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Flow field and thermal behaviour in swirling and non-swirling turbulent impinging jets

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ABSTRACT

The fundamental interaction between the mean velocity field, turbulence and the impingement surface characteristics are presented. The nozzle used achieves a seamless transition from non-swirling ($S = 0$) to highly swirling jets ($S = 1.05$). Convective heat transfer measurements on the impingement surface are performed using infrared thermography. Numerical simulations are carried out using ANSYS FLUENT 14.5 via SST k-ω turbulence model. The effect of swirl number and impingement distance ($H = 2D$ and $6D$) on the heat transfer characteristics are investigated at a Reynolds number (Re) of 35,000.

Results show the effects of swirl on impingement heat transfer depend on impingement distance. In the near-field ($H = 2D$), high jet turbulence ($u'u'$ and $w'w'$) close to the surface (0.8 mm upstream) correlate very well with Nusselt number peaks resolved on the heated surface. The occurrence of any pockets of low turbulent kinetic energy (k) near the surface may cause localised Nu trough, which can also be correlated with the presence of swirl induced recirculation zones if they stabilise on the surface. Alternatively, in the case of far-field impingement ($H = 6D$), swirl causes wider jet spread and hence turbulence levels are reduced. At this distance, non-swirling jets thus yield higher Nu zones at the surface.

1. Introduction

Turbulent impinging jets are used in numerous industrial applications due to their higher effectiveness in heat and mass transfer rates. The existence of three independent flow regions, surface interacting flow curvatures and near-wall turbulence makes impinging jet problems challenging, and are attracted to the numerical research for a test case of modelling methodologies. Swirling jets are also investigated in many studies for their strong mixing characteristics. They are often compared to their non-swirling counterparts with an aim of understanding how swirl affects heat transfer on the impingement surface. However, the use of various swirl generating mechanisms (even for the comparable upstream flow conditions) has led to inadequate deductions between swirl and heat transfer improvement (both magnitude and uniformity) on the impingement surface [1–4].

Numerous studies, including classical reviews and recent treatises [5–9], investigate fluid flow behaviour and heat transfer characteristics between a nozzle and an impingement surface for axisymmetric non-swirling impinging jets. The literature reveals the potential core length and the impingement region varies with nozzle-to-plate distance (H) when a turbulent jet impinges at a distance less than six nozzle diameters (D) i.e. $H < 6D$. Flow entrainment outside the conical potential core and vortical structures (due to shear layer instability) then affect the impingement and wall jet regions. For $H < 6D$, the heat transfer distribution on the surface does not show a monotonic decrease, with two Nusselt number maxima within a radial distance of $r/D < 2.5$. The outer peak is located in the radial range $r/D = 1.7$–2.5 [8,10,11], whereas the inner peak is found to occur either on the jet axis ($r/D = 0$) [12,13] or at $r/D \approx 0.5$ with a local minimum on the jet axis [2,14]. The exact reasons for these heat transfer peaks are not unique and several plausible physical explanations have been proposed experimentally in the literature [7,14–17]. More recently advanced numerical simulations were performed using LES [18,19] at $H/D = 2$ for better understanding the physical explanation of the Nu secondary peak, but no consensus was found for the exact reason of the peak. Whilst the former group argued the flow acceleration in the developing region of the boundary layer is a cause of the secondary peak,

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the latter group believed to be the enhanced turbulence production from wall-attached eddies in that region. Additionally, velocity profiles at the nozzle exit was also found a strong influence on the impingement heat transfer characteristics [7]. Contradictions in the results for non-swirling jets may primarily be caused by the varied nozzle exit conditions in these studies.

Existing research on swirling impinging jets predominantly used geometrically generated swirl (using helical inserts or guide vanes within a nozzle), and reported both a reduction [3,20–22] or an enhancement [23–25] of average (area integrated) heat transfer compared to non-swirling counterparts. Heat transfer reduction is largely ascribed to the geometry induced dead-zone, typically around the jet centre. In contrast, intense flow mixing and formation of vortices on the impingement surface are found to contribute to the heat transfer enhancements. For the radial uniformity of impingement heat transfer, the literature disagrees for the relationship between radial uniformity (flatness) of heat transfer and swirl [2,3,26]. Although geometry induced intricacies, such as flow blockages and perturbations can be avoided by aerodynamically generated swirl jets, but the limited number of these studies lead to poor understanding of the fundamental relationship between swirl and heat transfer. Moreover, contradictory results in relation to heat transfer improvement and radial uniformity for increasing swirl intensity also exist [25–27]. Substantially different Reynolds numbers, investigation of limited swirl intensities and lack of precise upstream conditions among these studies may contribute to such discrepancies in the results. Similar to the non-swirling jets, disagreement of the radial location of heat transfer peaks with swirl intensity also varies, regardless of the swirl generation. However, explanations for such occurrences of heat transfer peaks at different swirl intensities are not adequately addressed in the literature. This reinforces the need to investigate a swirling impinging jet for a wider range of swirl intensities with well-defined boundary conditions to improve the understanding between flow field characteristics and heat transfer.

Although extensive numerical research on non-swirling, turbulent impinging jets is available in the literature [9,28–32], the computations for swirling impinging jets are still scarce. Even for the non-swirling impinging jets, the simulations are found to be challenging to resolve complex flow behaviours near the impingement surface, such as steep pressure gradients and anisotropic flow nature due to the jet-wall interaction [31]. Moreover, choosing a turbulence model is also an issue since no turbulence model was found to predict accurately all the flow features of an impinging jet [33–35]. The inclusion of swirl into the jet exacerbates the modelling complexity of turbulent flow fields and heat transfer characteristics. The absence of highly resolved flow field data (for benchmarking) as well as clearly defined nozzle exit boundary conditions is another drawback of swirling impinging jets computations. Despite these difficulty, the limited works conducted recently on turbulent swirling impinging jets (geometrically generated swirl), are the studies by Ortega-Casanova [36], Amini et al. [37], and Wannassi and Monnoyer [38]. Likewise, similar to experimental investigations, numerical studies show disparity in outcomes in relation to heat transfer improvement.

Previous research shows contradicting heat transfer results regardless of the swirl generating mechanisms and/or investigation methods, i.e. experimental or numerical. A fundamental understanding for the relationship between swirl intensity and flow fields, and the underlying mechanism of heat transfer peaks appears inadequately reported. As such, this research experimentally and numerically investigates swirling jets (aerodynamically generated), which impinge on a surface located at $H = 2D$ and $H = 6D$ for a Reynolds number of 35,000. This paper uses well-defined nozzle exit conditions derived via Constant Temperature Anemometry (CTA) and provides the inlet boundary conditions for CFD simulations. Section 2 briefly details both the experimental and numerical methodologies. Section 3 discusses the results followed by the conclusions in Section 4.

2. Methodology

2.1. Experimental techniques

The air jets are obtained from a specially designed swirl nozzle [39] which can deliver non-swirling and (aerodynamically induced) swirling jets without the use of geometric inserts/vanes. The nozzle has two axial and three tangential ports to control both the total flow at the exit plane (Reynolds number) or their relative proportions so as to change swirl number (independent of Re), and vice versa. The two 12 mm axial ports are positioned diametrically opposed to each other and the three 12 mm tangential ports are circumferentially 120° apart from other. Tangential ports are 20° upwards (off the horizontal) and also have a 15° rotation about the tangential port axis. More details on the nozzle configuration, its internal cavities and general design features is available in the literature [39–41]. The nozzle exits with a straight section of diameter $D = 40$ mm and a sharp-edged termination with a wall thickness of 0.2 mm.

An X-wire probe (DANTEC, model: 55P61) was used to characterise the nozzle exit conditions via measurements of axial and azimuthal velocity components 1 mm above the nozzle exit plane. The detail of the hotwire measurement methodologies for low-to-high swirl intensity is demonstrated in our another paper [42]. For non-swirling and swirling flows, the estimated accuracy of CTA measurements was 2% and 4% of centreline velocity, respectively.

Convective heat transfer measurements between the jets and the impingement surface are performed by the steady-state heated thin foil technique [43], with surface temperatures being resolved via infrared thermography. In this regard, the jet flows vertically upward from the swirl nozzle and impinges on a heated horizontal surface. An infrared camera (FLIR systems, model: A325) is positioned above the surface, as shown in Fig. 1, to measure two-dimensional temperature distributions. A 25 μm thick and 320 × 200 mm stainless steel foil is used as the impingement surface and heated by a high current DC power source (Powertech, model: MP3094) so as to establish a constant surface heat flux through Joule heating at 40 A and 3 V. The thin foil is painted flat (matt) black on steel foil surface which faces the infrared camera to attain a high emissivity. The other side of the foil (nozzle facing) is left unpainted. The foil is considered isothermal across its thickness since the Biot Number is less than 0.1 [44]. Prior to data acquisition, the emissivity coefficients for both the unpainted and painted faces of the impingement surface are measured by a separate experiment, with black insulating tape (emissivity 0.98) as a reference [45,46]. In this regard, a foil strip was painted flat black at one end, left unpainted at the other end and the tape was mounted in the middle. The foil strip was then placed in a thermostatic-controlled constant temperature water bath (MATEST, Model: B051-01) that was heated from ambient to 60 °C (at steps of 10 °C). The emissivity of painted and unpainted surfaces were then determined by a method outlined in Ref. [46], and found to be 0.97 and 0.06, respectively. In the jet experiments, once steady-state conditions were established on the heated impingement plate (i.e. temperatures stabilise), data was acquired at a rate of 5 Hz for 30 s with a total of 150 thermal images averaged to represent each jet condition. This imaged data was then post-processed using MATLAB (version 2012b). Further detail of heat transfer measurement methods is discussed in Ref. [40] and is not repeated here.