ISSN 2049-7318

International Journal of Science and Technology Volume 3 No. 3, March, 2014

Design and Fabrication of Groundnut (*Arachis Hypogaea)* Roaster cum Expeller

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ABSTRACT

It is incumbent to find an effective, drudgery free and less expensive means of groundnut oil production. A combined groundnut roaster and oil extractor for medium scale expression of groundnut oil was designed, fabricated and tested to establish the influence of moisture content, heating time and temperature on percentage oil yield. The machine has a power rating of 5.5KW and extraction capacity of 4kg/hr with the screw shaft rotating at 60 revolutions per minute and a reduction gear to step down the speed of rotation to ratio of 25:1. The roaster was equipped with; a thermostat calibrated from 0-400^{o}C, a stirrer to create even distribution of heat. Four moisture content levels (6, 7, 8 and 9% wet basis) and heating duration (10, 20, 25 and 30 minutes) were used for the evaluation at four temperature settings 70, 80, 90 and 100^{o}C. The highest percentage oil yields were 41.6, 31.3, 25.5 and 21.2% at 6, 7, 8 and 9% moisture content wet basis respectively were all observed at 100^{0}C with exception of 21.2% observed at 70^{0}C. The machine gave the highest oil yield of 41.6%, an equivalent of 92% when compared to 45% oil content of groundnut, at 100^{o}C, 6% moisture content and heating time of 25minutes. The observed differences in percentage oil yield were found to be significant at 5% significance level. The overall cost of the machine was N160, 000, ($1066.7).

Keywords- *Extractor, Groundnut, Oil yield, Process parameters, Roaster.*

1. INTRODUCTION

Groundnut (*Arachis hypogaea*) contain up to 50% oil (although the usual range is 40% to 45%). ITDG (2011). The oil is known for its high concentration of monounsaturated fatty acids, considered by many as healthy oil. 1 cup of peanut oil, i.e. about 216 gm of this oil contains 99.79 gm monounsaturated fats, 69.12 gm polyunsaturated fats and 39.5gm saturated fats. It contains a high level of oleic acid, linoleic acid and palmitic acid. The oil has a high smoke point, which makes it ideal for cooking in high temperature without burning. It is known to possess a light nutty aroma and a pleasing taste.

The local roasting of groundnut produces uneven roast due to the unavailability of accurate temperature regulation devices and apart from exposing the groundnut to unhygienic conditions. It is a tedious process involving hand stirring and exposure to heat. Manual groundnut roasters with stirrer have being constructed (Lawal *et al.*, 1990) but a temperature regulatory device is still not inculcated because these roasters are powered by means of wood or charcoal. Thus, electrically powered roasters exists (Kabri *et al.,* 2006) but none in combination with an extractor. The combined groundnut roaster and oil extractor is aimed at removing drudgery at every stage of groundnut oil processing.

The objectives of this article is to design, fabricate and evaluate the machine performance in quest of knowing the best moisture content, heating time and heating temperature for higher oil yield.

2. METHODOLOGY

2.1 Description of the Machine

The orthographic drawing of the combined groundnut roaster and oil extractor is shown in figure 1 with the main components listed and fig 2a and b shows the plan and front view of the machine

1. Regulation Box containing the contact set.
2. Bevel and Pinion Gear Box
3. Feed Gate (Upper)
4. Roasting Chamber
5. Reduction Gear Box
6. Cage Bar
7. Cooling Unit
8. Support
9. Groundnut Oil Trough
10. Pinion Shaft.
11. Pulley and V-belt connection of the Bevel and Pinion gear to the Reduction Gear
12. Pulley and V-belt connection of the Electric Motor to the Reduction Gear.
13. Electric Motor.
14. Electric Motor Attachment to the Combined Groundnut Roaster and Oil Extractor

2.2. Basic Designs

Since the aim of this study was to design and fabricate a machine that will both roast and express oil from groundnut without jeopardizing efficiency and at minimal cost, hence, local, durable materials were used for the fabrication. The designed components include: expeller capacity, roaster volume, heater capacity, screw thread, power requirement, and belt and expression barrel.

2.2.1 Determination of Expeller Capacity

The mass of groundnut processed per hour,

$$\rho = density = \frac{M}{V} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

Where,

V- Volume of groundnut extracted per batch, m^3
M- Mass of groundnut, Kg

Using the packed density of groundnut to account for volume of void;

$$\frac{7}{12} \times 913 = \frac{532.6kg}{m^3}$$

10 litres of oil is assumed to be extracted per day, mass (m) of oil extracted;

$$m = V \times \rho = 1.83kg \quad \ldots\ldots\ldots\ldots\ldots\ldots \ (2)$$

Where,

V- Volume of groundnut, m^3
ρ- Density of groundnut, kg/m^3
Assume 1kg of oil is extracted per hour.
Since oil content of the groundnut is 40-45% (ITDG, 1995)

$$\frac{mass\ of\ oil}{mass\ of\ groundnut} = 0.45, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3)$$

+

$$mass\ of\ groundnut = 4kg$$

4kg of groundnut is extracted per hr/batch, in a continuous process

2.2.2 Design of Heating Chamber

Volume required, V = $\frac{mass\ of\ groundnut}{bulk\ density\ of\ groundnut}$ … (4)

$$V = \frac{4kg}{532.6kg/m^3} = 0.0075m^3 = 7.5 litres$$

$$V = \pi \times r^2 \times h \quad \ldots (5)$$

$$0.0075 = r^2 \times h,$$
$$r^2 = \frac{0.0075}{0.1275} = 0.059m^2$$
$$r = 0.24m$$

2.2.3 Determination of Roaster Volume, v,

In order to estimate the roaster volume, the area of cross-section was calculation as, Area of roaster cross-section (A)

$$A = 2\pi r^2 + 2\pi rh \quad \ldots (6)$$

$$A = 2 \times \pi \times 0.24^2 + 2 \times \pi \times 0.24 \times 0.1275 = 0.5542m^2$$

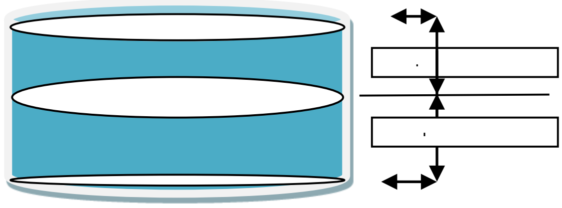
Calculated Volume (v) of roaster= $\pi \times r^2 \times h = \pi \times 0.24 \times 0.1275 = 0.0231\text{m}^3$, *where*

$r = radius\ of\ roaster\ cylinder(m),$

$h = height\ of\ cylinder(m)$

Calculated volume, v, is greater than the required volume V.

The total volume of the roasting chamber is $0.0231m^3$



2.2.4 Determination of Heater Capacity

Going by Fourier's law of conduction, (Osore, 1997)

$$Q = \frac{-KA(t_2 - t_1)}{x} \quad \ldots (7)$$

Heat flow by conduction in a cylinder,

$$Q = -KA\frac{dt}{dr} \quad \ldots (8)$$

$$Q\frac{dr}{r} = -2\pi Lkdt \quad \ldots (9)$$

$$Q\int_{r_1}^{r_2}\frac{dr}{r} = -2\pi LKdt \times \int_{t_1}^{t_2} dt$$

$$QIn\frac{r_2}{r_1} = -2\pi LK(t_2 - t_1)$$

$$Q = \frac{2\pi LK(t_2 - t_1)}{In\frac{r_2}{r_1}} \quad \ldots (10)$$

$$Q = \frac{\frac{(t_1 - t_2)}{In\frac{r_2}{r_1}}}{2\pi LK}$$

$$R_T = \frac{In\frac{r_2}{r_1}}{2\pi LK} \quad \ldots (12)$$

For continuity of flow, the heat transfer through each layer must be the same, hence

$$Q_{cond} = \frac{2\pi LK_A(t_1-t_2)}{In\frac{r_2}{r_1}} = \frac{2\pi LK_B(t_2-t_3)}{In\frac{r_3}{r_2}} \quad \ldots (13)$$

$$R_T = \frac{In\frac{r_2}{r_1}}{2\pi LK_A} = 0.0001691K/W$$

$$Q_{cond} = \frac{(t_1-t_3)}{R_T} = \frac{250-100}{0.0001691} = 887kW$$

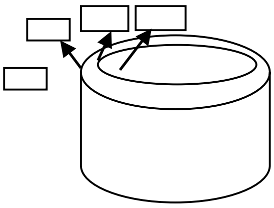


Fig 4: Roaster Thickness and Different Temperatures

To determine the heat transfer rate from the heater to the roaster made of 1.5mm thick mild steel with radius, r, 230mm and 255mm long, by conduction. Mild steel is of thermal conductivity, k_A, 48W/mk. The heat transfer coefficient, h_A, on the inside wall of the roaster is 3000W/m^{2}k.

r_1= 0.23m, cylinder bore
r_2= 0.235m, cylinder bore + thickness
r_3= 0.2465m, cylinder bore + thickness + insulation thickness
R_T= Thermal Resistance due to conduction
Q= rate of heat flow through a block of material, W or J/s
x= thickness of the roasting chamber through which heat is transferred, m
A= area of the roaster, m^2
T_3 (°C) = t_d= desired temperature of groundnut =80°C
T_2 = inner surface temperature (°C) where t_2>t_3
T_1=temperature of the outside surface of the roaster
T_4= atmospheric temperature = 27^{0}C
Assuming a temperature of 150°C for the outer surface temperature of roaster, T_1
K= thermal conductivity of mild steel = 48-58W/mk
L= height of a roaster chamber containing 4kg groundnut, 0.1275m

Treating the groundnut flow as fluid flow over a solid surface and considering the heat transfer from the roaster to the groundnut by convection.

Newton's law of cooling, (Osore, 1997)

$$Q = hA(t_s - t_f) \quad \ldots (14)$$

t_f = Fluid temperature
t_s = solid surface temperature
h =convection heat transfer coefficient, W/m^{2}K

$$Q = \frac{t_1-t_3}{R_T} \quad \ldots. (15)$$

$$R_T = R_A + R_1 \quad \ldots (16)$$

$$R_A = \frac{1}{2\pi r_1 L h_A} = 0.001809K/W \quad \ldots (17)$$

$$R_1 = \frac{In\frac{r_2}{r_1}}{2\pi LK_1} = 0.0001691K/W \quad \ldots (18)$$

$$R_T = R_A + R_1 = 0.0019782K/W$$

$$Q_{conv} = \frac{t_3-t_4}{R_T} = 2.741kW$$

A 3.0kW heater is required for roasting of the groundnut in the roaster.

2.2.5 Design of Temperature Control Capacity

To design and make use of the right temperature control for the roaster is important to avoid overloading thereby burning of the temperature control component.

$$Q = IV \ , I = \frac{Q}{V} = \frac{3KW}{230} = 13amps \quad \ldots\ldots.(19)$$

Where,

Q- Heater capacity of heating elements,kW
V- Electrical voltage entering into the temperature control, Volt
I-Required current capacity of the temperature control, Amps
The capacity of the temperature control and contact setting used was 40amps.

2.2.6 Design of Gear Unit and Stirrer Shaft,

Desired range of speed for the stirrer is 4 to 5rpm, gear design calculation at http://www.technologystudent.com/gears1/gears5.htm, the number of teeth knowing the rpm, can be selected as,

$$N = \frac{G_a}{G_b} \quad \ldots (20)$$

N= number of revolution per minute of the gear shaft, 5rpm
G_a= number of teeth of driver gear
G_b=number of teeth of driven gear

$$N = 5 = \frac{G_a}{G_b} = \frac{8}{40} = \frac{9}{45}$$

$$\frac{5}{1} = \frac{input\ movement}{output\ movement}$$

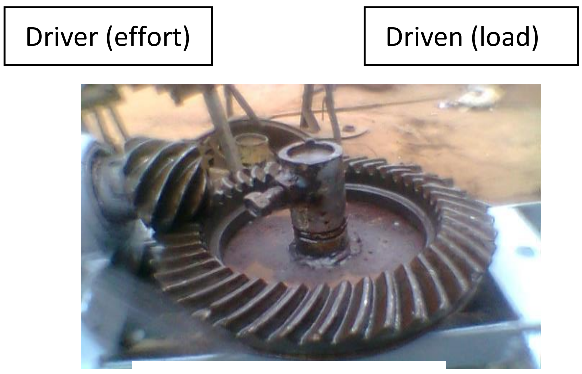


Plate 1: Gear Mesh

2.2.7 Screw Thread Design

The length to diameter ratio (L/D) of biopolymer extruder according to (Fayose *et al.* 2009) is 15.

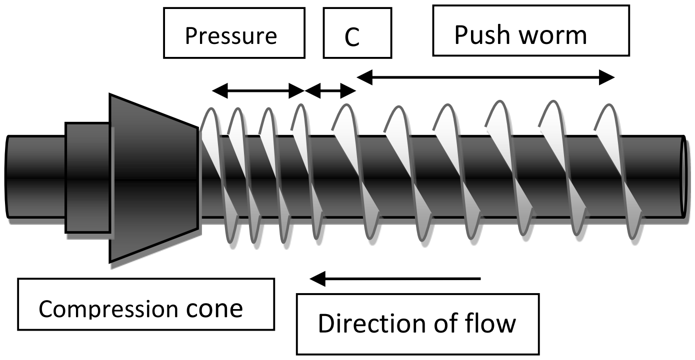


Fig 5: Threaded Screw Shaft

2.2.8 Handling capacity of the expeller

Deduced from roaster capacity = 0.0231m^3/h = 0.000385m^3/min .In one revolution, the expeller will handle $\frac{0.000385}{60} = 0.00000642m^3/min/rev$

2.2.9 Determination of Pitch between Worms

To determine the pitch in between the worms, the volume of the expression chamber is required and is calculated as;

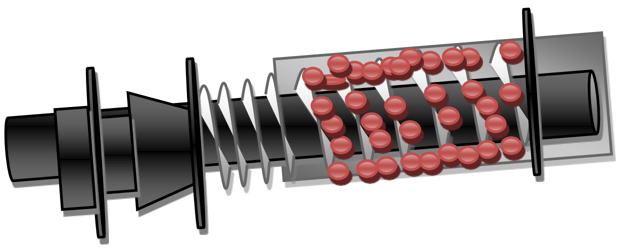


Fig 6: Groundnut in the Pressure Section of Threaded Screw Shaft

$V_C = \frac{\pi \times D_c^2 \times P}{4} = 0.001590P$ … (21)

V_T= Volume of shaft + Volume of thread

Volume of shaft $= \frac{\pi \times D_{io}^2 \times P}{4} = 0.0004524P$ … (22)

Volume of thread is gotten by assuming an unwrapped section of the thread;

Volume of thread $= \frac{\pi((D_i+2b)^2 - D_i^2)b}{4} = 0.00001629$ … (23)

V_T= 0.0004524P + 0.00001629

V_G=0.00000642=V_C – V_T = 0.001590P -0.0004524P – 0.00001629 … (24)

0.002042P – 0.00002271=0

P= 0.0111m = 11.1mm

2.3.0 Determination of Number of threads (n) on the screw

The number of thread considered here is found to be the minimum push worm required for the machine.

$n = \frac{lenght\ of\ screw\ threaded\ section}{pitch} = 6$ … (25)

2.3.1 Power Required by the Screw Shaft

With a chosen speed of 60rpm for the expeller shaft and 25:1 revolution of electric motor to expeller shaft. An electric motor capable of producing; Khurmi and Gupta (2009)

$$w = 60 \times 25 = 1500\text{rpm}$$

$$w = \frac{2\pi N_w}{60} = 157.1 rad/s \quad \dots (26)$$

$$T = \frac{P}{w} \quad \dots (27)$$

$$\frac{N_1}{D_1} = \frac{N_2}{D_2}, \quad \dots (28)$$

$$N_1 = 1500rpm, N_2 = 60rpm$$

$$since\ ratio\ of\ diameter\ of\ driver\ to\ driven\ is\ 1:2,$$

$$D_1 = 0.09m \ \ D_2 = 0.15m$$

Where;

w= angular speed of screw shaft, rad/s

T= torque transmitted by worm, Nm

N_w=number of revolution per minute of worm action, rpm

P=power transmitted by worm action, W

N_1=speed of driving (electric motor) shaft, rpm

N_2=speed of driven (expeller) shaft, rpm

D_1=diameter of driver pulley, m

D_2=diameter of driven pulley, m

2.3.2 Determination of Belt Length, L.

To get the angle of contact or lap for both pulleys, Khurmi and Gupta (2009)

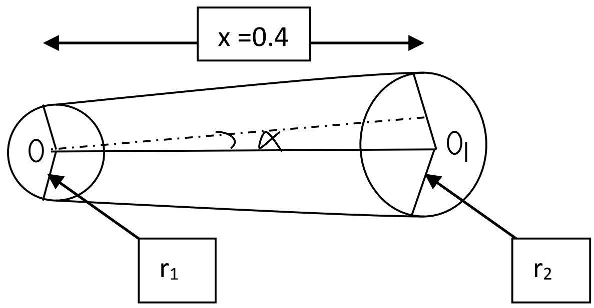


Fig. 7: Pulley-Belt Cross-sectional View

$$sin \propto = \frac{r_2 - r_1}{x} = 0 \quad \dots (29)$$

Hence $\propto = 0$

Angle of wrap,

θ= 180 – 2α=180° =3.142rad … (30)

T_1=Tension in the tight side of the belt

T_2=tension in the slack side of the belt

Assume the grove angle of the pulley,

2β =34° and β=17°

Ratio of belt tension is given by,

$$2.3 log\frac{T_1}{T_2} = \mu\theta cosec\ \beta = =3.01 \quad \dots (31)$$

T_1=tension in the tight side of the belt

T_2= tension in the slack side of the belt

$$log\frac{T_1}{T_2} = \frac{3.01}{2.3} = 1.3087$$

$$\frac{T_1}{T_2} = 20.36 \quad \dots (32)$$

Velocity of the belt, v,

$$v = \frac{\pi \times d_2 N_1}{60} = 11.78m/s \quad \ldots (33)$$

Power Transmitted by belt, $P_s = T_\gamma w$ according to Khurmi and Gupta (2009)

T_γ=shear stress of agro materials, given as 5.995
w=required angular speed of expeller shaft, 60rpm

$$P = 5.995 \times 157.1 = 785.5W = 0.786kW$$

$$P = (T_1 - T_2)v \quad \ldots (34)$$

$$(T_1 - T_2) = 111.12 \quad \ldots (35)$$
$$\frac{T_1}{T_2} = 20.36 \quad T_1 = 20.36T_2$$
$$20.36T_2 - T_2 = 111.12 \quad T_2 = 5.74N$$
$$T_1 = 111.12 + T_2 = 116.86N$$

At maximum power condition, maximum tension, T, is given by,

$$T_1 = \frac{2T}{3} \quad \ldots (36)$$

$$T = \frac{3T_1}{2} = 175.29N$$

Centrifugal tension in belt, T_c,

$$T_c = \frac{T}{3} = 58.43N \quad \ldots (37)$$

Cross sectional area of belt, A,

$$A = \frac{T}{\sigma} = 33.4 \times 10^{-6} m^2 \quad \ldots (38)$$

σ=permissible stress in belt material, given as, 1.75 x 10^{-6}

Required Belt length L,

$$L = 2c + \frac{\pi}{2}(D_1 - D_2) + \frac{D_2 - D_1}{4c} \quad \ldots (39)$$

c- center diameter between the pulleys = 0.4m

$$L = 2(0.4) + \frac{\pi}{2}(0.09 + 0.15) + \frac{0.15 - 0.09}{4(0.4)} = 1.2m$$

Since D_1=0.09m D_2=0.15

2.3.3 Determination of Center Distance, between Pulleys of the Expeller Shaft and the Electric motor

$$1.2 = 2c + \frac{\pi}{2}(0.09 + 0.15) - \frac{0.15 - 0.09}{4c}$$
$$4.8c = 8c^2 + 1.50796c + 0.06$$
$$c^2 - 0.41151c + 0.0075 = 0$$
$$c = \frac{-b \pm \sqrt{b^2 - 4ac_1}}{2a} \quad \ldots (40)$$

$c = 0.39$ Using +ve sign

2.3.4 Design of Expeller Shaft Diameter, d, and Machine Power Requirement, P

Using the calculated power required to drive belt, P=Tw

$$T = \frac{P \times 60}{2 \times \pi \times N} = 125Nm \quad \ldots (41)$$

According to PSG TECH (1982)

Using $P = QL\frac{W_0 \pm \sin\beta}{367}\frac{1}{\epsilon}$ … (42)

Where;
P=Power Required by Shaft
Roaster capacity, C = 0.0000064m^3/s
Mass flow rate, m= 532.6 kg/m^3
Q = volumetric capacity of the, 0.023ton/h
L= length of screw Shaft
W_o= material factor, 4 for powder
β=angle of inclination of screw shaft to horizontal, 3.01
€= gear reducer efficiency, 25:1
P=5.59kW +ve
P=5.45KW -ve

An electric motor of 5.5KW is needed for powering the machine.

$$number\ of\ belts = \frac{designed\ power}{power\ transmitted\ per\ belt} = \frac{5.45}{0.786} \cong 7 \quad \ldots (43)$$

2.3.5 Design of Shaft diameter, d,

Bending moment Calculation, M on the shaft due to belt tensions,

According to Khurmi and Gupta (2009),

$$M = (T_1 + T_2 + 2T_c)O_v \times n \quad \ldots (44)$$

$$M = (116.86N + 5.74N + 2(58.43N)) \times 0.08 \times 25 = 478.92Nm$$

n= number of belts

O_v= overhang distance of center of driven pulley to nearest bearing,

Equivalent twisting/torsional moment, M_T

$$T_e = M_T = \sqrt{T^2 + M^2} \quad \ldots (45)$$
$$T_e = M_T = \sqrt{244989.37} = 494.96Nm$$
$$d^3 = \frac{16}{\pi \times s_s} T_e = 0.03979 = 0.40m \quad \ldots (46)$$

d=0.40m

2.3.6 Design of the Expeller Barrel

The maximum tensile stress (hoop or circumferential stress), σ_h, the cylindrical barrel will be subjected to, at failure is considered, as;

$$\sigma_h = \frac{f}{tl} = \frac{Prl}{tl} = \frac{Pr}{t} \quad \text{(Lasisi, 1997)} \quad \ldots (47)$$

Where;
P=intensity of the internal pressure (N/m^2)
r=internal radius of the cylinder wall (m)
l=length of the cylinder wall (m)
t=thickness of the cylindrical wall (m)
f=hoop force (N)

$$t = \frac{D_{co} - D_{ci}}{2} \quad \ldots (48)$$

$$\sigma_h = \frac{2PD_{ci}^2}{D_{co}^2 - D_{ci}^2} \quad \ldots (49)$$

The yield or ultimate stress for mild stress is 140MN/m^2

The allowable stress considering factor of safety of 2 is 70MN/m^2

P=40 X 10^{-6}N/m^2
D_{ci} =inner diameter of cylinder = 45mm =0.045m

$$70 \times 10^6 = \frac{2 \times 40 \times 10^6 \times 0.045^2}{D_{co}^2 - (0.045)^2}$$

$$D_{co} = 0.066m$$

Thickness of the barrel $= t = \frac{0.066-0.045}{2} = 0.0104m$

2.3.7 Bending Stress determination

Bending load/bending stress acting on expeller shaft (tension or compression) is:

$$s_b = \frac{M_{b\times r}}{I} \qquad \ldots (50)$$

Hence,

$$\mathrm{S_b} = \frac{32\mathrm{M_b}}{\pi \mathrm{d}^3} = \frac{32\times 478.92}{\pi \times 0.40^3} = 76.2MN/m^2$$

Where;
S_b = bending stress
Mb = bending moment
d = shaft diameter
I = moment of inertia

Also, $I = \frac{\pi d^4}{64}$ for a cylindrical shaft (PSG Tech, 1982).

2.3.8 Torsional Stress

Torsional stress is determined using

$$\tau_{xy} = \frac{M_{T\times r}}{J} \qquad \ldots (51)$$

But $J = \frac{\pi \times d^4}{32}$ hence,

$$\tau_{xy} = \frac{16M_T}{\pi d^3} = \frac{16\times 494.96}{\pi \times 0.040^3} = 39.4MN/m^2 \ldots (52)$$

Where;
τ_{xy} = Torsional stress, N/m^2
M_T = torsional moment
r = radius of shaft
J = Polar moment of area
d = diameter of shaft

2.3.9 Bearing Selection

The machine has two bearings each located at both ends of the extractor shaft and three bearings on the roster, two on both ends of the pinion shaft, and the third on the stirrer shaft. Static and dynamic conditions as well as design life requirements are considered in selecting these bearing. Bearing must be selected based on its load carrying capacity, life expectancy and reliability (PSG Tech 1989). The relationship between the basic rating life, the basic dynamic rating and the bearing load is:

$$C = \frac{L}{L10} \times \frac{P}{K} \qquad \ldots (58)$$

$$\frac{C}{P}K = \frac{L}{L10}$$

$$L = \frac{60n}{106} \quad \text{million revolutions}$$

$$L10 = \frac{C}{P}K \times \frac{106}{60n}$$

Where;
L10 = life of bearing for 90% survival at one million revolutions;
L = required life of bearing in million revolutions (mr);
n = rotational speed, rev/min
C = basic dynamic load rating, N
P = equivalent dynamic bearing load, N
K = exponent for life equation with
K = 3 for ball bearing
K = 10/3 for roller bearing

Also, P = radial load + axial load

P = ($XF_r + YF_a$) … (53)

Where,
X = radial load factor for the bearing
Y = axial load factor for the bearing
F_r = actual radial bearing load, N
F_a = actual axial bearing load, N

The recommended life value in operation is as shown in Table 6. It is assumed that this machine is designed to operate for 6 hours per day intermittently and whose breakdown will have serious consequences. The bearing life in operating hours is chosen to be 9,000.

According to FAG standard table the dynamic loading, C, on the bearing can be approximated to 14 000, Hence a ball bearing of Designation 6205 is used. (FAQ, 2008)

2.4.0 Design of key-way and key selection

Chosing a value of shaft diameter between 38mm and 44mm, from the standard keys and key-ways code (Khurmi and Gupta (2009)),

The width and thickness of the key can be estimated as follows,

$$\frac{44-38}{40-38} = \frac{14-12}{w-12} \qquad \ldots (54)$$

$$width, w = 12.7mm$$

$$\frac{44-38}{40-38} = \frac{9-8}{t-8} \qquad \ldots (55)$$

$$thickness, t = 8.3$$

2.4.1 Principle of Operation of the Machine:

The machine works as a continuous process. Roasting of the groundnut is done with the aid of an electric roaster. A cylindrical shaped roasting chamber having two compartments, the pre-heating or warming is done at the upper chamber while the final roasting, to specific temperature is carried out in the lower chamber. The lower chamber roaster has an electric heater attached to its base while another electric heater is attached to the base of upper chamber roaster. A shaft is passed through the centre of the cylindrical roaster and attached to the shaft are stirrers for creating even heat distribution thereby creating even roasting effect and preventing burnt roast. A gate is located on base of both the upper and lower chamber of the roaster. The gate for the lower chamber is placed directly above the hopper of the screw extractor to maintain the groundnut temperature and allow for easy flow of roasted groundnut into the screw extractor hopper. The combined groundnut roaster and oil extractor is efficient and cost effective considering the energy and time conservative mode of operation.

3. RESULTS

The machine was test-run after fabrication in order to ascertain it performance. The result of the performance was summarized in table 1 and 2 below.

3.1 Discussion

A combine groundnut roaster and oil extractor machine was designed, fabricated and easy to operate with relative cost. The materials for fabricating the machine are sourced locally which make it maintenance friendly. Oil yield decreased with increase in moisture content from 6 to 7%, 7 to 8% and 8 to 9% (Table I). Oil yield increased with increased heating time to certain level and then decreased with further increase in heating time (Fig. 8-11). For all moisture content at 100ºC, oil yield increased with increased heating time until a heating time of (25 minutes) was reached. Increasing heating temperature to a highest value of 100ºC and a heating time of 25minutes increased the percentage oil yield to a value of 41.6%, obtained at 6% moisture content. The increased heating temperature and time also increased the colour intensity of the oil expressed which agrees with the finding on groundnut by **(**Adeeko and Ajibola. ,1980), (**Ajibola *et al.*, 1990)**. The expression efficiency is approximately 92% when compared to the 45% oil content of groundnut.

The colour of oil extracted was observed to be affected by the heating time and heating temperature as confirmed by (Makeri *et al*, 2011). Thus no substantial increase was observed in the percentage oil yield beyond 100^0C, this is in correlation with the findings of (Adeeko and Ajibola 1990).

4. CONCLUSION

A combined groundnut roaster and oil extractor was designed, fabricated and evaluated for efficiency. The result implied that the efficiency of the combined groundnut roaster and oil extractor is dependent on the heating temperature of extraction, heating time of extraction and feeding rate of extraction. The inclusion of a roaster apart from removing the drudgery and time consuming process involved in heating processes of groundnut before extraction also lead to basic improvement in the extraction efficiency. A slow feeding rate will ensure no wastage of groundnut paste through other outlets during extraction. The choice of material and overall cost of machine would encourage mass production and, hence, reduction in price and availability of non-contaminated and non-adulterated groundnut oil.

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