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ORC Finned-Tube Evaporator Design and System Performance Optimization

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Abstract

Finned-tube heat exchangers have been used in many applications systems (refrigeration, waste heat recovery…) for many years. A considerable improvement in the thermal performance of these heat exchangers is possible by choosing an appropriate geometrical configuration for the corresponding application. In the present study, using exergy analysis, the geometry of an Organic Rankine Cycle (ORC) evaporator has been derived, and optimized geometrical configurations have been found out. Results show that operation parameters (superficial air velocity, internal mass velocity), geometrical parameters (fin types, fin pitches, tube lengths, tube diameters, tube pitches) and evaporator circuitry can be optimized to ensure a compromise between exergy losses and evaporator compactness. The elaborated procedure can be used to design heat exchangers of various systems (refrigeration, heat pump, ORC …).

Keywords: finned-tube evaporator, heat transfer, pressure drop, heat-exchanger optimization procedure, entropy generation, exergy losses.

1. Introduction

The valorization of low-temperature waste heat within industrial processes undoubtedly plays a crucial role in improving overall energy efficiency of industrial processes. The European Union launched several projects that aim at the development of innovative technologies to generate electricity by recovering low-temperature waste heat (<120°C) using ORC. One of the basic concepts that will be investigated for heat extraction from gases is a direct heat exchanger, which is a finned-tube evaporator featuring a direct exchange between flue gases and the working fluid. For wet gas heat sources (Tdp > 50°C), the temperatures of tube and fin surfaces are generally below the water dew point temperature, leading to simultaneous heat and mass transfer tube and fin surfaces, then part of the finned-tube evaporator will operate in dehumidifying conditions.

In view of the operating temperature, the counter-current configuration must be carefully designed to minimize temperature differences between hot and cold fluids. Pressure losses can be minimized using compact designs. Several studies are related to finned-tube heat exchangers. Shah et al. (1978) and Van den Bulck (1991) employed optimal distribution of the UA value across the volume of cross flow heat exchangers and optimized different design variables like fin thickness, fin height, and fin pitch. Bejan (1978) and Bejan (1979) used the concept of irreversibility for estimating and minimizing the usable energy wasted in heat-exchanger design and presented an optimum design method for balanced and imbalanced counter-flow heat exchangers. He proposed the use of a “Number of Entropy Production Units” as a basic parameter in describing heat exchanger performance. Later on, Khan et al. (2006) optimized the design of tube banks in cross flow using Entropy Generation Minimization method (EGM). In the present study, the exergy analysis is used to derive optimal geometrical configuration of the ORC evaporator. The parameters of the studied heat source are shown in Table 1.

Table 1: Parameters of the studied heat source

<table>
<thead>
<tr>
<th>Tdb</th>
<th>Tdp</th>
<th>Vdrv @ normal conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>110°C</td>
<td>60°C</td>
<td>32,000 Nm³.hr⁻¹</td>
</tr>
</tbody>
</table>

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2. Exergy analysis

The use of the exergy analysis to optimize the operating and geometrical parameters of the finned-tube evaporator in the ORC allows the assessment of the combined effect of heat transfer and pressure drop through the simultaneous interaction with the HEX. However, the pressure drop in the evaporator affects the exergy losses in the other ORC components, especially the turbine. Thus, a system exergy analysis for the ORC is needed.

The evolution of the working fluid into the ORC consists in four operations as shown in Figure 1. The working fluid, sub-cooled liquid at the condenser outlet (point 1) is compressed by a pump from low to high pressure to enter the evaporator (point 2). The high-pressure liquid enters the evaporator in liquid phase where it is heated and vaporized at constant pressure by the entering flue gases. The vapor at the boiler exit (point 3) expands through the turbine and generates power. Finally, the vapor enters the condenser (point 4) where it is condensed at constant pressure. The flue gases are assimilated to hot air.

Figure 1: Simple ORC: a) Schematic diagram, b) (T-s) diagram.

The ORC process parameters used in this study are listed in Table 2.

Table 2: ORC process parameters

<table>
<thead>
<tr>
<th>Working fluid</th>
<th>R-245fa</th>
<th>$P_{\text{cond}}$ @saturated liquid</th>
<th>0.1824 MPa</th>
</tr>
</thead>
<tbody>
<tr>
<td>$m_r$</td>
<td>3.89 kg.s$^{-1}$</td>
<td>$T_{\text{SC}}$ cond</td>
<td>2 K</td>
</tr>
<tr>
<td>$Q_{\text{evap}}$</td>
<td>826 kW</td>
<td>$T_{\text{DP}}$ condenser pinch</td>
<td>2 K</td>
</tr>
<tr>
<td>$P_{\text{evap}}$ @saturated vapor</td>
<td>0.5021 MPa</td>
<td>$\eta_{\text{is. pump}}$</td>
<td>85%</td>
</tr>
<tr>
<td>$S_{\text{boiler}}$</td>
<td>0 K</td>
<td>$\eta_{\text{is. turbine}}$</td>
<td>80%</td>
</tr>
<tr>
<td>Evaporator pinch</td>
<td>3 K</td>
<td>$\eta_{\text{is. blower}}$</td>
<td>60%</td>
</tr>
<tr>
<td>$m_{\text{water condensate}}$</td>
<td>0.118 m$^3$.hr$^{-1}$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ambient air temperature ($T_0$)</td>
<td>20°C</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

For each component, the exergy losses are evaluated from the exergy balance. The exergy is defined by:

$$e_x = h - T_0 s$$  \hspace{1cm} (1)

a. Evaporator (2-3):

The exergy balance in the evaporator can be written as:

$$m_r (e_{x_1} - e_{x_e}) + m_r (e_{x_2} - e_{x_3}) - E_{\text{losses, evap}} = 0 \hspace{1cm} (2)$$

The exergy losses in the evaporator can be expressed by:

$$E_{\text{losses, evap}} = m_r (e_{x_1} - e_{x_3}) - m_r (e_{x_2} - e_{x_3}) \hspace{1cm} (3)$$

b. Condenser (4-1):

In the condenser, the condensing heat is transferred to cooling water, supposed at exergy reference temperature $T_0$. Then, the exergy losses in the condenser are equal to the exergy transferred from the working fluid to the cooling water and are given by:
Ex_{losses \, cond.} = m_r (ex_4 - ex_1) \tag{4}

c. Pump (1-2), turbine (3-4), and blower (z-e):
The entropy generation in each component can be written as:

\begin{equation}
S_{gen,k} = S_{out,k} - S_{in,k} \tag{5}
\end{equation}

The exergy losses in each component are given by:

\begin{equation}
Ex_{losses,k} = T_0 S_{gen,k} \tag{6}
\end{equation}

Where K = pump, turbine or blower.
The total exergy losses in the ORC are the sum of the exergy losses of each component and are expressed as:

\begin{equation}
Ex_{losses \, ORC} = Ex_{losses \, evap.} + Ex_{losses \, cond.} + Ex_{losses \, turbine} + Ex_{losses \, pump} + Ex_{losses \, blower} \tag{7}
\end{equation}

3. Heat and mass transfer model

The studied dehumidifying evaporator for source A will remove both moisture and sensible heat from entering flue gases. The flue gas (FG) to be cooled is a mixture of water vapor, nitrogen, oxygen, and CO₂; both loose sensible heat when in contact with a surface colder than the FG. Latent heat is removed through moisture condensation only on the coil parts where the surface temperature is lower than the dew point of the FG passing over it. When the coil starts to remove moisture, the cooling surfaces carry both the sensible and latent heat loads. As the FG approaches saturation, each degree of sensible cooling is nearly matched by a corresponding degree of dew-point decrease. The latent heat removal per degree of dew-point change is significantly greater.

Figure 2 shows a typical thermal diagram for the coil surface when it is operating partially dry (counter-flow architecture). Locations of the entering and leaving boundary conditions for both FG and working fluid are shown. The thermal diagram in Figure 2 shows three lines to illustrate local conditions for the FG, surface, and working fluid throughout a coil. The top and bottom lines in the diagram indicate, respectively, changes across the coil in the FG stream enthalpy hₐ and the working fluid temperature T_r. To illustrate continuity, the single middle line in Figure 2 represents both surface temperature T_s and the corresponding saturated FG enthalpy h_m, although the temperature and FG enthalpy scales do not actually coincide as shown. The differential surface area dA_w represents any specific location within the coil thermal diagram where operating conditions are such that the FG-surface interface temperature T_s is lower than the local FG dew-point temperature.

The dry-wet boundary conditions are located where the coil surface temperature T_{wb} equals the entering FG dew-point temperature T_{dp}. Thus, the surface area A_d to the left of this boundary is dry, while the remaining A_w of the coil surface area is wet. The potential or driving force for transferring total heat Q from the FG stream to the tube-side working fluid (coolant) is composed of two components in series heat flow: (1) an air-to-surface FG enthalpy difference (h_a - h_m) and (2) a surface-to-coolant temperature difference (T_s - T_r).

\begin{figure}
\centering
\includegraphics[width=\textwidth]{thermal_diagram.png}
\caption{Thermal diagram for general case when coil surface operates partially dry (ASHRAE, 2000).}
\end{figure}

The total heat transferred in the evaporator is calculated by:

\begin{equation}
Q_{evap} = m_a (h_{a, in} - h_{a, out}) = m_r (h_{r, out} - h_{r, in}) \tag{8}
\end{equation}

For dry case, the overall heat transfer is based on the temperature potential and given as follows:
\[ Q_w = (U_oA_o)_{D} \cdot DTLM \]  

Where the overall heat transfer coefficient in dry conditions \((U_oA_o)_d\) and DTLM are given respectively by:

\[
\frac{1}{UA} = \frac{1}{h_oA_o} + \ln\left(\frac{D_o}{D_i}\right) + \frac{1}{2\pi kL} \frac{1}{\eta_o h_o} 
\]

\[
DTLM = \left(\frac{T_{o,in} - T_{o,out}}{T_{o,out} - T_{o,in}}\right) \ln\left(\frac{T_{o,in} - T_{o,out}}{T_{o,out} - T_{o,in}}\right) 
\]

Where \(T_{r,in}\) and \(T_{r,out}\) are respectively the inlet and outlet working fluid temperatures and \(T_{a,in}\) and \(T_{a,out}\) are respectively the inlet and outlet FG temperatures.

In Equation (11), \(\eta_o\) is the overall outside surface efficiency and is given by:

\[
\eta_o = 1 - \frac{A_i}{A_o} \left(1 - \eta_{f,\text{dry}}\right) 
\]

For wet case, the used method is basically analogous to Threlkeld approach. Details of the reduction method process can be found from the previous studies by Wang et al. (1997) and Wang and Chang (1998). Note that the Threlkeld method is an enthalpy-based reduction method. A brief description of the method is given here below.

The overall heat transfer is based on the enthalpy potential and given as follows:

\[ Q_w = (U_oA_o)_w \cdot DHLM \]

In Equation (13), DHLM is given by:

\[
DHLM = \frac{(h_{w,in} - h_{w,out}) - (h_{w,out} - h_{w,in})}{\ln\left(\frac{b_{w,out} - b_{w,in}}{b_{w,in} - b_{w,out}}\right)} 
\]

Where \(h_{w,in}\) and \(h_{w,out}\) are respectively the saturated FG enthalpies at the inlet and outlet of working fluid and \(h_{w,in}\) and \(h_{w,out}\) are respectively the inlet and outlet FG enthalpies.

The overall wet heat-transfer coefficient is related to the individual heat-transfer resistances (Myers, 1967) as follows:

\[
\frac{1}{(U_oA_o)_w} = \frac{b_i}{h_iA_i} + b_p \ln\left(\frac{D_o}{D_i}\right) + \frac{1}{2\pi kL} \left(\frac{A_{w,\text{in}}}{b_{w,\text{in}}} + A_{f,\text{out}} \frac{\eta_{f,\text{out}}}{b_{w,f}}\right) 
\]

\[
h_{w,in} = \frac{1}{\frac{c_{w,in}}{b_{w,\text{in}}} + \frac{\delta_v}{k_v}} 
\]

\(\delta_v\) is the thickness of the condensed water film. A constant of 0.005 inch was proposed by Myers (1967). In practice, \(\frac{\delta_v}{k_v}\) accounts for only 0.5 - 5% compared to \(\frac{c_{w,in}}{b_{w,\text{in}}}\), and has often been neglected by previous investigators.

In Equation 15, there are four quantities \(b_i, b_p, b_{w,\text{in}}, b_{w,f}\) involving enthalpy temperature ratios that must be evaluated.

The definitions of these quantities are as follows:

\(b_i\) : slope of the FG saturation curved between the mean working fluid temperature and the mean inside surface tube temperature (J.kg\(^{-1}\).K\(^{-1}\))

\(b_p\) : slope of the FG saturation curved between the outside and the inside tube wall temperatures (J.kg\(^{-1}\).K\(^{-1}\))

\(b_{w,\text{in}}, b_{w,f}\) : slope of the FG saturation curved at the mean water film temperature of the tube surface (J.kg\(^{-1}\).K\(^{-1}\))

\(b_{w,f}\) : slope of the FG saturation curved at the mean water film temperature of the fin surface (J.kg\(^{-1}\).K\(^{-1}\))

Quantities of \(b_i\) and \(b_p\) can be evaluated as:

\[
b_i = \frac{h_{w,\text{in}} - h_{w,m}}{T_{p,\text{in}} - T_{r,m}} 
\]

\[
b_p = \frac{h_{w,\text{in}} - h_{w,m}}{T_{p,\text{in}} - T_{r,m}} 
\]
Where in Equations (17) and (18):

- $T_{p, i, m}$: mean inside tube surface temperature (°C)
- $T_{p, o, m}$: mean outside tube surface temperature (°C)
- $T_{r, m}$: mean working fluid temperature (°C)
- $h_{s, r, m}$: mean saturated FG enthalpy at the mean working fluid temperature (J.kg$^{-1}$)
- $h_{s, p,i, m}$: mean saturated FG enthalpy at the mean inside tube wall temperature (J.kg$^{-1}$)
- $h_{s, o, m}$: mean saturated FG enthalpy at the mean outside tube wall temperature (J.kg$^{-1}$)

Without loss of generality, $h_{s, f}$ can be approximated by the slope of the saturated FG enthalpy curve evaluated at the base surface temperature (Wang et al., 1997). Evaluation of $h_{s, f}$ requires a trial and error procedure. For the trial and error procedure, $h_{s, w, f, m}$ must be calculated using the following equation (Wang et al., 1997):

$$h_{s, w, f, m} = h_{w, m} - \frac{c_{p,w} h_{w,A_f} l_f}{b_{w,f} h_w} \left( 1 - U_{w,A} \left( \frac{h_p \ln \left( \frac{D_p}{D_f} \right)}{2\pi KL} \right) \right) \left( h_{w, m} - h_{s, r, m} \right) \tag{19}$$

Where in Equation (19), $h_{s, w, f, m}$ is the mean saturated FG enthalpy at the mean water film temperature of the fin surface.

Algorithms for solving the heat transfer model under both dry and wet conditions can be found by Pirompugd et al. (2008). In this study, the heat transfer model is done by dividing the finned-tube evaporator into many tiny segments and by applying the DTLM (resp. DHLM) method in dry case (resp. wet case). The references of the correlations used for heat transfer and pressure drop are listed in Appendix A.

4. Optimization of evaporator parameters

As stated previously, the evaporator will be a cross-flow heat exchanger with a continuous finned-tube criss-crossing the path of the flue gases. The copper tube – aluminum fins evaporator will be composed of many circuits ($N_C$). Each circuit has a single inlet and a single outlet and includes one or many horizontal rows ($N_H$). The number of vertical rows ($N_V$) will be determined to ensure the corresponding evaporator capacity. Furthermore, the evaporator can be divided into many parts along its length ($N_D$). Figure 3 shows a typical finned-tube evaporator composed of one circuit ($N_C = 1$) with four horizontal rows per circuit ($N_H = 4$) and two vertical rows ($N_V = 2$).

The design procedure consists of the following: for each internal mass velocity, the number of circuits is calculated. Since the FG mass flow rate is fixed, the FG velocity is calculated. In this section, the operation parameters (superficial FG velocity, internal mass velocity), geometrical parameters (fin types, fin pitches, tube lengths, tube diameters, tube pitches) and evaporator circuitry are optimized to ensure a compromise between exergy losses in ORC and evaporator compactness.

![Figure 3: Typical finned-tube HEX.](image)

The evaporator compactness is defined as follows:

$$Evap. compactness = \frac{Ex_{available, ORC} - Ex_{losses, ORC}}{V_{evap}} \tag{20}$$

Where $V_{evap}$ is the evaporator volume (m$^3$).
The available exergy for the FG is given by:

\[
Ex_{\text{available, }a} = Ex_{\text{available, }a \text{ max}} - Ex_{\text{remaining, }a}
\]  

(21)

In Equation (24), the remaining exergy and the maximal available exergy for the FG are given respectively by:

\[
Ex_{\text{remaining, }a} = (h_{out,a} - T_{e} s_{out,a}) - (h_{in,a} - T_{e} s_{in,a})
\]  

(22)

\[
Ex_{\text{available, }a \text{ max}} = (h_{in,a} - T_{e} s_{in,a}) - (h_{out,a} - T_{e} s_{out,a})
\]  

(23)

4.1 Tube length

The evolution of the ORC exergy losses and the frontal area of the evaporator with the internal mass velocity are shown in Figure 4a. The internal mass velocity \(G\) is defined by:

\[
G = \frac{m_i}{A_i}
\]  

(24)

Figure 4.a shows that by increasing the internal mass velocity, the ORC exergy losses increase and the frontal area of the evaporator decreases. For a given tube length, Figure 4.b shows the presence of an optimal internal mass velocity at which the evaporator compactness is maximal. In fact, for the same evaporator capacity, as the internal mass velocity increases, the frontal area decreases and the number of vertical rows for the evaporator \((N_V)\) increases leading to an increase in the evaporator depth. Thus, an optimal internal mass velocity exists at which the evaporator compactness is maximal. Figure 4.b shows also that the maximal evaporator compactness is reached for a tube length around 3 m. The corresponding internal mass velocity is around 400 kg.m\(^{-2}\).s\(^{-1}\).

![Figure 4](image)

(a) Exergy losses and evap. frontal area for \(L = 3\) m and (b) Evap. Compactness for different tube lengths, vs. \(G\) \((D_i = 13.84\) mm, \(P_{T} = 45\) mm, \(P_l = 38.97\) mm, \(F_{p} = 3\) mm, \(\delta_i = 0.2\) mm, wavy fins, \(N_H = 1, N_D = 1\) )

4.2 Fin types and fin pitches

The dominant resistance in the finned-tube evaporator usually is at the FG side. Hence, fin surfaces (plain, wavy, louver, and slit fins) are usually adopted for improving the overall heat transfer performance. In this section, the performance of wavy fins will be compared to plain fins. The slit and louver fins will be disregarded when recovering from industrial FG due to slugging reasons. As shown in Figure 5.a, the use of wavy fins instead of plain fins leads to less exergy losses in the ORC and to more compact evaporator. For the selection of the fin pitch, Figure 5.b shows a decrease in the exergy losses with the increase of the fin pitches until reaching a fin pitch of 3 mm where the exergy losses almost remain constant. On the other side, the evaporator compactness presents almost a linear decrease with the increase of the fin pitch and then high evaporator compactness can be reached for small fin pitch. However, for small fin pitch, slugging of the evaporator can occur due to dehumidification process. In this study, a 3-mm fin pitch will be selected as a compromise choice.
4.3 Tube diameters and tube pitches

As stated previously, the evaporator will be composed of continuous finned-tube criss-crossing the path of flue gases. The main available configurations (tube diameters and tube pitches) of continuous finned-tube evaporator for the industrial applications are listed in Table 3.

Table 3: Main available configurations of continuous finned-tube evaporator

<table>
<thead>
<tr>
<th>Configuration</th>
<th>D_i (mm)</th>
<th>Tube thickness (mm)</th>
<th>P_T (mm)</th>
<th>P_L (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Configuration 1</td>
<td>8</td>
<td>0.76</td>
<td>25</td>
<td>21.65</td>
</tr>
<tr>
<td>Configuration 2</td>
<td>10.92</td>
<td>0.89</td>
<td>37.5</td>
<td>32.476</td>
</tr>
<tr>
<td>Configuration 3</td>
<td>13.84</td>
<td>1.01</td>
<td>45</td>
<td>38.97</td>
</tr>
</tbody>
</table>

The evaporator compactness and the ORC exergy losses as a function of the internal mass velocity are shown in Figure 6. For each configuration, the tube length is calculated at the optimal evaporator compactness. By comparing configurations listed in Table 3, Figure 6.a shows that configuration 3 leads to more compact evaporator compared to configurations 1 and 2. At the optimal internal mass velocity at which the evaporator compactness is maximal (Figure 6.a), Figure 6.B shows that configuration 3 presents the lowest exergy losses in the evaporator. Thus, configuration 3 seems the most appropriate for the present application.

4.4 Evaporator circuitry

The main parameters for determining the evaporator circuitry are the number of horizontal rows per circuit (N_H) and the number of division along tube length (N_D) (refers to Figure 3). Table 4 shows the effects of N_H and N_D on ORC
exergy losses. For \( N_D = 1 \), the optimal internal mass velocity for \( N_H = 2 \) is twice that for \( N_H = 1 \), which leads to a slight increase in exergy losses due to the increase in the pressure drop inside tubes because the evaporator will operate with higher internal mass velocity. For \( N_D = 2 \), a two horizontal rows per circuit \( (N_H = 2) \) gives slightly lower exergy losses compared to \( N_H = 1 \) and \( N_H = 4 \). However, the value of exergy losses for \( N_D = N_H = 2 \) is slightly lower compared to that obtained for \( N_D = N_H = 1 \). In conclusion, operating with \( N_D = 1 \) and \( N_H = 1 \) seems the most promising choice. By dividing the evaporator along tube length into \( n \) parts \( (N_D = n) \), the optimal corresponding number of horizontal rows will be equal to \( n \) \( (N_H = n) \).

Table 4: Effects of \( N_H \) and \( N_D \) on ORC exergy losses.
\((L = 3m, D_i = 13.84 \text{ mm}, P_T = 45 \text{ mm}, P_L = 38.97 \text{ mm}, F_p = 3 \text{ mm}, \delta_f = 0.2 \text{ mm}, \text{ wavy fins})\)

<table>
<thead>
<tr>
<th>( N_D )</th>
<th>( N_H )</th>
<th>( G_{\text{optimal evap. compactness}} ) (kg.m(^{-2}).s(^{-1}))</th>
<th>Exergy losses (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>~ 400</td>
<td>142.8</td>
</tr>
<tr>
<td>1</td>
<td>2</td>
<td>~ 800</td>
<td>144.8</td>
</tr>
<tr>
<td>2</td>
<td>1</td>
<td>~ 200</td>
<td>144.2</td>
</tr>
<tr>
<td>2</td>
<td>2</td>
<td>~ 400</td>
<td>143.8</td>
</tr>
<tr>
<td>2</td>
<td>4</td>
<td>~ 800</td>
<td>145.3</td>
</tr>
</tbody>
</table>

4. Conclusions

The objective of this work was to study and optimize the geometric design and operation parameters of the ORC finned-tube evaporator recovering energy from a low-temperature waste heat source (\( T < 120^\circ \text{C} \)) and operating under dehumidifying conditions. The studied heat source is characterized by an inlet temperature of 110°C, a water dew point temperature of 60°C and a volume flow rate of 32,000 Nm\(^3\).hr\(^{-1}\). Using exergy analysis, geometry of an ORC finned-tube evaporator has been derived, and optimized geometrical configurations and operational parameters have been found out by ensuring a compromise between exergy losses and compactness of the evaporator. Results show that the optimal tube length will be around 3 m, and the optimal operating internal mass velocity will be around 400 kg.m\(^{-2}\).s\(^{-1}\) (corresponding superficial FG velocity around 3 m.s\(^{-1}\)). For the selection of fin types, wavy fins lead to more compact evaporator compared to plain fins. A 3 mm is selected as a compromised fin pitch. By comparing the main available continuous finned-tube configurations, the optimal compactness and minimal exergy losses are reached with the largest tube diameter (5/8”). Finally, operating with \( N_D = 1 \) and \( N_H = 1 \) seems the most appropriate choice for the evaporator circuitry.

Appendix A

This appendix shows the references of the correlations used in the finned-tube evaporator design for the present study.

Table A.1: References of the correlation used at the working fluid side

<table>
<thead>
<tr>
<th>Working fluid side</th>
<th>Heat transfer</th>
<th>Pressure drop</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Tubes</td>
<td>Tubes</td>
</tr>
<tr>
<td>One-phase flow</td>
<td>Dittus and Boelter correlation (Bigot, 2001)</td>
<td>Blasuis correlation (Ould Didi et al., 2002)</td>
</tr>
<tr>
<td>Two-phase flow</td>
<td>(Gungor and Winterton, 1986)</td>
<td>Friedel correlation (Ould Didi et al., 2002)</td>
</tr>
</tbody>
</table>

Table A.1 shows the correlations used to calculate the heat-transfer coefficient inside tubes and the pressure drop inside tubes and elbows at the working fluid side for both one-phase and two-phase flows. No heat transfer will occur in the elbows since they are considered as adiabatic. Table A.2 shows the correlations used to calculate the heat transfer coefficient and pressure drop at the FG side for plain and wavy fins in both dry and wet modes. The dry and wet fin efficiencies can be found from previous study by Pirompugd et al. (2008).
Table A.2: References of the correlation used at the FG side

<table>
<thead>
<tr>
<th>FG side</th>
<th>Heat transfer</th>
<th>Pressure drop</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Dry</td>
<td>Wet</td>
</tr>
<tr>
<td>Plain fins</td>
<td>(Gray and Webb, 1986)</td>
<td>(Wang et al., 2000a)</td>
</tr>
<tr>
<td>Wavy fins</td>
<td>(Wang et al., 2002)</td>
<td>(Wang et al., 1999)</td>
</tr>
</tbody>
</table>

Nomenclature

**Subscripts**

- a: FG
- b: boundary
- cv: control volume
- D: division
- d: dry
- db: dry-bulb
- dp: dew-point
- f: fins
- gen: generated
- in: inlet
- i: inside
- is: isentropic
- m: mean
- max: maximal
- o: ambient, outside
- out: outlet
- r: working fluid / coolant
- s: saturated, surface
- w: wet, wall, water film

Greek symbols

- \( \eta \): efficiency (%)
- \( \rho \): density (Kg.m\(^{-3}\))
- \( \delta_f \): fin thickness (mm)

Abbreviations

- Cond: Condenser
- Evap: Evaporator
- FG: Flue gas
- HEX: Heat exchanger
\[
\begin{array}{ll}
U_{o,w} & \text{Overall heat transfer coefficient (wet)} \\
V & \text{Volume}
\end{array}
\]

kg.m\(^{-2}\).s\(^{-1}\)
m\(^3\)

References


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