International Journal of Thermal Sciences 127 (2018) 288–293

Contents lists available at ScienceDirect

International Journal of Thermal Sciences

journal homepage: www.elsevier.com/locate/ijts

Introducing flow architecture in the design and optimization of mold inserts cooling systems

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ARTICLE INFO

Keywords:
Mold insert cooling
Flow architecture
Constructal principle
Flow resistance
Svelteness

ABSTRACT

Mold cooling is a critical phase of the molding cycle, comprising more than half of its total time. It affects the final quality of the plastic part and production rate. Therefore, the cooling channel system design is essential to achieve better control over the cycle time. Despite the improvements of conformal cooling, the flow configuration is still absent from cooling optimization criteria. In this work, using the constructal principle, we introduce in these criteria the flow systems svelteness as a global geometric design parameter expressing its architecture. In the configuration explored for cooling small-scale mold inserts, the flow enters through a primary channel and returns through secondary channels, equally spaced, similar to an umbrella shape. The evolution of flow configuration toward minimizing the flow resistance (local pressure losses) points to an architecture with smaller secondary channels and larger return angles. Depending on thermal requirements, it is possible to define optimal regions of interest presenting the cooling systems designer with an appropriate optional range.

1. Introduction

The mold cooling stage is one of the most important in the molding cycle. It plays a crucial role in injection molding by taking more than half of the molding cycle time and affecting the plastic part final quality and mold productivity [1]. It begins after the compaction stage when the plastic at entry point solidifies and ceases to flow into the mold. The time taken before we can eject the plastic part depends on the desired ejection bulk temperature and the heat removal capability of cooling the mold [2]. Therefore, an optimized design of cooling channels is essential to ensure a reduced cooling stage to increase productivity and maximize the quality of plastic parts.

One of the most challenging topics in the injection of plastic in a mold is the hardening of small thin wall plastic parts with complex geometries that increase the difficulty in designing appropriate cooling channels. The use of *Additive Manufacturing* allows a more precise control of mold design, and a better adaptation of cooling channels to highly complex geometries, known as conformal cooling [3,4]. The result is a more uniform cooling of the plastic part, which in turn reduces cooling time [5].

The manufacturing of inserts by additive manufacturing is capable of producing highly intricate geometries [4]. But, while we recognize the positive impact of conforming cooling channels to the mold insert, the optimization of their configuration is still semi-empirical and recurs to simulations tools.

When minimizing systems or components, the methodologies developed aim to improve mold inserts cooling without affecting the plastic part's characteristics. And the tendency to miniaturization in flowing systems points towards vascularization, as observed in nature. Through the application of constructal design, the configuration of channels should evolve to suit the inserts complex geometries better and achieve the intended thermal balance between the mold insert and heat dissipation requirements to form the plastic parts.

Therefore, the primary goal of optimizing the cooling system is to increase inserts thermal performance and improve the plastic part's quality, as well as the mold production rate. However, the flow architecture is still not a design parameter, not even included as one of the cooling system optimization criteria. This is why the goal of this work is an attempt to include the flow configuration in the design phase through an approach based on the constructal principle [6].

2. Cooling channels thermofluids model

"For a finite-size system to persist in time (to live), it must evolve in such a way that it provides easier access to the imposed currents that flow through it." [6]. This is the constructal principle. One of the novelties associated with a constructal theory approach is the global geometric property associated with the flow system designated as *Svelteness* first introduced

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https://doi.org/10.1016/j.ijthermalsci.2018.01.035
Received 31 October 2017; Received in revised form 20 December 2017; Accepted 26 January 2018
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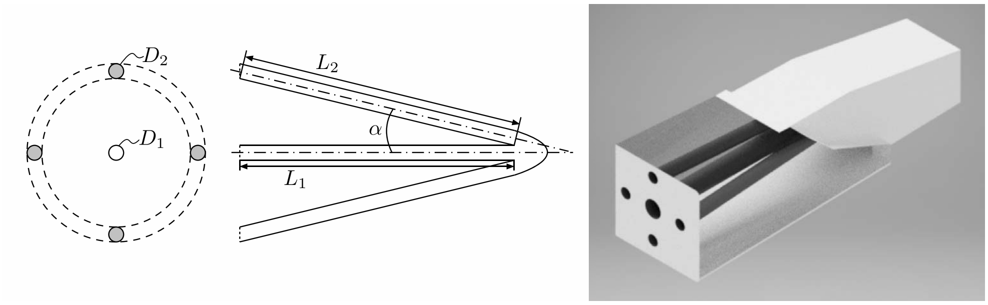

Fig. 1. Schematic example of the *umbrella* geometry with $n = 4$ return channels and the relevant geometrical parameters. And an example of its application in a mold insert.

by Lorente and Bejan [7]. The *Svelteness* is a parameter relating an external flow length scale with an internal one. In channel flows, it is possible to formulate this global geometric property as a function of the relative importance of local pressure losses in bends, junctions, contraction and expansions and friction pressure losses distributed along the channels. The purpose of formulating this relation is to find the geometric conditions where local pressure losses become negligible relatively to friction losses.

Given the typical shapes of mold inserts to produce plastic parts for the electronic industry, the configuration used to explore the introduction of the flow architecture in cooling optimization criteria has a shape similar to an *umbrella*. It cools the insert volume through a central channel until a tip where n channels depart making the fluid flow return to the initial sectional point of entry (see Fig. 1).

The first part of this section considers the flow development and establishes a relation between the ratio of local to distributed pressure losses and the *Svelteness*. And, in the second part, considerations made for the heat transfer involved allow introducing the flow architecture in the heat transfer relations which cooling system's designers may use to choose the configuration that best meets heat removal requirements in a mold insert.

2.1. Characterizing the flow architecture through its svelteness

Following a mass and momentum conservation, and using the same approach as Bejan and Lorente [6] (pp. 113–115), we apply a Lagrange multipliers method to find the condition of minimum flow resistance in both primary (D_1) and secondary (D_2) channels. This relation results in

$$\frac{D_1}{D_2} = n^{1/3} \tag{1}$$

where n is the number of secondary channels returning the flow. In the case of a single bifurcation $n = 2$, the result corresponds to the known Hess-Murray rule.

Using again the conservation of mass and momentum, considering a laminar flow and using the Darcy friction factor for circular channels ($f = 64/\mathrm{Re}_D$), which is a consequence of the Poiseuille flow, the resulting relation between local and friction pressure losses becomes

$$\frac{\Delta p_{local}}{\Delta p_{distr}} = \Lambda(n,\alpha)\cdot\frac{\mathrm{Re}_{D_1}}{64\left(\frac{L_1}{D_1}\right)} \tag{2}$$

with

$$\Lambda(n,\alpha) = \frac{1 + n^{-2/3} - 2\cos(\alpha)n^{-1/3}}{1 + n^{1/3}\sec(\alpha)}$$

In the result above, local pressure losses depended on the local pressure at the entrance of the configuration in D_1 and the term associated with the dynamic pressure. However, when modeling the local pressure losses through the momentum conservation, which mathematical developments are relatively complex, the terms associated with the static and dynamic pressures depend on different geometric factors,

$$\Delta p_{local} = \left(g(n,\alpha)\cdot p_1 + \frac{1}{2}\rho V_1^2\right) l(n,\alpha) \tag{3}$$

with

$$g(n,\alpha) = \frac{1 - n^{-1/3}\cos(\alpha)}{1 + n^{-2/3} - 2\cos(\alpha)n^{-1/3}}$$

and

$$l(n,\alpha) = 1 + n^{-2/3} - 2\cos(\alpha)n^{-1/3}$$

While the order of magnitude of $g(n,\alpha) \sim 1$, the static pressure at the entrance is $p_1 \sim 10^5$. A scale analysis of the dynamic pressure term based on $\mathrm{Re}_D \sim 10^3$ shows $\frac{1}{2}\rho V_1^2 \sim 10^7$. Thus, Eq. (2) also results from the assumption that the pressure term in the momentum conservation is negligible relatively to the dynamic pressure associated with the flow velocity.

To ensure negligible local head losses based on the order of magnitude of the Svelteness (Sv), we need to relate the ratio in Eq. (2) with Sv, defined as the relation between an *external length scale* (L_e) and an *internal length scale* (L_i),

$$\mathrm{Sv} = \frac{L_e}{L_i} \tag{4}$$

The relation $L_e = \sqrt{A_c}$ expresses the external length scale associated with the total area occupied by the construct (A_c) and expresses the internal length scale depends on the total volume of the channels (V_c).

Fig. 2 shows the total area occupied by the umbrella construct, including some geometrical considerations to formulate A_c. This space depends on two specific areas. One around the area between the inlet channel and each outlet channel, A_{c1}, and a second area defined between outlet channels (A_{c2}). Some characteristic parameters are the:

- distance between inlet and outlet sections: $b = L_1\tan(\alpha)$
- distance between consecutive outlet sections: $c = 2L_1\tan(\alpha)\sin(\gamma/2)$
- angle between consecutive distances defined by the inlet and outlet sections: $\gamma = \frac{2\pi}{n}$
- half-angle of A_{c2} close to the umbrella's tip: $\beta = \arcsin(\sin(\alpha)\sin(\gamma/2))$
- height of A_{c2}: $h = L_1\tan(\alpha)\sin(\gamma/2)$

The final result for the Svelteness is a function of the flow architecture through the divergence angle α, the number of channels n and the length-to-diameter ratio of the primary channel (L_1/D_1).

$$\mathrm{Sv} = \left(\frac{L_1/D1}{\psi(n,\alpha)}\right)^{2/3} \tag{5}$$

with

$$\psi(n,\alpha) = \sqrt{\frac{\frac{\pi}{4}(1 + n^{1/3}\sec(\alpha))}{\left[\tan(\alpha)\left[n\left(\sin(\pi/n) + \frac{\cot(\alpha)}{2}\right)\right]^{1/2}\right]^3}}$$

Expressing the ratio $\frac{L_1}{D_1}$ as a function of the Svelteness and using this in

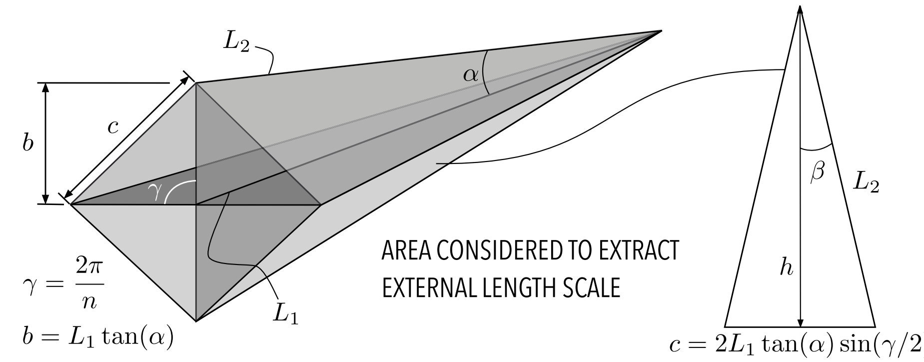

Fig. 2. Total area occupied by the umbrella construct.

Eq. (2) we finally obtain

$$\frac{\Delta p_{local}}{\Delta p_{distr}} = \frac{\Lambda(n, \alpha)}{64\psi(n, \alpha)}\left(\frac{\mathrm{Re}_D}{\mathrm{Sv}^{3/2}}\right) \tag{6}$$

The ratio between local and distributed head losses is the criterion used for optimizing the flow configuration. The physical modeling considers a laminar regime, since the risk of phenomena, such as cavitation, is higher in the turbulent regime. By ensuring that local losses are much lower than distributed losses due to friction, $\frac{\Delta p_{local}}{\Delta p_{distr}} \ll 1$, in practice, their effect becomes negligible. In this sense, the following analysis assumes local losses should be at least an order of magnitude lower than distributed losses, thus

$$\frac{\Delta p_{local}}{\Delta p_{distr}} = \Delta p^*_{crit} < 0.1 \tag{7}$$

And since the *Svelteness* is a geometrical property of the flow related to pressure losses, we can use it as the criterion to evaluate the flow efficiency.

2.2. Introducing the flow architecture in heat transfer relations

The plastic part requires a heat removal quantified as

$$Q_P = m_P\, c_{p,P}\, (T_{inj} - T_{ejt}) \tag{8}$$

where m_p corresponds to its mass, $c_{p,P}$ to the specific heat of the material, T_{inj} and T_{ejt} correspond to the plastic injection temperature and the ejection temperature of the fully formed plastic part, respectively. Optimizing the molding cycle implies reducing, as much as possible, the cooling period after compaction of the plastic injected, Δt_c. Reducing a fraction of this time interval in millions of plastic parts produced is significant. By equating heat dissipation requirements (Q_P) with the cooling potential (q_c) of the geometry, $q_c = q_P \Leftrightarrow q_c = \frac{Q_P}{\Delta t_c}$, it is possible to obtain the required temperature difference between mold surface (T_s) and coolant mean temperature (T_m): $\Delta T_{sm} = T_s - T_m$. This is an important quantity, since it translates the cooling potential in each cross-section of a channel in the configuration.

With Eq. (1) and the definition of Nusselt, we arrive at the equality $D_1 h_1 = D_2 h_2 = \mathrm{Nu}{\cdot}k_c$, where k_c corresponds to the coolant thermal conductivity. Thus, assuming a small enough difference between the inlet and outlet flow temperatures, the thermal energy removed in a section area in the entire circuit length during a production cycle (q_P) includes the primary channel (D_1, L_1) and n secondary channels (D_2, L_2): $q_c = q_1 + n\, q_2$, where

$$q_1 = \pi D_1 L_1 h_1 (T_s - T_m) = \pi \mathrm{Nu}{\cdot}k_c L_1 \Delta T_{sm} \tag{9}$$

$$q_2 = \pi n D_2 L_2 h_2 (T_s - T_m) = \pi \mathrm{Nu}{\cdot}k_c n{\cdot}L_1 \sec(\alpha) \Delta T_{sm} \tag{10}$$

considering that $L_2 = L_1 \sec(\alpha)$. Therefore, expressing this relation for the temperature difference between mold surface and fluid mean temperature in the channels (ΔT_{sm}) results in

$$\Delta T_{sm} = \frac{q_P}{\pi\ \mathrm{Nu}\ k_c\ L_1 (1 + n\sec(\alpha))} \tag{11}$$

We assume a steady state flow with a uniform heat flux as the boundary condition on the inner surface of each channel. Thus, the difference between the mold surface temperature and the mean fluid temperature is constant in the above formulation. The maximization of heat transfer depends on the largest possible number of secondary channels. Under the criterion of a minimum distance between secondary channels equal to their diameter (D_2) to ensure the mold-insert structural integrity, the maximum number of channels results in

$$n_{max} = floor\left[\frac{\pi L_1 \tan(\alpha)}{D_2}\right] \tag{12}$$

where *floor* represents a function rounding the result to the nearest integer less than or equal to that value. Fig. 3 exemplifies the application of this criterion for a fixed length of the primary channel and diameter of secondary channels, while considering three different return flow angles.

The expression for cooling potential of the umbrella configuration regarding enthalpy variation between the inlet and outlet of the flow is typically equated as

$$q_c = \dot{m}_c\, c_{p,c}\, \Delta T_m \tag{13}$$

Here, $\Delta T_m = T_{m,o} - T_{m,i}$ is the difference between outlet and inlet mean temperatures. We can relate q_c formulated from the enthalpic variation with forced convection in the channel section area. In this way, it is possible to define the ratio $\frac{\Delta T_m}{\Delta T_{sm}}$ as a functional relation of forced convection (St) with geometry ($\psi(n, \alpha)$) and configuration (Sv) through Eq. (5).

$$\frac{\Delta T_m}{\Delta T_{sm}} = 4\ \mathrm{St}\ \psi(n, \alpha)\ \mathrm{Sv}^{3/2} \tag{14}$$

The expression above includes the Stanton number, $\mathrm{St} = \frac{\mathrm{Nu}}{\mathrm{Re}_D \mathrm{Pr}}$ which measures the ratio of heat transferred to a fluid and its thermal capacity. And the dependence on the *Svelteness* (Sv) finally introduces the flow architecture in the heat transfer phenomena involved.

The temperature differences ΔT_{sm} and ΔT_m are essential factors to evaluate the validity of the cooling stage within the production cycle. The next section explores the effect of geometric parameters on the *Svelteness* and their implications for heat transfer.

3. Evolution of flow architecture in the umbrella configuration

The underlying idea of exploring a constructal approach is to devise an assessment strategy of how should a flow architecture evolve. The aforementioned global geometric property allowing this assessment is the Svelteness. According to Bejan [6], the evolution of a flow configuration increases the value of the Svelteness. Thus, it is essential to investigate the effect of geometric parameters on this parameter.

In micro-mold cooling with the umbrella configuration, there are

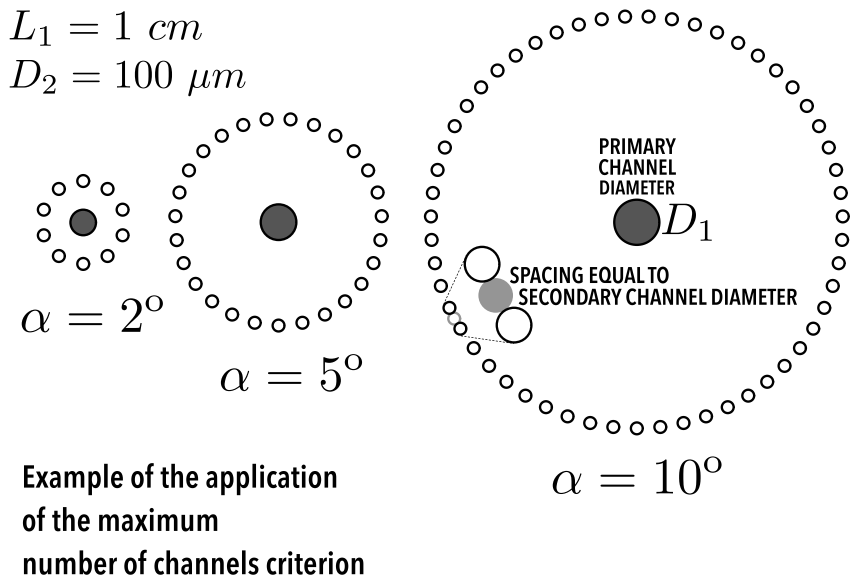

Fig. 3. Application of the criterion for the maximum number of channels with $L_1 = 1\,cm$, $D_2 = 100\,\mu m$ and three return flow angles ($\alpha = 2°, 5°, 10°$).

three independent geometric design parameters:

1. the size of secondary channels (D_2);
2. the return flow angle (α) of those channels;
3. and the length of the primary channel (L_1).

The size range for secondary channels (D_2) considered lies between $50\,\mu m$ and $500\,\mu m$. The return flow angle (α) varies from 1° to 10°. And the range for the length of the primary channel L_1 is between $0.5\,cm$ and $2.5\,cm$. Using Eq. (6) with the criterion that $\Delta p^*_{crit} \lesssim 0.1$, we calculate a minimum value for the Svelteness. For any $Sv > min\{Sv\}$, local pressure losses are negligible compared to friction losses in one order of magnitude.

Fig. 4 shows the variations induced by the return angle α and the diameter of secondary channels D_2 for a specified length of the primary channel L_1 on the minimum Svelteness. As earlier stated, an increasing Svelteness shows the direction in which the configuration should evolve. Therefore, results point to flow architectures with smaller secondary channels at higher return flow angles.

Fig. 5 show the effect of the primary channel length (L_1) for the smallest secondary channel size considered ($D_2 = 50\,\mu m$) and highest return angle ($\alpha = 10°$). Results evidence how altering this length does not alter the direction of how the Svelteness increases, thus, how the configuration evolves in its path of increasingly lower flow resistance.

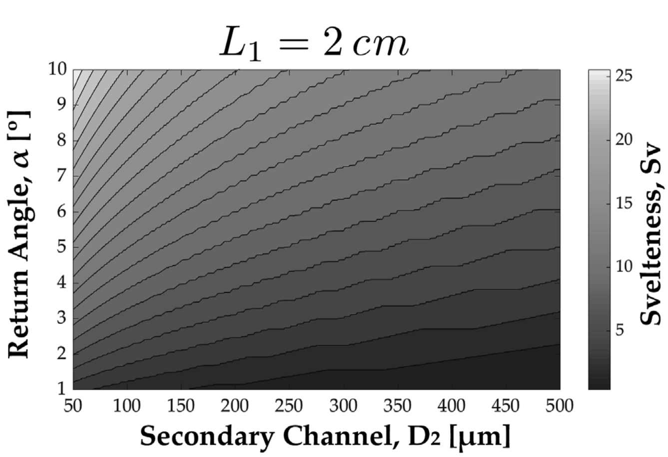

Fig. 4. Minimum *Svelteness* value for $1° \le \alpha \le 10°$, $50 \le D_2 \le 500\ [\mu m]$ and $L_1 = 2\,cm$.

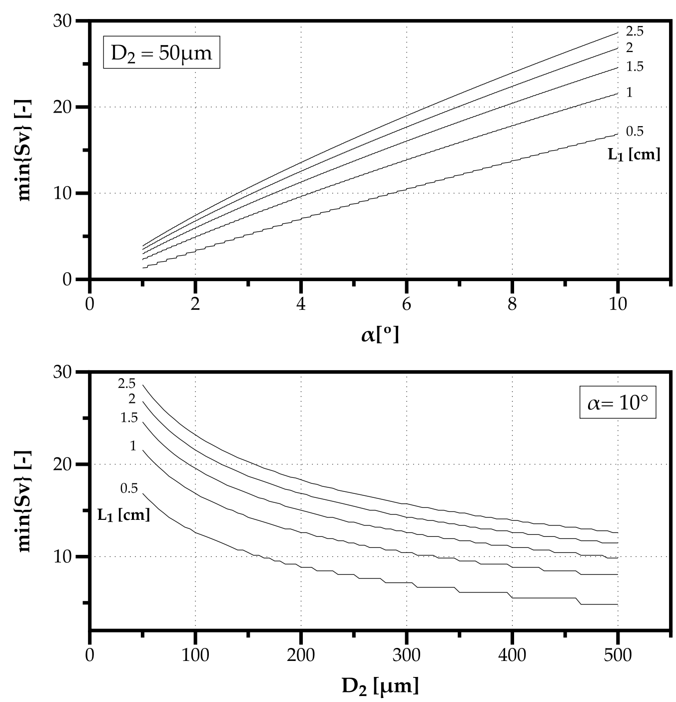

Fig. 5. Effect of the Primary channel length on the minimum *Svelteness* value for $D_2 = 50\,\mu m$ (top) and $\alpha = 10°$ (bottom).

Once the Svelteness provides information on the direction of evolution of the Umbrella configuration, it is important to relate it with the heat transfer involved in the cooling process.

4. Application of thermofluids model to a plastic part

Taking into account the considerations made in section 2.2, the implications of the evolving direction pointed by the Svelteness for heat transfer consider the case-study of a plastic part with the following characteristics:

- Plastic specific heat, $c_{p,P} = 1300\ [J\ kg^{-1}K^{-1}]$.
- Part weight, $m_P = 1[g]$.
- Plastic injection temperature, $T_{inj} = 280[°C]$.

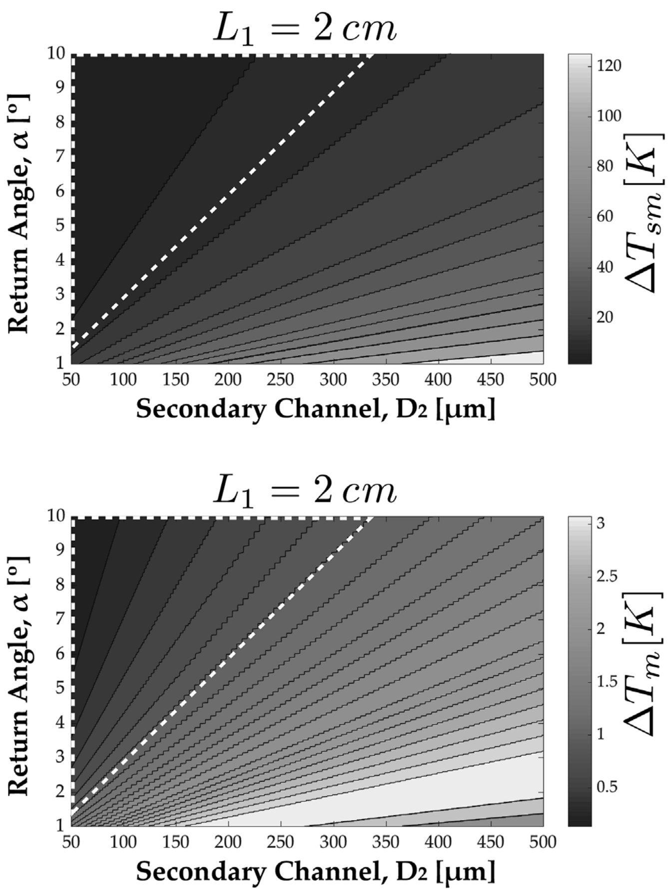

Fig. 6. TOP: ΔT_{sm} value for $1° \le \alpha \le 10°$, $50 \le D_2 \le 500$ $[\mu m]$ and $L_1 = 2\,cm$; BOTTOM: ΔT_m value for $1° \le \alpha \le 10°$, $50 \le D_2 \le 500$ $[\mu m]$ and $L_1 = 2\,cm$.

- Part ejection temperature, $T_{ejt} = 120[°C]$.
- Cooling time, $\Delta t_c = 3$ $[s]$.

The assessment of these implications considers the effect of the flow configuration in two temperature differences obtained from Eqs. (11) and (14). Fig. 6 shows the results obtained for these temperature differences. Menges [8] states the coolant temperature rise between inlet and outlet of the cooling circuits should not exceed $3\,K$ to $5\,K$ to ensure adequate homogeneous cooling along the cooling circuit. However, Kazmer [2] is stricter and state the usual temperature difference between outlet and inlet in mold cooling should be $1\,K$. Thus, the area marked with a dashed line corresponds to values where $\Delta T_m \lesssim 1\,K$, which is the order of magnitude found in other similar applications.

The demarcation in this region of interest aims at providing the cooling system's designer with an appropriate optional range. For example, a technique inserting circular channels in micro-mold inserts may have limitations regarding the minimum size of secondary channels for a given return flow angle. This optional range informs the designer about the compromise between manufacturing limitations and optimal thermal performance.

The same area is included in the results for the temperature difference in the channel's section area ΔT_{sm} (Fig. 6:TOP). The maximum values in this area for ΔT_{sm} are in the interval of $11.4 - 11.57\,K$ which is reasonable in these applications. The results for other lengths of the primary channel are similar, changing only in the magnitude as evidenced in Fig. 5. A final remark would be the compatibility between the evolution suggested by the approach based on constructal principle, through its global geometric parameter (Svelteness), and the best results regarding heat transfer, validating the applicability of the constructal theory as a design tool in the optimization of thermofluid systems.

5. Concluding remarks

Thermal management in micro-mold inserts is an essential element in the optimization of production cycles of plastic parts. Any improvement in the cycle time may represent a significant increase in production. However, given the complexity of geometries and the size scale of mold inserts, designing cooling circuits is challenging. The best way to uniformly cool a mold insert is to use a conformal design strategy embedding channels where cooling is necessary. This design results in a given flow architecture which does not belong to cooling optimization criteria. This is the main goal of the work presented here.

The approach developed based on the constructal theory introduced by Bejan [6] uses thermofluid principles to investigate how should the flow architecture evolve in time. In constructal theory, the Svelteness is a global geometric parameter characterizing the flow architecture. And an increase of the Svelteness points the direction of evolution of the flow configuration. In this work, we propose a new configuration for cooling mold inserts designated as *umbrella* and use it to introduce the flow architecture in the design of the cooling channels' system. The umbrella configuration consists of a primary central channel which transports the coolant toward the tip of the mold insert and returns the flow backward through angled and uniformly distributed secondary channels.

The first step while developing the physical model is to correlate the ratio between local and friction distributed pressure losses with the Svelteness (Sv). A high Sv evidences the configurations where local pressure losses are negligible compared with friction distributed losses, which is an important feature to discern the evolutionary path of the flow architecture. In the case of an umbrella flow configuration for cooling mold inserts, results show it should evolve toward smaller secondary channels and higher return flow angles.

We evaluate the implications of the umbrella configuration for heat transfer through the criterion of a maximum temperature difference between the coolant inlet and outlet of 1 K, pointing to regions of interest useful from the design point of view. The physical model presented in this work is the subject of ongoing numerical simulations and experiments. Thus, future work includes experiments on mold inserts made with Selective Laser Melting (SLM) additive manufacturing to assess the relation between the flow architecture evolution direction suggested by the Svelteness increase and the uniformity degree of the temperature distribution in the mold insert.

Nomenclature

A_c	Total area occupied by the construct $[m^2]$
A_{c1}	Area between primary and secondary channels $[m^2]$
A_{c2}	Area between secondary channels $[m^2]$
b	Distance between primary and secondary sections $[m]$
c	Distance between consecutive secondary sections $[m]$
$c_{p,c}$	Coolant specific heat $[J\cdot kg^{-1}K^{-1}]$
$c_{p,P}$	Plastic part specific heat $[J\cdot kg^{-1}K^{-1}]$
D_1	Diameter of primary channel $[m]$
D_2	Diameter of secondary channel $[m]$
f	Friction factor
h	Height of A_{c2} $[m]$
h	Heat transfer coefficient $[W\cdot m^{-2}K^{-1}]$
k_c	Coolant thermal conductivity $[W\cdot m^{-1}K^{-1}]$
L_1	Length of primary channel $[m]$
L_2	Length of secondary channel $[m]$
L_i	Internal length scale
L_e	External length scale
	Mass flow rate $[kg/s]$
m_P	Plastic part mass $[kg]$
n	Number of secondary channels
Nu	Nusselt number
p	Pressure $[Pa\ ]$

Q_P	Plastic part heat $[J]$
q_c	Geometry cooling potential $[W]$
q_P	Thermal energy removed in a section area $[W]$
q_1	Primary channel cooling potential $[W]$
q_2	Secondary channel cooling potential $[W]$
Re_D	Reynolds number
St	Stanton number
Sv	Svelteness
T_{ejt}	Temperature of material ejected of a mold $[^{\circ}C$ or $K]$
T_{inj}	Temperature of material injected in a mold $[^{\circ}C$ or $K]$
T_m	Coolant mean temperature $[^{\circ}C$ or $K]$
T_s	Mold surface temperature $[^{\circ}C$ or $K]$
V	Fluid velocity $[m/s\ ]$
V_c	Total volume of channels $[m^3]$

Greek symbols

α	Return angle $[^{\circ}]$
β	Half-angle of A_{c2} at the tip of the umbrella $[^{\circ}]$
Δt_c	Cooling period $[s]$
ΔT_m	Temperature difference between outlet and inlet mean temperatures $[^{\circ}C$ or $K]$
ΔT_{sm}	Temperature difference between mold surface and coolant mean temperature $[^{\circ}C$ or $K]$
Δp_{local}	Local pressure loss $[Pa]$
Δp_{distr}	Friction pressure loss $[Pa]$
γ	Angle between consecutive distances defined by the primary and secondary channels $[^{\circ}]$
ρ	Density $[kg/m^3]$

Acknowledgements

The authors would like to acknowledge project COOLGRAFENO (POCI-01-0247-FEDER-017883) co-financed by the program PT2020 Co-Promotion R&D for supporting this work and the fellowship of M.R. Clemente.

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