

DESIGN AND CONSTRUCTION OF A FISH FEED EXTRUDER

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Abstract- In fish farming, the consumption of feeds in pellet form aids the total consumption of the ingredients by fishes. Nevertheless, some local farmers in Nigeria are using pelletized feeds in their various farms because of the reduced cost. However, some readily identifiable disadvantages of such pelletized feeds exist. As such extruded feeds have recently become appealing to local farmers and those who use the extruded feeds rely heavily on importation. The aim of this project is to design and fabricate a fish feed extruder with improved qualities and affordable cost using locally sourced materials to aid the production of extruded feeds by local small scale farmers. The design was carried out using engineering principles with due consideration to cost, ease of operation, serviceability, durability and performance. Based on these, an extruder using 5HP, 3-phase electric motor with a production capacity of 300kg/hr was designed and fabricated using locally sourced mild steel materials. It consists of hopper, screw, barrel, die, drive system and heater. An element of 1000W was attached to the barrel surface so as to ensure cooking of resin pellet. The cylindrical pellets size produced was in the range 3–5mm diameter suitable for fish and poultry. Based on this evaluation, there was a drastic improvement in the quality of locally processed pellets. The designed and fabricated extruder performed smoothly as a fish feed extruder.

Index Terms- Extruded feeds, Pelletized feeds, Pellets, Fish feeds, Extruder.

1. INTRODUCTION

Fish feed extruder is a high temperature short-time bio-reactor which can transform a variety of raw ingredients into pellets for production of healthy wholesome fish feed. It is one of the prominent developments in fish farming which contributes to the huge step forward in aquaculture. Fish feed extruder comprises of hopper, a barrel which houses, the screw conveyor, the cutting knife, the die, a heater etc. It can be divided into wet and dry type mill. They both can produce floating fish feed pellets, slowing sinking pellets and sinking feed pellets. The main difference is whether need to connect a boiler/heater to send steam before extruding. Pellets can be produced from both pelletizer and extruder. Pelletizing was introduced into Europe about 1920 and into the United States feed industry in the late 1920's Schoeff, (1994) cited by Behnke,

(1994). Its popularity has grown steadily until about 80% of all feeds in the U.S. are currently pelleted. Yet, the demand for and production of pelleted feeds for fish are practically non-existent in many African countries in spite of the known feeding and handling advantages of pelleted feeds Anon, (1991). Today, the process is widely used because of both the physical and the nutritional benefits it provides. The physical benefits include improved ease of handling, reduced ingredient segregation, less feed wastage and increased bulk density. Nutritional benefits have been measured through animal feeding trials Falk, (1985) by Behnke, (1994). As a rule, feeding pelleted feed improves animal performance and feed conversion compared with feeding a meal form of a diet. The improvements in performance have been attributed to (Behnke, 1994). Pellets are considered highly desirable for feeding poultry, calves, rabbits, dogs and minks. Their main advantage is that all their ingredients are bonded together so that the animal finds it very difficult to separate the components and discard those things it finds less palatable. Pelletizing insures that the animal receives the full spectrum of nutrients that it must ingest for maximum growth, health, and production Matz, (1991). Pressure extruders which subject the mix to heat treatment and intensive mixing are now used extensively in the feed industry. They are generally

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more expensive and more difficult to maintain than pellet mills, but are capable of operating under a much wider range of conditions. The high heat treatment which can be applied in pressure extruders can make the pellets more palatable and also inactivate some undesirable factors found in oil meals and other ingredients. Extruders can also deliver firmer pellets; provide many different sizes and shapes of pellets, and process mixtures which cannot be satisfactorily handled by the pellet mill. Extruders can perform a cooking function, which does significantly improve the nutritive value of many raw materials, as they are concerned. *Moscicki and van Zuilichem, (1984)* reported that the average value for energy consumption of a pelletizer making ruminant feed pellets is 8 to 13 kWh per ton whereas an average single-screw extruder requires about 90 to 180 kWh for well gelatinized feeds. For twin-screw extruders, the power requirement would be much less, however, on the order of 9 to 14 kWh per ton. It has been

observed that the nutritive value of most grains is enhanced, at least for some animals, when their starch component is gelatinized.

It has been discovered by researchers that the consumption of feed in pellet form helps to aid the total consumption of the ingredients by fishes without selectively discarding the less palatable ingredients thus improving the nutritional value consumed. It is apparent that local fish farmers in Nigeria are mostly using pelletized feed for their fishes because of the reduced cost. This pelletized feed is not properly processed during its production thereby reduces growth inhibition in fishes, causes water pollution and promote harmful organisms and defectives enzymes in fish feeds. Therefore, there is need for development of machinery to produce and meet local demand. The research is aimed in designing, fabricating and evaluating a fish feed extruder using locally sourced materials capable of producing high quality feed pellets.

2. MATERIALS AND METHOD

Material Selection and Production Planning

The table below is a summary of design criteria and the materials used in the fabrication of the machine components.

TABLE 1
MACHINE PARTS SPECIFICATIONS

Component Part	Dimension	Material Used	Quantity
Hopper	ø 300 x 435 mm	Mild steel plate	1
Barrel	ø 133 x 800 mm	Mild steel (aco pipe)	1
Screw conveyor	ø 50 x 1000 mm (inner shaft), ø 125 x 800 mm (sleeve pipe)	High carbon steel Mild steel	1
Die head plate	ø 135 x 10 mm	Mild steel plate	1
Frame	2mm x 2mm	Angle iron	5
Pulley	ø 250 mm	Mild steel	1
Fasteners (bolts and nuts)	ø 15 x 50mm (for tightening), ø 10 x 50mm (for barrel casing)	Mild steel bold and nuts	8 20
Finishing and coating	-	Chemical (paint)	-
Pillow bearing	ø 50 mm	-	2
Flat bar	ø 40 mm x 800mm	Mild steel	1
Mild steel welding electrode	Gauge 10 and 12	-	2 packets
Grinding disk	-	-	2
Electric motor	5 horse power	-	1
V- belt	12 mm thickness	Rubber	2

After procurement of the parts, each component was examined to locate points where alterations would be carried out most easily. This information was used to carry out production planning and assembly planning. The machine tools and the major tools used in the fabrication of the extruder

include, lathe machine, drilling machine, welding machine, sniper, hammer and block, hacksaw vise and electrodes.

Machine Design

The design was carried out using principles of engineering design with due consideration to cost, ease of operation, serviceability, durability and

- (1) Design of Machine Components:** This was divided into fundamentals and basics. The fundamentals refer to the principles governing each component while the basics deals with
- (2) Design of Drive System:** This includes the speed reduction unit specification (motor-screw reduction phase) and the belt/pulley design

3. DESIGN ANALYSIS

Step 1: Hopper Design

The hopper designed is a combination of two shapes: a cylinder with a conical hopper at its base to enable storage of raw materials.

$$\text{Volume of hopper} = [\text{Volume of cylinder} + \text{volume of cone with a tip (frustum)}] \quad (1)$$

Where

$$\text{Volume of cone with a top (frustum)} \Rightarrow \frac{\pi h}{3} [(R_1^2 + R_1 r_1 + r_1^2) - R_2^2 + R_2 r_2 + r_2^2] \quad (2)$$

Where $(R_1^2 + R_1 r_1 + r_1^2) - (R_2^2 + R_2 r_2 + r_2^2)$ is wall thickness of frustum

Let h be the height, R the radius of the upper base, and r the radius of the lower base of the frustum.

Generally, volume, V, for a hollow cylinder is given by:

$$V = \text{Area} \times \text{height} = \pi(r_1^2 - r_2^2)h \quad (3)$$

Thickness of the cylindrical wall, t:

$$t = r_1 - r_2$$

Hence

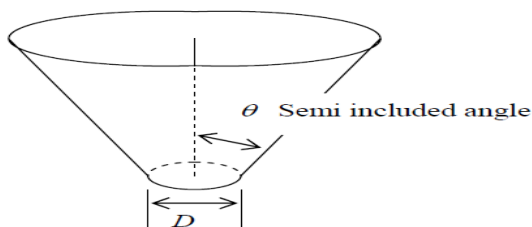


Fig 1: Conical hopper with outlet size D and semi Included angle θ

As commonly practiced, the coefficient of friction is expressed as the angle of wall friction given by φ as;

$$\phi = \text{arc tan}(\mu) \quad (7)$$

With known values of semi included angle θ, wall friction φ and the effective angle of internal

performance. The design of this machine was separated into two distinct phases:

thespecific calculations affecting the machine being designed. The components include the hopper, barrel, screw conveyer, die and frames.

$$\text{Weight of hopper} = \text{Volume of hopper} \times \text{Density of steel} \quad (4)$$

$$\text{Volume occupied by steel} = \text{Volume of the hopper} \times \text{Density of material} = 7850 \text{ kg/m}^3$$

$$\text{Mass of hopper} = \text{Volume of hopper} \times \text{Density of material to be considered}$$

$$\text{Weight of hopper} = \text{Mass of hopper} \times \text{Acceleration due to gravity}$$

Step 2: Determination of Conical Hopper Outlet Diameter

The opening of the diameter for conical hoppers is given by (Jenike, 1964)

$$D = H(\theta) \frac{CAS}{\rho g / g_c} \quad (5)$$

$$H(\theta) = 2 + \frac{\theta}{60} \quad (6)$$

Where g_c is gravity constant conversion factors to convert the result from units of mass to units of force $\frac{1 \text{ kgm}}{\text{Ns}^2}$. Typical value for H is about 2.4 (Jenike, 1964). Therefore θ semi included angle of the materials from equation (5) is 24°.

friction, the flow factor (ff) can be determined from the Jenike published charts for conical hoppers.

$$\text{For flow: } \frac{\sigma_c}{ff} > \sigma_y \quad (8)$$

Assuming σ_y and σ_c are related by the material flow function

$$\sigma_y = \sigma_c^{1.34}$$

Thus, the criterion for flow becomes

$$\left[\frac{\sigma_y^{1/1.34}}{ff}\right] > \sigma_y$$

And so the critical value of unconfined yield stress σ_{crit} is found when

$$\left[\frac{\sigma_y^{1/1.34}}{ff}\right] = \sigma_y \quad (9)$$

$ff = 1.54$ from the Jenike published chart for conical hoppers. The angle θ was reduced by 3° as a margin of safety (1964). Therefore θ used was 21° throughout. Hence solving for σ_y from equation (1) gives σ_{crit}

By applying law of indices we have,

$$\sigma_y^{(-0.34/1.34)} = 1.54$$

Therefore σ_y equals to σ_{crit} is 0.1823 kN/m^2

$$D = \frac{2.4 \times 0.1823 \times 1000}{900 \times 9.81}$$

Minimum diameter of circular outlet D is 0.0495

Step 3: Determination of Mass Flow Rate of Conical Hopper

The mass flow rate or discharge rate of a conical hopper was determined by the equation (Jenike, 1964):

$$\dot{m} = \rho^0 A \sqrt{\frac{Dg}{2(m+1)\tan\theta}} \quad (10)$$

$$\text{Where } A = \frac{\pi D^2}{4} \quad (11)$$

For conical hopper $m = 1$

The stress action on the hopper wall is given by:

$$P_v = \frac{\rho^0 g B}{4\mu K g c} \left(1 - \exp\left(-\frac{4z\mu K}{D}\right)\right) \quad (12)$$

Where B is the hopper maximum diameter and z is the cylinder height

To estimate the normal stress on the hopper wall Jassen's assumption was applied

$$P_w = K P_v \quad (13)$$

The material that can withstand the maximum stress acting on the wall of the hoppers should be selected.

Step 4: Capacity Design for a Given Electric Motor Rating

The design for motor output power enables appropriate selection of a motor with enough power to start and run the machine at full load.

$$\text{Power} = F \times V \quad (14)$$

Where

P = Power in watts

F = Rotational force acting on the shaft in Newton (N)

V = Linear velocity of the shaft in meters/seconds

$$\text{But } F = ma \quad (15)$$

M = Mass of rotating shaft in kilogram (kg)
a = Angular acceleration of the motor in radian/seconds square (rad/s^2)

$$a = w^2 r \quad (16)$$

w = angular velocity of the motor in radian/seconds

$$w = v/r$$

$$\text{Therefore } v = wr \quad (17)$$

By putting equation (16) into (15)

$$F = mw^2 r \quad (18)$$

By putting (18) and (17) into (14)

$$P = mw^2 r \times wr \quad (19)$$

$$\text{But } w = \frac{2\pi N}{60} \quad (20)$$

Putting (20) into (19)

$$P = 8M \left(\frac{\pi N}{60}\right)^3 r^2 \quad (21)$$

Step 5: Determination of Screw Conveyor Diameter and Power to Drive Conveyor

The diameter and power of the screw conveyor required for conveying material at a rate of 500 kg/hr for the capacity of continuous screw conveyors was calculated from the expression given by (Spirakorshy and Dyachkov, 1967).

$$D^2 = \frac{4Q}{60\pi S n \phi \rho c} \quad (22)$$

Where, Q = capacity of screw conveyor,

S = screw pitch

n = speed of convey = loading efficiency

p = free bulk density of the material,

c = loading factor depending on the inclined angle to the horizontal

The recommended values by Spirakorshy and Dyachkov, (1967) for slow flowing abrasive material are

$$S = 0.8D, \phi = 0.125 \text{ and } c = 1 \text{ for inclination angle } b_0 = 0.$$

The recommended minimum a maximum speed of conveyor is between $24 - 150 \text{ rpm}$.

$$D^2 = \frac{2000}{60 \times 3.142 \times 0.8D \times 120 \times 0.125 \times 900 \times 1}$$

$$D = \left(\frac{2000}{2036016}\right)^{0.3}$$

Therefore, the diameter of the screw conveyor is 0.125 m .

$$P_r = \frac{QL}{367} (\omega_0 + \sin \beta^\circ) \quad (23)$$

Where ω_0 is 4.0 for slow - flowing abrasive material, inclination angle of conveyor (β) is 0° and

L = length of conveyor

$$Pr = \frac{500 \times 0.8}{367} (4 + 0) = 0.00435 \text{ kW}$$

Step 6: Design of Drive System (Belt and Pulley Design)

The machine runs with a mere 600rpm motor which will produce a speed of 120rpm. The motor/shaft reduction was achieved with a pair of pulleys with diameters at ratio 1:5. This reduces the speed of the motor via a V - belt before it enters the shaft. The smaller pulley is adapted at the motor and connected to the bigger pulley on the shaft of the screw via a V - belt. The bigger pulley welded to the shaft of the screw which passes through two pillow bearing.

Step 7: Determination of Pulley Diameter

Speed of driving pulley versus speed of driven pulley can be expressed by (Khurmi, 2006) as

$$DeNe = DsNs \quad (24)$$

Where, De and Ne are the diameter and speed of driving pulley. Also, Ds and Ns are the diameter and speed of driven pulley. For a driving pulley of 50mm diameter, the driven pulley diameter was calculated from the above at 250mm (0.250m) and the input ratio is 1:5.

Step 8: Determination of Belt Length

With known pitch diameters of pulleys
Ds = 250mm, De = 50mm and center distance between motor/shaft pulley, C = 630mm.
The length of belt required is calculated using the (Fenner, 1994) by (Fadare, 2004) equation below:

$$L = 2C + \left(\frac{Ds - De}{4C}\right) + 1.57(Ds + De) \quad (25)$$

$$L = 2(0.630) + \left(\frac{0.250 - 0.050}{4(0.630)}\right) + 1.57(0.250 + 0.050)$$

Therefore the length of the belt require = 1,812.5mm.

From the catalogue the nearest B section belt to 1812.5mm length is 1819mm. Hence, B 1810 belt was selected.

Step 9: Center To Center Distance Design

The minimum distance between driving and driven pulleys for appropriate belt tension is obtained below

$$C = \frac{1}{2}(D_1 + D_2) + D_2 \quad (26)$$

C = the minimum distance between the pulleys in meters

D₁ = Pitch diameter of the driving pulley (motor pulley) in meters.

D₂ = Pitch diameter of the driven pulley

Step 10: Determination of Angle of Wrap

The angles of wrap on the small and large pulleys for open belt are determined using (Hall et al, 1982) equation by (Fadare, 2004).

$$\alpha_1 = 1800 - 2\sin^{-1} \left(\frac{R - r}{C}\right) \quad (27)$$

$$\alpha_2 = 1800 + 2\sin^{-1} \left(\frac{R - r}{C}\right) \quad (28)$$

Where, α_1 and α_2 angle of wrap on the small and large pulleys respectively, R and r are radius of large and small pulley.

For small pulley

$$\alpha_1 = 1800 - 2\sin^{-1} \left(\frac{0.125 - 0.050}{0.630}\right) = 161.7^\circ$$

For large pulley

$$\alpha_2 = 1800 + 2\sin^{-1} \left(\frac{0.125 - 0.050}{0.630}\right) = 198.26^\circ$$

Step 11: Determination of Tension in the Belt

Tension T₁ acting on the tight side of the belt and the tension T₂ acting on the slack side of the belt are calculated using the equation at Hall et al (1982) by Fadare, (2004).

$$\frac{T_1 - mV^2}{T_2 - mV^2} = e^{\frac{f\alpha}{\sin 0.5\theta}} \quad (29)$$

Where m = mass of a unit length of the belt

v = Linear velocity of belt;

α = Angle of wrap on pulley (rad)

f = Coefficient of friction between belt & pulley.

T₁ = Tension in the tight side of belt (N)

T₂ = Tension in the slack side of (N)

θ = Groove angle for v - belt (degree).

The maximum tension in the tight side of the belt depends on the allowable stress of the material.

For a B section belt the following parameters are given:

T₁ = allowable tension = 900N (Hall et al, 1982),

$\theta = 38 \pm 0.50$ (Fenner, 1994), f = 0.2 (Tangka, 1995)

The linear velocity V is given by

$$V = \frac{\pi \times Ds \times Ns}{60} \quad (30)$$

$$V = \frac{3.142 \times 0.250 \times 120}{60} = 1.571 \text{ m/s}$$

The right side of the equation [29] was determined for both small and large pulleys as follows:

For small pulley

$$\alpha = \alpha^1 = 161.7^\circ (2.82 \text{ rad}) \text{ using } \theta$$

$$= 38.5^\circ (0.672 \text{ rad})$$

$$e \frac{f\alpha}{\sin 0.5\theta} = 5.54$$

For large pulley,

$$\alpha = \alpha_2 = 198.26^\circ (3.46 \text{ rad})$$

$$e \frac{f\alpha}{\sin 0.5\theta} = 8.61$$

The small pulley with the highest value of 5.54 will be used as a basis for the design. Substituting the value of $T_1 = 900\text{N}$, $m = 0.19$ and V into the equation. The tension T_2 on the slack side can be calculated as:

$$\frac{900 - 0.19(1.571)^2}{T_2 - 0.19(1.571)^2} = 5.54$$

Therefore, T_2 equals 162.84N.

Step 12: Determination of Power Capacity per Belt

Power capacity per belt is determined using the equation (Hall et al, 1982) by (Fadare 2004).

$$P = (T_1 - T_2) V \quad (31)$$

$$P = (900 - 162.84)1.571$$

Hence the power capacity per belt equals 1.158Q (1.158kW)

Fish Feed Extruder Construction

Barrel Design and Construction: The barrel was constructed by machining a purchased round hollow mild steel pipe into appropriate length of 800mm. it has an outer diameter of 133mm and inner diameter of 130mm with a thickness of about 3mm. After the barrel was halved, four flat bars were welded to the edges. These flat bars were drilled into 10 holes each. The upper and the lower part were held together with the aid of 20 pieces of M12 bolts and nuts. This was done for the ease of maintenance and assembly of the inner components (i.e. screw and bearing replacement and adjustment). Furthermore, a circular opening of 30mm diameter was created a few meters away from the die plate. This serves as a vent to reduce overheating of resin pellets during the extrusion process. A U-shape element of 1000W was fixed at the bottom of the barrel surface to ensure thermo-mechanical cooking of the resin pellets. The element is mounted on the barrel by two insulators (clay and mild steel plate of 6mm thickness). This mild steel insulator was machined to size by an oxy-acetylene gas because of its high thickness. The stopped the heat from escaping to the surroundings other than the barrel alone.

Screw and Shaft Construction:It is the combination of the screw and the shaft. A 130mm screw conveyor was gotten which helped to eliminate the necessity of manufacturing the screw conveyor locally which would have been a complex task. It was machined by turning operation with a lathe into appropriate diameter. It was reduced to 125mm making a clearance of 5mm between screw and barrel. The shaft was step turned by 20mm at its two ends. The shaft protrudes out of the casing along the drive section of the machine and entered the bearing housing before it finally enters the larger pulley. The shaft also extends out of the die plate center and the second bearing is housed inside the barrel. A bearing of suitable dimension was also purchased for use in the extruder. The shaft is about 1000mm in length and 50mm in diameter.

Die Design and Construction:The extruder needed a die which could sustain the high pressure of material conveyed by the feed screw. The die was made using two layers of 3mm thickness mild steel plates which was welded together into appropriate dimension. It has a diameter of 136mm and is removable. Finally sixty (60) numbers of holes were drilled around the surface of the die plate to make way for the resin pellets to pass through. The drilled holes are 3-5 mm in diameter.

Hopper Design and Construction:This was designed and constructed locally. The construction was separated into two sections; the cylindrical and conical sections. The two shapes were achieved with the aid of a roller after cutting to the required dimension. The hopper outlet is 50mm and maximum diameter of 300 mm. Hopper height is 425mm and a thickness of 2mm.

Extruder assembling: After the construction of each part of the extruder had being completed, the various parts of the extruder were coupled as follows: firstly the upper and the lower barrel were held together by the 20 bolts and nuts. After the screw conveyor was placed into the barrel from the front side. The inner bearing was attached to the shaft and the back casing which has a drilled hole at the center was welded to barrel. Immediately the second bearing was also attached to the back of the casing along the drive system. The welding gauge was used to create a hole of about 50 mm on the barrel surface. The hopper was welded on this hole which was at a small distance from where the screw starts. This whole work was now welded on

two supports at the two ends of the barrel before finally welded on the constructed frame. The material used for the frame was angle iron. The larger pulley was fixed and the electric motor to which the smaller pulley was adapted was sitting firmly on the frame. A belt was fixed to the grooves of the pulleys giving a V shape belt. A guide which had been constructed with a mild steel plate initially was mounted to the frame with bolts and nuts. At the end, a ring was that was also halved was welded to the edge of the barrel. This ring was drilled and threaded with six holes on the face. The die plate was also drilled and threaded at the circumference which made it possible to screw the die with the ring on the edge of the barrel.

Working Principle of the Extruder and Test

In this system, the screw conveyor is used to convey raw materials to the melt zone at a steady rate. A three phase 5hp electric motor is used rotate the shaft. During the journey, resin pellets encounter friction from feed screw surfaces and barrel surface and each other. Prior to the production, the heater which is attached to the outside surface of the barrel will be switched on to get hot to a maximum temperature so as to heat the thick wall of the barrel properly before

extrusion begins. The heat sink is air. This heat facilitates more melting of resins. A test was carried out on the coupled extruder using fish feed purchased. Before the extrusion process mixing and milling were carried out on the feed from a grinding machine to ensure homogeneity of the fish pellets. The feed ingredients were propelled along the barrel of the extruder by the screw conveyor which rotates in the anticlockwise direction at a speed of 120rpm. The mechanical screw action and the friction it created sheared, blended and cooked the resins. During the extrusion process, the heater helped in cooking the starch in the resins completely and also served as a drier to remove excess moisture from the pellets. Because of the pressure found inside and outside the barrel, the resins expanded and according to the die shape (with several drilled holes) it created pellets in the range of 3- 5mm sizes. A multi meter scale was used to detect the temperature of heat loss to the surrounding during this test. Heat loss was due to conduction and convection. Heat was lost through the extruder from the barrel surface, hopper outlet and vent. The total heat loss was calculated as 110kW. Heat loss due to conduction was negligible in the calculation.

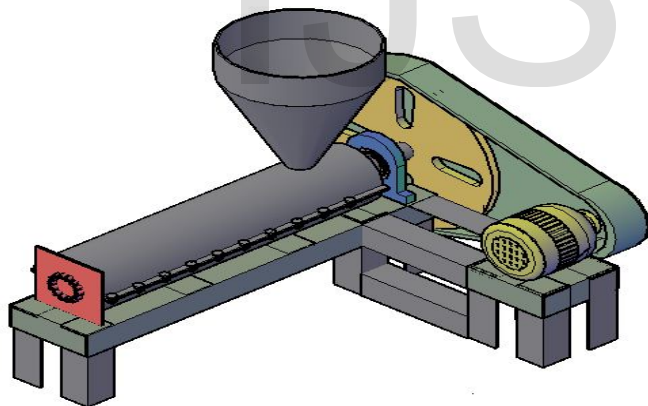


Fig 2: Schematic view of the Extruder

4 RESULTS AND DISCUSSION

The outcome of the construction is as follows: The purchased barrel and screw conveyor are shown in Fig 3. The die head plate is shown in Fig. 4 before drilling of holes on the surface. The larger pulley adapted to the shaft is shown in Fig. 5. The temporary arrangement of the extruder

components is shown in Fig. 6. Productions of pellets from the extruder are shown in Fig. 7- 8. The sideview and back view of the finished extruder are shown in Fig. 9 - 10. Finally, Fig. 11 demonstrated the use of multi scale meter in detecting the degree of hotness of the heater that was attached to the base of the barrel.





Fig 3: Barrel and screw conveyor before machining
Fig 4: Die head plate before machining operation
Fig 5 :Larger Pulley adapted to the shaft
Fig 6:Temporary assembling of components
Fig 7: Production of pellets with the extruder
Fig 8: Pellets produced of varying sizes
Fig 9: Finished extruder (side view)
Fig 10: Finished extruder (back view)
Fig 11: Multi meter used to determine heat loss to the surrounding

5. CONCLUSION

The selection, design, construction and testing of the fish feed extruder for a fish farm were undertaken. The extruder was tested and it worked smoothly by producing pellets in the range 3- 5mm in diameter. The completed extruder contained the major features to be found in most extruders. The machine can be classified as general purpose extruders since it performs all the functions possible on the typical commercial extruders.

6. ACKNOWLEDGEMENTS

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