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Thermal convection measurements inside aluminum foam and comparison to existing analytical solutions

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Abstract

Metal foams have high thermal conductivity and large surface area per unit volume. The internal structure of the foams promotes vigorous mixing of a moving fluid inside the foams. As such, metal foams are very suited for convection heat transfer designs. Clear models for forced convection heat transfer inside the foam, as well as reliable thermal measurements are indispensable for convection-based thermal system designs. This paper present direct experiment for Darcy airflow and fluid's temperature inside a heated aluminum cylinder filled with aluminum foam. The experimental fluid temperature is compared to the available analytical solutions for the two-equation model for fully-developed forced convection. Peculiar, physically-unexplainable behavior is displayed when plotting the existing analytical solution in the literature. An error was discovered and corrected. Good agreement is obtained with the correct solution.

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Keywords: Convection; experiment; fluid temperature; analyis ; heat transfer

1. Introduction

Due to the complex internal structure of the foam, exact solutions of the complete transport equations are virtually impossible. Examples of numerical, experimental and analytical work are given in Calmidi et al. (2000), Hwang et al. (2002), Lee and Vafai (1999), and Vafai and Kim (1989).

Lu et al. (2006) analyzed forced convection in a tube filled with a porous medium subjected to constant heat flux using the two-equation model. Zhao et al. (2006) presented an analytical solution for a tube-in-tube heat exchanger,

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for which the inner tube and the annulus were filled with metal foam. Analytical solutions in porous media continue to be sought (Xu et al. (2011), Qu et al. (2012)). Direct comparisons to experimental values of key variables, e.g., the fluid and solid temperatures inside the foam, is lacking in the literature. In a recent comprehensive review, Zhao (2012) indicated that there is a lack of reliable experimental heat transfer data for open-cell metal foam.

Researchers typically measure substrate (wall) temperature, and/or the temperatures at the inlet and outlet of the foam. Average surface heat transfer coefficient, volumetric heat transfer coefficient and/or Nusselt number are determined and used for comparing analytical and numerical results to experimental data (Calmidi et al. (2000), Hwang et al. (2002), Bhattacharya et al. (2002), Boomsma et al. (2003), Kurbas and Celik (2009), Zhao et al. (2004), and Kim et al. (2000)).

1.1. Solution of Two-Equation Model

A cylindrical porous medium of radius r_0 is bounded by a wall and heated by a constant heat flux$_{q''}$, Fig. 1. There is fully-developed one-dimensional flow in the z-direction with a volume-averaged velocity component u. From the Brinkman-Darcy equation, the velocity is (Vafai and Kim (1989), Dukhan (2012)):

$$U = 1 - \frac{I_0(\omega R)}{I_0(\omega)} \tag{1}$$

Where $R = r/r_0$,$Da = K/r_0^2$ (Darcy number), K is the permeability, $\gamma = \mu/\mu_e$, μ is the fluid viscosity, μ_e is the effective viscosity, $U = u/u_\infty$, u_∞ is the velocity outside the boundary layer, $\omega = \sqrt{\gamma / Da}$ and I_0 is the modified Bessel function of order zero.

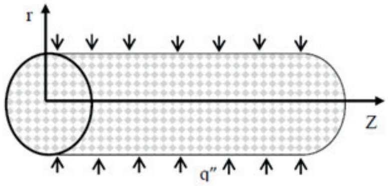

Fig. 1. Schematic of metal foam heated cylinder.

In the thermally fully-developed region, the volume-averaged two-equation model is (Calmidi and Mahajan (2000), Lee and Vafai (1999), Krishnan et al. (2004), and (Alazmi and Vafai 2002)):

$$\frac{1}{R}\frac{\partial}{\partial R}(R\frac{\partial \theta_s}{\partial R}) - Bi(\theta_s - \theta_f) = 0 \tag{2}$$

$$\frac{\lambda}{R}\frac{\partial}{\partial R}(R\frac{\partial \theta_f}{\partial R}) - Bi(\theta_s - \theta_f) = 2U \tag{3}$$

At R = 0 $$\frac{\partial \theta_s}{\partial R} = \frac{\partial \theta_f}{\partial R} = 0 \tag{4}$$

At R = 1 $$\frac{\partial \theta_s}{\partial R} = \lambda \frac{\partial \theta_f}{\partial R} = 1 \tag{5}$$ and $\theta_s = \theta_f = 0$

where $\theta_f = (T_f - T_w) / q''r_0 / k_s$, $\theta_s = (T_s - T_w) / q''r_0 / k_s$, $Z = z / r_0$, $R = r / r_0$, $\lambda = k_f / k_s$ and $Bi = h \sigma r_0^2 / k_s$ is the Biot number. Also, k_s and k_f are the effective thermal conductivities, T_s and T_f are temperatures of the solid and the fluid, respectively, h is the interstitial heat transfer coefficient, σ is the surface area per unit volume and ρ and c are the density and the specific heat of the fluid is the wall temperature which is not known a priori (Lee and Vafai