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Conceptual Design and Fabrication of an inclined feed mixer

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Abstract

In this study, an optimal fish feed mixer was designed and constructed. Several machining operations of cutting, turning, milling, etc. were employed. The component parts of the machine were designed individually and coupled by welding, soldering and screw joints. The material components were made from aluminum, mild steel, medium carbon steel, etc. These materials were sourced and selected locally in order to meet strength, accuracy and reliability requirements. The tumble feed mixer was designed for mixing ground beans, soybeans, palm oil, water and maize, combining them to form fish feed used in the agro industry on a daily basis. The capacity of the drum is 50kg maximum of fish feed materials and the electric motor selected was 3hp three phase electric motor with an efficiency of 98.3%.

Keywords: *Tumbling mass, tumble mixer, mixing chamber, speed reducer*

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1. INTRODUCTION

Fish cultivation has become worldwide business activity because it promotes food security and also a source of protein and oils to the human body development, however feeding is done manually in Nigeria (Osueke et al 2018). The importance of granular mixing to the economy cannot be overemphasized, for example, annual cost of inefficient industrial mixing in the US has been estimated to be as high as about US$ 10 billion (Moakher et al., 2000). Hence the need to design and develop an efficient mixing machine becomes imperative. In fish farming the consumption of feed in pellets form aids the entire consumption of ingredients by fishes (Odesoa et al 2016). Researches have tried to develop and improve feed mixers in recent times by changing the orientation of the mixing chamber (Adedeji M et al 2021)

Several agro-industries utilize the tumbling blenders in granular mixing operations, including medical/pharmaceutical, cosmetics, mining, energy, polymer, and semiconductor. Tumbling blenders are easy to operate, available in various capacities and are able to operate with shear sensitive or non-agglomerating materials. Their cleaning and emptying procedures are easy. Moreover, tumbling blenders are suitable for blending of dry and free flowing materials (Alexander et al., 2004; Kuo et al., 2005).

Powder mixing is a widely implemented unit operation in several particulate processing industries (e.g., food, pharmaceutical, chemical, etc.) for combining two or more raw materials in the required proportions into a final blend (mixture). Uniformity in the composition of the final blend is a key requirement and has a considerable influence on the quality of the final product. Since mixing dictates the uniformity in the composition of the blend, which is then sent for further downstream processing, the performance of this particular operation is critical to the operational efficiency of these industries (Alian et al, 2015).

Various industrial blenders with different mixing mechanisms are available and can be chosen based on the processing requirements. For example, tumbling blenders are often implemented for mixing granular materials, and a bin blender is the most commonly-used variation of the tumbling blender in the pharmaceutical industries, due to a high level of safety and convenience (Arratia et al, 2006).

The counter rotating of the vessel and the installation of internal baffles would also enhance the mixing of particles (Cullen, 2009).

To quantify the characterization of granular mixer, Saduh et al. (2002) investigated powder mixing in many kinds of tumbling blenders. On the other hand, significant research into understanding complex flows near blade impellers has been undertaken by Zhou et al. (2004).

2. METHODOLOGY

2.1 Design Considerations

Reliable trouble-free operation of mixers requires careful considerations of many factors during design. Some of the important design criteria are:

i. properties of materials to be mixed (such as abrasiveness, corrosiveness, bulk density and flow ability, ease of aeration, angle of repose, hydroscopic or damped materials; etc);
ii. mixer type
iii. mixer size and
iv. power required to mix the products.

2.2 Design of machine components

2.2.1 Design of mixing chamber

Given: mass of feed meal, m_f = 50kg and bulk density of feed meal, $\rho_f = 751.9 kg/m^3$

$$\therefore \text{volume of feed meal}, v_f = m_f \times \rho_f = 50 \times 751.9 = 0.0665\,m^3$$

Assuming that 50% of free space be provided in the mixing chamber for mixing, then we have:

$$\text{volume of mixing chamber}, \ v_c = 1.5v_f = 1.5 \times 0.0665 = 0.10m^3$$

Let ratio of height (h_c) to diameter (d_c) to be (say **2.5:1**), then we have:

$$v_c = \frac{\pi d_c^2 \times h_c}{4} = \frac{\pi d_c^2 \times 2.5d_c}{4} = \frac{2.5\pi d_c^3}{4} = 0.10$$

$$\therefore d_c^3 = \frac{0.10 \times 4}{\pi \times 2.5} = 0.05093m^3$$

$$\therefore d_c = \sqrt[3]{0.0509} = 0.370m$$

Use d_c = 0.37m = 370mm

Hence, $h_c = 2.5d_c$= 2.5 (0.37) = 0.925m= 925mm

2.2.2 Determination of total mass of mixing chamber

Mass of empty mixing chamber, m_e = density of mild steel, ρ_s x volume of empty chamber, v_e

$$\therefore m_e = \rho_s \times v_e = 7850kg/m^3 \times 3.87 \times 10^{-3} m^3 = 30.4kg$$

Hence, total mass of mixer, m_t = mass of feed meal + mass of empty chamber = 80.4kg

2.2.2 Determination of power requirement for the tumble mixer

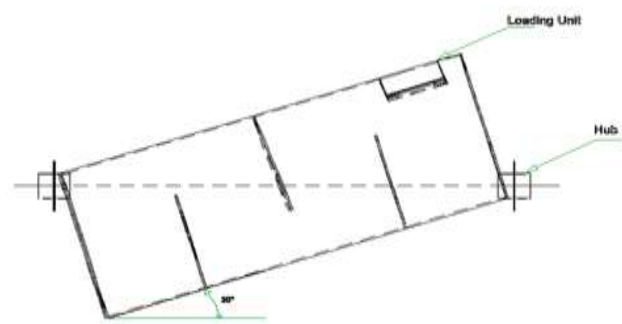

Fig. 2.1: Determination of power requirement of the mixer

Given: length of mixing chamber, L_{AB} = 996.3mm = 0.996m; mass of mixing chamber, m = 80.4kg, g = 9.81m/s^2; N = rotational speed of mixing chamber = 50rpm

$$\text{Torque transmitted}, T = W \times \frac{L}{2} = m_t \times g \times \frac{L}{2} = 80.4 \times 9.81 \times \frac{0.996}{2} = 392.8Nm$$

Power developed, P = Torque, T x angular velocity, ω

$$\therefore \text{Power}, P = T\omega = 392.8 \times \frac{2\pi \times 50}{60} = 2056.6W = 2.76hp$$

So, we take P =3hp

2.2.3 Design of speed reduction mechanism

(a) Electric motor/gearbox speed ratio

Assuming speed of electric motor to be 1440rpm and gearbox speed ratio of 15:1; then

Hence, gearbox output speed,

$$N_g = \frac{1}{15} N_m = \frac{1}{15} \times 1440 = 96rpm$$

(b) Speed reduction by gearbox output pulley/mixer shaft input pulley

$$\frac{\text{Speed of gearbox pulley}}{\text{Speed of mixer pulley}} = \frac{\text{diameter of mixer pulley}, d_2}{\text{diameter of gearbox pulley}, d_1} = \frac{96}{50}$$

$$\therefore \text{diameter of mixer pulley}, d_2 = \frac{96}{50} \text{ x } 100 = 192mm$$

2.2.4 Design of belt

Let, centre distance between mixer shaft pulley and speed reducer pulley be x = 600mm, α = angle between two pulleys, θ = angle of contact of belt, T_1 = Tension on tight side of belt, T_2 = Tension on the loose side of belt and μ = coefficient of friction between the belt and pulley.

For leather belt, μ = 0.25

$$\text{Sin } \alpha = \frac{r_2 - r_1}{x} = \frac{96-50}{600} = 0.077$$

$$\alpha = Sin^{-1}(0.077) = 4.420^0$$

$$\theta = 180 - 2\propto = 180 - 2\,(4.42) = 171^0 = 2.99\ rad$$

Assuming there is no groove, the angle of groove = 0, β = 0

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

$$\log\left(\frac{T_1}{T_2}\right) = \frac{0.25x2.99}{2.3} = 0.325$$

$$\frac{T_1}{T_2} = 10^{0.325} = 2.11$$

$$T_1 = 2.11T_2$$

Also, Torque transmitted by shaft is

$$T = (T_1 - T_2)\frac{d_2}{2} \quad 427.43 = (T_1 - T_2)\ 0.096$$

$$T_1 - T_2 = 4452.40$$

$$2.11T_2 - T_2 = 4452.40$$

$$:.\ T_2 = 4011.17N$$

$$:.T_1 = 2.11 \times 4011.17 = 8463.56N$$

Power transmitted per belt is given by:

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

$$V_b = \text{Velocity of belt} = \frac{\pi d_2 N}{60} = \frac{3.142 \text{ x } 0.192 \text{ x } 50}{60} = 0.503 \text{ m/s}$$

Thus power transmitted per belt

$$P = (T_1 - T_2)V_b = (8463.56 - 4011.17) \times 0.503 = 2239.55W$$

$$\text{No of belt} = \frac{\text{Designed power}}{\text{Power transmitted per belt}}$$

$$\therefore \text{No of belt} = \frac{2238}{2239.55} = 0.999, \text{implying that 1 belt is required.}$$

Length of belt, L_b is given by:

$$L_b = \pi\,(r_2 + r_1) + 2x + \frac{(r_2-r_1)^2}{x} = 3.142\,(96 + 50) + 2\,(600) + \frac{(96-50)^2}{600} = 1662.20mm$$

Use belt length = 1960mm

2.2.5 Design of shaft

2.2.5.1 Design shaft diameter

Two transverse forces in the vertical plane upon the shaft in the shaft in the same transverse section .the weight of the mixing chamber is assumed to be at the any point between the bearing supports while the sum of the belt tensions as well as the weight of the pulley act at pulley position.

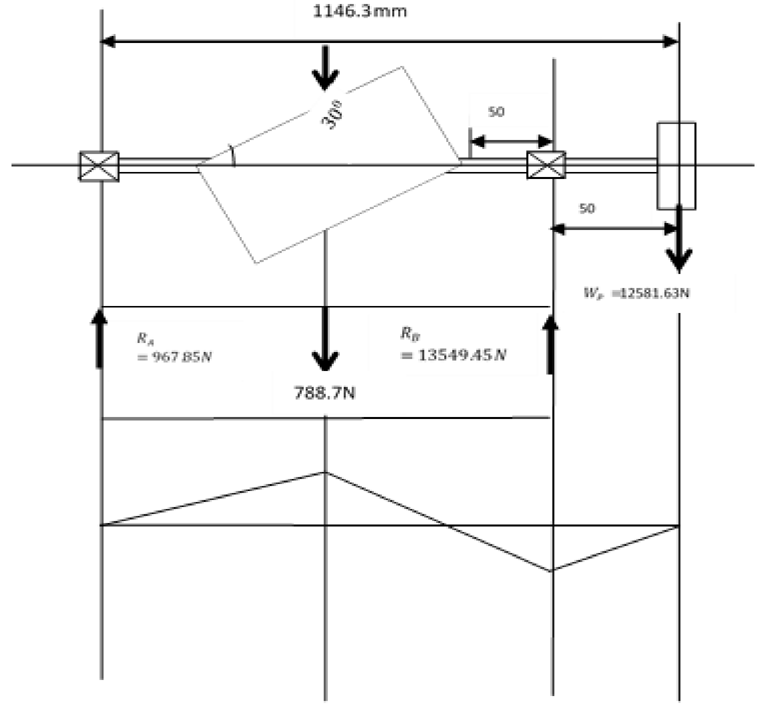

Fig. 2.2: Shaft design

Let, P_w = weight of pulley = volume of pulley x density of steel x g = 106.9N

Total vertical load on pulley, $W_p = P_w + T_1 + T_2$ = 106.9 + 8,463.56 + 4,011.17 = 12,581.63

$$R_A + R_B = 12{,}581.63N \qquad (1)$$

Taking moment about point A, we have:

$$R_B \times 1096.3 = 788.7 \times 548.15 + 12{,}581.63 \times 1146.3$$

$$\Rightarrow R_B = 13{,}549.45N$$

From eqn. (1), we have:

$$R_A = 12{,}581.63 - 13{,}549.45 = 967.85N$$

We know that bending moments (B.M) at A and C $= M_A = M_C = 0$

But, B.M. at mid-point $= M_C = R_A \times 0.593 = 329.6 \times 0.593$ = 195.45Nm

Also, B.M at D = Also B.M at D $= M_D = Wc \times 0.049 = 1373.9 \times 0.049 = 67.32Nm$

∴ Maximum bending moment, M=M_c= 195.45Nm

Torque transmitted by shaft, $T = \frac{P \times 60}{2\pi\pi} = \frac{1820 \times 60}{2\pi \times 50} = 347.59Nm$

Using the equivalent torque recommended by ASME for rotating shaft, with sudden load with minor shock, k_m=1.5, and k_t =1.5

Hence, the equivalent torque is $T_e = \sqrt{(K_m \times M)^2 + (K_t \times T)^2}$

$$T_e = \sqrt{(1.5 \times 195.45)^2 + (1.5 \times 347.59)^2} = \sqrt{85951.78 + 271842.32} = \sqrt{357,793.90}$$

$$\therefore Te = 598.16 Nm$$

Also we know that $T_e = \frac{\pi}{16} \tau_{all} d^3$

Where τ_{all} =allowable shear stress in the shaft (τ_{all}= 42Mpa)

$$\text{Hence}, 598.16 \times 10^3 = \frac{\pi}{16} \times 42 \times d^3$$

$$\Rightarrow d^3 = \frac{598.17 \times 16 \times 10^3}{42\pi} = 72533.27$$

$$\therefore d = 72533.27 = 41.70$$

$$\text{Take } d = 50mm$$

2.2.5.2 Calculation of Angular Deflection or torsional deflection of Shaft

The design of shaft for rigidity was based on the permissible angle of twist α. The amount of permissible angle of twist depends on the particular applications, and varies from about 0.3^0 per metre for machine tool shafts to about 3^0 per metre for line shafting (i.e. $0.3^0/m \le \alpha \le 3^0/m$). The deflection of shaft, α due to the load on it is given by:

$$\alpha = \frac{584 \times T \times l}{Gd^4}$$

$$\text{where } l = 1130mm \text{ and } d = 50mm$$

$$\therefore \alpha = \frac{584 \times 347.59 \times 1.13}{(0.05)^4 \times 80 \times 10^9} = 0.46^\circ / m$$

Since $\alpha = 0.0.46^0/m < 3.0^0/m$, therefore the design is satisfactory based on torsional rigidity.

2.2.5.3 Calculation of lateral deflection of shaft due to bending

The deflection at any point for a simply supported shaft of length l in which W is at mind point between the bearing supports is given as:

$$\delta = \frac{W l^3}{48EI}$$

$$\text{where } I = \frac{\pi d^4}{64} = \frac{\pi (0.04)^4}{64} = 3.07 \times 10^{-7} m,\ E = 207 \times 10^9 N/m^2, l = 1.13m \text{ and } W = 772.7N$$

$$\therefore \delta = \frac{772.7 \times 1.13^3}{48 \times 207 \times 10^9 \times 3.07 \times 10^{-7}} = 0.3655mm$$

Since $\delta_{max} < \delta_{all}$ then the design is satisfactory on the bases of lateral deflection.

2.2.5.4 Calculation of critical speed of shaft

Using Rayleigh – Ritz equation the critical speed of a shaft, may be calculated by

$$w_c = \frac{60}{2\pi}\sqrt{\frac{g}{\delta}}$$

Where g = 9.81m/s^{2}and $\sigma = \frac{0.3655}{1.13} = 0.3235\text{mm/m}$

$$\therefore w_c = \frac{60}{2\pi}\sqrt{\frac{g}{\sigma}} = \frac{60}{2\pi}\sqrt{\frac{9.81}{0.3235\times10^{-3}}} = 1662.91\,\text{rpm}$$

Since w_c = 1662.91rpm is far greater than 50rpm, then the design is satisfactory based on critical speed analysis.

2.2.6 Design of keys

In the design under consideration, the torque being transmitted from the diesel engine to the screw-shaft Via the gear box, is vibratory and heavy; hence taper keys were used; in accordance with BS4235: part 1:1972.

Since the diameters of the diesel engine shaft and the input gearbox shaft are both greater than 22mm; rectangular taper keys were selected for design, in accordance with BS 4235: part 1:1972.

Shear stress on the key, is

$$\tau = \frac{2M_t}{d_{shaft}\,1_d\,b}\ \text{on the key side, and}$$

Compressive (bearing) stress on the key, is

$$\delta_{bear} = \delta_{compressive} = \frac{2M_t}{(h-t_1)1_d\,d_{shaft}}$$

Since the rectangular taper key is wider than it is deep, the key will fail in compression,(bear) before it will fail in shear.

Hence, the limiting equation is:

$$\delta_{bear} = \frac{2M_t}{(h-t_1)1_d\,\text{dshaft}} \le [\delta_{bear}]_{allowable}$$

Where : Mt = Torque transmitted (N/m), d_{shaft} = shaft diameter (mm),

1_d = effective design length of key (mm), $4b \le 1_d \le 166$ (where b = width of key),

h = height of key (mm), t_1 = depth of key in hub (mm) and

for steel hubs $[\delta_{bear}]_{allowable}$ is taken as 70 Mpa or 70N/mm^2

2.2.6.2 Design of keys calculations

(a.) calculation of bearing stress in the key between the electric motor shaft and pulley hub

Given: Input power, P=3hp, motor speed, N_1 = 1440rpm, σ_{all} = 70mpa

$$l_d = 70\text{mm}, h = 10\text{mm}, t_1 = 6\text{m}, b = 16\text{mm}, d_{shaft} = 50\text{mm}$$

$$M_t = \frac{P}{\omega} = \frac{60P}{2\pi\pi_1} = \frac{60\times3\times746}{2\pi\times1440} = 14.84\,\text{Nm} = 14{,}841.20\text{Nmm}$$

$$\therefore \sigma_{bear} = \frac{2Mt}{(h-t_1)l_d d_{shaft}} = \frac{2\times 14841.20}{4\times 70\times 35} = 3.03\,N/mm^2$$

Since $\delta_{bear} < [\delta_{bear}]_{all}$, therefore the design is satisfactory from the stand point of bearing pressure.

(b.) calculation of bearing stress in the key between the gear box input shaft and the Pulley hub
Given that power, P=3hp =2238w.

$$N_2 = 600rpm, \lambda_d = 70mm, h = 10mm, \lambda, = 6mm, b = 6mm, d_{shaft} = 35mm, b = 16mm$$

$$\therefore m_\tau = \frac{60P}{P\pi\pi_2} = \frac{60\times 2238}{2\pi\times 600} = 35.6Nm = 35{,}618.88Nmm$$

$$\therefore \sigma_{bear} = \frac{2\times 35{,}618.88}{4\times 70\times 35} = 3.63N/mm^2$$

$$Since\sigma_{bear} \le [\sigma_{bear}]_{a\mu}$$

(d.) calculation of bearing stress (σ_{bear}) in the key between mixer shaft and pulley hub
Given=1820w

$$N_4 = 50pm, d_{shaft} = 50mm$$

$$\therefore M_t = \frac{60P}{2\pi\pi_4} = \frac{60\times 2238}{2\pi\times 50} = 427.43Nm = 427{,}426.52Nmm$$

$$\therefore \sigma_{bear} = \frac{2m_t}{(h-t_1)\lambda_d d_{shaft}} = \frac{2\times 127426.52}{(10-6)\times 70\times 50} = \frac{2\times 427426.52}{4\times 70\times 50} = 61.06N/mm^2$$

$$Since\ \sigma_{bear} \le [\sigma_{bear}]_{a11}$$

2.2.7 Design of Mixer Hub

From BS449, we have the following limit equation:

$$\frac{d_{hub}}{d_{shaft}} \ge 1.75$$

$$\therefore d_{hub} \ge 1.75 d_{shaft}$$

Take d_{hub} =1.8 d_{shaft}
Given that d_{shaft} =50mm

$$\therefore d_{hub} = 1.8\times 50 = 90mm$$

2.2.7.2 Design of weld thickness or size

The hub which is welded to the mixer chamber is secures to the shaft by means of keys on both ends of the mixer, hence the torque on the shaft is fully taken up by peripheral fillet weld.
This torque is given as ,

$$T = 2\pi\pi^2\tau h = \frac{\pi d^2}{2}\tau h$$

Where d=d_{shaft}=50mm, h=weld depth or weld throat = 0.707t
t =weld size and τ =allowable shear stress for the weld material = 42Mpa
$\therefore$ But T= 347.59Nmm

$$T = \frac{\pi(50)^2}{2} \times 42 \times h = 4271426.52$$
$$\therefore h = 2.59mm$$
$$\therefore \text{weld size or weld leg}, t = \frac{h}{0.707} = \frac{2.59}{0.707} = 3.67mm = 2.40mm$$

2.2.7.3 Design of key to secure hub mixer shaft

Bearing stress in the key between the mixer shaft and the hub weld to the mixer chamber is calculated as:

$$\sigma_{bear} = \frac{2TM}{(h-t)l_d} d\,\text{shaft}$$
$$\text{But } T = \frac{60P}{2\pi \times 50} = \frac{60 \times 2328}{2\pi \times 50}$$
$$\therefore T = 427{,}426.52Nmm$$
$$\therefore \delta_{bear} = \frac{2 \times 427426.52}{(10-6) \times 70 \times 50} = 61.66N/mm^2$$

2.2.8 Selections of bearings:

2.2.8.1 Determination of reaction at bearings supports

The drive for the tumble mixer is subjected to a considerable cantilever load, F_c (radial load) due to the belt transmission and carries a radial load in developed by the tumble mixer the support reactions are found from appropriate equilibrium equations.

$$R_{YA} + R_{YB} = W + Fc = 2146.8$$
$$\sum M_z = We + Fc(2l_1 + l_2) - R_y B(2l_1) = 0$$
$$= 772.9(593) + 1373.9\,(1235) - R_{yB}\,(1186) = 0$$
$$\therefore R_{YB} = \frac{772.9(593) + 1373.9(1235)}{1186} = 1817.1N$$
$$\Rightarrow R_{yA} = 2146.8 - 1817.1 = 328.9N$$

Since is no horizontal component of wand Fc (i.e. no axial components) than the overall radial reactions of the bearing are as follows:

For support A: $R_A = R_Y A = 328.9N$

For support B: $R_B = R_Y B = 1817.1N$

2.2.8.2 Determination of dynamic load rating of the heaving support bearing

The reaction at the support B is used to determine the dynamic load rating of the bearings since it is the most heavily loaded. The relationship between the basic rating life the dynamic load rating and the bearing load is given by formula (SKF bearing catalogue 3200-I/E, 2001).

$$L_{10h} = 500\, f_h^{\,p}$$
$$f_n = \left(\frac{33.3}{n}\right)^{1/p}$$

$$\text{where}: L_{10h} = \text{Basic rated life, hour}, f_h = \text{life factor}, f_h = \text{speed factor}$$

$$n = \text{rotational speed rpm}, P = 3 \text{ for ball bearings and } p = {}^{10}\!/_{3} \text{ roller bearings}$$

$$\text{The basic rated life can also be expressed as}:$$

$$L_{10h} = \frac{10^6}{60n}\left(\frac{Cr}{P}\right)$$

SKF recommends a life of 20,000hrs to 30,000hrs for machines in general in the mechanical industries, where machines are fully utilized for 8-hours service.

Therefore, assuming $L_{10h} = 30{,}000\text{hrs}$ and for speed of mixer shaft of50 rpm, we have:

Speed factor, $f_n = \left(\frac{33.3}{n}\right)^{10/3} = \left(\frac{33.3}{50}\right)^{10/3}$

$\therefore f_n = 0.8852$

For L_{10h} = 30, 000hrs, we have f_h = 3.40

But $C_r = P\frac{f_h}{f_n}$

Where p = P_r = 1817.1N for bearing support at B.

$$\therefore C_r = 1817.1X\frac{3.40}{0.8852} = 7{,}186.8N$$

For a bearing to be selected for application the calculated dynamic load rating, C_{rc} must be less than the required dynamic load rating, C_r.

Hence, the nearest to C_r to the calculated value for bore diameter d = 35mm is C_r = 22, 600N is selected. This bearing has the following feature D = 62mm, B =14mm, limiting speeds (grease =12000rpm and oil =15000rpm).

2. 3 Description of machine

The pictorial view of the tumble feed mixer machine are shown in Figure 3. The machine consists of the following main parts: the frame, the mixer drum, the shaft, the motor electric and the gearbox/mixer shaft pulleys and the feed receptacle. The shaft attached to the mixing chamber turns at a relatively slow speed of 50rpm to effect the proper mixing of the feed material. The mixing section consists of the tumble chamber inclined at 30° to the horizontal shaft. Both the mixing chamber and the feed receptacle were constructed using a mild steel sheet metal of 3 mm thickness. The feed inlet and outlet is located at the same end of the mixer chamber above the receptacle to discharge the mixed feed into the receptacle. All the parts that make up the machine are mounted on a frame robustly built with welded stands. An angle iron of 50 mm x 50 mm x 5 mm is used in the construction of the frame. The frame is constructed with the following dimensions: 1300 mm height, 1000 mm length and 500 mm width at its base.

3. RESULTS AND DISCUSSION

3.1 Material used for construction

Mild steel: mild steel is an alloy of iron which contains 0.1-0.25 carbon. The use of this material for the construction of the shell was informed by the good physical and chemical properties by the material such strength and its availability.

Electric motor: It is a 2hp 3 phase type of medium speed it is ruggedly designed for tough duties and sealed for moisture resistance that is very prevalent in the construction environment

Machine frame: The materials of the have good stress resistance ability, high strength, high ductility, and is creep resistance. Material selected for the manufacture of the machine frame is mild steel.

3.2 Method of construction

The breakdown of the construction of the concrete mixer is sectioned as follows:

A. Machine frame construction
B. Mixer drum construction
C. Motor unit construction
D. Gear box speed reducer
E. Assembly of components.

The mud steel angle bars, and metal sheet were cut with cutting tools. They were arranged and joined together, by appropriate joining method such as welding riveting and brazing.3 Assembly of components

The various parts were welded and bolted together. The electrical components assembly process followed. The machine frame was laid first, followed the drum, the motor and then all electrical component parts are assembled in the control panel.

3.3 Testing

The feed mixer was tested in a feed process industry used to mix 50kg of feeds to determine its effectiveness. The effectiveness of the machine was determined by noting the different produced by the mixer drum during the mixing process. Effective and thorough mixing was achieved after just 15 minutes of mixing. During the test, it was observed that the more viscous, the harder it was for the feed to mix thoroughly.

3.4 Process Technology

The mixing action is achieved by the thorough agitation of the feed which basically consists of soya beans, water, protein and oil. The tumbling action induced by the shaft rotation and the inner screw linings on the shaft; ensures that the feed is properly mixed.

4. CONCLUSION

The design and fabrication of the tumble feed mixer has been carried out using available local materials. The output is 50kg of mixing per mixing duration using a 3hp electric motor. The machine consists of the machine frame, the mixer drum, the electric motor, speed reducer and the pulleys. The capacity of the drum is 50kg maximum of fish feed materials and the electric motor selected is 3hp three phase electric motor and the machine efficiency was 98.3%..

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APPENDIX