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ON THE DETERMINATION OF HEAT TRANSFER COEFFICIENT BETWEEN PVC AND STEEL IN VACUUM EXTRUSION CALIBRATORS

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ABSTRACT

The design of vacuum calibrators for the cooling of complex PVC profiles is central to the production of high quality extrudates. One important parameter governing cooling efficiency is the heat transfer coefficient at the interface between the stainless steel calibrator and the PVC extrudate, whose value is often taken as constant regardless of the extrusion velocity and the applied pressure vacuum. In this paper, a method is proposed to evaluate the variation of the heat transfer coefficient over the entire calibrator length. The idea is to use temperature measurements together with heat transfer simulation to derive a heat transfer correlation that can be used in practical design cases.

KEY WORDS PVC Extrusion profile Calibration Heat transfer coefficient Numerical simulation Heat conduction

INTRODUCTION

PVC profiles are increasingly popular in the construction industry where they are used as building parts and in the manufacturing of window frames. These profiles are produced by extrusion, the main shaping process used in the plastics industry. A typical PVC extrusion line includes three main parts, namely: an extrusion screw (for polymer melting), a die (that ensures continuous shaping) and a calibration unit, made normally of several sections. Various types of calibrators can be selected[1] depending on the application. They all have the same role, i.e. to cool down the PVC extrudate uniformly in order to avoid profile warpage, to decrease the surface and bulk temperatures sufficiently so that the profile can be further manipulated (hauled-off, sawed, packed and shipped), and finally to adjust the shape and dimensions of the end-product within the prescribed tolerance. Due to their great shaping flexibility, PVC profile designs are rapidly growing in complexity forcing the calibration unit to cool down profiles faster while meeting more stringent specifications especially in terms of surface aspect.

The most common cooling process is vacuum calibration due to its greater efficiency. We show in *Figure 1* the principle of a vacuum calibrator. Its design is very challenging. Two issues must be addressed, namely: the selection of an adequate cooling channel layout (channels position and diameter, coolant temperature) and the determination of the overall calibration length. Considering the large cost associated with trial-and-error-based design, numerical simulation of the profile cooling appears to be an effective way to assess preliminary calibration strategies and determine some sort of "optimal" design guidelines before the actual fabrication of the calibrator. In order to model calibration accurately, it is important to identify the prevailing heat transfer mechanisms and the boundary conditions that govern the cooling process, specially at the interface between the calibrator and the profile.

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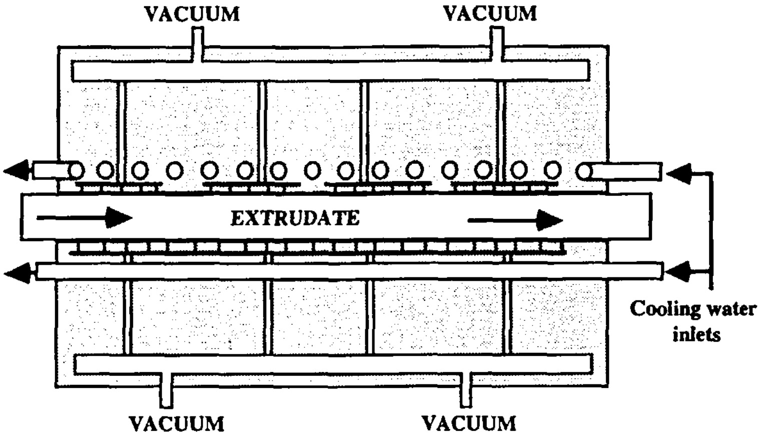

Side view

Figure 1

A literature survey revealed that not many papers have been published on PVC calibration issues. A rough estimate of the calibration length can be obtained with dimensional analysis. For this purpose, the Fourier number is introduced, namely:

$$Fo=\alpha t/H^2=\alpha L/(V_e H^2) \tag{1}$$

where H is the extrudate thickness, V_e the extrusion speed, α the polymer thermal diffusivity and L the length of the cooling section. Starting from a known calibration case in very simple topologies such as pipes and plates, the value of Fo can be readily computed. Its value is then used to calculate the length required for the design case at hand that would achieve comparable cooling[2,3].

In order to determine the temperature profile in the extrudate, the time dependent heat transfer equation must be solved. In simple geometries, it is possible to integrate the one dimensional heat transfer equation analytically [4–6]. However, some difficulties may arise when it comes to specifying appropriate boundary conditions at the interface between the calibrator and the profile. One alternative is to specify the coolant temperature assuming that there is no heat resistance at the interface. Unfortunately, this approach is too crude and generates unrealistic results[7]. Another possibility is to set a value of the heat transfer coefficient at the interface. This coefficient, which is related to the temperature difference between the profile surface and the cooling medium, is, however, very dependent on the calibrator geometry. From a practical standpoint, it is difficult to determine its value.

Calibration problems were also investigated using numerical simulation. The finite element resolution of the two dimensional transient heat transfer equation in the extrudate combined to a steady-state approach in the calibrator allowed the simultaneous determination of the temperature distribution in the profile and in the calibrator. A slice-wise approach consisting of computing the temperature field with a space-marching technique was proposed which simplifies significantly the problem resolution[8,9]. This method, called the Menges model in the forthcoming, is limited to purely two dimensional systems, i.e. systems in which the cooling channels are parallel to the profile. The authors neglected axial thermal fluxes in their study, an assumption which proves correct only when the cooling channels are located close to the interface. The Menges model has been generalized recently to account for axial heat flux. The new model, called the corrected slice method, allows therefore the treatment of any cooling layout topology[10].

In the Menges model, the heat transfer coefficient at the interface was defined using the temperature difference across the interface between the profile surface and the calibrator, removing the geometry dependence mentioned before. However, a constant value for this heat transfer coefficient was utilized in the simulations, regardless of the position along the calibrator and the vacuum pressure applied.

The objective of this work is to propose a method to compute the variation of the heat transfer coefficient at the interface between the calibrator and the profile in a vacuum calibrator. The basic principle is to use temperature measurements in the calibrator together with heat transfer simulation to determine the range of values along the extrusion axis, a piece of information of prime importance in calibrator design.

METHODOLOGY

Heat transfer in an industrial calibration unit is a three dimensional transport phenomenon which is governed by the following equations:

Profile:

$$k_p \operatorname{div}(\operatorname{grad} T) - \rho C_{pp}(v \cdot \operatorname{grad} T) = 0 \tag{2}$$

Calibrator:

$$k_c \operatorname{div}(\operatorname{grad} T) = 0 \tag{3}$$

where C_{pp} is the heat capacity of PVC and k_p and k_c the thermal conductivity of PVC and steel.

Using the same assumptions as in the Menges model, these two equations can be reduced to the following simpler two dimensional equations:

Profile:

$$k_p(\partial^2 T/\partial x^2 + \partial^2 T/\partial y^2) - \rho C_{pp}(\partial T/\partial t) = 0 \tag{4}$$

Calibrator:

$$k_c(\partial^2 T/\partial x^2 + \partial^2 T/\partial y^2) = 0 \tag{5}$$

As the extrusion speed is constant, the time derivative term in (4) and the second term in (2) are identical due to the time-space equivalence (see Reference 8, for instance).

The boundary conditions associated with these equations are shown in *Figure 2*. Natural convection inside the hollow profile channel is considered sufficiently low to be neglected, so that an adiabatic condition is specified. As for the cooling channels, two conditions can be imposed: either a channel wall temperature equal to the cooling temperature or a wall convective heat transfer condition. Finally, for mathematical closure, an additional condition is required

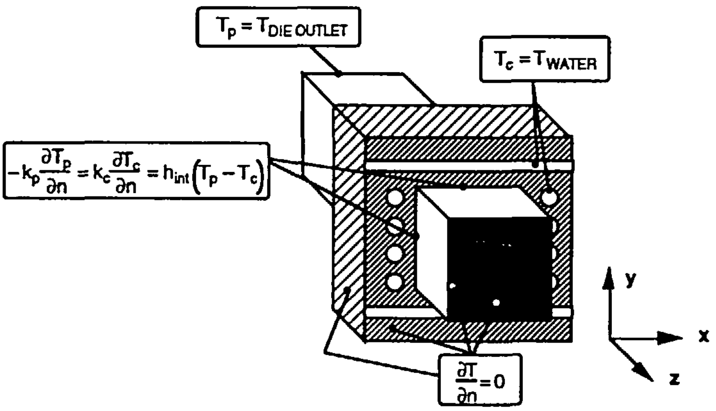

Figure 2

at the profile/calibrator interface which is nothing but a heat flux balance across this interface, namely:

$$k_c(\partial T/\partial n)_{interface} = -k_p(\partial T/\partial n)_{interface} = h(T_p - T_c)_{interface} \tag{6}$$

where n is the normal to the interface, T_p the profile surface temperature, T_c the calibrator surface temperature, and h the heat transfer coefficient.

As a result of the space-marching scheme, the profile temperature initial condition is set to the temperature of the profile before entering the calibrator, i.e. the die outlet temperature.

In this work, (4) and (5) are resolved using a standard Galerkin finite element method using six-noded quadratic triangular elements. No implementation details will be given here, and we refer the reader to Reference 10 for more information. As for the domain tessellation, the profile is first meshed using a highly structured pattern in order to keep the number of elements as low as possible while preserving computational accuracy. Then, an unstructured mesh generator[11] is used for the behaviour taking the profile mesh contour nodes as starting points. In this way, compatibility between the two meshes is ensured, allowing their merging into a unique grid corresponding to the overall calibrator/profile geometry.

The transient term $\partial T/\partial t$ in (4) is approximated using a first-order implicit Euler scheme for the first time step and a second-order Gear scheme for all subsequent steps. A double node technique is implemented[12] to deal adequately with the temperature discontinuity at the profile-calibrator interface. With this technique, two temperatures are computed at the same node, one in each medium. The presence of the cooling channels is dealt with as thermal boundary conditions by means of Lagrange multipliers as detailed in Reference 13.

HEAT TRANSFER COEFFICIENT

A series of eight experiments was conducted in order to track the effect of the two major variables influencing heat transfer in the problem namely: the extrusion speed and the vacuum pressure applied. In practical applications, only the extrusion velocity can be predicted from the knowledge of the extruder output and the profile cross-section. The vacuum level cannot be calculated nor estimated before the process is brought on line. For this reason, only the extrusion speed was considered in the correlation. The influence of the vacuum pressure should have to be assessed, as well, if it shows any effect on the heat transfer efficiency, a situation that did not occur in our tests.

The calibrator used in our experiments comprises three sections. It is displayed in *Figure 3* along with the location of the temperature measurement points. *Figure 4* illustrates the extrudate cross-section (hollow profile with a square cross-section) and shows how the temperature measurements were taken. Threaded thermocouples were carefully installed inside the calibrators, with their tip located in two fashions (A and B) as shown in *Figure 4*. In A, the profile surface temperature is measured with the tip perfectly aligned with the calibrator surface while in B, the calibrator temperature is taken as close as possible to the interface (1.45 mm). An infrared thermometer was also utilized to monitor the profile surface temperature, between the calibrator sections. PVC emissivity settings in the thermometer were adjused so temperatures taken with both thermometer and thermocouples would compare.

As for the experimental conditions, the setup was operated within a typical industrial window. The extrusion velocity was varied between 0.5 and 1.5 metre per minute and the absolute vacuum pressure between 40 and 80 kPa. Temperature-controlled water was used as the cooling medium and the temperature was set at 13°C.

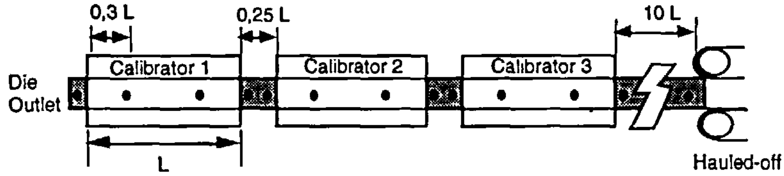

Figure 3

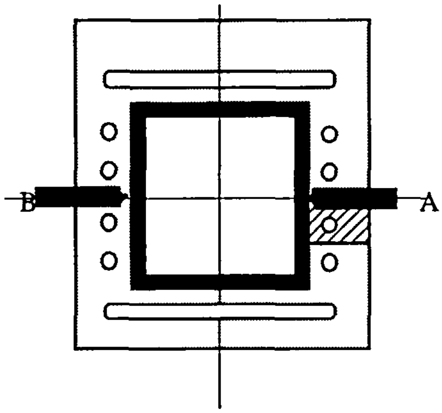

Figure 4

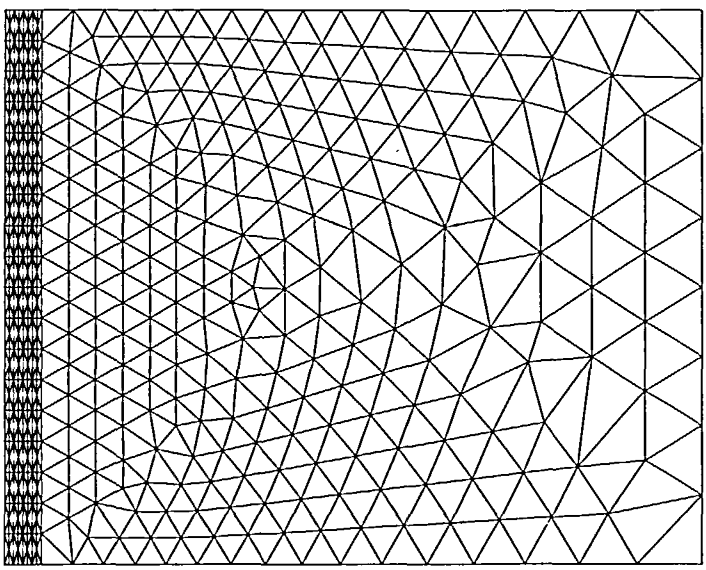

Figure 5

Physical properties used in the numerical work are those given by the various equipment and material suppliers except for the PVC heat capacity, that was experimentally determined by differential scanning calorimetry. For symmetry reasons and physical arguments (we assume that the cooling is only induced by the closest channel), only a small part of the calibrator was simulated (shown in *Figure 4*), allowing to reduce significantly the simulation burden. The corresponding mesh is shown in *Figure 5*. Two striking features of our numerical strategy are illustrated here, i.e. the highly structured pattern of the profile mesh and the absence of cooling channels. In fact, as mentioned before, the cooling channels are there, but they are dealt with in a "virtual" manner using optimization methods. In *Table 1*, the temperature results (equivalent to thermocouple reading) obtained in the full calibrator and the small area are compared. It can be noticed that the difference between the two results is not significant since it is within the

Table 1 Comparison between full calibration and cooling channel area temperatures

Geometry type	Temperature (°C)	Difference (%)
Full calibration with transverse channel	85.1	0.2
Full calibration without transverse channel	85.0	0.1
Simplified	84.9	—

Table 2 Comparison of temperature predictions and measurements

		Temperature (°C)			
Measurement	Point	Measured	Simulated (correl.)	Deviation (%)	Simulated (constant)
Die outlet		191.0	191.0	—	191.0
Outlet	1st calibrator	4.72	48.9	+3.6	47.5
Inlet	2nd calibrator	53.9	52.8	−2.0	51.6
Outlet	2nd calibrator	34.4	32.8	−4.7	31.8
Inlet	3rd calibrator	38.3	36.1	−5.7	36.2
Outlet	3rd calibrator	27.8	21.1	−24.1	20.2

infrared pyrometer precision of $\pm 0.5°C$. It is then clearly demonstrated that the presence of transverse cooling channels at the top of the calibrator does not have a significant influence on temperature measurements and can be neglected, which justifies the above physical argument. We should mention here that the full simulation of a calibrator comprising transverse cooling channels requires a special technique discussed in Reference 3.

In order to determine the actual variation of the heat transfer coefficient along the extrusion axis, an iterative approach was used, which consists of resolving (3) and (4), and adjusting the value of the heat transfer coefficient until a satisfactory fit is obtained between the temperature predictions and measurements[4,5,14]. The method is repeated for every calibrator slice bounded by temperature points. A value of 5 W/m^2-K was used to stimulate the contact with air between the calibration units[15]. In these areas, the heat conduction in the profile driven by the strong temperature gradients between the surface and its bulk (which is still quite hot), overwhelmed such a low cooling capacity. This is clearly showed by the measured temperatures recorded in *Table 2*. We can indeed notice a slight "re-heating" of the profile surface between calibrators.

We present in *Figure 6* the variation of the heat transfer coefficient with the Fourier number. The simulation method employed in this work coupled to the use of the Fourier number provides flexibility when dealing with variable material properties and/or extrusion speeds. It can be seen that the interface heat transfer coefficient decreases exponentially with an increasing Fourier number. A simple linear regression provided the intercept and slope. The mathematical expression of the correlation is:

$$h = \exp(9.2 - 1.3Fo) \tag{7}$$

The value of h varies from 10 to 10,000 W/m^2-K along the calibrator axis and depends on the extrusion speed and the calibration length. This range of value compares well with literature results. For instance, Haberstroh reported values of h between 500 and 2500 W/m^2-K for vacuum calibrators[14]. The wide variation of h measured here (up to three orders of magnitude) fully justifies the need for using a variable heat transfer coefficient, although a constant value of h was used in the simulation studies published so far. It should be mentioned that it was not possible to detect any vacuum influence on heat transfer in the range covered, i.e. from 80 kPa down to 40 kPa absolute. These values correspond to the vacuum at which the profile starts to collapse and the maximum operating pressure of the pumps, respectively.

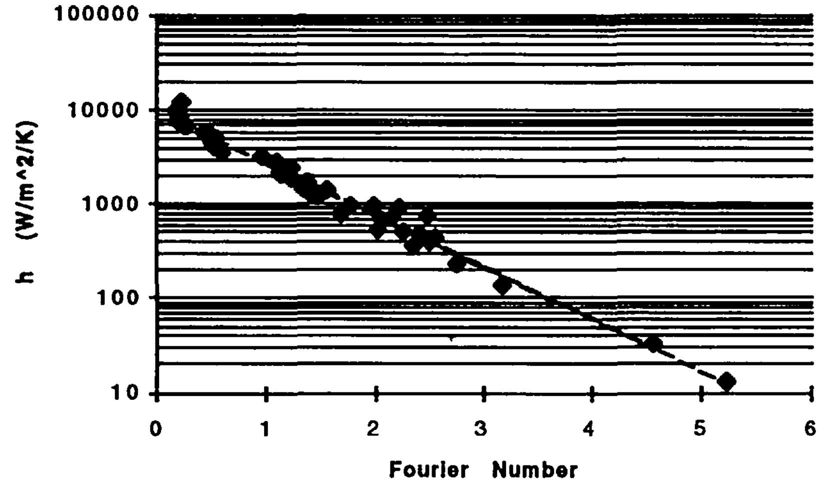

Figure 6

Remark: Despite the relative complexity of our calibrator (square shape, hollow channels), our results show a similar trend as the one obtained with a flat plate being extruded at constant velocity[16].

CASE STUDY

The above correlation was tested in a complex industrial extrusion problem, i.e. the cooling of a window frame profile. *Figure 7* displays the profile cross-section and the mesh used in the simulation. We show in *Figure 8* a typical temperature field that can be obtained with our method. This view corresponds to the temperature field at the first calibrator outlet. The slight irregularities in the cooling channel contour result from the optimization method used to represent them (discretization error).

A comparison between the temperature prediction and the experimental data is presented in *Table 2*. As it was impossible to install threaded thermocouples inside the calibrators used on the production line, temperature measurements were taken with an infrared thermometer on the right-hand side of the profile vertical wall at the inlet and the outlet of the three calibration sections. Accessibility, the presence of a flat surface with a constant temperature (see *Figure 8*) and the lack of internal walls guided our choice. We recall here that accurate temperature measurements with infrared pyrometers can only be made when the measurement area is at constant temperature.

In *Table 2*, it can be seen that the agreement between the various results is fairly satisfactory, except at the outlet of the third calibration section. This is undoubtedly due to a poor evaluation of the heat transfer coefficient in this area, which is believed to originate from the profile shrinkage. Indeed, the shrinkage phenomenon is not taken into account during the simulations although the calibrators are designed to take care of this shrinkage all along the cooling path. On the other hand, the third calibration unit is designed slightly larger than the part specifications to avoid the profile jamming inside. The adherence of the profile is then not as good as it is in the first cooling units. At this point, the high rigidity of the part is stronger than the vacuum applied to make the profile "sticking" to the calibrator walls. This phenomenon entrains a drastic reduction of the cooling efficiency, hence a measured temperature higher than the predicted one. We must also remember the earlier assumption of no heat flux at the interface profile/hollow cavity used as boundary condition (see *Figure 2*). This cavity which is not taken into account in the simulations, is an heat source which could be partly responsible for the deviation.

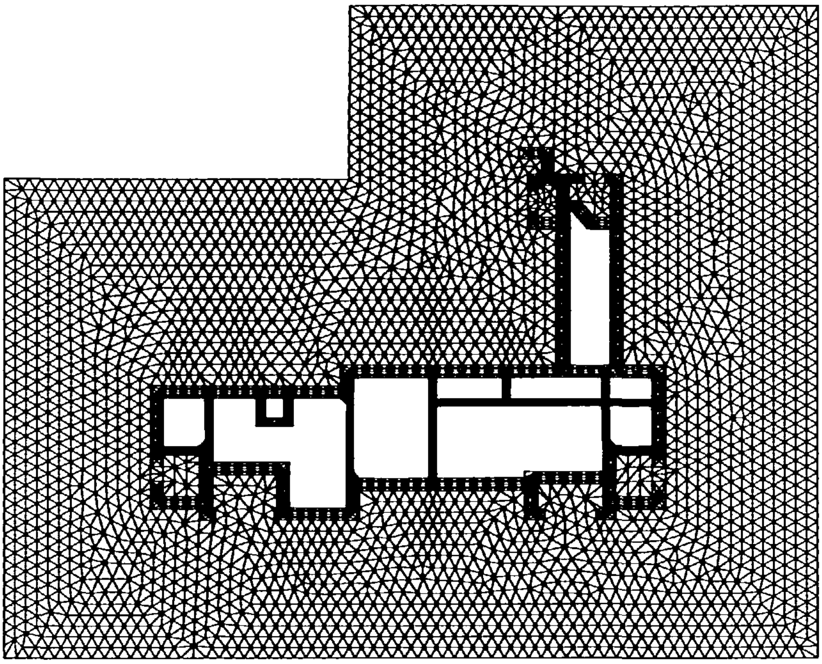

Figure 7

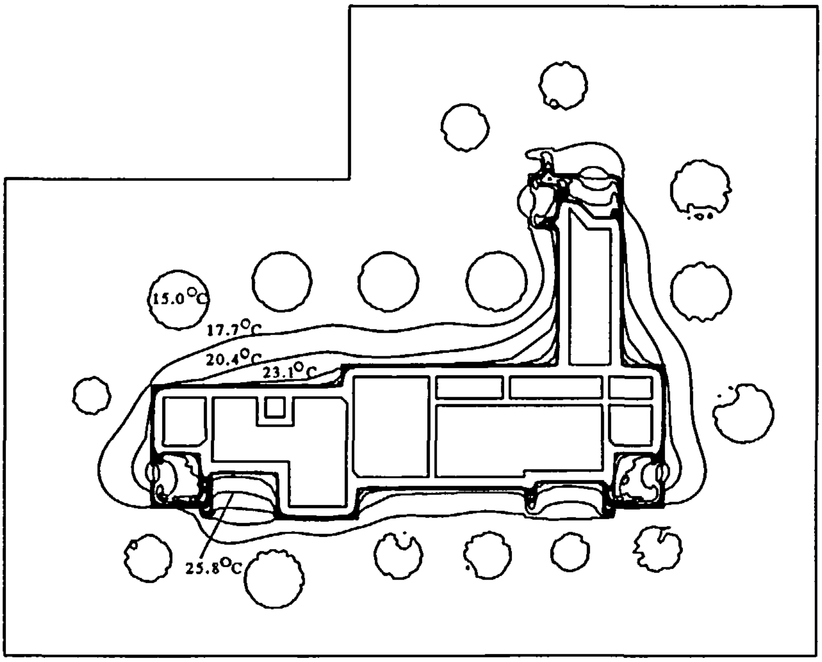

Figure 8

Table 3 Comparison of temperature difference between the inlet and the outlet of every calibration unit

	Measured difference (°C)	Calculated difference (°C)	Deviation (%)
First calibrator	−143.8	−142.1	−1.2
Second calibrator	−19.5	−20.0	+2.6
Third calibrator	−10.6	−15.6	+47.2
Complete line	−163.3	−170.4	+4.3

For sake of assessing the importance of considering a variable heat transfer coefficient, simulations were also performed with constant heat transfer value in each calibrator (*Table 2*). The average value of h used in the modelling was calculated with the above correlation for an average value of the Fourier number calculated at the inlet and outlet of every calibration section. Temperature results are fairly similar to the measured ones. Such results could be expected since typical Biot number values in extrusion processes are low (ranging from 1 to 25). In these situations, the polymer conductivity has much more influence on the cooling phenomenon than heat transfer at the interface. The average value calculated from the correlation provided a reasonable value for the simulation without any trial and error process[4,5,14]. In this case, the proposed correlation revealed itself as a performing tool in providing a way to estimate what should be the constant heat transfer coefficient value to be used for the design of each calibration section.

We show in *Table 3* these results in another perspective. Here, the temperature difference between inlets and outlets of every cooling unit is compared to the simulated value. This way of manipulating the results is more representative of the correlation capability to reflect the evolution of heat transfer along the extrusion axis. Here again a very good fit between the predictions and the measurements can be observed. As for the third calibrator, the error may seem large. However, considering the overall cooling duty, it comes out that the third unit accounts for only 6% of the total cooling process (10 degrees as compared to 163.3 degrees). This fact explains why the overall cooling calculated for the complete line is very satisfactory, which proves the adequacy of the heat transfer correlation.

CONCLUDING REMARKS

A correlation has been developed to describe the heat transfer at the interface between a calibrator and an extruded profile in the profile extrusion process. This correlation has been further used in the simulation of an industrial calibration unit, based on the Menges model. The predicted temperatures and the measurements exhibit a fairly good agreement, paving the way to calibrator optimal design.

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REFERENCES

1 Hensen, F., Knappe, W. and Potente, H. *Plastics Extrusion Technology*, Hanser Publishers, Munich (1988)
2 Ast, W. A simple method of designing cooling sections in plastics extrusion, *Kunststoffe*, **69**, 186–193 (1979)
3 Kurz, H. D. Fixing the dimensions of extruded high-MW polyolefin pipes during cooling, *Kunststoffe*, **78**, 1052–1058 (1988)
4 Worthberg, J., Haberstroh, E., Lutterbeck, J., Masberg, U., Shmidt, J. and Targeil, G. Designing of extrusion lines, *Advances in Polymer Technology*, **2**, 75–106 (1982)

5 Menges, G., Haberstroh, E. and Janke, W. Systematic lay-out of cooling lines for film, sheet and pipe extrusion plant, *Kunststoffe*, **72**, 332–336 (1982)
6 Haberstroh, E. and Menges, G. Design of cooling sections in extrusion lines, Technical Paper, *ANTEC*, **29**, 200–204 (1983)
7 Kleindienst, V. Effective parameters for vacuum calibration of extruded plastic pipes, *Kunststoffe*, **63**, 7–11 (1973)
8 Menges, G., Kalwa, M. and Schmidt, J. FEM simulation of heat transfer in plastics processing, *Kunststoffe*, **77**, 797–803 (1987)
9 Menges, G. and Kalwa, M. Heat transfer calculations in extrusion and welding, *Kunststoffe*, **79**, 37–40 (1989)
10 Sheehy, P., Tanguy, P. A. and Blouin, D. A finite element model for complex profile calibration, *Polym. Eng. Science*, **34**, 650–656 (1994)
11 Dannelongue, H. H. and Tanguy, P. A. An adaptive remeshing technique for non-Newtonian fluid flow, *Int. J. Num. Meth. Eng.*, **30**, 1555–1567 (1990)
12 Mavridis, H., Hrymak, A. N. and Vlachopoulos, J. Finite element simulation of stratified multiphase flows, *AIChE J.*, **33**, 410–422 (1987)
13 Sheehy, P., M.Sc. Thesis Laval University (1992)
14 Choplin, L., Rapport projet rhéologie, projet: *Conception de filières de profilés de PVC*, rapport interne (1991)
15 Haberstroh, E. Heat transfer: *Encyclopedia of Polymer Sci. and Eng.*, **7**, 665–685 (1987)
16 Mayer, A. Criteria for the design of extrusion systems, Advances in Polymer Technology, **4** (291–306)
17 Agassant, J.-F., Avenas, P., Sergent, J. Ph. and Carreau, P. J. *Polymer Processing: Principles and Modeling*, Oxford University Press, New York (1991)