AN OPTIMIZED SYSTEM OF A COMBINED TRANS-CRITICAL CO_2 REFRIGERATION AND MULTI EFFECT DESALINATION SYSTEM

AL-hasan Ali Abdulwahid[1], Hongxia Zhao[1*], Qian Xue[1], Essam E Khalil[2], Yanhua Lai[1], JitianHan[1]
1 School of Energy and Power Engineering, Shandong University, Jinan, 250061, P.R. China
2 Department of Mechanical Power Engineering, Cairo University, Giza, Cairo, 11321, Egypt

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 CO_2 refrigeration and boosted multi effect desalination system. The two systems are analyzed and compared thermodynamically. The optimized system with two boosters produces around $361.72\ m^3/day$ of fresh water, on the other hand the original system with only one booster module produces around $290.3\ m^3/day$ 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

B	Booster module
CO_2	Carbon dioxide
Com	Compressor
Cp	Specific heat (kJ kg^{-1} K^{-1})
D	distillate production rate (kg s^{-1})
EES	Engineering Equation Solver
eva	Evaporator
F	Feed seawater
GAC	Gas cooler
h_{CO2}	specific enthalpy of CO_2 (kJ kg^{-1})
h_{fg}	latent heat (kJ kg^{-1})
HTA	Heat Transfer Area (m^2)
$\dot{m}$	Mass flow rate (kg s^{-1})
MED	Multi Effect Desalination
$\dot{Q}$	Heat transfer rate (kW)
TAC	Total Annual Cost ($)

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].

* Corresponding author.
E-mail address: hongxia.zhao@sdu.edu.cn (Hongxia Zhao).

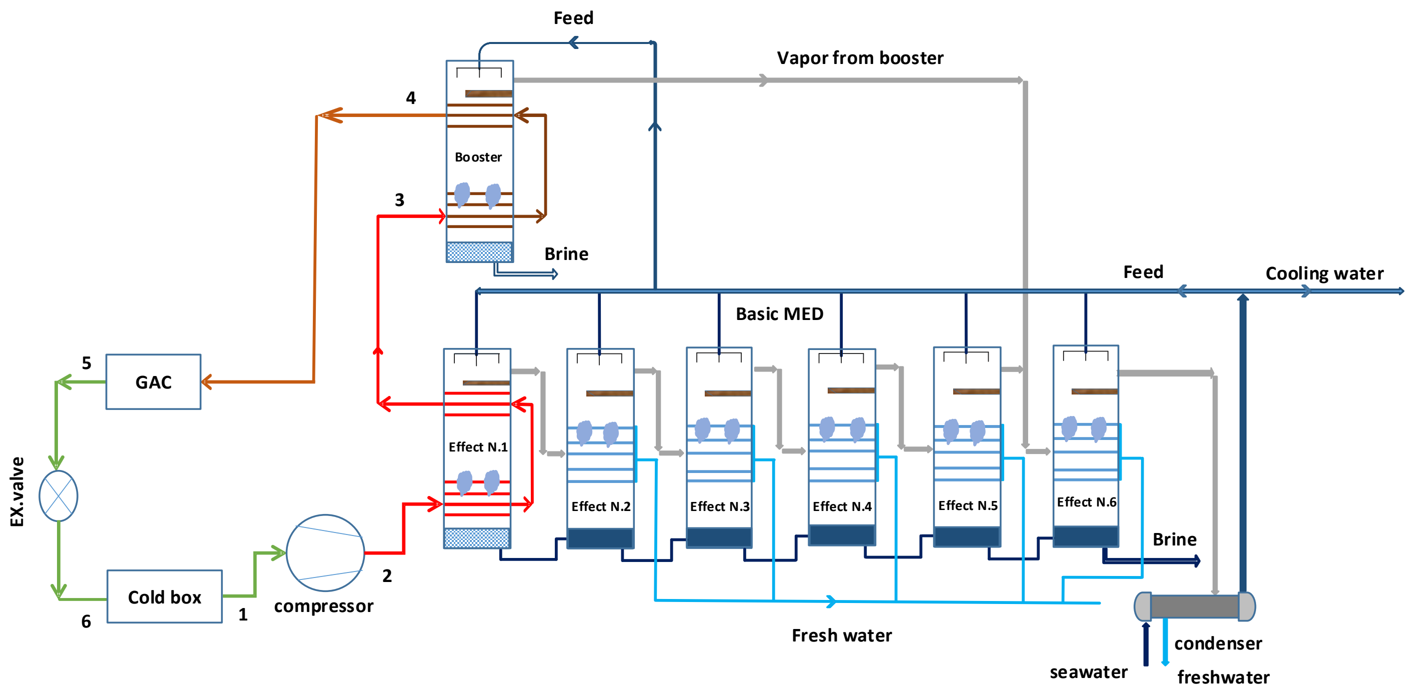

Fig 1 Schematic diagram of the combined B-MED system with trans-critical carbon dioxide refrigeration system (base system).

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 CO_2 Supercritical refrigeration system, a booster module and a basic MED system with 6 stages.

The high temperature CO_2 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℃), it is desirable to further use this energy into the booster module so as to increase the distillated 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.

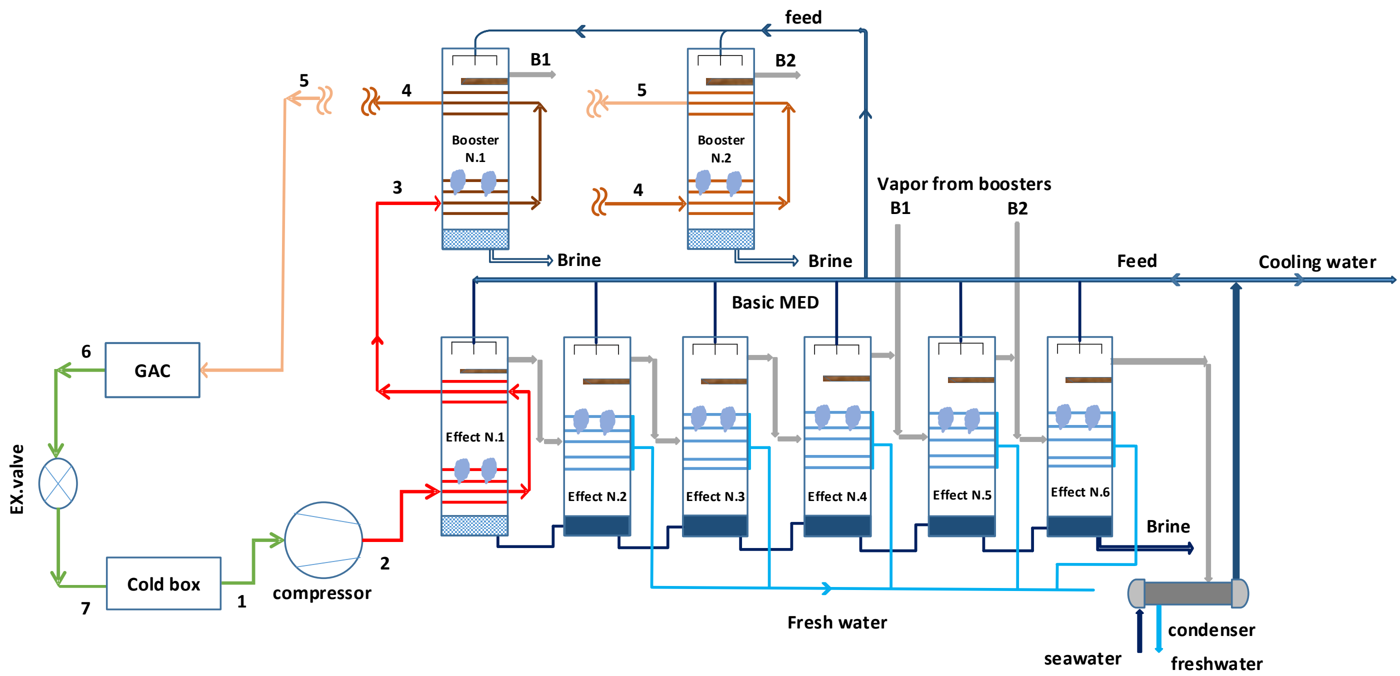

Fig 2 Schematic diagram of the combined system with two Booster Modules (the optimized system).

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:

$$\dot{Q}_{GAC} = \dot{m}_{CO2}\left(h_{CO2,4} - h_{CO2,5}\right) \quad (1)$$

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

$$h_{CO2,6} = h_{CO2,5} \quad (2)$$

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

$$\dot{Q}_{cold\ box} = \dot{m}_{CO2}\left(h_{CO2,1} - h_{CO2,6}\right) \quad (3)$$

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

$$\dot{W}_{com} = \dot{m}_{CO2}\left(h_{CO2,2} - h_{CO2,1}\right) \quad (4)$$

The COP of the CO_2 trans-critical refrigeration is defined as:

$$COP = \frac{\dot{Q}_{eva}}{\dot{W}_{com}} \tag{5}$$

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).

$$\dot{m}_{B1} = \dot{m}_{f1} - \dot{m}_{D1} \tag{6}$$

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

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 $2-n$)

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).

$$x_{sw}.\dot{m}_{f1} = x_{B(1)}\,.\dot{m}_{B(1)} \tag{8}$$

$$x_{sw}.\dot{m}_{fi} + x_{B(i-1)}.\dot{m}_{B(i-1)} = x_{B(i)}\,.\dot{m}_{B(i)} \tag{9}$$

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).

$$x_{sw}.\dot{m}_{f,B(1,2)} = x_{B,B(1,2)}\,.\dot{m}_{B,B(1,2)} \tag{10}$$

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

Where $\dot{m}_{f,B(1,2)}$, $\dot{m}_{B,B(1,2)}$ and $\dot{m}_{D,B(1,2)}$ 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):

$$\dot{Q}_{sensible,1} = \dot{m}_{f1}Cp_w\left(T_1 - T_f\right) = \dot{m}_{CO2}\left(h_{CO2,prim,3} - h_{CO2,3}\right) \tag{12}$$

$$\dot{Q}_{latent,1} = D_1\,h_{fg,1} = \dot{m}_{CO2}\left(h_{CO2,2} - h_{CO2,prim,3}\right) \tag{13}$$

$$\dot{Q}_{source} = \dot{Q}_{sensible,1} + \dot{Q}_{latent,1} \tag{14}$$

$$\dot{Q}_{source} = \dot{m}_{f1}Cp_w\left(T_1 - T_f\right) + D_1 h_{fg1} = \dot{m}_{CO2}\left(h_{CO2,2} - h_{CO2,3}\right) \tag{15}$$

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

$$\dot{Q}_{sensible,B(1,2)} = \dot{m}_{fB}Cp_w\left(T_{B(1,2)} - T_f\right) = \dot{m}_{CO2}\left(h_{CO2,prim,(4,5)} - h_{CO2,(4,5)}\right) \tag{16}$$

$$\dot{Q}_{latent,B(1,2)} = D_{B(1,2)}\,h_{fg,B(1,2)} = \dot{m}_{CO2}\left(h_{CO2,(3,4)} - h_{CO2,prim,(4,5)}\right) \tag{17}$$

$$\dot{Q}_{source,B(1,2)} = \dot{Q}_{sensible,B(1,2)} + \dot{Q}_{latent,B(1,2)} \tag{18}$$

$$\dot{Q}_{source,B(1,2)} = \dot{m}_{fB(1,2)}Cp_w\left(T_{B(1,2)} - T_f\right) + D_{B(1,2)}\,h_{fg,B(1,2)} = \dot{m}_{CO2}\left(h_{CO2,3} - h_{CO2,4}\right) \tag{19}$$

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

$$D_i h_{fgi} + D_B\,h_{fg,B} + Br_{i-1}Cp_w\left(T_{i-1} - T_i\right) = D_{i+1}h_{fgi+1} + \dot{m}_{fi}Cp_w\left(T_i - T_f\right) \tag{20}$$

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

$$\dot{Q}_{con} = D_n\,h_{fgn} = \left(\dot{m}_f + \dot{m}_{cw}\right).Cp_w.\left(T_{sea} - T_f\right) \tag{21}$$

Table 1
Input parameters as used in the present simulation [8].

Parameter		Value
Feed water inlet temperature	[°C]	28
Top brine temperature (TBT)	[°C]	70
Number of effects		6
Heating medium		CO_2
Mass flow rate of CO_2	[kg s^{-1}]	15
$P_{out,com}$	$[kPa]$	10570
Heat source inlet temperature	[°C]	110

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

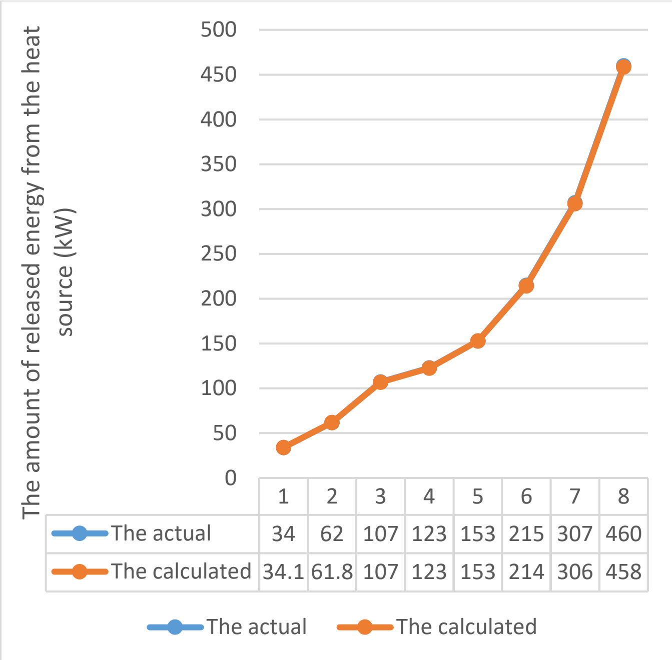

Fig 3 Comparison of the calculated amount of released energy from the heat source and the actual amount of released energy from Alfa Laval single effect freshwater generators.

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.

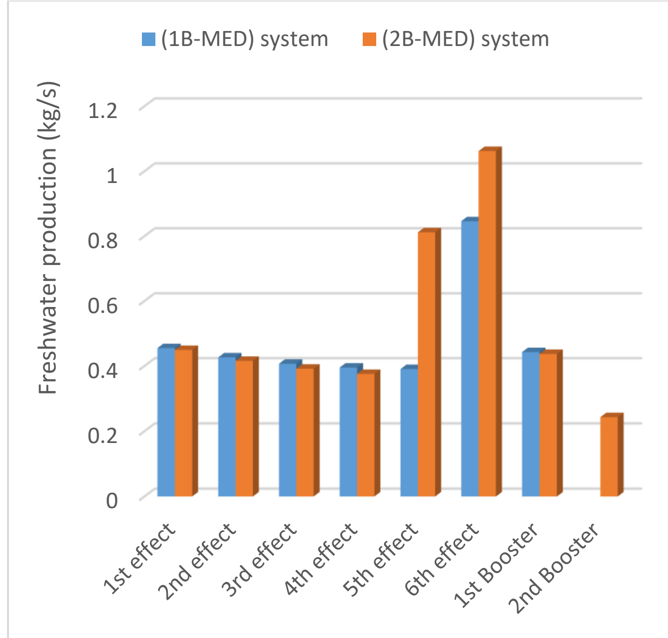

Fig 4 Comparison of the MED's fresh water production rate of the 2B-MED with that of 1B-MED.

According to Fig. 4 we can see that the 2B-MED system produces around $361.72\ m^3/day$ of freshwater, on the other hand the 1B-MED system produces around $290.3\ m^3/day$ 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 $50\ ℃$, 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 $44\ ℃$, 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)

Parameter	**(1B-MED)**	**(2B-MED)**
Condenser inlet temperature [℃]	28	28
Condenser outlet temperature [℃]	44	40.4
$\dot{Q}_{gas\ cooler}\ [kW]$	1059	472.5

Total distillated water $[m^3/day]$	290.3	361.72
Outlet temperature from the first booster [℃]	50	50
Outlet temperature from the second booster [℃]	-	44

5. CONCLUSION

An optimized system of a combined trans-critical CO_2 refrigeration and multi effect desalination system is studied based on thermodynamic analysis. The results show that the 2B-MED system produces around $361.72\ m^3/day$ of fresh water, while the 1B-MED system produces around $290.3\ m^3/day$ 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 ℃, 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 ℃, 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

This work is supported by the National Natural Science Foundation of China (Grant No. 51776110) and the Fundamental Research Funds of Shandong University (Grant No. 2017JC037).

REFERENCES

[1] Kalogirou SA. Seawater desalination using renewable energy sources. *Prog. Energy Combust. Sci.*, vol. 31, no. 3, pp. 242–281, 2005.

[2] Collins M *et al.*, *Long-term climate change: projections, commitments, and irreversibility*. 2013.

[3] Ghaffour N, Missimer TM, Amy GL. Technical review and evaluation of the economics of water desalination: Current and future challenges for better water supply sustainability. *Desalination*, vol. 309, no. 2013, pp. 197–207, 2013.

[4] Einav R, Harussi K, Perry D. The footprint of the desalination processes on the environment, *Desalination*, vol. 152, no. 1–3, pp.141–154,2003.

[5] Esfahani IJ, Yoo CA. highly efficient combined multi-effect evaporation-absorption heat pump and vapor-compression refrigeration part 2 : Thermoeconomic and fl exibility analysis, *Energy*, 2014.

[6] Farsi A, Mohammadi SMH, Mohammadi, Ameri M. Thermo-economic comparison of three configurations of combined supercritical CO 2 refrigeration and multi-effect desalination systems, *Appl. Therm. Eng.*, vol. 112, pp. 855–870, 2017.

[7] Janghorban I, Tae Y, Yoo C. A high ef fi cient combined multi-effect evaporation e absorption heat pump and vapor-compression refrigeration part 1 : Energy and economic modeling and analysis, *Energy*, pp. 1–15, 2014.

[8] Farsi A, Mohammadi SMH, Ameri M. An efficient combination of transcritical CO2refrigeration and multi-effect desalination: Energy and economic analysis, *Energy Convers. Manag.*, vol. 127, pp. 561–575, 2016.

[9] Hughes BR, Rezazadeh F, Chaudhry HN. Economic viability of incorporating multi-effect distillation with district cooling systems in the United Arab Emirates, *Sustain. Cities Soc.*, vol. 7, pp. 37–43, 2013.

[10] Wang Y, Lior N. Proposal and analysis of a high-efficiency combined desalination and refrigeration system based on the LiBr – H 2 O absorption cycle — Part 2 : Thermal performance analysis and discussions, *Energy Convers. Manag.*, vol. 52, no. 1, pp. 228–235, 2011.

[11] El-Dessouky HT, Ettouney HM. Fundamentals of Salt Water Desalination, Elsevier Science B.V., 2002.

[12] Alfa Laval marine & diesel product catalogue, single effect freshwater generator, model JWP-16/26-C series, 2003.