International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 2 Issue 3, March - 2013

Development And Performance Evaluation Of An African Breadfruit Shredder Machine

Amiebenomo, S.O[1], Omorodion I.I[3] and Igbesi F.C[2]
[1]Dept. of Mechanical Engg., Ambrose Alli University, PMB 14, Ekpoma- Nigeria;
[2]Dept. of Production Engg.,Ambrose Alli University,PMB14,Ekpoma-Nigeria
[3]Delta State Polytechnic, Otefe Oghara-Nigeria

ABSTRACT

Shredding of breadfruit is an essential aspect in the processing of the fruit to facilitate the fasting drying and thus preservation of the breadfruit. A motor power breadfruit shredding machine was designed, constructed and tested. Test results reveal a maximum mechanical efficiency of 94.8% and shredding efficiency of 64.5%without the use of a press weight, while results employing the press weight showed a maximum shredding efficiency of 78.1% and mechanical efficiency of 83.2% respectively. There were significant effect on the time to shred a mass of breadfruit applying a press weight on the machine at 5% level of significance and the absence of any interaction between the mass of breadfruit and the press weight on the shredding time. Sustainability of shredding device was achieved in terms ease of maintenance and repair due to minimal moving parts and complexity.

INTRODUCTION

The African breadfruit tree (*Treculia africana* Decn var africana) is native to many tropical countries like West Indies, Ghana, Sierra Leone, Nigeria and Jamaica1. Its seed, commonly called “afon” and “ukwa” by the Yoruba and Igbos of Nigeria, is popular as a traditional food item. Breadfruit is a fruit tree that is propagated with the root cuttings and the average age of bearing first crop is between 4 to 6 years (Amusa et al., 2002). It produces its fruit up to three times in a year and the number of fruits produced is very high. The fruit has been described as an important staple food of a high economic value (Soetjipto and Lubis, 1981). The breadfruit pulps are made into various dishes; it can be pounded, fried, boiled or mashed to make porridge; it can also be processed into flour and used in bread and biscuit making (Amusa et al., 2002). Breadfruit has also been reported to be rich in fat, ash, fibre and protein (Ragone, 1997). Despite the importance of this fruit, its production is faced with several problems including short shelf life and poor yield due to diseases (Cook, 1975). The fruits are thus utilized in Nigeria within 5 days of harvesting because of their short shelf lives.

Local processing entail manual means employing a basic hand grater and shredded by hand or sliced with a small knife. This is very time consuming, unproductive, and labour intensive. Past studies had focused on nutritional composition of the fruit and mechanical processing in terms of shelling and dehulling of the fruit. As,example, Arowora et al (2011)looked at the nutritional effect of storing breadfruit in powder form, while Akanbi et al(2009) determine the proximate, functional and pasting properties of breadfruit
starch. The works: Nwigbo et al(2008) and Etomaihe(2010) focused on development and design of machines for shelling and dehulling of the breadfruit. It can be observed that although several studies have been conducted on the breadfruit processing and the nutritional effects of storage in various modes, there is a dearth of studies focusing on increasing its shelf live in terms of shredding by employing mechanized processing machines.Thus the principal aim of this study is to design,fabricate and determined the performance of the develop machine in meeting its design objectives.

DESIGN CONCEPTS AND OPERATING PRINCIPLES

The Isometric view and orthographic front view of the motor power shredding machine is shown in fig 1and fig 2.below .The main components are the machine frame (column and beam), transmission shaft, shredder blade, feeder tube and centre divider, prime mover and motor base. Preliminary design analysis highlighted factors that affect the performance characteristics of the machine to include the following; Speed of rotation of the disc, diameter of the feeder tube, Press weight applied on the bread fruit and number of perforated holes on the shredding disc. The effect of the press weight was considered, while other parameters listed were made constant. The shredding action takes place when the breadfruit is placed in each half of the partitioned cylinder and the main shaft rotates along with the attached shredder disc power by the electric motor. The weight of the fruit forces it against the shredding disc, while the centre divider prevents the fruit from rotating along with the disc. The portion of the fruit which comes in contact the shredding disc is continuously shredded away as the disc rotates. The shredded slices pass through the perforated holes on the disc and drop into a receptacle below it

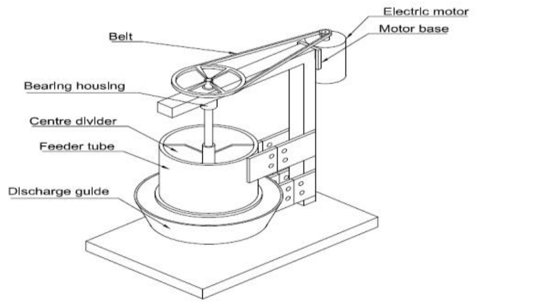

FIGURE 1.ISOMETRIC VIEW OF BREAD FRUIT MACHINE

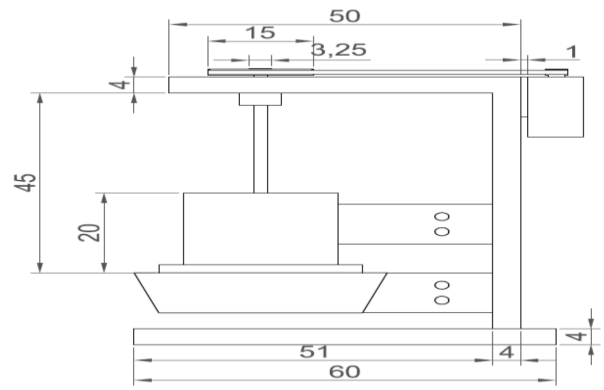

FIGURE2.ORTHOGRAPRAPHIC VIEW OF BREADFRIUT MACHINE

DESIGN SPECIFICATIONS

To facilitate accurate design and delineation of the bread fruit machine, the following specification will be followed.

i. Maximum bread-fruit size to be handled – 2kg
ii. Shredding force – 37N(turning force measured with spring scale).
iii. Average Shredding speed – 300 rpm
iv. Operating pulley diameters: small – 5cm.
vi. Average density of bread fruit – 0.66g/cm^3 (Lawal 0.S 2005)
vii. Drive pulley centre distance = 32cm
viii Drive belt cross-section; a= 12.5cm,b =6.5cm,c=8.5cm

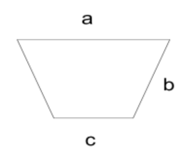

Figure 3: Belt cross section

DESIGN ANALYSIS AND CALCULATION OF MAJOR COMPONENTS

Preliminary analysis and calculations are carried out to determine the critical and actual parameter needed for construction and selection of the components of the bread fruit machine; some of these parameters are

i. Power need for shredding (for motor selection)
ii. Volume of bread fruit to be shredded (for determination of feeder tube size)
iii. Pulley diameter/size
iv. No of belts required (determines pulley groove required either single or double)

Determination of the Power Needed For Shredding

In shredding the fruit, the shredding disc, need to overcome the resistance of the bread fruit internal cohesive force from the specification given, the shredding force has been determined to be 37N. Now considering the disc and shaft arrangement; the force (f) sets off tensional resistance, which must be overcome by the torque Treq.

Thus; $T_{req} = f \text{ x } r$ ………………….(1)

Where f = Shredding force, r = radius of shredding disc = 0.11m and T_{req} is found to be 4.08N

Now, $power = T_{req}\, x\omega$ …………. (2)

But $\omega = \frac{2\pi N}{60} = \frac{2\pi\ X300}{60} = 31.4 rad/s$

Therefore power = 4.08 x 31.4 = 127.8w

Determination of Volume of Breadfruit Shredded

The volume of breadfruit is required in other to calculate the volume of the feeder tube. Base on the specification provided; maximum mass of breadfruit (2kg) and average density of breadfruit 0.66g/cm^3

Given that $Density = {}^{mass}/_{volume}$ substituting values, we have that the volume of breadfruit is 3030.3cm^3

Determination of The Driven Pulley Diameter

From the initial specifications given; Average shredding speed (N) = 300 rpm, Motor speed (n) = 1500 rpm, Motor pulley diameter (d) = 5cm .Therefore the relationship between the speed and pulley diameters is given as;

D x N = dxn ……………(3)

Hence, the driven pulley diameter (D) was found to be 25cm

Belt Analysis and Design

The belt selected is based on the readily available belt type and material (v –belt, rubbersized fabrics)

Accordingly the following initial parameters for further design of the belt are provided; Transmitted power = 127.8w

Driving speed = 1500rpm ,Pulley center distance = 32cm.

Required parameters are the Belt length (L_B) and Number of belt(s) (n_B)

Belt Length (L_B)

The general expression for the length of belt is given as

$$L_B = \pi(r + R) + \frac{(R-r)^2}{C} \qquad \text{………………..(4)}$$

Where r= radius of smaller pulley (2.5cm), R = radius of larger pulley(12.5cm),C=Centre distance of the two pulleys (32cm)

So, L_B=114.23cm

Determination of Number of Belts Required

The number of belts (n_b) needed to transmit the needed power is given as:

$$n_b = \frac{F_t}{k\,A} \qquad \text{………………….(5)}$$

Where, F_t =maximum load to be transmitted; K=effective initial stress = 2.5 for the selected belt material(shingley2009 and shaum series2003) ;A=cross – sectional area of belts.

From the given belt dimensions, A is found to be 80.75mm^2

$$F_t = \frac{2T_1 X 1000}{d_S} \qquad \text{…………………..(6)}$$

where d_s = diameter of smaller pulley (50mm),T_1 = Transmitted torque .

but $T_1 = {}^{power}\!/_{angular\ speed(\omega)}$ ……………..(7)

where power is 127.8w,while ω is found to be 25.1rad/s,so transmitted torque T_1 is 5Nm,employing this value into eqn(6),F_t =200N.

Therefore, considering eqn(5) number of belts(n_b) required is found to be one (1)

DESIGN OF THE FEEDER TUBE

Material selected: 2mm thick stainless steel plate. The feeder is analyzed for the minimum volume needed to hold the maximum weight of the breadfruit. For a maximum weight of 2kg;
Average volume (section) = 3030.3cm^3
For a breadfruit of volume 3030. 3cm^3, the feeder tube would be 1.3 x this volume

Thus let V_T = 3939.39cm^3

$Hence, V_T = \pi \frac{D^2}{4} xH$ …………… (8)

Where VT = Volume of feeder tube; D = diameter of the feeder tube = 22cm (Equivalent to the diameter of shredding disc); H = Height of the feeder tube; determined as 10.37cm

Total Surface of the Feeder Tuber

$Total\ area\ (A_r) = \pi DH$ .... (9)

This is found to be 716.3cm^2, while the area of centre divider (Ac) =93.33cm^2
For 2 centre divider = 2x93.33=186.66cm^3
The total area of material = 716.3+186.66 =902.96cm^2

MAIN SHAFT DESIGN

The main shaft is analyzed for: mechanical properties and dimensional analysis in terms of length and diameter. The material specification is stainless steel shaft. The shaft design analysis seeks to determine the appropriate material for the shaft and also appropriate dimension of the shaft to ensure that the shaft is capable of withstanding the working load on it.

The main shaft carries the shredding disc and holder, and the turning force is transmitted for the machine pulley (Large pulley) to the shredding disc; the forces acting on the shaft include:

(i).The tensile force due to the weight of the suspended disc and hold.
(ii).The twisting force due to the transmission of torque to the disc for shredding purpose.
(iii).The bending force extended by the belt tension on the pulley.
The tensile stress due to the weights is negligibly small thus the tensile force can be assumed to be zero. Also the shaft flexural rigidity is capable of resisting the bending stress exerted by the belt and thus the effect of bending is neglected.

Thus the shaft will be analyzed on the basis of twisting stress exerted on the shaft.

From Fig 4; length of shaft (L) = (H+h) – C .... (10)

Where H =Height of the frame = 63cm,

h=Height above the frame require to accommodate the large pulley,

C=Distance from the base to the shredding disc

Thus L = (63+5) – 18 = 50cm

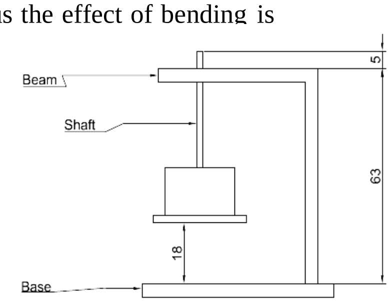

FIGURE 4.Schematic diagram of main frame and shaft

Determination of the Shaft Diameter

Shaft is subjected to stress ,the torque acting is related to the stress as;

$\frac{T}{J} = \frac{\tau}{r}$ .... (11)

Where, T=Twisting moment (or torque) acting upon the shaft, J=Polar moment inertia of the shaft about axis of rotation, τ =Torsional shear stress, r=Distance from neutral axis to the outer most fibre = $d/_2$where d is the diameter of the shaft

For a circular shaft,

$J = \frac{\pi d^4}{32}$ ....(12)

Substituting this eqn(12) into eqn(11),the following expression is obtained,

$T = \frac{\pi\ \tau d^3}{16}$ ....(13)

Based on this expression with τ=48N/mm(Shigley2011),diameter of shaft was found to be 17.5mm

MACHINE FRAME

ISSN: 2278-0181
Vol. 2 Issue 3, March - 2013

Fig 5. Machine Frame

The machine frame, which consist of the beam, the column and the base, carries and holds the functional components of the shredding machine; as a result, it is subjected to various stresses as imposed by the components attached to it.

The frame is designed to withstand the forces on it

and for optimum configuration for convenient use of machine

The frame analysis entails material selection of 4mm thick mild steel channel.

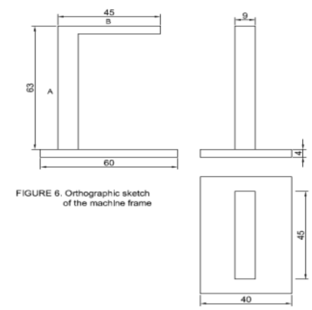

FIGURE 6. Orthographic sketch of the machine frame

(i)**To determine the total length of channe**l,

From fig 6, the total length (T_L) of the channel required for the frame is given as

T_L = A + B =108cm

Where, A=63cm,B=45cm

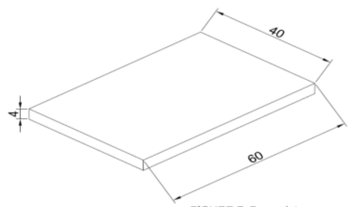

FIGURE 7: Base plate

(ii)**To determine the total area of base plate**

From fig 7;$P_{T=}$ 60x40 + 2 (4 x 60) =2880cm^2

(iii) **Stress analysis on frame**

Consider a cross section of the beam (fig 8)

FIGURE 8: A cross section of the beam.

The analysis on the frame is to determine if the stress induced on the beam is less,

equal or greater than the allowable stress (σ_{All}) for the material of the beam thus,

consider the section of the beam under load.

The basic relation for analyzing the bending moment and bending stress in straight beams is given as;

$\frac{M}{I} = \frac{\sigma}{Y}$ .... (14)

Where, M = Bending moment acting at the section, σ = Bending stress,Y =

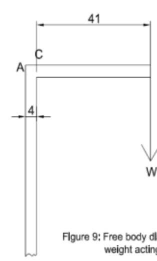

Figure 9: Free body diagram of weight acting on beam.

Thus considering the section of the beam under load(W) from fig 9,

maximum bending moment B_{max} at section C, is given by

Bmax = W x CB .... (15)

Where,CB=41cm
W=
$\sum mass\ of\ shaft, mass\ of\ pulley. mass\ of\ press\ weight, mass\ of\ breadfruit\ andmass\ of\ bearing\ and\ housing$

Therefore force exerted by the masses is 132.44N, hence Bmax= 54.3Nm The section modulus Z can be determined from the configuration of the beam section. Considering the section; the moment of inertia about the section X-x

$I_{XX} = \frac{Bh^3 - b(h-t)^3 + ah_1{}^3}{3}$ .... (16)

$h_1 = H - h = \frac{aH + bt}{aH^2 + bt^2}$ .... (17)

$Z = 2I_{XX}\left[\frac{aH+bt}{aH^2+bt^2}\right]$ .... (18)

we have that: a=8mm,H=40mm,B=90mm,t=4mm,b=82mm,and h_1=29.11mm.

Substituting this values into eqn(16) and eqn(17) we obtain I_{xx}=60397.72mm and Z=5546.73mm^3

Thus $\sigma_{max} = \frac{B_{MAX}}{Z} = \frac{54300}{5546.73} = 9.79N/mm$ .... (19)

Thus since the allowable stress σ all for the material (420N/mm^2) is greater than the stress induce on the beam by the bending stress, then the beam will not fail by shear.

ANALYSIS AND DESIGN OF THE BLADE MOUNT

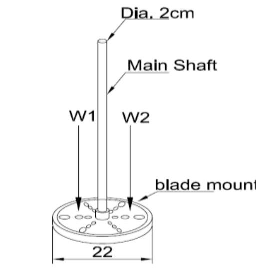

Figure 10: Blade mount

Considering blade mount loading figure 9

The force acting on the blade mount is due to the weight of the

bread fruit and the press weight

Given that: weight of bread fruit (W_B) = 2kg; Press weight (wp) = 6kg ; Total weight = 8kg

Therefore, force on the blade mount = 8x9.81=78.48N

In analyzing the effect of the force on the blade mount, we consider the free body diagram of the blade mount, under the effect of the applied load. We assume the load on the mount is concentrated and acts through the mid-point of the mount on either side ,that is (W_1=W_2) as shown in the fig 9. The effect of the forces is to set up bending moments across the blade mount with a maximum force acting at the point R (Fig 3.14).

The blade mount will be analyzed based on the basic relation relating the bending stress to the bending moment,

$\sigma_{max} = \frac{M_b}{Z}$ .... (20)

Where σ_{max} =Bending stress, M_b =Bending moment acting, Z=Section modulus

Thus, M_b =wL .... (21)

Given that L=22cm; w= 78.48N, hence M_b =17260Nmm

Now considering a cross-section of the blade mount, we assume the X- section to be square.

Therefore: $Z = \frac{hb^2}{6}$ for square section .... (22)

Inputting the eqn (22) into eqn (20), we obtain the following expression below,

$\sigma_{max} = \frac{6M_b}{b^3}$ .... (23)

Since b= h, hence for the blade mount material σ_{max} =150N/mm^2 (standard guage Iron),b was found to be 8.84mm

Thus the minimum section thickness must not be less than 12.5mm

MACHINE PERFORMANCE EVALUATION

The performance analysis on the machine was based

On the determination of the effect of the press weight on the

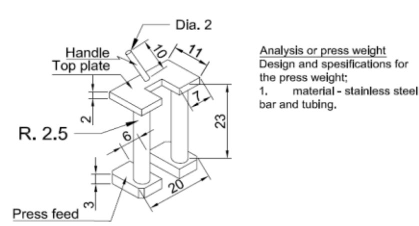

FIGURE11. Isometric view of Press Weight

ISSN: 2278-0181
Vol. 2 Issue 3, March - 2013

Shredding rate, the mechanical and shredding efficiency of the machine

TEST MATERIALS

In carrying out the test, 4 pieces of the bread fruit of varying mass were obtained.

A stop watch, weighing balance, volt-meter and Ammeter were also provided.

PROCEDURE

1. The bread fruit were tagged and weighed before being cut into two groups A and B for the experiment.
2. The fruits in group A were shredded without the press-weight while those of group B were shredded with the press-weight in use.
3. The respective time to completely shred a bread fruit was measured with the stopwatch, also the voltage and current across the power supply were also measured and tabulated.

Table 1; SHREDDING EXPERIMENTS

S/N	Weight of Bread fruit (g)	Shredding time without press weight (second)	Shredding time with press weight (second)	Voltage (volts)	Current output without press weight	Current output with pressweight	Shredded breadfruit without press weight(g)	Shredded breadfruit with press weight(g)
1	500	80	50	220	0.54	0.7	350	400
2	300	60	10	220	0.59	0.66	180	220
3	450	70	30	220	0.7	0.75	310	350
4	350	60	24	220	0.62	0.68	190	280

RESULT EVALUATION

(i) Power consumption without the press-weight

Average Current recorded (Table 1) $Av_A = \frac{0.54+0.59+0.7+0.62}{4} = 0.6125A$

Given that voltage supply = 220v,therefore, Power consumed(P_c) = 0.6125 x 220= 134.75W

(ii)Power consumption with the press-weight

Similarly from Table 1; Average current recorded = $\frac{0.7+0.66+0.75+0.68}{4} = 0.6975A$

For voltage of 220v; Power consumed (P_c) = 0.6975 x 220 = 153.45W

Mechanical efficiency of the machine

This refers to the percentage of the mechanical and electrical losses in the machine to the power output of the machine

The mechanical efficiency can be determined as the ratio of calculated theoretical shredding power(P_s) to perform the shredding and actual power consumed (P_c**),** needed for shredding

Given that the calculated (theoretical) shredding power, P_s=127.8W, mechanical efficiency obtained without the use of a press weight is

$$\cap_m = \frac{127.8}{\text{electric power}} = \frac{127.8}{134.75} = 94.8\% \qquad ....(24)$$

Mechanical efficiency with press weight

$$\cap_m = \frac{127.8}{153.45} = 83.2\% \qquad ....(25)$$

Shredding Efficiency ($\cap_s$)

The shredding efficiency was calculated considering mass of breadfruit before and after shredding considering the use of the press weight.therefore from table 1.

Average mass of breadfruit before shredding =400g

Average mass of breadfruit after shredding(with press weight) = 312.5

Average mass of breadfruit after shredding(without press weight) =257.5

$Shredding\ efficiency(\cap s_{press\ weig\,ht}) = \frac{312.5\ X100}{400} = 78.1\%$.....................(26)

While the Shredding efficiency without press weight was found to 64.5%.

An important component of the machine the press weight was developed and generally it was found to have improve the time to shred a given mass of breadfruit, which invariably was found to improve the shredding efficiency of the machine, as seen in the operational data obtained in the foregoing results obtained .The test procedures is summarized in an analysis of variance table as outputted from Design Expert below.

Response 1 TIME TO SHRED

ANOVA for selected factorial model

Analysis of variance table [Partial sum of squares - Type III]

Source	Sum of Squares	df	Mean Square	F Value	p-value Prob > F	
Model	3042.00	1	3042.00	16.56	0.0066	significant
B-PRESS WEIGHT	*3042.00*	*1*	*3042.00*	*16.56*	*0.0066*	
Residual	1102.00	6	183.67			
Cor Total	4144.00	7				

Final Equation in Terms of Coded Factors:

TIME TO SHRED = +48.00 +19.50* B

Response 1 TIME TO SHRED

ANOVA for selected factorial model

Analysis of variance table [Classical sum of squares - Type II]

Source	Sum of Squares	df	Mean Square	F Value	p-value Prob > F	
Model	536.00	3	178.67	0.20	0.8927	not significant
A-MASS	*536.00*	*3*	*178.67*	*0.20*	*0.8927*	
Residual	3608.00	4	902.00			
Cor Total	4144.00	7				

RESULTS AND DISCUSSION

The data obtained from the test has shown that mechanical efficiency and shredding efficiency obtained were effected by the application of a press weight, whose function is to ensure that breadfruit during the process of shredding has effective and consistent contact with the blade of the shredder. Results of the test reveal that shredding efficiency(78.1%) is obtained by use of the press weight and while decreasing the mechanical efficiency of the machine by 83.2%.

The press weight being a new product design was evaluated to study its effect on the shredding time by employing a two stage nested design, with press weight nested under various mass of breadfruit employed in the test.The linear statistical model for the two- stage nested design is

$$Y_{ijK} = \mu + \tau_i + B_{j(i)} + \epsilon_{(ij)k} \begin{bmatrix} i = 1,2, \dots \dots \dots \dots a \\ j = 1,2, \dots \dots \dots \dots . b \\ K = 1,2, \dots \dots \dots \dots . n \end{bmatrix} \qquad \dots \dots \dots \dots \dots \dots ..(27) Montgomery(2001)$$

That is,there are a levels of factor A,(mass),b levels of factor B(press weight) nested under each level of A, and n replicates.

Examining the P values of the Anova table at 0.05 significance level, we would conclude that there is no significant effect on shredding time due to the mass of breadfruit, but the time to shred due to the press weight for various mass of breadfruit does differ significantly. From Fig 12 the time to shred a mass of breadfruit employing a press weight is found to decrease consistently for various masses of breadfruit when compare to shredding time obtained without the use of the press weight. Employment of a diagnostic checked (residual analysis) to established if the variability within the press weight in terms of the time to shred is the same for the various mass of breadfruit was conducted. The plot of residuals against mass reveals that the spread of the residuals is about the same for all four mass of breadfruit. Hence we conclude that the press weight variability in the time to shred is about the same four all mass of breadfruit. Fig 13.Also Fig 14 reflects the lack interaction between the mass of the breadfruit and the press weight in determining the time to shred.

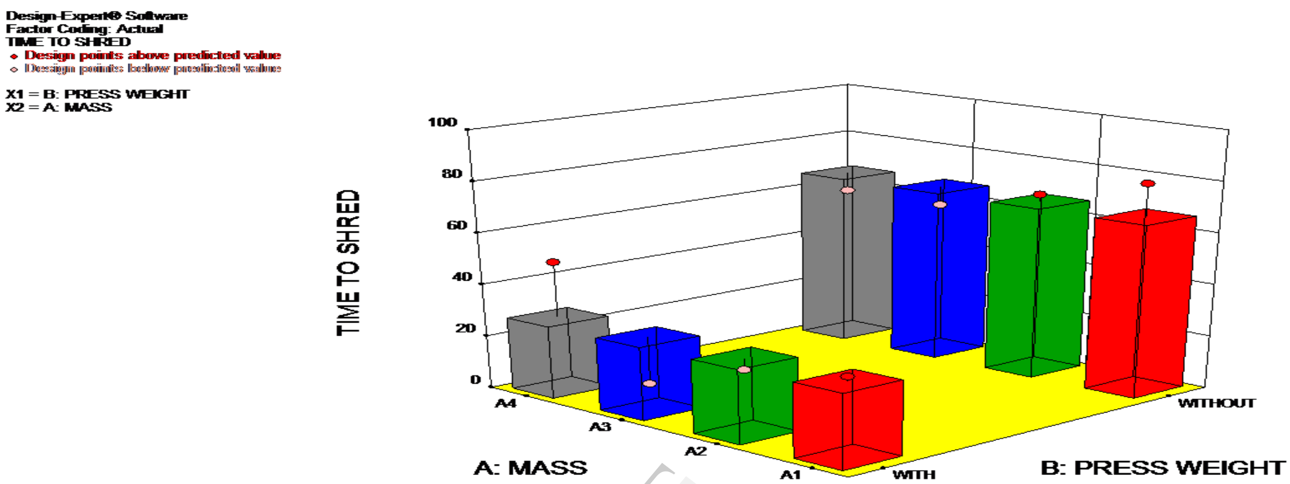

Figure 12.A plot of Mass and Press weight against Time to Shred

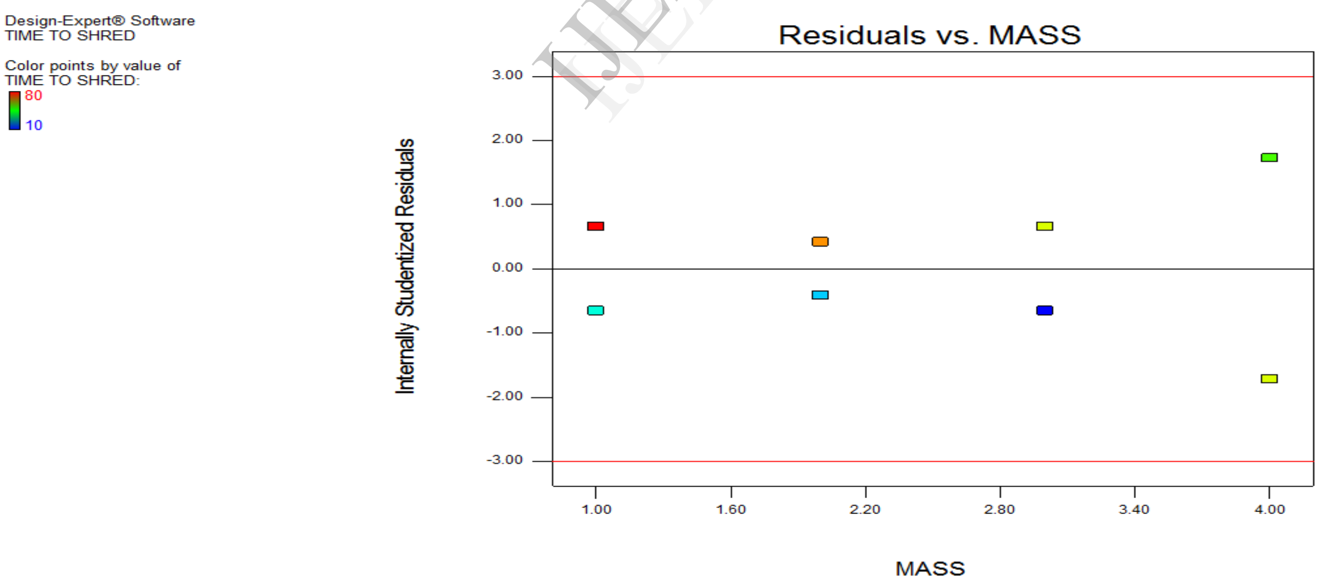

Figure 13.A Plot Residuals(e_J) against Mass

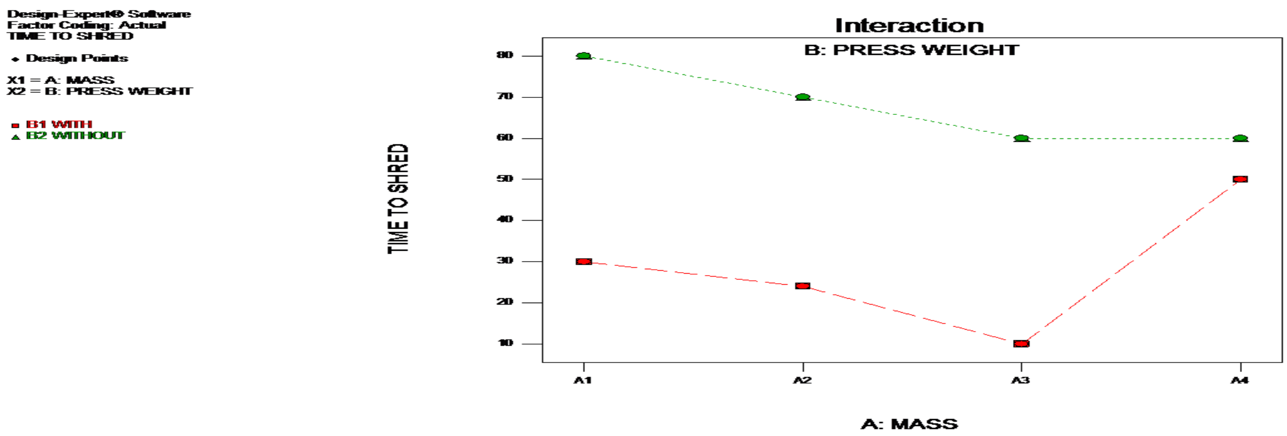

FIGURE14.GRAPH OF MASS AND PRESS WEIGHT INTERACTION

CONCLUSION

The breadfruit shredder machine employing a prime mover was developed and its performance found to be satisfactory. An element of the machine, press weight was found to have a significant effect on the time to shred and invariably the shredding efficiency of the machine.

RECOMMENDATION

WE recommend that subsequent work on breadfruit shredder should be focused on for further improvement and incorporation

of this equipment. The following suggestions can be adopted:

The use of diesel or petrol powered engine to eliminate dependency on the epileptic electric power supply.

REFERENCES

1.ADEBOWALE, K.O., OLU-OWOLABI, I. O., OLAWUNMI, E.K AND LAWAL, O.S (2005): Functional properties of native, physically and chemically modified breadfruit (Artocorpus altilis) starch. Industrial Crops and Products. (U.K) 21, 343-351.

2.Akanbi, T.O., Nazamid, S. and Adebowale, A.A.(2009). Functional and pasting properties of a tropical breadfruit (Artocarpus altilis) starch from Ile-Ife, Osun State, Nigeria International Food Research Journal. 16, pp 151-157

3.Amusa, N.A, Kehinde, I.A, Ashaye, O.A (2002). Bio-deterioration of breadfruit (*Artocarpus communis*) in storage and its effects on the nutrient composition. Afri. J. Biotechnol. 1(2): 57-60.
Arowora, K. A.1, Ogundele, B. A.1 and Ajani, A. O.(2012). Storability of breadfruit and its hazard analysis in Nigeria
Journal of Stored Products and Postharvest Research. 3(5), pp. 63 – 66,

4.Cook, A.A (1975). Disease of Tropical and sub-tropical fruits and nuts.Longman Co. Ltd. UK. p. 34.
Etoamaihe, U.J and Ndubueze, K.C(2010).Development and Performance Evaluation of Dehulling Machine for African Breadfruit (Treculia Africana).Journal of Engineering and Applied Sciences.5(4).pp312-315

5.Montgomery, D.C.(2001).Design and Analysis of Experiments. New York.5th Ed: John Wiley And Sons Inc,pp557-590

6.Nwigbo,S.C., Chinwuko, C.E., Achebe C.H and Tagbo D.A.(2008). Engineering Design of Breadfruit Shelling Machine. African Research Review.2(4),pp 1-16

7.Ragone ,.D (1997). Breadfruit *Artocarpus altilis* (Parkinson) Fosberg.Promoting the conservation and use of underutilized and neglected crops. Institute of Plant Genetics and Crop Plant Research, Gatersleben International Plant Genetic Resources institute, Rome, Italy

8.Shigley, J. E., & Mischke, C. R. (2011). Mechanical engineering design. (9th ed.). New York:McGraw-Hill Companies, Inc., (Chapter 18).

9.Soetjipto, N.N, Lubis, A.S (1981). Vegetables: IBPGR Secretariat, Rome.p. 330.

10.Tyler G. H (2004).Standard Handbook of Engineering Calculations (4th ed.).The Mcgraw-Hill Companies,Inc.,