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Design Analysis of a Pedal Powered Cassava Grinding Machine

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Abstract: *The purpose of this technical study was to do a design analysis in order to ascertain the performance and output capacity of a pedal powered cassava grinding machine that is very efficient and affordable for rural farmers (dwellers) who cannot afford petrol and diesel engines as a prime mover for garri processing. In the design, the human expended for an average age man of 70kg (1501b) at a cycling speed range of 16km/h – 24km/h or 233 r.p.m was used and after the analysis the efficiency and human power required to drive the shaft was calculated to be 56% and 1.02hp respectively. The machine which is very cheap and affordable is highly recommended for farmers as it can deliver an output capacity of 58:59kg of cassava per hour and can produce a mechanical advantage of 0.42 which is less than 1 as recommended under simple machines.*
Keywords: *Cassava, Pedal Powered, Rural Farmers*

I. Introduction

In Nigeria, cassava is very essential as it serves as a number one stable food, used to produce starch, garri flour and it peels are also used to feed livestock .The process of transforming cassava tubers into pulp form is known as grating. Peeled cassava tubers are feel into the hopper made of mild steel then to the cylindrical drum which the grind the cassava into a mashed form.The rate of grinding will depend on the cycling speed of the person riding the pedal. The pedal (attached to the crank) is that part that gives effort (energy) for motion to take place.The means of motion transmission is through a chain and then through a belt mounted on a pulley which is mounted on the transmission shaft supported by bearings and a flywheel at one end to restore energy lost from the system.

1.2 Purpose Of The Study

This study is aimed at ascertaining the performance analysis and output capacity of a humanly powered cassava grinds, m/c which will encourage rural farmers to invest into it and also provide a ground for researchers to improve on.

1.3 Benefit Of Study

This work will be useful in rural areas where farmers cannot afford to purchase petrol and diesel powered cassava grinding machine.

1.4 Materials And Method

The fabricated parts are; the hopper which is of mild steel, was welded, the frames (structural members) where welded and fastened using bolts and nuts.

Other parts such as shaft, pulley, and flywheel were turned and bored using the lathe machine. The grinding power comes from the rider through the belt and chain to the rotating element (shaft) supported by bearings and a flywheel which stores the energy when there is excess of energy and releases it when there is shortage of energy.

The average power produced by a 70kg man (non athlete) is 1.02 hp (757.76 watts) at speed range of (16 – 24) km/h.

II. Design Analysis

2.1 Design Data:

- Human energy expended say 70kg (1501b) Person: For cycling @ 15km/h (16 – 24km/h) = 1.62kJ/(KmKg).
- The average cycling speed = 15.5km/h
- The highest speed officially recorded for any human powered vehicle (HPV) on level ground = 133.78km/h
- The cycling speed in r.p.m = 133 rpm;

2.1.1 K.E Energy (K.E)

K.E = $\frac{1}{2}MV^2 \ (Translational)$……………………………………….(1)

K.E = $\frac{1}{2}1w^2 \ (Rotational)$……………………………………………(2)

But W = $\frac{V}{r} \ (Without \ Slipping) \ and \ I = mr^2$

$\Rightarrow$ From (2) K.E = $\frac{1}{2}Mr^2 . \frac{V^2}{r^2} \ \frac{1}{2}MV^2$ ……………………..(3)

TOTAL K.E$_T$ = K.E of rotational + K.E of (3) translational But since the is no translational motion:

$\Rightarrow$ **TOTAL K.E$_T$** = K.E of rotational = $\frac{1}{2}mv^2$

Where V = 16 – 24km/h (Average speed) and mass, M = 70Kg

$\therefore$ K.E$_T$ = $\frac{1}{2}$ *70*162^2 = **8960J** = **8.96KJ**

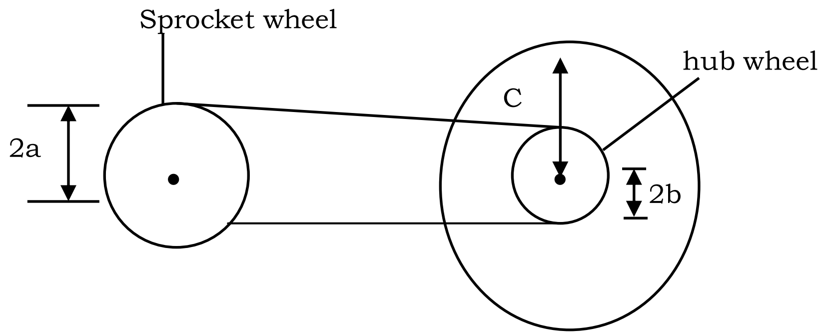
MECHANICAL ADVANTAGE

M.A = $\frac{Load}{Effort} = \frac{L}{E}$

But, L = 70 x 9.81 = 686.7 N = 0.687 KN
and E = 1.62KJ/(kmkg)

$\Rightarrow$ M.A = $\frac{0.687}{1.62} = 0.42 \ (less \ than \ 1)$

VELOCITY RATIO (V.R)



When the pedals and sprocket wheels complete a revolution, the hub wheel rotates through $\frac{2\pi a}{2\pi b}$ revolutions, and the back wheel does the same. Thus the black wheel has moved through a distance of 2 π C x $\frac{a}{b}$ When the pedal complete a revolution.

If L is the length of the pedal crank, the distance moved by effort in one revolution is
= 2and velocity ratio

V.R = $\frac{2\pi L}{\frac{2\pi ac}{b}} = \frac{bL}{ac}$ **N/B: The V.R and M.A is always less than 1**

$\therefore \ V.R = \frac{bL}{ac}$ V.R Using India standard, Khurmi and Gupta, 2005,

$\Rightarrow$ b = 50mm and hub dia = 100mm.

a = 125mm sprocket dia = 250mm
and length of crank L = 1.5m (Theory of machine by Khurmi, 2005)

Efficiency

Efficient $= \frac{M.A}{V.R} \; x \; 100\%$

The velocity Ratio V.R

$= \frac{bL}{ac} = \frac{0.05x1.5}{0.125x0.6}$

Where; C = 600mm $\Rightarrow \quad V.R = 0.75$ (less than 1)

$\therefore$ Efficiency $= \frac{M.A}{V.R} \; X \; 100\%$

$= \frac{0.42}{0.75} \; X \; 100\%$

= **56%**

The Pedal Powered Unit
Human power required to drive the cassava crushing unit

P = $gmVg(K_1 + 5) + K_2V_a^2 \, Vg^1$

Where P is a watts and g is earth's gravity, Vg is ground speed (m/s), m is bike/rider mass in the rider's speed through the air (m/s), K_1 is a lumped constant for all frictional lesser (tiers, bearing, chain) and is generally reported with a value of 0.0053. K2 is a lumped constant from aerodynamic drag and is generally report with a value of 0.185Ks/m.
N/B: If there is no wind, Vg, = Va
For rotational only: P = $K_2V_a^2 \, V_{g,}$ But Vg = Va

$\Rightarrow$ P = $K_2V_a^3$ where

K_2 = 0.185Kg/m and V_a
= 16 – 24Km/h

$\therefore$ P = 0.185×16^3

= 757.76 Watts

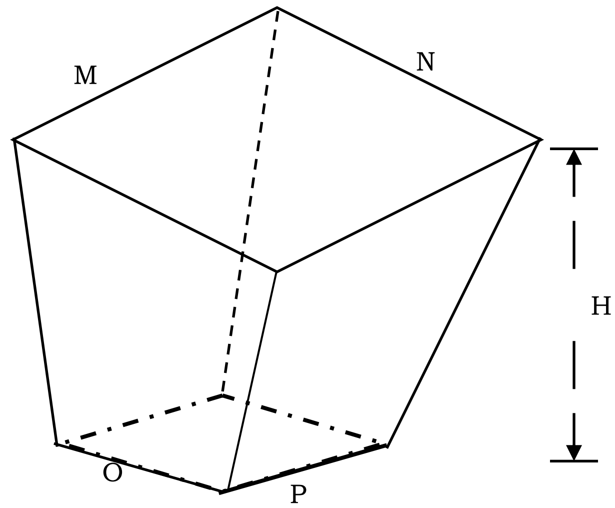
Ihp = 746 Watts

$\therefore$ 757.76 Watts $= \frac{757.76}{746}$
= **1.02hp**

$\therefore$ The Power P = **1.02hp**

$\therefore$ The minimum human power required to drive the cassava crushing unit is;
1.02HP

The Cassava Crushing Unit
⟶ The design of the hopper;



- Considering the upper rectangular dimension

Length (L) = N 1400mm = 1.4m
Width (W) = M 640mm = 0.64m

- Considering the lower rectangular dimension

Length (L) = O = 140mm = 0.14m
Width (P) = P = 64mm = 0.064m

- The height of the hopper, H

H = 510 = 0.51m
So, using the relationship;

Volume; V = $\frac{1}{3}H$ [(N^2*M – O^2*P)/(N – O)]

Substituting Values, We Have;

$$V = \frac{1}{3} * 0.5 \left[\frac{1.4^2 * 0.64 - 0.14^2 * 0.064}{(1.4 - 0.14)} \right]$$

$$= 0.17 \left[\frac{1.2544 - 0.001254}{1.26} \right]$$

= **$0.1691m^3$**

The Output Capacity

$$= \frac{Total\ mass\ of\ cassava\,(Kg)}{Total\ Time\ Taken\,(hr)}$$

Calculate the output capacity the table below should be considered

III. Result And Discussion

The machine makes use of both eh gravitational movement of eh cassava as well as gradual loading during grating as it does not grate when it start from root rather it is allowed to gather momentum before it is loaded.

The machine was tested for ten (10) different input values of mars of cassava and the time for each was taken and recorded as the output capacity was 58.59kg of cassava per hour. The **table 1;** Shown below gives a summary of the test carried out on the machine.

Table 1: Number of cassava loading and time taken to grate

NUMBER OF LOADING	MASS OF CASSAVA (KG)	TIME TAKEN (SEC)
1	0.6	32
2	1.1	63
3	1.6	95
4	2.1	127
5	2.6	159
6	3.1	191
7	3.6	223
8	4.1	255
9	4.6	287
10	5.1	319
TOTAL	**28.5**	**1751**

$\therefore$ The output capacity $= \dfrac{Total\ mass\ of\ cassava\,(Kg)}{Time\ take(s)}$

$$= \frac{28.5\ x\ 36009(Kg)}{1751\,(hrs)}$$

= 58.59Kg/hr

The volume of the rotor;

$$V_\alpha = \pi\alpha^2\ \mathrm{L}$$

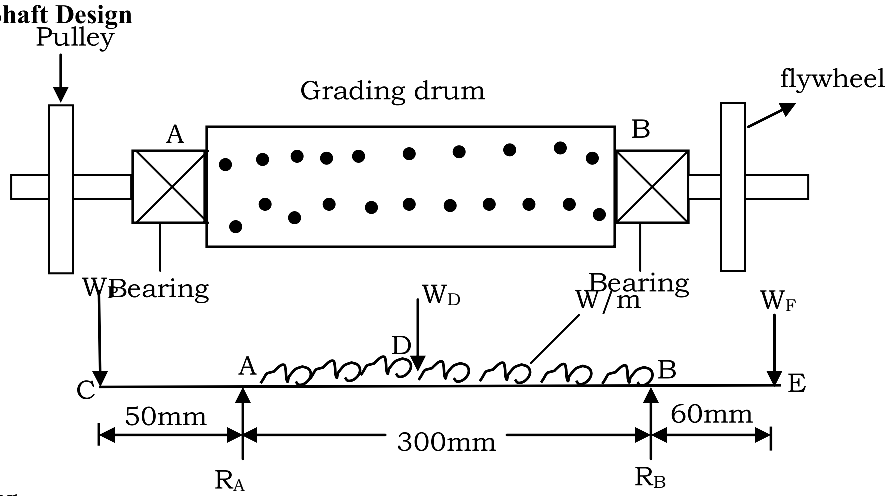
Where L = 300mm and α = 90mm

V^α = $3.143 \times 0.09^2 \times 0.3$

$\therefore$ The volume of rotor

= **0.00764m^3**

Shaft Design



Where;

WP = Weight of pulley

Wd = weight of grating drum

Wf = Weight of flywheel

(i) The weight of pulley; $W_p = M_p x g$ where M_p = mass of pulley

$\Rightarrow$ W_p = 2x9.81 = **<u>19.62N</u>** = 2kg

(ii) The weight of the grating drum; W_d

$\Rightarrow$ $W_d = V_a \times P_d \times g$
But $V_d = \pi\alpha^2 L$
Where;
D = 180mm = 0.18m
α = 90mm = 0.09m
L = 300mm = 0.3m and
P_d = 7930kg/m^3 (stainless)
= 7860kg/m^3 (mild steal)

$\Rightarrow$ V_d = 3.142x0.09x0.3
= 0.00763506m^3

Wd = 0.00763506 x 7860x9.81 = **<u>588.714N</u>**

(iii) The weight of flywheel; W_F
$W_F = M_F \times {}_g g$

Where, M_F = Volume x density
= 2π R*A*P

Where density,
P = 7260Kg/m$_3$ (cast iron)
= 7800kg/m3 (Cast steel)
And A = b x t and b = Width of rim
t = Thickness of rim
but the ratio of b/t = 2 $\Rightarrow$ b = 2t

$\Rightarrow$ A = 2t x t = 2t^2
$\Rightarrow$ A = 2 x 0.025^2
= **<u>0.00125m^2</u>**

If t = 25mm; = 0.025mm and
D = 300mm R = 150mm = 0.15m

$\therefore$ M_F = 2 x 3.142 x 0.5 x 0.00125 x 7260
= 8.55kg

$\therefore$ $W_F = M_F \times g$ = 8.55 x 9.81
= **<u>83.88N</u>**

The distributed loading due to the drum is = $\frac{588.714}{300}$

$\Rightarrow$ Wd = 1.962N/MM = **<u>1962.38N/M</u>**

Knowing, that the human power P = 1.02hp = 757.76 Watts

The Torgue = $\frac{757.76\ x\ 60}{2\pi\ x\ 233}$ The cycling speed = 233 r.p.m.

$\therefore$ Torgue 31.05Nm = 31050Nmm

The Shaft Diameter

The equivalent twisting moment, $T_e = \frac{\pi}{16} \times £ \times d_s^3$

Where the perminosible shear stress of the shaft material
£ = 1200N/mm^2 (Khurmi and Gupta, 2005)

$\Rightarrow \quad d_s^3 = \frac{Te \; x \; 16}{\pi \; x \; £}$ but Te = 19069Nmm as calculated below

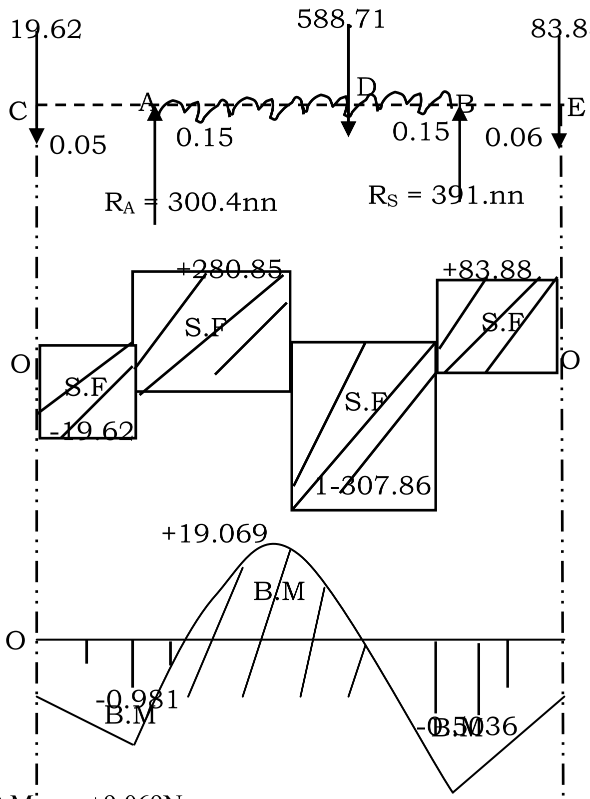
$$d_s^3 = \frac{19069 \; x \; 16}{3.142 \; x \; 1200}$$

$d_s = \sqrt[3]{80.920}$

d_s = 26mm

Therefore, we choose 25mm standard size from Khurmi and Gupta, mechanic design 2005

S.F And B.M Diagrams



8.F CALC

@ C, S. F_C = -1962N

@ A, S.F_A = -19.62 + 300.40n = 280.85N

@ D, S.F_D = -19.62 = 300.47 – 588.71 = -307.86N

@ B,S,F_B = 19.62 = 300.4n – 588.71 = 391.74 = +83.88N

@ E,S.F_E = +83.88 – 83.88 = 0

B.M CALC

@ C,B.M_C = O

@ A,B,M_A = 19.62 x 0.05 = -0.981Nm

@ D,B.M_D = -19.62*0.2 +

B.M_D = +9.069Nm

@ Pt. B., M_B = -19.62*0.35 + 300.α n*0.3 - 1962.38*0.8*0.15 = -6.867 + 90.1α 1 – 88.31 = -5.036Nm

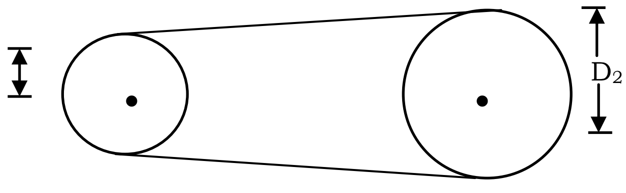
@ Pt.E;B.M_E = -19.62*0. α 1 +300. α n*0.36 1962.38*0.3*0.21 + 391.74*0.06 = -8.044 + 108.169 – 123.63 + 23.5044 = **0**

∴ The maximum bending moment will occur at point D which is = +19.069Nm = 19069Nmm

Belt Tensions

The torgue on the pulley is same as the torgue on shaft

$T_{AV} = (T_1 - T_2) \frac{D1}{2}$ D_1



But T_{AV} = 3105Nm = 31050Nmm but according to India standard, by Khurmi and Grupta, 2005, D_1 = 100mm and D_2 = 250mm

$\Rightarrow$ $31.05 = (T_1 - T_2)\dfrac{0.1}{2}$

$\Rightarrow$ $T_1 - T_2 = 621$ ……………………………….(1)

Also, $\dfrac{T_1}{T_2} = e^{N\theta}$ ………………………………………………….(2)

Where θ = (180 – 2 θ) and groove angle (θ) varies between 32^0 to 40^0 from Khurmi and Gupta, 2005.

So, using θ = 320^0

$\Rightarrow$ θ = (180 – (2x32)) = 180 – 64 = 116^0 and λ = 7an θ = 7an 32 = 0.62

So, substituting values into (2) we have; $\dfrac{T_1}{T_2} = 2.718 \; \dfrac{0.62X116}{180} = 2.7180^{0.3996} = 1.49$

$\Rightarrow$ $\dfrac{T_1}{T_2} = 1.49$; $T_1 = 1.49T_2$ ……………………………………..(3)

So, putting (3) into (1) we have; $1.49T_2 - T_2 = 621$

$\Rightarrow$ $(1.49 - 1)\, T_2 = 621$

$\Rightarrow$ $T_2 = \dfrac{621}{0.49} = 1267.35N$;

= 1.267KN

and T_1 = 1.49 x 1.267 = **1.888KN**

The Length Of Belt

$$L = 2C + \frac{\pi}{2}(D1 + D2) - \frac{D_2 - D_1}{4C}$$

Where, C = Centre distance, in $\Rightarrow$ $C = \left(\dfrac{D_2 + D_1}{2}\right) + D_1$

So, using D_1 = 100mm and D_2 = 250mm (Khurmi and Grupta, 2005)

$$C = \left(\frac{250+100}{2}\right) + 100 = 275mm$$

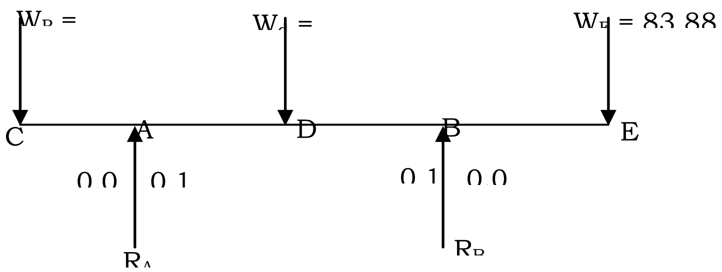
∴ The length at belt;

$$L = 2*275 + \frac{3.142}{2}(100 + 250) - \frac{(250-100)}{6*275}$$

∴ L = 550 + 549.85 – 0.1364 = 1099.71mm = **1.099m**

Bearing Selection

Static Load on Bearings



$R_A + R_B = W_P + W_d + W_F$

$R_A + R_B = 19.62 + 588.71 + 83.88$

$R_A + R_B = 692.21$ ………………………………….(1)

$M_B = O$

R_A x 0.3 + 83.88 x 0.06 = 19.62 x 0.35 + 588.71∗0.15

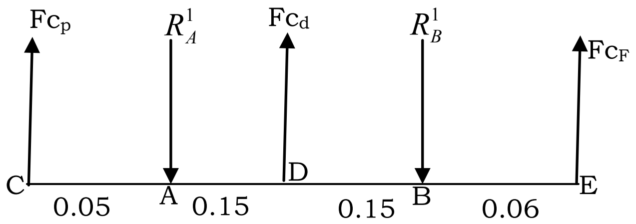
$0.3R_A$ + 5.033 = 6.867 + 88.307

$0.3R_A$ = 90.142

$_A$ = 300.67N = 0.3KN
from (1)
R_A = 692.21 – 300.47
= 391.74N = 0.39KN

Therefore, the static loads are;
R_A = 0.3KN
R_B = 0.39KN

The Dynamic Loads



$Fc_p = m_p r_p w^2$, Where r = $\frac{100}{2}$ = 2 x 0.05 x 2332 = 5426.9N = 5.43KN

$Fc_d = m_d r_d w^2$ = 58.87 x 0.09 x 233^2 = 287639.41N = 287.64KN

$Fc_F = M_F r_F W^2$ = 8.55 x 0.15 x 233^2 = 69625.64N = 69.625KN

Where;
Fc_P = Centrifugal force on pulley
Fc_d = Centrifugal force on drum
Fc_F = Centrifugal force on flywheel
W = 233 r.p.m cycling speed

$$Md = \frac{Wd}{10} = \frac{588.71}{10} = \underline{\mathbf{58.87kg}}$$

The Dynamic Reactions (Loads)

$R_A^1 + R_B^1$ = $Fc_p + Fc_d + Fc_F$ = 5.43 + 287.64 + 69.625

$\Rightarrow \quad R_A^1 + R_B^1 = 362.69$ ……………………………………….(1)

$\sum \quad M_B = O$

Fc_p x 0.35 + Fc_d x 0.15 = $R_A^1 * 0.3 + Fc_F \; x \; 0.06$

5.43∗0.35 + 287.64∗0.15 = 0.3 R_A^1 + 69.625∗0.06

1.9005 + 43.146 = 0.3 R_A^1 + 4.178

$\Rightarrow$ 0.3 = 40.869

$$R_A^1 = \frac{40.869}{0.3} = 136.23KN$$

and subst. into (1) we have;

$R_B^1 = 362.69 - 136.23$

= 226.46KN

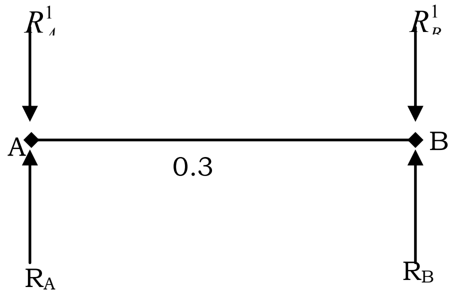
Therefore, the dynamic loads are;

R_A^1 = 136.23KN

R_B^1 = 226.46KN

- The maximum and minimum loads on the bearing; And will be determined when the line of action of the dynamic load coincides with that of the static loads.

Therefore, the set up now becomes;



The maximum load on Bearing A;

= R_A + R_A^1 = 0.3 + 1136.23

= **+136.53KN**

The minimum load on Bearing A;

= R_A - R_A^1 = 0.3 – 136.23

= **-135.93KN**

The minimum load on Bearing B;

= R_B + R_B^1 = 0.39 + 226.46

= **+226.85KN**

The bearing load on Bearing B;

= R_B - R_B^1 = 0.39 – 226.46

= **-226.07KN**

IV. Recommendation

This machine is highly recommended for domestic application especially for ruler dwellers since it is very simple to operate and does not require any fuel for its operation and it is very efficient and affordable. Therefore, efforts should be made to adopt and popularize this design.

V. Conclusion

The design analysis of a humanly powered cassava grinding machine was done and discovered to be 56% efficient with mechanical advantage and velocity ratios less than one (1) which confirms it to be a simple and easy machine to operate with an out put capacity fo 58.59kg of cassava per hour thus, the machine is very economical and viable for domestic application especially in rural areas.

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