

## Development and Testing of a Hammer Mill

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### *Abstract*

*The hammer mill was designed and constructed from locally available materials for grinding grain particles such as maize, millet, guinea corn and other coarse materials of cassava tuber, yam tuber, beans, etc. into small size enough to pass through the holes of the cylindrical sieve positioned beneath the hammer assembly.*

*The grinding process is achieved by the use of a hammer in beating the material fed into fine particles; the fineness aimed depends on the detachable screen with aperture sizes ranging from 87 $\mu$ m to 2 mm. Based on the power ratings and output shaft speed of the existing grinding machines in industries like flour mill, it was found that the main shaft speed of 700 rpm transmitted by a belt drive from a one-horse-power electric motor is suitable to mill effectively.*

*The machine was designed to be power-operated and portable with overall dimensions of 900 x 500 x 400 mm. The economic evaluation of the machine revealed that the material worth of US\$85.20 was used for its construction.*

**Keywords:** Grinding, grain particles, shaft, machine, fineness, shaft speed.

### **Introduction**

The hammer mill, which can otherwise be referred to as Cereal Miller, is designed for processing, grinding, and sieving all kinds of cereal grains, such as maize, wheat, millet, corn, sorghum, wheat. It can also process non-cereal materials such as dry cassava tuber and yam tuber.

Cereal processing is complex. The principal procedure is milling; that is, the grinding of the grain so that it can be cooked and rendered into an attractive foodstuff. In ancient time, the cereal grains were crushed between two stones and made into crude cake. But the advent of modern automated systems employing steel material such as hammer mill have revolutionized the processing of cereals and their availability as human foods and other purposes (Donnel 1983).

The machine is of hammer mill type. In this case, there is hammer-like projection mounted on a shaft. The hammer revolves at high speed and grinds the materials fed into pieces by beating. Moreover, the machine can mill only the dry materials.

The machine is incorporated with a detachable sieving mechanism to ensure fineness of cereal grain ground. The industrial screen - the main components responsible for sieving - is made of wire cloth with aperture sizes ranging from 870  $\mu$ m to 2 mm.

The machine cannot be operated manually. The electrical operation is effected by the use of one horsepower electric motor with speed of 1,400 rev/mm. The machine can handle 5 kg of cereal grains in a single operation lasting 15 min.

The entire construction is brought about by locally sourced material thereby making the cost not prohibitive. The machine elements are easily accessible and detachable to facilitate assembling and maintenance process. Although the machine is sufficiently rugged to function properly for a reasonable long period, it is cheap enough to be economically feasible.

The world population is increasing at the rate of 75 million a year. Of the present world population 6,850 millions, three quarter live in developing countries where cereals play the leading dietary role (Anon. 1980). It becomes imperative to have efficient milling machines

such as the universal milling machine to ensure the processed cereals availability not only in time but also in right quantity.

Moreover, cereals are important components of animal feeds. The cereals mainly used include maize, wheat, millet, etc. For large-scale animal feeds, the cereals are ground into paste for mixing with other food components. Without the grinding machine, the use of milled cereal in non-food products such as flour for manufacturing sticking paste and industrial alcohol, and wheat gluten for core binder in the casting of metal would be particularly impossible.

### Design Analysis

#### Determination of the Shaft Speed

To calculate the shaft speed, the following parameters are used:

$$\frac{D_1}{D_2} = \frac{N_2}{N_1} \dots\dots\dots 1$$

Where  
 N<sub>1</sub> = revolution of the smaller pulley, rpm.  
 N<sub>2</sub> = revolution of the larger pulley, rpm.

This shaft speed is only obtained when there is no slip condition of the belt over the pulley. When slip and creep condition is present, the value (700 rpm) is reduced by 4% (Spolt 1988)

#### Determination of Length of the Belt

Assume the center distance between the larger pulley and the smaller pulley = 600 mm, the pitch length of the belt is given by (John and Stephens 1984)

$$L = 2C + 1.57(D_2 + D_1) + \frac{(D_1 + D_2)}{4C} \dots\dots 2$$

Where  
 L = length of the belt, mm  
 C = center distance between larger pulley and the smaller one, mm  
 From the standard table, a belt designated as A60 was selected.

#### Determination of the Belt Contact Angle

The belt contact angle is given by equation 3:

$$\sin^{-1} \beta = \frac{(R - r)}{C} \dots\dots\dots 3$$

Where  
 R = radius of the large pulley, mm  
 R = radius of the smaller pulley, mm

The angles of wrap for the pulleys are given by:

$$\alpha_1 = 180 - \sin^{-1} \left( \frac{R - r}{C} \right) \dots\dots\dots 4$$

$$\alpha_2 = 180 + \sin^{-1} \left( \frac{R - r}{C} \right) \dots\dots\dots 5$$

Where  
 α<sub>1</sub> = angle of wrap for the smaller pulley, deg  
 α<sub>2</sub> = angle of wrap for the larger pulley, deg  
 Comparing the capacities, e<sup>μ a/sin 1/2 θ</sup> of the pulley,

Using μ = 0.25; θ = 40°  
 For the smaller pulley e<sup>0.25 x 3.04/sin20</sup> = 9.22  
 For the larger pulley e<sup>0.25 x 3.04/sin20</sup> = 10.68

Since that of smaller pulley is smaller, the smaller pulley governs the design.

#### Determination of the Belt Tension

The belt tension can use equation 6 below (Maitra and Prasad 1985):

$$T_2 = \frac{(T_1 - MV_2)}{\exp \left( \frac{\mu \alpha}{\sin \frac{1}{2} \theta} \right)} \dots\dots\dots 6$$

and

$$T_1 = SA \dots\dots\dots 7$$

Where  
 T<sub>1</sub> = the tension in the tight side of belt, N  
 T<sub>2</sub> = the tension in the slack side of belt, N  
 S = the maximum permissible belt stress, MN/m<sup>2</sup>  
 A = area of belt,  
 M = mass per unit length of belt  
 v = linear velocity of belt  
 mv<sup>2</sup> = centrifugal force acting on the belt

**Determination of the Torque and Power Transmitted to the Shaft**

Power transmitted to the shaft is given by

$$P = (T_1 - T_2)V \dots\dots\dots 8$$

Torque at the main shaft is given by Spolt (1988)

$$T = (T_1 - T_2)R \dots\dots\dots 9$$

**Determination of the Hammer Weight**

$$W_h = m_h g \dots\dots\dots 10$$

It can be seen that the action of the weight of hammer shaft on the main shaft is negligible.

**Determination of the Centrifugal Force Exerted by the Hammer**

Centrifugal force exerted by the hammer can be calculated from equation 11 as given by:

$$F_c = \frac{mv^2}{r} \dots\dots\dots 11$$

The angular velocity of the hammer is given by

$$\omega = \frac{2\pi r N}{60} \dots\dots\dots 12$$

**Determination of the Hammer Shaft Diameter**

The bending moment on the shaft is given by (Ryder 1996)

$$M_{b(max)} = \frac{Wl^2}{8} \dots\dots\dots 13$$

Since the bending moment that can be carried by a beam is a measure of the strength of the beam and this depend upon,  $I/y\mu a \theta$  (Ryder 1996).

$$\sigma_{s(allowable)} = \frac{M_b Y_{max}}{I} \dots\dots\dots 14$$

$$\frac{I}{Y_{max}} = Z \Rightarrow \sigma_{s(allowabl\theta)} = \frac{M_b}{Z} \dots\dots\dots 15$$

Where

$Y_{max}$  = distance from neutral axis to outer fibers  
 $I$  = moment of inertia  
 $Z$  = Section modulus

For a solid round bar:

$$I = \frac{\pi d^4}{64} \dots\dots\dots 16$$

$$Z = \frac{\pi d^3}{32} \dots\dots\dots 17$$

**Determination of the Maximum Bending Moment**

The position of the electric motor in relation to the main shaft is such that  $T_1$  and  $T_2$  act vertically downward and  $T_1 + T_2 = 148N$ .

The overall loading system on the shaft is as shown in Fig. 1

From the shear force diagram of Fig. 1, it is obvious that  $b$  is the point of maximum bending moment.

**Determination of the Shaft Diameter**

The ASME code equation for a solid shaft having little or no axial loading is:

$$d^3 = \frac{16}{\pi \sigma_s} \sqrt{(K_b M_b)^2 + (K_t M_t)^2} \dots\dots 18$$

**Construction Details of Major Parts**

**Main Shaft:** A 32 mm diameter rod was cut to a 480 mm length using power hacksaw. The shaft was then faced and center drilled. It was held between its centers and step turned to 25 and 30 mm. Keyway was cut on it using milling machine.

**Beater Shaft:** 5 mm diameter rod was cut, faced and turned to 12 mm diameter. The length of the shaft is 70 mm.

**Hammer:** A 3 mm thick bar of 30 mm width was cut into 70 mm pieces. A hole of 13 mm was drilled at the bottom of each hammer, using twist drill, to enable it to be put into position on the hammer shaft.

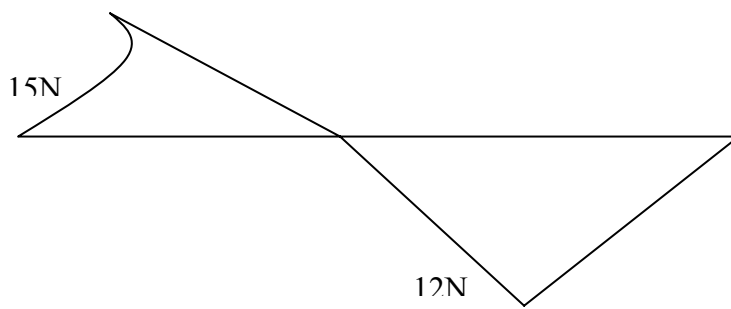
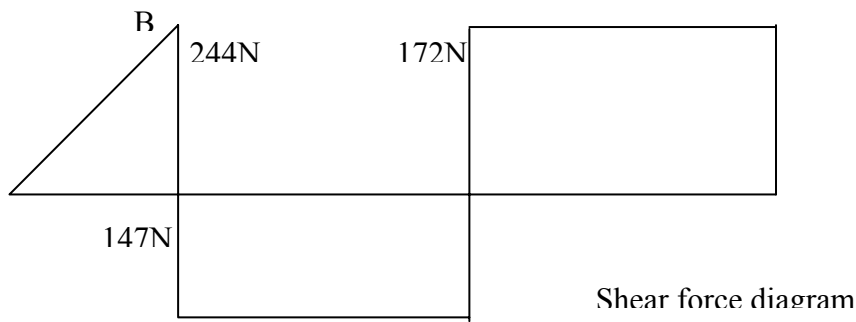
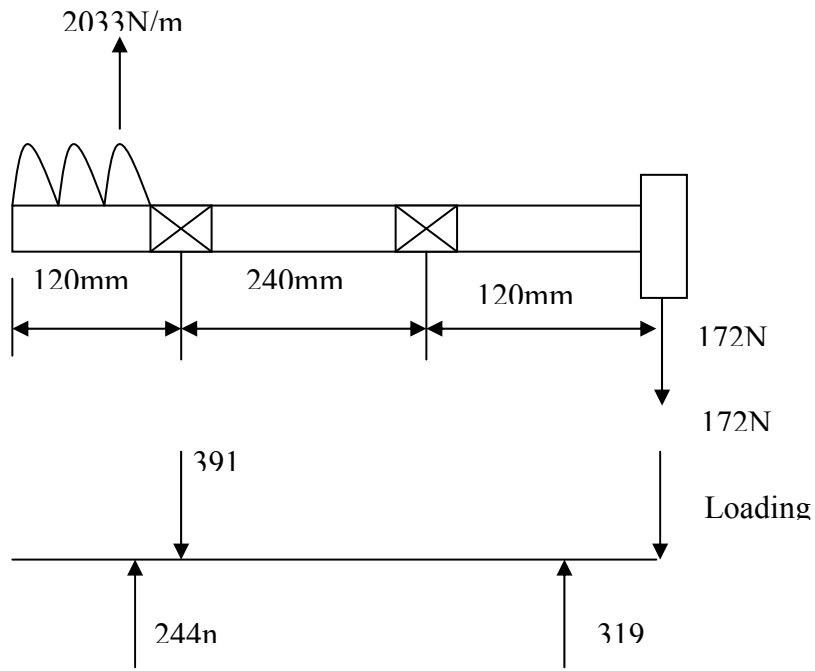


Fig. 1. Shear force and bending moment diagram.

Table 1. Results of the calculated parameters

Parameters	Symbol	Value	Unit
Shaft Speed	$N_2$	672	rpm
Length of Belt	L	1484.10	mm
Belt Contact Angle	B	2.87	Degree
Angle of wrap for smaller pulley	$\alpha_1$	174	Degree
Angle of wrap for larger pulley	$\alpha_2$	184.74	Degree
Tension in the slack side of belt	$T_2$	16.23	N
Tension in the tight side of belt	$T_1$	132.01	N
Torque transmitted to the shaft	T	7	Nm
Power transmitted to the shaft	P	490	W
Weight of hammer	$W_{hammer}$	0.47	N
Centrifugal force exerted by the hammer	C.F	244.04	N
Diameter of hammer shaft	D	8.7	mm
Weight of hammer shaft	$W_s$	0.243	N
Maximum bending moment	$M_b$ (max)	20.54	Nm
Diameter of main shaft	d	16	Mm

**Hopper:** It is pyramidal in shape and it was made from 1.8 mm thick plate. It was dimensioned 204 x 204 mm top opening, 50 x 50 mm base opening and height 240 mm. The plate was marked and cut to sizes and then welded together.

**Collar:** A hollow pipe of internal diameter of 13 mm was cut to 10 mm length.

**Supporting Bar (Stand):** Two 100 x 40 mm mild steel bar was cut on 3 mm thick plate using hacksaw. Three holes of 25, 12 and 12 mm were drilled.

## Testing

Testing is a vital step in the process of machine development. After the design and construction, testing is necessary in order to:

- determine the performance of the machine,
- expose defect and area of possible improvement, and
- appreciate the level of success in the research.

Thus, it is important to test run a machine to determine its work ability and efficiency.

### Testing Using Dry Cassava Tuber

A 5 kg of dry cassava was fed into the hopper and the hammer mill was switched on. The grinding tin was noted. This was repeated for four times and averages used for calculation.

### Test Using Dry Maize

The same procedure was reported using 5 kg of dry maize

## Results and Discussion

Table 1. Hammer mill test results using cassava

Trial	Mass of cassava before grinding (kg)	Mass of cassava after grinding (kg)	Time taken (min)
1	5	4.8	15
2	5	4.7	14
3	5	4.8	15
4	5	4.9	16
Aver	5	4.8	15

Average mass of the cassava before grinding = 5 kg  
 Average mass of the cassava after grinding = 4.8 kg  
 Average time taken = 15 min

$$Crushing\ efficiency = \frac{Mass\ of\ output\ material}{Mass\ of\ input\ material} \times 100$$

$$Losses = \frac{M_b - M_a}{M_b}$$

where

M<sub>b</sub> = Mass before grinding

M<sub>a</sub> = Mass after grinding

Table 2. Hammer mill test results using maize

Trials	Mass of maize before grinding (kg)	Mass of maize after grinding (kg)	Time taken (min)
1	5	4.7	15
2	5	4.6	14
3	5	4.8	16
4	5	4.7	15
Aver.	5	4.775	15

Average mass of the maize before grinding = 5 kg

Average mass of the maize after grinding = 4.775 kg

Average time taken = 15 min

Crushing capacity is defined as the mass of material ground in kg/hr (Mott 1980).

### Discussion

From the result of the test, the crushing efficiency of the machine was found to be 96 and 94% for dry cassava and dry maize, respectively. The slight difference may be because of cassava was softer than maize. It is clear from the crushing capacity and efficiency above that the performance of the machine is satisfactorily. The loss obtained was due to the sticking of the powdery materials to the wall of the crushing hammer and some strains that did not pass through the screen.

### Conclusion

The paper presents the design of an electrically operated universal milling machine for both domestic and commercial purposes. From the design consideration and analysis, portability, reliability, safety, serviceability and cost of construction were given due consideration.

The construction was successfully carried out and a universal-milling machine with the following parameters produced.

Crushing efficiency = 96%  
 Crushing capacity = 31 kg/hr  
 Crushing losses = 0.04

Taking into consideration the various size of material needed, the screen is detachable, making it possible to fix screens with different sieving apertures.

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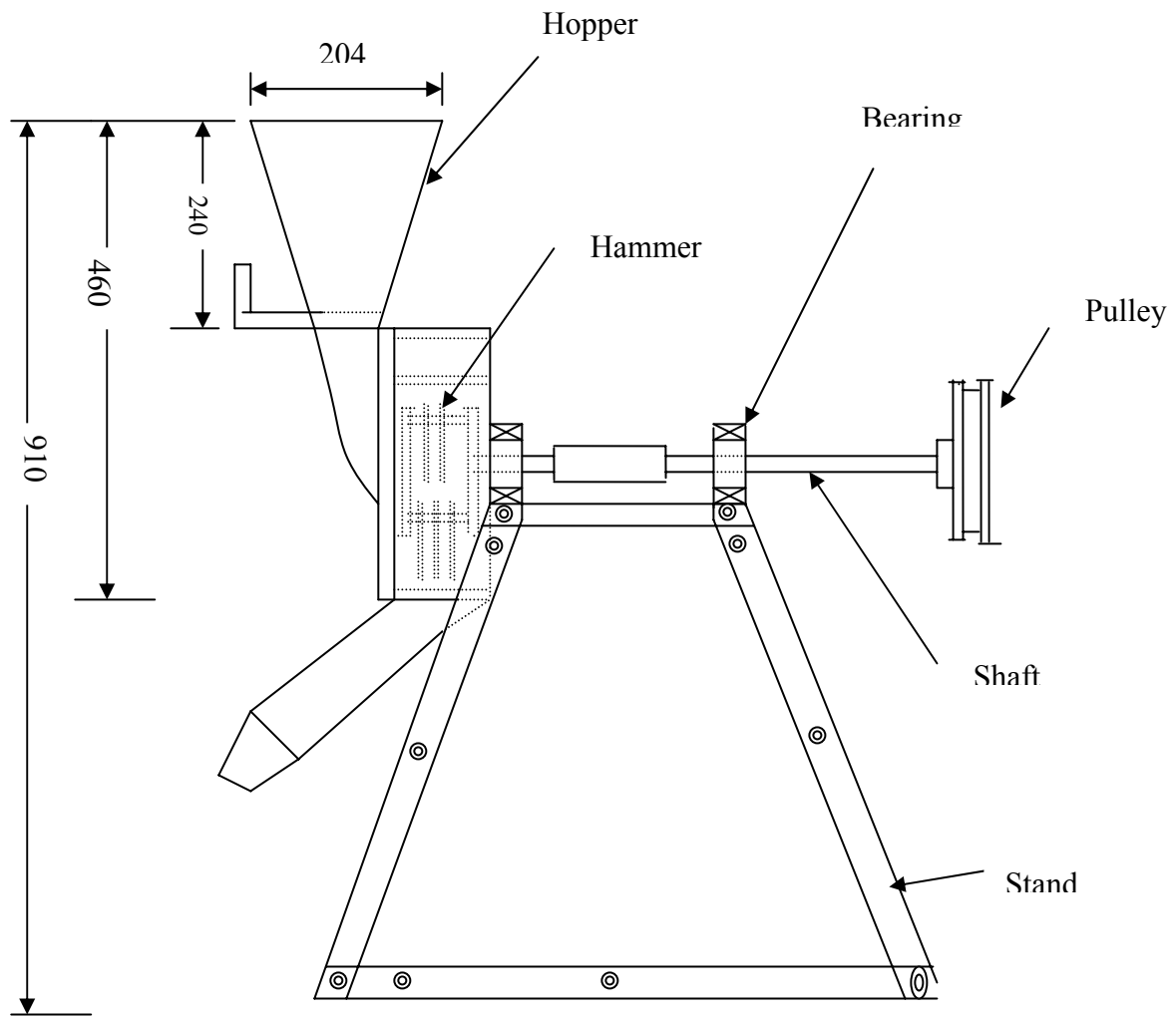


Fig. 2. Side view of the Hammer Mill

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