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The Estimation of the Convection Heat Transfer Coefficient in Annular Regime Transition Flow

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Abstract

An analysis of the convection heat transfer to flow in transition regime through the annular space is presented in this paper. Experimental data obtained for water – water heat transfer in a triple concentric tube heat exchanger, operated under laboratory conditions, were used in this respect. The values for the convective heat transfer coefficient, calculated by means of several criteria relations, were compared with those obtained from the equation of the Newton's law in order to identify the relation that best models the achieved heat exchange.

Key words: *transition regime, annular space, convective heat transfer coefficient*

Introduction

The transition regime flow in circular ducts or annuluar spaces is characterized by instability, with fluctuations in heat transfer and pressure drop. Due to this the flow behavior, the criteria relations for calculating the convective heat transfer coefficient (α) have a higher degree of uncertainty than those from laminar and turbulent regimes [1, 6].

The general formula for convection heat transfer is:

$$Q = \alpha \cdot A \cdot \left(t_w - t_f\right) \tag{1}$$

where Q is the heat transferred, A is heat transfer surface, t_w is the temperature of the solid surface (wall) and t_f is the average temperature of the fluid.

According to the literature, the convective heat transfer coefficient to flow in annular space can be calculated by using the criteria relations established for flow through constant circular sections, if the diameter is everywhere replaced by the equivalent diameter (d_h) as characteristic length [1 - 7]. The criteria relations define α through the Nusselt number (Nu), which represents the ratio between the convective and the conductive heat transfer:

$$Nu = \frac{\alpha \cdot l_c}{\lambda} \tag{2}$$

where l_c is the characteristic length and λ is thermal conductivity.

For the flow inside circular ducts in transition regime (2300 < Re > 10^4, where Re is Reynolds number), the convective heat transfer coefficient can be computed by using the criteria relations established for the transition annular flow, Ramm method or the graphical method proposed by Colburn.

Ramm method involves the calculation of α by using a relation specific to flow in a turbulent regime ($Re > 10^4$) and the correctioin of the obtained value with Ramm factor, f [2]:

$$f = 1 - \frac{6 \cdot 10^5}{Re^{1.8}} \tag{3}$$

where the formula for Re number is:

$$Re = \frac{w \cdot \rho \cdot l_c}{\mu} \tag{4}$$

In equation (4), the terms used have the following meanings: w – is the linear average velocity, ρ – represents density and μ - dynamic viscosity.

The graphical method proposed by Colburn is based on the correction curves obtained with Sieder - Tate relations for laminar ($Re < 2300$) and turbulent regimes, respectively, and it is less used [2, 3].

The purpose of this paper is to establish the easiest, most practical and precise method to estimate the convective heat transfer coefficient in case of transition annular flow. For this study, there were conducted a series of experimental tests on the water - water heat transfer in a triple concentric tube heat exchanger, operated under laboratory conditions. The values obtained for the convective heat transfer coefficient, by using several criteria relations, were compared with those obtained from the equation of Newton's law in order to identify the criteria relation that best models the achieved heat exchange.

Experimental Part

The heat exchanger used to carry out the experimental measurements is made of straight thin-walled copper tubes (1 mm), with the following outer diameters: 0.014 m for the inner tube, 0.028 m for the intermediate tube and 0.042 m for the outer tube. The length of the tubes is 1.193 m for the inner and the intermediate tubes (L_1) and 0.935 m for the outer tube (L_2) [6, 7]. In Figure 1 it is shown a cross section through the heat exchanger tubes.

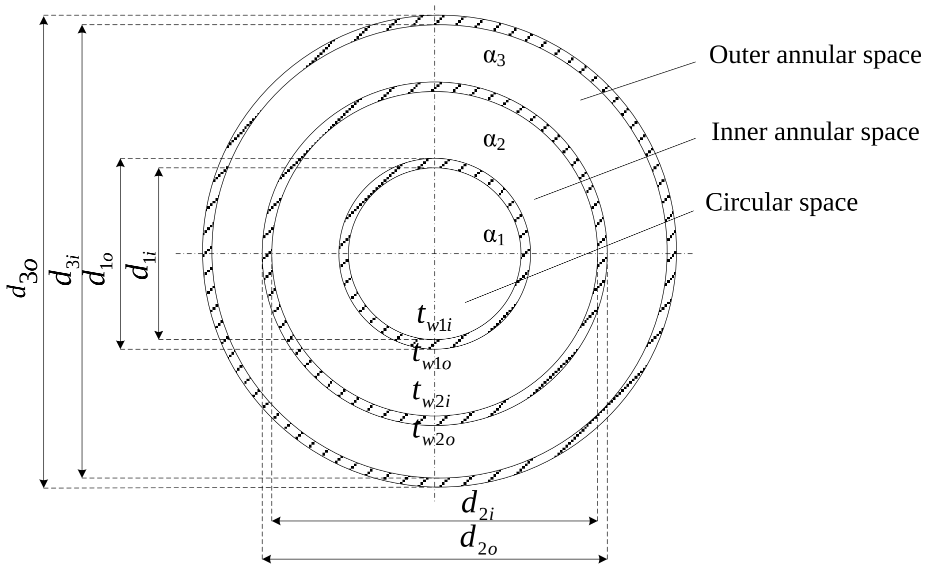

Fig. 1: The cross section through the heat exchanger tubes

In Figure 1: d_{1i}, d_{1o} – inner and outer diameters of the inner tube; d_{2i}, d_{2o} – inner and outer diameter of the intermediate tube; d_{3i}, d_{3o} – inner and outer diameter of the outer tube; t_{w1i}, t_{w1o} – inside and outside wall/s temperatures of the inner tube; t_{w2i}, t_{w2o} – inside and outside wall/s temperatures of the intermediate tube; α_1 - heat transfer coefficient for the heat transfer between the inside surface of the inner tube and the cold water stream C1, α_2 - heat transfer coefficient for the heat transfer between hot water and the outside surface of the inner tube and the inside surface of the intermediate tube, α_3 - heat transfer coefficient for the heat transfer between the outside surface of the intermediate tube and the cold water stream C2.

The hot water flows through the inner annular space and cold water flows through the inner tube (cold water stream C1) and through the outer annular space (cold water stream C2). The hot water transfers its heat simultaneously to cold water streams C1, C2 respectively.

For each of the three water streams, there were measured the flow rates and the inlet and outlet temperatures at the ends of the heat exchanger. The flow rate for the cold water stream C1 (V_{C1}) was kept constant to 100 l/h, whereas the flow rates for hot water (V_H) and cold water stream C2 (V_{C2}) varied between 145 and 350 l/h, for hot water, and between115 and 450 l/h for cold water stream C2. The inlet temperatures were 11.3 ºC, for cold water streams ($t_{C1\ in}$, $t_{C2\ in}$), and 60 ºC for hot water ($t_{H\ in}$).

The flow arrangement of fluids in the heat exchanger was in counter-current flow. In Table 1 there are given the outlet temperatures of the three fluids ($t_{C1\ out}$, $t_{C2\ out}$ - outlet temperatures of cold water streams C1 and C2, $t_{H\ out}$ - outlet temperature of the hot water) and the flow rates for hot water and cold water stream C2.

Table 1. Temperature and flow rates data

No. det.	Cold water stream C1	Hot water		Cold water stream C1	
	$t_{C1\ out}$, ºC	V_H, l/h	$t_{H\ out}$, ºC	V_{R2}, l/h	$t_{C2\ out}$, ºC
1	23.4	145	41.2	115	24.0
2	23.0	175	41.6	160	23.7
3	22.7	205	42.0	205	23.4
4	22.3	235	42.4	250	23.1
5	22.0	265	42.7	300	22.7
6	21.7	295	43.0	350	22.4
7	21.4	325	43.3	400	22.1
8	21.1	350	43.5	450	21.7

Results and Discussions

Flow regimes are: transitory in the inner tube (circular space) and the inner annular space, and laminar (Re < 2300) in the outer annular space.

The heat transfer coefficient on the inside of the inner tube was computed by using the following equation:

$$\alpha = \left(1415 + 21 \cdot t_{C1}\right) \cdot \left(\frac{w^{0.8}}{d_{1i}^{0.2}}\right) \tag{5}$$

where t_{C1} is the average temperature of the cold water stream C1 ($t_{C1} = t_{C1\ in} + t_{C1\ out}$).

Equation (4) refers to water flowing through circular sections and it is valid for $Re > 2000$ and water temperature ranging between 0 and 100 ºC [3].

The heat transfer coefficient for the heat transfer between hot water and the outside surface of the inner tube and the inside surface of the intermediate tube was calculated from the following Nusselt number correlations:

- Devis correlation established for the flow through an annular space [2, 3]:

$$Nu = 0.038 \cdot Re^{0.8} \cdot (Pr)^{1/3} \cdot \left(\frac{\mu}{\mu_w} \right)^{0.14} \cdot \left(\frac{d_{2i}}{d_{1o}} \right)^{-0.15} \tag{6}$$

where Pr is Prandtl number and μ_{w2} represents the dynamic viscosity of the fluid at the temperature of the tube wall.

Equation (5) is valid for $7 < Re < 180000$ and $1{,}2 < d_{2i}/d_{1o} < 6800$ [2]. The relation of Pr is

$$Pr = \frac{c_p \cdot \mu}{\lambda} \tag{7}$$

where c_p reprezents the specific heat and λ is the thermal conductivity of the fluid.

- Monrad and Pelton corelation for the turbulent flow through an annular space [5], corrected with the Ramm factor:

$$Nu = 0.020 \cdot Re^{0.8} \cdot Pr^{1/3} \cdot \left(\frac{d_{2i}}{d_{1o}} \right)^{0.53} \tag{8}$$

- Gnielinski corelation for the flow through a circular space [1]:

$$Nu = \frac{(f/8) \cdot (Re - 1000) \cdot Pr}{1 + 12{,}7\sqrt{f/8} \cdot \left(Pr^{2/3} - 1\right)} \cdot \left(1 + \left(\frac{d_{h1}}{L_1} \right)^{2/3} \right) \tag{9}$$

were f is the Darcy friction factor and can be computed from equation $f = (0.782\, ln\, Re - 1.51)^{-0.2}$ and d_{h1} is the equivalent diameter of the inner annular space, $d_{h1} = d_{2i} - d_{1o}$. Equation (8) is valid for $0{,}6 < Pr < 2000$ and $2100 < Re < 10^6$.

The heat transfer coefficient for the heat transfer between the outside surface of the intermediate tube and the cold water stream C2 was computed by using the following Nusselt number correlation [10]:

$$Nu = 0.33 \cdot Re^{0.5} \cdot Pr_{0.43} \cdot \left(\frac{Pr}{Pr_w} \right)^{0.25} \cdot \left(\frac{L_2}{d_{h2}} \right)^{0.1} \tag{10}$$

where Pr_w represents the Pr number for the physical properties of the fluid at the wall temperature and d_{h21} is the equivalent diameter of the outer annular space, $d_{h2} = d_{3i} - d_{2o}$.

In equations (5) and (7), the simplex $(Pr/\,Pr_w)^{0.25}$ considers the influence of the physical properties of the boundary layer on the heat transfer by convection and it is used especially for viscous liquids and larger differences between temperatures wall – fluid [9].

In the above-mentioned experimental conditions, both $(\mu/\mu_w)^{0.14}$ in equation (2) and the simplex $(Pr/\,Pr_w)^{0.25}$ were considered equal to 1.

The equation of Newton's cooling law was applied to compute a estimated value for α_2. The formula for the convection heat transfer in the inner annular space is:

$$\alpha_2 = \frac{m_H \cdot c_{pH} \cdot \left(t_{H\,out} - t_{H\,in}\right)}{A \cdot \left(t_H - t_w\right)} \tag{11}$$

where m_H is mass flow rate of hot water, c_{pH} – specific heat of hot water, A – heat transfer area of the inner annular space, t_H – average temperature of hot water ($t_H = t_{H\,in} + t_{H\,out}$) and t_w is considered as the average temperature of the inner annular space walls on the inside surfaces ($t_w = t_{w1o} + t_{w2i}$).

$$A = \pi \cdot \left(d_{1o} \cdot L_1 + d_{2i} \cdot L_2\right) \tag{12}$$

The equation of t_w for the heat transfer achieved by means of convection and conduction through cylindrical walls can be written as:

$$t_w = 0.5 \cdot \left[t_{C1} + Q_1 \cdot \left(\frac{1}{\alpha_1 \cdot A_{1i}} + \frac{1}{2\pi L_1 \lambda_t} ln \frac{d_{1o}}{d_{1i}} \right) + t_{C2} + Q_2 \cdot \left(\frac{1}{\alpha_3 \cdot A_{2o}} + \frac{1}{2\pi L_2 \lambda_t} ln \frac{d_{2o}}{d_{2i}} \right) \right] \tag{13}$$

where Q_1 is the received heat flow by cold water stream C1, A_{1i} – heat transfer area on the inside surfaces of the inner tube ($A_{1i} = \pi \cdot d_{1i} \cdot L_1$), λ_t – thermal conductivity of tube material, t_{C2} – average temperature of cold water stream C2 ($t_{C2} = t_{C2\,in} + t_{C2\,out}$), Q_2 – the received heat flow by cold water stream C2 and A_{2o} is the heat transfer area on the outside surfaces of the intermediate tube ($A_{2o} = \pi \cdot d_{2o} \cdot L_2$).

$$Q_1 = m_{C1} \cdot c_{pC1} \cdot \left(t_{C1out} - t_{C1in}\right) \tag{14}$$

$$Q_2 = m_{C2} \cdot c_{pC2} \cdot \left(t_{C2out} - t_{C2in}\right) \tag{15}$$

where: m_{C1} represents the mass flow rate of cold water stream C1, c_{pC1} – the specific heat of cold water stream C1, m_{C2} – the mass flow rate of cold water stream C2 and c_{pC2} – the specific heat of cold water stream C2.

The linear average velocities are 0.25 m/s for the cold water stream C1 and range between 0.11 - 0,26 m/s for the hot water and 0,05 – 0.20 m/s for the cold water stream C2. The values of Pr range between 7.52 – 7.80 for the cold water stream C1, 3.35 – 3.41 for the hot water and 7.45 – 7.72 for the cold water stream C2. The values for the received heat flows range between 1137 - 1404 W for Q_1 and 1694 - 5431 W for Q_2. In Table 3 there are shown the values for Re and the convective heat transfer coefficients (W/(m^2·°C)) for all fluids.

Table 3. Values of *Re* and the convective heat transfer coefficients

No. det.	Re			α_1	α_2 ec. (10)	α_3
	Cold water steam C1	Hot water	Cold water stream C2			
1	2752	2336	563	1402	1408	1424
2	2737	2827	780	1399	1708	1679
3	2726	3321	996	1396	1995	1899
4	2712	3818	1209	1393	2236	2096
5	2701	4315	1443	1391	2479	2294
6	2690	4814	1677	1388	2710	2477
7	2679	5315	1909	1386	2910	2646
8	2668	5732	2136	1383	3036	2804

In Figure 2 there is represented the variation of values α_2, obtained by means of three criteria relations and the Newton's law of cooling, with the linear average velocity of hot water, w_H.

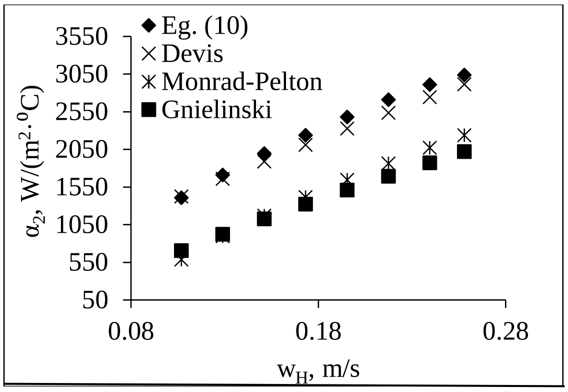

Fig. 2. The variation of α_2 with the linear average velocity of hot water

In this figure, it is shown that the values of α_2 calculated with Devis correlation are similar to those calculated with the equation for the Newton's law of cooling; their curves are very close, as opposed to the curves obtained for the values calculated with Gnielinski and Monrad – Pelton correlations. Therefore, it is considered that Devis correlation accurately/ adequately shapes/ models the heat transfer achieved under the presented experimental conditions and it is suitable for calculating the convective heat transfer coefficient in case of transitory flow through an annular space.

Conclusions

In this paper it was analyzed the convection heat transfer for a transition annular flow. There were used experimental data obtained for water-water heat transfer in a triple concentric tube heat exchanger. In the heat exchanger, the desired flow was obtained in the inner annular space for *Re* ranging between 2336 and 5732. The convective heat transfer coefficient in case of transition annular flow was calculated by using relations specific to several flow patterns, namely Devis correlation, for the flow in an annular space and $7 \leq Re \leq 1.8 \cdot 10^5$, Monrad and Pelton correlation, for the flow through an annular space in turbulent regime and corrected with the factor Ramm, and Gnielinski correlation, for the flow through a circular space and $2100 < Re < 10^6$. It was proven that only Devis correlation leads to similar values of convective heat transfer coefficient with those obtained from the equation for Newton's law of cooling, as opposed to Monrad - Pelton and Gnielinski relations that had led to unsatisfactory results. Therefore, the recommendation of the literature to use criteria relations for calculating the convective heat transfer coefficient is not generally valid, as most of the criteria relations are customized to the experimental system for which they were established.

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Estimarea coeficientului de transfer de caldura convectiv la curgerea printr-un spatiu inelar in regim tranzitoriu

Rezumat

În lucrare se face o analiză a transferului de căldură convectiv la curgerea în regim tranzitoriu printr-un spaţiu inelar. Datele experimentale obţinute la transferul de căldură apă - apă într-un schimbător de căldură de tri-concentric operat în condiţii de laborator au fost prelucrate în scopul realizării acestui studiu. Valorile coeficientului de transfer termic convectiv calculate cu mai multe relaţii criteriale au fost comparate cu cele obţinute din ecuaţia legii lui Newton pentru a identifica relaţia criterială care modelează cel mai bine schimbul de căldură realizat.