A STUDY ON THE THERMAL PERFORMANCE OF CALIBRATORS

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Abstract

In this work a numerical code able to model the heat transfer in calibrators for extruded profiles is described and validated. For assessment purposes the numerical predictions are compared with analytical models and numerical results obtained with a commercial software. The routines developed are then used to identify the main process parameters and to estimate their relative importance.

Introduction

A typical extrusion line for the production of thermoplastic profiles generally comprises an extruder, a die, a calibration/cooling system, a haul-off unit and a saw. The calibration/cooling system is responsible for the establishment of the final most relevant dimensions of the profile while cooling it down until a temperature that guarantees its shape along downstream stages [1,2]. The viscoelastic behaviour of the melt upstream, together with the expected slight variations of the operating conditions/polymer rheological properties, make it very difficult to produce the required melt extrudate cross-section. Moreover, as the profile progresses along the line is subjected to a variety of external forces (such as friction, buoyancy and compression), being necessary to guarantee that it is strong enough to withstand these forces without deforming.

From a thermal point of view, the calibration/cooling system should also ensure fast rate uniform cooling of the extrudate, in order to induce the adequate morphology and a reduced level of thermal residual stresses [3,4]. In practical terms, the temperature gradient along the profile contour and along its thickness must be minimized [3] and its average temperature at the calibration/cooling system outlet must fall bellow the solidification temperature (Ts), in order to avoid subsequent melting [1]. Furthermore, at every position, a of layer material must be cooled down to a temperature below Ts to ensure sufficient rigidity to support the imposed solicitations [1].

Calibration may be performed by internal pressure or external vacuum and be wet and/or dry [5,6], i.e., allowing or not direct contact between the cooling medium (generally, water) and the hot profile, respectively. In the particular case of profiles, the most common calibration process is the vacuum system using dry cooling or a combination of dry and wet cooling [5].

In this work a 3D code, based on the finite-volume method, developed to model the thermal interchanges occurring during the calibration/cooling stage of profile extrusion is described, validated and used with the view to designing calibration systems. This simulation software is part of a future optimisation code aiming the optimisation of the thermal performance of calibration systems. The motivation for the development of an own simulation software was the need of swift calculations, requisite particularly critical when optimisation algorithms are used. The finite-volume method is not so demanding in terms computational resources as methods based on finite-elements [7], being, therefore, the most adequate technique for this purpose. The need to control the simulation software during the optimisation process also justifies its development.

Process Modelling

The large number of parameters involved in the process, together with the geometrical complexity of some profiles, requires the use of numerical methods to determine the time evolution of the temperature fields. The first attempts to model the cooling of plastic profiles or pipes, made in the seventies and eighties (see, for example, [8-10]), were based in 1D models, being only adequate to characterize the cooling stage under ideal circumstances, i.e., uniform cooling conditions for constant thickness extrudates. Therefore, these models are only expected to generate accurate results when applied to large pipes or sheet cooling. *Menges et al.* developed a 2D FEM approach [11], sufficiently general to deal with any extrudate cross-section. However, this model is not adequate to emulate situations where axial heat fluxes are significant, being later extended by *Sheehy et al.* [12] using the so-called Corrected Slice Model (CMS). This is a hybrid 2D model that can cope with the three dimensionality introduced by the axial heat fluxes. There are also other works dealing with 2D extrudate cooling modelling, namely those including more realistic boundary conditions for the heat exchange occurring in the internal cavities of hollow profiles [13,14] and those predicting sag flow of thick wall pipes [15-17].

One major difficulty to face when modelling the cooling step is the selection of the proper heat transfer

coefficient, h, accounting for the heat transfer between the plastic profile surface and the cooling medium, i.e., calibrator internal walls, water or air. This coefficient is, therefore, a measure of the contact resistance. It was experimentally shown that the heat transfer coefficient might vary between 10 and 10000 W/m2K [18], depending on the location along the calibration system. Other researchers [19] use their practical experience, essentially based on the observation of the calibrator's wear pattern, to select different constant h values for each profile wall, depending on the effectiveness of the contact between the profile and the calibrator. Due to the great amount of factors affecting these coefficients and the lack of information available, it is a common practice to determine their values solving the inverse problem. Hence, knowing the real temperature fields (experimentally determined) simulations are performed in order to adjust the effective heat transfer coefficient values (see, for example, [15,20]).

Model Assessment

Direct confrontation between predictions and experimental data is difficult, since the practical measurement of the evolution of the extrudate's cross-section temperature profile along the calibrating /cooling system is cumbersome, requiring the use of thermocouples imbedded at different depths of the profile thickness, and thus moving with the profile along the line [15]. Profile surface temperatures between two consecutive calibrators are easier to monitor, but the quality of the measurements depends on the emissivity settings used in the non-contact infrared thermometers that are generally employed and also on the measuring depth, i.e., the thickness effectively reached by the radiation from the sensor [21]. However, most of the temperature measurements reported concern pipes [1,15-17,20] rather than profiles (and even in this case the data presented is insufficient for modelling purposes) [18].

Therefore, the model developed and presented in this work will be assessed confronting its predictions: i) with the analytical results derived for a simple geometry, ii) with the calculations provided by a general purpose finite element software (Polyflow [22]).

Analytical Solution

The case study considered is illustrated in Figure 1. It consists of two rectangular slabs, S1 and S2, with thermal conductivities k1 and k2, respectively, under contact through one of their faces. As shown also in the figure, the temperature is imposed on the remaining faces. The temperature distribution in each slab can be obtained from [23]:

$$T = T_{b1} + \sum_{n=1}^{\infty}\left[\frac{2}{\pi}\left(T_{b2}-T_{b1}\right)\frac{(-1)^{n+1}+1}{n} \frac{-k_2}{\left(k_2+k_1\right)}\sin\left(\frac{n\pi x}{W}\right)\sinh\left(\frac{n\pi y}{W}\right)\right] \tag{1}$$

for S1, and from

$$T = T_{b2} + \sum_{n=1}^{\infty}\left[\frac{2}{\pi}\left(T_{b2}-T_{b1}\right)\frac{(-1)^{n+1}+1}{n} \frac{k_1}{\left(k_2+k_1\right)}\sin\left(\frac{n\pi x}{W}\right)\sinh\left(\frac{n\pi(-y+2H)}{W}\right)\right] \tag{2}$$

for S2, for perfect contact case, and by:

$$T = T_{b1} + \sum_{n=1}^{\infty}\left[\frac{2}{\pi}h\left(T_{b1}-T_{b2}\right)\frac{(-1)^{n+1}+1}{n} \frac{1}{-k_1\dfrac{n\pi}{W}\cosh\left(\dfrac{n\pi H}{W}\right)-\left(\dfrac{k_2+k_1}{k_2}\right)\sinh\left(\dfrac{n\pi H}{W}\right)} \sin\left(\frac{n\pi x}{W}\right)\sinh\left(\frac{n\pi y}{W}\right)\right] \tag{3}$$

for S1 and from

$$T = T_{b1} + \sum_{n=1}^{\infty}\left[\frac{2}{\pi}h\left(T_{b1}-T_{b2}\right)\frac{(-1)^{n+1}+1}{n} \frac{1}{k_2\dfrac{n\pi}{W}\cosh\left(\dfrac{n\pi H}{W}\right)+\left(\dfrac{k_2+k_1}{k_1}\right)\sinh\left(\dfrac{n\pi H}{W}\right)} \sin\left(\frac{n\pi x}{W}\right)\sinh\left(\frac{n\pi(-y+2H)}{W}\right)\right] \tag{4}$$

for S2, when the interface is modelled with a contact resistance boundary condition.

The temperature distributions defined by Equations 1 to 4, were compared with those obtained with the numerical routines developed, using (see Figure 1) W=100 mm, H=50 mm, T_{b1}=100 ºC, T_{b2}=180 ºC, k_1=7 W/mK, k_2=14 W/mK and, for the case of contact resistance, h=500 W/m^{2}K. A comparison between the analytical and numerical results obtained is illustrated in Figure 2. As shown, the two sets of results are coincident.

Complex Layout

The predictions of the numerical routines developed were also compared with the results obtained by a commercial software Polyflow [22], for a problem that consists in determining the temperature distribution of a polymeric sheet (2 mm thick) that is moving at 0.01 m/s while being cooled by a calibrator (50 mm long and 10 mm thick) containing three transverse cooling channels, illustrated in Figure 3. The thermal and physical properties of the calibrator (c) and polymer (p) are: k_p=0.18 W/mK, k_c=23.0 W/mK, ρ_p=1400 kg/m^3 and c_p=1000 J/kgK. The thermal boundary conditions are identified in Figure 3. A comparison of the results obtained both by Polyflow [22] and the developed code are depicted in Figure 4, through the comparison of the cross-temperature profiles (y-

direction) at three locations downstream (z=7, 30 and 50 mm). The agreement is obvious.

Influence of Process Parameters

The availability of the numerical software provides the opportunity to estimate the significance of some numerical, geometrical and process parameters on the performance of calibrating/cooling systems. For that purpose the geometry of Figure 5 was considered. It involves the cooling of a rectangular hollow profile along a 750 mm long dry calibrator, under the general conditions summarized in Table 1. As shown in Table 2 and in Figure 6, several variations of the original layout were studied, in order to establish the influence of a number of parameters on the final profile cross-section average temperature ($\overline{T}$) and standard deviation (σ_T) value, which is a measure of the temperature gradient.

Figure 7 shows graphically the results obtained for the various parameters. For the system under consideration, changes in the heat transfer coefficient (h), cooling channel diameter (dw), distance of the cooling channel to the profile (cd) and cooling fluid temperature (tw) had almost no effect on the results considered. Moreover the effects are conflicting, i.e., values of those parameters that promote lower $\overline{T}$ induce higher σ_T and vice-versa. This is a consequence of the high Biot number (hδ/k) that characterises heat transfer in plastics, i.e., heat conduction in the bulk material is much slower than convection at the interface. Anyway, the most influencing parameter is the linear extrusion velocity (vp). Although it promotes variations of $\overline{T}$ and σ_T with the same trend, it is not surprising that a better cooling performance requires the decrease of the extrusion velocity.

The use of a zig-zag arrangement for the cooling channels favoured the decline of $\overline{T}$, but increased the value of σ_T. In practice, these advantages will be probably offset by the higher machining costs. Conversely, splitting the calibrator into several units (nc) creates an effect almost similar to that of reducing the extrusion velocity. It is well known that this is due to the reduction of the heat flux at the polymer surface occuring in-between two consecutive calibrators, which increases both the temperature homogeneity and the effectiveness of subsequent cooling, given the increase of the profile surface temperature.

Finally, positioning the cooling channels close to the profile corners (lb) causes a reduction of $\overline{T}$ and an increase of σ_T. This is a consequence of the fact that the profile corners are cooled down more efficiently, but not the sides. However, since in the reference problem the corners were already cooler than the sides, this option does not carry any improvement.

Conclusions

A 3D FVM code developed to model the cooling stage of an extrusion line was presented and validated. It is able to tackle practical situations such as the presence of several individual cooling units and the existence of a thermal resistance between the plastic profile and the cooling medium.

The effect on the cooling performance of a number of process parameters (geometrical, operational and material related) can be quite distinct. Usually, when a reduction of the profile average temperature is imparted, a lower temperature homogeneity is also obtained. Extrusion velocity and splitting the calibrator into several units seem to be the exceptions, but reducing the former affects the production rate. The benefit of adopting zig-zag cooling channels seems to be insufficient to overcome the increase in machining costs. However, it is important to note that while for the geometry considered the best location of the cooling channel is obvious, if another more complex geometry was considered, the effect of the parameters related to the geometry/layout of the cooling channels could be identified as more important.

Given the availability of a modelling tool such as that presented in this work, which is sensitive to changes in the process parameters and is relatively fast from a computational point of view, the next stage of development will consist in its integration in an automatic design algorithm.

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Table 1 – General conditions used in the simulations.

k_p	0.18 W/mK
k_c	14.0 W/mK
ρ_p	1400 kg/m^3
c_p	1000 J/kgK
Linear extrusion velocity	2 m/min
Profile thickness	3 mm
Cooling channels' diameter	8 mm
Melt inlet temperature	180 ºC
Room temperature	20 ºC
Cooling fluid temperature	18 ºC
Profile/air convection heat transfer coefficient (free convection)	5 W/m^{2}K
Profile/calibrator convection heat transfer coefficient (contact resistance)	500 W/m^{2}K
Inner profile boundary	Insulated
CD	12 mm

Table 2 – Changes to the reference problem

Parameter	Problem Code	Description
--	ref	Reference problem
Number of calibrators	nc	the calibrator of the reference problem was split into three individual calibrators, totalizing the same length (see Figure 6 (a))
Cooling channel layout	lb	Four cooling channels placed in the corner zones (see Figure 6 (b))
	lc	Top and bottom cooling channels in a dense zig-zag arrangement (see Figure 6 (c))
	ld	Top and bottom cooling channels in a sparse zig-zag arrangement (see Figure 6 (d))
Distance of cooling channel to profile surface (CD in Figure 5)	cd↓	CD = 8 mm
	cd↑	CD = 16 mm
Cooling channel diameter	dw↓	d = 4 mm
	dw↑	d = 12 mm
Cooling cooling temperature	tw↓	T_w = 12ºC
	tw↑	T_w = 24ºC
Profile/calibrator convection heat transfer coefficient (contact resistance)	h↓	h = 250 W/m^{2}K
	h↑	h = 750 W/m^{2}K
Profile velocity	vp↓	v_p = 1 m/min
	vp↑	v_p = 3 m/min

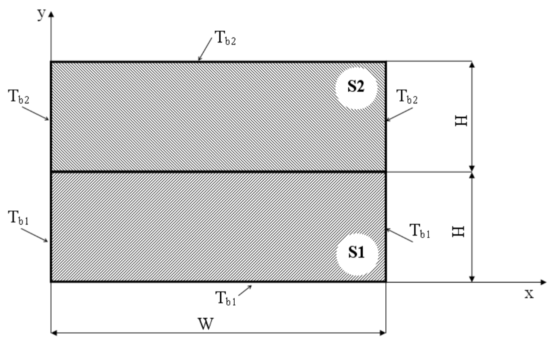
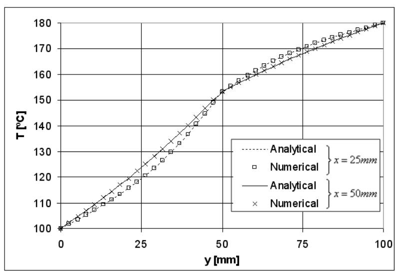
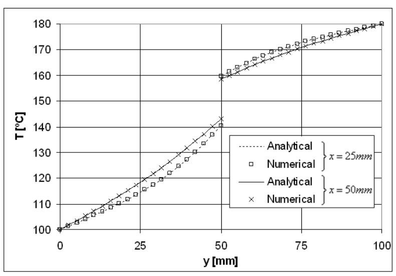


Figure 1 – Geometry and boundary conditions for the 'Analytical' problem..



(a)



(b)

Figure 2 – Analytical and numerical results for the temperature distribution in the case of the 'Analytical' problem: (a) perfect contact; (b) contact resistance.

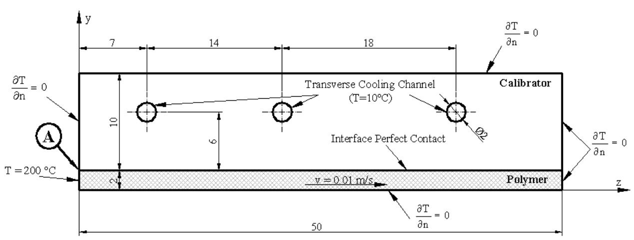


Figure 3 – Description of the 'Complex Layout' problem.

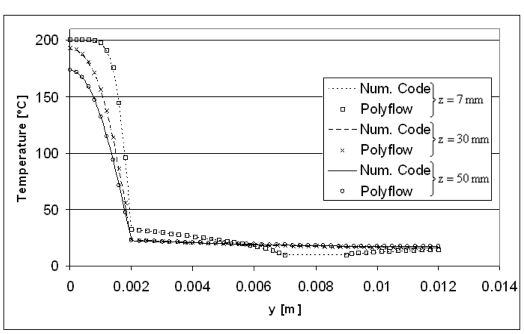


Figure 4 – Temperature distributions for the corrected 'Complex Layout' problem obtained with Polyflow and with the routine developed.

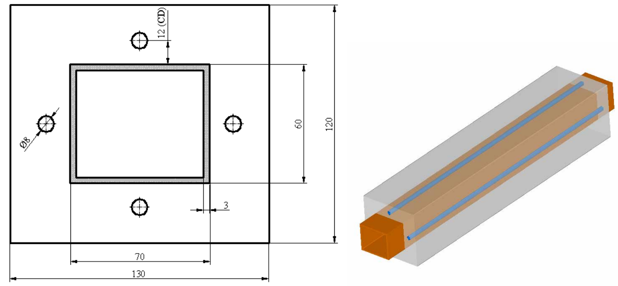


Figure 5 – Reference layout for the study of the cooling of a rectangular hollow profile.

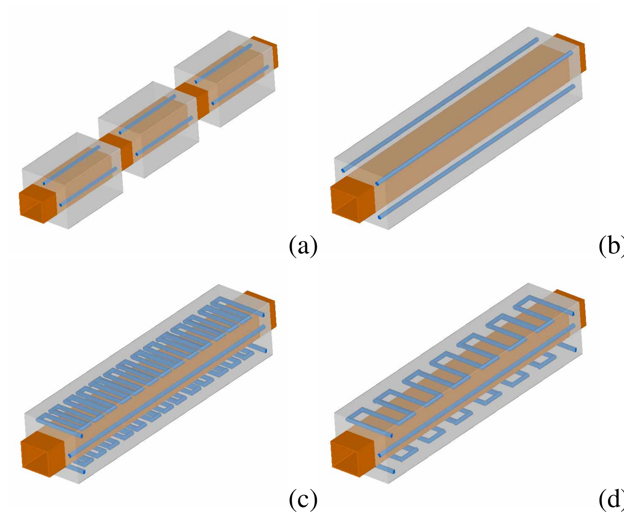


Figure 6 – Layout of the cooling system for some of the problems tested (see Table 2): number of calibrators - nc (a), linear axial water channels - lb (b), - lc, and (c) large pitch zig-zag arrangement - ld (d).

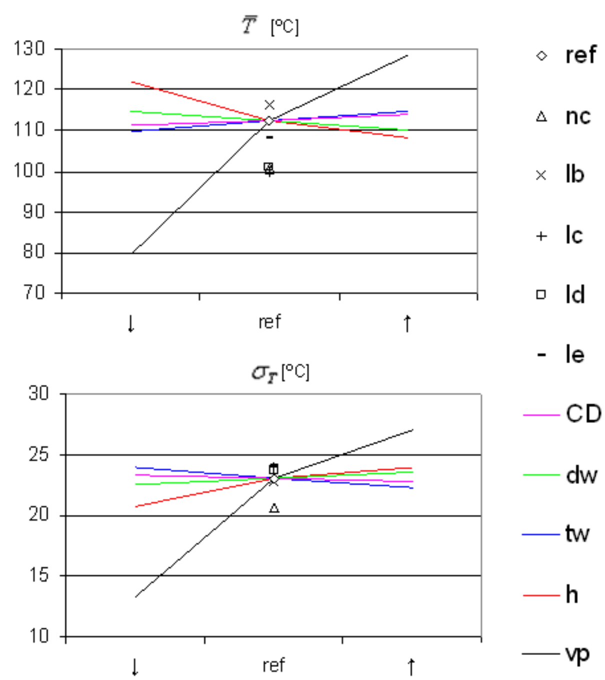


Figure 7 – Results for the case studies of Table 2.