# Numerical Investigation of Heat Transfer of Aluminum Metal Foam Subjected to Pulsating Flow

A. Bayomy and M. Z. Saghir
Ryerson University, 350 Victoria St, Toronto

**Abstract**: The rapid development of electronic devises leads to more demand for efficient cooling techniques. Porous media represent a convincing passive cooling enhancer due to its large contact surface area to volume ratio and intense mixing of fluid flow. In the present work, numerical studies have been investigated to study the heat transfer characteristics of aluminum porous metal foam subjected to both steady and oscillating flow using COMSOL software. Time average local surface temperature distributions, local Nusselt numbers and Temperature contours were illustrated. A good agreement between experimental and numerical technique has been achieved.

**Keywords:** Oscillating flow, Porous media, Pulsation flow, Heat sink, Electronics cooling

## 1. Introduction:

The development in the electronic devices for high-speed computers has led to achieve new and reliable methods of electronics cooling. Different kinds of heat sinks have been used in various industries at conventional scale; however, the studies and applications on the micro-scale heat transfer devices are still limited. As reported by Mahalingam and Berg [1] and Gochman et al. [2] the heat flux from desktop and mobile processors are 100 W and 30 W, respectively and the averaged dissipating heat flux can be up to 25 W/cm$^2$ for high-speed electronic components. However, the conventional natural or forced convection cooling methods are only capable of removing small heat flux per unit temperature difference. Therefore, it is imperative to look for new methods of cooling the modern high-speed electronic components.

One of the ways to solve this problem is to augment the convective heat transfer by using extended heat transfer surfaces as well as by increasing the cooling flow rate. To provide the extended heat transfer surface, the parallel-plate heat sinks have been widely used [3]. Thus, works on the forced convection design and the optimization of parallel-plate heat sinks have been extensively conducted up to date [2], [4].In an effort to enhance heat dissipation, recently, some studies have been performed concerning various types of heat sinks such as the strip-fin, the skive-fin, the pin-fin heat sink and so forth [3], [5]–[7].

Among the heat transfer enhancement schemes, one of the promising techniques is using the porous media subjected to steady or pulsating flow. Porous media has emerged as a convincing passive cooling enhancer due to its large contact surface area to volume ratio and intense mixing of fluid flow. Also, the aim of using pulsation flow or oscillating flow is to obtain a uniform temperature profile along electronic component as the reliability of transistors and the operating speed depend on uniform temperature along the surface. The temperature of hot region may effect on the performance level of the electronic component due to gate delay. Therefore, maintaining the uniformity of the temperature under certain limit is a big challenge by using pulsation flow.

Extensive investigations have been conducted on the flow and heat transfer of a channel filled with porous media. The early works related to heat dissipation are those by Cheng et al. [8], Kaviany [9] and Hunt and Tien [10]. Cheng et al. [8] studied the steady flow forced convection in a packed channel with asymmetric heating. Hunt and Tien [10] studied the heat transfer augmentation in a duct filled with foam material subjected to steady flow. More recently, due to increasing applications in electronics cooling, many researchers have studied the heat transfer enhancement of a channel filled with porous media. Tong et al. [11], Huang and Vafai [12], Hadim and Bethancourt [13], and Sozen [14] numerically studied the heat transfer enhancement of a channel or duct filled fully or partially with porous media subjected to steady flow. Fedorov and Viskanta [15] studied the conjugate heat transfer of a porous channel with discrete heat sources numerically. In the above literature, steady flow through the porous channel was investigated. However, as observed by Hwang and Chao [16], the local temperature of the substrate surface is more important than the averaged surface temperature in the application of electronic cooling. A temperature difference of more than 50°C between the most upstream and the most downstream locations of the channel was measured in the case of q=3.2 W/cm$^2$ in their experiments. This indicates that steady flow through a porous channel heat sink still yield a relatively high local surface temperature. Generally, the research that has been done on oscillating flow with porous media is really scarce and incomplete Leong and Jin[17,18] investigated an experimental study of heat transfer characteristics in aluminum metal foam subjected to oscillating air flow. The temperature distributions over the surface, pressure drop across metal foam and the velocity of flow inside the foam were measured. Fu et al. [19] investigated an experimental study to compare between oscillating and steady air flow through the aluminum metal foam channel. The local temperature distribution and local Nusselt number were analyzed.

In the present work, numerical studies have been investigated to study the heat transfer characteristics of aluminum porous metal foam subjected to both steady and oscillating flow using COMSOL software. Time average local surface temperature distributions and local Nusselt numbers were illustrated. For electronic cooling applications, time-averaged quantities are more important than instantaneous quantities. As instantaneous higher power dispersion or temperature increasing may not damage the electronics components, but a long period of higher temperature will decrease their performance or even destroy them.

## 2. Governing Equations

Prior to analyzing these cases, it is useful to state the assumptions on which the governing equations are based:

1. Forced convection dominates the thermal field of the channels, i.e., natural convection effects can be ignored.

2. The fluid passing through the channel is Newtonian.

3. The porous medium is homogenous and isotropic.

4. No heat generation occurs inside the porous medium.

Taking in account these assumptions, the system of governing equations of Brinkman equation and energy equation, which describe the fluid flow inside the porous media and heat transfer, respectively.

$$\frac{\rho}{\varepsilon_p}\left(\frac{du}{dt}+(U.\nabla)\frac{u}{\varepsilon_p}\right)=\nabla.\left(-p+\frac{\mu}{\varepsilon_p}(\nabla u+(\nabla u)^{T})-\frac{2\mu}{3\varepsilon_p}(\nabla.u)\right)\text{..(1)}$$

$$\frac{d(\varepsilon_p\rho)}{dt}+\nabla.(\rho u)=Q_{br}\text{.........................................(2)}$$

$$\rho c_p\frac{dT}{dt}+\rho c_p u.\nabla T=\nabla.(K\nabla T)+q\text{..........................(3)}$$

## 3. Model Description and Grid Sensitivity analysis

The schematic diagram of the experimental facility presented by Fu et al. [19] and the model, which has been simulated in COMSOL software, are illustrated in figure 1. The model consists of a channel with dimensions of 50*50*10 mm filled with aluminum metal foam, the properties of the metal foam shown in table 1.

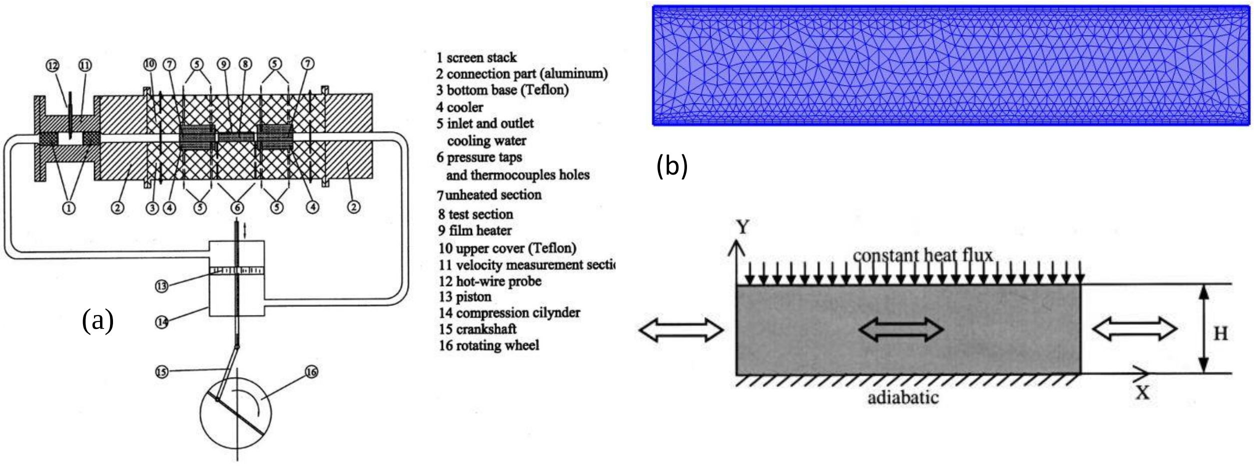


**Figure 1.** (a) Schematic diagram of experimental test section presented by Fu et al. [2], (b) COMSOL model

A film heater mounted on the surface of the top portion of the channel to supply different heat flux. The boundary conditions of the model are heat flux surface on the top portion between 0.8-1.6 W/cm$^2$, inlet velocity on left side in case of steady flow, inlet sinusoidal velocity with frequency ranged 2-8 Hz in case of pulsating flow, outlet velocity on right side, and adiabatic surface on bottom portion as insulation. In order to ascertain at what number of element the solution becomes grid independent. The solution for steady flow through the channel has been performed at different number of domain elements as shown in figure 2. It is noted that the solution reached constant with variation less that 0.1% at number of elements (N) = 10584 elements.

**Table 1.** Aluminum metal foam properties

| Test section material | ERG AL 40 PPI |
|---|---|
| Ligament diameter, $d_l$ [m] | 1.194 *10$^3$ |
| Permeability, K[m$^2$] | 3.36*10$^{-8}$ |
| Solid phase conductivity, k [ W/m.K] | 170 |

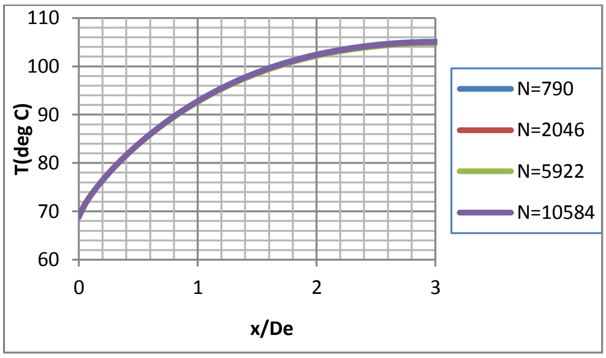


**Figure 2.**Grid sensitivity analysis

## 4. Results:

As mention before, the electronic components may be damaged or caused decreasing in performance such as gates delay. Therefore, the time average local temperature distribution along the surface has been predicted numerically and compared with previous experimental work presented by Fu et al. [2]. As shown in figure 3 the local average temperature distribution along the electronic at heat flux q"= 0.8 W/cm$^2$ and different Reynolds number, where ($Re = \frac{Ud_l}{U_f}$ where U is the Darcy velocity through the test section, $d_l$ is the ligament diameter of metal foam and $U_f$ is the kinematic viscosity of the fluid (air)). It is observed that the temperature increasing with dimensionless axial distant (x/De), where

(De) is hydraulic diameter of the channel (5H/3). This temperature is increasing due to the entry region of the fluid at the beginning of the channel which enhances the heat transfer by mixing.

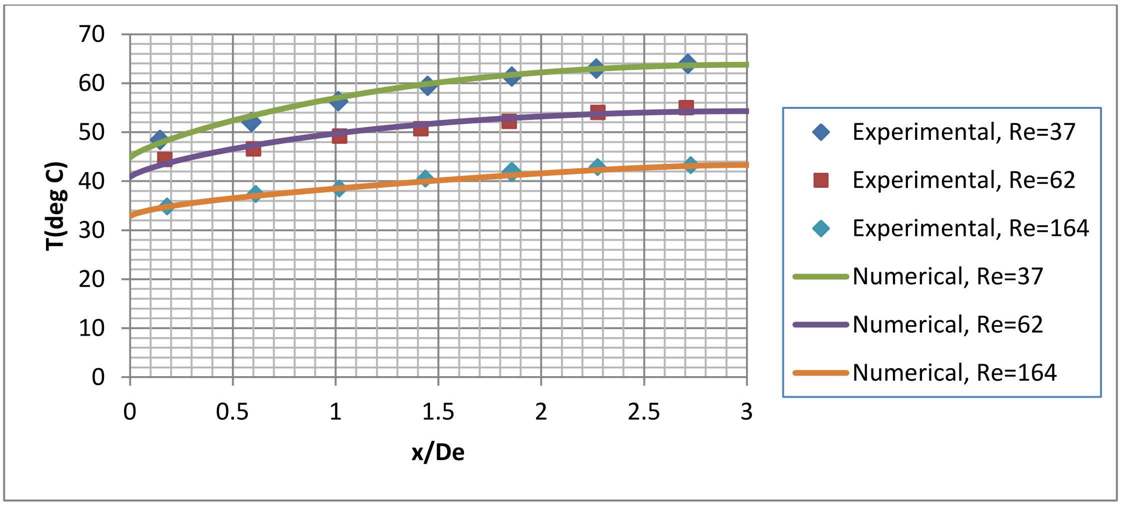


**Figure 3.** Temperature distribution along the surface at q"=0.8 W/cm$^2$

It is found that the temperature difference between most upstream and most downstream point is larger for low Reynolds number, Figure 4(a) shows the local Nusselt number versus dimensionless axial position for aluminum metal foam. The local Nusselt number was calculated by:

$$h_x = \frac{q}{T_w - T_i}$$

$$Nu_x = h_x D_e / k_f$$

Where $h_x$ is local heat transfer coefficient, $D_e$=5H/3 is hydraulic diameter of the channel, H is the height of the channel, $T_w$ is local temperature of the surface and $T_i$ is the inlet bulk temperature.

As shown in figure 4(a), the local Nusselt number decreasing while dimensionless axial position due to the entry region of the fluid at the beginning which increasing the mixing and enhance the heat transfer. Figure 4(b) illustrated the temperature contours at different times at q"=0.8 W/cm$^2$ and Re=37.The analysis of the steady flow through the channel filled with aluminum metal foam illustrated that the difference between most upstream and most downstream point along the surface reached to 40° C at low Reynolds number. Therefore, the pulsation flow through the channel filled with aluminum metal foam has been employed at frequency range (f=2-8 Hz) to achieve more uniform temperature distributions along the surface and enhance the heat transfer. Figure 5 showed the pulsation flow time average local temperatures along the surface. It is observed that the temperature acts as convex curve with maximum temperature at the center of test section, the temperature near both entrances are lower than that at the center due to reversing flow direction; there are two thermal entrance regions in the test section. Figure 6 shows the temperature contours for oscillating flow at different times, q"=0.8 W/cm$^2$ and Re=44. It noticed that the temperatures at both ends are lower than that in center.

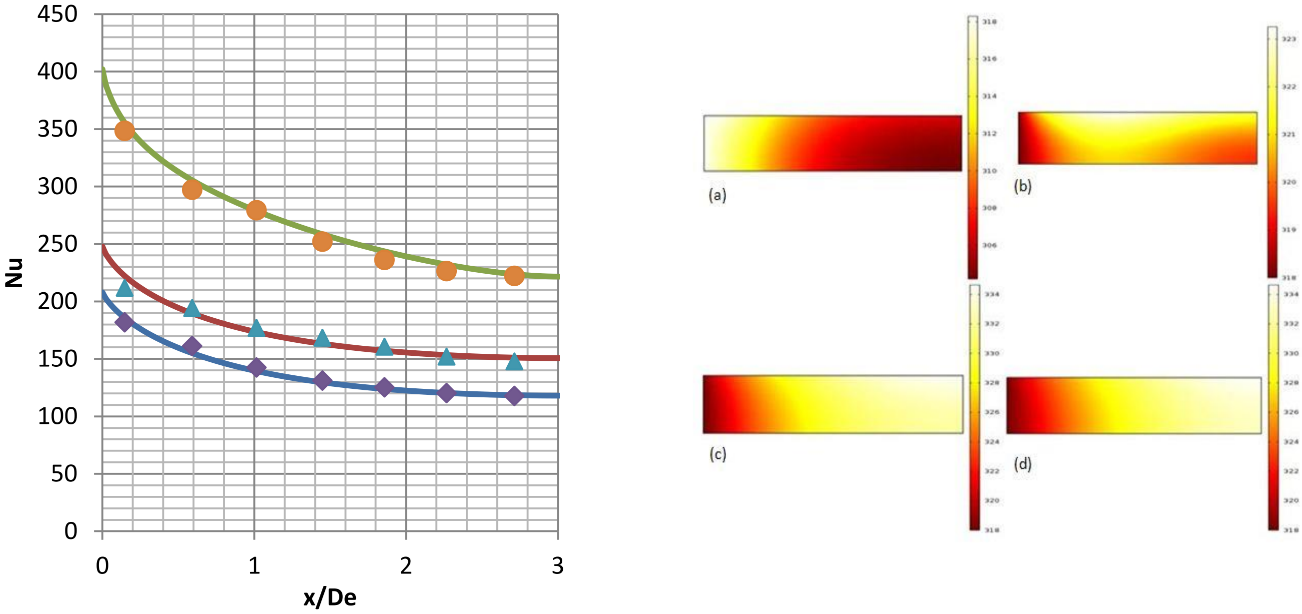


**Figure 4:** (a) Local Nusselt number along the surface at q"= 0.8 W/$cm^2$, (b) Temperature contours at different times at Re=37, q"=0.8

## 5. Conclusions

It is found that numerical results have good agreement with experimental data represented by Fu et al. [19] with a maximum relative errors for steady flow of 2%, and for oscillating flow of 1%.In comparison between steady and oscillating flow, for steady flow the local temperature increasing while dimensionless axial position is increasing until reached constant value when flow approaches the thermally developed region. It is observed that the difference in temperature between maximum and minimum temperature for steady flow is 1.5-3.5 times higher than that for oscillating flow. The surface temperature distribution for oscillating flow is more uniform than that for steady flow. For steady flow, the local Nusselt number decreases along the flow direction and approaches the thermally developed value. The time averaged local Nusselt number for oscillating flow; however, decreases first and then increases after the center point of the test section.

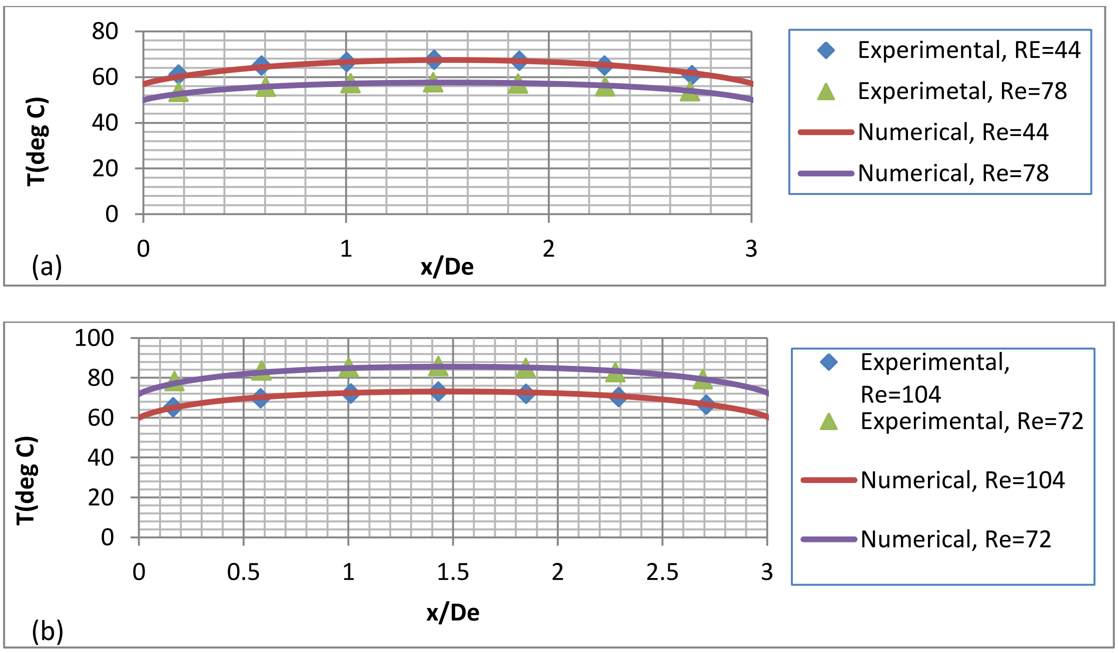


**Figure 5**. The local average temperature distribution (a) At q"=0.8 W/$cm^2$ (b) at q"=1.6 W/$cm^2$

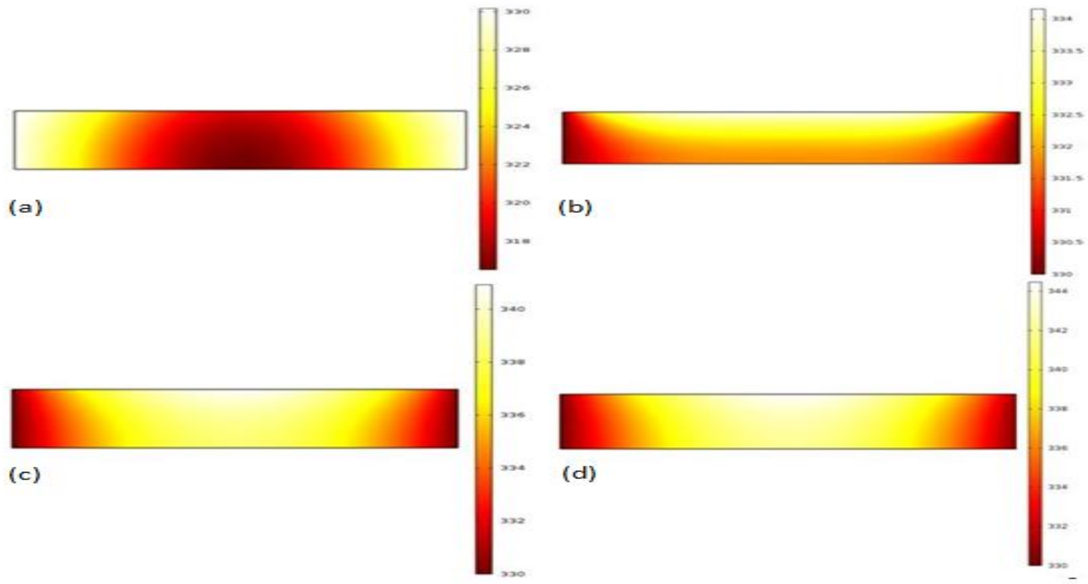

**Figure 6.**Temperature contours at different times at q"=0.8 W/cm$^2$

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