EXPERIMENTAL STUDY ON THERMAL AND HYDRAULIC BEHAVIOR OF MICRO-FIN TUBES IN SINGLE PHASE

Jacqueline Biancon Copetti
Universidade do Vale do Rio dos Sinos - Unisinos
Av. Unisinos, 950 Cep 93022-000 – São Leopoldo - RS
copetti@euler.unisinos.br

Mario Henrique Macagnan
Universidade do Vale do Rio dos Sinos – Unisinos
mhmac@euler.unisinos.br

Daiana de Souza
Universidade do Vale do Rio dos Sinos - Unisinos
dsouza@euler.unisinos.br

Rejane De Césaro Oliveski
Universidade do Vale do Rio dos Sinos - Unisinos
decesaro@euler.unisinos.br

Abstract. *This work provides heat transfer and friction characteristics for water single-phase flow in micro-fin tubes. The analysis of thermal and hydraulic behavior in a laminar to turbulent flow was carried out in an experimental setup with a 9,52 mm diameter micro-fin cooper tube. The tube was externally wrapped by an electrical resistance tape to supply a constant heat flux to its surface. Different operational conditions were considered in the tests (heat flux of 29,7 kW/m^2 and mass velocity of 151 to 2354 kg/sm^2). The inlet and outlet temperatures, differential wall temperatures along the tube, pressure drop and flow rate were measured. The transducers were connected to a computer data acquisition system that allows controlling the principal variables of the tests. The relations of heat flux and flow rate with heat transfer coefficient and pressure drop were analyzed. For the same conditions, comparative experiments with an internally smooth tube were conducted. The micro-fin tube provides higher heat transfer performance than the smooth tube (in turbulent flow $h_{micro\text{-}fin}$ / h_{smooth}=2.9). In spite of the increase in pressure drop ($\Delta p_{microfin}/\Delta p_{smooth}$=1.7), the heat transfer results were significantly higher (about 80%), showing the advantages of this enhanced configuration in thermal performance relating to conventional tubes. The smooth tube results were validated by comparison with the Dittus-Boelter and Gnielinski correlations. For the micro-fin tube an empirical correlation to heat transfer coefficient adjusted from the set of measured data is proposed, showing a good agreement with experimental results.*

Keywords*. Heat transfer enhancement, pressure drop, micro-fin tubes, single phase flow*

1. Introduction

The micro-fin tubes have been used in coolers, condensers or evaporators for cooling systems and air conditioning. These tubes allow to combine a larger efficiency with more compact heat exchangers and, therefore, require less refrigerant load. This is a technology that has been capable of benefiting the increment of the heat transfer without similar increase in the pressure drop, mainly for two- phase applications.

The cooper tubes with internal microfinned surface and small diameter were introduced for the first time in the end of the 70s by Hitachi Cable Ltd. Since then, these tubes have found a great application in the heat exchangers market, replacing the smooth tubes. According to Muzzio et al (1998) it is possible to estimate that the international market of micro-fin tubes corresponds to 30% of all the heat exchangers production, while in Brazil the utilization of these tubes amounts to 10% of the production. It is possible to observe a constant evolution in this area, which justifies the effort that is being made to characterize the behavior of these tubes in different industry applications.

The micro-fins promote a major turbulence, intensifying the heat transfer, but at the same time, they introduce an increment in the pressure drop. Thus, for heat exchangers design, it is important to determine the heat transfer coefficient and the friction factor for different flow characteristics of the applications. This process is basically experimental, because it depends on the micro-fin geometry, the fluid and mode of operation: single-phase or two-phase (evaporation or condensation).

As long as geometry is concerned, many micro-fin configurations, such as rectangular, triangular or trapezoidal, have been proposed in the last years. The principal geometric parameters that characterize these tubes are the external diameter, height (from 0.075 to 0.4 mm), helix angle (from 10 to 35°) and the number of fins (from 50 to 60). Besides, there are some important dimensionless parameters such as roughness height (height/internal diameter or e/D) and the roughness spacing (pitch/height or p/e). A micro-fin tube normally presents e/D=0.02-0.04 and p/e=1.5-2.5 (Brognaux et al., 1997). Tubes whose specifications are out of these ranges, would be considered finned tubes.

Concerning the type of fluid and thermal exchange, a lot of research about thermal and hydraulic performance of micro-fin tubes with different refrigerants has been done. According to the review of Jabardo et al. (2000) and Newell (2001), most experimental studies were achieved with HCFC refrigerants in two-phase, for example, the works of Kuo and Wang (1996), Chamra et al. (1996) and Schlager et al. (1990) with R22. With HFC refrigerants there are works with R134a (Singh et al., 1996 and Nídegger et al.,1997) and R410a (Cavallini et al., 2000 and Eckels, 1997).

The popularity of micro-fin tubes is due to the great augmentation in the heat transfer during evaporation and condensation (from 50 to 100% relating to smooth tubes) and the little increase in the pressure drop (from 20 to 40%) for the same processes.

Although many works in this area have been published, very little has been done to explain the single-phase heat transfer in micro-fin tubes. The potential use of these tubes in single-phase, with sensible heat exchange, is not well established, even though data in these conditions are interesting in the subcooling region of air, condensers, water coolers of steam cycles compression and water absorption cycles. A few works in this field were produced by Al-Fahed et al. (1993), Chiou et al. (1995, 1996) and Brognaux et al. (1997), who studied the characteristics of flow and heat transfer of water in single-phase to different geometries.

This work presents an experimental study to characterize the thermal and hydraulic performance of water in single-phase flow in a 9,52 mm micro-fin tube, which is the most usual dimension. For comparison, the tests are carried out in a smooth tube with the same dimensions and conditions. This is a reference to discuss the effects of micro-fins. The results were used to adjust theoretical correlations for heat transfer and as a base to understanding and expanding them to two-phase flow.

2. Experimental setup and operational conditions

The experimental setup allows to test and evaluate the flow in cooper tubes internally microfinned and smooth under different conditions. Each tube is heated by a resistance tape (9 Ω/m) wrapped on its surface to supply a constant and controlling heat flux. To avoid losses of heat, fiberglass insulation is used over the tubes. Measurements of temperature, pressure and flow rate were recorded in some points of the tube, as can be seen in the Fig. (1).

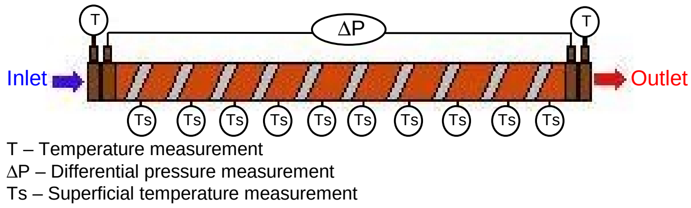

Figure 1. Diagram of tube with temperature and pressure measurements points.

Measurements of inlet and outlet fluid temperatures were carried out using RTD-PT100 sensors. These sensors were calibrated in an Isocal 6 Venus 2140 Plus bath, using a precision thermometer (0.1°C scale). In the tube surface ten thermocouples type T were fixed, whose assembly allows to use the cooper tube itself as an of the thermocouple elements. Constantan wires were welded over the tube in the longitudinal direction and joined with the cooper of the tube forming an assemblage denominated differential thermocouples (Fig. 2), which makes it possible to determine temperature gradients along the surface. Tests showed that these differential measurements temperature system minimize errors, when compared with conventional system of two wires thermocouples, since all the thermocouples measure the temperature relation to the same reference (Souza et al, 2001).

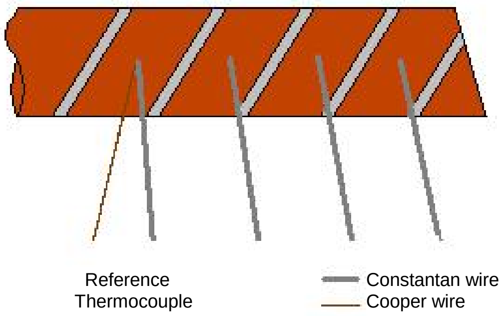

Figure 2. Thermocouples assemby for the differencial wall temperature measurements.

The pressure difference between inlet and outlet of the tube was measured using a pressure transducer with an inductive sensor (ABB - 600T series), with 0.31% of precision in the range from 0 to 100 kPa. The flow rate measurements were taken by an electromagnetic sensor (ABB type Magmaster with 0.1% of precision in the calibration range from 0 to 2 L/s). A frequency inverter (Toshiba, model VF-S9) was connected to the pump to control the flow water rate. The heating power is recorded by a Fluke power meter (Model 39).

The transducers were connected to an acquisition data system composed by a computer with a multimeter HP 34970, through an interface HP-IB. The program of the BenchLink Data Logger–HP was used to data acquisition, which allow us to follow and register the variation on different tests parameters.

The water flow in a closed circuit, goes trough the test section (micro-finned or smooth tube) where it is heated. Then it is cooled in a cooling tower returning to the system, such as we can see in the diagram of the Fig. (3).

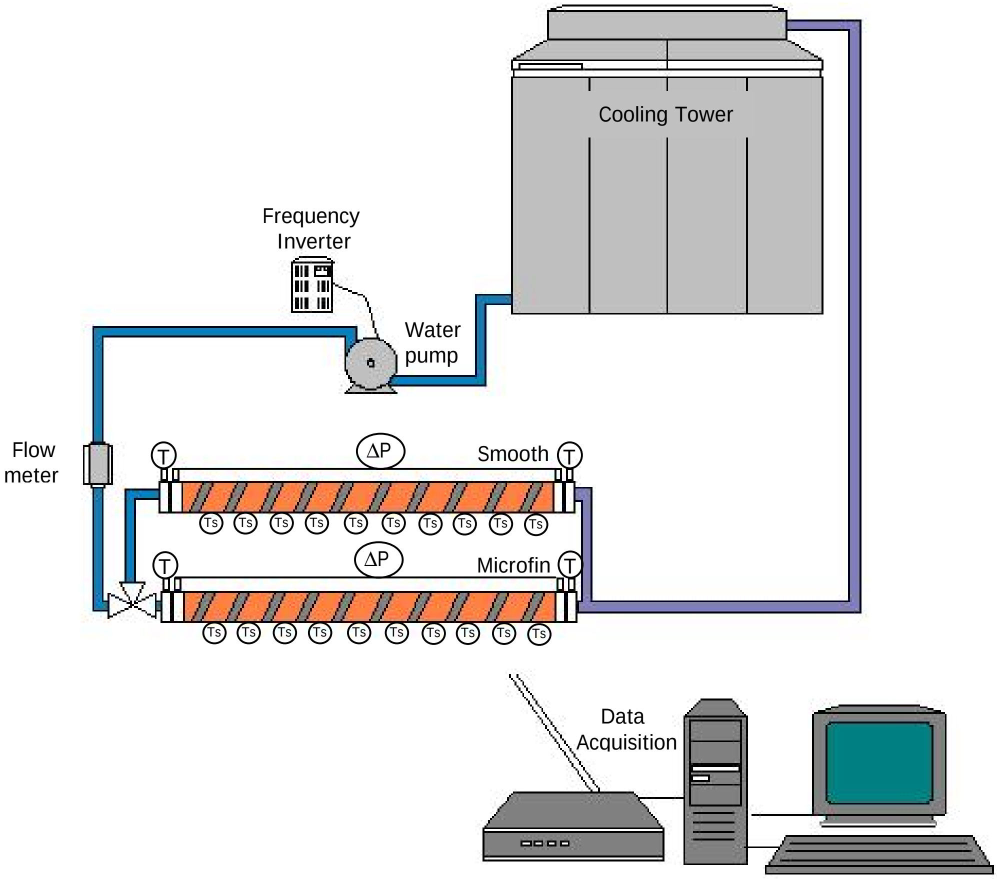

Figure 3. Schematic diagram of test facility.

Two tubes were tested, microfinned and smooth, and the main geometrical parameters of these tubes are listed in Tab. (1).

Table 1. Tested tubes geometry.

Geometrical parameters	Tubes	
	Microfinned	Smooth
Outside diameter (D_o) [mm]	9.52	9.52
Inside diameter (D_i) [mm]	8.95	8.87
Tube thickness (t) [mm]	0.29	0.33
Cross-sectional flow area (As) [cm^2]	0.61	0.62
Inside total surface area (Ai) [cm^2/m]	409.3	281.2
Perimeter (P) [mm]	40.93	28.12
Length (L) [mm]	1090	1090

In Fig. 4 the distinctive parameters of micro-fin tube are indicated, such as the height of the fins, e, the helix angle between the fin and the axial axis of the tube, α, and the apex angle, θ. The transversal section of the micro-fin is shown in the figure and it was obtained through a metallographic microscope (amplification in 20 times). The inside diameter of micro-finned tube (Di) was measured with the aid of a microscope and the parameters depend on it, for example the sectional tube area (As), used the equivalent diameter concept, according to Brognaux et al. (1997). Table

(2) presents more about micro-fin geometrical description including the dimensionless variables, such as roughness height, e/D, and the roughness spacing, $p/e=(\pi Di)/(n_f e)$.

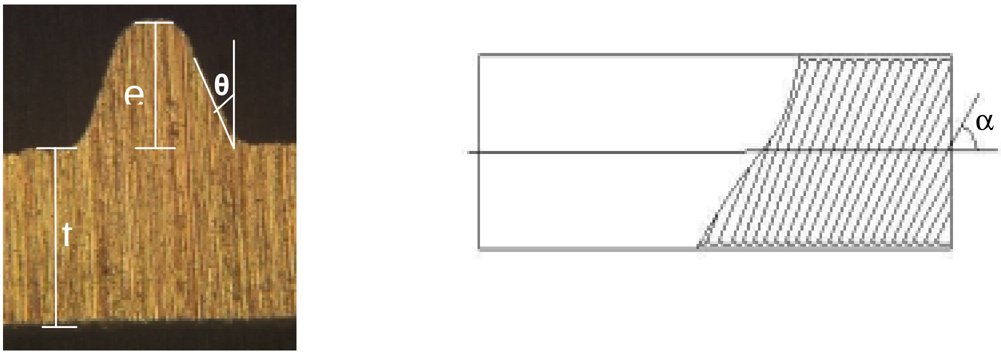

Figure 4. Micro-fin and the characteristics parameters.

Table 2. Micro-finned tube geometrical parameters.

Number of fins (n_f)	60
Fin height (e) [mm]	0.2
Helix angle (α) [degrees]	18
Apex angle (θ) [degrees]	40
Dimensionless fin height (e/D)	0.022
Dimensionless fin pitch (p/e)	2.34
Equivalent diameter [mm]	9.095

Many tests were carried out with the aim of verifying the thermal and hydraulic behaviour in different flow rates, from laminar to turbulent, for constant heat flux on the tube surface and for both micro-finned and smooth tubes. Experimental test conditions are shown in Tab. (3). Each test was carried out three times to verify the results repetitively.

Table 3. Test conditions.

Heat surface flux (q''_s) [kW/m^2]	33
Mass velocity (G) [kg/sm^2]	160 a 2400
Inlet temperature (T_i) [°C]	18

3. Data reduction

The data of temperature, flow rate and pressure drop were analysed using a computer data reduction program. From this analysis, the average convective heat transfer coefficient, h, and the friction factor were calculated and the behaviour of these parameters with flow rate was evaluated. Fluid properties were considered at the mean temperature along the test section.

3.1 Heat transfer

The thermal condition at the surface in this case can be approximated to the constant heat flux, realized by an electric resistance heating uniformly from all directions. Therefore, the mean fluid temperature in each point of the internal flow along of the tube can be expressed as the Eq. (1) (Incropera and DeWitt, 1998).

$$T_m(x) = T_i + \frac{q''_s P}{\dot{m}c} x \tag{1}$$

where Tm (x) is the mean fluid temperature in the x tube position, in °C, T_i is the inlet temperature, in °C, q''_s is the heat flux, in W/m^2, P is the perimeter, in m, $\dot{m}$ is the rate of mass, in kg/s, c is specific heat, in J/kg°C and x is the tube position where the surface temperature was measured. Note that the mean fluid temperature increases linearly in the flow direction, since the surface area (P.x) increases.

The local heat transfer coefficient, h(x), is obtained at any location on the tube from:

$$h(x) = \frac{q_s''}{T_s(x) - T_m(x)} \qquad (2)$$

where T_s (x) is the surface temperature measured in x's tube positions along the tube, as is indicated in Fig. (1).

The mean heat transfer coefficient for the tube, h, was attained considering the local coefficients, h(x), of Eq.(2). This procedure allows achieving a more precise value for h. It is also possible to verify the coefficient variation along the flow tube for different rates.

3.2. Pressure drop

The friction factor, f, is obtained from the pressure drop measurements, Δp, and is given by:

$$f = \frac{2\rho D(\Delta p - \Delta p_{loss})}{G^2 L} \qquad (3)$$

where Δp_{loss} is the head loss, which accounts the entrance and exit pressure losses due to the connection of the tested tube, and its value change with flow rate (from 0 to 13 kPa). D is the inside diameter for the smooth tube and equivalent diameter for the micro-finned one.

4. Results and discussion

Figure (5) presents the thermal behavior of the tubes, or the variation in the mean heat transfer coefficient, h, for different water mass velocity, G, for micro-fin and smooth tubes.

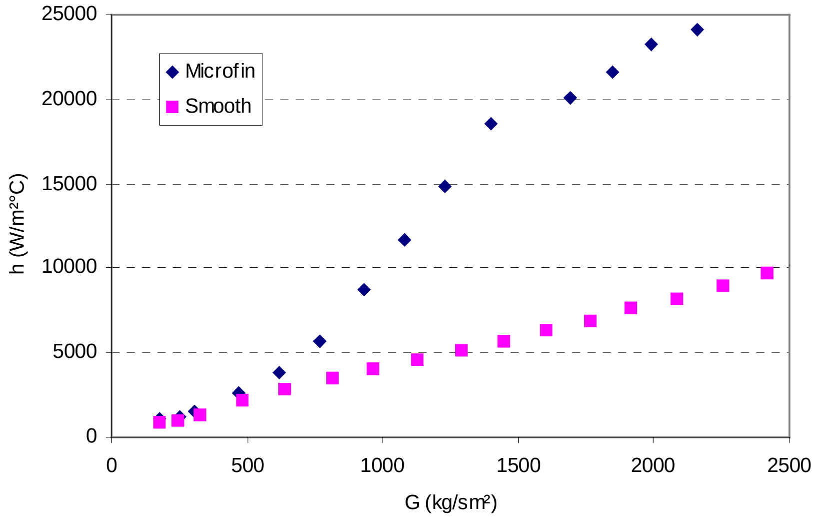

Figure 5. Heat transfer coefficients results of micro-finned and smooth tubes.

Analyzing the curves of Fig. (5), it is possible to note the increase in the heat transfer coefficient with mass flux, or the same as the flow rate, for both micro-finned and smooth tubes. However, the comparison between two tubes indicates the significant increment in the response of the micro-finned tube, mainly in the turbulent flow ($G>1000 kg/sm^2$). This increment can be quantified in terms of the enhance factor, F_h (Webb, 1994), which consists in a relation between the heat transfer coefficients, such as:

$$F_h = \frac{h_{micro\text{-}fin}}{h_{smooth}} \qquad (4)$$

The enhance factors for heat transfer in different flow rates calculated by Eq. (4) are indicated in Tab. (4). For the same flow rate the internal microfinned surface promotes a higher turbulency than the smooth surface, resulting in a greater heat transfer coefficient. The increase in the flow rate, from laminar to turbulent flows, affects the F_h factor, as can be seen in the Tab. (4).

Table 4. Enhance factors of the heat transfer and pressure drop.

Enhance factors	Flow rate		
	Laminar	Transition	Turbulent
Heat transfer - F_h	1.2	1.7	2.9
Pressure drop - $F_{\Delta p}$	-	1.2	1.7
Global – E	-	1.3	1.8

As was done with heat transfer, in Fig. (6) the tubes hydraulic behaviour is represented by the variation of pressure drop vs. mass velocity for both tubes.

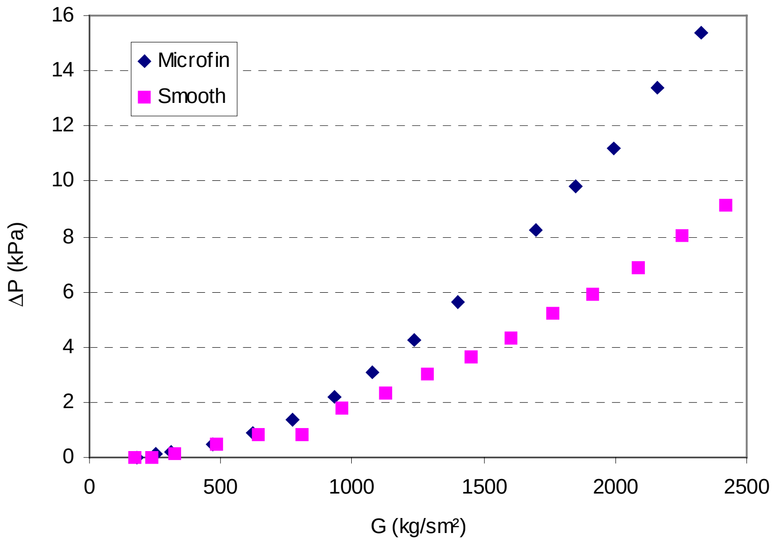

Figure 6. Experimental hydraulic behaviour of micro-fin and smooth tubes.

As could be expected, the pressure drop in the micro-fin tube is higher than in smooth tube, for $G>500kg/sm^2$ (transition flow). This is because the micro-fins promote more friction. The increasing in pressure drop for this tube is quantified by the enhance friction factor, $F_{\Delta p}$, as given by Eq. (5).

$$F_{\Delta p} = \frac{\Delta p_{\text{micro-fin}}}{\Delta p_{\text{smooth}}} \tag{5}$$

The values for $F_{\Delta p}$, shown in Tab. (4), indicate an augmentation about 70% for micro-fin tube in turbulent flow. In the laminar flow the results were not significant, since they are in the imprecision range of the measuring instrument.

The preceding results can be associated and the efficiency of micro-fin tube can be evaluated relating to smooth tube. The global enhance factor, E, according to Webb (1994), consists in the ratio between heat transfer enhance factor and friction factor, just as presented in Eq. (6):

$$E = \frac{h_{\text{micro-fin}} / \Delta p_{\text{micro-fin}}}{h_{\text{smooth}} / \Delta p_{\text{smooth}}} \tag{6}$$

The global enhance factors found for different flow rates are shown in Tab. (4). In turbulent flow an increasing of about 80% in heat transfer was achieved over the pressure drop.

5. Correlations

The experimental apparatus was qualified by running heat transfer and pressure drop experiments with smooth tubes. Performance of smooth tubes has been extensively reported and actual correlations predict very accurately heat transfer coefficient and friction factor. Figure 7 compares the smooth tube test results of the heat transfer and friction factor data. As seen in the figure, there is a good agreement between the present data and accepted correlations. The experimental results of heat transfer coefficient agree well with Dittus-Boelter's (1930) correlation, applied to Reynolds number > 10,000, and the Gnielinsk (1976) correlation, which is applied a larger range of Reynolds number, from 3000 to 5×10^6. The comparison between the experimental friction factors and the equations proposed by Blasius, for Re>3000, and Petukov (1970), for $3000<Re<5 \times 10^6$, is also shown in the figure.

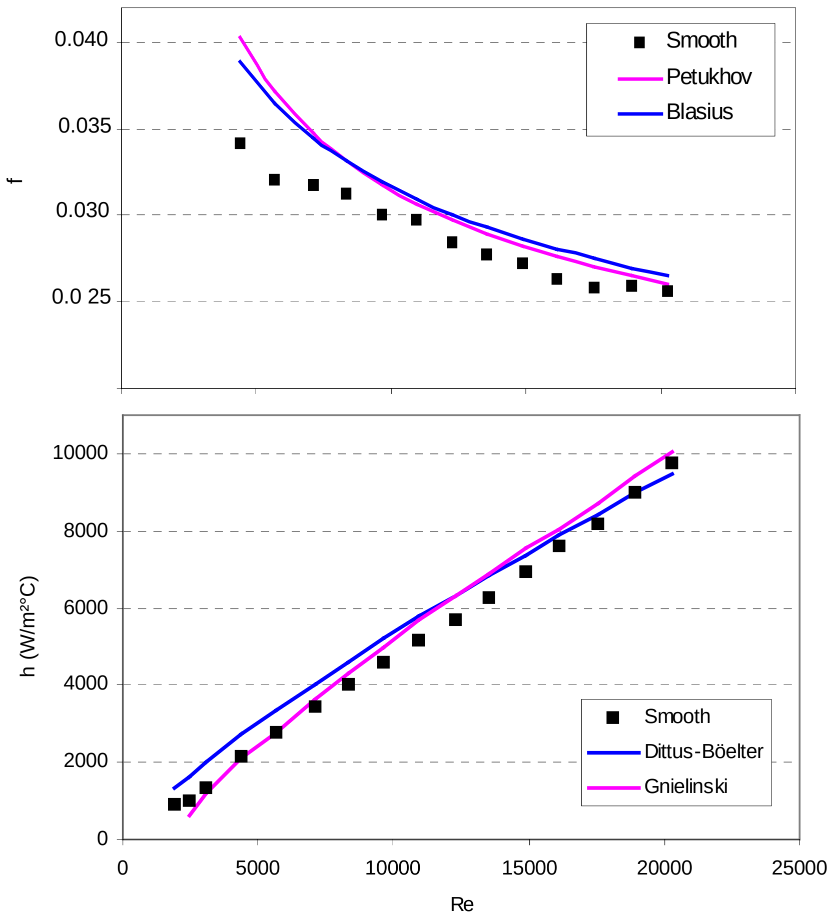

Figure 7. Friction factor and heat transfer coefficients measured and predicted for smooth tubes.

Concerning to micro-fin tube, only one type of microfin configuration was measured. First, a correlation for the heat transfer coefficient was adjusted in the form of Eq. (7), which does not include the variation of characteristics parameters of micro-fin tubes like e/D, n_f and α.

$$Nu = \frac{hD}{k} = a\,Re^b\,Pr^c \tag{7}$$

where a, b and c are constants to be determined from the experimental data. The constants a and b were adjusted using the Marquardt non-linear regression method. The parameter c was maintained constant and equal to 0.4, as in the Dittus equation, because the Prandtl number did not vary significatively. Equation (8) presents the adjusted parameters.

$$Nu = \frac{hD}{k} = 0.0034\,Re^{1.1}\,Pr^{0.4} \tag{8}$$

The previous equation was modified introducing the effect of the surface and fluid temperature difference on the fluid viscosity, as stated in the Sider-Tate equation, in order to get an better agreement to data, as shown in Eq. (9).

$$Nu = \frac{hD}{k} = 0.0013\,Re^{1.2}\,Pr^{1/3}\left(\frac{\mu}{\mu_s}\right)^{0.14} \tag{9}$$

where μ and μ_s are the fluid viscosities at mean fluid temperature and surface temperature, respectively.

Figure 8 shows the comparison between experimental and calculated heat transfer coefficient with Eq. (8) and (9). In the same figure the heat transfer coefficients predicted by the Wang and Chiou equation (1996), for the same tube and fluid, are presented.

The relative mean bias error, MBE, and root mean square error, RMSE, are used to indicate how closely the computed heat transfer coefficient agrees with measured data. The MBE represents the mean deviation between computed and measured data. The RMSE indicates the scatter between computed and measured data.

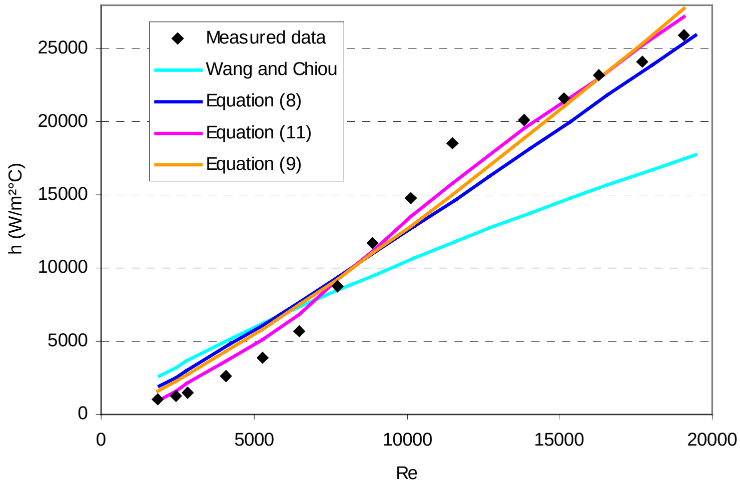

Figure 8. Heat transfer coefficient versus Reynolds number: experimental and predicted data.

Table 5. Statistical evaluation of predicted equations.

Predicted	MBE (%)	RMSE (%)
Equation 8	52.92	8.86
Equation 9	12.46	6.50
Equation 10	8.26	2.01

According to Tab. (5), the Eq. (9) presents lower values for MBE and RMSE when compared with Eq.(8). It can be observed in Fig. (8) that the predicted heat transfer coefficients by Eq. (9) are closer to measured data than the ones predicted by Eq. (8), mainly for Reynolds numbers greater than 10,000. There is a significative discrepancy between the experimental data and the Wang and Chiou correlation.

Although these results, Eq. (8) and (9) do not consider the geometry effects to flow and heat transfer. For rough surfaces Dipprey and Sabersky (1963) developed an analogy model parallels that of the smooth model (Petukov, 1970), which is based in heat transfer and momentum analogy, given by Eq. (10).

$$Nu = \frac{hD}{k} = \frac{\frac{f}{8} Re\,Pr}{1 + \left(\frac{f}{8}\right)^{1/2}\left[g(e^+)\,Pr^n - B(e^+)\right]} \tag{10}$$

where the Nusselt number, Nu, is a function of Reynolds and Prandtl numbers and the friction factor, which in turbulent flow depends on the enhancement dimensions (e/D, n_f e α) of the microfin tube. The correlating functions for rough tubes, $g(e^+)$ and $B(e^+)$, where e^+ is the roughness Reynolds number, will be different for different roughness types. In this experimental study a database with different microfin tubes is not available, thus it is not possible to adjust the equations for g (e^+) and B (e^+) functions. However, constant values for these functions were found allowing to represent the heat transfer coefficient for this tube geometry. Moreover, the Eq. (10) was modified to represent all the Reynolds range, resulting in Eq. (11).

$$Nu = \frac{hD}{k} = \frac{\frac{f}{8}(Re - 1000)\,Pr}{1 + \left(\frac{f}{8}\right)^{1/2}\left[8.05\,Pr^{-0.38} + 9.09\right]} \tag{11}$$

where the friction factor is given by f=0.014 $Re^{0.12}$ adjusted from experimental data (Eq. (3)).

According to Fig. (8) and Tab. (5), Eq. (11) showed the better results in the whole test range.

6. Conclusions

Water single-phase heat transfer and friction for micro-finned and smooth tubes at different flow rates were measured. The micro-finned tube tested allowed to increase significantly the heat transfer comparing to the smooth tube. Increments up to until 190% in the heat transfer coefficient at turbulent flow were observed. In laminar flow this increment was about only 20%, which does not justify its application, considering its higher costs comparing to smooth tubes.

The increase on pressure drop was also verified, but the enhance global factor showed that heat transfer augmentation is always superior. In a turbulent flow, this relation is of about 80%.

Although the microfinned tube was primarily designed for in-tube evaporation, this study clearly indicated that it could be used in a single-phase application promising results.

Empirical correlations for heat transfer were formulated for microfinned tube and compared with experimental data achieving a good agreement.

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