PERFORMANCE ANALYSIS OF DOUBLE-PIPE FINED HEAT EXCHANGER

Naser Sahiti  
University of Prishtina  
Faculty of Mechanical Engineering  
Kodra e diellit, 10000 Prishtina  
Kosovo

Kastriot Buza  
University of Prishtina  
Faculty of Education  
Rr. Agim Ramadani, 10000 Prishtina  
Kosovo

Florent Bunjaku  
University of Prishtina  
Faculty of Education  
Rr. Agim Ramadani, 10000 Prishtina  
Kosovo

ABSTRACT
Double-pipe heat exchanger as simplest solution of this heat transfer devices has found application in chemical industry for almost one century. There exists a variety of constructions of such heat exchangers such as double-pipe straight heat exchanger, double-pipe U-tube heat exchangers and multitube units. All this constructions might be equiped with fins in order to increase heat transfer. Fins applied for these purpose might be of longitudinal, circular, helical or other forms. Current paper tends to bring new insight regarding the optimization of double-pipe heat exchanger with longitudinal fins. Apart of analysis of thermal performance and fluid dynamic characteristics, an analysis from view point of thermodynamical irreversibilities and the overall heat exchanger performance is provided. The optimization aspects are discussed from view point of different equivalent diameter of annulus space, different flow length and different number of fins employed.

Keywords: Double-pipe heat exchanger, fins, performance

1. INTRODUCTION
The double-pipe heat exchanger, consisting of two coaxial pipes, is simple to fabricate and relatively easy for maintenance and cleaning. However, in order to increase the performance of such heat exchanger, commonly longitudinal straight fins have been applied. In that way, increase of heat transfer area is achieved, without excessive increase of pressure drop. Nevertheless, final overall performance depends on number of fins, thickness of fins and cross section area of annular flow space. This because, the increase of number of fins results on larger heat transfer area- means larger heat transfer rate, but it may end in extremely large pressure drop which may overcome benefits related to heat transfer. Hence, in designing process, there exists a kind of trade off between benefits related to heat transfer and drawback related to pressure drop. In order to properly address this issues, one needs to handle with corresponding equation related to heat transfer and pressure drop, but also with thermodynamics of heat transfer process such as analysis of related irreversibility’s.

2. PREDICTION OF HEAT TRANSFER AND PRESSURE DROP
Heat transfer and pressure drop of heat exchangers, is generally presented in terms of dimensionless variables such as Nusselt number \((Nu)\) and friction factor \((f)\). For the purpose of current paper, following expression were used to calculate heat transfer and pressure drop in double-pipe heat
exchanger. For the tube side Nusselt number is calculated based on Gnielinski equation preferred in the literature [1].

\[
Nu = \frac{h \cdot d}{k} = \frac{(\zeta / 8)(Re - 1000)Pr}{1.0 + 12.7(\zeta / 8)^{0.625} (Pr^{0.33} - 1)}, \quad 1 < Pr < 10^6 \text{ and } 3000 < Re < 5 \times 10^6
\]  

(1)

Where \(d\) = tube diameter, \(Pr\) = Prandtl number

Friction factor is calculated based on following Petukhov equation:

\[
\zeta = \frac{1}{(0.79 \cdot \ln Re - 1.64)^2}
\]  

(2)

For the finned annular space (shell side) following equation was used for Nusselt number [2]:

\[
Nu = 0.01783 \cdot Re^{0.835} \cdot Pr^{0.33}, \quad Re > 10000
\]  

(3)

Where \(d_h\) = hydraulic diameter of finned annular space, \(Pr\) = Prandtl number

Friction factor is estimated based on following equation[3]:

\[
f = 0.109 \cdot Re^{-0.255}, \quad Re > 2100
\]  

(4)

Equivalent diameter (alternatively known as hydraulic diameter) of finned annular space is calculated based on geometry of finned annular space (fig. 2).

Equivalent diameter for calculation of heat transfer is derived based on assumption that the tip of the fin is adiabatic:

\[
d_{eh} = \frac{4A_e}{P_{wh}} = \frac{4\left(\pi D_i^2/4 - \pi d_0^2/4 - n_f b h\right)}{\pi d_0 - n_f + 2hn_f}
\]  

(5)

Equivalent diameter for calculation of pressure drop:

\[
d_{ef} = \frac{4A_e}{P_{wh}} = \frac{4\left(\pi D_i^2/4 - \pi d_0^2/4 - n_f b h\right)}{\pi d_0 - n_f + 2hn_f + \pi D_i}
\]  

(6)

Minimum cross section flow area, relevant for calculation of fluid flow velocity which appears in \(Re\) is calculated based on following expression:
Heat exchanger is considered to be single pass, hence no pressure drop due to change of the flow direction is considered.

3. IRREVERSIBILITIES OF HEAT EXCHANGER AND OVERALL PERFORMANCE

In general thermodynamic irreversibility’s related to operation of a heat exchanger may be attributed to finite temperature differences between heat exchanger working fluids and to finite pressure drop of corresponding fluid streams. Such irreversibility’s results in entropy generation which may be used to quantitatively describe the quality of the heat transfer process within a given heat exchanger.

For current analysis, heat exchanger is analyzed under steady state conditions and is considered to be an open adiabatic system with respect to outside boundaries (normally heat exchangers are thermally insulated with respect to surrounding). By considering that both working fluids are incompressible liquids, flowing in opposite directions, following equation may be used for calculation of entropy generation rate on both fluid sides [3,4]:

$$\dot{S}_{gen} = m_h c_{ph} \ln \left( \frac{T_{ho}}{T_{hi}} \right) + m_c c_{pc} \ln \left( \frac{T_{co}}{T_{ci}} \right) + m_h \Delta P_h \frac{\ln \left( \frac{T_{ho}}{T_{hi}} \right)}{\rho_h T_{ho} - T_{hi}} + m_c \Delta P_c \frac{\ln \left( \frac{T_{co}}{T_{ci}} \right)}{\rho_c T_{co} - T_{ci}}$$

(8)

Where subscript \( h \) stays for the hot fluid and \( c \) for the cold fluid.

Assessing of heat exchanger performance based on level of irreversibility present, usually is performed by comparison of dimensionless entropy generation rate under different operating conditions. Dimensionless entropy generation known as entropy generation number within current paper is defined as follows:

$$N_s = \frac{\dot{S}_{gen}}{m_c c_{pc}}$$

(9)

In order to demonstrate the value of \( N_s \) for evaluation a heat exchanger, it is assumed that current double-pipe heat exchanger is made from steel pipes in counterflow arrangement and is used to heat Dowtherm (a heat transfer liquid) with an inlet temperature of 15 °C using waste hot water with inlet temperature of 95 °C. Further it has been assumed that colder fluid (Dowtherm) is flowing through annulus part of heat exchangers, employing longitudinal fins made from steel (fig. 2) whereas the hotter fluid (water) is flowing in opposite direction through inner pipe. Fluid flow velocity of dowtherm has been assumed to vary from 3 m/s to 14 m/s for which \( Re > 10000 \) are obtained it means eq. 3 might be used to derive \( Nu \) on annulus side. Fluid flow velocity of water has been assumed to vary from 0.5 m/s to 5 m/s, which are common in such systems and for which \( Re \) number within the range of eq. 1 are obtained. Fluid flow properties were considered constant. The values of such properties used for calculations were derived based on mean temperatures of fluids (table 1), [4]:

Table 1. Physical properties of heat exchanger fluids

<table>
<thead>
<tr>
<th></th>
<th>Cold fluid (dowtherm)</th>
<th>Hot fluid (water)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mean temperature (°C)</td>
<td>40</td>
<td>85</td>
</tr>
<tr>
<td>Density (kg/m³)</td>
<td>1044</td>
<td>969</td>
</tr>
<tr>
<td>Specific heat (J/kgK)</td>
<td>1622</td>
<td>4197</td>
</tr>
<tr>
<td>Thermal conductivity (W/mK)</td>
<td>0.138</td>
<td>0.676</td>
</tr>
<tr>
<td>Kinematic viscosity (m²/s)</td>
<td>2.59·10⁻⁹</td>
<td>3.21·10⁻⁷</td>
</tr>
<tr>
<td>Prandtl number (-)</td>
<td>31.7</td>
<td>1.93</td>
</tr>
</tbody>
</table>

Calculations were performed for different number of fins and different flow length. It should be noted that based on eq. 5 and 6, different number of fins results in different equivalent diameter. In fig. 3 are presented only results for 16 and 32 fins for flow length of 5 m, 10 m, 15 m and 20 m:
Overall performance of current double-pipe heat exchanger is assessed by direct plot of the heat transfer rate per unit volume \( \dot{q}_v \) versus the required pumping power per unit heat exchanger volume \( e_v \), [5]

4. CONCLUSIONS
Current work demonstrates that a minimum entropy generation number \( N_s \) can be found for each flow length of double pipe-heat exchanger and for each number of fins employed. Such minimum indicates the flow regime for which the heat exchanger perfumes optimally from view point of thermodynamics. In addition, fig. 4 demonstrates that heat exchanger configurations with lowest entropy generation number results in a better overall performance, means higher heat transfer rate for same pumping power.

5. REFERENCES