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*RESEARCH PAPER*

# DESIGN, FABRICATION AND EVALUATION OF FISH MEAL PELLETIZING MACHINE

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**ABSTRACT**
***A 113.1kg/h fish meal pellet processing machine which produced 4mm diameter pellet, with an average length of 6mm was designed and fabricated. Design values of $21^0$ was used for the maximum angle that the hopper wall formed with the vertical in the discharge zone, a critical stress of 1.3kPa of the ground particulate materials, and a density of 2.4521kg/$m^3$ of the particulate materials, were used to obtain a hopper smallest outlet diameter of 12.7cm with a capacity of 24,118$cm^3$ which proved efficient for the pelletizing machine. Two 3-cm diameter shafts carried the speed reduction gears with the perforated disc attached to the roller cutters on one end, while at the other end a 5hp motor was connected to the speed reduction gear by pulleys with diameters of 6cm and 12cm respectively. The speed reduction was 1:5 over a motor speed of 2000rpm. The fish meal pelletizing machine utilized 4kg of ingredients to produce 3.77kg pellets at an efficiency of 94.2%. The percentage loss due to unprocessed ground particulate materials was 5.8%. The moisture content of the fish meal pellets after 7days of drying in open air was 26.5% (wet basis). When tested for floatation, the pellets stayed afloat for 9days, while the un-dried pellets only remained afloat for 2days. A combination of the weight of the twin roller cutters and the addition of some starch to the ground particulate materials assisted the compacting and gelatinization of pellet formed. This machine will be useful to medium and small scale aquaculture farmers and also reduce the need for foreign sources of fish meal in the aquaculture industry, thus conserving foreign exchange.***

**Keywords**: *ground particulate materials, mixing, compacting, gelatinizing, pelletizing*

## INTRODUCTION

Currently, the annual importation of fish in Nigeria stands at about ninety billion naira which meets the national demand of three million tonnes (Punch 2014). The increasing growth in population in this nation and the need for good health, requires that improved methods of increasing fish production.. The first fish farm in Nigeria was established about 57 years ago at Onikan, Lagos (Olukunle, 2000). To grow the fish in aquaculture requires proper care and feeding, with fish meal made into pellets which contain ingredients that constitute a balanced diet for the fish. Such ingredients consist of soya beans (18-50% by weight), lipids which are fat (10-25% by weight), groundnuts and carbohydrates (15-20% by weight) and cereal, mostly maize is used in Nigeria (10% by weight). In addition, less than 5% of the following are mixed and added to the ingredients before being processed; blood meal, fish cramps, palm kernel cake, cotton seed, premix that provides the vitamin and mineral source for the fish (Atoyebi, 2007). There are two types of pellet making machines that are known universally. These are the disc-die type, and ring die type, (Kaankuka and Osu, 2013). The Chinese, Asians and Europeans made many modern disc die and ring die pellet machines (ASTW, 2014 and LIVI, 2014).These modern fish meal plants are elegant but very expensive. Consequently, there is a need for the local development of our own fish meal machines that will produce the same pellets efficiently, but at an affordable cost and simple to operate by our farmers. This will enable the nation to produce more fish to meet the protein, mineral and oil needs necessary for a healthy nation and also conserve foreign currency (Olusegun and Adekunle, 2009).

It was necessary to develop a fish meal pelletizer that would improve on the previous models, particularly those fabricated by Olusegun and Adekunle (2009) and Kaankuka and Osu (2013). It was envisaged that an improvement was possible if the good design features of each of the machines developed by these researchers could be integrated in a new fish meal pelletizing machine. Therefore, a machine with similar high capacity electric motor, a perforated disc die and roller cutter, powered through a gear reduction mechanism was the focus of this project. It is believed that the success of this project trough adoption of this machine would provide an increased quantity of fish meal pellets and hence an increase in fish production.

Olusegun and Adekunle (2009) developed a fish meal pelletizing machine that was of the screw type, which did not use steam to gelatinize the ingredients. The binding of the ingredients was achieved by the heat generated by the screw action of the machine and the chemical binder added to the ingredients. The machine had a die at the end of the screw chamber. The die sizes were interchangeable from 6 to 13mm to suit the size of pellet to be produced. The machine was powered by a 3hp electric motor with a speed of 1440rpm. .

The dual purpose fish pellet making machine designed and fabricated by Nwaokocha and Akinyemi (2010) could be operated electrically or manually. It was a low cost machine for making pellets for the purpose of research and for small scale production. The machine used a worm screw to propel the ingredients through the die. A hopper contained the ingredients which were conveyed by the worn screw that compacted it and expel it through the die. The pellets emerged at the end of the die. The machine capacity was 25kg/h and its electric and manual efficiencies were 92.6% and 91.4% when run by the electric motor and manually respectively. Kaankuka and Osu (2013) developed a revolving die and roller fish feed pellet machine. They observed that the higher pellet output obtained from a die speed of 761 rpm could be due to the higher heat generated during the production process which resulted in proper gelatinization of the carbohydrates in the ingredients. The gelatinized starch acted as a binder that reduced crumbling during pellet making.

The objective of this study is to design and fabricate a fish meal pellet making machine for rural application specifically. It is expected that it will be easy to operate and affordable. It is envisaged that when a number of these machines are produced and introduced into the market they will enhance development.

## MATERIALS AND METHODS

The choice of materials, design and fabrication was done in this section. The design of the perforated disc and rollers, the connecting shaft, the drive shaft, the pulley drive and the hopper were taken into consideration.

### Design of hopper

Freeman (2009) suggested that finely ground material flow in a hopper is governed by the shear properties of the material, wall friction, and compressibility, Flow stoppages are known to occur in hoppers as a result of, arching in the bin, size segregation and other reasons. When hoppers are properly designed they minimize and even suppress the problems mentioned above. The Jenike Theory proposed in 1964 is still in use today to help in the design of hoppers that are efficient (Jenike, 1964). The theory is applied to, the maximum angle that the wall needs to form with the vertical in the hopper discharge zone, he minimum outlet diameter required for appropriate, unbroken material flow during discharge, the influence of the hopper surface on the type of flow to be analyzed, and the final design to be tested (Fig. 1).

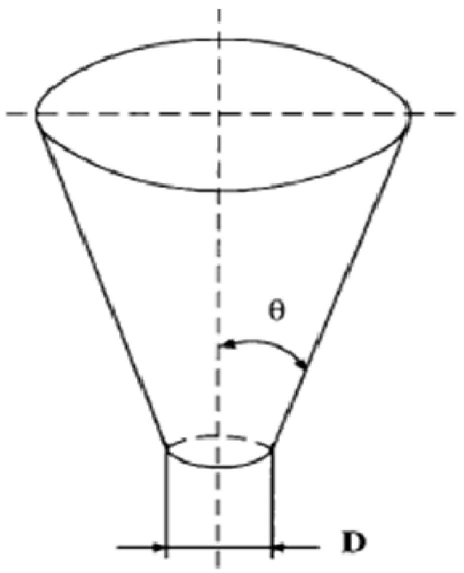


**Fig. 1: Hopper design variables**

According to Amoros *et al*, (2000), the equation for the smallest outlet diameter at which the hopper discharge occurs by uninterrupted mass flow is given by equation (1).

$$D = \left(2 + \frac{\theta_h}{60}\right) \times \frac{CAS}{\rho \times g} \quad (1)$$

where, D = the smallest outlet diameter at which the hopper discharges (m) $\theta_h$
the angle between the vertical and the hopper wall in the discharge zone ($^o$)
CAS = critical stress (Pa)

$\rho =$ particulate density $\left(\frac{Kg}{m^3}\right)$

g = the acceleration due to gravity $\left(\frac{m}{s^2}\right)$

The density of the ground particulate material was found by displacing 50g of its weight in water and measuring the volume thus the density of the ground particulate material was the 50g weight divided by its equivalent volume of water and was 2.4521 kg/m$^3$.

Tomas (1997) developed a series of curves on a graph of wall friction $\varphi_w$($^o$) against Hopper angle versus vertical angle $\theta_n$ ($^o$) for effective angle of internal friction $\varphi_e$. The cluster of intersecting curves was prominent at the range $\theta$ between 20$^0$ to 23$^0$. Also, Amoros (2000) confirm the findings of Tomas (1997). Therefore, this work chose $\theta_{h\,=}\,21^0$ as the hopper angle versus the vertical angle, as it is within the range of findings of the earlier researchers.

According to Freeman (2009), the point of similarity of normal stress and shear stress at a yield locus of 3kPa is 1.3kPa, which is assumed as the critical stress.

$$D = \left(2 + \frac{21}{60}\right) \times \frac{1.3}{2.4521 \times 9.81} = 0.127m$$

Hence, required hopper outlet discharge diameter is thus 12.7cm.

**Selection of speed ratio and motor speed**
Popp (2001), published a chart based on his research work on motor speed, speed ratio and horsepower for effective torque application on machines. A speed ratio reduction of 1:5 at a motor speed of 2000rpm and 5hp was selected from the chart. Hence, a design speed of 400rpm was used to provide the total torque required for this work.

**Design of the perforated disc and roller**
Fig. 2 shows the diagram of the perforated disc and rollers. The equation for force that will generate motion is given by Khurmi and Gupta (2006) as equation (2).

F = $I_\theta \ddot{\theta}$ where,

F = centrifugal force (N)
$I_\theta$ = moment of inertia of perforated disc ($m^4$)
$\theta$ = angular displacement (rad)

Ӫ=angul Ӫar acceleration $(\frac{rad}{s^2})$

$\omega = \theta$ = angular velocity $\frac{rad}{(s)}$

Let the solution of equation (2) be,

$$\theta = \hat{\theta} \sin \omega t \quad (3)$$

$$\therefore \dot{\theta} = \omega \hat{\theta} \cos \omega t \quad (4)$$

$$\therefore \ddot{\theta} = - -\omega^2 \hat{\theta}\, \omega t \quad (5)$$

Substituting the solution in equation (2) gives

$$F = \pm \omega^2 \hat{\theta} I_\theta \sin \omega t \quad (6)$$

At Maximum Force $\hat{F}\theta = \omega t = \frac{\pi}{2}$ (7)

Therefore equation (6) becomes,

$$\hat{F} = +\omega^2 \frac{\pi}{2}(I_G + mh^2)$$

where,

$I_G$ = moment of inertia acting from the center of gravity

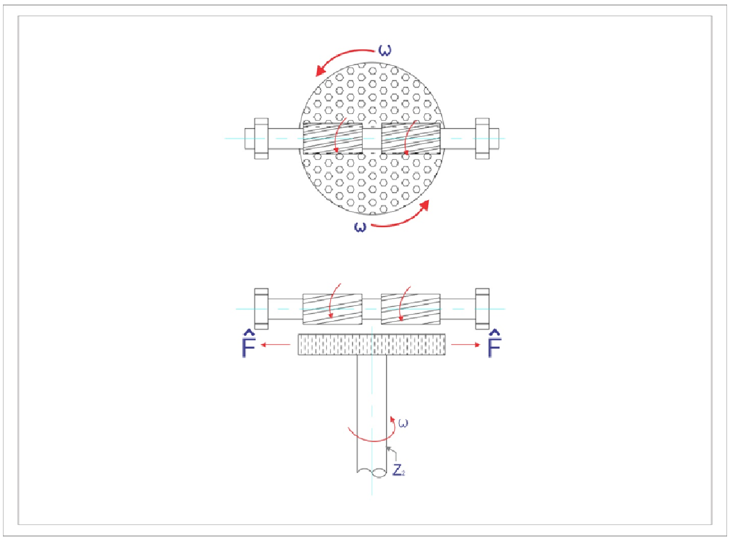


**Fig 2: Centrifugal force on perforated disc, angular rotation of disc and roller cutter**

h = distance of the c.g from the point of rotation (m) ( Lewis and Samuel, 1984)

$$\therefore \widehat{F} = \pm\omega^2 \frac{\pi}{2} \left(\frac{m}{2} r^2 + mh^2\right) \quad (8)$$

where,

m = total mass (kg) of perforated disc plus 2 roller cutters plus ground particulates in hopper when full

r = radius of the perforated disc (m)

Substituting in equation (8),

$$\widehat{F} = \pm 41.9^2 \frac{\pi}{2}\left(\frac{5}{2} 0.114^2 + 5 \times 0.03^2\right) = \pm 41.9^2 \times \frac{\pi}{2} \times 5\left(\frac{0.114^2}{2} + 0.03^2\right) = 102.01N$$

**Design of perforated disc shaft under combined twisting moment (T) and bending moment ($M_e$)**

From fig. 3, the force diagram shows $R_e$ due to the weight of ingredients, rollers and perforated disc, with

$\widehat{F}$ acting along the horizontal plain. Hence,

$$R_e = \sqrt{\{(mgh_e)^2 + \widehat{F}^2\}} \quad (9)$$

$R_e$ = Resultant Force acting at $45^0$ in the direction of the meshing gears. (N)

$h_e$ = total head of ground particulates in hopper (m).

Substitute into equation (9)

$$R_e = \sqrt{\{(5 \times 9.8 \times 0.54)^2 + 102.01^2\}} = 105.4N$$

Using a toque arm of 31cm (Oyeleke, 2006), the bending moment is,

$$M - 105.4 \times 0.31 - 32.7Nm$$

$$T = \frac{P \times 60}{2\pi N} = \frac{5 \times 746 \times 60}{2 \times \pi \times 400} = 89Nm \quad (10)$$

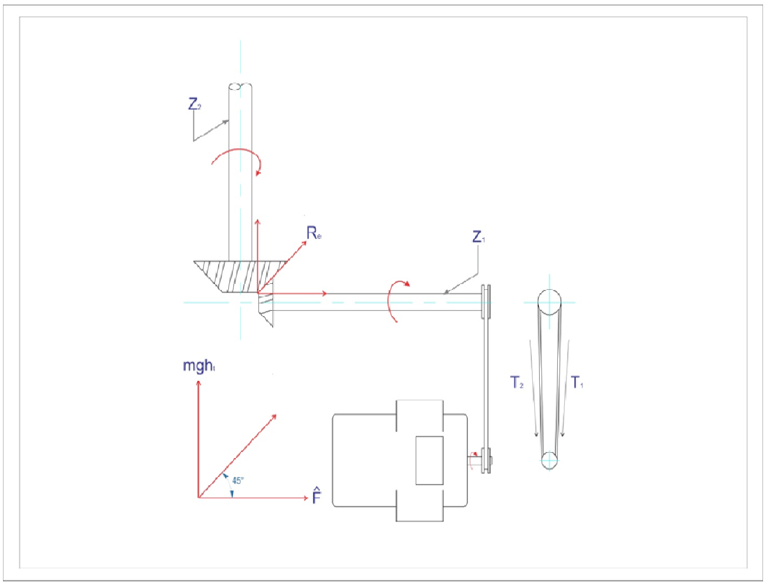


**Fig 3: Forces acting on the reduction gear and pulley shafts showing related forces and direction**

where, T= twisting moment (Nm)
P= motor power (W)
N= motor speed (rev/min)

For a shaft under combined twisting moment and bending moment, using the Theory of Tresca- Guest for mild steel, the maximum shear stress is given as,

$$\hat{\tau} = \frac{1}{2}\sqrt{\sigma_b{}^2 + 4\tau^2} \qquad (11)$$

$$\hat{\tau} = \frac{1}{2}\sqrt{\left(\frac{32M}{\pi d^3}\right)^2 + 4\left(\frac{16T}{\pi d^3}\right)^2} = \frac{16}{\pi d^3}\sqrt{M^2+T^2} \qquad (12)$$

$$\therefore \frac{\pi}{16} \times \tau \times d^3 = \sqrt{M^2+T^2} \qquad (13)$$

Maximum normal stress is given by,

$$\hat{\sigma_b} = \frac{1}{2}\sigma_b \div \frac{1}{2}\sqrt{\sigma_b{}^2 + 4\tau^2} = \frac{32}{\pi d^3}\left[\frac{1}{2}\left(M + \sqrt{M^2+T^2}\right)\right]$$

$$\therefore M_e = \frac{\pi}{32}\sigma_b d^3 = \frac{1}{2}\left[M + \sqrt{M^2+T^2}\right] \qquad (14)$$

where, $M_e$ = equivalent bending moment due to bending and twisting (Nm)

$\sigma_b$ = maximum permissible shear for mild steel shaft with key way and is 45MP$_a$ (Jones, 2002; and Khurmi and Gupta, 2006).

Substituting into equation (14), and making d the subject of the equation,

$$d^3 = \frac{16}{\pi \times \sigma_b} \times \left[32.7 + \sqrt{32.7^2 + 89^2}\right] = 10.73 \times 10^{-6}$$

$$\therefore d = 0.0221\text{m} = 2.21\text{cm}$$

The design for the diameter at the perforated disc end was taken as 3cm for safety and availability in the market.

**Design for reduction gear drive shaft under combined twisting and bending moment ($M_r$)**

The load acting on the reduction gear drive shaft is the perforated disc plus two roller cutters and the weight of the ground particulate materials. The torque arm was 31cm as used in the earlier work of Oyeleke (2006) and Olusegun and Adekunle(2009).

Hence the bending moment of the reduction gear drive shaft ($M_r$), is given as,

$$M_r = (mgh) \times L \qquad (15)$$

where, $M_r$ = bending moment of the reduction gear drive shaft (Nm)

L = the torque arm (m)
m, g, and h are defined earlier on in this work.

Substituting in equation (15),

$$M_r = (5 \times 9.8 \times 0.54) \times 0.31 - 8.2Nm$$

For the twisting moment of the reduction gear drive shaft ($T_r$), with the full motor speed,

$$\therefore T_r = \frac{5 \times 746 \times 60}{2 \times \pi \times 2000} = 17.81Nm$$

$$d^3 = \frac{16}{\pi \times 45 \times 10^6}\left(8.2 + \sqrt{8.2^2 + 17.81^2}\right) = 0.00000314m$$

$$d = 3cm$$

Hence a choice of 3cm diameter shafts was taken for the two shafts for design conformity and also because of market availability.

**Selection of pinion and wheel pulley diameter**

The pinion pulley and wheel pulley were selected with diameter ratio of 1:2 (Csavas *et al*, 1979), the sizes were 6-cm diameter for pinion and 12-cm diameter wheel.

**Fabrication of the fish meal pelletizing machine**

80-cm long mild steel rod of 3cm diameter was cut in two equal halves, 40cm length each. One end of each rod was reduced to a diameter of 1.5cm to fit the two protruding sleeve ends of the gear box. A 14 gauge steel sheet was used to fabricate a metal box 35cm in length, 33cm in breadth and 40cm in height inside which the gear box was fixed to the angle frame on which the metal box was also attached with nuts and bolts, Half way along the middle of the metal box, an opening was made for the exit of the fish meal pellets after they have been processed.

A stainless steel sheet 30cm long, by 30cm wide, by 6cm thick was marked with a scribe at its center. Radial lines were scribed at intervals of $5^0$ at the center point which gave 36 lines projecting from the center. A metal compass was used to scribe a 23cm diameter circle from the center point. Thereafter, other concentric circles were drawn at intervals of 5mm which gave 25 circles. These produced very many points of intersection at which 4mm perforations were made using a 4mm drill. The outer diameter of 23cm was cut with the use of a gas cutter which produced the perforated disc. The disc was cleaned with an electric grinder which reduced the final disc diameter to the required 22.9cm. The 3cm diameter end of the 40cm shaft was welded to the back of the perforated disc and at its center. The other 1.5cm end of the shaft was attached to the free end of the speed reduction box by means of the sleeve and bolts

A high carbon steel sheet of length 18.2cm, with a width of 10cm and thickness 4mm was grooved along its width to the depth of 2mm with the aid of a 2mm diameter tungsten carbide milling tool, the grooves of 2mm were spaced at intervals of 6mm to form the cutting edges. The grooved sheet was cut into two along the length to form two separate sheets 9cm in length by 10cm width and 4mm thickness. The cutting was done using a gas flame cutter and cleaning was done using an electric hand grinder. A mild steel shaft 18cm in length and 10cm diameter was cut into two halves, each 9cm in length. The shafts were bored along the diameter center line to a size of 3.1cm diameter to allow a roller shaft of 3cm to fit in with ease. The grooved sheet was folded and wrapped around the 9cm mild steel shaft. The two edges were tacked with a spot weld to hold them in position before they were finally welded together. The procedure was repeated for the second shaft. A shaft 30cm length and 3cm in diameter, made from mild steel was push into the 3.1cm diameter bore of the roller cutter. The two cutters were aligned with a 2.5cm space between them. The rollers were then welded on to the shaft making sure that the axial alignment was horizontal with the use of a steel angle. The weight of each roller cutter was 1Kg so that it could roll properly to perform the twin role of compacting the feed ingredients and cutting the pellets, (Talmadge, 1998; IITA, 2007). They were fitted with two ball bearings each for easier motion while cutting.. At the two ends of the roller cuter shaft where the ball bearings were located, there were adjusting nuts placed to control the clearance of the cutters and the perforated disc. The perforated disc, the roller cutter and the hopper were supported on a mild steel plate 6mm thick which was fastened to the mild steel box with four 17mm bolts 10cm long which had sleeves for the purpose of adjustment.

The hopper was constructed in two parts, the bottom and the top parts. The bottom half was fabricated using 14 gauge metal steel sheet. It was cut to a length of 72.3cm and width of 12.7cm. Then it was folded along the width to form a cylindrical shape. The top part of the hopper was made by marking out a trapezoid on 14 gauge metal steel sheet. The two parallel sides of the trapezoid were of length 129cm and 23cm respectively. It was then folded to form a cone-shape without a top, which when inverted on the cylindrical shape formed the hopper. The cylindrical bottom and the top half cone were panel-beaten to make the hopper. A cover which opened half way was fabricated and fitted to the top of the hopper to stop spillage of the ingredients.

**Installation of the electric motor to the machine**

The motor was mounted on one side of the rectangular frame 80.5cm in length, and 36cm in width. It was bolted down with four 17mm nuts and bolts. The metal box in which the gear box was fitted, the hopper, the roller cutter and the perforated disc, all rested on the angle frame on which the metal box stood and was bolted down.. The pulley size on the drive motor was 6cm in diameter. The pulley on the gear reduction box end was 12cm in diameter. Both pulleys were mounted using rectangular sunk keys. A belt size of 114cm was used to connect pinion and wheel.

**Testing the fish meal pellet machine**

Ingredients by weight were prepared as follows, soya bean meal (35%), fat (18%), groundnut meal (18%), maize (10%), blood meal (4%), fish crumbs (3%), palm kennel cake (3%), cotton seed cake (3%), premix (3%), with starch (3%) and little water as a binder, (James and Geoff, 2001, Ayodele and Ajani, 1999). They were properly mix with 1.8kg of water to an ingredient total weight of 3kg which gave a water to ingredient ratio of 1:1.7. The mixture was poured into the hopper and the machine was switched on. After adjusting *the* roller cutters with the adjusting screw, fish meal pellets compacted through the die were cut so that they dropped through the discharge tray to the container. The pellets were weighed and the unprocessed ingredients were also collected and weighed separately. The test was repeated with ingredient weights of 4kg and 5kg. The experimental results are repeated in Tables 2 and 3. The efficiency of the fish pellet machine, the percentage dryness after 7days drying in open air, and the production rate were determined.

**RESULTS AND DISCUSSION**

The test results are shown in Table 1. Table 1 shows that for the first sample, when 3kg of the ground materials were processed through the

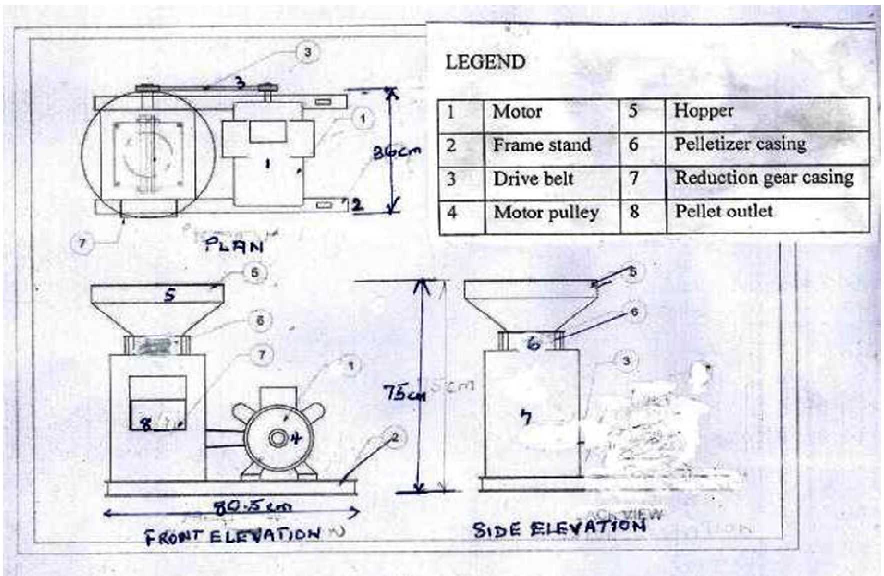


**Fig. 4: The designed and fabricated fish meal pellet machine**

**Table 1: Weight, efficiency and dryness of processed fish meal pellets**

| Sample Tested | Weight of Ingredients (Kg) | Weight of Fish Meal Pellets (Kg) | Weight of Residue Ingredients (Kg) | Efficiency of Fish Meal Machine (%) | Percentage Loss due to Ingredients not Pelletized (%) | Weight of Pellets after 7days Drying (kg) | Comparative Percentage Dryness after 7 days Drying (%) |
|---|---|---|---|---|---|---|---|
| 1 | 3 | 2.80 | 0.20 | 93.3 | 6.7 | 2.05 | 26.8 |
| 2 | 4 | 3.78 | 0.28 | 94.5 | 5.5 | 2.78 | 26.5 |
| 3 | 5 | 4.74 | 0.36 | 94.8 | 5.2 | 3.50 | 26.2 |
| Average | 4 | 3.77 | 0.28 | 94.2 | 5.8 | 2.78 | 26.5 |

**Table 2: Production rate of pelletizing machine**

| Sample tested | Weight of Ingredients (kg) | Weight of Fish meal pellets (kg) | Time taken to process Pellets (min) | Production Rate (kg/hr) |
|---|---|---|---|---|
| 1 | 3 | 2.80 | 1.5 | 112.00 |
| 2 | 4 | 3.78 | 2.0 | 113.40 |
| 3 | 5 | 4.74 | 2.5 | 113.76 |
| Average | 4 | 3.77 | 2.0 | 113.10 |

pellet mill, 2.8g of fish meal pellets were got with 60% dryness. A residue of 0.2kg of ingredient was not processed to pellets. Therefore efficiency of the machine was calculated 93.3%. The percentage loss due to ingredients not pelletized was 6.7%. A combination of the weight of the twin roller cutters and the addition of some starch to the ingredients, assisted the gelatinization, and reduced crumbling during pellet making.

The drier the fish meal pellets, the longer they stayed afloat on water to enable the fish to feed on. Hence, the processed pellets were dried in the open air for 7 days, after which they were weighed and the final weight of 2.05Kg was recorded as the weight of the drier pellets. The percentage dryness after 7days drying of the pellets in open air was calculated and recorded as 26.8% (Table 1). When tested for floatation, the pellets stayed afloat for about 9days, while the un-dried pellets only remained afloat for 2days.

The machine efficiency, percentage dryness and production rate were evaluated as follows:

Efficiency of the fish meal pellet machine

$$= \frac{weight\ of\ Fish\ Meal\ Pellets}{Weight\ of\ Ingredients} = \frac{2.8}{3} = 93.3\%$$

Percentage loss due to ingredients not pelletized

$$= \frac{Weight\ of\ Residual\ Ingredients}{Weight\ of\ Ingredients} = \frac{0.2}{3} = 6.7\%$$

Percentage dryness after days drying

$$= \frac{Wt\ of\ Fish\ Meal\ Pellets - Wt\ of\ Pellets\ after\ 7days\ drying}{Wt\ of\ Fish\ Meal\ Pellets} = \frac{2.8-2.05}{2.8} = 26.8\%$$

From Table 2,

The production rate $= \frac{2.8}{1.5} \times 60 = 112 kg/h$

The average production rate for the three samples was 113.1*kg/hr*

The above calculations were repeated for Samples 2 and 3. The average of the three calculations for the efficiency of the pellet machine, the percentage loss due to ingredients not pelletized, and the percentage dryness were used in deciding the final performance of the machine, (Sena and Treror, 1996 and LIVI, 2014). It was observed that the machine performance was best when 4kg of ingredients were loaded into the hopper to produce 3.78Kg of pellets at an efficiency of 94.5%. The moisture content of the pellets was 26.5% after 7 days drying in the open air. One good feature of the machine was that no waste was recorded as the residue ingredients were recycled.

## CONCLUSION

A fish meal pellet machine was designed, fabricated and evaluated. The shaft of the perforated disc was designed under combined twisting and bending. A resultant force of 105.4N, a bending moment of 32.7Nm, a twisting moment of 89Nm and a shear stress of 45MPa were used to achieve a shaft diameter of 3cm, under a reduced speed of 400rpm. The second shaft that linked the motor pulley to the gear reduction mechanism was likewise designed under combined twisting and bending, yielding a shaft of similar diameter of 3cm. A similar diameters shaft linked the speed reduction gear of 1:5 to the drive motor, and perforated disc which carried the total weight of the ground particulate materials, and roller cutter. The 5hp motor with a speed of 2000rpm reduced to 400rpm supplied enough torque to the machine which processed on average 4kg of particulate materials to produce 3.77kg of fish meal pellets in 2minutes at a production rate of 113.1kg/h. The average diameter and length of the pellets were 4mm and 6mm respectively. The machine operated with an average efficiency of 94.2% and a percentage loss of 5.8% due to some particulate materials which were not pelletized. The average moisture content of the pellets after 7days of drying in the open air was 26.5%. When tested for floatation, the pellets stayed afloat for about 9days, while the un-dried pellets were afloat for 2days. The design of this machine is an improvement over those of Kaankuka and Osu (2013), Nwaokocha and Akinyemi (2010) and Olusegun and Adekunle (2009).

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| NOMENCLATURE | Meaning | Units |
|---|---|---|
| ASME | American Society of Mechanical Engineers | |
| ASTW | A. S. Thai Works Ltd | |
| CPM | Control Program for a Microcomputer | |
| CAS | Critical stress | Pa |
| d | Diameter of shaft | m |
| D | Smallest outlet diameter at which the hopper discharges | m |
| F | Centrifugal force | N |
| $\hat{F}$ | Maximum Centrifugal Force | N |
| F.A.O. | Food and Agriculture Organization of the U. N. | |
| g | Acceleration due to gravity | $m/s^2$ |
| h | Distance of center of gravity from point of rotation | m |
| $h_e$ | Total head of ingredient in hopper when full | m |
| $T_1$, $T_2$ | Tensions on tight end and slack end of pulley | N |
| $T_r$ | Twisting Moment of reduction gear drive shaft | Nm |
| $\omega$ | Angular velocity | rad/s |
| y | Distance of shaft from neutral axis to the outer diameter | m |
| $\varphi$ | Phase angle | rad |
| $\theta_h$ | Angle b/w vertical & the hopper wall in the discharge zone | rad |
| $\Theta$ | Angular displacement | rad |
| $\ddot{\Theta}$ | Angular acceleration | $rad/s^2$ |
| $\hat{\theta}$ | Maximum angular displacement | rad |
| $\rho$ | Particulate density | $Kg/m^3$ |

## APPENDIX

**Materials cost**

| S/N | Components | Material | Quantity | Rate (N) | Amount (N) |
|---|---|---|---|---|---|
| 1 | Shaft | Mild Steel | 1 | 3500 | 3,500 |
| 2 | Frame | Angle Iron | 2 | 5000 | 10,000 |
| 3 | Machine Housing | 14 Gauge Steel Plate | $1\frac{1}{2}$ | 15,000 | 22,500 |
| 4 | Disc Die | Stainless Steel Plate | $\frac{1}{4}$ | 7,000 | 7,000 |
| 5 | Roller cutter | High Carbon Steel Plate | $\frac{1}{4}$ | 8,500 | 8,500 |
| 6 | Reduction Gear | High Alloyed Steel | 1 | 20,000 | 20,000 |
| 7 | 5hp Electric Motor | Motor | 1 | 25,000 | 25,000 |
| 8 | Ball Bearings | Ball Bearing | 6 | 800 | 4,800 |
| 9 | Pulleys | Aluminum Alloy | 2 | 600 | 1,200 |
| 10 | Housing and Frame | Welding Electrode | 1 Pack | 700 | 700 |
| 11 | Nuts and Bolts | Mild Steel | | | 800 |
| 12 | Miscellaneous | | | | 26,000 |
| 13 | | | | Total = | 130,000 |