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Design of single screw extruder for homogenizing bulk solids

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Abstract: The research was conducted to design a single screw extruder from locally available materials for the mixing and homogenization of bulk solids, such as the composites of cassava and yam starch-glycerol with nanoparticles. The design was made by computing the hopper outlet size, shaft diameter, screw geometry, barrel volume and the capacity of the conveyor, empirically. The stresses in the conical section of the hopper were also evaluated to assess its load requirement, thus avoiding any problem associated with the flow of materials through the hopper opening. The extruder was dynamically simulated to assess its throughput at the feeding, compression and metering zones. This was done by investigating the dynamic effect of the time of operation, with respect to the linear displacement, velocity and power, from the practical motion of the moving auger by Computational Fluid Dynamics method. The results showed that the vertical pressure acting downwards and the shear stress within the section were 37.02 and 6.44 kPa. The shaft diameter and screw geometry, which includes screw pitch and angle, were 20 and 56 mm, and 16.54o. The capacity of the extrusion conveyor and its power requirement were respectively, 18.46 tons/hour and 2.04 kW. The maximum linear displacement and velocity occur at the compression zone at every 3.03 rev/sec, which cause the bulk solid materials to melt, and are pushed by the resulting pressure into the metering zone. The relationship between the linear displacement and the time of operation obeys the power law. Consequently, a 5 hp electric motor was selected to power the single crew extruder.

Keywords: design, dynamic simulation, single screw extruder, homogenization, bulk solids

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1 Introduction

The interest and activities in nanocomposite researching are receiving serious boost in recent times considering the number of breakthroughs recorded yearly especially in the medical and food sectors (Weiss et al., 2006). The introduction of the science of nanotechnology in the packaging of fruits and vegetables has been proven to proffer solutions to the packaging challenges such as short shelf life and deterioration (Emamifar et al., 2011). The blend of biopolymers derived from renewable sources (proteins, cellulose, starches, and other polysaccharides and those synthesized chemically from naturally derived monomers such as lactic acid) with organic or inorganic particles smaller than 100 nm produces nanocomposite materials (Azeredo, 2009). Although, the blending of ordinary biopolymer materials can form essential composites for film casting, the introduction of nanoparticles in the mixtures will enhance the mechanical and barrier properties of the film (Lagaron and Sanchez-Garsia, 2008).

A screw extruder is usually used in extrusion process such as in the food industry, injection moulding and plastic production. It is used due to its simple geometry, effective control of product quality and wider application (Ficarella et al., 2006). Sorrentino et al. (2007) reported that effective blending of biopolymer material with organic or inorganic nanoparticles can be achieved using either a single or twin screw extruder. Extrusion of thermoplastics is a process in which the material is melted by external heat or frictional heat and conveyed forward by a screw to the opening of a die, which gives the shape of the required product. Extrusion process is a continuous process by which many products like films,

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raffia tapes, pipes, sheets, mono filaments, fibre and filaments can be manufactured (Jiang and Bi, 2009).

The choice of a screw extruder depends heavily on the flow ability of the bulk material fed through the equipment. Most screw extruders are designed for specific kind of bulk solid material (Jiang and Bi, 2009). Depending on the particle size characteristics of the bulk solids, extruder designs are carefully made to avoid likely flow problems associated with material handling (Fadeyibi et al., 2004). For more cohesive material such as thermoplastic starch, an arch or rathole may form if they are fed in screw extruders designed to accommodate less cohesive thermoplastic materials. Other flow problems related to the density of the bulk solid such as flooding may also occur during the material discharge (Prescott and Bernum, 2000).

Unfortunately, however, no work was reported on the design of screw extruder for handling of the particulate solids, such as the composites of cassava and yam starch-glycerol with nanoparticles in the literature. Consequently, the drive towards localizing the concept of nanotechnology has long been thwarted in the developing countries of Africa, especially in the food industry where it is most needed, and considering the high amount of annual fruits and vegetable wastage in the land. No wander this idea was restricted to the developed world of Europe and Asia, where the facilities are available and easily assessable. Therefore, the objective of this study was to design and perform the motion simulation of a single screw extruder for mixing and homogenisation of bulk solids.

2 Materials and methods

2.1 Auger description and design assumptions

According to Potente et al. (1994), the capacity of the extrusion conveyor of most screw extruders is 15 tons per h (approximately). The distance between the axis of flight screws was assumed 60 mm. The helix of most screw extruders is usually made up of a uniform pitch of 43 mm, and this is as well considered in the present design (Figure 1). The auger was designed tapered with screw diameter twice the pitch length. The sides of the screw flights are radial to the screw axis and the depth of the screw channel is constant across its width. Gravity and centrifugal inertia forces of the bulk solids are negligible in comparison with viscous and pressure forces. The auger has three distinct zones with varying channel width, and they are feed, compression and metering zones. The feed zone, which has the channel width same throughout the length, collects the bulk solids into the screw extruder. The compression zone, where the channel width gets progressively smaller, further mixed the bulk solids. The metering zone, which has a constant channel width, is where unmixed bulk solids are homogenized or mixed and melted to a uniform temperature and composition. Therefore, this information was used as a guide in the present design so as to achieve better design and performance.

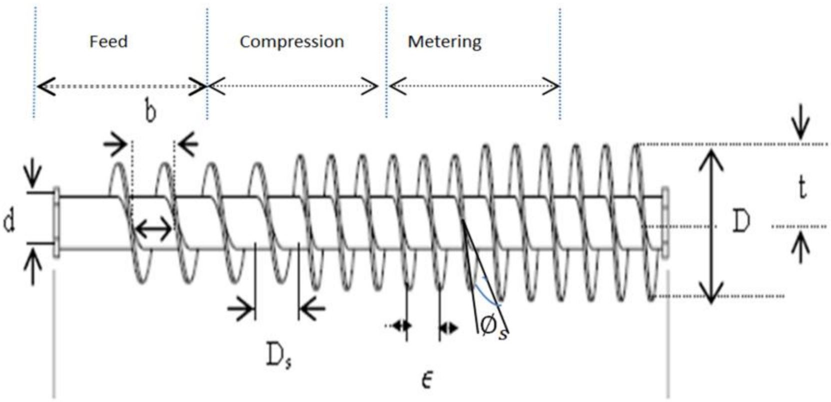

Figure 1 Auger Geometry [D = barrel diameter (mm), D_s = distance between axis of flight screws (mm), d = shaft diameter (mm), t = pitch length (mm), b = channel width (mm), ε = pitch of helix (mm), $\emptyset_s$ = helix angle (deg.)].

2.1 Design calculations

2.2.1 Extrusion barrel volume and capacity of the extrusion conveyor

The volume of the barrel was computed from the expression given by Khurmi and Gupta (2005) as shown in Equation (1).

$$V_{barrel} = \frac{1}{3}h\left(A_1 + \sqrt{(A_1A_2)} + A_2\right) \quad (1)$$

where, A_1 = the area at the feeding section and A_2 = the compression section of the barrel

A horizontal acting auger conveyor that operates inside a close fitted tapered extrusion barrel to be used for achieving composite conditioning and the subsequent extrusion through the die was design for the screw extruder. The auger was designed tapered with the pitch diameter of twice the pitch length. Its helices (distances between axes of flight screws) are designed with uniform pitch of 43 mm. Hence, the capacity of the horizontal acting auger conveyor was computed using the equation given by Khurmi and Gupta (2005) as shown in Equation (2).

$$Q = 60n\emptyset p\gamma(D^2 - d^2)\frac{\pi}{4} \quad (2)$$

where, Q = capacity of screw conveyor in the barrel, t/h, $\gamma =$ bulk density of conveyed material, 300 kg/m^3, n= number screw rotation (assume 500 rpm), $\emptyset$ = factor induced for horizontal conveyor, 1, p= conveyor pitch, 0.043 m, D = pitch diameter of conveyor, 0.112 m, d= shaft diameter, 0.02 m.

2.2.2 Hopper opening

The hopper outlet size was calculated from the Potente et al. (1994) as shown in Equation (3).

$$B = \frac{\sigma_c H(\theta)}{\rho g} \quad (3)$$

Where, B = outlet diameter (m), σ_c = consolidated stress generated in an arch at the outlet (kPa), $H(\theta)$ = friction that takes account of vibration in the arch thickness, hopper half angle and hopper geometric configuration, ρ = bulk density of material (kg/m^3), g = acceleration due to gravity.

The equation of a shaft having little or no axial loading was used to compute the diameter according to Khurmi and Gupta (2005) as shown in Equation (4).

$$d = \left(\frac{16}{\pi S_a}\sqrt{(K_bM_b)^2 + (K_tM_t)^2}\right)^{1/3} \quad (4)$$

Where, d= diameter of shaft, m, M_b = bending moment, Nm, K_b = combined shock and fatigue factor applied to bending moment, K_t = combined shock and fatigue factor applied to torsional moment, S_a = allowable stress, N/m^2.

For rotating shaft where load is suddenly applied (minor shock) Khurmi and Gupta (2005) stated that K_b and K_t range between 1.5 to 2.0 and 1.0 to 1.5, respectively.

2.2.2.1 Stress evaluation in hopper

The equations provided by Christensen (1997) were used to evaluate the stress in the cylindrical section of the hopper. The expressions are shown in Equations (5), (6) and (7).

$$P_v = \left(\frac{\gamma g D\mu K}{4}\right)\left(1 - \exp(-\frac{4H\mu K}{D})\right) \quad (5)$$

$$P_w = KP_v \quad (6)$$

$$\tau = \mu P_w \quad (7)$$

Where, γ = bulk density of the powdered solids, P_v = vertical pressure acting downwards, D = diameter of the hopper, μ = coefficient of friction of the material, H = height of the hopper, P_w = Janssen pressure equivalent, K = Janssen coefficient, assumed to be 0.4. It can vary according to the material but it is not often measured.

2.2.3 Screw geometry

The screw geometry which includes flight width, screw pitch, pitch angle, channel width and channel depth was designed using the equations provided by Potente et al. (1994), and the relationship between the outer screw diameter and the distance between the axis of a flight screw, intermeshing angle ($\propto$), flight angle (β), screw pitch (t), angle of pitch ($\emptyset_s$), channels width (b_{max}) and depth (h_{max}) were computed from the Equations (8) to (14).

$$\frac{a}{D_s} \leq \frac{\sqrt{2}}{2} \quad (8)$$

$$\propto = 2\cos^{-1}(\frac{a}{D_s}) \quad (9)$$

$$\beta = \frac{\pi}{i} - \alpha \quad (10)$$

$$t = \frac{2\pi L_{Kn}}{\delta j_{Kn}} \quad (11)$$

$$\emptyset_s = \tan^{-1}(\frac{t}{\pi D_s}) \quad (12)$$

$$b_{max} = t\cos\emptyset_s - \epsilon_{max} \quad (13)$$

$$h_{max} = D_s - a \quad (14)$$

where, L_{Kn} = 28 mm right handed standard screw, δ = 1.57 radians, j_{Kn} = standard screw element = 2, a = distance between axis of flight (mm).

2.2 Force interaction in the extrusion barrel

2.3.1 Thrust and tangential forces

The thrust force (W) moving the material inside the extrusion barrel of cross sectional area (A) and the force which lifts the composite as it moves inside the extrusion barrel was computed from the expression given by Khurmi and Gupta (2005) as shown in Equations (15) and (16).

$$W = PA \quad (15)$$

$$F_\theta = Wtan\emptyset_s \quad (16)$$

where, P = pressure inside the barrel, Pa, A = area of extrude, m^2, F_θ= tangential force lifting the composite material inside the barrel, N/m^2, W = thrust force, $\emptyset_s$ = pitch angle.

2.3.2 Die extrusion pressure

The extrusion pressure of composite through the die of the screw extruder varies with a constant feed rate, speed of extrusion, composite formulation and material density. An annular die with 5 mm and 7 mm as inner and outer diameters with 20 holes was considered in this design. Thus, the cross sectional area of the annular dies with 20 holes, and hence, the die extrusion pressure was computed from the following Equations (17) and (18).

$$A_{die} = n\frac{\pi}{4}({D_2}^2 - {D_1}^2) \quad (17)$$

$$P_{die} = \frac{W}{A_{die}} \quad (18)$$

where, A_{die} = area of annular die with 20 holes, m^2, D_1 = inner diameter of the die, m, D_2 = outer diameter of the die, m.

2.3.3 Material hold-up

The equation for computing the materials hold up in the barrel of a screw extruder as proposed by Khurmi and Gupta (2005) was used evaluates the material holdup in the extrusion barrel. This is expressed in Equation (19).

$$H = DV_{Tot} \quad (19)$$

where, H = material hold-up, m^3, D = degree of fill, 46%, V_{Tot} = reaction volume of the extruder = Volume of barrel − volume of screw

2.4 Pulley diameter, belt length and speed

The V-belt and pulley arrangement were used to transmit power from the electric motor to the shaft of the screw extruding unit. The flexibility, simplicity, cost of maintenance and the ability to absorb shocks against the effects of forces of vibration are the reasons behind the choice of the V-belt drive (Khurmi and Gupta, 2005). The diameter of the pulley of the screw shaft was computed from Equation (20).

$$D_2 D_1 = N_2 N_1 \quad (20)$$

where, N_1 = speed of motor, rpm, D_1 = diameter of the drive pulley (motor pulley), D_2 = diameter of the screw shaft pulley, mm, N_2 = speed of the screw shaft

Khurmi and Gupta (2005) provided the expressions which was used to compute the belt speed to drive the screw shaft and the length of belt of the extruder of the extruder as shown in Equations (21) and (22).

$$V_{belt} = \frac{\pi DN}{6\times10^4} \quad (21)$$

where, V_{belt} = belt speed, m/s, D = diameter of the drive pulley (motor pulley), 30 mm, N = speed of the motor, r/min$X = 2C + 1.57(D_1 + D_2) + \frac{(D_1+D_2)^2}{4C}$

(22)

where, X = belt length, m, C = center distance between pulley, mm (assumed), D_1 = pitch diameter of drive pulley, mm, D_2 = pitch diameter of driven pulley, mm.

2.5 Power requirement for motor selection

The power required by the screw conveyor was computed from the expression given by Khurmi and Gupta (2005), as shown in Equations (23) and (24).

$$P_{Sc} = 0.7355\text{ClQ} \quad (23)$$

where, P_{Sc} = power required by the screw conveyor, kW, C = constant coefficient for conveyed material, usually taken to be 0.3, l = length of the screw conveyor, taken be 500 mm since 50 mm was isolated for clearance from the extrusion barrel.

The pressure in the extrusion barrel was computed from the following Equation (24):

$$P = \frac{P_{Sc}}{Q_f} \quad (24)$$

where, P = pressure inside the extrusion barrel, N/m^2, P_{Sc} = power available to the screw conveyor, kW, Q_f = flow rate of material, kg/s.

2.6 Motion simulation of the machine

The motion simulation of the single screw extrusion machine was performed using Solid Works Design Package (2011) by the method of Computational Fluid Dynamics. The effects of time of operation of the machine on linear displacement, angular displacement, velocity and acceleration, translational kinetic energy and power consumption were investigated from the motion of the moving auger from the feed to the metering zone of the screw extruder.

2.7 Technical characteristics of the machine

The technical characteristics of the single screw extruder essentially designed for the mixing and homogenization of the starch-glycerol with nanoparticles, were presented in Table 1. The hopper outlet size, volume and discharge rate of cassava and yam starch-glycerol composites, as computed were also shown. The hopper outlet size was computed based on the flow properties of the bulk solids, including friction factor (2.48), consolidation stress (2.56 kPa) and hopper half angle (25^o) at 3 g /cm^3, as determined by Fadeyibi et al. (2014). The theoretical pressure acting downward, Johansson pressure equivalent and the shear stress of the composites in the conical section of the hopper were computed. The volume of the barrel, which houses the conveying auger, and the capacity of the conveyor together with its operating pressure, thrust and tangential forces, and the material holdup in the extrusion barrel were computed from the design expressions as given in Table 1. The belt speed and the shaft diameter of the machine were computed from the expressions given by Kurmi and Gupta (2005).

The screw geometry, which includes the distance between the axis of flight screws, flight angle, screw pitch, pitch angle flight and channel widths were computed from the expressions given in Equations (8) to (14). The gap, chamber and road areas, which define the pitch screw of the extruder, were carefully designed to meet specification and purpose. The gap area, otherwise known as clearance, is the distance between the inner surface of the barrel and the surface of screw flight. Meaningless material passing through this area and can be ignored. This area is often called the zero velocity area because, at this stage, the materials are momentarily at rest. In this design, the size of the gap area (0.5 mm) corroborates the range suggested by Siregal et al. (2014). The authors stated the size of the gap area in the range of 0.5--1.0 mm. It is important to note that, if the gap area is too large, the material is difficult to push forward but if it is too small, screw flight will rub against with the surface of the inner barrel lead to overheating. The chamber area, also called normal pitch, is a distance between blades, both above and below. This is the centre of circulation, which the material is transferred from one chamber to the other chamber. Additionally, this will get the pressure forced through the die section of the screw extruder. Therefore, high pressure is required to push the composite materials forward. In the design screw, however, the screw pitch was 56 mm, which correspond to 28 mm normal pitch. The normal pitch spacing was just enough to cause an increase in pressure sufficient to make the granular material flow. This is also in agreement with the works of Siregal et al. (2014), who suggested the range of 17--22 mm for better design. Finally, the road area corresponds to the surface of body of the screw.

Table 1 Technical characteristics of the single screw extruder

Parameters	Design Expression	Value
	Hopper design data	
Hopper Outlet Diameter, m	$B = \frac{\sigma_c H_a(\theta)}{\rho g}$	0.087
Hopper Volume, m^3	$V = \pi \frac{(D^2 - B^2)h}{12}$	0.019
Discharge Rate of Composite from Hopper, Kg/s	$W = 0.58\rho_b g^{0.5}(B - kd_p)^{2.5}$	6.075
Stress Evaluated in the Conical Section of the Hopper		
Theoretical Vertical Pressure acting downwards, N/m^2	$P_v = \left(\frac{\gamma g D\mu C}{4}\right)\left(1 - \exp(-\frac{4H\mu C}{D})\right)$	55.53
Jansson Pressure Equivalent, N/m^2	$P_w = CP_v$	22.21
Shear stress in the conical section of hopper, N/m^2	$\tau = \mu P_v$	9.658
Screw Design Geometry		
Barrel diameter, m	Assume $D_s = 60$	0.060
Distance between axis of flight screw, mm	$\frac{a}{D_s} \leq \frac{\sqrt{2}}{2}$	0.042
Flight angle, Deg/rad	$\beta = \frac{\pi}{i} - \alpha$	90^o (1.57)
Screw pitch, mm	$t = 2\pi L_{Kn}/\delta j_{Kn}$	56.00
Pitch angle, Deg/ rad	$\emptyset_s = \tan^{-1}(t/\pi D_s)$	16.54 (0.289)
Flight width, mm	$\epsilon_{max} = t\beta \cos\emptyset_s/2\pi$	0.013
Channel width, Mm	$b_{max} = t\cos\emptyset_s - \epsilon_{max}$	0.018
Extrusion Data		
Barrel volume, m^3	$V = \frac{1}{3}h\left(A_1 + \sqrt{(A_1A_2)} + A_2\right)$	0.0075
Capacity of extrusion conveyor, Tonnes/h	$Q = 15n\emptyset p\gamma(D^2 + d^2)\pi$	18.46
Power required by screw conveyor, kPa	$P_{Sc} = 0.7355ClQ$	2.037, 5HP
Operating pressure in the barrel, kPa	$P = P_{Sc}/Q_f$	397.53
Thrust force in the barrel, kN	$F_w = PA$	3.578
Tangential force in the barrel, kN	$F_\theta = Wtan\emptyset_s$	1.063
Theoretical pressure on composite through die, kPa	$P_{die} = W/A_{die}$	949.07
Material holdup in the extrusion barrel, m^3	$H = DV_{Tot}$	26.04
Rate of shear in the channel, s^{-1}	$\gamma_{Channel} = \frac{\pi DN}{60h}$	89.41
Shear rate between screw flight and Barrel wall, s^{-1}	$\gamma_{fligth/wall} = \frac{\pi DN}{60h}$	157.1
Shear rate in annular die, s^{-1}	$\gamma = \frac{6Q}{\pi(R_i + R_0)h^2}$	8704.06
Drive and Driven Pulley Data		
Drive pulley diameter, m	Assume $D_1 = 30$	0.030
Driven pulley diameter, m	$D_2D_1 = N_2N_1$	0.087
Belt speed, m/s	$V_{belt} = \pi DN/60 \times 10^4$	2.280
Shaft diameter, mm	$d^3 = 16M_{twist}/\pi\tau_{al}$	20.00

Variables in all equations retain their usual meaning (Potente *at al*., 1994; Khurmi and Gupta, 2005; Christensen, 1997)

2.8 Machine drawing

The assembly, isometric and third angle projection drawings of the single screw extruder are shown in Figures 2 to 4, respectively.

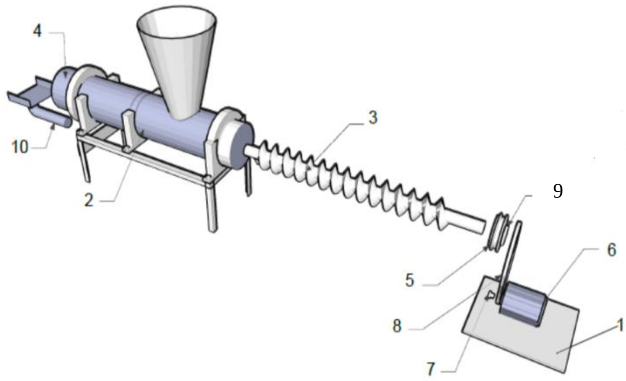

Figure 2 Assembly drawing of the single screw extruder

(1-Electric motor seat, 2- Extrusion barrel, 3- Auger, 4- Die head, 5- Pulley, 6- Electric Motor, 7- Motor pulley, 8- Belt drive, 9- Tip, 10- Collector)

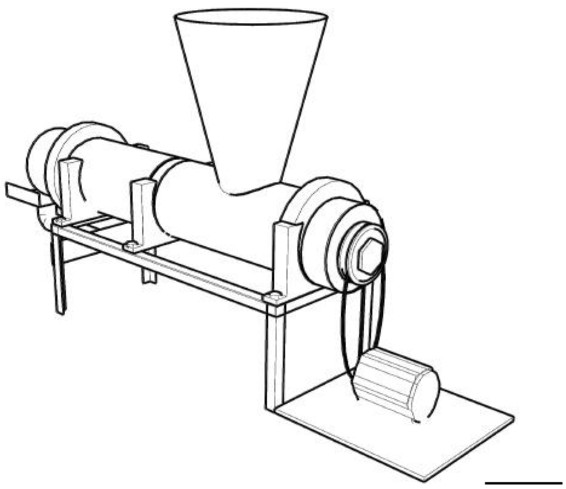

Figure 3 Isometric drawing of the single screw extruder

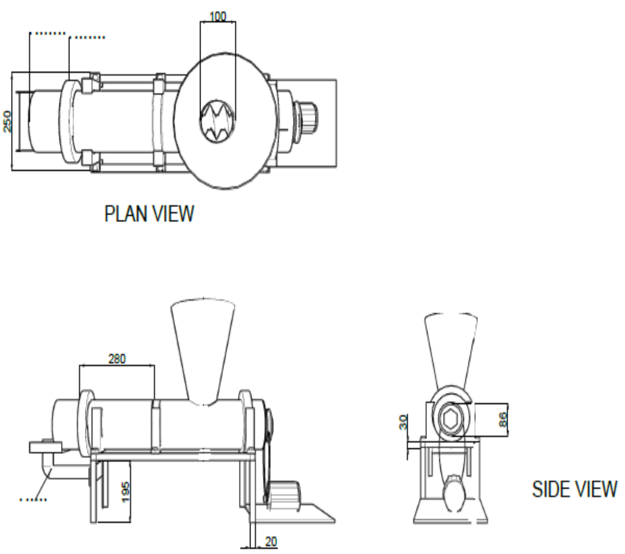

Figure 4 Projection (third angle) of the single screw extruder

3 Results and discussion

The motion simulation of the single screw extrusion machine was performed by the method of Computational Fluid Dynamics. The concept of velocity and local shear rate or displacement, as shown in Figures 5 and 6, described and verified the flow within the small gap between two concentric cylinders, one rotating at a constant speed and the other one stationary. The process begins when the screw was rotated in the barrel so that the bulk solids split and meshed to form a more compact composite. Thereafter, the material moves from one chamber to the next, due to the drag force exerted by the screw rotation on the z-direction (Dhanasekharan and Kokini, 2003). This then led to differences in the velocity magnitude and local shear rate. The simulation results were used to modify several variables to find the best expected results (Siregal et al., 2014). The velocity profile and local shear rate inside a screw extruder were the important factors in understanding the effect of pitch screw since they can be used to determine the effectiveness of the screw extruder. This agrees with the results obtained by Connelly and Kokini (2007) in their work on examination of the mixing ability of single and twin screw mixers using 2D finite element method simulation with particle tracking. Similarly, in their work on the finite element modelling of the mechanical behaviour of Jatropha seed under compression loading, Petru et al. (2012) agree that the force of compression splits and meshes granular materials until their surface areas are decreased.

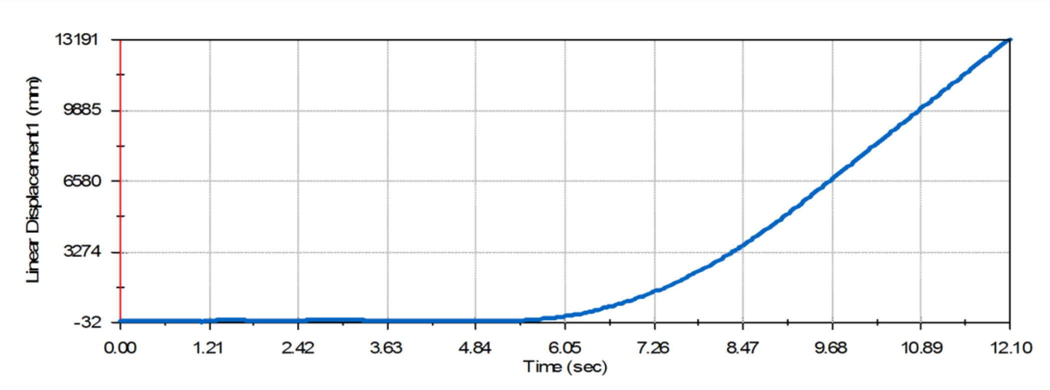

Figure 5 Effect of time of operation on linear displacement

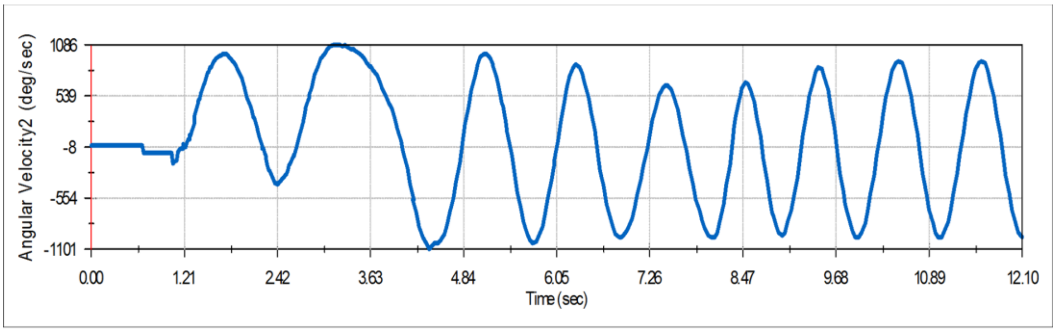

Figure 6 Effect of time of operation on angular velocity

The dynamic effects of time of operation of the extruder on the acceleration, kinetic energy and power were investigated from the motion of the moving auger from the feed to the transition regions of the screw extruder, as shown in Figures 7 to 9. As the shaft rotates from the feed zone through the compression to the metering zones, the auger will have completed many oscillations around the axis. Once the machine begins operation, the maximum acceleration and kinetic energy will occur in the compression zone of the screw extruder at 3.03 rev/sec. This effect will be essential for the effective mixing and homogenization of the granular materials at the compression zone, because at that stage, most of the composites will have melted. The melt flow pattern may be considered to have existed in the downstream of the melting zone occupying the screw channel and the melt pool region extending side by side with the solid bed in compression zone. Consequently, the materials will find their way through the metering or conveying zone, since the channel width decreases with an increase in the length of the barrel. Hence, according to Siregar et al. (2014) and Jiang and Bi (2009), the materials will acquire sufficient pressure to pass through the breaker plate and into die opening of the extruder.

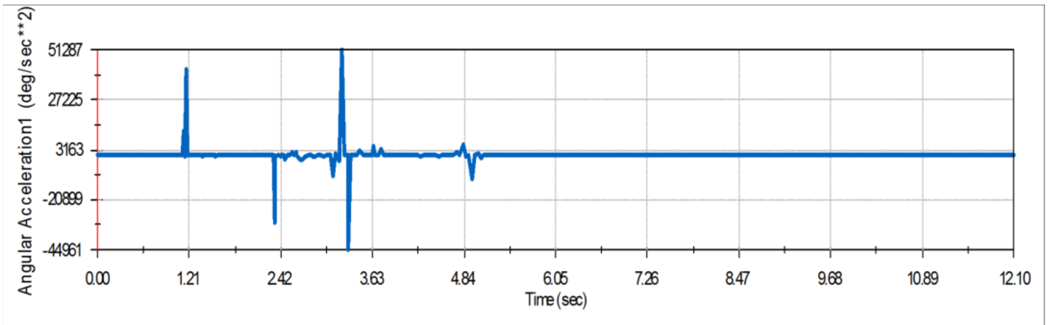

Figure 7 Effect of time of operation on angular acceleration

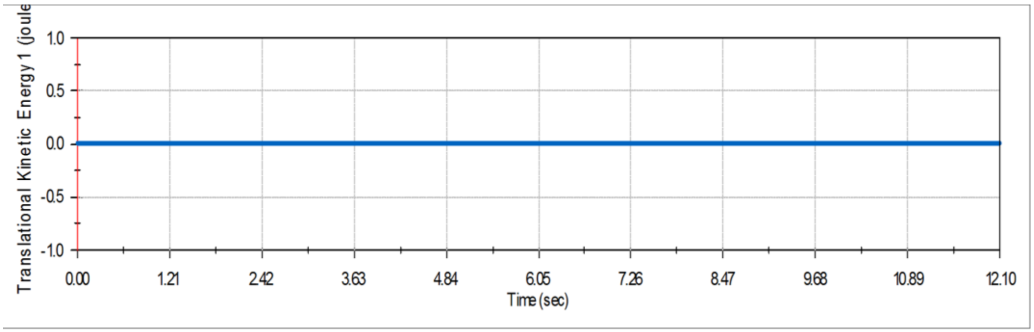

Figure 8 Effect of time of operation on translational kinetic energy

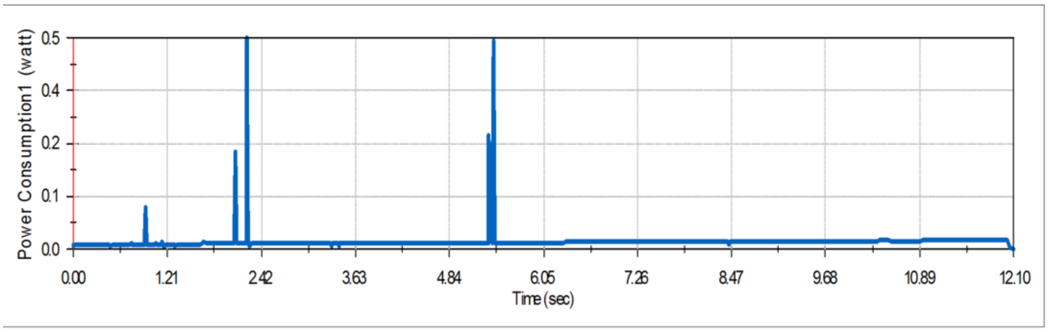

Figure 9 Effect of time of operation on power consumption

Furthermore, at the metering zone, which corresponds to the interval between 9.68 and 12.1 s operation periods, the acceleration and kinetic energy of the moving auger will remain steady at a constant channel depth. Consequently, the bulk materials will flow at a constant kinetic energy, sufficient enough to melt and mix the solid particles to a uniform temperature and composition. Moreover, it is expected that more power will be consumed by the moving auger at the feed and compression zones of the screw extruder; but at the metering zone, much of the work will have been done at the stages preceding it, and so the starch granules will simply flow at almost no power consumption through the die end of the machine. This corroborates the study by El-Sadi and Esmail (2005) on the simulation of complex liquids in micro-pump. They stated that the power consumption by a micro-pump operating at high speed is relatively constant.

4 Conclusions

A single screw extruder was designed for mixing and homogenization of bulk solids, such as the cassava and yam starch nanocomposites. The Computational Fluid Dynamics method was used to investigate the dynamic effect of time of operation on the linear displacement, velocity, acceleration, kinetic energy and power consumption. The vertical pressure acting downwards and the shear stress within the section were 37.02 and 6.44 kPa. The shaft diameter and screw geometry, which includes screw pitch and angle, were 20 and 56 mm, and 16.54^{o}. The capacity of the extrusion conveyor and its power requirement were respectively, 13.12 tons/hour and 1.447 kW. The maximum linear displacement and velocity occur at the compression zone at every 3.03 rev/sec, which cause the bulk solid materials to melt, and are pushed by the resulting pressure into the metering zone. The relationship between the time of operation and the linear displacement obeys the power law, while that with respect to the angular velocity was sigmoidal. The operational kinetic energy of the moving auger was constant with respect to the time operation. Consequently, a 5 HP electric motor was selected to power the single crew extruder.

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