ANALYSIS OF NEW COOLING SCREW

Chris Rauwendaal, Rauwendaal Extrusion Engineering, Inc.

Abstract

This paper presents a theoretical approach to the prediction of melt temperature profiles in cooling extruders. The effect of screw design on cooling performance in discussed. A new screw geometry is presented will substantially improved cooling capability. Initial performance data on a 200-mm cooling extruder are presented.

Introduction

Heat transfer is a critical issue in most polymer extrusion operations. In plasticating extrusion the objective is to add the right amount of heat to melt the polymer and to achieve the desired melt temperature. In some extrusion operations, however, the objective is to remove heat from the polymer. This is the case in tandem foam extrusion lines where the secondary extruder is used to cool down the mixture of polymer melt and blowing agent. Cooling extruders reduce the polymer melt temperature by a substantial amount, about 100°C, to achieve a melt consistency that is conducive for foaming

Little information has been published about the design of cooling screws in the open literature. This paper discusses basic issues related to the design of cooling screws and presents a new screw geometry to achieve improved cooling. As the foam extrusion industry faces pressure to move from HCFC (hydrogenated chlorofluorocarbon) blowing agents to nitrogen and carbon-dioxide (CO_2) the cooling capacity becomes more critical. CO_2 is less of a viscosity depressant than most HCFC blowing agents. As a result, with CO_2 more viscous heating occurs in the cooling extruder and more effective cooling is required to achieve the same reduction in melt temperature.

Standard Cooling Screws

Cooling screws have to be designed to remove heat efficiently from the gas-laden melt (GLM) while, at the same time, the viscous heat generation in the GLM has to be as low as possible. Generally, cooling screws have a large diameter (about 25% larger than the primary extruder), multiple flights, large helix angle, and deep channels. Cooling screws operate at low screw speed to minimize viscous dissipation. Figure 1 shows a typical cooling screw.

Axial Melt Temperature Profiles

The viscous heating is determined by the product of the melt viscosity (h) and shear rate ($\dot{g}$) squared. The shear rate can be approximated by the circumferential velocity divided by the channel depth of the screw. For a power law fluid with consistency index m and power law index n the viscous heating per unit volume (q_v) can be expressed as:

$$q_v = m\left(\frac{pDN}{H}\right)^{n+1} \qquad (1)$$

Variable D represents the diameter, N screw speed, and H the channel depth. A low screw speed (N) and a large channel depth (H) are beneficial in keeping the viscous dissipation low. Further, low values of the consistency index and power law index will result in low viscous dissipation. The consistency index is largely determined by the polymer; it also depends on temperature and the type and amount of blowing agent.

The power consumption (Z) is obtained from the product of q_v and the volume of the polymer melt. If the volume is approximated by $pDHL$ the power consumption becomes:

$$Z = \frac{m_r \exp[a(T_r - T)]\, L(pD)^{n+2} N^{n+1}}{H^n} \qquad (2)$$

The consistency index is made temperature dependent using an exponential dependence of temperature with a temperature coefficient of a. The consistency index m_r is the value at reference temperature T_r.

For realistic determination of melt temperatures we have to consider both viscous dissipation and conductive heat transfer through the barrel. When the screw is cooled we have to consider heat transfer through the screw as well. If the conductive heat transfer is constant, the temperature gradient can be expressed as:

$$\frac{dT}{dx} = B_1 e^{a(T_r - T)} - B_2 \qquad (3)$$

B_1 represents the contribution of viscous heating.

$$B_1 = \frac{m_r (pD)^{n+2} N^{n+1}}{H^n C_p \dot{M}} \quad (4)$$

where C_p is the specific heat and $\dot{M}$ the mass flow rate.

B_2 represents the contribution of conductive heat transfer.

$$B_2 = \frac{q_c pD}{C_p \dot{M}} \quad (5)$$

The units of B_1 and B_2 are [°C/m]; these are units of temperature gradient. Variable q_c is the heat flux through the barrel wall. Subject to boundary condition $T(x=0) = T_0$ the differential equation can be solved. The solution can be written as:

$$T(x) = \frac{1}{a} \ln\left[\left(e^{aT_0} - \frac{B_1}{B_2} e^{aT_r}\right) e^{-aB_2 x} + \frac{B_1}{B_2} e^{aT_r}\right] \quad (6)$$

The melt temperature is independent of distance when the conductive heat transfer equals the viscous dissipation. This limiting heat transfer q_{c0} can be expressed as:

$$q_{c0} = \frac{m_r (pDN)^{n+1} \exp[a(T_r - T)]}{H^n} \quad (7)$$

When $q_c > q_{c0}$ the melt temperatures will reduce with axial distance; when $q_c < q_{c0}$ the melt temperature will increase with axial distance. Obviously, in cooling extruders the actual heat transfer has to be greater than the limiting heat transfer. It is important to note that the limiting heat transfer is dependent on the actual melt temperature. As the melt is cooled along the extruder the effective viscosity will increase as the melt temperature is lowered. This means that the viscous dissipation will increase as the melt temperature reduces. As a result, the cooling will become less efficient as the melt progresses along the extruder. Therefore, increasing the length of the extruder does not necessarily improve the cooling capacity.

Expression 6 is valid for situations where the heat transfer is constant. If the barrel temperature is maintained at constant temperature the heat transfer rate will change as the melt cools down. We can analyze this situation by analyzing small length increments and adjusting the heat transfer rate at the start of each new increment. Figure 2 shows the axial temperature profile for a 200-mm cooling screw for six screw speeds, 3, 6, 12, 18, 24, and 30 rev/min. The barrel temperature is maintained at 100°C at a specified distance from the barrel internal diameter. The inlet temperature of the melt is 225°C.

At the start of the cooling process the melt temperature reduces quickly; however, the rate of cooling reduces along the length of the extruder. This is due to a reduced temperature gradient in the barrel and an increased level of viscous dissipation as the melt cools down. The effect of viscous dissipation is clearly shown by the increase in melt temperature with screw speed. Figure 2 clearly shows the benefit of operating the cooling extruder at low screw speed.

Cross Sectional Melt Temperature Distribution

The expressions developed describe the axial melt temperature profile as long as the heat flux through the melt equals the heat flux through the barrel wall. The expressions are essentially based on a finite volume approach. In order to the determine whether the heat flux through the melt is high enough to achieve efficient cooling we have to perform a 3D non-isothermal flow analysis to determine the cross section melt temperature distribution.

One of the main challenges in cooling is the low thermal conductivity of the melt. As a result, the cooling at the barrel surface affects only a relatively thin melt layer. This means that the outer recirculating melt layer is cooled effectively. However, the inner recirculating region is insulated from the barrel surface by a thick melt layer and the temperature in this region tends to be substantially higher than the barrel temperature. The insulated inner melt region leads to inefficient cooling particularly in screws with large channel depth.

Earlier studies (1, 2) on melt temperature distribution in extruder screws have found that high melt temperatures in the inner recirculating region are inherent in screw extruders. Figure 3 shows the temperature distribution in a 60-mm extruder screw running at 20 rpm with a fractional melt (MI=0.2) HDPE. This figure indicates that non-uniform cooling can result in highly non-uniform melt temperatures.

Figure 3 shows that the melt in the outer region of the channel is relatively cool while the melt in the center region is relatively hot. The inner recirculating region is insulated from the screw and barrel surface. As a result, heat removal from this region is very ineffective and this results in high melt temperatures in this region.

Improved Cooling by Screw Design

In order to improve cooling it is necessary to move melt from the inner region to the outer region. In the past this was done by machining slots in the flights of the screw; a large number of slotted flight geometries have been used (3, 4). However, slots generally do not achieve a very effective redistribution of the melt. Fogarty (5, 6) developed a screw with windows in the flights; this screw is called the Turbo screw. The windows are relatively large and allow melt to transfer from one channel to an adjacent channel improving heat transfer. Fogarty et al. (7) studied the mixing characteristics using the boundary element method and the heat transfer characteristics (8) using a finite volume method.

A recent development is the HHT (high heat transfer) screw by Rauwendaal (9, 10). This screw is designed to achieve an effective exchange of material from the inner region to the outer region and vice versa. The exchange is achieved by starting a new flight in the middle of the channel. The new flight splits the hot region; at the trailing side of the flight the hot region moves to the barrel surface while at pushing side of the flight the hot material moves to the screw surface. The net effect of the introduction of the new flight is that hot material in the inner region is forced to the outside and, at the same time, cold material from the outer region is forced to the inside. This is illustrated in figures 4, 5, and 6. Figure 4 shows the melt temperature distribution in a conventional screw. Figure 5 shows the change in melt temperature distribution when a new flight is introduced in the center of the channel. Figure 6 shows the melt temperature distribution after introduction of the new flight.

Figures 4-6 illustrate how the melt from the inner region is forced to the outside and the melt from the outside region to the inside. For that reason this screw is called the inside-out screw or HHT screw.

New Screw Performance Characteristics

The new screw was first applied to a tandem foam extrusion line for PS foam board, 60x700mm. The polymer was a blend of several melt index resins, 20, 30, and 2.5, with a small amount of talc, process aid, and flame retardant. The blowing agent was a mixture of two HCFCs with a small amount of alcohol added as a dispersion aid. The cooling extruder is a 200-mm extruder with a length to diameter ratio of 31:1. The HHT screw replaced a commercial cooling screw from Battenfeld. The throughput was 600 kg/hr and the screw speed was 9 rpm. The cooling capacity with the new screw improved 25% compared to the old screw. The product expansion was very uniform and significant better than with old screw. The uniform expansion is most likely due to the more uniform temperature distribution within the material. Figure 7 shows a schematic of the geometry of the new screw.

Conclusions

The effectiveness cooling extruders is limited by the fact that the melt in the center region of the channel is insulated from the barrel surface. Cooling can be improved significantly by using a screw geometry that achieves effective mass transfer from the center region to the outside region and vice versa.

A new screw geometry has been developed which forces high temperature melt in the center region of the channel to the barrel surface. This new screw has been used in polystyrene foam extrusion to improve the cooling capacity of the secondary extruder. The HHT screw improved the cooling capacity by 25% relative to the existing screw.

References

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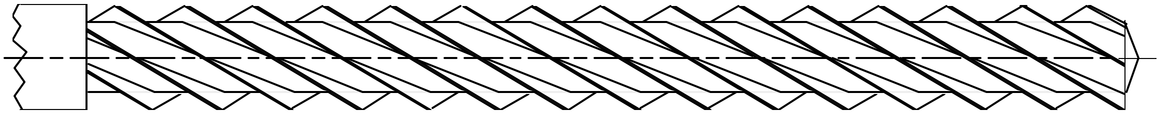

Figure 1, Standard cooling screw

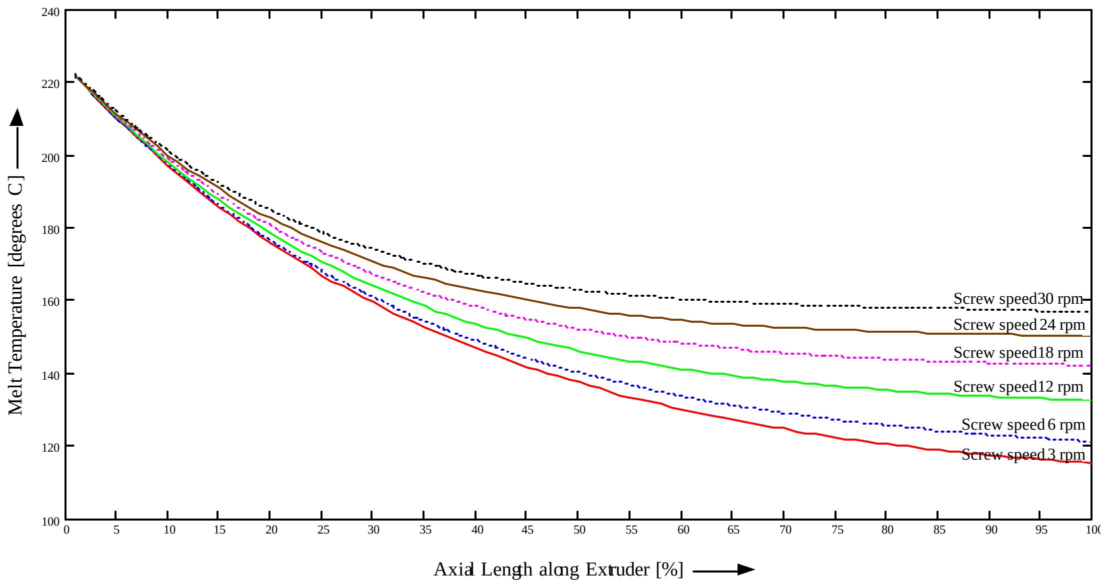

Figure 2, Axial melt temperature profile for 200-mm cooling screw at different screw speeds

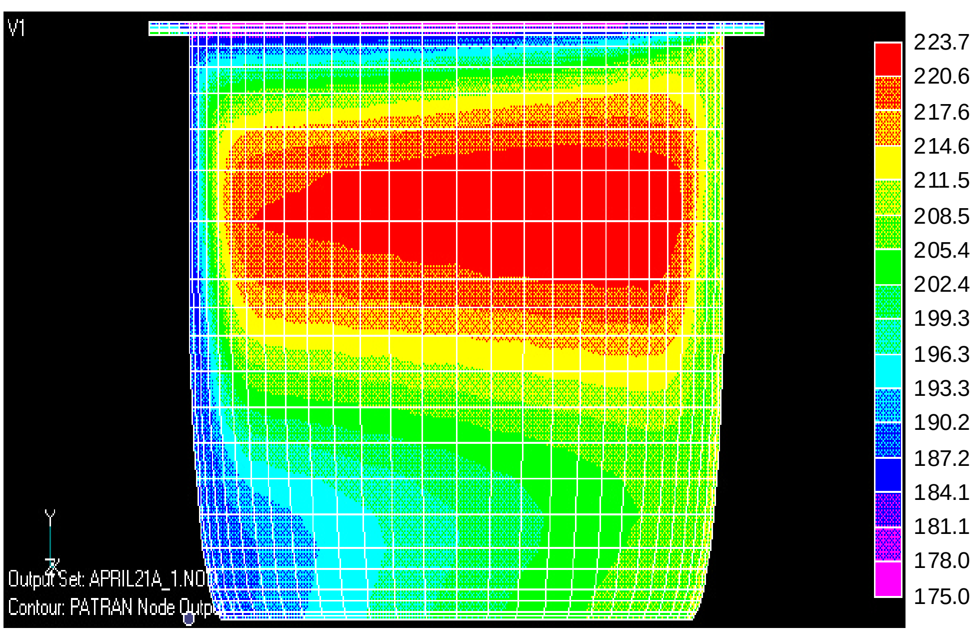

Figure 3, Melt temperature distribution in 60-mm extruder running a fractional melt HDPE

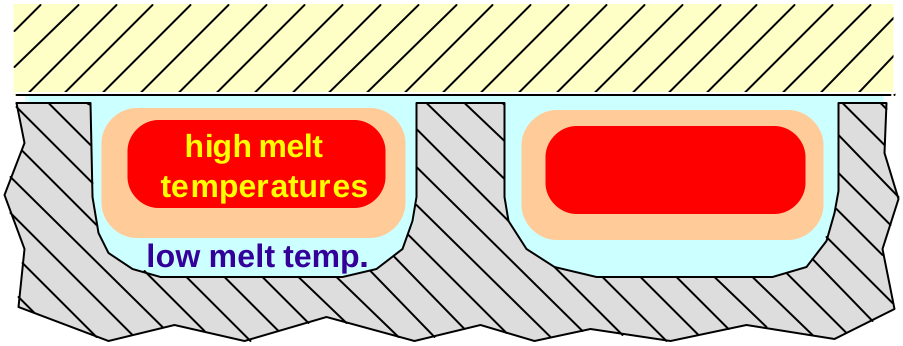

Figure 4, Melt temperature distribution in a conventional screw

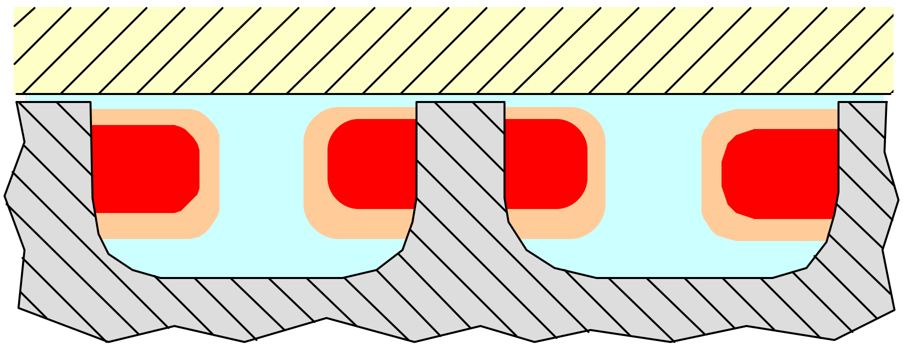

Figure 5, Melt temperature distribution at the point where new flight is introduced

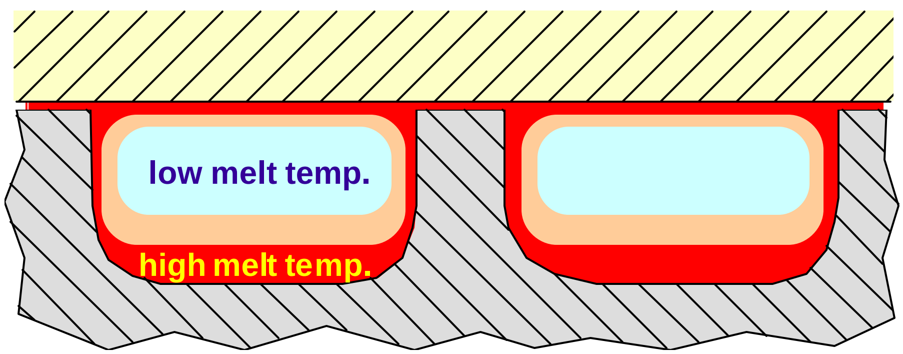

Figure 6, Melt temperature distribution after the new flight is introduced

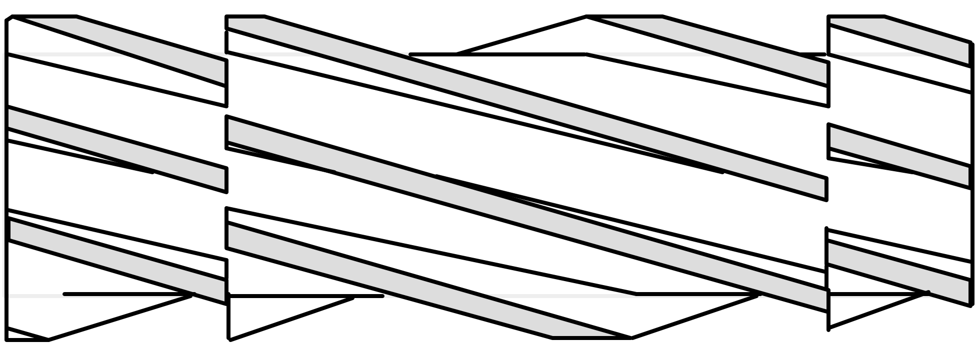

Figure 7, Schematic of HHT screw geometry