AN OPTIMIZED SYSTEM OF A COMBINED TRANS-CRITICAL CO₂ REFRIGERATION AND MULTI EFFECT DESALINATION SYSTEM

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ABSTRACT

The objective of this paper is to study and make a comparison between an original boosted-system with one Booster (B-MED) and an optimized system with two boosters (2B-MED) of a combined trans-critical CO₂ refrigeration and boosted multi effect desalination system. The two systems are analyzed and compared thermodynamically. The optimized system with two boosters produces around 361.72 m³/day of fresh water, on the other hand the original system with only one booster module produces around 290.3 m³/day of fresh water, which means that the optimized system with two boosters increased the fresh water production rate by 24.6 % in comparison with that of one booster module. In addition, the heat transfer rate of the gas-cooler to the environment in the original system is equal to 1059 kW, while it is equal to 472.5 kW in the optimized one, which means that the optimized system (2B-MED) improves the refrigeration system by decreasing the heat transfer rate of the gas-cooler by 55.38 %. This leads to the reduction of the heat transfer area (HTA) of the gas cooler and all that will lead to the decrease of the total annual cost (TAC) of the refrigeration system. So that, the optimized system with two boosters is thermodynamically better than the original one.

Keywords: Combined system, Trans-critical carbon dioxide refrigeration, Boosted-MED, optimized system, fresh water production rate.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>CO₂</td>
<td>Carbon dioxide</td>
</tr>
<tr>
<td>Com</td>
<td>Compressor</td>
</tr>
<tr>
<td>C_p</td>
<td>Specific heat (kJ kg⁻¹ K⁻¹)</td>
</tr>
<tr>
<td>D</td>
<td>Distillate production rate (kg s⁻¹)</td>
</tr>
<tr>
<td>EES</td>
<td>Engineering Equation Solver</td>
</tr>
<tr>
<td>eva</td>
<td>Evaporator</td>
</tr>
<tr>
<td>F</td>
<td>Feed seawater</td>
</tr>
<tr>
<td>GAC</td>
<td>Gas cooler</td>
</tr>
<tr>
<td>h_{CO₂}</td>
<td>Specific enthalpy of CO₂ (kJ kg⁻¹)</td>
</tr>
<tr>
<td>h_{fg}</td>
<td>Latent heat (kJ kg⁻¹)</td>
</tr>
<tr>
<td>HTA</td>
<td>Heat Transfer Area (m²)</td>
</tr>
<tr>
<td>m</td>
<td>Mass flow rate (kg s⁻¹)</td>
</tr>
<tr>
<td>MED</td>
<td>Multi Effect Desalination</td>
</tr>
<tr>
<td>Q</td>
<td>Heat transfer rate (kW)</td>
</tr>
<tr>
<td>TAC</td>
<td>Total Annual Cost ($)</td>
</tr>
</tbody>
</table>

1. INTRODUCTION

Water scarcity is the mismatch of demand and availability of freshwater in a particular location. It has become a worldwide issue with the pollution of existing water supplies, increasing population and industry activity, uneven freshwater to population distributions, and changing rainfall patterns. This implies that many regions containing populated centers are becoming less capable of meeting the water supply requirements of the residing populations [1][2][3][4].

Meanwhile the rise of global temperature renders refrigeration and air-conditioning demands to increase.

Fresh water and refrigeration are two important products that are usually required simultaneously in many regions with hot and dry climates such as Middle Eastern countries [5].

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The need for these two products usually coincides, in many regions, but usually they are provided separately [6][7][8]. In addition to the high capital cost to produce fresh water and cooling demand individually, the significant amount of waste heat from these systems leads to serious negative impacts on the environment and climate area [6][8]. Therefore, in order to decrease the product cost rate of portable water and refrigeration production and to enhance the performance of water desalination and refrigeration processes, integrated systems, which combines refrigeration and multi effect desalination systems, have been proposed and attracted attentions of many researchers [5][6][7][8][9][10].

An optimized system will be studied here, and this system is based on the original one proposed by Farsi et al. [8]. Optimization will be carried out in order to improve the original system. The performance of the original and the optimized systems are assessed thermodynamically and then a comparison between these systems will be applied in order to analyze the viability of the optimized system and its ability to improve the original one.

2. SYSTEMS DESCRIPTION

Fig. 1 illustrates a schematic diagram of the original combined system [8]. This system consists of a CO₂ Supercritical refrigeration system, a booster module and a basic MED system with 6 stages.

The high temperature CO₂ leaving the compressor enters the MED as a sensible heat source (process 2–3) and passes through the rest of the refrigeration cycle. Considering the fact that the refrigerant leaving the MED’s first effect still has a considerable amount of energy containment (the typical temperature is about 55–70°C), it is desirable to further use this energy into the booster module so as to increase the distilled water rate.

The prominent benefit of the booster module application at the MED is that there is no direct cost associated with the amount of energy consumed and rather than an optimal and a further use of the available energy potential that would be achieved [6][8].

Fig. 2 illustrates the optimized system with two booster modules in which the vapor from the first booster module is injected to the fifth effect with that from the fourth effect, while the sixth effect is injected by the vapor from the second booster with that from the fifth effect.
3. METHODOLOGY AND MODELING

We will compare the original system of B-MED with trans-critical carbon dioxide refrigeration against an optimized system which contains of two booster modules (2B-MED) thermodynamically. The objective of the optimized system is to improve the combined system (the refrigeration system and the B-MED system).

3.1 Properties and assumptions

Thermodynamic properties of the working fluids of these two systems (\(CO_2\) in refrigeration system; brine water, steam and pure (desalinated) water in MED) should be determined in order to analyze the systems’ performance. Table 1 shows the initial operating conditions. Also, for thermodynamic modeling of the combined system, several assumptions are considered as bellow:

- All the processes are assumed to be steady state and the startup and shutdown processes are not considered.
- The salinity of seawater and rejected brine is about 46,500 and 70,000 ppm respectively [11].
- The produced distillated water is assumed to be completely pure (\(w = 0\) kg/kg).
- The distillated water leaving each effect is saturated liquid (\(x = 0\)) and the number of effects is 6.
- Heat losses to the environment and pressure drops in piping and components in MED system are negligible.
- Minimum temperature difference over each effect (DT) is about 3°C.

3.2 Thermodynamic modeling of the trans-critical carbon dioxide refrigeration system

The thermodynamic modeling for all the parts will be taken according to Fig. 1 while the thermodynamic modeling for the second Booster Module will be taken according to Fig. 2.

The heat transfer rate of the gas-cooler to the environment is calculated as follows:

\[
Q_{GAC} = \dot{m}_{\text{CO}_2} (h_{\text{CO}_2,4} - h_{\text{CO}_2,5})
\]

The energy balance for the expansion valve is determined as follows:

\[
h_{\text{CO}_2,6} = h_{\text{CO}_2,5}
\]

The energy balance equation for the Cold box is given by Eq. (3):

\[
Q_{\text{Cold box}} = \dot{m}_{\text{CO}_2} (h_{\text{CO}_2,1} - h_{\text{CO}_2,6})
\]

The power consumption of the Compressor is given by Eq. (4):

\[
W_{\text{com}} = \dot{m}_{\text{CO}_2} (h_{\text{CO}_2,2} - h_{\text{CO}_2,1})
\]
The COP of the $CO_2$ trans-critical refrigeration is defined as:

$$ \text{COP} = \frac{\dot{W}_{\text{com}}}{\dot{Q}_{\text{sea}}} $$  \hspace{1cm} (5)

### 3.3 Thermodynamic modeling of the B-MED & 2B-MED systems

The mass balance equation for the first effect is given by Eq. (6). The mass balance equation for the other effects is given by Eq. (7).

$$ \dot{m}_{B1} = \dot{m}_{f1} - \dot{m}_{D1} \hspace{1cm} (6) $$

$$ \dot{m}_{B(i)} = \dot{m}_{f(i)} - \dot{m}_{D(i)} + \dot{m}_{B(i-1)} \hspace{1cm} (7) $$

Where $\dot{m}_{f1}$, $\dot{m}_{B1}$ and $\dot{m}_{D1}$ are the mass flow rate of the feed seawater, the brine stream and the distilled water production rate in the first effect respectively, and $i$ stands for the effect number (from 2 to $n$)

The salinity balance equation for the first effect is given by Eq. (8). The salinity balance equation for the other effects is given by Eq. (9).

$$ x_{sw} \cdot \dot{m}_{f1} = x_{B(1)} \cdot \dot{m}_{B(1)} \hspace{1cm} (8) $$

$$ x_{sw} \cdot \dot{m}_{fi} + x_{B(i-1)} \cdot \dot{m}_{B(i-1)} = x_{B(i)} \cdot \dot{m}_{B(i)} \hspace{1cm} (9) $$

Where $x_{sw}$ and $x_B$ are the salinity of the feed seawater and the brine stream respectively, 1 stands for the first effect.

The salinity balance equation for the first and the second booster is given by Eq. (10). The mass balance equation for the first and the second booster is given by Eq. (11).

$$ x_{sw} \cdot \dot{m}_{f1(B,1,2)} = x_{B,1,2} \cdot \dot{m}_{B,1,2} \hspace{1cm} (10) $$

$$ \dot{m}_{B,1,2} = \dot{m}_{f,1,2} - \dot{m}_{D,1,2} \hspace{1cm} (11) $$

Where $\dot{m}_{f,1,2}$, $\dot{m}_{B,1,2}$ and $\dot{m}_{D,1,2}$ are the mass flow rate of the feed seawater, the brine stream and the distilled water production rate in the first and the second booster respectively.

The energy balance equations for the MED’s first effect in which the refrigerant leaving the compressor is used as its sensible heat source are calculated by Eqns. (12) – (15):

$$ Q_{\text{sensible,1}} = \dot{m}_{f1} c_{p_w} (T_1 - T_f) \hspace{1cm} = \dot{m}_{CO2} (h_{CO2,prim,3} - h_{CO2,3}) \hspace{1cm} (12) $$

$$ \dot{Q}_{\text{latent,1}} = D_1 h_{f,g,1} = \dot{m}_{CO2} (h_{CO2,2} - h_{CO2,prim,3}) \hspace{1cm} (13) $$

$$ \dot{Q}_{\text{source}} = \dot{Q}_{\text{sensible,1}} + \dot{Q}_{\text{latent,1}} \hspace{1cm} (14) $$

$$ \dot{Q}_{\text{source}} = \dot{m}_{f1} c_{p_w} (T_1 - T_f) + D_1 h_{f,g1} = \dot{m}_{CO2} (h_{CO2,2} - h_{CO2,3}) \hspace{1cm} (15) $$

According to Fig. 1, the energy balance of the booster module is analogous to the first effect of the basic MED:

$$ Q_{\text{sensible,B(1,2)}} = \dot{m}_{FB} c_{p_w} (T_{B(1,2)} - T_f) \hspace{1cm} = \dot{m}_{CO2} (h_{CO2,prim,4} - h_{CO2,4}) \hspace{1cm} (16) $$

$$ Q_{\text{latent,B(1,2)}} = D_{B(1,2)} h_{f,g,B(1,2)} \hspace{1cm} = \dot{m}_{CO2} (h_{CO2,3} - h_{CO2,prim,4}) \hspace{1cm} (17) $$

$$ Q_{\text{source,B(1,2)}} = \dot{Q}_{\text{sensible,B(1,2)}} + \dot{Q}_{\text{latent,B(1,2)}} \hspace{1cm} (18) $$

$$ \dot{Q}_{\text{source,B(1,2)}} = \dot{m}_{FB(1,2)} c_{p_w} (T_{B(1,2)} - T_f) + D_{B(1,2)} h_{f,g,B(1,2)} = \dot{m}_{CO2} (h_{CO2,3} - h_{CO2,4}) \hspace{1cm} (19) $$

The energy balance for effects 2 – n can be defined as follows:

$$ D_i h_{f,g,i} + D_B h_{f,g,B} + B_{Ti-1} c_{p_w} (T_{i-1} - T_i) \hspace{1cm} = D_{i+1} h_{f,g,i+1} + \dot{m}_{f_i} c_{p_w} (T_i - T_f) \hspace{1cm} (20) $$

The energy balance equation for the MED’s condenser is in the form of Eq. (21).

$$ \dot{Q}_{\text{cond}} = D_n h_{f,g,n} = (\dot{m}_f + \dot{m}_{cw}) c_{p_w} (T_{sea} - T_f) \hspace{1cm} (21) $$

### Table 1

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Feed water inlet temperature</td>
<td>[°C]</td>
</tr>
<tr>
<td>Top brine temperature (TBT)</td>
<td>[°C]</td>
</tr>
<tr>
<td>Number of effects</td>
<td>6</td>
</tr>
<tr>
<td>Heating medium</td>
<td>$CO_2$</td>
</tr>
<tr>
<td>Mass flow rate of $CO_2$</td>
<td>[kg s⁻¹]</td>
</tr>
<tr>
<td>$P_{out,com}$ [kPa]</td>
<td>10570</td>
</tr>
<tr>
<td>Heat source inlet temperature</td>
<td>[°C]</td>
</tr>
</tbody>
</table>

### 3.4 Validation of process simulation model

We calibrated our model using single-effect distillation (SED) plant performance data available from the Alfa Laval Marine & Diesel product catalogue [12]. It is noteworthy that this proposed design simplifies to a cascaded single effect distillation plant when the number of effects is reduced to one. This enables us to validate the fundamental basis of our optimized design using the Alfa Laval SED plant data [12]. The optimized design is then applied to six effects distillation plants under different operation conditions which are the most common on the market. Fig. 3 depicts a comparison between the calculated amount of released energy from the simulation and the actual amount of released energy from SEDs reported in the Alfa Laval single effect
Freshwater generators catalog [12]. It is evident that our calculated data match the actual data very well.

4. RESULTS AND DISCUSSION

An optimized system is being studied here based on the original one proposed by Farsi et al. [8]. The performance of the original and the optimized systems is assessed thermodynamically. The viability of the optimized system and its ability to improve the original one is studied, an analysis is made for both systems thermodynamically, and in each system the refrigeration and the MED cycles are analyzed separately.

Fig. 4 shows the fresh water production rate for the both systems in each effect, and it is obviously noticed that for the fresh water production rate there is no big difference between the two systems in the first four effects (from 1 to 4) and the first Booster module. According to Fig. 2 we see that the vapor from the first booster module is injected to the fifth effect with that from the fourth effect, and the vapor from the second booster module is also injected to the sixth effect with that from the fifth effect, so that the fresh water production rate of the fifth and sixth effects in the 2B-MED system is much bigger than that in the 1B-MED which has only one injected effect.

According to Fig. 4 we can see that the 2B-MED system produces around 361.72 m³/day of freshwater, on the other hand the 1B-MED system produces around 290.3 m³/day of freshwater, which means that the optimized system with two boosters increases the fresh water production rate by 24.6% in a comparison with that of one booster module.

According to Table 2 the outlet temperature from the booster in the 1B-MED system is around 50 °C, so that the heat transfer rate of the gas-cooler to the environment in the original system is equal to 1059 kW. While the outlet temperature from the second booster in the 2B-MED system is around 44 °C, the heat transfer rate of the gas-cooler to the environment in the optimized system is equal to 472.5 kW, which means that the optimized system 2B-MED improves the refrigeration system by decreasing the heat transfer rate of the gas-cooler by 55.38%, and lowers the heat transfer area (HTA) of the gas cooler and the total annual cost (TAC) of the refrigeration system.

Table 2
Some outputs from the two systems (1B-MED) & (2B-MED)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>(1B-MED)</th>
<th>(2B-MED)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condenser inlet temperature [°C]</td>
<td>28</td>
<td>28</td>
</tr>
<tr>
<td>Condenser outlet temperature [°C]</td>
<td>44</td>
<td>40.4</td>
</tr>
<tr>
<td>$Q_{gas\ cooler}$ [kW]</td>
<td>1059</td>
<td>472.5</td>
</tr>
<tr>
<td>Total distilled water [m³/day]</td>
<td>290.3</td>
<td>361.72</td>
</tr>
<tr>
<td>Outlet temperature from the first booster [°C]</td>
<td>50</td>
<td>50</td>
</tr>
<tr>
<td>Outlet temperature from the second booster [°C]</td>
<td>44</td>
<td>44</td>
</tr>
</tbody>
</table>

5. CONCLUSION

An optimized system of a combined trans-critical CO₂ refrigeration and multi effect desalination system is studied based on thermodynamic analysis. The results show that the 2B-MED system produces around 361.72 m³/day of fresh water, while the 1B-MED system produces around 290.3 m³/day of fresh water, which means that the optimized system with two boosters increases the fresh water production rate by 24.6% in a comparison with that of one booster module.

The outlet temperature from the booster in the original system (1B-MED) is around 50 °C, so that the heat transfer rate of the gas-cooler to the environment in the original system is equal to 1059 kW. While the outlet temperature from the second booster in the optimized system (2B-MED) is around 44 °C, the heat transfer rate of the gas-cooler to the environment in the optimized system is equal to 472.5 kW, which means that the optimized system (2B-MED) improves the refrigeration system by decreasing the heat transfer rate of the gas-cooler by 55.38%, resulting in a decrease in the heat transfer area (HTA) of the gas cooler and the total annual cost (TAC) of the refrigeration system.

In the future we will study the economic part of the optimized system; the viability of adding preheating to the original system and comparing it with that of the two booster modules will also be carried out.

ACKNOWLEDGEMENT

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