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Effect of Hydraulic Jump on a Moving Surface by Slot Jet Impingement: A Numerical Approach

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ABSTRACT
This numerical investigation focuses on the effect of hydraulic jump on a moving flat surface due to jet impingement. Liquid jet impingement is broadly used in cooling applications because of its high heat transfer capability. Because it spreads radially after the jet impinges on a flat surface, the hydraulic jump affects the total amount of heat that is transferred, making it an essential component of the heat transfer study of jet impingement. The governing equations are solved numerically using the FVM method by ANSYS Fluent: a commercial CFD package. The results covered the effect of hydraulic jump on fluid flow and heat transfer characteristics on a flat surface for two different Reynold's number of 23,000 and 50,000. The effect of different plate velocities (U_p) is also considered for the simulation process to determine the hydraulic jump on the flat surface. The local Nusselt number and skin friction coefficient and pressure coefficient at the heated moving plate is presented. The study suggests that the heat transfer characteristics increased due to the effect of the hydraulic jump for higher plate velocity. The analysis also reveals that the jet exit Reynold's number and the plate velocity have a significant positive correlation with a substantial rise in the heat transfer phenomena. These key findings are engendered from the influence of hydraulic jump on thinning the boundary layer.

Keywords: Heat Transfer, Hydraulic Jump, Jet impingement, Cooling, Moving Plate.

1. Introduction

The study of jet impingement on static and moving surfaces is significant because it has a wide variety of technical applications. Some examples of these applications include hot rolling, extrusion cooling, quenching of steel and other metallic plates, cooling of turbine blades, and drying of textile products. The rate of heat transfer can be increased by jets impinging on the target. In most cases, the jet is ejected from a nozzle that has either a circle or a rectangular section, and it is referred to as either a circular jet or a slot jet depending on the shape of the section. The complexity of the problem, which arises from a sudden change in the direction of flow combined with a significant curvature and the motion of the plate, which causes abrupt changes in the velocity gradient near the impingement region which presents enormous challenges for finding a solution to the problem. Several works have been done on jet impingement, all with the purpose of analyzing the heat transfer characteristics. These works have varied the settings under which they were conducted. Despite this, it continues to be an active field of research due to the intricate fluid dynamics that are associated with it. During the very early days of numerical investigation on the impingement of jets, Polat et al. [1] published a detailed analysis of the heat transfer properties of impinging jets. There have been a lot of work to find out different heat transfer characteristics to contribute to improving the cooling processes. Some examples of such works are Morris et al. [1], Cziesla et al. [2], Shi et al. [3], Sahoo et al. [4]. etc. But, during all these mentioned works, the impingement plate was held stationary. Lee et al. [5] experimentally investigated the turbulent heat transfer characteristics in a stagnation point for an air jet impinging normal to a heated flat plate. The study of turbulent jet impingement on a stationary surface is difficult in and of itself, as was mentioned before. Amongst the complex fluid dynamics affiliated with it are as strong streamline curvature, flow recirculation, entrainment and boundary layer development on the impingement surface. So, further addition of surface motion of impingement in any definite direction embellishes the solution process with further complexities due to strong shear regions. There have been relatively fewer studies on the effect of the impingement surface motion on the flow field and heat transfer characteristics though there are numerous practical applications in industrial and manufacturing process. Huang et al. [6] experimented heat transfer under a turbulent slot-jet with cross flow and surface motion. His findings indicated that, when the surface was provided higher speed, Nusselt number was found to go down in places where the surface motion opposed the dividing jet flow. Also, the Nusselt number was found higher when the surface motion assisted the dividing jet flow. Chen et al. [7] and Zumbrunnen et al. [8] demonstrated the strong influence of moving surface on flow field and heat transfer. However, a numerical model was considered for convective heat transfer within an arrangement of submerged planar jets impinging on moving surface where the moving surface was applied with a constant heat flux. De facto, their study was considered for laminar regime. Later, Chattopadhyay et al. [9] solved a turbulent jet impingement problem on moving surface using Large Eddy Simulation (LES) for jet exit Reynolds number ranging between 500 and 3,000. The study outcome showed that the span averaged Nusselt number distribution is found to uniform while the total heat transfer reduces with increasing plate speed. However, the study was mainly focused on the velocity profiles, distribution of turbulent stresses and flow structures. Lee et al [5] experimentally investigated the turbulent heat transfer characteristics in a

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stagnation point for an air jet impinging normal to a heated flat plate.

There has not yet been sufficient attention paid to the problem of conducting a methodical and in-depth examination of restricted jet impingement and its myriad of properties under varying conditions of impingement surface velocity and Reynold's number. The sole purpose of this work is to analyze the effect of the hydraulic jump on the flow and heat transfer characteristics on a heated moving flat surface.

2. Methodology

2.1 Problem Description

The problem was solved using finite volume-based software ANSYS 2020R1. Below, **Fig. 1** shows the schematic diagram of the present work where a single impinging jet was considered inside a closed domain with pressure outlets on both horizontal sides. The direction of the motion of the moving plate is as demonstrated in the schematic. The impingement surface was maintained at a constant temperature. For this study, water was considered as the working fluid and the distance between jet nozzle and the impingement surface was 4 times the nozzle diameter, i.e., $H = 4D$.

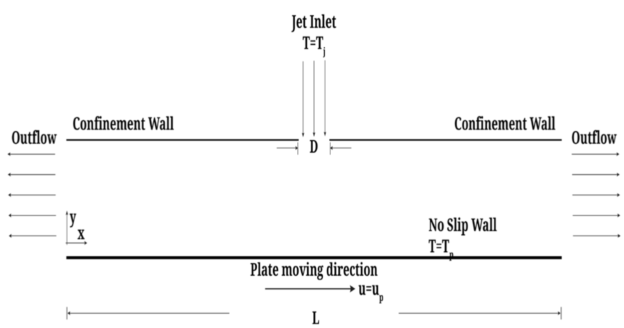

Fig. 1 A schematic diagram of current work

2.2 Computational Model

The mass, momentum and the energy equations are solved in a 2D planer domain using a RANS based approach. The incompressible continuity equation can be expressed as follows:

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

The momentum equation can be expressed as:

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

And the energy equation:

$$\frac{\partial}{\partial x_i}(\rho T) + \frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_i}\left(\frac{\gamma}{C_p}\frac{\partial T}{\partial x_i}\right) \tag{3}$$

For turbulence modeling the SST $k-\omega$ model was applied. Because, it can account for the transfer of the primary shear stress in boundary layers with an adverse pressure gradient and that goes with the current works' circumstances. Therefore, due to its high accuracy to expense ratio, it is the most prevalent model in the industry. The $k-\omega$ model equations are expressed as:

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial x}{\partial x_j}\left[\mu_k \frac{\partial k}{\partial x_j}\right] + G_k - Y_k + S_k + G_b \tag{4}$$

And

$$\frac{\partial}{\partial t}(\rho \omega) + \frac{\partial}{\partial x_i}(\rho \omega u_i) = \frac{\partial}{\partial x_j}\left[\mu_\omega \frac{\partial \omega}{\partial x_j}\right] + G_\omega - Y_\omega + S_{\omega v} + G_{\omega b} \tag{5}$$

Where, G_k= Generation of turbulence kinetic energy due to mean velocity gradient

G_ω = Generation of ω

Y_k, Y_ω = The dissipation of k and ω

σ_k, σ_k = Turbulent Prandtl number for k and ω

S_k, S_ω = User defined source terms

The local Nusselt number of the heat source on the surfaces can be defined as follows:

$$Nu = \frac{hd}{k} \tag{6}$$

where k represents the thermal conductivity of the fluid, h represents the heat transfer coefficient, and d represents the hydraulic diameter of the nozzle (also known as the jet diameter).

The average surface Nusselt number is calculated by:

$$\overline{Nu} = \frac{1}{A}\int_A Nu.dA \tag{7}$$

For the numerical simulations, the pressure-based coupled solver in Ansys Fluent was employed, which was used to apply all of the boundary conditions. In order to discretize the pressure, momentum, and turbulence, a second-order upwind technique was utilized, and a least-squares cell-based solver was used to the problem of calculating the gradient. In order to create the turbulence model, the SST $k-\omega$ model was applied with 5% turbulence intensity.

2.3 Boundary Conditions

At the inlet, water was defined at a fixed temperature of 300 K. The inlet was defined as a velocity inlet for the respected Reynold's number. Ambient boundary condition was applied at the pressure outlet. The moving plate was defined as a constant temperature wall at 325 K with a varying plate velocity.

2.4 Mesh Generation and Wall Treatment

A quadrilateral dominated structured finite-volume mesh is created for the two-dimensional axisymmetric flow domain with the assistance of the ANSYS Meshing. At this case, the turbulence is resolved in the viscous sublayer area. At the jet wall area and the impingement region, the mesh is denser. Biasing was employed at the impingement region and in the wall region with proper biasing factor. The y^+ value maintained near the impingement wall was about 0.87. The following **Fig.2** represents the generated mesh of the 2D domain.

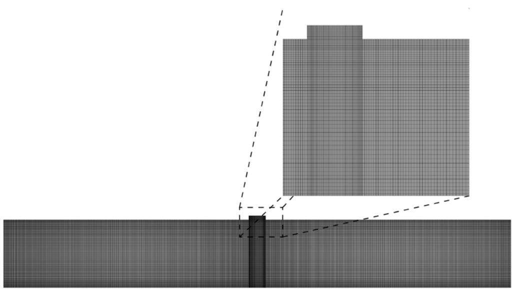

Fig. 2: Generated mesh of the 2D domain

2.5 Grid Independence Test

A number of simulations were carried out in order to discover mesh-independent solutions for the processes of impingement cooling of the moving heated flat plate. Grid independence study was carried out for Re = 23000 and U_p = 0 m/s condition. In order to proceed with the simulation procedure, meshes with numbers of elements of 1,02,300 was selected for the future simulation as the average Nusselt number ($\overline{Nu}$) didn't vary with the further mesh refinement. This meshing setup was also applied for the other geometrical configurations also.

Table 1: Grid independence test results for the 2D domain

Number of Elements	Average Nusselt Number ($\overline{Nu}$)
5,425	211.7754
12,030	199.6421
48,100	184.8198
1,02,300	186.799
1,23,789	186.887

2.6 Validation

The local Nusselt number data from the experimental work of Lee et al. [6] were found to be consistent with the present model. Consequently, the existing model was verified for the stationary heated flat surface. **Fig. 3** shows the comparison between the experimental and numerical data. Here it can be seen that the numerical value matches well with the experimental data. The secondary peak is captured also in the numerical simulation due to laminar to turbulent transition. Here the intermittency model was on to capture this phenomenon.

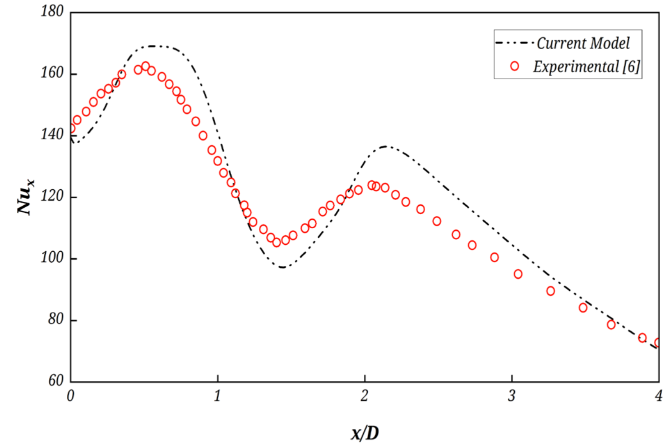

Fig. 3: Comparison of numerically local Nusselt number distribution along the stationary impingement plate with the experimental data [6].

3. Results & Discussion

For the steady-state analysis as shown in **Fig. 1** the moving heated plate confined by a parallel adiabatic wall at a nozzle to surface distance H = 4D is analyzed for two different Reynold's number. The predicted stream lines for the Reynold's (23,000 and 50,000) with the plate velocity (U_p = 0, 0.2, 0.5 & 1) are presented in **Fig. 4** and **Fig.5**.

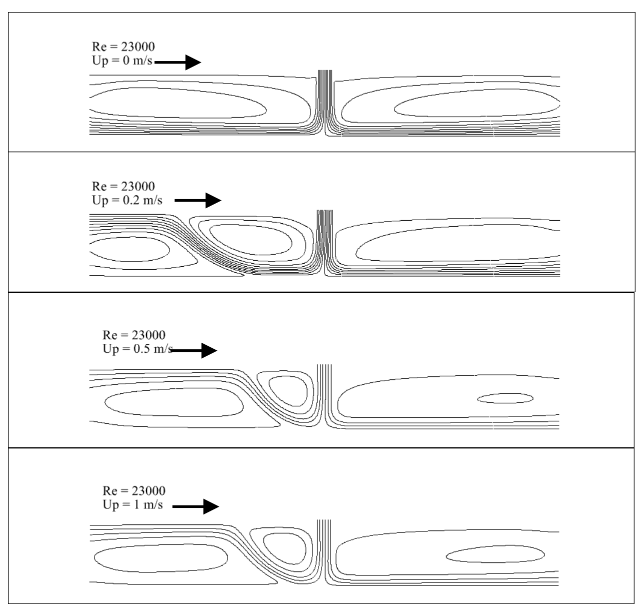

Fig. 4: Streamline contours for different plate velocities at Re=23,000.

For the case of stationary plate, $U_p = 0$ m/s as the jet hits the bottom plates, two counter rotating vortices are generated on the both sides of the jet due to entrainment of jet and confining effects of its closed domain. The vortex on the left side is clockwise and the vortex on the right side is counter clockwise.

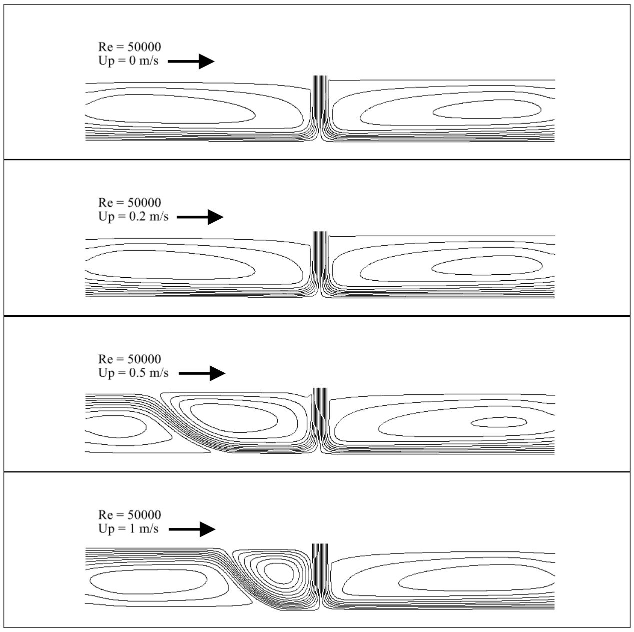

Fig. 5: Streamline contours for different plate velocities at Re=50,000.

As the bottom plate was given a velocity of 0.2 m/s, a hydraulic jump on the left side is seen developed due to the additional rightward drag exerted on the fluid. Due to the development of the hydraulic jump, two particular vortices are seen at the left side. The stream function contours for increased Reynolds number, (Re=50,000) is shown in **Fig. 5 .** Unlike, **Fig. 4** the hydraulic jump requires higher plate velocity for the case of increased Reynolds number. Also, it is evident from both **Fig. 4** & **Fig. 5** that the upper vortex of the jump gets skewed with the increase of plate velocity.

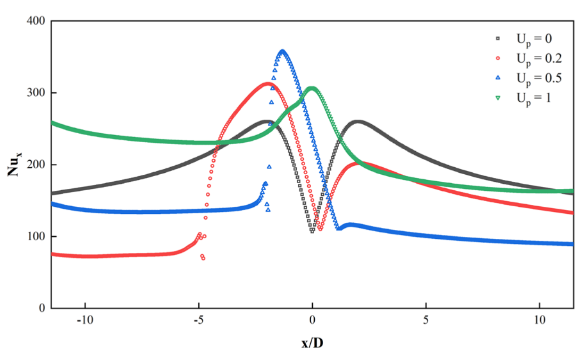

Fig. 6: Distribution of local Nusselt number along the hot moving plate for Re=23,000.

Fig. 6 depicts the local Nusselt number distribution for varied plate velocities at Re=23,000. From the figure, it's quite conspicuous that the hydraulic jump certainly has an impact on heat transfer as the peak value of local Nusselt number is lowest was the plate was held stationary. The maximum peak is achieved for the plate velocity at $U_p = 0.5$ m/s. From the figure it can also be seen that with zero plate velocity the Nusselt number distribution is similar for the left and right portion of the flow domain. But with the varying plate velocity, the local Nusselt number distribution is higher at the left portion of the heated moving plate than the right portion it. This is due to the double vortex creation at the left portion of the plate due to the hydraulic jump.

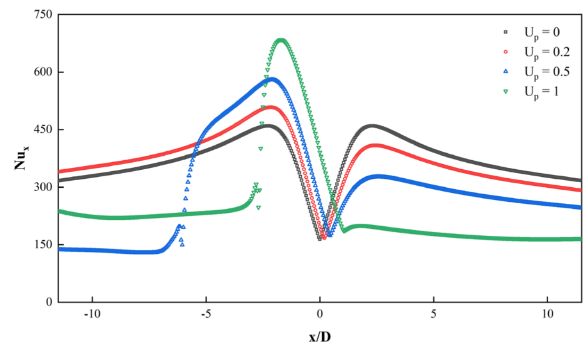

Fig. 7: Distribution of local Nusselt number along the hot moving plate for Re=50,000.

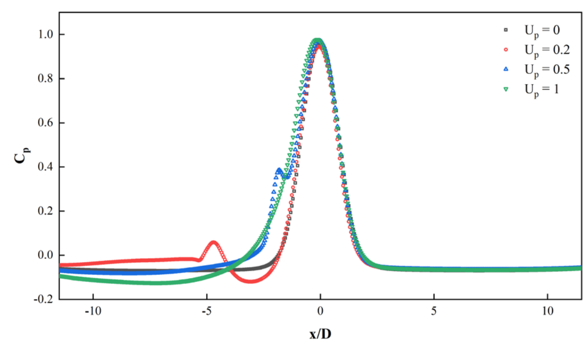

Fig. 8: Pressure coefficient (C_p) along the moving plate at Re = 23,000

From the **Fig. 7** it can be seen that the local Nusselt number distribution gets to a certain peak at $U_p = 0.5$ m/s. It has the maximum peak at $U_p = 0.5$ m/s. These overshot of the local peak Nusselt number is due to the open boundary which imitates a backflow in the fluid domain. Due to this the vortex separation is created and the flow separate from the flat plate resulting a hydraulic jump. Due to this the flow diffuses at the wall region and the local Nusselt number gets higher at the jet impingement region. But this effect produces a non-uniform distribution of temperature at the separation region which will eventually develop thermal stress on the flat plate.

The other hydrodynamic quantity like the pressure distribution on the moving surface is also studied. **Fig. 8** shows the distribution of coefficient of pressure variations at the impingement surface for Reynold's 23,000. The peak C_p occurs at the impingement point of the jet. There is a fluctuation in the C_p value observed for high plate velocity which indicates a sudden acceleration in the flow due to the double rotating vortex formation. And the negative value indicates the flow acceleration in the wall jet region.

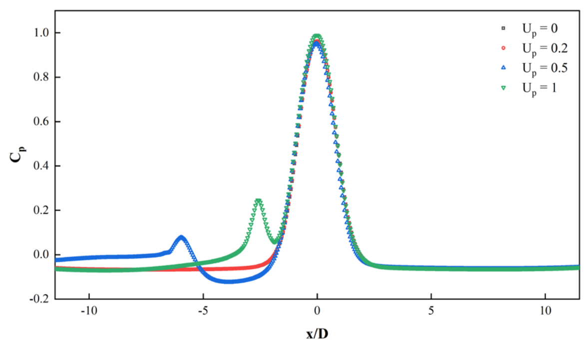

Fig. 9: Pressure coefficient (C_p) along the moving plate at Re = 50,000

Fig. 9 also shows the similar flow phenomena for the jet impingement process. There can be observed a secondary peak of this C_p value at higher plate velocity. This is due to the stronger vortex effect which accelerates the flow in that region and thus higher heat transfer is achieved at that location also.

4. Conclusion

A numerical study was conducted to explore the heat transfer that occurs as a result of jet impingement on a hot moving plate inside a limited domain for two different jet exit Reynold's numbers. In the case of a constant plate velocity, an increase in the Reynolds number, or jet exit velocity, causes a corresponding rise in the local Nusselt number. The linearity of the relationship between the moving plate velocity and the heat transfer characteristics is not maintained. However, there is a subtle relationship that can be noticed in between the jet exit velocity and the moving plate velocity for which the peak value of the local Nusselt number is obtained for a constant Reynold's number. The hydraulic jump effects the local thinning of the thermal and hydrodynamic boundary layers which increases the heat transfer rate at the near impingement region of the flat plate. But this process generates a local concentrated heat transfer region which will develop thermal stress in the impingement surface.

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Nomenclature

ρ : Density, kg/m^3
v : Velocity, m/s
C_p : Specific heat, J/kg.k
k : Thermal conductivity, W/m.k
μ : Kinematic viscosity, kg/m-s
H : Jet to Surface distance, m
U_p : Plate velocity, m
Re : Reynold's number
h : Heat transfer coefficient, W/m^2.K