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Experimental studies on heat transfer and friction factor characteristics of turbulent flow through a circular tube with small pipe inserts ☆

Tu Wenbin [a,b], Tang Yong [a,*], Zhou Bo [a], Lu Longsheng [a]

[a] Key Laboratory of Surface Functional Structure Manufacturing of Guangdong Higher Education Institutes, South China University of Technology, Guangzhou 510640, People's Republic of China
[b] Department of Electrical Engineering, Jiangxi University of Science and Technology, Nanchang 330013, People's Republic of China

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ABSTRACT

Heat transfer performance and pressure drop tests were performed on a circular tube with small pipe inserts. These inserts with different spacer lengths ($S = 100$, 142.9 and 200 mm) and arc radii ($R = 5$, 10 and 15 mm) were tested at Reynolds numbers between 4000 and 18,000. Tap water was used as working fluid. The use of pipe inserts allowed for a high heat transfer coefficient with relatively low flow resistance. The Nusselt number and friction factor increase with the decrease in spacer length. Optimal results were obtained for $S = 100$ mm ($R = 10$ mm). Heat transfer rates and friction factors were enhanced by 2.09–2.67 and 1.59–1.85 times, respectively, to those in the plain tube. Performance evaluation criterion (PEC) values were approximately 1.79–2.17. The Nusselt number and friction factor increase with the decrease in arc radius. Small pipe inserts with $R = 5$ mm and $S = 100$ mm show maximal heat transfer rates of 2.61–3.33 and friction factors of 1.6–1.8 times those of the empty tube. The PEC values were 2.23–2.7. Compared with other inserts, pipe inserts can transfer more heat for the same pumping power for their unique structure.

1. Introduction

The performance of forced convection heat transfer in tubes can be improved through the use of elements such as twisted tape, coiled wire and helical screw-tape [1–3]. The low cost, rapid installation and easy maintenance associated with this augment technique make it attractive compared with other enhancement techniques. Inserted elements typically function as vortex generators by generating swirl flow and modifying velocity distributions. This leads to fluid mixing and the redevelopment of the thermal boundary layers with a resulting enhancement in heat transfer. However, flow resistance is also increased. It is challenging to obtain a high heat transfer rate with low increase in pressure drop.

Researchers have designed various inserts to achieve a high heat transfer rate and low increase in pressure drop. Compared with full-length twisted tapes, multiple short-length twisted tapes can yield a lower pressure drop for the same twist ratio. Saha et al. [4] reported that regularly-spaced twisted tape performed significantly better than full-length twisted tapes at high Reynolds numbers and that the pressure drop decreased by 40%. Ferroni et al. [5] found that multiple short-length twisted tapes yielded pressure drops at least 50% lower than those of most well known full-length tapes. Previous work has adopted techniques such as disrupting the thermal boundary layer and filling of the porous medium to improve heat transfer performance. Murugesan et al. [6] investigated the thermal characteristics of V-cut twisted tape experimentally. The Nusselt number and friction factor for the tube with V-cut twisted tapes were 1.36–2.46 and 2.49–5.82 times, respectively, that of the empty tube. Zhang et al. [7] presented a numerical study on triple and quadruple twisted tapes where an increase of 171% and 182%, respectively, was obtained. However, the friction factors were 4.06 to 7.02 times, respectively, that of the plain tube. Naphon [8] studied the heat transfer characteristics and pressure drop of coil-wire inserts. This insert enhanced the heat transfer significantly, especially in the laminar flow region. Pavel et al. [9] showed that higher heat transfer rates can be achieved by using porous inserts at the expense of a reasonable pressure drop. Helical screw-tape is a modified form of a twisted tape. Unlike twisted tape, which generates swirling flow in two parallel flow directions, helical screw-tape provides a single smooth screw-like direction of motion [10] that greatly reduces flow resistance. Sivashanmugam et al. [11] studied the heat transfer and friction factor characteristics in a circular tube fitted with full-length helical screw elements of different twist ratio and spacer length. The helical screw inserts with spacer were suitable for heat augmentation only in turbulent flow with limited reduction in pumping power. Ibrahim [12] investigated the heat transfer and friction factor in horizontal flat tubes with full length helical screw inserts and found that the Nusselt

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* Corresponding author.
E-mail addresses: soccer_ren@sina.com (W. Tu), ytang@scut.edu.cn (Y. Tang).

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Nomenclature

A	surface area of test tube, m^2
c_p	specific heat of water, J/kg K
D_H	hydraulic diameter, m
d	tube diameter, m
f	friction factor
h	convective heat transfer coefficient, W/m^2 K
k	thermal conductivity, W/m K
l	the length of testing section, m
$\dot{m}$	water mass flow rate, kg/s
Nu	Nusselt number
Pr	Prandtl number
Δp	pressure difference, N/m^2
Q	mean value of heat transfer rate, W
Re	Reynolds number
R	arc radius, mm
R_w	conduction resistance through the tube wall
R_f	fouling resistance
S	spacer length, mm
t	temperature, K
ΔT_m	logarithmic mean temperature difference, K
U	overall heat transfer coefficient, W/m^2 K
v	mean velocity in the tube, m/s
ρ	fluid density, kg/m^3
μ	fluid dynamic viscosity N s/m^2

Subscripts

c	cold
h	hot
i	inside
o	outside
w	wall

number and friction factor decreased with the increase in spacer length and twist ratio for flat tubes. Some researchers realized the difference in flow velocity in different flow regions. They designed a new type of insert, a louvered strip insert, which is small in size in the core flow region where the flow velocity is relatively high and is relatively large in size in the vicinity of the boundary layers. This type of insert resulted in good enhancement and relatively low flow resistance. Fan et al. [13] investigated heat transfer and flow characteristics numerically in a circular tube fitted with louvered strip inserts. Eiamsa-ard et al. [14] investigated heat transfer and friction characteristics in a concentric tube heat exchanger fitted with louvered strips. The increase in the average Nusselt number and friction loss for the inclined forward louvered strip was 284% and 413%. Guo et al. [15] investigated the thermo-hydraulic performance of conical-strip inserts and found that the performance factor was enhanced by 36–61% that of the conical-ring inserts at the Reynolds range of 5000 to 25,000.

An increase in shear stress near the boundary layer inevitably results in an increase in the friction factor. However, it could be very different by changing the velocity and temperature field according to the velocity and temperature distribution in the tube. When fluid flows through a tube, the maximum flow velocity exists in the center and the minimum velocity near the wall of the tube. Temperature also has an uneven distribution during heat exchange. Mixing fluids with different temperatures and velocities will result in a significant heat transfer coefficient. Most studies focus on vortex intensity and disregard the distribution of the velocity and temperature field, which may lead to unnecessary disturbances and increase flow resistance. It is possible to devise a structure that distributes the temperature and velocity fields by design. In this paper, a new type of insert, consisting of small pipes, has been developed. The pipes enable fluid at different temperatures to flow easily and result in a uniform temperature. The geometries of the small pipe inserts are shown in Fig. 1, where some small pipes are mounted on a stainless steel rod and are placed inside the inner test tube. It is expected that a high heat transfer coefficient will be obtained with limited increase in pressure drop. An experiment was performed to investigate the pressure drop and thermal characteristics of the inserts.

2. Experimental setup

A schematic diagram of the experimental setup is shown in Fig. 2. It consists of a test section (heat exchanger), cooling water loop and hot water loop. The test section consists of two concentric tubes. Hot water flows through the inner tube and cold water flows through the annulus. The inner and outer tube diameters are 16 and 38 mm, respectively. The inner tube is copper (1300 mm long, 2 mm thick) and the outer tube is Plexiglas (1300 mm long and 5 mm thick). Water is heated in a hot water tank (HH-W600; HengFeng Electronics Manufacturing, Inc., JiangSu Province, China) and its temperature is maintained at 70 °C by a proportional integral derivative controller inside the water tank. The cold water temperature was maintained at ~30 °C. Water is pumped through the test section by a pump with flow rate controlled by a control valve. The inlet and outlet water temperatures are measured by four T-type thermocouples (Omeron Inc.). All thermocouples were calibrated before testing with an accuracy of ±0.1 °C. The water flow rate through the test section was measured using a rotameter with an accuracy of ±1%. The pressure in the test tube was measured by a pressure transmitter (P/N:DP1300-DP7E22M4B1N, Senex Inc.) with an accuracy of ±0.5%. Temperatures and pressures were recorded by using a data logger (Agilent Data Acquisition Unit 34970A).

The pipe inserts were copper (2 mm diameter and 0.5 mm thick) with the method of fabrication as shown in Fig. 3(a). A small pipe 20 mm in length was formed into an S-shape. Three such pipes were mounted on a 1000 mm stainless rod. The friction factor can be affected by spacer length [11]. Inserts with different arc radii have different slant angles and could have a significant effect on performance. The effect of slant angle on heat transfer performance is discussed in [14]. Based on these considerations, small pipe inserts with different spacer lengths ($S = 100$, 149.2 and 200 mm) and arc radii ($R = 5$, 10 and 15 mm) were tested in this work. The pipe insert geometries are shown in Fig. 3(b) and (c).

3. Data processing

Data processing of the measured results is summarized in the following procedures.

The heat transfer rate of cold water in the annulus, Q_C, can be written as:

$$Q_c = \dot{m}_c c_p \left(t_{c,o} - t_{c,i}\right) \tag{1}$$

where $\dot{m}_c$ is the cold water mass flow rate, c_p is the specific heat of water and $t_{c,i}$ and $t_{c,o}$ are the inlet and outlet cold water temperatures, respectively.

The heat transfer from the hot water in the test section, Q_h, can be expressed as:

$$Q_h = \dot{m}_h c_p \left(t_{h,i} - t_{h,o}\right) \tag{2}$$

where $\dot{m}_h$ is the mass flow rate of hot water and $t_{h,i}$ t and $t_{h,o}$ are the inlet and outlet water temperatures, respectively.