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Redesign and Fabrication of Hammer Milling Machine for Making Pasta

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Abstract : This study Aimed to redesign a Hammer mill machine for grinding,owing to the inability of existing mill to meet the demand of waste pastaflourin Dire Dawa food Complex. Rational design by drawing and calculations and Simulation in solid works. CAD software were used to bring this milltoreality. The modified pasta grinding milling machine efficiency will be 90%..Waste of Pasta, Macaroni, Biscuits, or Breads and there by impact cost of production through minimizing waste as well as inquests safe, hygienic & efficient way of grinding these by products at Dire Dawa food complex.

Key words : Pasta Grinding, Solidwork, Belt Drive, Design.

1.Introduction

Hammermills are more widely used in feed mills because they are easier to operate and maintain and produce particles spherical in shape (Kim et al., 2002; Amerah et al., 2007). To increase production capacity of hammer mills, they are equipped with an air-assist system that draws air into the hammer mill with the product to be ground, thus allowing particles to exit through screen holes (Kim et al., 2002).Hammer mills have been reported to produce a wide range of particle sizes compared to roller mills (Seerley et al., 1988; Douglas et al., 1990; Audet, 1995). During roller milling each grain passes through the mill independently of surrounding grains (Campbell et al., 2001). For that reason, the breakage patterns for each grain depends mainly on the interaction between the grain, the roller mill design and operation conditions (Campbell and Webb, 2000). Roller mills have at least two pairs of rollers, often accompanied by two or three additional pairs, that usually crush the grains as it passes between rollers (Laurinen et al., 2000). Reducing geometric standard deviation along with particle size has been achieved by using mills employing three high-rollers mills (Healy et al., 1994a). Alternatively, further reduction of geometric standard deviation has been reported when corn grains are firstly ground by roller mill and then milled using a hammer mill (Healy et al., 1994a; Wondra et al., 1995a). Adjustment of operating conditions within a mill can affect particle size distribution. In a milling trial with

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corn, reducing the hammer mill screen size from 9.6 mm to 1.2 mm resulted in a linear reduction in geometric standard deviation (Wondra et al., 1995b; Wondra et al., 1995c). When hammer mill screen size was plotted against geometric standard deviation, each 1 mm reduction in milling screen size resulted in a 0.09 reduction in geometric standard deviation (R2=0.82, n=7). Use of multiple stages of grinding using a roller mill has been reported to reduce geometric standard deviation. However a series of test grindings is required using a double pair roller mill to establish the optimal distance between the rollers to obtain the required particle size.(Svihus et al., 2004).Multi-stage milling processes have been shown to reduce the total energy required for milling (Dziki, 2011). In a study of wheat milling, initially grinding of the grains through a roller mill before hammer milling reduced the total energy required from 32.6-79.0 kJ/kg (hammer milling only) to 23.1-44.4 kJ/kg (roller milling and hammer milling). The change in particle size distribution in multi-stage milling processes is also of importance (Dziki, 2008). Soft wheat grains crushed before milling had a lower particle size and a higher proportion of material that was smaller than 200 µm than samples milled without crushing. Depending on the moisture content, some hard wheat samples also had a lower particle size and higher proportion of material smaller than 200 µm.

1.1 Draw back in the existing machine

As we know there are a lot of by-products of Pasta and Macaroni in the production room so to change this waste product into re use purpose, the company has two old grinding machine .these machines have a lot of unnecessary and bulky processes that leads to a high overall cost and also the first old machine uses 7 motors and its electrical consumption also high as its use these number of motors.Its working principle is including the new machine with some difference but the great difference is after blower flow or sends the flour. After that in old machine process the flour through pipe line get into coursepasta scrip storage silo with the help of air lock on the above of the silo. Then to the second grinding machine that is eliminated in modified machine. there is a step flow from roller then after milled in roller to sieve, there are two pipe lines after sieve the one which takes the milled enough flour to fine flour pasta silo or the one which back the wrong one (not milled enough flour) to roller. Then from fine flour pasta silo after that to cyclone.Then sends through pipe to mixer with the help of blower. The process flow of existing machine is given in Figure 1.

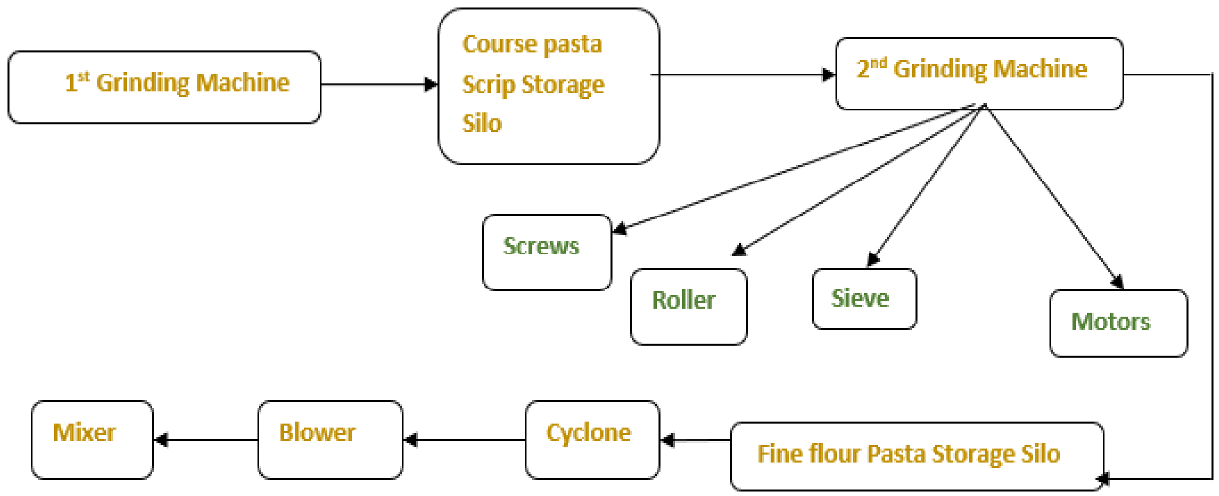
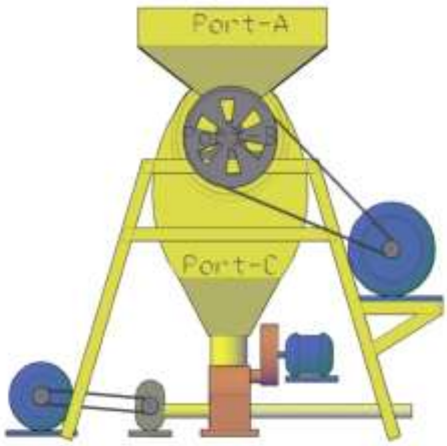


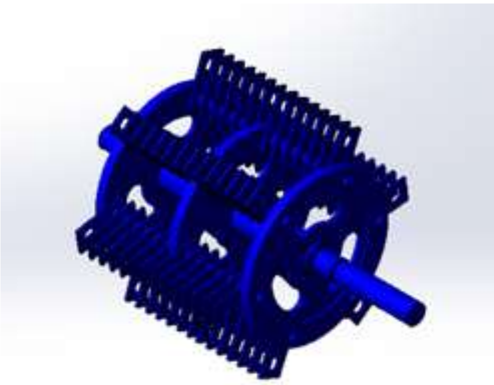
Figure 1Process of Existing Grinding Machine

1.2 Modified new machine

The proposed machine can feed 25kg by product of pasta and macaroni into port A. At port A there is a magnetizing rod, which is used to trap metallic material. Next process is the by-product goes to port B, at port B there is a flexible jaw welded on hallow shaft that rotates with high power by 5.5kw motor. And, there is sieve below shaft with a value of 0.025 mm diameter. The by-product goes to port B and as there is a high power of motor, then by product is easily changed into floury or the final product flour.The modified new machine and crusher are diagrammatically represented in Figure 2.



(a)



(b)

Figure 2 (a) Modified new pasta grinding machine (b) New design of Crusher

The Design and analysis of shaft and rotor assembly for hammer mill crusher of capacity 0.1 (100kg/hr) tones per hour transmitting 20 B.H.P and a speed of 750 rpm. The design is based on the standard design procedure. The diameter of rotor shaft of hammer mill crusher has been designed. The design should be safe when the values obtained from the present design procedure were compared with the values and results obtained from the analysis using solid-work package. When the shaft is rotated at rated optimum speed (rpm) and the loads applied to the shaft it should not bend during rotation. When the shaft is rotated under free conditions, deflections will be created due to the critical speed of the shaft. To compare this deflection shaft was designed such that the natural frequency and speed is under limits. In this project the shaft and rotor assembly of hammer mill crusher was modelled using Pro-e modelling. Meshing of the shaft model was done and the loads, stresses that were applied for the shaft to be checked out that the design should be safe one.

2.Critical Parts Design

2.1 Design Criteria

* Crusher Capacity = 100 Kg/h
* Motor Power = 3.7 KW = 5 hp
* Rotational speed N = 1440 rpm
* Frequency = 50 Hz
* Diameter of motor pulley D1 = 3 in = 75 mm

2.2 Power Transmission

A belt is a looped strip of flexible material used to mechanically link two or more rotating shafts. A belt drive offers smooth transmission of power between shafts at a considerable distance. The power transmission drives used for the machine are belt and pulley. Design for pulley or sheave: The rotor's pulley diameter was selected using the equation for speed ratio shown in Eq. (1):

Shaft Speed determination:

$$\frac{D_1}{D_2} = \frac{N_2}{N_1} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(1)}$$

Where

N_1 = revolution of the smaller pulley, rpm

N_2 = revolution of the larger pulley rpm

D_1 = diameter of smaller pulley, mm

D_2 = diameter of larger pulley, mm

The dimensions standard for V-belts taken for this design is listed in Table 1

Table 1. Dimensions of standard V-belts (Khurmi and Gupta, 2004)

Type of Belt	Power Ranges in Kw	Minimum Pitch Diameter of pulley (D) mm	Top Width (b) mm	Thickness (t) mm	Weight Per Meter Length in Newton
A	0.7-3.5	75	13	8	1.06
B	2-15	125	17	11	1.89
C	7.5 -75	200	22	14	3.43
D	20 -150	355	32	19	5.96
E	30 - 350	500	38	23	-

The selected pullies diameters and speeds are given in Table 2.

Table 2. Selected Pulley Diameter and Speed

D1 (mm)	D2 (mm)	N1 (rpm)	N2 (rpm)	Due To 4% Creep & Slip
75	200	1440	540	518.4

2.3 Selection of Belt Length

Assume the centre distance between the larger pulley and the smaller pulley =711 mm, then pitch length of the belt is given by standard table IS 2494-1974 Type A, x= 747 mm. Open belt drive selected for this design is given in figure 3.

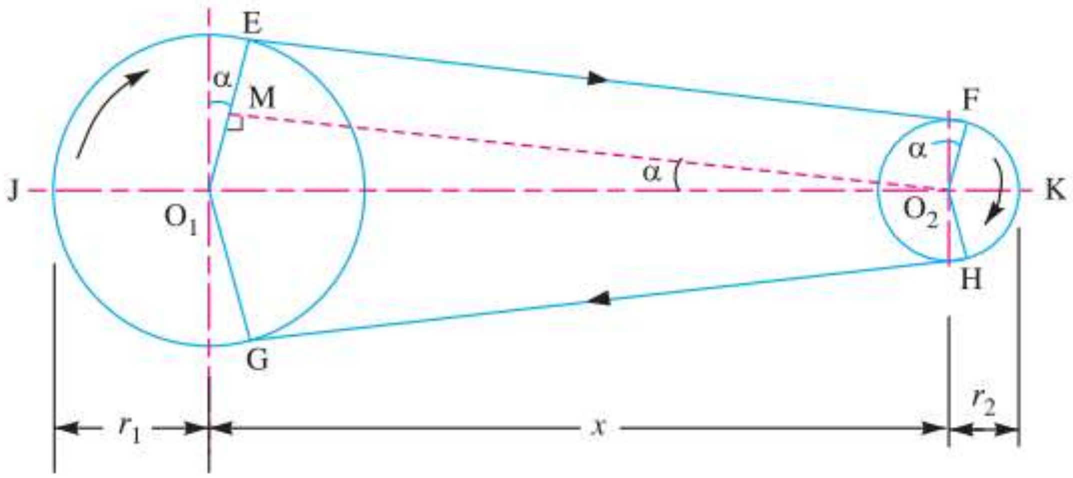


Figure 3 Open Belt Drive

$$L = \pi(r_1 + r_2) + 2x + \frac{(r_1 + r_2)^2}{x} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots.2$$

Where:

r_1 and r_2 = radii of the larger and smaller pulleys,
x = Distance between the center of two pulleys ($O1$ $O2$), and
L = Total length of the belt.

The standard pitch length of V belts according to IS: 2494-1974 is listed in table 3.

Table 3. Standard pitch lengths of V-belts according to IS:2494-1974

D1 (mm)	D2 (mm)	X (mm)	π	L (mm)
37.5	200	747	3.1415926	1867.901

The dimensions standard of V-belts is given in Table-4

Table-4 Dimensions of standard V-belts (Khurmi and Gupta, 2004)

Type of Belt	Standard Pitch Lengths of V-belts in mm
A	645,696,747,823,848,925, 950,1001,1026,1051,1102 1128,1204,1255,1331,1433,1458,1509,1560,1636,1661, 1687,1763,1814,1941,2017,2068,2093,2195,2322,2474, 2703,28880,3084,3287,3693,
B	932,1008,1059,1110,1212,1262,1339,1415,1440,1466, 1565,1694,1770,1821,1948,2024,2101,2202,2329,2507, 2583,2710,2888,3091,3294,3701,4056,4158,4437,4615, 7625,8387,9149,
C	1275,1351,1453,1580,1681,1783,1834,1961,2088,2113, 2215,2342,2494,2723,2901,3104,3205,3307,3459, 3713,4069,4171,4450,4628,5009,5390,6101,6863, 7625,8387,9149,
D	3127,3330,3736,4092,4194,4473,4651,5032,54136124,6886, 7648,8410,9172,9934,10 696,12 220,13 744,15 268,16 792,
E	5426,6137,6899,7661,8423,9185,9947,10 709,12 233,13 757,

2.4 Belt Rap & Contact Angle

The formulae used for calculation rapping angle is give in equation no 3.2 and for contact angles are in 3 and 3.1.

$$\alpha_1 = \pi - \sin^{-1}\left(\frac{r_1 - r_2}{x}\right) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots . . Small\ pulley \ldots\ldots\ldots\ldots\ldots\ldots 3$$

$$\alpha_2 = \pi + \sin^{-1}\left(\frac{r_1 + r_2}{x}\right) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots . Large\ pulley \ldots\ldots\ldots\ldots .3.1$$

$$\theta = (180° - 2\alpha)\frac{\pi}{180} rad = 40°\ ASME \ldots . . rapping\ angle \ldots\ldots\ldots .3.2$$

The rapping angle and contact angles which are calculated by aforesaid formulae is listed in table 5.

Table 5.Belt Rap & Contact Angle

r1 (mm)	r2 (mm)	x (mm)	π	α1	α2	θ
37.5	100	747	3.14159	3.225358931	3.32672	0.8391
				50.7446°	78.8°	40°

The force diagram of flat belt is represented in figure 5.

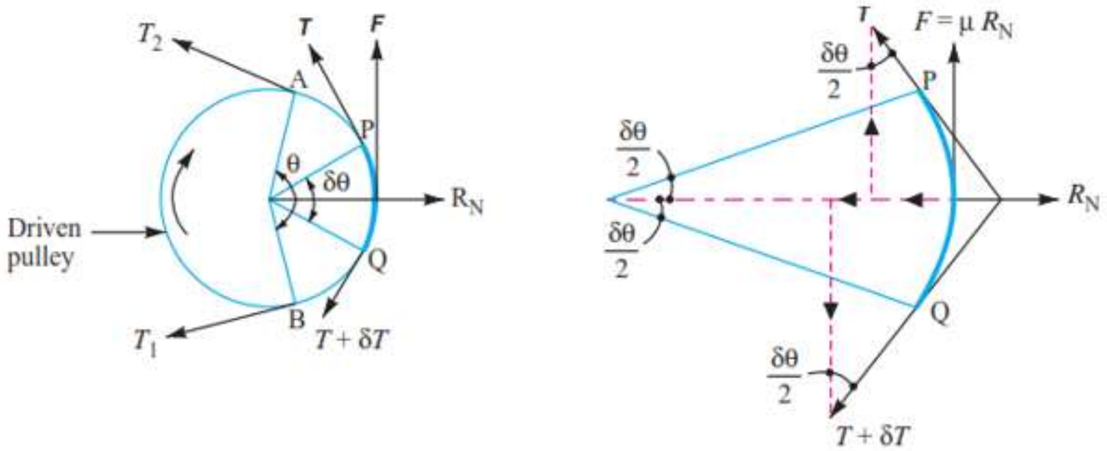


Figure 5 Ratio of Driving Tension for Flat Belt

Where;

T1 = Tension in the belt on the tight side,
T2 = Tension in the belt on the slack side, and
ϴ = Angle of contact in in radians (i.e. angle subtended by the arc *AB*, along of which the belt touches the pulley, at the center).
μ =coefficient of friction = 0.3

The value of coefficient of friction 0.3 selected from the table 6 is given below.

Table 6. Coefficient of friction between belt & pulley standard (Khurmi and Gupta, 2004)]

Belt material	*Pulley material*							
	Cast iron, steel			*Wood*	*Compressed paper*	*Leather face*	*Rubber face*
	Dry	*Wet*	*Greasy*				
1. Leather oak tanned	0.25	0.2	0.15	0.3	0.33	0.38	0.40
2. Leather chrome tanned	0.35	0.32	0.22	0.4	0.45	0.48	0.50
3. Convass-stitched	0.20	0.15	0.12	0.23	0.25	0.27	0.30
4. Cotton woven	0.22	0.15	0.12	0.25	0.28	0.27	0.30
5. Rubber	0.30	0.18	—	0.32	0.35	0.40	0.42
6. Balata	0.32	0.20	—	0.35	0.38	0.40	0.42

Belt tension

The tension of belt calculated by using equation no 4 and 4.1

$$\log_e\left(\frac{T_1}{T_2}\right) = \mu.\theta \quad \text{or} \quad \frac{T_1}{T_2} = e^{\mu.\theta} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots4$$

$$2.3 \log\left(\frac{T_1}{T_2}\right) = \mu.\theta \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots4.1$$

2.5 Design of Shaft

For a shaft subjected to twisting moment only, the diameter of the shaft was obtained by using the torsion equation given in equation no 5

$$T = \frac{\pi}{16} x \tau x d^3 \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots5$$

Where,

T= Twisting moment (Nm)
Torsional shear stress (N/m2)=42 MPa (Khurmi and Gupta, 2004).
D= Diameter of shaft (m)

Khurmi and Gupta (2004) developed equation for determination of Twisting (T) for a belt drive as

$$T = (T_1\text{-}T_2)\ R \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots6$$

Where;

T1 = Tight side tension (N)
T2 = Slack side tension (N)
R = Rdius of pulley (m)

Determination of T1 and T2:

Tension was gotten as:

T1 = Tm-Tc……………………………..7

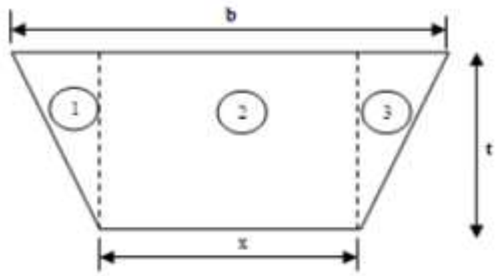


Figure 6 Cross-section of V-belt

where,
T_m = Maximum tension in belt (N)
T_c = Centrifugal tension (applicable for belt running at high speed).
T_m = Maximum stress x cross-sectional area of belt

$T_m = \sigma a$ ……………………………………..8

Determination of belts cross-sectional area,

a: The cross-sectional area of the belt was calculated by considering Fig. 6.

From Table 1, top width, b = 13 thickness, t = 8

$$a = \frac{t}{2}\left(\frac{b-x}{2}\right) + xt + \frac{t}{2}\left(\frac{b-x}{2}\right)$$ ……………….9

a = $(\frac{b-x}{2})t + xt$ … … … … … … … … … … … … … … … 9.1

x =7.2 mm => a = 80.8mm^2

Maximum allowable stress of belt, = 2.8 MPa (Also, centrifugal tension, T_C was determined using Eq.

$T_c = mV^2$ …………………………………10

Density of the material is selected from the table 7 given below.

Table 7. Density of belt materials (Khurmi and Gupta, 2004)]

Material of Belt	**Mass density in kg/m3**
Leather	1000
Canvass	1220
Rubber	1140
Balata	1110
Single woven belt	1170
Double woven belt	1250

Where m = mass of belt per unit length. It was calculated using:

m= pa
P = density of belt material (Rubber) (m3/s) ………………………..11
ρ = 1140 Kg/m^3
Also, V = linear speed of belt given as:
V = π DN / 60

…………………………………..12

For a V- belt drive, the tension ration is given by the

$$\frac{T_1 - T_C}{T_2 - T_C} = e^{\mu\theta \operatorname{cosec} \alpha/2}$$ ………………………………………13

where, μ= coefficient of friction b/n belt & pulley

θ = Angle of rap = 40° = 88.5679 rad
α = Groove angle

2.6 Power Transmitted by pulley

$\boldsymbol{P = (T_1 - T_2)v}$ ….14

The calculated values of Pulley and shaft are given in table 8

Table 8. Pulley and Shaft calculated parameters

L(m)	A(m^2)	P belt(kg/m^3)	R1()	N2	-- (N/m^2)	e^μ*Θ	π
1809	0.00008	1140	0.03	518.4	2.80E+06	2.7921	3.1415

M(m)	V(rpm)	Tc(N)	Tm(N)	T1(N)	T2(N)	T(N-m)	P(Watt)	n
0.0912	0.8143	0.060	291200	291200	14828	8291.15	3750.716255	0.99

2.7 Design of Shafts:

The shafts may be designed based on

1) Strength and
2) rigidity and stiffness

In designing shafts based on strength, the following cases may be considered:

1) Shafts subjected to twisting moment or torque only.
2) Shafts subjected to bending moment only.
3) Shafts subjected to combined twisting only
4) Shafts subjected to axial loads in addition to combined torsion & bending loads

Shafts subjected to twisting moment or torque only:

When the shaft is subjected to twisting moment (torque) only, then the diameter of the shaft may be obtained by using the torsion equation. We know that

$$\frac{T}{J} = \frac{f}{r}$$

Where,

T= Twisting moment acting on the shaft,

J = Polar moment of inertia of the shaft about the axis of rotation

Fs= Distance from neutral axis to the outer most fiber = d/2 where

We know for round solid shaft, polar moment of

$$J = \frac{\pi}{32} d^4$$

The equation may be written as

$T/\pi d4 = 2fs/32d$

$T = fs\pi d3/16$

Twisting moment (T) may be obtain by the following relation:

In S.I units, power transmitted (in watts) by the shaft,

$P=2\pi NT/60$ or

$T=(px60)/2\pi N$

Where,

T=Twisting moment in N-m

N=Speed of the shaft in RPM

Properties of selected ferrous alloys in an annealed state at room temperature is given in table 9

Table *9* Properties of Selected Ferrous Alloys in an annealed State at Room- Temperature

T (N-m)	N2 (rpm)	Fs(N/m^2)	P (Watt)	π
8291	518.4	84,000,000	3749	3.1415

Mechanical Properties of Materials referred for this design is listed in table 10

Table-10 Mechanical Properties of Materials

Material	Density (g/cm')	Modulus of Elasticity [(psi x10' (Gpa)]	Yield strength [ksi (MPa)]	Tensile Strength [ksi (MPa)]	Ductility (%EL in 2 in)	Poisson's Ratio	Electrical conductivity {(Ω-m)-1 x10-]	Thermal Conductivity (W/m-k)	Coefficient of Thermal Expansion [c) -1 x 10-5]	Melting Temperature or Range (C)
Iron	7.87	30(201)	19(130)	38(260)	45	0.29	10	80	11.8	1538
Gray Cast iron	7.15	Variable	-	18(125)	-	Variable	-1	46	10.8	"
Nodular cast iron	7.12	24(165)	40(275)	60(415)	18	0.28	-1.5	33	11.8	"
Malleable cast iron	7.20-7.45	25(172)	32(220)	50(345)	10	0.26	0.25-0.35	51	11.9	"
Low-carbon steel(1020)	7.86	30(207)	43(295)	57(395)	37	0.30	5.9	52	11.7	1495-1520
Medium-carbon steel(1040)	7.85	30(207)	51(350)	75(520)	30	0.30	5.8	52	11.3	1495-1505
High-carbon steel(1080) Stainless steels	7.84	30(207)	55(380)	89(615)	25	0.30	5.6	48	11.0	1385-1475
Ferritic, type 446	7.50	29(200)	50(345)	80(552)	20	0.30	1.5	21	10.4	1425-1510
Austenitic type 316	800	28(193)	30(275)	80(552)	60	0.30	1.4	16	16.0	1370-1400
Martensitic type410	7.80	29(200)	40(275)	70(483)	30	0.30	1.8	25	9.9	1480-1530

2.8 Weight of the hammer (Wh)

Weight of the hammer is calculated by using below mentioned formula.

$$W_h = m_h * g$$

Material => Mild Steel

ρ = 7.85 g/

Weight of each Hammer = (7.85*10*4*0.5) = 0.175 Kg

Number of Hammers = 6*8 = 48 hammers exert crushing or grinding force in Acceleration and speed up zone per revolution.

2.9 Centrifugal Force Exerted by the Hammer

Centrifugal force developed by the hammer during working is calculated below.

$$F_c = \frac{m * v^2}{r}$$

Where:

v =tip speed of the hammer,

m = mass of hammer, Kg

r = Radius of rotor = 13.5 cm

2.10 Hammer Tip Speed v

Hammer tip speed V is calculated by using below formula

$[v = r * \omega]$ = 12 m/s

Where

r =radius of rotor

ω =angular velocity of hammer $= \frac{\pi * r * N}{1800} rad/sec \ = \ 90\, rps$

Rotor's rpm =N= 384 RPM

$F_{ch} = n * m * r * \omega^2 = 9.1854\ KN$

Where:

n= number of hammers exerting Fc per revolution

Hammer Shaft Diameter Determination:

$$\sigma_s(\text{allowable}) = \frac{M_b Y_{max}}{I}$$

$$\frac{I}{Y_{max}} = Z = \sigma_s = \frac{M_b}{Z}$$

Where

Y max = Distance from neutral axis to outer fibers

I = Moment of inertia

Z = Section modulus

For solid round bar

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

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

Bending moment on the hammer shafts is given by:

$$M_{b(max)} = \frac{wl^2}{8}$$

$$= \frac{(191)(0.3^2)}{8} = 2.15\ N - m$$

Where

l = effective length of the hammer shaft =300 mm

w = loading on the hammer shaft

Main Shaft Diameter Determination;

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

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

Where:

ds= shaft diameter, *Mb*= bending moment, Nm,= torsional moment, Nm;

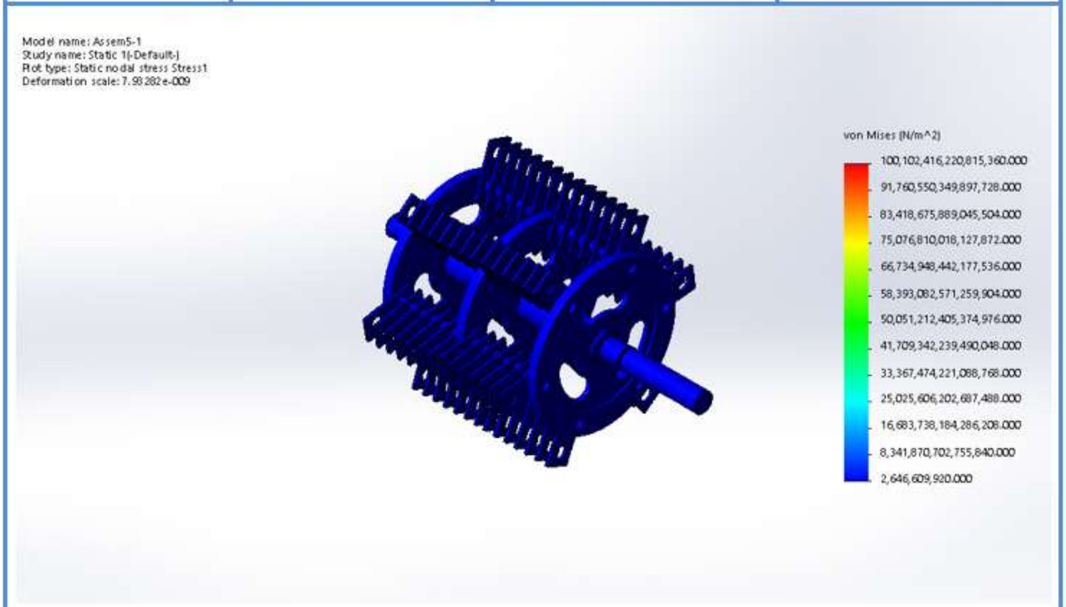
Kb= Shock and fatigue factor for bending moment and *Kt* = Shock and fatigue factor for torsional moment.

Since the load is suddenly applied with heavy shock, therefore,

Let *Kb* and Kt values be 2.0

The study of stress, displacement and strain were done by solidwork software is shown in figure 7, 8 and 9 respectively. In figure 7 the nodes which have induced minimum and maximum stresses are identified. In figure 8 the nodes whichdeform minimum and maximum displacements are illustrated. In figure 9 the nodes which induced minimum and maximum strain are exposed.

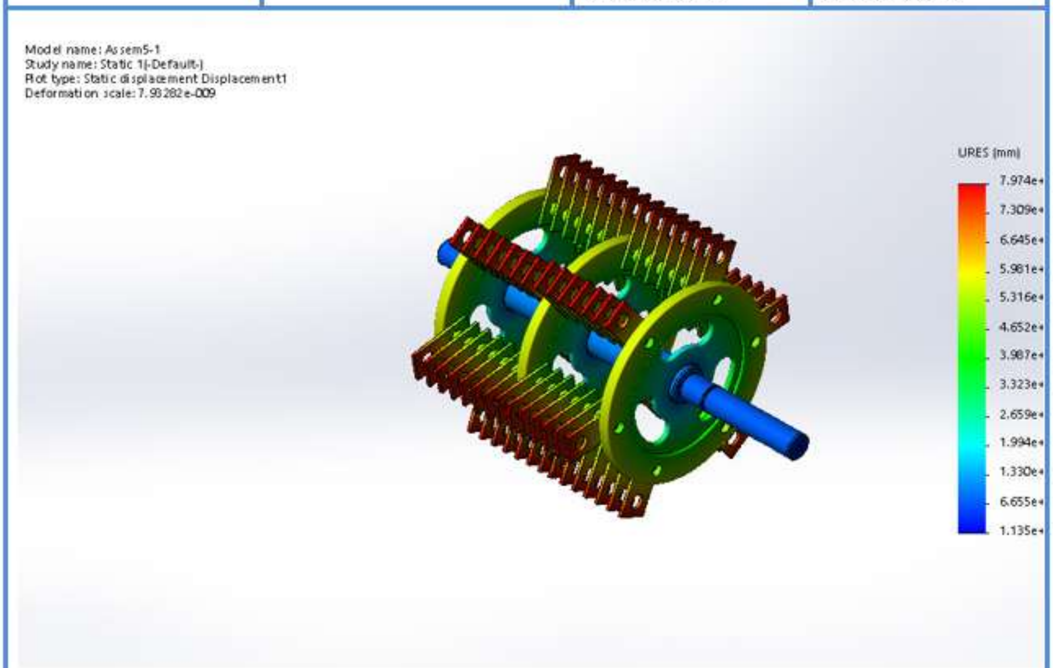
Name	Type	Min	Max
Stress1	VON: von Mises Stress	2.64661e+009 N/m^2 Node: 2949079	1.00102e+017 N/m^2 Node: 723584



Assem5-1-Static 1-Stress-Stress1

Figure: 7 Stress Analysis on Crusher

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	1.13493e+006 mm Node: 2143097	7.97368e+009 mm Node: 1770778



Assem5-1-Static 1-Displacement-Displacement1

Figure 8 Resultant Displacemet of Crusher

Name	Type	Min	Max
Strain1	ESTRN: Equivalent Strain	0.0116728 Element: 233326	390595 Element: 447818

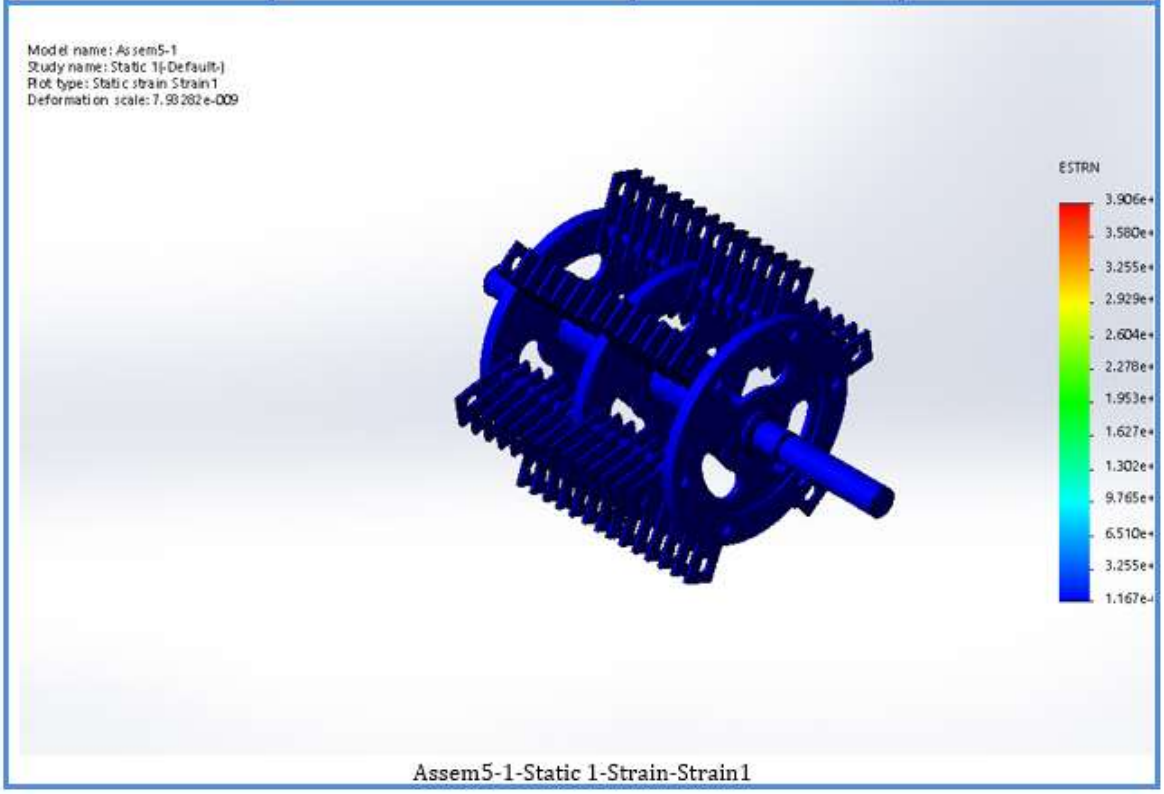


Figure 9 Strain Analysis on Crusher

3. Tests on New Designed Hammer Mill

The following parameters are calculated and used for new design is listed in table 11.

Table 11. Main parameters of new design

Parameter	Symbol	Value	Unit
Shaft speed	N_2	518.4	rpm
Length of belt	L	1814	mm
Angle of rap for small pulley	α_1	51	degree
Angle of rap for small pulley	α_2	79	degree
Tension in tight side of belt	T_1	291200	N
Tension in slack side of belt	T_2	14828	N
Torque transmitted to the shaft	T	8291	Nm
Power transmitted to the shaft	p	3749	W
Weight of Hammer	W_n	1.75	N
Centrifugal force by hammer	F_{ch}	9185.4	N
Diameter of the Hammer shaft	d_n	30	Mm
Weight of the Hammer shaft	W_{ns}	60	N

Table 12 Tested Performance of new design and existing machine

Sl No	Parameter	Existing Machine	New Design
1	Efficiency	82.3%,.	90%
2	Fineness modulus (fm)	2.32	0.31
3	Uniformity index (U)	4:1:5 (coarse: medium: fine)	0: 1: 9 (coarse: medium: fine)
4	Effective size (D	0.085 mm	0.075 mm

The modified pasta milling machine has a milling efficiency of 90 %,. The new design of the hammer mill produced was found to have a fineness modulus (fm) of 0.31, Uniformity index (U) of 0: 1: 9 (coarse: medium: fine) and effective size (D) of 0.075 mm which is better than that produced by an existing mill (hammer mill) of fineness modulus (fm) 2.32, uniformity index (U) of 4:1:5 and effective size (D10) of 0.085 mm

4. Conclusion:

The design and construction of a modified hammer milling machine was done, owing to the inability of existing mills to meet the demand of cassava flour in bakery industries. Rational design by drawing and calculations and fabrication in the Centre for Industrial Studies (CIS) FUTO were used to bring this mill to reality. The modified pasta milling machine has a milling efficiency of 90 %,.It is dust free and self-cleaning and due to proper air circulation does not destroy the cassava flour produced by overheating.

Future Scope : Higher capacity grinding machine can be designed and analyzed on the similar guidelines.

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