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Experimental determination of single-phase pressure drop and heat transfer in a horizontal internal helically-finned tube

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The single-phase pressure drop and heat transfer in an internal helically-finned tube and a plain tube are measured for aqueous ethylene glycol. Pressure drop data are collected under isothermal condition and heat transfer experiments are conducted in a flooded evaporator with R134a boiling outside the tube. The tests are performed with Prandtl number from 17 to 29 and Reynolds number from 5000 to 34,000. The main parameters of the internal helically-finned tube are nominal inside diameter (22.48 mm), fin height to the tube inside diameter ratio (0.022), number of fins (60), helix angle (45°). It is found that plain tube's results agree well with Filonenko and Gnielinski correlations. Friction factors of internal helically-finned tube increase with Re below 17,000, reaching a maximum at $Re_{cr} = 17{,}000$ and decrease above 17,000. This indicates that the conventional criterion $Re > 10{,}000$ for distinguishing turbulent flow isn't suitable for internal helically-finned tubes. A correlation is developed to predict the experimental heat transfer coefficient within 5%. In evaluation of the thermal performance, the j-factor for internal helically-finned tube is about 3.5 times of that for the plain tube and the efficiency index varies from 1.8 to 1.55. This study offers the reference on engineering design and later study.

1. Introduction

Internal helically-finned tubes have become one of the most popular passive heat transfer enhancement techniques because of their larger heat transfer enhancement relative to the increased pressure drop and low cost of the production [1]. Internally enhanced tubes with internal helically-finned on the water-side of large refrigeration evaporators and condensers are routinely used in refrigeration industry [2]. Efficient design of these heat exchangers requires detailed knowledge of the complex single-phase flow and heat transfer processes in the internal helically-finned tubes. This knowledge is strongly dependent upon our ability to analytically model these phenomena and to validate such models with accurate experimental measurements. Despite the extensive application of internal helically-finned tubes, the friction factor and heat transfer correlations are not well established.

The main geometric parameters described the internal helically-finned tube are fin height (e), tube inside diameter (D_i), helix angle (α), number of fins (N_s). Fujie et al. [3] invented the internal helically-finned tube in 1977. Since then, many experimental and theoretical studies have been conducted to investigate the single-phase flow and heat transfer characteristics in internal helically-finned tubes.

As to experimental side, many investigators have reported single-phase flow and heat transfer characteristics over a range of geometric parameters ($0.15\text{ mm} < e < 2.06\text{ mm}$, $6.46\text{ mm} < D_i < 24.41\text{ mm}$, $8° < \alpha < 49°$, $6 < N_s < 78$), Reynolds number ($1849 < Re < 163{,}000$) and Prandtl number ($0.7 < Pr < 200$) [1,2,4–19]. They obtained the variation of friction factor and Nusselt number (or j-factor). For $Re < 3500$, most of their results showed that the heat transfer coefficient of internal helically-finned tube is close to or even less than that of plain tube, but the friction factor of internal helically-finned tube is higher than that of plain tube (Meyer and Olivier [10,11], Tam et al. [12] and Raj et al. [13]). For $3500 < Re < 20{,}000$, there was a very stable secondary transition region for friction factors and j-factors of internal helically-finned tubes (Meyer and Olivier [10,11]). However the Reynolds number (Re_{cr}) which acts as the secondary transition ending point is not yet clear. For example, Meyer and Olivier [10] found Re_{cr} was approximately 10,000, but the experimental results of Tam et al. [12], Jenson and Vlakancic [14], Brognaux et al. [15], Li et al. [5], Siddique and Alhazmy [18] and Celen et al. [19] showed that Re_{cr} was approximately 8400, 12,000, 15,000, 16,500, 11,500, 18,500, respectively. For $Re > 20{,}000$, Webb et al. [2] found the friction factors of internal helically-finned tubes do not attain a constant

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Nomenclature

A	heat transfer area (m^2)
C_i	coefficient for internal heat transfer correlation (–)
C_o	constant for internal heat transfer correlation ($kW/m^2{\cdot}K$)
c_p	specific heat capacity (J/kg·K)
D	diameter (m)
e	fin height (m)
f	friction factor (–)
h_i	internal heat transfer coefficient ($kW/m^2{\cdot}K$)
$h_{i,Gni}$	internal heat transfer coefficient calculated by Gnielinski equation ($kW/m^2{\cdot}K$)
h_o	outside heat transfer coefficient ($kW/m^2{\cdot}K$)
j	Colburn j-factor: $j = Nu/Pr^{1/3}$
K	overall heat transfer coefficient ($kW/m^2{\cdot}K$)
l	tube total length (m)
L	test section length (m)
N_s	number of fin starts (–)
Nu	Nusselt number (–)
Pr	Prandtl number (–)
Q	heat transfer rate (kW)
q	heat flux (kW/m^2)
Re	Reynolds number (–)
Re_{cr}	critical Reynolds number (–)
R_w	thermal resistance of tube wall ($m^2{\cdot}K/kW$)
s	fin base width (m)
T	temperature (K)
u	flow velocity (m/s)
V	volumetric flow rate (m^3/h)

Greek symbols

α	helix angle (°)
θ	fin apex angle (°)
λ	thermal conductivity (W/m·K)
ρ	fluid density (kg/m^3)
ΔP	pressure drop (kPa)
ΔT_m	logarithmic mean temperature difference (K)
η	efficiency index (–)

Subscripts

ave	averaged
c	cold medium
exp	experiment data
F	fanning
f	fluid
h	hot medium
i	inside
in	inlet
o	outside
out	outlet
p	plain tube
pre	predicted by the equations
sat	saturation R134a
w	wall

value at high Reynolds number while Wang et al. [1] concluded that the fiction factors in internal helically-finned tubes show earlier transitions to a fully rough region for Reynolds number higher than 10,000 where the friction factors become independent of Reynolds number. Characteristics of flow inside internal helically-finned tubes are still not very well understood for experimental data are limited.

Based on the experimental study, many investigators had devoted great efforts to analytically model the single-phase flow and heat transfer phenomena. Using the heat-momentum transfer analogy, Wang et al. [1], Brognaux et al. [15] and Han and Lee [20] correlated heat transfer and friction factor correlations as functions of the roughness Reynolds number. Meanwhile, Al-Fahed et al. [4], Siddique and Alhazmy [18], Celen et al. [19] and Bilen et al. [21] correlated friction factor in Blasius type correlation and Nusselt number in Dittus–Boelter type correlation. The former two type correlations couldn't accurately figure out the effect of geometric parameters on the friction factor and heat transfer coefficient. A large number of investigators further developed heat transfer and friction factor correlations as functions of geometric variables and dimensionless parameters. (e.g. Webb et al. [2], Ji et al. [9], Meyer and Olivier [10,11], Jenson and Vlakancic [14], Wu et al. [16], Li et al. [17], Ravigururajan and Bergles [22], Carnavos [23], Mukkamala and Sundaresan [24], Zdaniuk et al. [8,25,26]) However, the existing theoretical models are specific to certain geometries, Reynolds numbers and Prandtl numbers. For $3500 < Re < 20{,}000$ and $D_i > 20$ mm, $\alpha \geqslant 45°$ and $N_s > 55$, none of the existing correlations can be used directly to predict the single-phase flow and heat transfer characteristics in the internal helically-finned tubes.

Internal helically-finned tubes ($D_i = 22.48$ mm, $e = 0.5$ mm, $\alpha = 45°$, $N_s = 60$) are recently used in ice storage system for aqueous ethylene glycol with Re (5000–34,000) and Pr (17–29). The main purpose of this paper is to experimentally investigate the aqueous ethylene glycol flow and heat transfer characteristics in this internal helically-finned tube. A Nusselt number correlation of Dittus–Boelter type is also developed for turbulent flow conditions in the range of $10{,}000 < Re < 32{,}000$ and $17 < Pr < 29$. The current study results can be used for industrial design immediately and provide academic reference to this field.

2. Experimental facilities and procedure

2.1. Experimental system

A schematic drawing of the apparatus used for the current experimental study is shown in Fig. 1. The test ring consists of a refrigerant loop, a hot medium circuit, a cold medium loop and a computer based control and data acquisition system. Refrigerant loop consists of two shell-tube heat exchanges. The upper one is a condenser and the lower one is a flooded evaporator. The refrigerant is driven by natural circulation. Specifically, liquid refrigerant flows down into the evaporator from the condenser while the vapor refrigerant flows up into the condenser from the evaporator. In experiment, the hot medium is the working fluid. The hot medium is obtained from a hot medium tank fitted with six industrial immersion electric heaters each of 2 kW capacity. A centrifugal pump is used to circulate the hot medium from the thermostatic water tank through the test tubes back to the tank. The hot medium is flowing through the inner side of the test tubes, which are placed in the test section. The cold medium is flowing through the inner side of the tubes placed in the condenser, and then flowing through the evaporator of a chiller unit (21 kW cooling capacity) before it returns to the cold medium tank. The cold medium loop is also powered by a centrifugal pump. Hot and cold medium temperatures are set for the electric heaters in hot or cold medium