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Design and Development of a Continues Palm Nuts Digesting Machine

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Abstract: *The design for the development of a continuous cooked palm nuts digester is conceptualized to reduce material handling in oil palm processing and increase productivity, save time and energy. From economic impact assessment, the machine reduces the cost of acquicission of two or more batch process cooked palm nuts digesters. Using CAE software, SOLIDWORKS the machine is designed and developed. The machine is designed with a barrel capacity of 0.1429*m^3 *and an ultimate load of 2.567KN. A worm –beater shaft at a 720rpm speed, 0.14m pitch transporting and apparent volume of 0.01588*m^3 *of palm fruits undergoing digestion per revolution.*

Key Words: *Beaters, worm drive, continuous digester, palm oil production*

INTRODUCTION

Local palm oil production has stages, after local heating/boiling in drums, digestion of palm fruit nuts is usually performed by the use of mortars and pestles or by barefoot mostly by women and young men. This has been on before the advent of locally fabricated machines like palm kernel cracker, digester etc. in 1997, to alleviate the hard working high palm oil producing community Enwan in Akoko Edo local Government Area of Edo State, Nigeria. It was gathered that besides the advent of mechanical means, the palm oil produced by this community has a shelf life of six (6) years without odour. These primitive methods of processing palm oil are usually characterized by slow and very low digestion, production rates, time-consuming, and drudgery and the fact is that a lot of human energy is usually expended during digestion even if it is termed unhygienic. The Oil Palm (*Elaeis guineensis*) is traced to have its origin from the tropical rain forest region of West Africa; this vegetation is found around Cameroon, Cote d'Ivoire, Ghana, Liberia, Nigeria, Sierra Leone, and Togo and the equatorial region of Angola and the Congo (Kwasi Poku, 2002). The palm fruit nut is made up of an outer skin (epicarp), a pulpy skin (spongy mesocarp) containing the palm oil in fibrous matrix, central nut consisting of stony shell carp and the kernel which itself contains palm kernel oil (Kwasi Poku, 2002).

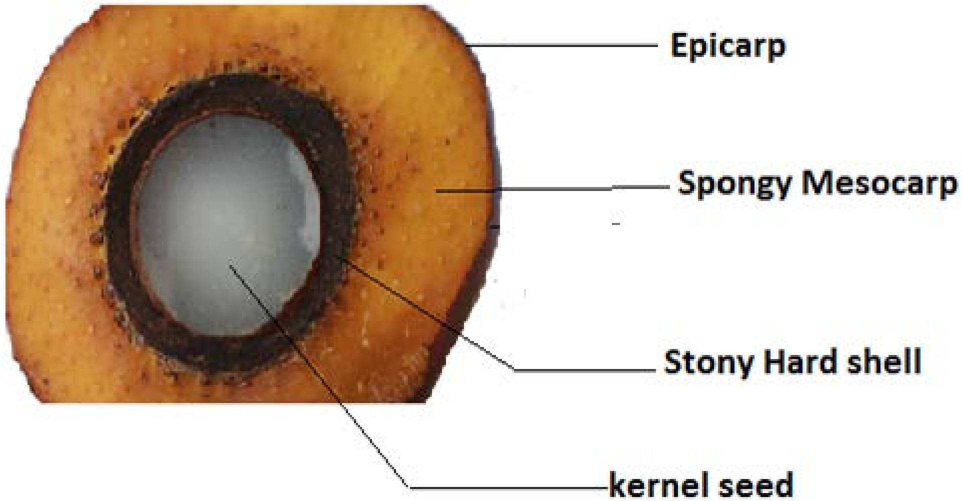


Fig.1. Section of the Palm Nut Structure

There are three varieties of palm fruit: the *dura, pisfera* and *tenera*. The *dura* has a thick pericarp or exocarp, 2–8 mm thick, a thin mesocarp (which is the reason for the low palm oil production of this variety), thick endocarp (shell) and generally large kernels (this makes dura suitable for kernel oil production). Its mesocarp content is 35–55%. The *pisfera* has fruits which possess thick mesocarp (with very little oil content), no endocarp (shellless) and with small kernel. The *tenera* type possesses thick mesocarp (much pulp), thin endocarp and a reasonably sized kernel. It is the product of the cross of *pisfera* and *dura*. It has high potential for the production of mesocarp (palm) oil but less kernel oil in comparison to *dura* variety (Opeke LK, 1982). Also, (Mathieu N, 2016) reported that the ratio of polyunsaturated fatty acids was higher in *pisfera* than *dura* and *tenera.* Processing of oil palm fruit is undertaken bearing in mind that the concern is mainly palm oil while palm kernel oil, shells, and fibres as by- products. The palm nut can be processed into shell, kernel oil and palm kernel cake, which have numerous domestic and industrial applications (Owolarafe OK et al, 2002). The basic process in the palm oil production process is digestion. Digestion is the process by which boiled or sterilized fruits are macerated for selective easy separation of oil from the fiber (Nchanji YK, et al, 2013). This involves crushing and detachment of the heat-weakened mesocarp from fruit nuts. Digestion and oil separation from the fibres and nuts are the most tedious and essential operations in traditional palm fruit oil production; therefore, early efforts are focused on improving these tasks (Adeniyi OR et al, 2014).
There are different types of digesters:

I. The beaters digesters
II. Screw digesters

The beaters types comprise shaft with beaters which digest boiled palm nuts to remove the oily spongy mesocarp from the kernel through collision and squeezing. The beaters types are also group based on the positioning of the shaft. There is a vertical shaft with augers or beaters and those of the horizontal shaft with auger or beaters. However, the locally available are vertical shaft, batch production machine with one or two through puts. Digestion involves the meshing up of the fruits under steam-heated conditions and pressing out the crude palm oil. Generally, twin-screw press is used to extract the oil from the digested mesh. The crude oil is further purified and dried for storage. The solid wastes from the milling operations are empty fruit bunches, palm fibres, and palm kernel shells (Nwankwojike BN et al, 2011). Previously, it is discussed that oil palm fruits digester comes in two categories and these are vertical and horizontal digesters (Stephen KA et al, 2009). In a vertical digester, digestion of fruit mesocarp is done by a vertical shaft fitted with beater arms at specified orientation and spacing on the shaft; while in horizontal digester, digestion is done by the beater arms which also transport macerated fruits from the digester inlet to the outlet (Ogblechi SR et al, 2014).

The oil extraction process involves several steps namely sterilization, bunch stripping or threshing, digestion of fruits, followed by extraction of oil and clarification and purification as shown in Figure 2. Fresh fruit bunches are steam sterilized at around 140°C for a period of 75 to 90 minutes. The sterilizing or steaming of the fruits are to facilitate separation and threshing of bunches to free the palm fruit. However, the main objectives are to deactivate hydrolytic enzymes responsible for the breakdown of oil to free fatty acid, to coagulate and facilitate the breaking of oil cells. In the bunch threshing step, the fruits are stripped and separated from fruit bunches in a rotary drum. Fruits are knocked out of the bunch as they passed through the bunch stripper (Nwankwojike BN et al, 2011). The detached fruits and sepals are collected and passed through a separator, clean fruits boiled and carried into the digester. Palm oil processing flow chart is conceptualized as this is based on the technology involved and the understanding of the person.

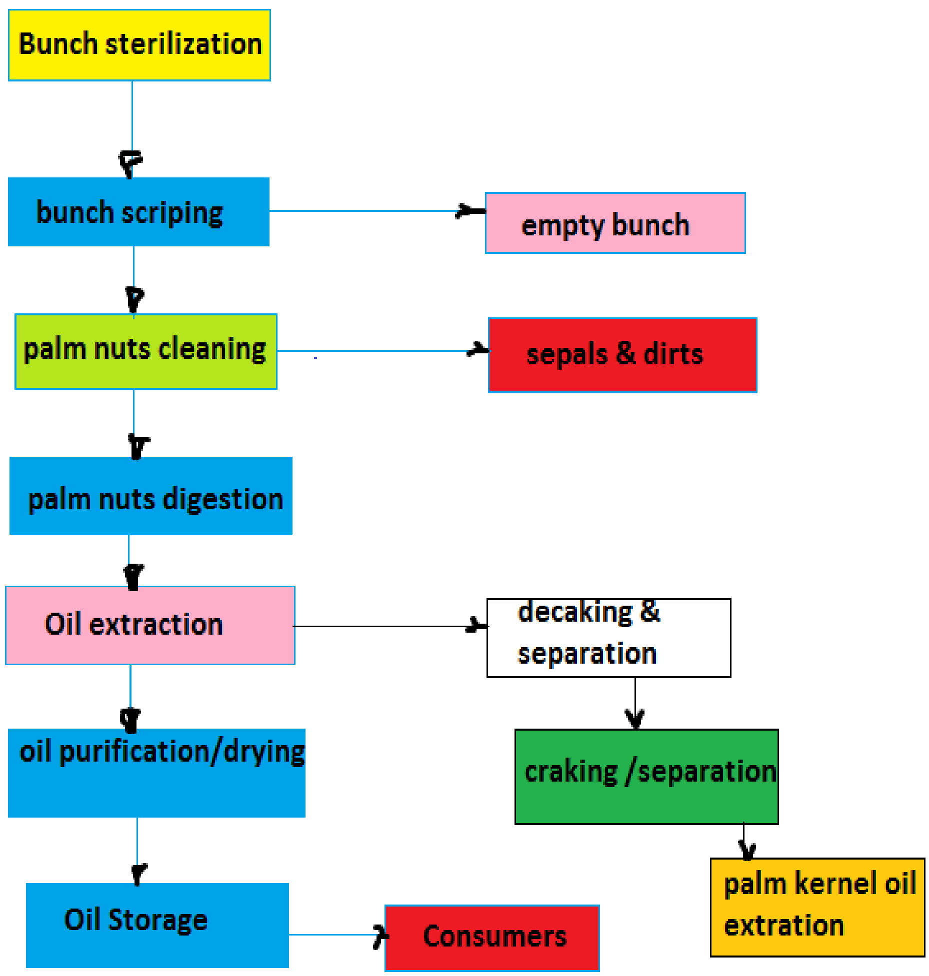


Figure 2: Flow diagram of palm oil extraction process

Table 1. Composition of oil palm fruit bunch

COMPOSITION	QUANTITY
BUNCH WEIGHT	23-27 Kg
Fruit /bunch	60-65%
Oil/bunch	21-23%
Kernel/bunch	5-7%
Mesocarp/bunch	44-46%
Mesocarp /fruit	71-76%
Kernel/fruit	21-22
Shell/fruit	10-11

Source (Poku K, 2002)

Table 2. Palm kernel oil composition

CONTENT	PERCEN(WEIGHT BASIS)
Free Fatty Acid(FEA)	3-4
Volatile matter including water	0.15-0.20
impurities	0.05-0.10
Peroxide(inch equivalent /kg)	2.0-4.0
Saponification value	242-222

Source (V ON, 1987)

Table 3. Proximate composition of palm kernel

COMPONENT	PERCEN(WEIGHT BASIS)
OIL	47-49
Crud protein	7.9-9.0
Extractable non-nitrogenous matter	23-24
cellulose	9
Ash	2
Water	7.5-9.0

Source (Hartely CWS, 1967)

2. Design Analyses

Methodology

The design for the development of the continuous boiled-palm kernel nuts digesting machine began by developing the concept and with computer aided engineering (CAE) - a virtual design is done with SOLIDWORKS 2013. The design was followed with design and material selection considerations.

Design Consideration

The functional parameters and component parts that are considered in this work are:

a. Density of boiled palm nut
b. Machine pulley speed (N_2)
c. Electric motor selection

d. Machine Torque
e. Belt
f. Size of shaft'
g. Bearing selection
h. Key and key way

Material Consideration

Materials needed to fabricate this machine are selected based on:

i. Nigeria's local content initiative policy.
ii. Availability of the material in the local markets
iii. Machinability
iv. cost
v. Hygiene

Hygiene is less considered since the oil will undergo clarification and drying. The materials selected are majorly Mild steel except the pulleys that are aluminum.

3. Mathematical analysis

***Volume and Weight Capacity*:**

The effective volume of this Machine comprises the volume of hopper and the barrel which are occupied by the palm fruit to be digested. It depends on the part of its trough that constitutes upper chamber where the palm fruits from the hopper is slacked before feeding to digesting unit at the lower chamber occupied by the worm shaft and beaters

i. **Hoper volume**

Feeding Unit/ Hopper Design

The feeding into the Hopper is continuous since the machine is a continuous processing machine. Calculating the volume of the Hopper:

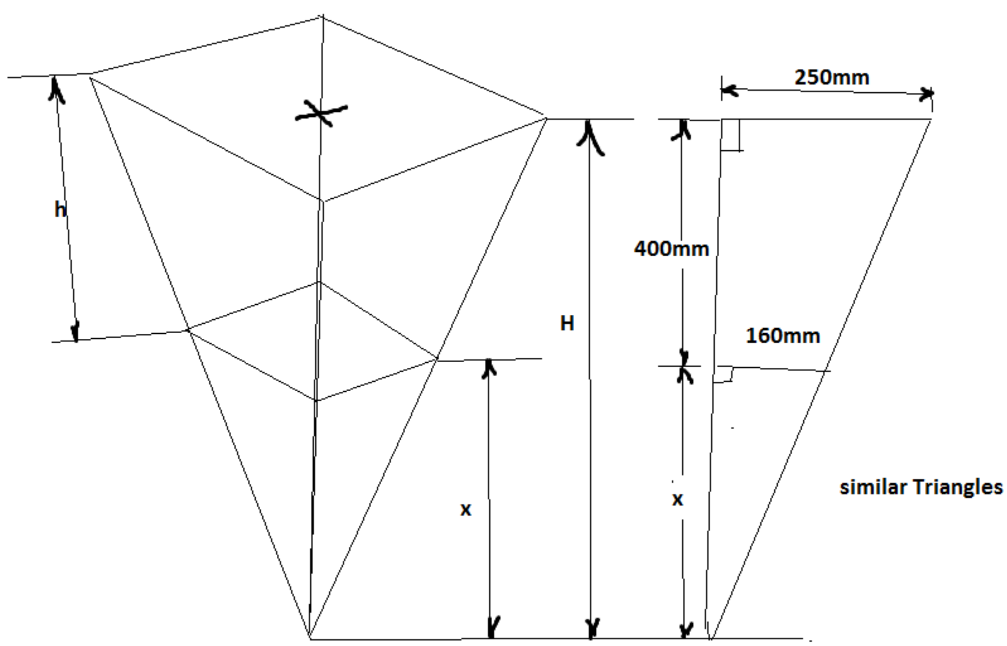


Fig.3a Pyramidal portion calculation using congruent triangle

$$The\ Volume\ of\ the\ Frustum\ =\ bigger\ volume - smaller\ volume.$$

Height of Hopper from calculation =h.
Also, height of the pyramid from calculation =H.

From figure 3. Using congruent triangle

$$\frac{250}{H} = \frac{160}{x}$$

$$\frac{250}{h+x} = \frac{160}{x}$$

Value of x is calculated

$Volume\ of\ frustrum\ =\ (1/3\ \times\ Base\ area\ \times\ Height) - (1/3 \times\ base\ area\ \times\ height).$ (1)

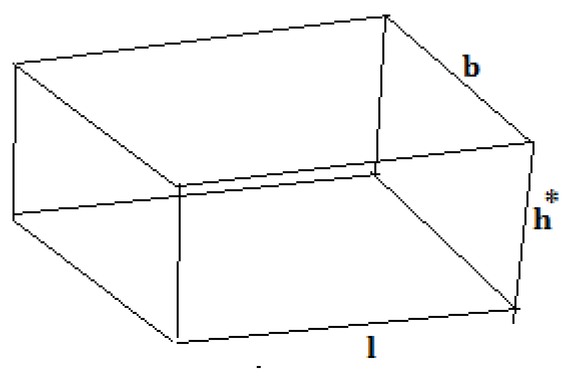


Fig3b. Cuboid Part of the Hopper

$Volume\ of\ the\ cuboid\ = Lbh^*$ (2)

$The\ \ volume\ of\ the\ hopper\ \ = volume\ of\ the\ truncated\ pyramid\ + volume\ of\ cuboid$ (3)

ii. Housing Unit / Barrel Design.
Calculating the volume of the barrel used for the design:
Length of the barrel, L
Internal Diameter of the Barrel, D
Internal Radius of the Barrel, R

Volume of the barrel is given as:
$V_b = \pi R^2 L$ (4)

iii. Full Load Condition

Density of boiled or cooked palm fruit =373.47kg/m^3
Volume of the barrel or drum = $\pi R^2 L$
Mass of the palm fruit =?

$$Mass(kg)\ =\ Density\ \times\ Volume.$$

Bulk density of digested palm fruits=1060kg/m^3 (Muthurajah, 2002)

Bulk density of conditioned palm fruit=708.66 kg/m^3 (Renel M .Alucilja et al 2013)

Bulk density of fresh palm fruit =707.55 kg/m^3 (Renel M .Alucilja et al 2013)

Load of palm fruits in the machine (W) = mg (5)

g= 9.81 m/s^2

The second type of load is the worm-beater shaft and pulley assembly. This load will be added to the load due to the palm fruits in the barrel to be carried. All of these loads are summed to be the applied load. Hence, for the sake of safety and over loading, a factor of safety of 2 is taken. Thus, the maximum load or ultimate load on which the design is based is given as:

$Maximum\ or\ ultimate\ load = applied\ load\ X\ 2$ (6)

Consider

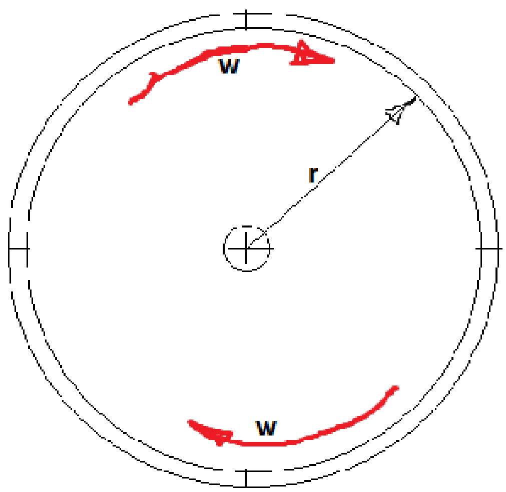


Fig 4. Descriptive Centrifugal Force Analysis

The force required for the shaft housing the worm and beaters(auger) to rotate the beaters

$F_B = Ma$ (7)

Where a m/sec^2 is acceleration

Centrifugal force (Fc) developed in the worm beater shaft in the barrel was deduced as given by:

$F_c = F_B$ (8)

Given that

$a = vr$ (9)

In a circular motion,

$v = \frac{N_1 \pi D_1}{60}$ (10)

$v = \omega r$ (11)

The centrifugal force F_c

$F_c = Ma = M\omega^2 r$ (12)

$\omega = \frac{2\pi N_1}{60}$ (13)

$F_c = M(2\pi N_1/60)^2 r$ (14)

iv. Torque Developed

The torque in the design is considered here on the basis of starting torque of the prime mover since most of the users are rural based and may not follow instruction on the use of the machine. Using the expression by Kurmi and Gupta (Khurmi RS et al, 2005) that torque (T) is equal to the product of force and radius, the value of the torque developed on the stirrer shaft was calculated using

$T = F_c r_2$(15)

Where Fc is the centrifugal force developed in the barrel and it is the maximum load to be carried by the prime mover, r_2 is the radius of stirrer shaft.

v. Power Requirement and Tension in the Belt

The power requirement of the digester was determined with the expression by Kurmi and Gupta (Khurmi RS et al, 2005) which states that power is the product of torque (T) and angular velocity(ω) as:

$P = T\omega$ (16)

$\omega = \frac{2\pi N_2}{60}$ (17)

$P = {2\pi N_2 T}/{60}$ (18)

Where P is the Power (Watt), N is the speed of shaft (rpm), T is the torque required to turn the shaft (Nm). Therefore, the prime mover was selected to give the required torque in the system.

vi. Tension In The V-Belt

The tension in the v-belt was deduced by applying Kurmi and Gupta (Khurmi RS et al, 2005) formula

$\frac{T_1}{T_2} = e^{\theta\mu}$(19)

Where T1 is the tension on the tight side of the belt, T2 is the tension on the slack side of the belt, μ is the coefficient of friction and θ is the angle of wrap in radians. But for v-belts, the maximum twisting moment on each of the shafts was determined using the relationship given by Khurmi and Gupta (2005), as;

$M_t = (T_i - T_j)\frac{D_2}{2}$(20)

The bending moments (*B.M.*) on each of the shafts were determined using the standard procedures as follows;

$T_c = mv^2$ (21)

$T_{max} = \sigma a$ (22)

$T_i = T_{max} - T_c$ (23)

Where Tc and Tmax are the centrifugal and maximum tension of the belts. The coefficient of friction, μ between the pulleys and the belts, mass per unit length, m maximum safe stress, σ and cross sectional area, a of the belts were obtained from standard tables cross sectional area, a of the belts were obtained from standard tables as 0.3, 0.108kg/m, 2.1N/mm2 and 81mm2 respectively (IS: 2494-1974-Khurmi and Gupta, 2005; Sharma and Aggarwal, 2006). The belt speeds, *v* for the motor and palm fruit digester drives were computed respectively from the following relations.

$$v = \frac{\pi N_2 D_2}{60} \quad (24)$$

$$\frac{N_1}{N_2} = \frac{D_2}{D_1} \quad (25)$$

N_1=Speed of driver or motor pulley
N_2 = the speed of driven pulleys consequently, the tension on slack side of each belt, *Tj* was determined using Equation. For the motor/auger and auger/cake breaker drives respectively

$$2.3 log_{10} \frac{T_i}{T_j} = \mu\theta cosec\beta \quad (26)$$

$$\alpha = Sin^{-1}\left(\frac{D_1 - D_2}{2x}\right) \quad (27)$$

$$\theta = \left(\frac{180 - 2\alpha}{180}\right)\pi \text{ radians} \quad (28)$$

v.Belt Design

The length of the belt was determined using the Kurmi and Gupta (Khurmi RS et al, 2005) formula below. The centre distances, *x* between the adjacent pulleys were determined using Equation 3 as 300mm and 150mm for the machine.

$$x = \frac{D_1 + D_2}{2} + D_1 \quad (29)$$

Where *D* 1 and *D*2 are diameters of driving and driven pulleys.

$$L = \frac{\pi(D_1 + D_2)}{2} + 2x + \frac{(D_1 - D_2)^2}{4x} \quad (30)$$

Where L is total length of the belt, D1 is the diameter of the driving pulley, D2 is the diameter of the driven pulley, and x is distance between the centers of the two pulleys.
From the table 4, the power requirement for the machine prime mover is 14.6kw; hence, tye C v-belt is selected belt from table 20.1 in (IS: 2494-1974-Khurmi and Gupta, 2005.pp728).

Table4. Dimensions of standard V-belt according to 2494-1974

Type of belt	Power rating in Kw	Minimum pitch diameter of pulley (D)mm	Top width (b) mm	Thickness(t) mm	Weight per meter length in newton
A	0.7- 3.5	75	13	8	1.08
B	2- 15	125	17	11	1.89
C	7.5- 75	200	22	14	3.43
D	20- 150	355	32	19	5.96
E	20 -350	500	38	23	-

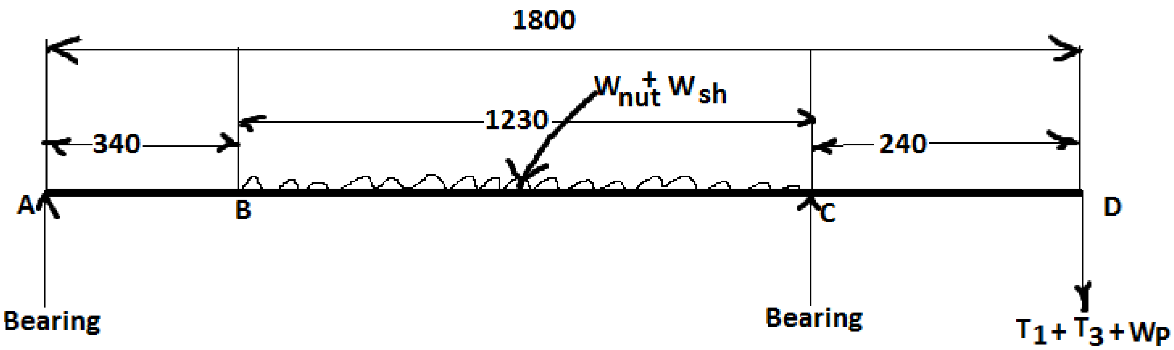
Culled from (IS: 2494-1974-Khurmi and Gupta, 2005 pp728)

vi. Determination of The Standard Length

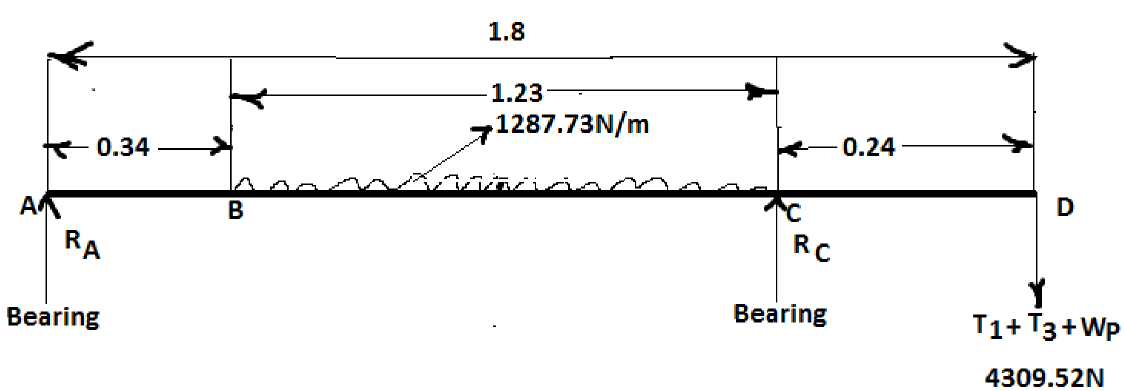
From table 20.3, (IS: 2494-1974-Khurmi and Gupta, 2005, pp729), the standard length for such belt, is based on the type. The calculated length of the belt is 1767.6643mm; hence, type C belt is of a standard length of 1783mm.

The number of belts required $= \frac{power\ to\ be\ transmitted}{power\ transmitted}$ (31)

vii. Shear Force and Bending Moment



a.



b.

fig 5. Loads on the shaft and reactions on the bearing

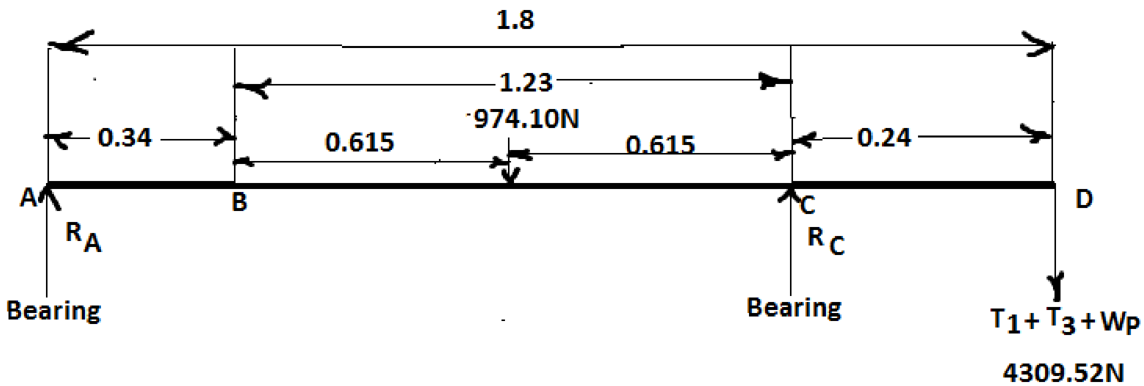


Fig 6. Equilibrium analysis

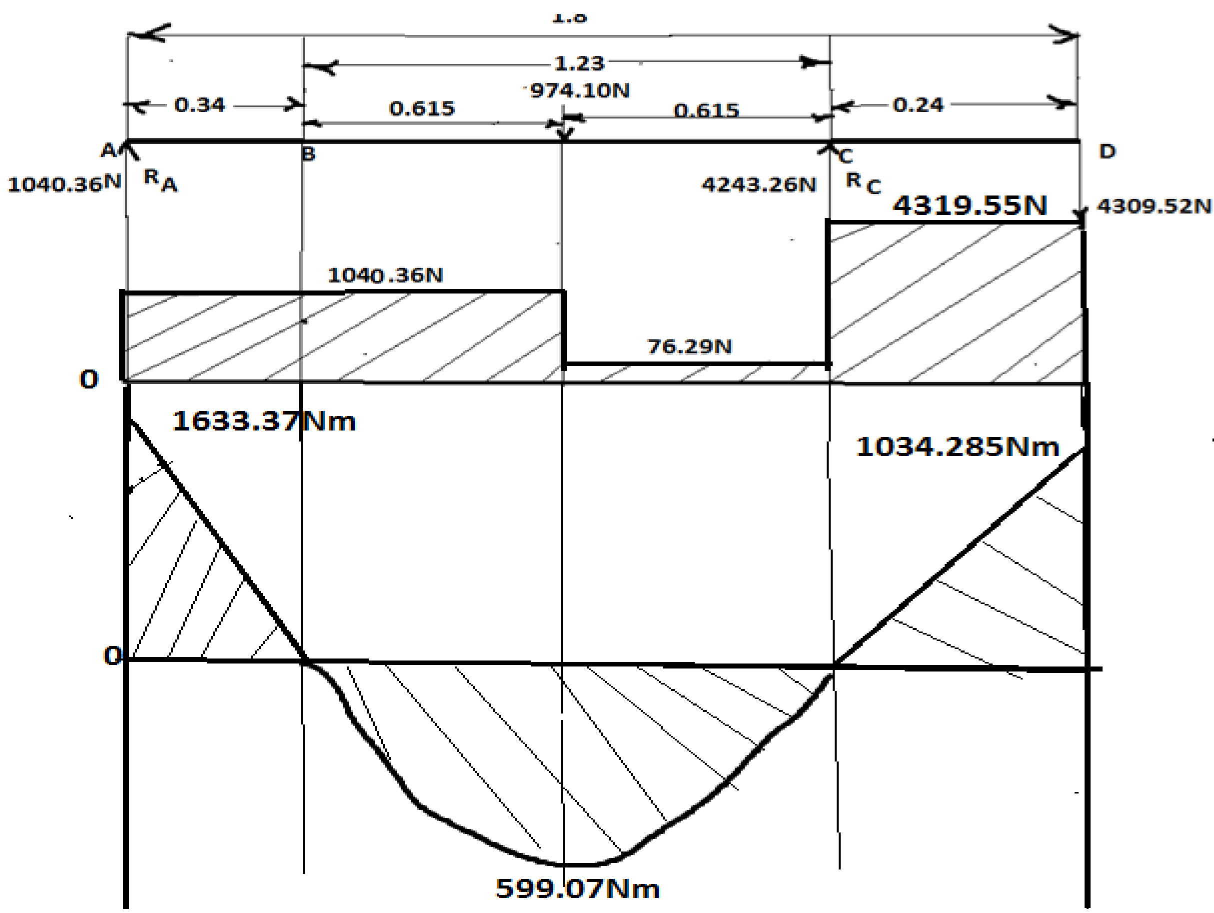


Fig 7. shear force and bending moment diagram

viii. Determination of shaft diameters:

The diameters, *d* of the Worm and beater shaft of this machine were determined using maximum stress relations given as

$$d = \left[\frac{16}{\pi\tau}((M_b K_b)^2 + (M_t K_t)^2)^{1/2}\right]^{1/3} \quad (32)$$

where:
τ =Allowable stress for steel shaft with provision for key ways = 42N/mm2
Mt = Maximum twisting moment on the shafts, Nmm.
Mb =Maximum bending moment on the shafts, Nmm.
Kb= Combined shock and fatigue factor for bending.
kt =Combined shock and fatigue factor for twisting.
Bending and twisting moments occur on shafts as a result of applied loads and belt tensions.

ix. Selection of Prime Move (Electric Motor):

The power required for the operation of this machine is the total sum of the power required to drive its units and the power required to overcome the drives friction. The power, *P* required in the palm fruits digestion respectively using Equation.

$$P = (T_i - T_j)v \quad (33)$$

Taking care of 10% possible power loss due to friction, the total power required to drive the developed palm fruits digesting machine was computed; therefore, a 20 HP electric motor was selected for the operation of this machine.

x. KEY WAY

The size of key is a function of the diameter of the shaft.

$$w = \frac{d}{4} \quad (34)$$
$$t = \frac{d}{6} \quad (35)$$

w and t are the width and thickness of the key where the key length is governed by the thickness of the pulley or hob of the pulley.

xi. Bearing Selection

The dynamic equivalent radial load (*W*) for radial and angular contact bearings, except the filling slot types, under combined constant radial load (*W*R) and constant axial or thrust load (*W*A) is given by:

$$W = X.V.WR + Y.WA \quad (36)$$

where,
V= *A* rotation factor,
V= 1, for all types of bearings when the inner race is rotating,
V= 1, for self-aligning bearings when inner race is stationary,
V= 1.2, for all types of bearings except self-aligning, when inner race is stationary.
The values of radial load factor (*X*) and axial or thrust load factor (*Y*) for the dynamically loaded bearings
The bearing that has the higher load value governs the design selection R_C=4243.26 N

$W_R=2R_C$ (37)
For X=1 AND Y=0

W_A=load on the shaft point load=974.10N

$\frac{W_A}{W_R} \le e$ =0.11478 (38)

From Table 27.4, deep groove ball bearing is chosen (Khurmi & Gupta, 2005: 1008). The bearing type based on ratio of w_a and w_r calculated considering the shaft diameter d=60mm.
From table 27.1, bearing number 312 with 60 bore, 130 outside diameter, and 31width are chosen.

xii. Life of the Bearing

The life rating of the bearing is calculated:

$$L = \left(\frac{C}{W}\right)^{1/K} * 10^6\ Revolutions \qquad (39)$$

C=64KN from table27.6 for single deep grooved ball bearing

K=3 for all ball bearing

From table 27.3 (IS: 2494-1974-Khurmi and Gupta, 2005: 1006), the life of the bearing is determined and the Machines required to work with high degree of reliability 24 hours per day. Therefore, a deep grooved ball bearing with bearing number 312. 60bore, 130 outer diameters and 31 widths is chosen for the design (IS: 2494-1974-Khurmi and Gupta, 2005: 1000). Bearing number is based on shaft diameter. From table 27.6, selection of the dynamic load is based on bearing number, (IS: 2494-1974-Khurmi and Gupta, 2005, pp 1013).

xiii. Stresses Built On the Cylindrical Barrel

The stresses built up in the barrel are

i) Hoop stress
ii) Longitudinal stress

From the relation connecting the hoop (circumferential) and longitudinal stresses, it is seen that hoop stress is greater than longitudinal stress.

Circumferential or hoop stress

$$\boldsymbol{\sigma_1 = \frac{Pr}{t}} \quad \textbf{(40)}$$

$$\boldsymbol{\sigma_2 = \frac{Pr}{2t}} \quad \textbf{(41)}$$

$$\boldsymbol{\sigma_1 > \sigma_2}$$

Note Area,

$$A = \frac{\pi D^2}{4} \quad \textbf{(42)}$$

δt = 186× 10^6N/mm^2 for stainless steel

But,

$P = F/A$ **(43)**

The thickness base on the value of the tensile strength of a stainless steel material is so minimum. Hence, the chosen barrel has a thickness of 10mm which shows it can withstand the pressure built up in the internal walls without tears. The essence of the hoop stress in the design is to know the maximum stress and thickness that can be used in the barrel or vessel.

Xv. Spring Selection

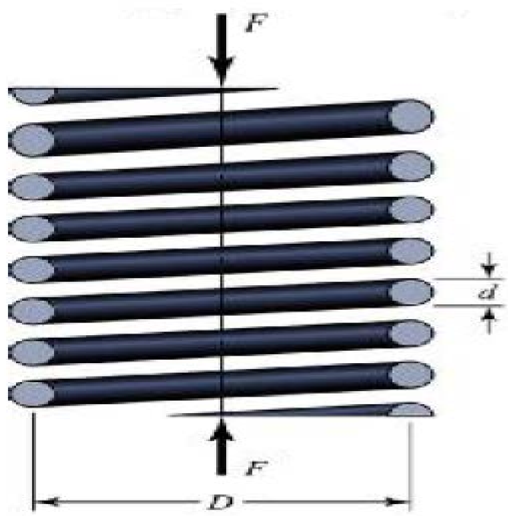


Fig spring loading

The two important parameters are

i. The mean coil diameter D
ii. Wire diameter d

$T = \frac{FD}{2}$ (44)

Spring index

$C = \frac{D}{d}$ (45)

Table 5..Design calculation results table

s/n	parameter	relation	value	unit
Volume of the hopper				
1	Pyramidal frustum	Eqtn 1	0.025138	m^3

2	Cuboid section	Eqtn 2	0.024576	m^3
3	Total hopper	Eqtn 3	0.049714	m^3
Volume of barel				
4	Internal diameter	From design	0.3800	m
5	length	From design	1.2450	m
6	Volume of barel	Eqtn 4	0.1429	m^3
Volume of hopper per volume of barrel				
7	Hopper per barrel		2.9 ≈ 3	hoppers
Power determination				
(Load) Weight to be carried by the prime mover				
8	Density of conditioned palm fruit	Ref: Renel M .Alucilja et al 2013	708.66	Kg/m^3
9	Volume of barrel	Eqtn 4	0.1429	m^3
10	mass of palm fruit in a filled barrel	Eqtn 5	101.267514	Kg
11	weight of palm fruit in a filled barrel	Eqtion 5	993.4343123	N
12	Weight of worm, beaters & shaft	From weighing machine	15 × 9.81 147.15	N
13	Total load	Eqtn 5	1,287.734312	N
14	Factor of safety	Assumed	2	-
15	Ultimate load	Eqtn 6	2575.468624	N
16	Diameter of driver pulley		0.3	m
17	Torque	Eqtn 15	386.32	Nm
Power requirement of the machine				
18	Speed of driver pulley(N_2)	Desired speed ratio	720	RPM
19	Power capacity of machine	Eqtn .18	29.13	kw
Size of shaft determinations				
20	Shaft length (l)	Eqtn 30	1.8	m
21	Center distance (x)	Eqtn 29	5.25	m
22	$\propto$	Eqtn 27	8.21^o	degree
23	θ	Eqtn 28	2.8554	radians
24	μ	Khumi & gupta	0.3	
25	Tension on slack side of belt T_2	Eqtn 26	F	N
26	Tension on tight side of belt T_1	Eqtn 26	2798.63	N
27	Maximum bending moment			Nm
28	K_t	Khumi & gupta	1.5	
29	K_b	Khumi & gupta	1	
30	Shaft diameter ,d	Eqtn 32	0.059 ≈ 60	m
Key ways and key size				
31	length		0.027	m
32	Width	Eqtn 34	0.015	m
33	thickness	Eqtn 35	0.010	m

		Bearing selection		
34	Dynamic equivalent load W	Eqtn 36	8486.52	N
35	Radial load W_r	Eqtn 37	8486.52	N
36	load on shaft W_a		974.10	N
	$\frac{W_A}{W_R} \leq e$	Eqtn 38	0.11478	
37	C	Khumi & gupta	64	KN
38	Bearing life,L		1.96	$10^6 revolution$
		Belt size and rating		
39	Length of belt	Eqtn 30	1783	mm
40	Belt type	Khumi & gupta	C	
41	Numbers of belt	Eqtn 31	1.999 ≈ 2	belts
		Electric motor requirement		
42	Speed N_1	Eqtn 25	1440	RPM
43	Belt velocityv_1	Eqtn 24	11.3112	m/sec
44	Power	Eqtn 33	14.566	kw
45	Power of the motor	X 0.746	19.53 = 20	Hp
46	Torque on motor shaft	Eqtn 18	96.58	Nm
47	Hoop Stress On The Barel σ	Eqtn 40	431,401.84	N/m^2
48	Cross sectional Area A	Eqtn 42	0.11343	m^2
49	Pressure P	Eqtn 43	22,705.36	N/m^2
50	thickness	Eqtn 40	10	mm
		Spring design and selection		
51	Wire diameter .dw		0.02	m
52	Major diameter ,D		0.075	m

4. Machine Models

The continuous cooked palm fruit nuts digester is designed bearing maintenance and operation for rural users and contemporary entrepreneurs in mind. The models of the machine are presented as follows:

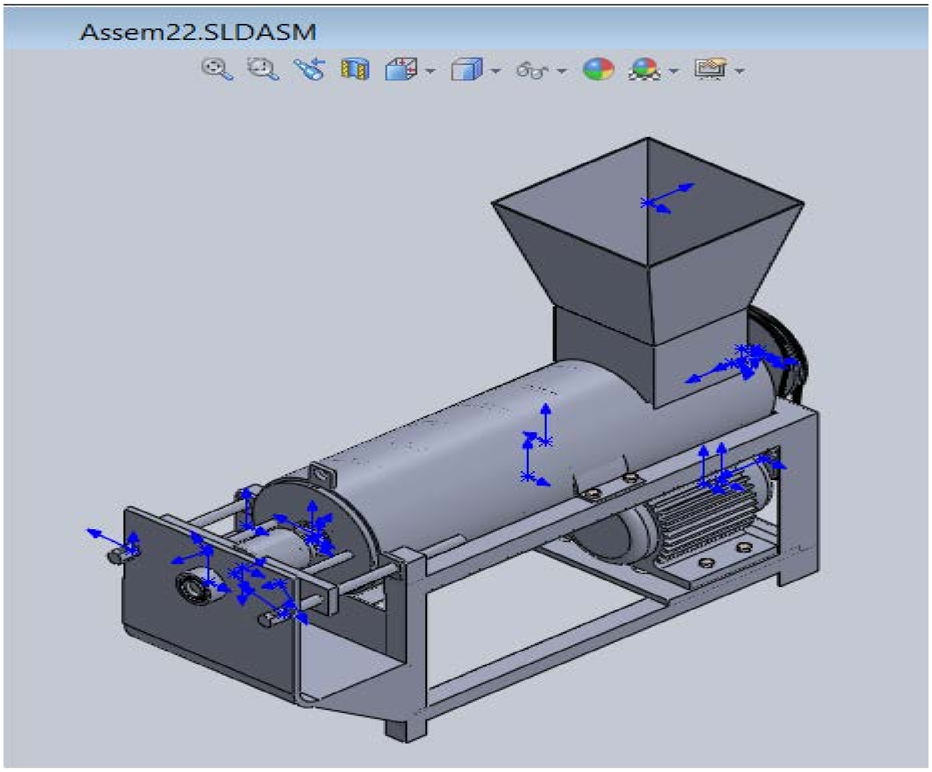


Fig 8. 3-D model of The continuous palm oil fruits digesting machine

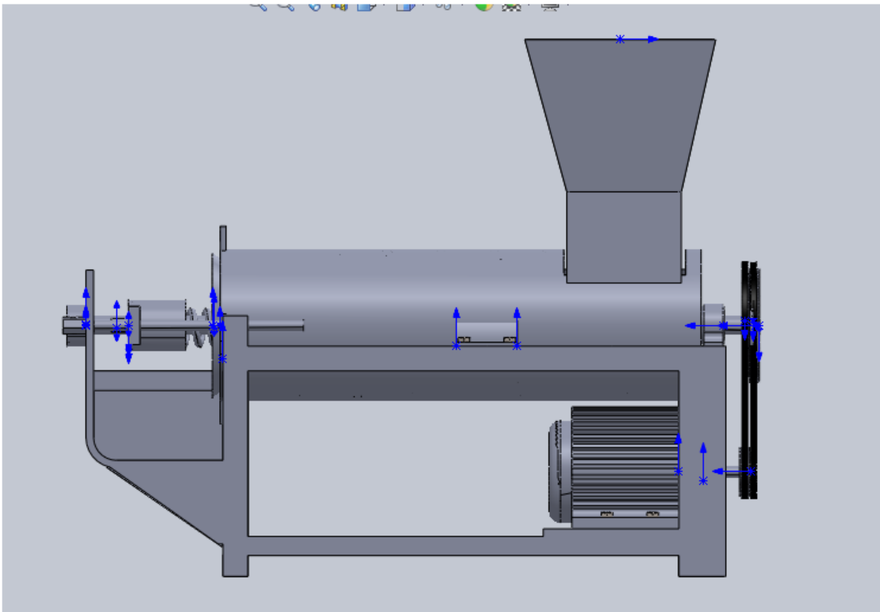


Fig 9. Side View of the Continuous Palm Oil Fruits Digesting Machine

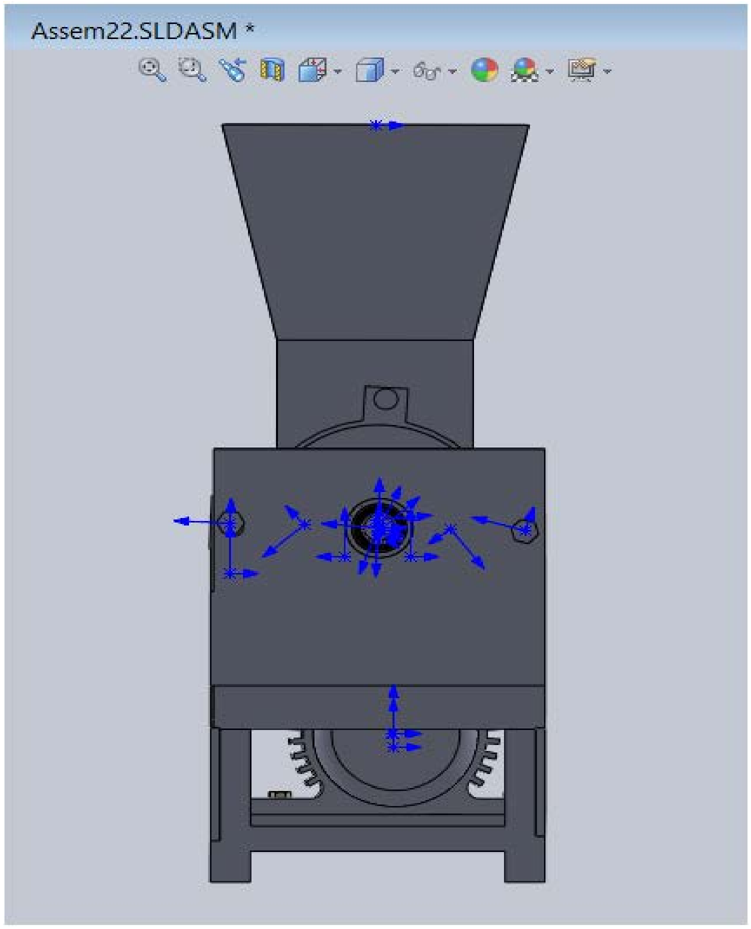


Fig 10. End View of the Continuous Palm Oil Fruits Digesting Machine

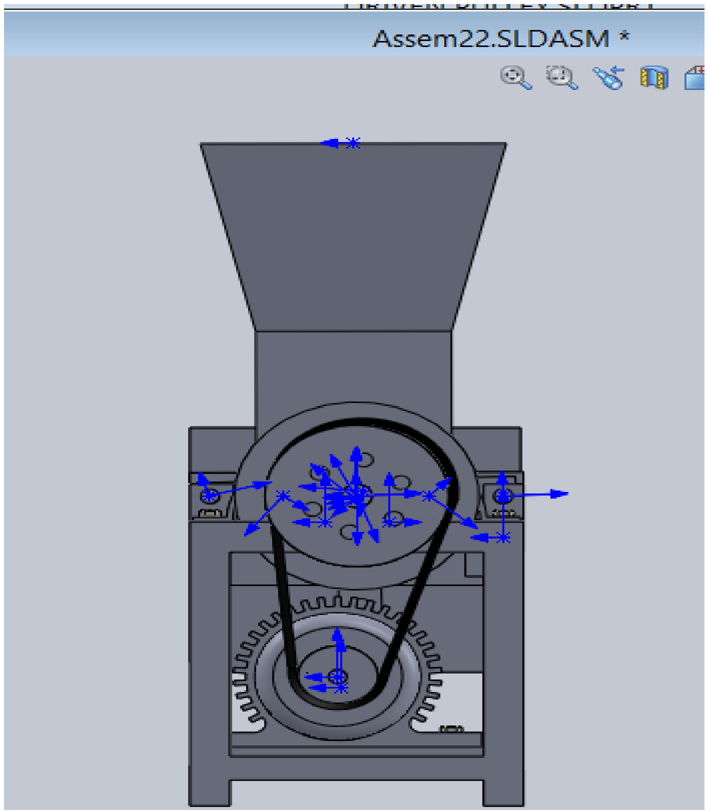


Fig11. Front View of the Continuous Palm Oil Fruits Digesting Machine

5. Discussion and Recommendations

Table 4 simply shows the values of the parameters used for this design work, and at a glance, creates path for the reference equations and formulas. This work is done to alleviate the problem of material handling i.e. palm fruit handling during batch processing and encourage the palm oil processing entrepreneurs and industries to improve production rate (Muthurajah RN, 2002). Besides, this economic impact assessment of the machine is designed to create job for local fabricators base on the local content policy to discourage importation and boost the country's GDP. It is therefore recommended that fabricators should be trained on the development of the machine and the improvision of some standard parts of this design (Sharma PC et al, 2006).

6. Conclusion

A continuous cooked palm fruit nuts digester is designed and developed with a CAE knowledge based software SOLIDWORKS and provisions are made for maintenance with the use of temporary joint fasteners and all standard parts were carefully selected with design bases. The machine is designed with a barrel capacity of 0.1429 m^3 and an ultimate load of 2.567KN. A worm–beater shaft at a 720rpm speed, 0.14m pitch transporting and apparent volume of 0.01588m^3 of palm fruits undergoing digestion per revolution (Renel M et al, 2013).

Future Work

It is expected that this machine will be fabricated and tested for performance and efficiency.

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