Development of a portable laboratory injection moulding machine

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Abstract

A portable laboratory injection moulding machine was designed and fabricated from locally available materials for teaching, learning and research purposes. The individual components were separately produced. The parts were then joined together either by welding or by bolt and nut assembly. The constructed plastic injection moulding machine was tested and was found to effectively serve the purpose for which it was constructed. The machine which is easy to operate and maintain is capable of processing 305g of plastics per hour.

Keywords

Injection moulding; Plastics; Shaft; Mould; Injection screw

Introduction

The quest for plastic in large quantities is on the increase due to its resistance to corrosion, light weight and low cost. The development of plastics can be regarded as one of the most important technical achievement of the twentieth century. In just 50 years, plastics have permeated virtually every aspect of our daily life, paving way for new inventions and replacing materials in existing products. The success of plastics has been based on their properties, resilience, resistance to moisture, chemical and photo and biodegradation, their stability and the fact that they can be moulded into any desired form [1].

The first man-made commercial plastic was invented in Britain in 1851 by Alexander Parkes. He publicly demonstrated it at the 1862 international exhibition in London, calling the material he produced “Parkesine” [2]. Derived from cellulose, Parkesine could be heated, moulded and retain its shape when cooled. It was however, expensive to produce, prone to cracking and highly flammable. In 1868, American inventor John Wesley Hyatt developed a plastic material he named celluloid, improving on Parkes invention so that it could be processed into finished form. Together with his brother Isaiah, Hyatt patented the first injection moulding machine in 1872. The industry progressed slowly over the years producing products such as collar stays, buttons and hair combs [3, 4].

The plastic injection moulding industry has evolved over the years from producing combs and buttons to producing a vast array of products for many industries including automotive, medical, aerospace, consumer products, toys, plumbing, packaging and construction. Injection moulding which is by far the most widely used process of forming thermoplastic materials [5-7] is used for producing parts from both thermoplastics and thermosetting plastic materials in which the required material is fed into a heated barrel, mixed and forced into a mould cavity where it cools and hardens to the configuration of the mould cavity [8].

A typical injection moulding machine consists of injection chamber, hydraulic, mould, clamping, and cooling and control systems as its major components [9]. In injection moulding machine the resin is injected into the mould by a reciprocating screw or a ram injector. The reciprocating screw offers the advantage of being able to inject a smaller percentage of the total shot (amount of melted resin in the barrel). The ram injector must typically inject at least 20% of the total shot while a screw injector can inject as little as 5% of the total shot. Essentially, the screw injector is better suited for producing smaller parts [10].

Although, there are sophisticated laboratory and industrial injection moulding machines which use electricity, pneumatic and hydraulic for power and automatically eject moulded parts [11], most of them are large, expensive [12,13] and difficult to accommodate in laboratories in tertiary institutions for teaching and research purposes in developing countries. There is, therefore, a need for small but efficient laboratory injection moulding machine which will aid the practical know how of students in various courses.

This work is aimed at the design, construction and testing of a portable plastic injection moulding machine for processing both thermoplastic and thermosetting resins.

Material and method

This section presents the determination of the parameters of the machine components which include the hopper, the screw shaft, the barrel, the nozzle, the heater, the bearing, the electric motor, the pulley drive, the mould and the cooling system as well as construction and performance evaluation of the machine. The machine is designed to handle a maximum of 2000g and 100g of plastics in the hopper and the injection chamber, respectively.

Determination of volume of feed hopper

The feed hopper has the shape of an inverted truncated cone and its volume is given by equation 1.

$$V = \pi h(R^2 + Rr + r^2)/3 \quad (1)$$

where R, r = radius of larger and smaller base, respectively, h=height of truncated cone.

The feed hopper was designed for the volume of plastic granule of a mass of 2000g and plastic density of $950 kg/m^3$.

$$V = M/\rho \quad (2)$$

where M = mass of plastics in hopper, M = 2000g, ρ = density of plastics, $\rho = 950 kg/m^3$.

Therefore, $V = 2.1 \times 10^{-3}\ m^3$

Taking R = 80mm, r = 25mm and substituting into equation 1, h = 225mm.

Power requirement and selection of motor

The power required to turn the plastics (P_p) in chamber may be determined from equations 3 -5.

$$P_p = T_p \omega_2 \quad (3)$$

$$T_p = W_p R_c \quad (4)$$

$$\omega_2 = 2\pi N_2/60 \quad (5)$$

where T_p, W_p = Torque for turning plastics and weight of plastics, respectively, ω_2 = angular speed of driven pulley, R_c = radius of injection chamber (barrel), R_c = 0.015m, N_2 = angular speed of the driven pulley, N_2 = 300rpm.

Substituting the above values into equations 3 – 5, we obtained ω_2 = 31.426rad/s, T_p = 0.15Nm and P_p = 4.71W. Based on the obtained value of power requirement and availability, 1hp motor with angular speed of 1400rpm was selected.

Therefore, angular speed, ω_m and torque of the motor, T_m were determined from the expressions in equations 3 and 5 as ω_m = 146.61rad/sec and T_m = 5.09Nm. This torque of motor is therefore satisfactory for the design. Similarly, T_2 = 23.74Nm.

Design of pulley drive

The diameter of the driving pulley was selected based on the prime mover (electric motor) power rating while that of the driven drive was determined.

The diameter of driving pulley, d_1 was chosen as equal to pitch diameter of 75mm for power rating of 0.746kW from the dimension of standard v-belts according to IS: 2494 - 1974 [14].

d_1 = 75mm,

Therefore, the diameter of the driven pulley d_2 is given by

$$d_2 = d_1 N_1 / N_2 \tag{6}$$

$$d_2 = 350\text{mm}$$

Design and selection of belt

V – belt design was used. In the design of the belt, the centre distance and the pitch length were determined after which the tight and slack tensions were determined.

A tentative minimum centre distance was determined from equation 7 [15].

$$C = (1.5 \rightarrow 2)(d_1+d_2) \tag{7}$$

Take tentative C = 750mm

The pitch length of the belt, Lp was determined from equation 8

$$Lp = 2C+1.57(d_2+d_1)+(d_2-d_1)^2/(4C) \tag{8}$$

where L_p = length of belt (pitch length), C = centre distance.

L_p = 2192mm

From the standard pitch length of V-belt according to IS: 2494 - 1974, the standard pitch length of belt adopted is L_p = 2195mm.

The actual centre distance was then calculated from equation 8 as C = 751mm.

The tension in the belt was determined based on the following procedure.

$$\sin\alpha = (d_2 - d_1)/(2C) \tag{9}$$

where α = Angle between the vertical axis and a line joining inside point of contact of belt with pulley to the centre of the pulley.

$\alpha = 10.55°$

$$\theta = (180 - 2\alpha)\pi/180 \tag{10}$$

where θ = Angle of contact (wrap) at the smaller pulley (in this case, driving pulley).

θ = 2.77 rad

For V-belt type A, the groove angle (2β) is usually between 32° and 38°. The groove angle of 34° was adopted.

Therefore, β = 17°

From the table of coefficient of friction between belt and pulley according to Khurmi and Gupta [14], the coefficient of friction μ was taken as 0.3.

$$T_1/T_2 = e^{(\mu\theta \operatorname{cosec}\beta)} \tag{11}$$

where T_1 and T_2=Tension in the tight side and in the slack side of the belt, respectively.

Substituting the values of μ, θ and β into equation 11, we obtained

$$T_1/T_2 = 17.15 \tag{12}$$

The tangential velocity of the belt is given as

$$V_b = d_1\omega_1/2 \tag{13}$$

$$V_b = 5.50\text{m/s}$$

Hence, centrifugal tension

$$T_c = mv^2 \tag{14}$$

where m = mass of belt per metre length

From dimension of standard v-belt according to IS: 2494 – 1974, the weight of belt per metre length is given as 1.06N/m

m = 1.06/9.81

m = 0.108kg/m

$T_c = 3.27N$

The power transmitted by the motor for one belt was obtained from equation 15

$$P = (T_1 – T_2)_v \tag{15}$$

$$T_1 – T_2 = P/v$$

$$T_1 – T_2 = 135.64 \tag{16}$$

From equation 12

$$T_1 = 17.15T_2 \tag{17}$$

Solving equations 16 and 17 simultaneously yielded $T_2 = 8.40N$ and $T_1 = 144.06N$. The initial belt tension T_o therefore is

$$T_0 = (T_1+T_2)/2 \tag{18}$$

$$T_0 = 76.23N$$

The true initial tension of belt at rest is given as

$$T_0^1 = T_0+T_c \tag{19}$$

$$T_0^1 = 79.50N$$

Design of shaft

Considering the weight of the shaft itself as negligible, the force analysis on the shaft is as shown in Figure 1.

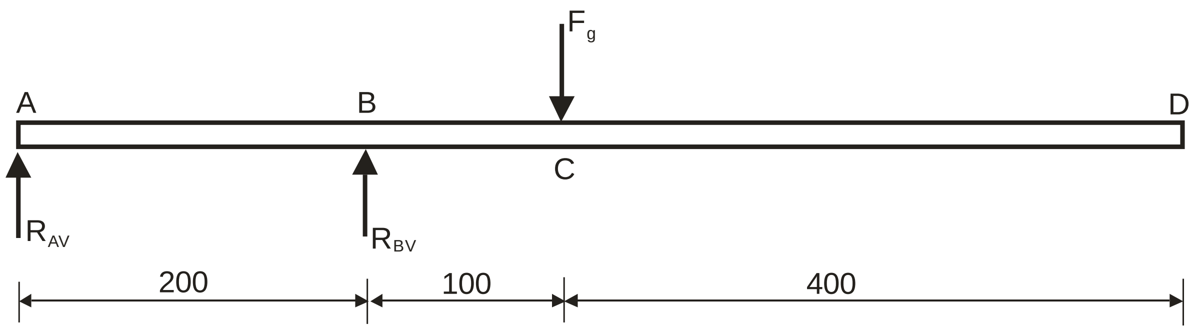

Figure 1. *Free body diagram of force distribution on shaft (vertical axis)*

F_g = mg = weight of plastic granule given as

$F_g = 2000×10^{-3}×9.81$

$F_g = 19.62N$

$$R_{AV} + R_{BV} = F_g \tag{20}$$

$$R_{AV} + R_{BV} = 19.62$$

where R_{AV}, R_{BV} = Reaction at bearings A and B, respectively.

Taking moment about point A

$$M_A = 0; \tag{21}$$

$0.3\ F_g - 0.2R_{BV} = 0$

$R_{BV} = 29.43N$ and $R_{AV} = -9.81N$

From Bending Moment diagram, $BMv = -1.96Nm$

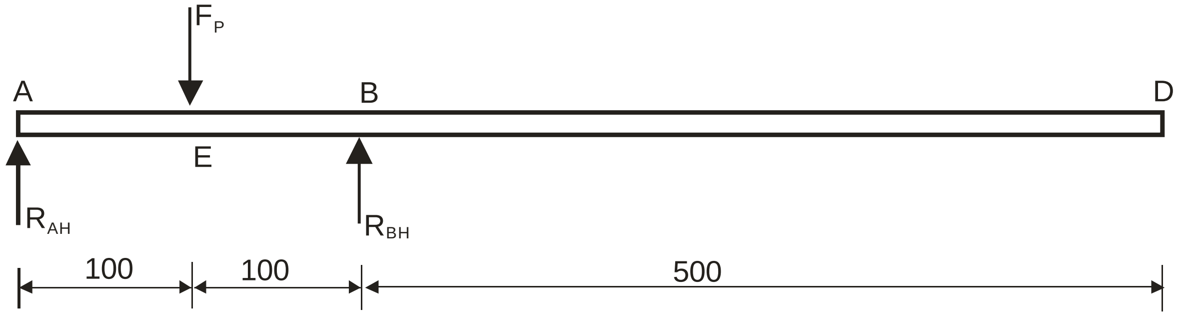

Figure 2. *Free body diagram of force distribution on shaft (horizontal axis)*

$$R_{AH} + R_{BH} = F_p \tag{22}$$

From tension in the belt

$$F_p = T_1 + T_2 \tag{23}$$

$F_p = 152.46N$

$R_{AH} + R_{BH} = 152.46$

Taking moment about point A

$$M_A = 0 \tag{24}$$

$0.1 \times F_p - 0.2 \times R_{BH} = 0$

$R_{BH} = R_{AH} = 6.23N$

From bending moment diagram, $BM_H = 7.62Nm$

The resultant maximum bending moment is given as

$$M = \sqrt{[(BM_V)^2 + (BM_H)^2]} \tag{25}$$

$M = 7.87Nm$

The equivalent twisting moment is given by

$$T_e = \sqrt{[(K_m M)^2 + (K_t T)^2]} \tag{26}$$

where $K_m = 1.5$, $K_t = 1.0$

$T_e = 26.51Nm$

Also,

$$d^3 = 16T_e/(\pi\tau) \tag{27}$$

The allowable shear stress τ for commercial shafting material is given as

$\tau = 84 \times 10^6 N/m^2$ (ASME CODE)

From equation 27, d = 11.71mm

Also equivalent bending moment due to both bending and torsional moment is given by

$$M_e = (K_m M + T_e)/2 \tag{28}$$

$M_e = 19.16Nm$

The allowable bending stress σ is given as $116\times10^6 N/m$

Shaft diameter from bending stress was determined by

$$d = [32Me/(\pi\sigma)]^{1/3} \tag{29}$$

d = 1.89mm

Since the shaft diameter due to bending stress is greater than that due to torsional stress, adopted value is 11.89mm. However, a standard shaft size of 25mm was chosen.

Design of injection screw

The shaft diameter which is 25mm is approximately equal to minor diameter of the screw. From the table of basic dimension for square threads for coarse series according to IS: 4694 – 1968 [14], d_c=24, d_0=34, P^l=10mm and h=5mm.

$$d_{is} = (d_o + d_c)/2 \tag{30}$$

where d, dc, do, P^1 and h = mean, core diameter (minor), nominal diameter (major), pitch and depth of thread, respectively.

$d_{is} = 29mm$

The helix angle α_h was determined as follows:

$$\tan\alpha_h = P^l/(\pi d_{is}) \tag{31}$$

where P^1=Screw pitch.

$\alpha_h = 6.26°$

Since the load is transported along the length of the screw shaft, it was therefore assumed to be conveyed in the forward direction. The effort required for this operation was obtained from equation 32.

$$P = W\tan(\alpha_h + Ø) \quad (32)$$

where P = effort required to convey the load, W = weight of the granule in injection chamber (load), Ø = friction angle.

The friction angle is related to coefficient of friction by

$$\mu = \tan Ø \quad (33)$$

For average quality material, average workmanship and running condition according to Khurmi and Gupta [14], $\mu = 0.13$.

Therefore, Ø = 7.41°

$$W = mg \quad (34)$$

m = 1000g = kg

$g = 9.81 m/s^2$

W =1×9.81 = 9.81N

Substituting the values of W, P, α_h and Ø into equation 32, we obtained P = 2.39N

Therefore, torque, T_s required is given by equation 35

$$T_s = Pd_{is}/2 \quad (35)$$

$$T_s = 0.035 Nm$$

Barrel design

In the design of barrel, it is required to calculate for the diameter of the barrel and its thickness, t.

$$t = r\{\sqrt{[(\sigma_t+P_b)/(\sigma_t - P_b)]} - 1\} \quad (36)$$

where t = thickness of shell in mm, r_1 = internal radius of the cylinder, r_1 = 18mm, P_b = intensity of internal pressure, $P_b = 100 N/mm^2$, σ_t = allowable stress for cylinder material, σ_t = 480MPa.

t = 4mm.

Bearing selection

Since the diameter of the shaft is 25mm, bearing with 25mm bore is required. From the table of principal dimension for radial ball bearing, bearing number 205 has the following parameters: Bore diameter, d = 25mm; Outside diameter, D = 52mm; Width, w = 15mm and Basic dynamic load capacity, C = 11kN

From the data obtained for the design of shaft, that is R_{AV} = -9.81N, R_{BV} = 29.43N, R_{AH} = 76.23N and R_{BH} = 6.23N, the resultant forces acting on the bearings were determined from equations 37 and 38.

$$R_A = \sqrt{[(R_{AV})^2 + (R_{AH})^2]} \tag{37}$$

$$R_B = \sqrt{[(R_{BV})^2 + (R_{BH})^2]} \tag{38}$$

Since the bearing 'B' has the greater resultant forces than 'A', bearing 'B' was used for the selection of both bearings with R_B = 81.71N.

The life expectancy under this load R_B = 81.71N was determined using equations 39 and 40.

$$L = (C/P)^K \times 10^6 \tag{39}$$

$$L = 60NL_H \tag{40}$$

where L = rated life in revolution, L_H = rated life in working hour, N = number of revolution of bearing, N = 300rpm, C = basic dynamic load rating, C = 11×10^3N, P = equivalent radial load rating, P = 81.71N, K = 3 for ball bearing.

Substituting the various values into equations 39 and 40 and solving simultaneously, we obtained $L_H = 135.54 \times 10^6$hrs.

Design of key

The width and the depth of the key were determined using equations 41 and 42.

$$w_k = d/6 \tag{41}$$

$$h_k = d/4 \tag{42}$$

Since diameter of shaft, d = 25mm then w_k = 4.2mm and h_k = 6.3mm.

Design of heater

The heat required, Q_{melt} to melt plastic pellets of mass m from their starting temperature for the plastics, T_p taken as 29.0° for this design, to the melt temperature, T_m, was estimated using equation 43.

$$Q_{melt} = C_p m(T_m - T_p) \tag{43}$$

where C_p = specific heat capacity of the plastic, C_p = 1470 J/(kg°C).

For this design, the range of melt temperatures is 180-260°C. Therefore, T_m = 220°C and Q_{melt} = 561.5kJ. A 2kW heater was chosen based on availability.

Mould design

The volume of the piece (flat plate) to be produced was determined by

$$V_c = \pi r_p^2 h_p \tag{44}$$

where r_p = radius of the plate (product), r_p = 40mm, h_p = thickness of the plate, h_p = 7mm.
$V_c = 3.52\times10^{-5}m^3$
The cooling time $t_{cooling}$ is given by Zhao *et al* [16] as

$$t_{cooling} = -h^2\{\ln[(T_e - T_a)/(T_m - T_a)]\}/(2\pi\alpha) \tag{45}$$

where T_m = 220°C (melt or injection temperature), $T_a = T_{mould}$ = 40°C (mould temperature), T_e = 50 °C (ejection temperature), $h = h_p$ = 7mm (average part thickness), α = 0.09mm²/s (thermal diffusivity).
$t_{cooling}$ = 4.2 min.

Design of cooling system

The cooling system comprises of a pump and a tank fixed at the lower end of the machine.

The tank is designed to be a rectangular shape. Area and volume of the tank is given by equations 46 and 47.

$$A = Lb \tag{46}$$

$$V = Lbh_t \tag{47}$$

where L = 300mm (length), b = 200mm (width), h_t = 400mm (height).
The volume of the tank was calculated to be 0.024m³.

According to Rajput [17], in selecting a pump, the power required to drive the pump and the discharge of the pump may be determined by

$$P = w_1QH/\eta \text{ (kW)} \tag{48}$$

where Q = rate of discharge, Q = 0.005m³/s, H = monometric head of water, H = 1.2m, w = weight of water, w = 9.81kN/m³, η = pump efficiency, η = 0.5.
P was determined as 11.77kW and the adopted power for pump is 0.5hp.

Nozzle design

A nozzle is a duct of smoothly varying cross sectional area in which a steadily flowing fluid can be made to accelerate along the duct. It is a hollow pipe threaded so that it can be screwed to the tapered end of the cylinder.

Therefore,

$$Q = C/t \tag{49}$$

$$Q = AV \tag{50}$$

$$A = \pi d_n^2/4 \tag{51}$$

where A = area of nozzle, d_n = diameter of nozzle, d_n = 10mm, C = screw design capacity, C = 1.60×10^{-4}m^3, V = velocity of plasticized material, t = time (sec), Q = volume flow rate (m^3/s)

From the design, the screw shaft has n = 50turns of thread, and the machine is designed for N_2 = 300rpm, therefore the time required for the material to be translated through the cylinder is given as

$$t_n = n/N \tag{52}$$

t_n = 0.167min (10s)

Time required to empty the cylinder is given as

$$t = n\times t_n \tag{53}$$

t = 500s

The volume flow rate, Q of the machine was calculated from equation 49 as $3.20\cdot10^{-7}$m^3/s, area of the nozzle was obtained as 7.85 $\times10^{-5}$m^2 while velocity was obtained as 0.00408m/s (4.08mm/s).

Construction and testing

The individual components of the machine were fabricated and assembled as shown in Figure 3 to get a complete functional unit of a portable laboratory injection moulding machine at a total cost of ₦67,390 ($406). The machine components fabricated include the frame, hopper, screw shaft, barrel, nozzle, mould unit and cooling system. The maximum size of the machine is 1210 × 800 ×1400mm for length, width and height, respectively.

After assembling, the machine was tested in two phases. In free rotation test, the machine was set in motion by application of torque from the motor to the shaft through the

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interconnection of the belt drive which rotates the screw shaft. In full load test, the heater was switched on to preheat the barrel to operating temperature of 220°C.

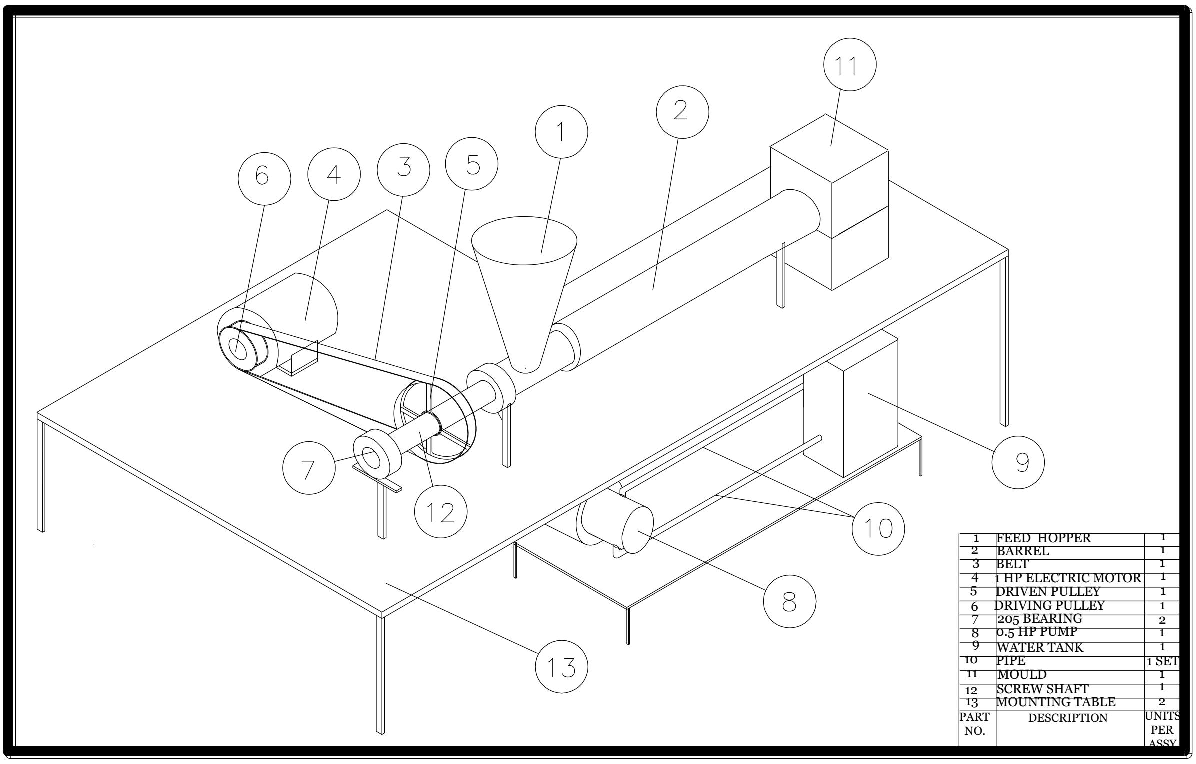

Figure 3. *Assembled injection moulding machine*

Thereafter, the machine was set in operation and fed with 2000g of plastic granules in the hopper. The plastic granules fed into the hopper were transported through the barrel where they were melted by the heat produced by the heating element. The rotational motion of the injection screw builds pressure at the inlet of the nozzle and forced the molten plastics into the mould through the outlet of the nozzle. Time taken for injection of the plastics into the mould and for the product to cool down to a temperature of 50° was recorded. Thereafter, the product (flat plate) along with the gating system was then removed from the mould after solidification and weighed.

Results and discussion

The results of performance test are given in Figure 4 and Table 1. Figure 4 shows the heating curve of the internal surface of the barrel while Table 1 gives the results of performance test. The result of free rotation test shows that the heater is capable of heating the barrel up to a temperature of 266°C making the machine suitable for processing of plastic materials such as Acrylonitrile butadiene styrene (ABS) and Polypropylene (PP) with melting temperature of about 250°C [18].

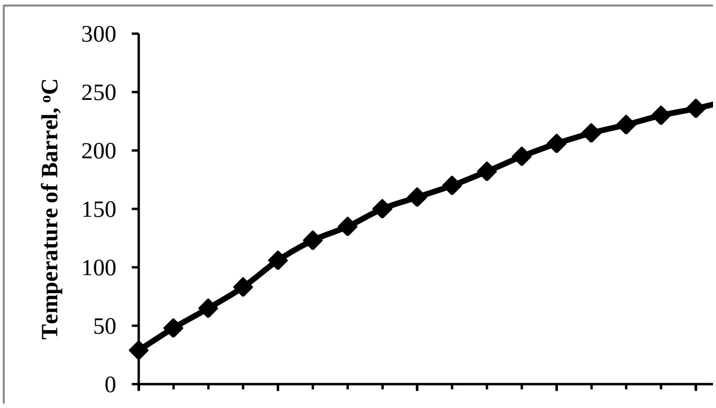

Figure 4. *Heating curve of internal barrel surface*

Table 1. Test results

S/N	Mass of plastics product (g)	Mould filling and cooling time (s)	Productivity (g/hr)
1	22.950	246	336
2	23.750	273	313
3	23.050	282	294
4	21.130	270	282
5	23.100	274	304
Mean	22.796	269	305

During the first cycle of the process of full load test, it took about 41.5 minutes for the electric heater element to heat the barrel to temperature of 180°C, sufficient enough to melt the plastic. Subsequent cycles took less heating time. The machine’s productivity is about 305g of plastics per hour.

The obtained productivity is low compared to those found in literature [19] for industrial machines. The low productivity is attributable to high heating time of the heater and high cooling time of the product. The achieved cooling time of about 4.2min is high compared to cycle time of 60 – 180s in literature [20].

With a maximum dimension of 1210 × 800 × 1400mm the machine can easily be accommodated in most small laboratories and hence it is portable. At the production cost of ₦67, 390.00 ($406), the machine is relatively cheap and affordable by most educational institutions. The machine will therefore be accessible by students and intending researchers who would like to do a further research study on this subject.

Conclusions

The designed and constructed potable plastic laboratory injection moulding machine has 305g/hr production capacity and has maximum dimensions of 1210 × 800 × 1400mm. It is suitable for conducting laboratory exercises in educational institutions. However, the performance can be improved through the use of higher capacity heater and cooling water pump.

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