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Optimised exergy efficiency of a combined flash spray desalinator recovering discharge thermal energy

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ABSTRACT

This study focuses on an individualised exergy analysis approach for optimising the exergy efficiencies of the high irreversible components for a new discharge thermal energy combined desalination (DTECD) system. DTECD is a newly introduced energy recovery system to use the latent heat of waste steam. It is a combination of closed and open thermodynamic cycles, which cogenerates power and pure water. Based on the extra steam in an ammonia plant, two scenarios were modelled with respect to the exergy performance, which were individualised to find the irreversibility of each component. The results showed that exergy efficiency of the entire system is about 50%. Also, it was found that the working fluid evaporator and vacuum flash desalinator were the most exergy destructive equipment in the closed power cycle and open water cycle, respectively. The performance of the DTECD system utilising a vacuum single-stage spray flash evaporator is compared with similar technologies. Finally, recommendations are provided as to how the exergy efficiencies of these low-efficiency pieces of equipment can be optimised by changing the operating parameters such as vacuum pressure and working fluid concentration.

Keywords: Exergy; Discharge thermal energy combined desalination (DTECD); Cogeneration; Vacuum flash evaporator

1. Introduction

Nowadays, there is more interest in using hybrid desalination technologies for cogenerating water and power. Exergy analysis of a power system coupled with a thermal desalination system is used to improve the overall thermodynamic performance and to calculate the components with the greatest losses [1–6]. However, there are a few articles with respect to utilising the low-grade energy in a desalination process [7–10]. Uehara et al. [11] introduced a single-stage ocean thermal energy conversion (OTEC) desalination system in combination with a power cycle. Use of a flashing spray results in a higher vaporisation rate compared with conventional pool or flash evaporation [12]. Thus, most of the low-temperature thermal desalination (LTTD) experiments utilised a single-stage vacuum spray flash evaporator to desalinate water [13,14]. This system is different from multi-stage flash

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and pool evaporation technologies, which are used by conventional desalination systems. This technology has been advanced to industrial scale as reported by Kathiroli and Sistla [15], and Mutair and Ikegami [16]. Existing methods in the literature are not adequate for analysing the performance of a new steam recovery dual-purpose system, which is proposed here.

Discharge thermal energy combined desalination (DTECD) utilising low-pressure steam (LPS) has been proposed by Hosseini Araghi et al. [17] and is a novel heat recovery system where the authors focused only on the overall performance and cost evaluation of the entire process. DTECD is a developed form of LTTD, which proposed utilising the Rankine Cycle, a single-stage spray flash evaporator and an easy operating liquid jet ejector for cogenerating potable water and power. However, performance of all pieces of equipment, to find the technical bottlenecks of the system, was not analysed. Thus, an individualised exergy approach, as described here, should be conducted to evaluate exergy destruction at the component level of this system to allow for optimisation of the operating conditions. Fig. 1 compares the operating temperature range of the low-grade energy source used in the DTECD technology with the other thermal recovery technologies.

2. Process description

Here, the Razi ammonia complex is used as an industrial process to couple with the DTECD system. The waste heat available for driving the proposed DTECD system is 60–90 ton/h LPS (3 bara and 134˚C), which is presently dumped to the environment through a cooling tower. DTECD has two subsystems which are closed power and open water cycles [17,18]. In this research, the sea water is replaced with cooling water as a heat sink and the low-grade energy is recovered by adding the proposed DTECD system to the existing plant. The thermodynamic concepts such as temperature levels and inputs/outputs block diagram of the proposed DTECD system are shown in Fig. 2(a) and (b), respectively.

The schematic process flow diagram of the proposed water–power cogeneration process is shown in Fig. 3. Working fluid in the closed cycle is a mixture of ammonia and water (70 wt.% NH_3), which is pumped (PU-101) to the evaporator (EV-101). The required heat is carried by LPS into the evaporator. The separated vapour of rich fluid (NH_3-H_2O) from the flash separator (FS-100) passes through the turbine (TU-100) to generate power and, afterwards, is mixed with the bottom stream of the separator in the diffuser/absorber (DF-100). The generated working fluid liquefies in the condenser (CO-101) and the output is then recycled through the process. Passing through the evaporator (EV-100) of the power-generating cycle, the saturated LPS condenses. Then, the first stream of sea water mixes with the low-grade energy condensate from condenser (CO-101) and is then injected to the vacuumed flash drum (VD-100) to separate the rich brine from the pure vapour. Furthermore, extra sea water is used as a coolant to condense the pure vapour in the condenser (CO-102). Unlike conventional thermal desalination systems with multi-effect or multi-stage evaporators, just a single-stage vacuum flash evaporator (drum) is utilised to recover the available waste low-grade energy and desalinating the sea water. A water jet ejector is applied instead of the steam jet ejector to avoid consuming the valuable steam and reduce the maintenance cost. The water is recycled to the liquid jet ejector (EJ-100) generating a vacuum inside the flash drum.

3. Thermodynamic approach

3.1. General equations

The merit of applying second-law analysis is to calculate how much energy quality deteriorates, and lost potentially for using this energy for other purposes. This section describes the equations that were used to estimate exergy flows, exergy destructions and efficiencies of the components. Following Kotas [19], the exergy flow is obtained by:

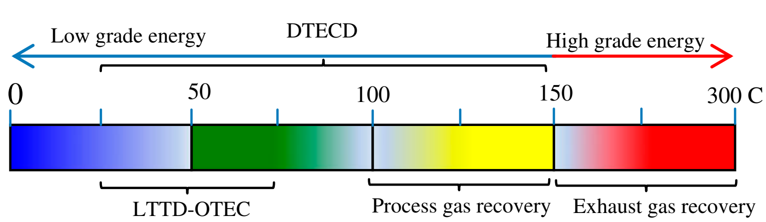


Fig. 1. Temperature range of different waste heat recovery technologies.

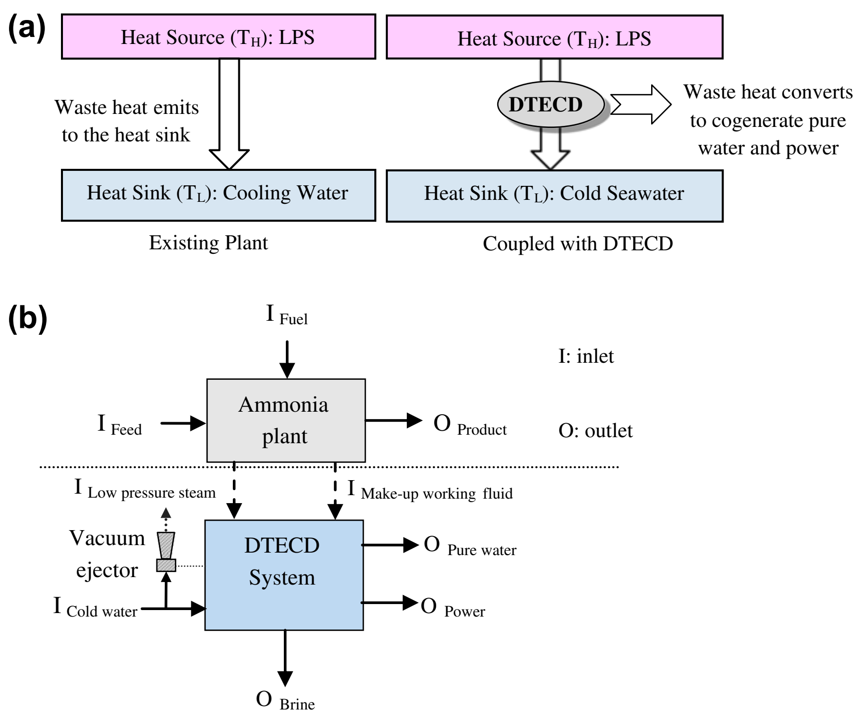


Fig. 2. Thermodynamic concepts of DTECD system: (a) DTECD system as a heat recovery and (b) Block diagram of DTECD system in interaction with an ammonia.

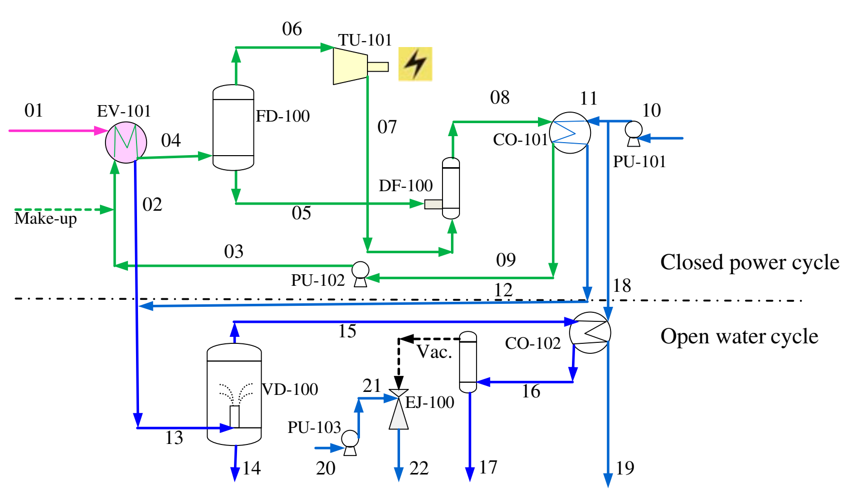


Fig. 3. Process flow diagram of the DTECD system.

$$\dot{E} = \dot{m}.(e^{\mathrm{tm}}+e^{\mathrm{ch}}) = \left[(h-h_0)-T_0(s-s_0)+\sum_{i=1}^{n} x_i(\mu-\mu_0)\right] \quad (1)$$

To calculate the enthalpy and the entropy of a saline water mixture as an ideal solution, the following equations could be considered, respectively [20]:

$$h_{\mathrm{mix}} = m_{\mathrm{f_s}}.h_{\mathrm{s}} + m_{\mathrm{f_w}}.h_{\mathrm{w}} \tag{2}$$

$$s_{\mathrm{mix}} = x_{\mathrm{s}}.s_{\mathrm{s}} + x_{\mathrm{w}}.s_{\mathrm{w}} \tag{3}$$

where x_{s} and x_{w} are defined as follows:

$$x_{\mathrm{s}} = \frac{M_{\mathrm{w}}}{M_{\mathrm{s}}(\frac{1}{m_{\mathrm{f_s}}} - 1) + M_{\mathrm{w}}} = \frac{1}{58.5(\frac{1}{m_{\mathrm{f_s}}} - 1) + 18} \tag{4}$$

$$x_{\mathrm{w}} = \frac{M_{\mathrm{s}}}{M_{\mathrm{w}}(\frac{1}{m_{\mathrm{f_w}}} - 1) + M_{\mathrm{s}}} = \frac{1}{18(\frac{1}{m_{\mathrm{f_w}}} - 1) + 58.5} \tag{5}$$

The salt enthalpy and entropy are obtained from the following equations:

$$h_{\mathrm{s}} = h_{\mathrm{s_0}} + c_{\mathrm{p_s}}(T - T_0) \tag{6}$$

$$s_{\mathrm{s}} = s_{\mathrm{s_0}} + c_{\mathrm{p_s}} \ln\left(\frac{T}{\mathrm{T}_0}\right) \tag{7}$$

For ideal solution, the component entropy is obtained by:

$$s_{\mathrm{i}} = s_{\mathrm{i,pure}}(T, P) - R_{\mathrm{u}} \ln x_{\mathrm{i}} \tag{8}$$

3.2. Individualised exergy equations

The destruction of exergy is caused by irreversibilities within the actual process and can be referred to as internal losses. On the basis of irreversibility approaches [19], the exergy equations of all internal components are individualised and presented in Table 1.

The total exergy efficiency of dual-purpose system is defined by:

$$\psi_{\mathrm{tot}} = 1 - \frac{\sum \dot{E}_{\mathrm{d}}}{\dot{E}_{\mathrm{in}}} \tag{14}$$

To overcome the complexity of the combined water and power model, some simplifying assumptions have been made. The proposed DTECD process is assumed to be under steady-state condition and that all pieces of equipment, except for the heat exchangers, operate adiabatically. Potential and kinetic energies were negligible in this study. In addition, efficiencies of the turbine and rotary pumps are 85 and 75%, respectively. Furthermore, it was assumed that there was not any air trapped in the system and that the piping pressure drop was negligible. Finally, the condensed water was assumed to be pure.

Table 1
The exergy equations associated with both cycles

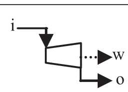
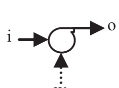
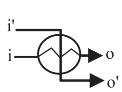
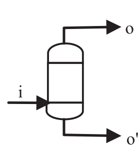
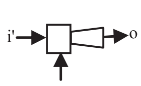
Operating unit	Box flow diagram	Exergy destruction	Exergitic efficiency	Formula no.
Turbine		$\dot{E}_{\mathrm{d}} = \dot{m}_{\mathrm{in}}e_{\mathrm{in}} - \dot{m}_{\mathrm{out}}e_{\mathrm{out}} - \dot{W}_{\mathrm{tu}}$	$\psi_{\mathrm{tu}} = \frac{\dot{W}_{\mathrm{tu}}}{\dot{E}_{\mathrm{out}} - \dot{E}_{\mathrm{in}}}$	(9)
Pump		$\dot{E}_{\mathrm{d}} = \dot{m}_{\mathrm{in}}e_{\mathrm{in}} - \dot{m}_{\mathrm{out}}e_{\mathrm{out}} + \dot{W}_{\mathrm{tu}}$	$\psi_{\mathrm{pu}} = \frac{\dot{E}_{\mathrm{out}} - \dot{E}_{\mathrm{in}}}{\dot{W}_{\mathrm{Pu}}}$	(10)
Heat exchanger		$\dot{E}_{\mathrm{d}} = \sum \dot{m}_{\mathrm{in}}e_{\mathrm{in}} - \sum \dot{m}_{\mathrm{out}}e_{\mathrm{out}}$	$\psi_{\mathrm{ex}} = \frac{(\sum \dot{E}_{\mathrm{out}} - \sum \dot{E}_{\mathrm{in}})_{\mathrm{cs}}}{(\sum \dot{E}_{\mathrm{in}} - \sum \dot{E}_{\mathrm{out}})_{\mathrm{hs}}}$	(11)
Vacuum drum		$\dot{E}_{\mathrm{d}} = \dot{m}_{\mathrm{in}}e_{\mathrm{in}} - \sum \dot{m}_{\mathrm{out}}e_{\mathrm{out}}$	$\psi_{\mathrm{vd}} = \frac{\dot{E}_{\mathrm{out}}}{\dot{E}_{\mathrm{in}}}$	(12)
Ejector		$\dot{E}_{\mathrm{d}} = \sum \dot{m}_{\mathrm{in}}e_{\mathrm{in}} - \dot{m}_{\mathrm{out}}e_{\mathrm{out}}$	$\psi_{\mathrm{ej}} = \frac{\dot{E}_{\mathrm{out}}}{\sum \dot{E}_{\mathrm{in}}}$	(13)

Table 2
Thermodynamic specifications of the streams (Refer to Fig. 3)

	Scenario 1					Scenario 2				
Stream	m_1 (ton/h)	T_1 (°C)	P_1 (kPa)	h_1 (kj/kg)	s_1 (kj/kg°C)	m_2 (ton/h)	T_2 (°C)	P_2 (kPa)	h_2 (kj/kg)	s_2 (kj/kg°C)
01	60	134	300	13,246	2.43	90	134	300	13,246	2.43
02	60	49	300	15,765	8.73	90	49	300	15,765	8.73
03	130	39.5	3,500	7,446	10.23	195	39.5	3,500	7,446	10.23
04	130	130	3,500	6,283	6.90	195	130	3,500	6,283	6.90
05	31	130	3,500	15,423	7.79	46	130	3,500	15,423	7.79
06	99	130	3,500	3,448	6.63	149	130	3,500	3,448	6.63
07	99	90	1,500	3,564	6.58	149	90	1,500	3,564	6.58
08	130	96	1,500	6,372	6.86	195	96	1,500	6,372	6.86
09	130	39	1,500	7,450	10.24	195	39	1,500	7,450	10.24
10	3,500	33	300	15,832	8.95	5,000	33	300	15,438	8.95
11	1,000	33	300	15,832	8.95	1,200	33	300	15,439	8.62
12	1,000	70	300	15,676	8.47	1,200	75	300	15,264	8.41
13	1,060	69	300	15,682	8.48	1,290	73	300	15,664	8.43
14	1,020	44	9	15,793	8.82	1,224	44	9	15,787	8.80
15	40	44	9	13,390	1.24	66	44	9	13,390	1.24
16	40	44	9	15,787	8.80	66	44	9	15,787	8.80
17	40	44	9	15,787	8.80	66	44	9	15,787	8.80
18	2,600	33	300	15,832	8.95	3,600	33	300	15,832	8.95
19	2,600	42	300	15,793	8.82	3,600	46	300	15,788	8.80
20	50	33	101.3	15,832	8.95	200	33	101.3	15,832	8.95
21	50	33.1	1,000	15,831	8.94	200	33.1	1,000	15,831	8.94
22	50	33.1	108	15,831	8.94	200	33.1	108	15,831	8.94
W_{Gross}	3.22 (MW)					4.83(MW)				
W_{Net}	3.20 (MW)					4.56 (MW)				
$\dot{m}_{pw}$	3,600 (ton/h)					5,000 (ton/h)				
ψ_{tot}	50%					50%				

4. Results and discussion

The thermodynamic properties such as enthalpy and entropy of the streams were modelled using ASPEN/HYSYS V.8.0. To simulate the new DTECD system, different equation of states (EoS) were selected. The electrolyte non-random two-liquid (ENRTL) model was applied for the open water cycle and the STEAM-TA model was selected to solve precisely the behaviour of the steam flows in the both cycles. Two scenarios were simulated based on the extra LPS in the ammonia plant to examine the feasibility of the DTECD model. In the first and the second scenarios, 60 and 90 ton/h LPS were consumed, respectively. All operating parameters, specifications and thermodynamic quantities of the process streams for both scenarios were calculated by simulator and recorded in Table 2. Furthermore, the amount of pure water production, the gross and net-work generations and total exergy efficiency of both scenarios are tabulated.

4.1. Exergy analysis

The exergy analysis was performed for the proposed DTECD system. The index line in Fig. 4 was set over the average amount of exergy destruction for all components and was used to highlight the highest exergy destructive components. Based on the exergy analysis, the most significant exergy-destroying component was the evaporator, due to the large difference between the temperatures of the inflow and the outflow streams. The second biggest contributor to destruction of the exergy was the vacuum drum. The significant irreversibility of the single-stage flash spray drum is because of the low-phase change rate of saline water to pure vapour. The exergy destruction in the turbine is low due to the utilisation of an ammonia–water mixture instead of the boiling feed water as working fluid, which increases its performance.

It has subsequently been found that the energy efficiency of shell and tube evaporator, flash spray desalinator and vacuum ejector are low, as shown in Fig. 5. The main reason for the irreversibility in the liquid jet ejector (eductor) is because of the high pressure drop required to produce a vacuum. However, the exergy destruction of eductor is negligible and its mechanical efficiency is acceptable compared to the other vacuum devices with respect to the operating conditions.

The exergy analysis showed that the performance of the base case can be enhanced by:

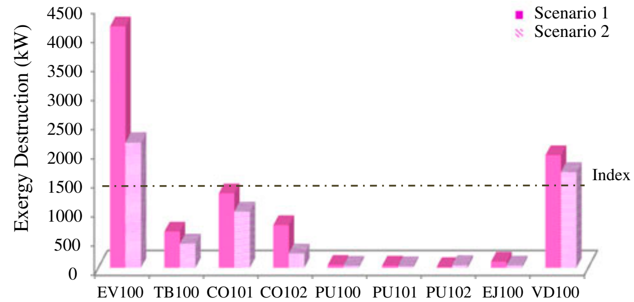


Fig. 4. Exergy destruction of the components.

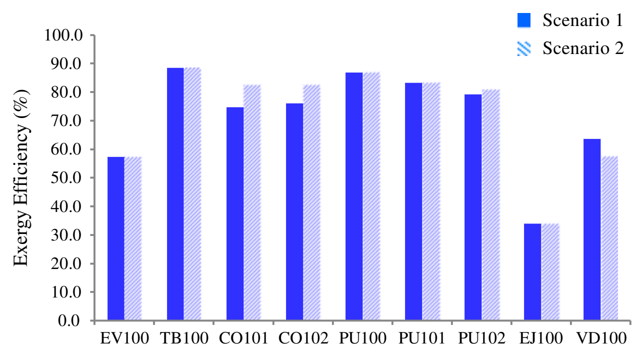


Fig. 5. Exergy efficiency of the components.

(1) recovering the exergy discharged by the LPS to the environment as a heat source in the proposed combined desalination and power system,
(2) utilising concentrated ammonia–water mixture as an alternative working fluid to increase the overall exergy efficiency of the closed power cycle,
(3) reducing the difference between the inlet temperatures of cold and hot streams in the heat exchangers as much as possible to reduce their exergy destruction,
(4) replacing the standard shell and tube exchangers with the high thermal efficiency plate and frame exchangers to improve the total thermal exergy efficiency and
(5) using rotary machineries such as turbine and pump with higher isentropic efficiencies to minimise the exergy destruction.

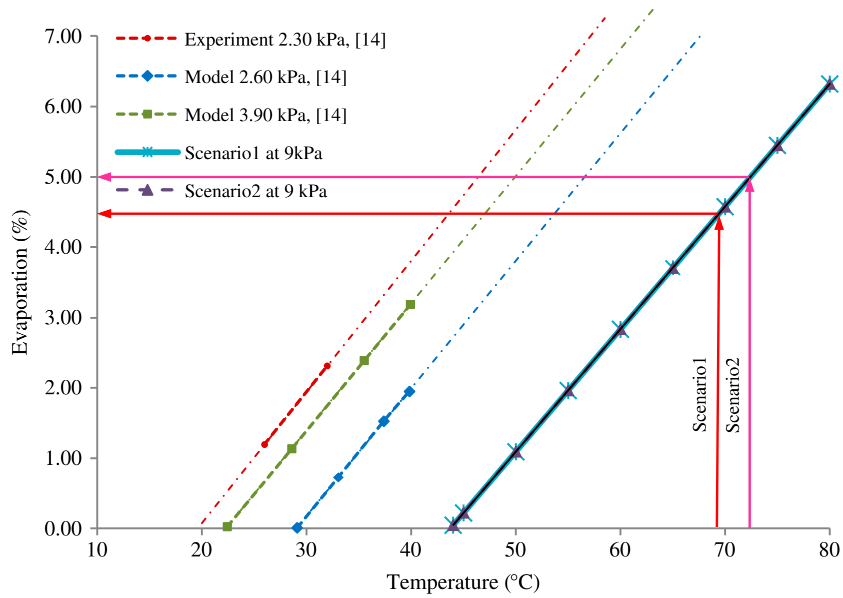


Fig. 6. Comparing the DTECD flash spray model with theoretical and experimental models.

Table 3
Comparison between DTECD and conventional thermal desalination technologies[a]

Parameter	DTECD	MSF	TVC
$\mathrm{GOR} = \frac{\dot{m}_{\mathrm{pw}}}{\dot{m}_{\mathrm{st}}}\left(\frac{h_{\mathrm{fg}(T_0)}}{h_{\mathrm{fg}(T_{\mathrm{st}})}}\right)$	1	1–10	3–15
$\mathrm{SEC} = \frac{\dot{Q}_{\mathrm{in}}}{\dot{m}_{\mathrm{pw}}}$ (kWh/m^3)	Negligible	13.5–25.5	11–28

[a]Table Ref. [21].

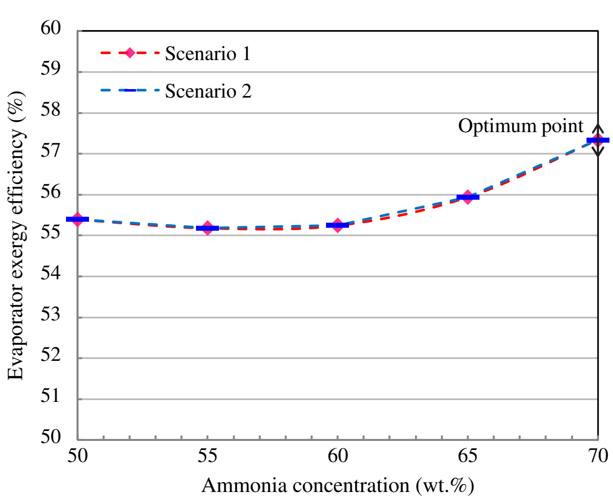


Fig. 7. The effect of ammonia–water concentration on the exergy efficiency of the evaporator.

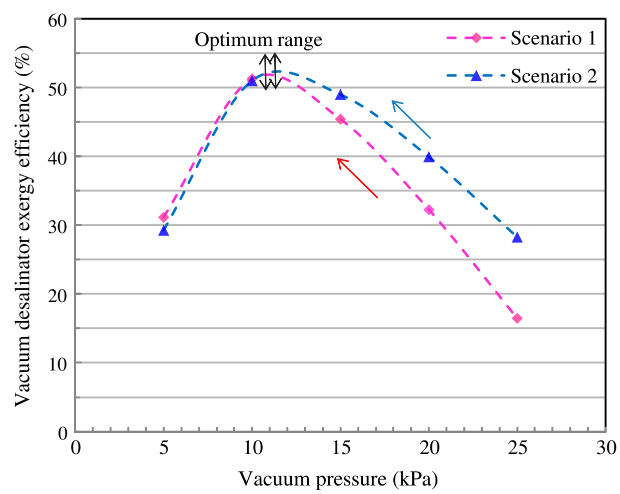


Fig. 8. The effect of vacuum pressure on the exergy efficiency of the flash spray drum.

4.2. *Performance of the single-stage spray flash desalinator utilising discharge thermal energy*

The DTECD system is a combined heat recovery system and it is not possible to compare all of its specifications with other high-capacity thermal desalination technologies. Thus, some meaningful aspects were only obtained from the proposed DTECD model to compare with the other technologies. Fig. 6 illustrates that the trend for model of a DTECD system at 9 kPa is in agreement with the theoretical models and previous experiment [14] regarding low-temperature vacuum spray flash evaporation. The operating conditions for both scenarios are specified in Fig. 6.

In Table 3, two performance terms such as gain output ratio (GOR) and specific energy consumption (SEC) of the DTECD technology were compared with those thermal desalination technologies, which utilise the flash evaporation technique. The DTECD system is a single-stage unit so its GOR in comparison with the multi-stage desalination system is low. However, its GOR is higher compared with the first stage of any pool flash evaporation. This system cogenerates power and water; therefore, the small amount of energy required by the system is supplied by its own closed power cycle.

4.3. Optimum operating range of the exergy destructive components

A parametric analysis was performed to verify the model results, understand the relationships between selected output and input parameters and optimise the efficiencies of the operating process. Note that the optimum operating range of the evaporator in the closed power cycle and the flash vacuum drum in the open water cycle were controlled by ammonia concentration and vacuum pressure, respectively. The other parameters such as waste steam temperature, working fluid flow and inlet temperature of the vacuum desalinator are fairly fixed.

Fig. 7 shows that the increase of the working fluid concentration leads to improve the evaporator exergy efficiency because the ammonia evaporation rate increases. However, the higher ammonia concentration may increase depreciation, manufacturing and safety costs.

As shown in Fig. 8, the decrease of vacuum pressure in the flash spray desalinator increases the exergy efficiency because the evaporation temperature of the saline water decreases, hence, the pure water production increases. Nonetheless, these upward trends towards the optimum points continue. Greater vacuum generation beyond the optimum range leads to a sudden decrease of the flash desalinator efficiency.

5. Conclusion

An individualised exergy methodology was performed for the first time to analyse all individual components of a DTECD system to find an optimum solution for improving its efficiency. Results showed that the evaporator and vacuum flash drum were the two most significant bottlenecks in this system with respect to the exergy destruction. By conducting parametric analysis to find the optimum range of significant parameters, it was found that working with 70% wt. of ammonia in the closed power cycle and 10–11 kPa of vacuum pressure in the open water cycle improves the irreversibilities of the evaporator and the vacuum drum. The maximum achievable exergy efficiency of the optimised heat recovery system was found to be 50%.

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Nomenclature

c_p	—	specific heat capacity (kJ/kg℃)
CO	—	condenser
E	—	specific exergy (kJ/kg)
$\dot{E}$	—	exergy flow (kJ)
EJ	—	ejector
EV	—	evaporator
H	—	specific enthalpy (kJ/kg)
h_{fg}	—	latent heat vaporisation (kJ/kg)
I	—	current Index
$\dot{m}$	—	mass flow rate (kg/h)
M	—	molecular weight (kg)
m_f	—	mass fraction
P	—	pressure (kPa)
PU	—	pump
$\dot{Q}$	—	heat rate (kJ)
S	—	specific entropy (kJ/kg℃)
T	—	temperature (℃)
TB	—	turbine
VD	—	vacuum drum
$\dot{W}$	—	work (kJ)
X	—	mole fraction

Subscripts

0	—	dead state or initial condition
d	—	destruction
ej	—	ejector
ex	—	heat exchanger
vd	—	flash drum
gen	—	generation
hs	—	hot stream
i	—	component i
max	—	maximum
min	—	minimum
mix	—	mixture
pu	—	pump
pw	—	pure water
s	—	salt
st	—	steam
tot	—	total
tu	—	turbine
w	—	water

Superscripts

ch	—	chemical
P	—	pressure
T	—	temperature
tm	—	thermomechanical

Greek

Φ	—	phase
μ	—	chemical potential (kJ/kmol)
ψ	—	exergetic efficiency (%)

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