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Effect of the plates geometry on the performance of a cross-flow recuperator for indirect evaporative cooling systems

Roberta Caruana, Luca Marocco, Stefano De Antonellis and Manfredo Guilizzoni

Department of Energy, Politecnico di Milano, Via Lambruschini 4, 20156 Milan, Italy

E-mail: roberta.caruana@polimi.it

Abstract. In recent years, Indirect Evaporative Cooling (IEC) is becoming a very interesting technology, as it can guarantee a good cooling effectiveness with low energy demand, thus helping to face the issues of climate change. In this work, a Computational Fluid Dynamics (CFD) model has been used to evaluate the effects of the plates geometry on the performance of a recuperator for IEC systems with dry primary and secondary channels. The model was validated against experimental data for a cross-flow IEC system with dimpled plates, and the validation results were quite satisfactory. Therefore, five types of dimpled plates differing from the original one in shape and size of the dimples were investigated. The results showed that a good trade-off between the improvement of dry-bulb effectiveness and restrained pressure losses along the channels is obtained with dimples in the shape of right prisms with either triangular, or pentagonal, or circular base section, depending on the application.

1. Introduction

Nowadays the impact of heating, ventilation, and cooling systems (HVAC) on overall building energy consumption is very significant [1], and cooling has become increasingly energy demanding due to the effects of climate change [2]. As a result, there is a growing interest in exploring Indirect Evaporative Cooling (IEC) as a potential solution, especially in regions with high temperatures and low humidity [3].

In an air-to-air IEC system, the latent heat of vaporization of water is used to extract thermal power from the air to be cooled, thus reducing the energy consumption with respect to traditional HVAC systems [4]. In particular, a IEC recuperator is composed of alternating primary and secondary channels, separated by thin conductive plates [5]. A pump delivers water, which can either form a film on the heat exchanger plates in the secondary channels [6], or pre-humidify the secondary air before entering the system. In this second case, water does not enter the recuperator, but it is only used to cool down the secondary air, to avoid problems of corrosion and aging of the plates.

The design of the plates geometry for these recuperators has the aim to enhance the heat transfer performance whilst avoiding an excessive increase of the pressure losses along the channels. Furthermore, the plates have to guarantee mechanical stiffness. Therefore, the fulfillment of these tasks requires a detailed analysis of the thermal-fluid dynamics behavior of the air streams inside the channels.

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A thorough description of the analytical and numerical models for the performance analysis of IEC systems is described in the review by Caruana et al. [7]. However, the possibility of changing the plate geometry has been investigated only by Adam et al. [8], and only with wet secondary channels. Therefore, a numerical performance analysis focused on plates shape of IEC recuperators with dry secondary channels is still missing.

The scope of this work is to use an experimentally validated numerical Computational Fluid Dynamics (CFD) model to evaluate the effect of the plates geometry on the performance of a cross-flow recuperator for IEC systems working in dry conditions. In particular, the study was conducted analyzing various dimples shapes, in order to assess one or more configurations able to guarantee the cited requirements.

2. Model description and validation

The model used in this work was developed using the open-source software OpenFOAM v9, and it relies on the following assumptions: 1) both primary and secondary channels work in dry conditions; 2) air is considered an ideal gas with constant specific heat capacities, dynamic viscosity, and Prandtl number; 3) the external plates of the recuperator are adiabatic; 4) gravity is negligible; 5) air inlet velocities are uniform; 6) water vapor in the primary air cannot condense; 7) radiative heat transfer is neglected.

As the computational domain is composed of two fluid regions, namely half primary and half secondary channel (exploiting a symmetry boundary condition), the *chtMultiRegionFoam* solver was used for the analysis. Therefore, governing equations that were solved for each iteration are: mass, momentum, and energy conservation for both fluids; ideal gas equation of state for both fluids; $k-\omega$ shear stress transport (SST) equations for the specific turbulent kinetic energy and the specific turbulent dissipation rate for both fluids [9]; and heat fluxes coupling between the two regions.

The simulations were performed using second order discretization schemes, in transient conditions. The time step was automatically selected by the software to keep the Courant number below 0.5, and every simulation was stopped after reaching a statistically-steady condition, in a time between 1.5 and 2 s.

The mesh was build using the *blockMesh* and *snappyHexMesh* utilities in OpenFOAM, and the mesh independence was assessed on two preliminary cases using seven meshes with a number of cells varying from 676k and more than 18M. After that, a mesh composed of about 4.9M cells was selected as the best compromise between a quite good accuracy and a still affordable computational time.

In order to validate the model, experimental data obtained for a commercial air-to-air cross-flow recuperator were used. The experimental setup, the adopted methodology, and the geometry of the recuperator are thoroughly described in [10]. It is worth mentioning that the plates which compose this recuperator show a pattern of concave and convex dimples in the shape of spherical caps on their surface, and the top of a dimple is always in contact with the top of the corresponding dimple on the adjacent plate, in order to guarantee the required mechanical stiffness of the system.

The model validation was based on the comparison between the numerical and experimental results of primary and secondary air outlet temperatures, and dry-bulb effectiveness, in two different cases, one with unbalanced volume flow rates (case 1), and one with balanced flow rates (case 2).

Table 1 summarizes the primary and secondary air inlet volume flow rates, $\dot{V}_{Pin}$ and $\dot{V}_{Sin}$, primary and secondary air inlet temperatures, T_{Pin} and T_{Sin}, and primary and secondary air inlet velocities, U_{Pin} and U_{Sin}, for the two analyzed cases.

The comparison between numerical and experimental primary and secondary air outlet temperatures, T_{Pout} and T_{Sout}, and dry-bulb effectiveness, ε_{DB}, for the two analyzed cases

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Case code	$\dot{V}_{Sin}$ [m^3/h]	$\dot{V}_{Pin}$ [m^3/h]	T_{Sin} [K]	T_{Pin} [K]	U_{Sin} [m/s]	U_{Pin} [m/s]
1	$1.0 \cdot 10^3$	$8.3 \cdot 10^2$	304.35	334.75	1.7	1.4
2	$1.1 \cdot 10^3$	$1.1 \cdot 10^3$	302.97	315.77	1.8	1.8

Table 1. Primary and secondary air inlet temperatures, volume flow rates, and velocities for the two analyzed cases.

is summarized in Table 2.

As the maximum difference between numerical and experimental results is 0.47% for the secondary air outlet temperature, 4.6% for the primary air outlet temperature, and 7.4% for the dry-bulb effectiveness, the model validation can be considered quite satisfactory.

Case code	Numerical T_{Sout} [K]	Experimental T_{Sout} [K]	Numerical T_{Pout} [K]	Experimental T_{Pout} [K]	Numerical ε_{DB} [−]	Experimental ε_{DB} [−]
1	312.62	312.75	314.28	313.05	67.34	71.38
2	310.21	310.27	308.35	307.76	57.97	62.59

Table 2. Comparison between numerical and experimental results of primary and secondary air outlet temperatures, and dry-bulb effectiveness, for the two analyzed cases.

3. Results and discussion

As the previously mentioned dimples have a base diameter of 20 mm, a height of 1.685 mm, and a spacing between two adjacent dimples of 30 mm, they are too flat, smooth, and spaced to guarantee a performance enhancement with respect to using flat plates without dimples, so they are necessary to ensure the required mechanical stiffness of the system, but they do not influence the thermal performance. As a consequence, recuperators using plates with five different types of dimples were simulated in the two cases used for the model validation, in order to choose the one which leads to the best performance enhancement, while maintaining restrained pressure losses along the channels. For simplicity, the plates manufacturability was not assessed in this analysis.

All the dimpled plates analyzed in this work are shown in Figure 1. In particular, panel A shows the original dimples in the shape of spherical caps, while panels B-F show the other types of dimples, that are all in the shape of right prisms, but with different base shapes. In fact, the dimples shown in panel B have triangular base section, the ones in panel C have squared section, the ones in panel D have pentagonal section, the ones in panel E have hexagonal section, and the ones in panel F have circular section.

All these new dimples in the shape of right prisms have a frontal size and a spacing between the extremities of two adjacent dimples of 6 mm, so that the air passages are narrow enough to cause flow mixing and potentially improve the heat exchange, but not so narrow as to excessively increase pressure losses.

Figure 2 shows the dry-bulb effectiveness (in purple) and maximum pressure loss along the channels (in orange) for all the analyzed cases.

From this figure it is possible to notice that using dimples in the shape of right prisms with 6 mm frontal size and spacing between the extremities always leads to a significant improvement of the thermal performance with respect to the original dimples, independently on the shape of the base section. In fact, these new plate geometries favor the flow mixing within the channels (as shown in Figure 3), thus improving the heat exchange between the two fluids. However, these dimple shapes cause a significant increase of the pressure losses along the channels, as they represent major obstacles to flow, thus potentially reducing the improvement due to a greater effectiveness.

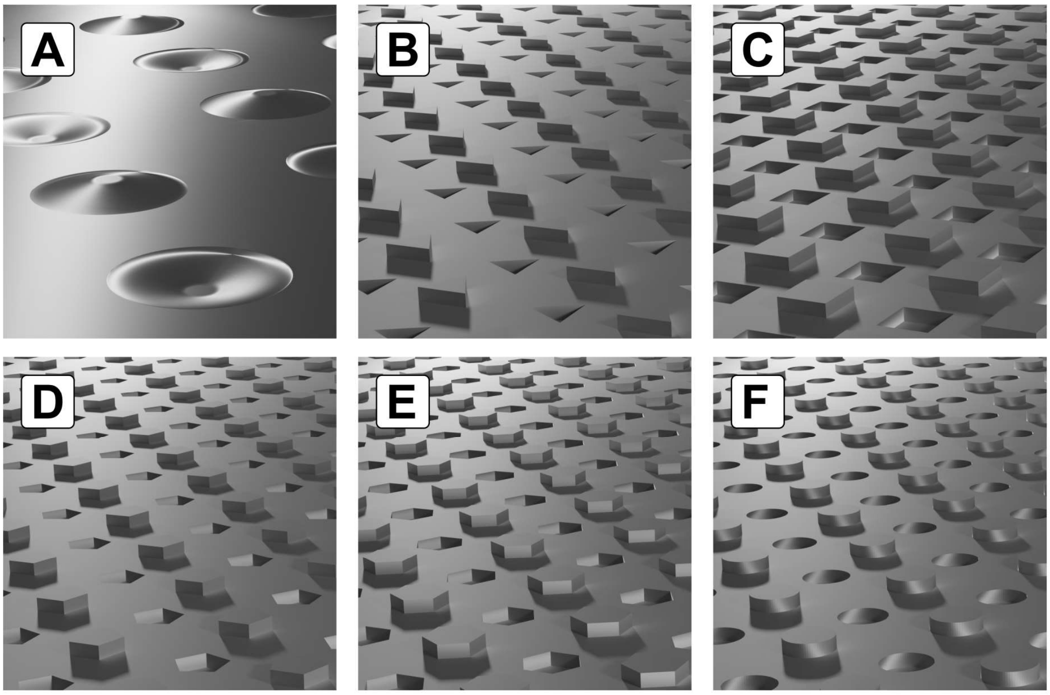

Figure 1. Sketches of the analyzed dimpled plates: original dimples in the shape of spherical caps (panel A), dimples with triangular base section (panel B), dimples with squared base section (panel C), dimples with pentagonal base section (panel D), dimples with hexagonal base section (panel E), and dimples with circular base section (panel F).

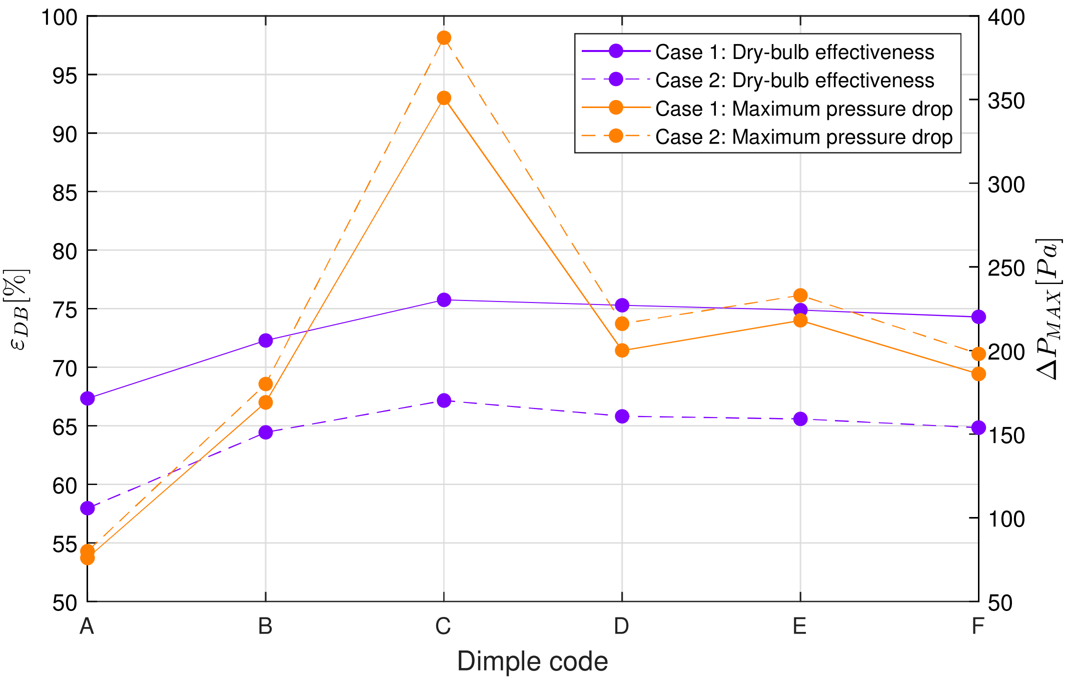

Figure 2. Dry-bulb effectiveness and maximum pressure loss along the channels for all the analyzed cases and plate geometries.

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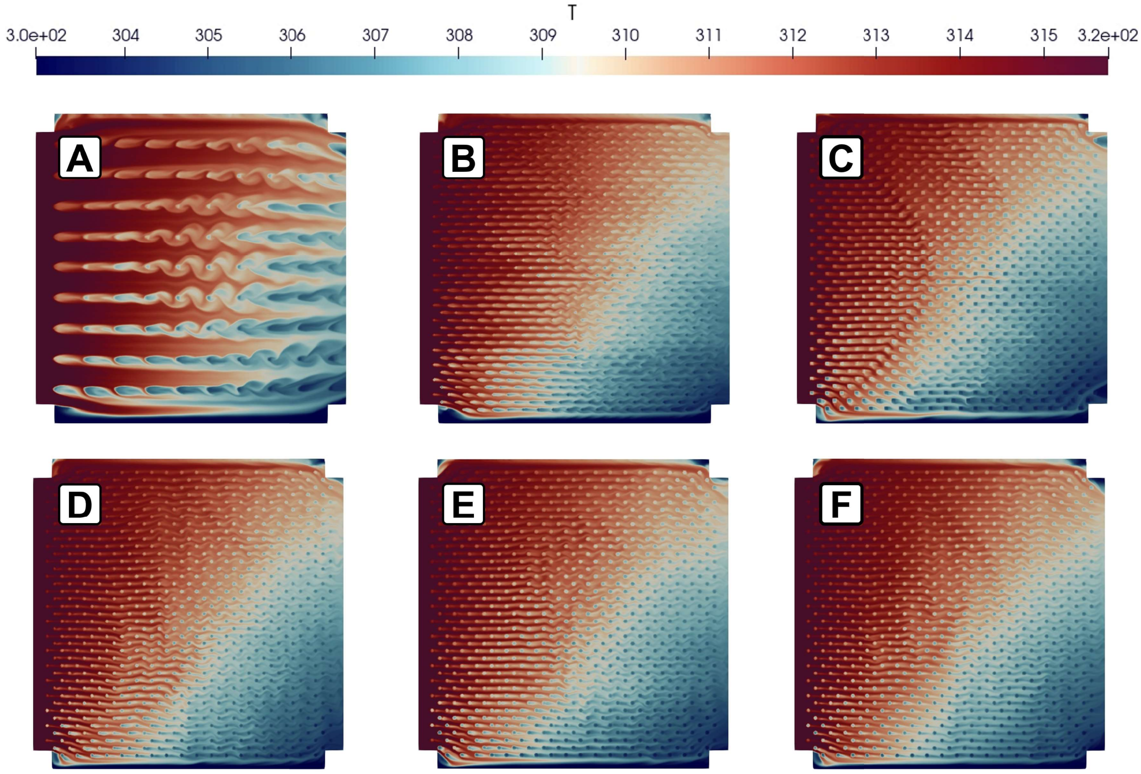

Figure 3. Primary air temperature field at the last time step of the simulation of Case 2 for all the analyzed plate geometries.

Therefore, according to these preliminary results, the choice of the dimple geometry to be used for the recuperators of IEC systems depends on the requirements of the considered application. In particular, the original dimples (A) are the best choice when the main aim is to have the minimum possible pressure loss, without a specific requirement on the effectiveness, while dimples in the shape of right prisms with squared base section (C) are optimal if the objective is to maximize the effectiveness, without strict limits on the pressure losses. Finally, dimples B, D, and F can be considered a good trade-off between the two requirements, thus they can be chosen based on the priorities of the specific application, while dimples E are always a sub-optimal choice.

4. Conclusions

In this study, a numerical CFD model has been used to evaluate the influence of the plates geometry on the performance of a recuperator for IEC systems with dry primary and secondary channels.

The model was validated against experimental data for a cross-flow recuperator with dimpled plates, showing a quite good agreement between numerical and experimental results.

After validation, five plates geometries, differing from the original one for the shape and size of the dimples, were investigated, in order to select one or more configurations able to improve the thermal performance without an excessive increase of the pressure losses along the channels.

The results showed that using plates with the original dimples in the shape of spherical caps leads to the lowest pressure loss, but also to the worst thermal performance, while using dimples in the shape of right prisms with squared base section leads to the best dry-bulb effectiveness, but to the highest pressure loss. A good trade-off between the two requirements can be obtained

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using dimples in the shape of right prisms with either triangular, or pentagonal, or circular base section, depending on the priorities of that particular application.

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