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RESEARCH ARTICLE

3D Numerical Modelling of Convective Heat Transfer through Two-sided Vertical Channel Symmetrically Filled with Metal Foams

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Abstract

A computational fluid dynamics analysis of forced convective heat transfer has been conducted numerically on the hydrodynamic and heat transfer of airflow through vertical channel. The effects of airflow Reynolds number, metal foam porosity and thermal conductivity on the overall Nusselt number, pressure drop, maximum temperature and temperature distribution were considered. The novelty of the present study is the use of metal foams in a two-sided vertical channel and the quantification of the heat transfer enhancement compared to an empty channel for different foam material. Based on the generated results, it is observed that the heat transfer rate from the heated plate is the same for aluminium foam (porosity of 0.948) and copper foam (porosity of 0.877) against equal velocity range and heat flux conditions. Furthermore, it is noted that increasing the airflow velocity reduces the maximum temperature; however, the decrement is not linear. Results obtained from the proposed model were successfully compared with experimental data found in the literature for rectangular metal foam heat exchangers.

Keywords

heat transfer enhancement, numerical modelling, metal foam, forced convection, porous medium, Computational Fluid Dynamic (CFD)

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1 Introduction

Recently, there has been an increasing focus on new class of materials with low densities and novel physical, mechanical, thermal, electrical and acoustic properties. Metal foams, which are a class of cellular materials improve efficiency and minimize the required weight and volume in energy producing or transferring systems. Due to their forms that have great strength to weight ratio, they are used in different engineering applications ranging from mechanical to thermal [1-3].

Considering the possible implementation of air and metal foam in engineering systems, in the last decade, airflow through porous medium has been largely investigated by several authors; Bastawros [4], Dai et al. [5], Dukhan and Chen [6], Calmidi and Mahajan [7], Hwang et al. [8], Hsieh et al. [9], Kim et al. [10], Boomsma et al. [11], Poulikakos and Kazmierczak [12], Noh et al. [13], Kurtbas and Celik [14], Cavallini et al. [15], Mancin et al. [16-21], Tamayol and Hooman [22].

Bastawros [4] provided experimental measurements and modelling of the heat transfer in porous metals subject to transverse airflow. Dai et al. [5] compared two configurations; a metal foam and a louver fins as extended surfaces for air cross flow heat exchangers under the same thermal and hydraulic conditions. Dukhan and Chen [6] numerically and experimentally studied heat transfer of air flowing through a block of aluminium foam, which is subjected to a constant heat flux on one side. He neglected conduction within the fluid and applied the local thermal equilibrium assumption for solid and fluid phases.

Calmidi and Mahajan [7] experimentally and numerically studied forced convection in high-porosity metal foam over a range of porosities and pore densities using air as the working fluid. In a similar study, Hwang et al. [8] showed that the volumetric heat transfer coefficient increased with decreasing foam density at a given Reynolds number.

Boomsma et al. [11] studied forced convective flow in various compressed aluminium foams. They showed that these performed well, not only in the heat transfer enhancement, but they also made a significant improvement in the efficiency compared to conventional compact heat exchangers. Poulikakos and Kazmierczak [12] studied forced convection in a duct

partially filled with a porous material. Mancin et al. [16-19] experimentally measured the heat transfer coefficient and pressure drop values during air forced convection through a horizontal channel. They found that foam thickness has no significant effect on the pressure drop. Moreover, Mancin et al. [20] investigated the heat transfer and fluid flow behaviour of five different 20 mm high copper foams. They also presented [21] the heat transfer and fluid flow measurements during airflow through twenty-one different aluminium and copper foams. Two correlations for the heat transfer coefficient and pressure drop calculations have been developed and validated. Tamayol and Hooman [22] numerically analysed forced convection heat transfer through metal foam heat exchangersa using a thermal resistances approach and estimated the heat transfer performance of metal foam heat exchangers.

Despite all the previously listed metal foam modelling and the high number of publications dealing with numerical and experimental analysis of metal foams, there are still some unaddressed issues of metal foam applications. The modelling investigations on metal foams as heat exchanger were conducted mainly for the unidirectional flow condition whilst published modelling studies on metal foams subjected to multichannel flow are rather scarce. The effect of Reynolds number, metal foam porosity and thermal conductivity on the heat transfer and pressure drop have not been adequately investigated.

In this work, a numerical model is established for a two-sided channel filled with metal foam. The proposed model first is presented and then is validated by comparing the velocity, pressure drop, temperature distribution and overall Nusselt number with an experimental study for rectangular metal foam heat exchangers under constant heat flux. Finally, the effect of airflow Reynolds number, metal foam porosity and thermal conductivity on the overall Nusselt number, pressure drop, maximum temperature and temperature distribution were considered to investigate their influences on the heat transfer through the porous channel compared to empty channel. In this work, more than 400% increase in the Nusselt number has been observed in some cases at high Reynolds numbers.

2 Numerical method

2.1 Description of heat transfer system

The schematic diagram of the heat transfer system is shown in Fig. 1. The channel is well insulated and air is entered from bottom and is withdrawn from the top of channel. The channel is divided into two part by a flat heater with constant power output 100W. A 3 mm thick aluminum plate was inserted on either side of plate heater as can be seen in Fig. 1.

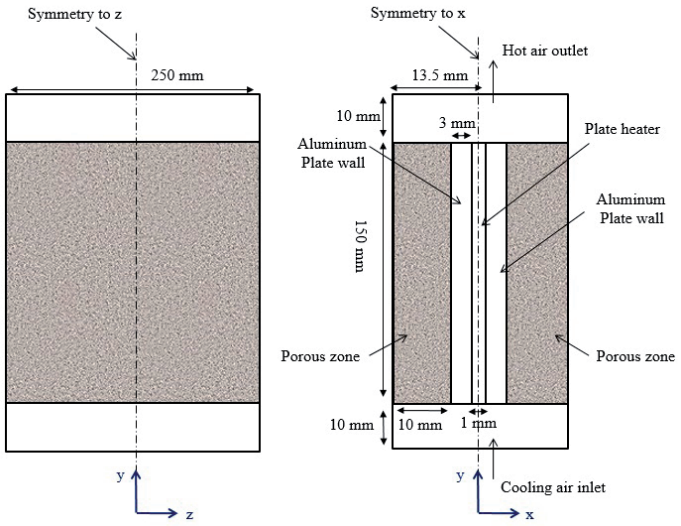


Fig. 1 Schematic diagram and coordinate system of the problem

The channel width 10 mm and height 150 mm on each side is filled with the metal foam block of dimensions 250(W) ×150(L) ×10(H) mm. In this work, we denote this section of heating system as test section. The metal foams used in the numerical study are aluminium and copper with 10 PPI and 10 mm thickness (Table 1).

Table 1 Aluminium and copper foams characteristics

Sample	W×L×H/2	ε	$K_p \times 10^7$	C_f	k_s
Aluminium 10 PPI	250×150×10	0.948	2.480	94.98	165
Copper 10 PPI	250×150×10	0.877	1.742	176.75	380

Studies have been done in different velocity of air. The inlet velocity is considered to be fully distributed in the transverse direction. In each case, the parabolic velocity profile is as follows:

$$V = V_{\max}\left(1-\left(\frac{x}{0.0135}\right)^2-\left(\frac{z}{0.125}\right)^2\right) \quad (1)$$

where $V_{\max}$ is in range of 0.4 – 3 m/s. This velocity range usually encountered in electronic cooling such as tall printed circuit boards and for large-scale applications like data centers. It should be noticed that the following assumptions are considered in our modelling:

- The fluid flow and heat transfer conditions are fully developed.
- The thermo physical properties of the fluid are constant.
- No fluid crosses the solid material surface.
- Thermal radiation effect is neglected.
- Heat conduction through the fluid is neglected.
- The effective properties of the porous medium are constant.
- Adiabatic wall condition is applied (heat loss is neglected).
- The porous medium is homogeneous and isotropic saturated.
- Local thermal equilibrium (LTE) condition is established.

2.2 Governing equations

The numerical modelling was performed in the computational domain (Fig. 2). The incompressible, laminar, steady state fluid flow and heat transfer in channel without metal foam (Region I and region III) is described by classical Navier–Stokes equations, together with the continuity and corresponding energy equation. These equations are as follows:

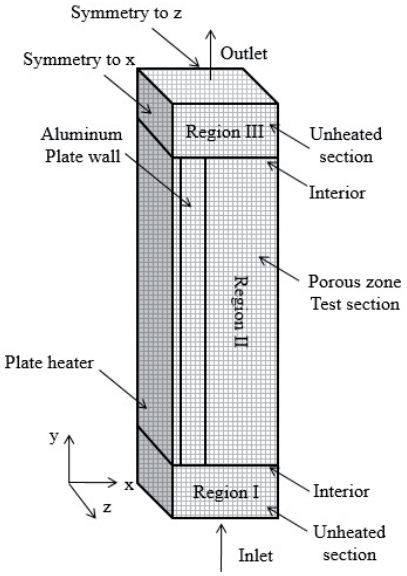


Fig. 2 Computational domain

Continuity equation

$$\nabla \cdot V = 0 \tag{2}$$

Momentum equations

$$\rho_f \left(V \cdot \nabla V\right) = -\nabla p + \mu_f \nabla^2 V \tag{3}$$

Energy equation

$$\rho_f C_{p\cdot f} \left(V \cdot \nabla T\right) = k_f \nabla^2 T \tag{4}$$

where $V = (u_x, u_y, u_z)$ is the flow velocity vector. They are also used for region II when the empty channel was modelled. In filled channel, the flow in porous medium (Region II) is governed by combination of continuity and momentum equations, which together form Darcy-Brinkman-Forchheimer model [23-25]. These equations can be written as follows:

The continuity Eq. (2) and Momentum equations

$$\frac{\rho_f}{\varepsilon}\nabla\left(\frac{V\cdot V}{\varepsilon}\right) = -\nabla p + \frac{\mu_f}{\varepsilon\rho_f}\nabla^2 V - \frac{\mu_f}{K_p}V - \frac{c_f\rho_f}{\sqrt{K_p}}|V|V \tag{5}$$

where μ_f is the dynamic viscosity of the fluid, V is the Darcy velocity vector, ρ_f is the fluid density, P is the pressure, ε is the porosity, K_p is the permeability of porous medium and C_f is the dimensionless form-drag coefficient. The apparent viscosity, μ in the Brinkman porous medium of the momentum equations should take a different value from fluid viscosity. Based on Lundgren experiment, the effective viscosity of saturated porous medium is equal to the viscosity of the fluid [26].

Energy equation

The heat transfer in porous medium was modelled using the following version of energy equation as the mathematical model for heat transfer in porous medium

$$\left(\rho_f C_{p,f}\right)_{eff}\left(V\cdot\nabla T\right) = k_{eff}\nabla^2 T \tag{6}$$

where ρ_f is the fluid density, $C_{p,f}$ is the fluid heat capacity at constant pressure, $(\rho_f, C_{p,f})_{eff}$ is the effective volumetric heat capacity at constant pressure, k_{eff} is the effective thermal conductivity and v is the fluid velocity field. The velocity field should be interpreted as the Darcy velocity vector, that is, the volume flow rate per unit cross-section area.

The average linear velocity, that is the velocity within pores, V_f, can be calculated as $V = \varepsilon V_f$ where ε is the porosity or fluid's volume fraction.

The effective conductivity of porous medium, k_{eff}, is volume average of fluid (k_f) and solid (k_s) medium conductivities:

$$k_{eff} = \varepsilon k_f + \left(1-\varepsilon\right)k_s \tag{7}$$

where indices f and s refer to fluid and solid parts of porous medium, respectively. The equivalent volumetric heat capacity is calculated by:

$$\left(\rho C_p\right)_{eff} = \varepsilon\left(\rho_f C_{p,f}\right) + \left(1-\varepsilon\right)\left(\rho_s C_{p,s}\right) \tag{8}$$

In heat transfer calculations, the average surface temperature of the aluminium plate is used. It is defined based on temperature excess, i.e. the difference between wall temperature and inlet ambient temperature of air, as follows:

$$\Delta T_{ave} = \left(1/n\right)\sum_{i=1}^{n} T_w - T_\infty \tag{9}$$

Where n is the number of data points. The overall heat transfer rate is calculated from average heat transfer coefficient defined in Eq. (10) for entire porous metal foam in the test section.

$$h_{ave} = \frac{Q - Q_{loss}}{2A_{heated}\Delta T} \xrightarrow{Q_{loss}=0} h_{ave} = \frac{Q}{2A_{heated}\Delta T} \tag{10}$$

where Q is the power input, A_{heated} is the heated area of aluminium plate (base area) and the factor '2' is to take into account the heat transfer that takes place from both sides of the aluminium plate.

$$N_u = \frac{h_{ave}H}{k_f} \tag{11}$$

where H is the foam thickness in the test section.

In this work, the model neglects conduction in the fluid and invokes the local thermal equilibrium (LTE) assumption for the solid and fluid phases in the foam. In the absence of LTE, the single energy equation needs to be replaced with two energy

equations, one for the solid and another for the fluid. Later Amiri and Vafai [27] investigated the validity of local thermal equilibrium (LTE) conditions for steady state, incompressible flow through a porous medium. In addition, in Lee and Vafai [28], a theoretical investigation of forced convective flow through a channel filled with a porous medium was presented. According to Minkowycz et al. [29], for most practical applications, the LTE is satisfied when Sparrow number (Sp) > 100. In this work, a Sp > 100 is obtained for all numerical experiments, which satisfies the assumption of LTE.

2.3 Computational domain and boundary conditions

The channel is symmetrical in both x and z directions. Therefore, a quarter of that was selected as computational domain. It is divided to sub domain consists of test section (Region II), plate heater, aluminium plate and a pair of unheated sections located at upstream and downstream (Region I and region III) as shown in Fig. 2. Uniform heat flux is applied to the heater. The air is entered to region I with constant temperature (T_{in}=303K) and a parabolic velocity profile (V_{in}), then it passes through the porous zone along the plate heater and warms up. The static outlet pressure is fixed, and the remaining flow variables are extrapolated from the inlet of computational domain. The appropriate boundary conditions are given in Table 2 and Table 3.

Table 2 Momentum boundary conditions for computational domain

Boundary condition		$V(u_x, u_y, u_z)$
Inlet (*mm*)	y = 0.0 0.0 ≤ x ≤ 13.5 0.0 ≤ z ≤ 125	$u_y = V_{in}$
Aluminium plate wall (*mm*)	10 ≤ y ≤ 160 x = 3.5 0.0 ≤ z ≤ 125	0 (No slip)
Heater wall (*mm*)	10 ≤ y ≤ 160 x = 0.5 0.0 ≤ z ≤ 125	0 (No slip)
Heater + Aluminium plate bottom wall	y = 10 0.0 ≤ x ≤ 3.5 0.0 ≤ z ≤ 125	0 (No slip)
Heater + Al plate top wall	y = 160 0.0 ≤ x ≤ 3.5 0.0 ≤ z ≤ 125	0 (No slip)
Symmetry to x (*mm*)	0.0 ≤ y ≤ 170 x = 0.0 0.0 ≤ z ≤ 125	$u_x = 0$ $\frac{\partial u_y}{\partial x} = \frac{\partial u_z}{\partial x} = 0$
Symmetry to z (*mm*)	0.0 ≤ y ≤ 170 0.0 ≤ x ≤ 13.5 z = 0	$u_z = 0$ $\frac{\partial u_x}{\partial z} = \frac{\partial u_y}{\partial z} = 0$
Outlet	y = 170 0.0 ≤ x ≤ 13.5 0.0 ≤ z ≤ 125	$\frac{\partial V}{\partial y} = 0, p = 0$

Table 3 Energy boundary conditions for the computational domain

Boundary condition		T
Inlet (*mm*)	y = 0.0 0.0 ≤ x ≤ 13.5 0.0 ≤ z ≤ 125	T_{in}
Aluminium plate wall (*mm*)	10 ≤ y ≤ 160 x = 3.5 0.0 ≤ z ≤ 125	Q = 100 (W)
Heater wall (*mm*)	10 ≤ y ≤ 160 x = 0.5 0.0 ≤ z ≤ 125	Q = 100 (W)
Symmetry to x (*mm*)	0.0 ≤ y ≤ 170 x = 0.0 0.0 ≤ z ≤ 125	$\frac{\partial T}{\partial x} = 0$
Symmetry to z (*mm*)	0.0 ≤ y ≤ 170 0.0 ≤ x ≤ 13.5 z = 0	$\frac{\partial T}{\partial z} = 0$
Outlet	y = 170 0.0 ≤ x ≤ 13.5 0.0 ≤ z ≤ 125	$\frac{\partial T}{\partial y} = 0$
Other walls		Adiabatic

2.4 Numerical Solution Procedure

In the present work, the 3D channel has been modelled by CFD code. Equations (2) − (4) and Eqs. (2), (5) − (7) are solved separately using a finite volume code based on collocated grid system to discretize the governing equations. Figure 2 shows the grids in the studied section. The grid consists of hexahedral cells. A denser grid was used at the boundaries (solid-solid and solid-fluid), in order to achieve greater numerical accuracy. The user defined functions (UDF) are linked to model equations in order to apply the inlet parabolic velocity profile. Based on the finite volume approach, the SIMPLE algorithm [30] is employed to deal with the problem of pressure-velocity coupling and the second-order upwind calculation scheme is used to determinate momentum and energy balances.

For each Reynolds number and each porous medium, the calculations were performed with three different meshes; the sizes of grids were 99,280, 103,594, and 106,643. The results of Nusselt number and pressure drop were compared for all three meshes. It was found that a grid with 106,643 cells is sufficient to accurately predict the basic characteristics of flow and heat transfer with and without metal foam inserts. The relative error between two successive iterations was specified by using a convergence criterion of 10^{-5} for each scaled residual component. To ensure the validity of numerical analysis, the numerical code was validated against the experimental results of Kamath et al. [31] for a vertical wind tunnel, containing symmetrically heated metal foams.

3 Results and Discussion

The study was conducted to discuss fluid flow and heat transfer in a channel with and without of metal foams. In Section 3.1, the hydrodynamic analysis and in Section 3.2, the heat transfer analysis have been explained.

3.1 Hydrodynamic analysis

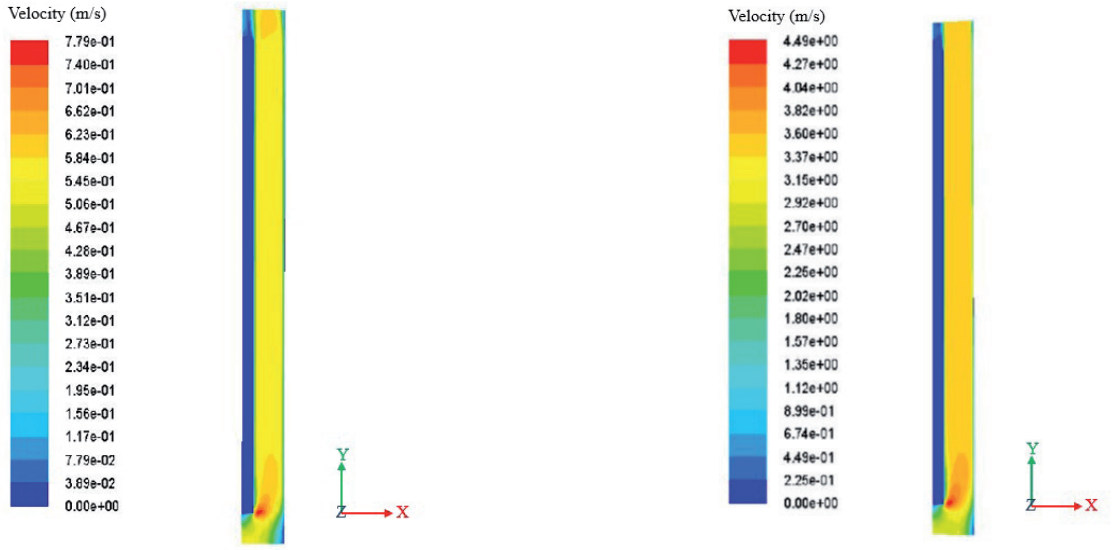
Hydrodynamic modelling was conducted with different inlet velocity and without energy calculations.

3.1.1 Velocity field analysis

Figure 3 (a) and 3 (b) show velocity distribution through aluminium foam for two maximum velocities (0.54 and 2.9 m/s). More mixing flow corresponding to high velocity is seen at the inlet of the porous zone. By decreasing the cross-sectional area of channel at the inlet of test section, the maximum velocity of the fluid increases significantly and shifts towards aluminium plate, which can be called velocity field rebuilding.

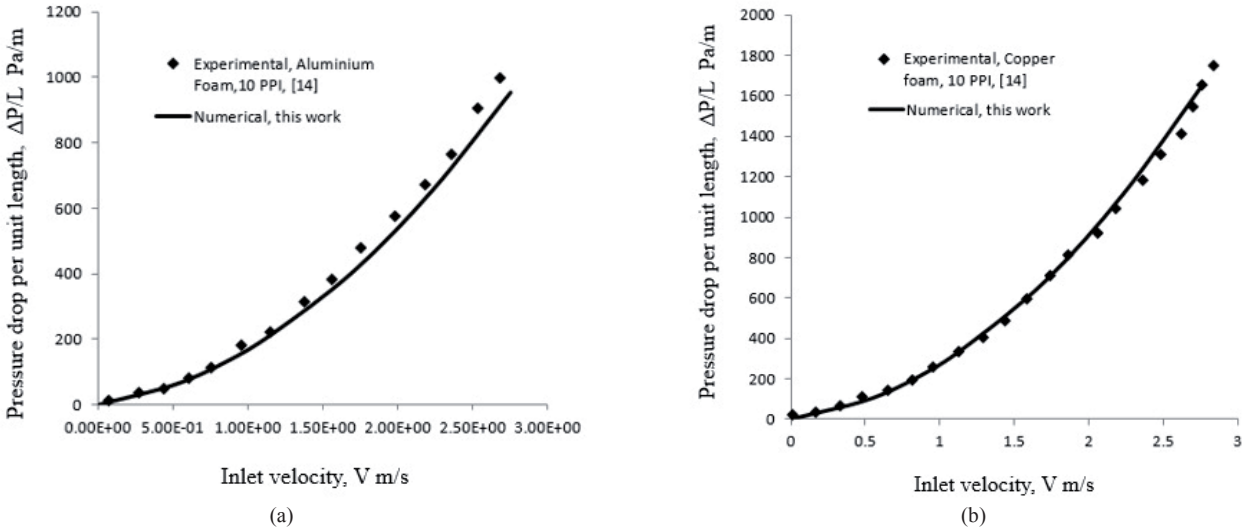
3.1.2 Pressure drop analysis

The pressure drop results are compared in Fig. 4 (a) and 4 (b) for aluminium and copper foams with different airflow velocity. There is a qualitative agreement between the model and the experimental data. The greater airflow velocity made more variation in the local pressure drop, due to $\Delta P = f(V2)$. It can be seen that local pressure drop for aluminium foam is lower than copper foam. It is due to lower form drag to viscous drag ratio of high porosity metal foam than low porosity metal foam.



(a) (b)

Fig. 3 Velocity distribution for aluminium foam, (a) V_{max}= 0.54 m/s (b) V_{max}= 2.9 m/s



(a) (b)

Fig. 4 Model validation: (a) variation of pressure drop in aluminium foam with 10 PPI versus velocity (b) variation of pressure drop in copper foam with 10 PPI versus velocity

3.2 Heat transfer analysis

3.2.1 Effect of velocity on temperature distribution through empty and metal foam filled channels

Figure 5 (a) − (c) presents temperature distribution of an empty channel along axial direction versus different inlet velocity. As the airflow velocity increases, the thermal boundary layer thickness on the aluminum plate decreases and the heat transfer from aluminum plate increases. However, the temperature distribution was decreased in the fluid region due to high velocity. It shows that, a large amount of air leaves channel without temperature change.

Figures 6 (a) − (f) present temperature distribution of filled channel along axial direction versus different airflow velocity for both aluminium and copper foams. This is obvious that heat transfer surface in the filled channel is much higher than empty channel. This is one of the advantages of employing metal foams that can provide a large surface area to volume ratio and it can intense mixing of flow through metal foam in the heat transfer systems. By using of metal foams, the surface temperature of the heated plate is significantly lower than that in the empty channel and temperature distribution gets smoother.

3.2.2 Effect of thermal conductivity of metal foams and Reynolds number on maximum temperature

The heat removal mechanism from the heater plate is conduction through the foam ligaments and convection through the foam. With increasing the airflow velocity, convective heat transfer rate increases and aluminium plate temperature decreases. Maximum temperature decreases with increasing airflow velocity and consequently, Reynolds number as shown in Fig. 7. However, the decrement is not linear.

As can be seen, after Re = 3230 (V = 2.5 m/s), temperature change is negligible and further increasing of velocity does not have a significant effect on the heat transfer rate. It is evident that by using of metal foam with low thermal conductivity, the heated plate cannot transfer heat well to metal foam and heat accumulation in the plate leads to increase surface temperature.

3.2.3 Effect of Reynolds number on the overall Nusselt number

The most important parameters affecting the heat transfer from heater are Reynolds number, porosity and permeability of metal foams. It can be observed from Fig. 8 and Fig. 9 that metal foams led to heat transfer enhancement. Overall Nusselt number for filled channel is higher than empty channel for both investigated cases. The maximum heat transfer rate was observed at Re=3970, which was significantly more than empty channel.

The heat transfer performance of aluminium and copper foams is compared in Fig. 10. The higher thermal conductivity of copper foam did not contribute significantly to increase heat transfer compared to aluminium foam. Consequently, copper foam with porosity of 0.877 and aluminium foam with porosity of 0.948 gave the same heat transfer performance for the investigated airflow velocity range and uniform heat flux conditions.

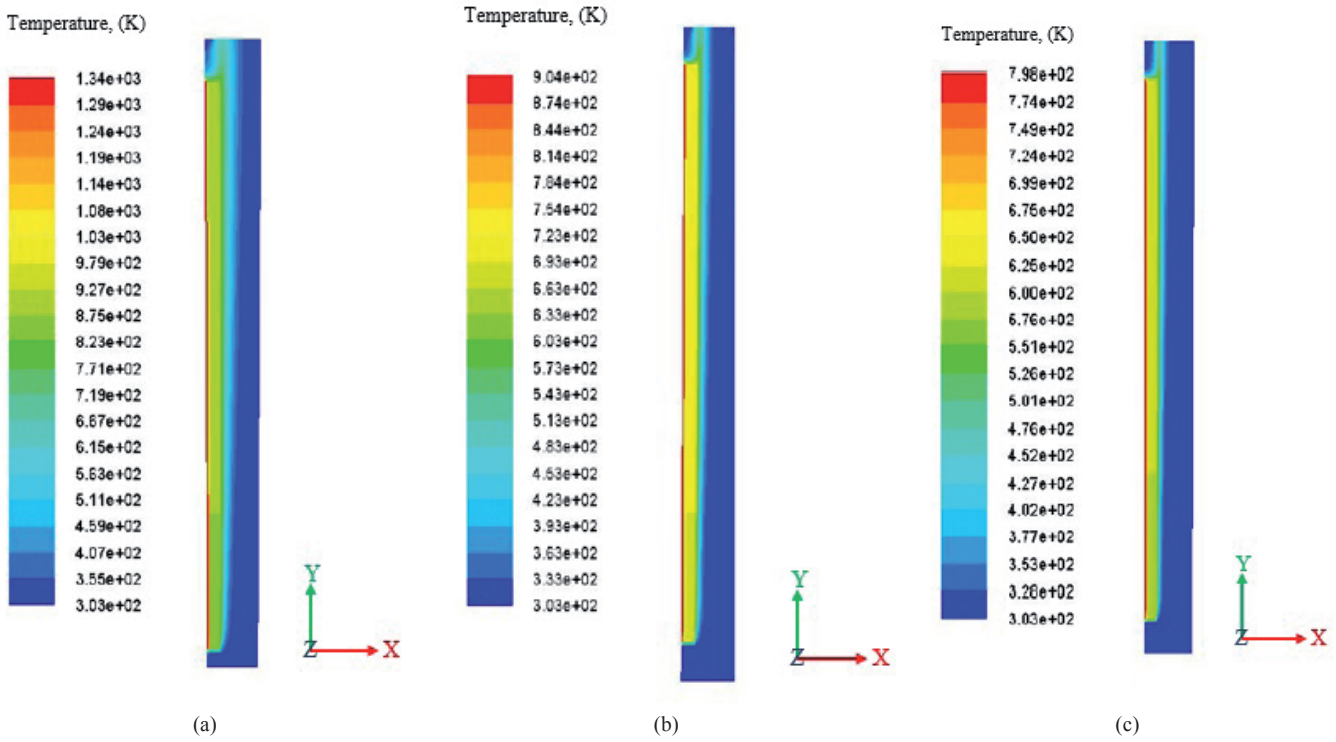


Fig. 5 Temperature distribution along axial direction for empty channel (a) V_{max} = 0.54 *m/s*, (b) V_{max} = 1.24 *m/s*, (c) V_{max} = 2.36 *m/s*

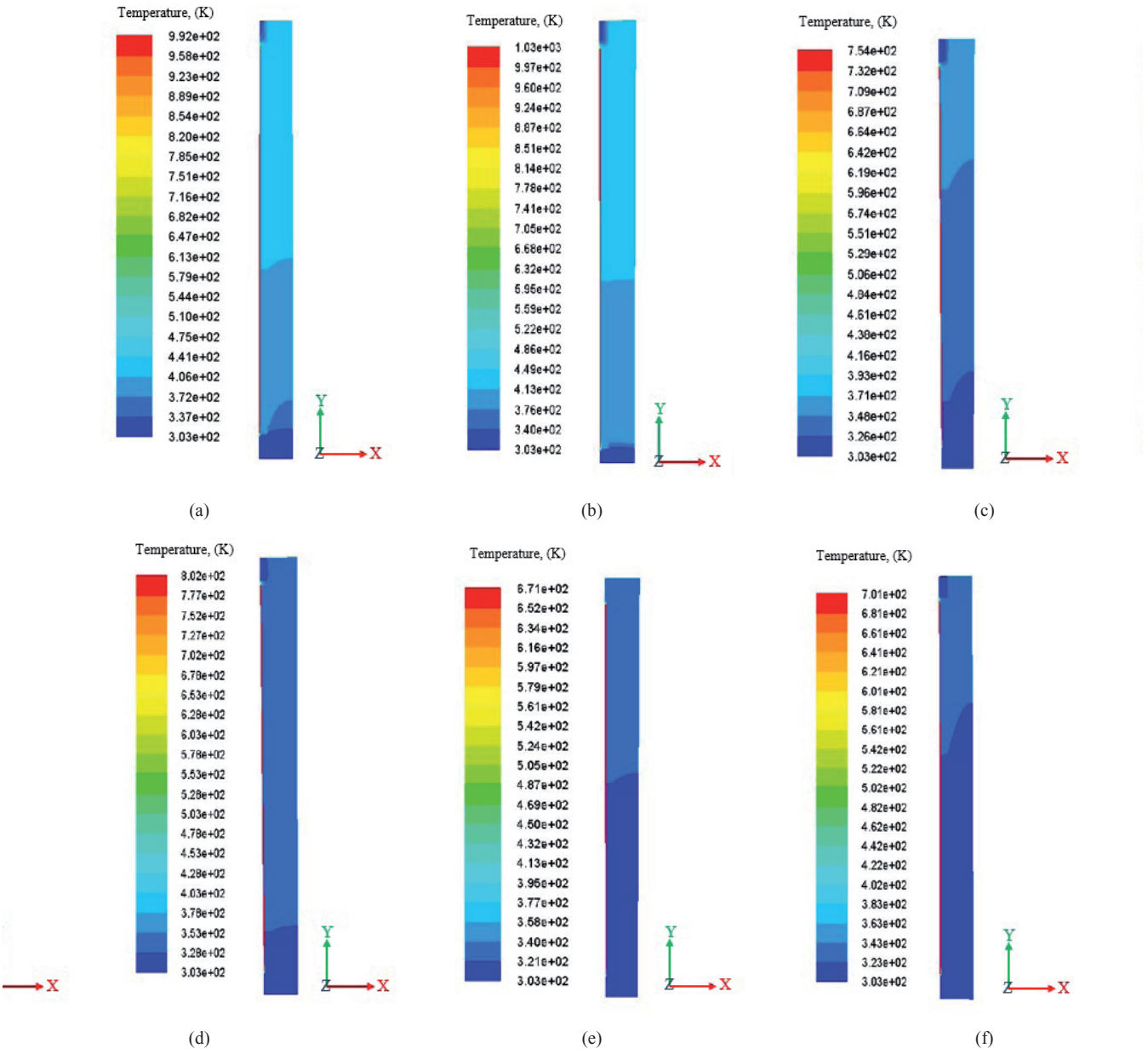


Fig. 6 Temperature distribution of filled channel along axial direction (a) copper foam with V_{max} = 0.54 *m/s*, (b) aluminium foam with V_{max} = 0.54 *m/s*, (c) copper foam with V_{max} = 1.24 *m/s*, (d) aluminium foam with V_{max} = 1.24 *m/s*, (e) copper foam with V_{max} = 2.36 *m/s*, (f) aluminium foam with V_{max} = 2.36 *m/s*

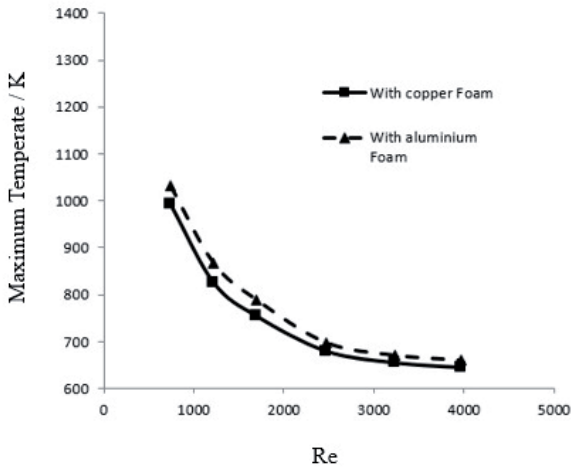


Fig. 7 Maximum temperature as a function of Reynolds number for aluminium and copper foams

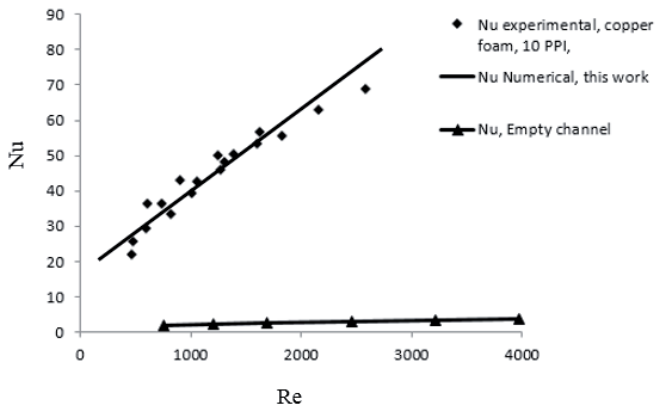


Fig. 8 Overall Nusselt number as a function of Reynolds number for empty channel and channel with copper foam

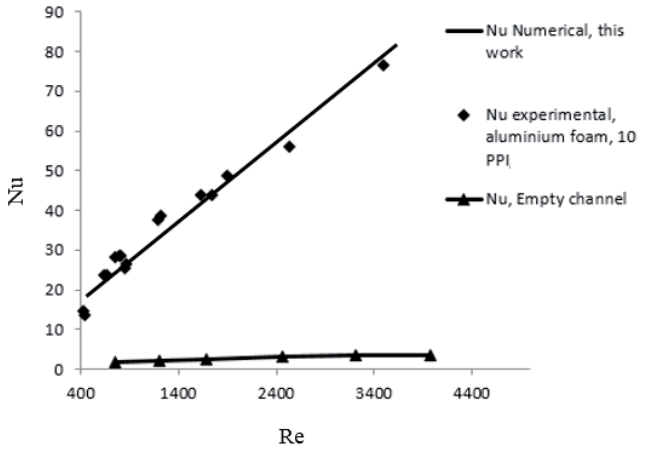


Fig. 9 Overall Nusselt number as a function of Reynolds number for empty channel and channel with aluminium foam

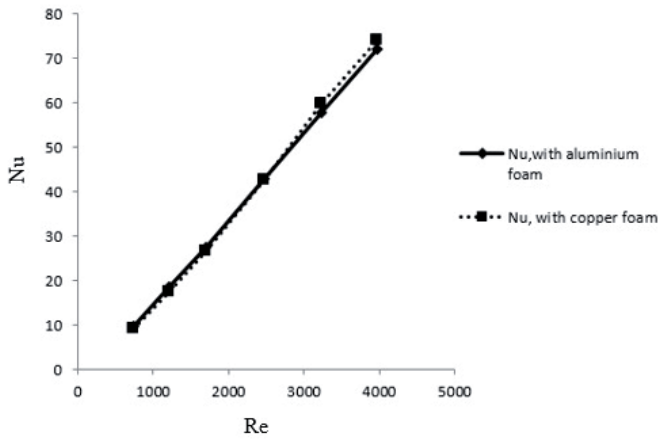


Fig. 10 Heat transfer performance of aluminium and copper foams

4 Conclusions

In the present work, heat transfer enhancement of fully developed laminar flow through a two-sided vertical channel which is filled with aluminium and copper foams is numerically investigated. The channel with and without metal foams were modelled using Darcy-Brinkman-Forchheimer, classical Navier-Stokes equations and corresponding energy equations. A finite volume method was utilized to solve the governing equations. The proposed model were successfully validated with experimental data found in the literature for rectangular metal foam heat exchangers.

The effect of Reynolds number, porosity and thermal conductivity on surface temperature distribution and overall Nusselt number had been analysed. Based on the results, the pressure drop increased by either increasing the airflow velocity or decreasing the metal foam porosity. The heat transfer factors, overall Nusselt number in the channel filled with metal foams are higher than empty channel. Therefore, a significant increase in heat transfer is obtained in two-sided channel filled with metal foams. Maximum temperature decreased with increasing of Reynolds number. It is shown that at high Reynolds numbers, the convection heat transfer has been dominated.

Moreover, the heat transfer enhancement through a metal foam filled channel increases with inlet velocity for studied ranges. Although, copper foam has higher thermal conductivity, but it could not contribute significantly to increase the heat transfer compared to aluminium foam. It is recommended that aluminium foam is better than the more expensive copper foam with similar PPI in this system. It is revealed that the proposed numerical model can efficiently provide useful information for design of multi-channel heat transfer system for a velocity range usually encountered in electronic cooling such as tall printed circuit boards and for large-scale applications like data centers.

Nomenclature

A	Surface area of the aluminium plate [m^2]
C_f	Form-drag coefficient [mm^{-1}]
$C_{p,f}$	Heat capacity of fluid [J kg^{-1} K^{-1}]
$C_{p,s}$	Heat capacity of solid [J kg^{-1} K^{-1}]
H	Foam thickness [mm]
h	Heat transfer coefficient [W m^{-2} K^{-1}]
K_p	Permeability of porous medium [m^2]
k_f	Thermal conductivity of fluid [W m^{-1} K^{-1}]
k_s	Thermal conductivity of solid [W m^{-1} K^{-1}]
k_{eff}	Effective thermal conductivity [W m^{-1} K^{-1}]
L	Foam height [mm]
LTE	Local Thermal Equilibrium
Nu	Overall Nusselt number
n	Number of temperature sample
P	Pressure [Pa]
PPI	Number of pores per linear inch[in^{-1}]
Q	Heat input [W]
Re	Reynolds number
Sp	Sparrow number
T	Temperature [K]
u	Velocity component [m s^{-1}]
V	Velocity vector[m s^{-1}]
V_{max}	Maximum velocity [m s^{-1}]
V_{in}	Inlet air velocity [m s^{-1}]
W	Channel width [mm]
x	Coordinate [mm]
y	Coordinate [mm]
z	Coordinate [mm]
ΔT	Excess temperature [˚C]
Δp	Pressure drop [Pa]

Greek symbols

ρ_f	Density of fluid [kg m^{-3}]
μ_f	Dynamic viscosity of fluid [kg m^{-1} s^{-1}]
μ	Viscosity [kg m^{-1} s^{-1}]
ε	Porosity of porous medium [−]

Subscripts

ave	Average conditions
eff	Effective
f	Fluid
in	Inlet
loss	Heat loss through the insulation
p	Permeability
p	Constant pressure
s	Solid
x	Coordinate
y	Coordinate
z	Coordinate
w	wall
∞	Ambient conditions

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