Flow and heat transfer investigation of a circular jet issuing on different types of surfaces

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Abstract. In the present study, flow and heat transfer characteristics of an impinging jet issuing from a circular cross-sectional pipe on a rectangular plate, convex and concave hemispheres were investigated numerically for various Reynolds numbers and jet-to-plate distances. Steady and three-dimensional Reynolds-Averaged Navier-Stokes equations were solved iteratively utilizing finite volume method. Turbulence model assessment study reveals that the transitional Shear Stress Transport $k-\omega$ turbulence model can be used for such a problem. Regardless of the geometry of the impinging surfaces, two different flow regions were detected around $y/D = 3$. The first region starts from the stagnation point ($y/D = 0$) to the edge of the rectangular plate and hemispheres ($y/D = 3$). In this region, the maximum heat transfer is obtained for the jet impinging on the rectangular plate. However, in the second region that covers the wall jet zone the impinging jet on the convex hemisphere provides the maximum while the concave causes the minimum heat transfer. The most efficient jet-to-plate distance is found as $H/D = 2$ in the first region while heat transfer does not change with distance in the wall jet zone. It was shown that Nusselt number increases with Reynolds number when the jet impinges on the concave and convex hemispheres as observed for the impinging jet on a flat plate.

Keywords. Impinging jet; heat transfer; computational fluid dynamics; turbulence; convex surface; concave surface.

1. Introduction

Impinging jets onto surfaces with various geometric configurations are used in numerous applications that high rates of heat and mass transfers are required; therefore, they are widely employed as an industrial application for cooling of electronics components and turbine blades, drying paper, textiles and food, metal sheet manufacturing and tempering of glass.

Impinging jet flows consist of three different flow zones: stagnation, wall jet, and free jet zones. The free jet zone is a combination of a potential core that depends on the geometry and flow conditions at the pipe exit [1], developing flow and fully developed zones. Potential core zone has significant effects on heat transfer. It was reported that when the jet-to-plate distance is higher than the length of the potential core, the non-dimensional heat transfer rate at the stagnation point shows maxima; however, it makes a secondary peak when the jet-to-plate distance is less than the length of the potential core [1, 2]. Therefore, the

potential core length becomes the crucial parameter for heat transfer. After potential core zone flow develops and spreads in the radial direction. The axial velocity of flow decreases due to dispersion. Then, the flow enters the stagnation region as it reaches the impinging surface. The axial momentum and turbulent normal stresses are transformed into the static pressure in the stagnation region and finally, the wall jet zone emerges due to 90 degrees of rotation of the flow. As the impinging fluid joins to the wall jet, the transition to turbulent flow regime occurs. The existence of the transition to turbulence is verified by a second peak in Nusselt number in the radial distance. However, it depends on both the Reynolds number and jet-to-plate distances. It was presented the transition to turbulence for two different impinging jets cases with Reynolds numbers of 21500 and 42600 [3]. Then, some studies were performed for a relatively low Reynolds number and a secondary peak in Nusselt number was shown that concludes that such a secondary peak in Nusselt number occurs in a wide range of Reynolds number [4].

The flow topology of the jet impinging on a concave hemisphere is not much different from the topology of the

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impinging jet onto a plane surface since it contains the same three regions. The collision of the jets after impingement produces a rather complex flow field that depends on several variables such as the type of the nozzle where the jets issue, jet-to-plate distance, nozzle shape, geometry of the impinging surface and Reynolds number. Since these variables affect significantly both heat transfer and fluid flow characteristics of impinging jets, lots of experimental and numerical studies have been reported so far.

Effects of nozzle shapes and jet-to-plate distance on a semi-circular concave surface for a wide range of Reynolds numbers were reported [5]. It was shown that in comparison with the impinging jet on a plane surface, impinging jet on a curved surface provides a higher rate of heat transfer. Heat transfer from the circular and chevron type impinging jets was compared experimentally for various Reynolds numbers and different jet-to-plate distances from $H/D = 2$ to 10 [6]. It was observed that using the chevron nozzle increases the heat transfer rate due to the development of streamwise vortices. In comparison with the rectangular impinging jet, the elliptic jet array provides higher heat transfer from a surface [7]. Yet, it was claimed that the thermal performance of a circular-shaped orifice is better than that of the non-circular one [8]. When a jet impinges on a hot surface the cooling performance drops rapidly as the fluid moves away from the impingement zone. To come up with such drawback jets are often used in arrays to increase the uniformity of the heat flux distribution [9]. Although relevant literature consists of lots of studies on the impinging jets on the plane surfaces, there are a few studies analyzed geometrically different impinging surfaces. Effects of Reynolds number, jet-to-plate distance, circumferential distance and surface curvature on turbulence characteristics of a slot jet impinging on different semi-circular convex surfaces at Reynolds number ranging from 5600 to 13200 and 12000 were measured [10, 11]. It was concluded that both streamwise and widthwise turbulence intensities along with the free jet centerline increase with the jet-to-plate distance. When slot jets impinge on a convex surface the local Nusselt number demonstrates a peak at the stagnation point for $H/D = 5 \sim 7$ while the peak Nusselt number inversely changes with the distance when slot jets impinge on the concave surface. It was reported that regardless of the H/D ratio a concave surface provides higher Nusselt numbers [12]. Convective heat transfer from a wedge-shaped concave surface revealed that changing the geometry of the impinging surface may improve the heat transfer [13]. Depending on the needs various jet-to-plate distance (H/D) ratios from 0.08 [14] to 20.1 [15] were investigated for various impinging jet configurations.

In the relevant literature, most of the studies have dealt with the impinging jet on flat plates and quite a few studies have analyzed the impinging jet on a curved surface. The primary aim of the present study is to numerically investigate the flow and heat transfer characteristics of an

impinging jet on not only a concave surface but also a convex surface and compare them with the rectangular plate for various Reynolds numbers and jet-to-plate distances.

2. Computational setup

In the present three-dimensional study, the steady-state continuity, Navier-Stokes and energy equations were solved iteratively. The governing equations were discretized by finite volume method that is employed by Ansys Fluent V19.0. The semi-implicit method for pressure linked equations (SIMPLE) algorithm was used for velocity–pressure coupling where the pressure is guessed and velocity is calculated from the momentum equations. The second-order upwind spatial discretization scheme for the convection terms of each governing equation was used. The convergence criterion for the solution of continuity, velocity components, and turbulence quantities was set to 10^{-3} [16]. For the energy equation, it was 10^{-8} .

The collision of the jet with these surfaces produces a rather complex flow field. Therefore, a three-dimensional computational domain was generated to get more realistic results that cannot be predicted by two-dimensional simulations (figure 1). The computational domain consists of a cylindrical pipe and a rectangular prism that covers $-60 < x/D < 60$, $-30 < y/D < 30$ and $0 < z/D < 20$. Such a domain was inspired by the experimental test case of Yao *et al* [17]. The length of the pipe was set to $L = 14D$ to ensure that the flow is fully developed. The dimensions of the rectangular, concave and convex impinging surfaces were kept constant as $D_h = 6D$. The vertical distance between the pipe exit and the impinging surface (H) was non-dimensionalized by dividing it to the pipe diameter ($D = 9$ mm). Effects of this ratio were investigated for three different distances; $H/D = 2, 6$ and 9.

The following boundary conditions were considered:

- Uniform velocity profile was applied to the circular cross-sectional pipe: $w = w_j$, $u = v = 0$.
- Pressure outlet boundary condition was assigned to all surfaces, except the bottom one.
- The no-slip condition was specified to the wall surfaces.
- Uniform wall temperature of 310 Kelvin was applied to the cooling surfaces as a thermal boundary condition.
- Turbulence intensity (I) and pipe diameter (D) were defined at the pipe exit as turbulence specification methods by considering $I = 0.16 \times (Re_D)^{-0.125}$ [18].

Air with a temperature of 298 K was chosen as the working fluid. It was assumed that the density of the air does not change with the temperature [19]. In the present study, simulations were performed for four Reynolds (Re)

Figure 1. Impinging jet onto (a) rectangular plate, (b) convex hemisphere, (c) concave hemisphere in the computational domain in (d) three coordinates and (e) yz -plane for ($H/D = 2$).

numbers ($Re = 4276, 7043, 11600$ and 23000) to analyze its effect on the flow and heat transfer from the surfaces. It should be noted that $Re = 7043$ was considered for flow structure only and the distribution of non-dimensional wall distance, y^+ .

The three-dimensional computational domain was split into small volumes called grids. The region where the jet flow impinges on the surface requires special attention during the meshing process due to large gradients; therefore, relatively denser mesh elements were used towards the impinging surface and the pipe (figure 2). Keeping this approach in mind, different grid resolutions were tested for $Re = 23000$ to ensure that the numerical results are grid-independent. Grid resolutions were compared in terms of Grid Convergence Index (GCI) by considering the Nusselt number. Although at least two grid resolutions can be used for GCI error estimation [20], in the present study three different grid resolutions called coarse, medium and fine were tested. Medium and fine grids were generated by higher refinement ratios than 1.3 as recommended [21]. Since the GCI is also used for numerical uncertainty it can be concluded that the numerical uncertainty in the present study is quite low (0.5 for the fine grid) as it can be seen in table 1. It is worth noting that y^+ was kept below unity at the impinging and bottom surfaces (figure 3). Grid independence study was also performed in terms of velocity and Nusselt number distributions. As shown in figure 4, both the velocity and Nusselt number deviate from the experimental data of Baughn *et al* [22] and Cooper *et al* [23]. Although medium and fine grids provide almost the same Nusselt number distributions, there are some minor discrepancies between them in terms of velocity change,

Figure 3. Distribution of y^+ for impinging jet onto rectangular plate, $H/D = 2$, $Re = 7043$.

particularly after $z/D = 0.38$. However, the medium grid was used in further simulations due to a little difference between the medium and the fine grids.

The governing equations include conservation of mass (Eq. 1), momentum (Eq. 2) and energy (Eq. 3) are shown below.

$$\frac{\partial U_i}{\partial x_i} = 0 \quad (1)$$

$$\frac{\partial (U_i U_j)}{\partial x_j} = -\frac{1}{\rho} \frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j} \left(\frac{\partial U_i}{\partial x_j} - \overline{u'_i v'_j} \right) \quad (2)$$

$$\rho u_i \frac{\partial T}{\partial x_i} = \frac{\partial}{\partial x_i} \left[\frac{\mu}{Pr} \frac{\partial T}{\partial x_i} - \rho \overline{u'_i T'} \right] \quad (3)$$

Figure 2. Grid distributions around (a) rectangular plate, (b) convex hemisphere and (c) concave hemisphere.

Table 1. Mesh properties, $H/D = 2$, $Re = 23000$.

Grid	Number of mesh elements	Refinement ratio	y^+	GCI relative error (%)
Fine	3235512	1.874	<1	0.5
Medium	1726813	1.903	<1	–
Coarse	907128	–	<1	–

Figure 4. Grid independence study for impinging jet on a flat plate in terms of (a) velocity and (b) Nu number distributions, $H/D = 2$, $Re = 23000$.

The fluctuation in the velocity and temperature is represented by u'_i and T' . The term $\overline{u'_i u'_j}$ and Pr in Eq. (3) is the Reynolds stress and the Prandtl number. The Reynolds stress needs to be modeled to close the equations. The Boussinesq approximation that relates the Reynolds stress to the strain rate was used (Eq. 4).

$$-\overline{u'_i u'_j} = \nu_t \left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) - \frac{2}{3} k \delta_{ij} \quad (4)$$

where ν_t and k stands for eddy viscosity and turbulent kinetic energy. δ_{ij} is the kronecker delta. One of the most important parameters in heat transfer is the Reynolds (Re) and Nusselt (Nu) numbers (Eqs. 5 and 6).

$$Re = \frac{u_{bulk} D}{\nu} \quad (5)$$

Figure 5. Comparison of the various turbulence models with the experimental data of [22] and [33], $H/D = 2$, $Re = 23000$.

Figure 6. Comparison of the present study with the experimental data [23] for impinging jet onto a plane surface, $H/D = 2$.

$$Nu = \frac{hD}{k} \quad (6)$$

where u_{bulk} , ν , D , h and k are the bulk velocity of the jet, kinematic viscosity of the fluid, pipe diameter and conduction heat transfer coefficient of the fluid.

Although there are some studies performed by direct numerical simulation (DNS) [24] and large eddy simulation (LES) [25], the majority of the numerical studies on the impinging jet employed Reynolds-Averaged Navier-Stokes (RANS) based turbulence models since both DNS and LES require huge computational resources and longer times. Several turbulence models such as Spalart-Allmaras (S-A) [26], standard k- ϵ [27], she stress transport (SST) k- ω and re-normalization group (RNG) k- ϵ [28], transitional SST and $\overline{v^2} - f$ ($v2f$) [29] and standard k- ω [30] were used for various impinging jet flow configurations. In the present study, among the aforementioned turbulence models only RNG k- ϵ , SST k- ω , transitional SST k- ω , and standard

Figure 7. Temperature contours (in K) and streamlines around (a) rectangular plate, (b) concave hemisphere and (c) convex hemisphere, $H/D = 2$, $Re = 7043$.

$\bar{v}^2 - f$. [31, 32] turbulence models were taken into account to find the most suitable turbulence model. Proposed by [33] as an extension of the standard $k-\epsilon$ turbulence model, the $\bar{v}^2 - f$ model solves two transport equations in addition to the turbulence kinetic energy (k) and its dissipation (ϵ). Claiming that the k is not the appropriate turbulent velocity scale, it introduces a new velocity scale ($\bar{v}^2$) and elliptic relaxation factor (f) that allows the model to scale the damping of turbulence near the walls. This approach makes the model more advantageous since it eliminates the need for damping or wall functions. Hence, it requires lower

computational costs than the other turbulence models. The performances of RNG $k-\epsilon$, SST $k-\omega$, transitional SST $k-\omega$, and standard $\bar{v}^2 - f$ were compared with the experimental data of Baughn and Shimizu [34] and Baughn *et al* [22] in terms of the Nusselt number distribution at $H/D = 2$, $Re = 23000$ for the impinging jet on a flat plate. As shown in figure 5 neither RNG $k-\epsilon$ nor SST $k-\omega$ turbulence models predict the Nu number desirably. Although the $\bar{v}^2 - f$ model is in good agreement with the experimental data, specifically after $r/D = 1.5$, the capability of transitional SST $k-\omega$ in estimating the Nusselt number is much better

Figure 8. Effects of Reynolds (Re) numbers on heat transfer in terms of Nusselt (Nu) number from different impinging surfaces, $H/D = 2$.

not only in the former stations but also in the subsequent stations after $r/D = 1.5$. The weakness of the $\overline{v^2} - f$ turbulence model at higher Reynolds number was also shown in the relevant literature. It was reported that the $\overline{v^2} - f$ turbulence model underestimates the Nusselt number at stations of $r/D < 1.0$ at $Re = 23000$ while it is in good agreement with the experimental data at the lower Reynolds numbers such as $Re = 12000$ [35]. Due to its overall favorable performance, the transitional SST $k-\omega$ turbulence model was utilized in the present study. The performance of turbulence models on a curved surface is controversial. It was shown that the most suitable turbulence model is the RNG $k-\epsilon$ turbulence model for impinging jet on a cylinder [36]. However, it was reported that the SST- $k-\omega$ turbulence model is applicable for impinging jet on both concave and convex surfaces [37, 38]. Similarly, for impinging jets on a flat plate and a concave surface it was proposed that like RNG $k-\epsilon$, the SST $k-\omega$ turbulence model predicts Nusselt number distributions better than the other turbulence models [39]. A brief overview of the tested turbulence models can be found in [1, 40]. The capability of the transitional SST $k-\omega$ turbulence model is shown in figure 6 by comparing changes in velocity with the experimental

data of Cooper *et al* [23]. As it is seen, the present study is in good agreement with the experimental data. It was reported in the literature that for an impinging jet on a flat plate a secondary peak occurs at low jet-to-plate distances, $H/D \leq 2$ as shown in figure 3b [41]. This secondary peak in Nusselt number is due to the increase in the turbulence and indicates the transition from laminar to turbulent flow in the wall jet.

3. Results and discussion

Although the flow topology of the jet impinging on a curved surface is not much different from the topology of the impinging jet onto a plane surface, the physics is little different due to the curvature effect on the flow. Due to the curvature counter-rotating vortices are formed and the momentum transfer and turbulence mixing are increased. When a jet impinges on a surface it causes the formation of complex flow structures. Such flow structure and temperature field developed over the impinging surfaces under consideration are shown in figure 7. It is seen that the recirculation regions develop over the surfaces in different

Figure 9. Nusselt (Nu) number distributions for various impinging distances to the (a) rectangular plate, (b) convex hemisphere, (c) concave hemisphere, $Re = 11600$.

sizes and takes various shapes according to the shape of the impinging surfaces. When the jet impinges on the rectangular plate two symmetrical and moderate-sized recirculation regions occur. Next the two main recirculation regions form and fill the entire vertical space between the jet and the ground surface. They stretch in the y -direction to $y/D = 2.7$. Furthermore, small-sized recirculating regions also

develop at the intersection of the rectangular plate and the ground. Sizes of the recirculation regions become quite large when the jet issues on the convex hemisphere. Unlike the flow fields in and around the rectangular and concave surfaces, there is not any secondary recirculation region on the convex hemisphere. This is due to the smooth connection of the convex surface to the ground surface that prevents the formation of any separation in the main flow. Since there is not any secondary recirculating flow region, the main recirculation region fills the entire flow field. The smallest main recirculating region occurs in the case of the impinging jet on the concave surface where it consists of counter-rotating vortices (CVP). A secondary vortex takes place in the region $3.0 < y/D < 9.0$ next to the CVP. Close to this one, a third but very small-sized vortex occurs. The temperature field and such flow configurations for each case will be discussed in the following sections.

As stated by [41] the prediction of the heat transfer behavior of impinging jets is based on the non-dimensional equations written in terms of non-dimensional numbers; such as Nusselt, Reynolds, and Prandtl; however, only Reynolds and Nusselt numbers were considered in the present study. Local Nusselt number variations with Reynolds numbers along with the rectangular plate, concave and convex hemispheres are illustrated in figure 8 for $H/D = 2$. First of all, it is indicated that regardless of the type of the impinging surface, the Nusselt number, accordingly the heat transfer, increases with Reynolds number for impinging jet onto surfaces regardless of their geometrical shapes. This is not surprising since the flow with a higher Reynolds number has more momentum and brings more energy on the stagnation point. Secondly, independent of the type of the impinging surface, the Nusselt number decreases along with the surface as the impinging fluid moves from the stagnation point because the thermal boundary layer thickens outward [42]. In comparison with the concave and convex hemispheres, the impingement on the rectangular plate provides higher heat transfer rate at any Reynolds number in the region $0 < y/D < 3$ where $y/D = 3$ is the edge of the flat plate and the hemispherical surface. After the station of $y/D = 3$, the cool air flows over the wall jet region and makes the concave impinging surface the most effective in terms of heat transfer. It is believed that the reason is the larger destabilization of the boundary layers and increased turbulent mixing. In the region of $0 < y/D < 3$ the convex hemisphere causes the least heat transfer, then it provides the highest in $3 < y/D < 9$ which is very consistent with the literature [43].

Effects of the jet-to-plate distances (H/D) on the heat transfer from different impinging surfaces are shown in figure 9 by plotting the local Nusselt number variation along the surface. It is seen that at $Re = 11600$ the distribution of the local Nusselt number is almost independent of the jet-to-plate distances for the impinging jet on the rectangular and concave surfaces after $y/D = 3$. In the stagnation point, the highest heat transfer rate is obtained by H/D

Figure 10. Pressure coefficient (C_p) distributions for different jet-to-plate distances, $Re = 23000$.

= 6, regardless of the type of the impinging surface. After that, $H/D = 2$ becomes as effective as $H/D = 6$ and leads the higher heat transfer due to the highest jet momentum. Although the second peak in the Nusselt number is observed for all jet-to-plate distance when the jet impinges on the convex hemisphere, such a peak does not occur for the other types of the impingement surfaces at $Re = 11600$. As it is shown in figure 7 the second peak in the Nusselt number occurs at $Re = 23000$ at any type of the impinging surface which implies that as Reynolds number decreases a second peak in the Nusselt number does not occur when the jet impinges on the rectangular plate and concave hemisphere. The second peak in Nusselt number distributions shows the transition from laminar to the turbulent flow. It was claimed that the optimum jet-to-plate distance coincides with the length of the potential core region and the maximum Nusselt number is obtained beyond this region in which an interaction occurs between the jet velocity and turbulence intensity [44]. As it is seen in figure 8,

regardless of the type of the impinging surface, the maximum heat transfer coefficient is obtained at $H/D = 6$.

Figure 10 shows the pressure coefficient (C_p) distributions for various jet-to-plate distances since pressure distribution affects the flow structure and heat transfer. A Gaussian profile that is independent of the jet-to-plate distance and type of the impinging surface is seen where the pressure coefficient at the stagnation point is the maximum and then decreases gradually as the flow spreads away from the impinging surfaces. Since subatmospheric regions occur along y -direction the pressure coefficients reduce to negative values that represent the acceleration of the flow along the impingement surfaces. The maximum value of the pressure coefficient changes with the jet-to-plate distance where the maximum pressure coefficient is obtained at the $H/D = 2$ regardless of the type of the impinging surface. However, due to the wider effective cooling area of the jet on the impingement surfaces higher jet-to-plate distances provide higher pressures [45]. The

pressure distributions of the impinging jet on the different surfaces depending on the jet-to-plate distance as reported for an impinging jet on a convex surface [16]. As aforementioned, the maximum heat transfer is obtained at the stagnation point where the pressure coefficient is the highest.

4. Conclusions

The present numerical study reveals the flow and heat transfer characteristics of an impinging cold jet on a hot rectangular plate, convex and concave hemispheres for the Reynolds numbers of 4276, 11600 and 23000 and jet-to-plate distances of $H/D = 2, 6$ and 9 . The comparisons with the experimental data found in the open literature revealed that the impinging jet flow simulations can be performed with the transitional SST $k-\omega$ turbulence model since there is a good agreement between them. A direct correlation between Reynolds and Nusselt numbers is found not only for impinging jet on the rectangular plate but also for concave and convex hemispheres. Regardless of the type of the impinging surfaces, the Nusselt number increases with Reynolds number. Like the Nusselt number, the maximum pressure coefficient is obtained at the stagnation point which implies the highest heat transfer. In comparison with the concave and convex hemispheres, impinging on the rectangular plate provides the highest heat transfer rate in the region $0 < y/D < 3$. After $y/D = 3$, using the convex hemisphere as the impinging surface causes the highest while using the concave surface induces the lowest heat transfer. Among three vertical jet-to-plate distances $H/D = 2$ provides the highest heat transfer rate in the stagnation region.

Nomenclature

D	jet diameter (m)
D_h	hydraulic diameter (m)
H	distance between the pipe and the impinging surface (m)
i, j	1, 2, 3
I	turbulence intensity (–)
k	turbulence kinetic energy (m^2/s^2)
K	thermal conductivity (W/mK)
L	length of the pipe (m)
Nu	Nusselt number (–)
number	
P	pressure (N/m^2)
Re	Reynolds number (–)
number	
T	temperature (K)
u, v, w	velocity components in x-, y- and z-directions, respectively (m/s)
x, y, z	streamwise, spanwise and normal directions, respectively (m)

Greek letters

δ_{ij}	Kronecker delta (–)
ε	dissipation rate of turbulence kinetic energy (m^2/s^3)
μ	dynamic viscosity of the fluid (kg/m.s)
ν	kinematic viscosity of the fluid (m^2/s)
ρ	density (kg/m^3)
ω	turbulence specific dissipation rate

Abbreviations

CFD	computational fluid dynamics
CVP	counter-rotating vortices
DNS	direct numerical simulation
GCI	grid convergence index
LES	large eddy viscosity
RANS	Reynolds-averaged Navier-Stokes
RNG	re-normalization group
SIMPLE	semi-implicit method for pressure linked equations
SST	shear stress transport

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