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Enhancement of Heat Transfer in a Circular Wavy-surfaced Tube with a Helical-tape Insert

S. Eiamsa-ard[*] and P. Promvonge[+1]

Abstract *- The purpose of the present study is to investigate experimentally the heat transfer and turbulent flow friction characteristics in a circular wavy-surfaced and constant heat-flux tube with a helical-tape insert. In the experiment, heat transfer augmentation is expected from both the turbulence of flow near the tube wall produced by the wavy surface and the swirling flow generated by the helical-tape. The experiments are based on Reynolds number at the tube inlet, ranging from 3000 to 9200. The experimental results obtained are compared with those from plain tubes in the literature. The results show that the heat transfer rate from using the wavy surface and helical tape insert is considerably higher than that from the plain tube. The Nusselt numbers and friction factors are found to be, respectively, 3.0 and 50 times over the plain tube for the tube with wavy- surfaced wall alone and to be 4.2 and about 110 times for the tube with combined wavy-surfaced wall and the helical-tape. The wavy-surfaced tube combined with the helical-tape provides higher heat transfer rate and friction factor than the wavy-surfaced tube alone around 57% and 125%, respectively. In addition, effect of insertion of the helical-tape with/without core-rod is also investigated.*

Keywords **- Heat transfer, Helical-tape, Turbulence flow, Wavy-surface wall, Swirl flow, Turbulence flow, Friction factor.**

1. INTRODUCTION

The process of improving convective heat transfer in tubes of a heat exchanger has been extensively investigated. Active and passive methods have been used to improve heat transfer in chemical reactors, heat exchanger and in flow systems for several decades. The principle of the passive technique involves in either surface treatment, such as coated surface, rough surface and extended surface or flow manipulation such as swirl flow and putting additive into the flow. One of the most favorable passive techniques is twisted-tape inserts because they are inexpensive and can be easily employed to the existing system. In the past researches, Royds [1] was the first to prove the useful effects of turbulence flow generators on heat transfer in 1921 with many experiments and types of turbulators. Kreith and Margolis [2], Kreith and Sonju [3] proposed that heat transfer can be enhanced by introducing swirl flow in the heat exchanger with tangential injection of the fluid at various locations along the tube axis. Marner and Bergles [4]-[7] reported experimental data for laminar flows of ethylene glycol and polybutene with a twisted-tape in an isothermal tube. Their experimental data over a wide Prandtl number range are of immense value in the twisted-tape-generated swirl flow heat transfer literature. They found that in very viscous liquid flows, swirl flows do not set in and the heat transfer enhancement is simply due to the duct partitioning.

In general, the twisted tape insert can help to generate the swirling flow and stronger turbulence in tubes. This causes a thinner boundary layer and longer resident time of the flow leading to an increase in the heat transfer coefficient. However, an increase in pressure drop is the penalty of the twisted tape technique. The straight tape twisted in geometry form of helical tape with similar geometry of the screw tape, called the helical tape was introduced in a research study [8]. The helical tape and the twisted-tape can create the swirling flow in the circular tube and both of them posses the different characteristics of flow. For the helical tape, the swirling flow goes in single way direction (a screw motion), while the twisted-tape shows the swirling flow in two ways direction simultaneously.

The reverse flow, sometimes called "re-circulation flow", devices or the turbulators are widely employed in thermal engineering applications. The effect of reverse flow and boundary layer disruption (dissipation) is to enhance the heat transfer coefficient and momentum transfer. The reverse flow with high turbulence can improve convection of the tube wall by increasing the effective axial Reynolds number, decreasing the flow cross-section area, and increasing the mean velocity and temperature gradient. It can help to produce the higher heat fluxes and momentum transfer due to the large effective driving potential force but also higher pressure drop. Yakut et al. [9] experimentally studied the influence of conical-ring turbulators on the turbulent heat transfer, pressure drop and flow-induced vibrations. Yakut and Sahin [10] again reported the flow-induced vibration characteristics of conical-ring turbulators used for heat transfer enhancement in heat exchangers. They pointed out that the Nusselt number increases with increasing Reynolds number and the maximum heat transfer

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is obtained for the smallest pitch arrangement. Durmus [11] also investigated the effect of cutting out conical turbulators inserted in a tube of the heat exchanger, on the heat transfer rates with four different types of turbulators and different conical-angles. Ayhan et al. [12] numerically and experimentally studied the heat transfer augmentation in a tube by means of truncated hollow cone inserts. Eiamsa-ard and Promvonge [13] studied effect of the V-nozzle turbulators on heat transfer and friction characteristics in a circular tube. They found that the nozzle-turbulators has a significant effect on the enhancement of heat transfer. Promvonge and Eiamsa-ard [14] again investigated the effect of conical-nozzle inserts and snail entrance on heat transfer and friction characteristics in a uniform heat flux tube and correlations of the Nusselt number and friction factor based on measured data was also reported.

In the present work, apart from improvement by swirl flows stemming from the centrifugal forces occurring from a helical tape insert in the tube, the goal is to increase the heat transfer further with the passive method by incorporating a wavy-walled surface in the tube. In this article the influences of (1) wavy-surfaced tube and (2) combination of the wavy-surface and the helical-tape, on heat transfer enhancements are experimentally investigated. All of the experiments are carried out at the same inlet conditions with the Reynolds number based on the inner tube diameter in a range of 3000 to 9200.

2. EXPERIMENTAL SET-UP

Materials

The experiments were carried out in an open-loop experimental facility as shown in Fig. 1. The loop consisted of a 7.5 kW blower, orifice meter to measure the flow rate, and the heat transfer test section. The copper test tube has a length of L = 1250 mm, with 47.5 mm inner diameter (D), 50.5 mm outer diameter (D_o), and 1.5 mm copper tube thickness (t) as depicted in Fig. 2. The tube was heated by continually winding flexible electrical wire provided a uniform heat flux boundary condition. The electrical output power was controlled by a variac transformer to obtain a constant heat flux along the entire length of the test section and by keeping the current less than 3 amps. The outer surface of the test tube was well insulated to minimize convective heat loss to surroundings, and necessary precautions were taken to prevent leakages from the system. The inner and outer temperatures of the bulk air were measured at certain points with a multi-channel temperature measurement unit in conjunction with the Chromel-constantan thermocouples as can be seen in Fig. 2. Fifteen thermocouples were tapped on the local wall of the tube and the thermocouples were placed round the tube to measure the circumferential temperature variation, which was found to be negligible. The mean local wall temperature was determined by means of calculations based on the reading of Chromel-constantan thermocouples. Fig. 2(a) represented the circular tube with wavy-surfaced wall arrangement used in the present experiment. The wavy-surfaced wall was made of Aluminum with l = 95 mm (2.0D) in length and its concave and convex diameters were 46 mm and 28 mm, respectively. Fig. 2 (b) represented the wavy-surfaced tube combined with a helical tape insert used in this test. In the experiments, the geometric conditions of the helical tape inserted were kept constant. The helical tape was made of stainless steel and has the geometric dimensions of W = 17 mm (0.95D), d = 5 mm (0.26D), P = 18 mm (0.95D), t = 1 mm (0.05D), respectively. In the test run, helical tapes were inserted in the tube core as shown in Fig. 2 (b).

Method

In the apparatus setting above, the inlet bulk air at 25°C from a 7.5 kW blower was directed through the orifice meter and passed to the heat transfer test section. The air flow rate was measured by an orifice meter, built according to ASME standard. Manometric fluid was used in U-tube manometers with specific gravity (SG) of 0.826 to ensure reasonably accurate measurement of the low pressure drop encountered at low Reynolds numbers. Also, the pressure drops of the heat transfer test tube with helical tapes were measured with inclined U-tube manometers. The volumetric air flow rates from the blower were adjusted by varying motor speed through the inverter, situated before the inlet of test tube. During the experiments, the bulk air was heated by an adjustable electrical heater wrapping along the test section. Both the inlet and outlet temperatures of the bulk air from the tube were measured by multi-channel Chromel-constantan thermocouples, calibrated within ±0.2°C deviation by thermostat before being used. It was necessary to measure the temperature at 15 stations altogether on the outer surface of the heat transfer test pipe for finding out the average Nusselt number. For each test run, it was necessary to record the data of temperature, volumetric flow rate and pressure drop of the bulk air at steady state conditions in which the inlet air temperature was maintained at 25°C.

The various characteristics of the flow, the Nusselts number, and the Reynolds numbers were based on the average of tube wall temperature and outlet air temperature. The local wall temperature, inlet and outlet air temperature, the pressure drop across the test section and air flow velocity were measured for heat transfer of the heated circular wavy tube with a helical tape. The average Nusselt numbers were calculated and discussed where all fluid properties were determined at the overall bulk mean temperature.

The uncertainties of the reduced data obtained experimentally are determined. The uncertainty in the data calculation is based on Ref. [15]. The maximum uncertainties of non-dimensional parameters are ±5% for Reynolds number, ±10% for Nusselt number and ±15% friction. The uncertainty in the velocity measurement is estimated to be less than ±7%, and pressure has a corresponding estimated uncertainty of ±5%, whereas the uncertainty in temperature measurement at the tube wall is about ±0.1%. The experimental results are reproducible within these uncertainty ranges.

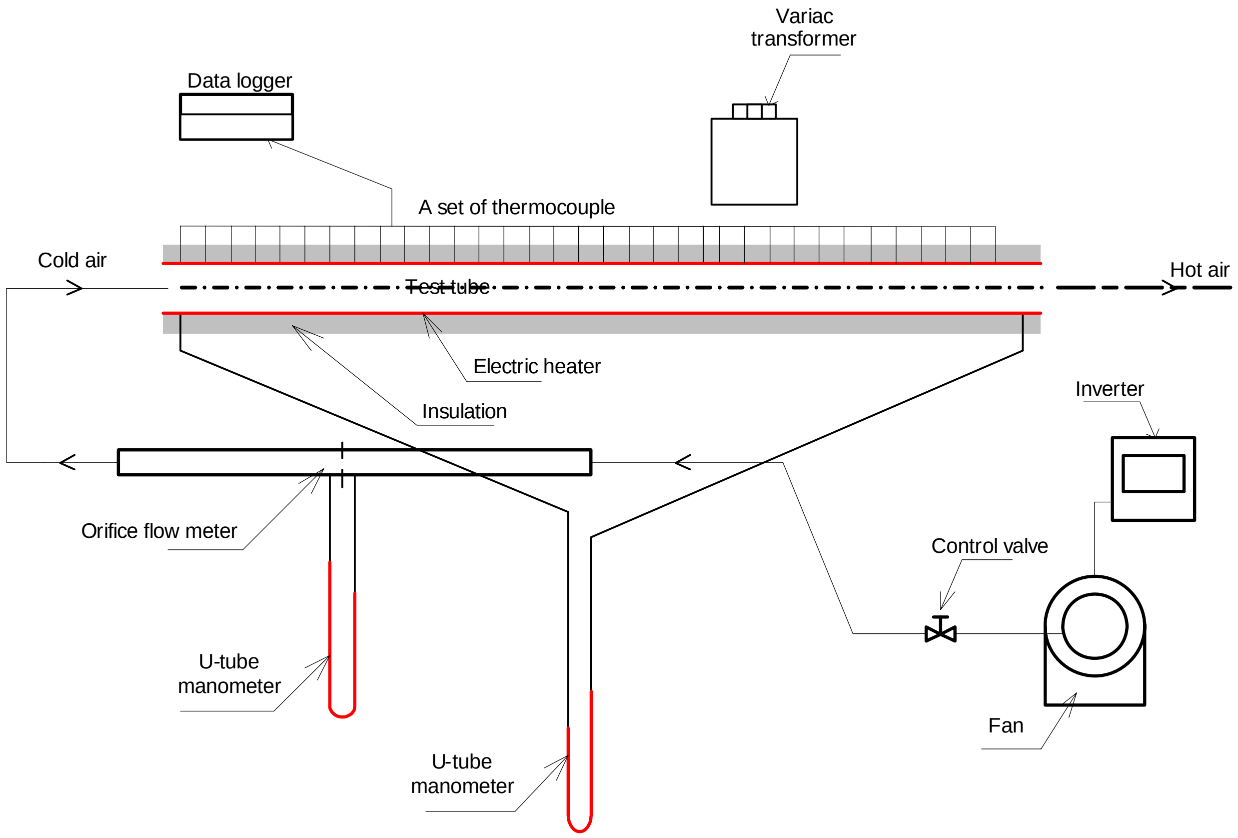
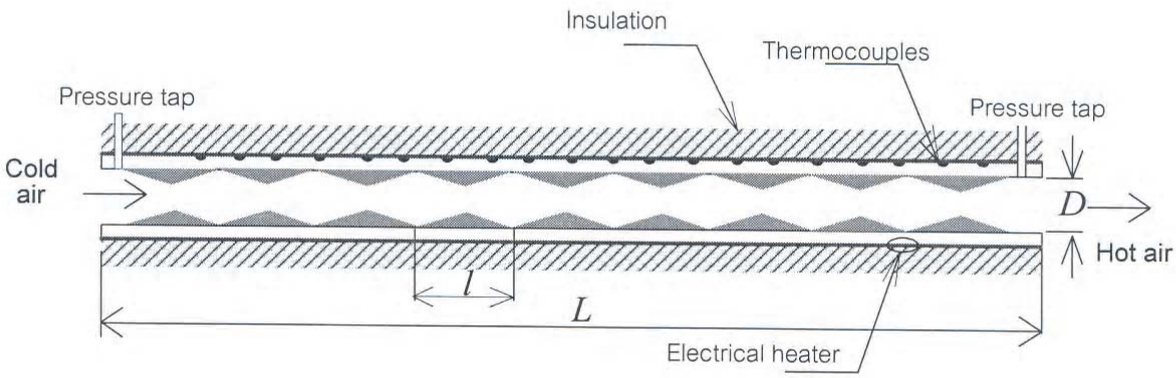
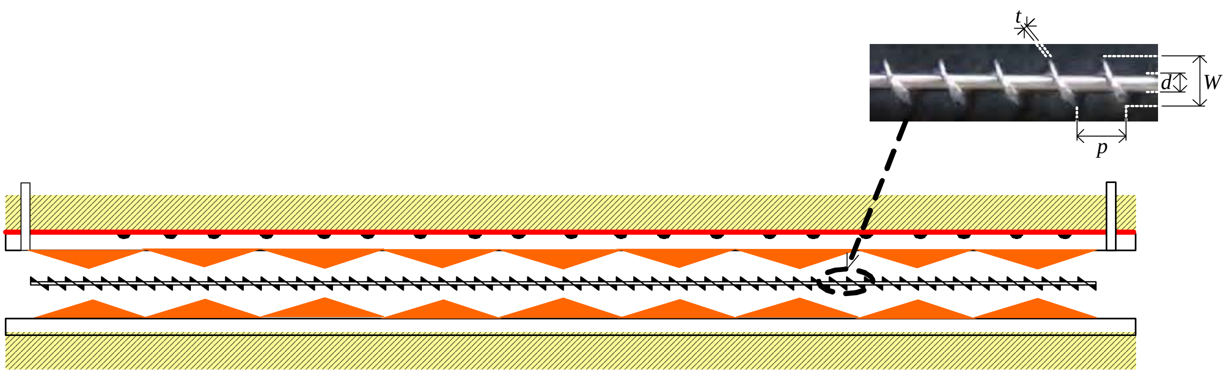


Fig. 1. Experimental apparatus.



(a) The test tube with wavy-surfaced wall



(b) The test tube with combined wavy-surfaced wall and helical-tape

Fig. 2. Test tube.

3. DATA REDUCTION

In the present work, the air is used as working fluid and flowed through a uniform heat flux and insulation tube. The steady state of the heat transfer rate is assumed to be equal to the heat loss from the test section which can be expressed as:

$$Q_{air} = Q_{conv} \quad (1)$$

in which

$$Q_{air} = \dot{m}C_{pa}(T_o - T_i) = VI \quad (2)$$

The local convection heat transfer coefficients,

$$h_x = q''/(T_{wx} - T_{bx}) \quad (3)$$

are obtained at each thermocouple location on the tube surfaces, T_{wx}, where, $q'' = Q_{conv}/A$ and since the heat is added uniformly along the tube length, the bulk temperature of the fluid at the measuring section, T_{bx}, is calculated by considering a linear variation with the axial direction.

The average heat transfer coefficient, h, may be obtained by means of

$$h = \frac{1}{L}\int_0^L h_x dx \quad (4)$$

The local Nusselt number, Nu_x and the average Nusselt number, Nu, are estimated as follows:

$$Nu = h_x D/k \quad (5)$$

$$Nu = hD/k \quad (6)$$

The Reynolds number is given by

$$Re = UD/v$$

Friction factor, f can be written a

$$f = \frac{\Delta P}{(L/D)\rho U^2/2} \quad (8)$$

in which U is mean velocity of the tube. All of thermo-physical properties of the air are determined at the overall bulk air temperature.

4. RESULTS AND DISSCUSION

Verification of Heat Transfer and Friction Loss

With the help of data obtained from the experiments and with the Nusselt number and friction factor were found and shown graphically. Beside this, the Nusselt number and friction factor were compared with those found using, correlations of Dittus-Boelter [16] and Petukhov [16] for turbulent flow in a cylindrical tube as seen in Figs. 3 and 4.

The Dittus-Boelter's correlation of the Nusselt number for the plain tube can be expressed as follows:

$$Nu = 0.023\ Re^{4/5}\ Pr^{0.4} \quad \text{for} \quad Re > 1\times 10^4 \quad (9)$$

Friction factor correlations:

Correlation of Petukhov,

$$f = (0.790 \ln Re - 1.64)^{-2} \quad \text{for } 3000 \le Re \le 5\times 10^6 \quad (10)$$

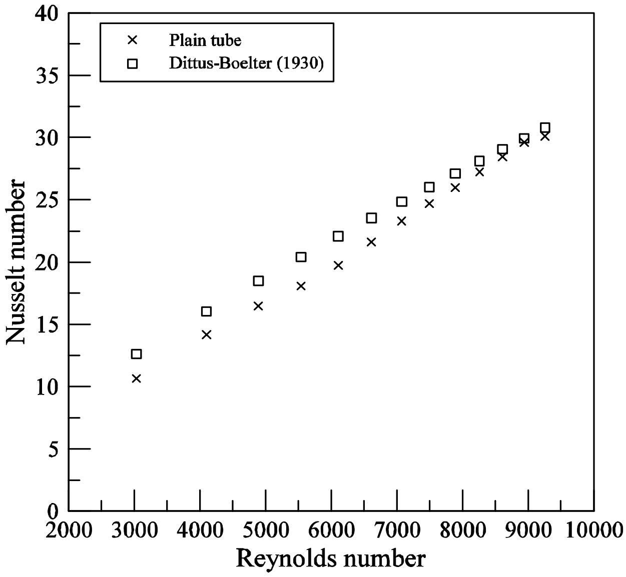


Fig. 3. Verification of Nusselt number for plain tube.

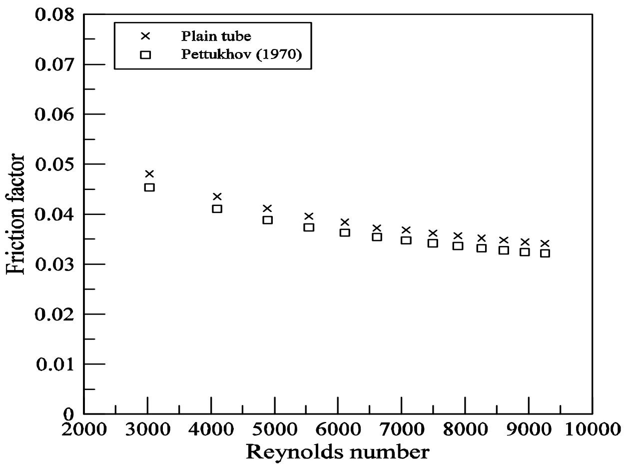


Fig. 4. Verification of friction factor for plain tube.

Figures 3 and 4 show a comparison between the present experimental work and correlations from the previous work. In the figures, results of the present work reasonably agree well with the available correlations within ±10% in comparison with friction factor correlation of Petukhov and within ±10% in comparison with Nusselt number correlation of Dittus-Boelter.

Effect of Wavy-surfaced Wall

The effect of using the wavy-surfaced tube wall on heat transfer characteristics is shown in Fig. 5. It is found that the tube fitted with the wavy-surfaced wall gives higher heat transfer rate than the plain tube. The maximum Nusselt number increases at about 300% when compared with those from the plain tube. This can be attributed to better mixing of fluids between the wall and the core regions from using wavy-surfaced wall causing the turbulence/reverse flow and pressure gradient in the radial direction. The boundary layer along the tube wall would be redeveloped along the wavy wall resulting in more heat flow through the fluid. Furthermore, the recirculation/reverse flow enhances the turbulence fluctuations, which leads to even better convection heat transfer. Thus, the higher the Reynolds number is, the greater the Nusselt number. The relationship between the friction factor and Reynolds number for using the wavy-surfaced tube wall is presented in Fig. 6. In the figure, it is worth noting that friction factor of the wavy tube is considerably higher than that of the plain tube. This is because of higher surface area and the dissipation of dynamic pressure of the fluid at high viscosity loss near the tube wall. The friction factor tends to reduce gradually for increasing Reynolds number.

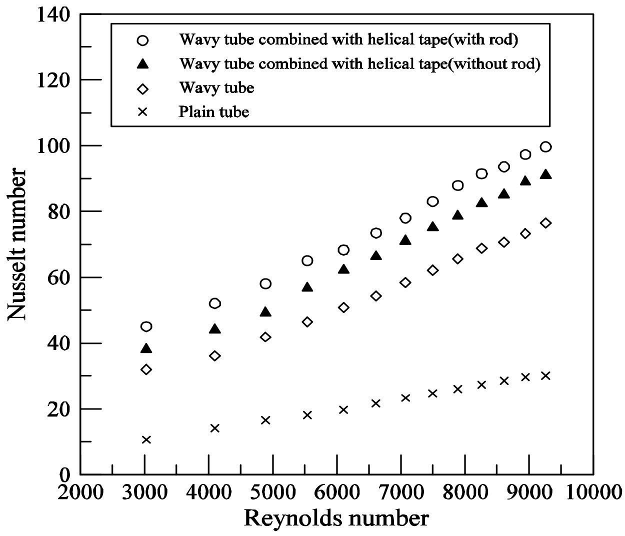


Fig. 5. Variation of Nusselt number with Reynolds number.

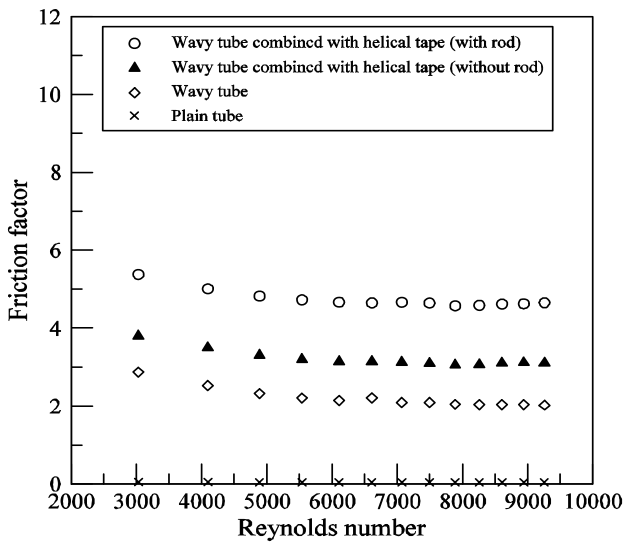


Fig. 6. Variation of friction factor with Reynolds number.

Effect of Combined Wavy-surfaced Wall and Helical Tape Insert

The results of the measurements with using the enhancement devices are also shown in Fig. 5. In the figure, it can be seen that use of the wavy-surfaced tube with the helical tape yields a higher heat transfer rate than that of the wavy-surfaced tube alone. The increase in Nusselt number can be explained by two mechanisms: firstly, the surface of the wavy tube may be act as extended heat transfer surfaces and secondly, it is due to strong turbulence/swirl flow created by the helical tape. In general, the average heat transfer rate for the wavy-surfaced tube with the helical tape is found to be 23 to 35% better than that for the wavy-surfaced tube alone. The corresponding increase in the mean Nusselt number of the wavy-surfaced tube with the helical tape is about 330% to 422% over the plain tube. However, it is apparent that the use of the wavy-surfaced tube along with the helical tape also provides higher friction factor than that of the wavy-surfaced tube alone due to larger contact surface areas and flow disturbance at the core. Besides, the presence of the wavy-surfaced wall together with helical tape reduces flow areas, resulting in a high speed rotating flow. This leads to the substantial pressure loss action of the fluid (hot air) higher than one without the helical tape. For the wavy-surfaced tube with the helical tape, the increase in friction factor is found to be around 50% above one without the tape.

Effect of the Helical Tape with/without Core-rod

Influence of using the helical tape with/without core rod in a wavy-surfaced tube on heat transfer characteristics is again depicted in Fig. 5. In the figure, the tape with core-rod yields higher heat transfer rate than that without core-rod. In general, the average heat transfer rate for employing the tape with rod is found to be 8 to 12% better than that for one without core-rod. The increases in Nusselt numbers in the tube with and without core-rod are about 422% and 356% respectively, in comparison with the plain tube

Relationship between the friction factor and Reynolds number for using the helical tape with/without core-rod in the tube is also presented in Fig. 6. A close examination reveals that use of the helical tape with core-rod leads to higher friction loss than that of one without core-rod due to larger contact surface areas and core-flow disturbance. Thus, for the tape without rod, the friction factor could be reduced around 50% below one with core-rod.

Figures 7 and 8 show comparisons between the present experimental data and the present predictions by the correlations below. In the figures, results of the present correlations reasonably agree well within ±10% in comparison with experimental data for the friction factor, and within ±10% for the Nusselt number.

A useful comparison between turbulent and straight flows can be made by comparing heat transfer coefficients at equal pumping power, since this is relevant to the operation cost.

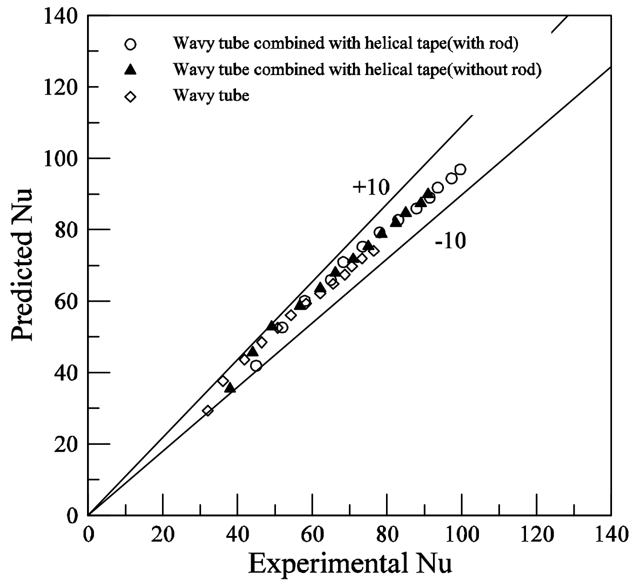


Fig. 7. Comparison of Nusselt numbers predicted by correlations with measured data.

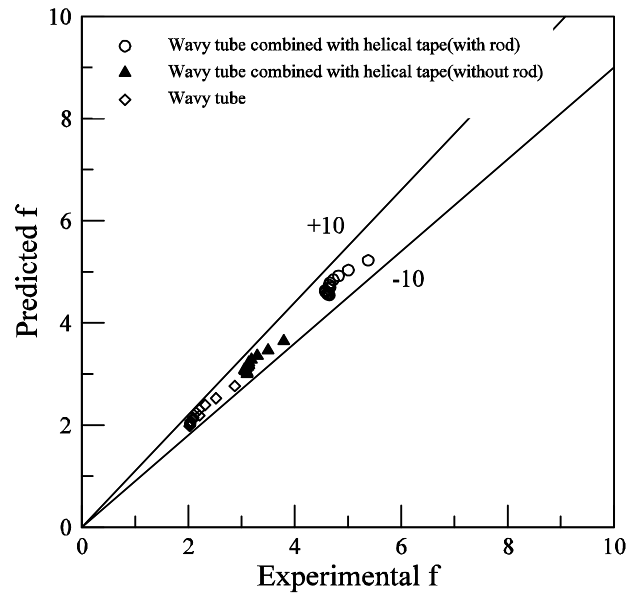


Fig. 8. Comparison of friction factors predicted by correlations with measured data.

For a constant pumping power

$$(\dot{V}\Delta P)_p = (\dot{V}\Delta P)_t \tag{11}$$

and the relationship between friction and Reynolds number can be expressed as:

$$(f\ Re^3)_p = (f\ Re^3)_t \tag{12}$$

The enhancement efficiency (η) at constant pumping power is the ratio of the convective heat transfer coefficient of the helical tape with/without core rod in a wavy-surfaced tube to the plain tube which can be written as follows:

$$\eta = \left.\frac{h_t}{h_p}\right|_{pp} \tag{13}$$

The present results are correlated with the Nusselt number and friction factor for the plain tube as follows:

$$Nu = 0.0135\ Re^{0.85}\ Pr^{0.4} \tag{14}$$

$$f = 0.423\ Re^{-0.275} \tag{15}$$

Correlation for wavy tube:

Nusselt number correlation;

$$Nu = 0.0224\ Re^{0.905}\ Pr^{0.4} \tag{16}$$

Friction factor correlation;

$$f = 30.6\ Re^{-0.3} \tag{17}$$

Enhancement efficiency:

$$\eta = 0.436\ Re^{0.0628} \tag{18}$$

Wavy tube combined with helical tape (without rod):

Nusselt number correlation;

$$Nu = 0.014\ Re^{0.96}\ Pr^{0.4} \tag{19}$$

Friction factor correlation;

$$f = 14.7\ Re^{-0.174} \tag{20}$$

Enhancement efficiency:

$$\eta = 0.448\ Re^{0.0685} \tag{21}$$

Wavy tube combined with helical tape (with rod):

Nusselt number correlation;

$$Nu = 0.014\ Re^{0.948}\ Pr^{0.4} \tag{22}$$

Friction factor correlation;

$$f = 14.238\ Re^{-0.12} \tag{23}$$

Enhancement efficiency:

$$\eta = 0.45\ Re^{0.055} \tag{24}$$

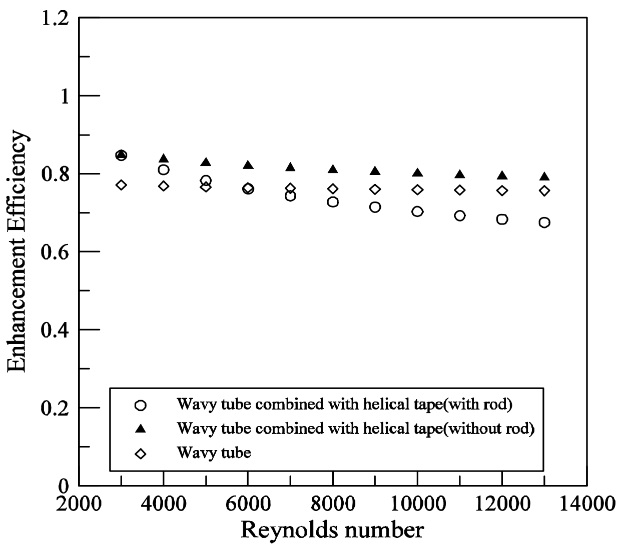


Fig. 9. Enhancement efficiency versus Reynolds number for different turbulators.

The influence of wavy-surfaced tube wall with/without a helical tape insert on heat transfer enhancement efficiency is presented in Fig. 9. It can be observed in the figure that use of the wavy tube fitted with the tape without core-rod leads to higher enhancement efficiency than those of the wavy tube alone and one with the tape with core-rod. Enhancement efficiencies varied between 0.68 and 0.85, 0.79 and 0.85, and 0.75 and 0.77 for the wavy tube with the tape with rod, the tube with the tape without rod, and the tube alone, respectively. The enhancement efficiency tends to reduce with the increase of Reynolds number. The wavy-surfaced tube with tape with core rod shows a faster decrease in the enhancement efficiency than the others for increasing Reynolds number. Close examination reveals that the enhancement efficiencies for all enhancement devices are less than unity. This suggests that the wavy-surfaced tube with/without the helical tape is not feasible in terms of energy saving, especially at higher Reynolds numbers.

5. CONCLUSIONS

An experimental study has been conducted to investigate haet transfer and friction loss behaviors in a circular wavy-surfaced tube with a helical-tape insert using hot air as the test fluid. The experimental result shows that apart from the friction factor, the heat transfer rate can be substantially improved by using both the wavy-surfaced wall and the helical tape. This is due to better mixing behavior between two streams from using the helical tape: swirling flow around the tape and the turbulence flow along the wavy-surfaced wall. The maximum increase in heat transfer rate and friction factor are found to be about 4.2 and 110 times the plain tube for the flow range studied.

NOMENCLATURE

A	heat transfer surface area, m^2
C_{pa}	specific heat capacity of air, kJ/kgK
D	diameter of the test tube, m
f	friction factor
h_x	local heat transfer coefficient, W/m^2K
h	average heat transfer coefficient, W/m^2K
I	current, amp
k	thermal conductivity of air, W/mK
L	length of test tube, m
l	pitch length, m
$\dot{m}$	mass flow rate, kg/s
Nu	Nusselt number
ÄP	Pressure drop, Pa
Pr	Prandtl number
Q_{air}	heat transfer rate of hot air, W
Q_{conv}	convective heat transfer rate, W
Re	Reynolds number
T	temperature, K
t	thickness of test tube, m
U	average axial velocity, m/s
V	voltage, volt
$\dot{V}$	volume flow rate, m^3/s

Greek symbols

μ	dynamic viscosity, Ns/m^2
η	enhancement efficiency
v	kinematic viscosity, m^2/s

Subscripts

b	bulk
i	inlet
m	mean
o	outlet
p	plain tube
p p	pumping power
t	turbulator
w	wall

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