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Research Article

HEAT TRANSFER OF SWIRLING JET IMPINGING ON A FLAT SURFACE WITH SWIRL GENERATORS

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ABSTRACT:
This paper presents a study on heat transfer enhancement by swirling impinging jets. The swirling jets were induced by inserting dual twisted tapes into jet nozzles. The effects of jet Reynolds number (4000 < Re < 16,000), jet-to-plate spacing (2 < L/D < 8), tape twist ratio (y/W = 3, 4, 5, 6) and swirling direction were experimentally investigated. Numerical visualization was also carried out to study a behavior of the flow phenomena of swirling jets.

Keywords: *Heat transfer, heat transfer enhancement, impinging jet, swirl jets, twisted tape*

1. INTRODUCTION

Jet impingement is one of the heat transfer enhancement techniques. This technique has been widely applied in many engineering applications such as cooling of hot steel plate, tempering of glass, drying of food products, papers, textiles and films, cooling of turbine blades, cooling of electronic equipment, cooling of outer wall of the combustion chamber. In common, an impinging jet brings extremely high heat transfer rate around a stagnation region. Then, the heat transfer rate rapidly decays in a wall jet region due to the development of thermal boundary layer on the impinged surface. Heat transfer performance of impinging jets is influenced by several factors as reported in published research papers. Martin [1] investigated the effect of the single round nozzle, arrays of round nozzles, single slot nozzle, and arrays of slot nozzles on the heat and mass transfer of impinging gas jets. Caliskan et al. [2] reported the influence of jet geometry (circular, elliptic and rectangular jets) on the flow and the heat transfer behaviors by using thermal infrared camera and Laser-Doppler Anemometry (LDA) system. They found that elliptic jets gave more efficient heat transfer than the circular jet due to a larger entrainment rate and larger scale coherent structure. Nanan et al. [3] study forced convective heat transfer on an impinged plate associated with swirling impinging jets. The swirling jets were induced using twisted tapes inserts with different twist ratios and jet-to-plate spacings. It was found that at the small jet-to-plate spacings (*L/D* = 2 and 4), swirling impinging jet provided higher heat transfer rate than conventional impinging jet while at large jet-to-plate spacings (*L/D* = 6 and 8), the opposed result was obtained. In addition, only swirling impinging jet induced by the twisted tape at a twist ratio of 6 consistently gave higher average Nusselt numbers than the conventional impinging jet. Wang et al. [4] studied the local heat transfer characteristics on a circular cylinder subject to a circular impinging jet in crossflow at different cylinder-to-jet diameter ratios. Their results revealed that a smaller cylinder behaved as if immersed in uniform free-stream-flow separation which caused local minimum heat transfer.

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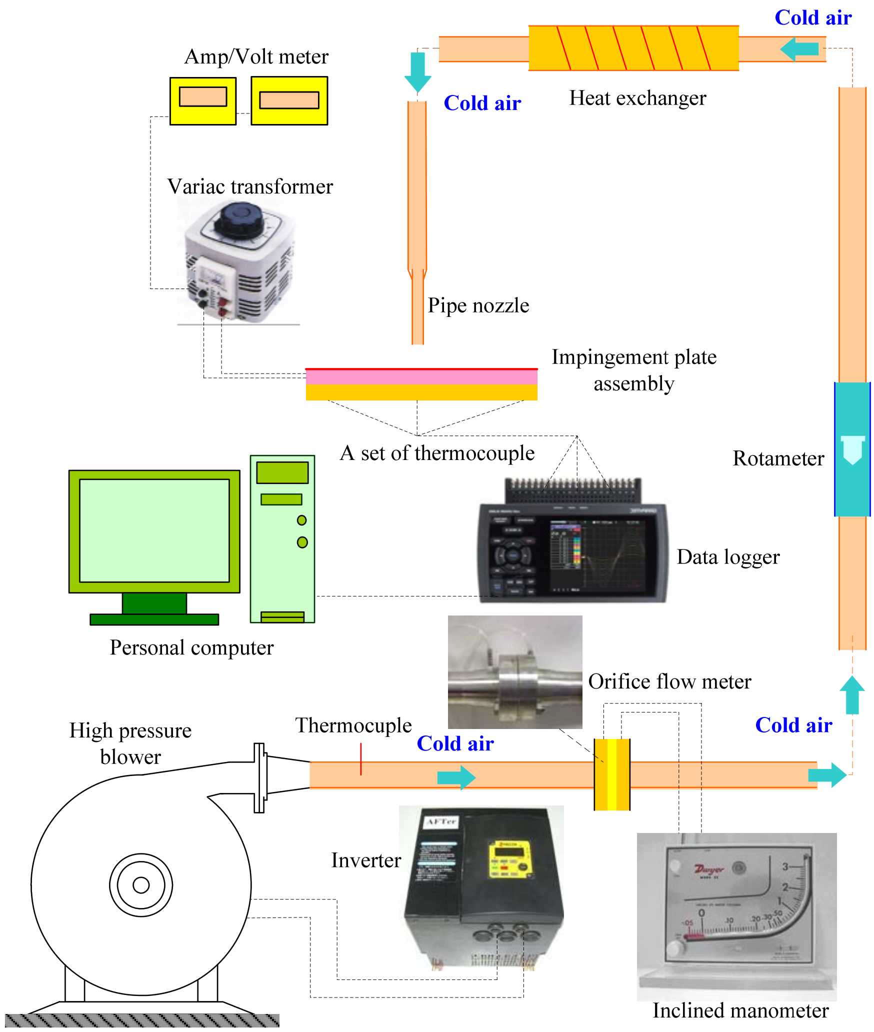
San and Chen [5] investigated the heat transfer distributions on a flat surface at five different confined circular air jets vertically impinging. The jets were arranged as an equilaterally staggered array (a middle jet and four neighboring jets). It was found that that heat transfer rate increased as *s/d* and *H/d* decreased, attributed to a stronger flow impact (jet interaction) on the impingement plate. In the present study, the heat transfer and flow behaviors of dual swirling jets on impinged surfaces are examined. The experiments were conducted with four different nozzle-to-plate spacings and twist ratios of the tape. The conventional impinging jets were also tested, for comparison. The flow patterns on an impinged surface were visualized using numerical technique. Experiments were performed for jet Reynolds numbers ranging 4000 to 16,000.

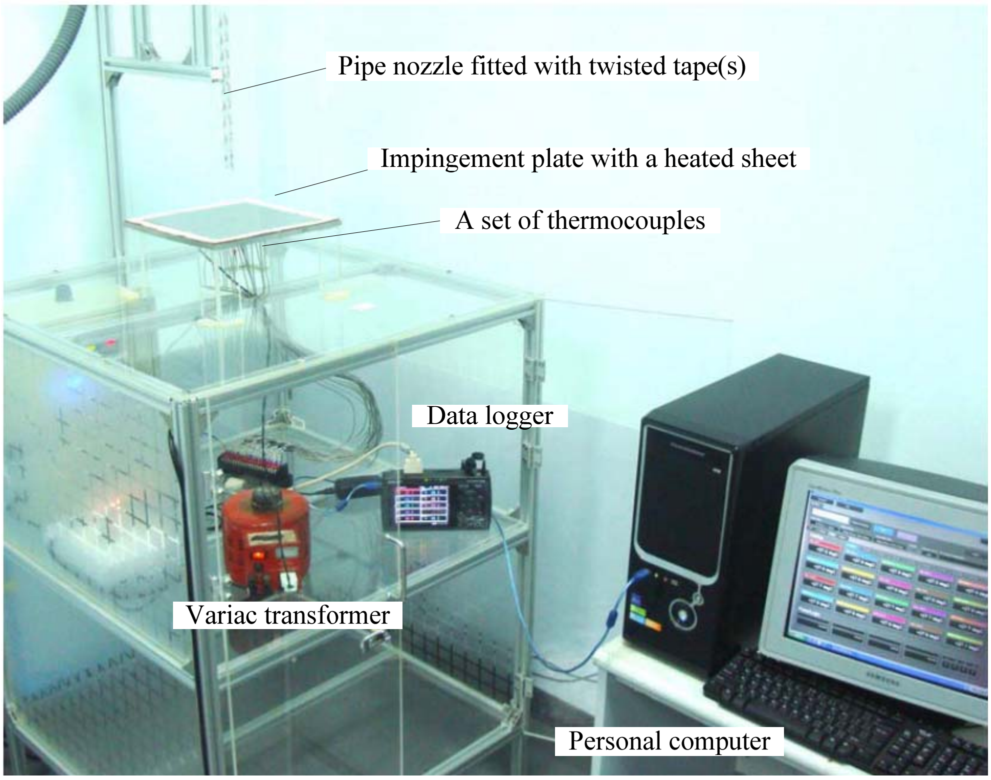
2. Experimental Facility

The experimental facility is shown in Fig. 1. The system consisted of (1), high pressure blower, (2) three phase inverter for controlling the speed of motor, (3) orifice flow meter for measuring the volumetric air flow rate, (4) rotameter for calibrating air flow rate, (5) heat exchanger for controlling an inlet air temperature (6) pipe nozzle (400 mm long and 2 mm thick), (7) twisted tape inserts (8) a set of impingement plate consisting of stainless sheet, heating sheet, insulation and a set of thermocouples, (9) data logger for collecting temperatures of the impinged plate, inlet air and ambient air, (10) variac transformer for controlling the voltage of the heating sheet, (11) amp and voltage meters for checking the inlet power, and (12) personal computer. The impinged plate (Fig. 2) consisted of (1) a stainless sheet or an impinged plate, (2) an electric heater sheet, (3) a thermocouple set, (4) an insulation material, and (5) a jet nozzle.



(a)

Fig. 1. Layout and photograph of experimental facility.



(b)

Fig. 1. Layout and photograph of experimental facility. (Cont.)

The impinged plate had a width of 250 mm and a length of 250 mm. The electric heater was made of stainless steel sheet with a width of 150 mm and a length of 300 mm. The heater sheet was clamped tightly and starched-stretched between 4 copper bus bars with the dimension of 300×300 mm^2 and 1 mm thick stainless steel sheet. Because the stainless steel heater plate was extremely thin, the lateral conduction was negligible and its surface was assumed to be under uniform heat flux condition. The heater sheet was embedded by 20 T-type thermocouples using a thermally well-conductive paste as an adhesive layer to diminish the disturbance on signal of the thermocouple by the electric heater sheet. The diameter of the thermocouple was 0.4 mm while the bead was smaller than 0.25 mm. The power supplied to the heater sheet was measured by an accurate digital power meter. The spacing between two adjacent thermocouples was 5 mm. The circular nozzle having a diameter of 21 mm (*D*) and a length of 1000 mm (48*D*), located above the impinged plate at four jet-to-plate spacing of L/D = 2, 4, 6 and 8. During experiments, power was supplied from DC power source to the stainless steel sheet with a uniform heat generation rate. Voltage taps were located 200 mm from the impinged plate for electric voltage measurement. The experiments were conducted in a temperature controlled room using an air conditioner.

3. Dual Impinging Jets with Co/Counter Swirl

In the experiments, twisted tapes were inserted into the pipe nozzle to induce swirl flow. The twisted tapes were made of aluminum sheets. For the dual tapes, each tape had thickness of 0.8 mm, length of 300 mm and width of 9 mm. In addition, a single twisted tape with the width of 20 mm (thickness and length were same as those of the dual ones) was also fabricated for comparative test. Twisted tapes were prepared with twist ratio of y/W = 3, 4, 5 and 6 where twist ratio is defined as twist length (180°/twist length) to tape width (*W*). The tapes were fabricated by twisting straight tapes, about their longitudinal axis, while being held under tension. In the present report, impinging jets are symbolized and defined as follows (1) single swirling impinging jet (SIJ) is a swirling jet induced by a single twisted tape (2) dual impinging jets with co swirling flows (Co-DSIJs) are the jets induced by dual twisted tapes arranged in the same direction (3) dual impinging jets with counter swirling flows (C-DSIJs) are the jets induced by dual twisted tapes arranged in opposite directions. The twisted tape arrangements are shown in Fig. 2.

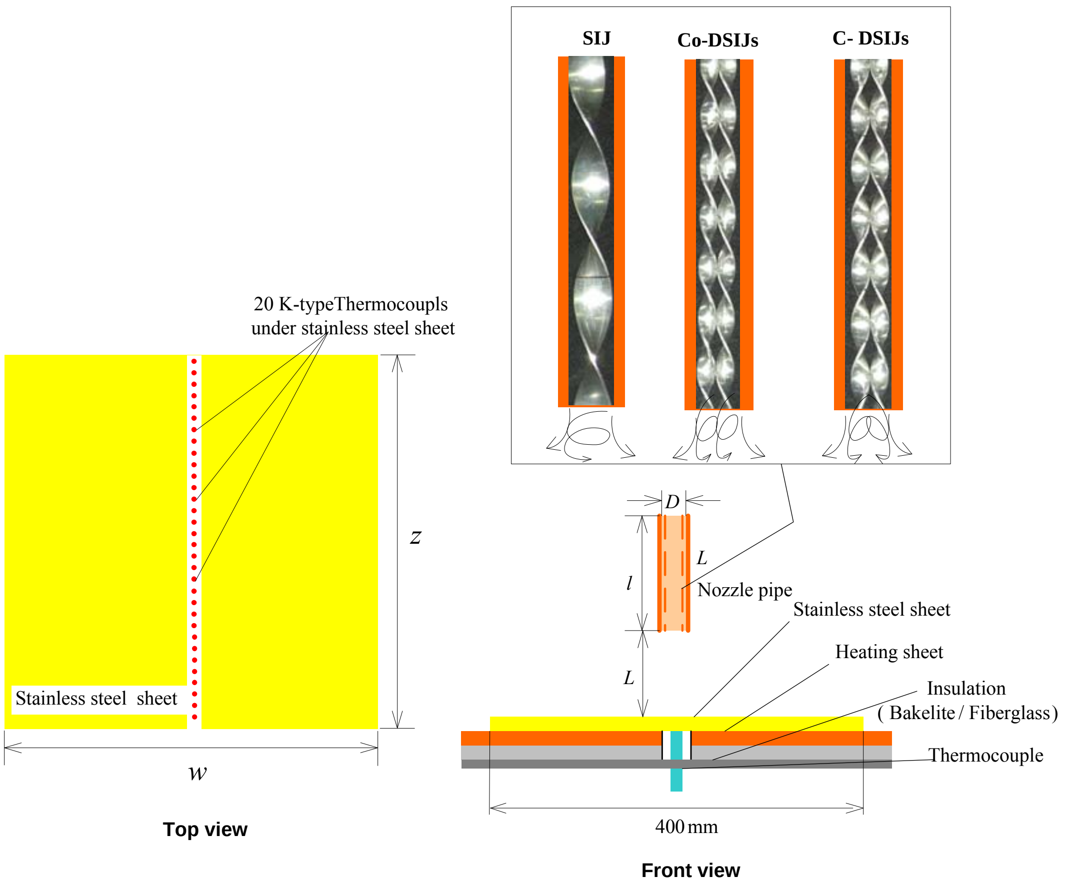


Fig. 2. Twisted tape arrangements, impinging jet and impinged plate components with their dimensions.

4. Data Reduction

The heat transfer rate in term of the local Nusselt number and local convective heat transfer coefficients at the impingement plate are calculated by

$$Nu = \frac{hD}{k} \tag{1}$$

$$h = q/A(T_w - T_j) \tag{2}$$

$$q = VI \tag{3}$$

where q/A is the heat flux obtained by power input from power supply to impinged wall, V is the voltage, I is the current and A is the surface area for smooth impingement surface.
Reynolds number is calculated from

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

where ρ is the density of air corresponding to the jet air temperature, U is the average velocity at the exit of the nozzle and μ is the viscosity of air.

5. Results

The experimental results as a relationship between mean Nusselt number and Reynolds number for all studied cases are shown in Fig. 3. For all cases, Nusselt number consistently increases with increasing Reynolds number due to the increase of axial flow velocity and thus kinetic energy of the impinging jet.

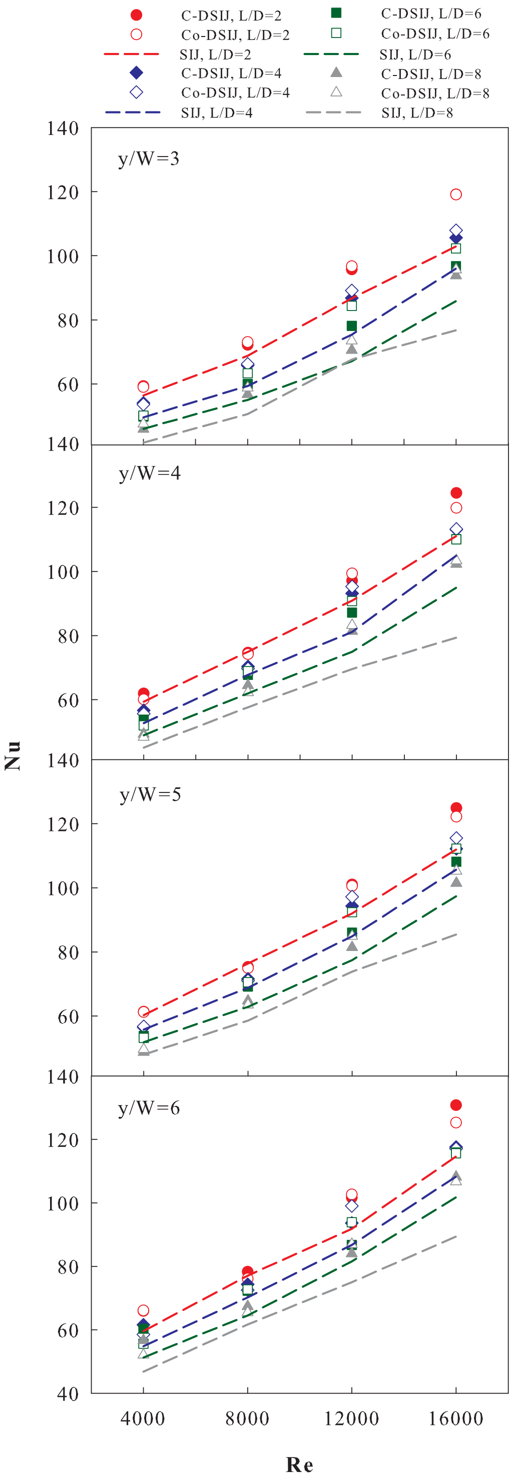
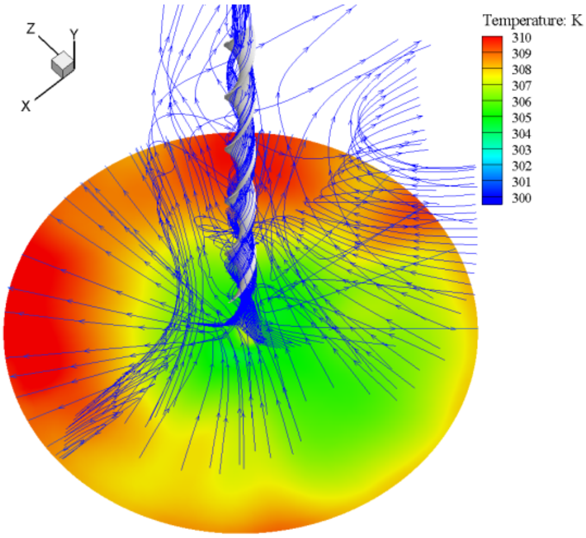
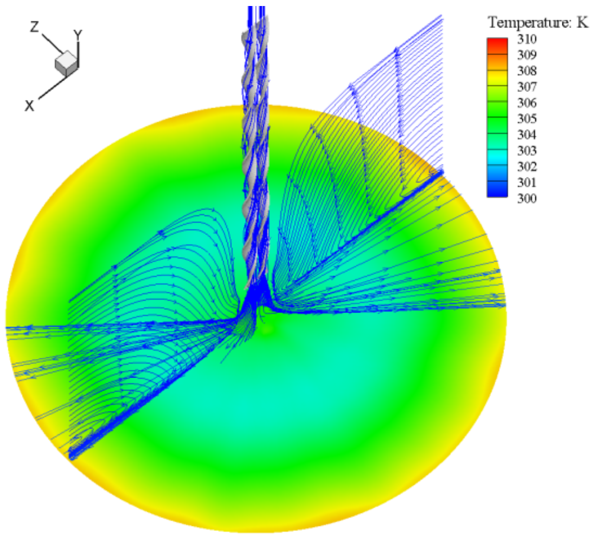


Fig. 3. Relationship between mean Nusselt number and Reynolds number, for the studied cases.

At a given Reynolds number, the swirling jets (SIJ, Co-DSIJs and C-DSIJs) with smaller nozzle-to-plate spacing yield higher heat transfer rates, due to a higher axial velocity. On the other hand, at a larger nozzle-to-plate spacing, although swirling flow enlarges the impinged region, the swirling jet spreading takes place along the axial direction prior to an impingement, this results in significant decrease of impinging velocity and thus poorer heat transfer. For the range examined, the smallest nozzle-to-plate spacing (*L/D* = 2) offer the highest heat transfer rate.

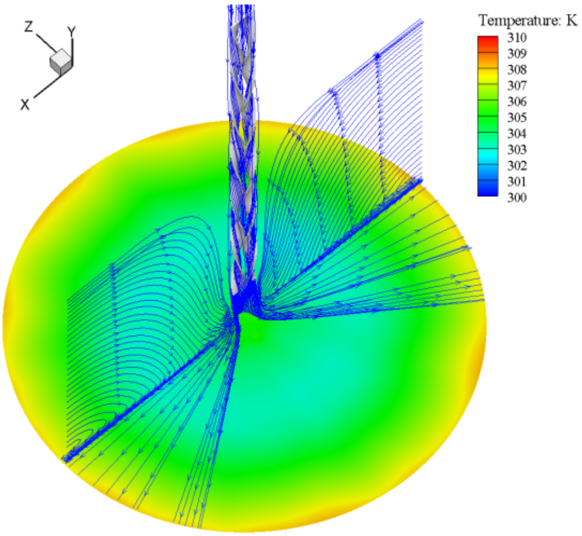


(a) SIJ



(b) Co-DSIJs [6]

Fig. 4. Temperature fields on the impinged surfaces which are impinged by (a) SIJ (b) Co-DSIJs (c) C-DSIJs for L/D = 2 and Re = 16,000.



(c) C-DSIJs

Fig. 4. Temperature fields on the impinged surfaces which are impinged by (a) SIJ, (b) Co-DSIJs and (c) C-DSIJs for L/D = 2 and Re = 16,000. (Cont.)

At similar conditions, Co-DSIJs and C-DSIJs yield higher Nusselt number than SIJ. This can be explained by the numerical visualization results of temperature fields on the impinged surfaces (Fig. 4). Apparently, Co-DSIJs and C-DSIJs offer larger impinged region and radially more uniform temperature fields than SIJ (Fig. 4b-c). This is primarily responsible by the interaction between the dual jets, resulting in higher turbulent intensity and more efficient radial spreading of the flow over the region away from stagnation point.

Effect of the tape twist ratio (*y/W*) on the heat transfer is shown in Fig. 5. It is found that at the use of tapes with smaller twist ratios (*y/W*) results in lower heat transfer rate especially at low Reynolds number (4000<Re<8000). This can be explained by the fact that at smaller twist ratio, larger tangential flow components are introduced into the main air flow resulting in more significant weakening axial flow, thus impingement velocity is lower and heat transfer is poorer.

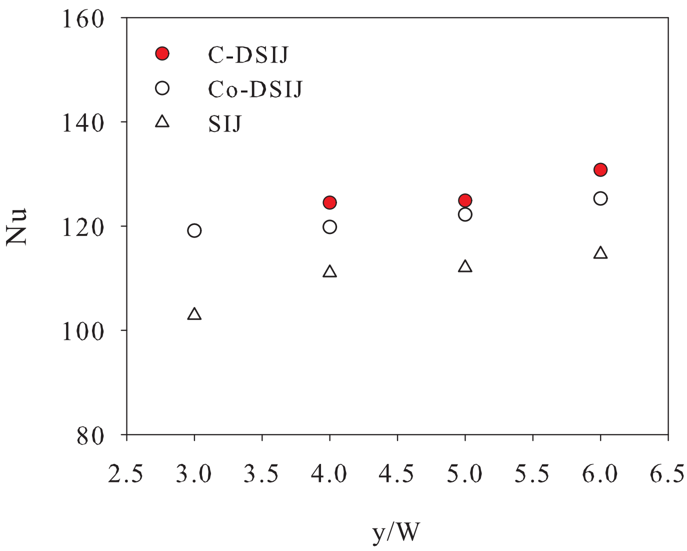


Fig. 5. Effects of twist ratio on average Nusselt number.

6. CONCLUSION

The present paper presents the heat transfer enhancement by means of co and counter-dual swirling jets. The effects of jet-to-plate spacing, tape twist ratio and swirling direction are reported. The obtained results reveal that heat transfer enhancement is improved with decreasing jet-to-plate spacing and increasing tape twist ratio. At similar conditions, C-DSIJs offer higher Nusselt number than the Co-DSIJs. The maximum Nusselt number is achieved by using C-DSIJs at the smallest jet-to-plate spacing (L/D = 2) and the largest twist ratio (y/W = 6.0). For the whole studied range, heat transfer by Co-DSIJs and C-DSIJs is superior to those by SIJ and CIJ.

7. ACKNOWLEDGEMENT

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