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Heat Transfer Characteristics of Aluminium and FeCrAlY Foam

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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 study both the hydraulic and thermal aspect performance. Pressure drop along air flow 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 14 metallic foams(7 Aluminium foams, 7 FeCrAlY foams).All these discussion is based on the same velocity u=2 m/s.

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

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equation

θ : non-dimensional temperature

κ : ratio of the effective fluid conductivity to that of the solid

λ : parameter defined by equation

ϵ : porosity

ρ : fluid density

Subscripts

eff : effective value

f : fluid phase

i : internal heat exchange

s : solid phase

w : wall

∞ : 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. Typically the properties of commercial available foam are given by the manufacture, these include PPI, porosity, effective conductivity, permeability, inertial coefficient, etc. Resent years some researchers have study on the calculative characteristics of metallic foam. Boomsma and Poulikakos 2001[1], (Bhattacharya, Calmidi et al. 2002 [(2)], Singh and Kasana 2004 [(3)] developed models to calculate the effective conductivity of foams, respectively. (Calmidi 1998 [(4)] porosed a specific formulation for permeability base on the experiment data. (Lu, Zhao et al. 2006 [(5)] present a formulation to get area density a.In this paper, we concern both of hydraulic and thermal aspect performance with the aluminum foam and FeCrAlY foam. Detail process is presented in the following.

2. Problem definition

2.1 Schematic description

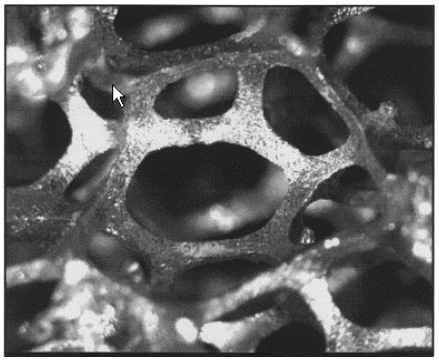

Fig. 1 Metallic foam

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 flow through the channel getting 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 Properties of metallic foam

The material scope of this paper is Aluminium and FeCrAlY. From now, we call these two foam A foam and F foam, respectively. The all data of Aluminium is documented in Calmidi (1998)[(4)] and list in Table 1. As for FeCrAlY , the reference David

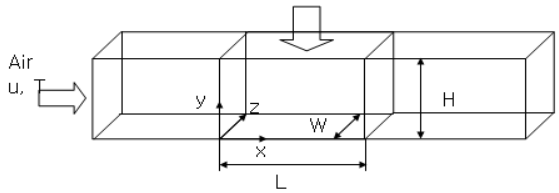

Fig. 2 Schematic of problem

problem definition , the length of foam is 0.195 m along the air velocity, through calculation, Fig 5 and Fig. 6 present the pressure drop when u=2 m/s. We notice that the most big value happened to F6, 8780 Pa pressure will lost, that means the higher fan power needed to get the air flow

Table. 1 Properties of Aluminium foam

	PPI	porosity	Kx10^7 (m^2)	f	kse (W/mK)	dfx10^5 (m)	dpx10^5 (m)	a (m2)
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

Table. 2 Properties of FeCrAlY foam

	PPI	porosity	Kx10$_7$ (m_2)	f	kse (W/mK)	dfx10$_5$ (m)	dpx10$_5$ (m)	a (m2)
F1	10	0.950	1.67	0.093	0.19	30.6	254	916.15
F2	10	0.850	0.50	0.13	0.69	38.7	254	1586.8
F3	30	0.950	1.00	0.15	0.19	10.2	84.7	2748.4
F4	30	0.900	0.50	0.164	0.44	11.2	84.7	3886.9
F5	60	0.950	0.33	0.24	0.19	5.1	42.3	5496.9
F6	60	0.850	0.11	0.49	0.69	6.45	42.3	9520.9
F7	30	0.925	1.00	0.2	0.32	10.5	84.7	3366.1

P. Haack[6] provide only PPI, porosity, permeability and inertial coefficient,
the value of effective conductivity and df , dp, a are calculated and listed in Table 2.

3. Pressure drop

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

$$\frac{\Delta p}{L} = au + bu^2 \qquad (1)$$

Fig. 3 and Fig. 4 show the unit pressure drop when the air velocity range from 0m/s to 3 m/s in Aluminium and FeCrAlY foams. In our through F6 at the velocity u=2 m/s.

4. Heat transfer

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. then to calculate the dimensionless temperature profiles in all A foams and F foams.

4.1 Acceptance of one thermal equation

Based on the above assumptions, the following

control equations are obtained from Amiri and Vafai and Amiri et[5,6].

Fluid phase

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

Solid Phase

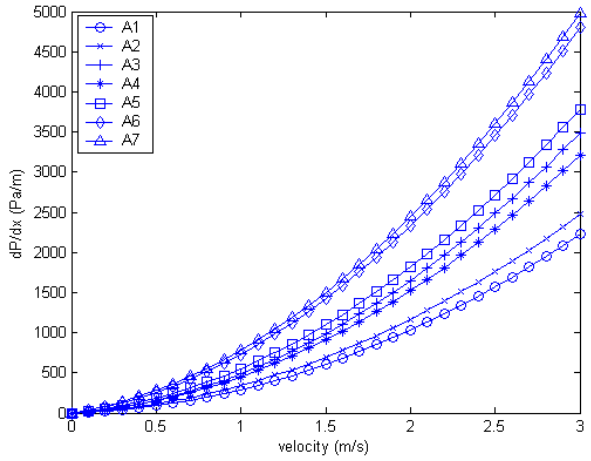

Fig. 3 Unit pressure drop in A foams

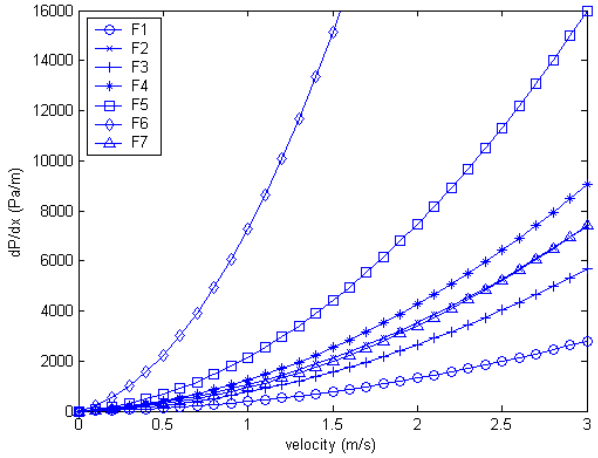

Fig. 4 Unit pressure drop in F foams

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

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

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

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

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

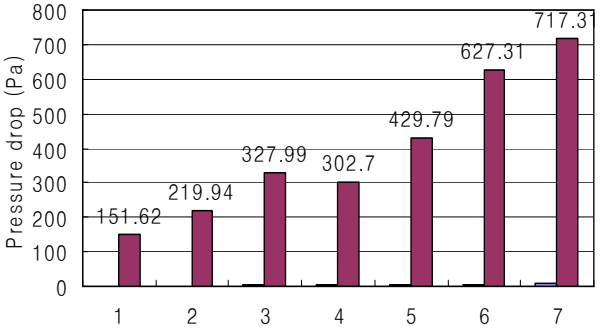

Fig. 5 A foam pressure drop at u=2 m/s

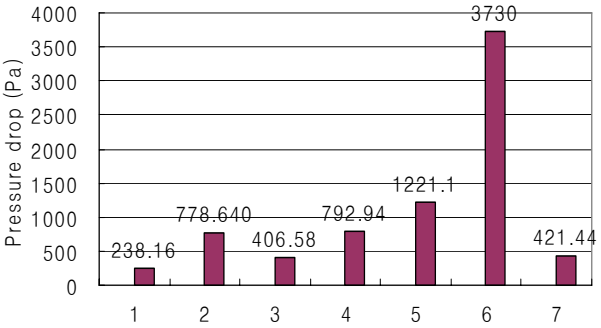

Fig. 6 F foam pressure drop at u=2 m/s

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

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

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

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

$$Bi = \frac{h_i \gamma a H^2}{k_{s,ef}}, \kappa = \frac{k_{f,eff}}{k_{s,eff}},$$

$$\lambda = \sqrt{Bi(1+\kappa)/\kappa}$$

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[8] , which is valid for staggered cylinders, is used to estimate h_i

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

where Re_d is the local Reynolds number,

$$Re_d = ud/v$$

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

Fig. 7 and Fig.8 represent A3 and F1 temperature profile respectively at u= 2m/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 our research scope. (as Fig. 7 and Fig. 8 show). Also, since the relative low thermal conductivity of F foams, the solid and fluid temperature is almost same.

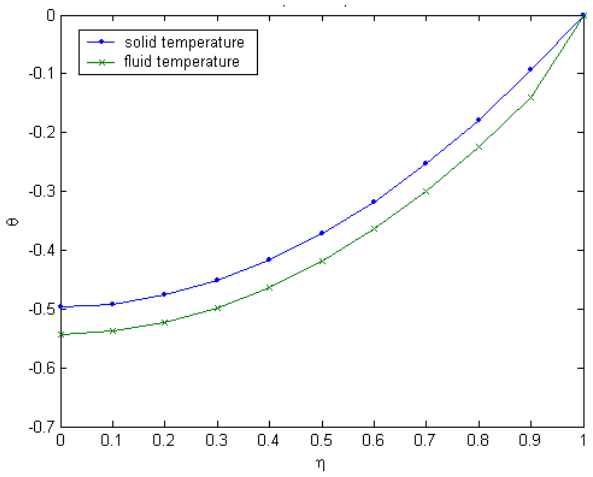

Fig. 7 Dimensionless temperature in A3 sample y direction

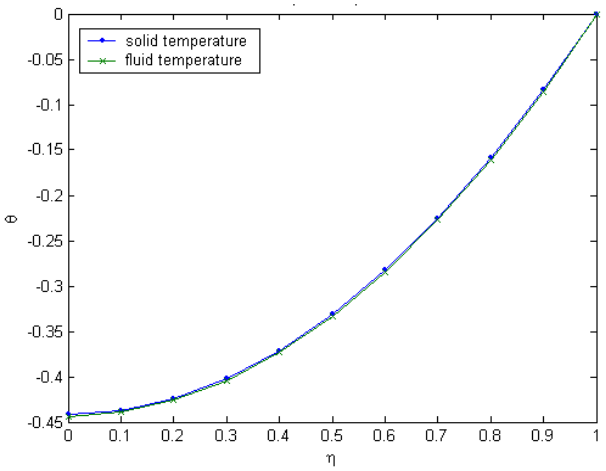

Fig. 8 Dimensionless temperature in A3 sample y direction

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 right hand. Dukhan and Chen[9] (2007) solved the equation

$$\Theta(X,Y) = \alpha X + \frac{1}{2}Y^2 - \frac{1}{6} - 2\sum_{n=1}^{\infty} \frac{(-1)^n}{\Lambda_n^2} e^{-\alpha \Lambda_{nn}^2 X} \cos(\Lambda_n Y) \quad (9)$$

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

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

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

Using Eq. (11) at u=2m/s, we can get the temperature inside the foam samples and F samples respectively. X means the direction of air flows, here we set X=1.0, 1.5,2.0,2.5.

Here present the temperature profiles in A3 and F1 foam (since the two foam have same PPI and almost same porosity, it's convenient to make comparison), shown in Fig. 9. Easy to find that A3 has much better heat transfer performance than F1 due to its relatively high effective conductivity 4.10 W/mK compared to F1 effective conductivity 0.19 W/mK.

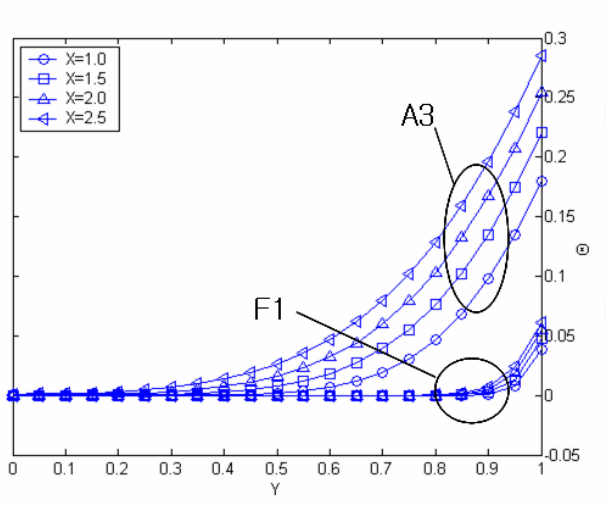

Fig. 9 Dimensionless temperature profile in A3 and F1 sample

To compare all of 14 foams, we compare their Nu number at the same velocity, u=2 m/s

$$h_w = \frac{q_w}{T_{f,bulk} - T_w} \tag{10}$$

$$T_{f,bulk} = \frac{\int_{y=0}^{y=H} T_f dy}{H} \tag{11}$$

$$T_w = T_f|_{y=H} \tag{12}$$

$$Nu = \frac{\overline{h_w} H}{k_f} \tag{13}$$

Fig. 10 and Fig 11 show the heat transfer coefficient along the X direction for 7 A foams and 7 F foams.

Fig. 12 compare the Nu between 14 foams. we can see that typically A foams performance better than F foams.

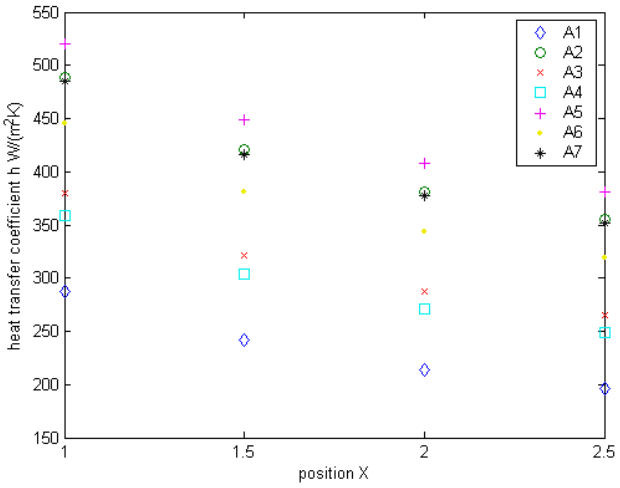

Fig. 10 Heat transfer coefficient along x direction in 7 A foams

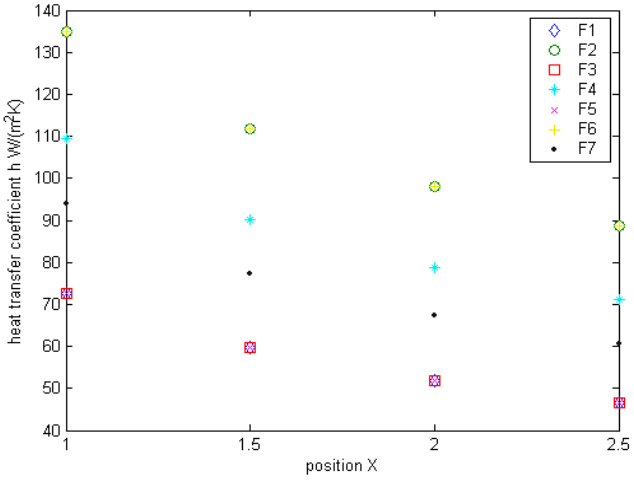

Fig. 11 Heat transfer coefficient along x direction in 7 F foams

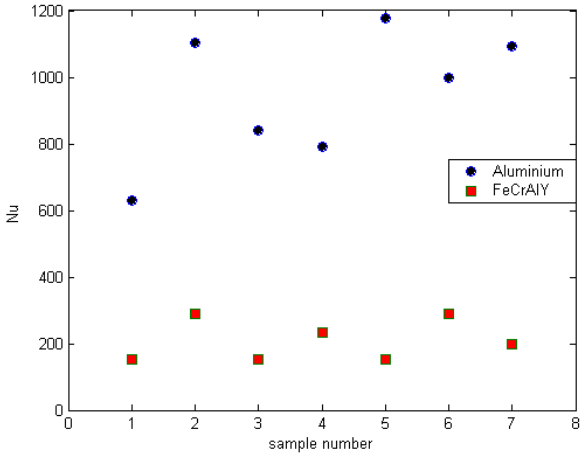

Fig. 12 Compare Nu in 14 Foams

5.Conclusion

The analysis result shows that the thermal equilibrium assumption is valid when metallic foam applying heat exchanger applications, especially for F foam. By comparing the Nu number of 14 foams (7 Aluminium foams, 7 FeCrAlY foams.) at the same velocity u= 2m/s, we conclude that A foams performance better than F foams due to its relatively high effective conductivity. By the mean while, we consider the air pressure drop under the same air velocity, u=2 m/s, it is because typically when apply to design the heat exchanger, pressure drop should be taken into consideration, this paper present the unit pressure drop when range u from 0 to 3 m/s, and specific pressure drop as our problem definition (L=0.195 m) at u= 2 m/s.

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