Experimental Characterization of Heat Transfer in Internal Flow of Corrugated Tubes

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November 2018

Abstract:

The main purpose of this work is the characterization of pressure drop and heat transfer in internal flows in smooth and corrugated tubes ($p = 6\ mm$ e $p = 12\ mm$). The tubes have internal diameters of 5.75 mm and a heat length of 0.38 mm. The working fluid used was water. An experimental setup was developed to reproduce and validate experimental measurements of pressure drop and heat transfer in laminar, transitional and turbulent regime. The Reynolds number varied between 429 and 6212. The heat flux imposed on the wall of the smooth and corrugated tubes ranged from 5.5 kW/m^2 to 21.1 kW/m^2. The friction factor was obtained for hydrodynamically fully developed flow. In laminar regime. The Nusselt number was obtained for thermally developing flow and in turbulent regime for thermally fully developed flow. The experimental results obtained for the friction factor and for the Nusselt number were validated for both regimes, laminar and turbulent, through correlations available in the literature. It was observed that the friction factor obtained for the corrugated tubes was higher than that obtained for the smooth tube, with the friction factor being higher for the $p = 6\ mm$ that for the $p = 12\ mm$. Although the Nusselt number is higher for the $p = 6\ mm$ tube in turbulent and transitional regime, in laminar regime the tube of $p = 12\ mm$ presented a higher Nusselt number. To overcome some of the difficulties encountered, the use of thermography as an alternative diagnostic technique is demonstrated here.

Keywords: Internal Flow, Transitional Regime, Corrugated Tubes, Friction Factor, Heat Transfer.

1. Introduction

Heat exchangers are widely used in engineering applications to exchange heat between two fluids that are at different temperatures and delimited by a solid wall.

According to Casella (2017) the use of water as working fluid has many advantages in these types of systems, since it does not involve high costs, it's an abundant resource that does not harm the environment, has low viscosity and high specific heat and finally, it has high thermophysical properties that benefit heat transfer.

The main purpose of a heat exchanger is to promote the largest heat transfer in minimum pressure drop conditions possible, since a larger pressure drop implies more work provided by the pump and thus higher operating costs.

In a heat exchanger performing in a Waste Heat Recovery System (WHRS), the preheating zone operates only in single-phase flow and corresponds to 70% of the total volume of the heat exchanger. Depending on the operation conditions, the flow inside the heat exchanger channels can be in laminar, transition or turbulent regime. The laminar and turbulent regime have been studied since 1883. However, it was not until the 1990s that studies on the transition regime began (Everts, 2014). Since then, authors such as Meyer and Ghajar have progressed on this subject, but more work must be done to fully understand transitional regime.

Ghajar *et al*. have studied mostly fully developed flow under constant imposed heat flux conditions using mixtures of ethylene glycol with a high Prandtl number. These studies made it possible to verify that the input geometry influences the value of the heat transfer coefficient and, the smoother the input and the higher the heat flux, the greater the Reynolds number for which the transition would occur. Thus, by changing the input geometry and the imposed heat flux, it is possible to change the Reynolds number for which the transition occurs (Everts, 2014)

Despite the scarce work still found in the literature, characterizing the flow inside tubes within the transition regime, many heat exchangers operate in this regime (Meyer, 2014). According to this author, this is an important regime to be explored with the use of corrugated tubes, since it can improve heat transfer significantly. Also, this regime allows a good compromise between low pressure drop and high heat transfer. In laminar regime, where pressure drop is relatively low, the heat transfer coefficients are also low (Meyer, 2014; Everts & Meyer, 2015). In turbulent regime heat transfer coefficients are higher, but the pressure drops have a higher order of magnitude than in laminar flow regime. (Meyer, 2014; Everts & Meyer, 2015). In laminar regime, the heat transfer coefficient is strongly influenced by secondary flow effects (mechanism that causes the fluid to move through differences in densities caused by temperature differences in the fluid itself), which are more significant with higher heat fluxes, as reported by Everts (2014). On the other hand, in a turbulent regime, the heat transfer coefficient depends neither on the entry region nor on the imposed heat flux (Everts, 2014).

The use of corrugated tubes in Shell-and-tube type heat exchangers has allowed the size and efficiency of the heat exchanger to be reduced (Vicente *et al*., 2004a; Vicente *et al*., 2004b).

As the corrugation of the tubes provides an increase in turbulence and consequently a better mixing of the flow, it is expected an increase in the heat transfer coefficient, h. (Vicente *et al*., 2004a; Vicente *et al*., 2004b). Corrugated tubes have been used in many industrial applications, such as evaporators, condensers and radiators (Vicente *et al*., 2004a), since the increase in the pressure drop induced by the corrugation is compensated by the increase of the heat transfer coefficient, and the production of a corrugated tube in addition to being inexpensive and simple, does not require more material than that used in the manufacture of smooth tubes (Vicente *et al*., 2004b).

There are two main types of corrugated tubes: internally concave; and externally convex corrugated tubes (Dizaji *et al.*, 2015). The present work is dedicated to the study of concave corrugated tubes in the interior.

Rainieri and Pagliarini (2002) used ethylene glycol to investigate the heat transfer process in the developing region of corrugated tubes with different pitches and to Reynolds numbers between 90 and 800. These authors concluded that for $Re > 200$ the corrugation induces significant swirl in the flow. On the other hand, Vicente *et al*. (2004a) studied the effect of pressure drop and heat transfer on corrugated tubes and found a 30% increase in the heat transfer coefficient and a 25% increase in the adiabatic friction factor, using corrugated tubes.

1.1 – Heat transfer and pressure drop in smooth tubes

The hydrodynamic entry region corresponds to the region that precedes the region of fully developed flow, which is fundamental for studies inside circular tube flows; hence the importance of knowing its length. Thus, for the calculation of the length of the entry region, it is necessary to consider several issues regarding the regime in which the flow is. According to Langhaar (1942, cit. by Incropera *et al.*, 2008), for laminar regime, the length of the entry region can be calculated through the expression:

$$x_{cd,h} \approx 0.05 Re D_h \tag{1}$$

where D_h is the hydraulic diameter of the tube. According to Kays and Crawford (1993, cit. by Incropera *et al.*, 2008), for turbulent flow this length is given by:

$$10D_h \lesssim x_{cd,h} \lesssim 60D_h \tag{2}$$

The friction factor was obtained by eq. (3):

$$f = \frac{\left(\frac{\Delta p}{\Delta x}\right) D_h}{\frac{\rho V^2}{2}} \tag{3}$$

where $\frac{\Delta p}{\Delta x}$ is the variation of the pressure drop along the channel, ρ is the specific mass and V is the mean velocity of the fluid. According to Incropera *et al.* (2008), in fully developed flow, the friction factor in laminar regime can be given by:

$$f = \frac{64}{Re} \tag{4}$$

For turbulent regime in fully developed flow in smooth tubes Incropera *et al.* (2008) proposes Blasius equation eq. (5) for $Re_D \lesssim 2 \times 10^4$ and Petukhov equation eq. (6) for $3 \times 10^3 \lesssim Re_D \lesssim 5 \times 10^6$ as follows:

$$f = 0.316 Re^{-\frac{1}{4}} \tag{5}$$

$$f = (0.79 \ln Re - 1.64)^{-2} \tag{6}$$

Hrycak and Andruskhiw (1974, cit. by Tam *et al*., 2013) developed a correlation to describe the behavior of a fully developed flow in the transition regime:

$$f = 4(-3.1 \times 10^{-3} + 7.125 \times 10^{-6} Re - 9.7 \times 10^{-10} Re^2) \tag{7}$$

This equation is valid for $2100 < Re < 4500$.

Similarly to the hydrodynamic analysis, the region that precedes the fully developed thermal region is called the thermal entry region and for laminar flow it can be calculated as:

$$x_{cd,t} \approx 0.05 Re Pr D_h \tag{8}$$

In a turbulent regime, the length of the thermal entry is given by:

$$x_{cd,t} = 10D_h \quad (9)$$

A very important dimensionless parameter in the heat transfer analysis is the Nusselt number, Nu, given by:

$$Nu = \frac{hD_h}{k} \quad (10)$$

where k is the thermal conductivity of the fluid, and h the heat transfer coefficient.

According to Incropera *et al*. (2008), Nusselt number takes an approximately constant value for fully developed laminar flow:

$$Nu \approx 4.36 \quad (11)$$

In a turbulent regime the correlation of Gnielinski (1976, cit. by Incropera *et al.*, 2008):

$$Nu = \frac{\left(\frac{f}{8}\right)(Re - 1000)Pr}{1 + 12.7\left(\frac{f}{8}\right)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}} - 1\right)} \quad (12)$$

has an error of approximately 10%, being valid for $3000 \lesssim Re \lesssim 5 \times 10^6$ and $0.5 \lesssim Pr \lesssim 2000$. The friction factor, f, can be calculated from (6), and the fluid properties should be calculated considering the mean temperature of the fluid.

Another correlation that Gnielinski (1975, cit. by Zhang *et al.*, 2013) proposes for turbulent regime is represented in (13) and is valid for $3000 \leq Re \leq 10^6$ e $1.5 \leq Pr \leq 500$.

$$Nu = 0.012(Re^{0.87} - 280)Pr^{0.4} \quad (13)$$

Li and Xuan (2002) also presented a correlation for turbulent regime, which for the case where the working fluid is water is reduced to:

$$Nu = 0.0059Re^{0.9238}Pr^{0.4} \quad (14)$$

This correlation is valid for $2500 \lesssim Re \lesssim 25000$ and presents a precision error of approximatley 8%.

Ghajar and Tam (1994) proposed a correlation that can be applied both in the entry region and in the fully developed region of laminar flow. The correlation is given by:

$$Nu = 1.24\left(\frac{RePrD_h}{x} + 0.024(GrPr)^{0.75}\right)\left(\frac{\mu_b}{\mu_s}\right) \quad (15)$$

where Gr represents the dimensionless number of Grashof, μ_b the dynamic viscosity at the bulk temperature of the fluid, and μ_s the dynamic viscosity at the mean temperature of the tube surface. This equation can only be applied if : $3 < \frac{x}{D_h} < 192$; $280 < Re < 3800$; $40 < Pr < 160$; $1000 < Gr < 2.8 \times 10^4$ e $1.2 < \frac{\mu_b}{\mu_s} < 3.8$.

Another correlation that can be applied in laminar flow under thermal development conditions is the correlation of Petukhov (1967, Minakov *et al.*, 2015) given by:

$$Nu = 1.55\left(\frac{RePrD_h}{L}\right)^{0.33}\left(\frac{\mu_b}{\mu_s}\right)^{-0.14} \quad (16)$$

Shah and London (1978, cit. by Utumo *et al.*, 2014) suggest a correlation to calculate local Nusselt number in laminar flow for constant heat flux.

$$Nu_x = \begin{cases} 1.302x^{*-\frac{1}{4}} - 1 & x^* \leq 0.00005 \\ 1.302x^{*-\frac{1}{4}} - 0.5 & 0.00005 \leq x^* \leq 0.0015 \\ 4.364 + 8.68(1000x^*)^{-0.506} & x^* > 0.0015 \end{cases} \quad (17)$$

where x^* is given by:

$$x^* = \frac{x/D_h}{RePr} \quad (18)$$

This equation is valid only for $500 \lesssim Re \lesssim 2000$ (Shah & London, 1978, cit. por Utumo *et al.*, 2014).

1.2 – Heat transfer and pressure drop for corrugated tubes

In the last 30 years, corrugated tubes have been the main target of some studies to understand how friction factor and heat transfer vary in relation to smooth tubes. (Vicente *et al.*, 2004a).

In turbulent regime, since the corrugation of tubes provides an increase in turbulence and consequently a better mixing of the flow, the heat transfer coefficient, h, is expected to increase (Vicente *et al*., 2004a, 2004b).

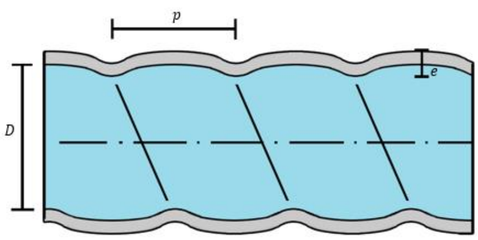

Fig. 1 - Longitudinal section of a corrugated tube (adapted from Vicente el al., 2004a).

Fig. 1 depicts the longitudinal section of a corrugated tube, where p symbolizes the helical pitch, e represents the height ridge and D the maximum tube diameter. The index parameter that characterizes the roughness of a corrugated tubes is called severity index and is given by:

$$\phi = \frac{e^2}{pd} \quad (19)$$

Vicente *et. al.* (2004a), tested water and ethylene glycol on corrugated tubes in laminar regime and obtained a correlation for the fricton factor for $200 < Re < 2000$:

$$f = 119.6\phi^{0.11}Re^{-0.97} \quad (20)$$

The authors in the development of this correlation used long corrugated tubes, with only one section of the tube being heated. All tubes have a maximum diameter of 18 mm na the parameter $\frac{p}{d}$ ranged from 0.886 e 1.158.

Vicente *et al.* (2004b) showed that in corrugated tubes the transition from laminar to turbulent flow occurs for lower Reynolds numbers than that those obtained for the smooth tube. Vicente *et al.*, (2004b) also verified that the transition Reynolds number is independent of the helical pitch, $\frac{p}{d}$, depending only on height ridge, $\frac{e}{d}$. Thus, they developed an expression for Re_{crit} with 15% accuracy, given by:

$$Re_{crit} = 2100\left(1 + 1.18 \times 10^{7}\left(\frac{e}{d}\right)^{3.8}\right)^{-0.1} \quad (21)$$

The study of Vicente *et al.* (2004b) shows that transition in smooth tubes occurs more abruptly than in corrugated tubes. Also, they reported that the pressure fluctuations occuring in internal flow in smooth tubes are damped using corrugated tubes because they promote a better swirl of the flow.

Vicente *et al*. (2004b) have developed a correlation for adiabatic friction factor, which gives good results for tubes with low severity index ($\phi < 10^{-3}$) and for $2000 \lesssim Re \lesssim 8000$ conditions, given by:

$$f = 6.12\phi^{0.46}Re^{-0.16} \quad (22)$$

On the other hand, if $\phi > 10^{-3}$, the authors recommend that the friction factor, f, take a constant value evaluated at $Re = 8000$.

Vicente *et al*. (2004b) studied heat transfer in turbulent regime for water and for ethylene glycol and proposed the following correlation:

$$Nu = 0.0344(Re - 1500)^{0.78}Pr^{0.37} \quad (23)$$

The authors concluded that the higher the number of Prandtl, the higher the heat transfer performance of corrugated tubes. For $Re < 10000$ the tubes with $\phi > 3 \times 10^{-3}$ presented better performance, while at $10000 < Re < 40000$, the corrugated tubes with intermediate corrugation ($1 \times 10^{-3} < \phi < 2 \times 10^{-3}$) presented better results. The authors also concluded that tubes with $\phi < 10^{-3}$ do not present advantages for practical applications.

Thus, Vicente *et al*. (2004b) refer that the selection of the corrugated tubes must be made according to the intended application, since corrugated tubes with higher ϕ have a better performance in lower Reynolds numbers, while for higher Reynolds numbers corrugated tubes with $1.5 \times 10^{-3} < \phi < 2 \times 10^{-3}$ present better results.

Rainieri *et al.* (1996) studied the effect of heat transfer and friction factor on internal flow in five different configurations of tubes with internal helical corrugation in laminar regime with constant heat flux. The tube diameters varied between 16 mm and 18 mm, the helical pitch between 10 mm and 18 mm and the severity índex between 0.006 and 0.014. The authors verified that the transition can occur in Reynolds numbers below 2000 and because of this the heat transfer increases considerably.

A study by Rainieri and Pagliarini (2002) with 14 mm diameter tubes, helical pitch from 16 mm to 64 mm and height of ridge of 1.5 mm showed that, for not very high Reynolds numbers ($Re \geq 200$), helical corrugated tubes induce significant *swirl*, though there is no significant increase in heat transfer. The authors report that regarding heat transfer in turbulent regime, the effect of the helical pitch is approximately negligible. Also, the placement of a development section before the test section had no significant effects on heat transfer, even at reduced Reynolds numbers. They also observed that the larger the helical pitch the greater the Reynolds number where transition occurs.

Barba *et al*. (2002) studied the effect of heat transfer and friction factor in corrugated tubes with a diameter of 14.5 mm, a helical pitch of 11.5 mm and a severity índex of 0.135, using ethylene glycol as working fluid for $100 < Re < 800$. The authors proposed a correlation for the friction factor obtained as follows:

$$f = 61.639Re^{-0.8602} \quad (24)$$

According to Incropera *et al*. (2006) the increase of the roughness of the tube's surface causes an increase of the friction factor. The authors mention that if friction factor is about four times higher than the value corresponding to the smooth tube, the heat transfer does not change significantly.

Pethkool *et al*. (2011) studied corrugated tubes with helical pitch of 4.5, 5.5 and 6.5 mm, with an internal diameter of 26 mm and, contrary to what was verified by Vicente *et al*. (2004a, 2004b), the greater the helical pitch and the height ridge, the greater the Nusselt number.

Therefore, there is still not much of a consensus in corrugated tubes studies, since many authors present results for the friction factor that only depend on the Reynolds number and other authors affirm that there is also a dependence on the severety index. In addition, the study by Vicente *et al*. (2004b) shows the existence of corrugated tubes optimized for a given range of Reynolds numbers that, when out of the optimal range, worsen their performance in relation to other tubes. Thus, it is important to carry out more studies in this subject, especially in the laminar and transitional regime, since the available literature is reduced.

2. Experimental setup

The experimental setup and all its components are shown in Fig. 2. The symbols T and dP refer to temperature sensors and pressure sensors, respectively.

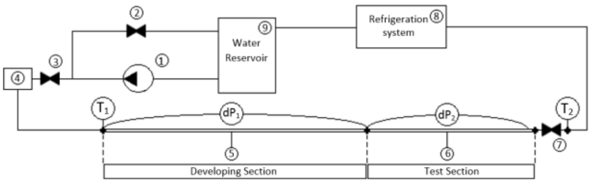

Fig. 2 - Schematic representation of the experimental setup: (1) Hydraulic pump (2) Valve (3) Valve (4) Flowmeter (5) Developing section (6) Test section (7) Fine adjustment ball valve (8) Mixing, heating and cooling system (9) Water reservoire.

In this experiment, a magnetic rotary blade pump (1) is used, through which the pressure of the working fluid is raised, thus forcing it to go through the experimental installation. The pump is connected to a frequency converter that allows the mass flow rate to be regulated at the installation. The valve (2) helps to regulate the flow rate aswell. Thus, when oppened, this valve forces a fraction of the flow to recirculate in the direction of the water reservoir (9) reducing the fraction of flow that goes through the remaining part of the experimental setup. Thus, as the valve (2) is closed, the recirculated flow fraction decreases towards the water reservoir and increases in the remain part of the experimental setup. Downstream of the pump, the fluid flows through a Coriolis flow sensor (4), in which the mass flow rate, density and temperature of the fluid is measured. Thereafter, the fluid flows through the developing section (5) consisting of a circular AISI 304 stainless steel tube with a length of 820 mm, an internal diameter of 5.75 mm and an external diameter of 6.5 mm. The length of the tube has been defined to ensure that the flow in the interior of the tube is hydrodinamicaly developed prior to entering the test section (6). The test section (6), shown in Fig. 3, is welded to the developing section (5) and may be composed by one of the AISI 204 stainless steel tubes shown in Table 1.

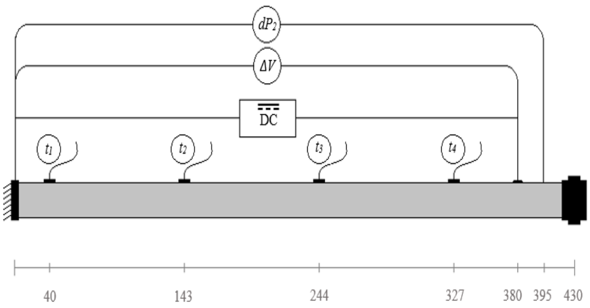

Fig. 3 - Schematic representation of the Test Section, where t_1, t_2, t_3 e t_4 represents the temperature sensor, DC represents the power supply, ΔV is the multimeter and dP_2 the differential pressure sensor.

Table 1
Test section tube characteristics, where D is the maximum internal diameter, p the helical pitch and e the ridge height.

Tube	D [mm]	p [mm]	e [mm]
Smooth	5.75	-	-
Corrugated	5.75	12	0.6
Corrugated	5.75	6	0.6

The test section is heated and connected directly to the DC power supply, to ensure a uniform heat flux on the tube surface (380 mm in length).

Throughout the surface of the test tube, four thermocouples of type K from Omega were mounted, thus allowing to determine the temperature of the fluid in four different points. In addiction, two thermocouples of type K were installed: (T_1) to the inlet of the developing section (5); and (T_2) to the outlet of the test section (6).

The fluid flows into a mixing, heating and cooling system (8), and then goes to the water reservoir (9) before returning to the hydraulic pump, and so completing the closed circuit.

3. Results Analysis

3.1 – Hydrodynamic and thermal conditions of the flow

Regardless of the Reynolds number, the test section always showed a hydrodynamicly fully developed flow. Since the developing section is not heated, all the tests in laminar flow were performed in thermal developing flow. In turbulent flow the thermal entry length was 86% fully developed.

The applied heat flux ranged between 5.5 kW/m^2 and 21.1 kW/m^2.

In this study, the secondary flow was so low that it could be neglected.

3.2 – Validation

The test section was validated for adiabatic friction factor and for laminar and turbulent Nusselt number, according to the corrrelations presented in the literature review. Fig. 4 shows the validation results.

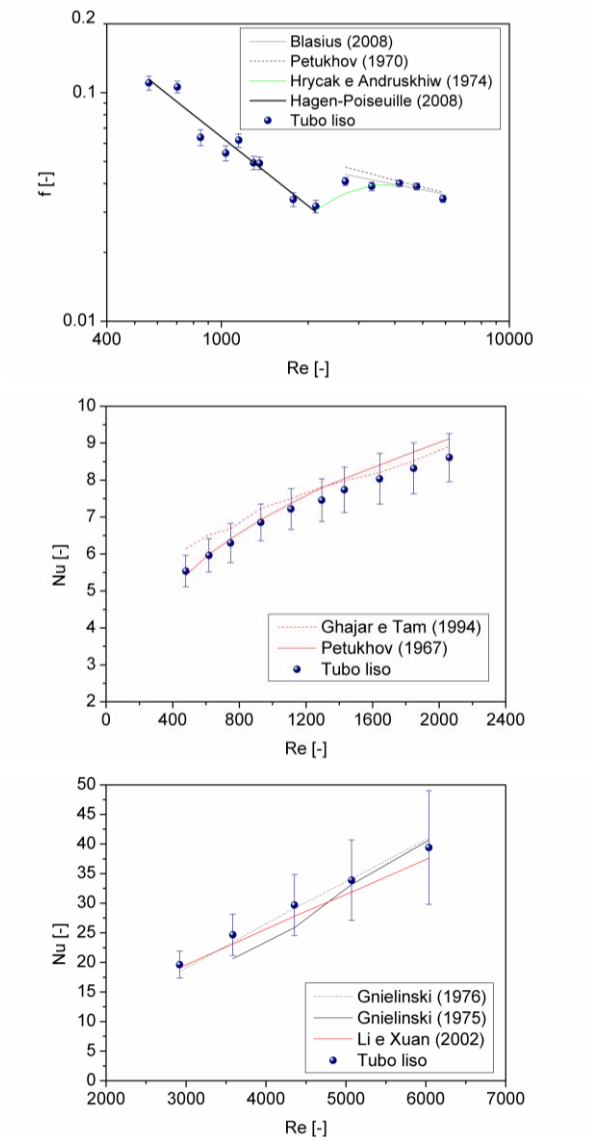

Fig. 4 – Validation of the test section based on the adiabatic friction factor and on mean Nusselt number correlations, as a function of the Reynolds number.

As shown in by Fig. 4, the test section was validated as it presents reliable results according to the literature regarding to hydrodynamic and thermal behavior of the flow.

3.3 – *Adiabatic friction factor*

To compare adiabatic friction factor results between the smooth tube and the two corrugated tubes some experiments were performed at ambient temperature and without heat flux applied to the surface of the tubes.

Fig. 5 shows the comparison of the adiabatic friction factor between the three tested tubes.

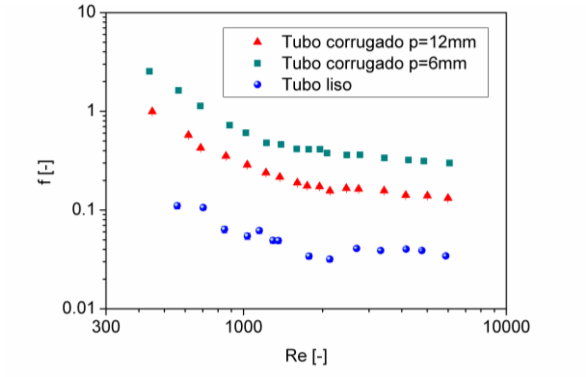

Fig. 5 – Adiabatic friction factor for the smooth tube and corrugated tubes, as a function of Reynolds number.

As expected, the smooth tube is the one with the lowest friction factor. It is verified that, in corrugated tubes, the obtained friction factor presents an ascending vertical displacement regarding the friction factor obtained in the smooth tube. The $p = 6\,mm$ corrugated tube is the one with the highest friction factor value.

The results show that the transition from laminar to turbulent regime occurs smoother for corrugated tubes than for the smooth tube. This phenomenon was also observed by Vicente *et al.* (2004a), who stated that this effect is due to the swirl effect produced in the flow by the helical shape of the corrugation. In addition, in the corrugated tubes, the transition begins at lower Reynolds numbers, as in Meyer and Olivier (2011). According to these authors this phenomenon occurs due to the resistance to the flow caused by the corrugation inherent in the tubes. As the increase of the friction factor in the transitional regime is damped for corrugated tubes, it becomes difficult to identify where it occurs exactly. However, Fig. 5 suggests that the transition in the $p = 6\,mm$ corrugated tube occurs at lower Reynold numbers than in $p = 12\,mm$ corrugated tube.

3.4 – *Diabatic friction factor*

The diabatic friction factor was obtained by applying a constant heat flux to the surface of the tubes. The results obtained were compared with the results of adiabatic friction factor.

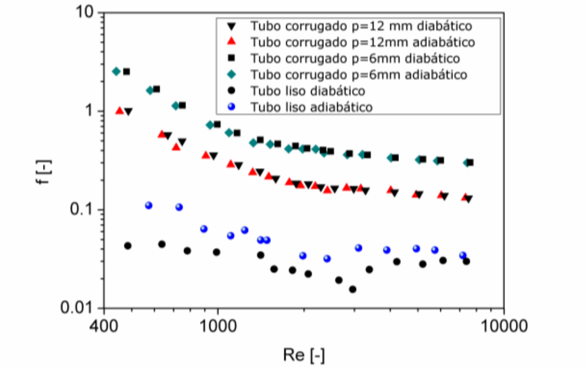

Fig. 6 – Diabatic friction factor fo the smooth tube and corrugated tubes as a function of the Reynolds number.

Fig. 6 shows that the adiabatic friction factor of the smooth tube is always higher than the diabatic friction factor, which agrees with the results obtained by Tam *et al.* (2013). This phenomenon can be explained by the decrease of the dynamic viscosity when the tube was heated. Thus, the velocity profile becomes more stable because because the shear stresses decrease, and so friction factor decreases aswell.

On the other hand, the transitional regime for the smooth tube's diabatic friction factor occurs for higher Reynolds numbers than those observed in the adiabatic case. Everts (2014) obtained similar results for transition and found that the diabatic and adiabatic friction factor did not present significant differences in turbulent regime.

Fig. 6 also shows an approach between the two friction factors in turbulent regime, which proves what Everts (2014) found.

On the other hand, when imposing a heat flux on the corrugated tubes surface, significant effects were not observed. Tam *et al.* (2013) found that applying a heat flux has a great influence on the friction factor in laminar and transitional regimes, and it has to do with secondary flow. Since in this study secondary flow was not verified, according to Tam *et al.* (2013), the results were expected.

3.5 – Heat transfer

To make an analysis based on heat transfer, a heat flux was applied on the surface wall of each tube and the mean Nusselt number was obtained from the difference of the inlet and outlet thermocouples.

Fig. 7 shows the mean Nusselt number corresponding to each teste tube.

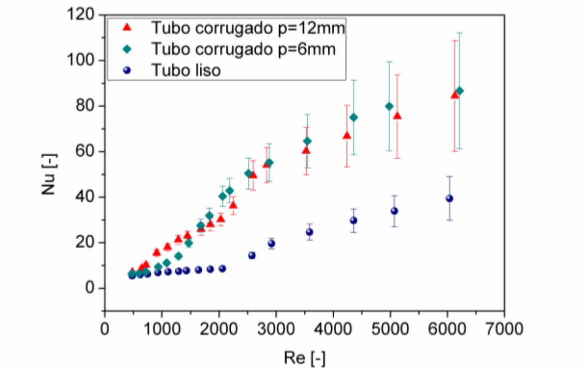

Fig. 7 – Nusselt number for the smooth tube and for the corrugated tubes as a function of the Reynolds number.

The Nusselt number obtained from the corrugated tubes is always higher than that obtained in the smooth tube.

The use of corrugated tubes, compared to smooth tubes, forces transition to occur at lower Reynolds numbers. Fig. 7 suggests that the transition corresponding to the $p = 6\,mm$ corrugated tube started at $Re \approx 1080$, while the transition corresponding to the $p = 12\,mm$ corrugated tube began at $Re \approx 1470$. Thus, the smaller the pitch, the lower the Reynolds number where transition occurs.

Fig. 7 shows that the Nusselt number associated with $p = 6\,mm$ corrugated tube is smaller than that of the $p = 12mm$ corrugated tube, within the range of $500 \lesssim Re \lesssim 1600$. It was only from $Re \gtrsim 1600$ that the heat transfer was higher in the $p = 6\,mm$ corrugated tube. Thus, although the dimensions and severity índex of the tubes used by Vicent *el al.* (2004b) are very different from those used in the present work, it was also verified that the $p = 12\,mm$ corrugated tube presented a higher performance than the $p = 6\,mm$ corrugated tube for a given range of Reynolds numbers, and outside of this range, the $p = 6\,mm$ corrugated tube presented better results.

Although the Nusselt number for $Re \gtrsim 1600$ is higher for $p = 6\,mm$ corrugated tube, the difference between both corrugated tubes is not very significant, which is in line with Incropera *et al.* (2008). The authors mention that since the friction factor is four times higher than the corresponding value for the smooth tube, the heat transfer does not vary significantly.

3.6 – Simultaneous analysis of pressure drop and heat transfer

A simultaneous analysis of j-Colburn factor and friction factor was performed, and it is shown by Fig. 8 that the tendency of the friction factor is quite similar the j-Colburn factor for the smooth tube.

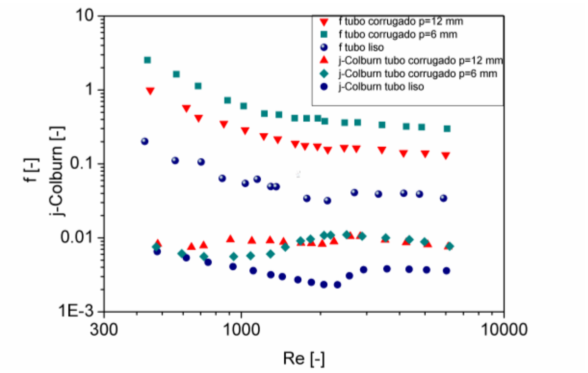

Fig. 8 – j-Colburn factor and friction factor obtained for all the test tubes as a function of the Reynolds number.

The $p = 12\,mm$ corrugated tube has a j-Colburn factor and a friction factor higher than the smooth tube. The Fig. 8 also shows that for $500 \lesssim Re \lesssim 900$ the friction factor is very high and heat transfer is very low, thus the $p = 12\,mm$ corrugated tube will have a bad performance operating in this regime. It is at the end of transition ($Re\ \approx 2600$) that these tubes can have the best performance, since heat transfer is at it's maximum and the friction factor didn't increase significantly.

The friction factor of the $p = 6\,mm$ corrugated tube has the highest value of all the tubes tested, and so in laminar regime it has the worst performance. On the other hand, when the flow reaches, approximately, the end of transition, heat transfer turns out to be higher than the one verified in the other tubes. These results show that the optimal performance of these tubes is obtained in the transitional regime, since heat transfer is high and friction factor is minimum. This confirms the studies of Meyer (2014) that states that it is in transitional flow that heat exchangers have better performance.

3.7 – Infrared thermal camera qualitative analysis for heat transfer in the test tubes

The Nusselt number obtained for the $p = 6\,mm$ corrugated tube at $500 \lesssim Re \lesssim 1600$ was lower than the obtained in $p = 12\,mm$ corrugated tube, and then in turbulent regime heat transfer turned out to be lower in the $p = 12\,mm$ corrugated tube. To understand this phenomenon a qualitative analysis was done using an Infrared thermal camera placed in a linear slide.

Four different sections (A, B, C and D), of each corrugated tube, were analysed and the results are showed in Fig. 9.

The Fig.9 shows how the temperature and the flow changed in the different sections of each tube for different Reynold number.

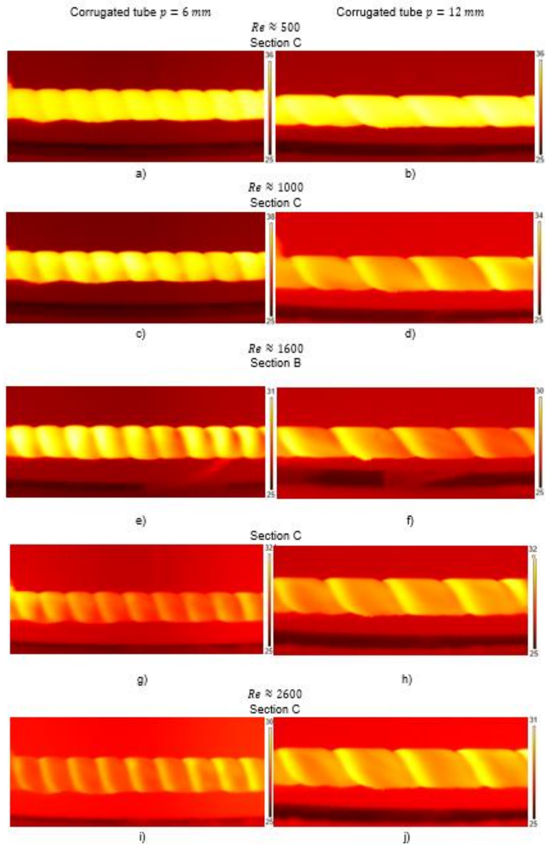

Fig. 9 – Flow representation in the two corrugated tubes with different Reynolds numbers

Through a) and b) it was found that in section C when $Re \approx 500$, the flow for both tubes were in laminar regime and no movement of the fluid inside the tubes was visible.

At $Re \approx 1000$, it was observed that the flow in the $p = 12\,mm$ corrugated tube showed a clear swirl effect while it was not noticeable with the $p = 6\,mm$ corrugated tube. Due to this, the temperature verified in d) tuned out to be lower than that verified in c). The fact that swirl occurs may explain the fact that the $p = 12\,mm$ corrugated tube has a higher Nusselt number in that Reynolds number, since the swirl effect promotes a better mixing of the flow, providing a better heat transfer. This may be due to the configuration of the $p = 6\,mm$ corrugated tube, which appears to force flow to separate at the ridge and to reattach again next to the following ridge. Since the fluid in this case only adheres to the surface of the tube just before the following ridge, the heat transfer between the fluid and the heated surface will be lower. This phenomenon was also observed by Kiml *et al.* (2004) when comparing the difference between tubes with transverse and helical corrugation. As discussed, the authors Kareem *et al.* (2015), also report that the corrugation induces swirl in the flow and this phenomenon is fundamental to improve the heat transfer.

For $Re \approx 1600$ it was found that in section B both flows show swirling effect, however only in f) the swirling effect is extended throughout the section. On the other hand, in section C, the $p = 6\,mm$ corrugated tube showed a much lower surface temperature than the surface of the $p = 12\,mm$ corrugated tube. This result implies that the local Nusselt number in g) was higher than the one showed in h) so it will certainly be a factor contributing to the increase in the mean Nusselt number of the tube.

In i) and j) the swirling effect can be seen from section A to section D, indicating that the flow is in turbulent regime.

This data showed that the $p = 6\,mm$ corrugated tube is not optimized to operate close to $Re \approx 1000$, since the $p = 12\,mm$ corrugated tube exhibits a greater heat transfer in that region. So, the choice of the corrugated tube to be implemented in a heat exchanger will depend on the Reynolds number to which it will operate.

The present work had as objective to characterize the hydrodynamic and termal behavior of the flow in smooth and corrugated tubes. The results obtained through the experimental installation were validated according to the correlations available in the literature, for laminar and turbulent regime. This validation was performed using the smooth tube as the test tube to gather information about the pressure drop and heat transfer along the tube.

Initially, a hydrodynamic and thermal input length analysis was performed for the smooth tube and the two corrugated tubes with 6 mm and 12 mm pitch.

In the analysis of the pressure drop for the smooth and corrugated tubes it was concluded that the smooth pipe presented a friction factor lower than that observed in the corrugated tubes and that the lower the pitch, the higher the friction factor will be. The curves of the friction factor of the corrugated tubes presented only a vertical displacement showing approximately parallel to the curve of the smooth tube's friction factor. It was found that the friction factor of the corrugated tubes presented a smoother transition than in the case of the smooth tube, probably due to the tangential velocity that is caused by the helical shape of the corrugation. It has also been shown that the smaller the pitch, the smaller the Reynolds number for which the transition regime starts.

For the smooth tube, the friction factor obtained without applied heat flux (adiabatic) was higher than that obtained with an applied heat flux (diabatic), with the transition regime for diabatic conditions starting at a Reynolds number higher than what was verified in the adiabatic case. In the smooth tube, the diabatic friction factor was lower, probably due to a decrease in the dynamic viscosity of the fluid, which led to a reduction of the shear stresses and uniformity of the velocity profile, thus leading to a lower diabatic friction factor. In the turbulent regime it was verified that the difference between the diabatic friction factor and the adiabatic friction factor decreases with the increase of Reynolds number.

For the corrugated tubes, the adiabatic and diabatic friction factor presented similar values, which can be explained by the fact that secondary flow effects were negligible in this study

The Nusselt number obtained for the corrugated tubes was always higher than that obtained for the smooth tube. The $p = 6\,mm$ corrugated tube showed, in the range of $500 \lesssim Re \lesssim 1600$, a Nusselt number lower than that observed with the $p = 12\,mm$ corrugated tube, however from $Re \gtrsim 1600$ the $p = 6\,mm$ corrugated tube showed a higher Nusselt number. Thus, the study suggests that a larger helical pitch ensures better performance in a certain range of Reynolds, while a smaller helical pitch guarantees better performance in another range.

4. Conclusions

In the presente work, some analyzes indicated that the optimum point of operation for a heat exchanger using corrugated tubes may well be in the transition regime, since it is there that was observed a higher difference of Nusselt number in relation to the smooth tube and is where the pressure drop is smaller.

In sum, the present work contributed to a better compression of the flow without and with heat transfer in laminar, transition and turbulent regime, for both smooth and corrugated tubes.

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