International Journal of Engineering & Technology
CENTRE OF PROFESSIONAL RESEARCH PUBLICATIONS
ISSN 2049-3444
International Journal of Engineering and Technology Volume 5 No. 2, February, 2015

Development of Palm Oil Extraction System

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ABSTRACT

This is an improved design of a palm oil extraction system with particular reference to Nigerian experience. Existing design conditions were carefully examined and necessary improvements made to enhance oil extraction efficiency. A detailed design of the oil extraction system was therefore done. The screw press design configurations, the effects of lagging, the casing of the press chamber, as well as optimal heating requirement of the system were carefully examined. A vertical oil palm digester design was made to feed the screw press by gravity. Detailed design of the screw press hereby presented. Much emphasis was however made in the use of available local construction materials. Performance test shows that the oil extraction efficiency of the screw press at an optimal temperature of 98ºC is 95.7 percent.

Keywords: *Palm oil extraction, improved design, optimal temperature, improved efficiency, lagging.*

1. INTRODUCTION

The oil palm tree is a tropical plant commonly found in warm climates at altitudes of less than 1600 feet above sea level (www.ask.com). 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. The tree produces one of the most popular edible oil of high nutritional value. Palm oil is rich in carotenoids (a pigment found in plants and animals) from which it derives its red colour and its major components of glycerides – a saturated fatty acid (palmitic acid). Hence palm oil is a viscous semi-solid substance even at tropical ambient and a solid fat in temperate climates (Poko, 1993). Because of its economic importance as a high yielding source of edible and technical oils, oil palm is now grown as a plantation crop in most tropical countries with high rainfall (minimum of 1600 mm per year). The palm bears its fruits in bunches varying in weight between 10 ~ 40 kilograms. The individual fruit (Fig.1.1) of weight between 6 ~ 20 grams is made of an outer skin (exocarp), a pulp (mesocarp) which entraps the palm oil in a fibrous matrix, a central nut consisting of a shell (endocarp) and the kernel-which itself contains an oil quite different from palm oil but resembling coconut oil (Poko, 1998). A typical composition of a Fresh Fruit Bunch (FFB) is as shown in Fig. 1.2.

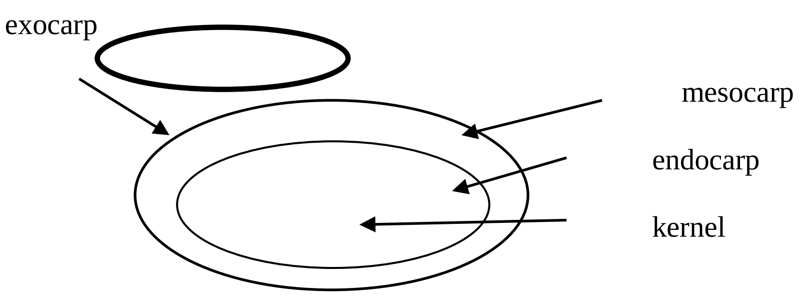

Fig. 1.1 Structure of Palm Fruit

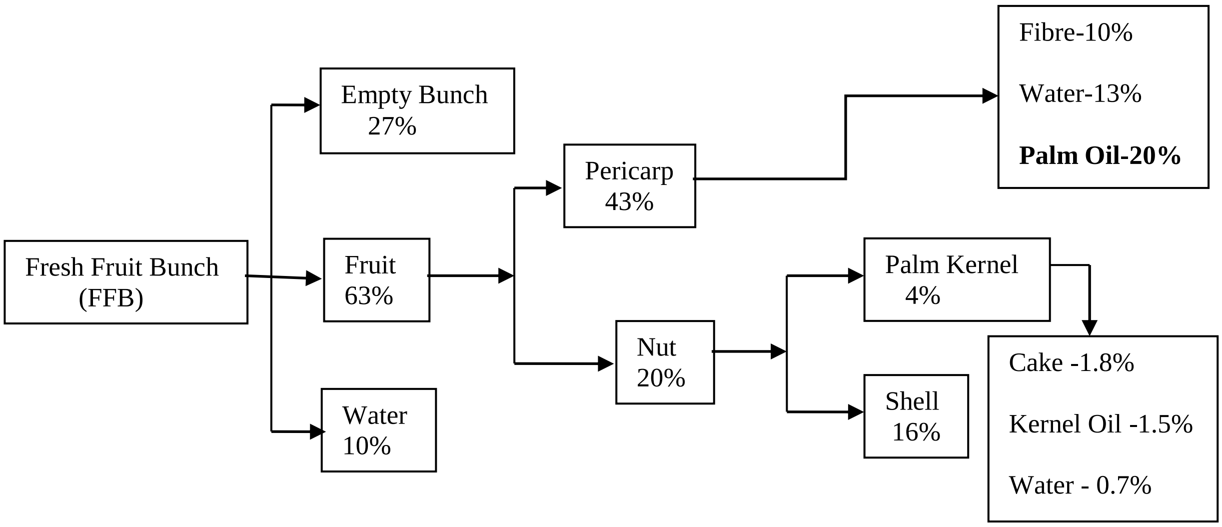

Fig. 1.2 Composition of Fresh Fruit Bunch

Conversion of crude palm oil to refined oil will involve the removal of products of hydrolysis and oxidation, colour and flavor. After refining, the oil is separated (fractionated) into liquids and solid phases by thermo-mechanical means (involving controlled cooling, crystallization and filtering), the liquid fraction (olein) being used extensively in the tropics as cooking oil. Palm oil extractors of all kinds incorporate these unit operational stages, differing only in the level of mechanisms that make the system batch or continuous. The primary interest however is how best to extract oil from the fruits by use of an integrated design of a fruit digester and screw press.

The oil content of the palm fruit is about 30 per cent by weight of the fresh fruit bunch (Hartley, 1977). Under normal conditions, oil extracted from palm fruit is light red. In addition to its primary use for edible purposes, it also serves as a major raw material in the production of margarine and soap. This makes its extraction a major concern for large scale production, considering oil loss arising from use of inefficient extraction methods. In order to ensure maximum oil extraction, it becomes necessary to consider as follows (Poko, 2002).

i. The best maturity time to cut the palm fruit bunch from the tree.
ii. The optimum period to keep the fresh (harvested) palm fruit bunch before processing.
iii. The optimum oil extraction temperature
iv. The effect of heat on the screw press surface and structure; and hence appropriate material selection.
v. Appropriate lagging of screw press chamber to minimize heat loss.
vi. Appropriate power selection to ensure optimum performance.
vii. Should the digester be incorporated to the screw press, considering that heat will be lost while transferring the digested palm fruits from the digester to the screw press? Also extra cost will be incurred. So will the integrated design actually guarantee higher efficiency?

The traditional design is simple, but the process involved is tedious and very inefficient (Poko, 2002). There are many palm oil extraction methods, but the question is; which among them is most efficient and economical? According to Cornelius (South, 1983), the use of screw press gives an oil extraction efficiency level as high as 95 per cent. However, a critical examination of the local press system revealed the following problems;

1. High viscosity of the extracted oil caused by low operating temperature and subsequent loss of heat during pressing operation.
2. Loss of heat while transferring digested palm fruits from the digester to the press chamber.
3. Increase in production cost owing to extra labour required to transfer digested palm fruits from the digester to the press chamber.
4. Inefficient extraction of entrapped oil.

2. CONCEPT DESIGN

Figure 2.1 shows the assembly view of the palm oil screw press, and Fig. 2.2 shows the orthographic views.

2.1. Description

The screw press consists of a screw in a cylindrical drainage cage with a main worm shaft which carries a worm assembly and which acts as palm oil expellers. Digested mash is introduced into the press screw via an inlet hopper. The press assembly has a provision through which hot water is sprinkled to the mash. This helps to keep the system at desired optimal temperature (can be as high as 98ºC). The higher the operating temperature, the lower the oil loss

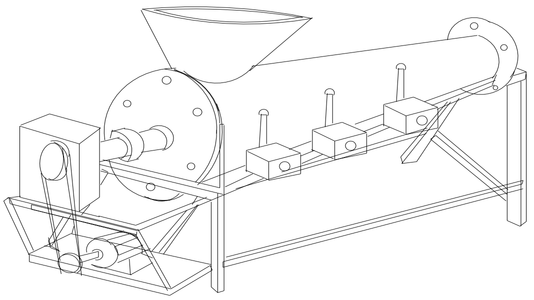

Fig. 2.1 Assembly View of the Screw Press

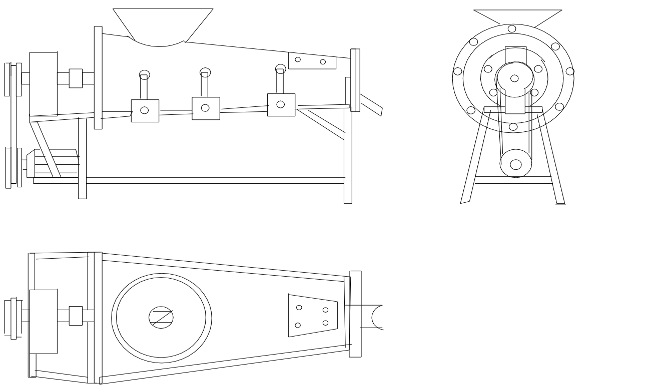

Fig.2.2 Orthographic View of the Screw Press

and the easier to extract the oil from the mash (South Worth, 1983). This is as a result of reduction in oil viscosity with increase in temperature. The drainage cage is lagged to prevent heat loss. The design incorporates a gear box and a variable speed pulley assembly for easy control of the operating speed. The screw press is powered through a reduction gear box. Power is transmitted from the electric motor through spur gears to an auxiliary shaft. The roller conveyor taps power from the motor through a crossed belt. The screw power shaft works both as a conveyor and as a power element. The screw is welded to the shaft with incrementally decreasing pitches. The press receives digested palm fruits from the digester via an inlet hopper and gradually moves it forward by aid of the screw threads. The decreasing screw pitch and screw height provide the compressing force needed to squeeze out the entrapped oil. Spur gears were used in the design of the reduction gear box. In general, screw presses are relatively simple in operation and much less costly to install than the solvent extraction systems. The screw method is reported to give oil extraction efficiency as high as 98 per cent (Hartley, 1977).

3. DESIGN ANALYSIS

3.1 Power Requirement Considerations

In the screw press design there are basically four areas where power is needed; these are:

i. Power required to overcome the inertia of the shaft and screw,
ii. Power required to convey digested palm fruits along the entire length of the press,
iii. Power required to effectively press and squeeze out entrapped oil from the digested palm fruits, and
iv. Power compensation for friction and other losses during operation.

3.1.1 Inertia of Drive Shaft

The shaft is cylindrical and hollow; thus volume, $V = \pi r^2 l$

Internal radius, r_i = 24mm; Outside radius, r_o = 30mm; Length, l = 1400mm

$V = 3.142\ (0.0009 – 0.0006) \times 0.14 = 1.32 \times 10^{-4}\ m^3$

Mass of the steel pipe, $M = 7850\ (kg/m^3) \times 1.32 \times 10^{-4}\ (m^3)$ = 1.04 kg

The press screw is made up of ten gradually decreasing flights cut and drawn to shape with a thickness of 4.9mm. The diameter of the largest flight is 270mm which progressively decreases by 9mm.

But the flights are connected by a 60mm hollow drive shaft, Thus net volume of flight = volume of flight – Volume of hollow shaft

Flight 1; $V_1 = \pi\ (0.00028) – 0.000014 = 0.0000266m^3$

Flight 2; $V_2 = \pi\ (0.0000834) – 0.000014 = 0.000248m^3$

Flight 3; $V_3 = \pi\ (0.0000778) – 0.000014 = 0.0023m^3$

Similarly, $V_4 = 0.00213m^3$; $V_5 = 0.000197m^3$; $V_6 = 0.00018m^3$; $V_7 = 0.000166m^3$; $V_8 = 0.000151m^3$

$V_9 = 0.000137m^3$; and $V_{10} = 0.000123m^3$

Total Volume, $V_T = 0.00191m^3$

Total mass of Flight, $M_T = 0.00191 \times 7850 = 15.001kg$

Mass of Flight + Mass of Shaft = 15.001 + 1.036 = 16.16kg

Power to overcome this mass = weight x mean peripheral velocity, v_m

But $v_m = T_mN/60$ (Spivakousky and Dyachkou, 1983)

Where T_m = mean lead of screw; N = RPM of the screw

$v_m = (0.111 \times 30) / 60 = 0.056$ m/s

Thus power required, $\mathbf{P_1}$ = (16.16 x 9.81) x 0.056 = **0.00887 kW**

3.1.2 Power Required to Convey Digested Palm Fruits

We first determine the throughput capacity of the press screws. The **Throughput Capacity** is given by,

$Q = (60\ \pi\ d^2tn\psi\rho c) / 4 = 47d^2tn\psi\rho c$ in t/hr (Spirakovski and Dyachkov, 1983)

Where d = screw diameter (225mm), t = screw lead (110mm), n = rotational speed (30rpm), c = correction factor based on the angle of inclination (0.8), ψ = filling coefficient of the screw cross section (0.125 for high abrasive materials), and ρ = density of digested palm fruit ($480kg/m^3$).

Thus, $Q = 47 \times (0.225)^2 \times 0.110 \times 30 \times 0.125 \times 480 \times 0.8 = 380.32kg/hr$

Power required to convey digested palm fruits,

$P_2 = QL\rho F_m / 168547$

Where Q is the conveying capacity; L is the projected length of the screw conveyor (1.2m), and F_m is the material factor per kW.

$\mathbf{P_2}$ = (380.32 x 1.2 x x 480 x 0.8) / 168547 = **1.4 kW**

3.1.3 Power Required to Press the Digested Palm Fruit

Experiment shows that it requires approximately 322kPa to crush and press out oil from palm kernel. Since high oil recovery is desired, a compression pressure slightly less than the crushing strength of the kernel is desired. During operation, the cage and the activities of its accessories (flights and the cone) add to the pressure needed to squeeze out the oil. Thus, let compression pressure required = 200kPa

But power required to press oil, $P = FV_m$

Where F = Force required to squeeze out the oil; V_m = Mean peripheral velocity of drive shaft

But F = Squeezing pressure x Total surface area in contact with the palm fruits

$V_m = t_mn/60$; where t_m = mean pitch of the screw

Vm = (0.111 x 30) / 60 = 0.056 m/s

Areas in contact with palm fruits are as shown in figure 3.1.

Curved surface area of drive shaft = $2\pi rl = 2 \times 3.142 \times 0.03 \times 1.2 = 0.226\ m^2$

Considering the flights an approximate circle joined by the shaft;

Area of first flight = $\pi\ (0.1350)^2 – \pi\ (0.03)^2 = 0.0545m^2$

Area of 2nd flight = $\pi\ (0.1305)^2 – \pi\ (0.03)^2 = 0.0507m^2$

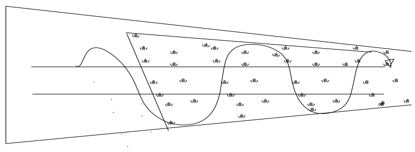

Fig.3.1 Areas in contact with palm fruits

Area of 3rd flight = $\pi\ (0.12605)^2 – \pi\ (0.03)^2 = 0.0471m^2$ and so on up to the 10th flight.

Total area of flights = sum total area of the ten flights

$= 0.391\ m^2$

We however observe that the effective area of the flights is equal to half of the total area therein;

Thus, Effective Total Area of Flights = 0.391 / 2 = 0.1955 m^2

For the curved surface area of the cone (Fig.3.2);

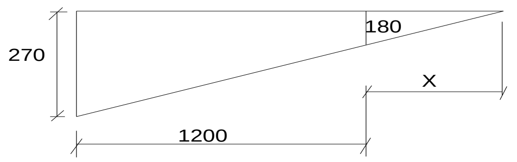

Fig.3.2 Curved Surface Area of the Cone

(X + 1200) / 270 = X / 180 (similar triangles)

X = 2400mm

Area of curved surface of the conical part of the cage = 142[(0.135 x 3.6) – (0.09 x 2.4)] = 0.848m^2

Effective area = 0.848 x 0.5 = 0.424m^2

Surface area of the end part = πr^2 = 3.142 $(0.09)^2$ = 0.026m^2

Both areas become 0.424 + 0.026 = 0.450m^2

Total Contacting Areas = (0.226 + 0.196 + 0.450)m^2 = 0.872m^2

Total Force Required = Pressure x Total Area = 200 (kPa) x 0.872 (m^2) = 174.40kN

Hence, Power required to press out palm oil, $\mathbf{P_3}$ = FV_m = 174.40 x 0.056 = **9.76 kW**

3.1.4 Power Required to Overcome Friction

Frictional Force, F = μN = 0.035 (174.40 x cos 30) = 5.2862 kN

Frictional Power, P_4 = 5.2862 x 0.056 = 0.296 kW

Overall Power Required, **P** = **0.00887 + 1.4 + 9.76 + 0.296 = 11.46 kW**

3.2 Selection of Electric Motor for the Screw Press

Power Required = 11.46 kW

Considering 10 % addition for compensation;

Power Required, P = 11.46 + 1.146 = **12.60 kW** = 12.60/0.746 = **16.8 hp**

Use 17 horse power electric motor

3.3 Design of the Drive Shaft

Forces acting on the shaft include:

i. Weight of digested palm fruits (distributed force)
ii. Weight of flights (also considered a distributed force)
iii. Force acting on the gear (concentrated force)
iv. Reactions at the bearings

Figures 3.3 and 3.4 show the load and free body diagrams respectively. In Fig. 3.4 the distributed forces arising from weights of digested fruits and flights are treated as concentrated forces. This helps to add to factor of safety of the design.

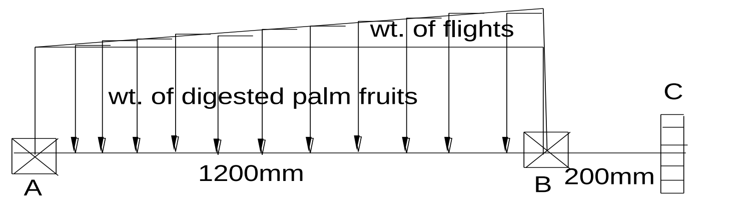

Fig. 3.3 Press Screw Shaft Loading

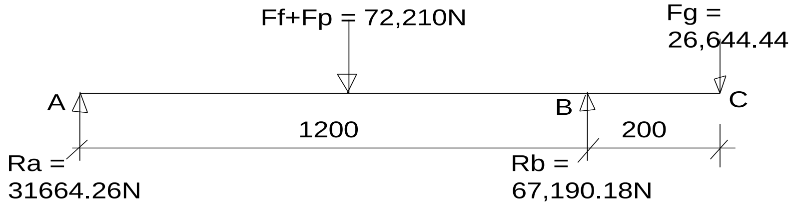

Fig.3.4 Free Body Diagram of Shaft Loading

Force due to weight of digested palm fruits, $F_p = 9.81(QL/V)$

= [9.81 (380.32/60 x 60) x (1.2/0.056)] = 22.21kN

Force due to weight of flights, F_f = 15.001 x 9.81 = 150kN

But since it is a distributed force, we choose one third of this value which is still very safe. Thus, F_f = 50kN

Thus, $F_p + F_f$ = 22.21 + 75 =72.21kN = **72,210 N** is considered a concentrated force acting at the centre.

Gear force, $F_g = T/r$ where T is the torque generated by the gear wheel, 2398 N-m

F_g = 2398/0.18 x 0.5 = **26,644.44 N**

Taking moment about A; (26644.44 x 1.4) + (72,210 x 0.6) – $1.2R_B = 0$

R_B = (26644.44 x 1.4) + (72,210 x 0.6)/1.2 = **67,190.18 N-m**

Also, $R_B + R_A$ = 72,210 + 26644.44; **R_A** =98854.44 – 67,190.18 = **31664.26 N**

Obviously, the maximum bending moment,(M_b) = 72,210 x 0.6 = **43326 N-m**

$d^3 = \{16/\pi S_a \sqrt{[(k_bM_b)^2 + (k_tM_t)^2]}$ (Hall, 1980)

where d = shaft diameter; S_a = allowable stress; Torsional moment, M_t = (9550 x kW)/RPM

K_b and K_t are shock correction factors for bending and torsional moments respectively, $k_b = k_t = 1$

M_t = (9550 x 12.60)/950 = 126.663N-m

$d^3 = \{(16/3.142 \times 120 \times 10^6)\sqrt{[(43326)^2 + (126.663)^2]}$

$d^3 = 0.001838$; $d = (0.001838)^{1/3} = 0.120$m

3.4 Gear Design for the Screw Press

Figure 3.5 shows the power transmission network of the screw press. Considering 20% power transmission losses,

Design Power, P = 12.6kW + 0.2 (12.6) = **15.20kW**

Speed of electric motor = 950rpm; motor sheave diameter = 180mm

Input shaft speed of gear box = (62/180) x 950 = 327.22rpm

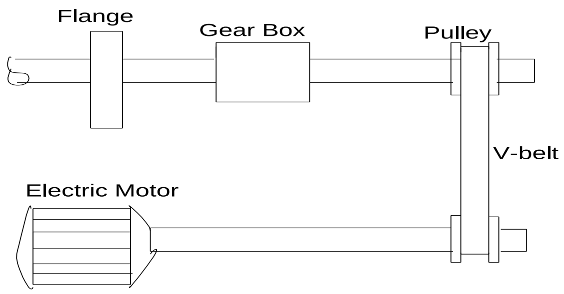

Fig. 3.5 Power Transmission

For the first two gears;

Number of teeth of pinion, N_1 = 18; Number of teeth of driven gear, N_2 = 72

Power = Fv where v is the pitch line velocity of the pinion

Whole depth of teeth = 2.157 x 4.5 = 9.7mm (where modulus, m = 4.5)

Minimum deddendum = 9.7 – 5.2 = 4.5mm

Pitch circle, PC = π x 4.5 = 14.137mm

Thus pc = π/14.137 = 0.222

Pitch diameter of pinion, $d_p = N_1 / 2$ = 18/0.22 = 81mm

Pitch diameter of gear, $d_g = N_2 / 2$ = 72/0.222 = 324mm

Centre to centre distance of the mating gears, $d_g/2 + d_p/2$ = 324/2 + 81/2 = 202.5mm

Pitch line velocity of pinion, $v_p = \pi d_p n/60$ where speed of pinion gear = Input shaft speed of gear box = 327.22rpm

v_p = (3.142 x 0.081x 327.22)/60 = 1.388 m/s

Force acting on pinion, F_p = power / v_p = 15.20 /1.388 = 10.90 kN

Induced stress on the pinion, s = S_o (3/ (3 x v_p) for v < 10 m/s where S_o is the allowable stress

Induced Stress, s = 800 [3/(3 x 1.3888)] = 576MPa

Since face width should not exceed $4p_c$ (Shigley and Chuke, 1989);

If b is the face width for pinion, then b = $4P_c$ = 4 x 0.014137 = 0.0565

But whole depth teeth, h = 9.7mm = 0.0097m

Induced stress, $s = 6Nh/bt^2$ where t is the thickness of gear teeth

$t^2 = 6Nh/sb = 6 \times 18 \times 0.0097 / 57600 \times 0.0565 = 0.000322$; $t = 0.018m = 18mm$

Thus, gear thickness of gear teeth, t = 18mm

3.5 Heat Calculations

Heat conducted through the materials will depend on;

i. The available time for heat transfer
ii. The type of material
iii. Thickness of the material
iv. The temperature difference, and
v. The area through which the transfer takes place

3.5.1 Total Area to be Heated

The screw press is in form of a frustum of a circular cone (Fig. 3.6).

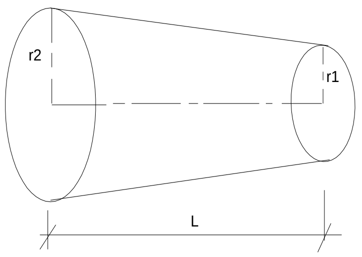

Fig. 3.6 The Press Chamber

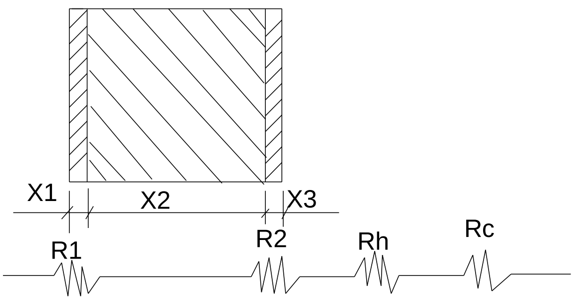

Fig. 3.7 Resistance to Heat Flow

Area = $\pi h_s (r_1 + r_2)$

Where h_s = slant height = $\sqrt{(r_2 - r_1)2 + L^2} = \sqrt{(0.270 - 0.170)2 + 1.380^2} = 1.384m$

Area, A = 3.142 x 1.384 x (0.27 + 0.17) = 1.913m^2

3.5.2 Determination of the Insulation Thickness

Proper insulation of heating equipment is important for efficient utilization of heat energy. Fiber glass was chosen (with a thermal conductivity of 39 W/m°C) because of its high sensitivity and ability to withstand temperatures up to the envisaged operating temperature. According to Woodcock and Mason (1987);

Quantity of heat transferred, $Q = KA (t_2 - t_1)2 / x$

Where K = Thermal conductivity of the material, 40W/mK; A = Area through which heat transfer takes place

t_1 = Initial temperature, 25°C = 298K ; t_2 = Final temperature, 98°C = 371K

x = Thickness of screw press chamber, 20mm = 0.02m

Q = 40 x 1.913 (371 – 298) / 0.02 = **27915 W**

ASSUMPTIONS

i. Steady state heat transfer exists
ii. One dimensional heat flow through the walls
iii. Effects of heat losses through edges and corners are neglected, and
iv. Material of the wall is considered to have constant thermal conductivity

$\Delta x_1 = \Delta x_3 = 0.005m$ where Δx_1 and Δx_3 are thicknesses of the inner and outer walls respectively.

Resistance, $R = \Delta x/K$

where K is the thermal conductivity.

Resistance of metal walls, $R_1 = 0.005/60 = 8.33 \times 10^{-5}$ W/K

where K for mild steel = 60W/mK

Resistance of aluminum wall, $R_3 = 0.005/52 = 9.62 \times 10^{-5}$ W/K

where K for aluminum = 52 W/mK

Resistance of insulating material (fiber glass), $R_2 = \Delta x_2/0.143 = 6.993\ \Delta x_2$ W/K

Where K for fiber glass = 39W/m°C = 0.143W/mK

R_n is the resistance due to surface conductance of the inner cage wall, and R_c is the resistance to heat flow of the outside wall.

$R_h = 1/h_h$; and $R_c = 1/h_c$ where h is the heat transfer coefficient of the surface with h and c representing hot and cold sides respectively.

Since heat transfer by convection is free or normal;

$h_c = h_h$; $1/h_c = 1/h_h = 1/10 = 0.1$mK/W

Total overall heat transfer coefficient, $V_c = 1/R_h + R_1 + R_2 + R_3 + 1/R_c$

$V_c = 1/0.1 + 8.33\times10^{-5} + 6.993\Delta x_2 + 9.62\times10^{-5} + 1/0.1$

$V_c = 20.0001 + 6.993\Delta x_2$

Rate of heat loss (Heat Flux), $Q/A = V_c\Delta T$ where ΔT = temperature difference = (373 – 298)K = 73K

A is the total heat transfer surface area = 1.9128m^2, and Q is the quantity of heat transferred = 27915W

$27915/1.912 = (20.0001 + 6.993\Delta x_2)\ 73$

$200 = 20.0001 + 6.993\Delta x_2$; $179.99 = 6.993\Delta x_2$; Δx_2 = 25.7mm

Thus, use insulation thickness of 26mm

4. PERFORMANCE RESULT

Performance tests were carried out at different operating temperature values. Digested fruits were fed into the screw press by gravity. Sprinkles of water at 100°C were added to the mash. Percentage of palm oil extracted was recorded at different operating temperatures.

Efficiency:

The oil extraction efficiency of the press can be defined as quantity of oil squeezed out by the press divided by the maximum oil content in the mash. According to Velayuthan and Chan (1983), the maximum oil content in palm fruits is 35% of whole weight. Thus 40kg of digested palm fruits was charged into the press chamber;

Total amount of realizable oil from the mash = (40/1) x (35/100) = 14kg of oil

Test 1: 12.20 kg of oil was extracted at an operating temperature of 92°C.

Efficiency, η_1 = (12.20/14) x 100 = 87.14%

Test 2: 12.96 kg of oil was extracted at an operating temperature of 96°C.

Efficiency, η_2 = (12.96/14) x 100 = 92.57%

Test3: 13.40kg of palm oil was extracted at an operating temperature of 98°C.

Efficiency, η_3 = (13.40/14) x 100 = 95.71%

5. CONCLUSION

It is observed that oil extraction efficiency is largely depended on the flow property of the oil which also depends on the viscosity of the oil and which in turn is a factor of the operating temperature. Maximum quantity of oil was extracted at an optimal temperature of 98°C.

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