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Technical Note

Forced convection from aluminum foam materials in an asymmetrically heated channel

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1. Introduction

With the pursuit of compactness for thermal systems, efficient removal of high-powered heat from the system has emerged as a crucial task. There have been lots of technical literatures on convective electronics cooling and heat-exchange problems for the reason [1]. A crux of these studies is how to enhance heat transports for given flow geometry and externally imposed physical constraints.

In an attempt to enhance convective thermal transport, we have focused on the utilization of aluminum foam materials in thermal systems. The motivation is attributed to the high surface area to volume ratio as well as enhanced flow mixing due to the tortuosity of aluminum foam. Furthermore, aluminum foams have excellent characteristics in the structural strength as well as the simple manufacturing process of foaming [2]. Therefore, it is believed that thermal performance of system can be substantially enhanced by using aluminum foam.

The present experimental study investigates the impact of the presence of aluminum foam on the flow and convective heat transfer in an asymmetrically heated channel. The aluminum foams tested in this experimental investigation are made of aluminum-6101 alloy that has three different permeability at a porosity of $\varepsilon = 0.92$. Aluminum foam is placed inside a channel, in which the upper wall is maintained at a constant temperature while the lower wall is thermally insulated. The simple correlation of the friction factor f and the average Nusselt number Nu of aluminum foams will be sought to provide a guide in practical applications.

2. Experimental apparatus and test procedure

A schematic diagram of the experimental apparatus is shown in Fig. 1. The experiments were conducted in a channel fabricated of Plexiglas of height $H = 9.0$ mm, width $W = 90.0$ mm. Test specimens of aluminum foams were made of height $H = 9.0$ mm, $W = 90.0$ mm and length $L = 188.0$ mm. Pressure taps were installed at 5 mm upstream and downstream of test section. The flow straightner was placed at 50 mm upstream of test section to make a unidirectional uniform flow. One end of the channel was open to atmosphere while the other end was connected to a calming chamber in which the compressed air of 3 atm is introduced through a rotameter. Frontal air velocities U_i tested in the present experiment ranged from 1.1 to 5.4 m/s to cover the operational range for commercial heat exchangers. The corresponding Reynolds number based on the channel height H was $Re = 570$–2800. To measure the pressure drop through the test specimen, an inclined manometer was placed in the channel.

A hot bath was mounted on the upper wall of the channel to provide a constant wall temperature condition. The water circulation loop supplied hot water to the bath, as seen in Fig. 1. Two channel walls were fastened together by using six C-shaped clamps to reduce thermal contact resistance between the bath wall and test specimen. As the compression loading was increased, convective heat transfer rate approached an asymptotic value that was considered as a condition for minimal contact resistance. Three copper–constantan thermocouples were inserted between the hot bath wall and the surface of test

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Nomenclature

A_f	frontal area of test specimen (m^2), WH
A_W	heat transfer area at the base of test specimen (m^2), WL
C_p	specific heat at constant pressure (J/kg K)
Da	Darcy number, K/H^2
f	friction factor, Eq. (1)
H	height of aluminum foam (m)
h	space-averaged heat transfer coefficient, Eq. (3) (W/m^2 K)
K	permeability of aluminum foam (m^2), $K = -\rho\nu U_i/(\Delta P/L)$
k	thermal conductivity of air (W/m K)
L	length of aluminum foam (m)
Nu	average Nusselt number, Eq. (2)
Pr	Prandtl number
Re	Reynolds number based on the channel height H, U_iH/ν
T_i	inlet air temperature (K)
T_o	outlet air temperature (K)
T_w	wall temperature (K)
U_i	frontal air velocity (m/s)
W	width of aluminum foam (m)

Greek symbols

ΔP	pressure drop of air flow (Pa)
ΔT	temperature difference of air between the inlet and the outlet, Eq. (4) (K)
ΔT_m	log mean temperature difference, Eq. (5) (K)
ε	porosity
ν	kinematic viscosity (m^2/s)
ρ	density of air (kg/m^3)

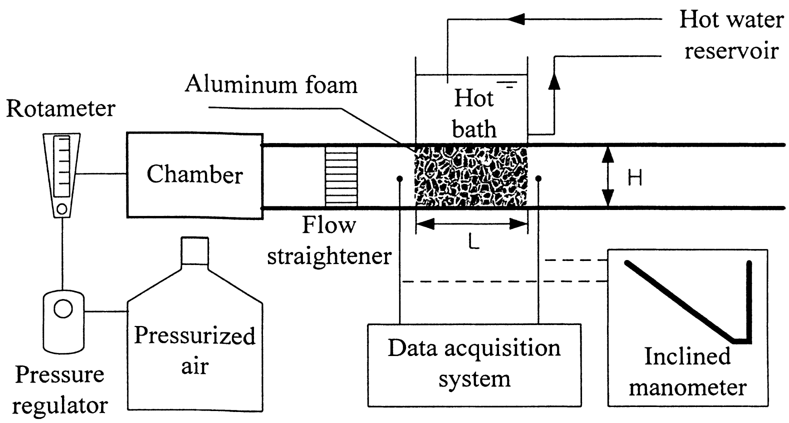

Fig. 1. Experimental setup.

specimen to insure the uniformity of the wall temperatures. The thermocouple wires were installed through the V-shaped grooves of 1.5 mm depth and 1.5 mm width that were fabricated into the surface of the hot bath wall. All the channel walls were carefully insulated except for the upper wall. Five copper–constantan thermocouples were vertically distributed at 5 mm downstream of test section to measure the bulk temperature of air at the exit.

Experiments were started by inducing controlled airflow to the channel and by maintaining a constant temperature at the hot bath wall. The pressure drop through the test section was measured from the inclined manometer. The temperatures were monitored during heat-up period by a data acquisition system (Yokogawa DR230). After reaching thermal steady-state, all the temperature data were recorded on the storage device for analysis.

3. Results and discussion

In the present study, the specimens of porosity $\varepsilon = 0.92$ were selected because the porosity is comparable to that of conventional fins used in air to air heat exchangers. The aluminum foams have three distinct pore densities, i.e, 10, 20 and 40 PPI (pores per inch). Hereafter, we refer them to aluminum foams (A), (B) and (C), respectively. As described at Table 1, the permeability K of aluminum foam gradually decreases with the increase of pore density (PPI). The permeability of aluminum foams was determined from the measured data of pressure drop. The effective thermal conductivity of the aluminum foam is little affected by the increase of pore density [3].

First, we define the friction factor f and Nu to assess the pressure drop and the heat transfer rate of the aluminum foams

Table 1
Flow parameters for aluminum foams

Aluminum foam	(A)	(B)	(C)
Materials	Al-6101	Al-6101	Al-6101
Porosity, ε	0.92	0.92	0.92
Pore density (PPI)	10	20	40
Permeability, K (m^2)	1.04×10^{-7}	0.76×10^{-7}	0.51×10^{-7}
Darcy number, $Da = K/H^2$	1.3×10^{-3}	9.4×10^{-4}	6.3×10^{-4}
Surface area to volume ratio (m^2/m^3)	790	1720	2740
Effective thermal conductivity, k_e (W/m K)	5.33	5.56	6.01

$$f = \left(\frac{\Delta P}{L}\right) H/\rho U_i^2, \tag{1}$$

$$Nu = \frac{hH}{k}. \tag{2}$$

The h denotes the space-averaged convective heat transfer coefficient that is determined from

$$h = \frac{\rho C_p U_i A_f \Delta T}{A_w \Delta T_m}. \tag{3}$$

Here, the temperature difference of air between the inlet and the outlet ΔT and the logarithmic mean temperature difference (LMTD) ΔT_m are

$$\Delta T = T_o - T_i, \tag{4}$$

$$\Delta T_m = \frac{\Delta T}{\ln[(T_w - T_i)/(T_w - T_o)]}. \tag{5}$$

To get a correlation of friction factors for the aluminum foams, the friction factor data are converted by using the non-dimensional groupings ($f\,Da^{1/2}$) and ($Re\,Da^{1/2}$), as demonstrated in Fig. 2. Then, it is found that the present experimental data generally agree to the correlation suggested by Beavers and Sparrow [4] for nickel foam metals. Consequently, we may give a general correlation of friction factors f for the aluminum foams used in the present study as follows:

$$f = \frac{1}{Re\,Da} + \frac{C_E}{Da^{1/2}}. \tag{6}$$

The inertia coefficient C_E varies around $C_E = 0.1$ for the present aluminum foams. It should be noted that the above equation implicates the Forchheimer-extended Darcy model for porous media, which includes the inertia effect [5].

The Nu as a function of Re is plotted in Fig. 3. The Nu is substantially augmented as the Re increases. The enhancement of Nu is much more pronounced for low permeable aluminum foam (C). This is attributed to large interstitial surface area of low permeable aluminum foam. The Nusselt numbers for an aluminum fibrous medium obtained by Hunt and Tien [6] show similar values. For all the aluminum foams, the uncertainty of Nu from the repeated experiments was less than 12.3%.

For a channel without foam material in which one wall is maintained at a constant temperature and the other wall is insulated, the fully developed Nusselt number is 2.43 [7]. Therefore, use of aluminum foam

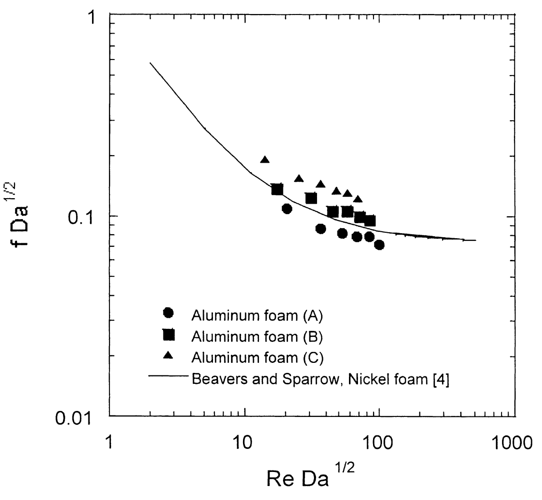

Fig. 2. Modified friction factors of aluminum foams.

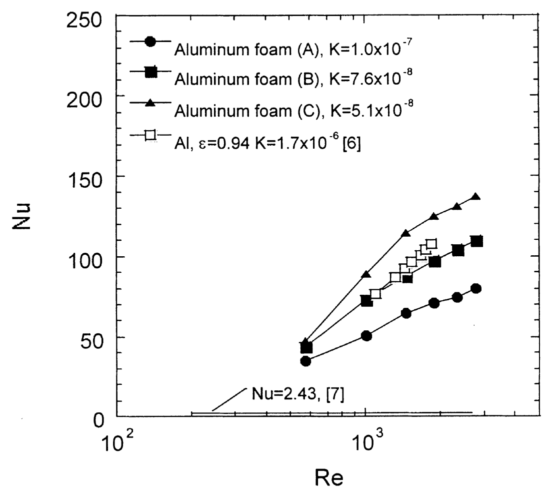

Fig. 3. Average Nusselt numbers of aluminum foams as a function of Reynolds number.

materials as heat dissipating elements may dramatically enhance the overall heat transfer rates from thermal systems.

In an attempt to get a Nusselt number correlation of the aluminum foams, the non-dimensional parameter $Da^{1/2}$ is applied again. When the Nusselt number data in Fig. 3 are converted using this non-dimensional parameter, the data merge to a single line in the log–log plot exhibited in Fig. 4. Then, it gives an empirical Nusselt number correlation with maximum 8% deviation for $1000 < Re < 3000$

$$Nu = 0.0159 Re^{0.426} Pr^{1/3} Da^{-0.787}. \quad (7)$$

The experimental data of the Nu for $Re < 1000$ in Fig. 4 were dropped from the derivation of the Nusselt number correlation, because of the saturated logarithmic mean temperature difference ΔT_m due to the long test specimen of the aluminum foam.

It should be noted that Eq. (7) does not include the effective thermal conductivity of the aluminum foam, and therefore, the thermal performance of aluminum foams can be predicted from measuring permeability according to the flow velocity. Owing to the high fin effectiveness of aluminum ligament, most solid portion of aluminum foam is maintained at a temperature close to the hot wall. Therefore, the heat transfer from the aluminum foam is mainly governed by the total heat transfer area of the aluminum foam rather than the thermal conductivity. In the present study, the total heat transfer area is a function of the Darcy number as seen in Table 1. Consequently, the Nusselt number correlation in Eq. (7) is not explicitly expressed by the effective thermal conductivity of the aluminum foams. If a foam material of low-thermal conductivity is considered, then the heat transfer may be significantly affected by the thermal conductivity of the foam due to its low fin effectiveness [8].

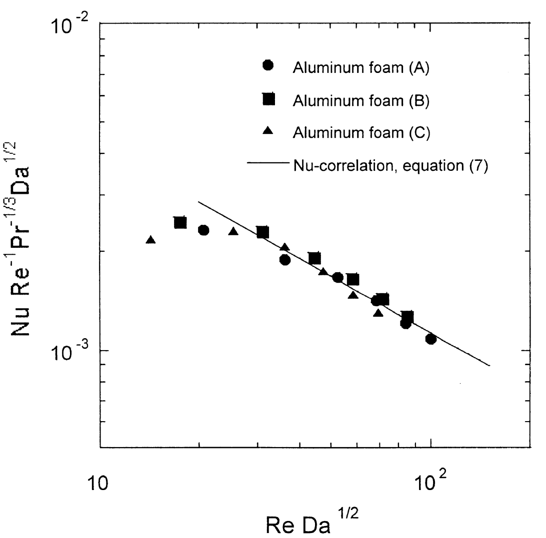

Fig. 4. Modified Nusselt numbers of aluminum foams.

4. Conclusion

An experimental investigation on the flow and convective heat transfer characteristics for aluminum foam in an asymmetrically heated channel has been performed. Three aluminum foams of various permeability and a porosity of $\varepsilon = 0.92$ were selected to provide the friction factor and heat transfer correlations for future design of thermal applications.

Experimental results indicate that the friction factor is much higher at the lower permeable aluminum foams while the significant enhancement in Nu is obtained. It is also found that the friction factor and Nu for aluminum foams merge to a single curve by using the non-dimensional parameter $Da^{1/2}$. Consequently, the friction factor and Nusselt number of aluminum foams can be obtained from the simple experimental measurements of permeability K.

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