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Enhancement of laminar and transitional flow heat transfer in tubes by means of wire coil inserts

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Abstract

This work presents an extensive experimental study on three wire coils of different pitch inserted in a smooth tube in laminar and transition regimes. Isothermal pressure drop tests and heat transfer experiments under uniform heat flux conditions have been carried out.

The friction factor increases lie between 5% and 40% in the fully laminar region. The transition from laminar flow to turbulent flow is continuous, without the instabilities and the pressure drop fluctuations that a smooth tube presents.

Heat transfer experiments have been performed in the flow ranges: $Re = 10$–2500, $Pr = 200$–700 and $Ra = 3 \times 10^6$–10^8. At Reynolds numbers below 200, wire coils do not enhance heat transfer with respect to a smooth tube. For Reynolds numbers between 200 and 1000, wire coils remarkably increase heat transfer. At Reynolds numbers above $Re \approx 1000$–1300, transition from laminar to turbulent flow takes place. At Reynolds number around 1000, wire inserts increase the heat transfer coefficient up to eight times with respect to the smooth tube.

A performance comparison between wire coils and twisted tape inserts has shown that wire inserts perform better than twisted tapes in the low Reynolds number range: $Re = 700$–2500.

Keywords: Heat transfer enhancement; Wire coil inserts; Heat exchangers; Turbulence promoters

1. Introduction

Heat transfer processes of viscous fluids usually take place in laminar or transitional regimes, where transfer rates are particularly low. Heat exchangers that work under these flow conditions are usually candidates to undergo an enhancement technique. Among the different techniques which are effective to improve the thermohydraulic behaviour in the tube-side in single-phase laminar flow, the insert devices stand out. The dominant literature (Bergles [1], Webb and Kim [2]) usually mentions five types: wire coils, twisted tapes, extended surface devices, mesh inserts and displaced elements. The main advantage of these types in respect to other enhancement techniques such as the artificial roughness by mechanical deformation or internal fin types is that they allow an easy installation in an existing smooth-tube heat exchanger.

Due mainly to its low cost, the insert devices which are most frequently used in engineering applications are wire coils and twisted tapes. It can be stated that both wires coils and twisted tapes, become candidates to update an existing tube exchanger. The two recent state-of-the-art revisions on insert devices carried out by Wang and Sundén [3] and Dewan et al. [4] focus mainly on these two devices. The thermohydraulic behaviour of twisted tapes in laminar regime has been widely studied. Design correlations to predict both the isothermal friction factor (Manglik et al. [5]) and the Nusselt number under uniform heat flux (Bandyopadhyay et al. [6], Hong and Bergles [7]) and constant wall

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Nomenclature

c_p	fluid specific heat (J kg^{-1} °C^{-1})
d	envelope (maximum inner) diameter (m)
e	wire coil diameter (Fig. 2) (m)
h	heat transfer coefficient (W m^{-2} K^{-1})
H	pitch for 180° rotation of twisted tape (m)
k	fluid thermal conductivity (W m^{-1} K^{-1})
k_w	tube wall thermal conductivity (W m^{-1} K^{-1})
l_h	length of the heat transfer section (m)
l_p	length of test section between pressure taps (m)
$\dot{m}$	mass flow rate (kg s^{-1})
p	wire coil helical pitch (Fig. 2) (m)
ΔP	pressure drop across the test section (Pa)
Q	overall electrical power added to the heating section (W)
Q_l	heat losses in the test section (W)
q''	heat flux $(Q - Q_l)/(\pi d l_h)$ (W m^{-2})
s	tube wall thickness (m)
t	temperature (K)
x_p	axial position of the measuring point (m)

Dimensionless groups

f	fanning friction factor $(\Delta P d^5 \pi^2 \rho/(32 l_p \dot{m}^2))$
Gr	Grashof number (Ra/Pr)
Nu	Nusselt number (hd/k)
Pr	Prandtl number $(\mu c_p/k)$
Ra	Rayleigh number $(g\rho^2 c_p \beta d^4 q''/(\mu k^2))$
Re	Reynolds number $(4\dot{m}/(\pi d \mu))$
Δt^*	dimensionless temperature difference (Eq. (12))
x^*	reduced length $(x_p/(d\, RePr))$
y	twisted tape twist ratio (H/d)

Greek symbols

β	fluid thermal expansion coefficient (K^{-1})
δ	thickness of twisted tape (m)
μ	fluid dynamic viscosity (Pa s)
ρ	fluid density (kg m^{-3})

Subscripts

b	based on bulk temperature
fc	forced convection
in	tube inlet
min	minimum value
max	maximum value
out	tube outlet
s	smooth tube
wi	based on the inside tube wall temperature
wo	based on the outside tube wall temperature
x	local value
∞	fully developed

Superscript

–	averaged value

temperature conditions (Manglik and Bergles [8]) are well known. The existence of these design correlations does not mean however that the twisted tape insert is the best insert device, such as Webb and Kim [2] points out. Even though twisted tapes can increase under certain conditions heat transfer at low Reynolds numbers, this effect entails usually a considerable pressure drop increase.

Wire coil inserts are devices whose reliability and durability are widely contrasted. In extreme applications such as the tube-side of fuel pyrotubular boilers with great fouling problems and with high variations in temperature that produce great dilatations, wires are used without any problem. This is a cheap enhancement technique and it is completely viable for many industrial applications. There is, however, an almost absence of laminar flow studies. This fact hinders a widespread use of wire coil inserts in industrial heat exchangers.

The difficulties for carrying out experimental studies on heat transfer in laminar flow is well known (Bergles [9]), as this flow is sensitive to the entry length effects, to the thermal boundary condition and to the natural convection. In fact the experimental works on enhancement techniques in laminar regime are scarce in comparison with those carried out in turbulent regime. Only three experimental studies on wire coil inserts are recognized: the one by Oliver and Shoji [10] on non-newtonian flows, and those by Uttarwar and Rao [11] and Inaba et al. [12] on newtonian flows. Inaba et al. [12] tested Reynolds numbers above 200 and they only studied heat transfer in forced convection. Nusselt number correlations proposed by Uttarwar and Rao [11] are those that Webb and Kim [2] recommends, though he points out that their experimental results are affected by the entry region.

Garcia et al. [13] carried out recently an experimental research on six wire coils. Laboratory work focused on the description of wire effects in turbulent regime. Experimental correlations were provided for turbulent flow at Reynolds numbers above 2000 (friction factor) and 1700 (heat transfer coefficient). In addition, results at lower Reynolds numbers were graphically presented and a smooth transition from laminar to turbulent flow was observed in plots of both friction factor and Nusselt number versus Reynolds number. The study confirmed the statement by Inaba el al. [12] by which the ratio between the pitch and the wire diameter p/e is the geometrical non-dimensional parameter that characterizes the thermohydraulic behaviour of wire coil inserts.

In the research on flow visualization in tubes with wire coils which the authors [14] carried out, different flow patterns in laminar and transition regimes were observed. It was accurately established the transitional Reynolds number to the turbulence. Likewise it was established how the flow patterns affect the pressure drop. Results from the visualization research have been also applied to the present experimental work. A new set of experiments has been performed on three wires of p/e between 16.4 and 44.3, relating the flow physical mechanism to the heat transfer enhancement in laminar and transition flows. Correlations for the wires heat transfer both for forced and mixed convection are also proposed, and the criterion to choose one of those is established. Finally the performance of wire coil inserts is compared to that of the twisted tapes inserts and the most appropriate conditions to use one or another enhancement technique are suggested.

2. Experimental set-up

A schematic diagram of the experimental set-up is shown in Fig. 1. It consisted of two independent circuits. The wire coils were installed in the main circuit. It was filled with the test fluid. The secondary circuit was used for regulating the tank temperature to a desirable value and it was filled with chilled water. Test fluid was pumped from an open reservoir tank by a variable-speed centrifugal pump. The flowrate was measured by a Coriolis flowmeter. All the instrumentation installed on the main circuit was connected to a HP 34970A Data Acquisition Unit.

The test section was a thin-walled, 4 m long, 316L stainless steel tube with a wire coil insert. The inner and outer diameters of the tube were 18 mm and 20 mm, respectively. Heat transfer experiments were carried out under uniform heat flux (UHF) conditions. The tube was heated by passing an ac current directly through the tube wall. Power was supplied by a 6 kVA transformer connected with copper electrodes to the tube. A variable auto-transformer in series with the main transformer was used for power regulation. The heat transfer test section was defined by the length between electrodes, which was varied up to $l_h/d = 150$. The axial position of the measuring point, x_p, was defined from the upstream electrode. This position could be set in a range from 30 to 120 diameters, to avoid the axial heat conduction effects close to the electrodes. The loop was insulated by an elastomeric thermal insulation material of 20 mm thickness and thermal conductivity 0.04 W/(m K) to minimize heat losses. The overall electrical power added to the heating section, Q, was calculated by measuring the voltage between electrodes (0–15 V) and the electrical current (0–600 A).

Fluid inlet and outlet temperatures t_{in} and t_{out} were measured by submerged type RTDs (*Resistance Temperature Detectors*). Since the heat was added uniformly along the tube length, the bulk temperature of the fluid at the measuring section, $t_b(x_p)$, was calculated by considering a linear variation with the axial direction. Average outside surface temperature of the wall $\overline{t_{wo}}$ was measured at one axial position x_p. The value of $\overline{t_{wo}}$ was calculated by averaging the temperatures which were measured by using twelve surface type RTDs peripherically spaced by 30°.

Two calibration tests with no electrical heating were performed: the first test was carried out to determine heat losses in the test section Q_l by measuring $(t_{in} - t_{out})$ at low flow rates and the second test at high flow rates (where $t_{in} \approx t_{out} \approx \overline{t_{wo}}$) to calculate the lay-out resistances of the surface type RTDs. Further details of the calibration tests are given in Vicente et al. [17].

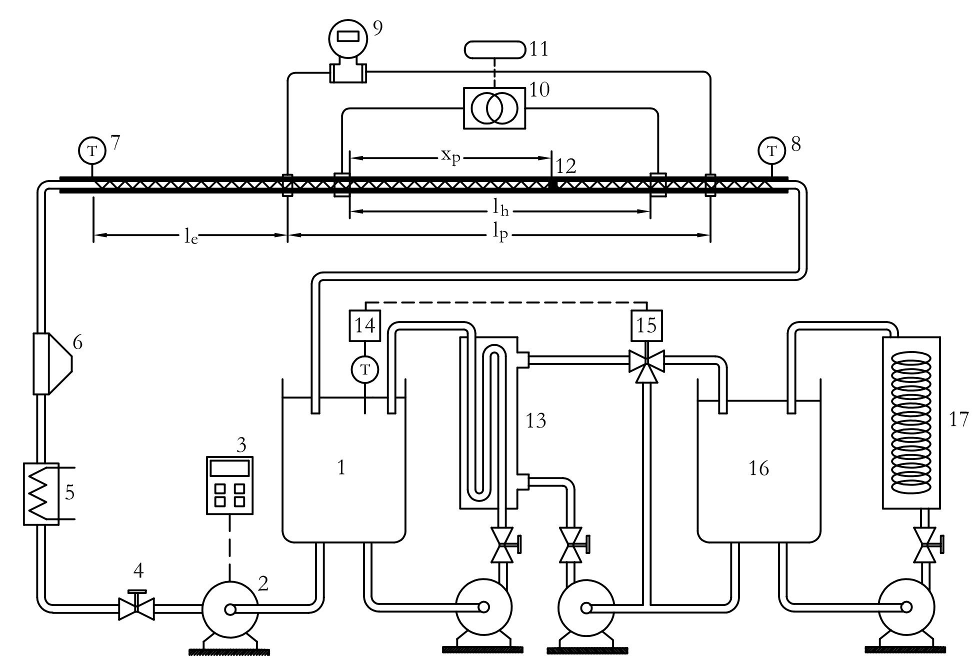

Fig. 1. Experimental set-up. (1) reservoir tank; (2) centrifugal pump; (3) frequency converter; (4) valve; (5) electrical heater; (6) Coriolis flow-meter; (7) inlet temperature; (8) outlet temperature; (9) pressure transmitter; (10) 6 kVA transformer; (11) auto-transformer; (12) surface type RTDs; (13) counterflow heat exchanger; (14) PID controller; (15) three way valve; (16) secondary reservoir tank; (17) cooler.

Heat flux added to the test fluid q'' was calculated by subtracting heat losses to the overall electrical power added in the test section. The inner wall temperature $\overline{t_{wi}}$ for each experimental point was determined by using a numerical model that solves the steady-state one-dimensional radial heat conduction equation in the tube wall from the following input data: $\overline{t_{wo}}$, Q, Q_l, and $t_b(x_p)$. The local Nusselt number was calculated by means of

$$\overline{Nu_x} = \frac{d}{k}\frac{q''}{\overline{t_{wi}} - t_b(x_p)}. \quad (1)$$

Laminar flow film coefficients in horizontal tubes under UHF conditions depend on the axial position in the thermally developing region and on the buoyancy effects in the fully developed region. Heat transfer experiments were performed at different axial positions x_p in order to study the thermally developing region. The way RTDs were installed (peripherically spaced by every 30°) allowed to perceive circumferential variations in wall temperature and to determine which experiences were affected by buoyancy effects. Nusselt numbers calculated by Eq. (1) were corrected by the factor $(\mu_{wi}/\mu_b)^{+0.14}$ (Shah and London [15]) to obtain correlations free from variable properties effects.

Pressure drop tests were carried out in the hydrodynamically developed region under isothermal conditions. The inner smooth tube diameter d was used as the reference diameter to calculate all friction factors. Fanning coefficients f were determined from fluid mass flow rate and pressure drop measurements by means of

$$f = \frac{\Delta P d^5 \pi^2 \rho}{32 l_p \dot{m}^2}. \quad (2)$$

Pressure drop ΔP was measured along the pressure test section ($l_p = 2.83$ m) by means of a highly accurate pressure transducer. Four pressure taps separated by 90° were coupled to each end of the pressure test section. Two differential membrane pressure transducers of different full scales assured the accuracy of the experiments.

The experimental uncertainty was calculated by following the "Guide to the expression of uncertainty in measurement" published by ISO [16]. Details of the assignment of the uncertainty level to the experimental data is given by the authors in [17]. Uncertainty calculations based on a 95% confidence level showed limit values of 4% for Reynolds number, 3.5% for Prandtl number, 4.5% for Rayleigh number, 6% for Nusselt number and 3% for friction factor.

2.1. Wire coils tested

Fig. 2 shows a sketch of a wire coil inserted in a tube, where p stands for helical pitch and e for wire diameter. These parameters can be arranged to define the wire geometry in non-dimensional form: dimensionless pitch p/d, dimensionless wire-diameter e/d and pitch to wire-diameter ratio p/e.

The experimental study was carried out on three wire coils fitted in a smooth tube. Table 1 shows geometrical parameters of wire coils. The inserts were manufactured from spring steel wire covered with plastic insulation sleeving. This plastic insulation avoided electrical contact between the steel wire and the inner tube wall, which was electrically heated. The experimental set-up was adjusted and verified through pressure drop and heat transfer experiments carried out on the smooth tube. This smooth tube was used afterwards for doing experiments in all wire coils.

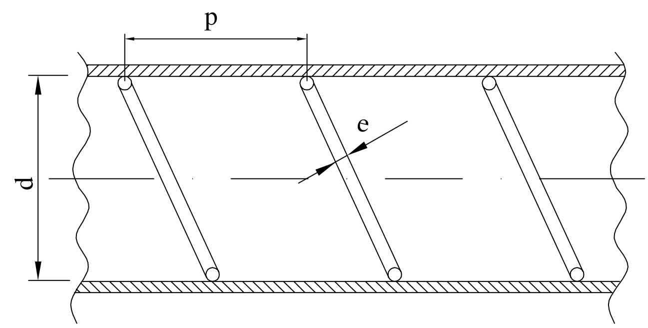

Fig. 2. Sketch of a wire coil fitted inside a smooth tube.

Table 1
Geometry of the wire coils tested

	d (mm)	p/d	e/d	p/e
W01, short pitch	18	1.25	0.076	16.4
W02, intermediate pitch	18	1.72	0.076	22.6
W03, long pitch	18	3.37	0.076	44.3

2.2. Test fluids

Pressure drop tests were performed with water, propylene glycol USP grade and a mixture of propylene glycol and water (50% by weight) at two different temperatures: 20 °C and 50 °C. Heat transfer tests were performed with propylene glycol USP grade at temperatures ranging from 18 °C to 44 °C.

3. Pressure drop results

Pressure drop tests were carried out in a smooth tube and in the same tube with three different wire coils. Friction factor results are provided in a range of Reynolds numbers from 40 to 80000 (Fig. 3). In this paper, attention will be directed to the laminar and transition regimes. The turbulent regime of six wire coils of geometrical parameters similar to those studied in the present work was analysed by the authors in a previous work ([13]).

Laminar flow friction factors were correlated in the *fully laminar region*. The following correlations are proposed:

$$\text{W01}: f = 14.5/Re^{0.93} \quad (Re < 400), \quad (3)$$

$$\text{W02}: f = 14.8/Re^{0.95} \quad (Re < 450), \quad (4)$$

$$\text{W03}: f = 13.3/Re^{0.97} \quad (Re < 700). \quad (5)$$

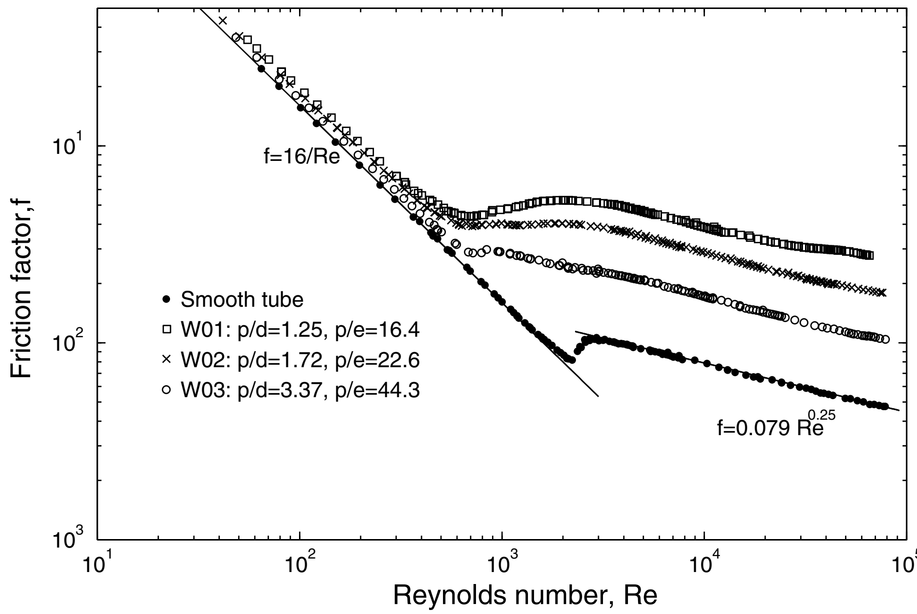

Fig. 3. Experimental results for friction factor in wire coils W01–W03.

These equations correlate 95% of the experimental data within 2% in the specified Reynolds number range.

The pure laminar behaviour persists in a wider range of Reynolds number in wires with longer pitches. In this region, friction factor increases between 5% and 40% (depending on the wire coil pitch) are observed; long pitch W03 shows the smallest friction factor values. In turbulent flow, friction factor increases are much higher than in laminar flow. They reach up to 400% as it was analysed by Garcia et al. [13].

Pressure drop results show that wire coils have a smooth transition from laminar to turbulent regime without the instabilities and the pressure drop fluctuations that the smooth and dimpled tubes have (Vicente et al. [17]). It can be stated that this effect is produced by the swirl velocity generated by the helical wire. In twisted tape inserts where the rotating component is clearly induced, transition from laminar to turbulent regime is produced continuously and it is also not possible to determine the point where transition occurs (Manglik and Bergles [18]).

4. Heat transfer results

Heat transfer tests for a smooth horizontal tube and for three wires in laminar flow have been carried out according to the experimental procedure described in Section 2. Tests were carried out under UHF conditions with the main aim of establishing whether the wires increase or decrease the heat transfer both in mixed and in pure forced convection.

These tests tried to determine the averaged local Nusselt number at the measuring point $\overline{Nu}_x$, which depends both on the measuring point and on the natural convection $\overline{Nu}_x = \overline{Nu}_x(x^*, Ra)$. In order to set the influence of these effects, tests were carried out at different positions x_p and flow rates (different x^*) and at different heat fluxes q'' (different Ra). Fluid temperature at the measuring point $t_b(x_p)$ was varied from 18 °C to 44 °C, in order to study the effect of Prandtl number on heat transfer enhancement. A wide range of flow conditions was achieved: $Re = 10\text{–}2500$, $Pr = 200\text{–}700, Ra = 3 \times 10^6\text{–}10^8$ and $x^* = 10^{-4}\text{–}10^{-2}$.

Tests were carried out with a considerable heat flux in order to obtain a wall to fluid temperature difference $(\overline{t_{wi}}(x_p) - t_b(x_p))$ high enough to be accurately measured (higher than 5 °C in each case). The buoyancy forces effect was stated by the circumferential variation that was observed in the external tube wall temperature at the measuring point. Many of the experimental data were influenced by the natural convection. However, tests at high flow rates where the measuring point was located close to the entry were not influenced by the natural convection. They were though influenced by the thermal entry region. In the experimental set-up, it is not possible to reach the condition of a fully developed flow in pure forced convection.

4.1. Smooth tube results

Tests started with the empty smooth tube and served both to check the facility performance and to verify the measuring uncertainties. Experimental results have been compared with the correlations for heat transfer on smooth horizontal tubes in mixed convection.

In the book by Petukhov and Polyakov [19] the following equation is proposed for the average Nusselt number in the *mixed laminar convection* region under UHF conditions:

$$\overline{Nu}_x = Nu_{fc}[1 + (Ra/B)^4]^{0.045}, \tag{6}$$

where Nu_{fc} is the local Nusselt number in *pure forced convection*, given by

$$Nu_{fc} = 4.36 + 1.31(x^*)^{-1/3} \exp\left(-13\sqrt{x^*}\right), \tag{7}$$

and B is a function of the dimensionless length x^*

$$B = 5.0 \times 10^3 (x^*)^{-1} \quad \text{if } x^* < 1.7 \times 10^{-3},$$
$$B = 1.8 \times 10^4 + 55(x^*)^{-1.7} \quad \text{if } x^* > 1.7 \times 10^{-3}.$$

In pure forced convection, the fully developed thermal region starts by $x^* \approx 0.1$; here the Nusselt number becomes constant to the already known value of $Nu_{fc} = 4.36$. However, in mixed convection the thermally developing region is much smaller ($x^* \approx 0.002$ at $Ra = 4 \times 10^6$). In the fully developed region, the Nusselt number shows an asymptotic behaviour ($x^* \to \infty, B = 1.8 \times 10^4$) and Eq. (6) is:

$$\overline{Nu}_\infty = 4.36[1 + (Ra/18000)^4]^{0.045}. \tag{8}$$

$\overline{Nu}_\infty$ is the Nusselt number in the *fully developed region*. From Eq. (7) it can be inferred that in the entry region the Nusselt number for pure forced convection, Nu_{fc}, is proportional to $(x^*)^{-1/3}$. On the other hand, Eq. (8) shows that in the fully developed region, $\overline{Nu}_\infty$ is proportional to $Ra^{0.18}$.

A first series of experimental tests was carried out with an intermediate position of the measuring point, $x_p/d = 55$. Experimental results both in forced and in mixed convection were achieved, as Fig. 4 shows. Results in mixed convection are well above to those predicted by Eq. (6). However results in pure forced convection agree to a great extent with Eq. (7), and they are correlated by the following equation:

$$Nu_{fc,s} = 1.32(x^*)^{-1/3}. \tag{9}$$

A second series of experimental tests with $x_p/d = 105$ was made to obtain further Nusselt number results in the fully developed thermal region in mixed convection. Fig. 5 shows results of this tests along with those of the tests at $x_p/d = 55$ which are in the fully developed region ($x^* > 4 \times 10^{-3}$). The experimental results are compared with the Petukhov and Polyakov [19] correlation (Eq. (8)) and with the Morcos and Bergles [20] correlation, which is:

$$\overline{Nu}_\infty = [4.36^2 + (0.145(RaPr^{0.35}pw^{-0.25})^{0.265})^2]^{0.5}. \tag{10}$$

The term $pw = (kd/(k_w s))$ from Eq. (10) represents the ratio between the peripheral and radial thermal resistances of the tube wall.

Both Petukhov and Polyakov [19] and Morcos and Bergles [20] worked under UHF conditions. Morcos and Bergles [20] correlation adjusts to Petukhov and Polyakov [19] correlation at typical Prandtl number values of water ($Pr = 2$–7). Morcos and Bergles [20] include a small influence of the Prandtl number on the heat transfer, $Nu_\infty \propto Pr^{0.09}$, which leads to $\overline{Nu}_\infty$ values above Petukhov and Polyakov correlation at high Prandtl numbers. Experimental results shown in Fig. 5 are within a Prandtl number range $Pr = 200$–700 and find themselve above those from Petukhov and Polyakov [19] and below those from Morcos and Bergles [20].

In order to estimate the real heat transfer increase obtained by the wire coils, the following correlation is proposed to predict the smooth tube Nusselt number in the fully developed thermal region:

$$\overline{Nu}_{\infty,s} = 0.2348Ra^{0.262}. \tag{11}$$

By using this expression, the smooth tube results were correlated with a deviation of 3% for 95% of the data in the Rayleigh number range from 2.3×10^6 to 3×10^7. Thus,

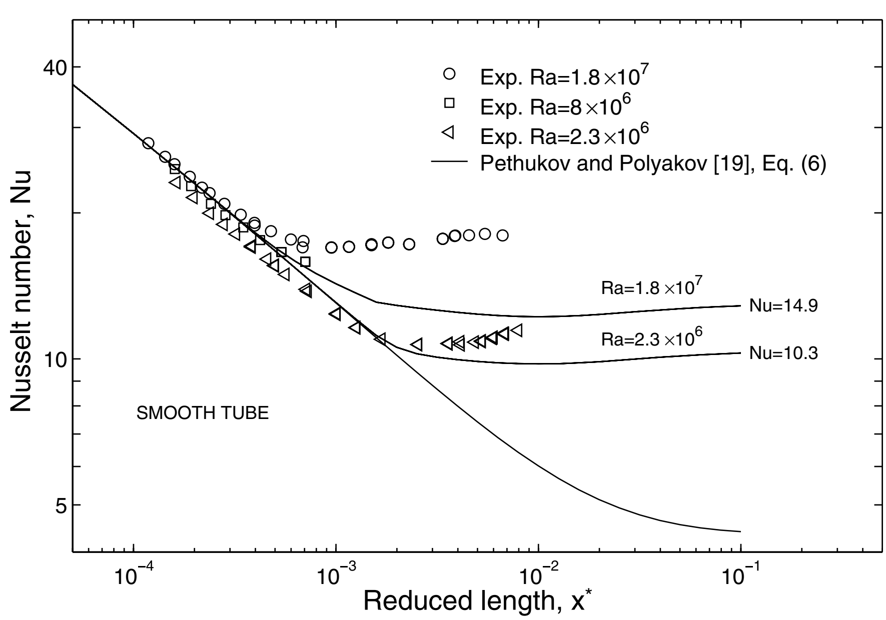

Fig. 4. Experimental results on heat transfer in forced and mixed convection in the smooth tube at $x_p/d = 55$. Local Nusselt number $\overline{Nu}_x$ as a function of Rayleigh number Ra and dimensionless length x^*. Comparison with the Petukhov and Polyakov correlation [19], Eq. (6).

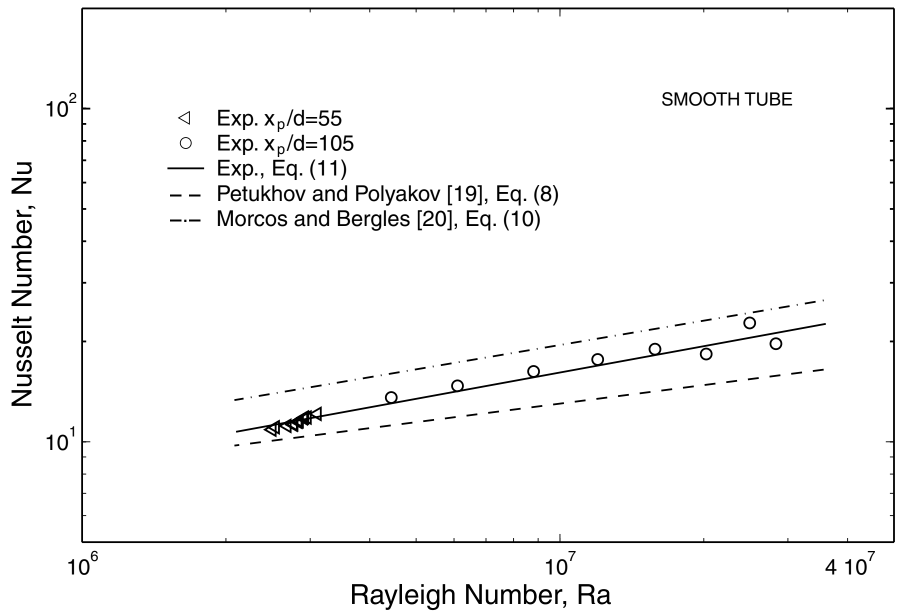

Fig. 5. Experimental results of heat transfer in natural convection in a smooth tube. Fully developed Nusselt number $\overline{Nu}_\infty$ as function of the Rayleigh number Ra. Comparison with the Petukhov's and Polyakov's [19] correlation, Eq. (8), and with the Morcos and Bergles [20] correlation, Eq. (10).

two experimental correlations for the smooth tube have been obtained: one for pure forced convection (Eq. (9)) and another for fully developed flow in mixed convection (Eq. (11)). To establish when buoyancy effects are predominant, that is to say, when Eq. (11) is to be applied instead of Eq. (9), every experimental result was processed. To this purpose, the parameter parameter Δt^* was used, which is the ratio between the maximum circumferential wall temperature difference and the wall to fluid temperature difference:

$$\Delta t^* = \frac{t_{\mathrm{wi,max}} - t_{\mathrm{wi,min}}}{\overline{t_{\mathrm{wi}}}(x) - t_{\mathrm{b}}(x)}. \tag{12}$$

This parameter was related to the dimensionless term Gr/Re^2, that is a measure of the ratio between the buoyancy forces and the inertia forces. It was taken as a criterion that at values of Δt^* above 0.1, the effect of buoyancy forces is important. It was obtained that all the experimental results with $Gr/Re^2 > 0.3$ had $\Delta t^* > 0.1$. Thus, in a smooth tube at values of Gr/Re^2 above 0.3, the heat transfer is affected in a meaningful way by mixed convection.

4.2. Wire coils results

Out of the analysis of the experimental heat transfer data from wires W01, W02 and W03, it has been confirmed a different behaviour between the results at Reynolds numbers below and above 200.

4.2.1. Re < 200

At Reynolds numbers below 200, it is perceived that the heat transfer shows the typical laminar flow features in a smooth tube. This area will be referred to as the *fully laminar region* and fits in the flow topology observed by the authors in [14], where the flow was not affected by the wires and a separating flow downstream of the wire did not occur.

Figs. 6–8 show the results for the averaged local Nusselt number $\overline{Nu}_x$ corresponding to W01, W02 and W03, respectively. Results correspond to positions $x_p/d = 27$ and $x_p/d = 55$ for two fixed Rayleigh numbers. The experimental data above $x^* \approx 0.003$ tend to the fully developed flow asymptote. It is also perceived that there is a region where data show the typical trend of forced convection in smooth tubes.

Results in the entry region in forced convection ($\Delta t^* < 0.1$), can be correlated by:

$$\mathrm{W01}: Nu_{\mathrm{fc}} = 1.258(x^*)^{-0.334}(Gr/Re^2 < 2.8), \tag{13}$$

$$\mathrm{W02}: Nu_{\mathrm{fc}} = 0.918(x^*)^{-0.375}(Gr/Re^2 < 0.7), \tag{14}$$

$$\mathrm{W03}: Nu_{\mathrm{fc}} = 0.961(x^*)^{-0.365}(Gr/Re^2 < 0.5). \tag{15}$$

In order to get data in the fully developed region, tests with $x_p/d = 92$ and Rayleigh numbers from 3×10^6 to 10^8 were performed. These results along with the results from the test with $x_p/d = 55$ that adhere to $x^* > 0.003$ are shown in Fig. 9 for Wire W01. Nusselt number data in the fully developed region with Δt^* higher than 0.1 were correlated for each wire. The following correlations are proposed:

$$\mathrm{W01}: \overline{Nu}_\infty = 0.0596Ra^{0.361}(Gr/Re^2 > 4.5), \tag{16}$$

$$\mathrm{W02}: \overline{Nu}_\infty = 0.0906Ra^{0.322}(Gr/Re^2 > 2.9), \tag{17}$$

$$\mathrm{W03}: \overline{Nu}_\infty = 0.1060Ra^{0.309}(Gr/Re^2 > 1.4). \tag{18}$$

By using the above expressions, the experimental results of wire coils W01–03 were correlated with a maximum deviation of 7% for 95% of the data in the Rayleigh number

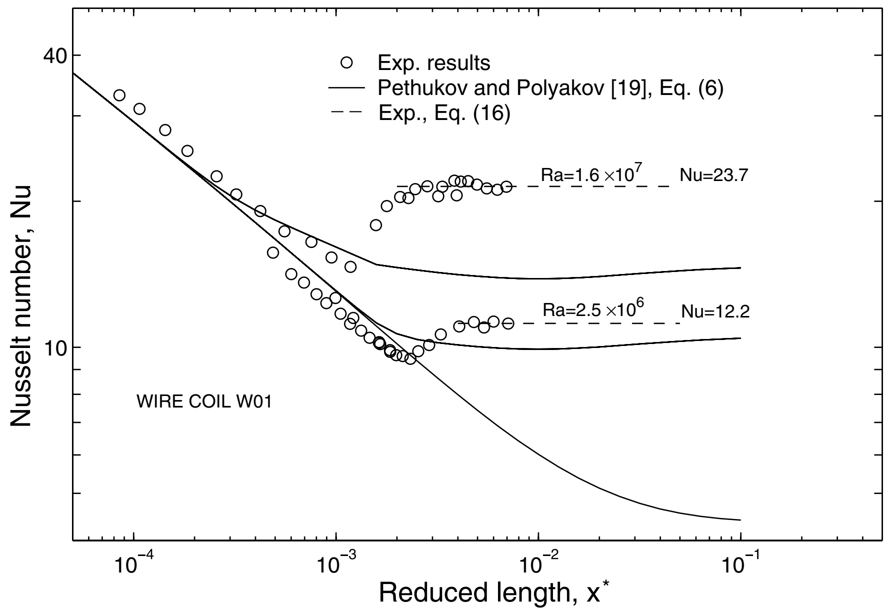

Fig. 6. Heat transfer experimental results in forced and mixed convection for wire W01. Local Nusselt number as a function of Rayleigh number and dimensionless length.

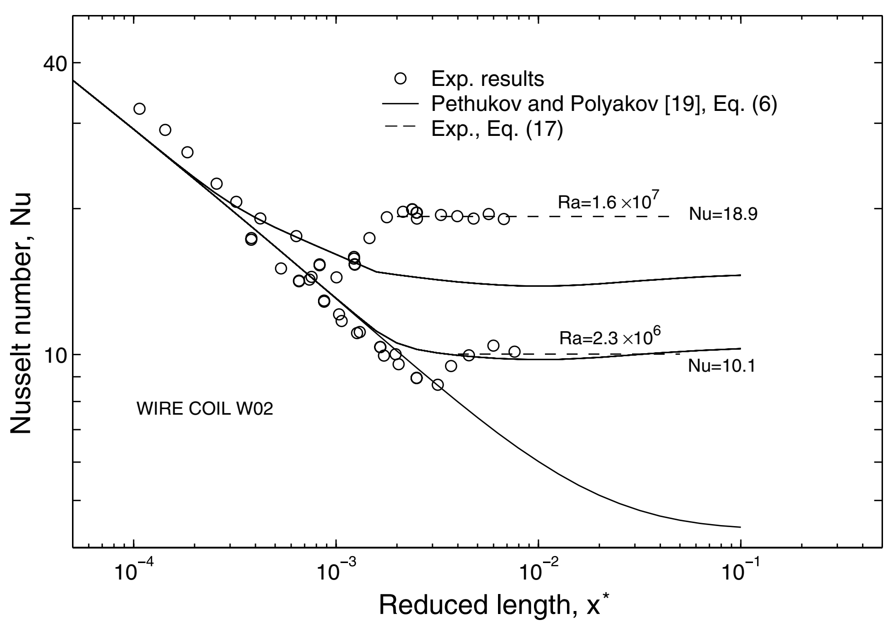

Fig. 7. Heat transfer experimental results in forced and mixed convection for wire W02. Local Nusselt number as a function of Rayleigh number and dimensionless length.

range from 3×10^6 to 10^8 for Reynolds numbers below 200.

For each wire it has been found out that between the limit values of the ratio Gr/Re^2 that guarantee a forced convection flow (Eqs. (13)–(15)) and those that guarantee a fully developed flow in mixed convection (Eqs. (16)–(18)), there is experimental data in both forced and mixed convection. If the term Gr/Re^2 lies between these limit values, it is recommended to use the forced convection equations as a conservative estimation.

The limit values of the ratio Gr/Re^2 show that wire coil inserts delay the development of mixed convection flow. In the smooth tube, at values of Gr/Re^2 so low as 0.3, there is a fully developed flow. This phenomenon becomes more noticeable in small pitch wires. It can be stated that the rotational component on the core flow produced in wire coils (authors [14]), hinders the establishment of the buoyancy-driven recirculations.

Fig. 10 compares the wire coils and smooth tube results for the fully developed Nusselt number, $\overline{Nu}_\infty$. Though wire

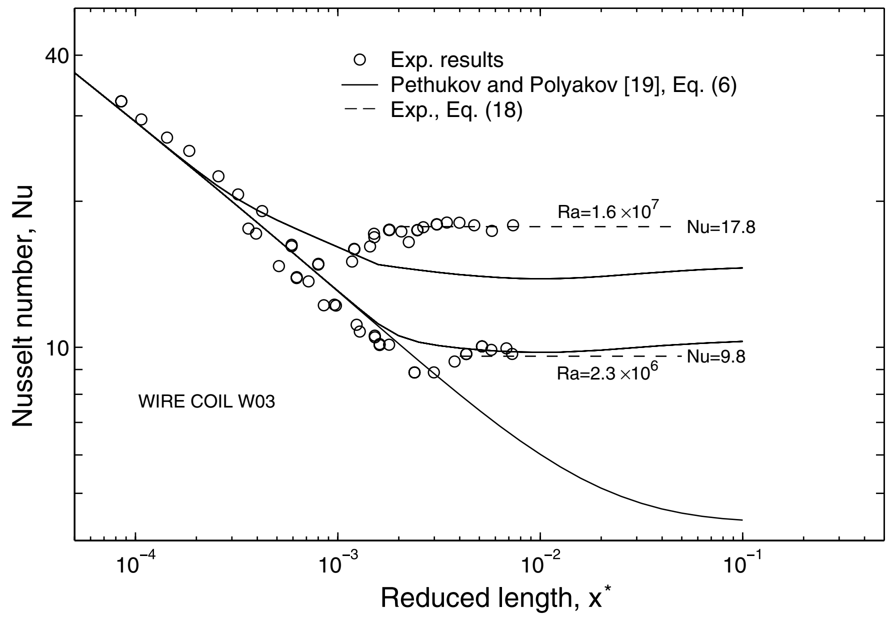

Fig. 8. Heat transfer experimental results in forced and mixed convection for wire W03. Local Nusselt number as a function of Rayleigh number and dimensionless length.

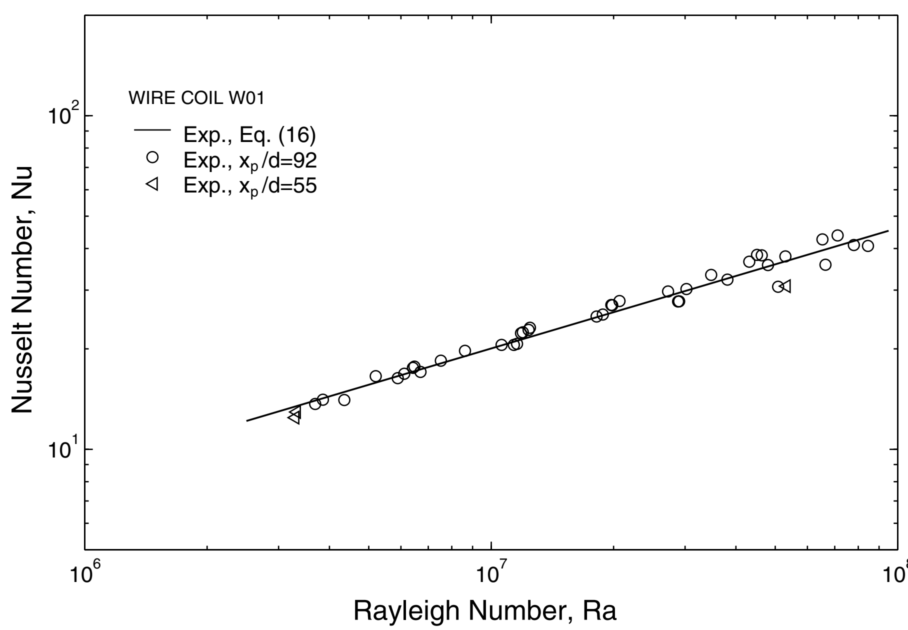

Fig. 9. Heat transfer experimental results in natural convection in wire W01. Fully developed Nusselt number $\overline{Nu}_{\infty}$ as a function of Rayleigh number Ra.

coils delay the appearance of mixed convection flow, it is observed that once it appears, the rate of increase of Nu with Ra is higher than that for the smooth tube. This type of behaviour has also been observed in corrugated tubes (Vicente et al. [21]) and in tubes with internal fins (Rustum and Soliman [22]).

4.2.2. Re > 200

Experimental tests on wires have shown that at Reynolds numbers above 200, heat transfer is not influenced either by the entry region or the natural convection. All the experimental results had Δt^* lower than 0.05. This suggests that the wire disrupts the flow enough to eliminate the onset of the recirculations that the buoyancy forces cause in horizontal tubes. Figs. 11–13 show the Nusselt number results in pure forced convection in the range $Re =$ 25–2500. The results shown for Reynolds numbers below 200 have Δt^* lower than 0.05. These results are compared with Eq. (9) for a smooth tube in the laminar region, the Gnielinski [23] correlation for a smooth tube in turbulent flow and the experimental correlation for turbulent flow in wire coils provided by Garcia et al. [13].

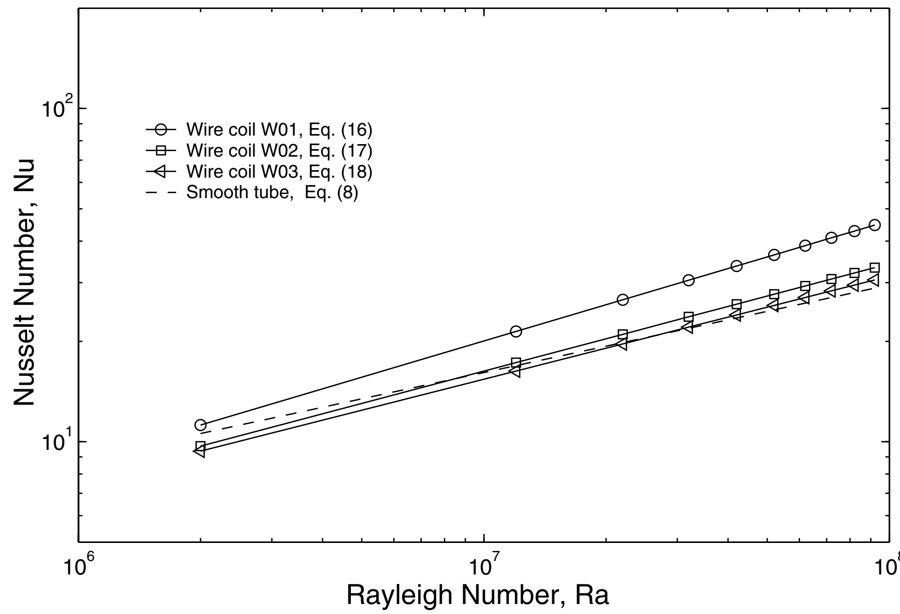

Fig. 10. Fully developed Nusselt number $\overline{Nu}_\infty$ as a function of Rayleigh number Ra. Comparison of results achieved in wires W01–03 and in the smooth tube.

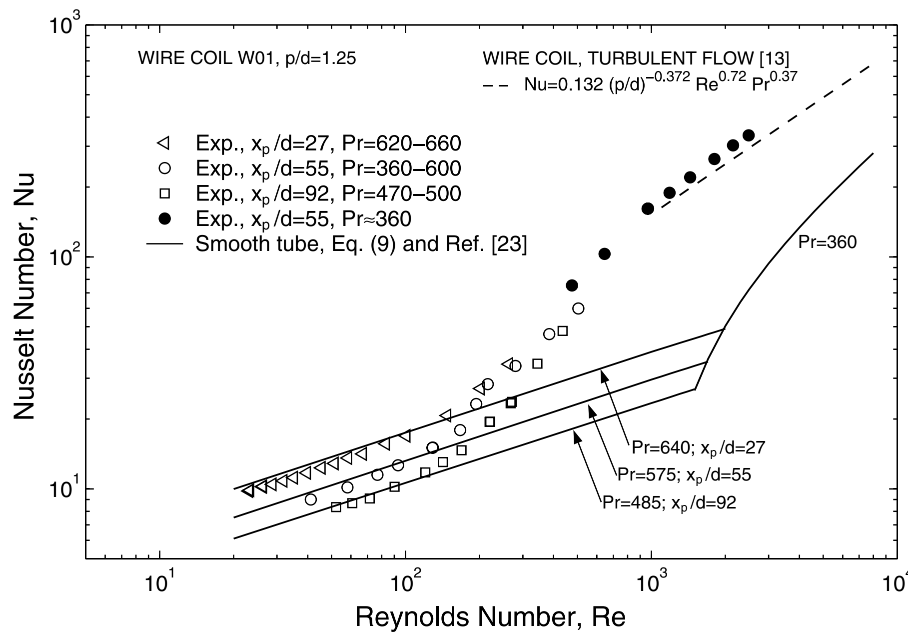

Fig. 11. Nusselt number in forced convection Nu_{cf} vs. Reynolds number Re. Results for wire W01.

First of all, results obtained for the small pitch wire W01 (Fig. 11) are commented in detail. In the fully laminar region ($Re < 200$), results only depend on the entry length. When these are compared to Eq. (9) for the smooth tube, ($Nu_{\mathrm{fc}} = 1.32(x^*)^{-1/3}$), it is observed that there are basically no differences. Results reveal clearly that at Reynolds numbers between 200 and 1000 there is a *laminar–turbulent transitional region* where the wire produces a flow perturbation which remarkably increases heat transfer. At Reynolds numbers above 1000 the flow is *fully turbulent* and Nusselt number results show the same trend as the one observed by the authors in their previous paper [13]. At $Re = 1000$ and $Pr = 360$ the heat transfer exceeds eight times the one that would be obtained with a smooth tube at $x_{\mathrm{p}}/d = 92$.

In the intermediate pitch wire W02 (Fig. 12), the Nusselt number increases are slightly lower than those of W01. Reynolds numbers at which the transition starts are slightly higher. From $Re \approx 1200$ on, the flow is fully turbulent.

The large pitch wire W03 performs the lowest heat transfer. Fig. 13 shows that for this wire the smooth tube tendency keeps within a wider Reynolds numbers range than in the other two wires, until $Re = 500$. The transition

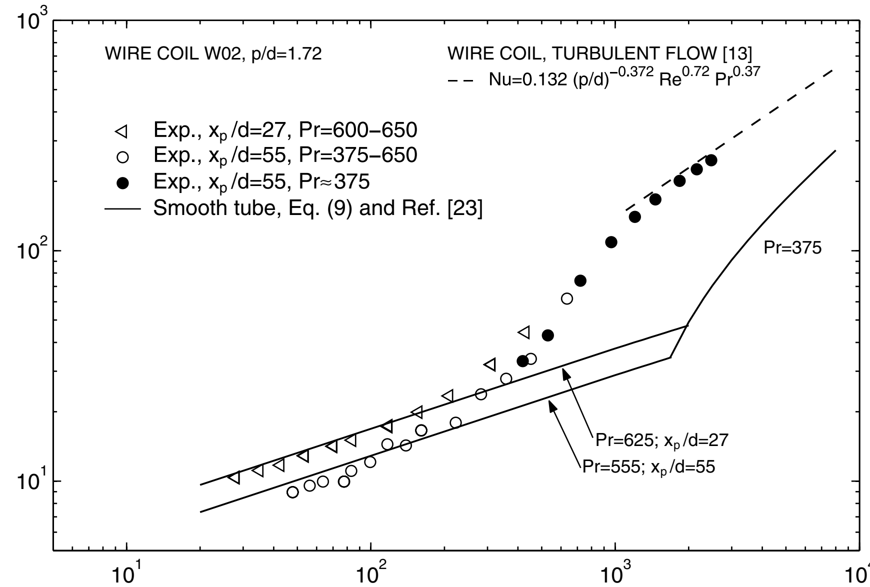

Fig. 12. Nusselt number in forced convection Nu_{cf} vs. Reynolds number Re. Results for wire W02.

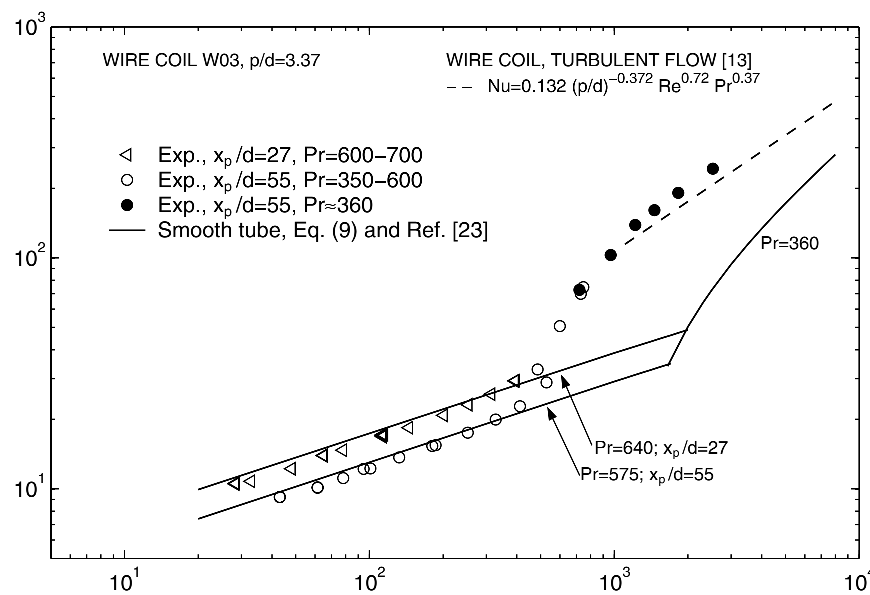

Fig. 13. Nusselt number in forced convection Nu_{cf} vs. Reynolds number Re. Results for wire W03.

region ranges from Reynolds numbers 500–1300. At Reynolds numbers higher than 1300 the observed tendency is the one of a fully turbulent flow.

Correlations for the three wires in the *laminar–turbulent transitional region* have been obtained:

$$W01: Nu = 0.104Re^{-1.01} \quad (Re = 200\text{–}1000, Pr = 200\text{–}650), \tag{19}$$

$$W02: Nu = 0.100Re^{-0.99} \quad (Re = 250\text{–}1100, Pr = 350\text{–}650). \tag{20}$$

$$W03: Nu = 0.077Re^{-1.03} \quad (Re = 500\text{–}1100, Pr = 350\text{–}600). \tag{21}$$

By using the above expressions, the experimental results of wire coils W01–03 were correlated with a maximum deviation of 8% for 95% of the data in the specified flow range. In the transition region, Nusselt number only depends on Reynolds number. It has not been found any meaningful dependency either in the Prandtl number or in the position of the measuring point.

4.3. Performance comparison: wire coils and twisted tapes

The choice of a specific enhancement technique to increase the thermal transfer of a heat exchanger in laminar regime is a decision that must be taken cautiously. The suitability of the chosen solution depends on variables such as the existence or not of fouling, the allowable head loss increase and the existence or absence of natural convection for the working conditions. Regarding the last point, it is quite frequent to find researchers or manufacturers who inform that their enhancement technique increases in several times the heat transfer with respect to the smooth tube in laminar regime. Authors refer these increases to the smooth tube in forced convection and obviate that at low Reynolds numbers it is quite likely that the thermal transfer would be carried out under mixed convection conditions. In warm-up processes, the natural convection notably benefits the global heat transfer. This even increases up to 5 times in horizontal smooth tubes, without a pressure drop increase. Except for the familiar technique of providing the smooth tube with extended surfaces, the passive enhancement techniques have little to offer to improve the heat transfer under free convection conditions (Bergles [24]).

The enhancements that the wire coil and the twisted tape inserts produce in laminar flow have been compared. Figs. 14 and 15 show the Nusselt number vs. Reynolds number results for wire W01, compared with the predictions of Manglik and Bergles [8] and Hong and Bergles [7] for two tapes of dimensionless thickness $\delta/d = 0.045$ and twist ratios $y = 2.5, 5$. The figures also show the Nusselt number prediction for the smooth tube both in forced and mixed convection at $Ra = 10^7$, according to the empirical

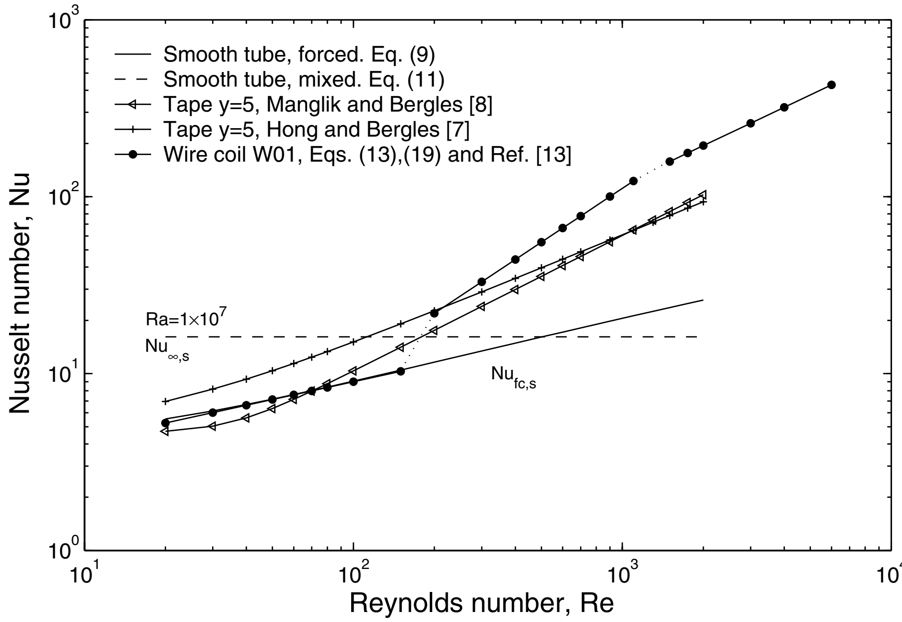

Fig. 14. Nusselt number vs. Reynolds number. Wire coil W01 and twisted tape with $y = 5$.

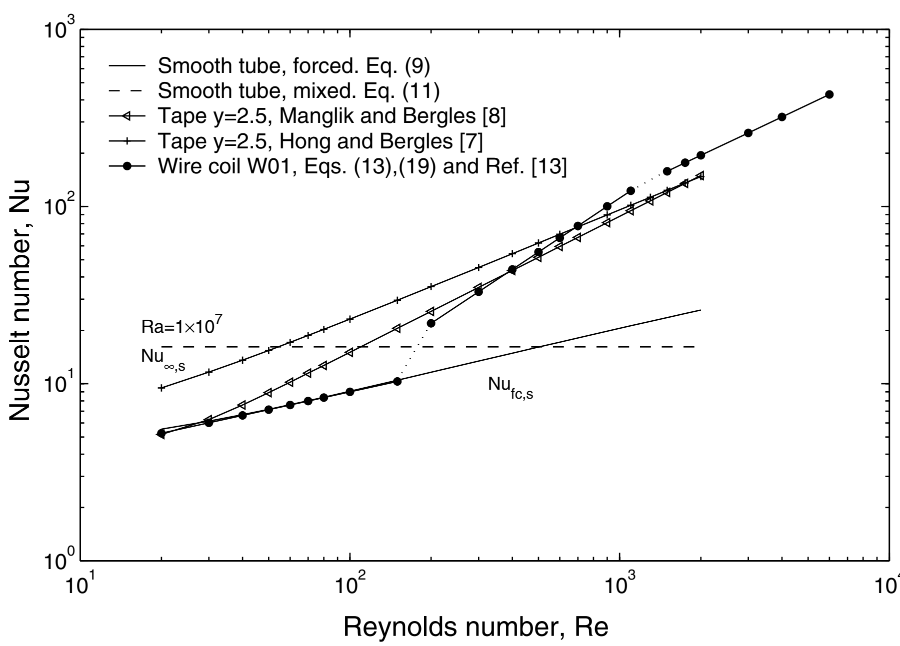

Fig. 15. Nusselt number vs. Reynolds number. Wire coil W01 and twisted tape with $y = 2.5$.

correlations obtained in this paper. The predictions have been obtained for a 1 m long horizontal tube, with inside diameter $d = 18$ mm under the flow conditions $Re = 20$–2500 and $Pr = 200$.

Figs. 14 and 15 show that the twisted tapes behave better than the wire coils at low Reynolds numbers ($Re < 200$). In this region a wire coil is almost ineffective. However, the use of a twisted tape would be clearly counterproductive if the smooth tube were under free convection influence at a Rayleigh number higher than 10^7. Here the smooth tube performance would get worse if either wire coils or twisted tapes were inserted. Both twisted tapes and wire coils increase heat transfer within the Reynolds number range between 200 and 1000. Here wire coil W01 performs better than the twisted tape of $y = 5$; wire W01 and tape of $y = 2.5$ increase heat transfer by percentages of the same order of magnitude. The latter represents the limits of manufacturing of twisted tape inserts, and at $Re = 300$ has a Nusselt number 40% higher than that of the wire W01. The head loss in twisted tapes is always very high: at $Re = 300$ the friction factor increases f/f_s for the wire W01 and twisted tapes of $y = 5, 2.5$ are 1.3, 3.3 and 4, respectively. At $Re = 700$ the wire W01 and the twisted tape of $y = 2.5$ have the same Nusselt number but the pressure drop in the tape is almost three times that in the wire coil.

For Reynolds numbers above $Re = 1000$, wire coil insert have higher heat transfer coefficients than any of the two studied twisted tapes. The difference is essential: whereas for this Reynolds number in the wire W01 the flow is already turbulent, in the twisted tapes the flow is still laminar. Whereas the twisted tapes *delay* the transition to turbulent flow up to Reynolds numbers higher than those of the smooth tube (Manglik and Bergles [8]), the wire coil inserts *bring forward* it (authors [14]). Even so, the pressure drop produced by the wire W01 is of the same order than the one introduced by any of the two studied twisted tapes. In this range of Reynolds numbers, Oliver and Shoji [10] also observed a better performance of wire coils with respect to twisted tapes for non-newtonian flows.

5. Conclusions

1. An extensive experimental study on three wires of different pitch inserted in a smooth tube has been carried out. Laminar and transition regions have been covered. Heat transfer tests have been performed under UHF conditions, and the flow ranges have been: $Re = 10$–2500, $Pr = 200$–700 and $Ra = 3 \times 10^6$–10^8. For each wire, experimental correlations of isothermal friction factor and Nusselt number under forced and mixed convection conditions have been obtained.
2. The friction factor increases in the fully laminar region lie between 5% and 40%. The transition from laminar flow to turbulent flow is smooth, without the instabilities and the pressure drop fluctuations that a smooth tube presents.
3. The fully laminar region covers up to $Re \approx 200$. Here, heat transfer in wire coils can be produced either in forced or in mixed convection. In this region, wire coils do not enhance the heat transfer with respect to the smooth tube.
4. At Reynolds numbers between 200 and 1000, wire coils significantly disturb the flow and remarkably increase heat transfer. The perturbation that the wire brings about hinders the establishment of the recirculations that the buoyancy forces cause in horizontal tubes, and the heat transfer always takes place in forced convection.
5. At Reynolds numbers above $Re \approx 1000$–1300 on, wire coil inserts promote the transition from laminar to turbulent flow. The Nusselt number results follow the correlation for turbulent flow developed by the authors. The heat transfer increase in this low Reynolds number region is very remarkable: in wire W01 at $Re = 1000$ and $Pr = 360$ the Nusselt number Nu is eight times the one of a smooth tube with $x_p/d = 92$.
6. Comparison between the experimental results obtained in wire coils and the most recognized correlations for the twisted tapes allows to state that both devices are ineffective to increase the heat transfer of a smooth tube in mixed convection at Re below 200. At Reynolds numbers between 200 and 700, both the pressure drop and the heat transfer increases need to be considered in order to select one of the two enhancement techniques. Finally, in a smooth tube equipment that works at Reynolds numbers between 700 and 2500, the heat transfer enhancement obtained with the wire coils will be quite higher than the one obtained with the twisted tapes.

Acknowledgements

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