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AN EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER ON THE ANNULAR SIDE OF HELICALLY COILED DOUBLE PIPE HEAT EXCHANGERS

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ABSTRACT

Despite its industrial importance, data on single phase heat transfer in helically coiled annular ducts are lacking. The present work present experimental data on average heat transfer coefficients for four annular-coil geometries, covering the laminar, transition and turbulent flow regimes, spanning the range $700 < Re < 2{,}5 \cdot 10^4$ and $4 < Pr < 13$. The effect of the centrifugal forces due to the coil curvature was evident, enhancing the heat transfer significantly in the laminar and transition regions. In the turbulent flow regime, the effects of curvature on heat transfer were small. Laminar flow was stabilized by secondary flow motion, shifting the transition to turbulence to higher Re. Heat transfer correlations were developed, taking into consideration the asymptotic behavior at high and low Re. The correlation reproduced 80% of the data to within ±15%.

INTRODUCTION

Helically coiled double-pipe heat exchangers have a wide field of applications in the food processing, refrigeration, HVAC and other process industries. In spite of its practical importance, the available information on heat transfer and pressure drop on the annular side of such heat exchangers is limited.

Flow in helically coiled circular tubes experience a centrifugal force promoting a secondary flow that enhances the heat transfer relative to straight tubes. Additionally, the secondary motion has a stabilizing effect on the flow, extending the laminar flow regime to higher Re. Also, the thermal entry-length in laminar flow is shortened due to the secondary flow. The increase in heat transfer is most pronounced in the laminar region, but is also evident in the turbulent regime. A similar behavior is observed for the friction factor; however, the increase in friction is often less than the increase in heat transfer in laminar flow, resulting in a net increase in heat transfer duty per unit pressure drop for coiled tubes as compared to straight tubes. A comprehensive review of the flow and heat transfer characteristics in coiled, circular tubes is given by Shah and Joshi (1987).

NOMENCLATURE

A	[m^2]	Heat transfer surface
D_c	[m]	Coil diameter
De	[-]	Dean number ($=\mathrm{Re}\sqrt{d_h / D_c}$)
d_i	[m]	Outside diameter of inner tube (See Fig. 2)
d_h	[m]	Hydraulic diameter (=4F/P)
d_o	[m]	Inside tube diameter of outer tube (see Fig. 2)
d_{ti}	[m]	Inner diameter, inner tube
f	[-]	Darcy friction factor
F	[m^2]	Flow area
h_a	[W/m^{2}K]	Heat transfer coefficient, annulus
h_t	[W/m^{2}K]	Heat transfer coefficient, circular tube
k	[W/mK]	Thermal conductivity
L	[m]	Tube length
$LMTD$	[K]	Logarithmic mean temperature difference
N_c	[-]	Number of coil revolutions
Nu	[-]	Nusselt number
P	[m]	Wetted perimeter
Pr	[-]	Prandtl number
Q	[W]	Heat duty
Re	[-]	Reynolds number
s	[m]	Coil pitch
U	[W/m^{2}K]	Overall heat transfer coefficient
Greek letters		
α	[-]	Annulus diameter ratio ($=d_i/d_o$)
Subscripts		
i		Inner
o		Outer
h		Hydraulic
w		Wall
$calc$		Calculated
exp		Experimental/Measured
Ann		Annular
L, lam		Laminar
$turb$		Turbulent

The flow and heat transfer conditions in the annular coiled passage has received significantly less attention. Choi and Park (1992) performed a numerical analysis of laminar flow in annular ducts, demonstrating the effect of the inner tube blockage on the secondary flow patterns. For small/moderate diameter ratios (α < ca. 0,7) two pairs of vortexes were observed, as shown in Figure 1. The flow close to the inner and outer tube wall was directed inwards (towards the centre of the coil), whereas the flow in the central core was directed outwards (in the direction of the centrifugal force).

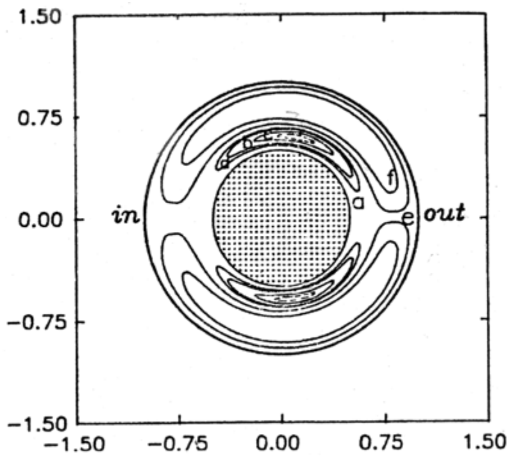

Figure 1 Stream functions showing secondary flow in annular coiled pipe. d_i/d_o=0.5, De=8000. Coil centerline at left. (Petrakis and Karaholis, 1996).

At larger diameter ratios (α >ca. 0,8) the smaller vortices close to the inner tube disappeared, the remaining vortex having an outwards direction close to the core tube wall. Similar results showing the pair of vortices at low/moderate diameter ratios were obtained analytically by Petrakis and Karahalios (1996). Choi and Park extended their numerical analysis to include heat transfer in fully developed laminar flow, however, they assumed heat transfer both at the inner and outer wall surfaces, with boundary condition constant axial heat flux and constant peripheral wall temperature. They concluded that the secondary flow enhances heat transfer relative to straight annular ducts, and that the main parameters influencing the average inner and outer wall heat transfer coefficient at negligible buoyancy were the centrifugal force, represented by the Dean number (De) and the diameter ratio, α. A similar case was subject to numerical analysis by Yang and Ebadian (1993), also concerning fully developed laminar flow with heat transfer at both the outer and inner walls having constant axial heat flux and constant peripheral wall temperature boundary conditions. They concluded that the heat transfer was enhanced relative to straight annular ducts by increasing flow velocity (i.e. increasing the axial pressure gradient), increasing the curvature ratio (d_h/D_c), and by decreasing the diameter ratio α. However, at larger α (α >0,6) the diameter effect becomes small. A common feature of the presented numerical results was that they did not adequately represent the boundary conditions encountered in double-pipe heat exchangers, where heat transfer takes place at the inner wall, the outer wall being adiabatic.

For coiled circular ducts it has been established that the thermal and hydraulic entrance lengths in the laminar flow region are shorter than for straight tubes (Shah and Joshi, 1987). For coiled annular ducts, however, no systematic studies are available. Choi and Park (1992) performed a numerical study of the development of the velocity profile, and concluded that the hydrodynamic inlet length is sensitive to the inlet boundary condition and that 'the flow in a curved annular duct is not necessarily fully developed earlier when the diameter ratio (α) is large owing to the complicated interaction between the viscous and the centrifugal forces'.

The only experimental work on heat transfer in annular coils found in the literature was by Garimella et al (1984), who presented average annular side heat transfer coefficients in the laminar and transition regions for two helical coil geometries using water on both the annular and tube sides. Their heat exchangers, having (d_i/d_o=0,81 ; d_h/D_c=0,0068) and (d_i/d_o=1,55 d_h/D_c=0,0198) showed significant heat transfer enhancement relative to straight annular ducts. However, their test geometries included helically wrapped wires with wire thickness equal to the gap opening in the annular channel, serving as spacers to centre the inner tube. The wires were wrapped with a pitch of 200-360 mm, and would be likely to have an impact on the secondary flow motion and hence the heat transfer coefficient. Recently, Louw and Meyer (2005) reported experimental results on two helically coiled double-pipe heat exchangers both having d_i/d_o=0,55 and d_h/D_c=0,0258, and having concentric, respectively eccentric positioning of the core tube. Their experiments showed that the annular-side heat transfer coefficient was positively affected when the core tube was eccentrically positioned, effectively touching the outer tube wall, yielding a 96% increase in the annular-side heat transfer coefficient. It was argued that this increase was due to a change in the annular side flow pattern, the increased effective heat transfer surface due to the contact between the inner and outer tubes, and due to the flattening of the outer tube due to the manufacturing process.

In view of the limited data available on annular side heat transfer in helical coils, an experimental program covering four geometries was performed. The annular side average heat transfer coefficients for the laminar and transition flow regimes are presented and compared.

EXPERIMENTS

Test geometries

The annular double pipe heat exchangers were as shown in Figure 2. Two concentric tubes were coiled to a helix sharing the same coil diameter (D_c) and coil pitch (s). The inner tube was positioned concentrically relative to the outer tube by means of a patented method using Ø3mm distancing pins, inserted through holes drilled in the outer annular tube wall at 3 circumferential positions (120° separating angle), and repeated approximately 3 times per revolution. The main heat exchanger dimensions are given in Table 1. The inner tube dimension was fixed at 13 mm inside diameter and 16 mm outside diameter for all heat exchangers. Both tubes were made from carbon steel, and the heat transfer surface was cleaned before each test run by circulating an acid solution on both the tube and annular sides, followed by clean water washing.

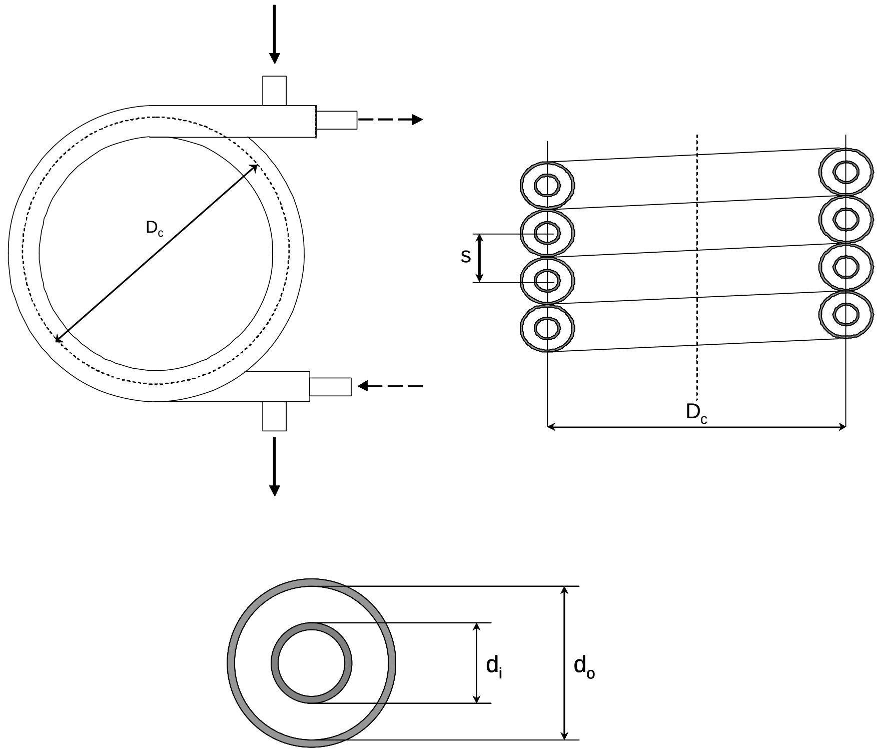

Figure 2 Test heat exchanger geometry

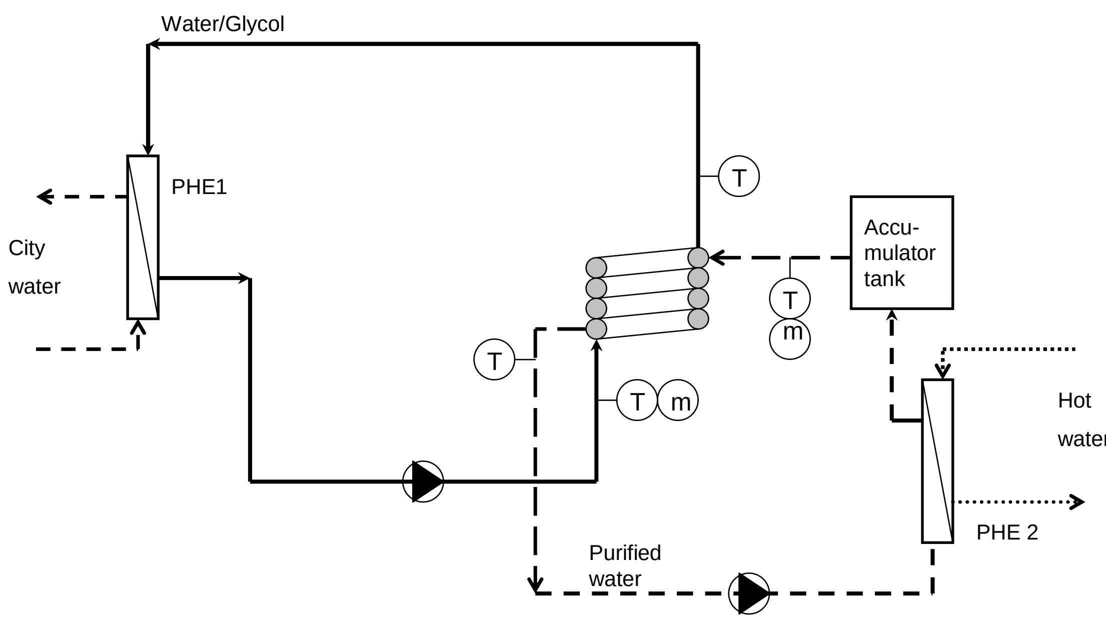

Figure 3 Test rig

Table 1 Main heat exchanger data

	D_c [mm]	d_o [mm]	d_i [mm]	s [mm]	L [mm]	N_c [-]	A_i [m^2]
Geometry 1	400	28,0	16,0	37	7690	6,117	0,3865
Geometry 2	600	28,0	16,0	37	7704	4,086	0,3872
Geometry 3	410	44,3	16,0	53	8573	6,650	0,4309
Geometry 4	610	44,3	16,0	53	7902	4,122	0,3972

Experimental setup

The experimental setup is shown in Figure 3. The annular-side fluid was purified water or a water/tri-ethylene-glycol mixture (30% ethylene glycol by mass) circulating in a closed loop. The annular side fluid was heated in the double-pipe heat exchanger and the heat was rejected to cold city water in a plate heat exchanger (PHE 1). A variable speed circulation pump was used to control the flow rate. Purified water was used as heat transfer fluid inside the coiled tubes, flowing countercurrent to the annular side fluid. The water was heated in a plate heat exchanger (PHE 2) by a hot water circuit connected to the central district heating system. Due to temperature fluctuations in the hot supply water, the circulating water was passed through a 0,1 m^3 accumulator tank downstream PHE 2 before entering the helical coil heat exchanger. The purpose of the accumulator tank was to dampen any temperature disturbances stemming from variations in the hot supply water temperature, thereby providing a stable inlet temperature to the test heat exchanger. The circulating hot water flow rate was adjusted so that an appreciable temperature difference was maintained through the entire heat exchanger, but at the same time maintaining the tube side heat transfer coefficient as high as possible.

The volumetric flow rates for both the tube side and annular side fluids were measured using MagFlo magnetic transmitters and the fluid inlet and outlet temperatures were measured using PT100 thermistors. Static mixers were provided at all temperature measurement stations in order to secure correct readings. All readings were recorded electronically by a data acquisition system.

The heat exchanger was insulated using 75mm glass-wool, minimizing heat losses. Overall heat balance deviations were typically within ±5%.

Data reduction

The heat duty for both the annular side and inner tube fluids were calculated from the measured flow rates and fluid temperatures. The arithmetic average of these was then used in determining the overall heat transfer coefficient as shown by Equation (1).

$$U = \frac{Q_m}{A_i \cdot LMTD} \tag{1}$$

Here Q_m is the arithmetic average of the tube side and the annular side fluid heat duties, A_i is the outside surface area of the inner tube, and LMTD is the logarithmic mean temperature driving difference for countercurrent flow. The tube side heat transfer coefficient (h_t) was calculated from the correlations recommended by Gnielinski (2002a). The annular side average heat transfer coefficient was determined from Equation (2).

$$h_a = \left[\frac{1}{U} - \frac{d_i}{d_{ti}} \cdot \frac{1}{h_t} - \frac{d_i}{2 \cdot k_w} \cdot \ln\left(\frac{d_i}{d_{ti}} \right) \right]^{-1} \tag{2}$$

Here, d_{ti} is the inner diameter of the core tube and k_w is the tube wall thermal conductivity (k_w=45 W/m·K). Taking into account the known uncertainties, primarily in heat duty and tube side heat transfer estimation, the uncertainty in the annular side heat transfer determination was estimated to be 10-15% at low and high flow rates, respectively.

For further generalization, the annular side characteristic length scale used in Nu, Re and De was taken as the hydraulic diameter (d_h=d_o-d_i). In the data reduction, all physical properties were evaluated at the arithmetic mean fluid operating temperature.

RESULTS AND DISCUSSION

The measured annular side heat transfer performance for each of the test geometries are shown in Figure 4 as function of the Reynolds number. Also shown in the figure are the Nusselt numbers for fully developed flow in straight annular ducts with the same diameter ratios. For straight annular ducts the laminar (Re <2 300) constant heat flux solution for Pr=5,0 (see e.g. Kays et al, 2005), and the turbulent flow (Re>10 000) correlation proposed by Gnielinski (2002b) are shown. In the transition region, a logarithmic interpolation between the laminar and turbulent flow solutions was used.

From Figure 4 it is obvious that no clear transition from laminar to turbulent flow was present, which is similar to observations made for coiled circular tubes (see e.g. Shah and Joshi, 1987). Further, it is observed that the heat transfer coefficient was significantly enhanced in the laminar and transition regions for all geometries, and that the turbulent flow solution for straight annular ducts was approached asymptotically as Re increases. For Geometries 1 and 2 (with the larger α) the straight annular tube heat transfer performance was met at approximately Re=10 000, whereas for the geometries with the smaller α the straight duct heat transfer coefficients were approached at slightly higher Re (ca 15 000).

Another observation worth noting is that $Nu \cdot Pr^{-1/3}$ was generally higher for the small diameter ratio geometries (Geometries 3 and 4) than for the larger diameter-ratio geometries. This was believed to be attributed to a higher degree of secondary flow in the small diameter-ratio geometries, where the blockage and skin friction caused by the central tube was less dominating, as also shown in the numerical/analytical works of Choi and Park (1994) and Petrakis and Karahalios (1996).

The observed Reynolds number dependency was, at least qualitatively, in agreement with experimental data for laminar flow in coiled tubes, where the exponent was found to be a function of the ratio (d_h/D_c) (Gnielinski, 2002b).

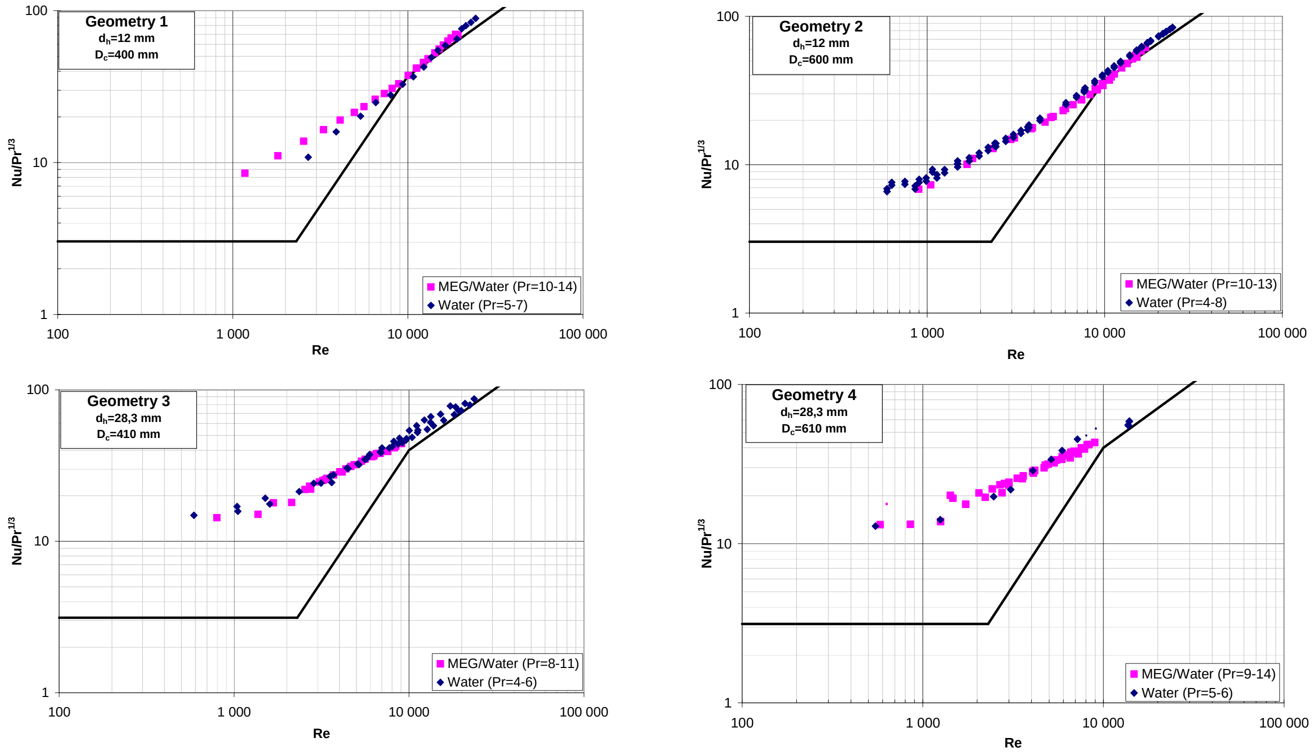

Figure 4 Experimentally obtained Nusselt numbers for all test geometries.

A linear regression analysis was performed to fit all the experimental data. As a basis for the analysis, the correlation developed by Gnielinski (2002b) for laminar flow in helically coiled tubes was used. However, correction terms taking into account the diameter ratio (α) and coil ratio (d_e/D_c) were included to reflect the effect on the secondary flow motion caused by the inner tube.

For low Dean numbers, where the centrifugal forces becomes negligible, Nu should asymptotically approach the value of a straight annular duct calculated according to Equation (3) (Gnielinski, 2002a).

$$Nu_{Ann,lam} = \left[3,66 + 1,2 \cdot \left(\frac{d_o}{d_i}\right)^{0,8}\right] + \left(1 + 0,14 \cdot \sqrt{\frac{d_o}{d_i}}\right) \cdot \frac{0,19 \cdot \left[\mathrm{Re} \cdot \mathrm{Pr} \cdot \left(\frac{d_h}{L}\right)\right]^{0,8}}{1 + 0,117 \cdot \left[\mathrm{Re} \cdot \mathrm{Pr} \cdot \left(\frac{d_h}{L}\right)\right]^{0,467}} \tag{3}$$

The first bracketed term in Equation (3) represents the solution for fully developed flow, and the second term represents the effect of the simultaneous development of the velocity tend temperature profiles on the average heat transfer coefficient. As discussed above there was no clear transition from laminar flow, and it was possible for the laminar flow to extend significantly beyond the straight tube critical value of about 2 300. Similar behavior has also been observed for coiled circular tubes (Shah and Joshi, 1987; Gnielinski, 2002b). The transition from laminar flow was in the present analysis set to Re=8 000, based on observations of the deviations between the developed correlations and the data)

The resulting correlation for the laminar region is shown in Equation (4).

$$Nu_L = Nu_{Ann,lam} + 1,658 \cdot 10^{-3} \cdot \left(\frac{d_o}{d_i}\right)^{0,754} \cdot \left(\frac{D_c}{d_h}\right)^{0,242} \cdot \mathrm{Re}^m \cdot \mathrm{Pr}^{1/3} \tag{4}$$

Equation (4) was used for $Nu_{Ann,lam}$, and the exponent *m* was taken from the correlation for helically coiled tubes of Gnielinski, shown in Equation (5):

$$m = 0,5 + 0,2903 \cdot \left(\frac{d_h}{D_c}\right)^{0,194} \tag{5}$$

For high Reynolds numbers, Nu was observed to be approaching the solution for fully developed turbulent flow in annular ducts, given by Equation (6) (Gnielinski, 2002a). Fully developed turbulent flow was considered reached at Re=15 000, which was in accordance with observations on coiled circular tubes, where fully developed turbulent flow is assumed for Re > 15 000 (ESDU, 2001) to Re> 22 000 (Gnielinski, 2002b).

$$Nu_{Ann,turb} = \frac{\mathrm{Re}\cdot \mathrm{Pr}\cdot \frac{f}{8}}{1+12,7\cdot\left(\mathrm{Pr}^{2/3}-1\right)\cdot\sqrt{\frac{f}{8}}}\cdot\left(0,86\cdot\left(\frac{d_o}{d_i}\right)^{0,16}\right) \qquad (6)$$

$$\frac{f}{8} = \left(1,82\cdot \log_{10}\mathrm{Re} - 1,64\right)^{-2}$$

For the transition region (8 000 < Re <15 000) a linear interpolation between the laminar flow (at Re=8 000) and turbulent flow (at Re=15 000) was used. The results are shown in Figure 5 to Figure 8. As observed, the overall agreement is acceptable; however there is a tendency of overpredicting the heat transfer coefficient in the low Reynolds number range for Geometries 1 and 2 (having the largest coil diameters) by 15% and 24%, respectively. This may be attributed to in inadequate correlation for the Re-exponent *m*, Equation (5), which in the present study was taken from a correlation for coiled circular tubes. 80% of the experimental data are, however, correlated to within ±15%, as illustrated by Figure 9.

The parameter range covered by the proposed correlation is:

$$700 < \mathrm{Re} < 25\,000$$
$$4 < \mathrm{Pr} < 13$$
$$0{,}361 < d_i/d_o < 0{,}571$$
$$0{,}02 < d_h/D_c < 0{,}069$$
$$279 < L/d_h < 642$$

Further correlation efforts in the laminar regime should draw experiences from the qualitative information obtained by numerical analysis, e.g. the influence of the centrifugal forces and viscous friction on the transition between one and two pairs of vortices as shown in Figure 1, and the extent of the thermal-hydraulic entry length, which apparently does not seem to be well understood for coiled annular duct flow.

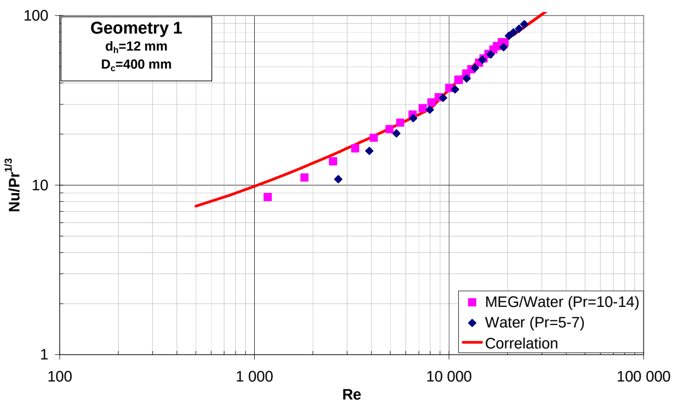

Figure 5 Comparison between experimental data and Equations (3)-(6) for Geometry 1. [d_o=28mm; d_i=16mm; D_c=400mm]

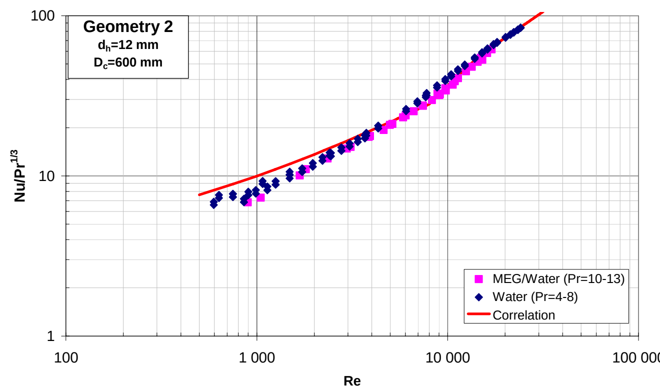

Figure 6 Comparison between experimental data and Equations (3)-(6) for Geometry 2. [d_o=28mm; d_i=16mm; D_c=600mm]

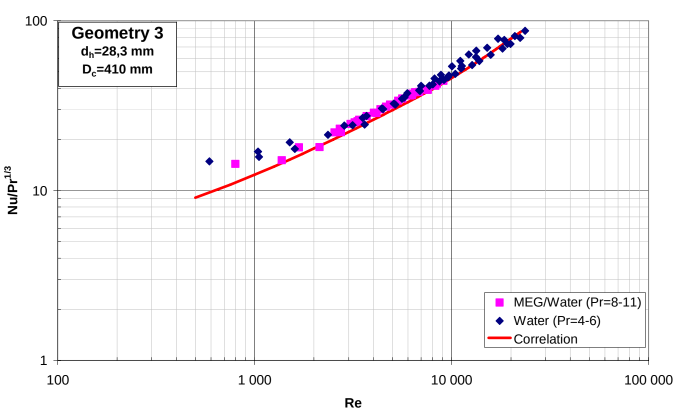

Figure 7 Comparison between experimental data and Equations (3)-(6) for Geometry 3. [d_o=44,3mm; d_i=16mm; D_c=410mm]

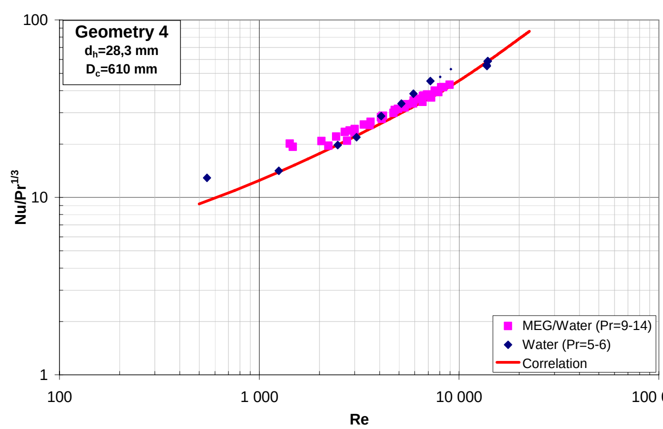

Figure 8 Comparison between experimental data and Equations (3)-(6) for Geometry 4. [d_o=44,3mm; d_i=16mm; D_c=610mm]

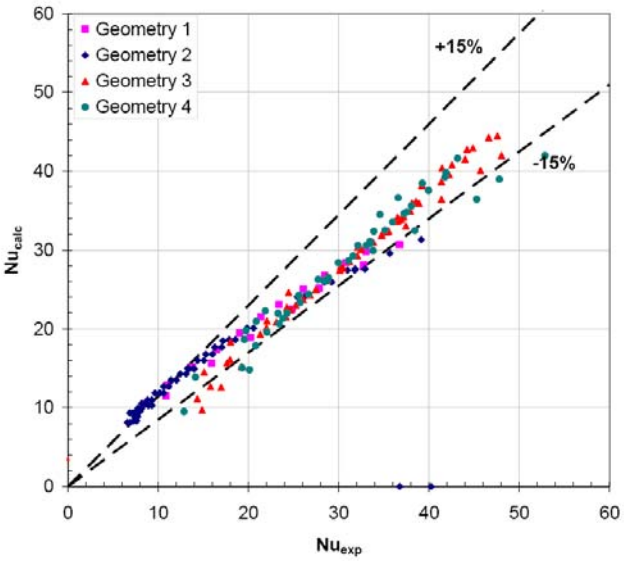

Figure 9 Error band for Equations (3)-(6).

CONCLUSION

New heat transfer data in the laminar, transition and turbulent flow regimes for helically coiled concentric annular tubes were obtained. The experiments covered the range 700 < Re < 25 000 and 4 < Pr < 13.

From the experimental data, the following were observed:

1. Heat transfer in the laminar region is significantly enhanced compared to straight annular tubes. This is due to the centrifugal forces setting up a secondary flow that enhances heat transfer.
2. No clear transition from laminar flow was observed. Based on the experimental data, laminar flow may persist up to Re=8 000.
3. In the laminar flow region, the Re-exponent was dependent on the coil ratio (d_h/D_c) and probably also the diameter ratio (d_o/d_i).
4. At Re > 15 000 only minor enhancement effects of secondary flow motion was observed; the heat transfer coefficients were adequately reproduced by correlations for turbulent flow in straight annular pipes.

A set of correlations were proposed, covering the laminar, transition and turbulent flow regimes. The correlations have straight annular duct heat transfer as asymptotes for high and low Re. The correlations reproduce the experimental data with a standard deviation of about 16%.

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