Evaluating and optimizing performance of shell and tube heat exchanger using excel-solver

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Abstract. Heat exchangers in the oil and gas industry play an important role as heat exchangers. In this research, shell-and-tube heat exchanger was chosen because this type of heat exchanger is the most widely used in the industry in general and in the oil and gas industry in particular. This research was conducted to determine the performance of shell and tube type heat exchangers in terms of thermal and hydraulic. Data acquisition is done by observing in the field. So that the data processing and calculation results are obtained as follows: Fouling factor operation 0.02823 (hr.ft$^2$.F/Btu) higher than design 0.00485 (hr.ft$^2$.F/Btu), Shell side pressure drop of 19.73 psi has over the maximum permissible of 10 psi, so it is time for cleaning, tube side pressure drop of 1.47 psi is still under the maximum permissible of 10 psi the effectiveness of the operation is 57.47% less than design value of 68.73%, and the simulation using excel-solver on the dirt (operating) condition result is effectiveness can be increased to 79.93%, shell side and tube side pressure drop 6.2 psi and 9.98 psi, shell side and tube side velocity 0.98 ft/s and 9.5 ft/s respectively.

1. Introduction

A heat exchanger specifically a shell-and-tube type is heat transfer mechanical equipment that is widely used in various industrial fields, such as petroleum, petrochemical, energy and others so. The heat exchanger is an important equipment of the many equipment components in the industry. Given the function of this equipment to transfer heat that will be used for the needs of the heating process, if the performance of this equipment has decreased, the heat transfer process will also experience a decrease too, so the condition of the performance of the equipment must always be monitored [1-4]. The function of the heat exchanger, as the name suggests, is for transfer’s heat from one fluid with a higher temperature to one fluid others with lower temperatures. The mechanism of heat transfer in the shell-and-tube type occurs by conduction and convection. Conduction heat transfer or known as conduction heat is the process of heat transfer if heat flows from a temperature high to a place where the temperature is lower with fixed heat conducting media. While heat transfer by convection is heat transfer occurs between a solid surface with fluid that flows around it with using a medium of liquid / gas fluid. By doing a simulation, by conducting a trial of changing certain parameters, it is expected to get the most effective heat exchanger operation that can be done.
2. Method

2.1. Shell and tube
In the shell and tube heat exchanger there are two sides of the fluid flow stream, namely the inside of the tube and the outside of the tube (shell side), where both sides are generally of different fluid types [3]. But in the calculation is almost the same, only different parameter values. Some things that need to be calculated on the side of fluid flow are as follows:

2.1.1. Reynolds number

2.1.1.1. Shell side and tube side Reynolds number (Re). The Reynolds number shell side and tube side determined by general equations bellow:

\[ \text{Re}_s = \frac{D_s G_s}{\mu} \]  \hspace{1cm} (1)

\[ \text{Re}_t = \frac{d_t G_t}{\mu} \]  \hspace{1cm} (2)

Where:
- \( d_i \) = inside tube diameter
- \( D_e \) = shell side equivalent diameter
- \( G \) = mass velocity
- \( \mu \) = dynamic viscosity of fluid.

2.1.1.2. Shell side equivalent diameter (\( D_e \)). Equivalent diameter equations on the shell side (\( D_e \)) can be selected as follows:

Square pitch tube lay-out [5]:

\[ D_e = \frac{4\left(P_t^2 - \pi d_i^2/4\right)}{\pi d_o} = \frac{1.27}{d_o} \left(P_t^2 - 0.785d_i^2\right) \]  \hspace{1cm} (3)

Triangular tube lay-out [5]:

\[ D_e = \frac{4\left(0.5P_t(0.86P_t) - 0.5\pi d_o^2/4\right)}{\pi d_o/2} = \frac{1.10}{d_o} \left(P_t^2 - 0.917d_o^2\right) \]  \hspace{1cm} (3)

Where:
- \( d_o \) = inside tube diameter
- \( P_t \) = tube pitch

2.1.2. Mass velocity

2.1.2.1. Fluid mass velocity (G). The general equation for determine of fluid mass velocity (\( G \)) in the shell and tube side is:

\[ G_s = \frac{m_s}{A_s} \]  \hspace{1cm} (4)

\[ G_t = \frac{m_t}{A_t} \]  \hspace{1cm} (5)

Where:
- \( m \) = mass flow rate
- \( A \) = cross sectional area
2.1.3. Cross sectional area. The cross sectional area of the shell and tube can be determined by the following equation:

\[ A_s = CB \frac{d_s}{P_t} = (P_t - d_o)B \frac{d_s}{P_t} \]  \hspace{1cm} (6)

\[ A_i = \frac{\pi d_i^2}{4} \]  \hspace{1cm} (7)

Where:
- \( d_i \) = inside tube diameter
- \( C \) = clearance
- \( B \) = baffle space
- \( d_s \) = inside shell diameter
- \( d_o \) = outside tube diameter
- \( P_t \) = tube pitch

2.2. Heat transfer

In the calculation of heat transfer both on the tube and shell sides are also almost the same, only differing in a few parameters and values. Below are some things that need to be calculated in the heat transfer.

2.2.1. Heat transfer coefficient. Calculation of heat transfer on the inside of the shell and the inside of the tube can be done as follows:

2.2.1.1. Inside tube heat transfer coefficient \( h_i \) for turbulent flow \((Re > 10,000)\). The heat transfer coefficient for turbulent flow inside tube can be determined by the Sieder-Tate equation as follows:

\[ h_i = \frac{kh}{d_i} \left( \frac{\mu}{\mu_w} \right)^{0.14} \]  \hspace{1cm} (8)

Where:
- \( Re \) = Reynolds number,
- \( Pr \) = Prandtl number
- \( \mu = \frac{\mu c_p}{k} \)
- \( d_i \) = inside tube diameter
- \( \mu \) = dynamic viscosity of fluid.
- \( \mu_w \) = dynamic viscosity of fluid base on the wall temperature.
- \( c_p \) = specific heat, heat capacity
- \( k \) = thermal conductivity of fluid

All of fluid properties base on bulk temperature except \( \mu_w \) base on the wall temperature.

2.2.1.2. Inside tube heat transfer coefficient \( h_i \) for transition region \((2,100 < Re < 10,000)\). On this region the heat transfer coefficient can be calculate by the equation as follows [5]:

\[ h_i = \frac{k}{d_i} 0.116 \left[ \frac{R_e^{2/3} - 125}{Pr^{1/3}} \right] \left[ 1 + \left( \frac{D}{L} \right)^{2/3} \right] \left( \frac{\mu}{\mu_w} \right)^{0.14} \]  \hspace{1cm} (9)
2.2.1.3. **Inside tube heat transfer coefficient (h<sub>i</sub>) for laminar flow (Re < 2,100).** For laminar flow, the heat transfer coefficient calculated by the equation as follows [5]:

\[
h_i = \frac{k}{d_i} = 1.86(\text{RePr})^{0.33}\left(\frac{d_i}{L}\right)^{0.33}\left(\frac{\mu}{\mu_w}\right)^{0.14}
\]  

(10)

Where: \(L\) = tube length

2.2.1.4. **Outside tube heat transfer coefficient (h<sub>o</sub>).** The magnitude of the convection heat transfer coefficient on the outside of the tube for heat exchangers with 25% baffle cut can use the following equation.

\[
h_o = \frac{k}{D_e}j_H \left(\frac{c\mu}{k}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}
\]

(11)

Value of \(j_H\) can be determine in the figure 28 [6], or can be use the following equation [7]:

For Reynolds number range: \(2 \times 10^3 < \text{Re} < 1 \times 10^6\)

\[
j_H = 0.36 \left(\frac{D_eG}{\mu}\right)^{0.55}
\]

(12)

For the Reynold number (Re < 2,100) from the figure 28 [6], can be approximated by the following equation:

\[
j_H = 0.5934 \text{Re}^{0.4822}
\]

(13)

2.2.2. **Heat duty**

2.2.2.1. **Heat balance.** In the heat exchange process, the hot fluid will release the heat and is received by the cold fluid, which is ideally the same amount of heat, so the equilibrium equation is:

\[
Q = m_c c_{pc} (T_{co} - T_{ci}) = m_h c_{ph} (T_{hi} - T_{ho})
\]

(14)

Where:

- \(m_c\) = mass flow rate of hot fluid, (kg/s)
- \(m_h\) = mass flow rate of cold fluid, (kg/s)
- \(c_{pc}\) = specific heat of cold fluid, (kJ/kg°C)
- \(c_{ph}\) = specific heat of hot fluid, (kJ/kg°C)
- \(T_{ci}, T_{co}\) = cold fluid temperature inlet, outlet, (°C)
- \(T_{hi}, T_{ho}\) = hot fluid temperature inlet, outlet, (°C)
- \(Q\) = heat duty, (kW)

2.2.3. **Overall heat transfer coefficient.** Overall heat transfer coefficient (\(U\)) calculated based on the outer surface area of the tube.

2.2.3.1. **Dirt condition.** Dirt overall heat transfer coefficient (\(U_f\)) at operating condition can be calculate by \(LMTD\) equation as follows:
Where: 
- \( Q \) = heat duty
- \( A \) = heat transfer surface area
- \( F \) = LMTD correction factor
- \( LMTD \) = Log Mean Temperature Difference

2.2.3.2. Clean condition. The clean overall heat transfer coefficient \((U_c)\) can be calculated using the following equation:

\[
\frac{1}{U_c} = \frac{1}{h_o} + \frac{d_o \ln(d_o/d_i)}{2k} + \frac{d_o}{d_i h_i}
\]

(15)

Where: 
- \( h_o \) = outside tube heat transfer coefficient
- \( h_i \) = inside tube heat transfer coefficient
- \( k \) = thermal conductivity of tube material
- \( d_o \) = outside tube diameter
- \( d_i \) = inside tube diameter

From the two (dirt and clean) overall heat transfer coefficient equations above, then, the amount of total fouling resistant \((R_f)\) can be expressed by the following equation:

\[
R_f = \frac{1}{U_f} - \frac{1}{U_c}
\]

(16)

2.3. Pressure drop
The amount of pressure drop can be determined using the following equation:

2.3.1. Inside tube pressure drop. Pressure drop on the inside of the tube can be calculated by the following equation:

\[
\Delta p_i = 4 fn \frac{L}{D_i} \frac{G_i^2}{2 \rho} \left( \frac{\mu}{\mu_w} \right)^a
\]

(17)

Where: 
- \( a \) = 0.14 (turbulent flow)
- \( a \) = 0.25 (laminar flow)
- \( L \) = tube length one pass tube
- \( n \) = number of pass tube
- \( G_i \) = mass flow velocity
- \( \rho \) = fluid density
- \( f \) = friction factor (fanning)

2.3.1.1. Friction factor. For laminar flow, \( R_e < 2100 \), the friction factor can be calculated by the following equation:
\[ f = \frac{16}{Re} \quad (18) \]

For Reynolds number \( Re > 2100 \), the friction factor equation as follows:

\[ f = 0.0014 + 0.125Re^{-0.32} \quad (19) \]

For multi pass tubes, the return flow of the tube will also experience a pressure loss whose magnitude is as follows:

\[ \Delta p_r = 4n \frac{G_i^2}{2\rho} \quad (20) \]

2.3.1.2. Total pressure drop inside tube. From the two pressure drop equations above, the magnitude of the total pressure drop of the tube side is:

\[ \Delta p = \Delta p_t + \Delta p_r \quad (21) \]

2.3.2. Shell side pressure drop. Pressure drop on the shell side can be calculated by the following equation (Kern Method) [6]:

\[ \Delta p_s = f \left( \frac{N_B + 1}{D_c} \right) \frac{d_s}{s} \frac{\rho w}{\mu} \left( \frac{\mu w}{\mu} \right)^{0.14} \quad (22) \]

Where:
- \( N_B \) = number of baffle
- \( D_c \) = shell side equivalent diameter
- \( d_s \) = inside shell diameter
- \( f \) = friction factor

2.3.2.1. Friction factor \( f \). Equations that can be used to determine the magnitude of the friction factor are as follows:

The friction factor for turbulent flow in the pipeline is used the Wilson, McAdams, and Seltzer equations as follows:

\[ f = 0.0035 + (0.264 / (Re)^{0.42}) \quad (23) \]

Or in another equation as follows:

- For \( Re < 500 \)

\[ f = \exp \left[ 5.1858 - 1.7645 \ln(Re) + 0.13357(\ln(Re))^2 \right] \quad (24) \]

- For \( Re > 500 \), \( f = 1.728Re^{-0.188} \)

- For fluid flow in the shell \( (400 < Re = \frac{G_sD_c}{\mu}) < 10^6) \):

\[ f = \exp(0.576 - 0.19 \ln Re) \quad (25) \]
2.4. Performance evaluation
The performance evaluation of the heat exchanger is carried out by looking at the thermal, hydraulic conditions and their level of effectiveness.

2.4.1. Thermal evaluation. For thermal evaluation, it is carried out by comparing the values of the operating fouling resistance ($R_f$) and the design fouling resistance ($R_d$) as follows:

- If $R_f < R_d$, then the heat exchanger is thermally suitable for operation.
- If $R_f > R_d$, then the heat exchanger is thermally not suitable for operation.

2.4.2. Hydraulic evaluation. For hydraulic evaluation, is performed by comparing the pressure drop of operating conditions with the pressure drop permissible.

- If the pressure drop operation condition ($\Delta p$) less than the pressure drop permissible ($\Delta p_r$), then the heat exchanger is suitable for operation.
- If the pressure drop operation condition ($\Delta p$) greater or equal to the pressure drop permissible ($\Delta p_r$), then the heat exchanger is not suitable for operation.

2.4.3. Effectiveness. The effectiveness of the heat exchanger can be determined by the following equation:

$$\varepsilon = \frac{Q_{act}}{Q_{max}}$$

Where:
- $\varepsilon$ = effectiveness of heat exchanger
- $Q_{act}$ = the actual heat that is transferred
- $Q_{max}$ = the maximum possible heat must be transferred

The actual heat that is transferred and the maximum possible heat must be transferred can be expressed in the following equations:

$$Q_{act} = m_{i}c_{pi}(T_{ci} - T_{co}) = m_{h}c_{ph}(T_{hi} - T_{ho})$$

$$Q_{max} = C_{min}(T_{hi} - T_{ci})$$

Where:
- $C_{min}$ = the smallest of $m_{c}$ between the hot or cold fluid
- $m$ = mass flow rate
- $c_p$ = specific heat of fluid
- $T_{c}$ = cold fluid temperature
- $T_{h}$ = hot fluid temperature
- $i, o$ = inlet, outlet

2.5. Optimization
From the evaluation results obtained whether or not the heat exchanger operates, and after calculating the effectiveness it is obtained by what % the level of the heat exchanger effectiveness means heat transfer.

The small value of effectiveness shows the ability to transfer heat is also small to what should be transferred. A large effectiveness value means that the ability to transfer heat is also large against what should be transferred [8,9]. By varying the operating parameters in accordance with the above effectiveness equation, it is expected to know the best operating conditions of the heat exchanger used.
From the effectiveness equation, to increase the ability to transfer heat (its effectiveness) can be done by raising the $Q_{act}$ or decreasing $Q_{max}$ by assuming the mass rate, fluid temperature can be changed and then simulated by calculating the heat transfer [6,10].

3. Results and discussion

For performance optimization in this research the data used is partly the same as the evaluation data (equipment data and some fluid data is fixed data that has not been changed) and some other changes must be made by trial and error so that the most optimal results are obtained. Table 1 is a data table for simulating.

Table 1. Fluid and HE data.

| Parameter                        | Symbol | Unit | Shell side | Tube side |
|----------------------------------|--------|------|------------|-----------|
| Equivalent, inside diameter      | $D_e,$ $d_i$ | ft   | 0.03       | 0.0487    |
| Outside tube diameter            | $d_o$  | ft   | 0.0625     |           |
| Tube length                      | $L$    | ft   | 40.66      |           |
| Baffle space                     | $B$    | in   | 14         |           |
| Cross sectional area             | $A_s,$ $a_t$ | ft$^2$ | 0.166   | 0.253    |
| Heat transfer area               | $A$    | ft$^2$ | 1.085.766  |           |
| Fluid                            |        |      |            | Decant Oil, Cooling Water |
| Mass flow rate                   | $m_h,$ $m_c$ | Lb/hr | 70,000     | 300,000   |
| Inlet temperature                | $T_{hi},$ $T_{ci}$ | $^O_F$ | 303.46     | 89.6      |
| Outlet temperature               | $T_{ho},$ $T_{co}$ | $^O_F$ | 170        |           |
| Average temperature              | $T_{av}$ | $^O_F$ |            |           |
| Fluid Density                    | $\rho$ | lb/ft$^3$ | 59.69     | 62.06     |
| Specific heat                    | $c_p$  | Btu/lb.$^O_F$ | 0.446   | 0.998     |
| Dynamic viscosity                | $\mu$  | lb/ft.hr | 26.61     | 1.715     |
| Thermal conductivity             | $k$    | Btu/ft.hr.$^O_F$ | 0.0625 | 0.3625   |

From the data table, the following steps to do optimization are:

- Assume capacity ($m_h,$ $m_c$),
- Assume the exit temperature (Tho, or Tco) and the equilibrium of heat and mass balance
- Determine the fluid's properties
- Calculate Reynolds numbers (Res, Ret)
- Calculate the heat transfer coefficient (hi, ho)
- Calculate the clean overall heat transfer coefficient (Uc)
- Calculate the dirt overall heat transfer coefficient (Ud) based on the fouling resistance (Rd) of the evaluation results
- $\epsilon$-NTU method
  - Calculate $C = m \cdot c_p$
  - Specify $C_{min}, C_{max}, C = C_{min} / C_{max}$
  - Calculate $Q_{max} = C_{min} (T_{hi} - T_{ci})$
  - Calculate NTU = U.A / $C_{min}$
  - Determine $\epsilon = f (NTU, C)$ [5,7,11]
  - Calculate $Q_{act} = \epsilon \cdot Q_{max}$
  - Determine $T_{ho}, T_{co}$ based on $Q_{act} = Q_h = Q_c = m \cdot c_p \cdot dT$
  - Use the $T_{ho}, T_{co}$ value to enter step 2)
- For manual iteration, repeat step 2) until it converges (there is no change), then continue by repeating step 1) with another number change.
- For automatic iteration (with solver), step 1) is changed automatically by the solver until the best effectiveness value is obtained [12,13].

From the steps above, we get the results as in table 2.

### Table 2. Evaluation and 1st trial optimization results.

| Parameter                      | Symbol | Unit      | Shell side | Tube side |
|--------------------------------|--------|-----------|------------|-----------|
| Mass flow rate                 | m      | Lb/hr     | 70,000     | 300,000   |
| Inlet temperature              | T      | F         | 303.46     | 89.6      |
| Outlet temperature             | T      | F         | 170        | 103.52    |
| Design Effectiveness           | \(\epsilon\) | % | 68.73 | |
| Operation Effectiveness        | \(\epsilon\) | % | 57.47 | |
| Effectiveness 1st trial        | \(\epsilon\) | % | 55.3 | |
| Design fouling resistant        | \(R_d\) | h.ft\(^2\).F/Btu | 0.00485 | |
| Operation fouling resistant     | \(R_d\) | h.ft\(^2\).F/Btu | 0.02823 | |
| Operation pressure drop        | \(\Delta P\) |Psi | 19.73 | 1.47 |
| Pressure drop 1st trial        | \(\Delta P\) |Psi | 21,384 | 3.65 |

From the data that has been processed in table 2, the following results are obtained: The heat exchanger is not operationally feasible, because the calculation fouling factor is 0.0285 h.ft\(^2\).F / Btu, the pressure drop on the shell side (19.73 psi) has exceeded the minimum requirement (10 psi), the tube side pressure drop (1.47 psi) is still below the minimum requirement (10 psi) from these results, it is concluded that the tool is due cleaning. As a result, the effectiveness of operations is only 57.47%, which is under the design of 68.73%. More complete results can be seen in table 3. The following results obtained using SOLVER, are seen in table 3.

### Table 3. Optimization result.

| Parameter                      | Method   | Unit      | Shell side | Tube side | Shell side | Tube side |
|--------------------------------|----------|-----------|------------|-----------|------------|-----------|
| Mass flow rate                 | DO       | Lb/hr     | 34,955.86  | 537,510.02| 34,956     | 537,510   |
| Inlet temperature              | Tube side| F         | 303.46     | 89.6      | 303.46     | 89.6      |
| Outlet temperature             | Tube side| F         | 132.52     | 94.57     | 132.52     | 94.57     |
| Design fouling resistant        | Evolutionary | h.ft\(^2\).F/Btu | 0.0285 | |
| Operation fouling resistant     | GRG Non Linear | h.ft\(^2\).F/Btu | 0.0285 | |
| Design pressure drop            | Evolutionary | Psi | 6.16 | 9.98 | 6.16 | 9.98 |
| Operation pressure drop         | GRG Non Linear | Psi | 10 | 10 | 10 | 10 |
| Pressure drop                   | Evolutionary | ft/s | 0.98 | 9.5 | 0.98 | 9.5 |
| Permissible Velocity            | GRG Non Linear | Ft/s | 0.98 to 3.3 | 3.3 to 9.8 | 0.98 to 3.3 | 3.3 to 9.8 |

Note: DO (Decant Oil), CW (Cooling Water).
4. Conclusion
From data processing and calculation results are obtained as follows: Fouling factor operation 0.02823 (hr.ft$^2$.F/Btu) higher than design 0.00485 (hr.ft$^2$.F/Btu), Shell side pressure drop of 19.73 psi has over the maximum permissible of 10 psi, so it is time for cleaning, tube side pressure drop of 1.47 psi is still under the maximum permissible of 10 Psi, the effectiveness of the operation is 57.47% less than design value of 68.73%, and the simulation using excel-solver on the dirt (operating) condition result is effectiveness can be increased to 79.93%, shell side and tube side pressure drop 6.2 psi and 9.98 psi, shell side and tube side velocity 0.98 ft/s and 9.5 ft/s respectively.

Further research is needed to create programs that can input properties fluid into a table so that it can be programmed and inputted into the add-in system on Microsoft excel, making it easier to calculate the simulation.

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