Design and Thermal Analysis of Shell and Tube Heat Exchanger

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Abstract: Heat exchangers are used in a number of applications in various industries. The present study focuses on calculation of shell and tube heat exchanger design with minimum pressure drop. Calculation of heat exchanger design has been done by considering two different coolant cases: water at 98°C entering the exchanger and its outlet at 60°C. In one case, water is used as coolant, and in another case, R134a is used. This study also includes different coolant temperature usage, and MATLAB software is used to analyze pressure drop distribution. When a heat exchanger is placed into a thermal transfer system, a temperature drop is required to transfer heat. The magnitude of this temperature drop can be decreased by utilizing a larger heat exchanger, but this will result in a large pressure drop, which increases the cost of the heat exchanger. Economic considerations are important in engineering design, and in a complete engineering equipment, not only the thermal performance characteristics but also the minimum pressure drop are required.

Keywords: overall heat transfer coefficient, heat transfer rate, heat transfer area, pressure drop, pressure distribution.

1. INTRODUCTION

A heat exchanger is a device in which heat is transferred between a warmer and a colder substance, usually fluid. There are various types of heat exchangers:

1. The type of heat exchanger (a) recuperator and (b) regenerator.
2. The type of heat exchanger process between the fluids (a) indirect or transmural and (b) direct contact,[1]
3. Thermodynamic phase or state of the fluids (a) single phase (b) evaporation or boiling (c) condensation,[1]
4. The type of construction or geometry (a) tubular (b) plate and (c) extended or finned surface.[1]

Figure 1. Shell and Tube Heat Exchanger with Segmental baffles: Two Tube Pass, One Shell Tube[1]

In the design of heat exchangers, it is important to specify whether the fluids are mixed or unmixed, and which of the fluid is mixed. It is also important to balance the temperature drop by obtaining approximately equal heat transfer coefficient on the exterior and interior of the tubes. If this is not done, one of the thermal resistances may be unduly large and cause an unnecessarily high overall temperature drop for a given rate of heat transfer, which in turn demands larger equipment and results in poor economics. The shell-and-tube heat exchanger illustrated in Fig (1) has fixed tube sheets at each end, and the tubes are welded or expanded into the sheets.[1]

This type of construction has the lowest initial cost but can be used only for small temperature differences between the hot and cold fluids because no provision is made to prevent thermal stresses due to the differential expansion between the tubes and the shell. Another disadvantage is the tube bundle cannot be removed for cleaning. These drawbacks can be overcome by modification of the basic design in Fig(2).

1. Shell cover
2. Floating head
3. Vent connection
4. Floating head backing device
5. Shell cover end flange
6. Transverse baffles or support plates
7. Shell
8. Tie rod and spacers
9. Shell nozzle
10. Impingement
11. Stationary tube sheet
12. Channel nozzle
13. Channel
14. Lifting ring
15. Pass partition
16. Channel cover
17. Shell channel end flange
18. Support saddles
19. Heat transfer tube
20. Test connection
21. Floating head flange
22. Drain connection
23. Floating tube sheet

Figure 2. Shell and Tube Heat Exchanger with Floating Head [1]

In this arrangement one tube sheet is fixed but the other is bolts to a floating-head cover that permits the tube bundle to move relative to the shell. The floating tube sheet is clamped between the floating head and flanges so that it is possible to remove the bundle for cleaning. The heat exchanger shown in Fig (2) has one shell pass and two shell passes.

In the design and selection of a shell-and-tube heat exchanger, the power requirement and the initial cost of the unit must be considered. Show that the smallest possible pitch in each direction result in the lowest power requirement for a specified rate of heat transfer, since smaller values of pitch also permit the use of a smaller shell, the cost of the unit is reduced when the tubes are closed packed. There is little difference in performance between inline and staggered arrangement, but the former are easier to clean. [1]

Heat transfer in a heat exchanger usually involves convection in each fluid and conduction through the wall separating the two fluids. In the analysis of heat exchangers, it is convection to work with an overall heat transfer coefficient $U$ that account for the contribution of all these on heat transfer.

The rate of heat transfer between the two fluids at a location in heat exchanger depends on the magnitude of the temperature difference at that location, which varies along the heat exchanger. In the analysis of heat exchangers, it is usually convection to work with the logarithmic mean temperature difference LMTD, which is an equivalent mean temperature difference between the two fluids for the entire heat exchanger. [5]

2. LOG MEAN TEMPERATURE DIFFERENCE

The temperature of fluids in a heat exchanger are generally not constant but vary from point to point as heat flows from the hotter to the colder fluid.

Even for a constant thermal resistance, the rate of heat flow will therefore vary along the path of the exchangers because its value depends on the temperature difference the cold fluid and hot fluid in that section. Fig (3) illustrate the changes in temperature that may in either or both fluid in a simple shell and tube exchanger. [1] The distances between the solid lines are proportional to the temperature differences $\Delta T$ between the two fluids.

$$\Delta T = (\Delta T_a - \Delta T_b) / \ln(\Delta T_a / \Delta T_b)$$

Where the subscripts a and b refer to the respective ends of the exchanger and $\Delta T_a$ is the temperature difference between the hot and cold fluids streams at the inlet while $\Delta T_b$ is the temperature difference at the outlet end as shown in Fig (3).

Figure 3 (a) Temperature Distribution in Single Pass Parallel Flow Heat Exchanger [1]

Figure 3 (b) Temperature Distribution in Single Pass Parallel Flow Heat Exchanger [1]
temperature difference $\Delta T$ for the entire heat exchanger, defined by

$$ Q = UA \Delta T $$

The ordinate of each is the correction factor $F$. To obtain the true mean temperature for any of these arrangements, the LMTD calculate for correction factor, that is

$$ \Delta T_m = \text{LMTD} \times F $$

The values shown on the abscissa are for the dimensionless temperature-difference ratio $P$ and $Z$:

$$ P = \frac{(T_{h,o} - T_{h,i})}{(T_{c,i} - T_{h,i})} $$

$$ Z = \frac{(T_{c,i} - T_{c,o})}{(T_{h,o} - T_{h,i})} $$

Where the subscripts $h$ and $c$ refer to the hot and cold fluid respectively.

$T_{h,i}$ = inlet temperature of hot fluid

$T_{h,o}$ = outlet temperature of hot fluid

$T_{c,i}$ = inlet temperature of cold fluid

$T_{c,o}$ = outlet temperature of cold fluid

$U$ = overall heat transfer coefficient of clean exchanger

$$ U = \frac{Q}{A \Delta T} $$

Where,

$$ h_i = \text{convection heat transfer hot coefficient} = 7500 \text{ W/m}^2\text{K} $$

$$ h_o = \text{convection heat transfer cold coefficient for water} = 8000 \text{ W/m}^2\text{K} $$

$$ A_i = \text{area of hot fluid} $$

$$ A_o = \text{area of cold fluid} $$

$$ A_k = \text{area of copper tube} $$

$$ x = \text{thickness of copper tube} = 5 \times 10^{-3} \text{ m} $$

$$ k = \text{thermal conductivity of copper tube} = 399.375 \text{ W/mK} $$

$$ U = 3692.04 \text{ W/m}^2\text{K} $$

4. FOULING FACTOR

The overall heat transfer coefficient of a heat exchanger under some operating conditions, especially in the process industry, often cannot be predicted from thermal analysis alone. During operation with most liquids and some gases, a deposit gradually builds up on the heat transfer surface. The deposit may be rust, boiler scale, silt, coke, or any number of things. Its effects, which is referred to as fouling, is to increase the thermal resistance. The manufacturer cannot usually predict the nature of the deposit or the rate of fouling. Therefore, only the performance of clean exchanger can be guaranteed. The thermal resistance of the deposit can generally be obtained only from actual tests or from experience. If performance tests are made on a clean exchanger and repeated later after the unit has been in service for some time, the thermal resistance of the deposit (or fouling factor) $R_d$ can be determined from the relation.

$$ R_d = \frac{1}{U_d} - \frac{1}{U} $$

Where,

$$ U = \text{overall heat transfer coefficient of clean exchanger} = 3692.04 \text{ W/m}^2\text{K} $$

$$ U_d = \text{overall heat transfer coefficient after fouling has Occurred} $$

$$ R_d = \text{fouling factor (or unit thermal resistance) of Deposit} = 0.0002 $$
5. CALCULATION OF HEAT TRANSFER RATE

Heat exchanger usually operate for long periods of time with no change in their operating conditions. Therefore they can be modeled as steady-flow devices. As such, the mass flow rate of each fluid remains constant, and the fluid properties such as temperature and velocity at any inlet or outlet remain the same. Also, the fluid streams experience little or no change in their velocities and elevations, and thus the kinetic and potential energy changes are negligible. The specific heat of a fluid, in general, changes with temperature. The streams experience little or no change in their velocities and elevations, and thus the fluid outlet remain the same. Also, the fluid with a small heat capacity rate will experience a large temperature change, and the fluid with a small heat capacity rate will experience a large temperature change. [5]

The idealization state above are closely approximated in practice, and they greatly simplify of a heat exchanger with little sacrifice of accuracy. Therefore, they are commonly used. Under these assumptions, the first law of thermodynamics requires that the rate of heat transfer from the hot fluid be equal to the rate of heat transfer to the cold one. That is,

\[ Q = m_c C_{ph} (T_{ci} - T_{co}) \]

Where,
- \( Q \) = heat transfer rate
- \( m_c \) = mass flow rate of cold fluid
- \( C_{ph} \) = specific heat of cold fluid
- \( T_{ci} \) = inlet temperature of cold fluid
- \( T_{co} \) = outlet temperature of cold fluid

\[ \Delta T_{av} = \frac{(T_{hi} - T_{co}) + (T_{co} - T_{hi})}{2} \]

Where the subscripts c and h stand for cold and hot fluids, respectively. Note that the heat transfer rate \( Q \) is taken to be a positive quantity, and its direction is understood to be from the hot fluid to the cold one in accordance with the second law of thermodynamics. In heat exchanger analysis, it is often convection to combine the product of the mass flow rate and the specific heat of a fluid into a single quantity.

The heat capacity rate of a fluid stream represents the rate of heat transfer needed to change the temperature of the fluid stream by 1°C as it flows through a heat exchanger. Note that in a heat exchanger, the fluid with a large heat capacity rate will experience a small temperature change, and the fluid with a small heat capacity rate will experience a large temperature change. [5]

That is, the heat transfer rate in a heat exchanger is equal to the heat capacity rate of either fluid multiplied by the temperature change of that fluid. Note that the only time of temperature rise of a cold fluid is equal to the temperature drop of the hot fluid is when the heat capacity rates of the two fluids are equal to each other. [5]

### 5.1. Calculation of Heat Transfer Area

Heat transfer of hot water is determined by the following equation,

\[ Q = m_c C_{ph} (T_{hi} - T_{co}) = U A F \Delta T_{av} \]

Where,
- \( U \) = overall heat transfer coefficient
- \( A \) = heat transfer area

### Table 1. Typical Fouling Factors [2]

<table>
<thead>
<tr>
<th>Type of Fluid</th>
<th>Fouling Factor, ( \text{m}^2\text{°C}/\text{W} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Seawater below 125°F</td>
<td>0.00009</td>
</tr>
<tr>
<td>above 125°F</td>
<td>0.002</td>
</tr>
<tr>
<td>Treated boiler feedwater above 125°F</td>
<td>0.0002</td>
</tr>
<tr>
<td>Fuel oil</td>
<td>0.0009</td>
</tr>
<tr>
<td>Quenching oil</td>
<td>0.0007</td>
</tr>
<tr>
<td>Alcohol vapors</td>
<td>0.00009</td>
</tr>
<tr>
<td>Steam, non-oil-bearing</td>
<td>0.00009</td>
</tr>
<tr>
<td>Industrial air</td>
<td>0.0004</td>
</tr>
<tr>
<td>Refrigerating liquid</td>
<td>0.0002</td>
</tr>
</tbody>
</table>

### Table 2. Comparison Result Table of H₂O and R 134a

<table>
<thead>
<tr>
<th>Name</th>
<th>Symbol</th>
<th>H₂O</th>
<th>R.134a</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coolant inlet temperature</td>
<td>( T_{ci} )</td>
<td>40</td>
<td>-41</td>
<td>°C</td>
</tr>
<tr>
<td>Coolant outlet temperature</td>
<td>( T_{co} )</td>
<td>55</td>
<td>-26</td>
<td>°C</td>
</tr>
<tr>
<td>Heat Loss</td>
<td>( Q )</td>
<td>326.87</td>
<td>326.87</td>
<td>kW</td>
</tr>
<tr>
<td>Convection heat transfer for cold fluid</td>
<td>( h_{col} )</td>
<td>8000</td>
<td>2261.307</td>
<td>W/m²K</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>( k )</td>
<td>399.375</td>
<td>403.425</td>
<td>W/mK</td>
</tr>
<tr>
<td>Over all heat transfer coefficient</td>
<td>( U )</td>
<td>3692.04</td>
<td>1700.8</td>
<td>W/m²K</td>
</tr>
<tr>
<td>Heat transfer area</td>
<td>( A )</td>
<td>2.82</td>
<td>1.71</td>
<td>m²</td>
</tr>
</tbody>
</table>

\[ \Delta T_{av} = \frac{(T_{hi} - T_{co}) + (T_{co} - T_{hi})}{2} \]

Where the subscripts c and h stand for cold and hot fluids, respectively. Note that the heat transfer rate \( Q \) is taken to be a positive quantity, and its direction is understood to be from the hot fluid to the cold one in accordance with the second law of thermodynamics. In heat exchanger analysis, it is often convection to combine the product of the mass flow rate and the specific heat of a fluid into a single quantity.

The heat capacity rate of a fluid stream represents the rate of heat transfer needed to change the temperature of the fluid stream by 1°C as it flows through a heat exchanger. Note that in a heat exchanger, the fluid with a large heat capacity rate will experience a small temperature change, and the fluid with a small heat capacity rate will experience a large temperature change. [5]

That is, the heat transfer rate in a heat exchanger is equal to the heat capacity rate of either fluid multiplied by the temperature change of that fluid. Note that the only time of temperature rise of a cold fluid is equal to the temperature drop of the hot fluid is when the heat capacity rates of the two fluids are equal to each other. [5]
5.2 Pressure Drop for Heat Exchanger

Pressure drop is a major constraint in thermal design of shell and tube heat exchanger is meaningful only when it is optimum and the extent of the optimality is constrained by the pressure drop. Optimization of thermal design requires maximization of overall heat transfer coefficient and effective mean temperature difference. So as to minimize the heat transfer area subject to constraints, pressure drop being the major one [4]. Tube side pressure drop can be lowered in following ways:

- Increasing the shell diameter. Increasing the shell diameter increases tube flow area due to increased number of tubes and, thereby reduces tube flow velocity and, hence, reduces tube side pressure drop. Further, it also means reduced tube length which, too, leads to reduced pressure drop.

- Increasing the tube diameter. Increasing the tube diameter reduces velocity and, thereby, reduce pressure drop. However, tube diameters are standardized and standard outside diameters are limited. Further tube outside diameters more than 1.0inch is generally not desirable as higher tube diameter means higher shell diameter to accommodate required number of tubes due to increased tube pitch which, in turn, means higher cost.

- Increasing the nozzle size. If the nozzles are too small in diameter, their diameter can be increased reasonably to lower the pressure drop.

- Using the shells in parallel. Multiple shells can be used in parallel so that total tube side flow is split and flow velocity is reduced. Consequently, pressure drop is reduce. However, it increases the cost due to the reasons as mentioned for shell side pressure drop.

The pressure drop can be relatively straightforward in a single-pass pipe-in tube heat exchanger or extremely difficult in, say a shell and tube exchanger. Table (3) and (4) are shown pressure drop for water and R134a coolant with varies tube length and diameter.

The pressure drop in a straight run of pipe, given by

\[ \Delta P = fL\frac{v^2}{2gD} \]

Where,
- \( f \) = friction loss
  - 0.023
- \( L \) = length of tube
- \( v \) = velocity of hot fluid
  - 1.24m²
- \( g \) = gravity
  - 9.81m/s²
- \( D \) = diameter of tube
- \( \Delta P \) = pressure drop

<table>
<thead>
<tr>
<th>Length of tube (m)</th>
<th>Volume of tube ( V_\text{tube} ) m³</th>
<th>Volume of cold fluid ( V_\text{cold} ) m³</th>
<th>Pressure drop for hot fluid ( \Delta P ) m</th>
</tr>
</thead>
<tbody>
<tr>
<td>38</td>
<td>0.192</td>
<td>0.012</td>
<td>2.69</td>
</tr>
<tr>
<td>22</td>
<td>0.011</td>
<td>0.022</td>
<td>1.56</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>No.</th>
<th>Length(m)</th>
<th>Diameter(m)</th>
<th>Pressure drop(m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>38</td>
<td>0.0254</td>
<td>2.6631</td>
</tr>
<tr>
<td>2</td>
<td>34</td>
<td>0.0279</td>
<td>2.2009</td>
</tr>
<tr>
<td>3</td>
<td>31</td>
<td>0.0305</td>
<td>1.8493</td>
</tr>
<tr>
<td>4</td>
<td>29</td>
<td>0.0330</td>
<td>1.5758</td>
</tr>
<tr>
<td>5</td>
<td>27</td>
<td>0.0356</td>
<td>1.3587</td>
</tr>
<tr>
<td>6</td>
<td>25</td>
<td>0.0381</td>
<td>1.1836</td>
</tr>
<tr>
<td>7</td>
<td>24</td>
<td>0.0406</td>
<td>1.0403</td>
</tr>
<tr>
<td>8</td>
<td>22</td>
<td>0.0432</td>
<td>0.9215</td>
</tr>
<tr>
<td>9</td>
<td>21</td>
<td>0.0457</td>
<td>0.8219</td>
</tr>
<tr>
<td>10</td>
<td>20</td>
<td>0.0483</td>
<td>0.7377</td>
</tr>
</tbody>
</table>

Table 4. Pressure Drop for R134a with Various Length and Diameter

<table>
<thead>
<tr>
<th>No.</th>
<th>Length(m)</th>
<th>Diameter(m)</th>
<th>Pressure drop(m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>19.5497</td>
<td>0.0254</td>
<td>1.3848</td>
</tr>
<tr>
<td>2</td>
<td>17.7725</td>
<td>0.0279</td>
<td>1.1445</td>
</tr>
<tr>
<td>3</td>
<td>16.2915</td>
<td>0.0305</td>
<td>0.9617</td>
</tr>
<tr>
<td>4</td>
<td>15.0383</td>
<td>0.0330</td>
<td>0.8194</td>
</tr>
<tr>
<td>5</td>
<td>13.9641</td>
<td>0.0356</td>
<td>0.7065</td>
</tr>
<tr>
<td>6</td>
<td>13.0332</td>
<td>0.0381</td>
<td>0.6155</td>
</tr>
<tr>
<td>7</td>
<td>12.2186</td>
<td>0.0406</td>
<td>0.5409</td>
</tr>
<tr>
<td>8</td>
<td>11.4998</td>
<td>0.0432</td>
<td>0.4792</td>
</tr>
<tr>
<td>9</td>
<td>10.8610</td>
<td>0.0457</td>
<td>0.4274</td>
</tr>
<tr>
<td>10</td>
<td>10.2893</td>
<td>0.0483</td>
<td>0.3836</td>
</tr>
</tbody>
</table>

Figure 6 shown the pressure drop with varies tube diameters and figure 7 shown the tube length with varies tube diameters.
6. CONCLUSION

In this paper, counter flow shell and tube heat exchanger design are calculated. Hot water enter at 98ºC and at exist 60ºC. Density of the working fluid is used average temperature. Tube material is copper. Tube diameter and length are varies with the calculation of pressure drop. Firstly, cold water is used for coolant inlet temperature 40ºC and outlet temperature 55ºC. The velocity of the hot water is 1.24m/s and flow is turbulent. The roughness of the copper pipe is 0.0015mm. Finally, the coolant is R134a. Inlet temperature of the coolant is -41ºC and outlet temperature is -26ºC. In this condition the heat transfer area of R134a is less than using the water coolant. Pressure drop is also less than coolant is water for the same diameter. Now a day in shell and tube heat exchanger, tube side pressure drop is 7m used.

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8. REFERENCES


