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HEAT TRANSFER ENHANCEMENT IN SMOOTH TUBE WITH WIRE COIL INSERT IN LAMINAR AND TRANSITIONAL NON-NEWTONIAN FLOW

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ABSTRACT

This work presents an experimental study on the heat transfer enhancement by means of a tube with wire-coil insert, for non-Newtonian laminar and transitional flow. The dimensionless pitch and wire diameter (based on the plain tube inner diameter) were chosen as p/D= 1 and e/D=0.09. Two pseudoplastic test fluids have been used: 1% by weight aqueous solutions of carboxymethyl cellulose (CMC) with high viscosity and medium viscosity. A wide range of flow conditions has been covered: Reynolds number from 10 to 1200 and Prandtl number from 150 to 1900. Isothermal pressure drop tests and heat transfer experiments under uniform heat flux conditions were performed on both the plain tube and the tube with wire-coil. The plain tube results have shown an excellent agreement with the available data. At Reynolds numbers below Re=400, the wire-coil insert has no effect in heat transfer. However, it becomes more effective as turbulence is established: at Re=1200, the wire-coil increases the heat transfer rate by 350%, with a pressure drop increase of 250%. At Reynolds number Re=1800, if a wire-coil is inserted in a smooth tube the heat transfer rate will be increased by 300% with no increase in pumping power.

INTRODUCTION

Heat transfer process of high-viscous fluids is commonly encountered in the chemical, food, pharmaceutical or petroleum industries. Due to its nature, most of these fluids are processed in the laminar or transitional regimes, where transfer rates are particularly low. Here, the various forms of insert devices are effective to enhance the heat transfer performance of tubular heat exchangers.

The dominant literature (Bergles [1], Webb and Kim [2]) usually mentions five devices: wire coils, twisted tapes, extended surface devices, mesh inserts and displaced elements. The main advantage of inserts, with respect to other enhancement techniques, is that they allow an easy installation in an existing exchanger of smooth tubes. Due mainly to its low cost, the insert devices which are most frequently used in engineering applications are wire coils and twisted tapes.

The difficulties for carrying out experimental studies on heat transfer in laminar flow is well known (Bergles [3]), as this flow is sensitive to entry length effects, the thermal boundary condition and the buoyancy forces effect. Moreover, in many cases a highly-viscous fluid exhibits a non-Newtonian behaviour; which implies that an appropriate rheological characterization needs to be done prior to the experimentation. Another problem that can arise when dealing with non-Newtonian fluids is thixotropy, where their apparent viscosity changes along the time of experimentation.

Consequently, few experimental studies have been reported on the enhancement of non Newtonian laminar heat transfer. Nazmeev [4] studied the enhancement of pseudoplastic fluid flows using twisted-tape inserts. He found a substantial increase in heat transfer (50-300%), similar to that observed in the studies with Newtonian fluids carried out by Hong and Bergles [5] and Marner and Bergles [6]. Manglik et al. [7] extended the available experimental data by performing new experiments with different tapes, working with aqueous solutions of Cellulose ether powder. More recently, Patil [8] presented an experimental investigation of heat transfer and flow friction of a power-law fluid in tubes with twisted tape inserts.

Wire coil inserts are an alternative to twisted tapes and other insert devices for heat transfer enhancement at moderate Reynolds numbers. However, very few authors have studied wire coils in non Newtonian flow so far. An early work from Igumentsev and Nazmeev [9] studied the effect of wire coils in the intensification of convective heat exchange, for anomalously viscous liquids (aqueous solutions of sodium carboxymethyl cellulose). However, the authors did not inform about the values of wire diameters, friction factor and Nusselt number results.

Oliver and Shoji [10] tested three types of insert devices (twisted tape, Cal Gavin patent wire matrix and wire-coil) with aqueous solutions of sodium carboxymethyl cellulose (SCMC) used as test fluids. Measurements of both isothermal pressure drop and heat transfer at constant wall temperature were made. They also gave attention to the degradation with time (thixotropy) of the pseudoplastic test fluid.

This work presents an experimental study on a wire coil inserted in a smooth tube using non-Newtonian fluids, in order to characterize its thermohydraulic behaviour in laminar and transition flow. By using different qualities (high and medium viscosity) of CMC (Carboxyl-methyl-cellulose) solution in water at several temperatures (taking into account the thixotropic effects), a wide range of flow conditions has been covered. This range of flow conditions were previously measured on the empty smooth tube, and compared to the Bird [11] and Mahalingam [12] solutions. The ranges of the investigated experimental variables were:

<u>1.0 % CMC high viscosity solution</u>
Heat flux: 7,320-37,541 W/m^2
Reynolds number, Re_{MR}: 8-400
Prandtl number, Pr: 490-1900
n values: 0.41 (25ºC)-0.43(45ºC)
Shear rate, $\dot{\gamma}$=87-778 s^{-1}
$\dot{m}$=150-1400 Kg/h

<u>1.0 % CMC medium viscosity solution</u>
Heat Flux: 7,936-66,463 W/m^2
Reynolds number, Re_{MR}: 58-1230
Prandtl number, Pr: 150-380
n values: 0.86 (25ºC)-0.94(45ºC)
Shear rate, $\dot{\gamma}$=74-700 s^{-1}
$\dot{m}$ = 150-1400 Kg/h

EXPERIMENTAL PROCEDURE

Geometric dimensions of the wire-coil insert

Fig. 1 shows a sketch of a wire coil inserted in a smooth 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*. In the present work, the wire coil was made of steel and covered with an insulating coating to prevent electrical conduction. The wire-diameter was *e*=1.62 mm and the pitch was *p*=18.4 mm (*p/D* =1.02, *e/D* = 0.09).

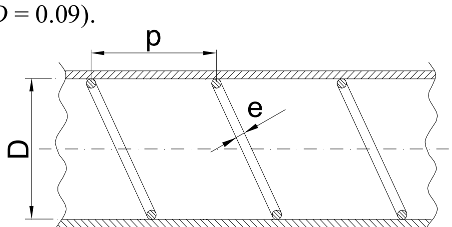

Figure 1 Geometric scheme of the wire coil inserted in a smooth tube.

Experimental set-up

The schematic diagram of the experimental setup is shown in Fig. 2. It consists of two independent circuits: the main circuit, which was filled with the pseudoplastic test fluid, and a secondary circuit with chilled water that was used for regulating the tank temperature to a desirable value. The test section was placed in the main circuit and consisted in 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. Two oversized, low-velocity gear pumps (one on each circuit) were used for circulating the working fluid, in order to minimize the fluid degradation during the tests. Mass flow rate was measured by a Coriolis flow meter.

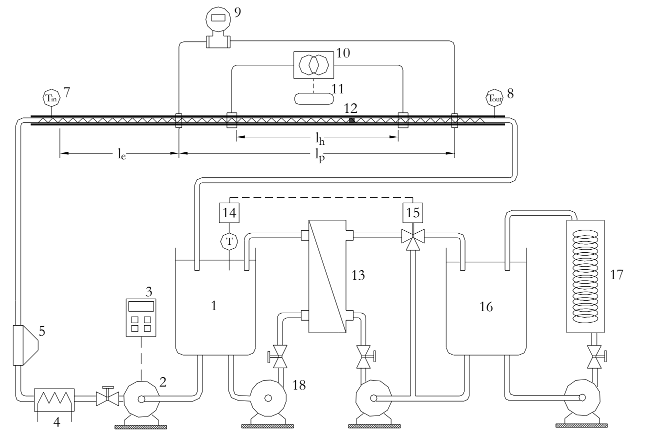

Figure 2 Schematic diagram of the experimental setup: (1) working fluid, (2, 18) gear pumps, (3) frequency converter, (4) immersion resistances, (5) Coriolis flowmeter, (7, 8) inlet and outlet immersion RTDs, (9) pressure transmitter, (10) electrical transformer, (11) autotransformer, (12) surface thermocouples (13) plate heat exchanger, (14) PID, (15) three-way valve, (16) cooling fluid, (17) cooling machine.

Pressure drop experiments were carried out in the hydro-dynamically developed region under isothermal conditions. Four pressure taps separated by 90º were coupled to each end of the pressure test section (L_p=1.85 m). Pressure drop ΔP was measured by means of a highly accurate pressure transmitter. The test section was preceded by 2 m of smooth tube with wire-coil insert, in order to ensure fully developed flow conditions. The fanning friction factor *f* was calculated from measurements of mean pressure drop and fluid mass flow rate as:

$$f = \frac{\Delta P}{L_p}\frac{D}{2\rho\bar{u}^2} = \frac{\Delta P}{L_p}\frac{\rho\pi^2 D^5}{32\dot{m}^2}. \qquad (1)$$

Heat transfer experiments were carried out under uniform heat flux (UHF) conditions, with hydrodynamically developed flow at the test section entry. Energy was added to the working fluid by Joule effect heating. A 6 kVA 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=1.49 m). The test section 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 power input added to the heating section was calculated by measuring the voltage between electrodes (0-15 V) and the electrical current (0-600 A). The heat input added to the test fluid, Q, was estimated after correcting the electrical power for heat losses through the outer wall. Fluid inlet and outlet temperatures T_{in} and T_{out} were measured by immersion RTD sensors. The axial position of the measuring point, x, was defined from the upstream electrode. In the present work it was fixed at a distance L_x=1.02 m. Since heat was added uniformly along the tube length, the bulk temperature of the fluid at the measuring section, T_b(x), was calculated by assuming a linear variation of mean fluid temperature with axial direction. The outside wall temperature at the measuring section, T_{wo}, was measured. At that location, eight surface thermocouples were placed 45º apart, circumferentially, and electrically insulated from the tube; T_{wo} was estimated by averaging the eight wall-temperature readings. The local Nusselt number was calculated as:

$$Nu_x = \frac{D}{k}\frac{q''}{T_{wi} - T_x}, \tag{2}$$

where q" stands for the heat flux at the inner wall and T_{wi} is the inner wall temperature, that was obtained from the numerical solution of the radial, 1D heat conduction across the tube and insulation.

Further details of the working apparatus and the calibration procedure are given in Garcia et al. [13]. The experimental uncertainly was calculated by following the "Guide to the expression of uncertainty in measurement", published by ISO [14]. Details of the uncertainty assignation to the experimental data are given by the authors in Vicente et al [15]. Uncertainty calculations based on a 95% confidence level showed maximum values of 3% for Reynolds number, 3% for Graetz number, 3% for Nusselt number and 4% for friction factor.

Test fluid characteristics

The non-Newtonian test fluids were 1% by weight aqueous solutions of carboxymethyl cellulose (CMC). CMC was supplied by Sigma-Aldrich Co. Two different test fluids were obtained by using two types of CMC (medium-viscosity and high-viscosity grade). The solutions were prepared by dissolving the polymer powder in distilled water and then raising the pH values of the solution to increase viscosity. All the fluid properties except the rheological parameters were assumed to be the same as pure water. Solutions of CMC in water at low concentrations are pseudoplastic in nature, and their constitutive relationship can be expressed as (Chabbra et al [16]):

$$\tau_W = K\left[\frac{8u}{D}\left(\frac{3n+1}{4n}\right)\right]^n, \tag{3}$$

where τ_w is the wall shear stress, K the flow consistency index, n the flow behavior index and ($8u/D$) is the velocity gradient at the wall for Newtonian fluids in fully developed laminar flow.

The values of n and K for the test fluids were obtained by using the smooth tube as a viscometer. The parameter ($8u/D$) was calculated from fluid flow rate, and τ_w was computed from isothermal pressure drop measurements at 25ºC, 45ºC and 60ºC by means of:

$$\tau_W = \frac{\Delta P}{L_p}\frac{D}{4}. \tag{4}$$

Fig. 3 shows the flow curves (τ_W vs. $8u/D$) for the test fluids used in the smooth-tube experiments. Fresh CMC solutions were prepared to be used in the wire coil experiments. The values of n and K for the wire-coil and the smooth-tube experiments are listed in Table 1. The uncertainty calculations based on a 95% confidence level have shown maximum values of 0.2% for n and 3% for K.

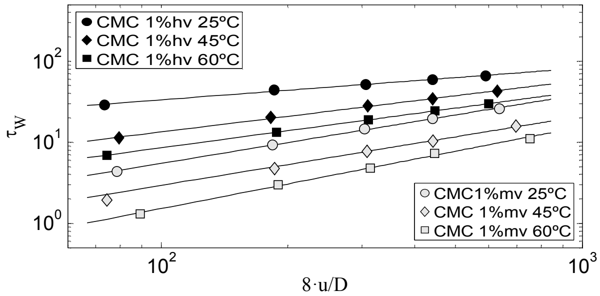

Figure 3 Flow curves for the CMC solutions in the smooth tube experiments.

		CMC 1.0% medium viscosity		CMC 1.0% high viscosity	
Test	Temp. ºC	n	K	n	K
Smooth tube	25	0.86	0.10	0.39	4.82
	45	0.94	0.04	0.41	2.91
	60	1.01	0.01	0.44	2.72
Wire coil	25	0.85	0.11	0.38	4.83
	45	0.96	0.05	0.42	2.93
	60	1.02	0.02	0.45	2.71

Table 1 Values of n and K for the test fluid used for wire coil tests and smooth tube tests.

Experimental details

The aqueous solutions of CMC degrade with shear and temperature because of breakage of polymer chains (thixotropy). In this work, the experimental program was designed to shorten the time of testing. The tests for a given geometry (plain tube or tube with wire-coil) with one type of aqueous solution of CMC (CMC high viscosity or medium viscosity grade) took about 450 minutes. A typical measurement cycle in a tube with wire coil consisted of: 1-Pressure drop and flow-rate measurements at T_b=25ºC (plain tube), 2-Pressure drop test at T_b=25ºC, 3-Heat transfer test at T_b(x)=25º, 4-Pressure drop and flow-rate measurements at T_b=45ºC (plain tube), 5-Pressure drop test at T_b=45ºC, 6-Heat transfer test at T_b(x)=45ºC, 7-Pressure drop and flow-rate measurements at T_b=60ºC (plain tube), 8-Pressure drop and flow-rate measurements at T_b=25ºC (plain tube). After a complete measurement cycle was performed, the test fluid was replaced with a fresh one.

The plain tube results (tests 1, 4, and 7) were processed to obtain the rheological fluid properties, K and n, as described in the preceding section. These properties were used to process the wire-coil results. In the heat transfer experiments, the power input was adjusted to control the wall temperature at the measuring section, $T_w(x)$. In the tests at $T_b(x)$=25ºC, $T_w(x)$ was fixed at 45ºC, and at $T_b(x)$=25ºC, $T_w(x)$ was 60ºC. Thus, in the data processing, the values of n, K and K_W were completely known. The difference between the values of K and n obtained from tests 1 and 8 (performed at a same temperature in an interval of about 450 minutes) give insight into how much the fluid degrades with time. Table 2 shows the values of K and n at the beginning and the end of each measurement cycle. A maximum fall of 38.9% in the value of K and 6.9% in the value of n is observed for the worst case (wire coil with CMC-mv). The arithmetic averaging procedure proposed by Joshi and Bergles [17] was employed, in order to estimate the concrete values of K and n for a given time.

		CMC 1.0% medium viscosity		CMC 1.0% high viscosity	
Test	Measure	n	K	n	K
Smooth tube	Initial	0.86	0.10	0.39	4.82
	Final	0.92	0.06	0.43	3.89
	Variation	7.0%	40.0%	10.3%	19.3%
Wire coil	Initial	0.85	0.11	0.38	4.83
	Final	0.92	0.06	0.43	3.89
	Variation	10.5%	50.0%	15.4%	20.3%

Table 2 Values of n and K for the test fluids, at the beginning and the end of 25ºC experiments.

RESULTS AND DISCUSSION

Friction factor results

The friction factor results for the plain tube and the wire-coil insert are presented in Fig. 4. The Reynolds number proposed by Metzner and Reed [18] for non-Newtonian power-law fluids has been used:

$$\mathrm{Re}_{MR} = \frac{8^{1-n} D^n u^{2-n} \rho}{K\left[\dfrac{3 \cdot n+1}{4n}\right]^n}, \qquad (5)$$

which for laminar flow in a smooth tube is related to the friction factor in the same way as is for Newtonian fluids ($f=16/Re_{MR}$). The smooth tube results shown in Fig. 1 are in excellent agreement with the analytical solution, with a maximum deviation of ±1.57%.

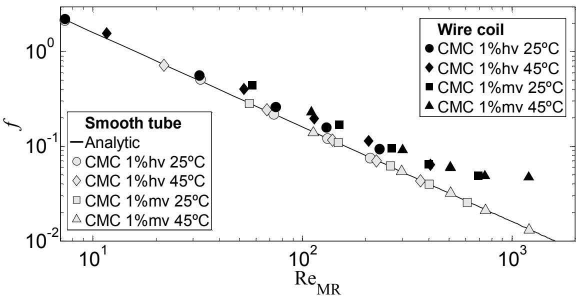

Figure 4 Fanning friction factor results. Smooth tube and wire-coil insert.

Fig. 5 shows the increase in friction factor f_w/f_s vs. Reynolds number Re_{MR}. The wire-coil produces a moderate increase in pressure drop at Reynolds numbers below 400 (f_w/f_s=1.5). Above Re_{MR}≈400, the wire-coil produces a perceptible pressure drop increase and at Re_{MR}≈1200, the friction factor with the insert is 3.5 times the plain tube value.

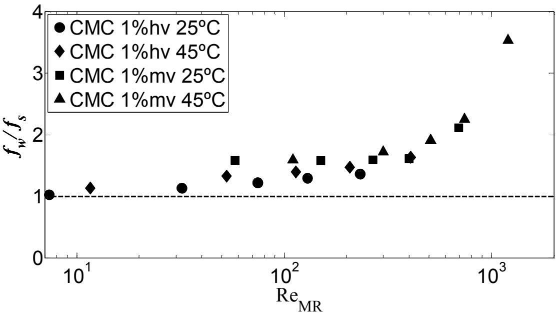

Figure 5 Fanning friction increase *fw/fs* vs. Reynolds number.

Nusselt number results

The heat transfer results for the smooth tube have been compared with the Bird et al's correlation [11] for forced convection heat transfer in laminar flow:

$$Nu = 1.41\Delta^{1/3} Gz, \qquad (6)$$

where Gz is the Graetz number:

$$Gz = \frac{\dot{m} c_p}{k L_x}, \qquad (7)$$

and the term Δ is expressed as:

$$\Delta = \frac{3n+1}{4n}. \qquad (8)$$

Both the heat transfer measurements in the plain tube and the tube with wire-coil have been corrected by factor $\Delta^{1/3}$, in order to show the results free of non-Newtonian effects. A consistency index correction has also been applied to account for radial temperature variation, according to Joshi and Bergles [17]:

$$\left(K_b / K_w\right)^{0.58-0.44n}, \qquad (9)$$

where K_b and K_w are the fluid consistency index evaluated at bulk and wall temperature, respectively. As mentioned in the "Experimental set-up" section, the outside wall temperature at the measuring point, T_{wo}, was measured with eight surface thermocouples placed 45º apart. This arrangement allowed to determine if a circumferential temperature variation existed, which would suggest a mixed convection heat transfer mode. Within the experimental range covered, no effect of buoyancy forces on heat transfer has been observed.

The heat transfer experimental data for both the plain tube and the tube with wire-coil insert is presented in Fig. 6, in terms of local Nusselt number, Nu, vs. Graetz number, Gz.

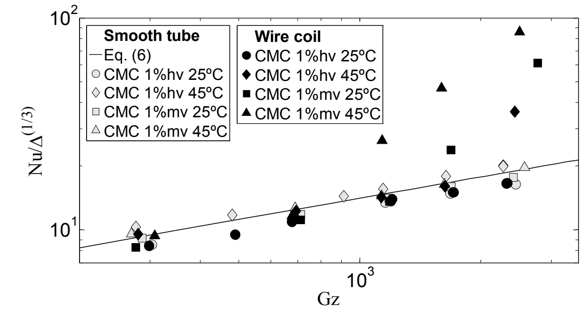

Figure 6 Heat transfer results for the plain tube and the wire coil. Nusselt number vs. Graetz number

As shown in Fig. 6, the Nusselt number results for the plain tube are in good agreement with Bird et al's correlation, with a deviation of ±6.52% for 95% of data.

Fig. 7 shows Nusselt number vs. Reynolds number for the plain tube and the tube with wire-coil insert. The four reference lines shown in Fig. 7 correspond to the solution of Eq. 6 at 25ºC and 45ºC, for each working fluid (CMC-hv and CMC-mv). At low Reynolds numbers, the wire coil has no effect in heat transfer. However, it becomes more effective as turbulence is established. This fact is more clearly seen if the heat transfer results are processed in terms of Nu_w/Nu_s ratio, which relates the heat transfer coefficient in the tube with wire-coil insert and in the plain tube, at a same Reynolds Number. Fig. 8 shows Nu_w/Nu_s ratio vs. Reynolds number. Above Re_{MR}≈400, the Nusselt number results for the wire-coil are significantly higher than the plain tube results. The Nu_w/Nu_s ratio increase with Reynolds number; a maximum value of Nu_w/Nu_s=4.5 is observed at Re_{MR}=1200.

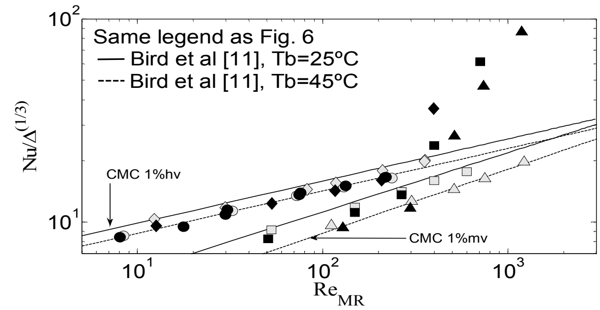

Figure 7 Heat transfer results for the plain tube and the wire coil. Nusselt number vs. Reynolds number.

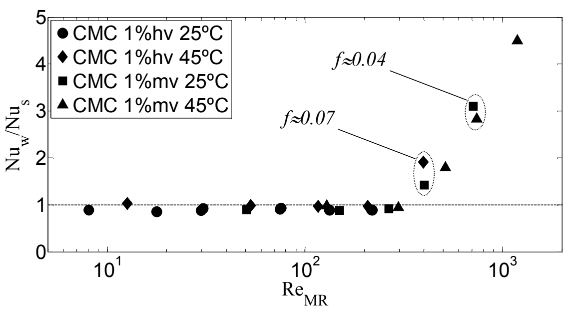

Figure 8 Nusselt number augmentation, Nu_w/Nu_s, vs. Reynolds number.

Fig. 8 also suggest, along with the friction factor results shown in Fig. 4, that the wire coil enhances more effectively heat transfer for fluids with a marked non-Newtonian behavior. At Re_s≈500 and Re_s≈700, the Nusselt number augmentation is higher for the fluid with a lower flow behavior index, but there are not differences in the values of friction factor. This highlights the need for more experimental data to confirm this trend.

Performance evaluation

The Criterion *R3* outlined by Bergles et al. [19] has been calculated to quantify the benefits from the wire coil insert. This Criterion yields the heat transfer augmentation (Nu_w/Nu_s) when the wire-coil is inserted in a smooth tube, for equal pumping power and heat exchange surface area. To satisfy the constraint of equal pumping power, Nu_s is evaluated at the equivalent smooth tube Reynolds number Re_s, which matches:

$$\mathrm{Re}_s^{\,3} = \frac{f_w}{f_s}\mathrm{Re}_{MR}\,. \tag{10}$$

Fig. 9 shows the results of R3 vs. the equivalent smooth tube Reynolds number, Re_s. The wire-coil insert has a poor performance at low Reynolds numbers (Re_s<400), where it adversely affect heat transfer (R3<1). However, the insert rapidly becomes more effective as the flow becomes turbulent. At Re_s≈700, R3 reaches a value of 1.7, whereas at Re_s≈1800, the R3 value rises to 4. This means that at Re_s≈1800 the heat transfer rate will be increased by 300% if the wire-coil is inserted in the smooth tube.

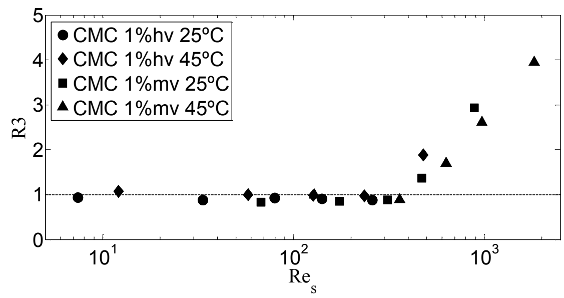

Figure 9 R3 Performance evaluation vs equivalent smooth tube Reynolds number

CONCLUSIONS

1. Isothermal pressure drop tests and heat transfer experiments under UHF for non-Newtonian flow have been performed with a wire coil inserted in a smooth tube. A wide range of flow range has been covered: Re_{MR}=10-1200 and Pr=150 to 1900. Two pseudoplastic test fluids have been used: 1% by weight aqueous solutions of CMC with high viscosity and medium viscosity. The wire coil geometrical parameters have been chosen as: p/D= 1 and e/D=0.09.

2. The experimental set-up has been used as a viscometer, in order to obtain the rheological parameters for the test fluids. The problem of thixotropy for the CMC solutions has been successfully solved by shortening the time of testing. The

smooth tube results have shown an excellent agreement with the available correlations (±1.57% for the friction factor and ±6.52% for the Nusselt number).

3. The wire-coil insert produces a moderate increase (50%) in pressure drop at Reynolds numbers below 400. Above Re_{MR}≈400, the wire-coil produces a perceptible pressure drop increase and at Re_{MR}≈1200, the friction factor with the insert is 3.5 times the plain tube value. With respect to Heat transfer, at low Reynolds numbers (below Re_{MR}≈400) the wire coil has no effect in heat transfer. However, it becomes more effective as turbulence is established: at Re_{MR}≈1200, the Nusselt number with the wire-coil is 4.5 times the smooth tube value.

4. According to the Criterion R3 outlined by Bergles et al. [19], the wire-coil insert has a poor performance at low Reynolds numbers, where it adversely affect heat transfer (R3<1). However, the insert becomes effective at Reynolds numbers above 400. At Re≈700, R3 reaches a value of 1.7, whereas at Res≈1800, the R3 value rises to 4. This means that at Res≈1800 the heat transfer rate will be increased by 300% if the wire-coil is inserted in the smooth tube.

NOMENCLATURE

D	[mm]	Tube inside diameter
e	[mm]	Wire diameter
f	[-]	Fanning friction factor
k	[W/mK]	Thermal conductivity
K	[Ns^n/m^2]	Flow consistency index
L	[m]	Tube length
$\dot{m}$	[kg/s]	Mass flow rate
n	[-]	Flow behaviour index
p	[mm]	Wire pitch
ΔP	[N/m^2]	Pressure drop
q''	[W/m^2]	Heat flux
R	[m^2K/W]	Interfacial thermal resistance
T	[ºC]	Temperature
ΔT	[ºC]	Temperature difference
u	[m/s]	Average velocity

Special characters

Δ	[-]	$(3n+1)/4n$
$\dot{\gamma}$	[s^{-1}]	Shear rate
ρ	[kg/m^3]	Density
τ_W	[N/m^2]	Wall shear stress

Dimensionless groups

Gr	Grashof number $(g\beta D^4\rho^2 q'')/(\mu^2 k)$
Gz	Graetz number $(\dot{m}\, c_p)/(kL)$
Gz_s	Equivalent Graetz number
Pr	Prandtl number $(c_p\mu/k)$
Nu	Nusselt number $(Dq'')/(k\Delta T)$
Nu_w	Nusselt number with wire coil
Nu_x	Local Nusselt number
Re_{MR}	Reynolds number Eq. 5
Re_s	Equivalent Reynolds number Eq. 10
$R3$	R3 criterion

Subscripts

MR	Meztner and Reed
h	Heat transfer test section
p	Pressure drop test section
in	Inlet conditions
out	Outlet conditions
x	Local conditions
b	Bulk conditions
W	Wall conditions
w	Wire coil conditions

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