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1. Abstract

The air/PCM heat exchanger described in this paper consists in horizontal PCM plates separated by an air flow. This is a storage system dedicated to insure a better summer thermal comfort. It stores freshness during the night and releases it in the house during the day. The purpose of presented work is to contribute in the improvement of the modeling of air / PCM convective exchanges. In this kind of exchanger, we are confronted to different flow patterns, which must be taken into account in the choice of the convective coefficient.

So a detailed study of flow patterns encountered in the exchanger is first performed and appropriate correlations from the literature are selected. These one are tested on a numerical model of the heat exchanger. The results are compared to experimental data. Here, a rigorous choice of the correlations, according to the flow patterns, does not seem to improve significantly the numerical results comparing to the use of a single correlation representative of the most covering flow structure. Other model errors must be tackled before we can appreciate the improvement due to the correlation choice.

Keywords: air/PCM exchanges, exchanger, convection, simulation, experimental results

Nomenclature

b	air gap thickness [m]	Re_{Dh}	Reynolds number
D_h	hydraulic diameter [m]	u_0	mean velocity of the fluid [m.s^{-1}]
f	friction factor	x_c	distance from which turbulent flow starts
h	convective heat transfer coefficient [W.m^{-2}.K^{-1}]		
L	length of the exchanger [m]		
L_m	entrance length [m]	λ_{air}	thermal conductivity of the air [W.m^{-1}.K^{-1}]
Nu_{D_h}	Nusselt number	ν	kinematic viscosity [m².s^{-1}]
Pr	Prandtl number		

2. Introduction

In order to better understand and optimize air / PCM heat exchangers' performances, developing numerical models is necessary. The models should take into account precisely the phenomena such as the PCM behavior or the convection coupling, in order to predict accurate heat flux. Through the numerical models, the influence of several parameters on performances can be carried out. The tested parameters are for instance air gap, PCM plate's thickness, exchanger length, or the convection coefficient itself. Dolado et al. [1] have tested many parameters including convective heat transfer coefficient with a numerical model. This model was validated comparing experimental power measurement to simulated power using the Gnielinski correlation (given below) to calculate the

Nussel number. This study shows that an increase in the convection coefficient leads to higher power peak and shorter melting time. Hed et al. [2] found out the same result. Yet, unlike Dolado et al. [1] the Nusselt number used for turbulent flow is calculated using the Colburn correlation (given below).

Thus, those studies have pointed out the influence of convective heat transfer coefficients on air / PCM heat exchangers' performances. However, no improvement in the knowledge of real coefficient has been performed in those publications. Only Garcia et al. [3] established an experimental correlation of the convective heat transfer coefficient. But their correlation corresponds only to separated boundary layers. Yet, convective heat transfer coefficient affects the performances since the exchanges between air and PCM plates are strongly linked to this coefficient. We can suggest a better knowledge of this coefficient could improve the accuracy of the model. Thus, the choice of the correlation must be done carefully, based on the flow patterns encountered in the exchanger. Moreover, in some cases the convective heat transfer coefficient can be determined locally.

The purpose of this work is to investigate the effect of the correlation used for convective heat transfer coefficients on the accuracy of the model. Thus, a study of flow patterns encountered in air/PCM heat exchanger with flat plates and corresponding correlation of convective heat transfer coefficients is first produced. Outcome of the model with different correlation is then compared to experimental data.

3. Materials and method

3.1. Theory and methodology for the study of flow patterns

The convection inside the air / PCM heat exchanger is forced internal convection. Thus, the convective heat transfer coefficient h is linked to the Nusselt number Nu_{D_h} through the equation (1):

$$Nu_{D_h} = \frac{hD_h}{\lambda_{air}} \quad (1)$$

The Nusselt number formulation depends on the speed profile (boundary layers or fully developed flow region) and flow regime between the flat plates.

When the fluid flows between parallel plates, the fluid velocity is null at the wall and boundary layers appear. The thickness of the boundary layers is defined as the distance from the wall to the point where the fluid velocity is equal to 99% of the velocity in the middle of the air gap. The boundary layers grow along the exchanger and merge at a distance L_m as displayed in Fig.1. The entrance length L_m can be expressed as given in equation (2). The value of convective heat transfer coefficient is different whether the boundary layers are separated or fully developed flow region is reached.

$$L_m = \left(\frac{b}{10}\right)^2 \frac{u_0}{\nu} \quad (2)$$

The Nusselt number formulation also depends on the flow regime (laminar or turbulent) determined by the Reynolds number based on the mean air velocity and the pipe section geometry. The transition between laminar and turbulent flow is:

For separated boundary layers, the laminar/turbulent transition is similar to the laminar/turbulent transition over a flat plate (Fig.2). Turbulent regime occurs when $\mathrm{Re}_{x_c} > 5 * 10^5$. The flow is turbulent when the downstream location x is over x_c defined as:

$$x_c = \frac{5*10^5 \nu}{u_0} \quad (3)$$

For a fully developed region, the flow regime is turbulent if $\mathrm{Re}_{\mathrm{Dh}} > 2000$. This is not a constrain on the downstream location, but on the channel width b. The flow is turbulent when b is over b_c defined as:

$$b_c = \frac{1000\nu}{u_0} \tag{4}$$

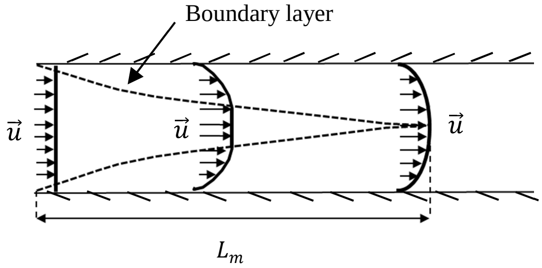

Figure 1 : speed profile in a pipe

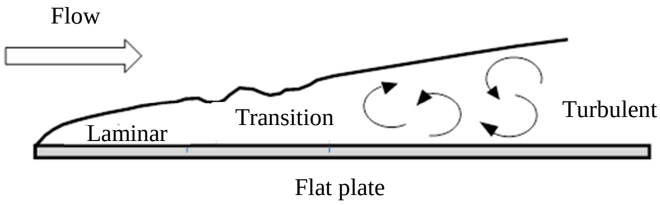

Figure 1 : flow regime over a flat plate

Thus, to target the convection coefficient formulation that best fits the situation, the flow patterns encountered in the PCM/air exchanger must be studied: the speed profile is first determined by comparing the entrance length L_m to the length of the exchanger. Then, for each region (separated boundary layers and fully developed flow region), the flow regime is analyzed based on the Reynolds number.

Then a list of correlations from the literature corresponding to the different flow patterns is established.

3.2. Numerical model and experimental data

Different correlations have been tested through a Matlab model and the results were compared to experimental data.

- The model used for the comparison has been established from two validated models produced by Borderon [4] and Arzamendia [5]. The model takes into account the PCM hysteresis and convection between air and PCM plates.
- The experimental data comes from experiments carried out by Borderon [4]. The air/PCM exchanger is composed of parallel plates and is 1.2m long.

Two sets of experimental data are used for the model validation. Their features are given in Table 1.

Table 1: sets of experimental data

	Type	*Period*	*Flow rate [$m^3.h^{-1}$]*	*Air gap [m]*	*Velocity [$m.s^{-1}$]*
Set 1	*Sine from 0 to 40°C*	*24h*	*330*	*0.018*	*2.12*
Set 2	*Square from 5 to 40°C*	*10h*	*370*	*0.018*	*2.38*

4. Results and discussion

4.1. Flow patterns in the air gap

There are five possible flow configurations represented on Fig. 3. Each configuration depends on the relative position of L, L_m, x_c, and the value of b_c.

The determination of the flow configurations implies four transitions which are summarized in Table 2. The transitions are expressed as functions of the air gap thickness b, the plate length L, and the mean velocity u_0 in the second column of the table, according to equations 2, 3 and 4.

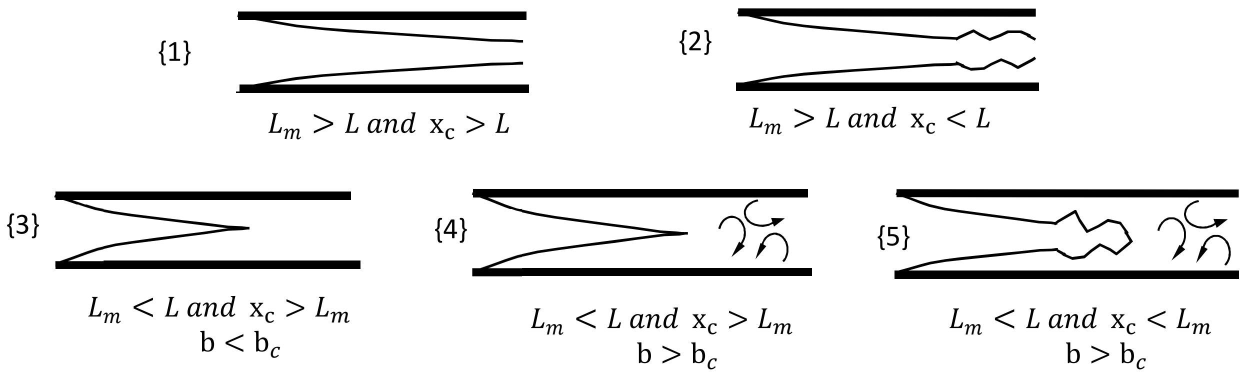

Figure 2 : flow patterns along the air gap

Table 2 : four transitions

(A) $L_m = L$	$b = 10\sqrt{\frac{L\nu}{u_0}}.$	Curve A
(B) $b = b_c$	$b = \frac{1000\nu}{u_0}.$	Curve B
(C) $x_c = L$	$u_0 = \frac{5 * 10^5 \nu}{L}$	Curve C
(D) $x_c = L_m$	$b = 10\sqrt{5 * 10^5}\frac{\nu}{u_0}$	Curve D

For a given length of exchanger, the five configurations can be localized on a b /u_0 mapping. Figure 4 shows the case of a 1.2m long exchanger, corresponding to our experimental data. The mean velocity in air/PCM exchangers is usually lower than 4m.s^{-1}. Thus the configurations which are likely to come across are the configurations {1}, {3} and {4}.

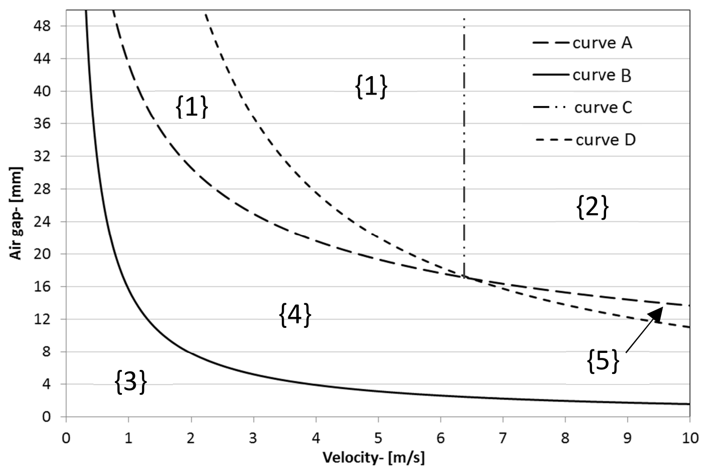

Figure 3 : flow pattern according to air gap and fluid velocity in the case of a 1.2m long exchanger

The two sets of experimental data were obtained with air gap of 0.018m and a velocity around $2.2 m.s^{-1}$. The flow pattern encountered is the configuration {4}: the flow remains laminar when boundary layers are separated; it is directly followed by a turbulent fully developed flow region.

4.2. Nusselt correlations

As long as the boundary layers are separated, the flow is similar to one on a flat plate and the convective heat transfer coefficient can be determined locally or globally.

The correlations depend on flow structure and the thermal boundary condition. Two thermal boundary conditions are proposed by the literature: constant surface temperature or constant surface heat flux. The Table 3 summarizes the available correlations.

Table 3 : convective heat transfer coefficient correlation from litterature

		Separated boundary layers		**Fully developed flow region**	
		Laminar	**Turbulent**	**Laminar**	**Turbulent**
Local	**Constant surface temperature**	Shah	-	-	-
	Constant surface heat flux	Hwang	-	-	-
Global	**Constant surface temperature**	Stephan	Colburn	Nu_{Dh}=7.54	Colburn Gnielinski
	Constant surface heat flux	-	Colburn	Nu_{Dh}=8.24	Colburn Gnielinski

The expressions of the correlations are given below:

Shah correlation (Bejan [6])

$$Nu_x = 7.55 + \frac{0.024\, x^{*-1.14}\left(0.0179 Pr^{0.17} x^{*-0.64} - 0.14\right)}{\left(1 + 0.0358 Pr^{0.17} x^{*-0.64}\right)^2} \qquad x^* = \frac{\frac{L}{D_h}}{Re_{D_h} Pr}$$

Stephan correlation [7]

$$\mathrm{Nu_x} = 7.55 + \frac{0.024\, \mathrm{x}^{*-1.14}}{1 + 0.0358 \mathrm{Pr}^{0.17} \mathrm{x}^{*-0.64}} \qquad \mathrm{x}^* = \frac{\frac{\mathrm{L}}{\mathrm{D_h}}}{\mathrm{Re_{D_h} Pr}}$$

Colburn correlation (Taine [8])

$$\mathrm{Nu_{Dh}} = 0.023 \mathrm{Re_{D_h}}^{0.8} \mathrm{Pr}^{0.23} \text{ ; if } \mathrm{Re} > 10^4, 0.7 < \mathrm{Pr} < 160 \text{ and } \frac{\mathrm{L}}{\mathrm{D}} > 60.$$

Gnielinski correlation [9]

$$\mathrm{Nu_{Dh}} = \frac{{}^{\mathrm{f}}/_{2}\left(\mathrm{Re_{D_h}} - 10^3\right)\mathrm{Pr}}{1 + 12.7\left({}^{\mathrm{f}}/_{2}\right)^{\frac{1}{2}}\left(\mathrm{Pr}^{2/3} - 1\right)} \qquad \mathrm{f} = 0{,}078 * \mathrm{Re_{D_h}}^{-\frac{1}{4}}$$

Hwang — Hwang table given by Taine [8]

As said, the experimental data were found corresponding to the configuration {4}. Thus, in the first part of the exchanger boundary layers are separated and the flow is laminar. The convective coefficient can be determined locally through the Shah correlation.

At the end of the exchanger, turbulent fully developed flow region appears. The database has shown that two correlations existed for turbulent flow: Colburn and Gnielinski correlations.

4.3. Sensitivity analysis of the model to the convective heat transfer coefficient correlations

The model is first tested with two global convective heat transfer coefficients: Colburn and Gnielinski correlations. The outlet temperature of the model using Colburn and Gnielinski correlations and the experimental data are displayed in Fig.5 a) and b) respectively for the set 1 and the set 2.

Fig.5 shows that the numerical outlet temperatures are quite similar. However, the outlet temperature of the model using Colburn correlation is lower at the beginning of heating stage. This is due to a higher value, in this case, of the convective heat transfer coefficient using Colburn correlation compared to Gnielinski correlation. Thus, the convective transfer is more important between the air and the PCM plates which leads to a lower outlet temperature.

At the end of the heating stage, the numerical curves exhibit the opposite trend. The PCM, more solicited with a higher convective heat transfer coefficient, becomes eventually less efficient.

During heating stage, the experimental outlet temperature is always below both numerical outlet temperatures. Thus the nearest numerical outlet temperature from experimental temperature is at the beginning given by the Colburn correlation and at the second stage by the Gnielinski correlation. Using either the Colburn correlation or the Gnielinski correlation does not lead to a significant better result. And both numerical models are less representative of the reality at the end of the heating stage.

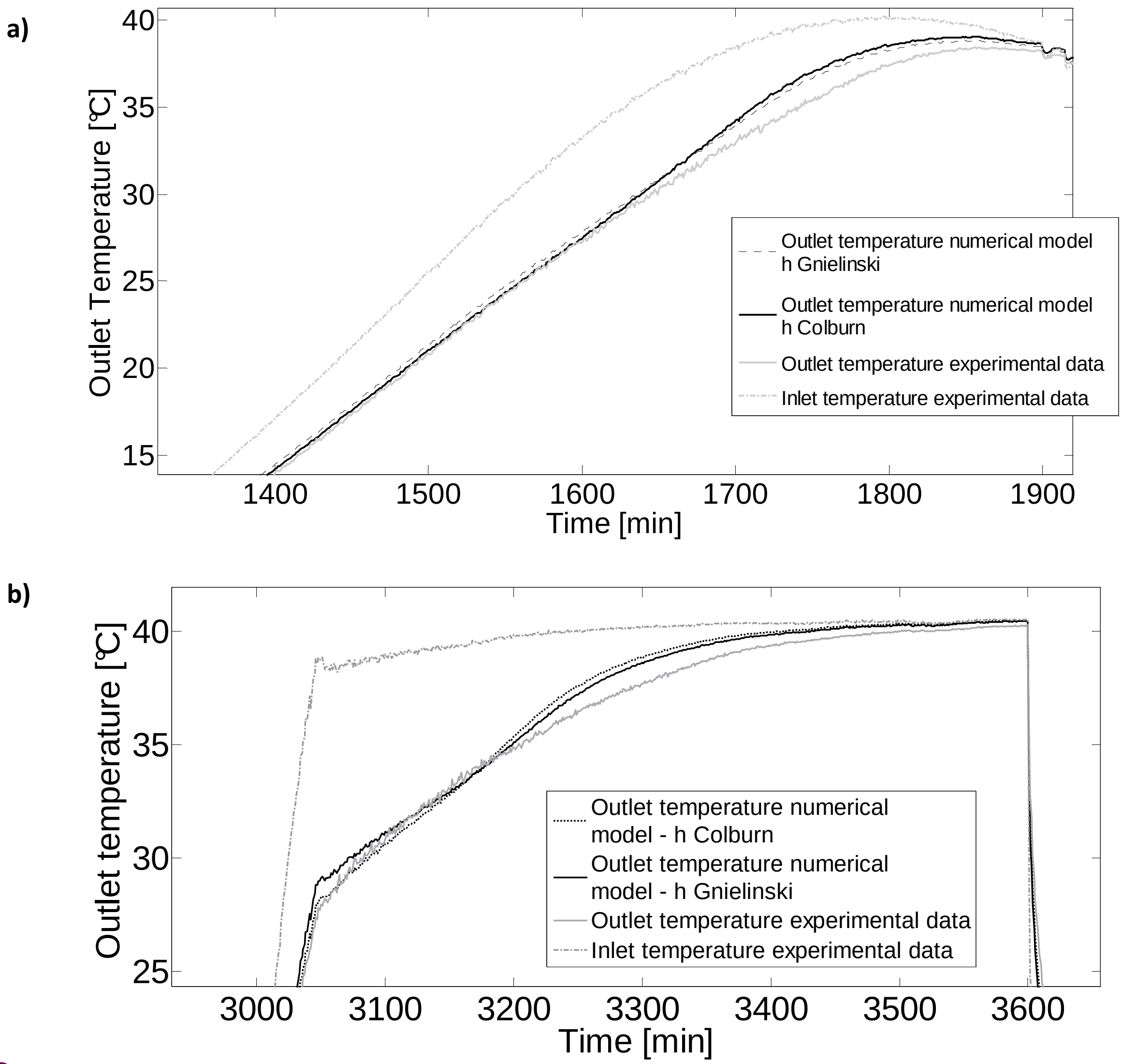

Figure 4 : Outlet temperature :numerical model and experimental data a) set 1 and b) set 2

The study of flow patterns and correlations from literature leads to a local convective heat transfer coefficient correlation at the beginning of the exchanger, when boundary layers are separated. In order to analyse the influence of this local correlation on the accuracy of the model, two cases are compared: the Gnielinski correlation constant along the exchanger and the local Shah correlation until fully developed region where Gnielinski correlation is also used. The value of the convective heat transfer coefficient along the exchanger is represented on Fig.6 for the two cases.

The comparison of the outlet temperatures has shown a difference lower than 0.05°C between the two cases. Thus, using the local correlation seems not to imply significant differences on the numerical outlet temperature.

In Fig. 7 is displayed the PCM surface temperature at 0.2m from the exchanger entrance. At that place, the values of the coefficient in the two cases are clearly different (see Fig 6.): the local coefficient is lower than the global one. Thus, logically, Fig. 7 shows that the PCM surface temperature is lower using the local correlation than the global Gnielinski correlation. However, compared to the experimental data, the local coefficient is less representative of the real PCM surface temperature.

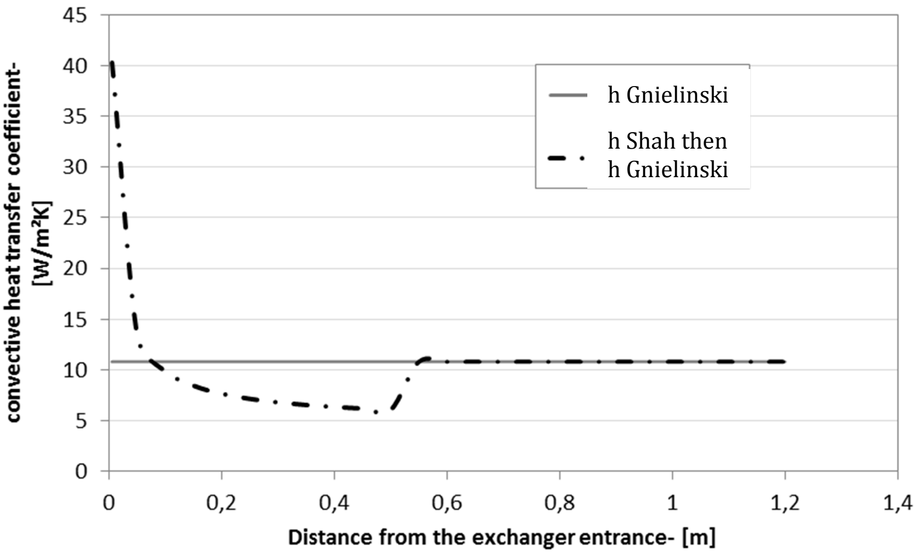

Figure 5 : convective heat transfer coefficient along the exchanger

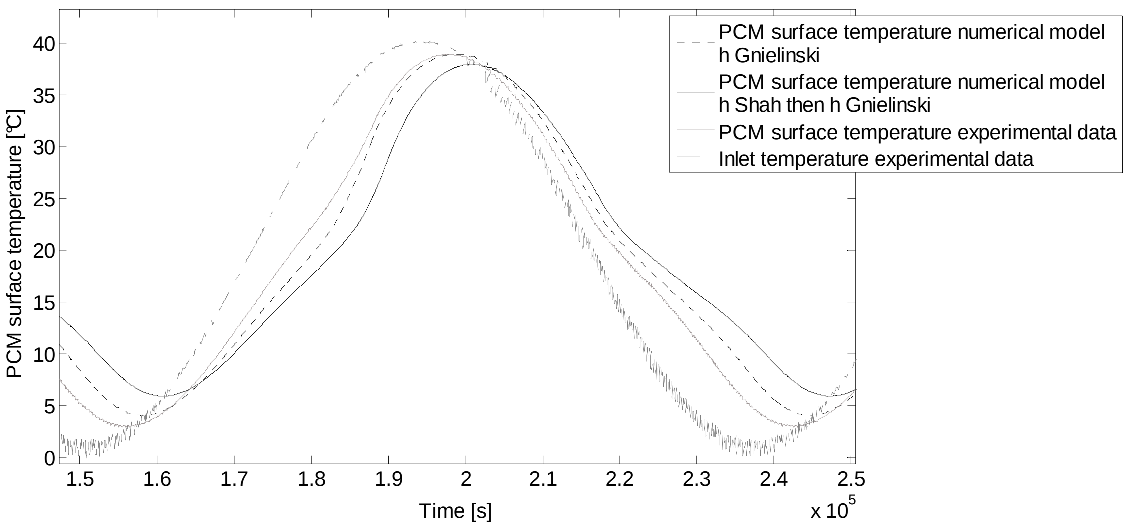

Figure 6 : PCM surface temperature at the entrance of the exchanger

5. Conclusion

In this study, a rigorous methodology is applied for selecting the most appropriate correlations for the convection coefficient in the channels of a PCM/air heat exchanger. Indeed, the flow regime evolves along the channel, which implies different correlations to be used at different locations.

To assess the improvements made by using proper correlations, we simulated an experiment and compared the numerical results to the experimental data. Here the most rigorous use of the correlations was to use a local correlation from Shah (laminar separate boundary layers) at the entrance of the channel, and to use the Gnielinski correlation (turbulent developed flow) right after.

Here, we must admit that the simulation results are not significantly improved when using the correlation in a rigorous way, when compared to the results with the Gnielinski correlation all along the plate. There are two interpretations for that. The first one is that the correlations presented here were developed for constant temperature or heat flux. This is not the case when using phase change material plates, but there is no simple correlation for that case.

The second interpretation is that the improvement made by using proper correlations is small compared to the global errors of the model. But there is a long path for reaching a really representative model of PCM heat exhangers.

6. References

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