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Thermal performance characteristics in plain horizontal tubes influenced by combined wire coil and twisted tape inserts

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Abstract. In this paper, heat transfer, friction factor behaviors in a tube equipped with the combined devices between two different types of twisted tape y=2.7 and y=3.0 and a wire coil pitch ratio are experimentally investigated. This paper presents heat transfer enhancement in single-phase transitional flow by wire-coil insert with geometrical parameters rib height-to-diameter ratio e/D=0.071, and rib pitch-to-height ratio p/e =7.5. The experiments are at different measured fluid temperatures in a turbulent flow regime with Reynolds numbers ranging from 1000 to 10000 using water as the test fluid. At Reynolds number 2000 the highest Nu* is of around 5.6

1. Introduction

The main purpose of heat transfer enhancement techniques is to augment heat transfer and increase the thermal performance of the system. The techniques can be diveded into two categories: active and passive methods. Supplying extra energy to the fluid is used in the active methods.

Rotating the surface, mechanical auxiliary elements, constituting electrostatic areas in the flow area, mixing fluids with mechanical accessories are active methods. The passive enhancement doesn't acquire any external energy. The approaches of this category take account of rough surface, surface area extension, coated surface, and turbulator, swirl generator device. One of the most promising techniques is insertion of turbulator or swirl generator devices. Reducing the thickness of the boundary layer is one of the major functions of turbulator or swirl generator which could introduce better fluid mixing [1].

Wire coils and twisted tapes have taken great attention due to their low cost and promising performance [2]. Numerous research works have been reported with taking into account the effect of coil pitch, on heat transfer and friction characteristics, coil-wire thickness for wire coil inserts or twist ratio and tape thickness for twisted tape inserts [2–16]. The research showed that twisted tape insert disturbs the entire flow field while the wire coil insert mainly perturbs the flow near the wall. In heat transfer enhancement point of view the twisted tape is more effective than wire coil inserts [18]. When we get a crucial restriction, wire coil seem to be more useful due to its less pressure drop penalty. According to Wang and Sunden, in order to gain better heat transfer enhancement it is useful for heat transfer enhancement device and also for consideration of some device combinations.

The purpose of this paper is to report the results from experimental study of heat transfer and pressure drop in the turbulent flow regime of heating water flowing in horizontal circular smooth tube.

2. Experiments

2.1. Experimental Set-up

A schematic diagram of the overall experimental set-up [17] for heat transfer and pressure drop experiments is shown in Fig. 1. The test section consisted of a tube-in-tube heat exchanger in a counter flow configuration, 1. The inner tube is a horizontal copper circular tube with an inside diameter of 14.0 mm and an outside diameter of 16.0 mm. The total length of the test section, for isothermal friction factor

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measurements, was 3 m, providing a maximum length-to-inside diameter ratio (L/D_i) of 214, while for the heat transfer and non-isothermal friction factor measurements the length of the heating test section was 2.0 m and this ratio was 143.

Water was used as a working fluid for both streams, with the inner fluid being cold and the annulus fluid being hot. The temperature of the tested fluid (cold water in the inner tube) was changed in the range 5.5-50^{o} C to insure the variation of the Prandtl number in the range $3.5 < \mathrm{Pr} < 10.0$. The temperature of the hot water in the annulus was changed in a such way to maintain the temperature difference between hot and cold water as small as possible. In this way, the variation of the wall temperature was never greater than 2^{o}C at the length of the tube. Cooling of the test fluid of the inner tube and heating the water in the annulus were performed by secondary flow loops containing water from large reservoirs. The 200-litre reservoir 2 for annulus water was heated by an electric heater, while the reservoir 3 for the inner fluid (also 200 litre) was heated by an electric heater or cooled by a chiller 4.

The test fluid was pumped through the system with two pumps 5 installed in parallel and used in accordance with the flow rate requirements. From the reservoirs 2 and 3, the water flowed through a set of flow meters 6 that measured the volume flow rate. A total of three flow meters were used, two in parallel for the working fluid, used according to the flow rate requirements. After the flow meters, the fluids flow through the experimental section and then back to into the reservoirs. The working fluid passed the mixing chamber 7 and smooth tube entrance section 8 and run through the smooth or rough test tube, followed by another mixing chamber 9 and cooler 10. An additional water tank 11 was used to sustain the lower temperatures of the working fluid.

At both the inlet and the outlet, the tube tested was equipped with two measurement stations 12, each one including four pressure taps set 90 deg apart in a cross section installed by drilling a 1 mm hole through the copper tube. The static pressure from the measurement stations (the average of the tap outputs) was connected to the differential pressure transducer 13 by nylon tubing.

For obtaining a representative inner-tube wall temperature, 18 of the 22 thermocouples were placed along the length of the inner-tube's outer-wall at nine, equally spaced stations. The remaining thermocouples were placed at the in- and outlets of the test section. All readings from thermocouples and differential pressure transducer were logged by data-acquisition system 14.

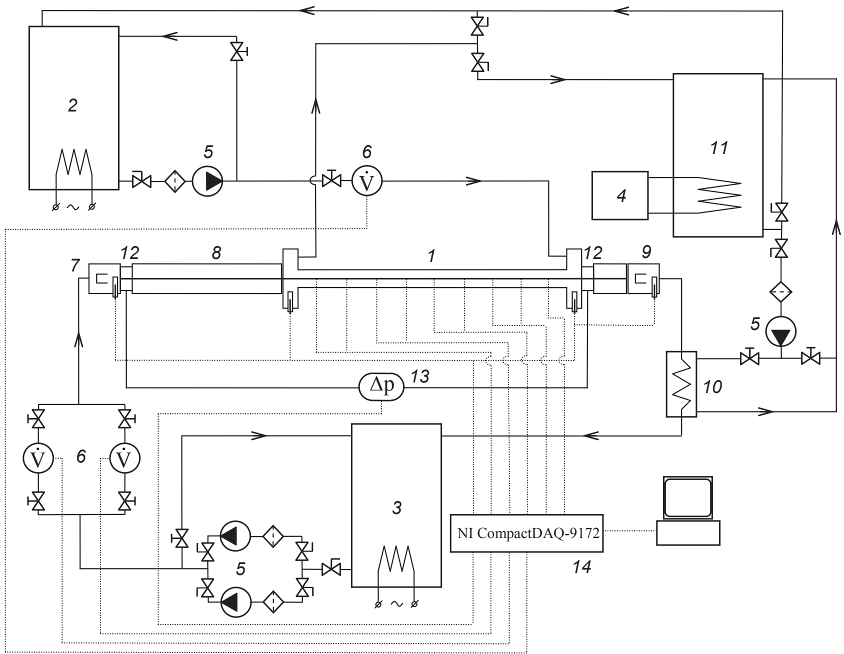

Figure 1. Experimental set-up.

3. Data Reduction

Experiments have been conducting in the laminar, transitional and turbulent regimes under isothermal and heating flow conditions. The experimental data were used to investigate the effect of heating on the entrance and fully-developed friction factors from the laminar to turbulent regions and to establish the validity of the isothermal and non-isothermal correlations wellestablished in the literature.

3.1. Fanning Friction Factor and Heat transfer

The isothermal and non-isothermal pressure drop studies were conducted at different temperatures of the water in the range 5.5-50 ºC over the range of Reynolds number $1.2\times10^3 < \mathrm{Re} < 1.7\times10^4$. The pressure drop data were converted to the Fanning friction factor by the equation

$$f = \frac{\pi^2 D_i^2}{32\rho L}\frac{\Delta p}{\dot{V}^2} \tag{1}$$

where Δp is obtained from the readings of the differential pressure transducer, and $\dot{V}$ - from the measurements of the volume flow rate. The fluid properties were calculated at the average inner-tube fluid temperature, except the water density which was calculated at the inlet temperature.

Heat transfer studies were carried out to obtain values for the inner's tube average heat transfer coefficient h_i and annulus heat transfer coefficient h_o . Since the wall temperature was measured, the individual coefficient was determined from the equation

$$\dot{Q} = h_i A_i \, \Delta T_{m,i} = h_o A_o \, \Delta T_{m,o}\,, \tag{2}$$

where $\dot{Q}$ is the average heat flow of $\dot{Q}_i$ (based on cooling water) and $\dot{Q}_o$ (based on heating water) given by

$$\dot{Q}_i = \dot{m}_i \, c_{p,..i}(T_{i,o} - T_{i,i}) \tag{3}$$

and

$$\dot{Q}_o = \dot{m}_o \, c_{p,o}(T_{o,i} - T_{o,o})\,. \tag{4}$$

Only those runs for which the heat balance error, given by

$$hb = \frac{\left|\dot{Q}_i - \dot{Q}_o\right|}{\left(\dot{Q}_i + \dot{Q}_o\right)/2} \times 100\%\,, \tag{5}$$

was less than $\pm 5\%$ were processed for evaluation of h_i and h_o. The log-mean temperature differences $\Delta T_{m,i}$ and $\Delta T_{m,o}$ were used to determine the heat transfer coefficients h_i and h_o from Eq. (2), where $\Delta T_{m,i}$ was defined by

$$\Delta T_{m,i} = \frac{\Delta T_{x=o} - \Delta T_{x=L}}{\ln\left(\dfrac{\Delta T_{x=o}}{\Delta T_{x=L}}\right)}, \tag{6}$$

where $\Delta T_{x=o} = T_w(o) - T_{i,i}$ and $\Delta T_{x=L} = T_w(L) - T_{i,o}$.

It became clear from the wall temperature measurements along the length of the tube that the variation of the wall temperature along the length of the tube could be outlined with a linear correlation $T_w(x) = a_1 + a_2 x$ with constants a_1 and a_2 calculated for each regime. The values of the temperatures $T_w(o)$ and $T_w(L)$ have been calculated by extrapolation of the linear correlation for $x = o$ and $x = L$. The influence of the wall resistance on the heat transfer coefficient in all cases was found to be negligible.

4. Exposition

4.1. Results and discussion

The experimental results are presented with isothermal friction factor results and heat transfer results. Fig. 2 show the experimental friction factor results at y=3.0 simultaneously at isothermal conditions.

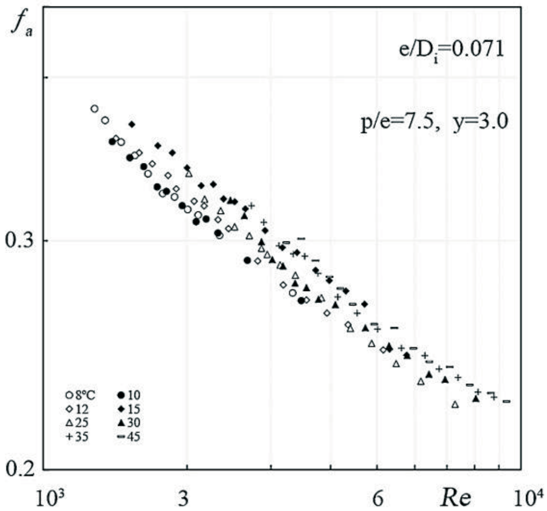

Figure 2. The variation of adiabatic friction factors with Reynolds number.

Fig. 3 shows friction factor augmentation $f_* = f / f_s$ produced by the wire coil. The tendency of friction factor behaviour commented in the foregoing discussion is very clean expressed.

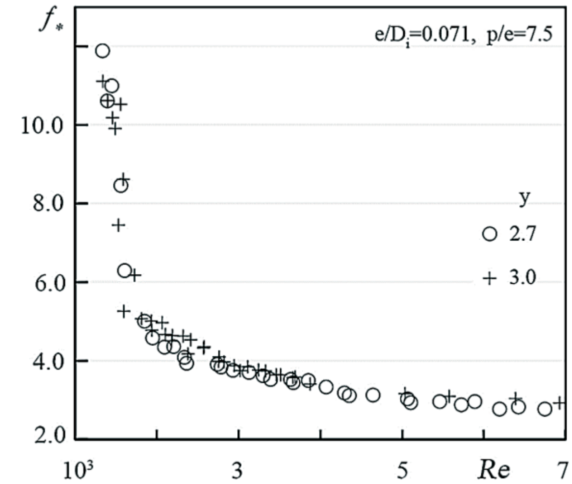

Figure 3. The variation of the ratio f_* with Reynolds number.

As seen, for Reynolds number in the range 1100– 2100, the friction factor ratio f_* decreases from 11.0 to 4.0. After that it gradually decreases with the increase of Re.

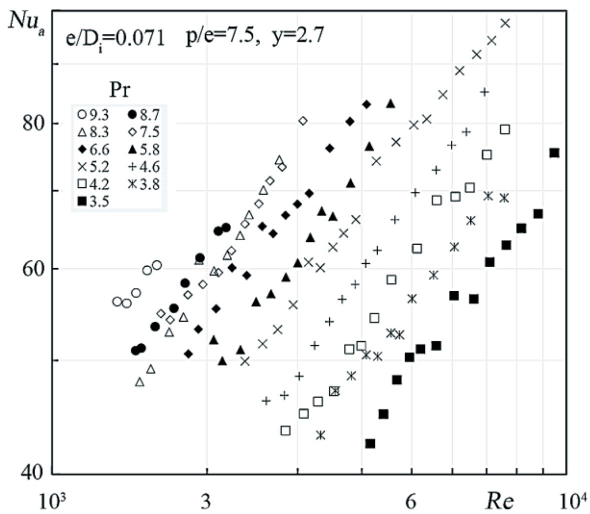

Figure 4. The variation of Nusselt number with Reynolds number for different Prandtl numbers.

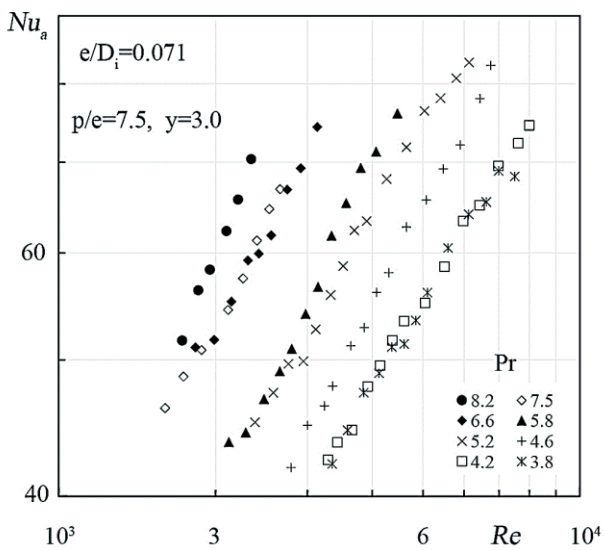

Figure 5. The variation of Nusselt number with Reynolds number for different Prandtl numbers.

Fig. 4 and fig.5 show the experimental Nusselt numbers for a fully developed hydrodynamic boundary layer at the inlet but with a thermal boundary layer still developing. A total of 150 data points represents the data in transition and turbulent regimes.

Nusselt number augmentation is defined by the ratio $Nu_* = Nu / Nu_s$ at the same Reynolds and Prandtl numbers. Fig. 6 shows the ratio Nu_* for the wire-coil inserts studied. In the transition region (1 100 < Re < 4 000) Nu^* reaches its maximum values 5.6 and after that it decreases gradually and then remains constant.

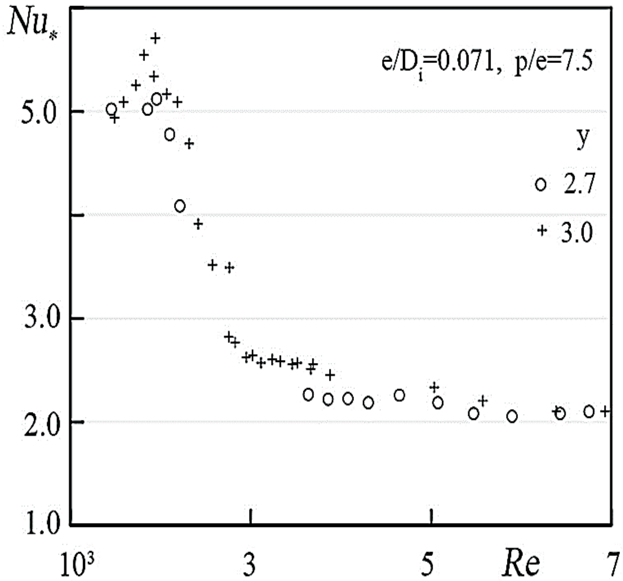

Figure 6. The variation of the ratio Nu_* with Reynolds number.

5. Conclusion

A comprehensive experimental study has been carried out on a wire coils inserted in a smooth tube, with geometrical parameters $e / D_i = 0.071$ and $p / e = 7.5$, covering transition and turbulent flow regimes: Re = 1 100-10 000 and Pr = 3.7-10. A huge number of experimental points for friction factor and heat transfer coefficient have been obtained. As seen from the Figs. 3 and 6, the ratios f_* and Nu_* have a maximum value at one and the same Reynolds number around 2100.

The variation of this ratios with Reynolds number are required to be able to evaluate the real benefit of the use of this particular heat transfer augmentation technique into practice.

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