \times 337 \text{ kg/h} = 326.9 \text{ kg/h} \quad (25)$$

The mass flow rate of the compressed cake through the constriction nozzle is determined as the difference between the mass flow through the screw and mass of oil required. Hence:

$$\text{Mass flow through nozzle} = 1000 - 326.9 = 673.1 \text{ kg/h} \quad (26)$$

The density of cake is assumed to be equal to the solid density of the seed which is 1040 kg/m^3 . Therefore, the volume flow of cake through the constriction nozzle is estimated as

$$\text{Volume flow} = \frac{\text{mass flow}}{\text{density}} = \frac{673.1}{1040} = 0.65 \text{ m}^3/\text{h} \quad (27)$$

Assuming velocity of cake through the constriction, v_c , is equal to the load propulsion speed of the screw = 0.23 m/s. The area of constriction, A_c , is

$$A_c = \frac{\text{Volume flow}}{v_c} = \frac{0.65}{0.23 \times 60 \times 60} = 7.85 \times 10^{-4} \text{ m}^2 \quad (28)$$

The maximum diameter of the constriction is estimated using

$$d_c = \left[\frac{4A_c}{\pi} \right]^{\frac{1}{2}} = \left[\frac{4 \times 7.85 \times 10^{-4}}{\pi} \right]^{\frac{1}{2}} = 0.022 \text{ m} \quad (29)$$

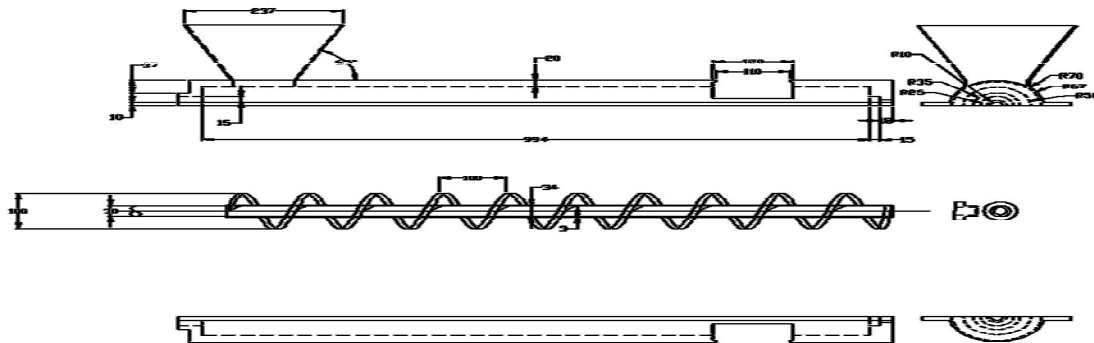


Figure 3: Assembly drawing of the screw press for the Jatropha oil expelling machine

Alternatively, using the continuity equation, $\rho_s A_s v_s = \rho_c A_c v_c$; and taking density of crushed seed, ρ_s , as 900 kg/m^3 ; load propulsion speed of conveyor, v_s , as 0.23 m/s; density of pressed cake, ρ_c , assumed to be equal to the solid density of the seed (1040 kg/m^3); area of the screw face, A_s , is estimated as $7.85 \times 10^{-3} \text{ m}^2$. Therefore, $A_c = 3.59 \times 10^{-4} \text{ m}^2$.

Hence, the constriction nozzle diameter is $d_c = 21 \text{ mm}$. A nozzle diameter of less than 21mm is preferred to allow for maximum oil extraction. A nozzle diameter of 20mm is selected.

The pressure required to overcome the constriction at the nozzle which lies at the end of the screw is estimated using

$$P = \frac{F_c}{A_c} = \frac{10756.5 \times 4}{\pi \times 0.02^2} \text{ N/m}^2 = 34.34 \text{ MN/m}^2 \quad (30)$$

Beerens (2007) noted that the end pressure of a screw press varies from 4 to 35 MPa.

Design of Hopper

Hopper design is based on the criterion of the angle of repose which is the maximum slope at which a heap of any loose or fragmented bulk material will stand without sliding (Chelecha, 2003). It is also the angle of friction of rest (Eugene and Theodore, 1996). The hopper uses the gravity discharge mechanism with the recommended angle for agricultural materials being 8° to 10° higher than their angle of repose (Chelecha, 2003). The average angle of repose, γ , of the *Jatropha* seed is taken to be 35° . Assuming radius of upper opening, r_o is 210 mm; radius of lower opening, r_i , is 120 mm; vertical height of the hopper, h , is 300 mm, the capacity of the hopper is estimated $9.3 \times 10^{-3} \text{ m}^3$

Design of Dosing Screw

The dosing screw is a conveyor which is required to meter the seeds from the hopper into the crusher ensuring that the system is not overloaded, either at the crusher or at the screw conveyor pressing the crushed seeds. It has been designed to have the same capacity as the screw conveyor which presses the crushed seed. The parameters considered for the dosing screw are similar to that of the screw conveyor in the press section. However, the length is shortened to reduce the power required to drive the screw. The capacity, Q_R , is 1000 kg/h; minimum speed n_{min} is 94.3 rpm.

Taking length of screw conveyor, L , as 200 mm (the thread is assumed to have 2 full forms); material factor, W_o , assumed to be 2.5; efficiency of the gear reducer, η , assumed to be 70%; angle of inclination of screw shaft to horizontal, β , as 0° ; friction factor of material on trough, μ , assumed to be 0.6. Therefore, the power required for driving the screw conveyor is 1.98 kW. The load propulsion speed is evaluated as 0.16 m/s. The load per meter length of conveyor is 17.4 kNm. The axial thrust on the screw is $F_a = 2083 \text{ N}$. The torque required to drive the conveyor is $T = 20.05 \text{ N}$.

The dosing screw is driven by a pulley on the shaft driving the crusher. The speed ratio of the pulleys is taken to be 1 since the speed of the feed screw must be greater than 94.3 rpm and less than 140 rpm. Thus the speed is taken to be 120 rpm. A standard pulley of 80 mm is selected. The belt required for the drive is a Type A belt. Where f is taken from tables and has a value of 3.5 for Type A belts, the diameters of the pulleys are $D_{out} = 87 \text{ mm}$ and $D_{in} = 73 \text{ mm}$.

Oil Outlet or Sieves

The quality of separation (Faluyi, 2001) is the criterion used in the design of the outlets. The oil

outlet is a small part located at the bottom of the barrel housing the screw conveyor press and is close to the caking section of the press. The diameter of each perforation is assumed to be 1.50 mm. The concave sieve is made from a rectangular shaped metal sheet. Where D = maximum diameter of the required holes assumed to be 1.5 mm; C_o = coefficient of opening assumed to be 2.5 mm (for oil), the distance, d , between two successive holes of the punched sieve having holes of circular shape is given by Faluyi (2001) as:

$$d = \left[\frac{D(3\pi - 2C_o)}{C_o} \right]^{1/2} = \left[\frac{1.5(3\pi - 2 \times 2.5)}{2.5} \right]^{1/2} = 1.63 \quad (31)$$

So, the standard of 2 mm is used for the second or lower sieve for further purification of the *Jatropha* oil. For the upper or first sieve, assuming $D = 2.5 \text{ mm}$ and $C_o = 2.5 \text{ mm}$ (for oil)

$$d = \left[\frac{D(3\pi - 2C_o)}{C_o} \right]^{1/2} = \left[\frac{2.5(3\pi - 2 \times 2.5)}{2.5} \right]^{1/2} = 2.1 \text{ mm} \quad (32)$$

Also, the standard of 2 mm is used for the first (upper) sieve for the purification or filtration of the crude *Jatropha* oil. Area, A_x of the sieve is given by:

$$A_x = l_x \times b_x \quad (33)$$

Where l_x = arc length of the concave sieve; b_x = breadth/width of the concave sieve. If R is the radius of concave, α = perpendicular angle of the lengths of the sieve. Therefore, using θ as 90° and R as 40 mm, then

$$l_x = \frac{\alpha}{360} \times 2\pi R = \frac{90}{360} \times 2\pi \times 40 = 62.8 \text{ mm} \approx 60 \text{ mm} \quad (34)$$

Letting b_x , 120 mm, then

$$A_x = l_x \times b_x = 60 \times 120 = 7200 \text{ mm}^2 \quad (35)$$

Volume Flow Rate of *Jatropha* Oil and Capacity of Oil Collector

The mass of oil recovered has been estimated as 326.9 kg/h. The density of *Jatropha* oil is 918 kg/m³ (Agarwal and Agarwal, 2007). The volume rate of oil recovered is $0.36 \text{ m}^3/\text{h}$. The capacity of the oil collector should be greater than the volume rate of the oil. For a day's job assumed to be 8 hours, the volume of oil collected is 2.88 m³, equivalent of 2880 litres per day work.

Electric Motor Selection

The electric motor for the *Jatropha* oil expeller selected based on the load characteristics of the machine is a three-phase 50Hz motor for industrial purpose. The power of the electric motor is the selected power output of the motor using gear box which enhances speed variation and is equal to the power required for conveyor and crusher coupled with pressing multiplied by the service factor that cares for the power loss during power transmission.

If P_m = maximum motor power; P_{cl} = power required to drive the conveyor = 14.7 kW; P_{c2} = power required to drive the crusher = 22.1 kW; and with a

selected correction factor, S, of about 1.1, the total power required is (Ismail, 2005; PSG, 1992)

$$P_m = (P_{c1} + P_{c2})S = (14.7 + 22.1)S = 40 \text{ kW} \quad (36)$$

An electric motor with a power rating of 40 kW was selected. In selecting the motor, it was assumed that the electric motor operates continuously for at least 8 hours.

Design of Transmission System

The transmission system was basically the use of V-belt drives because it can easily be installed and removed, and the possibility of powering a number of driven shafts from a single drive without the use of belt tightness (Sharma and Aggarwal, 2006).

Transmission between the Motor and Screw Conveyor

Design power is obtained by multiplying the power to be transmitted by the service factor. For screw conveyors, the service factor is from 1.0 to 1.5 (Sharma and Aggarwal, 2006). A service factor of 1.5 is selected.

$$\text{Design power} = 1.5 \times 14.7 = 22.05 \text{ kW} \quad (37)$$

A Type C belt corresponds to the power to be transmitted and has a minimum recommended diameter of 229 mm and minimum cross sectional area of 2.3 cm^2 (Sharma and Aggarwal, 2006). The speed of the electric motor is assumed to be 850 rpm. Where N_m is the speed of electric motor = 850 rpm; N_s is the speed of the screw conveyor = 140 rpm (maximum); Hence, the speed ratio of the drive between motor and screw conveyor is (Sharma and Aggarwal, 2006):

$$VR = \frac{N_m}{N_s} = \frac{850}{140} = 6 \quad (38)$$

The slip factor, s, has been selected to have a value of 0.03, therefore for a maximum belt speed, v, of 5m/s,

$$D_m = \frac{60v}{\pi N_m} = 0.112 \text{ m} = 112.3 \text{ mm} \quad (39)$$

A standard of 112 mm is selected. Therefore,

$$D_s = D_m(1-s) \cdot VR = 112(1-0.03) \cdot 6 = 651.8 \text{ mm} \quad (40)$$

A standard of 630 mm is selected. The centre distance between the pulleys can be estimated using (Sharma and Aggarwal, 2006):

$$C \leq 3(D_m + D_s); \quad C \leq 2226 \text{ mm}; \quad \text{Also,} \\ C \geq D_s; \quad C \geq 630 \text{ mm} \quad (41)$$

The centre distance is between 630 mm and 2226 mm. A centre distance of 1000 mm is selected. The pitch length of the belt is estimated using (Sharma and Aggarwal, 2006; Shigley and Mischke, 2003)

$$L = 2C + 1.57(D_s + D_m) + \frac{(D_s - D_m)^2}{4C} = 3232 \text{ mm} \quad (42)$$

A standard pitch length of 3205 mm is chosen. The angle of contact of the smaller pulley is estimated using (Sharma and Aggarwal, 2006):

$$\theta = \pi - 2 \sin^{-1} \frac{D_s - D_m}{2C}; \quad \theta \geq 2.1 \text{ rad}; \quad \theta = 2.61 \text{ rad} \quad (43)$$

The maximum belt tension is estimated from (Sharma and Aggarwal, 2006)

$$\text{Design power} = \frac{(F_2 - F_1)v}{1000} \text{ kW} \quad (44)$$

Taking μ is the coefficient of friction of belt on the pulley surface taken as 0.3 and β is the angle of groove taken as 40° . Hence,

$$\mu_e = \frac{\mu}{\sin \frac{\beta}{2}} = \frac{0.3}{\sin \frac{40^\circ}{2}} = 0.88 \quad (45)$$

Therefore

$$\frac{F_2}{F_1} = e^{\mu_e \theta} = 10; \quad F_1 = 10F_2 \quad (46)$$

The design power is thus

$$\text{Design power} = \frac{(10F_2 - F_2)v}{1000} = \frac{(9F_2)v}{1000} \text{ kW}; \quad (47)$$

$$F_2 = \frac{22050}{10 \times 11.2} = 198.6 \text{ N}; \quad F_1 = 1986.5 \text{ N} \quad (48)$$

The required number of belts is estimated using

$$n = \frac{F_1}{(ft - \rho v^2)AC_1C_2} \quad (49)$$

Where the following defined have values selected from tables

$$ft = \frac{1}{6} \text{ of } 10.5 \text{ MN/m}^2 = 1.75 \text{ MN/m}^2 \quad (50)$$

A = cross sectional area of belt profile chosen = $2.3 \text{ cm}^2 = 2.3 \times 10^{-4} \text{ m}^2$; C_1 = Arc of contact factor on the smaller pulley = 0.87; C_2 = Belt length correction factor = 1.11; ρ = density of belt material = 1400 kg/m^3 for rubber. Hence, $n = 5.2$.

Therefore, number of belts is taken as 5.

For the selected Type C belts, the diameters of the motor pulley are $D_{out} = 124 \text{ mm}$ and $D_{in} = 100 \text{ mm}$. The diameters of the pulley on the screw conveyor are $D_{out} = 642 \text{ mm}$ and $D_{in} = 618 \text{ mm}$. To design the shaft that should carry the pulley, the torque on the pulley is estimated as

$$T = \frac{(F_2 - F_1)D_s}{2} = \frac{(1986.5 - 198.6)0.63}{2} = 563.2 \text{ Nm} \quad (51)$$

Where r is the radius of the shaft and J is the section modulus of the shaft, and the design shear stress is taken as 55 MN/m^2 . Therefore, from

$$d = \left[\frac{16T}{\pi \tau} \right]^{1/3} = \left[\frac{16 \times 563.2}{\pi \times 55 \times 10^6} \right]^{1/3} = 0.037 \text{ m} = 37 \text{ mm} \quad (52)$$

A diameter of 40mm is selected.

Transmission to the Crusher

Design power is obtained by multiplying the power to be transmitted by the service factor. For mills and crushers, the service factor is from 1.4 to 1.6 (Sharma and Aggarwal, 2006). A service factor of 1.5 is selected. The design power is **33.15 kW**. From standards, a belt Type C corresponds to the power to be transmitted and has a minimum recommended diameter of 229 mm and minimum cross sectional

area of 2.3 cm^2 (Sharma and Aggarwal, 2006). The crusher is driven by a pulley on the shaft of the screw conveyor. Where N_c is the speed of crusher = 120 rpm; N_s is the speed of the screw conveyor = 140 rpm (maximum). Hence, the speed ratio of the drive between screw conveyor and crusher is 1.16. The pitch diameter of the smaller pulley, D_s , on the screw conveyor shaft is assumed to have a selected standard value of 250 mm. The slip factor, s , is selected to have a value of 0.03 and the diameter of the larger pulley on the crusher shaft is determined as $D_c = 281.3 \text{ mm}$.

A standard pulley diameter of 280 mm is selected. The estimated belt speed is 1.83 m/s . The centre distance is between 280 mm and 1590 mm. A centre distance of 500 mm is selected. The pitch length of the belt is estimated as 1833 mm but a standard pitch length of 1948 mm is chosen. The angle of contact of the smaller pulley is estimated using (Sharma and Aggarwal, 2006):

$$\theta = \pi - 2 \sin^{-1} \frac{D_c - D_s}{2C}; \quad \theta \geq 2.1 \text{ rad}; \quad \theta = 3.05 \text{ rad} \quad (53)$$

Where μ is the coefficient of friction of belt on the pulley surface taken as 0.3 and β is the angle of groove taken as 40° . Hence, $\mu_s = 0.00$; therefore $F_1 = 14.6F_2$. Therefore, belt tensions are $F_1 = 1835.7 \text{ N}$ and $F_2 = 244.8 \text{ N}$.

The required number of belts is estimated taking cross sectional area, A , of belt profile chosen as 2.3 cm^2 ; arc of contact factor on the smaller pulley, C_1 , as 0.87; Belt length correction factor, C_2 , as 1.11; density of belt material, ρ , = 1400 kg/m^3 for rubber.

Hence, number of belts is $n = 5.2$ and number of belts is taken as 5. For the Type C belts, the diameters of the motor pulley are $D_{out} = 262 \text{ mm}$ and $D_{in} = 238 \text{ mm}$. The diameters of the pulley on the crusher are $D_{out} = 292 \text{ mm}$ and $D_{in} = 268 \text{ mm}$.

Design of the Heating System

Heating systems are required at two levels, each performing different functions. A heating system is required for cooking the seeds before feeding into the crusher at a temperature that will not burn the seeds. It is required so that cell wall rupturing is enhanced and thereby facilitates the outflow of oil. A second heating system is provided at the area around the nozzle to decrease the viscosity of the paste inside the press and to reduce the solid content in the oil (Ferchau, 2000; Beerens, 2007). A suitable temperature in the range of 60 to 80°C has been proposed by Beerens (2007). The heating coils for both systems are connected to a thermostat for the

purpose of regulating the temperature within the specified range. An electric cable heating system is proposed.

Operation of the Jatropha Oil Expelling Machine

The Jatropha expelling machine has a hopper which is fed with preheated Jatropha seeds. The dosing screw feeds seeds to the crusher to prevent overloading of the crusher. Seeds passing through the crusher are crushed and some of the oil released. The mixture enters into the screw press chamber where the press moves the crushed seeds towards the die where pressing takes place. The expelled oil is collected at the oil outlet nozzles and the pressed cake is collected at the die opening. The working drawings of the machine are presented in figures 4 and 5. The capacity of the machine designed is such that it can process 1000kg of Jatropha seeds per hour to yield 2880 litres of oil per 8-hours day work. The designs of the crusher and heating systems before pressing will enhance the performance of the expeller and put the expected efficiency of the designed oil expelling machine which is improved by the addition of a crusher is 97% which should surpass the 60% efficiency of current oil expellers.

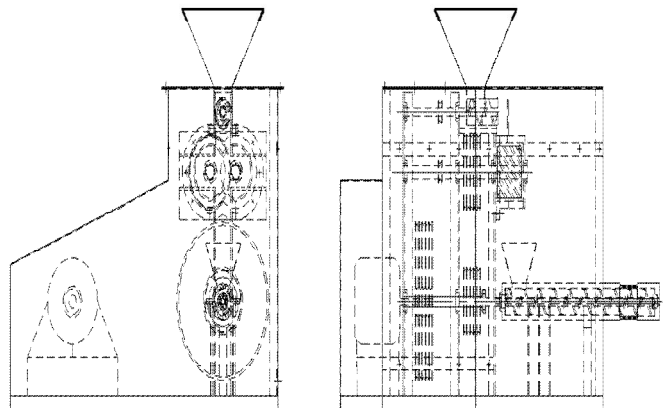


Figure 4: Assembly drawing of the Jatropha oil expelling machine

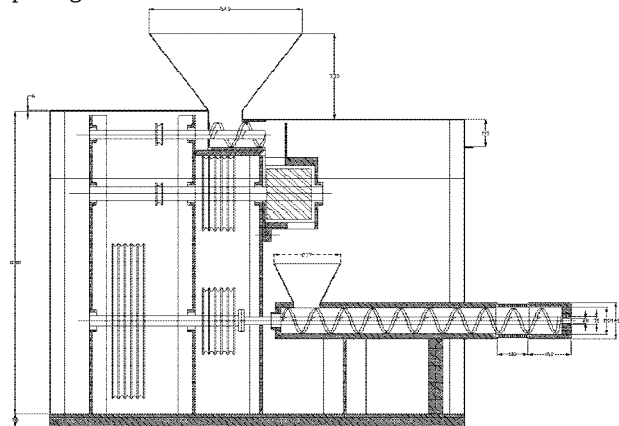


Figure 5: Sectioned view of the Jatropha oil expelling machine

CONCLUSION

A Jatropha oil expelling machine was designed. This machine is conceived as ideal, easy to maintain and economic for commercial uses. The expected capacity of the expelling machine is 1000kg/hr. The designs of the crusher and pre-heating systems before pressing will enhance the performance of the expeller and the machine is capable of delivering greater than 90% efficiency. The machine will expel oil at the rate of 2.2m³/hr. The machine is powered using electric motor of 40kW to be operated for 8 working hours. This expelling machine is conceived as an improvement over existing machine and presupposes appropriate technology for manufacture, operation and maintenance. The machine has not been manufactured but can be fabricated locally at a well equipped machine shop.

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