Journal of Multidisciplinary Engineering Science and Technology (JMEST)
ISSN: 2458-9403
Vol. 6 Issue 7, July - 2019

The Design And Construction Of A Single Screw Extruder

[1*]Ugboya, A. Paul, [2]Odiamenhi, A. Martins, [3]Aigbojie .O. Eddy
Dept. of Industrial and production Engineering, Ambrose Alli University Ekpoma, Edo State, Nigeria
Corresponding author's e-mail: upaigbe2002@yahoo.com

***Abstract*—The use of polymers in laboratories and industries is imperative resulting to the constant need for homogenize melting and mixing of materials. The efficiency, complexity and cost of already existing single screw extruder machine have discouraged individuals, institutions and industries from mass usage of the machine and this necessitated the development of this single screw extruder machine in view of evaluating and improving on the efficiency of existing machines. A single screw extruder was designed and constructed for melting and mixing of various types of polymer. The machine which is easy to use has a heating element incorporated into its screw and a thermostat to control and determine the amount of temperature in the barrel, so as to overcome the shortcomings observed on already existing single screw extruder machine. Results from the test conducted showed that the extruder is capable of handling 1000cm^3 of polymer of 353wt% in 16 minutes. Also, results showed that the machine provides a faster, easier and more profitable means of obtaining molten polymer. Results further showed that the efficiency of the machine using a molten polymer of 285wt% was 81%. In conclusion, the machine is very applicable for local production, operation, repair and maintenance. The operation of this machine manually makes it a unique type compare to others. The operation save energy and does not require high skilled labour.**

Keywords—Polymers, Melting time, Single screw extruder, Heating element, Thermostat, Temperature.

1. INTRODUCTION

For many years engineering practice has been made possible only by the use of metals. However, in recent years, the development and increasing use of plastic materials has necessitated the introduction of several techniques used to shape polymers (Ogorkiewicz, 1977). Plastic, as defined by the society of platic industry is "Any one of a large and varied group of materials wholly or in part of combinations of carbon with oxygen, hydrogen, nitrogen and other organic and inorganic elements which while solid in the finished state, at some stage in its manufacture is made liquid and thus capable of being formed into various shapes, most usually through the application, either singly or together, of heat and pressure" (Millet, 1997). Single screw extruder machine comes in different shapes and sizes, so defining the single screw extruder machine and matching it to the individual's physical characteristics and mixing goals are important. Therefore there is need to improve and develop on the existing mechanical and electrical components like the steel frame, screws, barrel, speed regulator, hopper, electric motor, stand etc. that need to work together to provide a realistic experience. However there are some differences in single screw extruder machine designed for mixing, heating and extruding. Single screw extruder machines have a small hopper and light frame construction, powered by an electric motor as the level of impact and stress on the mixing is considerably less than the extruding machine. It is imperative to give a deep consideration to the following factors such as duration and frequency of use per week, pre-existing effect, material weight and long term mixing goal as they influence the suitability of the type of laboratory plastic single screw extruder machine we should consider. The aim of this study is to design and develop a laboratory single screw extruder for compounding polymer and polymer composites. The specific objectives include:

a. To design a single screw extruder machine frame.
b. To evaluate the efficiency of the single screw extruder machine.
c. To construct the extruder using local resources.
d. To regulate the temperature of the extruder via a thermostat

Single screw extrusion is generally defined as a process to mix, homogenize and shape material by forcing it through a specifically designed opening (die). Single screw extruder combines a high-temperature short-time (HTST) cooking process with several other unit operations, such as conveying, kneading, heating, mixing and forming in a single unit. This versatile technology is finding increased application in the polymer and food industries as a means of transforming relatively low ingredients, into intermediate or semi-finished products.

Energy input into the material during extrusion is an important parameter because it relates to the physical and chemical transformation in the product. In a typical extrusion operation, the two main sources of energy associated with enthalpy change of the extrudate are (1) convection heat transfer between the hot cold barrel and the polymer or food material and

(2) viscous dissipation of the mechanical energy into heat inside the material. The rate of convected heat transfer is proportional to the amount of contact area between the barrel and the flowing material whereas the heat generated due to viscous dissipation is proportional to the volume of the material. In single screw extruder machine, viscous dissipation of the mechanical energy predominates, especially at low moisture contents, thus making the extrusion process highly energy efficient cost effective. This relatively high efficiency, coupled with other benefits such as flexibility and versatility, is the reason for the rapid growth of extrusion technology in the food and polymer industries (Godavarti and Karwe, 1997). The full-scale developments in the plastics industry and polymer processing technologies in Japan have started since 1950s, Japan Society of polymer Science (SPSJ), which mainly deals with polymer materials, was founded in 1952, but Japan Polymer Processing Society (JSPP) was established 48 years after this foundation. JSPP has three kinds of periodical meetings to be held every year (Annual Meeting, Regional Meeting and Asian Workshop). On the other hand, International Polymer Processing Society (PPS) was established as a global academic society at Akron (USA) in 1985, and till now, annual PPS meetings were held three times in Japan. Several polymer processing machines such as extruders have started to be manufactured in Japan in early 1950s. However, a very interesting thing is that the elementary components of these machines have not essentially change, although their size, control/monitoring equipment, output capacities drastically changed. For example, in case of single-screw extruders, the output of 65mm screw diameter, was only 30 kg/hr. in 1950's (at present, more than 500 kg/hr.) and the cable coating speed was 66m/min (now, more than 2000 m/min) (Seikei-Kakou, 2005). Recently, the capacity of processing machines was much upgraded in accordance with the expansion of the polymer industry in the world. For example a large scale single-screw extruder with diameter D = 700mm, a large scale intermeshing twin-screw extruder with screw and barrel length ratio L/D =100 and a large scale devolatilization twin-screw extrude with six vent holes. These machines were manufactured in Japan during 1990 – 2000 corresponding to higher demand for productivity and functionality in the polymer industry (Seikei-Kakou, 2005).

1.2 THE SINGLE SCREW EXTRUDER ANATOMY

In the extrusion of plastics, the raw compound material is commonly in the form of nurdles (small beads, often called resin) or pellet are gravity fed from a top mounted hopper into the barrel of the extruder. Additives such as colourants and UV inhibitors {in either liquid or pellet form) are often used and can be mixed into the resin or pellet prior to arriving at the hopper. The plastic enters through the feed throat (an opening near the rear of the barrel) and comes in contact with the screw. The single screw (normally turning at up to 563 rpm) conveys the material (plastic) forward into the barrel. In this process, a heating profile is set for the barrel in which the controlled heating element gradually increases the temperature of the barrel and screw from the rear (where the plastic enters) to the front. This allows the material to melt gradually as they are pushed through the barrel and lowers the risk of overheating which may cause degradation in material. At the front of the barrel, the molten plastic leaves the screw and travels through the die (small opening). The die is what gives the final product its profile and must be designed so that the molten plastic evenly flows from cylindrical profile to the product's profile shape. Uneven flow at this stage can produce a product with unwanted residual stresses at certain points in the profile which can cause warping upon cooling. Almost any shape imaginable can be created so long it is a continuous profile. The product may be cooled or allowed to cool before any further processing.

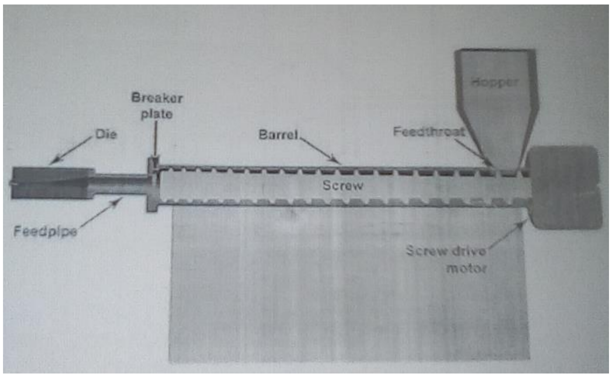

Figure 1: Single screw extruder machine

2. RESEARCH METHODOLOGY

The chapter entails the determination/calculations of the various parts/components of the machine adopted in this research.

2.1 DESIGN CALCULATION

i. THE CAPACITY OF THE SINGLE SCREW EXTRUDER

The single screw extruder machine was designed to have a capacity of extruding $1000cm^3$ of polymers in 16 minutes.

ii. ROTATIONAL SPEED OF SCREW

The single screw extruder machine was designed to rotate at a speed of 563 rpm for effective mixing and extruding.

iii. MINIMUM PRESSURE TO MIX POLYMER

A polymer variety requiring 2GPa pressure to mix the polymer was assumed.

2.1.1 DESIGN FOR VOLUME OF THE HOPPER (V_H)

The hopper has a base radius (r) of 19 mm, a top radius (R) of 72.5mm, and a height of 155 mm and the shape of a truncated cone.

Journal of Multidisciplinary Engineering Science and Technology (JMEST)
ISSN: 2458-9403
Vol. 6 Issue 7, July - 2019

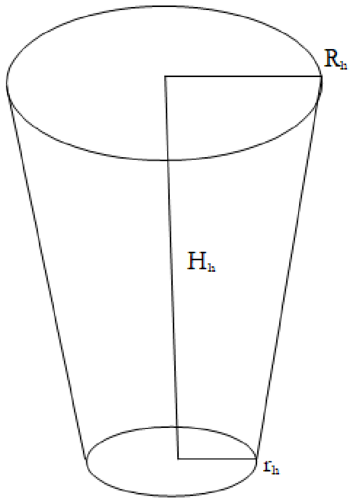

Figure 1: truncated cone

Using the truncated cone equation to calculate for the volume of the hopper,

$$\text{Volume of hopper } (V_h)$$

$$= \frac{1}{3} \text{ x } \pi \text{ x } H_h(R_h^2 + r_h^2 + R_h \text{ x } r_h)\ . \qquad (1)$$

Where,

H_h = Height of the hopper = 160mm = 15.5cm

R_h = Radius of the top of hopper = 72.5mm = 7.25cm

r_h = Radius of the base of hopper = 19mm = 1.9cm

V_h = Volume of the hopper

Therefore,

$$(V_h) = \frac{1}{3} \text{ x } \pi \text{ x } H_h(R_h^2 + r_h^2 + R_h \text{ x } r_h)$$

$$(V_h) = \frac{1}{3} \text{ x } 3.142 \text{ x } 15.5(7.25^2 + 1.9^2 + 7.25 \text{ x } 1.9)$$

$$(V_h) = \frac{48.701}{3}\ (52.5625 + 3.61 + 13.775)$$

$$(V_h) = \frac{48.701}{3}(69.9475)$$

$$(V_h) = 16.2337(69.9475)$$

$$(V_h) = 1135.506731$$

$$(V_h) = 1136\text{cm}^3$$

Therefore,

If 88% of the hopper is being utilized and 12% is allowed for the free space, so as to prevent deformation of barrel and too much load standing in the hopper, therefore, the final volume of the hopper (Vfh) will be,

V_{fh} =*Calculated volume x % being utilized*

$$\%\text{ being utilized} = 88\% = \frac{88}{100} = 0.88$$

V_{fh} = 1136 x 0.88

V_{fh} = 1000cm^3

Therefore, the hopper has a total volume of 1136cm^3 with a utilized volume of 1000cm^3 and a free volume of 136cm^3.

2.1.2 CALCULATING FOR THE VOLUME OF SPARE HOPPER (V_{SH})

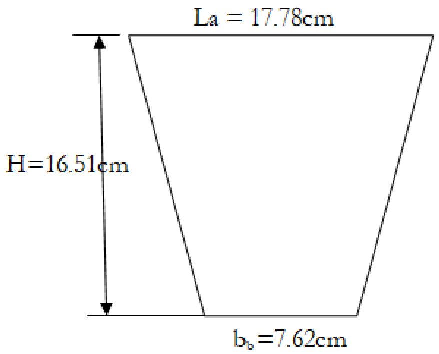

Fig.1a: front view of hopper

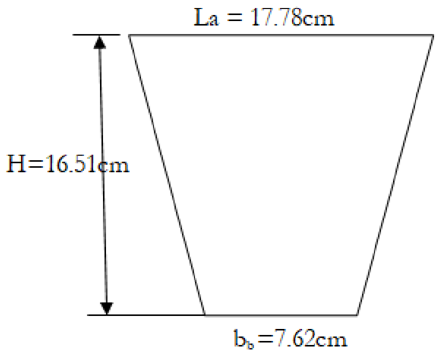

Fig.1b: Side view of hopper

The hopper has a square base of 7.62cm x 7.62cm and rectangular top of 17.78cm x 16.51cm and has the shape of the frustum of a pyramid (truncated).

Using the frustum of a pyramid (truncated equation) to calculate for the volume of the hopper, we have,

Volume of spare hopper (V_{sh})

$$= \frac{H}{3}\left(A_a + A_b + \sqrt{A_a A_b}\right)\ . \qquad (2)$$

Where,

H_{sh} = Height of the spare hopper = 16.51cm

A_a = Area of the top of spare hopper

= L_a x b_a = 17.78 x 16.51

= 293.55cm cm^2

A_b = Area of the base of spare hopper

= b_b x b_b = bb^2 = (7.62)^2

= 58.06cm^2

V_{sh}= Volume of spare hopper

Therefore,

$$(V_{sh}) = \frac{H}{3}\left(A_a + b_b^2 + \sqrt{A_a b_b^2}\right). \quad (3)$$

$$(V_{sh}) = \frac{H_{sh}}{3}\left(A_a + b_b^2 + b_b\sqrt{A_a}\right). \quad (4)$$

$$(V_{sh}) = \frac{16.51}{3}\left(293.55 + 58.06 + 7.62\sqrt{293.55}\right)$$

$$(V_{sh}) = 5.5(293.55 + 58.06 + (7.62 \text{ x } 17.13))$$

$$(V_{sh}) = 5.5(293.55 + 58.06 + 130.53)$$

$$(V_{sh}) = 5.5(482.14)$$

$$(V_{sh}) = 2651.77\text{cm}^3$$

If 76% of the hopper is being utilized and 24% is allowed for the free so as to prevent deformation of barrel due to load standing in the hopper before the volume of the hopper will be

V_{fsh} = *Calculated volume* x % *being utilized*

$$\% \text{ being utilized} = 76\% = \frac{76}{100} = 0.76$$

V_{fsh} = 2651.77 x 0.76

V_{fsh} = 2015.3452cm^3 = 2015cm^3

Therefore, the spare hopper has a total volume of 2651.77cm^3 with a free space of 636cm^3 and a utilized space of 2015cm^3.

2.1.3 DESIGN FOR VOLUME OF THE BARREL (V_B)

V_b = n x r^2 x L . (5)

Where,

r_{ib} = inner radius of the barrel

= 35.1mm = 3.51cm

L_b = length of the barrel

= 457mm = 45.7cm

V_b = 3.142 x (3.51)^2 X45.72

V_b = 3.142 x 12.3201x45.72

V_b = 1769.8cm^3

2.1.4 CALCULATING THE VOLUME OCCUPIED BY THE SCREW SHAFT (V_{SS})

Where,

Radius of screw shaft (r_{ss}) = 19mm ~ 1.9cm

Length of the screw shaft (L_{ss}) = 457mm = 45.7cm

V_{ss} = π x $_{rss}$^{2} x L_{ss} . (6)

V_{ss} - 3.142 x (1.9)^2 x 45.7

V_{ss} = 3.142 x 3.61 X 45.72

V_{ss} = 518.58cm$^{3.}$

2.1.5 CALCULATING FOR VOLUME OCCUPIED BY THE SCREW FLIGHT (V_{SF})

Where Number of screw ribs (n_{sf}) = 7 = 3.5 rev

Length of 1 screw rib (L_{sf}) =20mm = 2cm

Length of 3.5 screw rib (L_{sf}) = 2 x 3.5 = 7cm

Radius of the screw rib (r_sf) = 35 - 19 = 16mm = 1.6cm

V_{sr} = π x r_{sr} x L_{sr} . (7).

V_{sr} = 3.142 x (1.6)^2 x 7

V_{sr} - 3.142 x 2.56 x 7

V_{sr} = 56.3cm

2.1.6 CALCULATING FOR VOLUME OCCUPIED BY THE SCREW (V_S)

V_s= V_{ss} + V_{sr} . (8)

V_s = 518.58 + 56.3

V_s = *574.88cm^3*

2.1.7 CALCULATING FOR FINAL VOLUME OF THE BARREL (V_{FB}}

V_{fb} = V_b - V_s . (9)

V_{fb} = 1769.8 – 574.88

V_{fb} = 1194.92 = 1195cm^3

If 95% of the barrel is being utilized and 5% is allowed for the free space so as to prevent improper melting and mixing of polymer and also to prevent too much load in the barrel, therefore the new final volume of the barrel will be,

Vf_b = *Calculated volume* x % *being utilized*

$$\% \text{ being utilized} = 95\% = \frac{95}{100} = 0.95$$

V_{fb} = 1195 x O.95

V_{fb} = 1135.25cm^3 - 1135cm^3

2.1.8 POWER REQUIREMENT

Power can be expressed as

Power (P) = Torque resistance x Angular Speed

P = Tω . (10)

$$\text{But } \omega = \frac{2\pi N_1}{60}. \quad (11)$$

Where T = Torsional Stress

ω = Angular Speed

N_1 = Speed in revolution per minute (1425 r.p.m)

Substituting the value of N into equation (11)

We have,

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

$$\omega = \frac{2 \text{ x } 1.42 \text{ x } 1425}{60} = \frac{4047}{60} = 67.45\text{rad/sec}$$

The angular speed is 67.45 rad/sec

For the electric motor

1 h.p = 746 watts

3 h.p = 3 x 746 watts = 2238 watts

P = 2238 watts

Calculating the torque transmitted

From equation (10)

P = Tω

$$T = \frac{P}{\omega}$$

$$T = \frac{2238}{67.45} = 33.18\mathrm{N/M}$$

2.1.9 DESIGN OF SCREW

CALCULATION FOR SCREW GEOMETRY

$$\tan\varphi = \frac{\mathrm{Pitch}}{\pi D_s}$$

Where,

Helix angle (φ) =? (unknown)

Screw diameter (D_s) = 70mm

Screw pitch (p) = 48mm

$$\tan\varphi = \frac{48}{3.142 \times 70}$$

$$\tan\varphi = \frac{48}{219.94}$$

$$\tan\varphi = 0.2182$$

$$\varphi = \tan^{-1}(0.2182)$$

$$\varphi = 12.3 = 12^0$$

2.1.10 CALCULATION FOR COMPRESSION RATIO

$$CR = \frac{\text{Channel depth in feed section}}{\text{Channel depth in metering section}}$$

Where

Channel depth in feed section = 16mm

Channel depth in metering section = 16mm

$$CR = \frac{16}{16}$$

$$CR = 1:1$$

2.1.11 CALCULATION FOR LENGTH/DIAMETER (L/D) RATIO

$$L/D = \frac{\text{Screw flighted length}}{\text{Screw outside diameter}}$$

Where

Screw Flighted length = 419.2mm

Screw Outside diameter = 70mm

$$L/D = \frac{419.2}{70}$$

$$L/D = 6:1$$

2.1.12 DESIGN OF SCREW SHAFT (PARAMETERS)

W = 0.96KN

= 0.96×10^3 N,

L = 10.5mm,

x = 436.2mm, T = 33.18 *N/m* = *33180N/mm,*

d_{ss} = 38mm

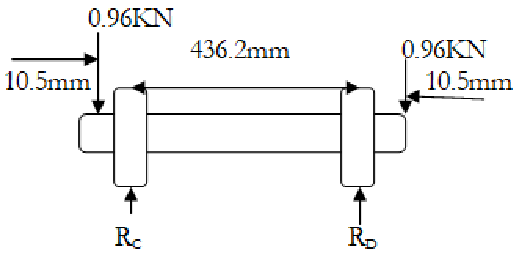

Figure 1: The shaft with bearings

A little consideration will show that the maximum bending moment acts on the shaft at C and D. Therefore maximum bending moment,

M = W x L . (12)

M = $0.96 \times 10^3 \times 2.34 \times 10^4$

M = 22464000

M = 22.46×10^6 *N -mm*

2.1.13. CALCULATING FOR SHAFT SUBJECTED TO COMBINED TWISTING MOMENT AND BENDING MOMENT:

Let, τ = Shear stress induced due to twisting moment, and

σ_b = Bending stress (tensile or compressive) induced due to bending moment.

According to maximum shear stress theory, the maximum shear stress in the shaft,

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\tau^2} \ . \quad (13)$$

Where

$$\tau = \frac{16T}{\pi d^3} \text{ and } \sigma_b = \frac{32M}{\pi d^3}$$

Substituting values of τ and σ_b into equation (13)

$$\tau_{max} = \frac{1}{2}\sqrt{\left(\frac{32M}{\pi d^3}\right)^2 + 4\left(\frac{16T}{\pi d^3}\right)^2} = \frac{16}{\pi d^3}\sqrt{M^2 + T^2}$$

$$\frac{\pi}{16} \times \tau_{max} \times d^3\sqrt{M^2 + T^2}$$

Therefore

$$\tau_{max} = \frac{16}{3.142 \times 38^3}\sqrt{(22.46 \times 10^6)^2 + (33180)^2}$$

$$\tau_{max} = \frac{16}{3.142 \times 54872}\sqrt{5 \times 10^{14} + 11 \times 10^8}$$

$$\tau_{max} = \frac{16}{172 \times 10^3}\sqrt{5 \times 10^{14}}$$

$\tau_{max} = 9.3 \text{ x } 10^{-3} \text{ x } 22.36 \text{ x } 10^6$

τ_{max} = 2079.48 N/mm

The expression $\sqrt{M2 + T2}$ is known as equivalent twisting moment and is denoted by T_e. The equivalent twisting moment may be defined as that twisting moment, which when acting alone, produces the same shear stress (τ) as the actual twisting moment. By limiting the maximum shear stress {T_{max}} equal to the allowable shear stress (τ) for the material, the equation (23) may be written as

$$T_e = \sqrt{M^2 + T^2} = \frac{\pi}{16} \text{ x } \tau \text{ x } D_{ss}{}^3 \text{ .} \quad (14)$$

Therefore,

$$T_e = \sqrt{M^2 + T^2}\sqrt{(22.46)^2 + (33180)^2}$$

$$T_e = \sqrt{5 \text{ x } 10^{14} + 11 \text{ x } 10^8} = \sqrt{5 \text{ x } 10^{14}}$$

T_e = 22.36 x 10^6 N/mm^2

Now according to maximum normal stress theory, the maximum normal stress in the shaft,

$$\sigma_{b(max)} = \frac{1}{2}\sigma_b + \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\tau^2} \text{ .} \quad (15)$$

$$\sigma_{b(max)}$$

$$= \frac{1}{2} \text{ x } \frac{32M}{\pi D_{ss}{}^3} + \frac{1}{2}\sqrt{\left(\frac{32M}{\pi D_{ss}{}^3}\right)^2 + 4\left(\frac{16T}{\pi D_{ss}{}^3}\right)^2}$$

$$\sigma_{b(max)} = \frac{32}{\pi D_{ss}{}^3}\left[\frac{1}{2}\left(M + \sqrt{M^2 + T^2}\right)\right]$$

$$\frac{\pi}{32} \text{ x } \sigma_{b(max)} \text{ x } D_{ss}{}^3 = \frac{1}{2}\left[M + \sqrt{M^2 + T^2}\right] \text{ .} \quad (16)$$

The expression $\frac{1}{2}\left[M + \sqrt{(M^2 + T^2)}\right]$ is known as equivalent bending moment and is denoted by M_e. The equivalent bending moment may be defined as that moment which when acting alone produces the same tensile or compressive stress (a_b) as the actual bending moment. By limiting the maximum normal stress [ab(max)] equal to the allowable bending stress (a_b); then the equation (iv) may be written as

$$M_e = \frac{1}{2}\left[M + \sqrt{M^2 + T^2}\right] = \frac{\pi}{32} \text{ x } \sigma_{b(max)} \text{ x } D_{ss}{}^3 \text{ .} \quad (17)$$

Therefore

$$M_e = \frac{1}{2}\left[22.46 \text{ x } 10^6 + \sqrt{(22.46 + 10^6)^2 + (33180)^2}\right]$$

$$M_e = \frac{1}{2}\left[22.46 \text{ x } 10^6 + \sqrt{5 \text{ x } 10^{14} + 11 \text{ x } 10^8}\right]$$

$$M_e = \frac{1}{2}\left[22.46 \text{ x } 10^6 + \sqrt{5 \text{ x } 10^{14}}\right]$$

$$M_e = \frac{1}{2}\left[22.46 \text{ x } 10^6 + 22.36 \text{ x } 10^6\right]$$

M_e = 22.41 x 10^6N/mm

2.1.14. DESIGN OF SPROCKET AND CHAIN

(Calculation for relation between Pitch and Pitch Circle Diameter)

$$P = D_{pe}\text{Sin}\left(\frac{360^0}{2T}\right) = D_{pe}\text{Sin}\left(\frac{180^0}{T}\right) \text{ .} \quad (18)$$

Or

$$D_{pc} = \text{pcosec}\left(\frac{180^0}{T}\right) \text{ .} \quad (19)$$

Where, P = 12.70mm = 0.0127m

d1 = 8.5mm = 0.0085m

Calculating for the pitch circle diameter of the smaller sprocket (D1)

Where T_1 = 15

Therefore,

$$P = D_1\text{Sin}\left(\frac{180^0}{T_1}\right)$$

$$0.0127 = D_1\text{Sin}\left(\frac{180^0}{15}\right) = D_1 \sin(12) = 0.2079D_1$$

$$D_1 = \frac{0.0127}{0.2079} = 0.061\text{m} = 61\text{mm}$$

The sprocket outside diameter (Do) for the smaller sprocket, for satisfactory operation is given by

$D_0 = D_1 + 0.8d_1$. (20)

D_0 = 0.061 + 0.8 (0.0085) = 0.061 + 0.0068

D_0 = 0.0678m = 67.8mm = 68mm

Calculating for the pitch circle diameter of the bigger sprocket (D_2)

Where T_2 = 38

Therefore,

$$P = D_2\text{Sin}\left(\frac{180^0}{T_2}\right)$$

$$0.0127 = D_2\text{Sin}\left(\frac{180^0}{30}\right) = D_2\text{Sin}(4.7368) = 0.0826D_2$$

$$D_2 = \frac{0.0127}{0.0826} = 0.154\text{m} = 154\text{mm}$$

The sprocket outside diameter (D_0) for bigger sprocket for satisfactory operation is given by

$D_0 = D_2 + 0.8d_1$

D_0 = 0.154 + 0.8 (0.0085) = 0.154 + 0.0068

D_0 = 0.1608m = 160.8mm = 161mm

Calculation for Velocity Ratio of Chain Drives

$$V.R = \frac{T_2}{T_1}$$

Where, T_1 = 15, T_2 =38

$$V.R = \frac{38}{15}$$

V.R = 2.5m/s

Calculating for the speed of rotation of larger sprocket

$N_1 = 1425$

$N_2 = ?$

$$\frac{N_1}{N_2} = \frac{T_2}{T_1}$$

$$N_1 x T_1 = N_2 T_2$$

$$N_1 x T_1 = N_2 T_2$$

$$N_2 = \frac{N_1 \text{ x } T_1}{T_2} = \frac{1425 \text{ x } 15}{38} = \frac{21375}{38} = 563 \text{ t.p.m}$$

2.1.15. CALCULATING FOR THE AVERAGE VELOCITY (V)

$$V = \frac{\pi DN}{60} = \frac{\tau \emptyset N}{60}$$

Therefore

Average velocity of smaller sprocket (V_1)

$$\frac{\pi D_1 N_1}{60} = \frac{3.142 \text{ x } 0.061 \text{ x } 1425}{60}$$

$$(V_1) = \frac{273.1184}{60} = 4.6 m/s$$

Average velocity of bigger sprocket (V_2)

$$\frac{\pi D_2 N_2}{60} = \frac{3.142 \text{ x } 0.154 \text{ x } 563}{60}$$

$$(V_1) = \frac{272.4177}{60} = 4.5 m/s$$

2.1.16. CALCULATION FOR DESIGN POWER AND LOAD ON CHAIN

Where Rated power = electric motor power = 2238w = 2.238kw

Pitch line velocity = average velocity of smaller sprocket = 4.6 *m/s*

Breaking load in Newton (W_B) = 31kN = 31000N

Service factor (K_s) = 1x1. 5x1 = 1. 5

2.1.17. CALCULATING FOR THE DESIGN POWER

Design power = Rated power x Service factor (K_s)

K_s = 2.238 x 1. 5 = 3.357kW

2.1.18. CALCULATING FOR THE LOAD ON THE CHAIN (W)

$$W = \frac{\text{Rated power}}{\text{Pitch line velocity}}$$

$$W = \frac{2.238}{4.6} = 0.487 kN = 487 N$$

2.1.19. CALCULATION FOR LENGTH OF CHAIN AND CENTRE DISTANCE

Where T_1 = Number of teeth on the smaller sprocket = 15

T_2 = Number of teeth on the larger sprocket = 38

ρ = Pitch of the chain = 12.70mm = 0.0127m,

x = Centre distance = 15.24mm = 0.01524m

K_d = number of chain links = 26

$L_c = K_{d.}\rho$

L_c = 26 x 0.0127 = 0.3302m = 330.2mm

L_c = 330mm

The number of chain links may be obtained from the following expression, i.e.

$$K_d = \frac{\tau_1 + \tau_2}{2} + \frac{2X}{P} + \left[\frac{\tau_2 - \tau_1}{2\pi}\right]^2 \text{ x } \frac{P}{x}$$

The value of K_d as obtained from the above expression must be approximated to the nearest even number.

The centre distance is given by

$$X = \frac{p}{4}\left[K_{cl} - \frac{T_1 + T_2}{2} + \sqrt{\left(K_{cl} - \frac{T_1 + T_2}{2}\right)^2 - 8\left(\frac{T_2 + T_1}{2}\right)^2}\right]$$

In order to accommodate initial sag in the chain, the value of the centre distance obtained from the above equation should be decreased by 2 to 5 mm.

2.1.20. DESIGN CALCULATION FOR BARREL

Calculating for the circumferential or hoop stress

Where p_b = Intensity of internal pressure = 2GPa.

Internal diameter of the cylindrical shell {d_{cs}) =70.2mm

Length of the cylindrical shell (L_{cs}) =457mm

Thickness of the cylindrical shell (t_{cs}) = 3mm

σ_{t1} = Circumferential or hoop stress for the material of the cylindrical shell,

total force acting on a longitudinal section of the shell

= Intensity of pressure x projected area

$= p_b {}^x d_{cs} \text{ x } L_{cs}$. (21)

the total resisting force acting on the cylinder walls

$= \sigma_{t1} \text{ x } 2t_{cs} \text{ x } L_{cs}$. (therefore of two sections). (22)

From equations (21) and (22), we have

$$\sigma_{t1} \text{ x } 2t_{cs} \text{ x } L_{cs} = p_b \text{ x } d_{cs} \text{ x } L_{cs} \text{ or } \sigma_{t1} = \frac{p_b \text{ x } d_{cs}}{2t_{cs}} \text{ or } t_{cs} \frac{p_b \text{ x } d_{cs}}{2\sigma_{cs}}$$

Therefore,

$$\sigma_{t1} = \frac{p_b \text{ x } d_{cs}}{2t_{cs}} = \frac{2 \text{ x } 10^9 \text{ x } 70.2}{2 \text{ x } 3} = \frac{1.404 \text{ x } 10^{11}}{6}$$
$$= 2.34 \text{ x } 10^4 \text{MPa}$$

2.1.21. CALCULATING FOR LONGITUDINAL STRESS

Where σ_{t2} = Longitudinal stress.

In this case, the total force acting on the transverse section

= Intensity of pressure x Cross-sectional area = $= p \text{ x } \frac{n}{4} (d_{cs})^2$. (23)

And the total resisting force = $\sigma_{t2} \text{ x } \prod \text{x } d_{cs} \text{ x } t_{cs}$. (24)

From equations (23) and (24), we have,

$\sigma_{t2} \text{ x } \prod \text{ x } d_{cs} \text{ x } t_{cs} = p_b \text{ x } \frac{n}{4} (d_{cs})^2 \quad \sigma_{t2} = \frac{p_b \text{ x } d_{cs}}{4t_{cs}} \text{ or } t_{cs} \frac{p_b \text{ x } d_{cs}}{4\sigma_{cs}}$. (25)

Therefore

$$\sigma_{t2} = \frac{p_b \text{ x } d_{cs}}{4t_{cs}} = \frac{2 \text{ x } 10^9 \text{ x } 70.2}{4 \text{ x } 3} = \frac{1.404 \text{ x } 10^{11}}{12}$$
$$= 1.17 \text{ x } 10^4 \text{MPa}$$

From above we see that the longitudinal stress is half of the circumferential or hoop stress. Therefore, the design of a pressure vessel must be based on the maximum stress i.e. hoop stress.

Calculating for the maximum shear stress

We know that according to maximum shear stress theory, the maximum shear stress is one-half the algebraic difference of the maximum and minimum principal stress. Since the maximum principal stress is the hoop stress a_{tl} and minimum principal stress is the longitudinal stress a_n, therefore maximum shear stress, 2

$\tau_{max} = \frac{d_{t2} - d_{t1}}{2}$. (26)

$$\tau_{max} = \frac{2.34 \text{ x } 10^4 - 1.17 \text{ x } 10^4}{2} = \frac{11700}{2}$$
$$= 5.85 \, x \, 10^3 \, GPa$$

2.1.22. FEED ZONE

The geometry of the feed zone of the screw is given by the following data:

Barrel diameter (D_b} = 76.2 mm = 0.0762m

Screw lead (5) = 48mm = 0.048m

Number of flights (V) =1

Flight width (W_{FLT}) - 20mm = 0.02m

Channel width (W_c) = 44.6mm = 0.0446m

Depth of feed zone (H_{fz}) = 16.2mm = 0.0162m

Conveying efficiency ($^n{}_f$) = 0.35

Screw speed N_2 = 563 r. p. m.

Bulk density of the polymer (P_0} = 950 Kg/m^3

Helix angle (φ) =12°

The solids conveying rate in the feed zone of the extruder can be calculated as,

$G = 60 \text{ x } P_o \text{ x } N_2 \text{ x } n_f \text{ x } \pi^2 \text{ x } H_{fs} \text{ x } D_b (D_b - H_{fs}) \frac{w_e}{w_e + w_{flt}} \text{ x } \sin\varphi \text{ x } \cos\varphi$. (26)

Therefore,

$$G = 60 \text{ x } 950 \text{ x } 563 \text{ x } 0.35 \text{ x } 3.142 \text{ x } 0.0162 \text{ x } 0.0762(0.0762 - 0.0162) \text{ x } \frac{0.0446}{0.0446 + 0.02} \text{ x } \sin(12) \text{x } \cos(12)$$

$$G = 60 \text{ x } 950 \text{ x } 563 \text{ x } 0.35 \text{ x } 9.7594 \text{ x } 0.0162 \text{ x } 0.0762 (0.06) \text{x } \frac{0.0446}{0.0646} \text{ x } 0.2079 \text{x} 0.9781$$

$$G = 1651 \text{kg/h}$$

2.1.23. ANALYSIS OF FLOW

i. Drag Flow (Q_d)

$Q_d = \frac{1}{2} \text{ x } \pi^2 \text{ x } D_s^2 \text{ x } N_1 \text{ x } H_{cd} \text{ x } \sin\varphi \text{ x } \cos\varphi$. (27)

Where,

Screw diameter (D_s) = 70mm = 0.07m

Screw Speed (N_2) = 563r.p.m

Channel Depth (H_{cd}) = 16.2mm = 0.0162m

Helix angle (φ) = 12^0

Therefore

$$Q_d = \frac{1}{2} \text{ x } (3.142)^2 \text{ x } (0.07)^2 \text{ x } 563 \text{ x } 0.0162 \text{ x } \sin(12) \text{ x } \cos(12)$$

$$Q_d = \frac{1}{2} \text{ x } 9.87 \text{ x } 0.0049 \text{ x } 563 \text{ x } 0.0162 \text{ x} 0.2079 \text{ x } 0.9781$$

$$Q_d = \frac{0.0897}{2}$$

Q_d = 0.0449m^3/s

ii. Pressure Flow (Q_p)

$\text{Pressure Flow } (Q_p) = \frac{\pi D_s H_{cd}{}^3 \sin 2\varphi}{12^n} \text{ x } \frac{P_a}{L_{esl}}$. (28)

Where,

Screw diameter (D_s) - 70mm = 0.70m

Channel depth *(H_{cd})* =16.2mm = 0.0162m

Helix angle *(φ)* = 12°

Fluid viscosity (η) = 0.280

Operation Pressure (P_a) =?

Effective screw length (L_{esl}) = 457mm — 0.457m

But

The pressure distribution of the flow in the extruder is the total output obtained

from the drag flow, back pressure flow and leakage. Assuming that there is no

leakage

$$Q_d = \frac{1}{2} \text{ x } \pi^2 \text{ x } D_s^2 \text{ x } N_2 \text{ x } H_{cd} \sin\varphi \, \cos\varphi - \frac{\pi D_s H_{cd}{}^2 \sin^2\varphi}{12} \frac{P_a}{L_{esl}} = Q_d - Q_p$$ (Crawford, 1998)

When there is no pressure build up at the end of the extruder, any flow is due to drag and maximum flow rate Q_{max} can be obtained. The equation then can be reduced to only the drag term as follows.

$$Q = Q_{max} = \frac{1}{2}\pi^2 D_s^2 N_2 H_{cd} \sin\varphi \, \cos\varphi \qquad (29)$$

Therefore,

$$Q = Q_{max} = \frac{1}{2} \text{ x } 3.142^2 \text{ x } 0.07^2 \text{ x } 563 \text{ x } 0.0162$$

$$\text{x} \sin 12 \, \cos 12$$

$$Q = Q_{max} = \frac{1}{2} \text{x } 9.87 \text{ x } 4.9 \text{ x } 10^{-3} \text{ x } 563 \text{ x } 0.0162 \text{ x}$$

$$0.2079 \text{ x } 0.9781$$

$$Q = Q_{max} = \frac{0.0897}{2}$$

$$Q = Q_{max} = 0.0449 m^3/s$$

Similarly, when there is a high pressure drop at the end of the extruder the output of the extruder, Q becomes equal to zero (Q=0) and the maximum pressure is obtained from the equation.

$$\frac{1}{2}\pi^2 \text{ x } D_s^2 \text{ x } N_2 \text{ x } H_{cd} \sin\varphi \, \cos\varphi = \frac{\pi D_s H_{cd}{}^2 \sin^2\varphi}{12^n} \frac{P_a}{L_{esl}} \ . \qquad (30)$$

$$P_a = \frac{12^n L_{esl} \pi^2 D_s{}^2 H_{cd} \sin\varphi\cos\varphi}{2\pi D_s H_{cd}{}^3 \sin^2\varphi}$$

$$P_a = \frac{6^n L_{esl} D_2 N_2 \cos\varphi}{H_{cd}{}^2 \sin\varphi}$$

$$Recall, \tan\varphi = \frac{\sin\varphi}{\cos\varphi} \therefore \frac{1}{\tan\varphi} = \frac{\cos\varphi}{\sin\varphi}$$

Hence,

$$P_a = \frac{6\pi L_{esl} D_2 N_2{}^n}{H_{cd}{}^2 \tan\varphi} \qquad (31)(\text{Crawford } 1998)$$

But

$$\eta = m(T^0 C)\Upsilon^{\eta-1} \qquad (32)(\text{Oswald, } 2009)$$

Where, m = consistency index = 2.00×10^4

n = power law index = 0.41

The power law is usually represented as $\eta = m\gamma^{n-1}$, where m is sometimes replaced by 'k' or other letter (Michaeli, 2003). The consistency index is said to include the temperature dependence of the viscosity whilst the power law index represents the shear thinning behavior of the polymer melt. "The limits of the law are zero (0) and infinity" (Osswald, 2009)

Therefore,

$$\eta = m\Upsilon^{\eta-1} \qquad (33)$$

But,

Shear rate for a quadratic cross section is given by

$$\Upsilon = \frac{6Q}{W_{FLT} H^2{}_{SFLT}} \qquad (34)$$

$$\Upsilon = \frac{6 \text{ x } 0.0449}{0.02 \text{ x } 0.0162^2}$$

$$\Upsilon = \frac{0.2694}{5.2488 \text{ x } 10^{-6}}$$

$$\Upsilon = 51326 \, s^{-1}$$

Therefore,

$$\eta = (2.00 \text{ x } 10^4)(51326)^{0.41-1}$$

$$\eta = (2.00 \text{ x } 10^4)(51326)^{-0.59}$$

$$\eta = 33.26 \, Pa.s$$

Therefore

$$P_a = \frac{6 \text{ x } 3.142 \text{ x } 0.70 \text{ x } 0.457 \text{ x } 563 \text{ x } 33.26}{0.0162^2 \text{ x } \tan 12}$$

$$P_a = \frac{112928.1753}{0.0002624 \text{ x } 0.2126}$$

$$P_a = \frac{112928.1753}{0.00005579}$$

$$P_a = 2024165179 Pa$$

P_a = 2 GPa

Therefore, $\text{Pressure flow } (Q_p) = \frac{\pi D_s H_{cd}{}^3 \sin 2\varphi}{12^\eta} \text{ x } \frac{P_a}{L_{esl}}$

$$\text{Pressure flow } (Q_p) = \frac{3.142 \text{ x } 0.07 \text{ x } 0.0162^3 \sin^2 12}{12 \text{ x } 33.26} \text{ x } \frac{2 \text{ x } 10^9}{0.457}$$

$$\text{Pressure flow } (Q_p) = \frac{9.35 \text{ x } 10^{-7} \text{ x } 0.0432 \text{ x } 2 \text{ x } 10^9}{12 \text{ x } 33.26 \text{ x } 0.457}$$

$$\text{Pressure flow } (Q_p) = \frac{80.784}{182.397}$$

$$\text{Pressure flow } (Q_p) = 0.4429$$

3. RESULT AND DISCUSSION

3.1 PERFORMANCE EVALUATION

The machine was tested with 1000cm^3 of high density polyethylene and it was found that the constructed machine was capable of handling 1000 cm^3 of polymers in 16 minutes. The machine can be used for compounding polymer and polymer composites.

3.2 EFFICIENCY OF THE MACHINE

$$\text{Efficiency} = \frac{\text{work output}}{\text{work input}} \text{ x } 100\%$$

Where,

Work input corresponds to the weight

Work input (1000cm^3) = 353g = 0.353kg

Work output = 285g = 0.285kg

$$\text{Efficiency} = \frac{0.285}{0.353} \text{ x } 100\% = 0.807 \text{ x } 100\% = 81\%$$

4. CONCLUSION

A viable machine for extrusion commonly found in Nigeria was fabricated from available locally sourced materials. The single screw extruder machine is very applicable for local production, operation, repair and maintenance. The operation of this machine manually makes it a unique type compare to others. The operation save energy and did not require high skilled labour. The machine performs different types of operations during the extrusion process that can be categorized as solid conveying, melting and melt conveying. This extruder is indisputably the most important piece of machinery in the polymer processing industry. The advancement obtained from the machine (single screw extruder machine) brings about high productivity. It also provides a more efficient and fast means of mixing polymeric materials. As a matter of fact, the project is just a starting point for the future need for such a machine, therefore the following is recommended to enhance a more improved single screw extruder machine;

1. The machine should be adapted for the processing of different varieties of polymers and food cookies.

2. Using stainless material for the construction of frame unit will make it more hygienic.

3. Incorporation of a pressure gauge device to measure the amount of pressure in the barrel

4. An improved screw design should be done for better mixing properties.

REFERENCES

[1]. Sakai T, Seikei-Kakou (2005), International polymer processing, Vol. 17, 20, Pages17-216; 109.

[2]. R. M. Ogorkiewicz (1977), The Engineering Properties of Plastics, Oxford University Press.

[3]. Kokini J. L. (1993), the effect of processing history on chemical changes in single-screw and twin screw extruders. Trends in Food Science and Technology, 4 (10), 324-329

[4]. Cassagnau.P, M.and Taha, J. (1996), Application. Polymer.Science, 60 (10).

[5]. Karwe M. V; Godavarti S. (1997), Accurate measurement of extrudate

temperature and heat loss on a twin-screw extruder. Journal of Food Science, 62(2), 367-372.

[6]. R. Millet (1997), Design and Technology of plastic, A Wheaton and Co. Ltd., Exeter.

[7]. Crawford R. (1998), Plastic Engineering, 3rd ed. Oxford: Butterworth - Heine-mann

[8]. Tim A. Oswald, Juan Pablo Hernandez-Ortiz (2009), Polymer processing [www.books.google.uk (online) Available, http://books.google.co.uk/books].

[9]. Rosato, Mariene G. (2000), Concise encyclopedia of plastics, Springer, page. 245, ISBN 978-0-7923-8496-0.

[10]. R. S. Khurmi and J. K. Gupta (2005). A Textbook of Machine Design, Eurasia Publishin House (PVT.) ltd.

[11]. S. Kalpakjian and S.R. Schmid (2008), Manufacturing Processes for Engineering Materials, 5th ed., Pearson Education.