Heat Transfer Coefficient Solver for a Triple Concentric-tube Heat Exchanger in Transition Regime

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The paper proposes a calculation algorithm used in heat transfer studies when triple concentric-tube heat exchangers are involved. The study consists in determining the partial coefficients of heat transfer afferent to three fluids that exchange heat between them based on experimental results. According to experimental mode, the flow in tubes of the heat exchanger is in counter-flow from the hot fluid to two cold ones, while the circulation of the cold fluids is in co-current flow. In the experiment water was used as working fluid, the heat exchange taking place without the phase transformation. The proposed algorithm allowed obtaining useful correlation of partial coefficients of heat transfer calculation for the hot fluid which circulates through the inner annular space, in the transition regime.

Keywords: heat transfer, partial coefficient of heat transfer, triple concentric-tube heat exchanger

Triple concentric-tube heat exchangers are frequently used in heat processes, cooling processes, pasteurization/sterilization, freezing and concentration processes applied especially in the food industry. Within the annular spaces of the triple concentric-tube heat exchanger high viscosity fluids, Newtonian fluids or liquids containing particles can be transferred. The technical and economical advantages that are ensured by the use of this type of exchangers, in comparison to the use of heat exchangers with two concentric tubes, are represented by the area of higher heat transfer per length unit and higher velocity due to the presence of annular spaces [1-4].

Triple concentric-tube heat exchangers have been analyzed in previous studies from the theoretical point of view [5-7]. Different papers have developed case studies for the design and indirect testing of a heat exchanger with concentric triple tubes used to steam milk sterilization [2, 3].

For these heat exchangers, mathematical models have been developed in order to determine the axial temperature distribution of fluids [1] and a numerical simulation has been made to observe the thermal and fluid-dynamic behaviour [8].

This paper is based on heat transfer studies in a triple concentric-tube heat exchanger upon the circulation in counter-current flow from a hot water stream towards two streams of cold water, with circulation in co-current flow, without phase transformation.

For the transfer of heat through the tubes and annular spaces, most literature data are for the laminar regime ($Re < 2100$) and fully developed turbulent, and for the case of the transition regime ($2100 < Re < 10^4$), these data are much more restrained, or have not been yet verified. In the transition regime, the flow is characterized by instability leading to fluctuations in pressure drop and heat transfer [9].

A good heat exchange for a heat exchanger is provided by a turbulent regime; however, in practice there are cases in which we meet the transition and even laminar regimes also. For these reasons, one of the objectives of the paper is to establish a correlation that allows partial coefficient of heat transfer determination for a hot fluid that circulates through the inner annular space of the heat exchanger in transition regime.

The partial coefficient of heat transfer determination for a hot fluid is achieved by starting from experimental data and by using the heat balance equations. Moreover, by using correlations already presented in literature for partial coefficients of heat transfer, this parameter has been determined for the cold streams.

Experimental part

Due to the thermal properties that can be determined easier and also because of its availability, safety and easy manipulation, water has been used as a medium for the presented experiment. Three different streams of water have been used, i.e. a hot water stream and two cold water streams, noted C1 and C2, respectively. The cold water streams were provided by the water supply network.

The experimental setup on which the experimental measurements were carried out is presented in figure 1.

In triple concentric-tube heat exchanger (1) (fig. 1), two cold water streams C1 and C2 flows in co-current flow while the hot water stream flows in counter-current flow with the ones two cold water streams; the hot water flows through the inner annular space; the cold water stream, C1 flows through the central tube and the cold water stream, C2 flows through the outer annular space. The water is heated in the thermostat bath (2) from where, by using a pump (3), it is transferred to rotameter R3 and enters in the heat exchanger. The hot water which flows

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Fig. 1. Experimental setup
1 - triple concentric-tube heat exchanger, 2 - thermostat bath, 3 - centrifugal pump, 4 - electronic thermometers, R1, R2, R3 - rotameters
through the inner annular space transfers heat to the cold water streams. The triple concentric-tube heat exchanger used is made up of 1mm thick industrial copper tubes and it is insulated with a layer of mineral wool. The geometrical characteristics of the tubes used are presented in table 1.

Although in the construction of the heat exchanger, the lengths of the tubes are different, there shall not be considered a significant influence of the difference of lengths on the heat exchange carried out in the experimental system.

Figure 2 presents the cross section through a triple concentric-tube heat exchanger with specific characteristic parameters of heat transfer (\( t_{c1}, t_{w1}, t_{h}, t_{w2}, t_{c2}, \alpha_{c1}, \alpha_{h}, \alpha_{c2}, Q_{c1}, Q_{c2} \)) and dimensional parameters (\( d_{i,j}, i=1,2,3, d_{ij}, j=1,2,3 \)).

Heat transfer mechanisms in the heat exchanger are convections from the fluid towards the wall and conduction through the cylinder walls.

Because the walls of the tubes are thin and the device has no deposit of dirt on the walls, the temperatures on the walls have been taken as the average temperature of their surfaces.

Inlet temperature of hot water in experimental research, was kept constant \( t_{Hi} = 55.5 \, ^{\circ}C \). Inlet temperatures of cold water for both circuits are equal, except that experimental researches were conducted on different days, \( t_{C1i} = t_{C2i} = 10.8, 13.0, 14.2 \, ^{\circ}C \).

Next water flow rates were used during the experiment: hot water flow rate, \( V_H = 150 – 490 \, L/h \); cold water flow rate through the central tube, \( V_{C1} = 100 – 300 \, L/h \); cold water flow rate through the outer tube, \( V_{C2} = 100 – 470 \, L/h \).

The proposed algorithm for calculating of partial coefficients of heat transfer for fluids exchanged in the triple concentric-tube heat exchanger follows almost the same steps as for the case of the calculation algorithm to be applied for a double concentric-tube heat exchanger [10].

**Proposed algorithm**

The calculated algorithm has the following steps:
- global thermal balance for the heat exchanger has been established;
- heat flows for the hot fluid and cold fluids \( C1 \) and \( C2 \) were determined;
- determination of the partial coefficient of heat transfer for the cold fluid \( C1 \) and the determination of the average temperature of the central tube wall;
- determination of the partial coefficient of heat transfer for the cold fluid \( C2 \) and the determination of the average temperature of the intermediate tube;
- average temperature of the inner annular space’s walls was determined;
- partial coefficient of heat transfer for hot fluid was calculated.

All physical properties of water in the used equations (density, kinematic viscosity, specific heat, thermal conductivity) were calculated at the average temperature between the inlet and the outlet temperatures for both hot and cold streams.

The examples of values of constant and variable quantities of experimental measurements used to calculate partial coefficients of heat transfer are presented in table 2.

The global heat balance for the heat exchanger presumes the establishment and determination of the heat flow transferred and the heat flows received:

\[
Q_H = Q_{C1} + Q_{C2}
\]

**Table 1**

<table>
<thead>
<tr>
<th>Geometrical characteristics</th>
<th>Symbol</th>
<th>Central tube</th>
<th>Intermediate tube</th>
<th>Outer tube</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outside diameter, mm</td>
<td>( d_{i,j} )</td>
<td>14</td>
<td>28</td>
<td>42</td>
</tr>
<tr>
<td>Inside diameter, mm</td>
<td>( d_{i,j} )</td>
<td>12</td>
<td>26</td>
<td>40</td>
</tr>
<tr>
<td>Length, mm</td>
<td>( l_{a-i,j} )</td>
<td>1193</td>
<td>1193</td>
<td>935</td>
</tr>
</tbody>
</table>

**Table 2**

<table>
<thead>
<tr>
<th>No. det.</th>
<th>Cold water (C1)</th>
<th>Hot water</th>
<th>Cold water (C2)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( V_i ) L/h</td>
<td>( t_{C1i} ) °C</td>
<td>( t_{C2i} ) °C</td>
</tr>
<tr>
<td>1</td>
<td>100</td>
<td>14.2</td>
<td>23.6</td>
</tr>
<tr>
<td>2</td>
<td>100</td>
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<td>25.1</td>
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<td>3</td>
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<td>27.1</td>
</tr>
<tr>
<td>4</td>
<td>140</td>
<td>13.0</td>
<td>21.0</td>
</tr>
<tr>
<td>5</td>
<td>210</td>
<td>13.0</td>
<td>19.2</td>
</tr>
<tr>
<td>6</td>
<td>310</td>
<td>13.0</td>
<td>18.4</td>
</tr>
<tr>
<td>7</td>
<td>200</td>
<td>10.8</td>
<td>21.8</td>
</tr>
<tr>
<td>8</td>
<td>300</td>
<td>10.8</td>
<td>19.1</td>
</tr>
</tbody>
</table>
The heat flows for the three fluids are calculated using the following calorimeter equations:

\[ Q_H = m_H \cdot c_{p,H} \cdot (t_{in} - t_{out}) \]  
\[ Q_{CI} = m_{CI} \cdot c_{p,CI} \cdot (t_{in} - t_{out}) \]  
\[ Q_{C2} = m_{C2} \cdot c_{p,C2} \cdot (t_{in} - t_{out}) \]

For the calculation of the partial coefficient of heat transfer \( \alpha_{CI} \) for cold water which circulates through the central tube for a transition regime there were applied the following correlations:

- \( Nu_{CI} = 0.027 \cdot Re_{CI}^{0.8} \cdot Pr_{CI}^{0.13} (\mu_{CI} / \mu_w)^{0.14} \) \[9,11,12\] \[5\]
- \( Nu_{CI} = 0.022 \cdot Re_{CI}^{0.8} \cdot Pr_{CI}^{0.4} \) \[9,11,12\]: \[6\]

\( Nu_{CI} = \frac{(f / 8) \cdot (Re_{CI} - 1000) \cdot Pr_{CI}^{0.13} \cdot (d_{in} / L_1)^{0.7}}{1 + 12.7 \cdot (f / 8)^{0.7} \cdot (Pr_{CI}^{0.13} - 1)} \) \[9\]

\( Nu_{CI} = 0.116 \cdot (Re_{CI}^{0.8} - 125) \cdot Pr_{CI}^{0.13} \cdot [(d_{in} / L_1)^{0.7} / (\mu_{CI} / \mu_w)]^{0.14} \) \[9\]

Equations (5) and (6), specific for turbulent regime for applicability in transition regime were corrected with the Ramm factor \( f \):

\( f = 1 - \left(6 \cdot 10^5 / Re^{1.6}\right) \)

The factor \( f \) in the equation (7) is the Darcy friction factor explained by Colebrook equation:

\( f = \left(0.782 \ln(Re - 1.51)\right)^{-1} \)

In equations (5) and (8) the simplex \((\mu_{CI} / \mu_w)^{0.14}\) was considered to be equal with 1.

Applying equations (5) - (8), Nusselt number was obtained with a difference up to \( \pm 14\% \).

Equations (5) and (6) were applied for the turbulent regime to select correlation as fairly to have no influence on heat transfer coefficient part of hot water.

It has been found that for the given conditions, the most suitable is the Gnielinski correlation.

For determining the flow regime in the central tube, the Reynolds number has been calculated via the following equation:

\( Re_{CI} = \frac{d_{in} \cdot w_{CI} \cdot \rho_{CI}}{\mu_{CI}} \)

\( w_{CI} = \frac{4 \cdot V_{CI}}{\pi \cdot d_{in}^2} \)

and Prandtl number has been calculated with equation:

\( Pr_{CI} = \frac{c_{p,CI} \cdot H_{CI}}{\lambda_{CI}} \)

Knowing the Nusselt number value, the partial coefficient of heat transfer for stream C1 is calculated with the equation (14).

\( \alpha_{CI} = \frac{Nu_{CI} \cdot \lambda_{CI}}{d_{in}} \)

Knowing \( \alpha_{CI} \), from Newton's law \( \alpha_{CI} \cdot A_{C1} \cdot (t_{in} - t_{out}) \), the average temperature of the central tube’s wall, \( t_{w1} \), is calculated with equation:

\( t_{w1} = t_{in} + \frac{Q_{C1}}{\alpha_{C1} \cdot A_{C1}} \)

where \( t_{w1} = (t_{in} - t_{out}) / 2 \), and \( A_{C1} = \pi \cdot d_{in} \cdot L_1 \).

In Table 3 are presented the main values obtained by experimental data processing for stream C1.

For calculating the partial coefficient \( \alpha_{C2} \) of heat transfer for cold water which circulates through the outer annular section, in laminar regime the following correlations have been used:

- \( Nu_{C2} = 5.66 + 1.2 \cdot (d_{out} / d_{in})^{0.8} + 0.19 \cdot \left[1 + 0.14 \cdot \left(d_{out} / d_{in}\right)^{0.7} \cdot \left(Re_{C2} \cdot Pr_{C2} \cdot d_{in} / L_2\right)^{0.8}\right] \)

\( 1 + 0.117 \cdot \left(Re_{C2} \cdot Pr_{C2} \cdot d_{in} / L_2\right)^{0.8} \)

- \( Nu_{C2} = 3.66 + 0.21 \cdot \left(d_{out} / d_{in}\right)^{0.8} \cdot \left(Re_{C2} \cdot Pr_{C2} \cdot d_{in} / L_2\right)^{0.8} \)

\( 1 + 0.04 \cdot \left(d_{out} / d_{in}\right)^{0.8} \cdot \left(Re_{C2} \cdot Pr_{C2} \cdot d_{in} / L_2\right)^{0.8} \)

- \( Nu_{C2} = 0.51 \cdot Re_{C2}^{0.2} \cdot Pr_{C2}^{0.47} \cdot \left(Re_{C2} / Pr_{C2}\right)^{0.2} \)

\( Nusselt number: Nu_{C2} = \frac{\alpha_{C2} \cdot d_{in}^2}{\lambda_{C2}} \)

<table>
<thead>
<tr>
<th>No. det.</th>
<th>m_{C1} x 10^{3}, kg/s</th>
<th>Q_{C1}, W</th>
<th>w_{C1}, m/s</th>
<th>Re_{C1}</th>
<th>Pr_{C1}</th>
<th>Nu_{C1}</th>
<th>\alpha_{CI}, W/m²°C</th>
<th>t_{w1}, °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>27.73</td>
<td>1090</td>
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<td>2867</td>
<td>7</td>
<td>22</td>
<td>1094</td>
<td>41.1</td>
</tr>
<tr>
<td>2</td>
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<td>1264</td>
<td>0.25</td>
<td>2923</td>
<td>7</td>
<td>22</td>
<td>1117</td>
<td>44.8</td>
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<tr>
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<td>27.73</td>
<td>1495</td>
<td>0.25</td>
<td>2999</td>
<td>7</td>
<td>23</td>
<td>1147</td>
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<tr>
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<td>0.34</td>
<td>3817</td>
<td>8</td>
<td>32</td>
<td>1576</td>
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<td>2397</td>
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<td>83.23</td>
<td>2891</td>
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<td>7737</td>
<td>7</td>
<td>68</td>
<td>3339</td>
<td>34.2</td>
</tr>
</tbody>
</table>

| Table 3 | CALCULATED VALUES FOR C1 COLD WATER STREAM |
Table 4
VALUES CALCULATED FOR C2 COLD WATER STREAM

<table>
<thead>
<tr>
<th>No. det.</th>
<th>mC2 10^{-3}, kg/s</th>
<th>QC2, W</th>
<th>wC2, m/s</th>
<th>ReC2</th>
<th>PrC2</th>
<th>NuC2</th>
<th>αC2, W/m²°C</th>
<th>tC2, °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2.773</td>
<td>1156</td>
<td>0.04</td>
<td>509</td>
<td>7</td>
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<td>1187</td>
<td>30.7</td>
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<tr>
<td>2</td>
<td>2.773</td>
<td>1287</td>
<td>0.04</td>
<td>517</td>
<td>7</td>
<td>24</td>
<td>1191</td>
<td>32.9</td>
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<td>1198</td>
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<td>0.09</td>
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<td>1690</td>
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<td>1389</td>
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<td>51</td>
<td>2489</td>
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<tr>
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<td>3112</td>
<td>0.20</td>
<td>2063</td>
<td>8</td>
<td>51</td>
<td>2492</td>
<td>28.8</td>
</tr>
</tbody>
</table>

Reynolds number: \( Re_{C2} = \frac{d_{in} \cdot w_{C2} \cdot \rho_{C2}}{\mu_{C2}} \)  

where \( d_{in} = d_{in1} - d_{out2} \).  

\( w_{C2} = \frac{4 \cdot V_{C2}}{\pi \cdot (d^2_{in} - d^2_{out})} \)

and Prandtl number: \( Pr_{C2} = \frac{C_{pC2} \cdot h_{C2}}{\lambda_{C2}} \)

The \( (Pr_{C2} / Pr_{w2})^{0.25} \) ratio from equation (18), where \( Pr_{w2} \) is the Prandtl number calculated with the physical characteristics of the wall temperature, \( t_{w2} \), was considered equal to 1.

Calculating \( \alpha_{C2} \) by using equation (19), the average temperature of the intermediate tube wall, \( t_{w2} \), has been calculated using equation (23).

\[ t_{w2} = t_{H2} + \frac{Q_{C2}}{\alpha_{C2} \cdot A_{out2}} \]  

where \( t_{C2} = (t_{C21} + t_{C2e}) / 2 \) and \( A_{out2} = \pi \cdot d_{out2} \cdot L_{2} \).

Due to the fact that the \( t_{w2} > t_{H} \) for equations (16) and (17), the \( t_{w2} \) values are not presented in the paper. In this paper are shown the calculated values by applying the relationship proposed by Zukauskas, equation (18).

In table 4 are presented the main values obtained by experimental data processing for C2 stream which flows through the outer annular space.

Table 5
EXPERIMENTAL AND CALCULATED MEASUREMENTS FOR HOT WATER STREAM

<table>
<thead>
<tr>
<th>No. det.</th>
<th>mh1 10^{-3}, kg/s</th>
<th>Qh1, W</th>
<th>wh1, m/s</th>
<th>th1, °C</th>
<th>ah1, W/m²°C</th>
<th>ReH</th>
<th>PrH</th>
<th>Nuexp</th>
<th>Nu_calc</th>
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</thead>
<tbody>
<tr>
<td>1</td>
<td>41.05</td>
<td>2227</td>
<td>0.11</td>
<td>35.9</td>
<td>1135</td>
<td>2360</td>
<td>3.51</td>
<td>20.63</td>
<td>20.93</td>
</tr>
<tr>
<td>2</td>
<td>68.42</td>
<td>2570</td>
<td>0.18</td>
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<td>1412</td>
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<td>3.39</td>
<td>25.54</td>
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</tr>
<tr>
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<td>3034</td>
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<td>7453</td>
<td>3.30</td>
<td>38.12</td>
<td>37.11</td>
</tr>
<tr>
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<td>3.70</td>
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<td>20.83</td>
</tr>
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<td>40.80</td>
<td>38.65</td>
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<tr>
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<td>6045</td>
<td>0.36</td>
<td>31.5</td>
<td>2172</td>
<td>7836</td>
<td>3.44</td>
<td>39.37</td>
<td>38.62</td>
</tr>
</tbody>
</table>

In order to determine the partial coefficient of heat transfer \( \alpha_{H} \) for the hot water, Newton’s law has been used:

\[ Q_{H} = \alpha_{H_exp} \cdot A \cdot (t_{H} - t_{w}) \]  

\[ A = A_{out1} + A_{in2} \]  

where: \( A_{exp1} = \pi \cdot d_{in1} \cdot L_{1} \), \( A_{w1} = \pi \cdot d_{out1} \cdot L_{2} \), \( t_{H} = (t_{H1} + t_{H2}) / 2 \) and \( t_{w} = (t_{w1} + t_{w2}) / 2 \).

From equation (24), the value of \( \alpha_{H} \) was calculated:

\[ \alpha_{H} = \frac{Q_{H}}{A \cdot (t_{H} - t_{w})} \]  

Applying equation (26), an experimental value of the partial coefficient of heat transfer is obtained. This allows the experimental calculation of Nusselt’s number, \( Nu_{exp} \), using the following equation:

\[ Nu_{exp} = \frac{\alpha_{H} \cdot d_{i}}{\lambda_{H}} \]  

where \( d_{i} = d_{out2} - d_{in1} \).

The experimental values obtained for \( \alpha_{H} \) and \( Nu_{exp} \) were mathematically processed in Excel spreadsheet application. The linear regression was applied to obtain the Nusselt number calculated, \( Nu_{calc} \), by using the correlation \( Nu_{H} = a \cdot Re_{H} \cdot Pr_{H}^{-0.25} \cdot \left( \frac{d_{i}}{L} \right)^{s} \).

The Reynolds criteria corresponding to the inner annular space used in the correlation is expressed as follows:
The value of the representative measurements obtained when processing the experimental data for hot water stream are presented in table 5.

Results and discussions

The correlation obtained following the mathematical modeling, by applying linear regression, contains the geometric simplex \((d_{h}/L)^{2}\) because it is considered that geometrical capacities, \(d_{h}\) and \(L\), influence the heat transfer in the studied heat exchanger. The exponent of the Prandtl criterion was considered equal to 1/3, value used in the majority of correlations. Through the attempts of modeling applied the following correlation has been obtained within a tolerance of \(\pm 8\%\):

\[ Nu_{H} = 2.718 \cdot Re_{H}^{0.597} \cdot Pr_{H}^{1/3} \cdot \left(\frac{d_{h}}{L}\right)^{2/3} \]

\[ \text{and Prandtl number is } Pr_{H} = \frac{c_{p} \cdot H}{\lambda_{H}} \]

The square average tolerance between \(Nu_{exp}\) and \(Nu_{calc}\) tends towards 1 as observed in figure 3, which proves that the obtained correlation verifies the actual heat exchange performed in the laboratory setup.

Conclusions

This paper established an algorithm for the calculation of partial coefficient of heat transfer for a fluid which flows through an inner annular space of a triple concentric-tube heat exchanger in transition regime. The flow regimes in heat exchanger are: transition in the central tube and the inner annular space and laminar in the outer annular space.

A new correlation developed for design purposes on heat transfer devices, such as triple concentric-tube or double concentric-tube heat exchangers was obtained. The correlation obtained is

\[ Nu_{H} = 2.718 \cdot Re_{H}^{0.597} \cdot Pr_{H}^{1/3} \cdot \left(\frac{d_{h}}{L}\right)^{2/3} \]

and it molds the heat exchange for Reynolds values that go from 2264 to 7893 and for velocities values between 0.11 and 0.36 m/s. The practical applicability of the obtained correlation in the study applies for Prandtl values between 3.30 and 3.70, space characterized by the diameter ratio of \(d_{out}/d_{in} = 14/26\) and relatively small tube lengths. For these conditions of applicability, the error percentage of the mathematical relation was determined as being \(8\%\). The obtained heat transfer coefficient, \(\alpha_{H}\) for hot water has values between 1116 and 2254 W/m\(^2\)°C.

Nomenclature

- \(V\) – volumetric flow rate, L/h;
- \(Q\) – heat flow rate, W;
- \(m\) – mass flow rate, kg/s;
- \(c_{p}\) – specific heat, J/kg·°C;
- \(t\) – temperature, °C;
- \(w\) – linear average velocity, m/s;
- \(L\) – length, m;
- \(d\) – diameter, m;
- \(A\) – heat transfer area, m\(^2\).

Subscripts

- \(C1\) – cold fluid in the central tube;
- \(C2\) – cold fluid in the exterior tube;
- \(H\) – hot fluid in the intermediary tube;
- \(i\) – inner;
- \(in\) – inlet;
- \(o\) – outer;
- \(out\) – outlet;
- \(w\) – wall;
- \(h\) – hydraulic;
- \(t\) – thermic;
- \(1\) – central tube;
- \(2\) – intermediate tube (inner annular space);
- \(3\) – outer tube (outer annular space).

Greek letters

- \(\rho\) – density, kg/m\(^3\);
- \(\mu\) – dynamic viscosity, kg/m·s;
- \(\lambda\) – thermal conductivity, W/m·°C;
- \(\alpha\) – partial coefficient of heat transfer, W/m\(^2\)°C.

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