Design of a Manual and Motorized Meat Grinding Machine

Bako Sunday*[1], Mohammed B. Ndaliman#[2]

[1]*Department of Mechanical Engineering, Nuhu Bamalli Polytechnic Zaria, Nigeria*

[2]*Department of Mechanical Engineering, Federal University of Technology Minna, Nigeria*

Abstract- Meat is one of the most essential nutritious food item needed for human consumption from which high quality proteins, minerals and essential vitamins are derived. However, certain categories of consumers such as the children, elderly and sick people might not be able to provide the requisite biting force for tough meat tissues. Thus they would only be able to consume meat in their grinded form. In order to address this problem, it becomes necessary to have a meat grinding machine; in order to ensure good and easy digestion of the meat in their system.

This paper presents the design of an efficient single meat grinding machine with both manual and motorized mode of operation, which can be used at any where (urban and rural area) and at any time (during electric power outage). This design provides the kinematic arrangement of forces, materials selection and proportion of parts to ensure maximum strength and functionality of the machine. To avoid failure of the machine, the working stress (21MN/m^2) of the machine is kept within the value of its ultimate stress (30MN/m^2).

Keywords- Shaft and Handle Design

I. INTRODUCTION

This paper is the first of its kind ever published. In this design, a single machine with both manual and motorized mode of operation was undertaken. The meat grinding machine is a machine that is used to force meat by means of rotating shaft under pressure through a horizontal mounted cylinder (shaft housing). At the end of the shaft housing there is a cutting system consisting of a cross-shaped knives rotating with the shaft and a stationary perforated disc (hole plate).

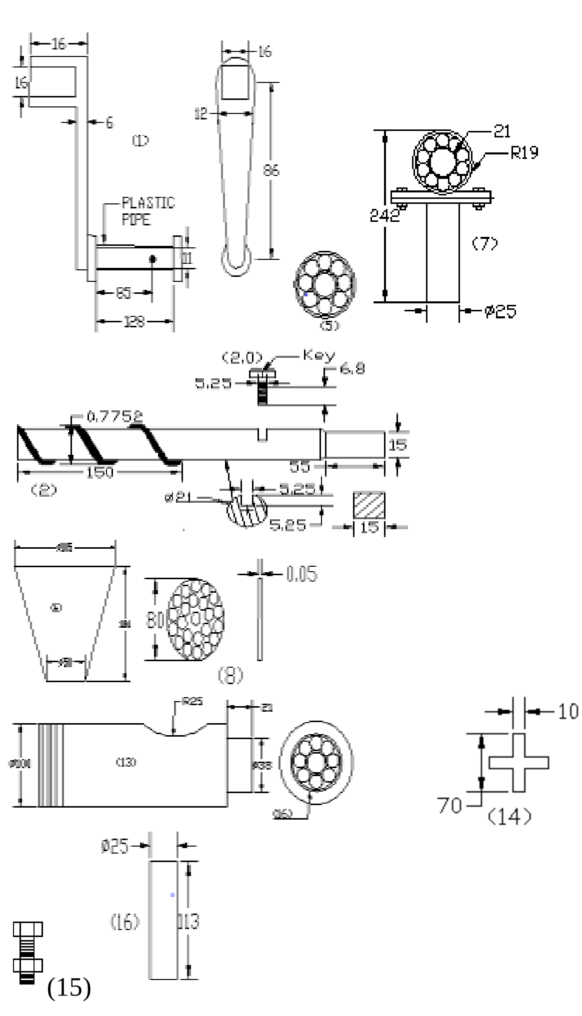
The perforation of the hole plate normally range from 1 to 13mm. the meat is compressed by the rotating shaft, pushed through the cutting system and extrudes through the hole in the hole plate after being cut by the revolving knives. The degree of grinding is determine by the size of the holes in the hole plate.

II. DESIGN ANALYSIS AND CALCULATIONS

The material selection and proportioning of parts of the meat grinding machine are controlled by their strength, rigidity, corrosion resistance and the fabrication method. The table I below shows components and their respective materials.

TABLE I. The Machine Components

The Machine Components



All Dimensions in mm, Scale 1:1

A. To Select the V-belt and Calculate the Torque,

The machine is designed to use two horse power (2hp) electric motor.

1hp (horse power) = 746W,
2hp = 1402W = 1.402KW

Since the power of the electric motor is 1.402KW; a "A" type of V-belt with a power range of 0,7-3,5 KW, a top width (b) of 13mm and thickness (c) of 8mm (as shown in figure 2) is selected according to Indian Standard (IS: 2494-1974).

Torque transmitted by electric motor, Tt is given by,

$$Tt = \frac{P}{\omega},$$

P = power transmitted by electric motor
ω = Angular speed of the electric motor

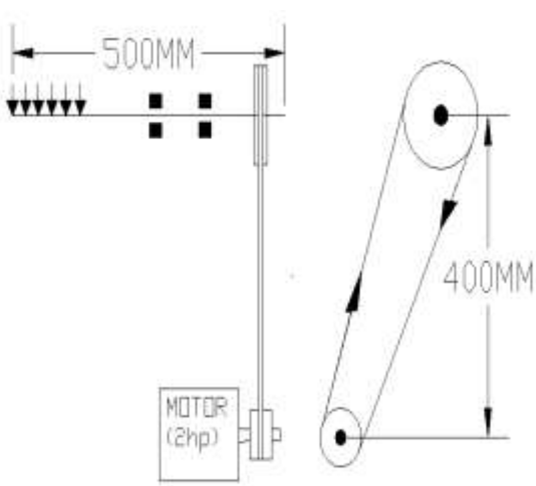
$$\omega = \frac{2\pi N}{60}$$

N = Speed of rotation of electric motor
= 1420rev/min

$$\omega = \frac{2 \times 3.142 \times 1420}{60} = 146.6 rad/sec$$

$$Tt = \frac{P}{\omega} = \frac{1492}{146.6} = 10.02 NM$$

B. To Calculate the Tightening Belt Tension T_1 and Slackening Belt Tension T_2,



(a
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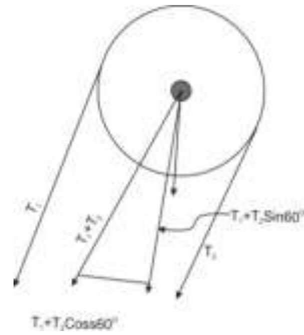


(b)\

Fig. 1: Power Transmission (a) and Analysis of the Pulley (b),

The above diagram shows the loads acting on the shaft and pulley.

Mg = The weight of the pulley,

$(T_1 - T_2)Sin60^O$ = Vertical load on the shaft,

$(T_1 + T_2)Cos60^O$ = Horizontal load on the shaft,

Torque supplied, Tt = $(T_1 - T_2)r$,

r = radius of the smaller pulley = 0.02275m

10.2 = $(T_1 - T_2)0.0275$

$T_1 - T_2$ = 370.9N (1)

Peripheral Velocity, V = ωr
= 146.6×0.0275 = 4.03m/s

Mass/unit length of belt m = *P*gA

P = Density Of belt = $980 kg/m^3$,
g = Acceleration due to gravity = $9.8 m/s^2$,
A = Cross-sectional area of belt m^2

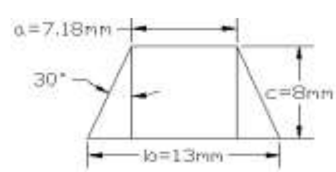


Fig. 2: Cross Section of V-belt

$$A = \frac{1}{2}(a+b)c$$

$$= \frac{1}{2}(7.18+13)8 = 80.72 \times 10^{-6} m^2$$

$$m = \rho g A = 980 \times 9.81 \times 80.72 \times 10^{-6} = 0.78 kg/m$$

Centrifugal Tension, $Tc = MV^2$

Tc = 0.78 × 4.03^2=12.7N

But $\frac{T_1 - T_c}{T_2 - T_c} = e^{\frac{r\alpha_1}{\sin\theta/_2}}$

μ = Coefficient of friction between belt and pulley = 0.2,
θ=Groove angle for V-belt = 30^O,
ρ=Density of belt materials = 980kg/m^3,
α_1 = Angle of wrap for smaller pulley (rad) = ?

$$\alpha_1 = 180^0 - 2Sin^{-1}\frac{R-r}{c}$$

$r = 0.027m$,

C = *Centre distance between pulley* = 400*mm* = 0.4*m*

Radius of the larger pulley, R =?

$$But\ D = \frac{N_1 d}{N_2}.$$

The machine is designed so that the speed of the shaft and the lager pulley is 50% of the natural speed of the system. *Therefore speed of the larger pulley* N_2

$$N_2 = \frac{50}{100} \times 1400 = 700 rev/\min$$

$$D = \frac{1400 \times 55}{700} = 110mm = 0.11m$$

$$R = \frac{D}{2} = 55mm = 0.55m$$

$$\alpha_1 = 180^0 - 2\sin^{-1}\left(\frac{0.055 - 0.0275}{0.4}\right)$$

$$= 171.1^0 = 3.0 rad$$

$$\frac{T_1 - 12.7}{T_2 - 12.7} = e^{\frac{0.2\times3}{\sin 30/_2}} = e^{2.6} = 13.5$$

T$_1$ = T$_2$13.5 – 158.8 (2)

From equation (1) *above*

T$_1$=370.9+T$_2$ (3)

Substituting equation (.3) into (2) we have;

370.9 + T$_2$ = T$_2$13.5 – 158.8

-12.5T_2 = -529.7

T$_2$ = 42.4N

From equation (3) we have;

T$_1$ = 370.9 – 42.4

T$_1$ = 328.5N

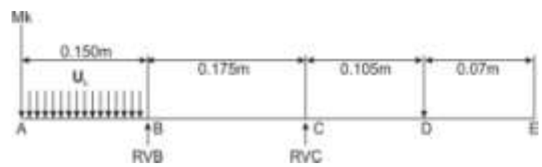


Fig. 3: Vertical Loading on the Shaft

Mk = Weight of the knife = 1.1N,

U_L = Uniform distributed load = 175.6N/M

RVB=Reaction of the bearing at B,

RVC= Reaction of the bearing at C,

mg + (T$_1$ + T$_2$)Sin60^O = Vertical Loading at the pulley

$$RvB + RHC = 1.1 + (175.6 \times 0.150) + 349.1$$

$$RvB + RvC = 376.5N \quad (4)$$

$\sum MB = 0$ (sum of moment at B)

$$-1.1 \times 0.510 - 175.6 \times \frac{0.150^2}{2} + 0.175 \times RHC - 0.28 \times 349.1 = 0$$

$$-0.165 - 1.98 + RHC0.175 - 97.7 = 0$$

$$RvC = 628.0N \quad (5)$$

Putting (5) into (4)

$$RvB = 376.5 - 628 = -251.5N$$

Moment at A $M_A = 0$

Moment at B $M_B = (-1.1 \times 0.150) - (0.150^2 \times 175.6) = -2.1N$

Moment at C $M_C = -349.1 \times 0.105 = -37.0N\text{-}M$

Moment at D $M_D = 0$,

Moment at E $M_E = 0$

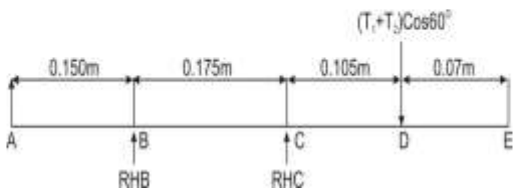


Fig. 4: Horizontal Loading on the Shaft

$(T_1 + T_2)Cos60^O$ = Horizontal loading at the pulley

RHB and RHC = are reaction of the bearing B and C for the horizontal loading

RHB+RHC = 218N (6)

$$\sum M_B = 0 \; RHC \times 0.175 - 218 \times 0.28 = 0$$

$$RHC \times 0.175 = 61.0$$

$$RHC = 348.8N \quad (7)$$

Putting (7) into (6)

RHB + 328.8 = 218N

RHB = 218 – 348.8 = -130.8N

Moment at A $M_A = 0$

Moment at B $M_B = -218 \times 0.28 - 348.8 \times 0.175 = 0$

Moment at C $M_C = 130 \times 0.175 = -23.0N\text{-}M$

Moment at D $M_D = 0$,

Moment at E $M_E = 0$

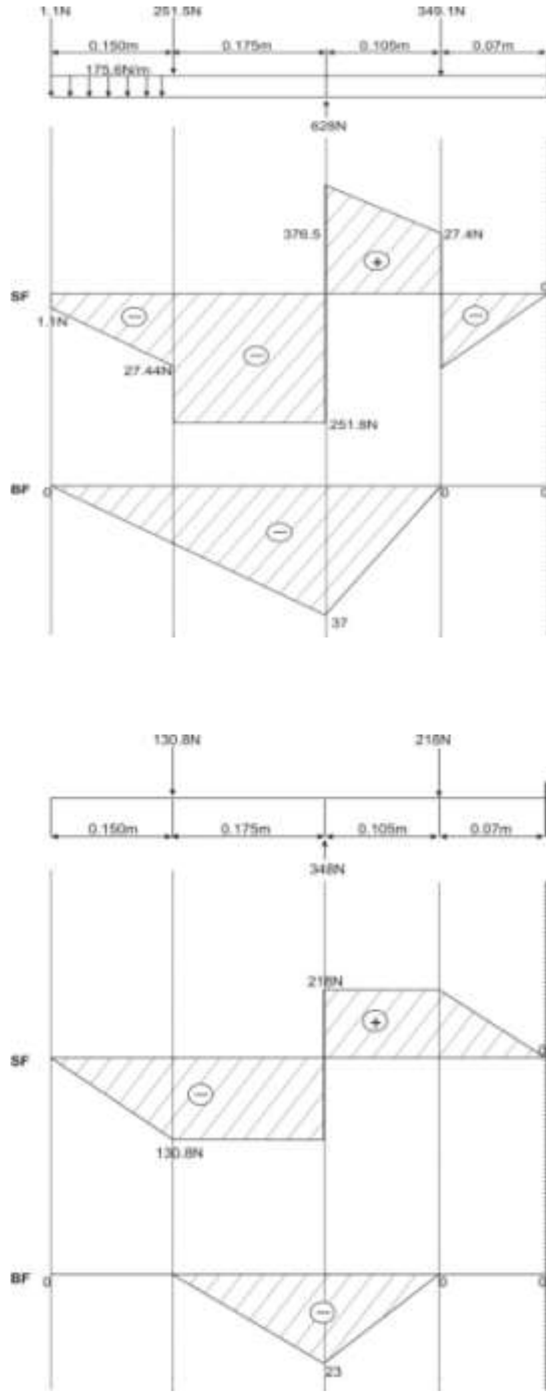


Fig.5: Bending Moment and Shear Force Diagram for Vertical (a) and Horizontal (b) Loading

C. To Calculate the Resultant Maximum Bending Moment, Mmax,

$$M\max = \sqrt{M_H{}^2 + M_V{}^2}$$

M_H = Maximum bending moment for horizontal loading = -23NM

M_v=Maximum bending moment for vertical loading = -37NM

$$M\max = \sqrt{(-23)^2 + (-37)^2} = 43.6NM$$

D. To Calculate the Shaft Diameter d_0,

In actual practice shafts are subjected to fluctuating torque and bending moment. For shafts subjected to combine bending and torsion, the shaft diameter, d_o is given by;

$$d_O^3 = \frac{16}{\pi\tau_s}\sqrt{(K_t M\max)^2 + (K_t.T_t)^2}$$

The shaft is design to have the maximum allowable shear stress of

$$\tau_s = 30 \times 10^{-6} N/M$$

K_b and K_t are Combine shock and fatigue factors for bending and torsion. The recommended value for K_t and K_b are 1.5 and 1.0 for steady loading (Khurmi 2005),

π = 3.142

$$d_O^3 = \frac{16 \times 10^{-6}}{3.142 \times 30}\sqrt{(1.5 \times 43.6)^2 + (1 \times 10.2)^2}$$

d_0 = 22.4mm

The diameter of the shaft for the meat grinder is 22mm.

E. Bearing Selection,

The resultant radial force (F) and the dynamic loading (C) are calculated.

$$F = \sqrt{F_H^2 + F_V^2}$$

$$FH = RHC = 348.8N$$

$$FV = RVC = 628.0N$$

$$F = \sqrt{348.8^2 + 628.0^2} = 718.4N$$

Bearing life $Ln = 30{,}000 hours$

Allowable shaft speed n,

$$n = \frac{50}{100} \times 1400 = 700 rev/\min$$

Constant for ball bearing, $P = 3$

$$C = F\left(\frac{Ln\,60\,n}{10^6}\right)^{\frac{1}{P}}$$

$$= 718.4\left(\frac{30{,}000 \times 60 \times 700}{10^6}\right)^{1/3} = 7.759KN$$

The bearing number 301 having the dynamic loading of 7.65KN is selected

F. To Calculate the Correct Centre Distance C between the Two Pulleys,

The shorter the belt the more often it will be subjected to bending stress while turning around the pulley at a given speed. (Sharma and Aggawal 2006).

The correct centre distance, C is given by:

$$C = L/4 - \frac{\pi(D+d)}{8} + \sqrt{\left(\frac{L}{4} - \frac{\pi CD + d)}{8}\right)^2 - \frac{(D-d)^2}{8}}$$

$$= \frac{1.053}{4} - 3.142\frac{(0.110-0.055)}{8} + \sqrt{\left(\frac{1.053}{4} - 3.142\frac{(0.11-0.055)}{8}\right)^2 - \frac{(0.11-0.05)^2}{8}}$$

$$C = 0.4824m = 482.4mm$$

G. To Calculate the Factor of Safety, FS,

$$FS = \frac{Ultimate\ Stress}{Working\ Stress}$$

Ultimate Stress = Maximum Allowable Stress in Shaft = 30 000000N/m^2

= 30MN/m^2

$$Working\ Stress = \frac{16T}{\pi d^3}$$

T = Twisting Moment (Maximum Bending Moment) = 43.6Nm

D = Shaft Diameter = 22mm = 0,022m

$$Working\ Stress = \frac{16 \times 43.6}{3.142} = \frac{20851257.2N}{m^2}$$

$$= 21MN/m^2$$

$$FS = \frac{30000000}{20851257.2} = 1.4$$

According to Sharma and Aggarwal (2006), the Factor of Safety (FS) is between 1.25 to 1.5 for exceptionally reliable material used under controlled condition and subjected to loads and stresses that can be determine certainly. Therefore since the Factor of Safety (FS) of this machine is within the range (1.25 to 1.5); the design is save.

III. KEY DESIGN

In this design, a round key inform of a bolt is selected. With the shaft diameter as a guide, the size of the key, which is the height (h) and the width (b) is determined by using the empirical design code relation for different types of keys.

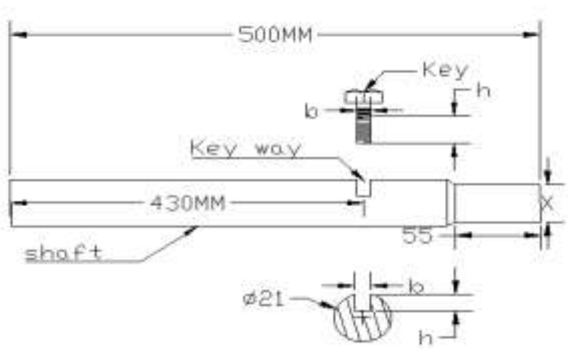


Fig. 6: Key Analysis and Position on the Shaft

A. To Calculate the Width/Diameter of the Key b,

b = nd

n = Ratio of the key width to the shaft diameter. The recommended value of n is 0.25

d = shaft diameter = 22mm

b = 0.25 × 22 = 5.5mm

b = 6mm

B. To Calculate the Depth/Height of the Key h,

h = mb

m = Ratio of height of the key to the key width. The recommended value of m is 1.30

b = 5.5mm

h = 1.3 × 5.5 = 7,15mm

h = 7mm

Therefore, a bolt of 6mm in diameter (width) and the more than 7mm height is selected.

IV. HANDLE DESIGN

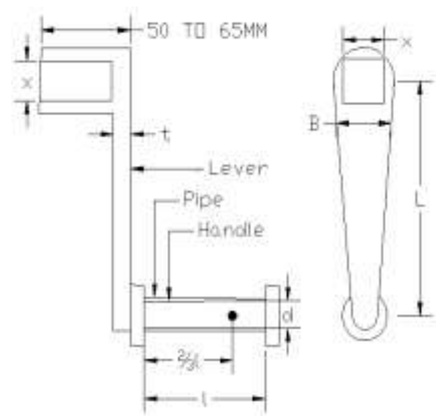


Fig. 7: Handle Analysis

The average of 170.6N effort is assigned to operate the machine during the manual operation.

The maximum force required for a single person to operate the handle is 400N with the handle length of 300mm, (Khurmi. 2005)

Therefore If 400N = 300mm,

$$170.6N = \frac{300 \times 170.6}{400} = 127.95mm$$

The handle length, l = 28mm = 0.128m

A. The Maximum Bending Moment of the Handle M_L,

$$M_L = P \times \frac{2l}{3}$$

$$M_L = 170.6 \times \frac{2 \times 0.128}{3} = 14.6N - M$$

B. The Selection Modulus of the Handle Z,

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

Where, d = diameter of the handle = 11mm = 0.011m and π = 3.142

The diameter of the handle is proportioned as 25mm for single person with the effort of 400N.

Therefore; $400N = 25mm$

$$170.6N = \frac{25 \times 170.6}{32} = 11mm = 0.011m$$

Therefore; $Z = \frac{3.142 \times 11^3}{32} = 130.7mm^3$

C. The Constant Twisting Moment T,

$$T = \frac{2}{3} \times p \times l$$

$$T = \frac{2}{3} \times 170.6 \times 0.128 = 14.6N\text{-}M$$

D. The Maximum Bending Moment M_L,

$$M_L = p \times L$$

$$L = \frac{M_L}{P} = \frac{14.6}{170.6} = 0{,}086m = 86mm$$

The length of the lever, L is 86mm

E. The Width near the Boss B,

$$B = 2t,$$

t = thickness of the lever

The section modulus Z for the level arm is also given by;

$$Z = \frac{1}{6} \times t \times B^2$$

Therefore $130.7 = \frac{1}{6} \times 4 \times t^3$

$$t^3 = 196.05$$

$$t = 6mm.$$

$$B = 2t = 2 \times 6 = 12mm$$

F. The Dimension or Length of the Square End, of the Handle x,

$x = \sqrt{\frac{D^2}{2}}$, D = Shaft diameter = 22mm

$$x = \sqrt{\frac{22^2}{2}} = 15.5mm$$

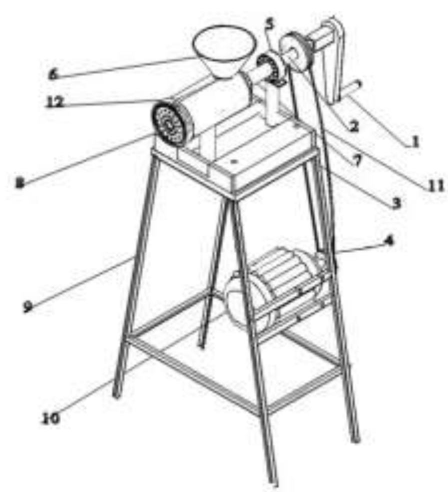
x = 16mm

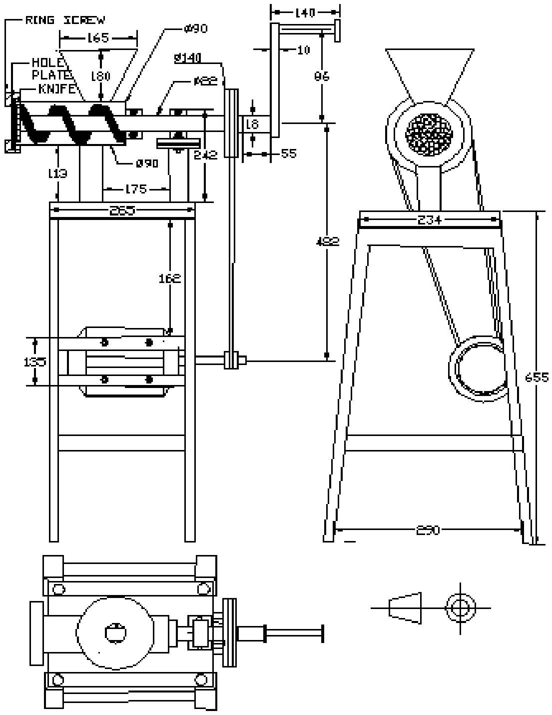
V. MACHINE ASSEMBLY

Table II, shows the position of the various machine components and their respective materials.

TABLE II. The Machine Assembly

The Machine Assembly Drawing





All dimensions in mm, Scale 1:1

S/N	Component	Material	Qty
1	Handle	MS, P	1
2	Shaft & Key	SS,MS	1
3	Grinder Base	MS	1
4	Motor Pulley	MS	1
5	Bearing	SS	2
6	Hopper	SS	1
7	Bearing Housing	MS	2
8	Hole Plate	A	3
9	Stand	Ms	1
10	Electric Motor	MS, R	!
11	V-belt	R, C, R	1
12	Ring Screw	A	1
13	Shaft Housing	SS	1
14	Knife	SS	1
15	Bolt & Nut M12x1,25 M16x1,5	 MS MS	 6 4
16	Grinder Leg	MS	1

A-Aluminum, C-Cord, F-Fabric, MS-Mild Steel
SS-Stainless Steel, R-Rubber

VI. CONCLUSIONS

To avoid failure of the machine, the working stress is kept within the ultimate stress of the machine. From the Shear Force (SF), Bending Moment (BM) diagram and the value of the Factor of Safety (FS), we conclude that the design is save. We hereby recommend this machine for construction and call on food Engineers and Technologist to develop different types of meat products that can be produced from grinded meat. We also call on entrepreneurs to develop new business model that will utilize the use of this machine for entrepreneurship in order to create jobs and to the reduce the rate of unemployment in the societies,

Finally, if this machine is utilized for the production of grinded meat products: Hospitals, Nurseries and Primaries schools would be a good market for selling of these products.

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