INTRODUCTION

As shown in Fig. 1, a typical plastics extrusion line for the production of profiles comprises an extruder, a die, a calibration/cooling table (which can include several units), a haul-off and a saw (or, alternatively, a coiling device). The viscoelastic nature of the polymer melt, together with unavoidable fluctuations of the operating conditions (which affect the rheological behavior and flow dynamics), makes it very difficult to produce an extrudate with the required cross section. Moreover, as the profile progresses along the production line, it is subjected to a variety of external forces (such as friction, gravity, buoyancy and compression), which can cause important deformations, unless efficient cooling ensures enough profile strength (1, 2). Therefore, the calibration/cooling step has a double objective: it prescribes the final dimensions of the profile, while cooling it fast to solidify the outer layers of the extrudate to ensure sufficient rigidity during the remainder cooling steps (1). This is seen in Fig. 1, after the first calibrator, where a layer with thickness \( \delta \) has been cooled to a temperature below the solidification temperature \( T_s \). After calibration, the average profile temperature should also be lower than \( T_s \) to avoid subsequent remelting (1). Cooling of the extrudate should be as uniform as possible, meaning that the temperature gradients along the profile contour and thickness should be minimized, in order to induce adequate morphology development and a reduced level of residual thermal stresses (3, 4). Therefore, the objective is to
minimize both the profile average temperature and the

The calibration can be carried out by applying either

and/or dry, i.e., either direct or indirect contact takes

place between the cooling medium (generally, water)

and the hot profile, respectively (5, 6). Usually, several

calibrators are used in series, separated by relatively

short air zones (7, 8), where the temperature tends
to equalize, to minimize the internal thermal induced
stresses and increase the heat transfer efficiency in
the next calibrator. For high-speed profile extrusion, vac-
uum-assisted dry calibration has proved to be particu-
larly reliable (5). Sometimes, a combined wet/dry system
is used, consisting of a dry vacuum block unit followed
by a series of water-cooling blocks.

Given the above, the parameters that influence the
thermal performance of the calibration system may be
grouped as follows:

i) system geometry—number of calibrating units,
unit length, separating distance and layout of the
cooling channels (the latter involves such quanti-
ties as number, diameter, type of arrangement,
distance between consecutive channels and dis-
tance to the profile) (6, 9);

ii) cooling conditions—temperature of the inlet water,
flow rate, flow direction and wet versus dry con-
tact with the profile (1, 2);

iii) vacuum conditions—number and location of vac-
uum holes and vacuum pressure;

iv) extrusion conditions—mass flow rate and cross-
temperature profile field at die exit;

v) polymer thermo—physical properties-thermal dif-
fusivity and thermal expansion coefficient;

vi) properties of the calibrator material—thermal con-
ductivity and surface roughness;

vii) profile cross section—thicknesses, number and
location of hollow sections, etc.

Despite their obvious practical relevance, calibrating
and cooling systems have attracted relatively little at-
tention in the scientific literature. Most available re-
ports concern the calculation of the time evolution of
the extrudate temperature (8—10), the exception being
the work of Fradette et al. (3), in which the model pre-
viously developed (9) was integrated in an optimization
routine used to determine the optimal location and size
of the cooling channels. However, a thorough study of
the influence of the above geometrical, material, pro-
cess and operational parameters on the cooling perfor-
mance is apparently not available. In fact, the existing
results are either qualitative or concentrate on a few
variables (10, 11), ignoring, for instance, the effect
of boundary conditions. Moreover, no methodology for
the design of calibrators has yet been proposed.

The authors aim to develop and validate an algorithm
for the thermal design of calibrators for thermoplastic
extrudates. As for other plastics-processing equipment,
an optimization approach seems well suited for this
purpose (12, 13). It should comprise an objective func-
tion quantifying the calibrator performance, an optimi-
,zation algorithm assessing and generating increasingly
more efficient solutions, and a modeling package de-
scribing the process response. For this purpose, this
work presents and validates a 3D code based on the fi-
nite-volume method (FVM) to model the thermal inter-
changes during the calibration and cooling stage of
profile extrusion. FVM software is faster and requires
less computational resources than its FEM counterpart
(14), which is essential for the recurring use required
by the optimization algorithm. With a view to design, a
study of the influence of the boundary conditions, geo-
metrical and operating parameters on the performance
of cooling is also carried out. The actual design meth-
odology will be discussed in a forthcoming paper.
PROCESS MODELING

The first attempts to model the cooling of plastic profiles or pipes were made during the 1970s and 1980s with 1D models (see, for example (7, 15, 16)), which were applicable only to idealized conditions, such as uniform cooling and uniform thickness extrudates. Menges et al. (17) developed a 2D FEM approach that could deal with any extrudate cross section, but ignored axial heat fluxes. Inclusion of axial diffusion was addressed by Sheehy et al. (9), who proposed the Corrected Slice Model (CMS), which is a hybrid 2D model that can cope with the three-dimensionality introduced by the axial heat fluxes. Other 2D modeling studies of extrudate cooling addressed other specific aspects, such as the inclusion of more realistic boundary conditions for the heat exchange within the internal cavities of hollow profiles (11, 18), or the prediction of sag flow in thick wall pipes (19).

A major difficulty facing the modeling of the cooling process is the adequate knowledge of the heat transfer coefficient, \( h \), between the profile surface and the cooling medium, i.e., calibrator internal walls, water or air, which must include the effect of the contact resistance. It was experimentally shown that \( h \) can vary significantly (20), depending on the location along the calibration system. Other authors estimate \( h \) empirically, considering the local effectiveness of the contact between the profile and the calibrator from observations of the wear pattern of the calibrator (10). Finally, values of \( h \) can also be estimated using an inverse problem strategy, i.e., determining the values of the coefficient that match numerical simulations with the corresponding experimental temperature fields (8, 21). Table 1 summarizes the values of \( h \) reported in the literature for calibration/cooling systems.

Table 1. Values Reported for the Heat Transfer Coefficient (h).

<table>
<thead>
<tr>
<th>Geometry</th>
<th>Situation</th>
<th>( h ) [W/m²K]</th>
<th>Determination/Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Profile</td>
<td>Polymer/calibrator (dry)</td>
<td>200</td>
<td>Empirical evaluation (10)</td>
</tr>
<tr>
<td></td>
<td>Polymer/calibrator (wet)</td>
<td>500</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Polymer/calibrator (air gap)</td>
<td>50</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Polymer/calibrator (poor contact)</td>
<td>Reduction of 2/3 of the corresponding good contact</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Polymer/air free convection (inner cavities)</td>
<td>10</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Polymer/water (good circulation)</td>
<td>250</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Polymer/water (poor circulation)</td>
<td>100</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Polymer/calibrator</td>
<td>1000</td>
<td>(9)</td>
</tr>
<tr>
<td></td>
<td>(along the calibrator axis)</td>
<td></td>
<td>Matching of numerical simulations with experimental results (20)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Annealing zones</td>
<td></td>
<td>5</td>
<td></td>
</tr>
<tr>
<td>Pipe (Ø 63–315 mm)</td>
<td>1st Vacuum tank (spray cooling)</td>
<td>700–2000</td>
<td>Matching of numerical simulations with experimental results (8, 21)</td>
</tr>
<tr>
<td></td>
<td>2nd Vacuum tank (spray cooling)</td>
<td>170–750</td>
<td></td>
</tr>
<tr>
<td></td>
<td>3rd Vacuum tank (spray cooling)</td>
<td>120–550</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Annealing zones</td>
<td>14–35</td>
<td></td>
</tr>
</tbody>
</table>

Outline of the Numerical Procedure

In this work, the thermal field in the calibrating and cooling system is calculated by a 3D computational code based on the finite-volume method. This code was initially developed for the computation of isothermal viscoelastic flows, and has been recently extended to the case of non-isothermal flows (22). The details of the numerical algorithm and of its implementation have been described elsewhere (23, 24). The code is used to numerically calculate the variation of the temperature field within the extrudate as well as within the calibrator. Therefore, the energy conservation equations to be solved here can be written as:

\[
\frac{\partial}{\partial x} \left( k_p \frac{\partial T_p}{\partial x} \right) + \frac{\partial}{\partial y} \left( k_p \frac{\partial T_p}{\partial y} \right) + \frac{\partial}{\partial z} \left( k_p \frac{\partial T_p}{\partial z} \right) - \frac{\partial}{\partial z} (\rho_p c_p u T_p) = 0 \tag{1}
\]

for the profile, and as...
for the calibrator, where $T$ is the medium temperature, $\omega$ is the longitudinal velocity component (extrusion direction) in a Cartesian coordinate frame, $p$ is the fluid density, $k$ is the thermal conductivity and $c$ is the specific heat. The subscripts $p$ and $c$ denote polymer and calibrator, respectively.

In order to account for real processing conditions, various temperature and heat flux boundary conditions were implemented. At the interface between the profile and the calibrator, either perfect contact, assuming both temperature and heat flux continuity,

$$ (T_p = T_c) \, \text{interface} $$

(3)

$$ k_p \left( \frac{\partial T_p}{\partial n} \right) \, \text{interface} = - k_c \left( \frac{\partial T_c}{\partial n} \right) \, \text{interface} $$

(4)

or the existence of a temperature discontinuity (i.e., a thermal contact resistance (20))

$$ k_p \left( \frac{\partial T_p}{\partial n} \right) \, \text{interface} = - k_c \left( \frac{\partial T_c}{\partial n} \right) \, \text{interface} = h_i (T_p - T_c) \, \text{interface} $$

(5)

was considered. Here, $h_i$ is the interface heat transfer coefficient and $n$ is the normal vector of the interface. At the interface between the outside walls of the calibrator and the surrounding air, or between the external extrudate surface and the surrounding air, adiabatic or natural convection and radiation boundary conditions were set up. Figure 2 summarizes the boundary conditions considered in a typical problem.

Equations 1 and 2 are discretized following a finite-volume approach, and the resulting sets of linear algebraic equations are solved iteratively and sequentially, assuming an imposed heat flux at the polymer/calibrator interface. The coupling between the temperature fields in the polymer and calibrator domains is dealt with as follows. At each iteration step, the interface temperatures obtained for both domains are used to update the interface heat flux values (which depend on the type of boundary condition assumed at the interface), by using either Eq 4 or Eq 5, and the whole procedure is repeated until the temperature field converges.

**Geometry and Mesh Generators**

Given that the numerical procedure outlined above will be used intensively for design purposes, where it is necessary to evaluate the performance of various tentative calibrator designs (this will be dealt with in a future publication), specialized routines were developed to generate automatically the geometrical layout of the calibrator and the corresponding mesh.

The former requires information on the profile cross section, number of calibrators and, for each calibrator, the corresponding dimensions, location, number and layout of the cooling channels (these can be machined longitudinally, transversally or in a zigzag arrangement). When the mesh is generated, the presence of the cooling channels is initially ignored. Subsequently, the cells corresponding to the latter are removed and the adequate boundary conditions are imposed to their neighbor cell faces. This approach generates a stepwise approximation of the geometry of the cooling channels (25), akin to the “virtual boundary conditions” employed by Sheehy et al. (9) for the same purpose. Figure 3 depicts an example of an automatically generated mesh corresponding to the geometry shown in Fig. 2.

The finite-volume routine requires information on the computational mesh ($x$, $y$, and $z$ coordinates of all mesh points) and connectivity arrays to allow the identification of all control volumes surrounding a given computational cell. With this information, it is possible to evaluate cell face areas, cell volumes and distances that are needed for the discretization of the governing differential equations.

**MODEL ASSESSMENT**

Direct confrontation between predictions and experimental data is difficult, since the practical measurement
of the temperature profiles within and along the extrude cross section, especially when the extrudate moves along the calibration/cooling system, is extremely difficult, requiring the use of thermocouples imbedded in the material, at different depths of the profile thickness, and moving with the profile (8). 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 noncontact infrared thermometers that are generally employed and also on the measuring depth, i.e., the thickness effectively reached by the radiation from the sensor (26). However, most of the temperature measurements reported concern pipes (1, 8, 19, 21, 27) rather than profiles (20), and, even in this case, the data presented are insufficient for modeling purposes.

Therefore, the model developed and presented in this work is assessed by comparing its predictions with: i) the analytical results derived for a simple geometry, ii) the results reported by Sheehy et al. (9) for a more complex layout, and iii) the calculations provided by a general purpose FEM software (Polyflow (28)).

**Analytical Solution**

The first case study considered is illustrated in Fig. 4. It consists of two rectangular slabs, S1 and S2, with thermal conductivities $k_1$ and $k_2$, respectively, in contact through one of their faces. As shown also in the figure, the temperature is imposed on the remaining faces. The energy equation (2) controlling the temperature distribution on each slab, for constant conductivity, takes the form:

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} = 0$$

As described by Nóbrega (29), the temperature distribution in each slab can be obtained from:

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

(7)

for S1, and from

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

(8)

for S2, for perfect contact case, and by:

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

$$- \frac{k_1}{W} \sin \left( \frac{n\pi y}{W} \right) \sinh \left( \frac{n\pi x}{W} \right)$$

$$\sin \left( \frac{n\pi x}{W} \right) \sinh \left( \frac{n\pi y}{W} \right)$$

(9)

for S1 and from

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

$$- \frac{k_2}{W} \sin \left( \frac{n\pi y}{W} \right) \sinh \left( \frac{n\pi x}{W} \right)$$

$$\sin \left( \frac{n\pi x}{W} \right) \sinh \left( \frac{n\pi y}{W} \right)$$

(10)

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

The temperature distributions defined by Eqs 7–10, both for perfect contact or thermal resistance, are compared in Fig. 5 with those obtained in the developed numerical routine, using $W = 100$ mm, $H = 50$ mm, $T_{b1} = 100^\circ$C, $T_{b2} = 180^\circ$C, $k_1 = 7$ W/mK, $k_2 = 14$ W/mK and, for the case of contact resistance, $h_i = 500$ W/m²K. It is clear that the two sets of results are virtually coincident, hence giving confidence on the correct implementation of the thermal routines.

**Complex Layouts**

The predictions of the numerical routines developed were also compared with the results of Sheehy et al. (9) for the problem shown in Fig. 6, which was used by those authors to validate the Corrected Slice Model (CSM). The problem consists of the determination of the temperature distribution in a 2-mm-thick polymeric sheet moving at 0.01 m/s while being cooled by a 50-mm-long and 10-mm-thick calibrator containing three transverse cooling channels. The thermal and physical properties of the calibrator (subscript c) and polymer (subscript p) are: $k_c = 0.18$ W/mK, $k_p = 23.0$ W/mK, $\rho_c = 1400$ kg/m³ and $c_p = 1000$ J/kgK. The thermal boundary conditions are also identified in Fig. 6. According to Sheehy et al. (9), the rigorous thermal solution of the problem is that shown in Fig. 7a. It is seen that the isotherms close to the calibrator inlet are perpendicular to the interface, indicating that there is no
Design of Calibrators for Extruded Profiles. I

heat exchange across the interface, i.e., the temperature gradient in direction normal to the interface looks to be zero, a physically unrealistic scenario. This is a consequence of assuming that the temperature of the calibrator at the corner next to the extrudate (point A in Fig. 6) is equal to the melt inlet temperature. Given the higher conductivity of the calibrator, the inlet melt temperature would spread out easily to the corner neighborhood, and, under these conditions, no heat exchange would occur between the calibrator and the profile. It is probable that the high temperature imposed at the calibrator corner was set because both domains
(calibrator and polymer) share the same node. This hypothesis was confirmed with Polyflow: its results, shown in Fig. 7b, are very similar to those of Fig. 7a, and were obtained considering a single node at point A with an imposed temperature of 200°C.

Given the above discussion, the problem studied by Sheehy et al. (9) was slightly modified here. Now, the polymer domain, where the temperature is imposed, begins 1 mm ahead of the calibrator entrance, as represented in Fig. 8. The new temperature distribution determined by Polyflow (see Fig. 9a) shows, as expected, heat exchange between the polymer and the beginning of the calibrator.

A more realistic modeling of the original problem of Sheehy et al. (9) requires the use of a coincident double-node technique, one belonging to the polymer and the other to the calibrator, to allow for a temperature discontinuity between the two domains. The results of the calculations with our code are shown in Fig. 9b, using the double-node technique, and should be contrasted directly with those of Polyflow in Fig. 9a. A good agreement is observed between both solutions, and this is further confirmed in Fig. 10, which compares temperature profiles across the sheet at three different axial locations (z/L = 7/50, 30/50 and 50/50, where L is the length of the calibrator).

It is worth mentioning that despite the problems discussed above, the results of Sheehy et al. (9) using the CSM are reasonably accurate: the average polymer outlet temperature was predicted as 122.3°C, with the model referred to as rigorous by those authors, and 120.3°C, with the CSM; we obtained 120.6°C.

Finally, the numerical code developed in this work was also assessed for the third test case study illustrated in Fig. 11, representing the behavior of the polymer sheet downstream from the calibration, in the annealing zone that further exposes the extrudate to cold air and homogenizes the polymer temperature distribution, as a consequence of heat fluxes from hotter to cooler regions. This situation is quite relevant for the study of systems having more than one cooling/calibrating unit (see Fig. 12b). The temperature distribution at the end of the annealing zone, shown in Fig. 10, corresponds to z/L = 75/50. The predictions of our finite-volume code and those of Polyflow are compared, and again, the agreement is excellent.

INFLUENCE OF BOUNDARY CONDITIONS, PROCESS, AND GEOMETRICAL PARAMETERS

Next, the code is used to investigate the effect of some boundary conditions, process, and geometrical parameters on the behavior and performance of calibrating/cooling systems. For this purpose the cooling of the rectangular hollow profile shown in Fig. 12 was studied under the general conditions summarized in Table 2. A calibration length of 600 mm was fixed but it corresponded to either a single or three consecutive calibration units. The results obtained with the various systems were compared in terms of heat fluxes at the geometry boundaries and of minimum ($T_{\text{min}}$), maximum ($T_{\text{max}}$) and average ($\bar{T}$) temperatures and the temperature distribution standard deviation ($\sigma_T$) calculated at...
the end cross section of the polymer extrudate. The latter is computed as:

$$\sigma_T = \frac{\sum_{i=1}^{n_f} (T_i - T)^2 A_i}{A_T}$$  \hspace{1cm} (11)$$

where $n_f$ is the number of computational cell faces on the profile outlet boundary, $T_i$ is the face temperature, $A_i$ is the face area and $A_T$ is the area of the profile cross section. Therefore, $\sigma_T$ is a measure of the temperature nonhomogeneity at the final cross section.

It is worth mentioning that the results obtained for this profile geometry under study cannot be directly extrapolated to other geometries, but, nevertheless, the results provide information on the qualitative effect of the main variables involved in the process and their relative importance.

**Boundary Conditions**

An important issue for modeling the profile cooling stage is the definition of the boundary conditions at the profile and calibrator surfaces. In the literature, the outer surfaces of both profile and calibrator are modeled either as adiabatic (9) or having a defined convective heat flux (8), in both cases neglecting radiation. For the interface between the profile and the calibrator, either perfect contact or contact resistance is used; some authors argue that contact resistance is the better choice (1), but in reality we are unaware of any practical or computational quantification of its relevance. Thus, in order to study the influence of this parameter, cooling of the profile presented in Fig. 12a was modeled, adopting the layout shown in Fig. 12b, where the cooling length of 600 mm was divided into three 200-mm-long calibrators, separated by 75-mm-long annealing zones. In this figure, Os1 to Os4 represents the profile outer surfaces, i.e., exposed to the surrounding environment along the cooling line. The set of case studies considered is described in Table 3, while the computed results are summarized in Tables 4 and 5.

**Table 4** compares the heat fluxes at the various boundaries. Starting with the reference case study c1r1, which accounts both convection and radiation at the outer boundaries, the table shows losses of 32.0, 20.7, 15.0 and 7.5 W through the extrudate outer surfaces Os1 through Os4, respectively, and losses of 1076.1, 736.3 and 589.8 in the three calibrators mostly via the cooling channels. **Table 5** contains data for temperature at the cross section at the end of the extrudate. As shown in **Table 4**, most of the heat is removed from the profile through its interface with the calibrator, and then from the calibrator through the cooling channels. Consequently, the values in **Table 5** are little affected by the type of boundary condition considered at the extrudate and calibrator outer surfaces, and, in contrast, the conditions at the interface between the extrudate and the calibrator are fundamental as can be seen in the total loss. In fact, changes in the resistance coefficient ($h_i$) have the highest impact on the total heat loss (compare $h_i^{↑}$, $h_i^{↓}$ and pc with the reference case). The effects of radiation and convective heat transfer are similar, but since most of the cooling takes place via the
cooling channels, the type of outer boundary has a negligible impact upon the total loss. However, the usual procedure of considering only convection is inadequate. Anyway, if detailed knowledge of temperature along the extrudate is required, then it is important to consider both the effects of convection and radiation in the annealing zones.

The above computations ignored the contribution of radiation heat exchange between the profile and the transversal external surfaces of the calibrator, as this would make the calculation algorithm much more complex. However, since only a minor part of the heat emitted by radiation by the profile would reach those calibrator surfaces, owing to the low view factors, this omission is irrelevant.

Finally, it is worth noting that a 50% change of $h_i$, which lies within the practical range of variation, yields

\begin{table}[h]
\centering
\begin{tabular}{|l|c|}
\hline
Parameter & Value (default) \\
\hline
$k_p$ & 0.18 W/mK \\
$k_c$ & 14.0 W/mK \\
$\rho_p$ & 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°C \\
Room temperature & 20°C \\
Cooling fluid temperature & 18°C \\
Profile/air convection heat transfer coefficient (free convection) & 5 W/m$^2$K \\
Polymer emissivity $\varepsilon_p$ & 0.9 \\
Calibrator emissivity $\varepsilon_c$ & 0.25 \\
Profile/calibrator convection heat transfer coefficient (contact resistance) & 500 W/m$^2$K \\
Inner profile boundary & Insulated \\
CD & 12 mm \\
\hline
\end{tabular}
\caption{General Conditions Used in the Simulations.}
\end{table}

Fig. 12. Cooling of a rectangular hollow profile problem (dimensions in mm): cross-section geometry (a), three-calibrator layout (b), and one-calibrator layout (c).
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Table 3. Case Studies Considered to Study the Influence of the Boundary Conditions.

<table>
<thead>
<tr>
<th>Code</th>
<th>Boundary for Outer Surface of Profile + Calibrator</th>
<th>Calibrator/Profile Interface</th>
<th>Calibrator/Profile Interface</th>
</tr>
</thead>
<tbody>
<tr>
<td>c1r1</td>
<td>convection + radiation</td>
<td>h = 500</td>
<td></td>
</tr>
<tr>
<td>c0r0</td>
<td>adiabatic</td>
<td>h = 500</td>
<td></td>
</tr>
<tr>
<td>c1r0</td>
<td>convection</td>
<td>h = 500</td>
<td></td>
</tr>
<tr>
<td>h↑ (+50%)</td>
<td>convection + radiation</td>
<td>h = 750</td>
<td></td>
</tr>
<tr>
<td>h↓ (-50%)</td>
<td>convection + radiation</td>
<td>h = 250</td>
<td></td>
</tr>
<tr>
<td>pc</td>
<td>convection + radiation</td>
<td>Perfect contact</td>
<td></td>
</tr>
</tbody>
</table>

a lower than 10% change in $T$ of the extrudate at the end of the cooling zone, in agreement with the variations in the total heat loss. Clearly, this parameter is the major factor affecting the thermal performance of the production line.

Process and Geometrical Parameters

To assess the influence of process and geometrical parameters on cooling, the conditions specified in Table 2 and the layout shown in Fig. 12c (i.e., single 600-mm-long calibrator) were considered. Table 6 presents a number of case studies, investigating changes in the conditions of Table 2, which will be taken as the reference case (ref) for comparison of the results. Figure 13 illustrates the changes in the layout of the cooling channels, and the corresponding results are listed in Table 7.

The effect of the cooling fluid temperature ($tc$) is much smaller than that of the profile velocity ($vp$). Additionally, the effect of $tu$ with respect to $T$ and $σ_T$ is conflicting, i.e., values that promote a lower $T$ will induce higher $σ_T$ and vice versa, as a consequence of the high Biot number ($hδ/k$) that characterizes heat transfer in plastics, i.e., heat conduction in the bulk is much slower than convection at the interface. Conversely, $vp$ promotes the simultaneous increase or decrease of both $T$ and $σ_T$, which is advantageous for optimization purposes. However, a better cooling performance (which requires low values of $T$ and $σ_T$) involves, not surprisingly, the decrease of $vp$, i.e., of the production rate.

In the case of geometrical parameters, the use of a zigzag arrangement for the cooling channels ($lc$, $ld$), or the increase of the number of cooling channels ($lb$), favors the decline of $T_{nin}$, $T_{max}$, and $T$, but again increase $σ_T$. The improvements obtained by narrowing a zigzag arrangement ($lb$) are negligible compared with the use of a wider one ($ld$). In practice, these marginal advantages will eventually be offset by the higher machining costs, i.e., simpler channels are the best choice for the present case study.

The distance between the cooling channel and the profile surface ($cd$) is relatively unimportant; its reduction ($cd↑$) has almost no effect on the results, while its increase ($cd↓$) reduces the cooling efficiency. This indicates a limiting $cd$ value below which the increase in cooling efficiency is negligible.

Table 4. Boundary Heat Fluxes [W] Computed for the Case Studies Listed in Table 3.
the extrusion velocity ($v_p \downarrow$), but without affecting the production rate. In terms of the values obtained for $T$ and “total heat removed,” it can be concluded that this option has a performance similar to that of layout $lb$, which employs a double number of cooling channels.

Finally, having the cooling channels close to the profile corners ($la$) reduces $T_{\text{min}}$ but increases $T_{\text{max}}$, because the profile corners cool more efficiently than the middle. However, since the former were already cooler than the latter, this option does not promote any improvement (in fact, it reduced the total heat removed from the system). This can also be seen in Fig. 14, where the predicted downstream profile temperature distribution is plotted both for the reference (ref) and $la$ case studies.

**CONCLUSIONS**

A 3D FVM code developed to model the cooling stage of an extrusion line was presented and validated prior to being used for investigating the effect of various process and geometrical parameters on the efficiency of calibration/cooling units. The code is able to handle accurately various 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.
Detailed investigation of the calibration unit has shown that most of the heat is removed at the calibrator via the cooling channels and that the contact resistance at the interface is the most important parameter affecting the performance of the unit. Additionally, it was shown that boundary conditions on the calibrator/ extrudate outer surfaces have negligible impact.

The effect of process and geometrical parameters on the cooling performance can be quite distinct. Often, when a reduction of the profile average temperature is imparted, lower temperature homogeneity is also obtained, but exceptions are variations in the extrusion velocity and splitting the calibrator into several units. Since the extrudates are characterized by high Biot numbers, significant increases in the heat transfer removal at the extrudate surface quickly reach a limiting behavior (in terms of efficiency), and this should be taken into consideration when designing calibration/cooling units. For instance, the benefits of adopting zigzag cooling channels are clearly insufficient to overcome the increase in machining costs. The effect of other geometrical parameters was not very important, but this may be related to the characteristics of the profile considered.

Table 7. Results Computed at the End Cross Section of the Extrudate and Total Heat Lost for the Case Studies of Table 6 (V-Value, D-Relative Difference to Reference Problem).

<table>
<thead>
<tr>
<th>Code</th>
<th>$T_{min}$ [°C]</th>
<th>$T_{max}$ [°C]</th>
<th>$\bar{T}$ [°C]</th>
<th>$\sigma_T$ [°C]</th>
<th>Total Heat Removed [W]</th>
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<tr>
<td>ref</td>
<td>48.7</td>
<td>142.9</td>
<td>111.9</td>
<td>23.3</td>
<td>2340.5</td>
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<td></td>
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<tr>
<td>$t_{w\downarrow}$</td>
<td>V</td>
<td>44.3</td>
<td>141.6</td>
<td>109.5</td>
<td>24.1</td>
</tr>
<tr>
<td>D</td>
<td>−9.1%</td>
<td>−0.9%</td>
<td>−2.2%</td>
<td>3.5%</td>
<td>3.6%</td>
</tr>
<tr>
<td>$t_{w\uparrow}$</td>
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<td>144.2</td>
<td>114.4</td>
<td>22.5</td>
</tr>
<tr>
<td>D</td>
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<td>0.9%</td>
<td>2.2%</td>
<td>−3.5%</td>
<td>−3.6%</td>
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<tr>
<td>$v_{p\downarrow}$</td>
<td>V</td>
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<td>99.0</td>
<td>79.1</td>
<td>13.4</td>
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<tr>
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<td>13.3%</td>
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<td>15.3%</td>
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<tr>
<td>$n_c$</td>
<td>V</td>
<td>48.9</td>
<td>136.1</td>
<td>107.9</td>
<td>21.0</td>
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<tr>
<td>$l_a$</td>
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<td>3.6%</td>
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<td>8.6%</td>
<td>9.7%</td>
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<tr>
<td>$c_d\downarrow$</td>
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<td>142.8</td>
<td>110.9</td>
<td>23.5</td>
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<td>−0.9%</td>
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<td>143.5</td>
<td>113.5</td>
<td>22.9</td>
</tr>
<tr>
<td>D</td>
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<td>0.4%</td>
<td>1.4%</td>
<td>−1.5%</td>
<td>−2.3%</td>
</tr>
<tr>
<td>$d_w\downarrow$</td>
<td>V</td>
<td>52.3</td>
<td>144.4</td>
<td>114.2</td>
<td>22.7</td>
</tr>
<tr>
<td>D</td>
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<td>1.0%</td>
<td>2.1%</td>
<td>−2.3%</td>
<td>−3.4%</td>
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<tr>
<td>$d_w\uparrow$</td>
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<td>46.1</td>
<td>141.6</td>
<td>110.0</td>
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</table>
Given the availability of this modeling tool, which is easily modified to investigate a wide range of processing and geometrical conditions and is relatively fast from a computational point of view, the next stage of development will be its integration into an algorithm for the automatic design of cooling units.

ACKNOWLEDGMENTS

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1. V. Kleindienst, Kunststoffe, 63(1), 7 (1973).
5. B. Endrass, Kunststoffe-German Plastics, 83(8), 584 (1993).

Fig. 14. Temperature fields of the lower-right corner of extrudate cross section downstream, for the reference (ref) and for layout a (la) case studies (Table 6, Temperature in °C).