Heat transfer characteristics of a gas cooler

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Abstract. The paper deals with the design and simulation of a gas-water heat exchanger used for heating purposes. The heat exchanger represents the gas-cooler/water heater of a transcritical CO₂ heat pump. The log-mean temperature difference and the NTU-ε methods are comparatively analyzed. The heat transfer area can be calculated through both methods but the true strength and what makes the difference when using the NTU-ε method is that based on it the simulation of the behaviour of an existing heat exchanger can be easily performed.

1. Introduction
A heat pump is an energy-efficient equipment because 75% of the total energy is derived from natural resources like water, air, soil, and the remaining 25% energy is the mechanical work converted into heat. The heat pump systems have been the subject of intensified research in recent years. Huseyin Benli [1] perform a thorough analysis of a heat pump with horizontal serpentines buried in the soil. Yingbai et al. [2] analyzes the destruction of exergy from the gas cooler in a carbon dioxide heat pump installation.

Although refrigerants with high ODP and GWP are currently being used, according to the EU Regulation [3], and in a relation with greenhouse gases policy it is obligatory that by 2050 these substances must be reduced by 80 to 95% compared to 1990’s to limit climate changes.

Carbon dioxide has a major disadvantage compared to other working agents used in heat pump installations (R407A, R245fa, R600a, R717) and it has been found that the heat pump installations with carbon dioxide carrier operate in transcritical mode. It is in the researchers’ attention because this fluid does not have a negative impact on the environment. Thermophysical properties and environmental impact factors for carbon dioxide are shown in Table 1 [4].

| Property                     | Carbon Dioxide |
|------------------------------|---------------|
| Normal boiling point [°C]    | -78.4         |
| Critical temperature tₖ [°C] | 30.97         |
| Critical pressure pₖ [bar]   | 73.33         |
| ODP                          | 0             |
| GWP                          | 1             |
This paper analyzes the behavior of the gas cooler built (or sealed) in plates, in terms of heat transfer process and the heat exchange area. Such heat exchangers are of a compact type (size) with 1500-2200 m²/m³ built from corrugated or smooth thin plates, forming fluid flow channels where heat exchange processes take place. On one side of the plate distributes the cooling fluid and on the other side the warming one [5]. This type of process is very well studied, as well as in various application aspects. Junyub and Kang [6] analyze the heat transfer characteristics to the condensation of R245fa refrigerant in a plate heat exchanger from a heat pump installation. Byung and Chung [7], Giovanni and Claudio [8] analyze the heat transfer and the pressure drop in heat exchangers with brazed plates during condensation of refrigerants R-1233zd and HFC1234yf.

Two methods are used to determine the heat exchange area. The first method is based on the logarithmic mean temperature difference, and it is the most known and used method for determining the heat exchange area. Major disadvantage of the method is that the inlet and outlet temperatures of the two fluids in the heat exchanger must be known. Also in this analyze, for the correct evaluation of the heat exchange area, must be determined according to each heat exchanger separately a correction coefficient, coefficient determined by a graphical method which most of the time introduces additional errors in the calculation [5, 9].

The second method used to determine the heat exchange area is the method based on the number of transfer units NTU. This method has a different approach to evaluate the heat exchange area by using two parameters, respectively, effectiveness and number of the heat transfer units. The equations applied for determining these parameters are different depending of the type of heat exchanger, so, the results obtained via this method are more reliable than one based on the logarithmic mean temperature difference approach [10, 11, 12].

2. Description of the heat pump installation

Figure 1.a shows the schematic diagram of an air-to-water heat pump installation composed of compressor Cp, gas cooler Gc, throttling valve Tv and evaporator Ev. Figure 1.b shows the transcritical evolution cycle of the heat pump installation in the pressure-enthalpy diagram. It can be seen (fig.1.b) that the 2-3 process is performed outside the saturation curve, so at the end of the gas cooling process (state 3) carbon dioxide is in the gaseous state.

![Figure 1.a. Heat pump diagram.](image1)

![Figure 1.b. p-h diagram of the cycle.](image2)

The pressures and working temperatures at the characteristic points of the heat pump cycle are shown in Table 2. All data presented below was determined using the Engineering Equation Solver software [4].
Table 2. State parameters in the key points of the heat pump cycle.

| No. | \( t[^{\circ}\text{C}] \) | \( p[\text{bar}] \) |
|-----|-----------------|-----------------|
| 1   | 7               | 36.73           |
| 2   | 108.3           | 120             |
| 3   | 65              | 120             |
| 4   | 2               | 36.73           |
| 5   | 55              | 1.5             |
| 6   | 60              | 1.5             |

3. Analysis of gas cooler operation

Figure 2 shows the temperature variations of the two fluids which are distributed in the gas cooler; so the Carbon Dioxide, is cooled from temperature \( t_2 \) to \( t_3 \) and water warms up from temperature \( t_5 \) to \( t_6 \).

![Figure 2. Counter flow gas cooler. Temperatures change across the heat exchanger.](image2)

The gas cooler presented in Figure 3 is equipped with plates [6].

![Figure 3. Configuration of the gas cooler.](image3)

In Table 3 are presented the geometric characteristics of the gas cooler from Figure 3.
Table 3. Geometry of the proposed gas cooler.

| Parameter                        | Unit     | Value |
|----------------------------------|----------|-------|
| Total length                     | $L_{tot}$ [m] | 0.3   |
| Port to port length              | $L_{port}$ [m] | 0.24  |
| Total width                      | $W_{tot}$ [m] | 0.13  |
| Port to port width               | $W_{port}$ [m] | 0.07  |
| Port diameter                    | $D_{port}$ [m] | 0.025 |
| Distance between plates          | $b$ [m] | 0.007 |
| Thickness of the plate           | $d$ [m] | 0.002 |
| Hydraulic diameter               | $D_e$ [m] | 0.013 |
| Plate area                       | $A_e$ [m$^2$] | 0.037 |
| Area of the flow channel section | $A_f$ [m$^2$] | 0.0009 |

4. The mathematical model

The heat transfer characteristics of both fluids - carbon dioxide and water, were estimated at the mean working temperature of the fluid given by equations (1) and (2):

$$T_{m_{CO_2}} = \frac{T_2 - T_3}{\ln \left( \frac{T_2}{T_3} \right)}$$  \hspace{1cm} (1)

$$T_{m_w} = \frac{T_5 - T_6}{\ln \left( \frac{T_5}{T_6} \right)}$$  \hspace{1cm} (2)

The equivalent diameter was calculated with equation:

$$D_e = \frac{4 \cdot A_e}{P_e}$$  \hspace{1cm} (3)

The convection heat transfer coefficient on the carbon dioxide side is calculated using the criterial equation [5]:

$$Nu_{CO_2} = 0.021 \cdot Re_{fCO_2}^{0.8} \cdot Pr_{fCO_2}^{0.43} \cdot \left( \frac{Pr_{fCO_2}}{Pr_{wall}} \right)^{0.25}$$  \hspace{1cm} (4)

The convection heat transfer coefficient on the waterside is calculated using the criterial equation [6, 13]:

$$Nu_w = 0.0508 \cdot Re_{fw}^{0.7304} \cdot Pr_{fw}^{0.33} \cdot \left( \frac{\mu_{fw}}{\mu_{wall}} \right)^{0.14}$$  \hspace{1cm} (5)

The overall heat transfer coefficient is given by the following equation:

$$U = \frac{1}{\frac{1}{h_{CO_2}} + \frac{1}{k_{wall}} + \frac{1}{h_w}}$$  \hspace{1cm} (6)

Flow and heat transfer characteristics as well as convection coefficient values are presented in table 4.
Table 4. Flow and heat transfer characteristics.

| Parameter          | Carbon dioxide | Water |
|--------------------|----------------|-------|
| \( t_m \) [°C]    | 86.2           | 57.5  |
| \( t_{wall} \) [°C] | 71.8           | 71.8  |
| \( k \) [W/m²K]    | 0.03697        | 0.6386|
| \( \mu_f \) [m²/s] | 8.862 \times 10^{-8} | 4.924 \times 10^{-7} |
| \( \mu_{wall} \) [m²/s] | 7.945 \times 10^{-8} | 4.037 \times 10^{-7} |
| \( a_f \) [m²/s]   | 6.435 \times 10^{-8} | 1.551 \times 10^{-7} |
| \( a_{wall} \) [m²/s] | 4.419 \times 10^{-8} | 1.591 \times 10^{-7} |
| \( \dot{m} \) [kg/s] | 0.2064         | 0.956 |
| \( Re_f \)  | 123338         | 28803 |
| \( Pr_f \)  | 1.377          | 3.175 |
| \( Pr_{wall} \) | 1.798          | 2.537 |
| \( Nu \)    | 261.6          | 122.3 |
| \( h \) [W/m²K] | 742            | 5881  |
| \( U \) [W/m²K] | 655.2          |       |

Two different methods for determining the heat transfer are used: the logarithmic mean temperature difference method and number of heat transfer units and efficiency method.

4.1. Method based on the logarithmic mean temperature difference
The relation of calculating the thermal power of the gas cooler is given:

\[
\dot{Q} = U \cdot A \cdot LMTD
\]

where, the mean logarithmic temperature difference is:

\[
LMTD = \frac{\Delta t_{max} - \Delta t_{min}}{\ln\left(\frac{\Delta t_{max}}{\Delta t_{min}}\right)}
\]

The necessary area for the heat exchange determined by the method based on the logarithmic mean temperature difference is:

\[
A_{LMTD} = \frac{\dot{Q}}{U \cdot LMTD}
\]

The mean-log temperature difference method is used mostly for designing or selecting a heat exchanger. In this case the flow rates, specific heat capacities, the inlet and outlet temperatures of both fluids and the overall heat transfer coefficient are known.

4.2. Method based on the number of transfer units and heat exchanger thermal efficiency
The NTU-ε method is very useful for simulations of heat exchanger's efficiency based on design and materials. For the case the heat transfer area and the overall heat transfer coefficient are known. The simulation for the heat exchanger are related to prediction of outlet temperatures at different inlet temperatures of the flows and thermal characteristics of the equipment.

For the heat pump heater (gas cooler) three equations for the rate of heat transfer could be written (Fig. 2):

\[
\dot{Q} = C_{CO_2}(t_2 - t_3)
\]

\[
\dot{Q} = C_{w}(t_6 - t_5)
\]
\[ \dot{Q} = U \cdot A \cdot LMTD = U \cdot A \left( \frac{t_2 - t_6}{\ln \frac{t_2 - t_6}{t_3 - t_5}} \right) \]  
(12)

If \( C_{CO_2} \), \( C_w \), \( U \), \( A \) (or \( U \cdot A \)) and the inlet temperatures \( t_2 \) and \( t_5 \) are known, Eqs. 10-12 contain as unknown parameters \( \dot{Q} \) and the two outlet temperatures \( t_3 \) and \( t_6 \).

Combining Eq. 10 and 11, Eq. 13 results:

\[ C_{CO_2} (t_2 - t_3) = C_w (t_6 - t_5) \]  
(13)

\[ C_{CO_2} (t_2 - t_3) = U \cdot A \left( \frac{t_2 - t_6}{\ln \frac{t_2 - t_6}{t_3 - t_5}} \right) \]  
(14)

Substituting \( t_6 \) from Eq. (13) into Eq. (14) gives:

\[ \ln \frac{t_2 - t_6}{t_3 - t_5} = U \cdot A \left( \frac{1}{C_{CO_2}} - \frac{1}{C_w} \right) \]  
(15)

Denoting \( P \) as:

\[ P = U \cdot A \left( \frac{1}{C_{CO_2}} - \frac{1}{C_w} \right) \]  
(16)

then Eq. (15) becomes:

\[ \frac{t_2 - \left[ t_5 + \frac{C_{CO_2}}{C_w} (t_2 - t_3) \right]}{t_3 - t_5} = e^P \]  
(17)

In Eq. (17) the only unknown parameter is \( t_3 \) that can be written as:

\[ t_3 = t_2 - (t_2 - t_5) \frac{e^P - 1}{e^P - \frac{C_{CO_2}}{C_w}} \]  
(18)

\( C_{CO_2} < C_w \) the heat exchanger efficiency becomes:

\[ \varepsilon = \frac{t_2 - t_3}{t_3 - t_5} \]  
(19)

Combining Eq. (18) and Eq. (19) the following results:

\[ \varepsilon = - \frac{e^P - 1}{e^P - \frac{C_{CO_2}}{C_w}} \]  
(20)

where (Eq. 16) \( P = U \cdot A \left( \frac{1 - \frac{C_{CO_2}}{C_w}}{C_{CO_2}} \right) \)

Denoting by

\[ NTU = U \cdot A \frac{C_{CO_2}}{C_{min}} = \frac{U \cdot A}{C_{min}} \]  
(21)
it results that

\[ P = NTU \left( 1 - \frac{C_{\min}}{C_{\max}} \right) \]  \hspace{1cm} (22)

Accounting for Eqs. (20) and (22) the efficiency \( \varepsilon \) becomes a function of NTU and \( \frac{C_{\min}}{C_{\max}} \):

\[ \varepsilon = f \left( NTU, \frac{C_{\min}}{C_{\max}} \right) \]  \hspace{1cm} (23)

For the specific case \( C_{\min}, C_{\max}, U \) and the inlet and outlet temperatures are given. When the size of the heat transfer area is needed, the following steps have to be applied:

1. Calculate \( \varepsilon \) (Eq. 19)
2. Determine \( P \) (Eq. 20) and NTU (Eq. 22)
3. Determine \( A \) (Eq. 21)

| LMTD  | \( A_{LMTD} \) [m\(^2\)] | \( \varepsilon \) [-] | NTU [-] | \( A_{NTU} \) [m\(^2\)] |
|-------|-----------------------------|------------------|--------|-----------------------------|
| 24.32 | 1.254                       | 0.8124           | 1.78   | 1.254                       |

Obviously (Table 5) with both methods the same value for the heat transfer area \( A_{LMTD} = A_{NTU} \) is obtained.

For simulation the behaviour of a given (existing or in design phase) heat exchanger (\( A \) and \( U \) stated), the NTU-\( \varepsilon \) method is able to provide the outlet temperatures for different inlet temperatures and different sets of mass flow rates and fluid properties (\( C_{\min}, C_{\max} \)).

For the specific countercurrent heat exchanger figures 4, 5 and 6 give the NTU-\( \varepsilon \), \( C_{CO_2} - t_3 \) and \( t_6-t_3 \) diagrams.

**Figure 4.** NTU - \( \varepsilon \) diagram.

**Figure 5.** \( C_{CO_2} - t_3 \) diagram.
5. Conclusions
The study points out the differences in calculating the heat transfer area based on the log-mean temperature difference between two streams and the NTU-ε method.

The input data for the log-mean temperature difference method are the two streams thermal capacities, the inlet temperatures and one outlet temperature. The second outlet temperature is obtained through energy balance of the heat exchanger.

For the NTU-ε method are required the thermal capacities of the two heat streams and information about only the inlet temperatures. The efficiency of the heat exchanger gives one of the outlet temperatures and the energy balance gives the second one.

The heat transfer area can be calculated with the same accuracy by both methods but the true strength and what makes the difference when assessing an existing heat exchanger using the NTU-ε method less input parameters are needed in order to perform the calculations.

Nomenclature

\[
\begin{align*}
A & \quad \text{heat transfer area} \quad [m^2] \\
a & \quad \text{thermal diffusivity factor} \quad [m^2/s] \\
C & \quad \text{thermal capacity} \quad [J/K] \\
\Delta t & \quad \text{temperature difference} \quad [{^\circ}C] \\
h & \quad \text{specific enthalpy} \quad [kJ/kgK] \\
k & \quad \text{thermal conductivity} \quad [W/mK] \\
\dot{m} & \quad \text{mass flow rate} \quad [kg/s] \\
Nu & \quad \text{Nusselt’s number} \\
p & \quad \text{pressure} \quad [\text{bar}] \\
P_e & \quad \text{perimeter} \quad [m] \\
Pr & \quad \text{Prandtl’s number} \quad [-] \\
Re & \quad \text{Reynolds’s number} \quad [-] \\
T & \quad \text{temperature} \quad [K] \\
t & \quad \text{temperature} \quad [{^\circ}C] \\
U & \quad \text{overall heat transfer coefficient} \quad [W/m^2K] \\
\end{align*}
\]

Greek symbols

\[
\begin{align*}
\nu & \quad \text{kinematic viscosity} \quad [m^2/s] \\
\end{align*}
\]

Indices

\[
\begin{align*}
gc & \quad \text{gas cooling} \\
CO_2 & \quad \text{carbon dioxide} \\
\end{align*}
\]
ev evaporator
f fluid
h hot
k critical point
m mean
max maximum
min minimum
w water

Abbreviations
GWP global warming potential
LMTD logarithmic mean temperature difference
NTU number of heat transfer units
ODP ozone depletion potential

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