Investigation of natural convection from intermittent foam blocks in a cavity

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Abstract: Natural convection heat transfer from intermittent open-celled aluminum metallic foams in a cavity has been investigated experimentally. Air was used as the working fluid. The test section was equipped with 3x3 aluminum-foam blocks placed on copper blocks subjected to uniform heat flux. The convective heat transfer rate from smooth heated surfaces without foam blocks is compared with the results obtained from 10 PPI aluminum foam blocks. The averaged temperature and the averaged Nusselt number distributions on the heating surface were tested under different heating power. The experimental study was made for modified Grashof numbers 1.19×10^6 to 7.45×10^6. As a result of comparisons, the aluminum-foam blocks with the pore density 10 PPI showed about 86% higher averaged- Nusset number than that without the foam blocks in a cavity.

Keywords: Natural convection, Aluminum-foam blocks, Electronics cooling.

Nomenclature

A_s	heat source surface area, m^2
g	gravitational acceleration, m/s^2
Gr^*	modified Grashof number
k	thermal conductivity, W/mK
L	length and width of metal foam sample or heater (m)
Nu_{avg}	average Nusselt number
q_c	convection heat flux, W/m^2
Q_c	convection heat transfer rate, W
Q_{cond}	conduction heat transfer rate, W
Q_{tot}	total power dissipation, W
R	electrical resistance, Ω
T_{avg}	average surface temperature, ^{0}C
T_0	ambient temperature, ^{0}C
V	voltage, V

Greek symbols

β	Thermal expansion coefficient, 1/K
ε	Porosity
ν	Kinematic viscosity, m^2/s

Introduction

Because of the excellence physical characteristics and good mechanical properties, open-celled metal foam is a kind of new engineering materials. The main advantage is the high surface-area-to-volume ratio, which leads to enhanced heat transfer and miniaturization of thermal system. Use of metal foams in electronic cooling applications is novel. Overheating which causes failure is mainly a major problem for electronic equipments. Almost all of the electrical energy consumed by electronic devices appears as heat energy, the power density that must be dissipated by individual chips called heat sources. Natural convection in a confined space has extensive application prospects, such as, compact heat exchangers for electronic cooling, nuclear reactor, porous radiant burner and heat pipes.

There are relatively few investigations of the heat transfer with free convection in very high porosity media ε~0.9, such as metal foams. Xu et al. (2011) comprised an experimental study for natural convection from horizontally-positioned copper metallic foams having different porosities and pore densities. They quantified the effects of porosity and pore density on the total thermal resistance of the foam sample. It is found that porous surface enhanced natural convection and reduced thermal resistance by about 20% in comparison with a smooth surface. The porosity influenced on the heat transfer performance was more remarkable when the pore density was higher. Zhao et al. (2005) investigated natural convection and its effect on overall heat transfer in highly porous, open-celled cellular FeCrAlY foams experimentally and numerically. They found that natural convection is very significant in metal foams due to the high porosity and inter-connected open cells, contributing more than 50% of the effective conductivity at the ambient pressure. Zhao et al. (2004) measured the effective thermal conductivity of five FeCrAlY foam samples with different pore sizes and relative densities using a guarded-hot-plate apparatus under both vacuum and atmospheric conditions. The obtained results showed that effective thermal conductivity increases rapidly as temperature raises, especially in the higher temperature range where the thermal radiation dominated the transport. Chamkha et al. (2002) considered thermal buoyancy-induced, hydromagnetic flow of an absorbing fluid along an inclined, semi-infinite, ideally transparent flat plate embedded in a variable porosity porous medium due to solar radiation. Phanikumar and Mahajan (2002) resented present numerical and experimental results for buoyancy-induced flows in high porosity metal foams heated from below. They obtained Brinkman–Forchheimer-extended Darcy flow model and a semi-heuristic two-equation energy model by relaxing the local thermal equilibrium (LTE) assumption are adopted. Experiments conducted under natural convection conditions for the same configuration were used to test the numerical model and the validity of the thermal equilibrium assumption for metal foams. They found that heat transfer rate for a given Rayleigh number decreased as the pore density increased from 5 to 40 PPI.

Natural convection flow of micro-polar fluid from a permeable uniform heat flux surface in porous medium was considered by Hassanien et al. (2004). It was found that the enhancement of the wall heat transfer represented by increasing in the Nusselt number. But the values of the skin friction and couple stress decreased as the suction parameter increased. Also, it was seen that the Darcy number and inertia effect tend to decrease the skin friction, Nusselt number, and couple stress. The micro-polar parameter enhanced the skin friction and reduced the heat transfer rate. Bhattacharya et al. (2002) presented experimental results on buoyancy-induced convection in aluminum metal foams of different pore densities [corresponding to 5, 10, 20, and 40 pores per in. (PPI)] and porosities (0.89–0.96).The obtained results showed that compared to a heated surface, the heat transfer coefficients in these heat sinks were five to six times higher. However, when compared to commercially available heat sinks of similar dimensions, the enhancement was found to be marginal. The experimental results also showed that for a given pore size, the heat transfer rate increased with porosity, suggesting the dominant role played by conduction in enhancing heat transfer.

As a result of an extensive literature survey, there are few natural convection investigations of metal foams and few works have been reported for air natural convection of aluminum foam in a cavity. The objective of this research is to investigate experimentally the buoyancy induced flow in open-celled aluminum-foam blocks heated from below and surrounded by air. What makes this work realistic is to choose 3x3 configurations in a cavity.

Experimental set-up

A summary of that information is presented below. Fig.1. shows a schematic representation of the experimental set-up. The test section is stationed inside a large plexiglass housing (42x25x25cm) with two windows on both sides 5x25cm and a window on the top with 10x25cm in dimensions. The test section is made of 5 mm thick pure Teflon (PTFE) (k =0.25 W/m K) in which the heat sources extended by aluminum foam block are embedded. This heated section was insulated with 20 mm Glasswool (Izopan) and 50 mm Styrofoam (k=0.028 W/m K). Distribution of aluminum-foam blocks in the test section is shown in Fig. 1 with all the necessary dimensions. Aluminum-foam blocks were arranged 3x3 in the test section. The block dimensions were 25x25x20mm. The heater sections of the teflon substrate were cut using an industrial laser.

Here, 25x25x20mm aluminum-foam blocks placed on 25x25mm copper plates (k =386 W/m K) were used and tightly fit to the teflon substrate. The area of resistance wire heating elements was the same as the copper plate. The heating elements were electrically insulated and the resulting assembly was screwed to the copper plates using a heat sink compound providing the least possible contact resistance. The thermocouples on the heated section were inserted through holes drilled in the insulation, and were pushed into drilled cavities placed inside the copper plates and soldered for rigidity. All thermocouples were separately calibrated. Signals from the thermocouples were collected, processed, stored and analyzed with a data acquisition system. It was observed that experimental conditions reach a steady-state condition after approximately 5 hours. After conditions had been steady for some time and differences in temperatures between two intervals became negligible ($\Delta T < 0.1\ ^0C$), all temperatures were collected, averaged and stored.

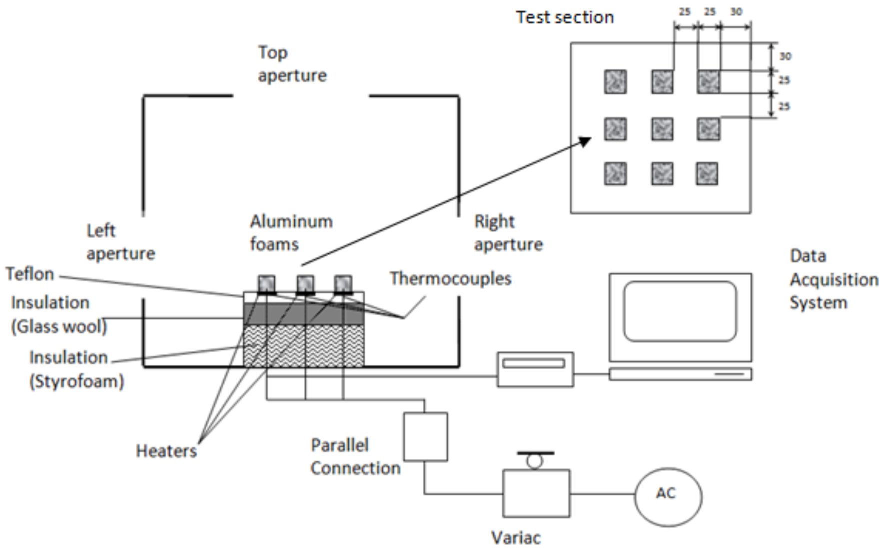

Figure 1: Schematic diagram of the experimental set-up, heat sinks layout and assembly (the measures are in mm).

The porous materials that situated in the test section are aluminum-foams having pore density of 10PPI. The aluminum foam block is shown by scaled photographs in Fig. 2, and also its particulars are displayed in Table 1. The parameters to specify the aluminum-foam block are the porosity, i.e., the volumetric fraction of void inside aluminum-foam, the pore density in pores per inch (PPI) and the permeability, i.e., the flow conductance (Kim et al. 2000). The permeability was determined by a direct measurement of pressure drop through the specimen installed in an air wind channel without air bypass.

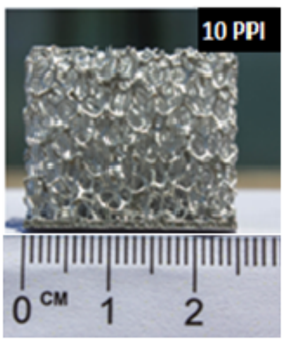

Table 1: Particulars of the Aluminum-foam Block.

Specimen	Porosity (ε)	Pore density (PPI)	Permeability(m^2)
(Al-6101)	0.912	10	$7.73\text{x}10^{-8}$

Figure 2: Pictures of 10 PPI aluminum-foam samples

Processing of the experimental data

In the analysis of heat transfer performance of the aluminum foams, averaged Nusselt number was defined as,

$$Nu_{avg} = \frac{Q_c L}{A_s (T_{savg} - T_0) k_{air}} \tag{1}$$

Here T_0 is the ambient temperature and all thermophysical properties were evaluated at this temperature. T_{savg} is the averaged surface temperature calculated from the thermocouple measurements. A_s is the total heater area. Q_c is the corrected heat dissipation rate, which represents the heat transferred to the fluid directly by convection from the aluminum foam blocks. Q_c was calculated from an energy balance as given below:

$$Q_c = Q_{tot} - Q_{cond} \tag{2}$$

Conduction losses (Q_{cond}) calculated from one dimensional Fourier's law.

In these calculations, first the total heat addition from the resistance heaters was calculated from:

$$Q_{tot} = 9\frac{V^2}{R} \tag{3}$$

where V is the voltage drop across the heater and R is the resistance of the heater. From this value Q_c was calculated by subtracting losses due to conduction.

Other dimensionless number affecting the heat transfer is the modified Grashof number:

$$Gr^* = \frac{g\beta q_c L^4}{k_{air} \nu_{air}^2} \tag{4}$$

where, q_c is the average convection heat flux for all the heaters.

In order to determine the reliability of the experimental results an uncertainty analysis was conducted on all measured quantities as well as the quantities calculated from the measurement results. Uncertainties were estimated according to the standard procedures reported in the literature. Overall, the uncertainty in the Nusselt number is around ±4.9% and for the Grashof number it is around ±4.1%, which is primarily due to uncertainties in the convective heat flux values.

Results and Discussion

Fig. 3 shows the relation between the average temperature distribution and Grashof numbers for with 10 PPI and without aluminum foam blocks (i.e. smooth surface) in a cavity. The average temperatures are considered in the calculations since the aluminum-foam block temperatures have almost the same values after the system has reached the steady state conditions. As can be seen from the figure, the averaged temperature monotonically increases with increasing Grashof numbers. It is obvious that for different Grashof numbers, average surface temperatures for no aluminum foam blocks is much higher than that of 10 PPI aluminum foam cases. Namely, the foam surfaces can be operated in higher heat flux range. The advantages of large surface-area-volume ratio and intense mixing of flow as a result of aluminum foam resulted in a much lower surface temperature distribution.

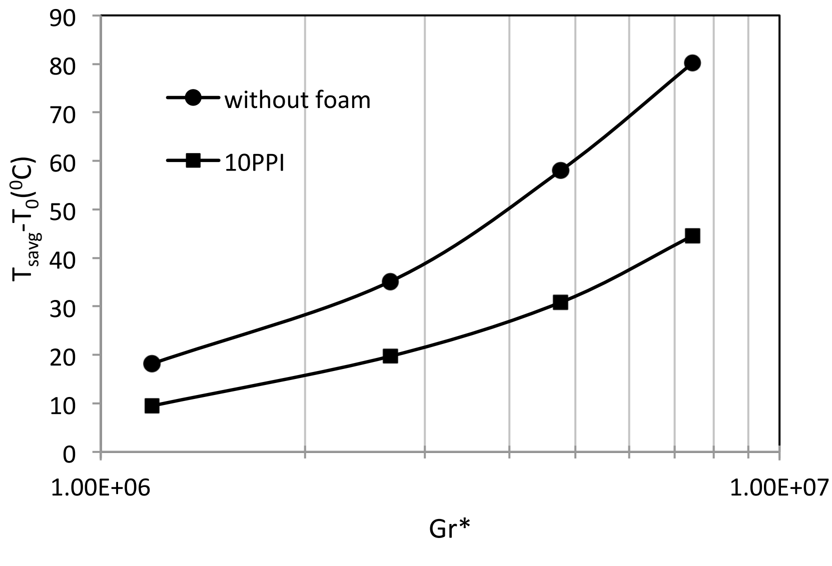

Figure 3: Effect of Grashof numbers on the average surface temperature for with 10 PPI and without foam blocks

Fig. 4 shows the average Nusselt number versus Grashof number for with 10 PPI and without aluminum foam blocks in a cavity. The average Nusselt number Nu_{avg} is obtained by averaging the separate-averaging the block Nusselt number Nu_b over the heater surface area. As anticipated, Nu_{avg} values substantially increase with the increase of Grashof number both with and without foam case. Although form drag and viscous drag tend to suppress the natural convection and weakened the heat transfer, the heat transfer is finally enhanced by aluminum-foams due to significantly extended heat transfer surface area. The 10 PPI aluminum foam blocks offered the most superior heat transfer performance and its average Nusselt number was approximately 2 times higher than that of the without foam blocks or smooth surfaces in the range of $1.19x10^6 \leq Gr^* \leq 7.45x10^6$.

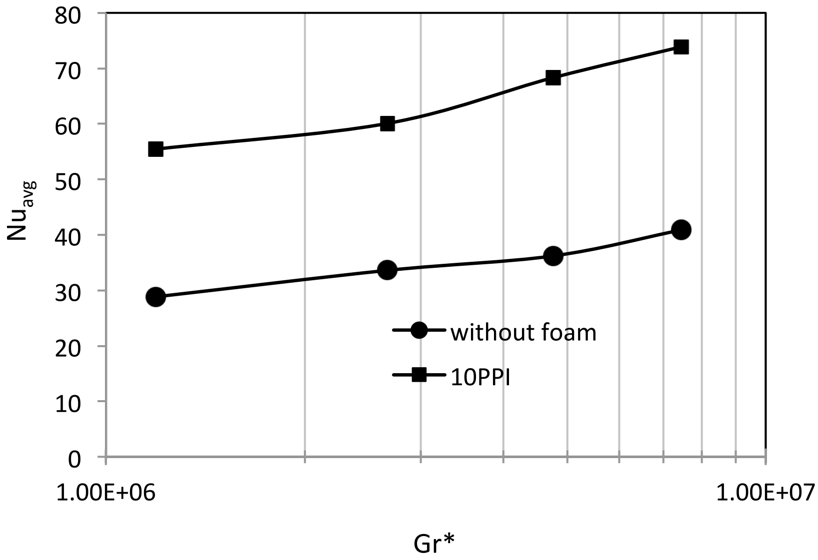

Figure 4: Effect of Grashof numbers on the average Nusselt number for with 10 PPI and without foam blocks

Conclusions

The experiments were carried out to investigate the thermal performance of aluminum-foam blocks for electronics cooling. The obtained results showed that the foam blocks with the pore density of 10 PPI showed about 86% higher average thermal performance that that of the smooth surfaces in the range of $1.19x10^6 \leq Gr^* \leq 7.45x10^6$.in a cavity.

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