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Conjugate Effect on the Heat Transfer Coefficient

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ABSTRACT

A transient computational fluid dynamics (CFD) investigation to determine the conjugate heat transfer (CHT) effect on the stagnation and local thermal characteristics due to an impinging air jet has been carried out in this study. It is shown that the thermal characteristics are influenced by the thermal conductivity and thickness of the impingement target. The CHT process enhances the local convective heat transfer at fluid-solid interface up to a certain redial extent from the stagnation point compared to the process with no CHT. The surface temperature distribution becomes more uniform as the thermal conductivity of the disc increases due to the enhancement of the conductive heat transfer inside the metal. The objective of the current study is to numerically examine the effect of wall conduction and thermal boundary strength on the convective heat transfer coefficient due to the jet impingement process. Numerical simulations, using the CHT approach to couple the fluid-solid heat transfer, are performed for fully-developed circular air jet, impinging perpendicularly onto a heated flat plate with different thicknesses and materials. A different jet Reynolds numbers and boundary heat fluxes are employed in the current study to provide an improved picture for the convection mechanism.

Keywords:
CFD simulation; Conjugate heat transfer; Nusselt number; Jet impingement; Turbulent model

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1. Introduction

Jet impingement heat transfer is commonly used in various engineering and industrial applications. Due to its high localized heating and cooling rates, a turbulent jet of gas or liquid directed to the target can efficiently heat up or cool down a particular region. When an axisymmetric jet strikes a plane surface, a very thin viscous layer forms on the surface normal to the impingement axis. This layer exhibits little resistance to heat flow, where the convective heat transfer coefficient can reach a large value. Following impingement, the flow resembles a wall jet which spreads thinner as it travels radially, i.e., the thickness of the liquid film adjacent to the wall decreases with radius. This decrease brings the growing boundary layer into contact with the surface of the fluid film, at which point the fluid film thickness begins to increase at larger radii due to the viscous drag, which slows down the flow and thickens the liquid layer.

Nasif et al. [1-3], Steven and Webb [4], Lee and Lee [5] and Liu et al. [6] have extensively studied the thermal and flow characteristics associated with liquid jets impinging on surfaces. These studies are in relatively good agreement with one another. The maximum heat transfer coefficient (HTC) due

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to the jet impingement occurs in a small region around the stagnation point [3]. Nasif et al. [1] have used CFD simulations to show that for a given Reynolds number, the temperature distribution on the impinging surface will be more uniform for larger nozzles compared to smaller nozzles. Smaller nozzles provide more efficient convective cooling in the region around the stagnation point, while the larger nozzles cool the surface more uniformly. A numerical investigation to evaluate the effect of nozzle geometry on thermal characteristics for long jet impingement has been undertaken by [7]. The investigation showed that the effect of nozzle geometry is insignificant on thermal characteristics. The viscosity tends to normalize the jet velocity profile in the radial directions for long jets after some distance from the nozzle exit, which results in a constant velocity gradient at the stagnation point. Experiments were performed in [8] to study the characteristics of an impinging turbulent jet onto a fixed surface subjected to constant heat flux, using different nozzle diameters and a wide range of Reynolds number. The investigations showed an obvious dependence of the stagnation zone Nusselt number on Reynolds number, Prandtl number and velocity gradient and less dependency on nozzle to plate spacing. Zhu et al. [9] investigated the wall effect on the HTC of an impinging jet. They concluded that the CHT approach redistributes the boundary heat flux and changes it from a uniform heat flux boundary to a nearly isothermal boundary. The heat redistribution is also driven by the non-uniform distribution of the HTC on the impinged surface. The degree of heat redistribution is related to both conductive thermal resistance in the solid and convective thermal resistance at the interface. The study also revealed that the CHT prediction may have a negative impact on the local Nusselt number (Nu), indicating a decrease in the local Nu as the thermal conductivity of the solid decreases. Mensch & Thole [10] used a conjugate heat transfer approach to account for the combined effects of both internal and external cooling. The geometry that was employed in the latter study is a turbine blade endwall that includes impingement, film cooling and heat conduction through the endwall. The conclusion from this study revealed that internal HTC of impingement geometries is sensitive to geometric parameters, while the average temperature of the endwall external surface is not particularly sensitive to geometric parameters.

In all previous studies, the effects of the nozzle size, jet Reynolds number, nozzle-to-target distance and working fluids on the thermal characteristics were extensively investigated. However, the conjugate effect has not been adequately addressed in these studies. Conjugate heat transfer (CHT) is a crucial issue in many engineering problems, which can be examined in different ways. Analytical approaches generate good results to identify the main parameters of the problem and to verify the codes. However, the applications of the analytical methods are restricted to very simple configurations [11-13]. Experiments, which are an alternative approach to the analytical methods, are considerably expensive and cannot be relied on in the industry. The modern computational CHT was developed after computers came into a broad application to replace the empirical expressions of proportionality of heat flux to temperature difference with heat transfer coefficient (HTC). The state-of-the-art of the computational method involves coupling the conduction in the solid and convection in the fluid to predict the HTC at the interface. The coupled approach is more reliable and common than a decoupled solution [14]. In the computational CHT approach, two separate simulations are set up, one for fluid analysis and another for solid thermal analysis. Assuming the temperature distribution on the wall boundary, the fluid flow problem is solved to determine the local HTC distribution on the wall. The HTC distribution with the reference temperature is applied to the solid thermal simulation to re-evaluate the temperature distribution in the solid. The wall temperature distribution predicted by the solid thermal analysis is fed back to the transient flow simulation and applied as a wall boundary condition to re-evaluate the modified HTC distribution at the interface. The iteration process continues until the solution is obtained with a suitable accuracy.

The objective of this study is to numerically investigate the effect of the conjugate heat transfer process on the thermal characteristic. For this end, CFD simulations of fully-developed circular air jets impinging orthogonally on a heated flat plate with different thicknesses are employed. The stagnation and local thermal characteristics are compared for the different cases in this study.

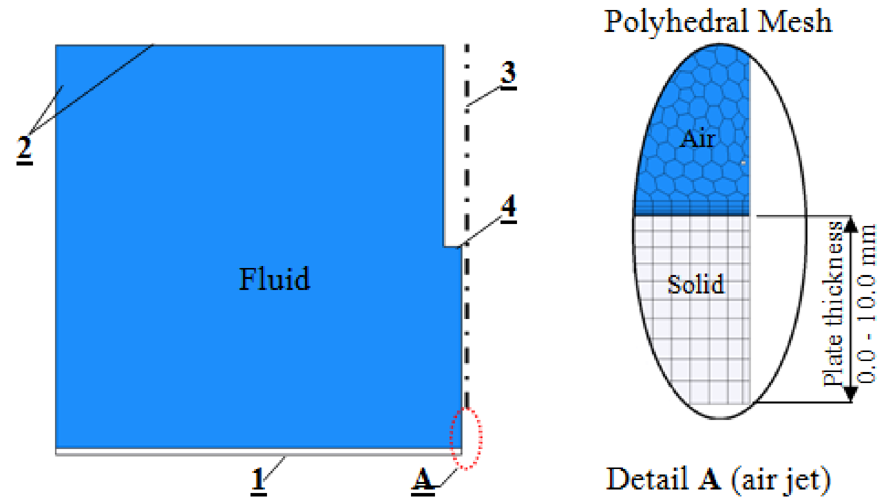
2. Problem Modelling

Figure 1 shows the computational domain with relevant boundary conditions. The different cases are simulated using CD-adapco's STAR-CCM+ commercial code with an unstructured polyhedral mesh for the fluid domain. The major advantage of polyhedral cells is that they generally have many neighbours, so gradients of the variable at cell centers can be much better approximated compared to other mesh types. The solid part in the computational domain is created by extruding the interface to a certain thickness (for the CHT simulations), hence this region includes structured elements as shown in Figure 1. The governing equations for transient analysis include continuity, momentum and energy equations. Each of these equations can be described in a general way by the transport of a particular scalar quantity ϕ, represented in a continuous integral form as [15]:

$$\frac{\partial}{\partial t}\int_{\mathrm{CV}}^{0}\rho\phi d\mathrm{V}+\oint_{A}^{0}\mathbf{n}.(\rho\phi\mathbf{u})dA=\oint_{A}^{0}\mathbf{n}.\left(\Gamma_{\phi}\boldsymbol{\nabla}\phi\right)dA+\int_{\mathrm{CV}}^{0}S_{\phi}\,d\mathrm{V} \tag{1}$$

where, CV is the control volume, A is the surface area of the control volume, $\mathbf{n}$ is the unit outward normal vector to the surface element dA, $\mathbf{u}$ is the velocity vector and ρ is the density. The terms in equation (1), from left to right are, the rate of change the property ϕ in the control volume, the rate of change the property ϕ due to the convection flux across the boundaries of the control volume, the rate of change the property ϕ due to the diffusive flux across the boundaries of the control volume, and the source term. The source term in equation (1) contains the effects of the pressure gradient and all types of body forces. The set of transport equations is obtained by selecting appropriate expressions for the diffusion coefficient $\Gamma_{\emptyset}$ and source term S_{ϕ} and setting the variable ϕ in equation (1) to velocity vector components for momentum equations, and i for energy equation, where i is the internal energy of the fluid or solid. The integral form of the mass conservation equation can also be obtained from equation (1) by setting $\phi = 1$ and the source term $S_{\emptyset} = 0$.

Grids independent study was carried out in the earlier stage to select the optimum mesh count. The criteria for choosing the cell count in the current study are based on the validation process, i.e., the numerical results for many grids and many parameter settings were checked and compared with experimental results. Prism layers are clustered at the jet trajectory and the fluid-solid interface to better resolve the wall effect, producing a dimensionless wall distance value of $y^+ < 3.0$ at the solid-fluid interface. First-order implicit time marching and second-order spatial differencing are used to discretize the governing equations, within a finite volume framework. The *k-ω* SST eddy viscosity model has been selected as the turbulent model in this study. The CHT is used to couple the heat transfer solution between fluid and solid. The time step for the simulations was set at $1\text{x}10^{-3}$ s with twenty internal iterations. Pipe nozzle sizes of d = 25.0 mm, with nozzle-to-target distance h/d = 6.0 and various bulk velocities at 20°C, are used in the study. The fully-developed velocity profile at the nozzle exit is extracted from a separate simulation and mapped at to ensure a fully developed velocity profile, pipes with a length to diameter ratio of $L/d = 50$ are used in separate simulations. The effect of the conjugate heat transfer (CHT) is investigated by using three different materials, i.e., copper (Cu), aluminum (Al), and stainless steel (316SST). The thermal properties of these materials are given in Table 1. Two plate thicknesses are used in the study, i.e., 5.0 mm and 10.0 mm while different heat fluxes are employed in the simulations as a thermal boundary (along boundary 1 in Fig. 1). The computational results of the CHT simulations are compared with each other and with the computational results from the jet impingement with no CHT, i.e., plate with zero thickness, to investigate the conjugate effect on the HTC. In the current study, the computational results are considered to have converged when the continuity, momentum and energy scaled residuals fall below 10^{-6}.



1: Constant heat flux; 2: Pressure outlet; 3: Axisymmetry boundary; 4: Nozzle exit

Fig. 1. Computational domain with relevant boundary conditions

Table 1
Physical properties of the investigated metals

	Density ρ (kg/m^3)	Thermal Conductivity k (W/m.K)	Specific Heat c_p (J/kg.K)
Copper (Cu)	8940	398	386
Aluminum (Al)	2702	237	903
Stainless Steel (316SST)	7990	15.4	500

3. Validation

The normalized local Nusselt number from the CFD simulations is compared with experimental data [5] at different radial locations from the stagnation point as shown in Fig. 2. In this figure, the Nu_0 and Nu represent the stagnation point and local Nusselt number, respectively. The Nusselt number is calculated based on the nozzle diameter and the air temperature at the nozzle exit, i.e., 20 ˚C. Three jet Reynolds numbers for nozzle diameter of d = 25.0 mm and h/d = 6.0 are used in the validation process, i.e., Re = 5000, 15000 and 30000. The validation process is performed by neglecting the conjugate effect to mimic the experiment setup [5], i.e., the jet impinges a flat plate with zero thickness. The difference between the computational and experimental results increases with the Reynolds number. It is shown that the computational model can reproduce the experimental data with a maximum difference of less than 10% for Re = 30000. Therefore, the present simulations can satisfactorily predict the heat transfer performance of the impinging jet. The validation process was also performed for h/d = 4.0 and 10.0; the results were comparable to what is shown in Fig. 2.

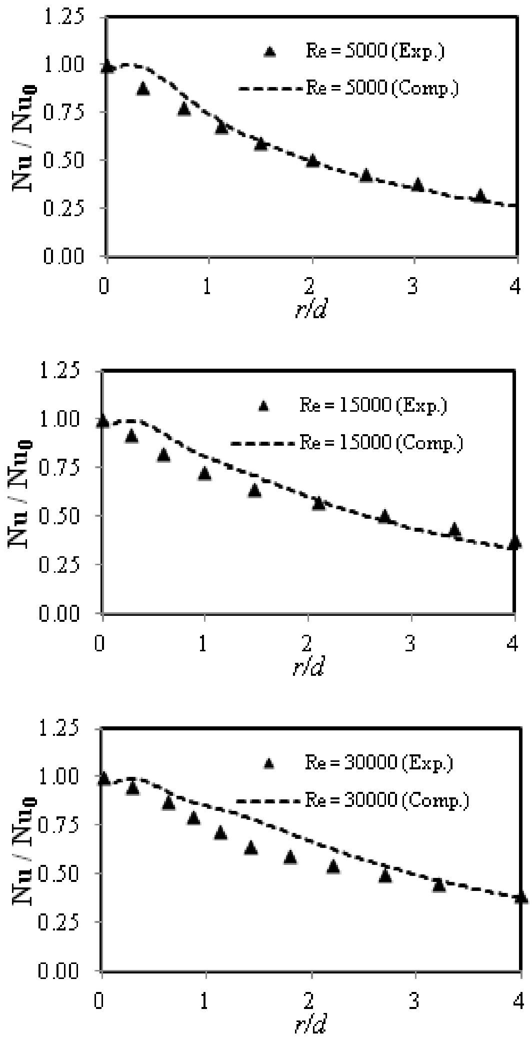


Fig. 2. Comparison between computational and experimental results at *h/d* = 6.0

4. Results

In this section, the jet Reynolds number of Re = 20000 and nozzle-to-target distance at *h/d* = 6.0 are used as sample operating condition to present the results. A constant heat flux of q_B = 1000 & 1500 W/m^2 are used as a thermal boundary condition at the bottom surface of the plate (boundary 1 in Fig. 1). The case of no CHT process is used as the benchmark to evaluate the relative thermal performance of various conjugate scenarios.

When the solid plate has a finite thickness, the conductive heat transfer inside the solid has an influence on the convective heat transfer from the plate surface. The CHT process will act to redistribute the uniform boundary heat flux inside the solid and make a difference not only at stagnation, but also in the local Nusselt. As the plate thickness increases, the thermal resistance in the radial direction $(R_r\ \alpha\ 1/\kappa.t_p)$ decreases, while the thermal resistance in the axial direction $(R_a \alpha\ t_p/\kappa)$ increases. Therefore, the conductive heat transfer in the radial direction towards the impingement point (heat sink) increases with the plate thickness, while the axial conductive heat transfer towards the surface (interface) decrease as the plate thickness increases. The boundary heat flux will not remain uniform at the interface as in the case of the zero plate thickness. In the convective heat transfer problem that involves a CHT process, the HTC profile at the interface depends on the thermal properties and thickness of the plate beside the other flow parameters. Figure 3 shows the conjugate effect on the normalized local Nusselt number of the jet impingement process. For all three materials that were used in the simulation (see Table 1), the Nu/Nu_0 profiles deviate downwards from the one with no CHT process as shown in Fig. 3a. The thermal conductivity of the metal has an effect on the CHT process and subsequently on the normalized Nu/Nu_0 profile. The profile is shifted up towards the profile with no CHT as the metal thermal conductivity decreases (316SST). The effect of the disc thickness is presented in Fig. 3b. The conclusion that can be drawn from this figure is that the disc thickness has insignificant effect on the Nu/Nu_0 profiles for the cases that are used in the current study.

In the current simulation, a range of heat fluxes is used as a thermal boundary condition to investigate the effect of thermal boundary condition and CHT on the Nu_0 as shown in Fig. 4. The subscripts (*B*) and (*T*) in Figs. 4 & 5 represent the heat fluxes at the bottom (constant heat flux boundary) and the top (convective heat flux at the interface) surfaces of the plate, respectively. These fluxes are identical for jet impingement onto a plate with zero thickness, i.e., $q_B = q_T$. It is clearly shown in Fig. 4 that for a given heat flux; the Nu_0 descends with the thermal resistance of the metal. In jet impingement problems, the difference in the Nu_0 is more obvious at higher heat fluxes for the CHT process compared with a zero thickness plate. The Nu_0 profile that implies a CHT process approaches the one with no CHT process as metal thermal conductivity decreases (316SST).

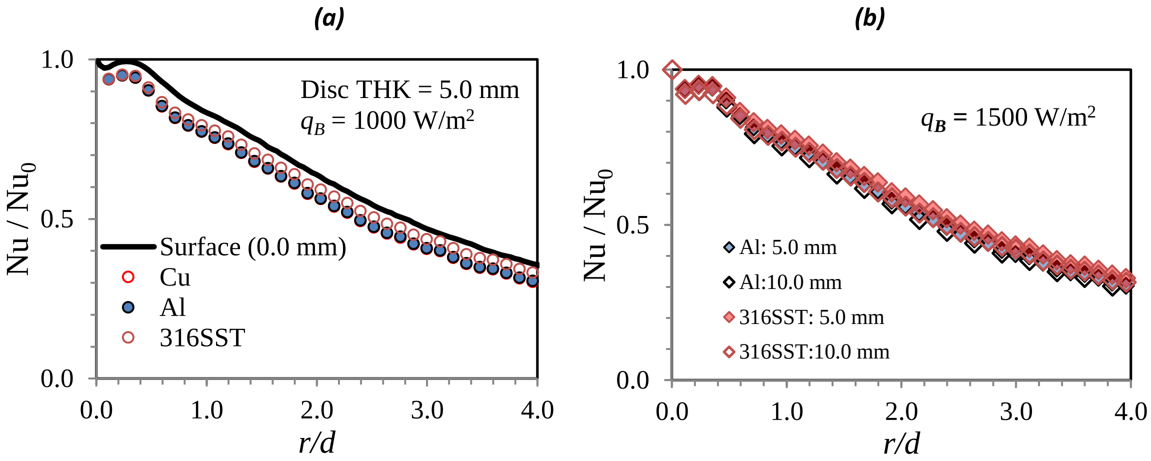


Fig. 3. The effect of (a) metal type, (b) metal thickness on the conjugate heat transfer (CHT)

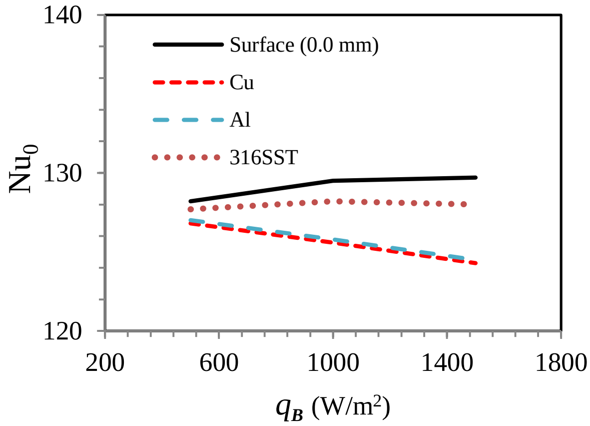


Fig. 4. The effect of the CHT process on stagnation point Nusselt number (Nu_0)

One of the CHT consequences is to enhance the conductive heat transfer in the radial direction towards the stagnation point. This is due to two combined effects: (i) reduction in the radial thermal resistance (R_r) of the solid, and (ii) increase in the axial thermal resistance (R_a) of the solid. Therefore, this will act to improve the local convective heat transfer at the stagnation point as shown in Fig. 5. In this figure, the local convective heat flux is given for the same operating conditions but different metals (316SST). The convective heat flux (q_T) profile becomes more uniform as $R_a \rightarrow \infty$, and becomes more non-uniform as $R_a \rightarrow 0$ with distinct maxima at the stagnation point. The local convective heat transfer profile at the solid-fluid interface drops below the uniform profile with no CHT process at downstream location of r/d = 6.0 for all metals that are used in the current simulation.

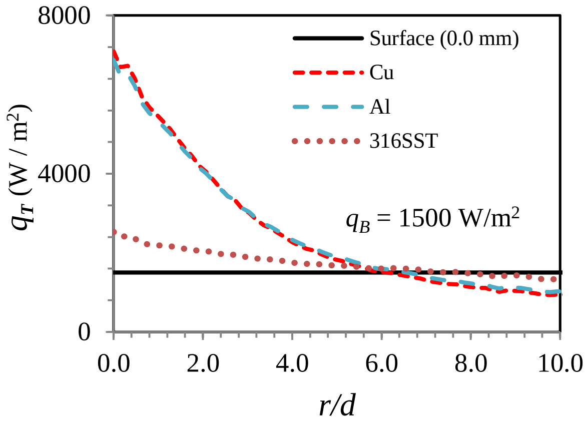


Fig. 5. The effect of the CHT process on local convective heat flux, Disc THK = 5.0 mm

The effect of the CHT process on the minimum disc temperature (T_{min}) is illustrated in Fig. 6. In this figure, two disc thicknesses are used over a wide range of boundary heat flux to investigate the combined effect of thermal boundary and CHT process on the T_{min}. The CHT acts to increase the minimum temperature at the stagnation point for all metals that are used in the simulation. The minimum temperature increases with the metal thermal conductivity (Al & Cu), where the maximum convective heat transfer is taken place. The disc thickness effect on the T_{min} is more apparent for the metal with lower thermal conductivity (316SST) as shown in Figure 5. The T_{min} increases with the thickness for the metal with lower thermal conductivity, while the thickness has insignificant effect for the metal with higher thermal conductivity as shown in Fig.6.

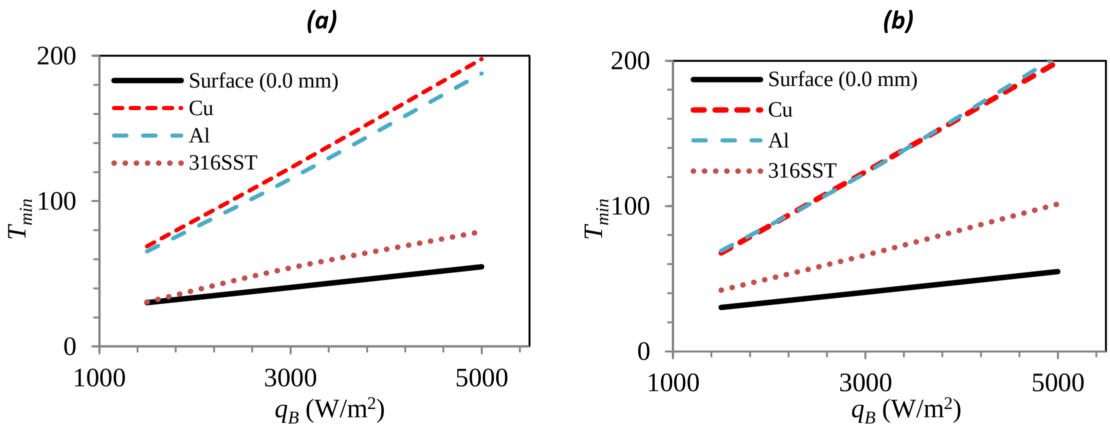


Fig. 6. The effect of the CHT process on the stagnation point temperature (a) disc THK = 5.0 mm, (b) disc THK = 10.0 mm

4. Conclusions

A transient CFD study has been carried out to evaluate the CHT effect on the thermal characteristics of the jet impingement process. The following results are concluded from the current study:

- The CHT process is governed by both components of the metal thermal resistance.
- The metal thickness and type have an effect on the CHT process. The Nu/Nu_0 profile is shifted downwards from the one with no CHT as the thermal conductivity of the metal increases.
- For a given boundary heat flux, the Nu_0 decreases as the thermal conductivity of the metal increases. The decrease in the Nu_0 is more obvious at higher heat fluxes.
- The CHT process enhances the local convective heat transfer at fluid-solid interface up to a certain redial extent from the stagnation point compared to the process with no CHT.
- The CHT process acts to increase the minimum temperature at the stagnation point. The stagnation point temperature increases with the metal thermal conductivity.

References

[1] G. Nasif, R.M. Barron, R. Balachandar and O. Iqbal. Simulation of jet impingement heat transfer. *Proceedings of ASME 2013 Internal Combustion Engine Division (ICED), Fall Technical Conference, Paper No. ICEF2013-19050,* Dearborn, MI, USA, 2013.

[2] Nasif, R.M. Barron and R. Balachandar. Heat transfer due to an impinging jet in a confined space. *Trans. ASME*, *J. Heat Trans.*, 136(11), 2014; doi: 10.1115/1.4028242.

[3] G. Nasif, R.M. Barron and R. Balachandar. Simulation of jet impingement heat transfer onto a moving disc. *Int. J. Heat Mass Transfer*, 80:539-550, 2015.

[4] X. Liu, L.A. Gabour, J.H. Lienhard, Stagnation-point heat transfer during impinging of laminar liquid jets: Analysis including surface tension. J. Heat Transfer, 115, pp. 99-105, 1993.

[5] J. Lee and S.J. Lee. Stagnation region heat transfer of a turbulent axisymmetric jet impingement. *Experimental Heat Transfer*, 12(2):137-156, 1999.

[6] X. Liu, J.H. Lienhard and S. Lombara. Convective heat transfer by impingement of circular liquid jets. *Trans. ASME*, *J. Heat Trans.*, 113(3):571-582, 1991.

[7] G. Nasif, R.M. Barron and R. Balachandar. Jet impingement heat transfer. International Conference on Advancements and Futuristic Trends in Mechanical and Materials Engineering, Kapurthala, Punjab, India, 2013.

[8] J. Stevens, and B. W., Webb. Local heat transfer coefficients under an axisymmetric, single phase liquid jet. J. Heat Transfer, Vol. 113(1), pp. 71-78, 1991.

[9] X.W. Zhu, L. Zhu, J.Q. Zhao, An in-depth analysis of conjugate heat transfer process of impingement jet. Int. J. Heat and Mass Transfer, 104, pp. 1259-1267, 2017.

[10] A. Mensch, K.A. Thole, Conjugate heat transfer analysis of the effects of impingement channel height for a turbine blade endwall, Int. J. Heat and Mass Transfer, 82, pp. 66-77, 2015.

[11] A. Pozzi and R. Tognaccini, Time singularities in conjugated thermo-fluid-dynamic phenomena, *J. Fluid Mechanics*, 538: 361-376, 2005.

[12] A. Pozzi and R. Tognaccini, Coupling of conduction and convection past an impulsively started semi-infinite flat plate, *Int. J. Heat and Mass Transfer*, 43(7): 1121-1131, 2000.

[13] B. Fourcher and K. Mansouri, An approximate analytical solution to the Graetz problem with periodic inlet temperature, *Int. J. Heat and Fluid Flow*,18(2): 229-235, 1997.

[14] E. Radenac, J. Gressier and P. Millan, Methodology of numerical coupling for transient conjugate heat transfer, *Computers & Fluids*, 100: 95-107, 2014.

[15] H.K. Versteeg and W. Malalasekera, An Introduction to Computational Fluid Dynamics: The Finite Volume Method, 2nd ed., Pearson Education Ltd., Harlow, UK, 1995.