Heat Transfer Coefficient for Hydrocracked Oil Flow in Laminar Regime through an Annular Space

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In this paper it is presented a correlation based on experimental data for the prediction of the laminar heat transfer coefficient when cooling hydrocracked oil in the inner annulus of a horizontal triple concentric-tube heat exchanger. The cold fluid is water and the heat exchanger is operated under counter-current flow conditions. The test section used in the experiment was made of copper tubes with inner diameters of 12 mm, 26 mm and 40 mm, and a length of 1193 mm. Heat transfer coefficient values experimentally obtained were further compared with those calculated from the correlations existing in the literature for flowing through annular and circular spaces in laminar flow regime, acceptable deviations being obtained.

Keywords: heat transfer coefficient, laminar annular flow, triple concentric-tube heat exchanger

Cooling and heating of the petroleum products that flow through the annuli of various concentric-tube heat exchangers, such as double tube or the triple concentric-tube heat exchangers, are of great interest for the determination of the heat transfer coefficients.

For the calculation of heat transfer coefficients in annuli, according to several authors [1 - 3] it is recommended to use the correlations established for the circular tube, in which the characteristic length in Reynolds (*Re*) and Nusselt (*Nu*) numbers is the equivalent hydraulic or heated diameter. Taking into consideration that, in the concentric heat exchangers a laminar regime ($Re < 2300$) is often met, especially for high viscosity products, the use of the well-established correlations for circular tubes involves grave errors, especially if the cross section has sharp corners. For the forced convection in the laminar flow inside a circular tube, there can be mentioned the correlations established by Sieder and Tate (cited by Serth [1], Şomoghi [2], Mehrabian [3] and Poh-Seng et al. [4]), M. Rubinstein, Miheev (cited by Şomoghi [2]) and Hausen (cited by Lienhard et al. [5]). However, for the laminar annular flow it is mentioned especially the correlation developed by Gnilelinski (cited by Serth [1] and Koiki [6]). Nevertheless, the heat transfer coefficients in the laminar flow regime of the petroleum products through the annuli made of short tubes cannot be properly predicted by using the existing correlations, as they are more suitable for longer tubes and a general flow pattern. For this purpose, a new correlation was developed in this paper in order to design and to control technologically the heat exchangers with concentric tubes. The experimental data obtained when cooling a hydrocracked oil with water in an experimental triple concentric-tube heat exchanger were used to establish the new correlation. Hydrocracked oil was chosen due to its specific characteristics versus the conventional mineral oils (in terms of composition, high viscosity index, low content of sulfur and nitrogen and a special sensitivity to additivation, purity, protection etc.).

Using the same heat exchanger and experimental setup, Radulescu et al. presented the results obtained in the study of water - water heat transfer [7] and proposed a new Nusselt number correlation for the flow through annulus in transition flow regime [8].

Several correlations previously developed in the literature [1 - 11] were applied for the calculation of the heat transfer coefficients in the heat exchanger, in order to find the best predictive correlations. Therefore, the calculated values for the experimental heat transfer coefficients were compared with those obtained by using the existing correlations.

Experimental part

The triple concentric-tube heat exchanger used within the experiments was made up of cooper circular smooth tubes with a thickness of 1 mm and insulated with mineral wool. The dimensions of the heat exchanger were: $d_{1,i}$ = 0.012 m, $d_{1,o}$ = 0.014 m, $d_{2,i}$ = 0.026 m, $d_{2,o}$ = 0.028 m, L_1 = 1.193 m, $d_{3,i}$ = 0.040 m and L_2 = 0.935 m. The experimental setup was the one presented by Radulescu et al. in [7, 8], namely: the heat exchanger (the test section), a thermostatic bath and measurement instrumentation (flow meters and digital probe thermometers). Oil was heated in the thermostatic bath and then circulated through the inner annulus of the heat exchanger. Cold water was circulated through the inner tube and the outer annulus and it was supplied by the network.

For the section tested, during the experiments there were measured the inlet and outlet temperatures of the water and the outlet temperatures of the oil, for certain established flow rates of the fluids and a determined inlet oil temperature. The inlet temperature of the oil to the test section was adjusted between 60 - 86.4°C, while the water inlet temperature varied between 10.9 - 19.3°C, depending on the climate conditions.

The flow rates of the fluids were established to 90, 100 and 110 l/h for the water stream that flows through the inner tube, to 50, 120, 150 and 180 L/h for the oil and to 50, 100 and 120 L/h for the water stream that flows through the outer annulus.

Table 1 presents eight sets of experimental data resulted from the tests performed.

The calculation of heat transfer coefficients and the establishment of a new Nusselt correlation

Within the experimental measurements, the heat transfer from the oil to the cold water streams takes place in two opposite directions. One direction is for the heat

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No.	Inner Tube			Inner annulus			Outer annulus		
	m_{C1}, kg/s	$t_{C1,in}$, °C	$t_{C1,out}$, °C	m_H, kg/s	$t_{H,in}$, °C	$t_{H,out}$, °C	m_{C2}, kg/s	$t_{C2,in}$, °C	$t_{C2,out}$, °C
1	0.031	12.1	14.5	0.043	60.0	52.1	0.033	12.1	14.7
2	0.028	12.6	14.9	0.029	60.4	50.3	0.028	12.6	15.2
3	0.028	11.3	13.9	0.036	60.2	51.2	0.028	11.3	14.2
4	0.028	10.9	14.0	0.036	70.3	59.6	0.028	10.9	14.4
5	0.028	11.3	15.4	0.035	86.4	72.3	0.031	11.3	15.6
6	0.025	13.4	16.4	0.028	70.4	58.2	0.028	13.4	16.7
7	0.025	17.2	18.7	0.012	60.5	45.0	0.028	17.2	19.0
8	0.025	19.3	20.8	0.012	60.5	46.2	0.014	19.3	22.4

Table 1
TEMPERATURE AND MASS FLOW RATE MEASUREMENTS

exchange between the oil and the water stream that circulates through the inner tube and the other direction is for the heat exchange between the oil and the water stream that circulates through the outer annulus. Figure 1 shows the longitudinal section of the heat exchanger tubes.

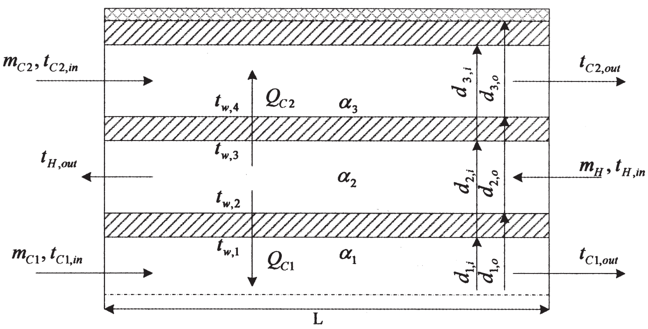

Fig. 1 The longitudinal section of the heat exchanger tubes

The heat transfer coefficients in a triple concentric-tube heat exchanger are: the heat transfer coefficient for the inside surface of the inner tube, α_1, the heat transfer coefficient for the outside surface of the intermediate tube, α_3 and, for the heat transfer between the hot fluid - outer surface of the inner tube and the inner surface of the intermediate tube, there can be estimated one heat transfer coefficient α_2 [8, 12 - 14].

The heat transfer analysis is based on the following heat balance equation:

$$Q = Q_{C1} + Q_{C2} \tag{1}$$

where the received heat flow rates can be calculated as follows

$$Q_{C1} = m_{C1} \cdot c_{p,C1} \cdot \left(t_{C1,out} - t_{C1,in}\right) \tag{2}$$

$$Q_{C2} = m_{C2} \cdot c_{p,C2} \cdot \left(t_{C2,out} - t_{C2,in}\right) \tag{3}$$

For the calculation of experimental α_2 it is used the following expression:

$$m_H \cdot c_{p,H} \cdot \left(t_{H,in} - t_{H,out}\right) = \alpha_2 \left(A_{1,o} + A_{2,i}\right)\left(t_C - t_w\right) \tag{4}$$

where:

$A_{1,o} = \pi \cdot d_{1,o} \cdot L_1$, $A_{2,i} = \pi \cdot d_{2,i} \cdot L_2$ and t_w is considered as $t_w = 0.5\left(t_{w,2} + t_{w,3}\right)$.

In equation (4) the temperature of the outer surface of the inner tube, $t_{w,2}$ and the temperature on the inner surface of the intermediate tube, $t_{w,3}$ can be written as

$$t_{w,2} = t_{w,1} + \frac{Q_{C1}}{2\pi L_1 \lambda_{Co}} \ln \frac{d_{1,o}}{d_{1,i}} \tag{5}$$

$$t_{w,3} = t_{w,4} + \frac{Q_{C2}}{2\pi L_2 \lambda_{Co}} \ln \frac{d_{2,o}}{d_{2,i}} \tag{6}$$

where the temperature of the inner surface of the inner tube, $t_{w,1}$ and the temperature on the outer surface of the intermediate tube, $t_{w,4}$ can be calculated from Newton's law of cooling written for α_1 and α_3. In these equations, it was used the thermal conductivity of cooper, λ_{Co} = 372.16 W/m°C.

For the experimental data presented in table 1, the flow regimes are transitional in the inner tube and laminar in annuli.

For the calculation α_1 there were applied the following correlations:

Sieder-Tate: $Nu = 0.027 \cdot Re^{0.8} \cdot Pr^{1/3}\left(\mu / \mu_p\right)^{0.14}$, (7)

$Re \geq 10^4$, $0.5 \leq Pr \leq 100$ [1, 10].

Dittus-Boelter: $Nu = 0.023 \cdot Re^{0.8} \cdot Pr^n$, (8)

$Re \geq 10^4, 0.6 \leq Pr \leq 160, L/d \geq 10, n = 0.3$ for cooling and 0.4 for heating [1, 2, 5, 9, 11].

Hausen: $Nu = 0.116 \cdot \left(Re^{2/3} - 125\right) \cdot Pr^{1/3} \cdot \left[1 + \left(d/L\right)^{2/3}\right] \cdot \left(\frac{\mu}{\mu_w}\right)^{0.14}$ (9)

$2200 < Re < 10^4$ [1, 2, 4].

Gnielinski: $Nu = \frac{(f/8) \cdot (Re - 1000) \cdot Pr}{1 + 12.7 \cdot (f/8)^{0.5} \cdot \left(Pr^{2/3} - 1\right)} \cdot \left[1 + \left(d/L\right)^{2/3}\right]$ (10)

$2100 < Re < 10^6$, $0.6 < Pr < 2000$ [1, 4, 5, 11].

The factor f in the equation (10) is the Darcy friction factor, as detailed in Colebrook equation: $f = (0.782 \ln Re - 1.51)^{-2}$.

The criteria relations established by Sieder-Tate and Dittus-Boelter for turbulent regime, for an applicability in the transitional regime (2300 < Re < 10^4) were corrected with the Ramm factor f [2]: f[2]: $f = 1 - (6 \cdot 10^5 / Re^{1.8})$.

The correlations used to calculate α_3 and α_2 are the following:

Sieder-Tate:

$$\text{Sieder-Tate: } Nu = 1.86 \cdot (Re \cdot Pr \cdot d/L)^{1/3} \cdot \left(\frac{\mu}{\mu_w}\right)^{0,14}, \quad (11)$$

$Re < 2100$, $0.5 < Pr < 17000$, $(Re \cdot Pr \cdot d/L)^{1/3}(\mu/\mu_w)^{0.14} > 2$ [1 - 4].

M. Rubinstein: $$Nu = c \cdot \left(Re \cdot Pr \cdot \frac{d}{L}\right)^{1/3}, \quad (12)$$

$Re < 2100$, $c = 1.60$ for cooling and 2.40 for heating.

Miheev: $$Nu = 4.366 \cdot \left(1 + 0.032 \cdot Re \cdot Pr^{5/6} \cdot \frac{d}{L}\right) \cdot \left(\frac{Pr}{Pr_w}\right)^{0.25} \quad (13)$$

(Re . d / L) > 10000, 0.7 < Pr < 1000 and a constant wall heat flux [2].

Hausen: $$Nu = 3.657 + \frac{0.0668 \cdot (Re \cdot Pr \cdot d/L)^{1/3}}{0.04 + (Re \cdot Pr \cdot d/L)^{-2/3}} \quad (14)$$

$(Re \cdot Pr \cdot d/L) < 1000$ [5].

Gnilelinski:

$$Nu = 3.66 + 1.2 \cdot \left(\frac{D_i}{d_o}\right)^{0.8} + \frac{0.19 \cdot \left[1 + 0.14 \cdot \left(\frac{D_i}{d_o}\right)^{0.5}\right] \cdot \left[Re \cdot Pr \cdot \frac{d_h}{L}\right]^{0.8}}{1 + 0.117 \cdot \left[Re \cdot Pr \cdot \frac{d_h}{L}\right]^{0.467}} \quad (15)$$

Re < 2100 [1, 6].

Reynolds, Prandtl (Pr) and Nusselt numbers were calculated using the following equations:

$$Re = \frac{w \cdot \rho \cdot L_c}{\mu} \quad (16)$$

$$Pr = \frac{c_p \cdot \mu}{\lambda} \quad (17)$$

$$Nu = \frac{\alpha \cdot L_c}{\lambda} \quad (18)$$

The characteristic length, L_c chosen for the calculation of Re and Nu numbers in the annuli is the equivalent hydraulic diameter d_h.

The equivalent hydraulic diameter for the inner annulus, $d_{h,2}$, is $d_{h,2} = d_{2,i} - d_{1,0}$, and for the outer annulus, $d_{h,3}$, is $d_{h,3} = d_{3,i} - d_{2,0}$. In the equation (16) the linear average velocity, w for the three fluids has the following equations:

$$w_{C1} = 4 \cdot m_{C1} / (\pi \cdot d_{1,i}^2 \cdot \rho_{C1}) \quad (19)$$

$$w_H = 4 \cdot m_H / (\pi \cdot (d_{2,i}^2 - d_{1,o}^2) \cdot \rho_H) \quad (20)$$

$$w_{C2} = 4 \cdot m_{C2} / (\pi \cdot (d_{3,i}^2 - d_{2,o}^2) \cdot \rho_{C2}) \quad (21)$$

The physical properties of the fluids were calculated at the arithmetic average between the inlet and outlet temperatures. The simplex $(\mu/\mu_w)^{0.14}$ in equations (7), (9) and (11) and $(Pr/Pr_w)^{0.25}$ in equation (13) were considered equal to 1.

Oil specific heat and thermal conductivity were estimated by using the following equations [2]:

$$c_p = [(2.964 - 1.332 d_{15}^{15}) + (0.006148 - 0.002308 d_{15}^{15})] \cdot (0.0538K + 0.3544), \text{ kJ/kg}\cdot{}^\circ\text{C} \quad (22)$$

where d_{15}^{15} is 0.885 and the characterization factor, K is 11.8.

$$\lambda = \frac{0.1172 - 6.33 \cdot 10^{-5} \cdot t}{d_{15}^{15}}, \text{ W/m}\cdot{}^\circ\text{C} \quad (23)$$

The density and kinematic viscosity of the oil were calculated using the equations established based on experimental determinations of these properties:

$$\nu = 0.034 \cdot t^{-1.8722}, \ R^2 = 1 \quad (24)$$

$$\rho = -0.0006 \cdot t + 0.8942, \ R^2 = 0.9974 \quad (25)$$

The Gnielinski correlations [1, 4 - 6, 11] used for the calculation of α_1 (eq. (10)) and α_3 (eq. (15)) give the best results in accordance with experimental data.

The calculated values for the linear average velocities, physical properties, Re, Pr, Nu numbers, heat transfer coefficients, the received heat flow rates and the wall

Table 2
VALUES OF α_1 CALCULATED BY USING GNIELINSKI CORRELATION, EQUATION (10)

No.	w_{C1}, m/s	ρ_{C1}, kg/m^3	c_{pC1}, J/kg·°C	$\mu_{C1} \cdot 10^5$, kg/m·s	λ_{C1}, W/m·s	Re_{C1}	Pr_{C1}	Nu_{C1}	α_1, W/m^2·°C	Q_{C1}, W	$t_{w,1}$, °C
1	0.27	999.5	4187	119	0.583	2713	8.6	21.9	1062	307	19.7
2	0.25	999.4	4187	118	0.584	2497	8.5	19.3	939	267	20.1
3	0.25	999.6	4188	122	0.581	2418	8.8	18.6	901	302	20.1
4	0.25	999.6	4188	122	0.581	2408	8.8	18.5	896	361	21.4
5	0.25	999.6	4187	119	0.583	2470	8.6	19.1	925	477	24.8
6	0.22	999.3	4185	114	0.586	2321	8.2	17.0	832	314	23.3
7	0.22	998.7	4182	105	0.593	2520	7.4	18.7	925	157	21.7
8	0.22	998.4	4180	100	0.597	2662	7.0	19.9	988	156	23.6

No.	w_{C2}, m/s	ρ_{C2}, kg/m^3	c_{pC2}, J/kg·°C	$\mu_{C2}\cdot 10^5$, kg/m·s	λ_{C2}, W/m·s	Re_{C2}	Pr_{C2}	Nu_{C2}	α_3, W/m^2·°C	Q_{C2}, W	$t_{w,4}$, °C
1	0.05	999.5	4187	119	0.583	524	8.6	5.8	283	363	29.0
2	0.04	999.4	4187	117	0.584	443	8.4	5.8	281	302	27.0
3	0.04	999.6	4188	121	0.581	428	8.7	5.8	280	337	27.4
4	0.04	999.6	4188	122	0.581	427	8.8	5.8	280	407	30.3
5	0.05	999.6	4187	119	0.583	481	8.5	5.8	282	550	37.2
6	0.04	999.3	4185	114	0.587	457	8.1	5.8	282	383	31.6
7	0.04	998.7	4182	105	0.593	496	7.4	5.8	285	209	27.0
8	0.02	998.4	4179	98	0.599	266	6.8	5.6	282	180	28.6

Table 3
VALUES OF α_3 CALCULATED BY USING GNIELINSKI CORRELATION, EQUATION (15)

No.	w_H, m/s	ρ_H, kg/m^3	c_{pH}, J/kg·°C	$\mu_H\cdot 10^5$, kg/m·s	λ_H, W/m·s	Re_H	Pr_H	Q, W	t_w, °C	$\alpha_{2,exp}$, W/m^2·°C
1	0.13	858.2	1994	1558	0.128	88	242	676	24.4	166
2	0.09	858.0	1991	1596	0.128	57	247	575	23.5	140
3	0.11	858.1	1992	1577	0.128	72	245	641	23.7	156
4	0.11	852.0	2030	1175	0.128	97	187	771	25.9	153
5	0.11	842.4	2088	800	0.127	141	132	1034	31.0	166
6	0.09	852.0	2027	1198	0.128	76	190	702	27.4	148
7	0.04	857.9	1980	1750	0.129	22	269	366	24.4	100
8	0.04	857.9	1983	1712	0.129	22	264	338	26.1	96

Table 4
VALUES OF $\alpha_{2,exp}$ CALCULATED BY USING EQUATION (4)

temperatures are shown in table 2 for the inner tube and in table 3 for the outer annulus.

Since the tubes walls are thin, there were obtained $t_{w,1}$ $t_{w,2}$ and $t_{w,4} \cong t_{w,3}$. The calculated values for the average linear velocities, physical properties, *Re* and *Pr* numbers, the heat flow rates, the wall temperatures in inner annulus and the experimental values of α_2 (α_{2exp}) obtained from equation (4) are shown in table 4.

From equations (11) - (15) used for the calculation of α_2, the correlations established by Sieder-Tate [1-4], Rubinstein (for heating) and Miheev [2] for the flow through circular ducts lead to acceptable results.

The correlation proposed for the oil laminar flow through the annulus has the general formula $Nu = c\cdot\left(\frac{Re\cdot d_h}{L}\right)^m\cdot Pr^n$ in which to the exponent *n* there was assigned 1/3, as commonly found in most criteria relations.

In correlation, there was introduced the simplex (d_h/L) because it was considered the direct influence of the annulus, with the length characteristic d_h, and of the tube length, *L*, on the heat transfer.

By applying the linear regression, there were obtained $c = 2.635$ and $m = 0.413$. Thus, the correlation obtained is

$$Nu = 2.635\cdot\left(\frac{Re\cdot d_{h2}}{L_1}\right)^{0.413}\cdot Pr^{1/3} \qquad (26)$$

In order to compare experimental and predicted data, there was calculated the average deviation (δ_{avg}) with the following equation:

$$\delta_{avg} = \frac{1}{N}\cdot\sum_{N}^{1}\left(\alpha_{exp}-\alpha_{calc}\right)/\alpha_{calc} \qquad (27)$$

Results and discussions

For the water stream flowing through the inner tube in the transitional regime, the errors between *Nu* numbers calculated with the Gnielinski correlation (eq. (10)) [1, 4, 5, 11] and *Nu* numbers calculated with equations (7) - (9) are between 15 - 28 % and, for water stream flowing through the outer annulus in laminar regime, the errors between *Nu* numbers calculated with Gnielinski correlation (eq. (15)) [1, 6] and *Nu* numbers calculated with equations (12) and (14) are between -6 - 19 %.

The experimental conditions for oil can be summarized, as follows:

- Laminar flow regime ($22 < Re_H < 141$);
- Neglect of the conduction in the axial direction ($Pe_H = Re_H{\cdot}Pr_H >> 100$);
- Neglect of the natural convection effects as the density values into and out of the tube are very close;
- $L_1/d_{h,2} = 99.4$ and $d_{2,i}/d_{1,o} = 1.86$.

The proposed correlation for the flow through the annular space verifies the values of the experimental heat transfer coefficients. The average deviations of the experimental values in comparison with the predictive values, obtained from the existing correlations for the flow through circular ducts, are 6 % for the Rubinstein and Miheev [2] and 37 % for Sieder-Tate [1 - 4]. On the other hand, the Hausen [5] correlation for the flow through circular spaces and Gnielinski [1, 6] correlation for the flow through annular

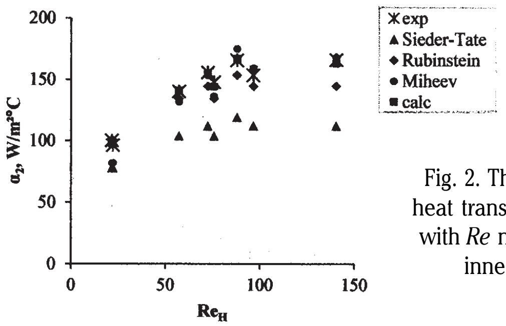

Fig. 2. The variation of heat transfer coefficients with *Re* numbers in the inner annulus

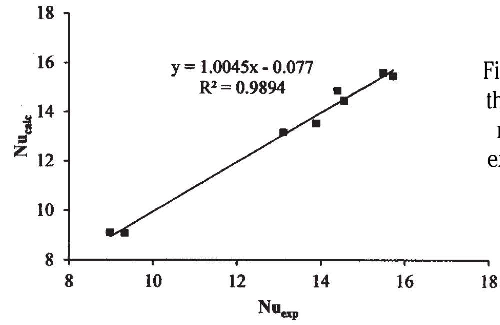

Fig. 3. Diagram of the predicted *Nu* number vs the experimental *Nu* number

space give unsatisfactory results. It is found that these correlations are suitable for water and less suitable for oil.

Figure 2 shows the variation of values $\alpha_{2,exp}$, where α_2 was calculated by using Sieder-Tate, Rubinstein and Miheev correlations and the correlation established in this paper ($\alpha_{2,calc}$) with *Re* number.

Figure 2 illustrates that experimental results have a similar profile to the values predicted with Rubinstein and Miheev correlations, that are recommended for the flow through circular spaces and with the correlation established in this paper.

The results of the experimental *Nu* number, Nu_{exp}, and the one calculated with the established correlation, Nu_{calc}, are plotted in figure 3.

An acceptable curve is obtained for the parameters represented. More than 98 % of the data could be captured by the curve fit and, therefore, an accurate description of the *Nu* number can be determined. The errors in using Nu_{calc} as opposed to Nu_{exp} are ± 4 %.

Conclusions

In this paper, it was analysed the hydrocracked oil - water heat transfer in an experimental triple concentric-tube heat exchanger. The heat exchanger is operated under the following conditions: counter-current flow, transitional flow regime in the inner tube and laminar flow regime in the inner and outer annuli. Based on experimental data, a Nusselt number correlation for the heat transfer coefficient calculation at the flow of hydrocracked oil through the inner annulus in laminar flow regime is established. The conditions and range of validity for the proposed correlation are: $22 < Re < 141$, $132 < Pr < 269$, $L_1/d_{h,2} = 99.4$, $d_{2,i}/d_{1,o} = 1.86$ and horizontal smooth circular tubes with shorter lengths. The experimental values of the heat transfer coefficient are in concordance with the results obtained from Rubinstein and Miheev [2] correlations, recommended for the flow through circular space, and for which the characteristic length considered in Reynolds and Nusselt numbers is the hydraulic diameter. For the experimental tests presented there were obtained values of α_1 and α_3 between 280 and 1062 W/m²°C, and α_2 between 96 and 166 W/m²°C.

Nomenclature

A – heat transfer area, m²;
c_p – specific heat, J/kg°C;
D – diameter of the outer tube, m;
d – diameter / diameter of the inner tube, m;
K – characterization factor;
L – length, m;
m – mass flow rate, kg/s;
N – number of points;
Nu – Nusselt number;
Pe – Peclet number;
Pr – Prandtl number;
Q – heat flow rate, W;
Re –Reynolds number;
t – temperature, °C;
w – linear average velocity, m/s.

Subscripts

1 – inner tube;
2 – intermediate tube / inner annulus;
3 – outer tube / outer annulus;
C1, C2 – cold fluids;
Co – copper;
c – characteristic;
calc – calculating;
exp – experimental;
H – hot fluid;
h – hydraulic;
i – inner;
in – inlet;
o – outer;
out – outlet;
w – wall.

Greek letters

α – heat transfer coefficient, W/m²·°C;
λ – thermal conductivity, W/m·°C;
μ – dynamic viscosity, kg/m . s;
ν– kinematic viscosity, m²/s;
ρ – density, kg/m³.

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