Effect of Thermal Boundary Condition on Heat Dissipation due to Swirling Jet Impingement on a Heated Plate

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Abstract: This paper reports on experimental research conducted to investigate the convective heat transfer characteristics for jet impingement with and without swirl. A series of different nozzle to surface heights, ranging from 0.5D to 10D, and Reynolds numbers, ranging from 8000 to 20000, were examined to determine their influence on the heat transfer characteristics for varying degree of swirl. Two separate experimental setups, having different thermal boundary conditions, were used in order to determine the effect that the thermal boundary condition might have on the findings. The results obtained indicate that the Nusselt number distributions are largely similar for the two boundary conditions considered, with the method based on a uniform wall temperature with hot-film sensor measurements showing greater local variation and spatial resolution in the stagnation region. For the range of parameters tested, the stagnation region heat transfer is enhanced with the addition of swirl to the jet flow.

1. Introduction

Impinging jets are well-known for their uses in mechanical and electronic systems, attributable to the high rates of heat transfer which can be achieved. The formation and development of the flow field which is employed in impinging jets has been examined many times in the past and the structure of the jet before and after impingement has been examined in detail to help explain what causes the high heat transfer rates.

Case studies of normal impinging jets are the most common type of impinging jet investigated in past research. Normal impinging jets are designed so that there is no obstruction in the flow which can distort it in any way. A review by Jambunathan et al. [1] illustrates the key factors in the flow structure of a normal impinging jet such as the potential core of the flow and the locations of high turbulence which lead to high levels of heat dissipation.

While normal impinging jets provide effective cooling mechanisms for heated surfaces, by altering the jet flow field even higher heat transfer rates can be achieved. One such method of potentially increasing the heat transfer is through inducing a swirling motion in the fluid flow, either by introducing a tangential flow into an axial flow or by using guide vanes.

The first method, introducing a tangential flow into an axial flow, has been used in past work [2-5] to avoid any blockages that may occur in the flow when guide vanes are used. The use of guide vanes, however, is the more common method of generating a swirling flow. Research into swirling impinging
jets that employ guide vanes to control the level of swirl has been conducted by Bakirci and Bilen [6], Huang and El-Genk [7], Hee Lee et al. [8] and Sheen et al. [9] to name but a few. Results commonly show that introducing a swirling motion into the flow can both enhance and reduce local heat transfer, depending largely on the distance at which the jet is positioned from the impingement surface.

This report details experimental heat transfer findings for a swirling jet impacting upon a heated horizontal surface. In order to determine whether the effects of swirl were sensitive to thermal boundary conditions, or to the associated measurement techniques employed, two experimental setups were used.

The comparisons between the results found using these two setups will give an indication about how the experimental techniques used to evaluate heat transfer due to impinging jets can influence the results. Following this a further detailed analysis into the optimisation of swirl generators can be conducted using the most appropriate method and can allow associations to be made between past research which may have been performed using different measurement systems.

2. Experimental Apparatus

2.1. Nozzle geometry
The impinging jet geometry consists of a contoured nozzle with a 5mm diameter, illustrated in Figure 1. Air enters a chamber above the nozzle and passes though a mesh to evenly distribute the fluid down through the chamber towards the nozzle. For a non-swirling impinging jet, the fluid passes downwards to the nozzle where the contoured geometry narrows the flow to the diameter of the nozzle.

2.2. Swirl Generators
To generate a swirling impinging jet, the use of guide vanes was employed. Special swirl generating inserts were designed using AutoCAD™ and created using a rapid prototyper. Four generators were produced, with two basic shapes and sizes. Each of the four generators has a set of six guide vanes, angled at 45° to direct the fluid flow towards the nozzle exit in the centre. The vanes were mounted on a flat disc to contain the fluid and ensure the direction of motion.

Two of the generators, one large (A) and one small (B), also include an additional section known as the swirl core. This has a further four vanes, which are twisted axially to create an angle of 45° with the horizontal. The final two inserts, C and D, have a simple curved cone at the centre to direct the fluid. The swirl generating inserts were fixed to the contoured jet nozzle such that the fluid flow would enter from the side and pass through the sets of guide vanes, changing the direction of the flow accordingly. The four swirl generators produced are shown in Figure 2 and a cross-sectional representation of the impinging jet including swirl generator ‘A’ placed in position is shown in Figure 3.

![Figure 1. Elevation of Jet Nozzle and photograph showing Nozzle Geometry](image1)

![Figure 2. (From left to right) Swirl Generating Inserts A, B, C and D.](image2)
2.3. Experimental Rig

Two separate experimental rigs were used to evaluate the convective heat transfer characteristics with and without swirl. Each system used a different thermal boundary condition.

2.3.1. Uniform Wall Flux Thermal Boundary Condition. To provide a uniform wall flux boundary condition an electrically heated thin foil was used, in conjunction with temperature measurement by infrared thermography.

A 25µm thick stainless steel foil, with one side coated in a layer of matt black paint, is ohmically heated. The thickness of the foil and paint has been proven to be thermally thin using Biot theory, such that the temperature on the upper side of the foil is regarded to be the same as that on the under side. The foil is mounted horizontally, with the painted side facing downwards and each end clamped between a pair of copper bars. The bars are connected to a power supply and a spring mechanism which keeps the foil taut at all times during testing. The impinging jet is positioned above the foil so that the flow impinges in the centre of the foil. A thermal imaging camera [FLIR Systems™ Thermovision™ A40] is placed underneath the foil, facing the painted layer, in line with the jet.

The convective heat transfer coefficients are determined using the analysis outlined in section 3.1 to take into account the effects of radiation, lateral conduction and natural convection from the underside of the foil. As well as the heated foil tests, the thermal imaging camera records a set of images of the unheated foil during jet impingement; these are taken as adiabatic wall temperature measurements. The images display the foil surface temperature over an area of approximately 110mm x 150mm corresponding to a camera resolution of 240 x 320 pixels. A series of temperature maps are averaged for each of the testing parameters, i.e. height of jet from the impingement surface, Reynolds number and swirl configuration. The resulting temperature data are then used in a Matlab program written to implement the analysis described in section 3.1.

2.3.2. Uniform Wall Temperature Thermal Boundary Condition. The second experimental rig approximates a uniform wall temperature thermal boundary condition and uses a hot film sensor for surface heat flux measurements.

The rig operates using point measurements by way of a hot-film sensor, with a geometric area of 0.1mm x 1.4mm, flush mounted on a copper plate which is heated uniformly by a heating mat placed underneath and moves laterally under the impinging jet in order to record data at different positions relative to the jet. This setup creates a nominally uniform wall temperature boundary condition. A Senflex® hot-film sensor operates in conjunction with a Constant Temperature Anemometer (CTA) to measure the local heat flux from the plate to the impinging jet. The hot film is maintained at an overheat above that of the copper plate using a Dantec StreamLine CTA. The power required to maintain the film at this overheat is equal to the heat actively being dissipated from the film. The CTA essentially acts as a Wheatstone bridge where the hot-film acts as one resistor in the bridge. The resistance of the film varies with temperature and therefore, the film temperature can be controlled by
varying a decade resistance which forms another arm of the Wheatstone bridge. The square of the voltage required to maintain the film at a constant temperature is proportional to the heat transferred to the air.

A normal impinging jet has been tested using this method in the past [10, 11] and these results validate the measurement technique. The swirling flow, however, has not previously been tested in this way, so it is anticipated that the greater spatial resolution associated with these measurements may provide additional insight to explain the effect of swirl.

3. Analysis

3.1. Uniform Wall Flux Thermal Boundary Condition

The convective heat flux to the impinging jet is determined from equation (1), which describes the energy balance of the system. This analysis subtracts heat losses such as natural convection, radiation and lateral conduction from the heat generated within the foil. The heat flux generated in the foil is calculated using the controlled current applied across the foil and the resistance of the material.

\[ q_{\text{conv}}^{\prime} = q_{\text{gen}}^{\prime} - q_{\text{conv}}^{\prime} - q_{\text{rad}}^{\prime} + q_{\text{lc}}^{\prime} \]  

Equation (1)

Natural convection heat loss from the underside of the foil is calculated using a Rayleigh number, Ra, correlation specific to a uniform wall flux thermal boundary condition. This correlation estimates the Nusselt number which is then converted into the equivalent heat flux to be used in equation (1). The level of heat lost from the bottom of the foil through natural convection is approximately 8% of the heat flux generated. The radiative heat loss from both the uncoated stainless steel upper side and the lower matt black paint coated side is estimated with an expression which takes into account the emissivity of the material and the temperatures of the material and the surroundings to give the heat flux dissipated. Since the foil has a layer of matt black paint on the lower surface, different values of \( \varepsilon \), the material emissivity, for each side of the foil must be used. Heat losses from the foil and the paint are estimated to be about 2% and 11% of the heat flux generated respectively.

The lateral conduction term, as discussed by Geers et al. [12], can be quantified by:

\[ q_{\text{lc}}^{\prime} = k \cdot t \left( \frac{\delta^2 T}{\delta x^2} (x, y) + \frac{\delta^2 T}{\delta y^2} (x, y) \right) \]

Equation (2)

where \( t \) is the thickness and \( k \) is the thermal conductivity of the material being examined. The lateral conduction heat flux is calculated for both the foil and the layer of matt black paint.

The resulting value of \( q_{\text{conv}}^{\prime} \) represents the heat flux due to the forced convection of the impinging jet on the upper surface of the foil. The equivalent heat transfer coefficient, \( h \), based on Newton’s law of cooling, is calculated. The Nusselt number is calculated once the heat transfer coefficient has been determined and has been estimated to have an uncertainty of approximately 8% at a 95% confidence level.

3.2. Uniform Wall Temperature Thermal Boundary Condition

As mentioned previously, the heat transfer for the UWT boundary condition is assessed using a hot-film sensor and a constant temperature anemometer. To calculate the heat flux using this device, the voltage produced by the sensor was first calibrated against a correlation outlined by Liu and Sullivan [13]:

\[ N_{u_o} \left( \frac{P_{o}^{0.4} \cdot R_{e}^{0.5}}{D} \right)^{-1} = 0.585 \]

Equation (3)

where \( N_{u_o} \) is the Nusselt number at the stagnation point of a steady impinging jet. In a paper by O’Donovan et al. [11], a detailed explanation into the method of using a hot-film sensor to estimate the heat dissipated is featured. This explanation includes comprehensive calculations of the effective area of the hot-film sensor which must be accurately determined. The effective area of the hot-film sensor being examined.
sensor is commonly found to be a number of times larger than the geometric area of the sensor since it operates at a higher temperature than the surface on which it is placed, known as the overheat. The effective area of the hot-film sensor used in this study is calculated using characteristics specific to the CTA, such as resistances of the Wheatstone bridge, to be of the order of five times the geometric area of the sensor. Combined with the corresponding voltage produced by the sensor to keep it at its set overheat during adiabatic and heated tests the resultant Nusselt number distribution for each scenario is generated and is estimated to have an uncertainty level of approximately 7%.

The resulting Nusselt number distributions from both series’ of experiments are then compared to one another to determine the possible effect the different thermal boundary condition may have.

4. Results & Discussion

For both experimental set-ups, Nusselt number distributions were produced up to a radial distance of ten nozzle diameters from the stagnation point, $0<r/D<10$. Representative results from the full test programme are presented here. In sections 4.1 to 4.3, the effect of the thermal boundary condition is explored for the plain jet with no swirl whereas the influence of swirl is assessed in section 4.4.

4.1. Uniform Wall Flux Thermal Boundary Condition: Case 1

For the uniform wall flux case, Figure 4 shows the effect of varying H/D for a fixed Reynolds number whereas Figure 5 shows the effect of varying Reynolds number at a fixed jet to surface spacing. Both sets of data refer to the plain jet with no swirl.

Figure 4 shows how the Nusselt number distribution changes form as the height of the nozzle above the impingement surface is increased. At low nozzle to surface distances, common features include a local minimum at the stagnation point and a secondary peak due to the impinging flow’s transition into a turbulent flow along the surface and shedding vortices [1]. In some cases a tertiary peak is also produced. It is believed that this additional peak can be attributed to the geometry of the jet nozzle and a confinement issue which may arise due to it. As the jet moves away from the surface these characteristics become less prominent and a bell-shaped curve with a single local maximum at the stagnation point is produced.

Figure 5 shows the expected increase in Nusselt number as the Reynolds number is increased, with all curves containing the characteristic secondary peaks associated with the low H/D of 2.
4.2. Uniform Wall Temperature Thermal Boundary Condition & Hot-Film Sensor Analysis: Case 2

The hot-film sensor technique allows the same study to be repeated for a uniform wall temperature thermal boundary condition. Figure 6 shows the results for a Reynolds number of 14000 at varied heights, while Figure 7 shows the Nusselt number distribution for a height of 2D at each Reynolds number tested and the relative change that occurs as the Reynolds number is increased. Figure 6 clearly shows the local minimum at the stagnation point followed by a secondary and tertiary peak for a low H/D. As the height is increased these features give way to a bell-shaped curve with a local maximum at the stagnation point.

![Figure 6. Non-Swirling Jet at Re=14000 for H/D=0.5 & 8 for Case 2](image1)

![Figure 7. Non-Swirling Jet at H/D=2 for all Reynolds numbers for Case 2](image2)

The results show common trends with varying Reynolds number and H/D for both cases. With the height fixed, both Figure 5 and Figure 7 show an increase in heat transfer with increasing Reynolds number. Likewise, Figure 4 and Figure 6 both show the reduction in the local minimum at stagnation, and in the secondary peaks, as H/D is increased.

4.3. Comparison of Thermal Boundary Conditions for each Impinging Jet Configuration

Figures 8 to 12 represent the measured Nusselt number distributions from the two measurement set-ups for the five impinging jet configurations, one with no swirl and four with swirl. Each figure shown here includes the results obtained with infrared thermography (uniform wall flux) and hot-film sensor analysis (uniform wall temperature). For the full range of tests conducted, both cases provide broadly similar findings with regard to local maxima and minima but there is a discrepancy in magnitudes, mainly in the stagnation region.

Preliminary indications show that a possible reason behind these local differences lies more in the spatial resolution of the measurement technique being used than in the thermal boundary condition. This contention is based on the fact that the main differences between the results, seen in figures 8 to 12, lie predominantly in the stagnation region while the boundary condition would be a more significant factor with increasing distance from the stagnation point.

In this study, the hot-film sensor has a higher special resolution in the radial direction, ~0.1mm, than the thermal imaging camera, where one pixel represents ~0.5mm. Further examinations involving higher resolution thermal imaging cameras with Case 1 and lower resolution MicroFoil heat flux sensors with Case 2 are planned to provide support for this hypothesis.
4.4. Comparison of Swirling and Non-Swirling Results

When comparing the swirling impinging jets to a non-swirling jet, the results have shown that the swirling impinging jets produce higher levels of heat transfer around the stagnation region, primarily for low nozzle to surface heights and high Reynolds numbers. Depending on the measurement technique used, however, the level of enhancement may change. Figures 13 and 14 illustrate this point. Figure 13 shows a comparison between the five jet configurations using infrared thermography at a height of 0.5D and a Reynolds number of 16000 and figure 14 shows the same comparison based on the hot-film technique. The infrared technique suggests a clear enhancement in heat transfer for each of the swirling jets over the jet with no swirl. In Case 2, however, this result is not as apparent. Thus, there is still some enhancement at r/D~0.75, but the hot film has also detected a reduction in the stagnation point Nusselt number for some of the swirling jet results.

This reduction is attributed to recirculation of the fluid as a result of the radial motion of the swirling jets, which has been seen using flow visualisation techniques [14]. This recirculation causes a localized drop in heat transfer which is not as easily identified from the thermal camera used in this study due to its lower spatial resolution.
5. Conclusion

Using different experimental techniques to characterise heat transfer for impinging jet flows with and without swirl, a number of findings have been made:

The thermal boundary condition which is used for experimental investigations is not considered to significantly influence an impinging jet’s performance; rather the results obtained from experimental tests are sensitive to the measurement technique used and its resolution.

With regard to the effect of swirl in an impinging jet, it has been found that a swirling impinging jet enhances the heat transfer of a system primarily in the stagnation region, especially at low nozzle to surface heights and high Reynolds numbers. At high nozzle to surface spacings, the flow from a swirling impinging jet is dispersed over a larger area and its impact on the impingement surface is not as effective as a jet without swirl.

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