

Study on the Thermal Performance of Heat Exchanger with Using Metallic Foam

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ABSTRACT: Since metallic foam will increase the performance of heat exchanger, it has caused many researchers' attention recently. Our research based on the model that metallic foams applied to heat exchanger. In this case, there is three kinds of heat transfer mechanisms, heat conduction in fibres, heat transfer by conduction in fluid phase, and internal heat change between solid and fluid phases. In this paper we study both the hydraulic and thermal aspect performance. Pressure drop along airflow direction will be presented. As thermal aspect, we first discuss the acceptance of applying thermal equilibrium among the two phases, then to calculate the dimensionless temperature profile, the heat transfer coefficient and Nu number in 7Aluminum foams.

Keywords: Metallic foam, Pressure drop, Heat transfer, Forced convection

NOMENCLATURE

a	Interfacial area per unit volume of porous media [m^{-1}]	ε	Porosity
Bi	Biot number defined in equation	η	non-dimensional transverse coordinate defined in equation
c_p	Specific heat [J/kgK]	γ	Geometric constant defined by equation
h_i	Heat transfer coefficient [$\text{W/m}^2\text{K}$]	θ	Non-dimensional temperature
K	Permeability	κ	Ratio of the effective fluid conductivity to that of the solid
$k_{e,ff}$	Effective conductivity [W/mK]	ρ	Density of air
PPI	Pores per inch		
q	Heat flux [W/m^2]		
u	Volume-averaged pore velocity [m/s]		
u_D	Darcian velocity [m/s]		
u_0	Average tunnel velocity entering [m/s]		
μ	Viscosity		

1. INTRODUCTION

Metallic foams have a distinct but continuous and rigid solid phase, and a fluid phase. as shown in Fig. 1. The use of open-cell metal foams have been widely increasing given by it diverse properties in various areas including aerospace, electronic

and automotive. They are typically available in high porosities; also have high thermal

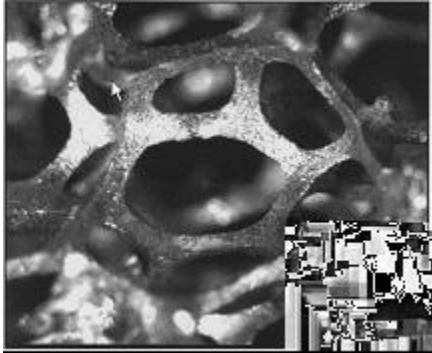


Fig. 1 Micro photo of metallic foam

conductivity and large area per unit volume. The properties make the material very promising applications include compact heat exchangers for airborne equipment, air-cooled condenser towers and compact heat sinks for power electronics.

Resent years some researchers have study on the calculative characteristics of metallic foam, (Boomsma and Poulikakos 2001) (Singh and Kasana 2004) and (Bhattacharya, Calmidi et al. 2002) developed models to calculate the effective conductivity of foams, respectively. (Calmidi 1998) porposed a specific formulation for permeability base on the experiment data. (S. Y. Kim 2000)experimented with the impact of porous fins on the pressure drop and heat transfer characteristics in plate-fin heat exchangers. In this paper, we concern both of hydraulic and thermal aspect performance with the aluminium foam, detailed process will be presented in the following.

2. Problem definition

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 (19.5 cm) in the flow direction, Width W (14 cm) and height H (7 cm), as shown in Fig. 2. The air flows through the

channel getting heat away from heating plate at velocity u m/s. To simplify the

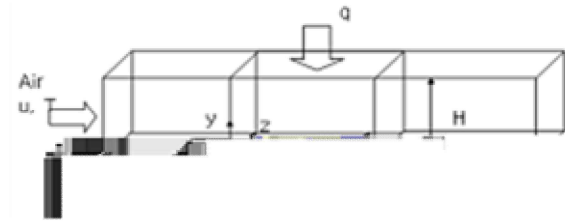


Fig. 1 Schematic diagram of research model

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 direction.
4. The flow is steady and fully developed

The properties such as PPI, porosity, permeability, inertial coefficient, effective conductivity, fibre diameter, pore size, area ratio are listed in Table 1. All of these data are documented in (Calmidi 1998)

Table 1 Properties of 7 Aluminium foam

	PPI	porosity	$K \times 10^7$ (m ²)	f	k_{se} (W/mK)	$df \times 10^5$ (m)	$dp \times 10^5$ (m)	a (m ²)
A1	5	0.973	2.70	0.097	2.48	50	402	415.42
A2	5	0.912	1.80	0.085	6.46	55	380	917.55
A3	10	0.949	1.20	0.097	4.10	40	313	799.63
A4	20	0.955	1.30	0.093	3.71	30	270	756.07
A5	20	0.901	0.90	0.088	7.19	35	258	1305.1
A6	40	0.927	0.61	0.089	5.48	25	202	1390.1
A7	40	0.913	0.53	0.084	6.37	25	180	1850.6

3. Pressure drop

The Forchheimer extended Darcy's equation is accepted here for this homogeneous, uniform, and isotropic metallic foam (P. Forchheimer)

$$-\frac{dp}{dx} = \frac{\mu u}{K} + \frac{\rho f}{\sqrt{K}} u^2 \quad (1)$$

Fig. 3 show the unit pressure drop when the air velocity ranges from 0m/s to 3 m/s in Aluminium foam. In our problem definition, the length of foam is 0.195 m along the air velocity, through calculation, Fig. 4 presents

the pressure drop when $u=2$ m/s. Generally, high PPI leads higher pressure loss, the

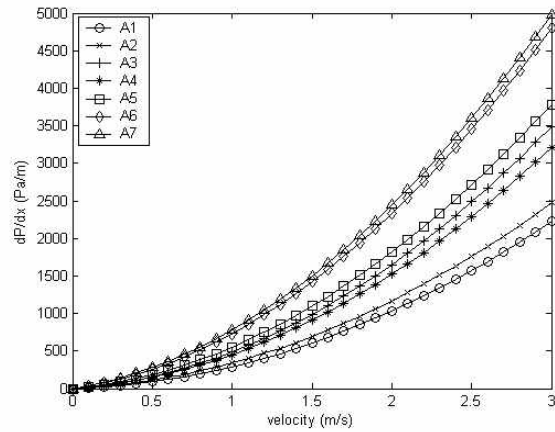


Fig. 3 Pressure drop vs velocity

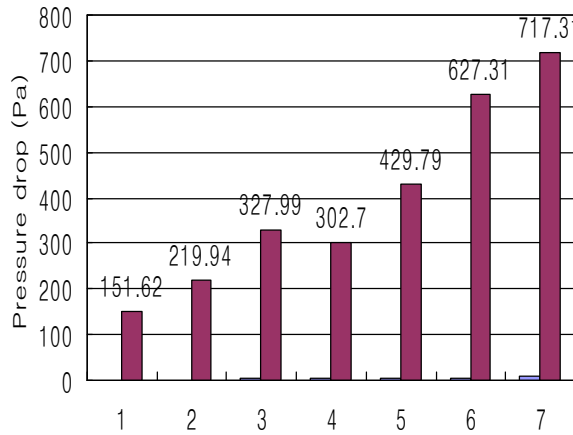


Fig. 4 Pressure comparison in 7 foams

pressure drop in 40 PPI foam is almost 3 ~ 4 times of that in 20 PPI foam. At a fixed porosity (for example A2 and A7, porosities of them are almost the same at the value of 0.912), decreasing the cell size increased the surface area to volume ratio which therefore increased the flow resistance by lowering the permeability and increasing the pressure drop. (Paek, Kang et al. 2000) did experimental work with aluminium foams, the work result agree with our analysis.

4. Convection in metallic foam

Energy equation in porous media has been considered as two or one by different

researchers. Here we first discuss the acceptance of applying thermal equilibrium among the two phases, and then calculate the dimensionless temperature profiles of foam and thermal performance

4.1 Acceptance of one thermal equation

Applying conservation of energy for steady state conditions to the solid and the fluid with a control volume inside the foam, we obtain from (Lu, Zhao et al. 2006)

fluid phase

$$k_{f,eff} \nabla_y^2 T_f + h_t a (T_s - T_f) = \rho c_p u \frac{\partial T_f}{\partial x} \quad (2)$$

solid phase

$$k_{s,eff} \nabla_y^2 T_s - h_t a (T_s - T_f) = 0 \quad (3)$$

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

$$\frac{\partial T_f}{\partial y} \Big|_{y=0} = \frac{\partial T_s}{\partial y} \Big|_{y=0} = 0 \quad (4)$$

The governing equation can be rendered dimensionless using the following non-dimensional variables:

$$\theta = \gamma \frac{k_{s,eff} (T - T_w) / H}{q}, \eta = \frac{y}{H} \quad (5)$$

where, $\gamma = \frac{D}{4H}$, in which D is the hydraulic diameter of the channel.

the temperature distribution was obtained by D.Y. Lee, the resultant equations are

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

$$\left[\frac{1}{2} (\eta^2 - 1) + \frac{\kappa}{Bi(1+\kappa)} \left\{ 1 - \frac{\cosh(\lambda \eta)}{\cosh(\lambda)} \right\} \right] \quad (7)$$

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

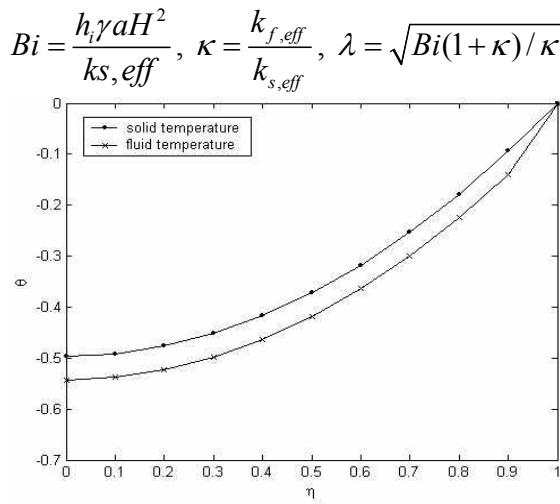


Fig. 5 Temperature distributions for solid and fluid domain

For foamed materials, there is no general model for the interfacial heat transfer coefficient, so the following correction developed by (Zukauskas 1987) which is valid for staggered cylinders, is used to estimate h_i

$$Nu_{sf} = 0.52 Re_d^{0.5} Pr^{0.37} \quad (8)$$

where $Re = ud / \nu$,

For metal foams, the cross-section of the fibers is not circular and to account for this the shape factor, is introduced

$$d = (1 - e^{-(1-\varepsilon)/0.04}) d_f$$

Fig. 5 represents A3 temperature profile respectively at $u = 2\text{m/s}$. From them, we can notice that the temperature difference between solid and fluid phase is very small, that means the thermal equilibrium is valid. For aluminium foam cooled by air, the temperature of the solid and fluid are similar. (Kim and Jang 2002) stated that the thermal equilibrium assumption is always valid when the solid conduction is the dominant heat transfer mode

4.2 One thermal equation

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

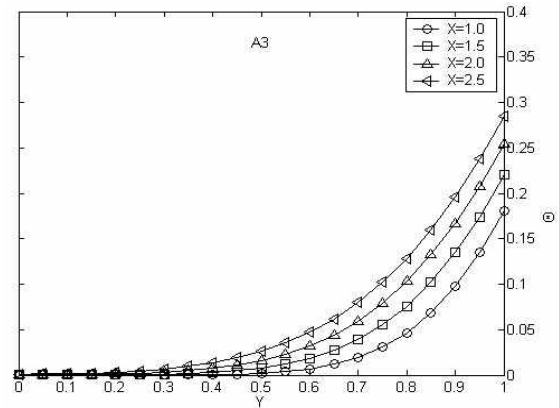


Fig. 6 Temperature profile in A3 foam

right hand. following single governing equation for the heat transfer inside the metal foam is obtained.

$$k_{s,eff} \frac{\partial^2 T}{\partial y^2} = \varepsilon \rho c_p u \frac{\partial T}{\partial x} \quad (9)$$

the boundary conditions are:

at $y = 0$

$$\frac{\partial T}{\partial y} = 0 \quad (10)$$

at $y = H$,

$$k_{s,eff} \frac{\partial T}{\partial y} = q \quad (11)$$

at $x = 0$,

$$T = T_\infty \quad (12)$$

the solution of above equations is available in reference.

$$\Theta(X, Y) = \alpha X + \frac{1}{2} Y^2 - \frac{1}{6} - 2 \sum_{n=1}^{\infty} \frac{(-1)^n}{\Lambda_n^2} e^{-\alpha \Lambda_n^2 X} \cos(\Lambda_n Y) \quad (13)$$

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

$$X = x / H, Y = y / H, \alpha = \frac{k_{s,eff}}{\varepsilon \rho c_p u H}, \text{ and}$$

$$\Lambda = \lambda_n H = n\pi, \text{ and } n=1,2,3,\dots$$

we equate the volume averaged pore velocity u to the Darcean velocity u_D which is defined

$$u_D = u_0 / \varepsilon \quad (14)$$

Here the velocity is all mentioned as Darcean velocity.

Using Eq. (11) at $u=2\text{m/s}$, we can get the temperature inside the foam samples. X means the direction of air flows, here we set $X=1.0, 1.5, 2.0, 2.5$. Fig. 6 show A foam. To compare all of 7 foams, we compare their Nu number at the same velocity, $u=2\text{ m/s}$

$$h_w = \frac{q}{T_{f,bulk} - T_w} \quad (15)$$

$$T_{f,bulk} = \frac{\int_{y=0}^{y=H} T_f dy}{H} \quad (16)$$

$$Nu = \frac{\bar{h}_w H}{k_f} \quad (17)$$

Fig. 7 show the heat transfer coefficient along the X direction for 7 A foams and 7 F foams.

Fig. 8 compares the Nu number between 7 foams. We can see that not like the hydraulic performance (pressure drop), thermal performance do not simply have the same trend with PPI, it is influenced by the effective conductivity. At same u , higher effective conductivity lead to higher Nu number.

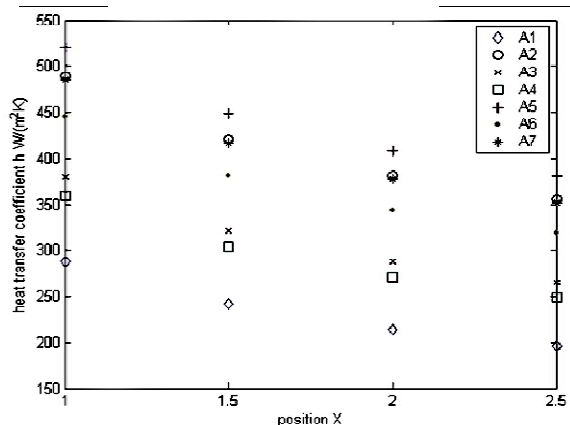


Fig. 7 Heat transfer coefficient profile along x direction in 7 foams

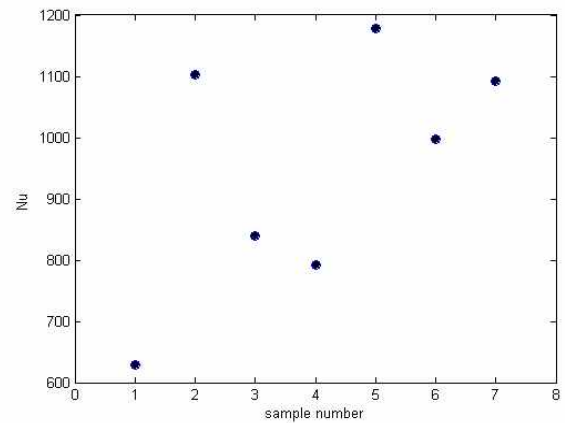


Fig. 8 Nuselt number comparison in 7 foams at $u=2\text{ m/s}$

5. CONCLUSION

In this paper, we took a study on the pressure drop and heat transfer in metallic foams subjected to constant heat flux. 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 assumption is valid when applying heat exchanger applications. Take both of hydraulic and thermal performance into consideration, foam 2 is the optimal choice in our case, which have high Nu but relatively low pressure drop compared to other foam.

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