Impingement Cooling by Round Jets with a Longitudinal Swirling Strip

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***Abstract*—The objective of this research is to study the effect of swirling impinging jets on the heat transfer in an impinged plate using a thermochromic liquid crystal (TLC) sheet. In the experiments, a nozzle with a diameter of 20 mm (D) was installed with a four-channel twisted tape as a longitudinal swirling strip. Jet impingement was performed at jet-to-plate distances (L/D) of 2, 3, 4, 5, 6, and 7 and a constant Reynolds number (Re) of 5,000. The experimental results showed that the swirling jet (SJ) induced by the four-channel twisted tape gave higher heat transfer rates than the conventional jet (CJ) at the same operating conditions. The highest average heat transfer rate was obtained at a jet-to-plate distance (L/D) of 2.**

***Index Terms*—Heat transfer, thermochromic liquid crystal, swirling impinging jet, swirl generator.**

I. Introduction

The manufacturing of automotive parts is currently growing fast, and intensive quenching is regarded as an important issue. Rapid cooling helps improve the mechanical properties and reduce cracking and distortion of the parts. Heat transfer enhancement techniques offer numerous benefits in automotive applications [1–10]. One of the effective intensive quenching techniques is jet impingement. This technique has been extensively developed and reported [1–10]. One of the effective intensive quenching techniques is jet impingement. This technique has been extensively developed and reported [1–3, 5, 7, 9]. Colucci and Vikanta [11] developed two modified nozzles: a sharp-edged orifice and a hyperbolic nozzle for jet impingement. They performed their experiments at a jet-to-plate distance (L/D) of 0.25–6.0 and a Reynolds numbers (Re) ranging from 10,000 to 50,000. They found that the two modified nozzles offered superior heat transfer to conventional nozzle and that the heat transfer enhancement increased with increasing Re. Huang and EL-Genk [12] investigated heat transfer using multichannel swirl generators with swirl angles of θ = 15°, 30°, and 45°. They reported that the nozzle installed with swirl generators gave a higher heat transfer rate than the conventional nozzle. The highest heat transfer rate was achieved at a swirl angle (θ) of 15°, which was in accord with that found by Lee et al. [13]. Wen and Jang [14] studied the heat transfer of circular jet nozzles fitted with longitudinal swirling strips with two and four channels. They observed that the swirling strip with four channels gave higher heat transfer than the one with two channels. Alekseenko et al. [15] studied the local structure of turbulent swirling impinging jets using a stereo PIV technique with advanced pre- and postprocessing algorithms. They reported that the swirling impinging jet possessed greater spread rates and faster decay in absolute velocity compared with conventional jets (CJs). Bakirci and Bilen [16] studied the effect of impinging swirling flow on heat transfer characteristics using a liquid crystal technique. They found that the optimum result was achieved at a swirl angle of θ = 50° and a jet-to-surface distance (L/D) of 14. Wongcharee et al. [17] reported the effect of swirling impinging jets with TiO_2-water nanofluids on the heat transfer rate. Their experimental results revealed that nanofluids with concentrations below 2.0 %vol. provide higher heat transfer rates than base fluids, whereas nanofluids with concentrations above 2.0 %vol. showed the opposite trend. According to a literature review, multichannel swirl generators show a promising performance in heat transfer enhancement. However, explanation of the heat transfer behavior is rarely reported. In the present work, a thermochromic liquid crystal (TLC) sheet was used to record the approximate wall temperature and the heat transfer rate of the swirling jet (SJ) impingement for a better understanding of the heat transfer behavior.

II. Experimental Facility

A schematic of the experimental setup for heat transfer visualization with a TLC sheet is shown in Fig. 1. The facility consisted of a high-pressure blower that pressurized air into the pipe nozzle, where the air flow rate was controlled by adapting the motor's speed via an inverter. The air flow rate was measured by an orifice flow meter coupled with a digital pressure gauge to measure the pressure drop. The volumetric air flow rate

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through the pipe nozzle was kept constant, corresponding to a Reynolds number of 5,000 at an air temperature of 27 ℃. The air temperature was controlled by a Silicon-Controlled Rectifier (SCR) via a temperature controller connected to an RTD for measuring the inlet air temperature at the pipe nozzle's entry.

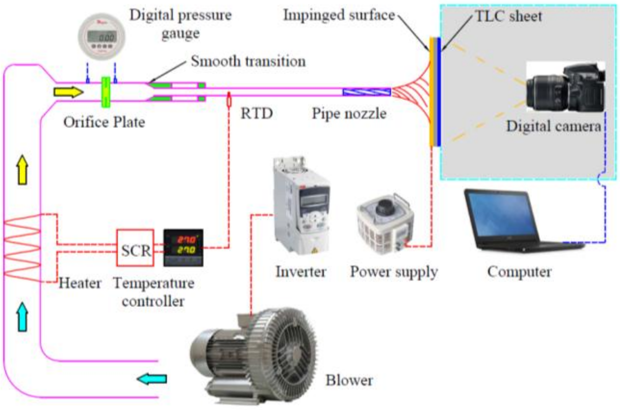


Figure 1. Schematic diagram of the experimental apparatus.

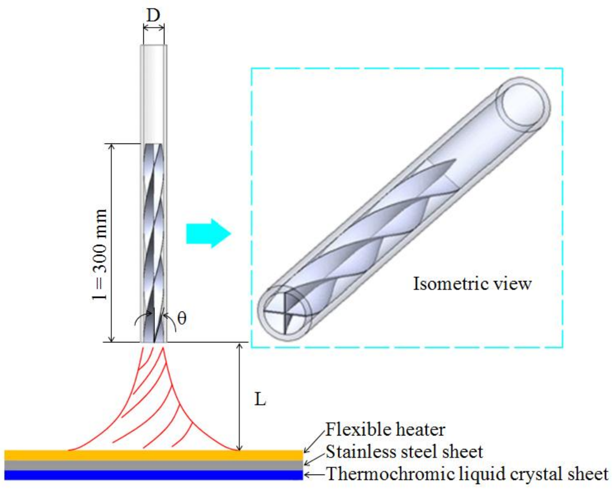


Figure 2. Round jet with a longitudinal swirling strip.

A pipe nozzle with a diameter of 20 mm was made of clear UPVC with a length of 1,000 mm (50D). The swirl generator (twisted tape) had four channels with a swirl angle (θ) of 15° and a jet-to-plate distance (L/D) of 2, 3, 4, 5, 6, and 7, as shown in Fig. 2. The impingement plate was made of a stainless steel sheet with a thickness of 0.15 mm that was attached to a flexible heater (Omega, KH-1212/10), which had a width of 300 mm and a length of 300 mm. The rear of the impingement plate was attached to a TLC sheet (Omega, LCS-95) for recording the wall temperature (T_w) pattern via a digital camera. Then, the temperature data were transferred to a personal computer. Images of a color pattern on the TLC sheet were converted from the RGB (red, green, and blue) system to the HSI (hue, saturation, and intensity) system. The hue (H) component was then correlated to the temperature via a temperature calibration equation obtained from the TLC sheet calibration experiment.

III. Data Reduction

Electrical energy dissipated in the test section can be calculated from Joule's effect as follows:

$$\dot{q}_{input} = \frac{I^2 R}{A}. \tag{1}$$

where I, R, and A are the supplied electrical current, the electrical resistance of the heater sheet, and the heat transfer surface area, respectively. The input electrical power dissipating on the heated surface was fixed at 319 W/m^2 for all experiments.

Consequently, the local heat transfer coefficient by forced convection (h), due to jet impingement, can be evaluated from the energy balance in the small element of the impinged plate, which is in conjunction with the TLC sheet as

$$h = \frac{\dot{q}_{input} - \dot{q}_{loss,convection} - \dot{q}_{loss,radiation}}{T_w - T_j}. \tag{2}$$

The local Nusselt number can be expressed as

$$Nu = \frac{hD}{k}. \tag{3}$$

The Reynolds number can be calculated from

$$\mathrm{Re} = \frac{\rho U D}{\mu}. \tag{4}$$

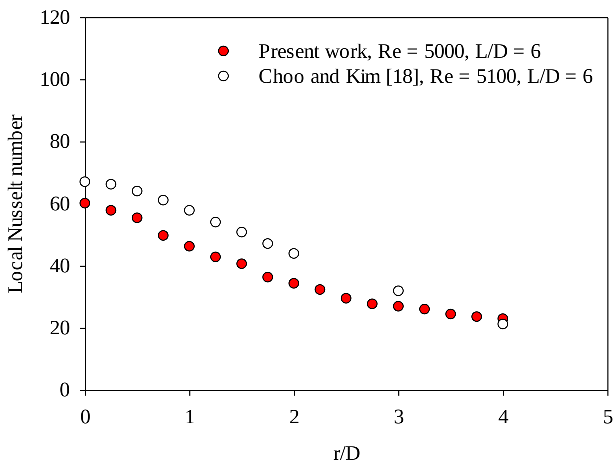
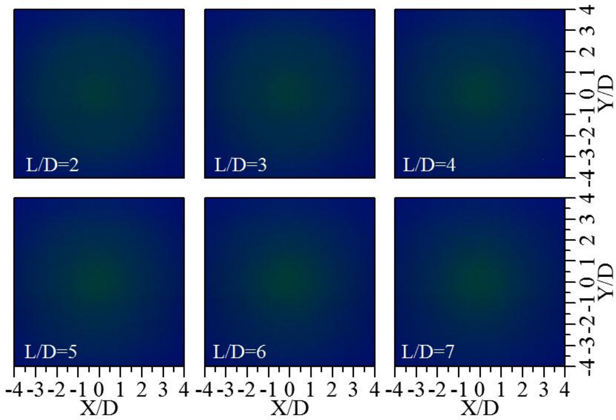


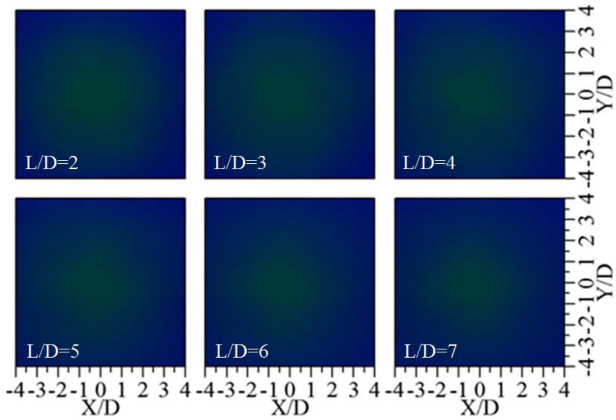
Figure 3. Comparison of the local Nusselt number with the data of Choo and Kim [18] for CJs.

IV. Experimental Results

To assess the reliability of the results obtained in the present work, the heat transfer rates in terms of Nusselt number (Nu) of a conventional impinging jet (Re = 5,000, L/D = 6) were compared with those of Choo and Kim [18] under similar operating conditions (Re = 5,100, L/D = 6), as shown in Fig. 3. The comparison revealed that the present data of both local and average Nusselt numbers were in good agreement with those of Choo and Kim [18].

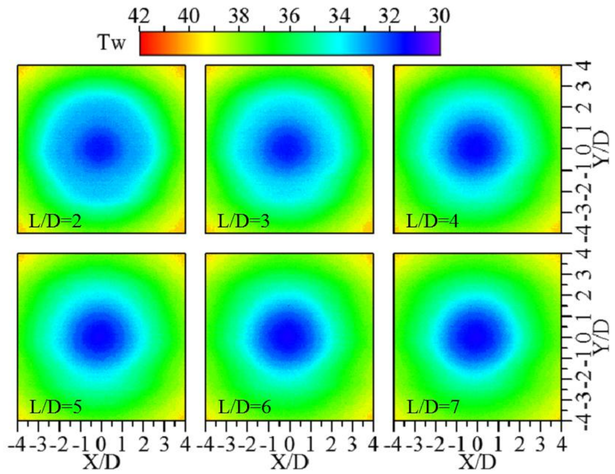


(a) Conventional impinging jet

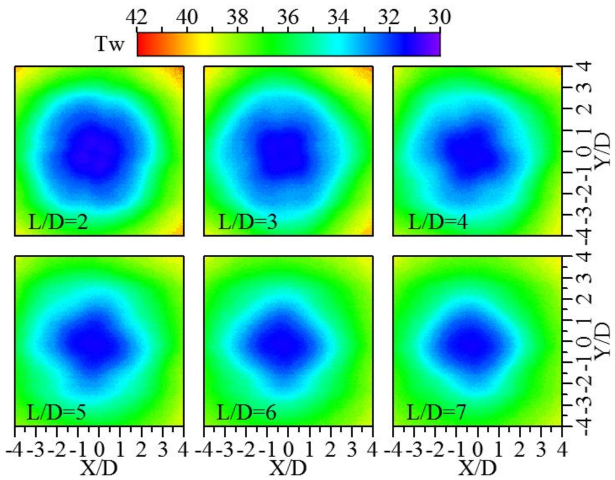


(b) Swirling impinging jet

Figure 4. Temperature distribution on the impinged plate by TLC.



(a) Conventional impinging jet



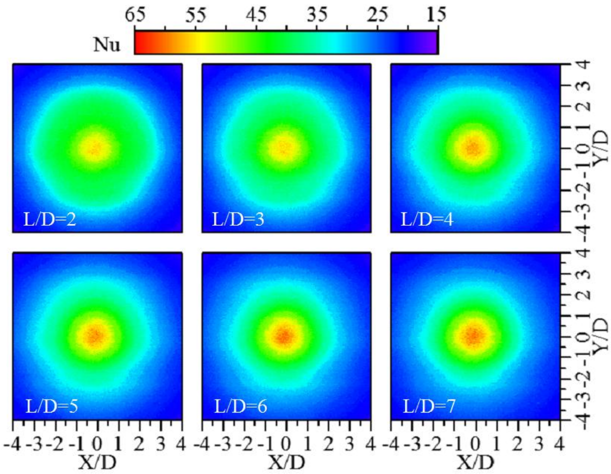
(b) Swirling impinging jet

Figure 5. Temperature distribution on the impinged plate.

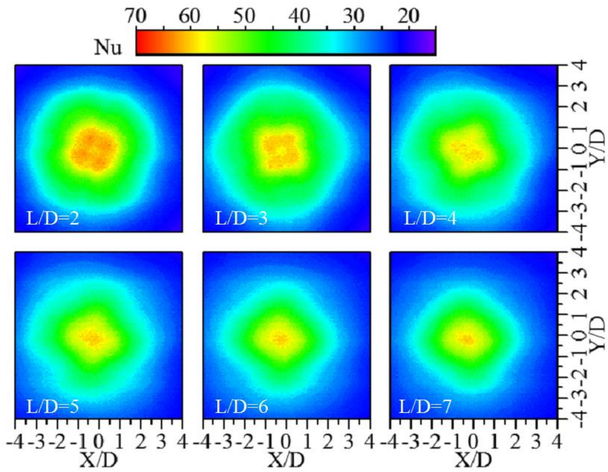
Figures 4(a), 5(a), 6(a), and 7(a) present the temperature and Nusselt number distributions in the impinged plate for a conventional impinging jet. Evidently, low-temperature regions existed around stagnation points (X/D = 0.0, Y/D = 0.0) as demonstrated in Figs. 4(a) and 5(a), which correspond to the high local Nusselt numbers (Figs. 6(a) and 7(a); due to the direct cooling effect). In addition, Nusselt number diminished along the radial direction, and the secondary peak heat transfer appeared in a donut shape [Figs. 6(a) and 7(a)], especially at a low jet-to-plate distance (L/D). This is attributed to the transition of the low turbulence in a stagnation region to a turbulent wall jet.

Figures 4(b), 5(b), 6(b), and 7(b) demonstrate the temperature and Nusselt number distributions in the impinged surface with the swirling impinging jet. Interestingly, the impinged area for the swirling impinging jet was in an asymmetrical shape because of the swirling effect. The intensified heat transfer regions were separated, attributed to the blockage by the twisted tape ridge (swirl generator). The separation became notable as the jet-to-plate distance decreased. It should also be mentioned that the formation of recirculation flow near the wall because of the swirl flow helps reduce the Nusselt number difference between the stagnation region and the surrounding area, resulting in more uniform distributions of Nusselt number, as compared to those found in CJ.

The effect of the jet-to-plate distances (L/D = 2, 3, 4, 5, 6, and 7) at the same Reynolds number on the average Nusselt number is shown in Fig. 8. It was found that the average Nusselt number given by swirling impinging jets and conventional impinging jets increased with decreasing jet-to-plate distances (L/D). At the smallest jet-to-plate distance (L/D) of 2, the swirling impinging jets and conventional impinging jets yielded the maximum average Nusselt number of 33.4 and 30.5, respectively. For CJs, the average Nusselt number given by the jets with jet-to-plate distances (L/D) of 2, 3, 4, 5, and 6 was, respectively, 10.1%, 7.9%, 7.2%, 6.1%, and 4.7% higher than that at L/D = 7. For SJs, the average Nusselt number given by the jets with jet-to-plate distances of L/D = 2, 3, 4, 5, and 6 was, respectively, 11.3%, 11.0%, 9.3%, 6.3%, and 2.7% higher than that at L/D = 7. At L/D = 2, 3, 4, 5, 6, and 7, the SJs yielded higher average Nusselt numbers than the corresponding CJs by around 9.5%, 11.4%, 10.4%, 8.5%, 6.2%, and 4.5%, respectively. The results indicated the poorer enhancement of SJs as compared to the CJs with increasing jet-to-plate distances (L/D). From this, it can be explained that, at large L/D, a swirling (spreading) effect causes a dramatic reduction of the axial velocity before impingement. This leads to the diminishment of both impingement momentum and impinged area, resulting in poor heat transfer on the impinged surface.

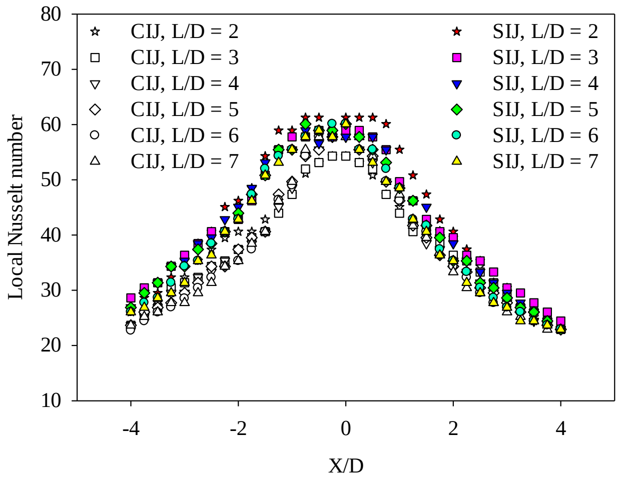


(a) Conventional impinging jet



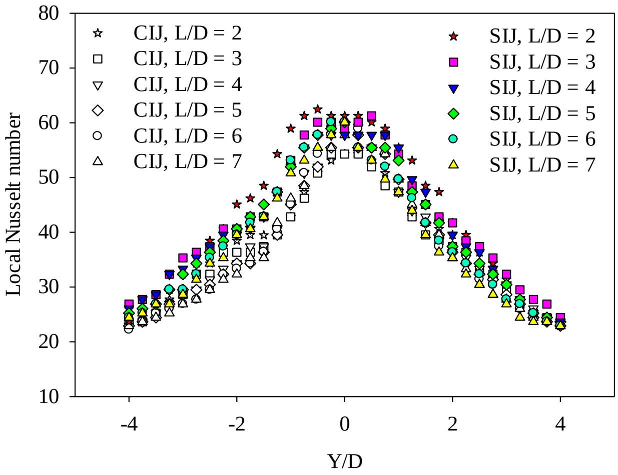
(b) Swirling impinging jet

Figure 6. Nusselt number distribution on the impinged plate.



(a) *x*-axis

Figure 7. Effect of jet-to-plate distance (L/D) on the local Nusselt number distribution.



(b) y-axis

Figure 8. Effect of jet-to-plate distance (L/D) on the local Nusselt number distribution (continued).

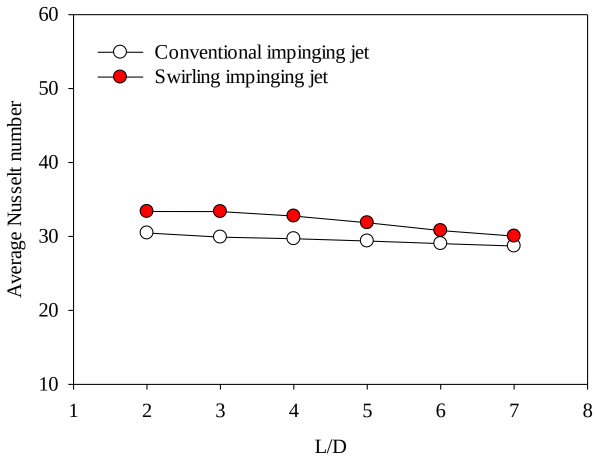


Figure 9. Average Nusselt numbers on the impinged plate.

V. Conclusions

The heat transfer behaviors of SJs and CJs were comparatively studied using a TLC sheet and image processing technique. At similar operating conditions, the SJ consistently yielded higher heat transfer rates than the CJ. At L/D = 2, 3, 4, 5, 6, and 7, the average Nusselt numbers given by the SJ were higher than those of the CJ by around 9.5%, 11.4%, 10.4%, 8.5%, 6.2%, and 4.5%, respectively. For SJs, the Nusselt number peaks at L/D = 2, 3, 4, 5, and 6 were, respectively, 6.1%, 4.7%, 3.4%, 2.0%, and 1.4% higher than that at L/D = 7.

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