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Laminar forced convection of a confined slot impinging jet in a converging channel

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ABSTRACT

In this article, a confined impinging slot jet in a converging channel is numerically investigated. The flow is laminar and a constant temperature is applied on the impingement surface. The governing mass, momentum and energy equations are solved using finite volume method. In order to consider the effect of converging angle on the flow and temperature fields in the channel, the numerical simulations were performed for different angles of $0°-5°$. Also different geometrical parameters and also the Reynolds number have been considered to study the behavior of the system in terms of stagnation point, average and local Nusselt number and stream function contours. The results showed that the intensity and size of the vortex structures increase with raising jet-impingement surface distance ratio (H/W), Reynolds number and also converging angle. The maximum Nusselt number occurs at the stagnation point and the highest values correspond to about $H/W = 5$. Also, as the converging angle increases, the average Nusselt number and skin friction increase due to raising velocity gradient along the channel.

1. Introduction

Forced convection is typically used to increase the rate of heat exchange. Many types of mixing also utilize forced convection to distribute one substance within another. Forced convection also occurs as a by-product to other processes, such as the action of a propeller in a fluid or aerodynamic heating. Fluid radiator systems, and also heating and cooling of parts of the body by blood circulation, are other familiar examples of forced convection. Various ways exist for enhancement of heat transfer such as, changing flow geometry, boundary conditions, or by enhancing the thermal conductivity of the fluid. Changing geometry, can be created with porous media, micro scale channels, increasing the surface and placing a shackle which will increase the total heat transfer. Using converging channel is one of the best commodious techniques for enhancement of heat transfer, which will be used in this paper.

One other factor for enhancing heat transfer is impinging jet. Because of their high momentum transfer rates and high efficiency, impinging jets are greatly utilized in many industrial and engineering applications such as cooling and heating processes, drying of textiles, film and paper, and food, freezing of tissue in cryosurgery and manufacturing, cooling of gas turbine components and the outer walls of combustors coating and tempering of glass and metal. Over the past 15 years, jet impingement has been used in cooling microelectronics. It is used in some application in automotive industries, such as the cooling of IGBTs (insulated-gate bipolar transistors), utilized in hybrid automobiles. Recently, a more attention has been devoted to the impinging liquid jet for much more the heat transfer rates than gas jets. Gauntner et al. [1] investigated flow characteristics of a single turbulent jet impinging on a flat plate. They reported that maximum heat transfer occurs at the tip of the potential core of the free jet. The extent of this core is 6–7 nozzle diameters for circular nozzles and is 4–7 slot widths for slot nozzles. Jambunathan et al. [2] reviewed published experimental data for the rate of the heat transfer when a circular jet impinges orthogonally on a flat surface. The results showed that the Nusselt number can be expressed in terms of the Reynolds number raised to an exponent which depends on the nozzle exit-to-plate spacing-and the distance from the stagnation point. Recently, investigations on heat and mass transfer with slot jet impingement have attracted more attention. According to Chen et al. [3], slot jet impingements offer many more beneficial features, such as higher cooling effectiveness, greater uniformity and more controllability. Slot jets have more applications than circular jets in industry especially in electronics cooling because of larger impingement domain. Also slot jets surpass multiple circular jets because of obstruction of flow between jets in the latter.

Koseoglu and Baskaya [4] experimentally investigated effects of jet inlet geometry and aspect ratio on local and average heat transfer characteristics. They reported that average heat transfer decreases with increasing jet to plate distance and heat transfer rate of

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rectangular jets reduce with increasing the cross-sectional area. Lee et al. [5] numerically studied two-dimensional laminar slot jet in the presence of an applied magnetic field to control vortices and enhance heat transfer. Hsieh et al. [6] did some visualizations and measurements on heat transfer characteristics at different Reynolds numbers. They reported that as Reynolds numbers increase vortex flow becomes unstable due to the inertia driven flow instabilities.

Naphon and Wongwises [7] reported different techniques that can be applied for heat transfer enhancement in jet impingement such as adding foams or fins. Recently, Lee et al. [8] experimentally investigated the heat transfer and overall visualized flow characteristics of confined, laminar milli-scale slot jets, as they impinge upon an isothermal impingement surface. The results showed that the maximum Nusselt Number occurs at stagnation point, decreasing with increasing jet-impingement surface distance. Also the results illustrated the sensitivity of Nusselt number for laminar boundary layer and laminar slot jet flow to thermal boundary conditions.

Lee et al. [9] numerically studied the unsteady two-dimensional fluid flow and heat transfer in a confined impinging slot jet using finite volume method. The results showed that in the steady region, the time-averaged Nusselt number at the stagnation point increases monotonically with increasing Reynolds number for different height ratios. Also they reported the average skin-friction coefficient in the steady flow decreases with increasing Reynolds number. Guo and Wood [10] made measurements along the stagnation streamline of a plane jet impinging on a flat plate and maintained the spacing in such a way that the stagnation streamline remained in the potential core of the jet. They reported that in the region close to the plate the attenuation of streamwise velocity caused a large velocity gradient which contributed to increased level of normal stress.

Garg and Maji [11] numerically investigated the flow inside a diverging-converging channels. The results showed that the converging/diverging channel configuration is an effective technique for enhancing heat transfer. However, such a technique is accompanied by undesired increase in pressure drop. Wang and Chen [12] investigated the effects of the Reynolds number and the Prandtl number on the rates of heat transfer for flow through a sinusoidally curved converging/diverging channel using a simple coordinate transformation method and spline alternating-direction implicit method. They saw that as the Reynolds number increases, total heat transfer and local Nusselt number were enhanced in the converging/diverging part of the channel. Tolentino et al. [13] experimentally investigated the fluid flow in a converging channel with wavy walls. They studied the flow behavior for different Reynolds numbers and converging/diverging angles. They found that a diverging channel has a better effect in a chaotic mixture as flow is more stable for a converging channel even at higher Reynolds numbers. Rahimi-Esbo et al. [14] numerically studied the turbulent forced convection of jet flow in a converging sinusoidal channel. They reported that by increasing the Reynolds number, aspect ratio and amplitude, the average Nusselt number and length of re-circulation region increase.

A systematic comprehensive study of confined slot impinging jet has not been conducted with different geometric parameters yet. In this paper two-dimensional laminar flow of confined impinging jet in a converging channel is numerically investigated. The scope of this study is to obtain an investigation of the velocity and temperature distribution, Nusselt number along bottom wall and stagnation point of the impinging jet inside converging channel. Moreover, the performance of the jet flow at various Reynolds numbers is investigated. Also the effect of converging different angles on thermal and flow characteristics and the effect of jet-to-impingement surface distances ratio on dimensionless stream function contours and the intensity and size of the vortex and heat transfer properties are investigated.

2. Model description and mathematical formulation

2.1. Geometrical configuration

Fig. 1a shows the impinging jet on a heated wall in the converging channel. The two-dimensional model has a length (L) equal to 0.8 m, height (H) and converging angles (α) changing from 0.02 to 0.1 m and 0° to 5° respectively and jet inlet width (W) equal to 0.01 m. So the value of H/W ratio, changes from 2 to 10. The considered fluid is water.

2.2. Boundary conditions

At the channel inlet section, uniform velocity, V_{jet} and temperature, $T_{jet} = 293\ K$, are assumed. At the channel exit section, pressure outlet condition (a fixed pressure at the section) is considered and other variables are extrapolated where the flow is outwards [15]. On the bottom wall, the no-slip condition is applied and the wall has a constant temperature $T_w = 340$ K. For the upper wall, adiabatic condition has been considered and the velocity components are equal to zero.

2.3. Governing equation

The continuity, momentum and energy equations for the 2D flow problem are given by:

1. Continuity:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \tag{1}$$

2. Momentum:

$$\frac{\partial}{\partial x_j}(\rho u_i u_j) = -\partial p/\partial x_i + \frac{\partial}{\partial x_j}\left[\mu\left(\partial u_i/\partial x_j + \partial u_j/\partial x_i\right)\right] \tag{2}$$

3. Energy:

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_j}\left(\frac{\mu}{pr}\frac{\partial T}{\partial x_j}\right) \tag{3}$$

The nondimensional flow and heat transfer parameters are defined as.

$$Re = 2\rho V_{jet} W/\mu \tag{4}$$

$$Nu = \frac{2W\partial T/\partial y}{T_w - T_{jet}} \tag{5}$$

$$C_f = \frac{\tau_w}{2\rho V_{jet}^2} \tag{6}$$

In the above equations, Re, Nu and C_f represent the Reynolds number, Nusselt number and friction coefficient, respectively.

3. Discretization method

Fig. 3 shows the solution domain subdivided into a finite number of contiguous quadrilateral control volumes (CV). The control volumes are defined by coordinates of their vertices, which