ADAPTATION OF IMPACT HAMMER MILL FOR CRUSHING COCOA PODS HUSK AS A LIVESTOCK FEED CONSTITUENT

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ABSTRACT

Dried Cocoa Pod Husks (CPH) have been identified as a substitute for maize in livestock feeds production. Conventional hammer mills in which beaters are made of low carbon steel does not provide sufficient impact force to crush dried cocoa pod husks (CPH). The impact hammer mill was adapted for this purpose using locally available materials. Beaters were made of heat-treated high carbon-steel to resist abrasion, distortion and improve overall performance. Crushing efficiency, percentage losses and throughput of the machine were 98.99%, 1% and 239.09 kg/h respectively and it costs One thousand and twenty US Dollars fifty cents only (USD1020.50) to fabricate the machine one-off. This work is a step further in cocoa processing; CPH having hitherto constituted a waste in the industry. It promises additional revenue for primary processors of cocoa especially in tropical Africa where the bulk of world cocoa is produced.

KEYWORD: Cocoa pod husk, hammer mill, crushing, beaters, heat-treated high carbon steel, livestock feed.

1. INTRODUCTION

Cocoa (*Theobroma cacao),* introduced to West Africa in the 16th century by the Spanish, is now largely grown in West Africa, Latin America and Southeast Asia (Owolarafe *et al.,* 2007). Cocoa fruit (pod) contains seeds (also called cocoa beans), which when fermented and dried provide valuable material for beverages, confectionaries and cosmetics. However, cocoa pod husk (CPH) which is over 70% (w/w) of the whole fruit is a grossly under-utilised by-product in cocoa processing. In 2004, cocoa cultivation in the world covered 3.5 - 4.5 million ha of land, yielding over 3.6 million tons of cocoa beans and about 10 million tons of CPH which ended up in the garbage (Owolarafe *et al*., 2007). The CPH contains a high proportion of potassium and constitutes 52-75% of the pod wet weight (Fagbenro, 1988) and at present it is an undesirable waste product yet to be economically exploited in cocoa processing. Cocoa-pod husk is known to contain sizeable amounts of useful organic nutrients and have been successfully substituted for maize in livestock feed rations (Fagbenro, 1988). Atuahene *et al*., (1984) and Sobamiwa (1998) established that maize may be replaced with 10% crushed CPH in livestock feeds production without any adverse effect on animal weight gain and carcass quality. It may also be burnt into potash for making soft soap (Adomako, 1995; Spore, 1998; Owolarafe *et al*., 2007). Previous research efforts on cocoa processing focused on machines for pod breaking and beans extraction (Faborode and Dirinfo, 1994), very little literature exists on the utilization and processing of CPH, the disposal of which constitutes a menace to the industry. Cocoa pod husks processing involve crushing the material in a hammer mill, drying and particle size reduction to enhance proper mixing with other feed components.

In ancient times, milling of cereal grains was carried out by the abrasion generated when rubbing two stones against each other. This later developed into the attrition mill and later hammer mill (Donnel, 1983; Nasir, 2005). There are various types of hammer mill designs mostly based on rotor/beater mechanisms and hopper configuration. In Nigeria and other countries in sub-Sahara Africa where cocoa cultivation is major, hammer mills are locally fabricated using low-carbon steel. However, dried CPH being an excessively hard and abrasive material breaks the rotor/beater mechanisms of the mills incessantly (Adekomaya and Samuel, 2014), resulting in high maintenance costs. From literature review, a design that is capable of crushing CPH efficiently is hardly found. In this work, the conventional hammer mill was adapted for crushing CPH using beaters made of high-carbon steel with low abrasion rate.

2. METHODOLOGY

2.1 Existing Hammer Mill Designs

Two common types of hammer mill designs were selected as a basis for modification: an imported design in the Department of Chemical Engineering, Obafemi Awolowo University, Ile-Ife (Fig. 1a) and a locally fabricated commercial design in Bodija Market, Ibadan (Fig. 1b) of capacities 0.125 and 0.150 ton/h respectively.

2.2 Design Modifications

The experimental machine was conceived basically as a device that will generate sufficient impact to crush CPH into desired particle sizes as required in livestock feeds production and achieve a milling capacity of 0.25 ton/h. The main components of the modified machine are: Hopper, Milling chamber, Discharge unit and Frame.

2.2.1 Design of the Milling Chamber

The milling chamber of the existing machines has the following components: separator disc, pivot rod, sieve, beater and rotor shaft (Fig. 2) and the following dimensions of the components (Fig. 3) were taken: separator diameter, D_o; sieve diameter, S_d; beater length, $H_{L;}$ beater width, W_o and beater thickness, T_o; beater swing length, S_L and pivot rod diameter, P_d. From the foregoing, the relationship between these dimensional indices may be described by the following expressions:

Fig. 1a. An imported hammer mill in the Department of Chemical Engineering, Obafemi Awolowo University, Ile-Ife.

Fig.1b. A locally fabricated commercial hammer mill in Bodija Market, Ibadan

$$S_d = 2D_o \quad (1)$$
$$S_L = 2W_o \quad (2)$$

$$P_d = 2T_o \quad (3)$$

$$H_L = S_L + \left(\frac{p_d}{2}\right) + P_d \quad (4)$$

Based on the foregoing, the relationship between throughput and dimension of components of the milling chamber were calculated as shown in Table 1. It was established that as throughput varied from 0.125 to 0.15 ton/h (at 0.025 ton/h interval), sieve diameter increased from 325 to 350 mm; and that 25 mm increase in sieve diameter translates to 0.025 ton/h increase in throughput. For the experimental machine to develop a throughput of 0.25 ton/h, the separator diameter, sieve diameter, beater length, beater width, beater thickness, swing length and pivot rod diameter of the milling chamber are obtained as: 225, 450, 160, 65, 10, 150, 20 mm respectively. This gave useful information for determining the rotor shaft diameter, power rating of the electric motor and pitch length of the V-belt.

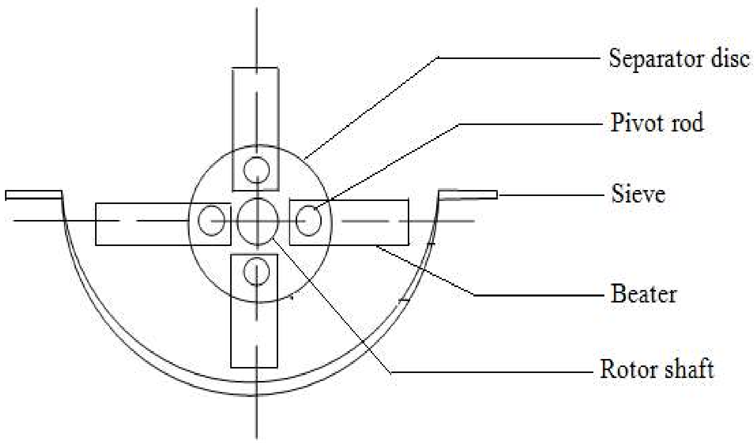

Fig. 2. Schematic drawing showing the internal components of the milling chamber

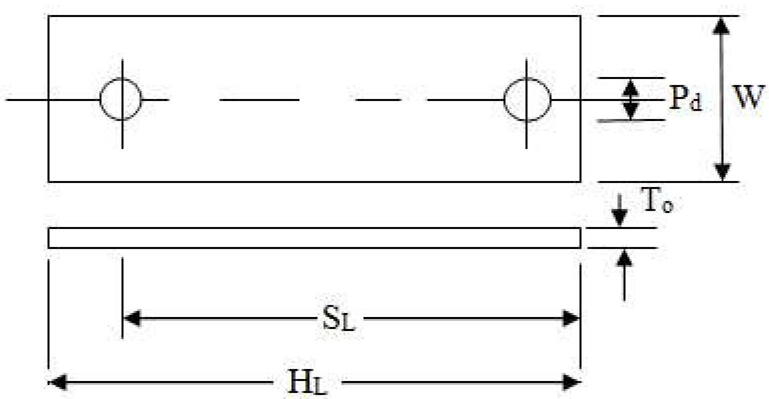

Fig. 3. Schematic drawing showing the dimensions of components of the milling chamber

Table 1. Relationship between throughput and dimension of components in the milling chamber

Throughput Ton/h	Beater Thickness T_o (mm)	Beater Width W_o (mm)	Pivot rod Diameter P_d (mm)	Beater Swing Length S_L (mm)	Beater length H_L (mm)	Sieve Diameter S_d (mm)	Separator Diameter D_o(mm)
0.100	5.00	40.00	10.00	80.00	95.00	300.00	150.00
0.125	5.00	45.00	10.00	90.00	105.00	325.00	162.50
0.150	6.00	50.00	12.00	98.00	117.00	350.00	175.00

0.175	7.00	52.00	14.00	104.00	125.00	375.00	187.5
0.200	8.00	58.00	16.0	116.00	140.00	400.00	200.00
0.225	8.50	60.00	17.0	120.00	145.00	420.00	212.50
0.250	10.00	65.00	20.00	130.00	160.00	450.00	225.00
0.275	10.00	70.00	20.00	138.00	168.00	475.00	237.50
0.300	10.50	70.00	21.0	140.00	172.00	500.00	250.00
0.325	11.00	70.00	22.00	145.00	178.00	525.00	262.50
0.350	12.50	70.00	23.00	166.00	200.00	600.00	300.00

2.2.2 Determination of Rotor Shaft Speed

The rotor shaft speed was obtained from the speed ratio relationship:

$$\left(\frac{d}{D}\right)=\left(\frac{N}{n}\right) \tag{5}$$

where, n = Speed of the driving pulley on the electric motor in rpm; which has diameter “d” mm
N = Speed of the driven pulley (rotor shaft pulley) in rpm which has diameter “D” mm

The speed of a 3-phase, 50 Hz induction electric motor intended for use in this design is taken as 1450 rpm (Enemuoh *et al.,* 2013). If n = 1450 rpm and d = 75 mm, since half of the electric motor speed was desired on the rotor, it follows then that. N= 725 rpm and D = 150 mm.

2.2.3 Pitch llength of V-belt on the Rotor Shaft Pulley

The distance ‘C’ between the centres of the two pulleys is assumed as 860 mm being the smallest distance between the rotor and the electric motor pulley. Shigley *et al.* (2004) expressed the pitch length of an open belt drive in mm as:

$$L = 2C + \frac{\pi(D-d)}{2} + \frac{(D-d)^2}{4C} \tag{6}$$

Applying equation (6), L = 2075.1 mm = 81.7 inches. This is equivalent to belt number B-81 using the standard Table for v-belt and load selection (Shigley *et al.,* 2004).

2.2.4 Determination of Belt Tension, Power and Torque Transmited

For uncrossed belt drives, it is known that the contact angle α_1 and α_2 for the small and large pulley respectively may be expressed as follows:

$$\alpha_1 = 180^o - 2\theta = 180^o - 2Sin^{-1}\left[\frac{R-r}{C}\right] \tag{7}$$

$$\alpha_2 = 180^o + 2\theta = 180^o + 2Sin^{-1}\left[\frac{R-r}{C}\right] \tag{8}$$

$$\text{and } \theta = \text{ the angle of wrap} = Sin^{-1}\left[\frac{R-r}{C}\right] \tag{9}$$

Substituting the values of “R”, “r” and “C” in equation (5), $\theta = 2.52^o$; hence using equations (7) and (8), $\alpha_1 = 3.09$ rad. and $\alpha_2 = 3.23$ rad.

When two pulleys of different diameters are connected by an uncrossed belt, the angle of contact of the small pulley should be considered for determining tensions "T_1 and T_2" of the tight and slack sides of the belt respectively (Khurmi and Gupta, 2006).

$$2.3 Log \frac{T_1}{T_2} = \mu\theta \tag{10}$$

For rubberized v-belts, $\mu = 0.30$; therefore, the ratio $\frac{T_1}{T_2} = 2.02$ (11)

The driven pulley pulls the belt from the tight side and delivers it to the slack side at speed v_1 and v_2 respectively.

where, $v_1 = \frac{\pi dn}{60} = 5.6 m/s$, $v_2 = \frac{\pi DN}{60} = 11.39 m/s$

According to Khurmi and Gupta (2006), the rubberised Class B type of v-belt material that was selected has the following specifications density, ρ_{belt} = 1000 kg/m^3; shear stress, σ_{max} = 2 Mpa, and belt dimensions, b = 25 mm and t = 15 mm. The belts' pitch length $l = 2075.1$ mm. Therefore mass of the belt, $M = b \times t \times l \times \rho_{belt}$ = 0.78 kg

However, the maximum tension, T of a belt = $\sigma \times b \times t$

= $2 \times 10^6 \times 0.025 \times 0.015$ = 750 N.

For greater power transmission, the belt speed, $v_{max} = \sqrt{\frac{T}{3M}} = 18 m/s$

For maximum power to be transmitted, centifugal tension $T_C = \frac{T}{3} = 250$ N

Consequently, belt tension "T_1" on the tight side $T_1 = T - T_C = 750 - 250 = 500$ N.

Using equation (11) the tension T_2 on the slack side of the belt, = 247.5 N

The maximum power that the belt can transmit $P = (T_1 - T_2)v_{max}$ = 4.55 kW

Based on the above, the electric motor with the nearest power rating of 5 kW was selected.

Hence, the torque, T_r of the rotor shaft is expressed as $T_r = \frac{P \times 60}{2\pi N}$ = 65.87 Nm.

2.2.5 Shearing Force and Bending Moment Diagrams

The rotor shaft was considered as been supported at two ends by uniformly distributed load of the beaters gang; the shearing force and bending moment diagrams of which are shown in Fig. 4. The maximum bending moment, M_{max} was taken at point "G" (Khurmi and Gupta, 2006).

$$M_{max} = 15.3 \times (60 + 180.75) - \left[\frac{0.859(180.75)^2}{2}\right] = 23.36 \text{ Nm}$$

$$\frac{M_{max}}{I} = \frac{\sigma_b}{y} \tag{12}$$

where $I = \frac{\pi d^4}{64}$, $y = \frac{d}{2}$ and for solid steel shaft the maximum shear stress, σ_b = 100 N/mm^2

Substituting "I" and "y" in equation (10), $d = \sqrt[3]{\frac{32 \times M_{max}}{100\pi}}$ = 13.5 mm

Khurmi & Gupta (2006) recommended a factor of safety of 4 for a steel material under steady load condition. The diameter of the selected rotor shaft = d × factor of safety= 4 × 14 mm = 54 mm, hence a rotor shaft Ø50 mm was selected. The summary of all machine design parameters are shown in Table 2.

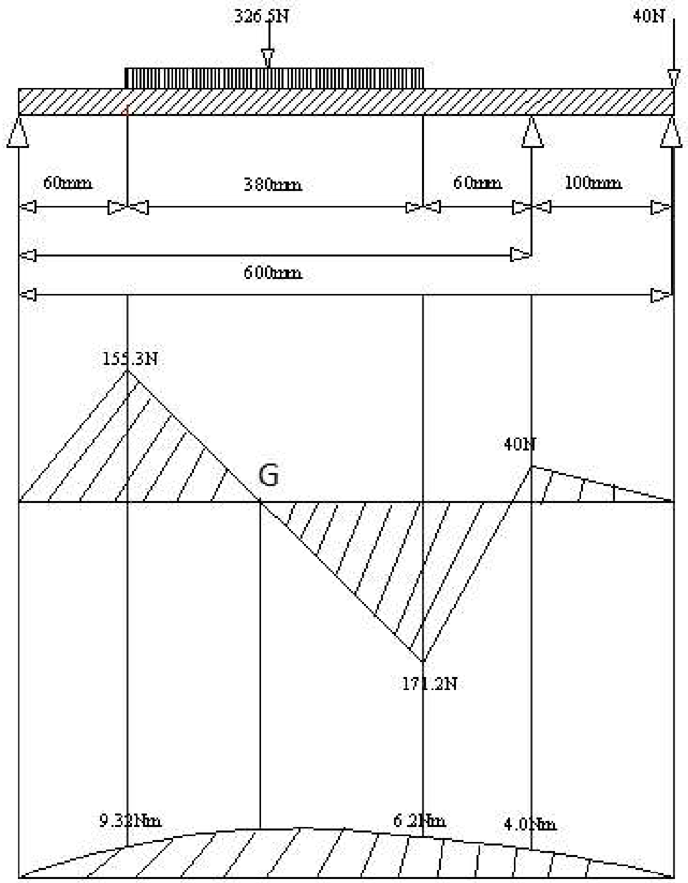

Fig. 4. Bending moment and shear force diagram

Table 2. Summary of machine design parameters

Parameter	Value
V– Belt length, L	2075/81.7 mm/in
Speed of Electric Motor, n	1425 rpm
Speed of Rotor shaft, N	712.5 rpm
Angle of wrap, θ	2.52°
Contact angle for small pulley, α_1	174.96°
Contact angle for big pulley, α_2	185.20°
Tension in the slack side of the belt, T_2	247.5 N
Tension in the tight side of the belt, T_1	500 N
Power transmitted to the shaft, P	5.0 kW
Torque transmitted to the rotor shaft, T_r	67 Nm
Max. velocity of the pulley & belt drive, V_m	18 m/s
Centrifugal force exerted by Beater, F_c	1.1 kW
Weight of Beater, W_b	106 N
Diameter of the rotor shaft, D	50 mm
Weight of the separator plate, W_{sp}	180 N
Weight of the pivoted shaft, W_{ps}	40.5 N

Max bending moment, M_{max}	23.4 Nm
Max shearing force, F_s	171.2 N
Diameter of pivoted shaft, D	24 mm

2.3 Machine Parts Description

Hopper: The hopper is the feeding unit of the mill through which dried CPH is fed into the machine. It is a truncated pyramid made of 6 mm thick mild-steel plate having dimension of 300 × 400 mm at the top, 160 × 400 mm basal chute and vertical height of 200 mm. The hopper is inclined at 45° to the horizontal.

The Milling Chamber: The milling chamber is a gang of 27 beaters, swinging on pivoted shafts born by four separator discs which were mounted at 120° space along the circumference and anchored on a rotor shaft. The beaters were held in position by thirty six spacers. The beaters were made of heat-treated high-carbon steel plate of 60 N/mm^2 strength (Juzt and Scharkus, 2003), each having a dimension of 130 × 55 × 8 mm and Ø25 mm hole drilled on both ends through which it is pivoted and fixed on the separator disc for easy swinging. Each spacer was a ring, 12 mm in length cut from a seamless steam pipe with inner and outer diameters of Ø25 and Ø35 mm respectively. Separator discs made of medium-carbon steel plate; 15 mm thickness and cut in circular shapes of Ø235 mm each were welded permanently on the rotor shaft. Below these beaters is a half circle-shaped screen of 3.5 mm aperture. The screen is made of a 6 mm thick mild steel plate cut to a size of 420 × 1320 mm and rolled into a half cylinder. Holes of Ø3.5 mm were drilled at 6 mm square distance on the entire plate. This was suspended internally on the bottom of the milling chamber at a radius of 210 mm and 5 mm clearance from the tips of the beater. The rotor shaft, driven by a 7.5 hp, 1450 rpm three phase electric motor; was made of a medium-carbon steel rod of Ø55 × 700 mm, turned to Ø50 × 600 mm and supported by two plummer bearings fixed on the frame. A key-way of 10 mm × 5 mm was milled on the rotor shaft at a 100 mm distance from one end to fix the driven pulley in place. A transition fit was maintained on both ends for bearing positioning.

The Discharge Unit: The discharge unit is an inclined 200 × 150×150 mm chute made of 6 mm thick mild-steel plate welded to the bottom chamber corresponding to the suspended inner plate in which the crushed material flows.

The Frame: The frame is the member which supports the machine. It is made of heavy duty angle-iron cut into 350 mm length and welded to the bottom of the milling chamber at a height of 250 mm to the base. The machine frame occupies a dimension of 1500 × 1120 × 600 mm after assembly.

The orthographic and isometric views of the machine are shown in Fig. 5. The exploded view of the experimental machine and a prototype are shown in Figs. 6 and 7 respectively.

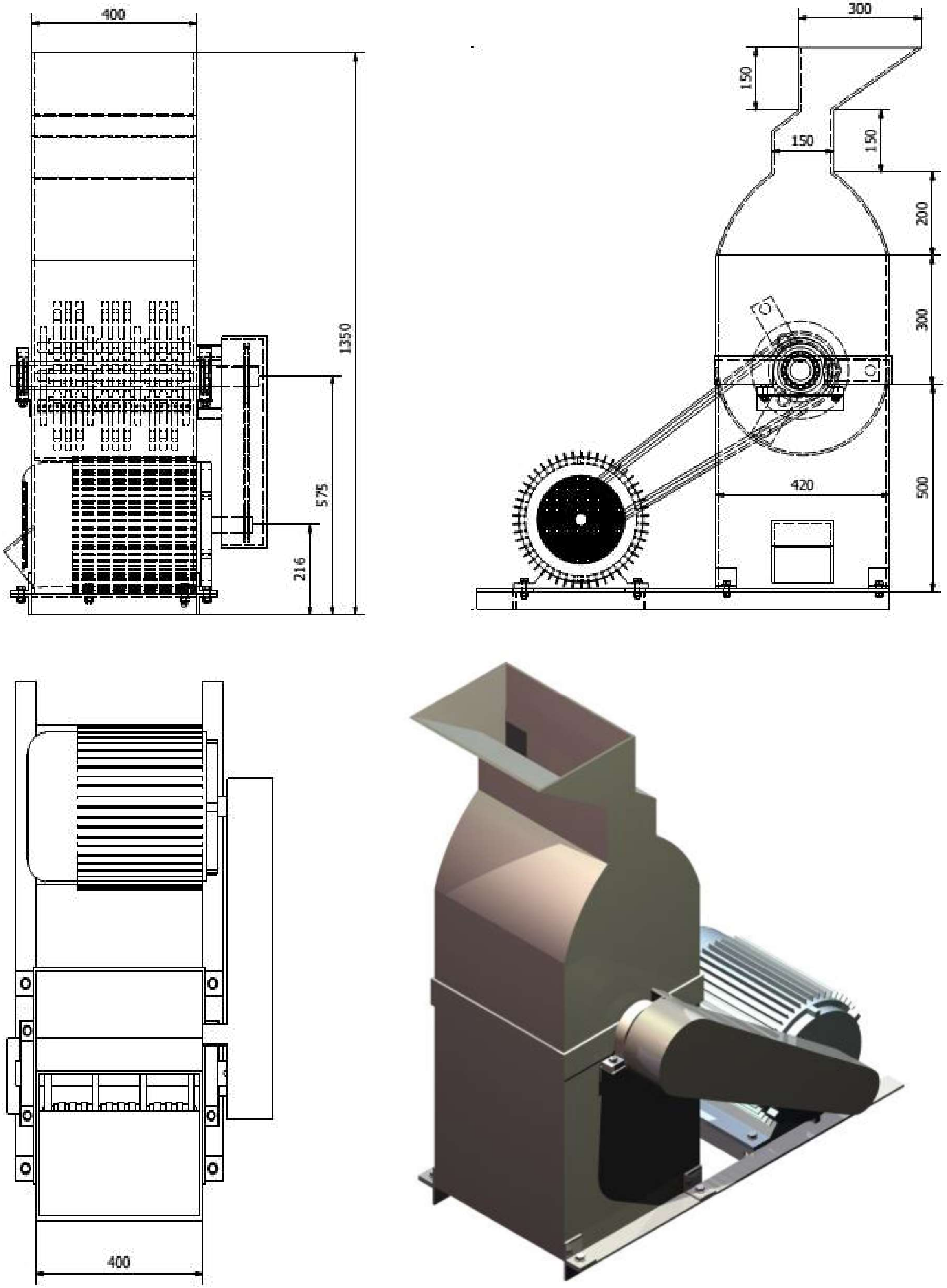

Fig. 5: Orthographic and Isometric Projections of the experimental machine

PARTS LIST			
ITEM	QTY	PART NAME	MATERIAL
1	1	Hopper	Steel, Mild
3	1	Milling Chamber	Steel, Mild
4	1	Rotor, Separator Discs & Beaters Sub-assy	
5	2	Rolling bearing 21310 C GB/T 288-94	Steel, Mild
6	2	Bearing Housing	Steel, Mild
8	8	ISO 7089 - 10 - 140 HV	Stainless Steel
9	4	ANSI B18.2.3.5M - M10 x 1.5 x 35	Steel, Mild
10	4	ISO 4032 - M10	Stainless Steel, 440C
13	1	Sieve	Steel, Carbon
14	1	Discharge Unit	Steel, Carbon
15	1	V-Belt	Rubber
16	1	Grooved Pulley1	Steel
17	1	Grooved Pulley2	Steel
20	1	Frame	Steel, Carbon
23	1	Electric Motor	Steel, Mild
24	1	Belt Guard	Steel, Carbon
25	16	ISO 7089 - 12 - 140 HV	Stainless Steel
26	4	ANSI B18.2.3.5M - M12 x 1.75 x 30	Steel, Mild
27	8	ISO 4032 - M12	Stainless Steel, 440C
28	4	ANSI B18.2.3.5M - M12 x 1.75 x 40	Steel, Mild

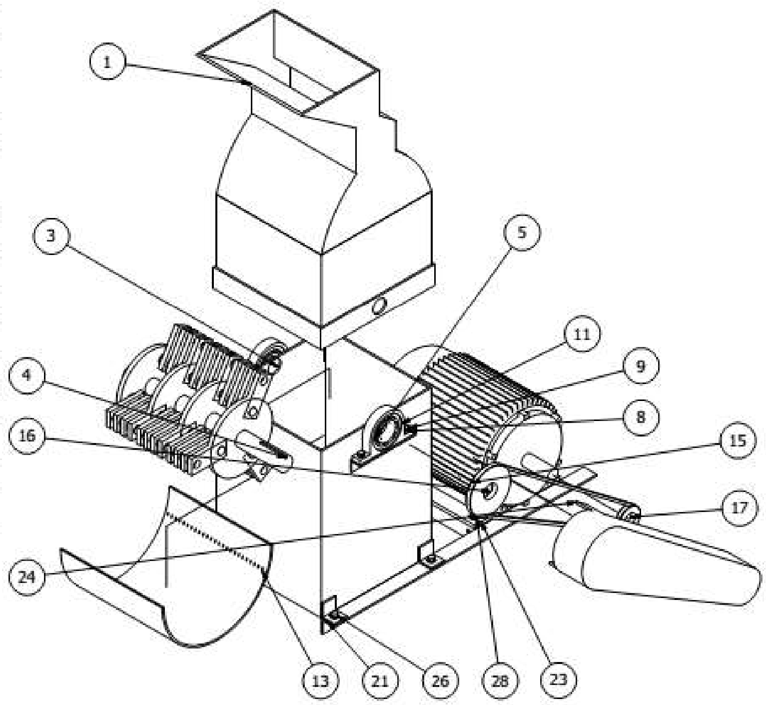

Fig. 6. An exploded view of the experimental machine

Fig. 7. Prototype of the experimental machine

2.4 Performance Evaluation Procedure

The size of pulley on the electric motor was varied in order to test the machine for efficient running speed; the effect of varying speed on machine performance is shown in Table 3. With the same pulley size of 150 mm diameter on both ends**,** it was observed that the machine vibrated excessively and bulk of input materials (CPH) were thrown out of the hopper. As the pulley size on the electric motor was reduced from 150 to 105 mm, the rotor speed reduced from 1450 to 1015 rpm and there was vibration of machine parts but no throwing of input materials occured. As the diameter of pulley on electric motor was reduced further at an interval of 15 mm, there was a corresponding reduction in the rotor speed. However the efficient machine speed was observed when the electric motor pulley diameter was 75 mm; this gave a corresponding rotor speed of 712.5 rpm. At this point, the machine ran smoothly and there was no vibration and material loss was minimal. Further reduction in electric motor pulley diameter below 75 mm resulted in poor machine performance and excessively low output. Therefore the efficient running speed of the machine is 712.5 rpm with an electric motor pulley diameter of 75 mm.

About 950 kg Dried CPH (at 9.56%) obtained from a village near Ipetu-Modu, Osun state, Nigeria were used to test the machine. Machine performance was evaluated on the basis of crushing efficiency, throughput and percent material loss. The electric motor pulley diameter was fixed at 75 mm and the rotor speed at 712.5 rpm during testing. At a feed rate of 4 kg/min, the machine was fed in six successions of 8, 12, 16, 20, 24, 30 and there were three replicates in each run. Crushing efficiency, percentage losses and throughput of the machine were calculated using equations (11), (12) and (13).

$$\text{Crushing efficiency, } \eta = \frac{M_o}{M_i} \times 100\% \tag{11}$$

$$\text{Percentage losses } = \frac{M_i - M_o}{M_i} \times 100\% \tag{12}$$

$$\text{Thoroughput} = \frac{M_o}{Time} \, kg/h \tag{13}$$

Where M_o = mass of material output and M_i= mass of material input

3. PERFORMANCE EVALUATION RESULTS

The results of the performance evaluation are shown in Table 4. Crushing efficiency, percentage loses and through put obtained are 98.99%, 1% and 239.09% respectively. Percentage loss of 1% may be attributed to particles that flew off out of the hopper and discharge chute during crushing. A thin layer of unrecoverable cocoa pod dust was also noticed on rotor shaft, beaters and the wall of the crushing chamber. Total cost of fabricating the machine is USD 1020.50.

Table 3. Effect of varying speed on machine performance

Diameter of pulley on electric motor, d (mm)	Diameter of pulley on rotor shaft, D (mm)	Electric motor speed, n rpm	Rotor speed N rpm
150	150	1425	1425
105	150	1425	1015
95	150	1425	920
75	150	1425	712.5
60	150	1425	580

Table 4. Test results of machine performance evaluation

Nos of run	Mass of CPH before crushing (kg)	Mass of CPH after crushing (kg)	Time taken (min)	Milling efficiency (%)	% Losses	Throughput (kg/h)
1	8	7.85	2	98.13	1.88	235.50
2	12	11.78	3	98.17	1.83	235.60
3	16	15.81	4	98.81	1.19	237.15

4	22	21.89	5	99.50	0.50	262.68
5	24	23.95	6	99.79	0.21	239.50
6	30	29.88	8	99.60	0.40	224.10
Average	18.67	18.53	4.67	98.99	1.00	239.09

4. CONCLUSIONS

A hammer mill for crushing dried cocoa pods into livestock feed has been developed. The machine, which crushes by impact has beaters made of heat-treated high carbon-steel as the major modified component; and was developed from locally available materials. The crushing efficiency, percent losses and throughput were: 98.99%, 1% and 239.09 kg/h respectively. The total cost of producing the machine one-off manufacture was estimated as One thousand and twenty US dollars, fifty cents only (USD1020.50). This work is a significant step in cocoa processing. It offers cocoa processors extra income through value addition to CPH, which hitherto had been a waste in the industry; and lessens competition between man and livestock for maize. Overall, this promises additional source of revenue for cocoa farmers in tropical sub-Saharan Africa where the bulk of world cocoa is produced.

REFERENCES

Adekomaya, S.O. and Samuel, O.D. Design and development of a petrol-powered hammer mill for rural Nigerian farmers. *Journal of Energy Technologies and Policy*, 4(4): 65-72.

Adomako, D. 1995. Non-traditional uses of cocoa in Ghana. Eighth meeting of the advisory group on the world cocoa economy, ICCO, pp. 79–85.

Atuahene, C.C., Adams, C. and adomako, D. 1984. Cocoa pod husk instarter diets of broiler chicken. In: Cros, E. (Ed) Proceedings of 9th International Cocoa Research Conference, Lome Togo, Feb. 12-18, 1984.

Donnel, C.H. 1983. *Farm Power and Machinery*. Mc-Graw Hill Book Company, New Delhi, India. pp 90-95

Enemuoh, F.O., Okafor, E.E., Onuegbu, J.C. and Agu, V.N. Modelling, simulation and performance analysis of a variable frequency drive in speed control of induction motor. *International Journal of Engineering Inventions*, 3(5): 36-41

Faborode, M.O. and Dinrifo, R.R. 1994. A mathematical model of cocoa pod deformation based on Hertz theory force. *International Agrophysics,* 8*:* 403–409.

Fagbenro, O.A. 1988. Results of preliminary studies on the utilization of cocoa-pod husks in fish production in south-west Nigeria. *Biological Wastes*, 25(3): 233–237

Juzt, H. and Scharkus, E. 2003. Westernman Table for Metal Trade Skip Series Vol 3. Third Edition. Pp 127. New Age International, New Delhi, India.

Khurmi, R.S. and Gupta, J. K. 2006. *Machine Design*, S.I. Unit Edition. Pg. 102, 514-711, New Delhi, India. pp 650-667

Nasir, A. 2005. Development and testing of hammer mill, *Australian Journal of Technology*, 8(3): 124-130.

Owolarafe, O.K., Ogunsina, B.S., Gbadamosi, A.S and Fabunmi, O.O. 2007. Application of coefficient of friction to the separation of cocoa husk–beans mixture. *Journal of Food Process Engineering*, 30: 584–592.

Sobamiwa, O. 1998. Performance and egg quality of laying hens fed cocoa husk based diets. *Nigerian Journal Animal Production,* 25: 22-24.

Spore, J. 1998. Farmers and fish love cocoa. ICCO *76*, 8.

Shigley, J.E., Mischke, A.R. and Budynas, R.G. 2004. *Mechanical Engineering Design*, McGraw-Hill, New York. pp 880-893