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Mixed convection on jet impingement cooling of a constant heat flux horizontal porous layer

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A B S T R A C T

In the present study, numerical investigation of jet impingement cooling of a constant heat flux horizontal surface immersed in a confined porous channel is performed under mixed convection conditions, and the Darcian and non-Darcian effects are evaluated. The unsteady stream function-vorticity formulation is used to solve the governing equations. The results are presented in the mixed convection regime with wide ranges of the governing parameters: Reynolds number ($1 \leq Re \leq 1000$), modified Grashof number ($10 \leq Gr^{*} \leq 100$), half jet width ($0.1 \leq D \leq 1.0$), Darcy number ($1 \times 10^{-6} \leq Da \leq 1 \times 10^{-2}$), and the distance between the jet and the heated portion ($0.1 \leq H \leq 1.0$). It is found that the average Nusselt number (Nu_{avg}) increases with increase in either modified Grashof number or jet width for high values of Reynolds number. The average Nusselt number also increases with decrease in the distance between the jet and the heated portion. The average Nusselt number decreases with the increase in Da for the non-Darcy regime when Re is low whereas Nu_{avg} increases when Re is high. It is shown that mixed convection mode can cause minimum heat transfer unfavorably due to counteraction of jet flow against buoyancy driven flow. Minimum Nu_{avg} occurs more obviously at higher values of H. Hence the design of jet impingement cooling through porous medium should be carefully considered in the mixed convection regimes.

1. Introduction

The jet impingement cooling through horizontal porous layer are important from theoretical as well as application points of view. The buoyancy driven phenomena in porous media has attracted researchers interests due to number of technical applications, such as, fluid flow in geothermal reservoirs, insulation of buildings, separation processes in chemical industries, dispersion of chemical contaminants through water saturated soil, solidification of casting, migration of moisture in grain storage system, crude oil production, solar collectors, electronic components cooling, etc. The heat removal of high power density encountered especially in microelectronic devices can be done effectively using jet impingement of cold fluid. To enhance the heat transfer, porous heat sinks are used in the printed wire board and porous inserts are used in cooling channel of injection mold. These devices could generate high heat flux which induces buoyancy effect. Hence it is important to understand the flow and thermal characteristics of the jet impingement cooling through porous medium. Comprehensive literature survey concerned with this subject is given by Gebhart et al. [1], Kaviany [2], Nield and Bejan [3], Pop and Ingham [4], Bejan and Kraus [5], Ingham et al. [6]. Bejan et al. [7] and Vafai [8]. The literature shows that the jet impingement through pure (non-porous) fluid has been studied extensively (see, for example Al-Sanea [9], Chou and Hung [10], Seyedein et al. [11], Chiriac and Ortega [12], Chung and Luo [13], Sahoo and Sharif [14] and Sivasamy et al. [15]).

Recently many researchers considered the impinging jet through porous media. Fu and Huang [16] investigated numerically the effects of a laminar jet on the heat transfer performance of three different shape (rectangle, convex and concave) porous blocks mounted on a heated plate. They neglected the buoyancy effects and considered the forced convection mode only. Their results show that the heat transfer is mainly affected by a fluid flowing near the heated region. For a lower porous block, all the three type of porous blocks enhance the heat transfer. However, for a higher porous block, the concave porous block only enhances heat transfer. A detailed flow visualization experiment was carried out by Prakash et al. [17] to investigate the effect of a porous layer on flow patterns in an overlying turbulent flow without heat transfer. They studied the effect of the parameters such as the jet Reynolds number, the permeability of the porous foam, the thickness of the

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porous foam and the height of the overlying fluid layer. Jeng and Tzeng [18] studied numerically the air jet impingement cooling of a porous metallic foam heat sink in the forced convection mode. They found the porous aluminum foam heat sink could enhance the heat transfer from the heated horizontal source by impinging cooling. Their results show that the heat transfer performance of the aluminum foam heat sink is 2–3 times larger than that without it. Saeid and Mohamad [19] studied numerically the jet impingement cooling of heated portion of an isothermal horizontal surface immersed in a fluid saturated porous media in the mixed convection regime. It was found for high values of Peclet number at increasing either Rayleigh number or jet width lead to increase the average Nusselt number. Narrowing the distance between the jet and the heated portion could increase the average Nusselt number. Recently Jeng et al. [20] carried out experimental investigation on heat transfer associated with air jet impingement on rotating porous Aluminum foam heats sink. They investigated the effects of jet Reynolds number (Re) in the forced convection mode, the relative nozzle-to-foam tip distance (C/d), the rotational Reynolds number (Re_r) and the relative side length of the square heat sink (L/d). They found that, when Re and L/d were small and C/d was large, the increase in Re_r increases the average Nusselt number.

Tong and Subramanian [21], Lauriat and Prasad [22], and Prasad et al. [23] demonstrated the Brinkman-extended Darcy flow model in the numerical investigation on natural convection in a vertical porous layer. They highlighted the importance of Brinkman equation in convection of porous media. Hadim and Chen [24] reported a numerical study of buoyancy-aided mixed convection in an isothermally heated vertical channel filled with a fluid saturated porous medium using the Darcy–Brinkman–Forchheimer model. Their results show that, the effect of decreasing Darcy number is, however, important only at low values of Darcy number (in the Darcy Regime). At large Darcy number, the flow in this region is dominated by forced convection and the Nusselt number is almost independent of Da. Wong and Saeid [25] numerically investigated the jet impingement cooling of heated portion of an isothermal horizontal surface immersed in a confined porous channel under mixed convection conditions with Brinkman-extended Darcy model. The results were presented in the mixed convection regime with wide ranges of Rayleigh number (Ra), Péclet number (Pe), jet width and Darcy number (Da) in Darcy regime and non-Darcy regime. The average Nusselt number decreases with the increase in Da for the non-Darcy regime when Pe is low. When Pe is high, the average Nusselt number increases with the increase in Da for the non-Darcy regime. Variation of Da in Darcy regime has negligible effect on the heat transfer performance.

In many application the hot surface may be at constant heat flux instead of being isothermal, for example, embedded electronic component on a circuit board. Whenever there is a large temperature difference between the working fluid and the horizontal heated surface with constant heat flux source and the flow velocity is not significantly high, there might be some effect of thermal buoyancy force. It is important to understand the effects of buoyancy in the heat transfer and flow characteristics for designing the cooling system when impinging jet cooling process operates in mixed convection regime. This is the main motivation for this study. It is interesting to investigate the effect of the Darcy number for the jet impingement cooling of a constant heat flux horizontal channel with fluid saturated porous medium.

2. Problem description

In the present study, the effect of the buoyancy on the jet impingement cooling of a constant heat flux horizontal surface immersed in a fluid saturated porous media is considered as shown in Fig. 1. The objective of the present study is to characterize the thermal performance of the jet impingement cooling in porous media in the mixed convection regime.

The governing parameters in the present problem are the half jet width d, the jet velocity V_0, the distance between the jet and the heated portion h, and the heat source length $2L$ in additional to the physical properties of the porous media and the fluid. These parameters can be reduced to a number of dimensionless groups as given in the next section. The physical properties are assumed to be constant except the density in the buoyancy force term which is satisfied by the Boussinesq's approximation. Further it is assumed that the temperature of the fluid phase is equal to the temperature of the solid phase everywhere in the porous region, and local thermal equilibrium (LTE) model is applicable in the present investigation.

Under these assumptions, the conservation equations for mass, momentum and energy for the two-dimensional unsteady flow can be written as: [25]

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{1}$$

$$\rho\frac{\partial u}{\partial t^*} + \rho u\frac{\partial u}{\partial x} + \rho v\frac{\partial u}{\partial y} = -\frac{\partial P}{\partial x} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) - \frac{\mu}{K}u \tag{2}$$

$$\rho\frac{\partial v}{\partial t^*} + \rho u\frac{\partial v}{\partial x} + \rho v\frac{\partial v}{\partial y} = -\frac{\partial P}{\partial y} + \mu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right) - \frac{\mu}{K}v + \rho\beta g(T - T_\infty) \tag{3}$$

$$\frac{\partial T}{\partial t^*} + u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \alpha\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \tag{4}$$

The following dimensionless parameters are adopted in the present study:

$$t = \frac{t^* V_0}{L} \quad X = \frac{x}{L} \quad Y = \frac{y}{L} \quad U = \frac{u}{V_0} = \frac{\partial \Psi}{\partial Y} \quad V = \frac{v}{V_0} = -\frac{\partial \Psi}{\partial X}$$

$$\theta = \frac{T - T_c}{\Delta T} \quad \Delta T = \frac{q'' L}{k}$$

From Equations (1)–(4), the following non-dimensionless equations are obtained:

$$\frac{\partial^2 \Psi}{\partial X^2} + \frac{\partial^2 \Psi}{\partial Y^2} = -\omega \tag{5}$$

$$\frac{\partial \omega}{\partial t} + U\frac{\partial \omega}{\partial X} + V\frac{\partial \omega}{\partial Y} = \frac{1}{Re}\left(\frac{\partial^2 \omega}{\partial X^2} + \frac{\partial^2 \omega}{\partial Y^2}\right) - \frac{1}{ReDa}\omega + \frac{Gr^*}{\mathrm{Re}^2 Da}\frac{\partial \theta}{\partial X} \tag{6}$$

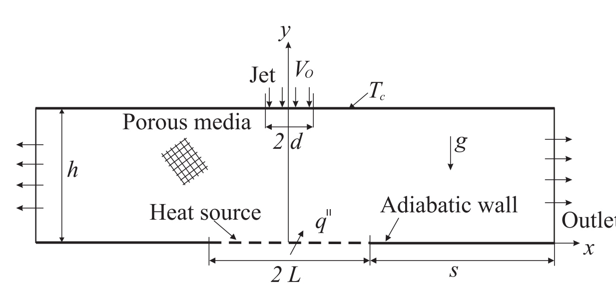

Fig. 1. Schematic diagram of the physical model and coordinate system.