Proceedings of the 7th World Congress on Momentum, Heat and Mass Transfer (MHMT'22)
~~Lisbon, Portugal~~ – April 07 – 09, 2022
Paper No. ICMFHT 175
DOI: 10.11159/icmfht22.175

Porous Media Model Limit in Low Fin Packing Density Channel

Yohanna HENROTEL[12], Damien SERRET[1], Joseph JABBOUR[1]
[1]Research and Development Engineer/Temisth
Bâtiment TEAM Henri Fabre Rue Jacqueline Auriol Technoparc des FLorides, 13700, Marignane, France
yohanna.henrotel@temisth.com; joseph.jabbour@temisth.com; damien.serret@temisth.com
[2]Aix-Marseille Université, IUSTI UMR 7343
5 rue Enrico Fermi, 13013 Marseille, France

Abstract – Porous media have been widely used in industrial applications such as car radiator, electronic cooling, geothermal system, and others. channels with high fins density can be considered as a porous media represented by a porous matrix.It is defined by its porosity, permeability, and the form drag coefficient. That means that it is not resolved explicitly but represented by a volumetric porous zone where a volume-averaged pressure gradient drag term is applied to the Navier–Stokes momentum equation. Porous media in heat exchanger industry is mostly used to gain on computational power, a compromise between classic CFD, which is the most precise but costly and empirical correlation, which are simple but imprecise and limited by geometries. Those method are applied to single-phase systems as well as phase changing systems. This paper focuses on the limits of the porous model for channel with low fin density.

Keywords**:** Porous media – Heat exchanger – CFD – Single-phase flow – StarCCM+

1. Introduction

The single porous volume approach is widely used for several configurations and industrial Heat exchanger (HX) and Heat Sink (HS). It is applied and specialy invested in geometries provided with extension area or 'fins'. Structure such as arrays of tubes or fins and can therefore be modelized by a single porous volume [1][2], it is important to link the structure of the real heat exchanger and the simplified porous medium. The extension area thus called fins are introduced to augment the exchanged area between working fluid. It is spatially used on the side where the thermal resistance is highest. It aims to enhance the heat exchanged duty by increasing simultaneously the exchanged area and the heat transfer coefficient. When dealing with plate-fins HX, higher base contribution can lead to a wrong equivalent porous media, since the base contribution is not modelized by the porous model that only account for the array of fins. The aim of this paper is to show the point where base heat exchange cannot be neglected in the volume averaging of the porous model.

2. Fundamentals

The performance of a thermal system is generally represented by its efficiency, here fin efficiency η and effectiveness ε:

$$\varepsilon = \frac{Q_{with\ fins}}{Q_{without\ fins}} \#(1)$$

$$\eta = \frac{Q_{T\ ideal}}{Q_{T\ real}} \#(2)$$

Fins efficiency depend on the temperature gradient along the fin, as in an ideal case it would be at the same temperature as the base, in some cases where the fin material is highly conductive for example, fins can be considered "ideal".

The first characteristic of a porous medium is its porosity: this is the ratio between the volume not containing solids and the total volume of the medium V.

$$\chi = \frac{V_f}{V} \#(3)$$

The Darcy-Forchheimer law corrects Darcy's law for large fluid flows in porous media by considering inertial effects:

$$\frac{\delta P}{\delta x} = -\frac{\mu}{k}u - \frac{\rho}{k_1}u^2 \#(4)$$

This law defines the pressure losses of a flow at Re>10 [3] through a porous medium, with k the permeability and k_1 the inertial permeability. The permeability of a porous material is its capacity to let a fluid pass through it. A good permeability therefore implies a good porosity, but the opposite is not necessarily true, it has been shown to be intrinsic to the solid matrix of the porous medium and therefore only a function of its topology and its associated geometric properties.

In porous media heat transfer between solid and fluid can be treated considering a local thermal equilibrium hypothesis (LTE) for example, in forced convection, the literature shows that this hypothesis is valid for lower Darcy, Reynolds and Prandtl number, as well as lower solid phase thermal conductivity; however, it becomes invalid for higher effective fluid thermal conductivity or lower interstitial heat transfer coefficient (the heat exchange between the fluid stream and the solid matrix of the porous medium)[4]. Therefor the local thermal non-equilibrium (LTNE) is preferred.

Thermal non-equilibrium energy equations:

$$\frac{\partial(\chi\rho_f E_f)}{\partial t} + \nabla.(\chi\rho_f H_f u) = -\nabla.(\chi Q_f) + \nabla.(\chi T.u) + Ah(T_f - T_s) + S_u^e \#(5)$$

$$\frac{\partial((1-\chi)\alpha_s\rho_s E_s)}{\partial t} = -\nabla.((1-\chi)\alpha_s Q_s) + Ah(T_s - T_f) + S_u^e \#(6)$$

With E_f and H_f the total energy an enthalpy of the fluid, a the interaction area density, h the heat transfer coefficient, T the stress tensor and S_u^e an energy source or sink Q_f and Q_s are the conduction heat flux the fluid and solid phases.

In (5) and (6) the solid temperature T_s both diffusion term and unsteady term is volume-averaged temperature of solid region while the T_s in the heat source term should be the surface temperature of the solid, porous model being used to modelized the fins array, it can lead to significant error between the volume-averaged temperature and the surface temperature of solid matrix for cases where a major part of the heat transfer is by the fins.

3. CFD simulations

The simulations were carried out on the software StarCCM+. The simulations used a three dimensional, steady, and were computed on both laminar and turbulent k-Ɛ model since this Reynold number is at the turbulent limit, but no significant difference was observed on the results. The test case consists in a classic rectangular mini channel with offset rectangular fins. Fins are the same height as the channel, they have no tip clearance. The working fluid is water at a temperature of 323.15K at the inlet and different masse flow corresponding to Re=3000 for each configuration. The outlet pressure condition is of 10^5Pa.

The boundary condition at the bases (top and bottom of the channel) is a heat flux of -1000W/m². In these conditions the outlet water is cooled by around 0.01K. A symmetry condition is applied to the plane cutting the middle of each offset fin on the sides of the channel to gain on simulation time (Figure 1), Sf is the fluid section and Sa the fin section.

A range of finned channels are studied from 0.5mm to 5mm (Table 1) with the same solid section to fluid section ratio of 25%, the length (0.05m) and height (0.01m) of the channel also remains, three row of offset fins are considered to have a developed flow.

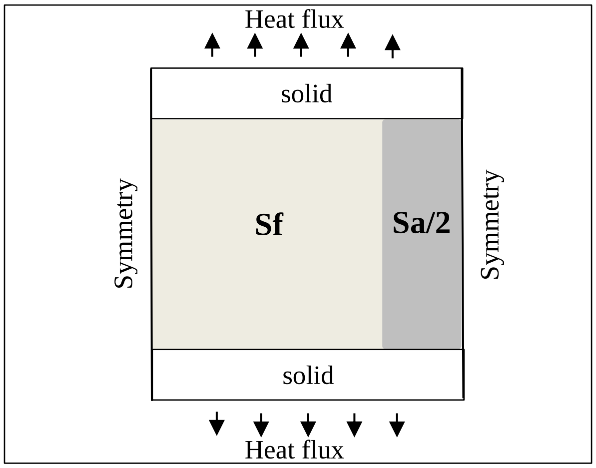

Figure 1: section of the modeled channel with boundary conditions

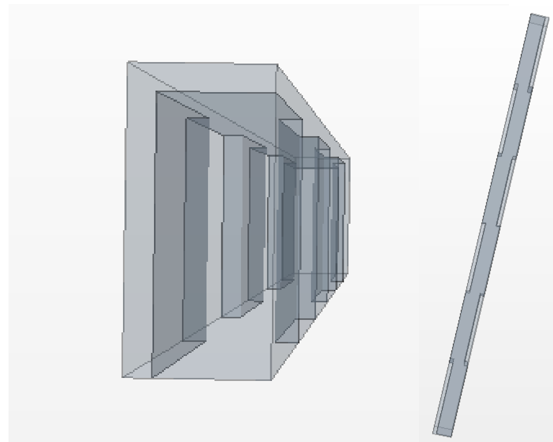
Figure 2: 3D CAD view of the fluid domain in StarCCM+

.

Table 1 : Channel geometries and flow velocities

Fins thickness	*0.5*	*1*	*2*	*5*	(.10^3) m
Fins exchange surface	*0.325*	*0.35*	*0.4*	*0.55*	(.10^3) m²
Base exchange surface	*0.12*	*0.24*	*0.48*	*1.2*	(.10^3) m²
External exchange surface	*0.15*	*0.3*	*0.6*	*1.5*	(.10^3) m²
Percentage of fin surface	*73*	*59*	*45*	*31*	%
Hydraulic diameter	*2.86*	*4.44*	*6.15*	*8*	(.10^3) m
Reynolds	*3000*	*3000*	*3000*	*3000*	-
velocity	*0.562*	*0.361*	*0.261*	*0.201*	m/s
Mass flow	*5.62*	*7.22*	*10.4*	*20.1*	(.10^3) kg/s

Power exchanged by the different parts of the channel and average heat transfer coefficient (HTC) in laminar forced convection are retrieved and analyzed bellow.

3. Results and discussions

This paper is focusing on the limits of the porous medium in the design of plate fin heat exchanger with lower fin density, the test channel is modelized entirely but analyzed as if it was a porous volume.

Results shows that from a fin percentage of under 45% the base contribution account for half of the power exchanged in the channel. The fins contribution closely follows the percentage of fin surface though it is slightly lower which implies that looking simply at the percentage of fin exchange area to get an idea of their contribution to the total exchange will tend to overestimate it.

The HTC is lower at the base than at the fins as fins locally increases the flow velocity, it is around 30% lower at the base but can be variable with turbulence (in a mini instead of micro channel for example [5]) or fin shape.

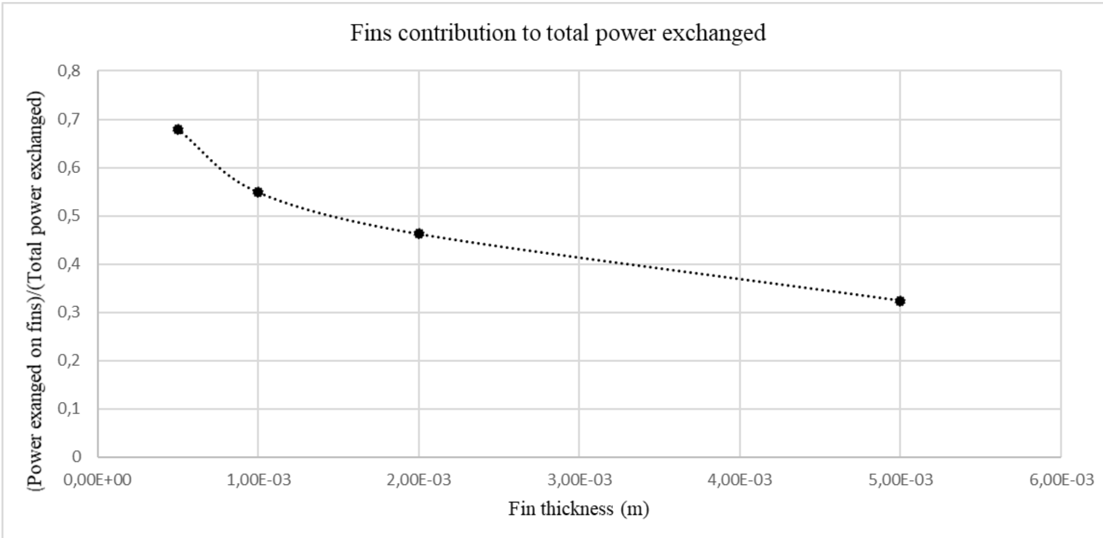

Figure 3: Fins contribution to the total power exchanged in the channel

Table 2: Average heat transfer coefficient (HTC)

fins average HTC	9.9	7.14	7.64	3.29	(.10^3) W/K.m²
base average HTC	8.14	5.8	5.02	2.39	(.10^3) W/K.m²

4. Conclusion

This short study has shown that in a standard rectangular channel with a sufficient spacing between the fins the volumetric average of the porous model is invalid as it is accounting for the fin matrix only and therefor assumes that most of the heat transfer is exchanged through the fins. A next stage to this study would be to carry it at higher Reynolds and with different channel geometries to observe turbulence effect on base contribution. Another point would be to carry this work in two-phase conditions.

Acknowledgements

This project has received funding from the Clean Sky2 Joint Undertaking (JU) under grant agreement No 886698. The JU receives support from the European Union’s Horizon 2020 research and innovation program and the Clean Sky 2 JU members other than the Union. This paper reflects only the author's view and that the JU is not responsible for any use that may be made of the information it contains.

References

[1] B. Çetin, K. G. Güler, and M. H. Aksel, “Computational modeling of vehicle radiators using porous medium approach,” in Heat Exchangers - Design, Experiment and Simulation, InTech, 2017.

[2] D. Juan and Z. Hai-Tao, “Numerical simulation of a plate-fin heat exchanger with offset fins using porous media approach,” Heat Mass Transf., vol. 54, no. 3, pp. 745–755, 2018.

[3] G. Schneebeli, “Expériences Sur la Limite de Validité de la Loi de Darcy et L’apparition de la Turbulence Dans un Écoulement de Filtration,” Houille Blanche, vol. 41, no. 2, pp. 141–149, 1955.

[4] G. F. Al-Sumaily, A. Al Ezzi, H. A. Dhahad, M. C. Thompson, and T. Yusaf, “Legitimacy of the local thermal equilibrium hypothesis in porous media: A comprehensive review,” Energies, vol. 14, no. 23, p. 8114, 2021.

[5] G. L. Morini, “Laminar-to-turbulent flow transition in microchannels,” Microscale thermophys. eng., vol. 8, no. 1, pp. 15–30, 2004.