CHAPTER 1

Introduction and Literature Review

A directed stream of fluid emerging from the nozzle exit into an ambient fluid is known as a jet. The ambient fluid may be stagnant or moving depending on the application. The flow at the exit of the nozzle develops into a parabolic velocity profile with moderate turbulence after passing through a tube. Somewhat flat velocity profile with less turbulence can be obtained by applying a pressure difference across a thin and flat orifice. When the fluid from the nozzle mixes with the ambient fluid (stagnant or moving) freely without the effect of any solid surface, it is known as an unbounded jet or a free jet. Presence of a wall makes the flow more complicated. In the wall bounded flow, there is a formation of a viscous sublayer very close to the wall where momentum and energy transport takes place due to viscous mechanism. Wall jet, offset jet, a combination of wall jet and offset jet (known as dual jet), impinging jet are some of the examples of wall bounded jet. The fluid from the nozzle flowing tangential to a solid surface is known as a wall jet. When the axis of the jet is parallel to and offset from a solid surface, it is known as an offset jet. A directed fluid coming out from a nozzle and impinging on a solid wall normally or obliquely is known as an impinging jet. The flow in an impinging jet can either be laminar or turbulent depending on the Reynolds number. It is found that impinging jets become turbulent with Reynolds number beyond 2000 (Marple et al. (1974), Deshpande and Vaishnav (1982), Jambunathan et al. (1992)).

A schematic diagram of an impinging jet has been shown in Figure 1.1. The three main regions in an impinging jet are free jet region, stagnation region and wall jet region. The jet, after emerging from the nozzle, passes through the surrounding fluid as a free submerged jet. The free jet region may have three different sub-regions, viz. potential core region, developing region and developed region. The flow may be either developing or fully developed by the time it reaches the impingement plate depending on the nozzle-to-plate spacing. Shear layers on the edges of the jet are formed due to the presence of velocity difference between the jet and its surrounding thus transferring momentum laterally outward. The jet entrains the surrounding fluid due to which the jet mass flow increases and it loses energy. The velocity profile of the jet gets widened laterally. The interior region between the progressively widening shear layers, known as potential core region, remains unaffected from momentum transfer where the inertia force is more dominant than the viscous force. The jet centerline velocity does not decrease in the streamwise direction within the potential core region due to the absence of viscous effect. The viscous effect penetrates to the jet centerline at the end of the potential core region beyond which the jet centerline velocity starts to decrease substantially and this region is known as the developing region. The developing region is characterized by the presence of velocity profiles resembling a Gaussian curve that gets wider and shorter in the streamwise direction (Zuckerman and Lior (2006)). The flow beyond the developing region can be characterized by the self similar velocity profiles and the region is known as the developed region. The lengths of these sub-regions depend on the shape of the jet, nozzle exit conditions like velocity profile, turbulence and Reynolds number and nozzle-to-plate spacing. The fluid flow is subjected to strong curvature and high strain due to the presence of impingement wall. The fluid gets decelerated due to an unfavorable pressure gradient in the stagnation region. The fluid accelerates along the impingement wall after the impingement. The impingement region extends to the point beyond which the pressure gradient on the target plate is zero. The flow, after leaving the stagnation region, proceeds as wall jets on both sides and decelerates in the flow direction with the boundary layer thickness increasing monotonically. The impingement plate motion makes the flow more complex by getting the strong shear regions come in to picture (Zumbrunnen (1991), Chattopadhyay and Saha (2003)). The behavior of the two wall jets become different to each other depending on their flow with reference to the impingement plate movement.

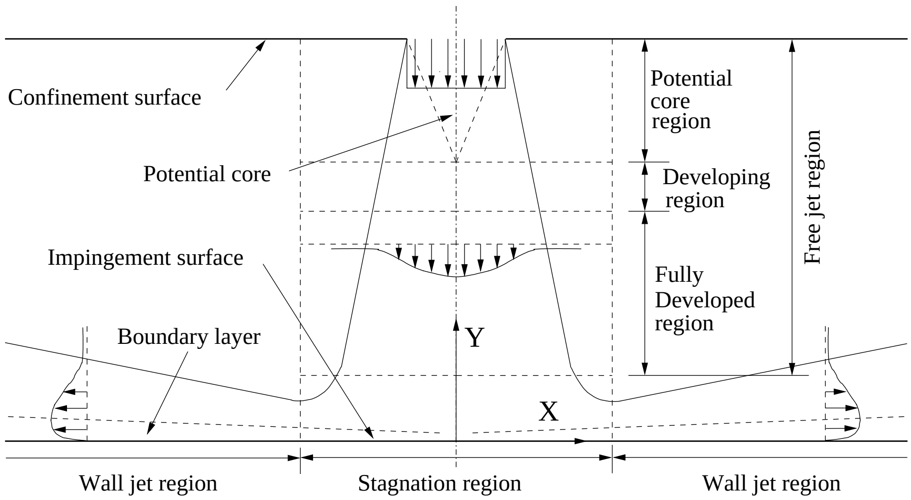

Figure 1.1: Schematic diagram of an impinging jet.

Investigation of impinging jet is one of the important problems in the field of heat and mass transfer due to its industrial and fundamental importance. The presence of very high convective heat and mass transfer in the impingement region (Antonia et al. (1983), Viskanta (1993), Sakakibara et al. (1997)) enable them to be used in industrial applications where large local heat transfer is required. Impinging jets have superior heat and/or mass transfer characteristics in single phase heat transfer methods compared to those obtained for fixed mass and momentum flux rates at the nozzle exit flowing parallel to the target surface. The flow required for an impinging jet device may be two orders of magnitude smaller than that required for a cooling application using a free wall-parallel flow for a given heat transfer coefficient (Zuckerman and Lior (2005)). The impinging jet has the ability to fine control the heat transfer rate by changing the operating conditions like nozzle exit velocity, nozzle size and shape, angle of inclination of jet impingement, nozzle-to-surface spacing etc. This has led to its increasing use in the industry for heating, cooling, and drying of surfaces. Tempering operations, turbine blade cooling, drying of paper, lumber and textiles or other thin films, secondary cooling of continuous casting of steel, cooling of high power density electronic components, preparation of printed wiring boards, baking and freezing of food items, de-icing of aircraft wings, heating of optical devices for defogging etc. are some of the industrial applications of impinging jets due to their highly favorable heat and mass transfer characteristics. The air curtain devices in HVAC (heating ventilation and air conditioning) employ impinging jets to protect a region from its surroundings. STOVL (short take off and vertical landing) aircraft produces vertical air jets to generate lifting force at zero/low forward speed. Analysis of vertical air jets impinging on the ground is essential for the design of this kind of aircraft. In the cooling of high power density microelectronics in compact enclosures, the fluid emerges from the orifice present in the plate held parallel to the target plate. The plate having the orifice in it acts as the confinement plate and the fluid is made to flow between the two plates after the impingement. Presence of a confinement wall makes free jet behavior coupled with the fluid behavior in a channel flow resulting in a more complicated flow. The strong streamline curvature, recirculation, boundary layer development along the impingement surface are some of the features that make the flow very complex. Numerous studies on fluid flow and heat transfer in an impinging jet could not defy from the fact that this field still remains one of the active domains of research and acts as a benchmark problem to evaluate the performance of different turbulence models due to its complex nature of fluid flow within a somewhat simple geometry.

1.1 Literature Review

Some of the review works on impinging jets have been done by Jambunathan et al. (1992), Viskanta (1993), Weigand and Spring (2011) and Dewan et al. (2012). Jambunathan et al. (1992) have reviewed on the experimental works of single circular jet impinging orthogonally onto a plane surface for nozzle-to-plate distances from $1.2-16$ nozzle diameters. The nozzle exit Reynolds number ranged from 5000 to 124000. The Nusselt number was expressed as a function of nozzle exit Reynolds number raised to a constant exponent in existing correlations found in literature for local heat transfer coefficient. However, the authors suggested that this exponent should be a function of nozzle-to-plate spacing and of the radial distance from the stagnation point. They also suggested that the Nusselt number is independent of nozzle-to-plate distance up to a value of 12 nozzle diameters beyond radii of six nozzle diameters from stagnation point. Viskanta (1993) reviewed on heat transfer characteristics of single and multiple isothermal turbulent air and flame jets impinging on surfaces. He identified areas in need of research such as cross flow and simultaneous motion of the impingement surface, curved impingement surface along with the emphasis on physical phenomena. Weigand and Spring (2011) have done a review on the heat transfer characteristics of systems of multiple impinging air jets where they compared the results with those of single impinging jets. They also analyzed the suitability of different CFD (computational fluid dynamics) tools in predicting the heat transfer rate for multiple impinging jets system. Dewan et al. (2012) reviewed on the current status of computation of turbulent impinging jet. Due to the the lack of generality in the reported data they could not assess the accuracy of different LES (large eddy simulation) results. They found that the hybrid RANS (Reynolds averaged Navier-Stokes)/LES gave good results compared to the simple RANS based models and the use of an appropriate SGS (subgrid-scale) model gave accurate prediction. This is due to the assumption of isotropy in eddy viscosity-based model that is not valid in the impinging region. The poor results of RANS based models may be due to the involvement of a number of arbitrary coefficients. An optimized selection of coefficients may give good result in one region and fail to do so in the other region. In addition to this, poor performance of wall function in the stagnation region and the methodology of time averaging are also the reasons for the poor performance of RANS based models.

1.1.1 Literature review related to circular impinging jet

In an experimental work of a circular jet impingement, Pamadi and Belov (1980) attributed the inner peak of the radial distribution of heat flux to the non-uniform turbulence in the jet. In an analytical study of an axisymmetric free impinging jet on a solid flat surface, Wang et al. (1989b) found that by increasing the wall temperature or wall heat flux with the radial distance reduces the stagnation point Nusselt number and by decreasing the wall temperature or wall heat flux with radial distance enhances the heat transfer at the stagnation point. Barata et al. (1992) investigated experimentally the effect of velocity ratio between the jet and the crossflow in a single confined axisymmetric jet using laser-Doppler anemometry. The deflection of the impinging jet by the crossflow is small for the high jet-to-crossflow velocity ratio. The experimentally found results were compared with the numerical results using $k - \epsilon$ turbulence model. They attributed the difficulty of assessing the turbulence model performance due to the intrusion of numerical diffusion errors. Using a thermal imaging technique, Lytle and Webb (1994) analyzed experimentally the local heat transfer characteristics of air jet impingement at jet-to-plate spacings of less than one jet diameter. They gave relationship of stagnant Nusselt number with Reynolds number and jet-to-plate spacing as $\mathrm{Nu_{st}} \sim \mathrm{Re}^{1/2}$ and $\mathrm{Nu_{st}} \sim (z/d)^{-0.288}$ respectively. Tawfek (1996) concluded that the static pressure distributions along the impingement surface, impinged by a circular jet, were similar and closer to the heat transfer variations for the same configurations. The local and average heat transfer are strong functions of radius of the jet and the jet-to-surface spacing. Ashforth-Frost and Jambunathan (1996a) used LDA (laser-Doppler anemometry) and liquid crystal thermography in an experimental study to assess the effect of nozzle geometry and semi-confinement on the potential core of a turbulent axisymmetric jet. The jet potential core was 7%

longer for the fully developed jet exit profile when compared to the flat jet exit profile. Presence of semi-confinement resulted in extending the potential core by up to 20% owing to limited entrainment and spreading of the jet and reducing the stagnation point heat transfer by up to 10%. Placing the impingement plate at the end of, and just downstream from the potential core (based on a 95% criterion) resulted in highest level of stagnation point heat transfer. In an extension to the earlier work, Ashforth-Frost and Jambunathan (1996b) numerically investigated the problem with standard $k - \epsilon$ eddy viscosity model with inlet boundary conditions based on measured profiles of velocity and turbulence. In the developing wall jet, where the isotropy prevails, the numerical results of heat transfer compared to within 20% of experiment. On the contrary, the stagnation point heat transfer was overpredicted by about 300%, which is attributed to the turbulence model and limitation of the wall function. Garimella and Nenaydykh (1996) conducted an experiment to determine the effect of nozzle geometry on the local heat transfer coefficients from a small heat source to a normally impinging liquid jet. They found that for small nozzle aspect ratio (< 1), the heat transfer coefficients were the highest. The heat transfer coefficients dropped sharply with increase of aspect ratio to $1 - 4$. But with further increase in aspect ratio up to $8 - 12$, the heat transfer coefficients gradually increased. However, this effect was not significant as the nozzle-to-target spacing was increased. Colucci and Viskanta (1996), in an experimental work for an axisymmetric jet, concluded that the local heat transfer coefficients for confined jets are more sensitive to Reynolds number and jet-to-surface spacing than those for unconfined jets. For very low separation distance, the location of the first maximum was observed to remain fixed, but the second maximum displayed a dependence on the Reynolds number.

In a simulation of flow and heat transfer in circular confined and unconfined impinging jet configurations by $v^2 - f$ model, Behnia et al. (1999) concluded that confinement leads to a decrease in the average heat transfer rates. On the other hand, the local stagnation heat transfer coefficient is unchanged. The effect of confinement is significant only for low nozzle-to-plate distance (< 0.25). Brignoni and Garimella (2000) investigated experimentally the effect of nozzle geometry on the pressure drop and heat transfer distribution on confined air jet impingement on a heat source. They compared the heat transfer and pressure measurement using chamfered nozzles with those obtained using square-edged (non-chamfered) nozzle of same diameter. The narrow chamfering (60^0) gave the better performance as the ratio of average heat transfer coefficient to pressure drop is enhanced by 30.8%. The effect of jet-to-jet spacing on local Nusselt number for confined circular air jets impinging normally on a flat plate was investigated experimentally by San and Lai (2001). They concluded that the two factors affecting the heat transfer were jet interference before impingement and jet fountain after impingement. The jet interference before impingement reduces the jet strength, thereby reducing the overall heat transfer. Chatterjee and Deviprasath (2001) concluded that upstream flow development because of vorticity diffusion, more so at small nozzle-to-plate distance, gives rise to the off-stagnation point heat transfer maximum in laminar axisymmetric impinging jets. This is true regardless of Reynolds number for both confined and unconfined jets. The off-stagnation point maxima is not because of local acceleration of the mean radial flow. The off-stagnation point maxima disappeared at dimensionless nozzle-to-plate distances larger than $3/8$. Local Nusselt number increased by a perforated plate installed between a circular impinging jet nozzle and the target plate in an experimental work by Lee et al. (2002). The rate of heat transfer increased as nozzle-to-target plate distance, nozzle-to-perforated plate distance and hole diameter on perforated plate decreased. For the same hole area, Nusselt number was higher for the square hole than the round hole. The effect of several key parameters on the hydrodynamics and heat transfer of an impinging liquid jet had been studied by Tong (2003a). He found out that the stagnation point Nusselt number increased with the Reynolds number and was directly proportional to the square root of the Reynolds number. The average Nusselt number is lowest in the case of uniform velocity profile though the local Nusselt number distribution decreases monotonically in all the three inlet velocity profile cases, parabolic and one-seventh power law being the other two inlet velocity profiles. Chattopadhyay (2004) compared the performance of an annular jet with a standard circular jet having the same values of mass and momentum efflux at the nozzle exit in a numerical investigation in laminar jets impinging on a surface. He found that the heat transfer from the annular jet is about 20% less compared to the circular jet. The distribution of Nu for annular jet scales with Re$^{0.55}$. Analyzing the velocity field and turbulence fluctuations in a hexagonal array of circular jets using PIV (particle image velocimetry), Geers et al. (2004) found that the jet-to-jet interaction was the result of strong upwash flow due to the collision of wall jets. The jet at the center had the shortest core and the highest turbulent kinetic energy, which indicated the strong interaction with a large number of surrounding jets. On the contrary, the outer jets had longer cores and lower kinetic energy levels due to smaller number of neighboring jets. Shuja et al. (2005), in a numerical study for conical jet impingement on to a flat plate, found that increase in the cone angle results in radial acceleration of the flow in both the viscous sublayer and the turbulent boundary layer. This, in turn, enhances the heat transfer rates as compared to a pipe flow situation. However, the location of maximum skin friction coefficient along the radial direction remains almost unaffected for all nozzle cone angles.

Baydar and Ozmen (2005) investigated experimentally and numerically the flow field of a confined jet issuing from the lower surface and impinging normally on the upper surface. At small nozzle-to-plate distance (< 2), a subatmospheric region occurs on the impingement plate. The subatmospheric region becomes stronger with decrease in nozzle-to-plate spacing and it moves radially outward from the stagnation point with increase in nozzle-to-plate spacing. In another similar study, the same authors (Baydar and Ozmen (2006)) carried out experimental investigation on the impinging jet flow for Reynolds number up to 50000 at various nozzle-to-plate spacings for both confined and unconfined configurations. The effect of confinement on flow structure is found to be significant for nozzle-to-plate spacing less than 2. The subatmospheric region occurs on both impingement and confinement surfaces nearly at the same locations. For unconfined jet, no subatmospheric region was found on the impingement surface. It was concluded by Baonga et al. (2006) that hydraulic jump radius increased with increased Reynolds number in an experimental work of a circular impinging jet. The radius at which the liquid layer depth increases beyond the parallel flow is termed as the hydraulic jump radius. Hadziabdic and Hanjalic (2008) described the time and spatial dynamics of the vorticity and eddy structures by the LES generated instantaneous velocity and temperature fields. They attributed the strong jet flapping and precessing, not the turbulence, as a reason for the peak in the local Nusselt number distribution. Tsujimoto et al. (2009) investigated, using DNS (direct numerical simulation), the heat transfer in impinging jets by controlling the vortical structures. They noticed that the heat transfer at the wall did not get strengthened by superposing the two cases of perturbations on the inflow boundary conditions in comparison to no excitation case. The vortical structures generated by excitation got mixed away from the wall not contributing to the heat transfer enhancement. The influence of natural convection on the temperature field at increasing temperature differences between the jet and the target plate was analyzed numerically and experimentally by Koseoglu and Baskaya (2009). They concluded that buoyancy induced natural convection might have opposing or assisting influence on local heat transfer at different locations of the target plate. At low jet inlet velocity, the average heat transfer coefficient at the highest modified Grashof number was higher than the value corresponding to the lowest Grashof number by 37%. Tummers et al. (2011), in an experimental work of turbulent flow in the stagnation region of a single impinging jet issuing from a round pipe, found out the wall shear stress distribution using LDA. They established a relation between the instantaneous flow reversal near the wall region with the formation of small secondary vortices. Rohlfs et al. (2014) investigated, numerically and theoretically, the effect of Reynolds number, nozzle-to-plate distance, Prandtl number, thermal conditions of impingement wall on heat transfer under free-surface jet impingement. They established correlations for predicting the stagnation-zone heat transfer for a wide range of parameters. Wilke and Sesterhenn (2015) justified the reason for the primary and secondary maximum in the local Nusselt number profile as the presence of secondary vortex rings that increase the heat transfer locally. Dairay et al. (2015) established the relation between the vortical structures and the secondary maximum in the radial distribution of mean Nusselt number using DNS of an impinging round jet. They used the space- and time-resolved DNS results to understand the unsteady features of the flow as it is difficult to do so by analyzing the mean flow.

1.1.2 Literature review related to slot impinging jet

In one of the early experimental study involving the turbulent structure along the centerline of a two-dimensional impinging jet, Gutmark et al. (1978) found a selective stretching of vortices along the direction in which the streamlines spread near the wall, causing anisotropy in the region. They got the distribution of energy among various frequencies from spectral measurement. From these measurements, the existence of a neutral frequency was established above which the energy was reduced by viscous dissipation and below which the energy was amplified by a vortex-stretching mechanism. Vader et al. (1991) conducted an experimental work for the surface temperature and heat flux distribution on a flat, upward facing, constant heat flux surface cooled by a planar impinging jet. They found that the results were sensitive to the variations in the stagnation line velocity gradient and the Prandtl number. Al-Sanea (1992) studied three cases of slot jet impingement on an isothermal flat surface viz. free-jet impingement, semi-confined-jet impingement and semi-confined-jet impingement through a crossflow. He found that a fully developed parabolic velocity distribution produced Nusselt number much higher than those produced by uniform velocity profile. Local convection heat transfer coefficient distribution along a constant heat flux surface experiencing impingement by two, planar, free-surface jets of water was obtained in an experiment by Slayzak et al. (1994). Two velocity ratios were considered keeping the other parameters constant. They found that with decreasing velocity ratio, impingement heat transfer coefficients beneath the weaker jet were reduced by the effects of crossflow imposed by the stronger jet. They used a range of Preston tubes and Stanton probes out of which the smallest probe (Stanton probe of size 0.05 mm) gave the best result for the wall shear stress. In an experimental study for a two-dimensional air jet impinging onto a vertical impingement plate, Tu and Wood (1996) used a wider range of Reynolds number, nozzle-to-impingement plate height to nozzle gap ratio as compared to the previous studies. They found the pressure distribution nearly Gaussian that was independent of Reynolds number. Cziesla et al. (1997) concluded that the velocity profile at the nozzle exit as a reason for the deviation of the stagnation point Nusselt number from the experimental values. An increase in local Nusselt number at the edges of abscissa was observed due to thinning of boundary layer caused by head-on collision between neighboring wall jets. Lin et al. (1997), in an experimental study on heat transfer behaviors of a confined slot jet impingement, found that the stagnation, local and average Nusselt numbers were affected by jet Reynolds number while it was insignificantly influenced by the nozzle-to-plate spacing. Yang and Shyu (1998) concluded that the positions of maximum local Nusselt number and the maximum pressure move downstream if the confinement plate inclination is increased. The local maximum Nusselt number was observed to decrease and the local Nusselt number in the downstream location was found to increase with an increase in the inclination of the confinement plate. Apart from this, they found that the inclination has a significant effect on the recirculation region. Voke and Gao (1998) simulated a thermally inhomogeneous turbulent plane jet, through an enclosed pool, impinging on a solid plate using LES. They found that the lateral conduction in the solid plate has no significant effect on the transfer of thermal fluctuations from the fluid to the plate. By this means, they justified a simple one-dimensional model of the thermal interaction between the media. Beitelmal et al. (2000) concluded that the region of maximum heat transfer shifts towards the uphill side of the plate and the local maximum Nusselt number decreases as the inclination angle decreases, the maximum inclination angle being 90^0 keeping the plate normal to the slot jet. The location of maximum heat transfer falls between 0 and $3D$, D being the the hydraulic diameter, uphill from the geometrical impingement point and unaffected by the Reynolds number in the range from 4000 to 12000. For low values of inclination angle, the local Nusselt number from the maximum heat transfer point is insensitive to jet-to-plate distance.

In an LES study, Cziesla et al. (2001) found negative production rate of turbulent kinetic energy in the near wall region. Maurel and Solliec (2001) developed a test bench with variable geometry; they used LDV (laser Doppler velocimetry) and PIV to analyze the development of the jet for different geometrical configurations. They concluded that the characteristic height of the impinging zone remained close to 12% to 13% of the jet-to-plate spacing, irrespective of the Reynolds number and the jet width. The flow in a confined two-dimensional slot jet impinging on an isothermal plate becomes unsteady at a Reynolds number between 585 and 610 when the jet-to-plate spacing and Prandtl number of the fluid are kept at 5 and 0.7, respectively. This was found from a numerical investigation with finite-difference approach by Chiriac and Ortega (2002). The distribution of heat transfer in the wall jet region is influenced by the flow separation caused by re-entrainment of the spent flow back into the jet. The time mean of area-averaged heat transfer coefficient is higher compared to what it would have been in the absence of jet unsteady effects. Chung and Luo (2002) studied the unsteady heat transfer caused by a confined slot impinging jet using DNS. They found a fluctuation of as high as 20% of the time-mean value in the impingement Nusselt number. They noticed that these fluctuations are mainly caused by impingement of the primary vortices originating from the jet nozzle exit. The Kelvin-Helmholtz instability is behind the nearly periodic generation of the primary vortices and thereby resulting in impingement heat transfer fluctuations. However, these quasi-periodic fluctuation become more chaotic and non-linear with increase in Reynolds number. The local Nusselt number away from the stagnation point is influenced by the secondary vortices arising due to the interaction between the primary vortices and the wall jet. The flow field of plane impinging jets at moderate Reynolds numbers was computed using LES with dynamic Smagorinsky model by Beaubert and Viazzo (2002). They studied the mean velocity, the turbulence statistics along the jet axis and at different vertical locations. The effect of the jet exit Reynolds number on near and far field structure was found to be significant between 3000 to 7500. Shi et al. (2002) used standard $k-\epsilon$ and RSM model and found that the turbulent kinetic energy and local Nusselt number at the stagnation region increased by 32.9% and 10.7% respectively when the turbulence intensity at the nozzle exit was increased from 2% to 10%. But a larger effect on turbulent kinetic energy (126% increase) and local Nusselt number (53% increase) at the stagnation zone was observed when the turbulence length scale was increased from $0.07D$ to D, D being the slot jet width.

Tong (2003b) studied numerically the hydrodynamics and heat transfer of the impingement process of an oblique liquid plane jet. Due to the asymmetric geometry of an oblique plane jet, the flow becomes complex. Both the local maximum Nusselt number location and the maximum pressure location shifted upstream from the geometrical impingement point of the jet. The extent of the shift increased with increase in inclination of the jet. Shi et al. (2003) carried out a numerical experiment of a semi-confined laminar slot jet to examine the effect of Prandtl number on heat transfer. The Nusselt number increased with an increase in Prandtl number in the range of 0.7 to 71. In addition to this, they found that gases with similar Prandtl number exhibited similar values of local Nusselt number, but different values of the surface heat transfer coefficient due to their thermal conductivity. In a numerical study of plane turbulent impinging jet in a confined space using DNS, Hattori and Nagano (2004) noticed that for low nozzle-to-plate distances, a second peak appears in the local Nusselt number and skin friction coefficient distribution along the impingement surface. This trend of secondary peak vanishes with increase in nozzle-to-plate distance. The mechanism for the occurrence of the second peak in the local Nusselt number is due to the development of the wall-normal heat-flux near the wall. Sahoo and Sharif (2004), in a numerical study of heat transfer characteristics in the slot jet impingement cooling of a constant heat flux surface, found that for a given domain aspect ratio and Reynolds number, Nusselt number does not change significantly with Richardson number. This indicated that the buoyancy effects are not significant in the overall heat transfer for the jet Reynolds number considered. Lou et al. (2005) carried out a numerical investigation to test the effects of geometric parameters on the confined laminar impinging jet heat transfer. They found that the Nusselt number and the pressure drop from the inlet to the outlet increased with increase in jet width and jet-to-plate spacing. The laminar jet heat transfer decreased with increase in surface roughness as the working fluid got trapped as recirculation bubbles in the cavities of the rough plates. In a numerical study of two-dimensional laminar confined impinging slot jet, Li et al. (2005) interpreted and discussed the bifurcation mechanism. They got two steady flow patterns under identical boundary conditions but with different initial flow fields. The dynamic and thermal behaviors were similar in the stagnation region. But the different flow patterns affected the heat transfer significantly, particularly in the downstream half of the target surface. Wang and Mujumdar (2005b) investigated numerically the flow pattern and mixing characteristics of three-dimensional confined turbulent unequal opposing jets in an in-line static mixer. The mixing effectiveness was found to improve with increase in mass flow rate ratio and decrease in inlet channel width ratio for a fixed total mass flow rate.

1.1.3 Literature review related to use of different turbulence models to solve impinging jet flow

A handful number of studies are available in the literature where several types of turbulence models have already been used to check the accuracy of the results compared with the available experimental results and to find the suitability of the turbulence models used. Craft et al. (1993) used four turbulence models comprising of one $k-\epsilon$ eddy viscosity model and three second-moment closures for the numerical simulation of turbulent impinging circular jets. The $k-\epsilon$ model and one of the Reynolds stress models gave too large levels of turbulence near the stagnation point resulting in high heat transfer coefficients and turbulent mixing with the surrounding fluid. For the numerical simulation of two-dimensional flow field and heat transfer impingement due to a turbulent single heated slot jet discharging normally into a confined channel, Seyedein et al. (1994) used both low-Reynolds number and high-Reynolds number versions of $k-\epsilon$ turbulence models. They found, from the low-Reynolds number model study, that models presented by LB (Lam-Bremhorst) and LS (Launder-Sharma) show very good agreement with the available experimental data. In another study, Seyedein et al. (1995) used LB low-Reynolds number and the standard high-Reynolds number versions of $k-\epsilon$ turbulence models to simulate the steady turbulent flow field and impingement heat transfer due to three and five turbulent heated slot jets discharging normally into a confined channel. LB model overestimated the normalized heat transfer coefficient, while the high-Reynolds number model underestimated it. Hosseinalipour and Mujumdar (1995) compared the performances of the standard high Reynolds number two equation $k-\epsilon$ with standard wall function approach and five low Reynolds number versions of the $k-\epsilon$ model. They compared their results of local Nusselt number distribution with the experimental data of Ichimiya and Hosaka (1989). They obtained better results by including Yap correction (Yap (1987)) in some of the low Reynolds number models. Heyerichs and Pollard (1996) evaluated a number of variants of $k-\epsilon$ and $k-\omega$ two-equation turbulence models and their ability to predict convection heat transfer in channel flow, impinging slot jet flow and flow downstream of a backward facing step. The Wilcox model, which does not require the calculation of the wall shear stress and distance from the wall, is the easiest to implement. The Wilcox model showed the highest correlation ($r \geq 95\%$) to the heat transfer for the impinging slot jet. Tzeng et al. (1999) used eight turbulence models, including one standard and seven low Reynolds number $k-\epsilon$ models, to test the prediction of heat transfer performance of multiple slot impinging jets. They concluded that the prediction by each turbulence model depended on grid distribution and numerical scheme used in the spatial discretization. The QUICK (quadratic upstream interpolation for convective kinematics) scheme incorporated with the AKN (Abe, Kondoh and Nagano) turbulence model generated more accurate prediction.

Angioletti et al. (2005) extensively investigated the flow field behavior in the vicinity of the stagnation region. Later, by using commercial CFD package, they evaluated the suitability of three different turbulence models by comparing the numerical results with the experimentally obtained results. They found that the $k - \omega$ SST (shear stress transport) model gave good result for lower Re and $k - \epsilon$ RNG (renormalization group) or RSM (Reynolds stress model) performed better for high Re. El-Garby and Kaminski (2005) compared the performance of standard k–ϵ model and the Yang-Shih (YS) model for a three-dimensional numerical simulation of impingement with cross flow. The jet angle was varied between 30^0, 60^0, and 90^0 as measured from the smooth flat impingement surface. The standard k–ϵ model, a high Reynolds number turbulence model, is best applicable to core flow regions and does not apply very near the wall where viscous effects are dominant. The YS model predicted average Nusselt number within $2 - 30\%$ and the standard $k - \epsilon$ model predicted average Nusselt number with $0 - 60\%$ error in 30 test cases. They tested the Yap correction (Yap (1987)) to reduce the turbulence length scale in the near wall region with the low Reynolds number $k - \epsilon$ models. Wang and Mujumdar (2005a) analyzed by comparing five versions of low Reynolds number $k - \epsilon$ models for the prediction of the heat transfer under a two-dimensional turbulent slot jet by comparing their results with the experimental work of van Heiningen (1982). The inclusion of Yap correction (Yap (1987)) improved the prediction in the stagnation region and far away downstream regions, but failed to do so in the intermediate downstream regions.

Zuckerman and Lior (2005) and Zuckerman and Lior (2006) compared the relative strengths and drawbacks of $k - \epsilon$, $k - \omega$, Reynolds stress model, algebraic stress models, shear stress transport, and $v^2 - f$ turbulence models for impinging jet flow and heat transfer. In their findings, they highlighted that though the computational cost for $k - \epsilon$, $k - \omega$, realizable $k - \epsilon$ and other $k - \epsilon$ variations, and algebraic stress model are low, impingement jet transfer coefficient prediction are poor. On the other hand, they concluded that shear stress transport, $v^2 - f$ and DNS/LES time variant models have moderate to high computational cost giving fair to excellent prediction of impingement jet heat transfer coefficient. The heat transfer on a cylindrical target exposed to radial impinging slot jets, for the design of an impinging jet device, was investigated by Zuckerman and Lior (2007) using shear stress transport, standard and realizable $k - \epsilon$, $v^2 - f$ and Reynolds stress model turbulence models. Based on the validation, they chose $v^2 - f$ model for further work. Static pressure rise and thereby, flow separation on the surface of the cylindrical target was caused by the interaction of adjacent opposed wall jets. This separation and the fountain flow between the two wall jets increased the Nusselt number. Hofmann et al. (2007) compared the capabilities of 13 widely spread RANS based turbulence models to assess the heat transfer and flow structure at different Reynolds number and different nozzle-to-plate distances. All the models examined were found suitable for the prediction of wall jet heat transfer. On the other hand, the SST $k-\omega$ model was found to predict the local heat transfer near the stagnation region correctly. SST $k-\omega$ model predicted the secondary maximum of the local Nusselt number distribution, which occurs at small nozzle-to-plate spacing. Numerical performance and accuracy of four $k-\epsilon$ and seven $k-\omega$ models were carried out in both plane and round impinging jets in a numerical work by Jaramillo et al. (2008). It was found that NLEVM (nonlinear eddy viscosity model) predicted local Nusselt number better than LEVM (linear eddy viscosity model) at the stagnation point. Models with good performance in the round jet configuration showed poor results in the plane jet configuration. Isman et al. (2008) reported the most satisfactory results with non-linear algebraic stress model of Shih-Zhu-Lumely in the stagnation region. But the overall performance of RNG and standard $k-\epsilon$ models were found to do better in comparison to other models considering entire region of flow. They also concluded that inclusion of property variation and buoyancy effect reduced the discrepancy with experimental results. Most recent evaluation of model performance includes the works by Dutta et al. (2013) and Afroz and Sharif (2013). Dutta et al. (2013) used eight different RANS equations based turbulence models to check the performance of the computation of turbulent jet impingement flow. They found that the accuracy of turbulence models is highly sensitive to the flow conditions. Both the standard and the SST $k-\omega$ models showed the best agreement with the experimental data in terms of secondary peak of local Nusselt number distribution for small nozzle-to-plate spacing (4). In the case of high nozzle to plate spacing (9.2), the standard k–ω and the standard k–ϵ models only showed good agreement with the experimental data in terms of local Nusselt number. The performance to predict the flow and thermal fields for a normal confined slot jet impingement of RNG $k-\epsilon$ model and SST $k-\omega$ model were compared by Afroz and Sharif (2013). They found that the local Nusselt number distribution predicted by the SST $k-\omega$ model agrees better with the experimental data. Subsequently, the authors employed the SST $k-\omega$ model to study the twin oblique impinging jet heat transfer. Results indicated that the local peak Nusselt number gradually reduced and its location slightly shifted away from the jet axis as the inclination angle of the jet was reduced from 90^0 (normal impingement). The average Nusselt number also decreased with decrease in the impingement angle for any combination of Reynolds number, jet-to-jet separation distance and nozzle-to-plate distance.

1.1.4 Literature review related to conjugate heat transfer in impinging jet

Conjugate heat transfer analysis of a laminar impinging jet on a laterally insulated disc was studied by Wang et al. (1989a). They found analytically that the Nusselt number depends on Prandtl number, the conductivity ratio of fluid-to-solid, aspect ratio of the thickness-to-radius of the disc and the non-uniform wall temperature or wall heat flux. In addition to the findings from their previous study, it was also found that for thick disc (aspect ratio=1), the non-uniform wall temperature or wall heat flux has little effect on the local heat transfer coefficient. However, for thin disc, the effect is considerable. They also found an obvious result that for very small aspect ratio (0.001) the result is same as that where the boundary condition is imposed on the impingement surface. From the results of numerical simulation of a free jet of a high Prandtl number fluid impinging perpendicularly on a solid substrate of finite thickness, Rahman et al. (1999) concluded that the disc, beyond a certain thickness, showed one-dimensional heat conduction in regions away from the impingement surface and did not exert any significant influence on the convective heat transfer process. Bula et al. (2000a) and Bula et al. (2000b) and investigated the conjugate heat transfer in slot jet and circular jet impingement. Bula et al. (2000a) carried out computation of a free jet of high Prandtl number fluid impinging perpendicularly on a solid substrate of finite thickness containing small discrete heat sources on the opposite surface to investigate the influence of different operating parameters such as jet velocity, heat flux, plate thickness, and plate material. The disc thickness and location of discrete sources showed to have strong influence on the maximum temperature and the average heat transfer coefficient. An ideal disk thickness for best performance was found where the heat transfer coefficient and the Nusselt number attained peak values. Bula et al. (2000b) investigated the influence of different operating parameters such as jet velocity, heat flux, plate thickness, plate material, and the location of the heat generating electronics in the analysis of the conjugate heat transfer from discrete heat sources to a two-dimensional jet of a high Prandtl number fluid from a slot nozzle. They found that there is an ideal design plate thickness, at which the heat transfer is optimized, depending on the material properties. The position of the discrete heat sources has a crucial role in the determination of the temperature and the heat transfer rate. Yang and Tsai (2007) carried out a transient conjugate heat transfer study of a flat circular disc, impinged by a circular impinging jet, using a low Reynolds number $k-\omega$ turbulence model with different Reynolds number, temperature or heat flux of the disc and orifice-to-heat source spacing.

They found that the jet Reynolds number has a significant effect on the hydrodynamics and heat transfer in the stagnation region in the way that the time required to reach the steady-state condition decreases as the Reynolds number increases. It was also found that the stagnation heat transfer is influenced by the induced turbulence from the surrounding around the jet. Rahman et al. (2008) computationally studied the conjugate heat transfer in a semi-confined liquid jet impingement from a rotating nozzle on a uniformly heated spinning solid disk of finite thickness and radius. They found that the local heat transfer coefficient increased reducing interface temperature difference over the entire disk surface with increase in Reynolds number. The rotational rate also increased local heat transfer coefficient under most conditions. Rahman and Lallave (2009), using Galerkin finite element method, presented the transient conjugate heat transfer characterization of a free liquid jet impinging on a rotating solid disc of finite thickness and radius. They found that the duration of the transient period increased with disc thickness and decreased with Reynolds number and thermal conductivity ratio. Panda and Prasad (2011) investigated both computationally (SST $k-\omega$ turbulence model) and experimentally a shower head of air jets impinging on the top surface with a constant heat flux imposed on its bottom surface. The spacing-to-orifice diameter ratio, the jet Reynolds number and the plate thickness-to-diameter ratio were varied as the independent parameters. Dependence of local variation of heat transfer rate on spacing-to-orifice diameter ratio is found to be significant whereas it is less sensitive to the thickness ratio.

1.1.5 Literature review related to impinging jet with moving impingement wall

In one of the very early experimental work dealing with heat transfer from moving impingement surface, Raju and Schlunder (1977) determined of heat transfer rate from continuously moving belt to air jet impinging normally on it using an infrared thermometer. They used an endless PVC (polyvinyl chloride) belt on seven rollers which was connected to a variable speed motor. They converted mass transfer results to heat transfer results using heat-mass transfer analogy. They found that the average heat transfer coefficients increased with belt speed initially to a maximum value and then remained almost constant for all higher belt speeds and the maximum heat transfer coefficients were about 1.5 to 2.0 times higher than those predicted for the stationary surface. Chattopadhyay and Saha (2001) compared the performance of horizontal knife jet with a standard axial jet using LES. The heat transfer rate decreased with increase in surface velocity. They found that the heat transfer rate for the case of knife-jet is more than axial jet when the surface-to-jet velocity ratio is of the order of 0.5. However, the trend got reversed for low Reynolds number. For higher Reynolds number, the heat transfer rate for the two types of jets becomes comparable for higher surface-to-jet velocity ratio beyond the order of 0.5. In another study involving laminar jet, the same authors (Chattopadhyay and Saha (2002)) achieved 30% more heat transfer in axial jet in comparison to knife-jet. The dependence of different components of turbulent production rate on impinging surface velocity has been investigated by Chattopadhyay and Saha (2003). They concluded that the turbulent production rate as a whole increases with increase in surface velocity up to a value of 1.2. Beyond a surface velocity of 1.2, it starts to decrease similar to the case of heat transfer. But the turbulent kinetic energy increased with increase in impingement surface velocity. In an experiment using PIV, Senter and Solliec (2007) found that the flow field remained independent of the jet Reynolds number in the range $5300 - 10600$ at a given surface-to-jet velocity ratio. The changes in the flow field, compared to that when the impingement plate is stationary, got increased with increase in surface-to-jet velocity ratio. They also found the turbulence intensity in the stagnation region getting increased with increase in surface-to-jet velocity ratio.

1.2 Objectives of the thesis

A detailed literature review revealed that most of the conjugate heat transfer problems that have been solved are for circular impinging jet. Though Bula et al. (2000b) analyzed the influence of different operating parameters in the analysis of the conjugate heat transfer from discrete heat sources in a slot jet of a high Prandtl number fluid, the Reynolds number is in the range from 550 to 2200. To the best of the author's knowledge, the study of conjugate heat transfer in a slot jet impinging on a moving plate is probably missing. Therefore, the present computational study attempts to investigate the conjugate heat transfer in a turbulent slot jet impinging a plate with a relatively high Reynolds number at the nozzle exit, the plate being stationary or in motion. The strong streamline curvature, recirculation, boundary layer development along the impingement surface are some of the features that make the flow very complex. Presence of complex fluid flow within a simple geometry makes it a benchmark problem to evaluate the performance of different turbulence models. Many researchers have used commercial softwares to find varying degrees of accuracy by different turbulence models in predicting the fluid flow and heat transfer in an impinging jet. So the performance of four different RANS based turbulence models in the prediction of impingement jet flow of air using an in-house code has been done in the present research.