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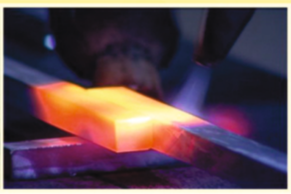


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Development Of A Multipurpose Mix-sieving Machine

Olaiya N.G; Oke P.K. and Mogaji P.B.

Department of Industrial and Production Engineering

Federal University of Technology Akure

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A B S T R A C T

This study focuses on mechanized process of mixing and sieving variety of flour (such as yam, plantain, cassava, etc.) with ease and good finished quality, especially when large quantity is required. This is achieved by a combination of a channel fitted with a conveyor blade which is set in a spiral manner on a shaft to produce thorough mixing, and a net in a wooden frame driven by pulley and connecting rod to perform the sieving action. The designed machine is to be driven by a compound belt drive from an electric motor of 1h.p at 1400rpm and its throughput capacity is 13.4 kg/hour. The availability of both mixing and sieving compartment in the design has distinguished the machine from the regular sieving or mixing machine. The machine is affordable and simple to maintain and therefore, it is recommended for small holders, local processors and home use.

1.0 INTRODUCTION

Sieving and mixing as primary operations in food processing are time consuming and labour intensive especially when the product is required in large quantity. A simple definition of sieving is the separation of fine material from coarse ones by means of meshed or perforated vessel (Fayose, 2008). Also, mixing is to combine ingredients by putting them together or blending them to make a single new substance (Microsoft Corporation, 2009). Sieving of yam or plantain flour before processing is a must do procedure if one is to enjoy the meal after preparation.

Consider a situation where a producer decides to obtain high nutrient of Elubo by mixing and then sieving yam and plantain flour before preparation, and if this is to be carried out manually, mixing will not be thorough, it will be energy and time consuming, tedious and strenuous. Also, dust generated by the fine foodstuffs during mixing and sieving process is both unhealthy and a discouragement to producers.

Correspondence:

E-mail: ngolaiya@futa.edu.ng; pkoke@futa.edu.ng; pbmogaji@futa.edu.ng

The multipurpose mix-sieving machine is conceived out of the desire to reduce to the barest minimum the rigours involved in mixing and sieving crushed food stuff. It takes into cognizance the quality of the end product as compare with the one prepared manually.

The machine can mix or sieve similar fine products apart from yam and plantain flour for instance, Cassava flour, Soybean flour, etc. with ease and good finished quality. This makes the taste of the design unique and appreciable.

In food industries, mixing of flour has been a major operation in their production process. Even in many homes, mixing of flour for baked foods has become necessary; hence the need for an affordable flour mixing machine is on the increase (Okafor, 2015). Despite so many dough mixers in the market, many small and medium scale productions in developing economies still use the traditional method of hand mixing of dough for economic reasons (Okafor, 2015).

Okafor (2015), in his work, designed the Power Driven Dough Mixing Machine by improving on the design of the available number of machines in the market. He tested for the performance of the machine and the result shows that proper dough mixing was achieved in a comparatively shorter time and the cost is quite affordable with 86 per cent process efficiency.

There are a lot of wet sieves around e.g. Gari and cassava mash sieves (Nweke, Ikpi, & Ezurnah, 1986). In Nigeria, and many African countries, small and medium scale processing of food slurry such as those for corn, sorghum, soya beans and millet have been done manually over the years with little or no technological development (Simolowo & Adeniji, 2011). These manual processes are associated with obvious constraints such as excessive length of time, tedious and boredom coupled with inherent unhygienic conditions surrounding these processes (Simolowo & Adeniji, 2011).

Research work has been focused on the development of a suitable mechanical system for wet sieving of agricultural products. This is developed as a means of reducing human effort and time involved in sieving operation. But the majority of these local sieves are batch operated and do not incorporate a mixing compartment (Fayose, 2008).

One of such designs is the suction sieving machine for corn slurry which was designed in stages (Agbonson, 1999; Adeleke & Gbolalade, 2000; Simolowo & Nduka, 2002). It made use of vacuum pressure to bring about a pressure drop that enhanced the sieving of the corn slurry. However, deficiencies of this suction sieving machine included: inadequate pressure drop (suction pressure); interference of filtrate flow stream with the suction line at the outlet; low sieving rate (Simolowo & Adeniji, 2011).

Simolowo & Adeniji (2011) in their work develop an entirely different sieving machine based on vibration principles and thereby formed the primary objective of their work. Advantages of the vibration sieving machine over the suction sieving equipment include: higher rate of filtration due to adequate agitation in the design; simplified assembling and dismantling of the operational units for easy serviceability; maintenance of high hygienic standards by the use of non-corrosive materials at points of liquid-metal contact. In addition, the food-slurry sieving machine designed herein has a multipurpose and extensive application for processing various slurry products (Simolowo & Adeniji, 2011).

Fayose (2008) in his research was able to develop a Multi-Purpose Wet Food Sieving Machine. This is a motorized starch extracting machine which is based on shaking mechanism. He tested the performance of the machine and the volumetric flow rate, the capacity of the machine obtained is 0.0206 m^3/h and 22.45 kg/h respectively. The test considered concentration at three levels 12.2 %, 14.44 % and 22.77 %. The study showed that the machine performance coefficients and sieving capacity increased with decreasing concentration. Also, highest performance coefficients of 98% was obtained for sieving of maize while sieving capacity of 16.90g/s was obtained when the machine was used to sieve cassava.

2.0 MATERIALS AND METHOD

2.1 Design Considerations

A variety of factors were put into consideration before the design of the machine was done. The factors considered include the following (Ayodeji & Abioye, 2011):

i. ease of assembly and disassembly of the machine parts
ii. the height of the machine
iii. the suitable material properties
iv. reduction in drudgery associated with the traditional/primitive method of sieving large quantity of flour
v. to make the mixing and sieving compartment an integral part of the machine
vi. to synchronise the operation of the mixer and the sieve

2.2 Design Concept

Figures 1 and 2 show the assembly drawing and the orthographic views of the machine, respectively. Table 2.1 shows the component parts of the machine.

[insert Figure 1,Figure 2 and Table1]

2.3 Operation Principle of the Machine

Yam and plantain flour measured in a desired amount is charged into the mixing chamber through the machine hopper made of 2mm mild steel sheet in turn. In the mixing chamber, a blade made of galvanised steel set in a spiral manner at a pitch of 80mm on a centrally placed horizontal shaft, performs the mixing action and directs the blended flour to the machine outlet. The outlet is then opened so as to allow the flour to channel via a discharge duct to the vibrating sieve. The end product is collected in a container located under the sieve. On the end side of the conveyor shaft, a pulley is fitted to the conveyor shaft to provide the drive from the second driver pulley via the v-belt, and the first driver pulley is on the electric motor of 1 horse power rotating at 1400 rpm. Set of connecting rods are attached from the last driven pulley to the sieve in such a way that the rotation of the pulley is converted to reciprocating motion of the sieve. The shaft is made up of 25mm gauge mild steel.

2.4 Design Analysis

2.4.1 Required capacity

2.4.1.1 Hopper design

As shown in Fig 2.1, it is a square base frustum. The volume of the hopper material (V_1) is calculated from equation 2.1.

$$V_1 = \frac{1}{3}[(B_1 - B_2)H_1 - (b_1 - b_2)h_1] \quad (2.1)$$

Where: B_1 and B_2 are the external and internal base area of the big pyramid (mm^2), H_1 is the big pyramid height(mm), b_1 and b_2 are the external and internal base area of the small pyramid (mm^2) and h_1 is the small pyramid height(mm).

$$but\ \frac{h_1}{(80^2+80^2)^{1/2} \div 2} = \frac{h_1 + 130}{(150^2+150^2)^{1/2} \div 2} (similar\ triangle)$$

$$\therefore h_1 = 148.6\ m$$

$$\therefore V_1 = 1.1805 \times 10^{-4^3}$$

The hopper capacity (H_C) is determined by

$$H_C = \frac{1}{3}(B_2H_1 - b_2h_1) \quad (2.3)$$

$$H_C = 1.6544 \times 10^{-3} m^3$$

Say maximum of $1654400\ mm^3$ $(i.e\ 0.97612\ kg)$ of flour can be loaded in the hopper.

The total surface area (A_1) is calculated from equation 2.4.

$$.A_1 = 2(a+b)h \quad (2.4)$$

Where: a, b and h are the upper length (mm), lower length(mm) and height (mm) of the trapezium respectively.

$$\therefore A_1 = 0.0598\ m^2$$

Say $59800\ mm^2$ $(i.e\ 245\ mm \times 245\ mm)$ mild steel sheet should be used for the fabrication.

2.4.1.2 Trough Design

As presented in Fig 2.1, it is a cylinder (alsosee appendix). The volume of the trough material (V_2) is calculated from equation 2.5.

$$V_2 = \pi(R^2H - r^2h) \quad (2.5)$$

Where r and h is the internal radius and internal height; R(mm) and H (mm) is the external radius and external height of the cylinder respectively.

$$\therefore V_2 = 5.1335 \times 10^{-4}\ m^3$$

The trough capacity (T_C) is determined by

$$T_C = \pi r^2 h \quad (2.6)$$

$$T_C = 4.7881 \times 10^{-3}\ m^3$$

The total surface area (A_2) is calculated from equation 2.7

$$A_2 = 2\pi R(R+H) \quad (2.7)$$

$$A_2 = 0.1767\ m^2$$

2.4.1.3 Duct Design

To determine the total surface area (A_3) of duct, equation (2.8) is used

$$A_3 = ab + b \quad (2.8)$$

Where: a = width of the bigger rectangle (mm), b = length of the bigger/smaller rectangles (mm), and c = width of the smaller rectangle (mm).

$$\therefore A_3 = 0.0644\ m^2$$

2.4.1.4 Collecting Bowl Design

As shown in Fig 3.6, it is a rectangular base frustum. To determine the bowl capacity (V_3), equation (2.9) is used

$$V_3 = \frac{1}{3}(B_3H_3 - b_3h_3) \quad (2.9)$$

Where: B_3 is the internal area of the big pyramid (mm^2), H_3 is the big pyramid height (mm), b_3 is the internal area of the small pyramid (mm^2) and h_3 is the small pyramid height (mm).

$$but\ \frac{h_3}{(135^2+250^2)^{1/2} \div 2} = \frac{h_3 + 200}{(300^2+400^2)^{1/2} \div 2}\ (Similar\ triangle)$$

$$\therefore h_3 = 263.4\ mm$$

$$\therefore V_3 = 1.5129 \times 10^{-2} m^3$$

The total surface area (A_3) is calculated from equation 2.11.

$$A_4 = h[(a+b)+(c+d)] + bd \quad (2.11)$$

Where a and b are the upper length (mm) and lower length(mm) of the first set of trapezium; c and d are the upper length (mm) and lower length (mm) of the second set of trapezium; h is their respective height (mm); b and d are also the dimension of the rectangular base.

$\therefore A_4 = 0.2508\ m^2$

2.4.1.5 Sieve Frame Design and Specifications

As shown in Fig 1, the sieve frame is square in shape, the specifications for the sieve frame is given below:

i. length or breath of frame, $L = 260\ mm$
ii. depth of frame, $d = 50\ mm$
iii. frame thickness, $t = 3\ mm$
iv. support pipe diameter, $D = 40\ mm$

Centre distance between support pipes is determined as

$$C_f = (L - 2t) - D \quad (2.12)$$

$$\therefore C_f = 214\ mm$$

2.4.2 REQUIRED STRENGTH

The critical design components are:

i. selection of belt and pulley systems,
ii. power requirement and frequency of oscillation
iii. the shafting system
iv. the support angle bars and machine stands

2.4.2.1 SELECTION OF BELT, PULLEY SYSTEM, POWER REQUIREMENT, AND FREQUENCY OF OSCILLATION IN THE SYSTEM

Pulley and belt are used to transmit power from one shaft to another, especially from electric motor to shaft (Oladugba, Obire, Peter, Wilfred, & Omoseyin, 2013; Ajayi, 2008). Because of small distance between centres of pulleys, larger transmission of power required in the system, and in order to allow good grip, v-belt was selected for this machine. The pulleys are made of cast iron because its lower cost and better wear resistance.

A compound belt (consisting of two belts) drive is used to transmit power through a number of pulleys of the machine in order to achieve optimum shaft speed of *51mm* and *57mm* diameters respectively.

2.4.2.1.1 Design of belt drive

According to Khurmi & Gupta (2005), Velocity ratio of the belts drive is given by:

$$V_r = \frac{Speed\ of\ last\ driven}{Speed\ of\ first\ driver} = \frac{product\ of\ diameters\ of\ drivers}{product\ of\ diameters\ of\ driven}$$

$$or\ \ \frac{N_4}{N_1} = \frac{d_1 \times d_3}{d_2 \times d_4} \quad (2.13)$$

Where: d_1, d_2, d_3, d_4 are the electric motor diameter, the first belt driven pulley diameter, the second belt driver and driven pulley diameters respectively.

$$\therefore N_4 = \frac{d_1 \times d_3 \times N_1}{d_2 \times d_4} = \frac{51 \times 57 \times 1400}{176 \times 176} = 131.4\ rpm$$

Length of the first belt drive is determine by,

$$L_1 = \frac{\pi}{2}(d_2 + d_1) + 2x_1 + \frac{(d_2 - d_1)^2}{4x_1} \quad (2.14)$$

$$x_1 = central\ distance$$

$$\therefore L_1 = 1206\ mm$$

Length of the second belt drive is determine by,

$$L_2 = \frac{\pi}{2}(d_4 + d_3) + 2x_2 + \frac{(d_4 - d_3)^2}{4x_2} \quad (2.15)$$

$$x_2 = central\ distance$$

$$\therefore L_2 = 842\ mm$$

According to Khurmi & Gupta (2005), torque acting on electric motor pulley:

$$T_m = \frac{60 \times rated\ power}{2\pi N_1} \quad (2.16)$$

$$T_m = 5.05\ Nm$$

According to Okafor (2015), torque acting on electric motor pulley is also given by:

$$T_m = \frac{(T_1 - T_2)d_1}{2} \quad (2.17)$$

$$5.05 = \frac{(T_1 - T_2) \times 51 \times 10^{-3}}{2}$$

From which the difference in the first belt tension, $T_1 - T_2 = 198.04\ N$ (2.18)

Where d_1, T_1, T_2 are: the electric motor diameter, the first belt tension in the tight side and slack side respectively

From the table of standard dimensions of v-belt, and coefficient of friction between belt and pulley, the following properties are selected for the belt drives (Khurmi and Gupta, 2005).

Type of belt selected $= Type\ A$

Grove angle $(2\beta) = 34°$

Coefficient of friction $(\mu) = 0.30$

Angle of contact of the belt on the electric motor pulley is given by,

$$\theta_1 = 180° - 2\sin^{-1}\left(\frac{d_2 - d_1}{2x_1}\right) \quad (2.19)$$

$$= 162.88° = 2.843\ rad$$

Ratio of driving tension for the first belt is,

$$2.3\ log\left(\frac{T_1}{T_2}\right) = \mu.\theta_1\ cosec\ \beta \quad (2.20)$$

$$2.3\ log\left(\frac{T_1}{T_2}\right) = 0.3 \times 2.843 \times cosec\left(\frac{34}{2}\right)$$

From which,

$$\frac{T_1}{T_2} = 18.55 \quad (2.21)$$

Solving equations (2.18) and (2.21), we have,

$$T_1 = 209.32\ N\ \ and\ \ T_2 = 11.28\ N$$

Velocity of the first belt drive,

$$v_1 = \frac{\pi d_1 N_1}{60}\ or\ \frac{\pi d_2 N_2}{60} \quad (2.22)$$

$$= 3.74\ m/s$$

Power transmitted by the first belt drive,

$$P_1 = (T_1 - T_2)v_1 \quad (2.23)$$

$$= 740.7\ Watt$$

Speed of driven pulley for the first belt (N_2) is the same as the speed of the driver pulley for the second belt (N_3),

$$N_2 = N_3 = \frac{d_1 N_1}{d_2} \quad (3.24)$$

$$= 405.68\ rpm$$

Difference between the tension in tigth side and slack side of the second belt is given by,

$$T_3 - T_4 = \frac{60 \times P_1}{\pi d_3 N_3} \quad (2.25)$$

$$= 611.76\ N$$

Angle of contact of the belt on the driver pulley is given by,

$$\theta_2 = 180° - 2\sin^{-1}\left(\frac{d_4 - d_3}{2x_2}\right) \quad (2.26)$$

$$= 150.01° = 2.618\ rad$$

Ratio of driving tension for the second belt drive is,

$$2.3\ log\left(\frac{T_3}{T_4}\right) = \mu.\theta_2\ cosec\ \beta \quad (2.27)$$

From which,

$$\frac{T_3}{T_4} = 14.72 \quad (2.28)$$

Solving equations (2.25) and (2.28), we have,

$$T_3 = 656.35\ N\ \ and\ \ T_4 = 44.6\ N$$

Velocity of the second belt drive,

$$v_2 = \frac{\pi d_3 N_3}{60}\ or\ \frac{\pi d_4 N_4}{60} \quad (2.29)$$

Where: N_4 is the shaft optimum speed required for the mixing and sieving operation as calculated earlier.

$$\therefore v_2 = 1.21\ m/s$$

Power transmitted by the second belt drive,

$$P_2 = (T_3 - T_4)v_2 \quad (2.30)$$

$$= 740.22\ Watt$$

2.4.2.1.2 Power required in the system

Total power transmitted in the system or the design rated power is given by,

$$P_{total} = P_1 + P_2 \quad (2.31)$$

$$= 1481\ Watt = 1.481\ k\ Watt$$

Therefore, the total power transmitted in the system is twice the electric motor rated power, and that is more effective.

2.4.2.1.3 Design of the pulley

From the table of standard dimensions of v-grooved pulley (Khurmi and Gupta, 2005), face width of the pulley is determined by,

$$B = (n-1)e + 2f \qquad (2.32)$$

Where: n, is the number of grooves in the pulley; e and f are standard dimensions (mm) in the v-grooved pulley. For 'type A' belt and single grooved pulley, $n = 1$, $e = 15$, $f = 10$

$$\therefore B = 20\ mm$$

2.4.2.1.4 Frequency of oscillation in the system

The machine is designed in such a way that the number of revolution of the last driven pulley correspond to the number of oscillation of the sieve frame. The frequency of oscillation of the sieve frame is determined by,

$$f = \frac{N_4}{60} \qquad (2.33)$$

Where $N_4 =$ speed of drive shaft in revolution per minute.

$$\therefore f = \frac{131.4}{60} = 2.19\ H_Z$$

Therefore, the sieve is designed to perform two oscillations per second, and that is more effective.

2.4.2.2 SHAFT DESIGN

The shafts are cylindrical with circular cross sections (Stephen & Emmanuel, 2009), one shaft has screw conveyor with a pulley mounted on it, while the other is short in length and a pulley mounted on it, and last two shafts are used to support the sieve frame. Since the shaft with screw conveyor is more loaded, the design of the shafts is based on it. The shaft will be subjected to fluctuating torque and bending moments. Because the shaft is load gradually with flour while rotating, therefore combined shock and fatigue factors (K_b=1.5) for bending and (K_t=1.0) for torsion are taken into account .

Bending Moment can occur as a result of the applied loads on the shaft and belt tension . The pull on the drive shaft due to belt tension acts horizontally while; all other loads are acting vertically.

Total load on the drive shaft due to belt tension,

$$b_t = T_3 + T_4 \qquad (2.34)$$

Where, T_3 , T_4 are: the second belt tension in the tight side and slack side respectively.

$$\therefore b_t = 701\ N$$

Weight of pulley,

$$W_p = \frac{g\rho_c \pi {d_p}^2 B}{4} \qquad (2.35)$$

Where g, ρ_c, d_p, B are: acceleration due to gravity(m/s^2), density of cast iron(kg/m^3), pulley diameter (m) and width of pulley (m) respectively.

$$\therefore W_p = 34.7\ N$$

Estimated distributed load,

$$E_d = \frac{gT_c\rho_f}{l} \qquad (2.36)$$

Where, T_c, ρ_f, g, l is: capacity of trough (m^3), density of flour (kg/m^3),

$$(R_{VB} \times 0.2841) = (34.7 \times 0.393) + \left(9.57 \times \frac{0.2841}{2}\right) \qquad (2.38)$$

From which,

$$R_{VB} = 52.78\ N\ (upward)\ and$$
$$R_{VC} = 8.51\ N(downward)$$

Bending moment(BM) at point A and C,

$$BM_{VA} = BM_{VC} = 0 \qquad (2.39)$$

BM at point B,

$$BM_{VB} = 34.7 \times 0.1089 \qquad (2.40)$$
$$= 3.78\ Nm$$

BM at point D,

BM at point D,

$$BM_{VD} = 34.7\left(0.1089 + \frac{0.2841}{2}\right) - R_{VB}\left(\frac{0.2841}{2}\right) + \frac{9.57}{2}\left(\frac{0.2841}{2}\right)^2 \qquad (2.41)$$
$$= 1.31\ Nm$$

Therefore, the maximum bending moment is at point B. The bending moment due to acceleration due to gravity (m/s^2) and span length (m) respectively.

$$\therefore E_d = 33.7\ N/m$$

The space, vertical loading, and horizontal loading diagram of the shaft are shown in figur3(a), 3(b), and 3(c) respectively

[insert figure3,figure 4 and figure 5]

2.4.2.2.1 Determination of maximum bending moment due to vertical loading.

From the vertical loading diagram, sum of the reactions at the bearings is given by,

$$R_{VB} + R_{VC} = 44.27\ N \quad (2.37)$$

Taken moment about point C,

vertical loading diagram of the shaft is shown in figure 3(d).

2.4.2.2.2 Determination of maximum bending moment due to horizontal loading.

From the horizontal loading diagram, sum of the reactions at the bearings is given by,

$$R_{HB} + R_{HC} = 701\ N \quad (2.42)$$

Taken moment about point C,

$$R_{HB}(0.2841) = 701\ (0.1089 + 0.2841) \quad (2.43)$$

From which,

$$R_{HB} = 969.7\ N\ (tension)\ and$$
$$R_{HC} = 268.7\ N(compression)$$

Bending moment(BM) at point A and C,

$$BM_{HA} = BM_{HC} = 0 \quad (2.44)$$

BM at point B,

$$BM_{HB} = 701 \times 0.1089 \quad (2.45)$$

$$= 76.34\ Nm$$

BM at point D,

$$BM_{HD} = R_{HC}\left(\frac{0.2841}{2}\right) \quad (2.46)$$

$$= 38.17\ Nm$$

Therefore, the maximum bending moment is at point B. The bending moment due to horizontal loading diagram of the shaft is shown in figure 3(e).

2.4.2.2.3 Determination of resultant bending moment.

According to Khurmi & Gupta (2005), resultant BM at point B is given by,

$$M_B = \sqrt{(BM_{VB})^2 + (BM_{HB})^2} \quad (3.47)$$
$$= \sqrt{(3.78\ )^2 + (76.34)^2} = 76.43\ Nm$$

Resultant BM at point D is given by,

$$M_D = \sqrt{(BM_{VD})^2 + (BM_{HD})^2} \quad (2.48)$$
$$= 38.19\ Nm$$

Therefore, the highest of these values is used in calculating the shaft diameter. The resultant bending moment diagram of the shaft is shown in figure 3(f).

2..4.2.2.4 Determination of drive shaft diameter.

According to Khurmi & Gupta (2005), torque acting on drive shaft can be calculated from:

$$T = T_3\left(1 - \frac{T_4}{T_3}\right) \times \frac{d_4}{2} \quad (2.49)$$

Where d_4, T_3, T_4 are: the drive shaft pulley diameter (mm), the second belt tension (N) in the tight side and slack side respectively.

$$\therefore\ T = 656.35 \times \left(1 - \frac{44.6}{656.35}\right) \times \frac{176}{2}$$
$$= 53.83\ Nm$$

Maximum bending moment,

$$M = M_B = 76.43\ Nm$$

According to Asoiro & Udo (2013), diameter of the drive shaft can be calculated from

$$d^3 = \frac{16}{\pi s_s}[(K_b M_b)^2 + (K_t M_t)^2]^{1/2} \quad (2.50)$$

Where: d is the diameter of shaft (m), s_s is the maximum permissible shear stress of mild steel without key way $(= 55\ MN/m^2)$, K_b is the combined shock and fatigue factor applied to bending moment$(= 1.5)$, M_b is the maximum bending moment $(= 76.43\ Nm)$, K_t is the combined shock and fatigue factor applied to torsional moment $(= 1.0)$ and M_t is the maximum torsion moment $(= 53.83\ Nm)$

$$\therefore d^3 = 11730\ mm^3 \quad (2.51)$$

According to Khurmi & Gupta (2005), for hollow shaft of the same material,

$$\left(\frac{{d_o}^4 - {d_i}^4}{d_o}\right) = d^3 \quad (2.52)$$

Where d_o and d_i are: the external and internal diameters (m) of the shaft respectively.

By taking,

$$\frac{d_i}{d_o} = 0.64 \quad (2.53)$$

Combining equations (2.51), (2.52) and (2.53), we obtain,

$$d_i = 15.5\ mm\ , \qquad d_o = 24.2\ mm$$

Therefore, hollow shaft of outside diameter (25 mm) and inside diameter (16 mm) was selected for the machine.

2.4.2.3 STRENGTH OF THE MACHINE STAND ASSUMPTIONS

The following assumptions are made (Khurmi and Gupta, 2005):

i. the bars are taken as long column and in form of cantilever
ii. the loads acting are eccentric to the centroidal axis
iii. the bar material is stressed with in the elastic limit
iv. the cross-section is uniform throughout its length

The sieve frame and angle bars are welded perpendicularly to the machine stand. According to Khurmi & Gupta (2005), this form an offset connecting link, and their weight are eccentric to the centroidal axis of the machine stand.

Estimated load of sieve frame is calculated from,

$$W_s = \left[\frac{1}{2}(L^2 - l^2)d + \frac{\pi D^2 L}{4}\right]\rho_s g \quad (2.54)$$

Where, $L,\ l,\ d,\ D, \rho_s, g$ is: external length of sieve frame (m), internal length of sieve frame (m), depth of sieve frame (m), support pipe diameter (m), density of mild steel (kg/m^3), and acceleration due to gravity (m/s^2) respectively.

$$\therefore W_s = \left[\frac{1}{2}(0.26^2 - 0.254^2)0.05 + \frac{\pi \times 0.04^2 \times 0.26}{4}\right] \times 7850 \times 9.81 = 31.1\ N$$

Estimated load of pulley support angle bar is calculated from,

$$W_a = T_1 + T_2 + w \quad (2.55)$$

Where, T_1, T_2, w are: the first belt tension in the tight side and slack side, and weight of pulley respectively.

$$\therefore W_a = 209.32 + 11.28 + 34.7 = 255.3\ N$$

Estimated load on trough support angle bar is calculated from,

$$W_t = g\left[(V_1\rho_s + H_c\rho_f) + \frac{1}{2}(V_2\rho_s + T_c\rho_f)\right] + W_p \quad (2.56)$$

Where: $V_1, V_2, H_c, T_c, \rho_f, \rho_s, g, W_p$ is the volume of hopper material(m^3), volume of trough material(m^3), capacity of hopper(m^3), capacity of trough(m^3), density of flour(kg/m^3), density of mild steel(kg/m^3), acceleration due to gravity(m/s^2) and weight of pulley(N) respectively

$$\therefore W_t = 9.81 \begin{bmatrix}(1.1805 \times 10^{-4} \times 7850)\\ +(1.6544 \times 10^{-3} \times 590)\end{bmatrix} + \left(9.81 \times \frac{1}{2}\right)\begin{bmatrix}(5.1335 \times 10^{-4} \times 7850)\\ +(4.7881 \times 10^{-3} \times 590)\end{bmatrix} + 34.7$$
$$= 52.3 + 34.7 = 87\ N$$

The distance between the point of action of the eccentric load and the centroidal axis of the column is referred to as the eccentricity (Khurmi and Gupta, 2005).

Eccentricity of the sieve frame is calculated from,

$$e_s = \frac{t + B}{2} \quad (2.57)$$

Where: $t =$ width of machine stands cross-section (mm), $B =$ length of sieve frame (mm)

$$\therefore e_s = \frac{40 + 260}{2} = 150\ mm$$

Eccentricity of the pulley support angle bar is calculated from,

$$e_a = \frac{t}{2} + l_a \quad (2.58)$$

Where: l_a = length of pulley support angle bar (mm)

$$\therefore e_a = 320\ mm$$

Eccentricity of the trough support angle bar is calculated from,

$$e_t = \frac{t + l_t}{2} \quad (2.59)$$

Where: l_t = length of trough support angle bar (mm)

$$\therefore e_t = 92.5\ mm$$

The space diagram and cross-section of the machine stand is presented in figure 2.4(a) and 2.4(b) below

Cross-sectional area of column,

$$A = B^2 - b^2 \quad (2.60)$$
$$= 304\ mm^2$$

Total load acting on the column,

$$P_T = 255.3 + 87 + 31.1$$
$$= 373.4\ N \quad (2.61)$$

Direct stress in the column,

$$\sigma_o = \frac{P_T}{A} \quad (2.62)$$
$$= 1.23\ N/mm^2 = 1.23\ MPa$$

Moment of inertia of the column section about the centroidal axis,

$$I_{YY} = \frac{B^4 - b^4}{12} \quad (2.63)$$
$$= 73.37 \times 10^3\ mm^4$$

Distance between the extreme side fibre of the column and centroidal axis

$$y = \frac{B}{2} \quad (2.64)$$
$$= \frac{40}{2} = 20\ mm$$

Section modulus,

$$Z = \frac{I_{YY}}{y} \quad (2.65)$$
$$= \frac{73.37 \times 10^3}{20} = 3.67 \times 10^3\ mm^3$$

Bending moment in the column was determine by the vector sum of load multiplied by eccentricity, and that is,

$$M = (W_a \times e_a) + (W_t \times e_t) - (W_s \times e_s) \quad (2.66)$$
$$= (220.6 \times 320) + (87 \times 92.6) - (31.1 \times 150)$$
$$= 73.98 \times 10^3\ Nmm$$

Bending stress in the column,

$$\sigma_b = \frac{M}{Z} \quad (2.67)$$
$$= 20.16\ N/mm^2 = 20.16\ MPa$$

According to Khurmi & Gupta (2005), since the bending stress (σ_b) is greater than the direct stress(σ_o), therefore, the left hand side of the column will be subjected to tensile stress and the right hand side will be subjected to compessive stress.

Maximum compressive stress,

$$\sigma_c = \sigma_o + \sigma_b \quad (2.68)$$
$$= 21.39\ N/mm^2 = 21.39\ MPa$$

Maximum tensile stress,

$$\sigma_t = \sigma_b - \sigma_o \quad (2.69)$$
$$= 20.16 - 1.23$$
$$= 18.93\ N/mm^2 = 18.93\ MPa$$

The stress diagram is presented in figure 5.

According to Rankine's formulae (Khurmi and Gupta, 2005), buckling load,

$$W_{cr} = \frac{\sigma_y \times A}{1 + a\left(\frac{L}{k}\right)^2} \quad (2.70)$$

Where: σ_y = yield stress in compression ($N/mm^2 or\ MPa$), A = cross-sectional area of the column (m^2), a = Rankine's constant, k = least radius of gyration for the column cross-section(m), $and\ L$ = equivalent length of column (m).

Least radius of gyration,

$$k = \sqrt{\frac{I_{YY}}{A}} \quad (2.71)$$

$$= 241.35\ mm$$

For a column in form of cantilever, equivalent length,

$$L = 2l \quad (2.72)$$

Where: l = actual length of column. For the machine stand, $l = 600\ mm$

According to Khurmi & Gupta (2005) for mild steel material, Rankine's constant and yield stress:

$$a = \frac{1}{7500},$$

$$\sigma_y = 320\ N/mm^2$$

From equations (2.70) to (2.72), we obtain the buckling load($W_{cr} = 96.96\ kN$) for the machine stand.

The buckling stress is calculated from,

$$\sigma_{cr} = \frac{W_{cr}}{A} \quad (2.73)$$

$$= \frac{96.96 \times 10^3}{304} = 318.95\ N/mm^2$$

$$= 318.95\ MN/m^2$$

Therefore, the maximum stress (σ_c = $21.39\ N/mm^2$) developed in the column, is much less than the buckling stress (σ_{cr} = $318.95\ MN/m^2$).

3.0 MATERIALS SELECTION

The following factors were considered before proper material was selected for each machine component (Ayodeji & Abioye, 2011). The factors include:

i. the material cost
ii. availability
iii. materials properties
iv. weight of the materials

The summary of the material selection process for each machine component is presented in Table 2.

4.0 ASSEMBLING PROCEDURE

The ease of transportation, ease of dismantling and assembling has purposely been considered in the design and fabrication (Ajayi, 2008). The through, sieve, pulleys, stands, discharge duct, collecting bowl can be separated and carried with ease to the location where the machine is to be used in such a manner that the heaviest single component is less than 10kg and this is the machine stands. The assembly can be done by one or two people.

The procedure for assembly is by starting with the stand then fix the trough in which the screw conveyor is mounted, then follow by the driven pulley attached to the conveyor shaft, then connect to the first v-belt horizontally through the driven pulley to the driving pulley then follow by the second v-belt connected through the second driven pulley to the electric motor pulley then follow by the discharge duct and the sieve, and finally placing the collecting bowl to machine base connected with helical springs. The mounting of the machine can be done within 5-10 minutes. The pictures figure 1 show the assembled and disassembled view of the machine.

5.0 CONCLUSION AND RECOMMENDATION

The benefits of the mix-sieving machine are immense. The rigours involved in sieving large quantity of flour are eliminated. This reduces the risk of producing poor quality of mixed and sieved flour and ensures safety in the course of production due to human inefficiency and ergonomic. The efficiency of the machine is over 95% when the right size of sieve is used with specific grain size of flour. An overall reduction in operation cost is enhanced by its portability. At a unit cost of about N 30,000:00(without the cost of electric motor), is considered affordable to targeted users.

A part from social settings, restaurants and religious retreat where flour would be needed in a very large quantity, the machine is also recommended for private individuals who find it rigorous to sieve flour manually or by hand before preparation, especially when he or she wishes to entertain others with the meal.

However, the following aspect of the machine needs improvement as observed during the process of evaluating the performance of the machine:

I. a wiper should be provided on the sieve frame, and set to spread the flour over the net when flour is introduced to the sieve, this will reduce the time taken for the sieving operation

ii. the sieve area should be more enclosed to prevent flower from blowing away when the machine is operated in an open atmosphere

iii. the collecting bowl should be made dipper and close to base of the sieve, this will also prevent flower in the bowl from escaping

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Table 1:Machine part list

S/N	QTY.	PARTS	MATERIAL
1	1	Top Hopper	Mild Steel
2	1	Trough	Mild Steel
3	1	Conveyor Shaft	Galvanize Steel
4	1	Trough Cover	Mild Steel
5	1	Cover Handle	Mild Steel
6	1	Sieve	Galvanize Steel
7	1	Sieve Frame	Wood
8	1	Electric Motor	Mild Steel
9	2	Square Pipe With Hole	Mild Steel
10	2	Helical Springs	302 Stainless Wire
11	2	Bolts	Mild Steel
12	1	Sieve Connectors	Mild Steel
13	2	Nuts	Mild Steel
14	4	Spring Holders	Mild Steel
15	5	Connecting Rods	Mild Steel
16	2	Square Pipe	Mild Steel
17	8	Angle Bar	Mild Steel
18	1	Collecting Bowl	Mild Steel
19	1	Discharge Duct	Mild Steel
20	2	V-Belt	Rubber Black
21	1	Shaft	Mild Steel
22	4	Bearing	Mild Steel
23	3	Pulley	Cast Iron

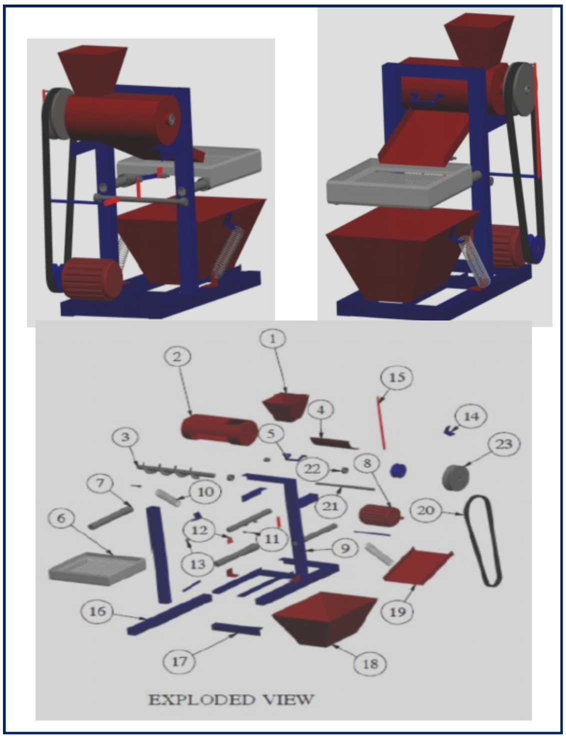

Figure 1: Assembly drawing of the machine

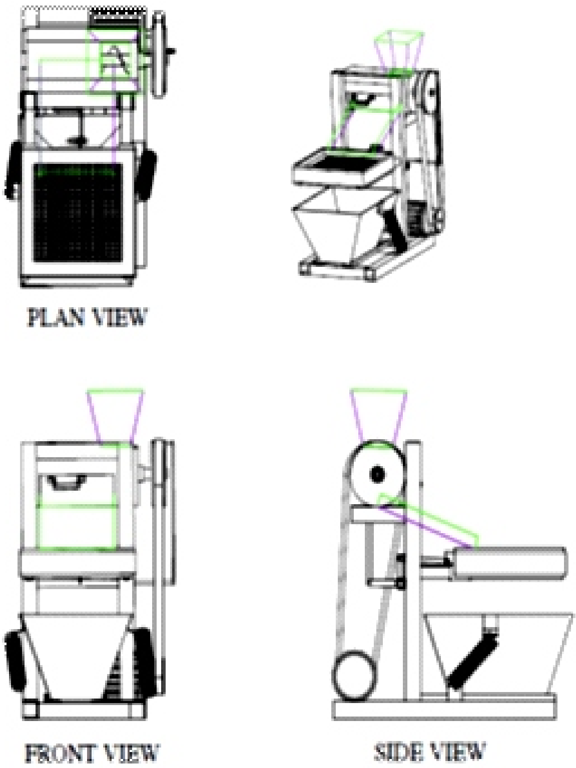

Figure 2.2: Orthographic View

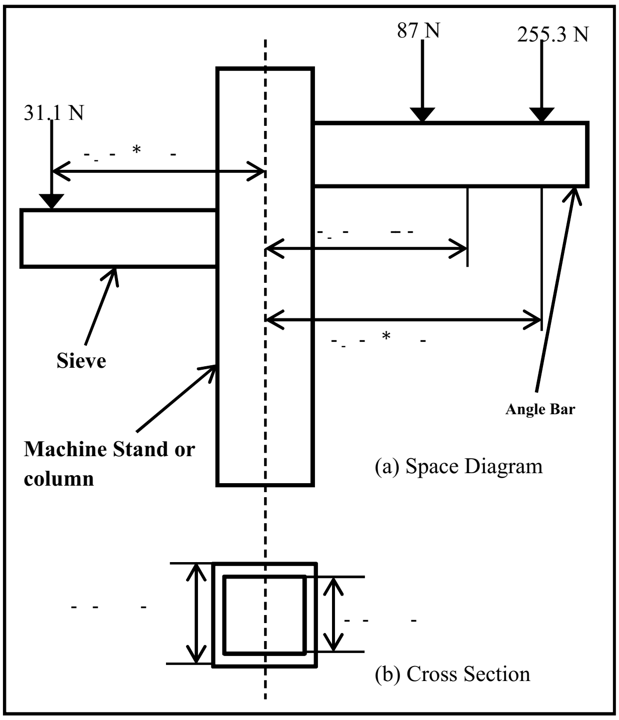

Figure 4: Space diagram and Cross-section of the machine stand

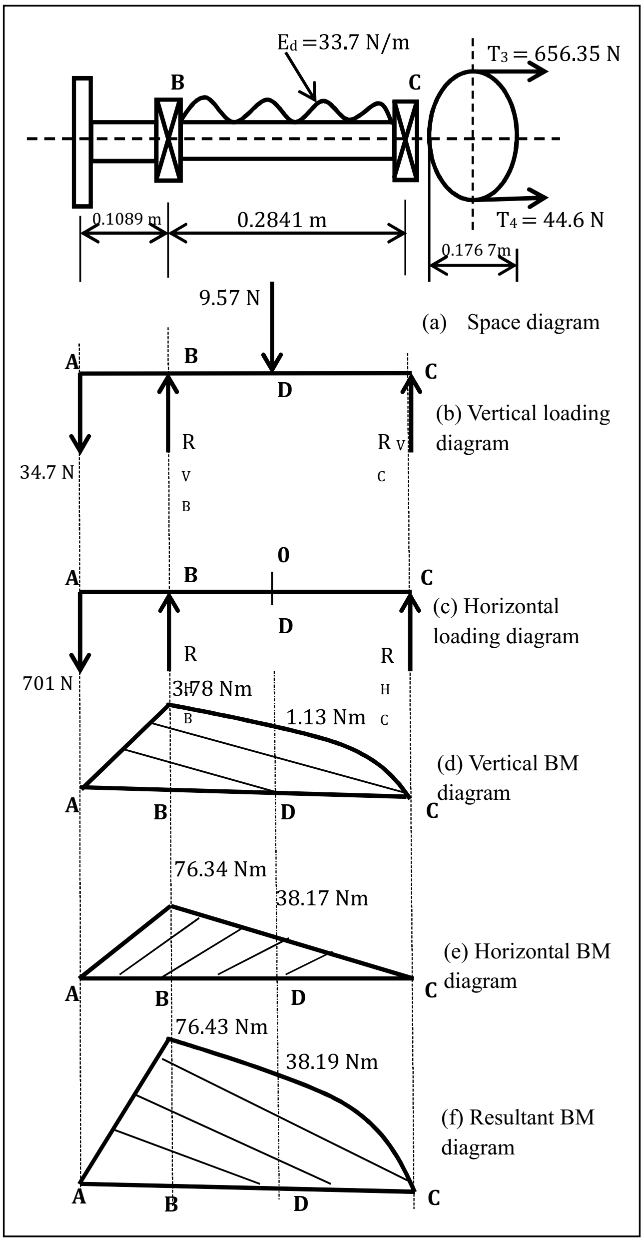

Figure 3: Space, Loading, and BM diagram for the shaft

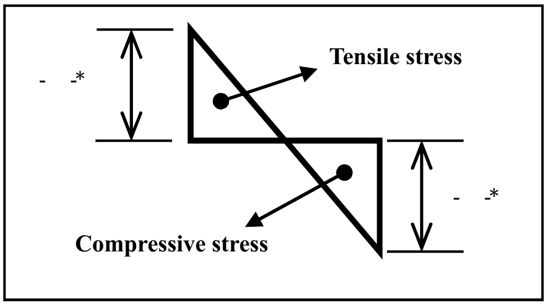

Figure 5: Stress diagram for the column