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Comparison of metal foam heat exchangers to a finned heat exchanger for low Reynolds number applications

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ABSTRACT

Due to its high porosity and large specific surface area, open-cell metal foam is an attractive material for heat transfer applications. In this article the performance of metal foam heat exchangers is compared to the performance of a bare tube bundle and the performance of an existing conventional louvered fin heat exchanger. A macroscopic model consisting of the Darcy–Forchheimer–Brinkman flow model and the thermal non-equilibrium energy model is used to perform two-dimensional simulations on metal foam heat exchangers. Because thermal design of heat exchangers is always a trade-off between heat transfer and pressure drop, both are considered together when evaluating the heat exchangers' performance. The foamed heat exchangers show up to 6 times higher heat transfer rate than the bare tube bundle at the same fan power. If the fins are replaced by metal foam while keeping the overall dimensions the same, the finned heat exchanger shows in all cases the best performance. However, a metal foam heat exchanger can outperform the finned heat exchanger if the frontal area is changed. Optimization is required to select the best foam parameters, material and dimensions. This clearly shows the potential of open-cell metal foam for high performance heat exchanger designs.

1. Introduction

In many industrial and domestic processes energy is transferred as heat. Hence, heat exchangers are important elements as contributors to increased energy efficiency in industry, transport and buildings. In many applications air is one of the working fluids (e.g. heat pumps, air conditioning devices, refrigeration, compressed air cooling, etc.). When exchanging heat with air, the main thermal resistance is located at the air-side of the heat exchanger. Commonly, the heat transfer rate is increased by adding fins at the air-side. The current state-of-the-art fins are complex interrupted designs, such as louvered fins and slit fins [1], or surface protrusions, such as vortex generators [2]. Further improvements are possible by combining existing enhancement techniques [3]. An example of such compound heat exchangers is the combination of louvered fins and vortex generators [4].

Heat exchanger manufacturers are continuously searching for new and better designs. A promising option is the use of open-cell metal foam as alternative for the conventional fins. This porous structure consists of a network of solid ligaments (or struts) around the pores. Open-cell foam is characterized by a high volumetric porosity (>0.85; ratio of the air volume to the total volume) and thus low weight, high surface-to-volume ratio (up to 1500 m^2/m^3) and excellent fluid mixing due to the complex network of struts [5]. These properties, in combination with the high thermal conductivity of the metal (e.g. aluminum or copper), make these foams a promising structure for heat transfer applications [6–11].

Due to time constraints, microscopic analysis of metal foam is usually restricted to a limited number of cells [12]. Microscopic models are thus not appropriate to simulate actual metal foam applications which contain thousands of cells. As metal foams can be treated as porous media, a macroscopic analysis is possible using the volume averaging technique (VAT): the details of the original structure are replaced by their averaged counterparts [13–14]. The governing macroscopic equations for the phase averaged variables can be solved much faster than the traditional transport equations for local variables, which require direct numerical simulations (DNS). However, because the details of momentum and energy transfer between the fluid flow and solid structure are lost during the averaging, closure relations are required. These include relations for the interstitial heat transfer coefficients [15–17], the inertial loss factor and the permeability (which determines the viscous loss factor) [18–22]. Also relations for the macroscopic (or effective) properties as function of the microscopic

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Nomenclature

A_c	minimum cross sectional flow area [m^2]
A_d^*	relative heat exchanger frontal area (Eq. (18)) [m]
A_o	overall heat transfer surface area [m^2]
$C_{\min}$	minimum heat capacity [W/K]
c_p	specific heat capacity [J/kgK]
D_h	hydraulic diameter [m]
D_o	outer tube diameter [m]
f	Fanning friction factor [–]
F_d	heat exchanger flow depth [m]
G_c	mass flux through the minimum cross section [kg/m^{2}s]
h	convective heat transfer coefficient [W/m^{2}K]
H	heat exchanger height [m]
j	Colburn j-factor [–]
k	thermal conductivity [W/mK]
$\dot{m}$	mass flow rate [kg/s]
NTU	number of transfer units [–]
P	pressure [Pa]
P_d^*	relative fluid pumping power (Eq. (20)) [m^{-2}]
P_l	longitudinal tube pitch [m]
Pr	Prandtl number [–]
P_t	transversal tube pitch [m]
Q	heat transfer rate [W]
R	thermal resistance [K/W]
Re	Reynolds number (Eq. (5)) [–]
T	temperature [K]
t_w	tube wall thickness [m]
U	[W/m^{2}K] overall heat transfer coefficient
$\vec{v}$	superficial velocity [m/s]
v_c	maximum velocity in the heat exchanger (Eq. (8)) [m/s]
V_d^*	relative heat exchanger volume (Eq. (19)) [m^2]
W	heat exchanger width [m]

Special characters

β	inertial loss factor [m^{-1}]
ε	effectiveness [–]
ϕ	porosity [–]
η_o	surface efficiency [–]
κ	permeability [m^2]
μ	dynamic viscosity [Pas]
ρ	density [kg/m^3]
σ	contraction ratio [–]
σ_o	specific surface area [m^2/m^3]

Subscripts and superscripts

e	effective
f	fluid
in	inlet
m	mean
out	outlet
s	solid
sf	interstitial

parameters are needed [23]. These include the effective thermal conductivities and thermal dispersion [15,18,24–25] as well as the effective viscosity [21]. These relations are obtained from experiments, analytical modeling or CFD simulations of a representative heat exchanger volume with a very fine mesh. They are only valid for the specific geometry and flow conditions under consideration. A detailed review on existing fluid and thermal transport models for open-cell metal foams can be found in [26–27].

When evaluating the performance of a metal foam heat exchanger it is important to compare the results to the performance of today's used cooling solutions (e.g. finned heat exchangers, grooved tubes, etc.) to judge the metal foam's potential. The closed macroscopic model can be used to examine metal foam heat exchangers. Under some assumptions (fully developed flow, constant thermophysical properties, neglecting the Forchheimer contribution, among others), analytical solutions for the velocity and temperature distributions can be deduced. Xu et al. [28] analytically examined a channel filled with metal foam. They used a Brinkman-Darcy model and two equation energy model (i.e. local thermal non-equilibrium). Based the performance criterion $j/f^{1/3}$, they concluded that the foam channel has a higher performance than the empty channel in the porosity range 80–95%. They also studied a partially filled channel with foam on the upper and bottom plate of the channel [29]. Here the minimum foam thickness was determined which results in a higher $j/f^{1/3}$ compared to the empty channel. In both studies no comparison to an internally finned/grooved channel is reported. Lu et al. [30] investigated a metal foam filled tube and concluded that the heat transfer performance can be improved up to 40 times compared to a plain tube, but at the expense of a higher pressure drop. Zhao et al. [31] extended this work and studied tube-in-tube heat exchangers with the inner tube as well as the annulus filled with open-cell metal foam. They showed that metal foam filled tube-in-tube heat exchanger outperforms the finned tube heat exchanger (inner grooved tube with external fins) from a heat transfer point of view. However, pressure drop results were not reported. In contrast to the analytical approach of the previous papers, numerical simulations using the closed macroscopic model are also possible. The computational time is acceptable as the macroscopic model is fast running. Chen et al. [32] used the Darcy-Brinkman-Forchheimer flow model and the two-equation energy model to study the heat transfer from multiple metal foam heat sinks in a horizontal channel under forced convection. They concluded that the cooling significantly improves if metal foam is mounted on the heat sources. The metal foam also causes a pressure drop penalty. However, these pressure drop results were not linked to the heat transfer results. Also a comparison with a conventional finned heat sink is missing. An air-cooled metal foam heat exchanger under a high speed laminar jet was numerically investigated by Ejlali et al. [33]. They showed that the metal foam outperforms a pin finned surface without increase of weight or pressure drop. A metal foam wrapped cylinder in cross-flow was examined by Odabaee et al. [34,35]. They assumed local thermal equilibrium, even though Lee and Vafai [36] showed that local thermal non-equilibrium yields more accurate predictions due to the large difference in thermal conductivity between the air and the solid foam material. The optimal foam layer thickness was determined. Comparison to a finned tube showed much higher heat transfer rate with reasonable pressure drop penalty. This numerical work was extended to a metal foam wrapped tube bank [37]. The effect of the tube pitches, foam thickness and foam parameters was studied. It is observed that the area goodness factor of the metal foam tube bundle is significantly better than that of the conventional finned tube heat exchanger. This higher performance was also confirmed by the experiments of Chumpia and Hooman [38]. They compared five foam wrapped tubes to a finned tube as benchmark. They found that a foam wrapped tube provides more heat transfer while keeping the pressure drop at the same level as that of the finned tube if the proper foam thickness is selected. T'Joen et al. [7] also compared metal foam wrapped tubes to finned tubes using