SINGLE-PHASE HEAT TRANSFER AND PRESSURE DROP PERFORMANCCE IN A HORIZONTAL INTERNAL HELICALLY-FINNED TUBE

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ABSTRACT

Enhanced tubes with internally helical fins have been extensively used in the large heat exchangers. This paper experimentally determined the single-phase friction factor and heat transfer coefficients in an internal helically-finned tube. Parameters of the test tubes are nominal inside diameter (22.48mm), fin height to the tube inside diameter ratio (0.022), number of fins (60), and helix angle (45°). Pressure drop data were collected under isothermal condition and heat transfer experiments are conducted in a flooded evaporator. The aqueous ethylene glycol was used as the working fluid and the tests are performed with the Prandtl number from 18.1 to 29.5 and Reynolds number from 7,000 to 37,000. The heat transfer enhancement ratio $h_{i\text{-}en}/h_{i\text{-}p}$ and friction enhancement ratio f_{en}/f_p increased with Reynolds number and reached maximum at Re>17,000, while efficiency index $(h_{i\text{-}en}/h_{i\text{-}p})/(f_{en}/f_p)$ decreased with Reynolds number and reached the minimum at Re>17,000. The maximum friction enhancement ratio of 2.13 and the maximum heat transfer enhancement ratio of 3.47 were obtained, while the corresponding minimum efficiency index was 1.62. The maximum efficiency index of 1.86 occurred at the minimum Re in the heat transfer experiments. This study offers a reference on the design of the heat exchangers.

INTRODUCTION

Internally helical fin has been an effective positive technique for improving the single-phase heat transfer inside the tube. They enhanced the heat transfer coefficients accomplished with pressure drop enlargement. Many investigators have conducted investigations on the single-phase flow and heat transfer performance of the internal helically-finned tubes.

NOMENCLATURE

A	m^2	Heat transfer area
C_i	-	Coefficient for internal heat transfer correlation
C_o	$kW/(m^2 \cdot K)$	Constant for internal heat transfer correlation
c_p	$J/(kg \cdot K)$	Specific heat capacity
D	m	Diameter
e	m	Fin height
f	-	Friction factor
h	$kW/(m^2 \cdot K)$	Heat transfer coefficient
K	$kW/(m^2 \cdot K)$	Overall heat transfer coefficient
L_{HT}	m	Tube length for heat transfer experiment
L_{PD}	m	Tube length for pressure drop experiment
L_T	m	Tube total length
N_s	-	Number of fin starts
Pr	-	Prandtl number
q	kW/m^2	Heat flux
Re	-	Reynolds number
T	K	Temperature
u	m/s	Flow velocity
V	m^3/s	Volumetric flow rate
Special characters		
Δp	Pa	Pressure drop
ΔT_m	°C	Logarithmic mean temperature difference
α	°	Helix angle
η	-	Efficiency index
ρ	kg/m^3	Fluid density

λ	W/(m·K)	Fluid thermal conductivity
Subscripts		
en		Enhanced tube
Gni		Calculated with the Gnielinski equation
i		Inside
in		Inlet
o		Outside
out		Outlet
p		Plain tube
sat		Saturation R134a

Internal helically-finned tube was invented by Fujie et al. in 1977 [1] and was originally employed to two-phase flow and heat transfer. Then it was soon used for single-phase flow in the tube. Carnavos [2] performed the experiment on the single-phase pressure drop and heat transfer performance of the internal helically-finned tube. The developed correlations could predict the turbulent heat transfer and pressure loss performance in the ranges of $0<\alpha<30°$, $10,000<Re<100,000$ and $0.7<Pr<30$. Ravigururajan [3] obtained the pressure drop and heat transfer data of internal helically-finned tube for a wide range of tube parameters and flow parameters. Furthermore, they developed several general correlations for the operation conditions. Jensen and Vlakancic [4] reported fifteen internal helically-finned tubes experimental results and classified the test tubes as high-fin tube and micro-fin tube. They presented the different correlations for these two type tubes applicable for wide flow parameter and geometric parameter range. Webb et al. [5] analyzed that the internal helically-finned tube has both characteristics of internal fin tube and rough tube. In addition, they also presented j-factor and friction factor correlations of for $20,000<Re<60,000$. Wang et al. [6] have correlated the single-phase heat transfer data by heat-momentum transfer analogy. Heat transfer correlations could predict 85.2% of the experimental data within 10% and the friction factor correlations could predict 95.8% of the data within 10%. Zdaniuk et al. [7] have conducted the flow and heat transfer experiment in the Re range of 12,000 to 60,000 for eight internal helically-finned tubes and presented pow-law correlations for Fanning friction factor and Colburn-j factor. In addition, Zdaniuk et al. [8] developed the correlations by the artificial neural networks approach.

Through many investigations have been conducted on the single-phase flow in the internal helically-finned tube, three important factors which affected the tube performance gradually appeared. They are inlet effect, Prandtl number dependency and critical Reynolds number. Meyer and Oliver [9,10] and Tam et al. [11] have reported their works on the inlet effect on the tube performance. Their results showed the smoother inlet such as bell-mouth inlet would delay the occurrence of the transition than square-edged and re-entrant inlets. The investigations on the Prandtl number dependency could be found in refs[12-14]. They investigated mainly on the effect of geometric parameters on the Prandtl number dependency. Meyer and Olivier [9] has found that the secondary transition existed in the transition flow. It was also certificated by the results of Jensen and Vlakancic [4], Tam et al. [11], Li et al. [13], Brognaux et al. [12], Siddique and Alhazmy [15] and our previous work [16]. For the limit knowledge of the tube performance, the experimental data for the internal helically-finned tube are also necessary.

The main purpose of this paper is to determine the single-phase pressure drop and heat transfer performance of an internal helically-finned tube. In addition, the effect of the internally helical fin on the tube performance was discussed.

EXPERIMENT

Experimental System

A test bench was built for obtaining the single-phase flow and heat transfer characteristics in the horizontal tube. This one is the bench introduced in our previous study [16] and was depicted in Figure 1.

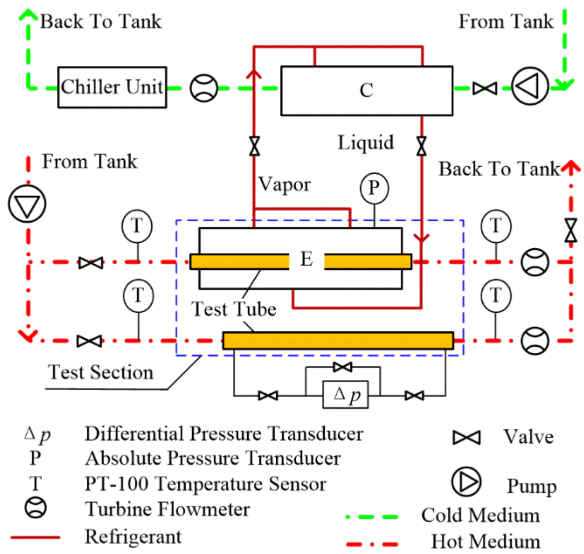

Figure 1 Schematic diagram of experimental system

The refrigerant loop consisted of the upper condenser and the lower evaporator. The refrigerant vapor flowed upward and the liquid flowed downward. The hot and cold resource for the refrigerant loop was the hot medium and cold medium, respectively. The hot and cold medium was provided by the respective tank with constant outlet fluid temperature.

The test section for heat transfer experiment was a flood evaporator. The working fluid flowed in the tube with refrigerant pool boiling on the surface of the tube to provide a uniform wall temperature for heat transfer. The test section for pressure drop experiment was a horizontal tube in which the working fluid flowed isolated with circumstance.

The pressure of the test section was measured with the absolute pressure transducer (0-0.7MPa, 0.1%FS) and was used to calculate the evaporation temperature. The temperatures at the inlet and outlet of the test tube were obtained by several four-wires temperature sensors(PT-100) calibrated within 0.05°C and the flow rates of the flowing fluid were measured with turbine flowmeters (0.6-6m^3/h, 0.5%R). The pressure drop data were obtained with three differential pressure transducers (0-10kPa, 0-20kPa, 0-40kPa, 0.1%FS).

The geometric parameters of the test internal helically-finned tube were the inside diameter(D_i), fin height(e), helix angle(α) and number of starts(N_s) of 22.48mm, 0.5mm, 45° and 60, respectively. The tube total length(L_T), the length of heat transfer(L_{HT}) and that of pressure drop(L_{PD}) are 3045mm, 2940mm and 2945mm, respectively.

The working fluid was water-ethylene glycol mixture and the refrigerant was R134a. The heat transfer experiment was conducted under 10000<Re<34000 and 18.1<Pr<29.5. The pressure drop experiment was performed with 7000<Re<37,000.

Data Reduction

The measured fluid temperature, saturation temperature, flow rates and pressure drop data are used to compute the friction factor and the average heat transfer coefficient, respectively.

Friction factor is determined by the following equation:

$$f = \frac{2\Delta p D_i}{\rho L_{PD} u^2} \tag{1}$$

where Δp is the overall pressure drop, u is the flow velocity and ρ is the density of the working fluid.

The Wilson plot method is used for obtaining the average heat transfer coefficient eventually. The Wilson plot method is shown below:

$$Y = \frac{1}{C_i} X + \frac{1}{C_o} \tag{2}$$

$1/C_i$ is the slope and $1/C_o$ is the intercept. The main purpose is to obtain C_i by keeping the intercept constant. Y and X in eq.(2) are calculated by following equations:

$$Y = 1/K \tag{3}$$

$$X = A_o / A_i / h_{i,Gni} \tag{4}$$

where K is the overall heat transfer coefficient and $h_{i,Gni}$ is the inside heat transfer coefficient calculated by Gnielinski equation. A_o and A_i are the outside and inside heat transfer area, respectively. K is calculated by the log mean temperature difference (LMTD) method:

$$K = q / \Delta T_m \tag{5}$$

$$q = \rho c_p V (T_{in} - T_{out}) / A_o \tag{6}$$

$$\Delta T_m = (T_{in} - T_{out}) / \ln((T_{in} - T_{sat}) / (T_{out} - T_{sat})) \tag{7}$$

In the eq.(5), q is the heat flux and ΔT_m is the logarithmic mean temperature difference. In the eqs.(6) and (7), c_p is the specific heat capacity and V is the flow rate. In addition, T_{in} and T_{out} are the inlet and outlet temperatures of the working fluid, respectively. T_{sat} is the saturation temperature and $h_{i,Gni}$ is computed by the following equation[17]:

$$h_{i,Gni} = \frac{\lambda}{D_i} \frac{(f/8)(Re-1000)Pr}{1+12.7\sqrt{f/8}(Pr^{2/3}-1)} \left[1 + \left(\frac{D_i}{L_{HT}}\right)^{2/3}\right] \tag{8}$$

where λ is the thermal conductivity of the working fluid. Eventually, the inside heat transfer was determined with the correlation of C_i:

$$h_i = C_i h_{i,Gni} \tag{9}$$

Uncertainty Analysis and Validation

The experimental uncertainty analysis for our experimental apparatus has been conducted in our previous work [6]. The uncertainty of experimental friction factor was 3% and the value of the heat transfer coefficient was 7.5%. The friction factor and heat transfer coefficients of the plain tube has been determined and showed good agreement with classical equations.

RESULTS AND DISCUSSIONS

Figure 2 depicted the pressure drop with different flow rate in the enhanced tube. There existed a great discrepancy among the pressure drop with different flow rate while the pressure drop in a same flow rate varied slightly with the Reynolds number. Especially, for V=1.65m^3/h, pressure drop increased with Reynolds number of 7,000 to 15,000. For V=2.50m^3/h, pressure drop increased with *Re* below 16,000 and decreased with *Re* above 16,000. For V=3.35, 4.20 and 4.75m^3/h, pressure drop decreased with Re above 16,000.

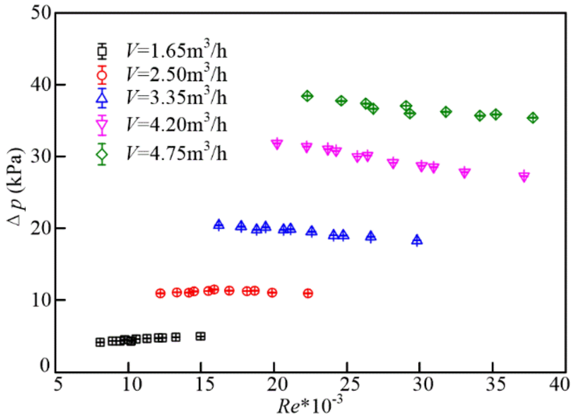

Figure 2 Pressure drop results for the enhanced tube

Figure 3 showed the inside heat transfer coefficients of the enhanced tube. The heat transfer coefficients showed a similarly linear relationship with Reynolds number. The heat data could be predicted by the average totally within 7%. Furthermore, enhanced tube showed larger heat transfer coefficients than the plain tube.

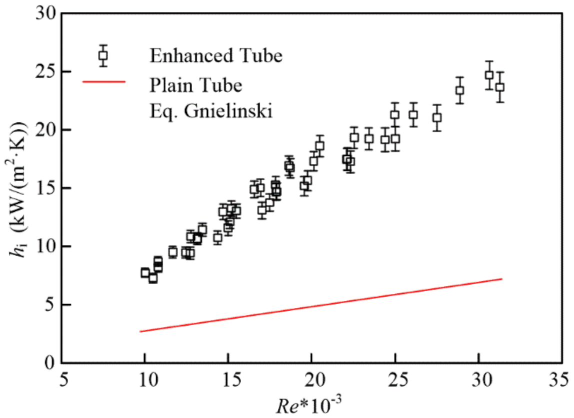

Figure 3 Inside heat transfer coefficients for the enhance tube

To discuss friction enhancement of the enhanced tube, Figure 4 presented the ratio between friction factor of the enhanced tube and that of plain tube. The enhancement ratio of 1.27 to 2.13 increased with Reynolds number before Re=17,000. This indicated that reducing the Reynolds number could be effective to get less pressure drop. In addition, the enhancement ratio got the maximum at Re>17,000. For Re>17,000, the enhancement ratio reached a constant value of 2.13. This indicated that the friction factor of the enhanced tube for *Re*>17,000 could be predicted by the plain tube friction equation correlated by the constant value of 2.13.

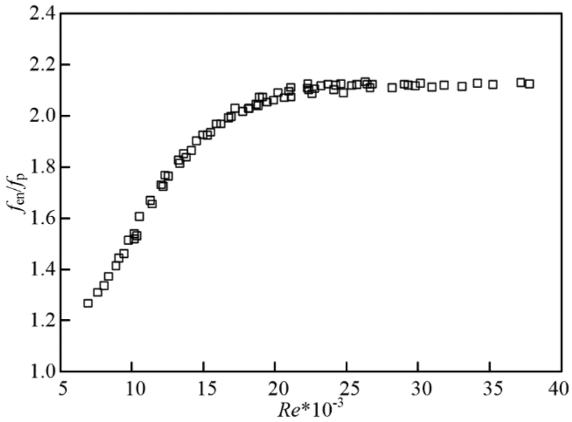

Figure 4 Friction enhancement ratio for the enhanced tube

Figure 5 depicted the heat transfer enhancement ratio for the enhanced tube. The heat transfer enhancement ratio of 2.8 to 3.47 increased with Reynolds number before *Re*=17,000, which is similar to the friction enhancement ratio above. To get higher heat transfer coefficient, the larger Reynolds number should be adopted. The heat transfer enhancement ratio was obviously higher than the friction enhancement ratio. It should be noted that the internally helical fin affects the tube performance positively.

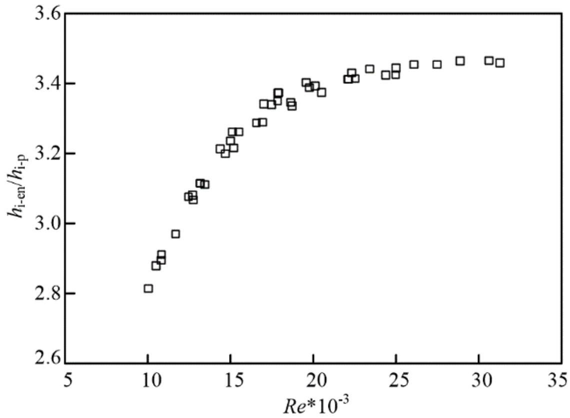

Figure 5 Heat transfer enhancement ratio for the enhanced tube

To compare the heat transfer and friction factor ratios, the efficiency index η defined by $\eta=(h_{i\text{-}en}/h_{i\text{-}p})/(f_{en}/f_p)$ was depicted in Figure 6. The efficiency index decreased with Reynolds number and eventually achieved the minimum at *Re*>17,000. The value ranged from 1.86 to 1.62 and the maximum achieved at the minimum Reynolds number of 10,000. It would be reasonably deduced that the efficiency would be higher at Reynolds number lower than 10,000.

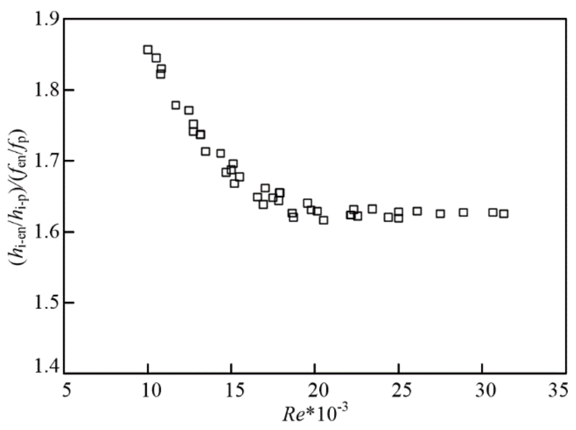

Figure 6 Efficiency index for the enhanced tube

CONLUSIONS

The isothermal pressure drop data for the internal helically-finned tube were obtained and the heat transfer coefficients were presented. Based on the experimental data, the analysis of heat transfer enhancement and pressure drop enhancement was conducted.

The pressure drop in different flow rates showed a significant difference and the value with the same flow rate was almost a constant. The heat transfer coefficients of the internal helically-finned tube showed an approximately linear relationship with Reynolds number.

The friction enhancement ratio f_{en}/f_p increased with Reynolds number and achieved the maximum value of 2.13 at Re>17,000. The heat transfer enhancement ratio $h_{i\text{-}en}/h_{i\text{-}p}$ also increased with Reynolds number and reached the maximum value 3.47 at Re>17,000. However, the efficiency index $(h_{i\text{-}en}/h_{i\text{-}p})/(f_{en}/f_p)$ decreased with Reynolds number and reached minimum value of 1.62 at Re>17,000. The maximum efficiency index of 1.86 occurred in the minimum Re (10, 000) in the heat transfer experiments.

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