Design and Development of Impact Hammer Mill for Limestone Crushing for Acidic Soil

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Abstract
Soil acidity becomes a serious threat to crop production in most highlands of Ethiopia particularly in Western parts of Oromia. Frequent tillage, removal of crop residues and mono-cropping and heavy rainfall contributes to soil acidification by leaching of cations. Agricultural limestone raises soil pH and reduces solubility of potentially toxic elements such as hydrogen, aluminum (Al^{3+}) and manganese (Mn) at optimum nutrient uptake by crops. In an effort to alleviate the problems associated with soil acidity, a motorized agricultural limestone crusher was designed and fabricated. The prototype of limestone crusher machine has a feed table, hammer plate, concave sieve, discharge chute, and supporting frame.
Keywords: Hammer mill, limestone, particles size and crushing.
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1. INTRODUCTION

A productive and sustainable agricultural system is fundamental to the well-being of a nation and a cornerstone of its development. Agriculture employs more than 80% of the population, contributes 40% to the GDP and more than 60% to exports in Ethiopia (Tegbaru, 2015). It is the major source of income, employment, food security and survival for the majority of the population. However, decreasing soil fertility due to soil erosion, continuous cropping, soil acidity and inadequate sustainable soil fertility management lead agricultural sector to low productivity (Van Straaten, 2002; Rutunga *et al.*, 2007; Nyirinkwaya B., 2013). The disruption of fertilizer supplies by economic, political and other pressures can seriously impede the development and livelihood of rural inhabitants.

Soil acidity becomes a serious threat to crop production in most highlands of Ethiopia in general and in the western part of the country in particular. Studies show that about 43% of the total land area of Ethiopia is affected by acidity (Tegbaru, 2015). About 28% of the cultivated agricultural land is acidic (pH<5.5) with significant impact on productivity for more than 3 million hectare of land. Areas which are most prone to acidity are the North-western, South-western, Southern and central regions of the country. Most of the acid affected areas are in the highlands where wheat, maize and teff are predominantly grown. Total strongly acid affected area in Oromia, SNNPE and Amhara were 1,437,887, 1,071,400, 582,600 hector respectively.

Kefeni (1992) found that the loss of nutrients from the eroded soil in a 100-ha catchment area in Anjeni in the Amhara region was about 210 kg (N), 680 kg (P) and 160 kg organic matter per hectare per year. Specific studies of Mesfin (2007) revealed that soils around Asosa and Welega in aggregate, 67% had pH values less than 6 and were very strongly to strongly acidic and 2.2 percent were extremely acidic (pH<4.5). Thirty four percent were very strongly acidic (pH = 4.5 to 5.0) 32.8% were strongly acidic (pH = 5.1 to 5.5) and 27% were moderately acidic with pH range of 5.6 to 6.0. Of the total, only three percent were slightly acidic (pH = 6.1-6.5), and 1% neutral.

Soil acidification is a natural process however, some of the major causes that speeds up soil acidification include: frequent tillage, removal of crop residues and mono-cropping. Frequent application of urea also would enhance soil acidity in the long term. Soil acidity restricts crop production mainly by impairing root growth and limiting nutrient and water uptake. It also creates toxic soil solution that hinders the growth of roots and micro-organism activity. Crops that are grown in acidic soils have a significantly stunted growth rate and are not very responsive to fertilizers. Lime can shift soil acidity towards neutral state and render nutrients more available to crops. Lime amounts of 2-5t/ha is typically needed to neutralize acid soils sufficiently for crop production, depending on the type of soil and levels of acidity (Tegbaru, 2015). The Ethiopian government is undertaking initiatives and planned to rehabilitate 226,000 ha of agricultural land by the end of the GTP II period. To achieve this, it is planned to produce 450,000-900,000 tons of lime. However, up to now, the achievement is quite low.

Limestone resource is a geological nutrient asset that could sustain and enhance crop production is necessary for soil amendments (Bosse *et al.*, 1996). Agro-minerals are physically modified by crushing and grinding. Hammer mill is the most widely used in mineral processing industries because of its desirable characteristics which include the ability to handle a wide variety of raw materials, ability to handle hard stray objects and its robustness (Ajaka and Adesina, 2014). Most of the existing lime stone hammer mill machines are designed for very large-scale production by the national companies. Therefore, to supplement the existing limestone production industry, small scale limestone crushing machine which can substitute import, produced and maintained locally, minimize cost (consider buying capacity of our farmers), simple, durable, efficient and effective machine should be developed.

Mesozoic limestone, dolomite and marl are largely deposited in western and northern Ethiopia. Large areas of central plateau near Ambo town, in Didessa valley and smaller deposits occur in Kella area south of Addis Ababa.

Objectives

- To design and develop impact hammer mill machine prototype

2. Methodologies used

2.1. Research Site

The experiment was conducted at BAERC, which is located in East Wollega Zone of Oromia National Regional State, Ethiopia. The machine was fabricated at BAERC work shop. The Center lies between 9^0 04'45" to 9^0 07'15"N latitudes and 37^0 02' to 37^0 07'E longitudes.

2.2. Materials used

A prototype of the designed impact hammer mill was made from the drawings using basic manufacturing tools and equipment such as engineers try square, steel ruler, clamps, scriber, vernier caliper, dot punch, lathe machine, fixed grinder, bending and rolling machines, arc welding machines, hand drill and grinder, power hacksaw, milling machine and spraying gun and different thickness of sheet metal (1.5, 2, 3, 4 mm) , angle iron (40x40x4mm), flat iron, pulley, bearings, engine, round bar, shaft Ø (20 and 40 mm), concave sieve (2, 4, 6 mm hole diameter) bolt and nut. Instruments such measuring tape, weighing balance, oven dry, tenso-meter, impact tester, different aperture size of sieve was used during prototype construction and data collection.

2.3. Design Analysis and Calculation

Design Consideration: The major components of the machine include the hammers, shaft, belt and pulley, casing and motor power.

2.3.1. Impact Bending Stress

For the maximum feed rates of (10.5 kg/min = 0.174kg/s) and maximum revolution of 900 rpm of the rotor, there were 15/s impacts by 4 rotors in one second.

$$\text{Tonnage/impact weight (w)} = \left(0.174kg/s \times 9.81m/s^2\right)/15/s = 0.11N$$

a. When impact hammer mill blow bar was subjected to a concentrated load at the mid of its span as indicate in (figure 3.4) the following equation was used according to Khurmi and Gupta, (2008).

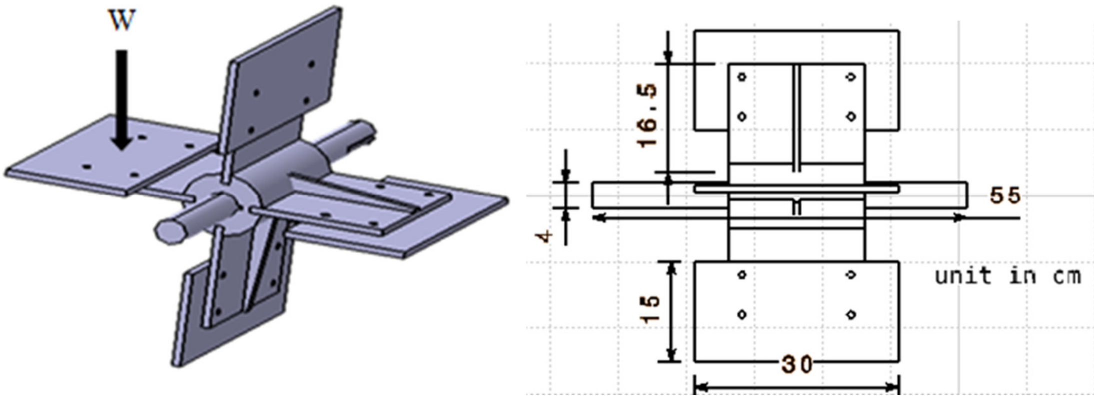

Figure 2.1 Hammer mill cantilever subjected to a concentrated load at the mid of its span.

$$W(h+y) = \frac{Py}{2} \quad where\ , \quad P = \frac{3EIy}{l^3}, \qquad 0.11(0.3+y) = \frac{3EIy^2}{2l^3} \qquad (Equation\quad 1)$$

$$I = \frac{bd^3}{12} = \frac{0.3 * 0.165^3}{12} = 4.32 * 10^4\ mm^4$$

Young's modulus E=165Gpa=$165*10^3$N/mm^2

Yield stress $\sigma_{ys} = 350\ Mpa = 350\ N/mm^2, \rho = 7.8 g/cm^3$

Height of fall = 350 mm

Weight of each hammer = 7.25 kg

The hammer was considered to act like a cantilevered beam fixed on a rotor shaft for

$EI=165*10^3 N/mm^2*4.32*10^4 mm^4 = 7.128*10^9 N/mm^2$

$$0.11(350\,mm + y) = \frac{3\times 7.128\times 10^9 \times y^2\,Nmm^2}{2\times 300^3}$$. By quadratic equation,

$$y = \frac{-b \pm \sqrt{b-4ac}}{2a} = 0.352\ mm \qquad (Equation\quad 2)$$

$$P = \frac{3EIy}{l^3} = \frac{3\times 7.128\times 10^9\,Nmm^2 \times 0.352\,mm}{300\,mm^3} = 278.78\,N \qquad (Equation\quad 3)$$

So, maximum moment = 278.78Nx165/2 = $23*10^3$ Nmm

Now we have allowable stress $\sigma_{ys} = 500Mpa$, so maximum allowable moment was determined as follows;

$$\sigma ys \times Z = \frac{\sigma}{\frac{d}{2}} = 500\times 300\times 12\times 12\times \frac{2}{12} = 36\times 10^5\ Nmm$$

where,

P = Equivalent static force (N)
y = bending (mm)
I = the moment of inertia (mm^4)
E = Modulus of elasticity (N/mm^2)
Yield stress $\sigma ys = 350Mpa,\quad \rho = 7800kg/m^3$
Height of fall (0.35m)
Weight of each hammers (7.25kg)
The ratio *I/y* is known as ***section modulus*** and is denoted by *Z.*

Since, Maximum allowable moment > Maximum moment developed by the machine and the design was safe for this condition.

a. When the impact hammer mill plate blow bar was subjected to a concentrated load at the tip it was calculated using equation(3) as shown in the figure 3.5,

$$P = \frac{3EIy}{l^3} = 278.78\,N \qquad (Equation\quad 3)$$

Maximum moment induced by machine = Pxl=278.78Nx165mm = $4.6x10^4$Nmm

Maximum allowable moment = $\sigma \times Z = \frac{350\times 300\times 12^2}{6} = 2.52\times 10^6\ Nmm$ since maximum moment developed by machine was < maximum allowable moments, hence the design was safe!

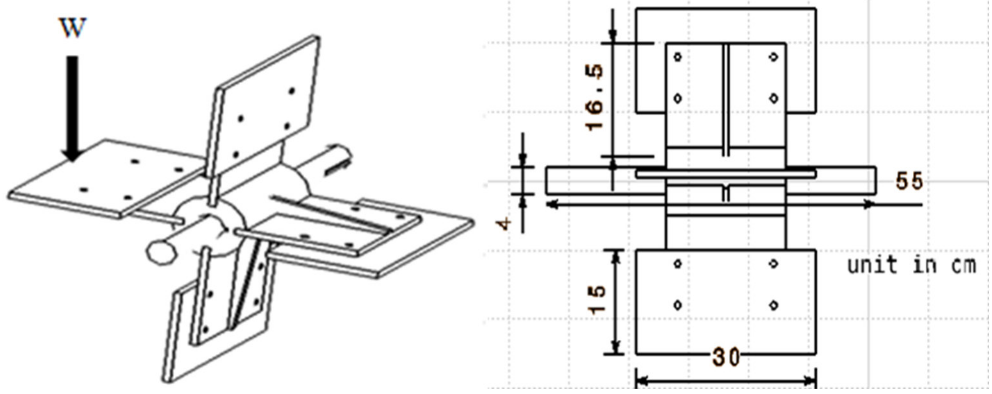

Figure 2.2. Cantilever blow bar subjected to a concentrated load at the tip

b. Impact bending stress due to cantilever beam subjected to uniformly distributed.

Total tonnage /hammer/ impact = 0.11 N, Length of exposed blow bar l=165(2/3) = 66 mm, Height of fall h = 350 mm. Since the weight was distributed uniformly over the l = 66 mm, Small work done due to, Impact distributed load was

$$W = \frac{w}{l}(hl - \frac{wl^5}{20\,EI}) = 48.99\ Nmm \qquad (Equation\quad 4)$$

Also static work done $= -\frac{Pwl^5}{40\,EI}$, so we get $48.99 = \frac{-P\times 0.11\times 66^5}{40\times 7.128\times 10^9}$, $P = 79688.25\ N$

Maximum bending moment $= \frac{Pl}{2} = 79688.25 \times \frac{66}{2} = 2.6 \times 10^{6}\ Nmm$,

Maximum stress induced by machine was $\sigma b = \frac{M}{Z} = \frac{2M}{Id} = 146.0.65\ \frac{N}{mm^{2}}$

The maximum allowable stress $= 500\ \frac{N}{mm^{2}}$ so, the design was safe in accordance to this condition too. In static load shearing, work done, $w = \frac{Fys}{2}$, $ys = \frac{6Pl}{10bGd} = \frac{6 \times 79688.25 \times 66}{10 \times 300 \times 12 \times 80 \times 10^{3}}$, P = 79688.25N, G = 80Gpa, ys = 0.01mm where, ys = displacement.

2.3.2. Determination of power requirement

The power required to operate the hammer milling machine was determined according to Bond, (1952) and was given as;

$$E = 0.3162 \times Wi\left(\frac{1}{\sqrt{D_{pa}}} - \frac{1}{\sqrt{D_{pb}}}\right) \qquad (Equation\ 5)$$

Wi = 12.74KWhr/tone = 45.864kWs/kg (for D_{pa} and D_{pb} Appendix A1).

$$E = 0.3162 \times \frac{45.864\text{kWs}}{\text{kg}} \times (1.9) = \frac{27.55\text{kJ}}{\text{kg}}$$

The power required to operate the hammer milling machine was calculated using the following equation and the maximum crushing capacity, CC = 0.174Kg/s was used to determine the power.

$$P_{p} = 0.3162\ qWi\left(\frac{1}{\sqrt{D_{pa}}} - \frac{1}{\sqrt{D_{pb}}}\right) \qquad (Equation\ 6)$$

P = 6.5 hp

Considering 10% power loss due to friction, the total power required to crush was = 6.5x0.1 + 6.5 = 7.15HP

where;

E = energy required to crush limestone (J/kg)
P = Crushing power (kw)
q = feed rate in (kg/min)
D_{pb} = average particle diameter before crushing (mm)
D_{pa} = average particle diameter after crushing (mm)
Wi = work index (Kwh/t)

So, a diesel or petrol engine of more than 7.15HP can be used to operate the machine. An electric motor of the same power rating can also be used where electrical power is available.

2.3.3. Selection of pulley

The machine required two pulleys; one driving pulley were mounted on the crank shaft of the engine and the driven pulleys were mounted on impact hammer mill shaft. Pulleys made from cast iron with 0.138 m diameter for driving pulley and 0.48 m diameter for driven pulley were selected based on its availability, low cost and high performance. Based on the required revolution per minute, the diameter of driven pulley was determined according to Khurmi R.S. and Gupta J.K., (2013)

$$N_{1}D_{1} = N_{2}D_{2} \qquad (Equation\ 7)$$

where;

D_1 = diameter of the driver = 0.138m
D_2 = diameter of the driven = 0.46m
N_1 = speed of the driver = 3000rpm
N_2 = speed of the driven = 900 rpm

2.3.4. Belt selection and determination of its length and center distance

V-belt and pulley arrangements were adopted in this work to transmit power from the engine to the shaft of impact hammer mill. The main reasons for adopting the v- belt drive was its flexibility, simplicity and low maintenance costs. Additionally, the v- belt has the ability to absorb shocks there by mitigating the effect of vibratory forces (Khurmi and Gupta, 2005). Appendix B Table 1 gives list of V-belts (A, B, C, D and E) and their standard dimensions and power carrying capacity. Length of the open belt was calculated according to Khurmi and Gupta (2005) as given below:

$$Lp = 2C + \frac{\pi}{2}(Dp + dp) + \left(\frac{Dp + dp}{4C}\right)^{2} \qquad (Equation\ 8)$$

$$2 \times 0.6 + \frac{\pi}{2}(0.138 + 0.46) + \frac{(0.138 + 0.46)^{2}}{4 \times 0.6} = 2.288\ m$$

where,

L_p= effective length of the belt, m
C = center distance, m

$$\frac{D_1 + D_2}{2} + D_1 \le C \le 2(D_1 + D_2) \qquad (Equation \quad 9)$$

$$0.437\ m \le C \le 1.2m$$

The closest standard length of the belt was selected from standard table and found to be 2.312m (B 91 V-Belt) and C = 0.6m taken value.

2.3.5. Determination of belt contact angle

The belt contact angle was given by the following equation (Khurmi and Gupta, 2005).

$$\sin^{-1}\beta = \frac{R - r}{c},$$

$$\sin^{-1}\beta = \frac{0.23 - 0.069}{0.6}, \qquad \beta = 15.56^0 \qquad (Equation \quad 10)$$

where;

R = radius of the larger pulley, m
r = radius of the smaller pulley, m

Wrap angle were determined using the equations given below.

$$\alpha_1 = 180 - 2\sin^{-1}\left(\frac{R - r}{c}\right)$$

$$= 180 - 2\sin^{-1}(0.268) = 148.87^0 = 2.6 rad \qquad (Equation \quad 11)$$

$$\alpha_2 = 180 + 2\sin^{-1}\left(\frac{R - r}{c}\right)$$

$$= 180 + 2\sin^{-1}(0.268) = 211.13^0 = 3.68 rad \qquad (Equation \quad 12)$$

where;

α_1= angle of wrap for the smaller pulley, rad.
α_2= angle of wrap for the larger pulley, rad.
C = center to center distance between small and large pulley, mm

2.3.6. Determination of power transmitted to the shaft

Power transmitted per belt to the shaft was determined according to Barber (2003).

$$V = \frac{\pi DN}{60} = \frac{\pi \times 0.46 \times 900}{60} = 21.68\, m/s$$

$$P = (T_1 - T_2)V$$

$$715 = (T_1 - T_2) \times 21.68\, m/s$$

$$357.5 = (T_1 - T_2) \qquad (Equation \quad 13)$$

2.3.7. Determination of the belt tension

The belt tension developed in the slack side was determined according to Barber (2003)

$$2.3 \log \frac{T_1}{T_2} = \frac{\mu\alpha_1}{\sin(\theta/2)} \qquad (Equation \quad 14)$$

$$\frac{T_1}{T_2} = e^{1.23} = 3.43,$$

$$T_1 = T_2 \times 3.43$$

$$P = 3.43T_2 - T_2$$

$$T_2 = 147.12\ N$$

$$T_1 = 3.43 \times 147.12 = 504.62\ N$$

$$T_r = R(T_1 - T_2)$$

$$T_r = 0.23(504.62 - 147.12) = 82.22\ Nm$$

where;

T_r = resultant torque
T_1 and T_2 = tension in tight and slack side of the belt, N
R = radius of the bigger pulley, m
θ = Groove angle = 34°

m = mass per unit length of belt, (kg/m),

$m = bt\rho = 204 \times 10^{-6} m^2 \times 1140\ kg / m^3 = 0.23\, kg / m$

= 0.42 (coefficient of friction between rubber belt and pulley,

Khurma and Gupta, (2005)

2.3.8. Determination of the impact hammer shaft diameter

The design of shaft was based on combined shock and fatigue, bending and torsional moment. The diameter of the main shaft for a solid shaft having little or no axial loading was calculated according to Khurmi and Gupta, (2008). The tight and slick sides tensions in the belt was 504.62N, 147.12N and weight of pulley 16kg x 9.81 m/s^2 = 156.96 N respectively and ($T_1 + T_2$) = (504.62 + 147.12 + 156.96) N = 808.7 N and for hammer load (Wh) = (41 kg x 9.81 m/s) = 402.21 N.

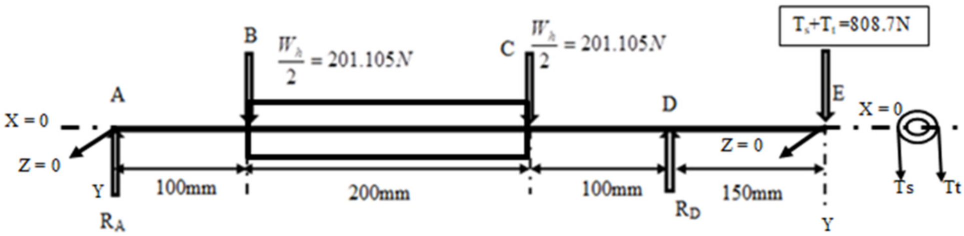

Figure 2.3. The hammer mill shaft and forces acting on it

where:

W_h = hammer weight

T_t = tension in the belt of tight side

T_S = tension in the belt of slack side.

∑MA=0, 201.105x100+201.105x300–RDx400+808.7x550 = 0

R_D =1313.07N

∑FY = 0

$R_A + R_D$ = 201.105+201.105+808.7

R_A = 1210.91 – R_D

RA = -102.16N (-ve sign shows the direction of action is down ward)

The resultant bending moments on the shaft were as follows;

∑MA = 201.105x0.1 + 201.105x0.3 – RDx0.4 + 808.7x0.55 = 0Nm

∑MB = 102.16 × 0.1 = 10.22Nm

∑MC = 102.16x0.3 + 201.105x0.2 = 43.28Nm

∑MD = 102.16x0.4 + 201.105x0.3 + 201.105x0.1 = 121.3Nm

∑ME = 102.16x0.55 + 201.105x0.45 +201.105x0.25 – R Dx0.15 = 0Nm

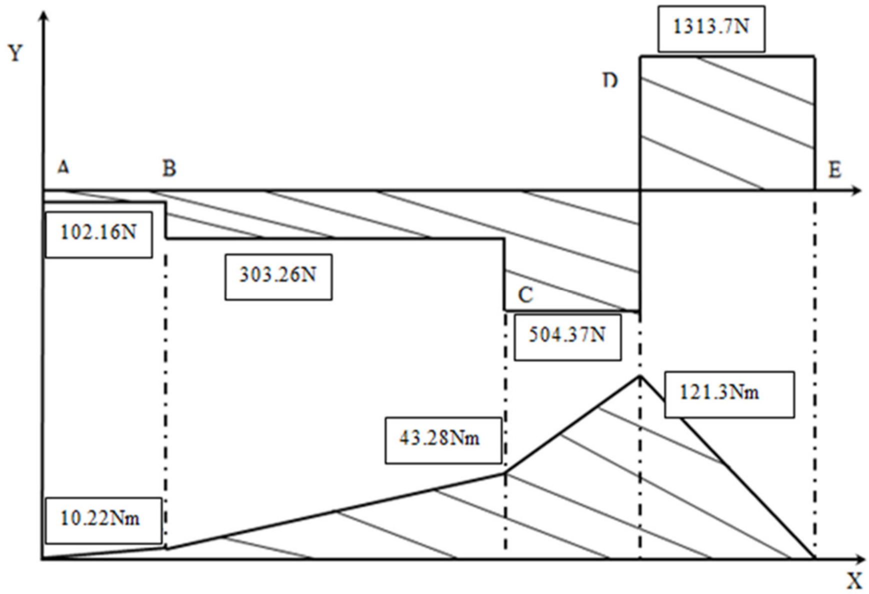

Figure 2.4. Shear and bending moment diagrams on the shaft

$$ds^3 = \frac{16}{\pi\tau}\sqrt{(K_b M_b)^2 + (K_t M_t)^2} \qquad (Equation\ 15)$$
$$= \frac{16}{\pi \times 40 \times 10^6}\sqrt{(3 \times 121.3)^2 + (2 \times 82.22)^2}$$
$$ds = 0.037\ m$$

Therefore, 40 mm shaft diameter was selected.

where,

ds = shaft diameter, m;
M_b = bending moment, Nm;
M_t = torsional moment, Nm;
Kb = Combined shock and fatigue factor applied to bending moment
Kt = Combined shock and fatigue factor applied to torsional moment
τ = Allowable shear stress of the shaft material, MN/m^2

The values of K_b and K_t were taken as 3 and 2 respectively for the suddenly applied load on the rotating shaft and the allowable shear stress of the shaft (τ) as 40 MN/m^2 based on American Society of Mechanical Engineers **(ASME)**.

Permissible angle of twist caused by torque on the shaft was determined according to the following equation.

$$\theta = \frac{584\ RtL}{Gd^4} = \frac{0.123^0}{m} \qquad (Equation\ 16)$$

Where;

θ = angle of twist, in degree
M_t = torsional moment, Nm;
L = length of shaft = 0.55 m
G = modules of rigidity 84×10^9
d = diameter of shaft, m

Note that the maximum permissible angle of twist = 1°/m., hence the shaft with in safe limit. The calculated angle of twist was less than the permissible angle of twist (1°/m). Hence, torsional deflection was satisfied.

2.4. Working Principles of Hammer Mill

Hammer mills operate on the principles of impact and pulverization (Lynch *et al.*, 2005). The limestone was fed into the hammer mill through the feed hopper. The feed hopper was chamfered to facilitate the unidirectional flow of the limestone by gravity to the milling chamber and the hammers strike the dried lime stone and broke them into small pieces. The pulverized material was prevented from leaving the milling chamber until it was reduced to fine particles. On subsequent impact by the hammers, the larger particles or uncrushed materials was recycled to the crushing chamber. Dried limestone 3.5 kg, 7 kg and 10.5 kg was feed into the chamber through the hopper and the duration of the milling process was recorded. The hammer mill consists of four fixed hammers on a drum with steel shaft, which rotates at a high speed in the housing. On the bottom of the housing and close to the tip of the hammers, it has a sieve. The fine particles passing through the sieve and was collected and the fineness of the particles was controlled by using sieves of different mesh sizes.

2.5. Description of the Machine Components and construction of prototype

The main components of the impact hammer milling machine include conveyor and feed table, impact hammer mill, sieve, delivery unit and frame as shown in the figure 3.1.

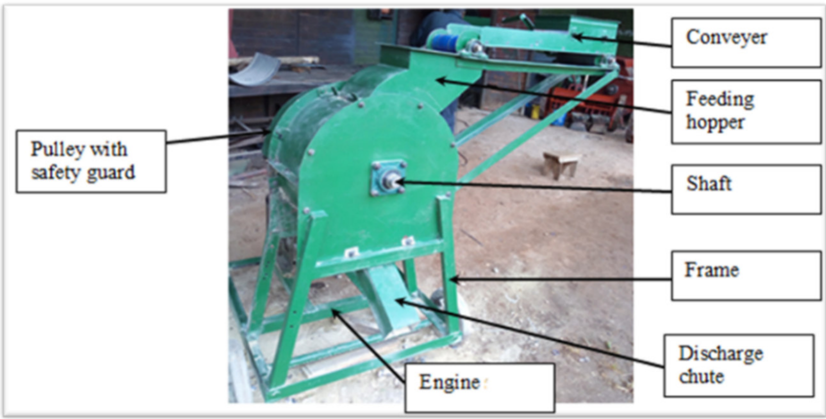

Figure 2.5. Prototype of impact hammer mill machine

3. Conclusion and recommendation

The design and construction of impact hammer mill machine have been carried out using available materials on the local market. The machine consists of pulley with safety Gard, engine, discharge chute, frame, shaft, feeding hopper and conveyer. The machine has best performance in terms of (capacity 630.32 kg/h, efficiency 99.61 %, geometric mean diameter 0.26 mm, fuel consumption 26.37ml/kg). It is recommended to look into the possibility of improving on the design and fabrication with a view of going into mass production which can also create a possibility for income generation.

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