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Experimental study of heat transfer enhancement with wire coil inserts in laminar-transition-turbulent regimes at different Prandtl numbers

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Abstract

Helical-wire-coils fitted inside a round tube have been experimentally studied in order to characterize their thermo-hydraulic behaviour in laminar, transition and turbulent flow. By using water and water–propylene glycol mixtures at different temperatures, a wide range of flow conditions have been covered: Reynolds numbers from 80 to 90,000 and Prandtl numbers from 2.8 to 150. Six wire coils were tested within a geometrical range of helical pitch $1.17 < p/d < 2.68$ and wire diameter $0.07 < e/d < 0.10$. Experimental correlations of Fanning friction factor and Nusselt number as functions of flow and dimensionless geometric parameters have been proposed. Results have shown that in turbulent flow wire coils increase pressure drop up to nine times and heat transfer up to four times compared to the empty smooth tube. At low Reynolds numbers, wire coils behave as a smooth tube but accelerate transition to critical Reynolds numbers down to 700. Within the transition region, if wire coils are fitted inside a smooth tube heat exchanger, heat transfer rate can be increased up to 200% keeping pumping power constant. Wire coil inserts offer their best performance within the transition region where they show a considerable advantage over other enhancement techniques.

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

1. Introduction

In the last decades, significant effort has been made to develop heat transfer enhancement techniques in order to improve the overall performance of heat exchangers. The interest in these techniques is closely tied to energy prices and, with the present increase in energy cost, it is expected that the heat transfer enhancement field will go through a new growth phase. Although there is need to develop novel technologies, experimental work on the older ones is still necessary. The knowledge of its performance shows a large degree of uncertainty which makes their industrial implementation difficult.

Tubeside enhancement techniques can be classified according to the following criteria: (1) additional devices

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Nomenclature

d envelope (maximum inner) diameter (m)
d_h hydraulic diameter, 4× (free volume/wetted surface) (m)
e wire diameter (Fig. 1) (m)
h heat transfer coefficient (W m^{-2} K^{-1})
I electric current (A)
k thermal conductivity (W m^{-1} K^{-1})
l_p length of test section between pressure taps (m)
l_h length of the heat transfer section (m)
$\dot{m}$ mass flow rate (kg s^{-1})
p helical pitch (Fig. 1) (m)
ΔP pressure drop across the test section (Pa)
Q overall electrical power added to the heating section (W)
Q_l heat looses in the test section (W)
q'' heat flux $(Q - Q_l)/(\pi d l_h)$ (W m^{-2})
$R3$ performance criterion
t temperature (K)
$\overline{t_{wi}}$ average inside surface temperature of the wall (K)
$\overline{t_{wo}}$ average outside surface temperature of the wall (K)
V voltage (V)
x_p axial position of the measuring point (m)

Dimensionless groups
f fanning friction factor
Nu Nusselt number
Pr Prandtl number
Ra Rayleigh number
Re Reynolds number

Greek symbols
μ dynamic viscosity (Pa s)
ρ fluid density (kg m^{-3})

Subscripts
a augmented tube (wire coil fitted inside a smooth tube)
b based on bulk temperature
in tube inlet
0 equivalent smooth tube value
out tube outlet
s smooth tube
w based on wall temperature (inside surface)

which are incorporated into a plain round tube (twisted tapes, wire coils) and (2) non-plain round tube techniques such as surface modification of a plain tube (corrugated and dimpled tubes) or manufacturing of special tube geometries (internally finned tubes).

In applications like petrochemical industry where specifications codes are required, insert devices can be used since they do not modify round tube mechanical properties as integral roughness does. They can be used when it is required to increase the heat transfer rate of an existing heat exchanger: there is no need to replace the tube bundle and they can be installed in a routine maintenance stoppage.

Wire coil inserts are currently used in applications as oil cooling devices, preheaters or fire boilers. They show several advantages in relation to other enhancement techniques:

(1) Low cost.
(2) Easy installation and removal.
(3) Preservation of original plain tube mechanical strength.
(4) Possibility of installation in an existing smooth tube heat exchanger.

Fig. 1 shows a sketch of a wire coil inserted in close contact with the inner tube wall, where p stands for

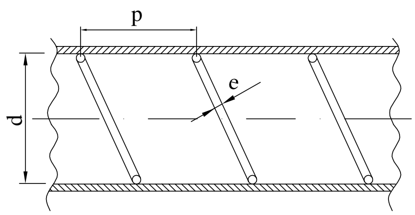

Fig. 1. Sketch of a helical-wire-coil fitted inside a smooth tube.

helical pitch, e for the wire-diameter and d is the tube inner 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 tubeside flow pattern is modified by the presence of a helically coiled wire as follows:

(1) If the wire coil acts as a *swirl flow generator*, a helical flow at the periphery is produced. This rotating flow is superimposed upon the axially directed central core flow and causes centrifugal forces. In most of liquids, where density decreases with temperature, centrifugal forces produce a movement of the heated fluid from the boundary

layer towards the tube axis, which produces a heat transfer augmentation.

(2) If the wire coil acts as a *turbulence promoter*, flow turbulence level is increased by a separation and reattachment mechanism. Besides, whenever wire coils are in contact with the tube wall, they act as roughness elements and disturb the existing laminar sublayer.

Depending on flow conditions and wire coil geometry, the heat transfer rate will increase through one or both of the mechanisms mentioned earlier. However, it is expected that wire coils will act as artificial roughness at high Reynolds numbers. Webb [1], Ravigururajan and Bergles [2] consider that in turbulent regime, wire coils disturb the flow in a similar way that corrugated or ribbed tubes do.

Experimental works which focus on wire coil inserts are few in comparison to those that study twisted tapes. This fact has been pointed out from the works of Kumar and Judd [3] or Sethumadhavan and Raja Rao [4], to the more recent ones of Silva et al. [5] or Shoji et al. [6]. Wire coils are interesting insert devices, and therefore predictable correlations are needed to extend the use of this technique. Twisted tapes may not necessarily be the best insert devices [1], but they are more used than wire coils because design correlations are well established in laminar, transition and turbulent flows.

Table 1 summarizes the most significant experimental works which have been carried out on wire coils in single-phase flow of Newtonian fluids. Most of these works were focused on turbulent flow, where air or water was used as the only test fluid and heat transfer dependence on Prandtl number was not established. In turbulent flow, only the work from Sethumadhavan and Raja Rao [4] extended the Prandtl number range by making use of water and glycerol as test fluids. In laminar and transition flows the only studies which are available in the open literature are those from Uttarwar and Raja Rao [7] and Inaba et al. [8]. For these flow regimes insert devices are expected to be an effective enhancement technique, but the current available data on wire coils is very limited. Uttarwar and Raja Rao [7] studied 7 wires by using servotherm oil as test fluid. They covered a moderate Reynolds number range from 30 to 700. Their heat transfer results are affected by the entry region [1], and they did not provide a friction factor correlation. Experimental work from Inaba et al. [8] is the only one that shows results in the laminar-transition-turbulent region, within a Reynolds number range from 200 to 6000 by using water as test fluid.

Wire coils have been mainly studied in water or air turbulent flow. In order to compare the results from different authors, their experimental correlations have been solved for a wire coil of geometry $p/d = 1.2$ and $e/d = 0.1$ at flow conditions: $Pr = 6$ and 150 and $Re =$ 4000–90,000. A significant dispersion among friction factor correlations is observed as shown in Fig. 2. Predictions of Inaba et al. [8] and Zhang [9] show differences up to a factor of 3. Rabas [10] compiled friction factor data taken from several sources. He concluded that there was no obvious explanation to select any of the available friction factor correlations. Besides, he stated that there is a marked lack of pressure drop data since many authors do not provide them. Nusselt number correlations are plotted in Fig. 3. Results present less deviation than friction data do, but disagreements reach up to factors of 2 at $Pr = 150$. Differences increase at high Pr, since most of heat transfer data were obtained at low Prandtl numbers.

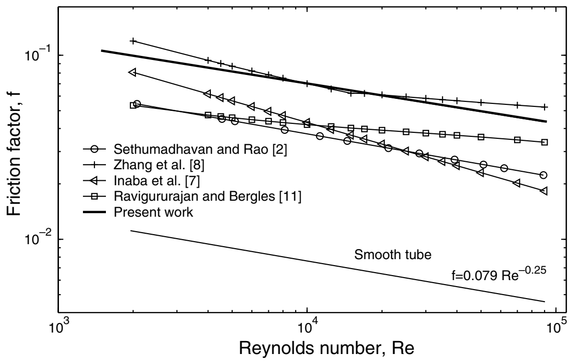

Fig. 2. Comparison of experimental correlations proposed to determine Fanning friction factor in turbulent flow for a wire coil of $p/d = 1.2$ and $e/d = 0.1$.

Table 1
Experimental range covered by the studies published in the literature available

Paper	N_{wire}	p/d	e/d	p/e	$Re \times 10^3$	d [mm]	Test fluid (Pr)
Kumar and Judd [3]	15	1.05–5.50	0.11–0.15	8.0–47.0	7.0–100	12.2	Water (4.3)
Klaczak [11]	13	0.68–2.88	0.10–0.22	6.6–13.0	1.7–20	6.8	Water (2.5–9.0)
Sethumadhavan and Raja Rao [4]	8	0.40–2.64	0.08–0.12	3.3–33.0	4.0–100	25.0	Water–glycerol (5.2–32)
Uttarwar and Raja Rao [7]	7	0.40–2.62	0.08–0.13	5.0–33.0	0.03–0.7	25.2	Servotherm oil (300–675)
Zhang et al. [9]	14	0.35–0.48	0.04–0.09	6.6–28.6	6.0–80	56.3	Air (0.7)
Ravigururajan and Bergles [2]	5	0.60–1.12	0.02–0.05	16–32.0	5.0–25	68	Air (0.7)
Inaba et al. [8]	19	0.30–6.50	0.12–0.19	1.6–52.0	0.2–6.0	16	Water (8.2–3.9)

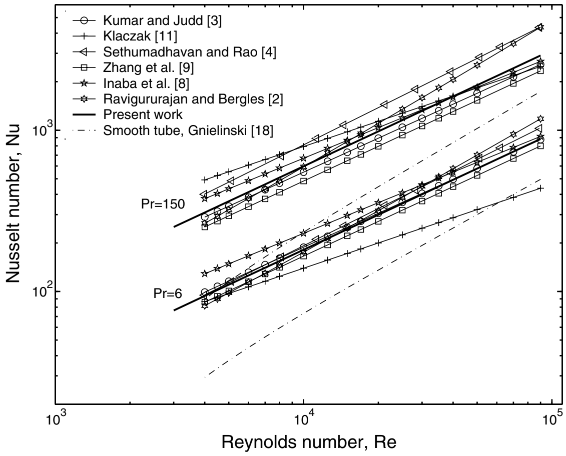

Fig. 3. Comparison of experimental correlations proposed to determine Nusselt number in turbulent flow for a wire coil of $p/d = 1.2$ and $e/d = 0.1$ at $Pr = 6$ and $Pr = 150$ and constant fluid properties.

The main aim of the present paper is to extend the experimental data available on wire coil inserts' thermohydraulic behaviour. Heat transfer and pressure drop results from six wire coils of geometry: $1.17 < p/d < 2.68$, $0.07 < e/d < 0.10$ and $14 < p/e < 33$ are shown in laminar-transition-turbulent flow within a wide flow range: $150 < Re < 90{,}000$ and $2.8 < Pr < 150$. Nusselt number and isothermal friction factor correlations are proposed for turbulent regime in terms of Re, Pr and inserts' geometry. The wide range of Prandtl number has allowed to establish its influence on heat transfer. Finally, a performance evaluation criterion has been used in order to assess the real benefit which wire coil devices offer both in laminar and turbulent flows.

2. Experimental programme

2.1. Wire coils tested

The experimental study was carried out on 6 wire coils fitted in a smooth tube. Table 2 shows wire coils and smooth tube geometrical parameters. 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 setup was adjusted and verified through pressure drop and heat transfer experiments on the smooth tube. The same tube was used for every wire coil. The geometrical range which has been covered in the present work is compared to those from other experimental investigations in Fig. 4.

2.2. Test fluids

Water and two mixtures of propylene glycol and water (90% and 50% by weight) at temperatures 40 °C and 65 °C were used as test fluids. This allowed to extend the experimental range as well as to perform the experiments with the optimum level of accuracy which the experimental apparatus offered. Test fluids are listed in Table 3 along with the experimental flow range which has been covered.

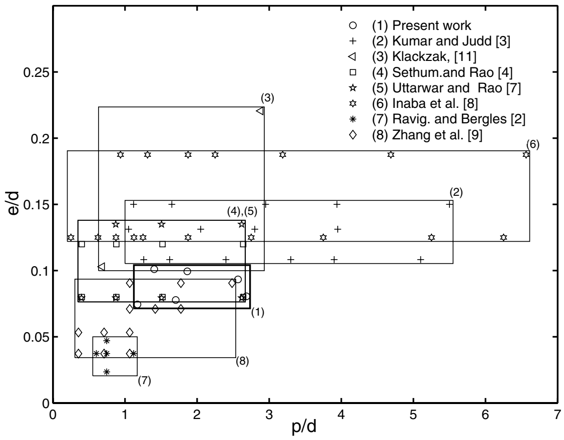

Fig. 4. Dimensionless pitch p/d vs. dimensionless wire-diameter e/d. Geometrical range of present work in comparison to other experimental papers.

Table 2
Characteristic dimensions of the helical wire coils

Wire coil no.	d [mm]	d_h [mm]	p [mm]	e [mm]	p/d [–]	e/d [–]	p/e [–]
Smooth tube	18.00	18.00	–	–	–	–	–
W01	18.00	14.61	21.12	1.34	1.17	0.074	15.76
W02	18.00	15.85	48.32	1.45	2.68	0.081	33.32
W03	18.00	15.28	30.66	1.40	1.70	0.077	21.90
W04	18.00	15.47	46.22	1.68	2.57	0.093	27.51
W05	18.00	14.77	33.57	1.79	1.86	0.099	18.75
W06	18.00	14.07	25.31	1.82	1.41	0.101	13.91

Table 3
Test fluids and flow range

Fluid	Temperature (°C)	*Re*	*Pr*
WT 100%	65	4000–90,000	2.8
WT 100%	40	1500–60,000	4.3
PG50%–WT50%	65	500–20,000	18
PG50%–WT50%	40	250–10,000	36
PG90%–WT10%	65	100–7000	75
PG90%–WT10%	40	80–3000	160

Water (WT) and mixtures of propylene glycol (PG) and water.

2.3. Experimental setup

A schematic diagram of the experimental setup is shown in Fig. 5. It consisted of two independent circuits: the main circuit where the wire coils were installed and the secondary circuit which was used for regulating the tank temperature to a desirable value. All the instrumentation was connected to a HP 34970A Data Acquisition Unit.

Heat transfer experiments were carried out under uniform heat flux conditions, where energy was added to the working fluid by Joule effect heating. A 6 kV A transformer was connected to the smooth tube by copper electrodes and power supply was regulated by means of an auto-transformer. The length between electrodes defined the heat transfer test section ($l_h = 1.37$ m). This section was insulated by an elastomeric thermal insulation material of 20 mm thickness and thermal conductivity 0.04 W/m K to minimize heat loses. 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 resistance temperature detector (RTDs). 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 located at 50 diameters from the upstream electrode. The value of $\overline{t_{wo}}$ was calculated by averaging the temperatures measured by using twelve surface type RTDs peripherically spaced by every 30 °C.

Two calibration tests with no electrical heating were done: the first test was carried to determine heat loses in the test section Q_l by measuring $(t_{in} - t_{out})$ at low flow rates, and the second test at high flow rates to calculate the lay-out resistances of the surface type RTDs ($t_{in} \approx t_{out} \approx \overline{t_{wo}}$).

Heat flux added to the test fluid q'' is 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

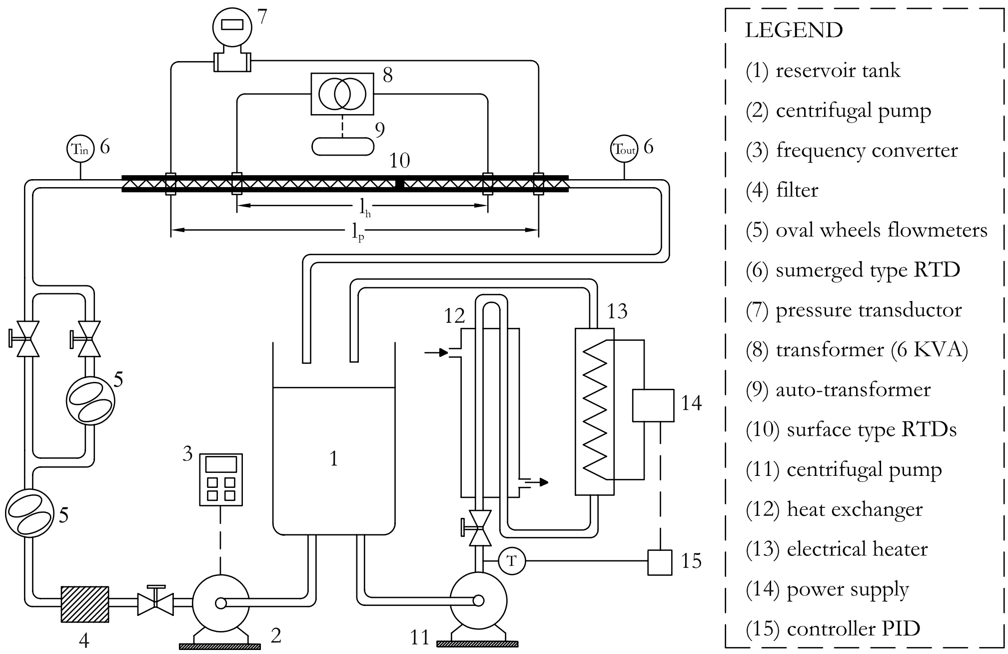

Fig. 5. Experimental setup.

tube wall and insulation 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

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

In turbulent flow, even for low Reynolds numbers, flow is fully developed at $x/d \approx 15$. The local Nusselt number was calculated at $x/d \approx 50$ and therefore this is the asymptotic Nusselt number. The entry region under turbulent flow is very small and in many practical applications it can be neglected. Therefore it can be assumed that Nusselt number given by Eq. (1) is approximately the mean Nusselt number. The Nusselt numbers calculated by Eq. (1) were corrected by the factor $(\mu_w/\mu_b)^{+0.14}$ [12] to obtain correlations free of 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

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

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 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 [13]. Details of the assignation of the uncertainty level to the experimental data is given by the authors in [16]. Uncertainty calculations based on a 95% confidence level showed maximum values of 4% for Reynolds number, 3.5% for Prandtl number, 4.5% for Rayleigh number, 4.5% for Nusselt number in turbulent flow, 6% for Nusselt number in laminar flow and 3% for friction factor.

3. Results and discussion

3.1. Pressure drop

Isothermal pressure drop experiments were carried out by employing water and water–propylene glycol mixtures. Fanning friction factors have been obtained in a continuous Reynolds number range from 100 to 90,000. Before performing experiments over the wire coils, the smooth empty tube was tested in order to adjust the experimental setup and check its uncertainties. Smooth tube friction factor results have been compared to the analytical solution ($f_s = 16Re^{-1}$) in the laminar region and to the widely known Blasius equation ($f_s = 0.079Re^{-0.25}$) in the turbulent region. The measurements deviation (<3%) was in accordance with the uncertainty analysis, and assured a proper instrumentation adjustment.

Results are shown in Fig. 6 for the 6 types of wire coils and the smooth tube in laminar, transition and turbulent flows. At Reynolds numbers below 500, friction

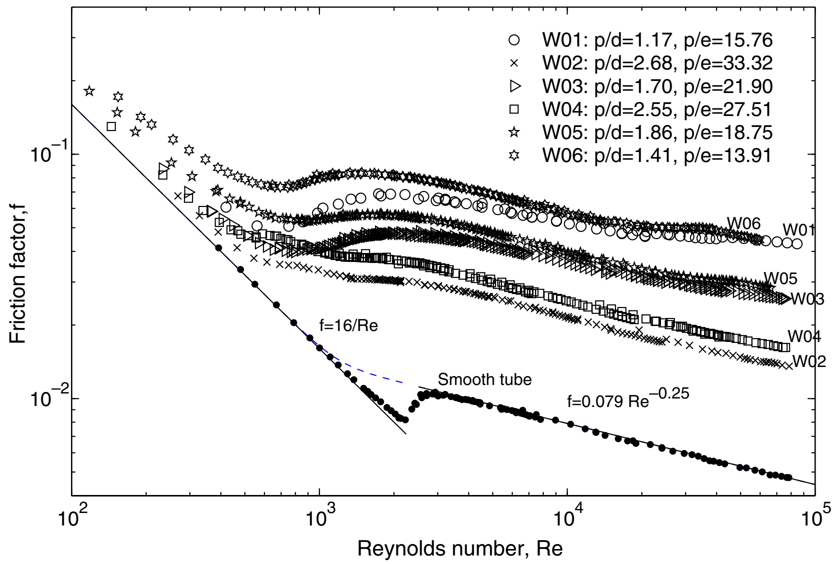

Fig. 6. Fanning friction factor vs. Reynolds number for wire coil inserts in laminar-transition-turbulent flow.

factor values are proportional to Re^{-1}, which means a pure laminar flow. Transition to turbulence regime occurs in a gradual way, in contrast to the abrupt transition observed in smooth tubes.

Friction factor experimental results from the 6 wires at Reynolds numbers from 2000 to 30,000 have been correlated by the following equation

$$f_a = 5.76(e/d)^{0.95}(p/d)^{-1.21}Re^{-0.217}. \tag{3}$$

Results are correlated with a similar accuracy if only pitch to wire-diameter ratio p/e is employed. The following correlation is finally proposed

$$f_a = 9.35(p/e)^{-1.16}Re^{-0.217}. \tag{4}$$

The regression equation's coefficients were assessed with the help of classical the least square method. Eq. (4) leads to a deviation of 8% for 95% of friction factor experimental data in the region: $Re = 2000$–30,000. At high Reynolds turbulent flow, ($Re = 30{,}000$–80,000), the use of this equation overpredicts the experimental values.

Fig. 7 shows friction factor augmentation (f_a/f_s) produced by the wire coils. For pure laminar flow ($Re < 500$), moderate friction factor augmentations between 1.2 and 1.8 are observed. For pure turbulent flows these values are much higher and at $Re = 80{,}000$ remain between 2.8 for W02 and 9 for W01.

An equation for friction factor augmentation for Reynolds numbers from 2000 to 30,000 has been obtained as the ratio between Eq. (4) and Blasius equation

$$f_a/f_s = 118.35(p/e)^{-1.16}Re^{0.033}. \tag{5}$$

Eq. (5) shows that pressure drop augmentation within the range $2000 < Re < 30{,}000$ depends mainly on pitch to wire-diameter ratio p/e.

3.1.1. Results discussion

This experimental work extends the currently available data on pressure drop produced by wire coils. Approximately 900 experimental friction factor data have been obtained for 6 wire coils of geometries p/d from 1.17 to 2.68 and e/d from 0.074 to 0.1 within a Reynolds number range from 100 to 90,000. Additionally a friction factor correlation has been proposed within the Reynolds number range from 2000 to 30,000 (Eq. (4)).

A review of the open literature shows that many authors do not correlate their experimental friction factor results. Besides, there is a significant discrepancy among the few available sources. Rabas [10] suggested as possible causes the vibrations of the coil and the tube and the looseness that sometimes exists between the coil and the tube wall.

Fig. 2 compares Eq. (4) to the correlations proposed by other sources in turbulent regime for a wire coil of geometry $p/d = 1.2$ and $e/d = 0.1$. The correlations by Sethumadhavan and Raja Rao [4], Ravigururajan and Bergles [2] and Inaba et al. [8] underpredict the results of the present work. Rabas [10] reviewed data from different open and private sources and concluded that the friction factor correlation by Ravigururajan and Bergles [2] underpredicted most of that data. Less discrepancies (around 25%) have been found with Kumar and Judd [3], who presented only graphical results. Finally Eq. (4) agrees with the results by Zhang et al. [9] whose wire

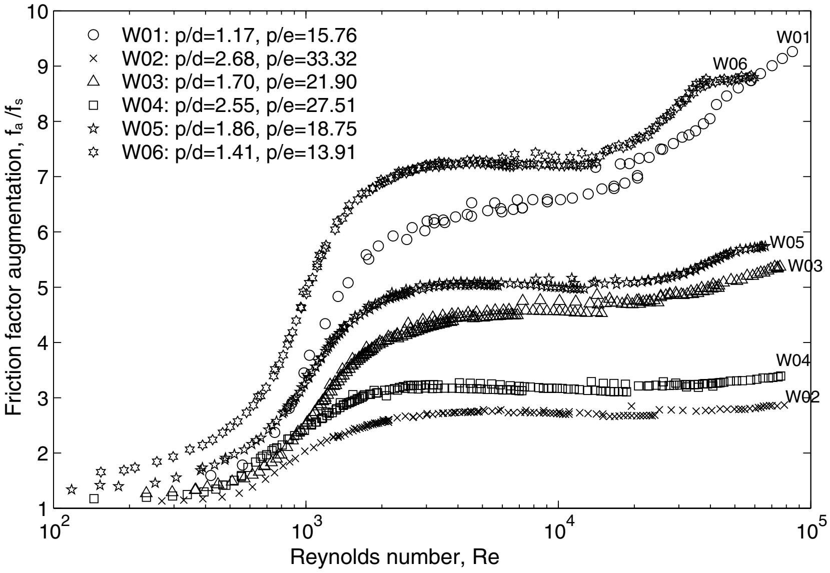

Fig. 7. Fanning friction factor augmentation vs. Reynolds number for wire coil inserts in laminar-transition-turbulent flow.

diameter range covers the one which has been studied in the present work.

Wire coils have been tested in the laminar and transition regimes in order to analyse the benefits that this technique offers for viscous fluids. For pure laminar flow ($Re < 500$) moderate friction factor augmentations are observed with values of f_a/f_s between 1.2 and 1.8. Fig. 6 shows that wire coils with the lowest hydraulic diameters d_h present the highest friction factors. This fact was also observed by Uttarwar and Raja Rao [7] and clearly suggests that wire coils fitted inside tubes behave mainly as a smooth tube in the laminar regime.

Transition to turbulent flow takes place at low Reynolds numbers ($Re \approx 700$) and in a gradual way. Fig. 6 shows that the critical Reynolds number is not clearly marked, which suggests that a swirl flow mechanism occurs: in twisted tapes, where a rotating component is clearly induced, transition from laminar to turbulent flow is produced steadily without being possible to determine the point where transition occurs [15]. This behaviour differs from what the authors found in corrugated and dimpled tubes, where sudden transitions were observed and the critical Reynolds numbers were defined by a local minimum in the friction factor curve (authors [14,16]).

3.2. Heat transfer

Heat transfer tests under uniform heat flux condition were carried out in a smooth empty tube and in the same tube with 6 wire coil inserts. Following the procedure described in Section 2, a wide range of flow conditions was covered: $Re = 80$–90,000 and $Pr = 2.8$–150.

The set of tests started with the empty smooth tube. These experiments allowed to check the experimental setup, to verify the procedure and to con. rm the calculated uncertainties. Fig. 8 shows Nusselt number results vs. Reynolds number for the smooth tube in laminar, transition and turbulent regimes. In laminar flow, heat transfer was produced under mixed convection. Local Nusselt numbers were measured in the fully developed region and therefore they depend only on Rayleigh number. Results at Reynolds number below 2300 are compared with Petukhov and Polyakov [17] equation

$$Nu_s = 4.36[1 + (Ra/18,000)^4]^{0.045}. \tag{6}$$

Experimental results agree to a great extend to Eq. (6). Heat transfer experiments were carried out at Rayleigh numbers between 2 and 3.5×10^7, which corresponds to $Nu_s = 15.4$ and 17 (horizontal lines in Fig. 8).

Turbulent Nusselt number results in Fig. 8 are compared to Gnielinski [18] equation,

$$Nu_s = \frac{(f_s/2)(Re - 1000)Pr}{1 + 12.7\sqrt{f_s/2}(Pr^{2/3} - 1)}. \tag{7}$$

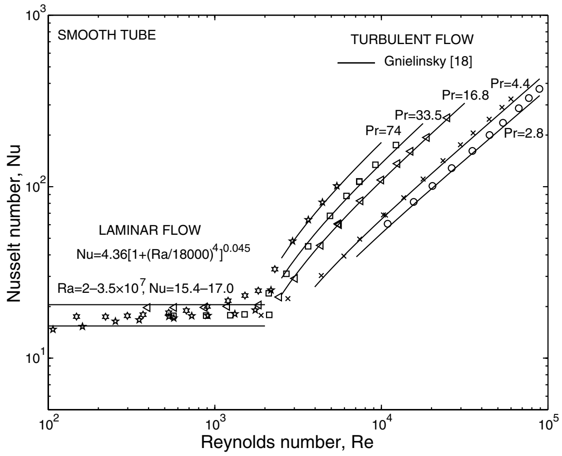

Fig. 8. Nusselt number vs. Reynolds number in laminar-transition-turbulent flow. Experimental smooth tube results compared with Petukhov [17] and Gnielinski [18] equations.

Experimental results are slightly higher than those predicted by Eq. (7) (1–6% above). In order to accurately determine the real heat transfer augmentation produced by the wire coils, the following equation is proposed:

$$Nu_s = 0.0147(Re - 1000)^{0.86}Pr^{0.39}. \tag{8}$$

This equation correlates 95% of the measured smooth tube Nusselt numbers within ±5% in the Reynolds number range from 2500 to 90,000 and Prandtl number range from 2.8 to 150.

Laminar, transitional and turbulent heat transfer experiments were carried out for the wire coil inserts described in Table 2. Approximately one hundred experimental points were taken for each wire coil in order to determine the influence of Re and Pr on Nu_a Fig. 9

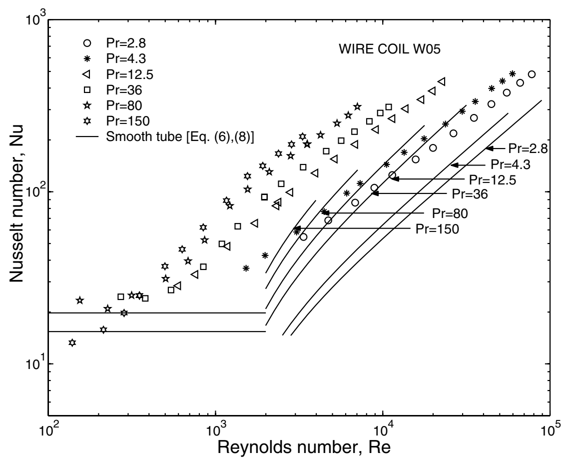

Fig. 9. Nusselt number vs. Reynolds number in laminar-transition-turbulent flow for wire coil W05.

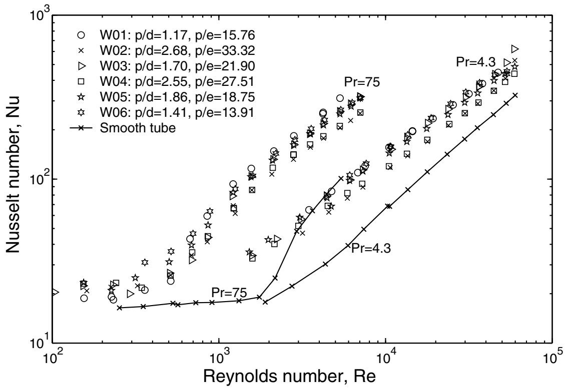

Fig. 10. Nusselt number vs. Reynolds number for wire coils W01–W06 at Prandtl numbers $Pr = 4.3$ and $Pr = 75$.

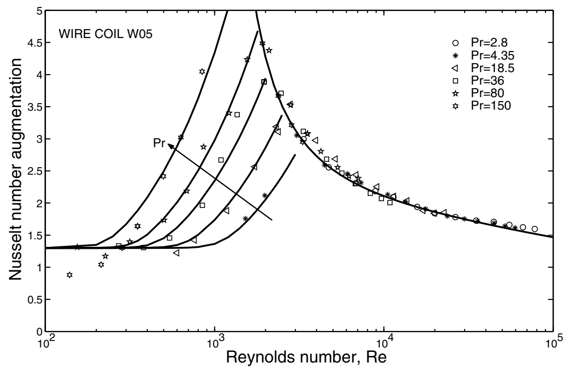

Fig. 11. Nusselt number augmentation vs. Reynolds number for wire coil W05.

shows Nusselt numbers measured for wire coil 05, as an example of the experimental work carried out on the six wire coils.

Heat transfer results for the whole set of wire coils are shown in Fig. 10. In order to avoid confusion, only Nusselt numbers at $Pr = 4.3$ and 75 are presented. In pure laminar regime ($Re < 500$), Nusselt numbers are close to smooth tube results under mixed convection. In fact, considerable circumferential temperature differences were measured in the tube wall, which indicates that the flow was affected by the buoyancy forces. A smooth transition to turbulent flow is observed and, at Reynolds numbers above 1000, Nusselt numbers show a tendency similar to a pure turbulent flow.

A Nusselt number equation in the form $Nu_a = Nu_a(Re, Pr, e/d, p/d)$ has been obtained through curve-fitting of heat transfer results for the six wire coils:

$$Nu_a = 0.303(e/d)^{0.12}(p/d)^{-0.377}Re^{0.72}Pr^{0.37}. \tag{9}$$

Eq. (9) shows that the wire-diameter makes a slight influence on heat transfer. Thus heat transfer in wire coil inserts is mainly influenced by the reduced pitch p/d. The following general equation correlates 95% of experimental data within a deviation of 9% within a Reynolds number range from 1700 to 80,000 and Prandtl number range from 2.5 to 170

$$Nu_a = 0.132(p/d)^{-0.372}Re^{0.72}Pr^{0.37}. \tag{10}$$

Nusselt number augmentation is defined by the ratio between Nu_a and Nu_s at the same Reynolds and Prandtl numbers. Fig. 11 shows Nu_a/Nu_s for wire coil 05. In the transition region ($500 < Re < 3000$), flow regime in a smooth tube is laminar and heat transfer depends neither on Reynolds nor on Prandtl number. However if a wire coil is inserted, flow becomes turbulent and Nusselt number depends on both non-dimensional numbers ($Nu \propto Re^{0.72}$ and $Nu \propto Pr^{0.37}$). Therefore, in the transition region, heat transfer augmentation depends strongly on Prandtl number. For $Re > 3000$ experimental measurements show that Prandtl number influence on heat transfer augmentation is negligible ($Nu_s \propto Pr^{0.39}$ and $Nu_a \propto Pr^{0.37}$).

Fig. 12 shows Nusselt number augmentation produced by the whole set of wire coils. In order to avoid confusion, only results at two different Prandtl numbers are plotted: for $Re < 4000$, Prandtl number is $Pr = 78$ and for $Re > 4000$, $Pr = 4.3$. There is no gap between the two series, which confirms that in turbulent flow heat transfer enhancement does not depend on Prandtl number. In the laminar region Nu_a/Nu_s has been evaluated by calculating smooth tube Nusselt number Nu_s through Eq. (6) at Rayleigh number $Ra = 3.5 \times 10^7$, which corresponds to $Nu_s = 17$. At $Re < 500$ values of Nu_a/Nu_s below 1.4 are always observed. At Reynolds numbers $Re > 500$, wire coils produce considerable heat transfer enhancement and those with the lowest pitch to wire-diameter ratio p/e show the highest Nusselt number augmentations. At turbulent flow, Nu_a/Nu_s decreases with

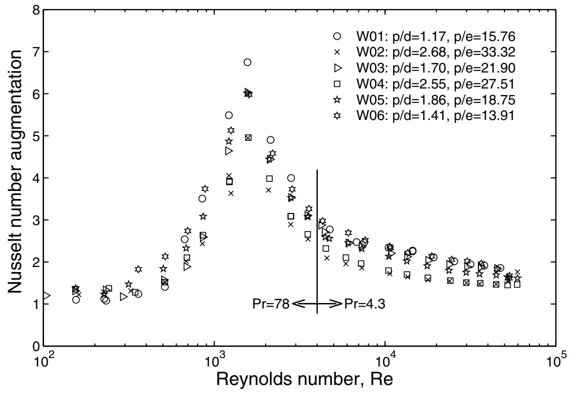

Fig. 12. Nusselt number augmentation vs. Reynolds number for wire coils W01–W06 at Prandtl numbers $Pr = 4.3$ and $Pr = 75$.

Reynolds number: $Nu_a/Nu_s = 2$–2.8 at $Re = 5000$ and $Nu_a/Nu_s = 1.5$–2.1 at $Re = 20{,}000$. In this region, wire coils with the lowest dimensionless pitch p/d show the highest heat transfer augmentation.

3.2.1. Result discussion

Results have shown that wire coils hardly provide any enhancement in pure laminar flow ($Nu_a/Nu_s < 1.4$). This was also observed by Inaba et al. [8], who performed heat transfer experiments in presence of buoyancy forces. The heat transfer experiments carried out in the present work were affected by natural convection. Therefore augmented Nusselt numbers have been compared to the smooth tube Nusselt number under mixed convection. Uttarwar and Raja Rao [7] obtained higher Nusselt number augmentations. However, they estimated that the natural convection contribution to heat transfer was negligible and they compared their results to the Sieder–Tate [19] smooth tube equation. Besides, their results were affected by the entry region and their conclusions should be carefully considered.

Since wire coils accelerate transition to turbulent flow, they show a considerable heat transfer increase within the low Reynolds number region ($500 < Re < 2000$). Here, the flow in a smooth tube would be laminar while in wire coils it is turbulent, which yields high Nusselt number augmentations (Fig. 10).

It can be stated that in wire coils transition from laminar to turbulent flow takes place steadily. This produces continuous curves of friction factor f and Nusselt number Nu vs. Reynolds number (Figs. 6 and 10). When transition from laminar to turbulent regime occurs within a plain tube heat exchanger, it is impossible to predict where transition arises, and therefore the performance of the heat exchanger can not be determined accurately. If wire coils are used, the design equations in the transition regime are continuous and therefore the behaviour can be predicted correctly. Therefore, wire coils have a ''predictable'' behaviour in the transition region, which implies a considerable advantage over other enhancement techniques.

For turbulent flow experimental results have shown that heat transfer augmentation decreases with Reynolds number. The values of Nu_a/Nu_s are similar to those found by Sethumadhavan and Raja Rao [4], Kumar and Judd [3] and Zhang et al. [9]. Fig. 3 plots Eq. (9) for a wire of $e/d = 0.1$ and $p/d = 1.2$ at Prandtl numbers 6 and 150 in comparison to the correlations by other sources. At $Pr = 6$, results from Eq. (9) are very close to those predicted by Kumar and Judd [3] and agree with the results from Sethumadhavan and Raja Rao [4] and Zhang et al. [9]. At $Pr = 150$ Eq. (9) agrees with Kumar and Judd [3] (although they only used water as test fluid) but underpredicts results by Sethumadhavan and Raja Rao [4]. That work and the present one are the only that have tried to determine Prandtl number influence on heat transfer. These works have reached different conclusions: Sethumadhavan and Raja Rao obtained $Nu \propto Pr^{0.45}$ whereas this work proposes $Nu \propto Pr^{0.37}$. The other works only used water or air as test fluids. Therefore the use of their correlations is not recommended at high Prandtl numbers.

Finally, results have shown that dimensionless wire-diameter e/d exerts a negligible influence on heat transfer in turbulent flow. This was also reported by Sethumadhavan and Raja Rao [4] and Kumar and Judd [3] within a similar e/d range. However it is expected, as suggested by Webb [1], that wire diameter makes an influence on heat transfer at low values of e/d. Zhang et al. [9] studied wire coils with e/d as low as 0.04 and concluded that $Nu \propto (e/d)^{0.372}$. The present work has found a slight dependence of heat transfer on e/d and it has been decided to correlate augmented Nusselt number only in terms of dimensionless pitch p/d (Eq. (10)).

4. Performance evaluation

Bergles et al. [20] and Webb [21] proposed several performance criteria to evaluate the thermohydraulic performance of the enhancement techniques. In this paper, criterion $R3$ outlined by Bergles et al. [20] has been calculated to quantify the benefits from wire coil inserts.

The criterion $R3$ is defined by $R3 = Nu_a/Nu_0$ where Nu_a is the heat transfer obtained with the wire coils and Nu_0 is the heat transfer obtained with a smooth tube for equal pumping power and heat exchange surface area (in wire coils $A_a = A_0$. To satisfy the constraint of equal pumping power, Nu_0 is evaluated at the equivalent smooth tube Reynolds number Re_0, which matches

$$f_a Re_a^3 = f_0 Re_0^3 \;\Rightarrow\; Re_0^3 = \frac{f_a Re_a^3}{f_0}. \tag{11}$$

Fig. 13 shows the performance parameter $R3$ for the six wire coil inserts at Prandtl number $Pr = 7$. In

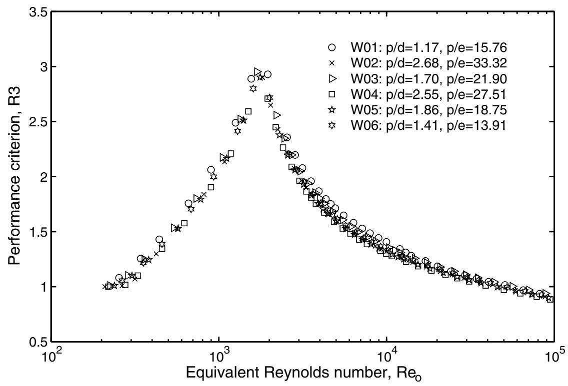

Fig. 13. Performance evaluation criterion $R3$ vs. equivalent smooth tube Reynolds number Re_0 for wire coils W01–W06.

laminar flow, wire coils show higher enhancements at higher Prandtl numbers. However heat transfer enhancement does not depend on Prandtl number in turbulent flow. In laminar regime the performance improves with Reynolds number. At $Re = 500$ heat transfer enhancements ranges from 30% to 40% ($R3 = 1.3$–1.4). At $Re = 1500$ heat transfer enhancements from 140% to 160% ($R3 = 2.4$–2.6) are obtained when wire coils are inserted in a smooth tube. On the contrary, the performance worsens with Reynolds number in turbulent flow. At $Re = 4000$, $R3 = 1.6$–1.7 and at $Re = 10{,}000$, $R3 = 1.3$–1.4. At $Re > 30{,}000$, the analysis allows to state that wire coils are not advantageous under $R3$ criterion. This fact differs from Sethumadhavan and Raja Rao [4] results. They observed $R3$ values that ranged between 1.15 and 1.85, depending on wire geometry but not on Reynolds number.

5. Conclusions

(1) A comprehensive experimental study has been carried out on six wire coils inserted in a smooth tube, covering the laminar, transition and turbulent regimes: $Re = 80$–$90{,}000$ and $Pr = 2.8$–150. General correlations have been proposed for Fanning friction factor and Nusselt number as functions of wire geometry and flow conditions.

(2) In laminar flow, results show that wire coils behave mainly as a smooth tube. Transition to turbulent flow takes place at low Reynolds numbers ($Re \approx 700$) and in a gradual way. Wire coils have a predictable behaviour within the transition region since they show continuous curves of friction factor and Nusselt number, which involves a considerable advantage over other enhancement techniques.

(3) In turbulent flow, wire coils cause a high pressure drop increase which depends mainly on pitch to wire-diameter ratio p/e. These show considerable heat transfer augmentations: at $Re = 10{,}000$, $Nu_a/Nu_a = 1.7$–2.5, depending mainly on dimensionless pitch p/d. However Nusselt number augmentation decreases rapidly with Reynolds number. For pure turbulent flow, it can be stated that Prandtl number does not exert an influence on heat transfer augmentation. On the contrary, when working with high Prandtl number fluids within the transition region, wire coils produce the highest heat transfer increase.

(4) If wire coils are compared with a smooth tube at constant pumping power, an increase in heat transfer rate is obtained at Reynolds numbers below 30,000. Although large differences have been observed among the six analysed wire coils, their performance evaluated under criterion $R3$ is quite similar: at $Re = 4000$, $R3 = 1.6$–1.7 and at $Re = 10{,}000$, $R3 = 1.3$–1.4. The best operating regime is found at the transition region where heat transfer enhancement calculated by $R3$ reaches 200%.

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