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Heat Transfer in Metallic Foam Subjected to Constant Heat Flux

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ABSTRACT: Since metallic foam will increase the performance of heat exchanger, it have caused many researcher's attention recently. Our research base on the model that metallic foams applied to heat exchanger. In this case, there is three kind of heat transfer mechanisms, heat conduction in fibers, heat transfer by conduction in fluid phase, and internal heat change between solid and fluid phases. In this paper, we first discuss the acceptance of applying thermal equilibrium among the two phases. then to calculate the dimensionless temperature profile along 7 metallic foams. The 7 samples have different characteristics, such as area ratio, effective conductivity, porosity, etc.

Key words: Metallic foam, Heat transfer model, Thermal equilibrium, Temperature profile

Nomenclature

a : interfacial area per unit volume of porous media [m^{-1}]
Bi : Biot number defined in equation
c_p : specific heat of the fluid [$J\ kg^{-1}K^{-1}$]
D : hydraulic diameter of the channel [m^2]
h : heat transfer coefficient[$Wm^{-2}K^{-1}$]
H : height of foam sample [m]
k : conductivity [$Wm^{-1}K^{-1}$]
Pr : Prandtl number of air
ppi : pores per inch
Re : Reynolds number based on foam height
q : heat flux [Wm^{-2}]
T : temperature [K]
L : length of foam sample in flow direction [m]
W : width of foam sample [m]
u : velocity, m/s

Geek symbols

ϵ : porosity
η : non-dimensional transverse coordinate defined in equation
γ : geometric constant defined by equation
θ : nondimensional temperature
κ : ratio of the effective fluid conductivity to that of the solid

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λ : parameter defined by equation
ϵ : porosity
ρ : fluid density

Subscripts

eff : effective value
f : fluid phase
i : internal heat exchange
s : solid phase
∞ : ambient

1. Introduction

Metallic foams have a distinct but continuous and rigid solid phase, and a fluid phase. as showed in Fig. 1. They are typically available in high porosities, also have high thermal conductivity and large area per unit volume. Resent years some researchers have study on the characteristics of metallic foam. Boomsma,K (2001), Bhattacharya, A.,V. V. Calmidi, Bhattacharya, A., V. V. Calmidi, et al. (2002), Singh, R. and H. S. Kasana (2004) developed model to calculate the effective conductivity of foams, respectively. Energy equation in porous media has been considered as two or one by different researchers. In this paper, we first discuss the acceptance of applying thermal equilibrium among the two phases. then to calculate the dimensionless temperature profile along 7 metallic foams. The properties of metallic foam we used in this paper are listed in Table 1.

2. Problem definition

2.1 Schematic description

Consider a rectangular block of open-cell metal foam and heated from above with constant heat flux q, and the other three faces is thermal insulated. The block has a length L (25 cm) in the flow direction, Width W (15 cm) and height H (10 cm), as shown in Fig. 2.The air flow through the channel getting

Table 1 Properties of foams used here

	ppi	porosity	df (m)	dp (m)	$ks_{,eff}$ (W/m²K)	$kf_{,eff}$ (W/m²K)	a (m^{-1})
1	5	0.9726	0.00050	0.00402	2.48	0.0256	415.42
2	5	0.9118	0.00055	0.00380	6.46	0.0237	917.55
3	10	0.9486	0.00040	0.00313	4.10	0.0248	799.63
4	20	0.9546	0.00030	0.00270	3.71	0.0250	756.07
5	20	0.9005	0.00035	0.00258	7.19	0.0233	1305.30
6	40	0.9272	0.00025	0.00202	5.48	0.0242	1390.10
7	40	0.9132	0.00025	0.00180	6.37	0.0237	1850.60

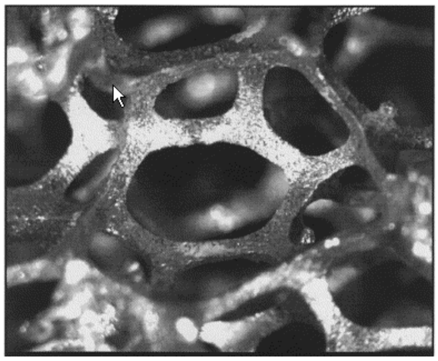

Fig. 1 Metallic foam

heat away from heating plate at velocity u m/s. To simplify the problem, we make the following assumptions:

1. Radiation effect is neglected.
2. Constant thermal properties of the solid and fluid phases.
3. The foam properties are constant and independent of diredtion.
4. The flow is steady and fully developed.

2.2 Heat transfer model

Based on the above assumption, the following control equations are obtained from Amiri and Vafai and Amiri et.[5,6]

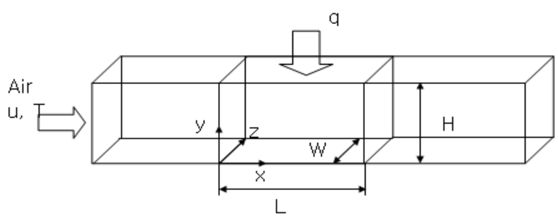

Fig. 2 Schemic of problem

Fluid phase

$$k_{f,eff}\nabla_y^2 T_f + h_i a(T_s - T_f) = \rho c_P u \frac{\partial T_f}{\partial x} \qquad (1)$$

Solid Phase

$$k_{s,eff}\nabla_y^2 T_s - h_i a(T_s - T_f) = 0 \qquad (2)$$

The boundary condition at the bottom of the channel can be written as

$$\frac{\partial T_f}{\partial y}|_{y=0} \frac{= \partial T_s}{\partial y}|_{y=H} = 0 \qquad (3)$$

The governing equation can be rendered dimensionless using the following nondimensional variables:

$$\theta = \gamma \frac{k_{s,eff(T-T_w)/H}}{q_w}, \eta = \frac{y}{H} \qquad (4)$$

where $\gamma = \frac{D}{4H}$,

in which D is the hydraulic diameter of the channel.

The temperature distribution was obtained by D.-Y. Lee(1999). The resultant equations are

$$\theta_f = \frac{1}{(1+\kappa)}\left[\frac{1}{2}(\eta^2 - 1) - \frac{1}{Bi(1+\kappa)}1 - \frac{\cosh(\lambda\eta)}{\cosh(\lambda)}\right]$$
(5)

$$\theta_s = \frac{1}{1+\kappa}[\frac{1}{2}(\eta^2 - 1) + \frac{\kappa}{Bi(1+\kappa)}1 - \frac{\cosh(\lambda\eta)}{\cosh(\lambda)}]$$
(6)

where the three parameters, Bi, κ and λ are defined as

$$Bi = \frac{h_i \gamma a H^2}{k_{s,ef}}, \kappa = \frac{k_{f,eff}}{k_{s,eff}},$$
$$\lambda = \sqrt{Bi(1+\kappa)/\kappa}$$

The non-dimensionalized bulk mean temperature of the fluid, $<\theta_f>$ can be obtained from equation as

$$<\theta_f> = -\frac{1}{1+\kappa}[\frac{1}{3} + \frac{1}{Bi(1+\kappa)}1 - \frac{1}{\lambda}tanh(\lambda)]$$
(7)

Nusselt numbers based on the channel hydraulic diameter, D, and the effective fluid conductivity can be presented as

$$N_w = \frac{h_w D}{k_{f,eff}} = \frac{4\gamma^2}{\kappa(-<\theta_f>)} \qquad (8)$$

where, $h_w = \frac{q_w}{T_w - <T_f>}$

For foamed materials, there is no general model for the interfacial heat transfer coefficient, h_i V. V. Calmidi (2000) So the following correlation developed by Zukauskas [17], which is valid for staggered cylinders, is used to estimate h_i

$$N_{sf} = 0.52 Re_d^{0.5} \mathrm{Pr}^{0.37} \qquad (9)$$

where Re_d is the local Reynolds number, $Re_d = ud/v$

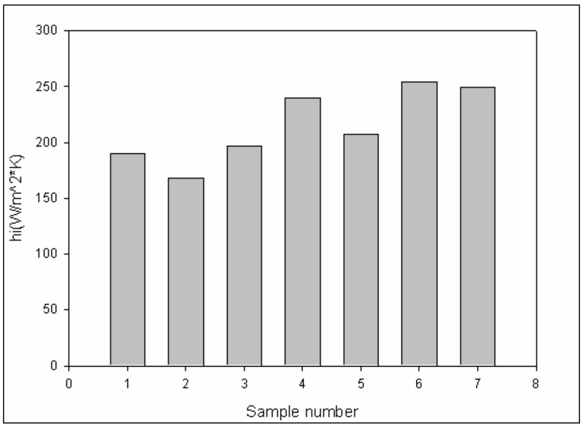

Fig. 3 Inertial heat transfer coefficient

For metal foams, the cross-section of the fibres is not circular and to account for this the shape factor,$d = (1 - e^{(-(1-\epsilon)/0.04)})df$, is introduced. (V. V. Calmidi[10]).

Setting the air velocity to be 2 m/s, using Eq. (9), the inertial heat transfer coefficient can be got as Fig. 3 shown.

From Fig. 3, we can see that the most thin fiber material (df=0.00025 m), sample 6 has the most high value of inertial heat transfer coefficient at the same velocity.

under the same condition(u=1 m/s, u=2 m/s u=3 m/s) for every sample, using Eq. (5) and Eq. (6) we can got the non-dimensionalized temperature of both solid phase and fluid phase. Results for Sample 1 and 2 are shown in Fig. 5 and Fig. 6, respectively.

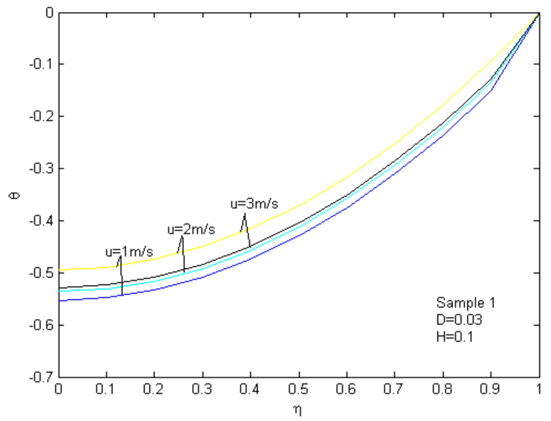

Fig. 5 Solid and fluid phase temperature profiles (Sample 1)

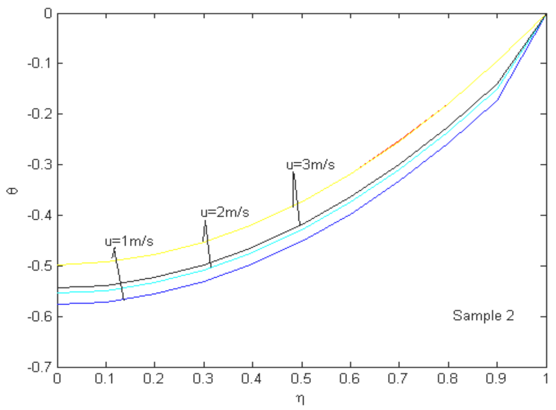

Fig. 6 Solid and fluid phase temperature profiles (Sample 2)

From above, we can notice that the temperature difference between solid fluid phase is very small, that means the thermal equilibrium is valid for our research scope. And that difference will be smaller as the velocity increase (as Fig. 5 and Fig. 6 show)

2.3 One thermal equation Model

From analysis above we know that, it is acceptable to use thermal equilibrium model to our case. The thermal equilibrium assumption allows us to replace the fluid temperature by the solid temperature on the right hand. N. Dukhan (2007) solve the equation :

$$\Theta(X,Y) = \alpha X + \frac{1}{2}Y^2 - \frac{1}{6} - 2\sum_{n=1}^{\infty}\frac{(-1)^n}{\Lambda_n^2}e^{-\alpha\Lambda_{nn}^2 X}\cos(\Lambda_n Y) \qquad (11)$$

where $\Theta(X,Y) = \dfrac{T-T_\infty}{qH/k_{s,eff}}$, (12)

$X = x/H$, $Y = y/H$, $\alpha = \dfrac{k_{s,eff}}{\epsilon\rho C_p uH}$, and

$\Lambda = \lambda_n H = n\pi$, and n=1,2,3...

Using Eq. (11) at u=2m/s, From Fig. 7 to Fig 13 are plotting the temperature profiles inside the 7 foam samples, respectively.

X means the direction of air flows, here we set X=0.5, 1.0, 1.5 and 2.0.

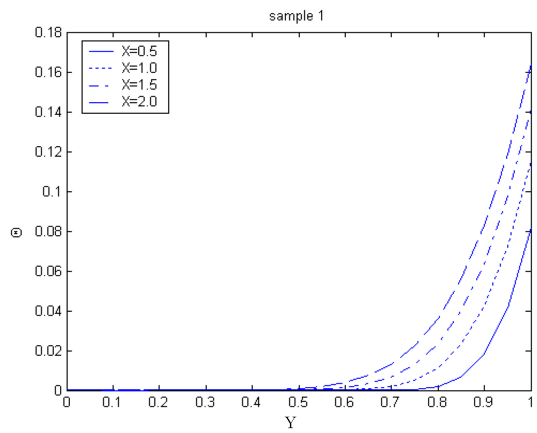

Fig. 7 Temperature distribution for sample 1 at u=2

From Fig.7 to Fig. 13, we noticed that the main trend is that higher effective conductivity cause higher heat transfer performance. That means among the properties of metallic foams, the main factor that influence the heat transfer when the fluid phase velocity is equals to each other is effective conductivities of foams.

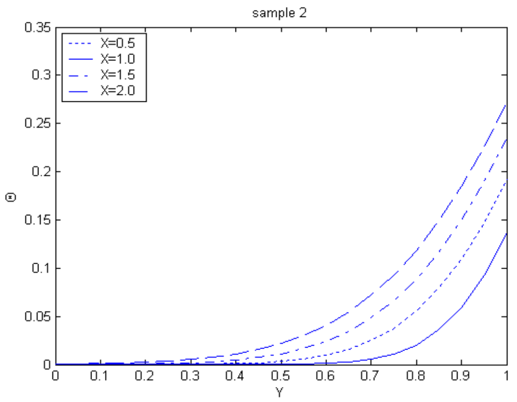

Fig. 8 Temperature distribution for sample 2 at u=2

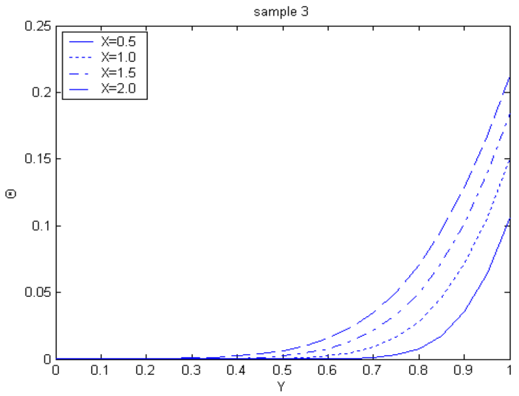

Fig. 9 Temperature distribution for sample 3 at velocity u=2

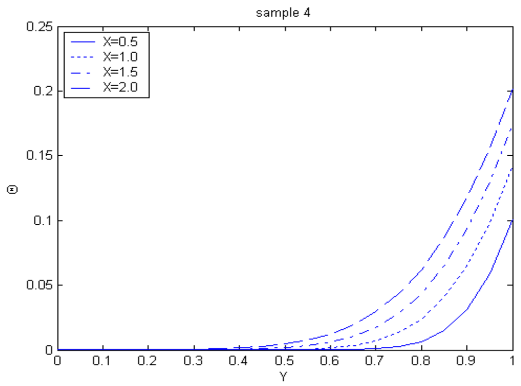

Fig. 10 Temperature distribution for sample 4 at u=2

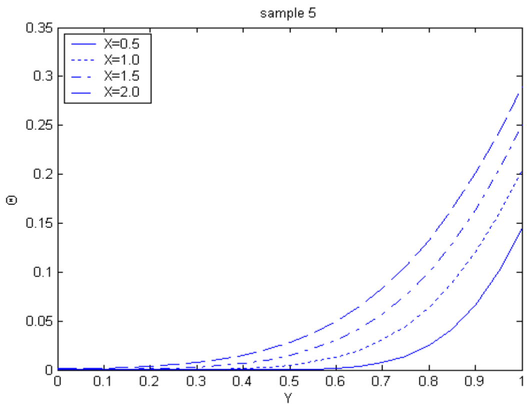

Fig. 11 Temperature distribution for sample 5 at u=2

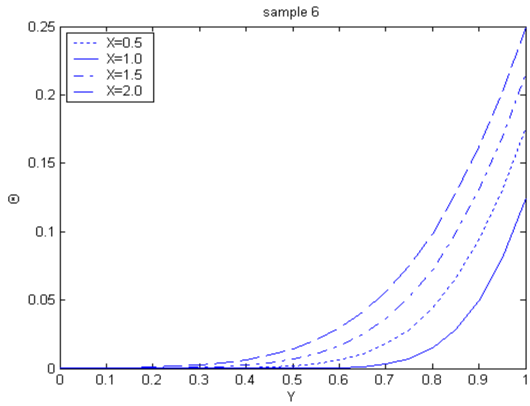

Fig. 12 Temperature distribution for sample 6 at u=2

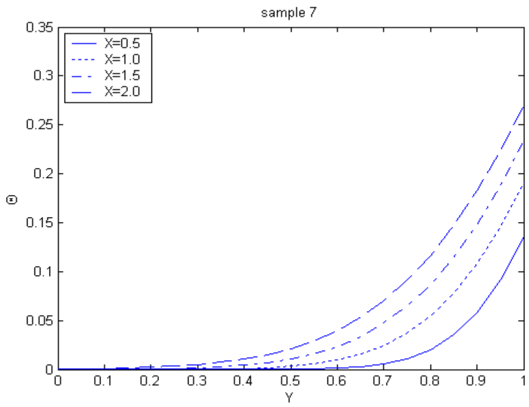

Fig. 13 Temperature distribution for sample 7 at u=2

3. Conclusions

In this paper, we took a study on the heat transfer in metallic foams subjected to constant heat flux. There are three kinds of heat transfer mechanisms exit, heat conduction in fibers, heat transfer by conduction in fluid phase, and internal heat change between solid and fluid phases. Firstly, we discuss the acceptance of applying thermal equilibrium among the two phases. then to calculate the dimensionless temperature profile along 7 metallic foams (The properties shown in Table 1). The result shows that the thermal equilibrium sumption is valid when applying heat exchanger applications. The result got under that sumption is that the effective conductivity of foams is main factor that influences heating performance.

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