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Numerical Investigation on Slot air Jet impingement Heat Transfer between Horizontal Concentric Circular Cylinders

Arash Azimi[1], Mehdi Ashjaee[2], Morteza Khayat[3]

[1]Department of Mechanical and Aerospace Engineering, Science and Research Branch, Islamic Azad University, Tehran
Tehran, 14778-93855, Iran, azimi.arash84@gmail.com,

[2]Department of Mechanical Engineering, University of Tehran
Tehran, Karegar street, Iran, ashjaee@ut.ac.ir

[3]Department of Mechanical and Aerospace Engineering, Science and Research Branch, Islamic Azad University, Tehran
Tehran, 14778-93855, Iran, mkhayat@srbiau.ac.ir

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Corresponding author: Arash Azimi, azimi.arash84@gmail.com

Abstract

A numerical study has been carried out for slot air jet impingement cooling of horizontal concentric circular cylinders. The slot air jet is situated at the symmetry line of a horizontal cylinder along the gravity vector and impinges on the bottom of the outer cylinder which is designated as θ=0°. The outer cylinder is partially opened at the top with a width of W=30mm and is kept at constant temperature T= 62°C. The inner cylinder which is a part of the slot jet structure is chosen to be insulated. The effects of jet Reynolds number in the range of $100 \leq Re_j \leq 1000$ and the ratio of spacing between nozzle and outer cylinder surface to the jet width for H=4.2 and H=12.5 on the local and average Nusselt numbers are examined. In the numerical study, FLUENT CFD package is used and validated by comparing the results with the experimental data at the same Reynolds number. It is observed that the maximum Nusselt number occurs at the stagnation point at (θ=0°) and the local heat transfer coefficient decreases on the circumference of the cylinder with increase of θ as a result of thermal boundary layer thickness growth. Also, results show that the local and average heat transfer coefficients are raised by increasing the jet Reynolds number and by decreasing the nozzle-to-surface spacing.

Keywords: Heat transfer, Impingement cooling, Slot-jet, Concentric cylinders

1. Introduction

Due to high rates of localized heat transfer, impinging jet flows is employed in a wide variety of applications of practical interest, such as high temperature gas turbines, surface coating and cleaning, cooling of electronic components, metal cutting and forming, fire testing and building materials, turbine blade cooling, drying of textiles, veneer, paper and film materials, aircraft wing leading edge heating for anti-icing application, etc. Even though the flow geometry is simple in jet impingement heat transfer problems, the physics of the flow is very complex due to the shear layer development at the free jet and wall jet boundaries, boundary layer development at the impingement surface, and very high streamline curvature near the impingement location. The surface curvature has a strong effect on the overall flow as well as in the wall jet development in the circumferential direction due to the additional streamline curvature and associated centrifugal and coriolis forces. It is confirmed by most of the related literature that where the jet impinges on a concave circular surface, the flow can be divided into the following regions: (a) free jet, (b) stagnation, (c) wall jet.

There have been numerous studies conducted relating to jet impingement flow and heat transfer, most of which considered impinging jet flow on flat surfaces [1-5]. There has not been much investigation into the flow of jets impinging upon curved surfaces. In general, the limited work that has been done mainly focuses on experimental heat and/or mass transfer measurements and their theoretical interpretation. Yang et al. [6] investigated the air jet impingement cooling on a semi-circular concave surface with three different shapes of slot nozzle, including round, rectangular and 2D contoured nozzle ones. Choi et al. [7] performed an experimental study on fluid flow and heat transfer for air jet impingement cooling on a semi-circular concave surface. The local and average Nusselt numbers for various jet Reynolds numbers ranging from 1780 to 7100 and nozzle-to-surface spacing ranging from 0.4 to 16 have been reported. Kayansayan and Kucuka [8] performed an experimental and numerical study on the air slot jet impingement cooling of a concave channel, where the slot-jet is located at the symmetry line of the semi-circular channel. Their jet Reynolds number ranged from 200 to 11000, while the normalized jet-to-impingement surface spacing ranged from 2.2 to 4.2. In addition, a correlation is presented for the stagnation region heat transfer. They did not perform computations for $35° \leq \Theta$ from the stagnation point. The nonlinear flow and heat transfer characteristics for a slot jet impingement on a slightly curved concave surface are experimentally studied by Eren et al. [9]. Their Reynolds number ranged from 8617 to 15415 and the jet-to-impingement surface distance is kept constant at 8. Fenot et al. [10] presented an experimental investigation of heat transfer due to the row of air jet impingement on a concave semi-circular surface. Heat thin foil technique and infrared tomography are used to measure the heat transfer characteristics. Terekhov et al. [11] experimentally investigated the fluid flow and heat transfer characteristics for jet impingement cooling of obstacles in the form of single spherical cavities. The distribution of the flow velocities between the nozzle and the obstacle, the pressure field and the heat transfer coefficients inside the cavities are measured. Kumar and Prasad [12] numerically investigated the flow and heat transfer from a row of circular jets impinging on a concave cylindrical surface using the Fluent CFD code and the SST $k-\Theta$ turbulence model. Hu and Zhang [13] experimentally and numerically studied the effects of the water jet impingement exit velocity and the nozzle-to-surface distance on the heat transfer characteristics on a convex hemispherical surface. Their results show that for lower Reynolds numbers, the local Nusselt number decreased from the stagnation point which the maximum Nusselt number occurs at this point. Sharif and Mothe [14] numerically investigated the convective heat transfer from the concave cylindrical surface for turbulent air slot-jet impingement. A constant heat flux is assumed for the surface boundary condition. The effect of jet-exit Reynolds numbers, the surface curvature and nozzle-to-surface spacing on the heat transfer and the flow structure are considered. Recently, Öztekin et al. [15, 16] performed an experimental and numerical investigation for slot jet impingement cooling and hydrodynamic characteristics on concave plates with varying surface curvature. The local and average Nusselt numbers for $3423 \leq Re \leq 9485$ and $\Theta \leq 60°$ from the stagnation point have been reported. These studies have mainly focused on the concave surfaces.

All of the mentioned investigations used a semi-circular concave surface less than or equal to 90° with respect to vertical symmetry line as target area. Also, the local convection heat transfer coefficient has been reported up to 90° form stagnation point. The aim of current study is to investigate the effects of air slot-jet impingement and nozzle-to-surface spacing on the heat transfer characteristics from horizontal concentric circular cylinders numerically. The computations in this study are performed using the Fluent version 6.3 commercial flow solver code. The results of the parametric study are presented in terms of isotherm plots and Nusselt number between cylinders surface up to 110° from the stagnation point (θ=0°).

2. Mathematical Formulation and Model

The characteristic parameters in the present problem are the angle from the stagnation point Θ, the distance between the jet and the outer cylinder bottom surface Z and the width of the slot jet B which are shown in Figure 1. The inner cylinder is chosen to be insulated and the outer cylinder is kept at a constant temperature of 62°C. The Reynolds number and the temperature of the jet are Re_j and T_j respectively. The cylinder opening width W=30mm which correspond to a circumferential angle of Δθ=50.8°. So, heat transfer occurs in an annulus and the impingement air jet. In the numerical study, FLUENT CFD package is used. This package uses a technique based on control volume theory to convert the governing equations to algebraic equations so that they can be solved numerically. The control volume technique works by performing the integration of the governing equations about each control volume, and then generates discretization of the equations which conserve each quantity based on control volume [17]. The governing equations (conservation of continuity, momentum and energy equations) for the fluid for laminar flow can be written as follows:

Continuity equation:

$$\frac{\partial \rho}{\partial t}+\frac{\partial}{\partial x}(\rho u)+\frac{\partial}{\partial y}(\rho v)=0 \tag{1}$$

Momentum equation in x direction:

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

Momentum equation in y direction:

$$\rho\left[\frac{\partial v}{\partial t}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}\right]=-\frac{\partial P}{\partial y}+\rho g+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right) \tag{3}$$

Energy equation:

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

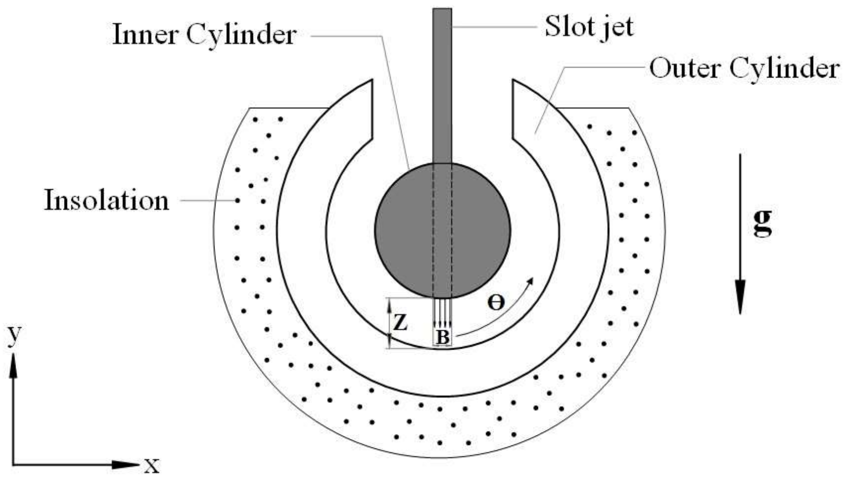


Fig. 1. A schematic of geometry.

The GAMBIT mesh generator associated with the solver has been used to plot and mesh the 2D model of the cylinders. The solution grid created is shown in figure 2 and the corresponding boundary conditions are specified in Table 1. Since the temperature of the cylinder does not change in the axial direction, problem is modeled and solved in two dimensional spaces at steady-state conditions. In order to obtain a converged solution, several iterations of the solution loop must be performed because the governing equations are nonlinear (and coupled). The standard laminar viscous flow model and surface to surface radiation model were used. For pressure-velocity coupling discretization, the SIMPLE (Semi-Implicit Method for Pressure Linked Equations) algorithm has been used. For continuity and momentum, the residual values were taken 10^{-8} and for energy 10^{-12}.

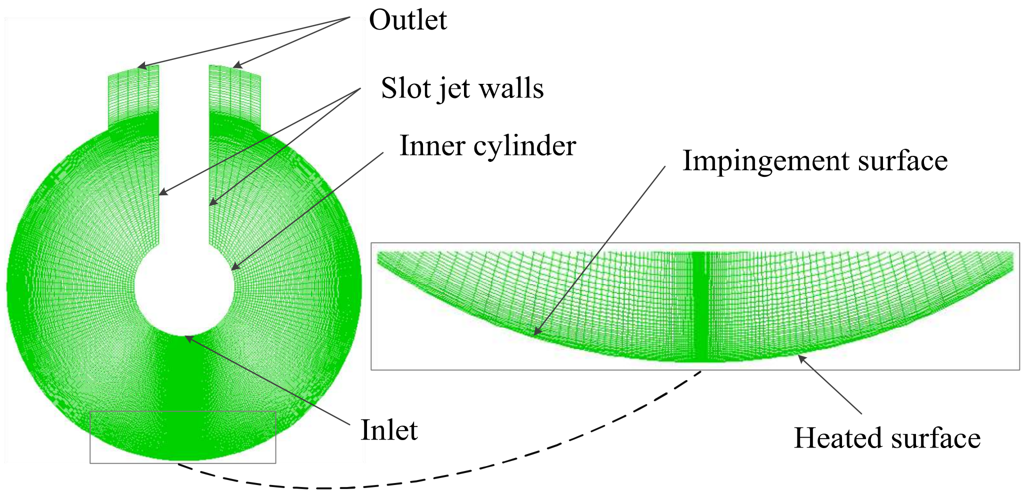


Fig. 2. The solution grid of the 2D model of the concentric cylinders.

Table 1. Boundary conditions

	Boundary condition	u (m/s)	v (m/s)	T (°C)
Inlet	Velocity Inlet	0	U_j	T_j= constant
Outlet	Pressure Outlet	-	-	-
Outer cylinder	Wall	0	0	T_s= constant
Inner cylinder	Wall	0	0	$q''=0$
Nozzle walls	Wall	0	0	$q''=0$

3. Grid Dependency

To ensure that the results are independent of the computational grid, grid sensitivity analysis is carried out. Generally, the accuracy of the solution and the time required for the solution are dependent on mesh refinement. In this study, the optimum grid is searched to have the appropriate run-time and enough accuracy. Figure 3 shows the results of a typical grid independence study for the case of Re_j=920 and H=4.2. A close examination of the plots reveals that grid distribution in excess of 25600 does not produce any significant change in average Nusselt number. So this grid distribution is used for further computations of this case. Similarly, for each case, a systematic grid convergence study has been done to make sure the proper grid resolution for grid independent result is selected.

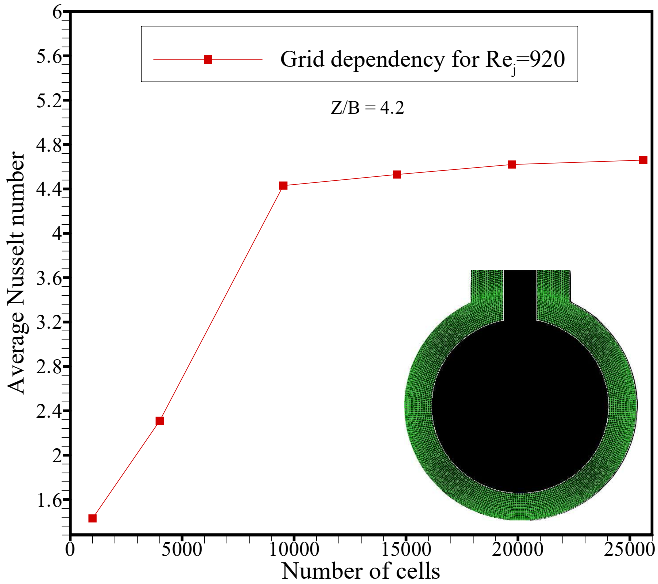


Fig. 3. Grid dependency.

4. Code Validation

In order to validate the numerical results, the local Nusselt number is compared with the benchmark experimental data by Kayansayan and Kucuka [8]. Also, for enough accuracy, the experimental study for Re_j=920 and H=4.2 is carried out using the Mach-Zehnder interferometry (MZI) technique. The Mach-Zehnder interferometer has been used extensively in studies of gas flow, combustion, plasma density, and diffusion, where changes in the refractive index occur that can be related to changes in pressure, temperature, or the relative concentrations of different components of a mixture. In this study, a Mach–Zehnder interferometer is used to observe the temperature field between cylinders. Further information about MZI can be found in [18, 19]. Comparison between results is shown in figure 4. As shown in this figure, an excellent agreement between the present numerical data with the published results [8] is noticed. Also, it is seen that numerical and experimental results for $10° \leq \Theta$ are in a good agreement. In addition, numerical and experimental results for isotherms in the case of Re_j=750 and H=4.2 are compared in figure 5.

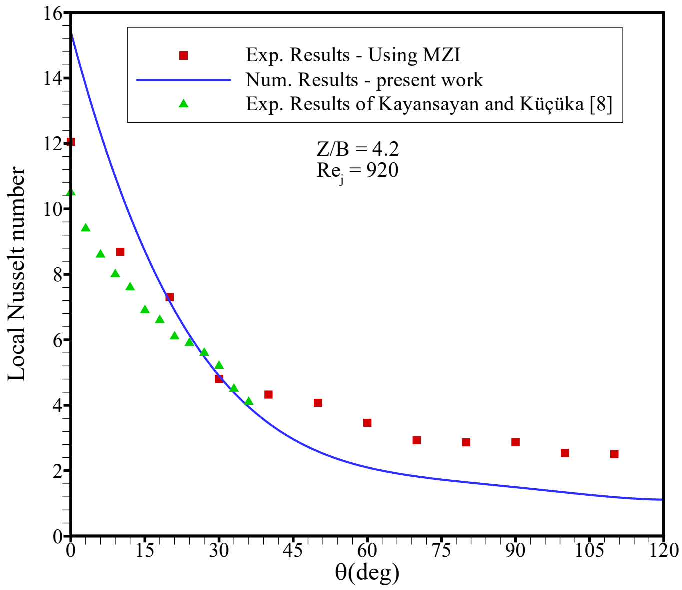


Fig. 4. Comparison between present numerical and experimental results and the previous work [8].

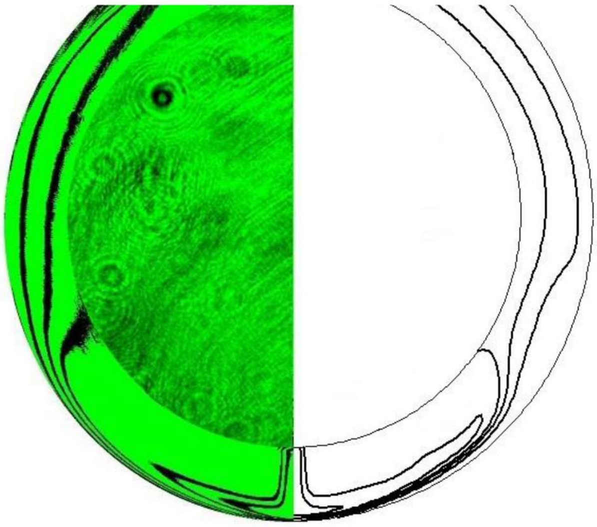


Fig. 5. Comparison between numerical isotherms (right) and experimental isotherms visualized by MZI (left).

5. Results and Discussion

In this study, the jet Reynolds number and nozzle-to-surface spacing are changed and their effects on local and average convection heat transfer coefficients are studied numerically. The effect of Re_j values on the local Nusselt number at a constant value of nozzle-to-surface spacing is shown in figure 6 and 7. At first, the slot air jet impinges on the heated surface at the stagnation point (Θ=0°) where a maximum convection heat transfer has been obtained. Then it gets divided to two wall jets flows between each half of the annulus. A thermal boundary layer develops from stagnation point to Θ=110° on the each half. These figures show the significant effect of the jet Reynolds number on the local Nusselt number, especially for the region close to the stagnation point. Numerical isotherms for H=4.2 and H=12.5 for different values of Re_j are shown in Figures 8 and 9 respectively. The distance between the isotherms along the circumferential positions increases from Θ =0° to Θ=110° which is an indication of growth of thermal boundary layer are decrease in convection heat transfer coefficients. As shown, due to the presence of high temperature gradients in the region around the stagnation point, the local Nusselt number is relatively high. After the impingement of the slot jet on the stagnation line, as the wall jet continues its motion following the cylinder surface, the local Nusselt number decreases rapidly due to the lowering of momentum and the growth of the thermal boundary layer.

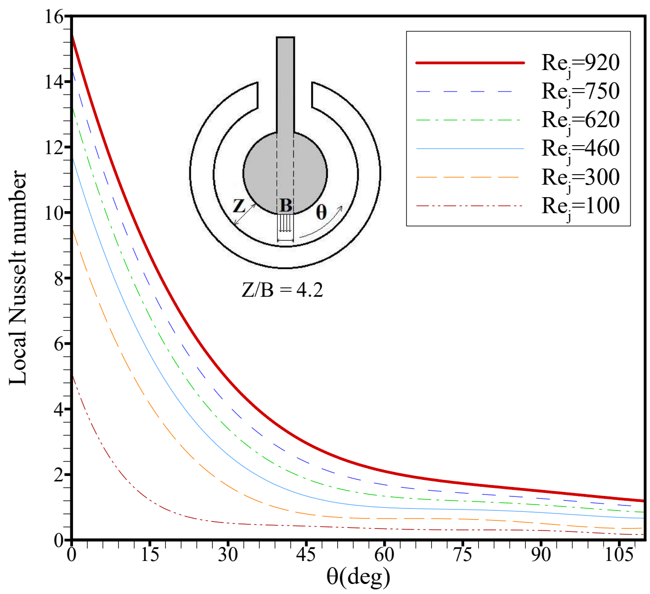


Fig. 6. Variation of local Nusselt number from Θ=0° to 110° at different values of Re_j, H=4.2.

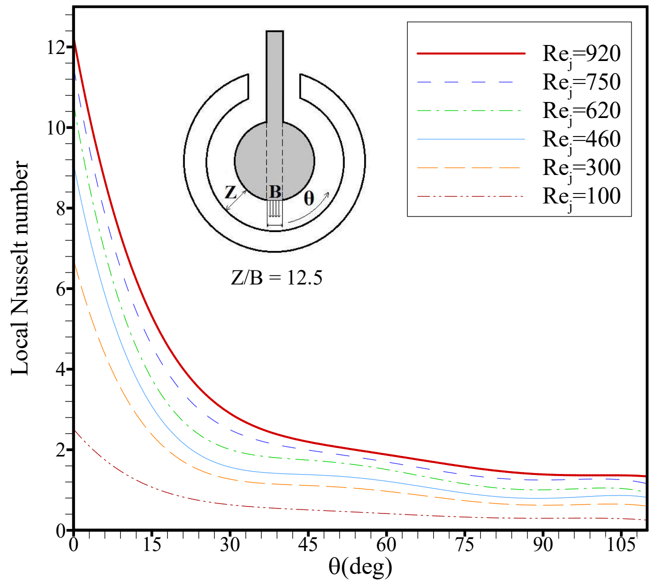
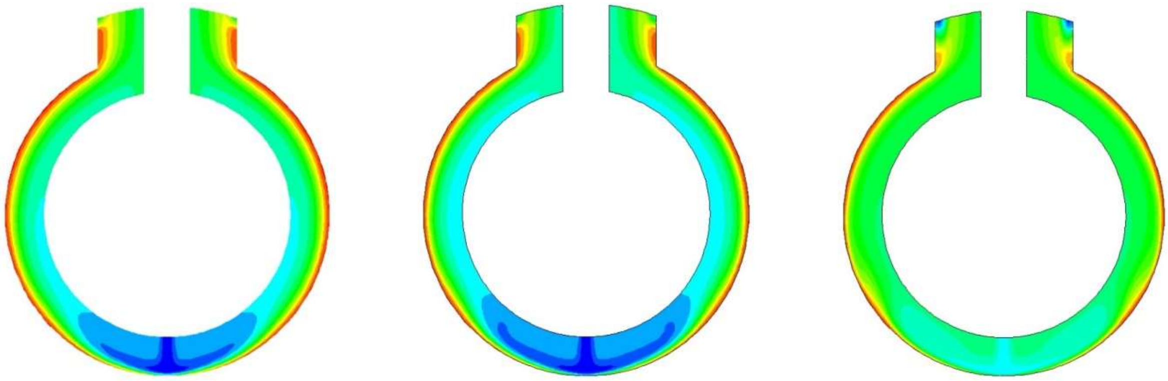


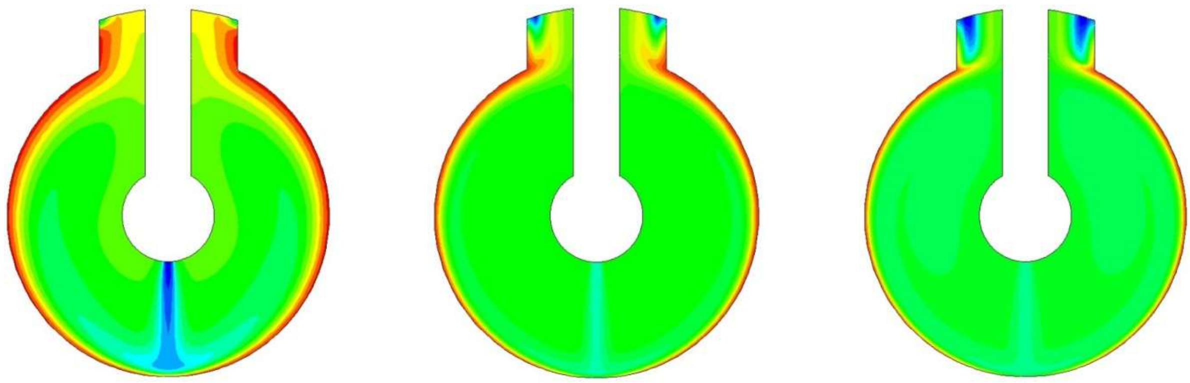
Fig. 7. Variation of local Nusselt number from Θ =0° to 110° at different values of Re_j, H=12.5.

Furthermore, it can be seen that the local Nusselt number decreases by increasing the nozzle-to-surface spacing especially for the region close to the stagnation point. The reason for this decrease is that by increasing the value of H, the air impinges the stagnation point with a lower velocity which figures 8 and 9 show this effect with an increase between the isotherm lines. So, the stagnation Nusselt number decreases by raising the value of H. After the region approaches the stagnation point, the effect of different nozzle-to-surface spacing on the local Nusselt number is weak.



Re_j=100 **Re_j=460** **Re_j=920**

Fig. 8. The temperature distribution on the heated outer cylinder for different values of Re_j and H=4.2.



Re_j=100 **Re_j=460** **Re_j=920**

Fig. 9. The temperature distribution on the heated outer cylinder for different values of Re_j and H=12.5.

It can be seen that due to the very high temperature gradient around the stagnation point, the maximum convection heat transfer has occurred (Θ=0°). At low jet Reynolds numbers, the isotherms pattern are very similar to the natural convection isotherms, particularly for the case of H=4.2 (Fig. 8).

Figure 10 illustrates the variation of the average Nusselt number as a function of jet Reynolds number for H=4.2 and H=12.5. As shown, the average Nusselt number increases by increasing the jet Reynolds number. Also, at a constant value of the jet Reynolds number, the average Nusselt number increase as the value of the nozzle-to-surface spacing decrease. This increase in the average Nusselt number is mainly due to large increase in the local Nusselt number around the stagnation point.

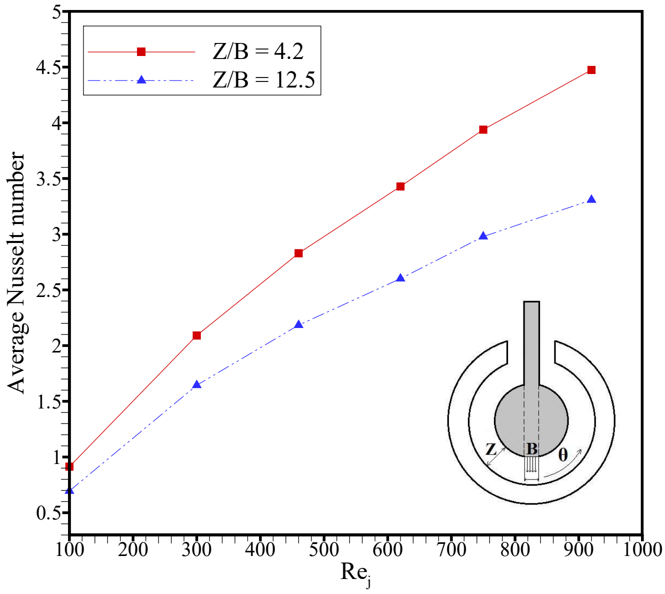


Fig. 10. Variation of average Nusselt number at different values of Re_j

The variation of the Nusselt number at the stagnation point versus jet Reynolds number is illustrated in Figure 11. It can be seen that the Nusselt number at the stagnation point is relatively high. As shown, at specified H, by increasing the jet Reynolds number, the Nusselt number at stagnation point increases. In addition, by increasing the value of H, the Nusselt Number at the stagnation point decreases.

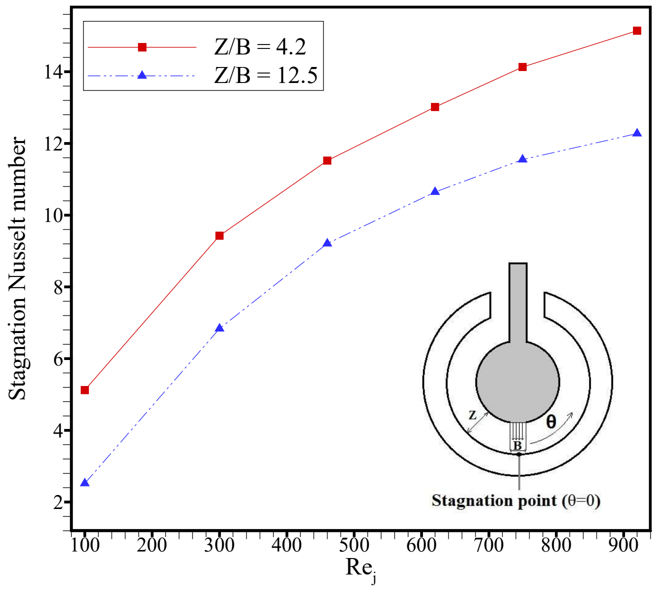


Fig. 11. Variation of stagnation Nusselt number at different values of Re_j

6. Conclusion

Heat transfer in an annulus by slot jet impingement has been studied numerically. In order to validate numerical results, an experimental setup is made. The experiments are carried out using Mach-Zehnder interferometer. The effects of the jet Reynolds number (Re_j) and the dimensionless nozzle-to-surface spacing (Z/B) on the local, average and stagnation point Nusselt number have been studied. The results are presented in terms of isotherm contours, the variation of the local and average Nusselt number on the circumferential direction of a concave surface in the annulus and the stagnation point Nusselt number.

The main results can be summarized as follows:

1. As the slot air jet impinges the outer cylinder surface at the stagnation point Θ =0°, there is a high value for the local Nusselt number at this location. Then the thermal boundary layer develops, which causes the local Nusselt number to decrease.
2. The local Nusselt number decreases by increasing the nozzle-to-surface spacing (H) especially for the region close to the stagnation point.
3. By increasing the value of jet Reynolds number at a constant H, the local Nusselt number increases.
4. At a constant value of H, the average Nusselt number and the Nusselt number at the stagnation point increase by increasing the jet Reynolds Number.

Nomenclature

B	Slot jet width [mm]	u	Horizontal velocity component [m/s]
g	Gravitational acceleration [m/s^2]	v	Vertical velocity component [m/s]
h	Local heat transfer coefficient [w/m^{2}K]	W	Width of opening of the outer cylinder [mm]
H	Dimensionless nozzle-to-surface spacing (Z/B)	x	Horizontal coordinate
Nu	Nusselt number [h×2B/k]	y	Vertical coordinate
P	Pressure	Z	Distance between the jet exit and the impingement surface [mm]
K	Air thermal conductivity [w/mK]	ρ	Density [kg/m^3]
Re_j	Jet Reynolds number [ρ× U_j ×2B/ μ]	α	Thermal diffusivity [m^2/s]
T	Temperature [°C]	Θ	Angle from the stagnation point [°]
q″	Heat flux [w/m^2]	μ	Kinematic viscosity [kg/ms]
U_j	Jet exit velocity [m/s]		

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