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Study on Laminar Heat Transfer Enhancement by Twisted-Inserts

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Key words: Laminar flow, Twisted insert, Heat transfer enhancement, Swirl flow

Abstract

In order to understand the laminar heat transfer enhancement by swirl flow, the effects of heat transfer in a circular pipe with twisted inserts are investigated experimentally. In the present study, a uniform heat flux condition is considered. Laminar heat transfer correlations are developed using least square fit method from surface temperature distributions of an electrically-heated pipe and flow properties. Average Nusselt number correlations with twisted inserts are expressed as a function of swirl parameter, Reynolds number and Prandtl number. When the twisted ratio is 6.05, mean Nusselt number and friction factor increase by approximately 500% and 300%, respectively, compared with the values for a pipe without inserts.

Nomenclature

A : cross-sectional flow area, $\pi d^2/4$ [m^2]
C : heat capacity [J/kgK]
d : inner diameter [m]
D : outer diameter [m]
f : friction factor
H : 180° tape pitch [m]
I : current [A]
h : heat transfer coefficient [W/m^{2}K]
k : thermal conductivity [W/mK]
L : test section length [m]
$\dot{m}$: mass flow rate [kg/s]
Nu : Nusselt number, hd/k
Pr : Prandtl number, $\mu c/k$
q'' : heat flux [W/m^2]
Q : heat transfer rate [W]
Re : Reynolds number, $\rho Vd/\mu$
Re_a : Reynolds number based on axial velocity, $\rho V_a d/\mu$
Re_s : Reynolds number based on swirl velocity, $\rho V_s d/\mu$
Sw : swirl parameter, $\mathrm{Re}/\sqrt{y}$
V : mean axial velocity [m/s]
V : voltage [V]
V_a : mean axial velocity with twisted inserts [m/s]

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V_s : swirl velocity at tube wall [m/s]

x : distance in axial direction [m]

y : twist ratio, H/d

Greeks

α : thermal diffusivity [m^2/s]

δ : tape thickness [m]

μ : viscosity [$N \cdot s/m^2$]

ρ : density [kg/m^3]

Subscripts

b : average fluid temperature

c : cross-section

i : inlet or inner wall

m : mean

o : outlet or outer wall

s : swirl flow

w : wall

1. Introduction

As efficiency of most energy systems can be increased by enhancing heat transfer, heat transfer enhancements have been of many researchers' interests. Basically, enhancement of heat transfer can be achieved by increasing heat transfer coefficient and heat transfer area. An increase in surface roughness of heat transfer tube, an increase in surface area, vibrations of working fluid and tube, application of electric and/or magnetic field can be used on both sides of heat transfer tubes to enhance heat transfer. In order for application of these methods, it needs to change considerably the shape of heat transfer tubes. However, a heat transfer enhancement method such as twisted inserts, which generate swirl flow, can be easily adopted by existing shell and tube heat exchangers. This method is currently being used in various fields such as residual heat recovery systems of high temperature steam flow, domestic water heaters, steam generators, boilers, heating or cooling of viscous liquids in heat exchangers.

A lot of recent researches of heat transfer enhancement using twisted inserts are involved with laminar flow while researches in this field were focused on turbulent flow of water and air in its early stage. Date and Singham,[1] Date,[2] Hong and Bergles[3] performed research of heat transfer enhancements of laminar viscous flow under a uniform heat flux condition. Carrying out numerical analyses, Date and Singham[1] and Date[2] took twist ratio and fin effect into account but not the effect of tape thickness. Hong and Bergles[3] measured properties of water and glycol flowing through a joule-heated pipe and suggested an empirical correlation of Nusselt number which can be applied to a fully-developed flow condition. Researches of laminar flow heat transfer of air,[4] oil,[5] and non-Newtonian fluid[6] have been reported as well.

In general, heat transfer is enhanced by various mechanisms such as fluid mixing due to swirl flow, increases in flow velocity and path, and fin effect of tape. If a twisted insert is installed in a circular pipe, convective heat transfer is augmented but pumping work and pressure drop are increased. In order to optimize these contrary effects, much understandings of flow in a pipe with twisted inserts are necessary. In addition, general design parameters such as heat transfer coefficients and friction factors should be given as well.

In this study, effects of twisted inserts on heat transfer are studied experimentally. A twisted insert in a heat transfer tube develops swirl flow which has tangential velocity component and increases convective heat transfer coefficient. The present study aims to understand the effects of twist ratio, inlet velocity and temperature on heat transfer in a vertical circular tube with twisted inserts (y=6.05 and

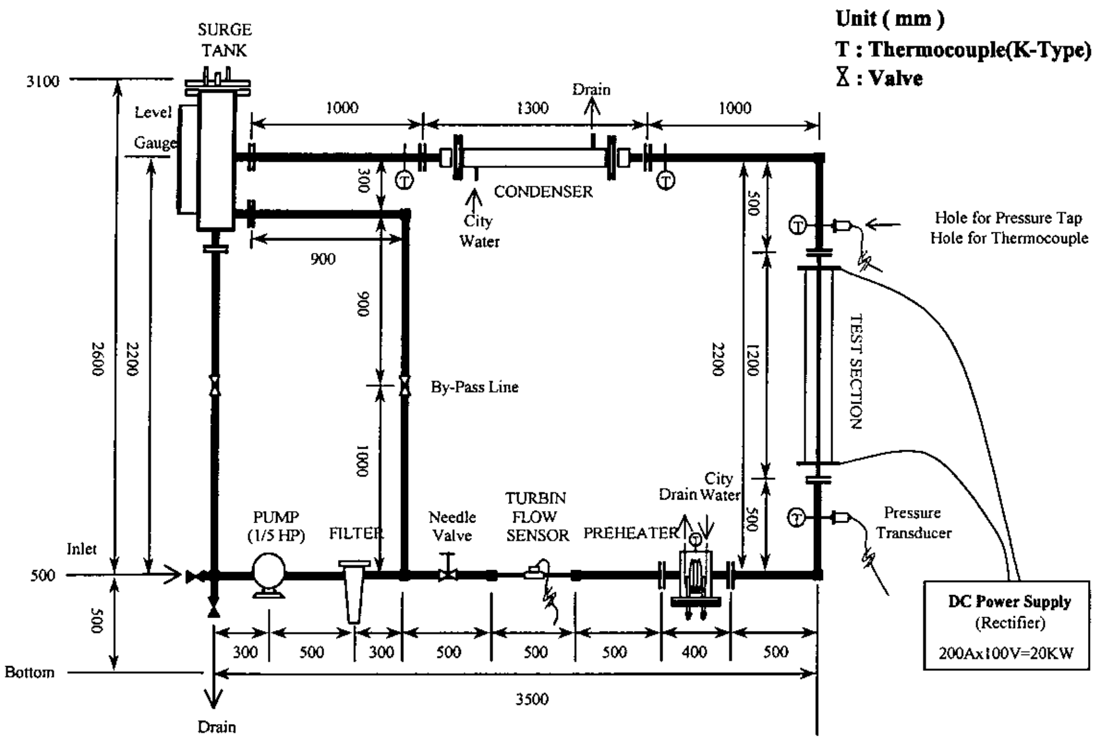

Fig. 1 Schematic diagram of test loop.

$y=\infty$). The test section is uniformly heated. The measurements of surface temperature, inlet and outlet temperatures, and flow parameters are used to develop an empirical correlation of Nusselt number in terms of Re, Sw, and Pr number.

2. Experiments

2.1 Experimental apparatus

Figure 1 shows a schematic diagram of the present experimental apparatus designed for testing the effect of twisted inserts on heat transfer in a circular pipe. The details of twisted inserts are given in Fig. 2. The experimental loop consists of a uniformly-heated test section, a preheater for fluid inlet temperature control, a flow control valve, a surge tank for fluid volume control, and a condenser. A turbine flowmeter, K-type thermocouples, pressure transducers, filter, and a centrifugal pump are installed in the loop. Water is used as a working fluid. All components of the loop are made of SUS304 stainless steel. Geometric dimensions of the test section and thermal properties are given in Table 1. The test section is a circular pipe of which inner diameter, thickness, and length are 10.9 mm, 1.8 mm, and 95.5 mm, respectively. A twisted insert is made of copper and whose thickness and width are

Table 1 Specification of test section

Material	SUS304
Electrical Resistivity, ρ	$72\times10^{-8}\ \Omega\cdot m$
Thermal Conductivity, k	$15\ W/m\cdot K$
Length, L, m	0.955000
Outside Diameter, D, m	0.012700
Inside Diameter, d, m	0.010922
Position of Thermocouples, x, m	0.128, 0.225, 0.326, 0.424 0.536, 0.636, 0.732, 0.835

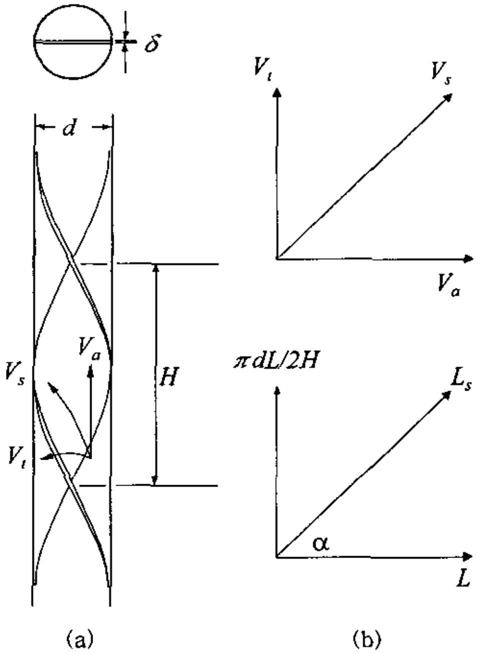

Fig. 2 A twist insert: (a) Geometrical characteristics, (b) Resolution of fluid swirl velocity and flow length into its axial tangential components.

0.3 mm and 10 mm, respectively. Twist ratio (y) is defined as $y = H/d$. H and d denote 180° tape pitch and pipe diameter, respectively. The condition of $y = \infty$ means the case that a circular pipe is divided into two semi-circular ducts.

The test section is joule-heated to apply a uniform heat flux onto the heat transfer area. As the electric resistance of the test section is so low that a low voltage/high current dc power supply is used.

2.2 Experimental procedures and data processing

Three types of inserts are tested: none, straight-type, twisted-type. Surface heat flux is adjusted at a pre-determined value by changing current and voltage of DC power supply in the beginning. When the surface heat flux reaches a steady state, flow rate, temperature, and pressure of working fluid, current and voltage of DC power supply are measured at various Prandtl and Reynolds numbers. After that, the surface heat flux is increased to an another value and the measurements are made again. The experiments are performed in the range of Reynolds number from 100 to 1500, inlet temperatures at 25, 30, 40, 50, 60, and 70℃. Reliability and repeatability are confirmed by repeating measurements several times at the same conditions.

The heat transfer rate and heat flux of the test section are calculated by equations (1) and (2).

$$Q = I \cdot V \tag{1}$$

$$q'' = \frac{Q}{\pi DL} \tag{2}$$

The uncertainty of the heat flux of the test section is evaluated by the methods of Moffat,[8] Wang & Simon[9] and found to be within ±5%.

Temperatures at inlet, outlet and 8 elevations of the test section are measured by K-type thermocouples. In addition, a variation of temperature in circumferential direction is measured at two elevations (0.326 m, 0.636 m). Six thermocouples are installed at each two elevation and they are equivalently spaced in circumferential direction. The uncertainty of temperature measurements are within ±0.2℃. The test section is thermally insulated to minimize heat loss.

Table 2 shows test parameters of the present experiments. Manglik and Bergles[10] reported that a swirl flow in the range of Sw< 1400 is to be considered as laminar flow. Judging based on this claim, it is believed that the fluid flows are in laminar flow regime through the present experiments in which twist ratio is 6.05 since the swirl parameter corresponding to the range of Reynolds number (100-1500) in the table 2 varies from 40.6 to

609.8. Terminal blocks (SCXI-1303, SCXI-1320) and modules (SCXI-1102, SCXI-1121) of National Instruments Co. are used for data acquisition and signals are processed by a PC-embedded I/O board (Lab-PC+). All data acquisition processes are controlled by a LabVIEW program. The measurement errors of the K-type thermocouples welded on test section surface are estimated to be within ±0.2℃ and overall error of the measurement system is evaluated to be within ±0.5℃ when tested in operation range. Maximum error in temperature measurement by ANSI/ASME PTC 19.1[11] is estimated to be ±0.8℃. Fluid flow rate is measured by a turbine flow meter (Magnetoflow Co., VISION 2008) whose accuracy is ±0.8%. System pressure is measured by a differential pressure cell (Data Instrument, DG) whose accuracy is ±1.0%.

Heat transfer coefficients and Nusselt numbers are calculated based on inner wall temperature, mean fluid temperature, and heat flux. Temperature variation in circumferential direction appeared to be negligible due to fluid mixing and high thermal conductivity of tube wall. The mean temperature remained at 2.1℃ and assumed to be constant. With this assumption, axial variation and mean heat transfer coefficient are measured. Nusselt number is expressed as a form of $Nu_m = f(Re, Pr, y)$. If Reynolds number and twist ratio are combined to produce Sw, Nusselt number can be expressed as $Nu_s = C\,Sw^n Pr^m$. Experimental measurements are regressed to get the constants, C, n, m, and finally empirical correlation for Nusselt number.

3. Results and discussions

Figure 3 shows a Nusselt number with respect to the reduced axial distance, x/dRe_sPr, in case whose twist ratio is y=6.05. A solid line represents Hong and Bergles's[12] analytical model developed for forced convection in a semi-circular pipe with a twist insert whose thickness is negligible. The model predicts smaller value than the present measurements. If the thickness of tape is considered, however, the model's prediction will become larger since the flow path surrounded by tape insert and tube wall gets narrower than a semi-circle. Figure 3 indicates that Nusselt number decreases with increase in reduced axial distance for a fixed twist ratio, y. Also, it shows that local Nusselt number at around tube inlet is larger than that at fully developed region. This is a similar trend with those previously reported. Figure 4 shows a relation between Prandtl number and mean Nusselt number

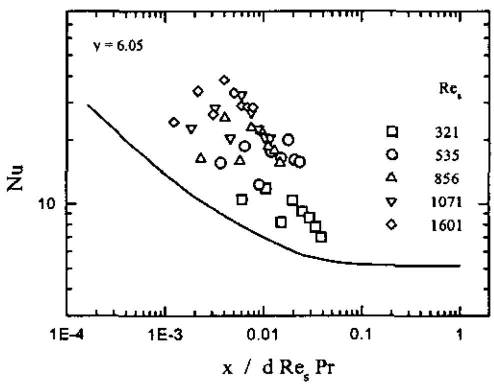

Fig. 3 Nusselt number vs. reduced axial distance (y=6.05).

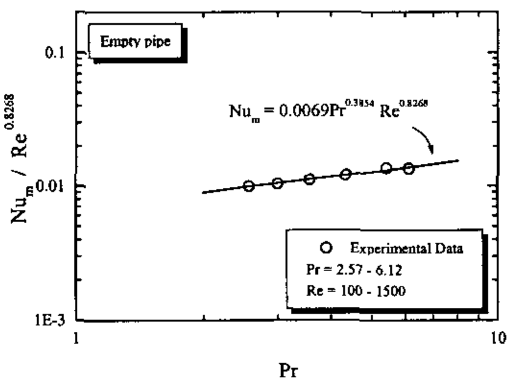

Fig. 4 Heat transfer correlation for empty pipe.

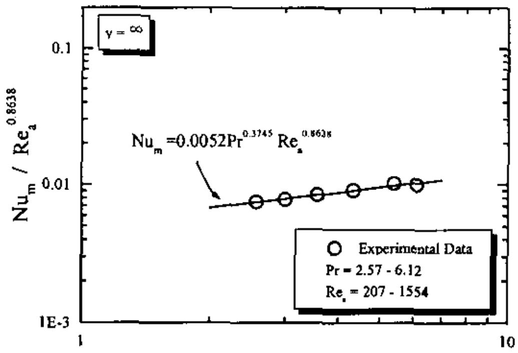

Fig. 5 Heat transfer correlation for straight tape insert ($y=\infty$).

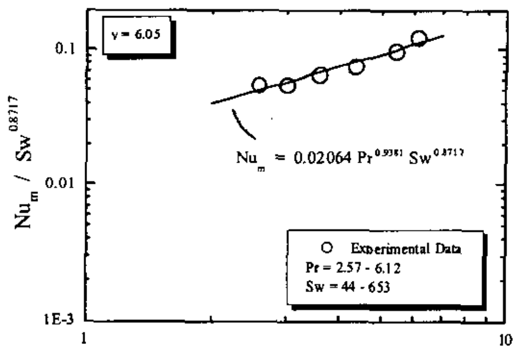

Fig. 6 Heat transfer correlation for twisted insert (y=6.05).

normalized by Reynolds number for empty pipe. The mean Nusselt number increases proportionally to Prandtl number. A linear regression of these values produces the following correlation and whose standard deviation is 7.8%.

$$\mathrm{Nu}_m = 0.0069\,\mathrm{Re}^{0.8268}\,\mathrm{Pr}^{0.3854} \tag{3}$$

A similar plot for the case of $y=\infty$ is shown in Fig. 5. In this figure, Reynolds number is calculated based on mean axial velocity (V_a) which is a realistic velocity considering the reduction of flow path due to tape insert. A regression result suggests the following correlation of Nusselt number and whose standard deviation is 1.1%.

$$\mathrm{Nu}_m = 0.0052\,\mathrm{Re}_a^{0.8638}\,\mathrm{Pr}^{0.3745} \tag{4}$$

Figure 6 shows a relation between Prandtl number and mean Nusselt number normalized by swirl parameter for a pipe with a twist-tape insert whose twist ratio is 6.05. In a swirl flow developed by a twisted insert, centrifugal force due to rotating flow and inertia force in axial direction are balanced with viscous force. Therefore, the swirl parameter, Sw, which represents the intensity of secondary flow, can be expressed as follows.

$$\mathrm{Sw} = \left(\frac{\mathrm{Re}_s}{\sqrt{y}}\right) = \left(\frac{\mathrm{Re}}{\sqrt{y}}\right)\left(\frac{\pi}{\pi - 4\frac{\delta}{d}}\right)\left\{1+\left(\frac{\pi}{2y}\right)^2\right\}^{\frac{1}{2}} \tag{5}$$

This parameter explains the effect of tape thickness, twist ratio, and increased spiral flow velocity. In cases where twist-tape insert is installed, Nusselt number shows an increasing trend with increases in Reynolds number and Prandtl number and a decrease in twist ratio. These observations mean that Re_s and y can be combined to be a single parameter, $\mathrm{Sw} = \mathrm{Re}_s/\sqrt{y}$. A correlation of Nusselt number based on this parameter can be expressed as follows and standard deviation is 2.0%.

$$\mathrm{Nu}_m = 0.02064\,\mathrm{Sw}^{0.8717}\,\mathrm{Pr}^{0.9381} \tag{6}$$

As can be seen from Figs. 4 to 6, Nusselt number appears to increase as the pipe condition changes from empty pipe to $y=\infty$ and finally to y=6.05. When compared under the same inlet conditions, Nusselt number for the case of y=6.05 appears to be five times larger than that for empty pipe case.

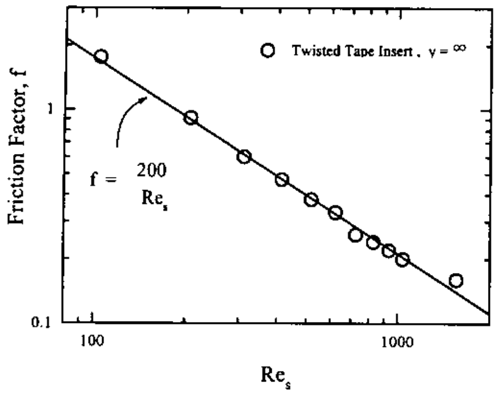

Fig. 7 Friction factor correlation for $y=\infty$.

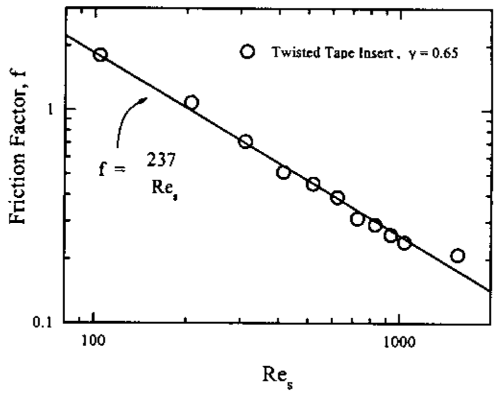

Fig. 8 Friction factor correlation for y=6.05.

Figures 7 and 8 show variations of friction factor with respect to Re_s for the case of $y=\infty$ and y=6.05, respectively. Friction parameter appears to be inversely proportional to Re_s. The slope for the case of y=6.05 is evaluated to be 18% larger than that for the case of $y=\infty$. This result claims that friction factor is affected by the twist ratio. Friction factor measured in a test section with a twisted insert has been found to be larger than that in a fully developed circular pipe, which is correlated as f=64/Re, by more than three times.

4. Concluding remarks

Experiments have been carried out to investigate the enhancement of laminar heat transfer by swirl flow. Swirl flow was generated by twisted inserts and three cases, empty pipe, y=0.65 and $y=\infty$, were tested under uniform heat flux condition. Based on the present measurements empirical correlations for Nusselt number and friction have been developed.

It has been found, under the same inlet conditions, that mean Nusselt number for twist ratio of 0.65 increases by more than five times compared with that for empty pipe. This observation means that decreasing in twist ratio improves heat transfer significantly. However, it was also found that friction factor for the case of y=6.05 increases by three times compared with the empty pipe case.

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References

1. Date, A. W. and Singham, J. R., 1972, Numerical prediction of friction and heat transfer characteristics of fully developed laminar flow in tubes containing twisted tapes, ASME Paper No. 72-HT-17.
2. Date, A. W., 1974, Prediction of fully-developed flow in a tube containing a twisted-tape, International Journal of Heat and Mass Transfer, Vol. 17, pp. 845-859.
3. Hong, S. W. and Bergles, A. E., 1976, Augmentation of laminar flow heat transfer in tubes by means of twisted-tape inserts, Transactions of the ASME Journal of Heat Transfer, Vol. 98, pp. 251-256.
4. Watanabe, K., et al., 1983, Heat transfer augmentation in tubular flow by twisted tapes at high temperature and optimum performance, Heat Transfer, Japanese Re-

search, Vol. 12, pp. 1-31.

5. Bandyopadhyay, P. S., et al., 1991, Influence of free convection on heat transfer during laminar flow in tubes with twisted tapes, Exper. Thermal and Fluid Science, Vol. 4, pp. 577-586.
6. Manglik, R. M., et al., 1988, Augmentation of heat transfer to laminar flow of non-newtonian fluids in uniformly heated tubes with twisted-tape inserts, Exper. Heat Transfer, Fluid Mechanics and Thermodynamics, Elsevier Science Pub., New York.
7. Marner, W. J. and Bergles, A. E., 1989, Augmentation high temperature heat transfer inside tubes with constant wall temperature, Exper. Thermal and Fluid Science, Vol. 2, pp. 252-267.
8. Moffat, R. J., 1985, Using uncertainty analysis in the planning of an experiment, J. Fluid Eng., Vol. 107, pp. 173-1822.
9. Wang, T. and Simon, T. W., 1988, Development of a special-purpose test surface guided by uncertainty analysis: Introduction of a new uncertainty analysis step. AIAA 26th Aerospace Sciences Meeting, pp. 1-9.
10. Manglik, R. M. and Bergles, A. E., 1993, Heat transfer and pressure drop correlations for twisted-tape inserts in isothermal tubes: part I-laminar flows, Transactions of the ASME Journal of Heat Transfer, Vol. 115, pp. 881-889.
11. ANSI/ASME PTC 19.1, 1985, Measurement Uncertainty.
12. Hong, S. W. and Bergles, A. E., 1974, Augmentation of laminar flow heat transfer in tubes by means of twisted-tape inserts, ISU Heat Transfer Laboratory Report HTL-5.