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EFFICIENCY OF A COMBINED DESALINATION AND POWER SYSTEM UTILISING A MULTI-STREAM HEAT EXCHANGER

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Keywords: Discharge Thermal Energy Combined Desalination (DTECD) and Power System, Compact Multi-Stream Condenser, Exergy Analysis, Parametric Optimization.

ABSTRACT: The aim of this study is to simplify the process of Discharge Thermal Energy Combined Desalination (DTECD) with Power System by integrating the two existing heat exchangers (condensers) into a new multi-stream one. This system is a heat recovery unit, which is used to cogenerate water and power. Two shell & tube condensers operate in a closed power cycle and a desalination system for cooling an ammonia mixture (70% wt.) as a working fluid and condensing a pure vapor respectively. Here, a multi-stream plate condenser is utilized instead of the two low exergy efficiency shell & tube condensers. The results proved that the proposed technique leads to improvement in the system performance. The performance of the proposed process was analyzed by applying parametric optimization.

1 INTRODUCTION

There is an interest to utilize alternative energy for running thermal desalination plants so that energy can be conserved in the associated environmental benefits (Eltawil 2009, Gude 2010). Uehara developed a hybrid cycle, which converted ocean thermal energy as a low temperature energy source to desalinate saline water (Uehara 1996). Some more low temperature processes for cogenerating water and power have been introduced by different researchers (Kamal 1997, Kronenberg 2001, Kumar 2005, Li 2012). One of the novel steam recovery systems is Discharge Thermal Energy Combined Desalination (DTECD) and Power Cycle, which cogenerates potable water and power by utilizing the Rankin Cycle with an ammonia mixture as working fluid (Hosseini Araghi 2014).

On the other hand, the compact multi-stream heat exchanger is a pioneering technology, which can exchange heat with small temperature differences between the streams in a low temperature processes. Applying this technology leads to significantly improve the overall efficiency and space (Abu-Khader 2012). Synthesis of a multi-stream heat exchanger network has been investigated in different articles (Wang 2001, Xiangkun 2008). Also, it is

found that the compact heat exchanger can be designed as an individual heat recovery system (Franco 2005, Picón-Núñez 2006, Arsenyeva 2011). The aim of this study is to investigate the effect of utilizing a compact multi-stream condenser on the heat recovery DTECD and power cycle. Furthermore, an exergy analysis and parametric optimization will be performed to find the performance of the proposed system.

2 PROCESS DESCRIPTION

DTECD is a steam recovery system, which is proposed to couple with the Razi Ammonia Plant to cogenerate water and power. This technology utilizes waste low pressure steam (134 °C and 300 kPa) as a heat source, and two ammonia-water mixture as the working fluid (70% wt.) and refrigerant (80% wt.) in the closed power and open water cycles, respectively (IIES 2012, Hosseini Araghi 2014). Fig. 1 illustrates the differences between the proposed system with the previous DTECD and Power Cycle. In the new system the two previous condensers (CO-101 and CO-102) in both cycles were replaced by one compact multi-stream heat exchangers (HXMS-100) and ammonia mixture used as refrigerant instead of the cooling water in the (CO-102).

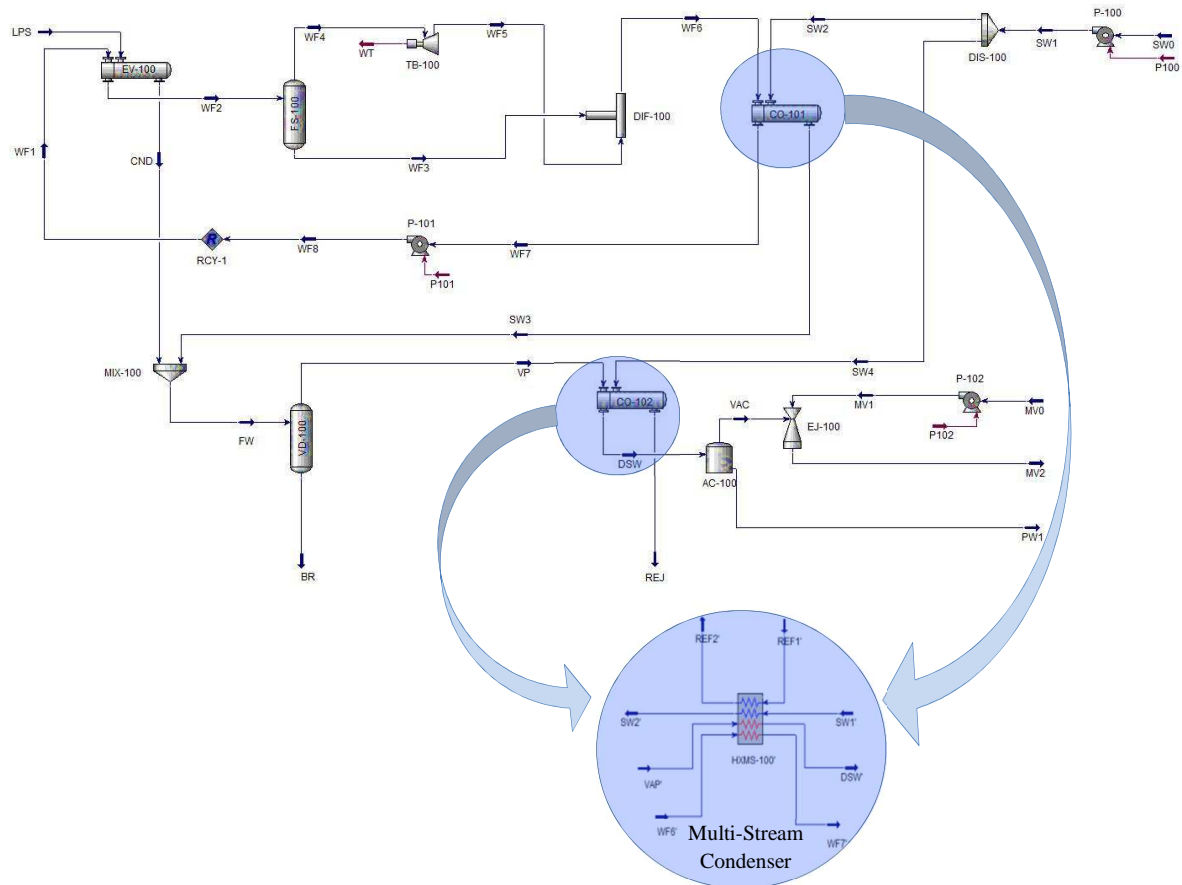


Figure 1: Utilizing a multi-stream exchanger in the DTECD and Power Cycle

3 MODEL AND EQUATIONS

3.1 Heat exchanger model and equations

Fig. 2 shows the schematic view of the temperature distribution model within the proposed four passes multi-stream condenser.

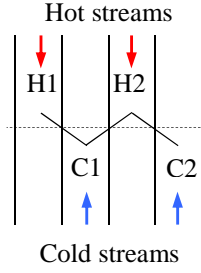


Figure 2: Temperature distribution model of the multi-stream heat exchanger

The required thermal equations, surface calculation (Picon-Nunez 2002, Shah 2003, Polley 2005) for designing the two-stream compact heat exchanger were investigated and summarized as below.

The basic heat transfer equation for two-stream heat exchanger is obtained by Eq. (1) to (3).

$$\dot{Q} = UAF\Delta T_{LM} \quad (1)$$

Where,

$$\Delta T_{LM} = \frac{(T_{H,i} - T_{C,o}) - (T_{H,o} - T_{C,i})}{\ln\left(\frac{T_{H,i} - T_{C,o}}{T_{H,o} - T_{C,i}}\right)} \quad (2)$$

$$\frac{1}{U} = \frac{1}{h_H} + \frac{\delta}{k} + \frac{1}{h_C} \quad (3)$$

The hydraulic diameter and cross section surface can be calculated from Eq. (4) and Eq. (5).

$$d_h = 4 \frac{LA_c}{A} = 4 \frac{V}{A} \quad (4)$$

$$A_c = \frac{St \cdot \dot{m} C_p}{h} \quad (5)$$

The pressure drop across the core of a compact heat exchanger is shown by Eq. (6).

$$\Delta P = \frac{2fL\dot{m}^2}{\rho d_h A_c^2} \quad (6)$$

All above equations should be correlated for designing the two-phase multi-stream heat exchanger. The phase change properties of the ammonia refrigerant are available in the literature (Corberán 1998, Ayub 2003, Sterner 2006). The heat balance of the condenser is obtained from the summation of the sensible heat transfer and the latent heat, which presents as Eq. (7).

$$\dot{Q} = \dot{m}(C_p \Delta T + H_{fg}) \quad (7)$$

Aspen/Hysys V.8.0 is used for modelling the proposed complex multi-stream condenser and simulating it in the DTECD heat recovery system. The ENRTL is selected as the equation of state, because all streams are electrolyte, e.g. saline water and ammonia-water mixtures.

3.2 Exergy equations

The exergy term presents the maximum workability of a system and it is an appropriate approach to analyze the performance of a thermal system such as proposed here. Subsequently, the exergy flow Eq. (8) can be decomposed to thermo-mechanical and chemical exergy equations (Tsatsaronis 1993).

$$\dot{E} = \dot{m} \cdot (E^{tm} + E^{ch}) = [(h - h_0) - T_0(s - s_0) + \sum_{j=1}^n x_j (\mu - \mu_0)] \quad (8)$$

The entropy of components in an ideal solution can be obtained by Eq. (9).

$$S = S_{j,pure}(T, P) - R_u \ln x_j \quad (9)$$

The governing exergy equations in this study are extracted from reference books (Kotas 1985, Bejan 1995). The entropy changes between two states for incompressible and compressible flows are calculated from Eq. (10) and Eq. (11), respectively.

$$S_{\Delta T} = C_p \ln\left(\frac{T_2}{T_1}\right) \quad (10)$$

$$S_{\Delta P} = C_p \ln\left(\frac{T_2}{T_1}\right) - R \ln\left(\frac{P_2}{P_1}\right) \quad (11)$$

Exergy destruction of each component can be derived from the exergy balance Eq. (12).

$$\dot{E}_d = \sum \dot{E}_i - \sum \dot{E}_o - \dot{W} + \sum_j (1 - \frac{T_0}{T_j}) \dot{Q}_j \quad (12)$$

The exergy efficiency of a system is obtained from Eq. (13).

$$\psi = 1 - \frac{\dot{E}_d}{\sum \dot{E}_i} \quad (13)$$

4 RESULTS AND DISCUSSION

4.1 Simulation results

Tab.1 presents the stream properties such as mass flow rate, temperature, pressure, enthalpy and entropy of the primary DTECD system. The net power generation and pure water production rate are 4.56 (MW) and 66 (ton/h), respectively. The total exergy efficiency of the heat recovery system is about 50%. The stream properties of the proposed multi-stream condenser are recorded in Tab. 2. The simulation results show that both the power generation and water production rate do not change significantly and remained constant after replacing the two shell & tube condensers with the proposed multi-stream one. However, the cooling water consumption rate reduces significantly by 70%, only by using small amount of the ammonia-water mixture (80% wt.) as a refrigerant.

| stream | m | T | P | H | S |
|--------|---------|------|-------|---------|------------|
| | (ton/h) | (°C) | (kPa) | (kJ/kg) | (kJ/kg °C) |
| LPS | 90 | 134 | 300 | 13246 | 2.43 |
| CND | 90 | 49 | 260 | 15765 | 8.73 |
| WF1 | 195 | 39.5 | 3500 | 7446 | 10.23 |
| WF2 | 195 | 130 | 3460 | 6283 | 6.90 |
| WF3 | 46 | 130 | 3460 | 15423 | 7.79 |
| WF4 | 149 | 130 | 3460 | 3448 | 6.63 |
| WF5 | 149 | 90 | 1500 | 3564 | 6.58 |
| WF6 | 195 | 96 | 1500 | 6372 | 6.86 |
| WF7 | 195 | 39 | 1460 | 7450 | 10.24 |
| SW1 | 5000 | 33 | 300 | 15438 | 8.95 |
| SW2 | 1200 | 33 | 300 | 15439 | 8.62 |
| SW3 | 1200 | 75 | 260 | 15264 | 8.41 |
| FDW | 1290 | 73 | 300 | 15664 | 8.43 |
| BRN | 1224 | 44 | 9 | 15787 | 8.80 |
| VAP | 66 | 44 | 9 | 13390 | 1.24 |
| DSW | 66 | 44 | 9 | 15787 | 8.80 |
| SW4 | 3600 | 33 | 300 | 15832 | 8.95 |
| REJ | 3600 | 46 | 260 | 15788 | 8.80 |
| MV0 | 200 | 33 | 101.3 | 15832 | 8.95 |
| MV1 | 200 | 33.1 | 1000 | 15831 | 8.94 |
| MV2 | 200 | 33.1 | 108 | 15831 | 8.94 |

Table 1: Stream properties of the primary DTECD system (Refer to Fig. 1)

| | WF6 | SW1 | VAP | REF1 | WF7 | SW2 | DSW | REF2 |
|---------------------------------|--------|--------|--------|-------|--------|--------|--------|-------|
| Temperature (°C) | 96.4 | 33.02 | 29 | 20 | 38.2 | 46 | 29 | 25 |
| Pressure (kPa) | 1500 | 300 | 4 | 1000 | 1475 | 275 | 4 | 975 |
| Mass Flow (ton/h) | 195 | 2000 | 62 | 103 | 195 | 2000 | 62 | 103 |
| Molar Enthalpy(kJ/kgmole) | 110332 | 285217 | 285525 | 85264 | 128988 | 284241 | 285525 | 85088 |
| Molar Entropy (kJ/kgmole °C) | 119 | 161 | 162 | 65 | 177 | 158 | 162 | 64 |

Table 2: Stream properties of the proposed multi-stream condenser (Refer to Fig. 1)

Tab. 3 presents the specifications of the old and new condensers.

| | HXMS-100 | CO-101 | CO-102 |
|-----------------------|----------------|----------------|----------------|
| Duty (kJ/h) | 210,677,874.20 | 211,490,182.18 | 158,601,107.06 |
| UA (kJ/°C h) | 44,932,039.81 | 19,917,454.38 | 60,021,618.53 |
| Min. Approach (°C) | 0.20 | 4.34 | 0.19 |
| LMTD (°C) | 4.69 | 10.62 | 2.64 |
| Hot Pinch Temp. (°C) | 38.62 | 37.36 | 43.79 |
| Cold Pinch Temp. (°C) | 38.42 | 33.02 | 43.59 |

Table 3: Specifications of the old and new condensers

4.2 Exergy analysis

Fig. 3 illustrates that the exergy destructions of the shell & tube heat exchangers such as evaporator and condensers are higher comparing with the other components in the primary DTECD system.

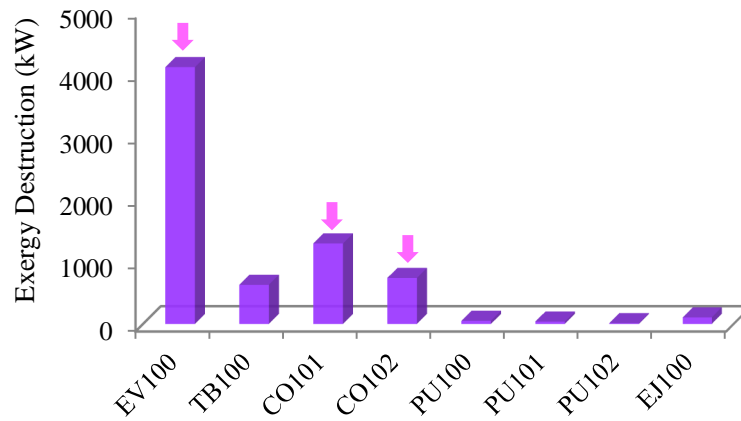


Figure 3: Components exergy destruction in the DTECD system

Based on the exergy analysis, it is found that the exergy efficiency of the heat transfer unit increases by utilizing the multi-stream heat exchanger. The comparison between the exergy efficiencies of the conventional condensers with the new integrated one is shown in Fig. 4.

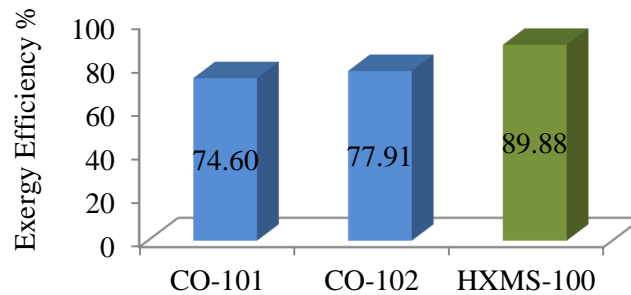


Figure 4: Exergy efficiencies of the old and new condensers

4.3 Parametric optimization

Parametric analysis was performed to find how the key parameters such as refrigerant concentration, flow rate, temperature affect the performance of the multi-stream condenser. An increase in ammonia concentration as a refrigerant has not significant effect on the overall exergy performance of the new condenser. However, enriched ammonia refrigerant leads to a reduction in its required flow rate for cooling the system as shown in Fig. 5.

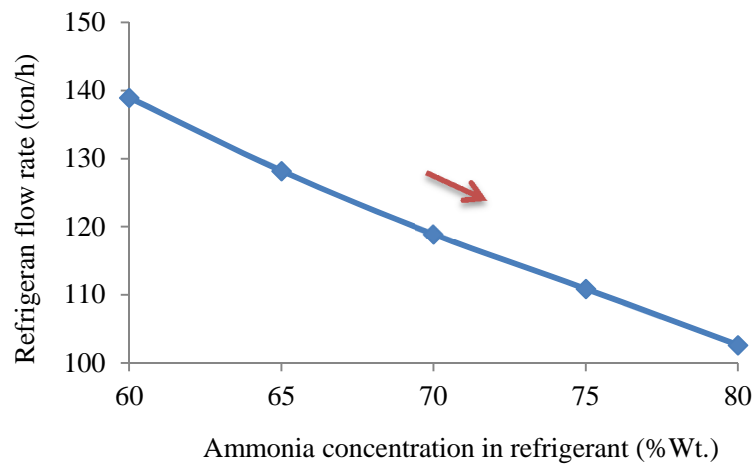


Figure 5: Effect of ammonia concentration on the refrigerant flow rate

The proposed multi-stream exchanger can operate with a lower pressure drop and refrigerant flow rate. The effects of refrigerant temperature and cooling seawater temperature on the required amount of the refrigerant flow rate are shown in Fig. 6. It is found that the minimum operating flow rate of the refrigerant is achieved by setting both temperatures at their minimum points.

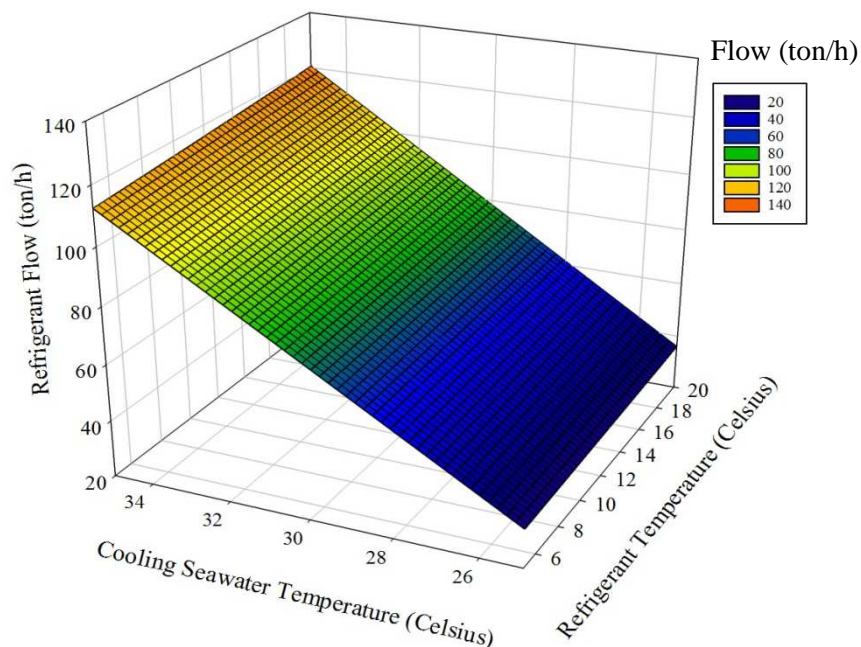


Figure 6: Effects of different temperatures on the refrigerant flow rate

5 CONCLUSION

In this study, utilizing a multi-stream condenser instead of two conventional shell & tube condensers in the DTECD heat recovery system was proposed. The result proved that this condenser leads to improve the exergy efficiency more than 15% and decrease the cooling water about 70%. Utilizing enriched ammonia refrigerant needs less flow rate. Operating the process with the minimum available temperatures of the refrigerant and cooling seawater leads to consume less amount of the refrigerant.

Nomenclature

| | |
|-----------|--|
| A | area (m ²) |
| A_c | cross section area (m ²) |
| C_p | specific heat capacity (kJ/Kg °C) |
| d_h | hydraulic diameter (m) |
| f | friction factor |
| F | correction factor |
| E | specific exergy (kJ/kg) |
| \dot{E} | exergy flow (kJ) |
| H | specific enthalpy (kJ/kg) |
| H_{fg} | specific latent heat (kJ/kg) |
| h | heat transfer coefficient of hot side (kJ/m ² °C) |
| L | exchanger length (m) |
| \dot{m} | mass flow rate (kg/h) |
| P | pressure (kPa) |
| \dot{Q} | heat rate (kJ) |
| R_u | universal gas constant (kJ/kgmol K) |
| S | specific entropy (kJ/Kg °C) |
| St | Stanton number |
| T | temperature (°C) |
| U | overall heat transfer coefficient (kJ/m ² °C h) |
| V | volume (m ³) |
| \dot{W} | work (kJ) |
| x | mole fraction |

Subscripts

| | |
|-----|---------------------------------|
| 0 | dead state or initial condition |
| C | cold stream |
| H | hot stream |
| i | inlet |
| j | number of element |
| o | outlet |

LM log mean

Abbreviations

| | |
|------------|----------------------|
| <i>BRN</i> | brine |
| <i>CPW</i> | condensed pure water |
| <i>FDW</i> | feed water |
| <i>FRW</i> | fresh water |
| <i>LPS</i> | low pressure steam |
| <i>MW</i> | motive water |
| <i>REJ</i> | rejection |
| <i>WF</i> | working fluid |
| <i>VAP</i> | vapor |

Greek

| | |
|----------|-------------------------------|
| μ | chemical potential (kJ/kgmol) |
| ψ | exergetic efficiency, % |
| δ | thickness (m) |
| ρ | density (kg/m ³) |

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