HEAT TRANSFER ENHANCEMENT FROM A HORIZONTAL SURFACE BY IMPINGING SWIRL JETS

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Abstract

Impinging jets are widely used in industry to increase heat transfer from hot surfaces in applications such as electronics cooling, glass tempering and paper drying. The current research is concerned with the design of a swirl jet which aims to enhance the surface heat transfer from a heated surface in a radially uniform manner. Tests were conducted for a wide range of parameters including Reynolds number \((Re = 10000\) to \(30000\)), dimensionless nozzle to surface spacings \((H/D = 0.5\) to \(6.0\)) and degree of swirl \((S = 0\) to \(2.25°/mm\)). Results have shown that the swirl generator enhances the heat transfer overall but this is more likely to be due to the increased jet turbulence which has a greater influence on the surface heat transfer than the swirling motion of the flow. At low \(H/D\), the surface heat transfer is reduced in the stagnation zone due to the swirl generator blockage of the jet flow. It has also been found that the optimum degree of swirl from a heat transfer perspective is a function of the nozzle to impingement surface spacing.

Nomenclature

- \(A\): Foil Area, \([m^2]\)
- \(D\): Diameter of Jet, \([m]\)
- \(h\): Convective Heat Transfer Coefficient, \([W/m^2K]\)
- \(H\): Height of Nozzle above Impingement Surface, \([m]\)
- \(I\): Current, \([A]\)
- \(k\): Conductivity, \([W/mK]\)
- \(q\): Rate of Heat Transfer, \([W]\)
- \(q''\): Heat Flux, \([W/m^2]\)
- \(r\): Radial distance from Stagnation Point, \([m]\)
- \(Re\): Reynolds Number \((\rho U_jD/\mu), [-]\)
- \(S\): Swirl, \([°/mm]\)
- \(S_n\): Swirl Number [-]
- \(T\): Temperature, \([K]\)
- \(U_j\): Jet Velocity, \([m/s]\)
- \(V\): Voltage, \([V]\)
- \(\rho\): Density, \([kg/m^3]\)
- \(\mu\): Viscosity, \([Ns/m^2]\)
- \(\delta\): Sensor Thickness, \([m]\)

1 Introduction

Impinging jets have been widely used for increasing heat transfer in engineering applications such as cooling of hot steel plates and turbine blades, tempering of glass, drying of papers and films and cooling of electronic components. Heat transfer to conventional (i.e. no swirl) impinging air jets has been the subject of many investigations and comprehensive reviews of the literature have been compiled by Martin (1977) and Jambunathan et al. (1992). In general, the distribution of the heat transfer from a heated surface subject to an impinging jet is shown to be a maximum at the stagnation point and reduces with increasing radial distance. At low nozzle to impingement surface spacings \((H/D \leq 2)\), however, secondary peaks have been observed by O'Donovan and Murray (2007) and others in the near wall jet region. It has been shown, by Lee et al. (2002) and others, that swirl can enhance the overall heat transfer and also make the distribution of the heat transfer more uniform.
Numerous techniques have been proposed to enhance the overall cooling performance of an impinging air jet and Ward and Mahmood (1982) introduced the idea that a swirling impinging jet had the potential to increase surface heat transfer. They found the radial distribution of the surface heat transfer was slightly more uniform than that of a conventional impinging jet, but its value was significantly lower, particularly in the vicinity of the stagnation point. Lee et al. (2002) investigated a swirling impinging jet that was achieved by inserting a swirl generator at the jet nozzle exit. It was found that swirl has the potential to enhance the overall heat transfer; however, it often came at a cost of a lower stagnation point heat transfer coefficient. The reduction of the heat transfer in the stagnation region was, in part, attributed to the significant blockage of the jet flow by the swirl generator. Huang and El-Genk (1998) used a swirl generator made of a cylindrical plug with four narrow channels to provide swirl to a single and multiple impinging air jets. This design effectively created a multi-channel jet and it was found that the surface heat transfer to the multi-channel jet was much higher than for the conventional impinging jet, particularly at and near the stagnation point. The surface heat transfer in the stagnation zone was reduced by swirl at high $H/D$ but at greater radial distances the opposite was true. Swirling jet flows achieved a more uniform distribution of the surface heat transfer overall, a practical benefit for many applications such as electronics cooling.

Lee et al. (2002) studied the effect on the local and average heat transfer distribution of a swirl jet generated by eight narrow channels. For a small jet-to-surface spacing ($H/D = 2$), the average Nusselt numbers of the swirling jet flows were larger than those without swirl for all swirl angles, but for $H/D > 10$, the effect of the swirling jet flows was less significant. Lee et al. (2002) found that the stagnation point Nusselt Number has a strong dependence on the swirl number ($S_n$) defined by equation 1:

$$S_n = \frac{2}{3} \left[ \frac{1}{l} \left( \frac{r}{R} \right)^3 \right] \tan \theta$$

where $\theta$ is the angle between the swirl vane and the vane axis and $r$, $R$ and $l$ are the inner radius, outer radius and length of the swirl generator respectively. It was observed that the stagnation point Nusselt number was higher with $S_n = 0$ than without a swirl generator up to $H/D = 10$. This is attributed to the fact that for $S_n = 0$, the flow passing through the swirl generator resembles multiple jets, issuing from multiple nozzles, thereby enhancing the heat transfer rate by the interaction between the jets. Two competing effects were identified to influence the stagnation point heat transfer. While increased swirl increases the interaction of the flow and thereby turbulence, it also reduces the arrival flow velocity.

Bakirci and Bilen (2007) visualised the temperature distribution and evaluated heat transfer rate on the impingement surface for a swirling, multi-channel and conventional impinging jets using thermochromic liquid crystals. They found the local Nusselt numbers of the multi-channel impinging jet were generally much higher than those of the swirling and conventional impinging jets. Heat transfer experiments demonstrated that as the swirl angle increased, every channel tended to behave like an individual jet, each forming a separate impingement region. Swirling impinging jets with angles $41^\circ$ and $50^\circ$ degrees produced saddle-shaped radial distributions of the local Nusselt numbers, which were more pronounced at high Reynolds numbers and/or small jet spacing. When swirl angle increased the locations of the heat transfer peaks were significantly shifted from the stagnation point; this radial shift was attributed to the higher tangential velocity component. The local and area averaged Nusselt number values decreased as swirl angle increased, and the highest swirl angle of $50^\circ$ gave a more uniform heat transfer rate on the surface. Increasing Reynolds number increased the heat transfer rate on the entire surface but had no significant effect on the position of the individual impingement regions, however it emphasised the saddle shaped heat.
transfer distribution. In general, the radial uniformity of these Nusselt numbers improved as the swirl angle and/or jet spacing was increased.

The current research is concerned with the design and test of a new swirl jet generator which minimises the blockage of the jet flow. Tests were conducted for a wide range of parameters including uniform wall flux and uniform wall temperature boundary conditions, Reynolds numbers \((Re = 10000 \text{ to } 30000)\), degree of swirl \((S = 0 – 2.25 \degree/mm)\) and dimensionless nozzle to surface spacings \((H/D = 0.5 - 6.0)\).

2 Experimental Rig

The experimental test rig consists of a jet nozzle and a heated impingement surface as indicated in figure 1. Air is supplied to the nozzle chamber from the building compressor through four orthogonally positioned inlet points. An Alicat Scientific Inc. Precision Gas Flow Meter is installed on the compressed air line to monitor both the air volume flow rate and temperature. The air flows through a swirl generator before issuing from the jet nozzle and impinging on the heated surface. The degree of swirl, \(S\), is defined as the angle through which the flow has been turned per millimetre length of the swirl generator. Figure 2 (a) presents a schematic of the swirl generator that was manufactured using a rapid prototyping technique; this technique has enabled the design of a swirl jet that minimises the blockage of the flow. The significant blockage of the jet flow created by the swirl generator used by Lee et al. (2002) (figure 2(b)) was shown to reduce the stagnation point heat transfer and so an objective of the current research is to minimise blockage.

The surface heat transfer to the impinging swirl jet has been measured using an RdF Micro-Foil® heat flux sensor which is flush mounted on a heated copper plate that approximates a uniform wall
temperature boundary condition. As this is a point measurement technique, the heated surface is mounted on a carriage which allows the sensor to be placed relative to the impinging jet so that a one-dimensional distribution of the surface heat transfer can be acquired. An RdF Micro-Foil® Heat Flux Sensor contains a differential thermopile that measures the temperature above and below a known thermal barrier. The heat flux through the sensor is therefore defined by equation 1.

\[ \dot{q} = k_s \frac{\Delta T}{\delta} \]  

(1)

where \( k_s \) is the thermal conductivity of the barrier (kapton) and \( \Delta T \) is the temperature difference across the thickness (\( \delta \)) of the barrier. A single pole thermocouple is also embedded in this sensor to measure the temperature locally. The RdF Micro-Foil® heat flux sensor was calibrated in situ against a reference stagnation point heat transfer coefficient presented by Shadlesky (1983). The calibration of the sensors compared favourably with that provided by the manufacturer’s specification. A complete calibration and uncertainty analysis for the Micro-Foil® set-up is presented by O'Donovan (2005).

An ohmically heated thin foil technique was employed to measure the two dimensional temperature distribution on the impingement surface; this case approximates a uniform wall flux thermal boundary condition. A layer of thermo-chronic liquid crystal, backed by a layer of black paint was applied to the thin electrically heated foil mounted on a 10 mm thick Perspex sheet. The foil used is 25 micron thick AISI 321 stainless steel supplied by Goodfellow Ltd. Both the black paint and the liquid crystal (Hallcrest: BM/R3C7W/S40) layers are applied using an Aztek A4702 artist's airbrush in conjunction with a compressed air supply at 1.5 bar. A voltage was applied across the foil through two large copper terminals at either end of the foil. Four strips of high intensity light emitting diode (LED) illuminated the liquid crystals on the test surface and an NAC Hi-Dcam II digital colour camera was used to record the colours of the liquid crystals with a resolution of 1280 \( \times \) 1024 pixels per frame. The Hue Saturation Intensity (HSI) of the images under isothermal conditions was used to calibrate against a reference RTD thermocouple. The resulting calibration function was used to convert colour images to temperature distributions on the surface. Surface temperature maps are used in the current investigation to visualise the 2-dimensional distribution of the surface heat transfer.

3 Results & Discussion

The surface heat transfer to an impinging swirl jet is presented for a wide range of experimental parameters along the measurement line indicated in figure 3. This line bisects the angle created between two guide vanes of the swirl generator. The flow and surface heat transfer is not axisymmetric as the swirl generator has the effect of splitting the main jet flow into four separate flows. The effect of this asymmetry is particularly evident at low H/D as the jet streams do not merge.

![Figure 3: Location of Surface Heat Transfer Measurements](image)

Figures 4 to 6 present distributions of the Nusselt number along the measurement line (indicated in figure 3) and some equivalent surface temperature maps of the impingement surface. The temperature measurements were acquired using liquid crystal thermography and are primarily used
to understand the distribution of the surface heat transfer from the surface subject to the impinging swirl jet. Temperatures which are outside the liquid crystal range appear as a shade of blue as seen at high values of $r/D$ in figure 4-6 (b).

Figure 4: Re = 20000, H/D = 0.5

Figure 5: Re = 20000, H/D = 3

Figure 6: Re = 20000, H/D = 6
Figures 4 to 6 compare the surface heat transfer of a swirling impinging jet to a conventional impinging jet for a Reynolds number of 20000 and a range of nozzle to impingement surface spacings ($H/D = 0.5, 3$ and $6$). It is immediately apparent that the swirl generator has a significant influence on the surface heat transfer. At $H/D = 0.5$ (figure 4) it can be seen that the heat transfer distribution of a conventional impinging air jet (no swirl) is drastically different to the zero swirl case. For the conventional impinging air jet the heat transfer peaks at the stagnation point ($r/D = 0$), and decreases with increasing radial distance before increasing to a secondary peak at approximately $r/D = 1.2$. This secondary peak has been attributed to an abrupt increase in turbulence in the wall jet by Gardon and Akfirat (1965), Lytle and Webb (1994) and others. Beyond the secondary peak, the heat transfer simply decays monotonically with increasing radial distance. In the case of zero swirl, however, where a straight swirl generator is in place, the heat transfer distribution is vastly different. The stagnation point heat transfer is now a local minimum and the heat transfer distribution exhibits two secondary peaks. It can be seen from the two dimensional temperature map in figure 4(b) that the likely cause of the low heat transfer at the stagnation point is the blockage of the impinging jet flow by the swirl generator. The first peak can then be attributed to the stagnation point of one of the four jet streams issuing from the swirl generator and the second peak to the abrupt increase in turbulence in the wall jet as with the conventional impinging jet. The temperature map in figure 4(b) also indicates that at $H/D = 0.5$ the jet streams impinge on the heated surface independently. With increasing $H/D$, as can be seen in subsequent figures 5(b) and 6(b), the four jet streams begin to merge before impingement.

It can be seen from each of the figures 4 to 6 that the heat transfer to an impinging air jet with a swirl jet present is higher overall than for the conventional jet. However most of this enhancement can be directly attributed to the enhanced mixing and turbulence due to the swirl generator and not to the swirling motion of the jet. In fact, at a low $H/D$ of $0.5$, it is difficult to discern if the degree of swirl has any influence on the local or area averaged heat transfer. At $H/D = 3$ (figure 5) the difference between the heat transfer distributions is more significant. Figure 5(a) indicates that there may be an optimum degree of swirl at $S = 1.5^\circ/\text{mm}$ to achieve high cooling rates. It can be seen in figure 5(b) that the jet streams are beginning to merge and the temperature distribution is becoming more axisymmetric, but there still exists a definite blockage at $r/D = 0$ at this value of $H/D$.

Figure 6 presents data for the largest nozzle to impingement surface spacing investigated ($H/D = 6$). It is apparent from figure 6(b) that the 4 separate jet streams must have merged before impacting on the heated surface at this height as the temperature map can be considered axisymmetric and the blockage of the jet flow is no longer evident, even at the stagnation point. Figure 6(a) shows that the heat transfer is a maximum at the stagnation point for each swirl condition. Swirl at $H/D = 6$ has enhanced the heat transfer at the stagnation point over that of a conventional impinging air jet marginally. This has come at the cost of marginally lower heat transfer in the wall jet region.
The effect of Reynolds number and nozzle to impingement surface spacing on heat transfer to a swirling impinging air jet \((S = 2.25°/mm)\) is presented in figure 7. As expected the mean heat transfer increases with increasing Reynolds number, as can be seen in figure 7(a). Interestingly, the relative magnitude of the secondary peaks increases with increasing Reynolds number. This is also the case for a conventional impinging air jet and is attributed to the elongation of the potential core length with increasing jet velocity. Both the magnitude and shape of the heat transfer distribution to a swirling impinging jet is highly influenced by the nozzle to impingement surface spacing as indicated in figure 7(b). The stagnation point heat transfer changes from a local minimum to a maximum with increasing \(H/D\) as the merging of the jet streams replaces the effects of the swirl generator blockage. Otherwise the heat transfer distributions behave similarly to those found for a conventional impinging air jet. The secondary peaks reduce with increasing \(H/D\) and as the jet spreads and the arrival velocity decreases the heat transfer distribution flattens somewhat.

![Figure 8: Area Averaged Nusselt number, Re = 10000](image)

All of the effects discussed so far contribute to the overall cooling performance of a swirling jet. Figure 8 shows how the Nusselt number averaged over an area extending to 5 diameters from the stagnation point varies with nozzle to impingement surface spacing and the degree of swirl for \(Re = 10000\). Similar trends are found for Reynolds numbers of 20000 and 30000. Figure 8(a) indicates that there exists an optimum degree of swirl for heat transfer, that increases with \(H/D\). At \(H/D = 0.5\), the optimum swirl angle is \(0.75°/mm\) and this increases to \(1.5°/mm\) for larger \(H/D\). Further discretisation of this range is required to determine this relationship between optimum degree of swirl and \(H/D\). Figure 8(b) shows that the area averaged heat transfer decreases with increasing \(H/D\) but that the heat transfer to the swirl jet is superior overall. It is not clear from the presented data that the enhancement of heat transfer can be attributed to the swirling flow however as the heat transfer is enhanced similarly for the zero swirl case. Therefore the increased heat transfer is largely attributable to the enhanced mixing due to the addition of the swirl generator.

### 4 Conclusions

Heat transfer distributions to an impinging air jet have been presented for a range of experimental parameters including Reynolds numbers from 10,000 to 30,000, nozzle to impingement surface spacing from 0.5\(D\) to 6\(D\) and degree of swirl from 0 to 2.25°/mm. From the data presented, the following can be concluded:

By incorporating a swirl generator at the jet nozzle, the flow issuing from the nozzle is partially blocked. This has the effect of reducing the heat transfer in the stagnation region as this is the location of most significant blockage. Despite the advanced manufacturing technique which has
reduced the size of the guide vanes in the swirl generator, the heat transfer distribution and the area averaged heat transfer are still affected by the blockage caused. This is particularly apparent at the lower range of $H/D$.

The flow from a swirling jet is three dimensional and not axisymmetric as in a conventional air jet. As four, essentially distinct, jet streams unite at the nozzle exit the heat transfer distribution cannot be considered axisymmetric. At larger $H/D$, the heat transfer at the stagnation point is greater for the swirl jet. This has been attributed to the extra mixing created between the four jet streams as they merge.

The effects of swirl are best appreciated by comparing the heat transfer subject to a swirling impinging jet to that of a jet impinging with a straight swirl generator ($S = 0^\circ/mm$). In this case an optimal degree of swirl has been found but is seen to be both $H/D$ and Reynolds number dependent.

5 References


Shadlesky, P. S., 1983, Stagnation point heat transfer for jet impingement to a plane surface, AIAA Journal, 21, 1214 - 1215.