DESIGN AND VALIDATION OF A SELF-COMPENSATING MELT REGULATOR FOR PLASTICS EXTRUSION

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Abstract

A design for a self-adjusting melt pressure regulator is presented for use between an extruder and one or more dies. The regulator is derived from a low force valve design that enables the outlet pressure to be directly regulated by a provided force on a valve pin without need for pressure sensors or a closed loop control system. Analytical and experimental results indicate an excellent level of response and consistency given the simplicity of the design.

Introduction

Extruders are highly efficient machines for plasticizing and pressurizing melt for subsequent forming through a die. However, the surging of the melt pressure due to the screw beat is a common issue that limits the consistency of extruded products. Similarly, the use of a single extruder to deliver flow to multiple dies has been limited. In response, practitioners are increasingly utilizing statistical analysis and optimization to improve the process performance [1, 2]. Alternatively, many practitioners are seeking to improve the inherent consistency of the process by incorporating additional components into the extrusion line design.

Gear pumps are positive displacement devices consisting of two intermeshing spur or helical gears in a housing with inlet and discharge ports. In many extrusion applications, gear pumps have been beneficially utilized to increase the consistency and pressure of the melt to the die. A literature review and industry survey both indicate that gear pumps are more frequently used to provide consistent melt pressure than a significant boost in melt pressure [3-12]. However, there are certainly specialty applications (e.g., thin films and fiber spinning) that require unusually high melt pressures for which gear pumps and other custom designs are utilized [13-15].

While gear pumps are viable components in many extrusion systems, there are issues related to purchase cost, reliability, and support [16]. Some alternatives to gear pumps have been proposed to improve the process consistency including a surge suppressor based on a dynamic seal [17-19], enlarged decompression chambers between the extruder and die, and improved screw designs [20-23]. Each of these alternatives has significant limitations related to size, performance, and/or their ability to be retrofitted to existing extrusion applications.

Previous research related to the dynamic control of injection molding led to the development of a system for dynamic actuation of multiple valve pins in a hot runner system, thereby achieving individual and dynamic control of the melt pressure at each gate [24-26]. Subsequent research has led to the development of a new valve design, shown in Figures 2 and 3 of [27], which requires negligible actuation forces to control at arbitrary melt pressures. A modification of this design is shown in Figure 1 of this paper. There are two important properties of this valve design. First, the resultant force on the valve pin is related predominantly to the pressure at the outlet which acts on the bottom surface of the valve pin. Second, the small mass of the valve pin provides for very fast reaction times in which the position of the valve pin can be adjusted to regulate the outlet pressure given fluctuations in the inlet pressure.

Analysis

The goal of the analysis is to develop and predict the performance of a self-regulating valve with minimal pressure drop, minimal shearing of the melt, minimal size, and fast response times. The primary design variables for another design of the regulator are identified in Figure 2 and include the inlet/outlet diameter, $2R$, the outer diameter, aR, the inner diameter, bR, the extension diameter, cR, the annulus length, dR, and the valve pin position, eR.

<u>Valve Sizing</u>: Flow channels in the valve must be designed such that the pressure drop is acceptable during operation. On the other hand, larger flow channel diameters are undesirable due to the increase in the valve size and valve pin inertia. Given a Newtonian fluid of viscosity η flowing at a rate, Q, through a pipe of length fR, the pressure drop is [28]:

$$\Delta P_{pipe} = \frac{8\eta fRQ}{\pi R^4} \tag{1}$$

There will also be a pressure drop through the annulus of the valve [28]:

$$\Delta P_{annulus} = \frac{8\eta dRQ}{\pi (aR)^4}\left[\left(1-\left(b/a\right)^4\right)-\frac{\left(1-\left(b/a\right)^2\right)^2}{\ln\left(a/b\right)}\right]^{-1} \tag{2}$$

In the design of the valve, allow the pressure drop to be equally distributed across the inlet, annulus, and outlet sections. Assuming for now that f, a, and d equal 2 with b/a equal to 0.5, then the nominal dimension, R, can be derived for a given viscosity, flow rate, and pressure drop. Figure 3 shows the relationship between the size and pressure drop at varying flow rates for a melt viscosity of 400 Pa·s. For this paper, consider an application requiring a flow rate of 100 kg/hour (27 cc/s) and a nominal pressure drop of 0.85 MPa

(123 psi) for which a nominal dimension, *R,* of 5.5 mm is selected.

Valve Pin Design: Polymer melts are viscous in nature, so there will be shear stresses which would tend to pull the valve pin in the direction of flow, as well as a related pressure differential that would tend to push the valve opposite the direction of flow. Since these forces counteract, it is possible to design the valve pin such that the forces resulting from polymer flow through the valve are small during operation. Accordingly, the force on the valve pin due to the pressure drop along the length of the orifice is estimated as:

$$F_{\Delta P} = \Delta P_{annulus} \pi R^2 \left(a^2 - b^2\right) \tag{3}$$

where $\Delta P_{annulus}$ is provided by eq. 2. As the polymer melt flows along the valve pin, a shear stress, τ, will be exerted which is approximately:

$$\tau = \frac{\Delta P_{annulus}}{2d}\left[b - \frac{\left(a^2 - b^2\right)}{2b\ln(a/b)}\right] \tag{4}$$

The resulting force on the valve pin due to this shear stress is estimated as:

$$F_\tau = \pi dbR^2\tau = \Delta P_{annulus}\pi R^2 \frac{b}{2}\left[b - \frac{\left(a^2 - b^2\right)}{2b\ln(a/b)}\right] \tag{5}$$

By comparing the forces due to the pressures and shear stresses acting on the valve pin, the valve pin may be designed to minimize the actuation force and sensitivity to fluctuations in viscosity and flow rate. Inspection of these equations indicates that the pressure forces predominate, so it is desirable to minimize the pressure drop. Continuing the example, Figure 4 shows the estimated forces acting on a valve pin as a function on inner to outer diameter, *b*/*a*. It is observed that the forces due to the pressure differential across the valve increase more quickly with *b/a* than the forces due to the shear stress on the valve pin. In particular, it is observed that the forces increase precipitously for values *b/a* greater than 0.5, so this value is selected.

Juncture Loss: Depending on the valve pin position, the pressure drop across the valve will vary due to the restriction of flow between the valve pin and the outlet. The pressure drop across this juncture is modeled as:

$$\Delta P_{juncture} = \frac{8\eta Q}{\pi a (eR)^k} \tag{6}$$

where *eR* is the valve pin position and *k* is an empirically derived index typically on the order of 2 for hydraulic fluids [29] and the subject of ongoing research. Current experience indicates that a value of *k* equal to 3 approximates the observed behavior and that the juncture loss becomes negligible for valve pin positions, *e*, greater than 0.5. As such, this value is selected to limit the valve pin travel. To scale, the final system design is shown in Figure 5.

Dynamics: The net force on the valve pin, *F*, is a function of the forces on the valve pin due to the pressure drop and shear stresses discussed above, as well as the control forces acting on the valve pin, $F_{control}$, and the force acting at the end of the valve pin, F_{outlet}:

$$F = F_{control} + F_{\Delta P} + F_\tau + F_{outlet} \tag{7}$$

where the force on the end of the valve pin is defined by the pressure at the outlet and the end diameter as:

$$F_{outlet} = \Delta P_{juncture} \pi R^2 \left(b^2 - c^2\right) \tag{8}$$

Provided that these forces do not balance, the valve pin will tend to move according to the equation [30]:

$$F = m\frac{d^2 eR}{dt^2} \tag{9}$$

where *m* represents the mass of the valve pin and connected transmission mechanisms.

Results

Since the position of the valve pin changes according to the foregoing equations, a constant control force to regulate the outlet pressure regardless of variations in the inlet pressure. There are several different concepts for providing a control force to regulate the outlet pressure. One very simple design is to use a pneumatic cylinder with a supply pressure related to the outlet melt pressure. Another simple alternative is to use a spring that is pre-loaded to correspond to the outlet pressure. Alternatively, a closed loop approach can be provided to vary the control force or valve pin position in response to the measured outlet pressure.

While closed loop control strategies are possible, it is an objective of this research to provide a low cost design that does not require any pressure sensors or active control. In fact, it is believed that the described design, which is inherently self- regulating, can provide better control in an open loop mode than a closed loop control strategy which is subject to errors related to pressure sensor calibration, noise, and response as well as additional issues related to the controller stability and response. As such, the performance of the described design is predicted and validated according to the control forces provided by a spring and pneumatic cylinder.

In this design, a pneumatic cylinder with a push area, *A*, provides a control force corresponding to the pressure at the outlet. The specified supply pressure, P_{supply}, can be calculated from the desired outlet pressure as:

$$P_{supply} = P_{outlet}^{desired} \pi R^2 \left(b^2 - c^2\right) / A \tag{10}$$

Given a constant supply pressure, the control force will be constant regardless of the valve pin position. For *c* equal to zero (no valve pin extension), 10 MPa melt pressure would correspond to a control force of 950 N.

Continuing the example, the system of equations (1-9) was solved by numerical simulation. Accordingly, the desired outlet pressure was selected as 10MPa, with the inlet pressure varying according to the function:

$$P_{inlet} = 15 + 2\sin(2\pi\omega t) \tag{12}$$

where *t* is measured in seconds and ω is selected as 4 s^{-1} to correspond to a screw beating at a fast 240 RPM. The mass of the valve pin and pneumatic piston was estimated as 0.5 kg. The results are shown in Figures 6 and 7. Figure 6 shows the melt pressure at the inlet varying as a sinusoid with the screw beat. Starting with the valve pin in a fully open position, there is only a small pressure drop through the valve so the outlet pressure is above the desired 10 MPa. Accordingly, the force due to the pressure acting on the valve pin at the outlet exceeds the control force, so the valve pin begins to close as shown in Figure 7. After approximately 3 ms, the valve pin moves to a state where the forces balance and the outlet pressure approaches the desired 10 MPa. Subsequent cycling of the inlet pressure does not significantly affect the outlet pressure since the valve pin is constantly self-regulating its position and the related juncture loss. The average absolute error in steady state operation between the outlet pressure and the desired pressure set-point of 10 MPa is 0.066 MPa (10 psi) even with a 4 MPa (580 psi) oscillation in the inlet pressure due to the screw beat.

To validate the performance of the valve design, the regulator was built and experiments were conducted on a hydraulic molding machine to precisely control the inlet pressure. Figures 8 and Figure 9 plot the outlet pressure for pulsed inlet pressures of 7 and 10 MPa, respectively, with a control force of 200N supplied by a pneumatic cylinder with a constant air pressure. It is observed that while the inlet pressures varied widely, there was no significant change in the outlet pressure. Figure 10 shows that an increase in the control force to 400 N produced an increase in the outlet pressure with no change in the inlet pressure.

Discussion

The analytical models and experimental data have shown the feasibility of the self-compensating melt regulator. The established level of control is remarkable given the simple system operating in an open loop fashion with a constant pneumatic supply pressure. While closed loop control is certainly possible, such an open loop approach is preferable in practice given their lower cost and greatly improved reliability.

It is observed that there are disparities between the experimental and analytical results. In particular, the experimental system has a slower response and greater variance than the analytical model. It is believed that these disparities may be due to several factors including 1) inaccurate inertia estimates of the piston in the cylinder, 2) unaccounted sliding friction of the valve pin and piston, and 3) improper modeling of the melt dynamics, junction losses, and non-Newtonian effects. Further numerical research is on-going to model the non-Newtonian, compressible flow through the valve including acceleration effects. Further empirical research will also acquire the inlet pressure and valve pin position at a variety of conditions. These additional insights will enable the rapid development of optimal valve designs.

Conclusions

A self-regulating valve was presented which can automatically compensate for fluctuations in inlet pressure to provide very consistent outlet conditions. The self-regulating valve is enabled by a new valve design, which has very low actuation forces and thereby allows a balance between a control force and exerted force from the melt pressure. With its simple design and inherent stability, it is hoped that these devices will be adopted across the industry.

Key Words

Melt regulator; gear pumps; plastics processing; extrusion.

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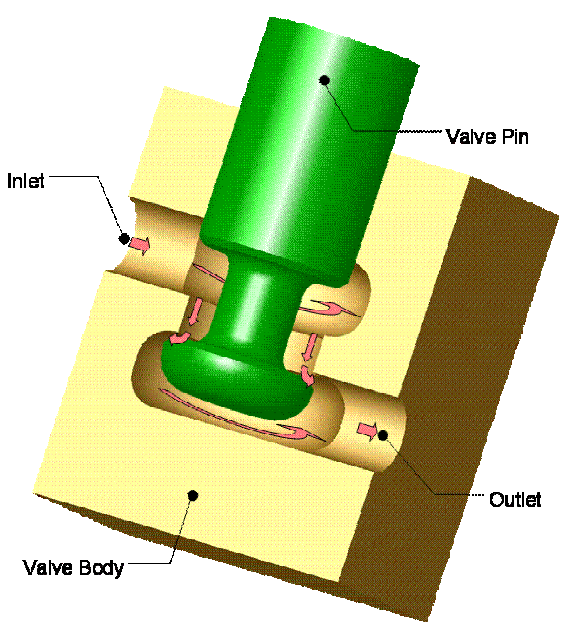

Figure 1: Melt regulator design

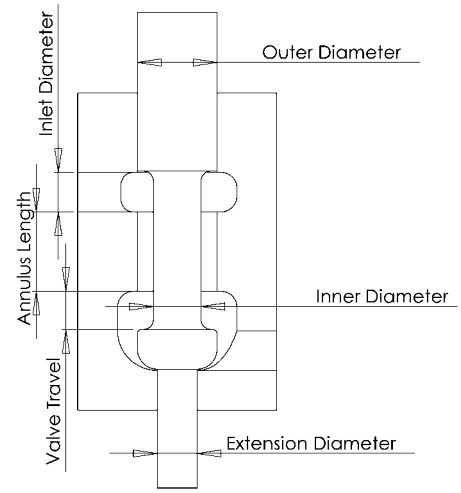

Figure 2: Valve design parameters

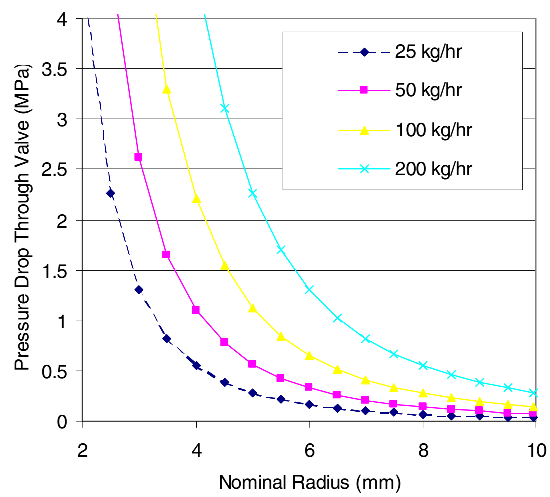

Figure 3: Pressure drop as a function of size and flow rate

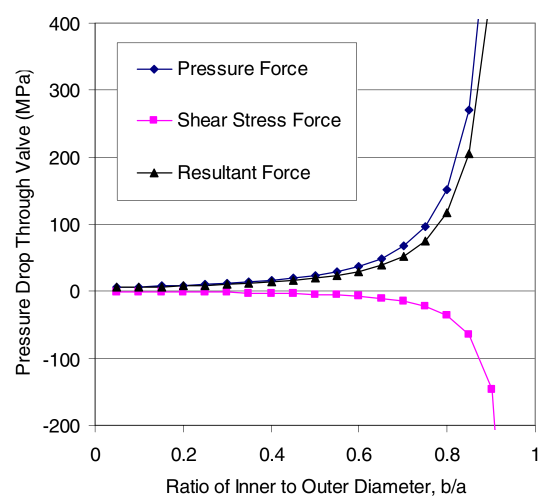

Figure 4: Forces as a function of the ratio b/a

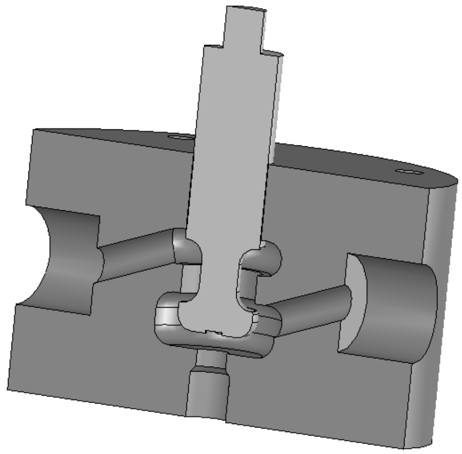

Figure 5: Implemented Design

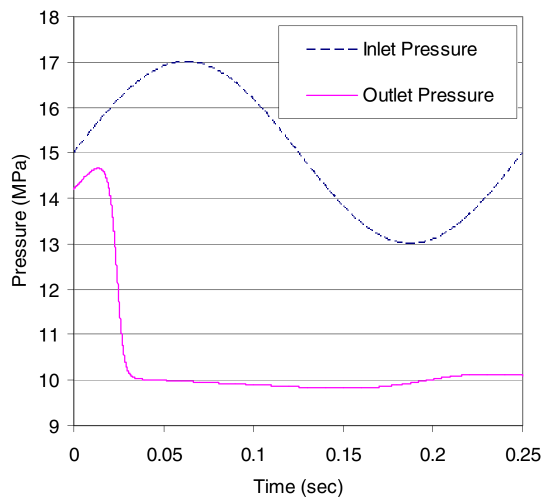

Figure 6: Inlet and outlet pressures with pneumatic control

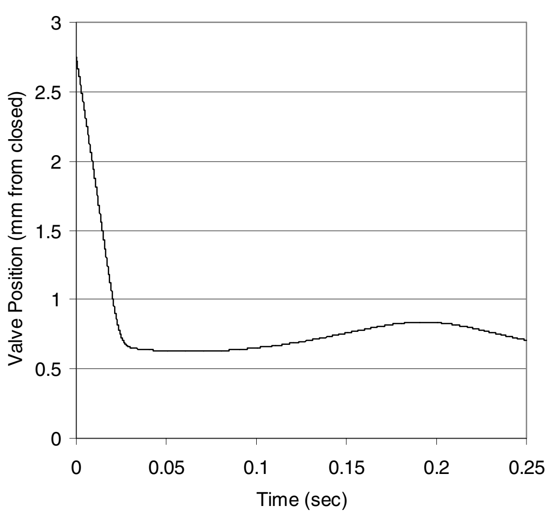

Figure 7: Valve pin position with pneumatic control

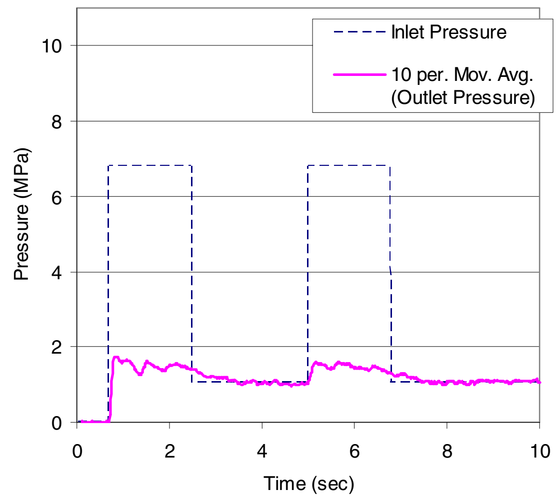

Figure 8: Outlet pressure (P_{inlet}=7Mpa, $F_{Control}$=200N)

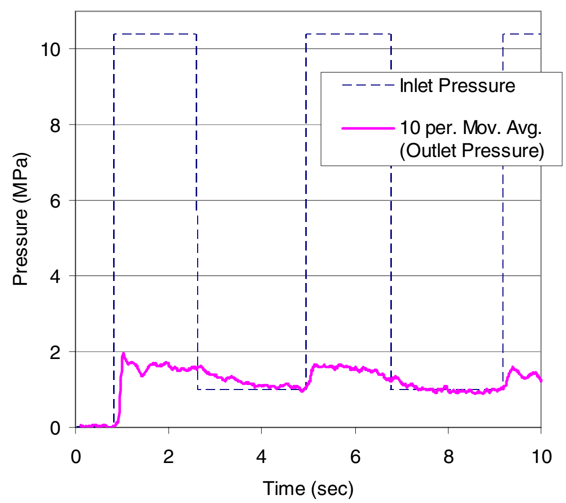

Figure 9: Outlet pressure (P_{inlet}=10Mpa, $F_{Control}$=200N)

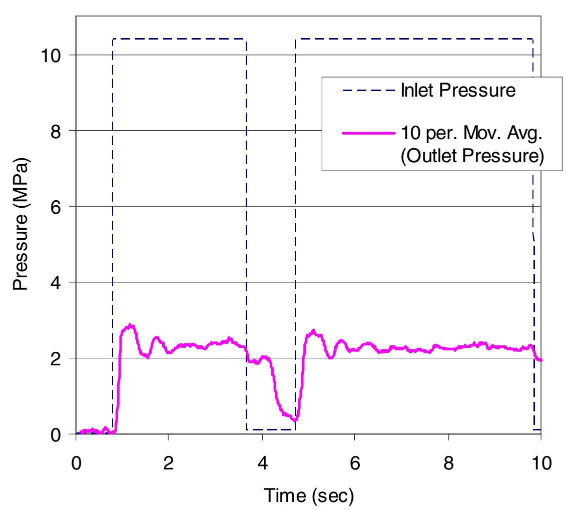

Figure 10: Outlet pressure (P_{inlet}=10Mpa, $F_{Control}$=400N)