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Chapter

A Novel Low-Temperature Thermal Desalination Technology Using Direct-Contact Spray Method

Qian Chen, Muhammad Burhan, Muhammad Wakil Shahzard, Raid Alrowais, Doskhan Ybyraiymkul, Faheem Hassan Akhtar, Yong Li and Kim Choon Ng

Abstract

Due to the emerging water crisis, the global desalination capacity has been expanding exponentially in the past few decades, leading to substantial amount of primary energy consumption. Therefore, the exploration of energy-efficient desalination processes and alternative energy sources has been the subject of great research interests. The spray-assisted low-temperature desalination (SLTD) system is a novel method for desalination that enables efficient renewable energy utilization. It works on the direct-contact spray evaporation/condensation mechanism and uses only hollow chambers. The merits include enhanced heat and mass transfer, lower initial and operational costs, and reduced scaling and fouling issues. This chapter presents a study on the SLTD system driven by sensible heat sources. The working principle of the system will be introduced first. Then a thermodynamic analysis will be presented to obtain the freshwater productivity under different design and operational conditions. Additionally, the energy utilization level will be quantified to highlight the energy wastage when operating with sensible heat sources. Afterward, the system configuration will be modified to maximize the utilization of sensible heat sources and promote productivity. Finally economic viability of the modified design will be evaluated.

Keywords: direct-contact spray, thermal desalination, sensible heat source, thermodynamic analysis, internal heat recovery

1. Introduction

Freshwater is the key resource for the continuation of human society. With the growth of world population, the world water consumption has been increasing exponentially in the past decades [1, 2]. Meanwhile, freshwater resources on the earth are limited, and they are degrading and depleting due to overexploration and environmental pollution [3]. Consequently, the global water deficit is becoming

more and more severe. By 2030, the world water deficit is expected to reach 2700 billion m^3/year [4], and the population that will suffer from water shortage will exceed 1.6 billion [5]. Therefore, it is of ultimate importance to develop new and sustainable sources for freshwater supply.

Seawater desalination is one of the most promising solutions to the issues associated with water shortage [6]. It is the process that separates a portion of freshwater from the seawater with the input of energy or work. The great potential of seawater desalination lies in the wide availability of seawater. More than 70% of the earth surface is covered by oceans. More importantly, most of the cities that are facing water shortage are located along the coast [7], and nearly 40% of the world population lives near the sea [8]. With the advances in desalination technologies, the great potential of desalination has been translating into an expanding global desalination capacity. So far, more than 16,000 desalination plants have been installed in nearly 150 countries, and the overall desalination capacity has exceeded 90 million m^3/day [9].

Existing desalination technologies in the market can be divided into two categories, namely, membrane-based processes and thermally driven processes. The membrane-based process, represented by reverse osmosis (RO), uses a semipermeable membrane to separate freshwater from seawater. The membrane only allows water molecules to pass, leaving behind the salt. In a RO system, the seawater is pressurized to overcome the osmotic pressure and drive the diffusion of water molecules. RO systems not only exhibit low-energy consumption but also have small plant footprint, making it the dominating technology in the desalination market. By 2016, the market share of RO has exceeded to 63% [9].

Thermally driven processes, as presented by multi-effect distillation (MED) and multistage flash distillation (MSF), separate water and salt by evaporation and subsequent condensation. Since salts are not volatile, thermally driven processes are able to reject almost 100% of the dissolved salt and achieve a very high distillate quality. The energy consumption of MED and MSF is much higher than RO due to the high latent heat of vaporization. However, thermally driven processes utilize low-grade heat instead of electricity. With various waste heat available in different industrial processes, thermal desalination processes are sometimes more appealing than RO. Moreover, thermal processes are less sensitive to the change of feed salinity, and they are able to handle harsh feed conditions where RO is not applicable.

One major barrier that hinders the wider application of MED and MSF is the high initial plant cost, which limits them to large-scale operations. However, cost-effective low-grade heat sources, such as industrial waste heat, are often available in a small amount. Therefore, it is of great impetus to develop small-scale thermally driven processes. Humidification-dehumidification (HDH) processes and membrane distillation (MD) are two emerging technologies that are suitable for small-scale operation. But both processes are facing key challenges that should be overcome before wider application is possible. The productivity of HDH is limited by the small vapor-carrying capability of air, and the footprint size is relatively large due to small heat and mass coefficients between wet air and condenser surface [10]. On the other hand, MD are facing the issues of small distillate flux, membrane degradation due to scaling and fouling, and relatively low thermal efficiency.

While on-going research efforts are being conducted to address the issues faced by HDH and MD, the development of more advanced thermal processes is also of importance. The spray-assisted low-temperature desalination (SLTD) technology is a recently proposed method that mitigates the issues faced by conventional thermal processes. It employs direct-contact evaporation/condensation method, thus eliminating metallic surfaces inside the system. The merits include (1) promoted heat and mass transfer due to direct contact between water and vapor, (2) reduced

scaling and fouling potential, and (3) lower initial and operational costs. In our previous publications, the technical viability of the direct spray method has been demonstrated experimentally [11, 12] and analytically [10, 13]. The thermodynamic performance [14–17] and economic viability [16] have also been evaluated. However, all of these studies employ steam as the heat source, while none of them have ever considered sensible heat sources.

This chapter specially investigates an SLTD system driven by sensible heat sources. Different from steam, sensible heat sources suffer from a temperature drop in order to release energy. To sustain a temperature difference for heat transfer, the heat source is always at a higher temperature than the medium that is being heated. Consequently, a portion of enthalpy in the heat source is usually left unused. Therefore, conventional energy efficiency measurements are not applicable for sensible heat sources since they only look into heat extraction from the heat sources and neglect un-extracted energy (which is negligible in steam-driven systems). In the following sections, the productivity of the SLTD system will be evaluated under different design and operational conditions. The energy utilization level will also be calculated to quantify the amount of unused enthalpy in the heat source. Then a modified configuration will be proposed to enhance energy utilization and boost freshwater production. A cost analysis will also be conducted to evaluate the economic viability of the proposed configuration.

2. System description and mathematical modeling

Figure 1 shows a simplified schematic of the proposed system, consisting of a series of evaporator-condenser stages, three sets of heat exchangers, and a vacuum

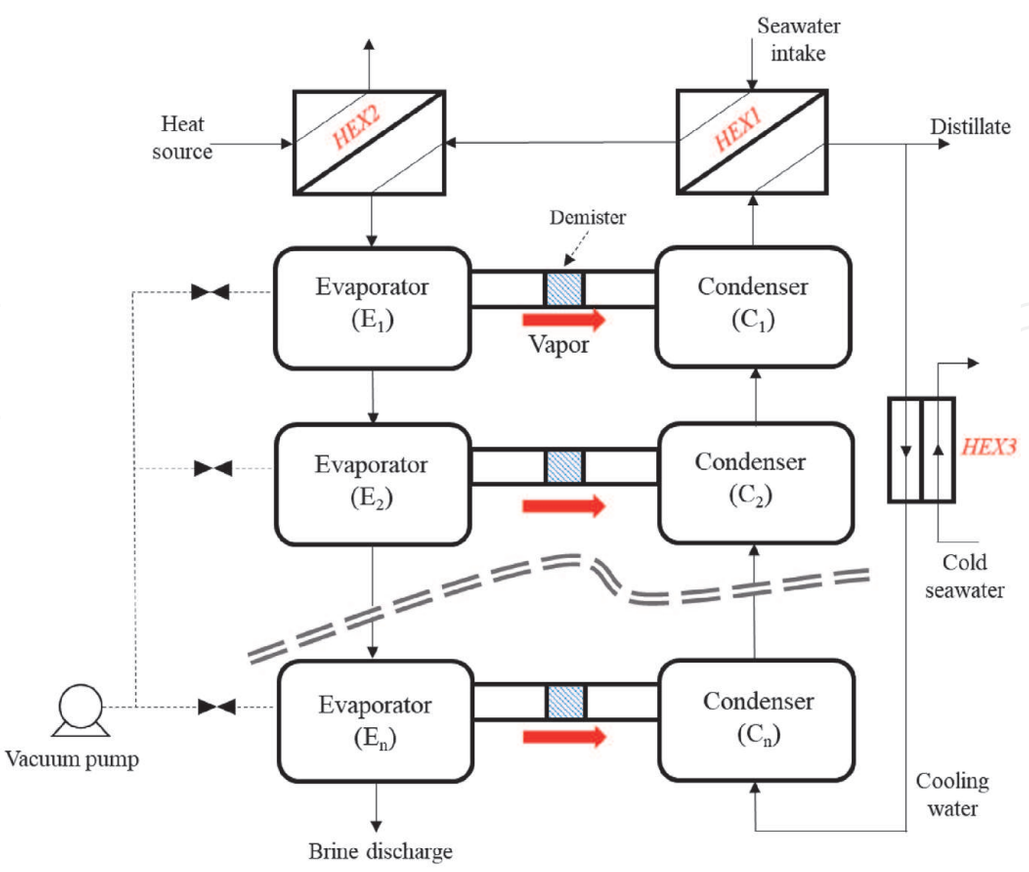


Figure 1.
Schematic of the spray-assisted low-temperature desalination system.

pump. Both evaporators and condensers are empty vessels operating under vacuum conditions. During operation, seawater is preheated externally and then sprayed into the evaporators, while cold cooling water is sprayed into the adjacent condensers. Driven by the partial vapor pressure difference, a portion of water evaporates from the seawater surface, travels to the condenser, and is condensed by the cooling water. The unevaporated seawater is then sprayed into the following evaporator, while the cooling water enters the previous condenser. The production stages are subjected to sequentially lowered pressure conditions so that the evaporation/condensation cycle is repeated. Finally the brine is disposed in the last evaporator, while the mixture of cooling water and distillate leaves from the first effect. Due to the accumulation of condensation heat, the mixed stream has a high temperature and is allowed to exchange heat with the intake seawater in *HEX1* to recover the condensation heat. Then the distillate is separated, and the cooling water is further cooled down in *HEX3* using the seawater before returning back to the stages. Meanwhile, the preheated seawater is directed to *HEX2* to be further heated to the desired temperature using an external heat source. The vacuum pump creates an initial vacuum condition at the beginning and removes the non-condensable gases dissolved in the seawater during operation.

The performance of the system can be predicted by analyzing heat and mass transfer between water and vapor in the vacuum environment as well as heat and mass balances among different components. **Figure 2** shows the schematic of the system components. The symbols that are used in the mathematical model are also included in **Figure 2**, while the governing equations are summarized in **Table 1**.

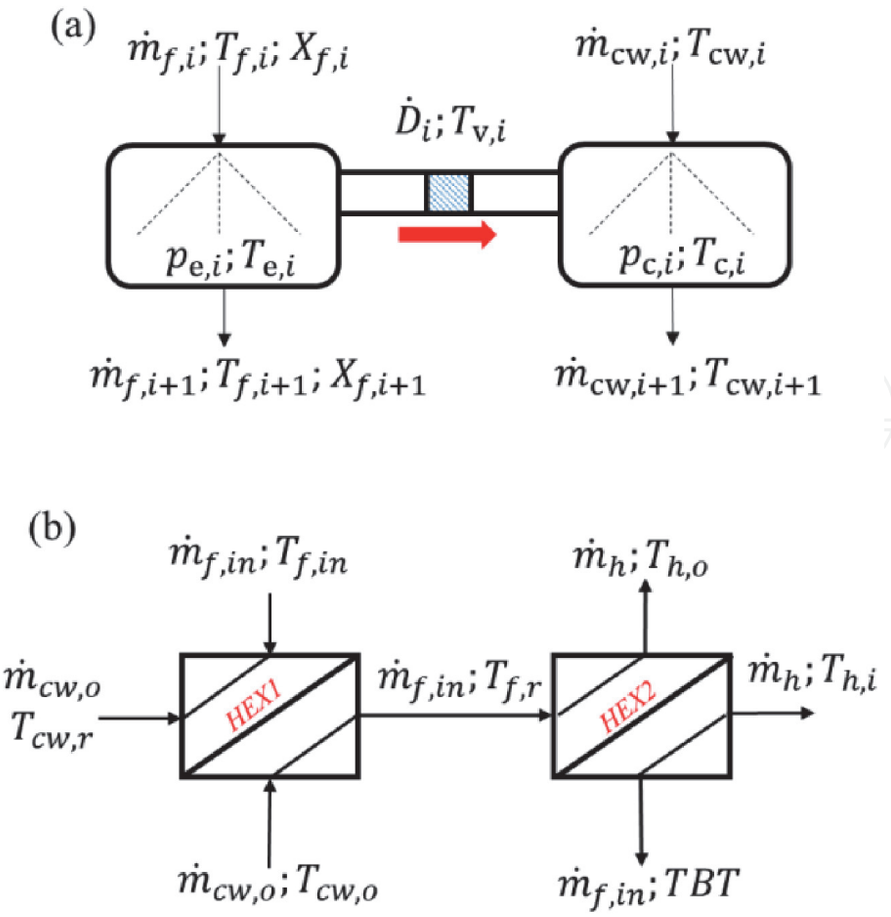


Figure 2.
Schematic of the system components with the main parameters: (a) production stages, and (b) heat exchangers.

Component	Equation	No.	Remarks
Evaporators	$T_{el,i+1} = T_{ev,i} + (T_{el,i} - T_{ev,i})\theta_{c,i}$	(1)	Seawater temperature change in evaporator; θ represents completeness of evaporation process [13]
	$T_{el,1} = TBT$	(2)	Inlet condition at the first evaporator
	$\dot{D}_{v,i} = \dot{m}_{el,i}\frac{c_{pel,i}\left(T_{el,i} - T_{el,i+1}\right)}{h_{fge,i}}$	(3)	Amount of vapor produced in evaporator
	$\dot{m}_{el,i+1} = \dot{m}_{el,i} - \dot{D}_{v,i}$	(4)	Seawater mass conservation
	$\dot{m}_{el,i+1}X_{l,i+1} = \dot{m}_{el,i}X_{l,i}$	(5)	Salt mass conservation
Condensers	$T_{cl,i} = T_{cv,i} - (T_{cv,i} - T_{cl,i+1})\theta_{c,i}$	(6)	Cooling water temperature; θ represents the completeness of condensation process
	$T_{cl,n+1} = T_{cw,in}$	(7)	Inlet condition at last condenser
	$T_{cv,i} = T_{ev,i} - BPE_I - T_{loss}$	(8)	Temperature drop of vapor due to (1) boiling point elevation caused by dissolved salt and (2) temperature drop across the demister due to pressure drop
	$\dot{D}_{v,i} = \dot{m}_{cl,i}\frac{c_{pcl,i}\left(T_{cl,i} - T_{cl,i+1}\right)}{h_{fgc,i}}$	(9)	Amount of vapor condensed on cooling water surface
	$\dot{m}_{cl,i} = \dot{m}_{cl,i+1} + \dot{D}_{v,i}$	(10)	Cooling water mass conservation
Heat exchangers	$\dot{m}_{sw,in}c_{p,sw}(T_{sw,r} - T_{sw,in}) = \dot{m}_{cw,o}c_{p,cw}(T_{cl,o} - T_{cl,r})$	(11)	Heat balance in *HEX1*
	$\dot{m}_{sw,in}c_{p,sw}(TBT - T_{sw,r}) = \dot{m}_h c_{p,h}(T_{h,i} - T_{h,o})$	(12)	Heat balance in *HEX2*

Table 1.
Model equations for the spray-assisted low-temperature desalination system.

3. Performance analysis

Employing the developed model, a 10-stage system operating at a top brine temperature of 70°C is firstly analyzed. Without loss of generality, the flowrate of the seawater is considered to be 10 kg/s. The flowrate of the cooling water and the heat source (considered to be hot water in this study) is equal to the feed flowrate in order to achieve the optimal system performance [14, 15]. The intake seawater is assumed to have a temperature of 25°C, and it will cool down the cooling water to 30°C in the counterflow heat exchanger (*HEX3*). **Figure 3** shows the temperatures for seawater and cooling water as well as freshwater productivity in each effect. It is obvious that seawater temperature drops successively along the stages and finally the brine is disposed at 33°C. On the other hand, the cooling water temperature increases in the reverse direction after absorbing the condensation heat and approaches the top brine temperature when leaving the first stage. The productivity also shows a descending trend, which is attributed to the drop of seawater flowrate after partial evaporation. The overall productivity is calculated to be 0.61 kg/s.

The productivity is expected to change when varying the number of stages and the top brine temperature. The former will change the efficiency of energy utilization, while the latter determines the total amount of energy that is available for

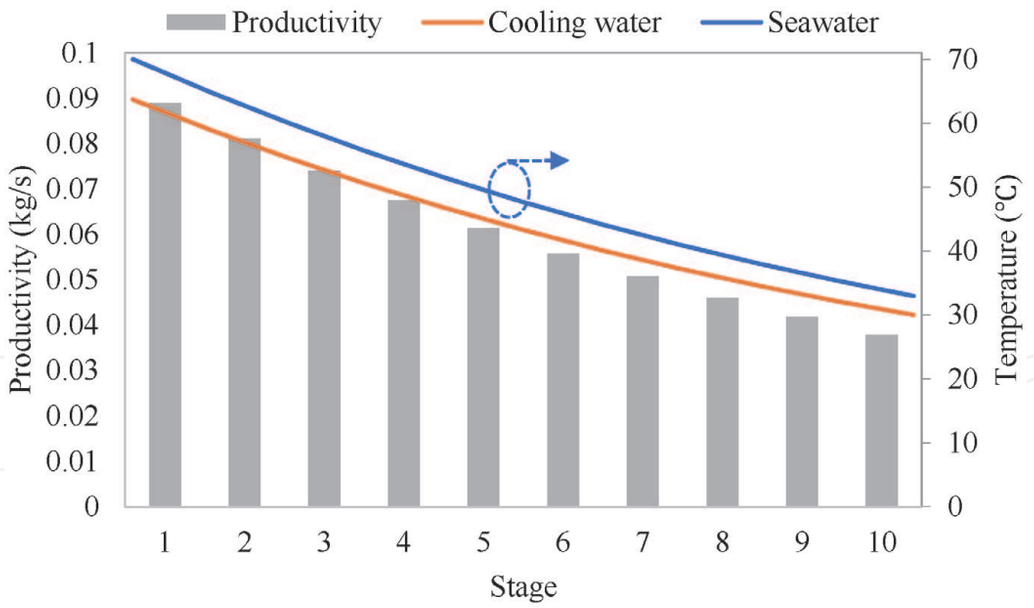


Figure 3.
Temperature and production profile for a typical system with 10 production stages.

evaporation. **Figure 4** shows the productivity under different conditions. The productivity firstly increases when the system has more operating stages, and the increasing trend gradually gets saturated. The reason is that increasing the number of stages improves only the energy efficiency, while the total amount of available heat is fixed. On the other hand, a higher top brine temperature will lead to remarkable improvement in productivity due to the availability of more heat source.

The majority of heating requirement is satisfied by the recovered condensation heat in *HEX2*, and the external heat source only undertakes a small portion. Take the 10-stage system operating at a *TBT* of 70°C as an example. As shown in **Figure 3**, the cooling water leaving the first condenser has a temperature of 63.7°C and is able to heat up the intake seawater to 58.7°C, considering an approach

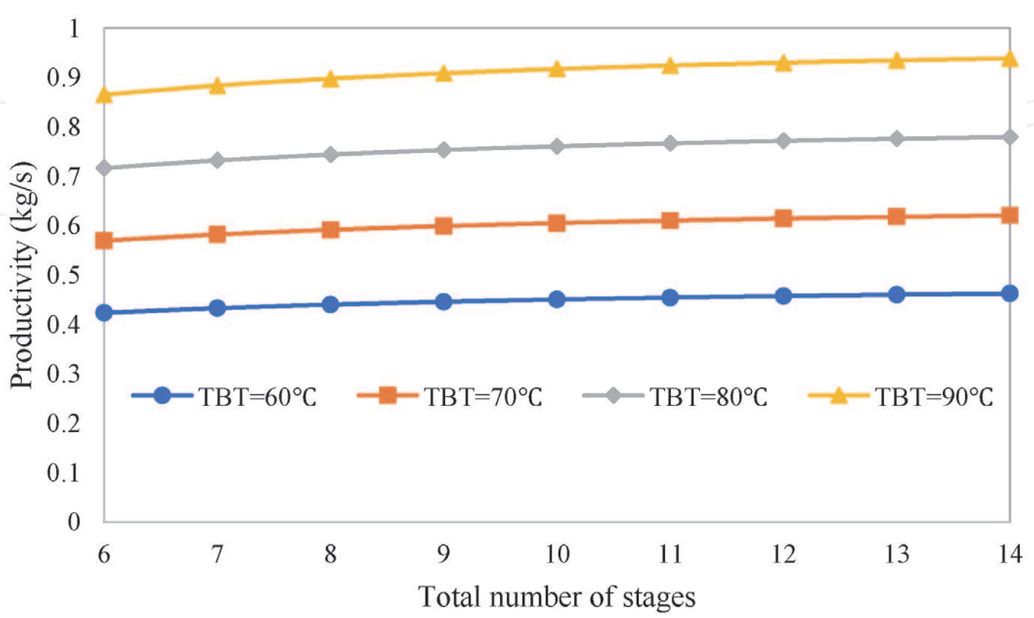


Figure 4.
Productivity under different top brine temperatures and operating stages.

temperature difference of 5°C in the heat exchanger. Therefore, the heat source only needs to heat up the seawater from 58.7 to 70°C, and its temperature drops from 75 to 63.7°C. Under such situation, only a very small portion of enthalpy is utilized from the heat source, while the remaining is left unused.

To quantify the efficiency of energy utilization, the level energy extraction from the heat source is calculated using the following equation:

$$Energy\ utilization = \frac{\dot{Q}_{utilized}}{\dot{Q}_{available}} = \frac{\dot{m}_h c_{p,h}(T_{h,i} - T_{h,o})}{\dot{m}_h c_{p,h}(T_{h,i} - T_{amb})} \tag{13}$$

The values of energy utilization under different design and operational conditions are plotted in **Figure 5**. Under all of the conditions, <20% of enthalpy is extracted from the heat source, while >80% is left unused. Such a low level of energy utilization results in significant wastage of the heat source and requires further optimization of the system.

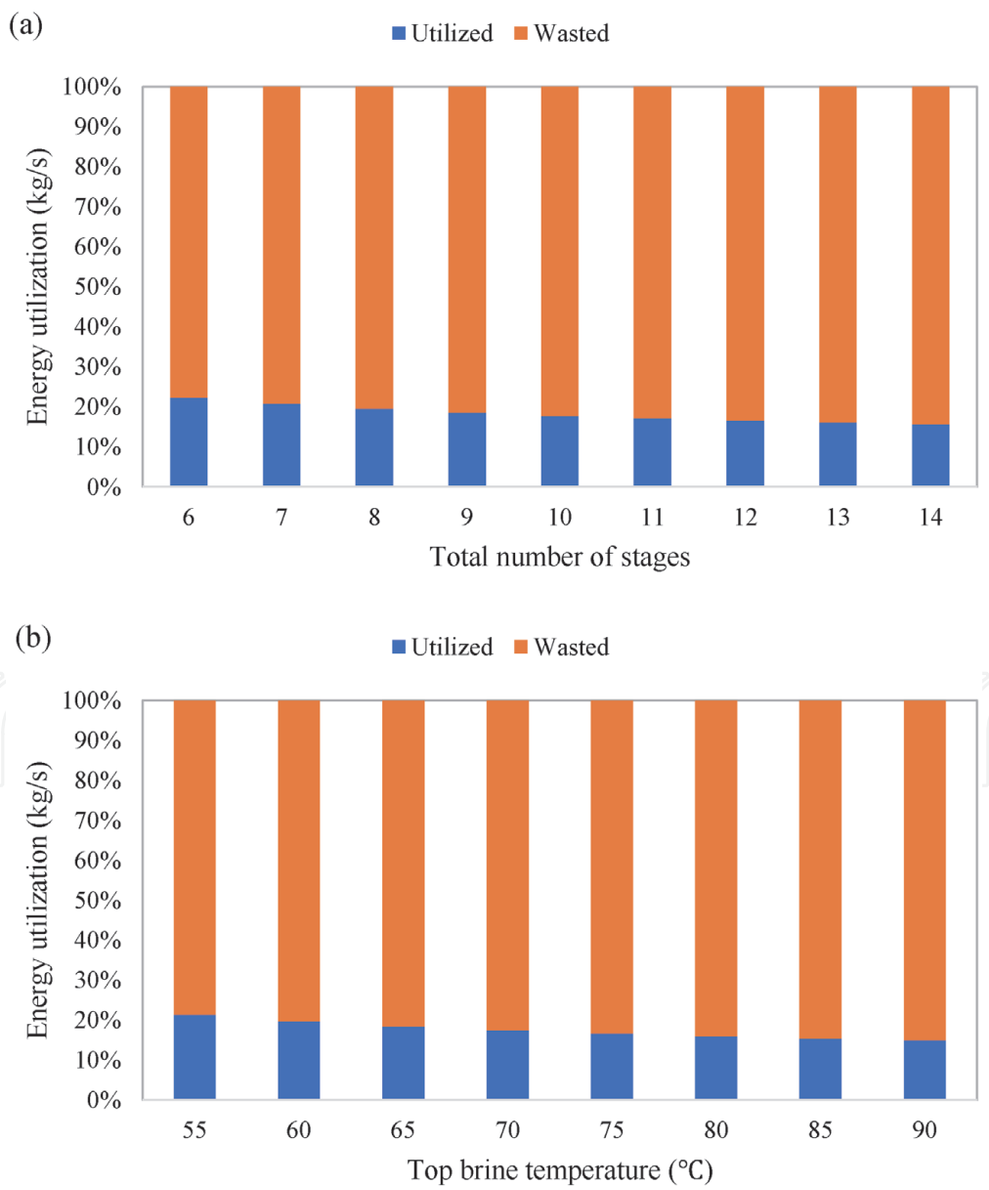


Figure 5.
Level of energy utilization under different (a) numbers of operating stages and (b) top brine temperatures.

4. System optimization

From the temperature profiles shown in **Figure 3**, it can be found that the cooling water in the earlier stages has a high temperature than the seawater in the later stages. For example, the cooling water leaving the second condenser has a temperature of 58.9°C, while the seawater enters the fifth evaporator at 51.4°C. If these two streams could exchange heat with each other, both the first and the fifth stages will have larger operating temperature differences, leading to a promoted productivity.

To facilitate such heat recovery, additional heat exchangers are added to the system, as demonstrated in **Figure 6**. For the first few condensers, the cooling water leaving the *i*th condenser is employed to preheat the seawater that is to be supplied into the *(i+NR)*th evaporator. To conserve heat exchanger area, a minimum temperature difference is set between the seawater and the cooling water. If the temperature difference between the two streams is below this value, no heat exchanger will be added for this evaporator-condenser pair.

The temperature variation for the seawater and cooling water, which are previously described by Eqs. (1) and (6), is now expressed as

$$T_{el,i+1} = \begin{cases} T_{ev,i} + (T_{el,i} - T_{ev,i})\theta_{e,i} & (i < NR) \\ T_{ev,i} + (T_{el,i} - T_{ev,i})\theta_{e,i} + \dfrac{\dot{Q}_{i-NR}}{\dot{m}_{el,i}c_p} & (i \geq NR) \end{cases} \tag{14}$$

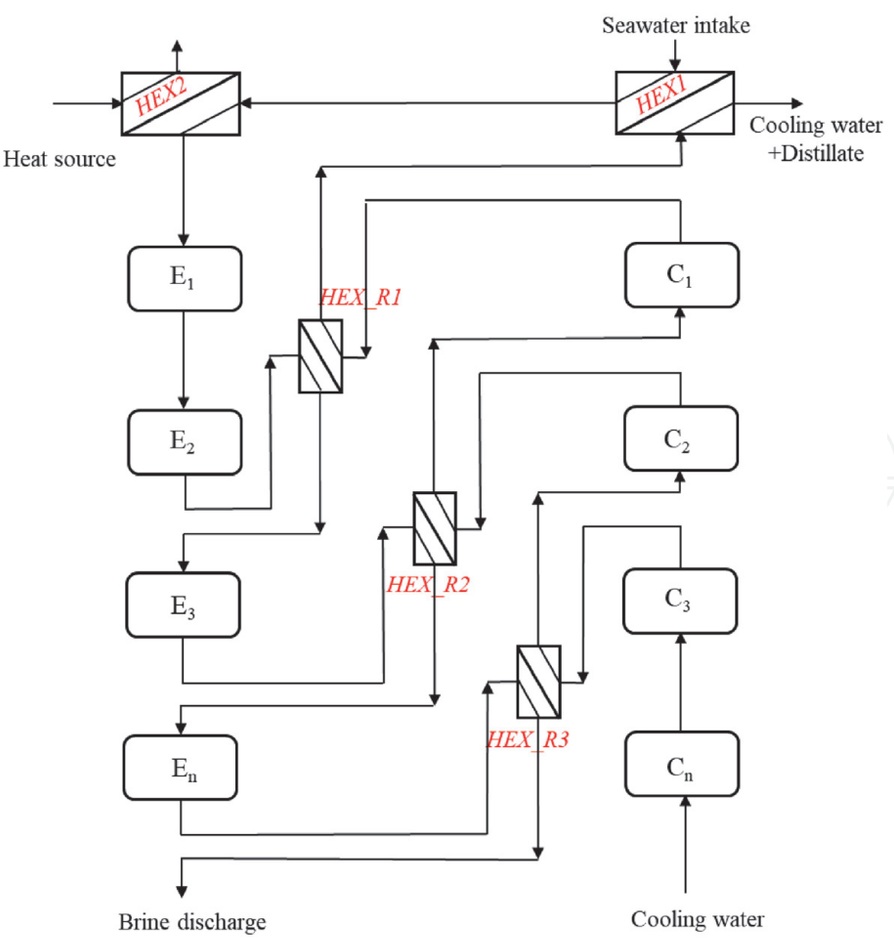


Figure 6.
Modified system with additional heat exchangers to enable internal heat recovery.

$$T_{cl,i} = \begin{cases} T_{cv,i} - (T_{cv,i} - T_{cl,i+1})\theta_{c,i} & (i > N_{total} - NR) \\ T_{cv,i} - (T_{cv,i} - T_{cl,i+1})\theta_{c,i} - \dfrac{\dot{Q}_i}{\dot{m}_{cl,i}c_p} & (i \le N_{total} - NR) \end{cases} \quad (15)$$

$$\dot{Q}_i = \begin{cases} \dot{m}_{cl,i}c_p(T_{cl,i} - T_{el,i+NR-1} - \Delta T_{HR})\theta_{c,i} & (T_{cl,i} > T_{el,i+NR-1} + \Delta T_{HR}) \\ 0 & (T_{cl,i} \le T_{el,i+NR-1} + \Delta T_{HR}) \end{cases} \quad (16)$$

Figure 7 shows the temperature profile for the modified system with 10 stages and a *TBT* of 70°C. The optimal value of *NR* is found to be 3 for this configuration. It is clearly shown in **Figure 7** that the feed seawater entering the 4th, 7th, and 10th evaporators are preheated by the cooling water leaving the 1st, 4th, and 7th condenser, respectively. As a result, productivity in the 4th, 7th, and 10th stages is significantly improved, as compared to the original system whose productivity shows a descending trend along the stages. The overall productivity has also been increased due to the elevated temperature gradient, from 0.61 kg/s to 0.74 kg/s.

Figure 8 presents the improvement of productivity for the modified system under different design and operational conditions. **Figure 8(a)** shows the performance under different numbers of operating stages. The *TBT* is kept fixed at 70°C, and the values of *NR* are varied according to the number of operating stages to achieve maximal fresh water yield. The modified system overperforms the original one with an increase ranging from 20 to 45%. Another difference is that the productivity does not increase monotonously with the number of operating stages for the modified system. This is attributed to the discontinuous values of optimal *NR* under different numbers of operating stages [18]. **Figure 8(b)** presents the system performance under different *TBT* for a 10-stage system. The productivity is improved by 55–78% compared to the original configuration. The higher the *TBT*, the more pronounced the improvement is.

As a result of the heat recovery action, the cooling water temperature leaving the first condenser is much lower. Consequently, less recovered condensation heat is available for seawater preheating in *HEX2*, and more heat input is required from the heat source. Therefore, the outlet temperature of the heating medium will be lower,

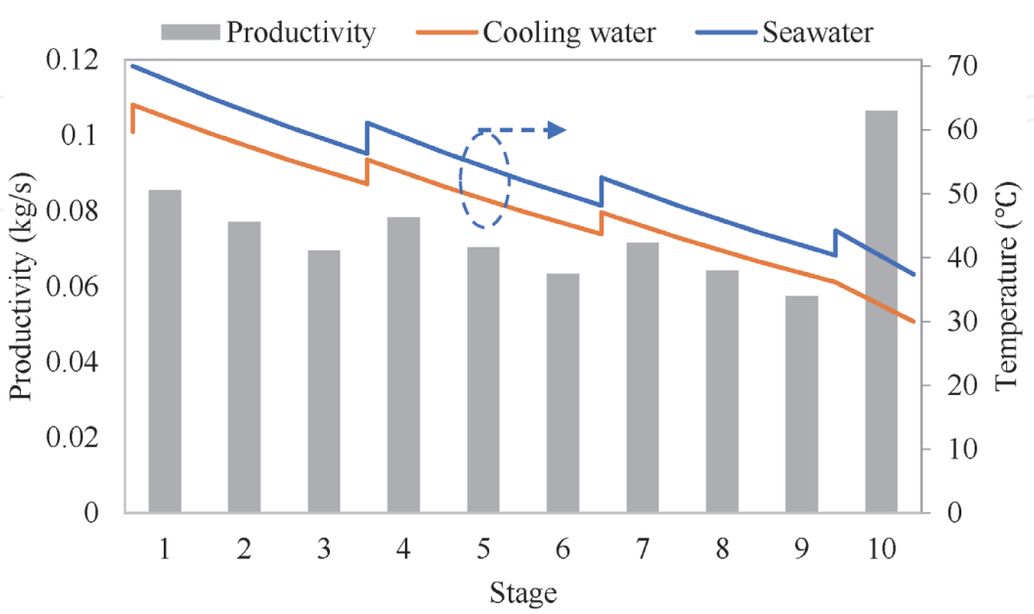


Figure 7.
Temperature and productivity profile for the modified system.

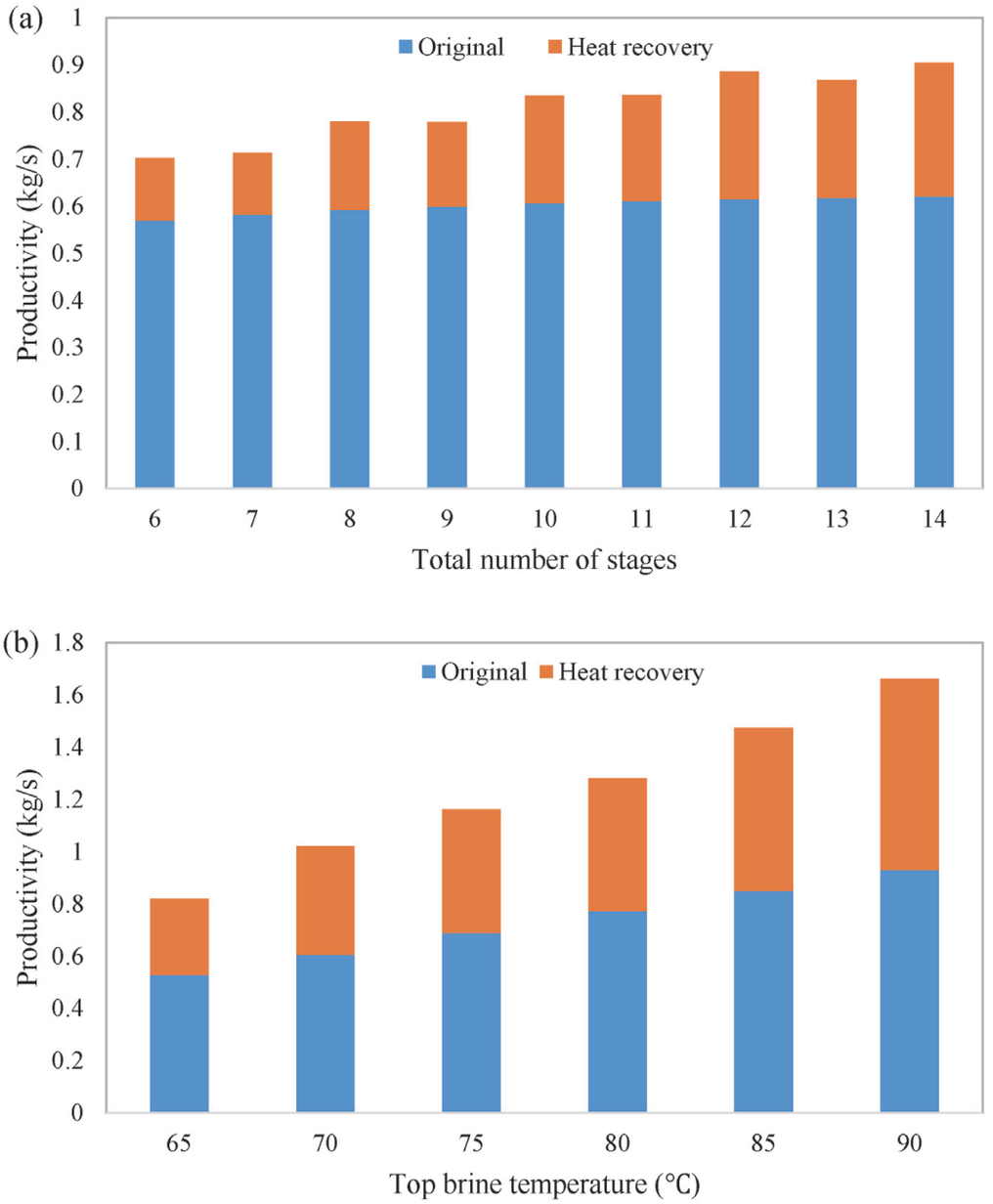


Figure 8.
Productivity improvement for the modified system under different (a) numbers of operating stages and (b) top brine temperatures.

indicating more heat extraction. Since the remaining energy cannot be further used, such change will not lead to additional energy cost and is acceptable. **Figure 9** compares the level of energy utilization before and after the heat internal heat recovery is conducted. It is clearly shown in **Figure 9** that the level of energy utilization is promoted by more than twice under all of the operating temperatures considered.

To evaluate the economic viability of the modified system, the costs for the system components are calculated for the original and the modified systems, as shown in **Table 2**. The calculation considers a 10-stage system operating at a *TBT* of 70°C. Compared with the original system, additional heat exchangers are added for internal heat recovery. Also, the pumping capacity will be higher to overcome the pressure drop across the newly added heat exchangers. As a result, the initial plant cost is increased by 9%. With an overall production increase of 23%, the unit cost for desalinated water will be lowered by 11%, making the modified configuration economic viable.

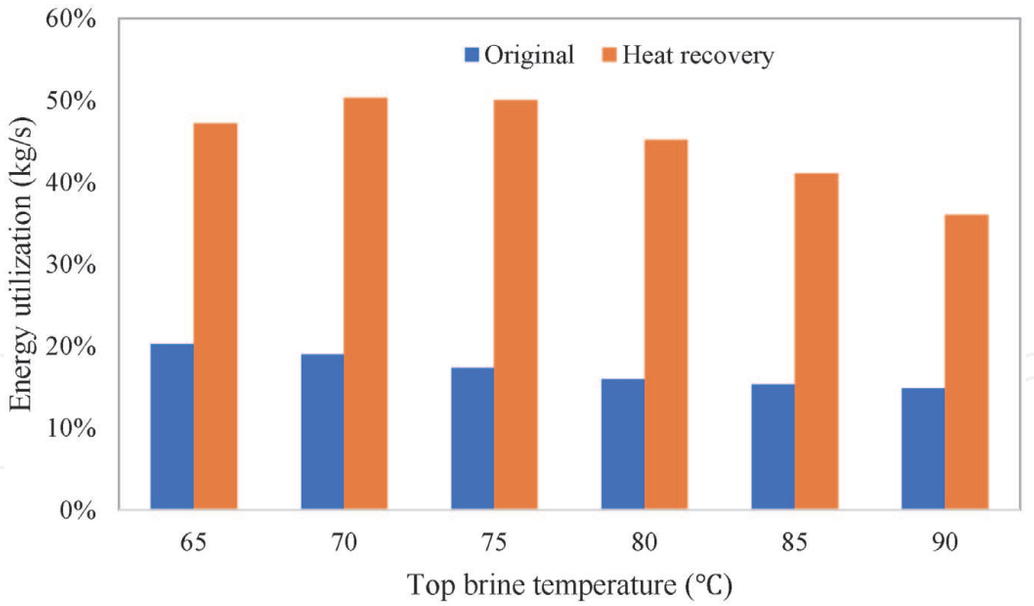


Figure 9.
Comparison of energy utilization for the heat source before and after heat recovery.

Component	Original	Modified	Difference
Evaporators ($)	32,785	32,785	–
Condensers ($)	9900	9900	–
Pumps ($)	3553	3560	0.2%
Heat exchangers ($)	23,574	30,065	28%
Total cost ($)	69,811	76,309	9%
Productivity (kg/s)	0.61	0.74	23%
Unit cost ($/(kgs^{-1}))	115,295	102,621	-11%

Table 2.
Costs of system components before and after modification.

5. Conclusions

A spray-assisted low-temperature desalination system-driven source has been proposed and evaluated. The system employs hollow chambers for evaporation and condensation, which not only promotes heat and mass transfer but also reduces the plant cost. The system is analyzed, employing sensible heat as the energy source. The key findings of this study are summarized as follows:

1. Both the water temperatures and the productivity decrease monotonously along the stages in the original SLTD configuration. A higher cooling water outlet temperature results in a high level of heat recovery and reduces the heat input requirement.

2. The freshwater productivity is proportional to the top brine temperature due to more heat source available. More numbers of operating stages also boost productivity because of the promoted heat utilization. However, less than 20%

of enthalpy is extracted and utilized from the heat source, and the majority is left unused.

3. By using the cooling water to heat up the seawater in the intermediate stages, the temperature gradient in each stage becomes greater, and the productivity is promoted under different design and operating conditions. The level of energy utilization is also increased by more than twice.

4. By adding extra heat exchangers for internal heat recovery, the initial plant cost is increased by 9%, while the productivity is boosted by 23%. As a result, the unit freshwater cost is expected to decrease by 11%.

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Conflict of interest

The authors declare no conflict of interest.

Nomenclature

BPE	boiling point elevation, °C
c_p	specific heat, J/kg
$\dot{D}$	production rate, kg/s
h_{fg}	latent heat of evaporation/condensation, J/kg
$\dot{m}$	mass flowrate, kg/s
N	total number of stages
NR	stage difference in heat recovery
Q	heat flux, W
X	salinity, kg/kg
T	temperature, °C
TBT	top brine temperature, °C

Greek letters

Δ	difference
θ	evaporation/condensation completion level

Subscripts

c	condenser/condensation
cl	cooling water
cv	vapor phase in condenser
cw	cooling water

e	evaporator
el	seawater in evaporator
ev	vapor in evaporator
h	heat source/hot water
HR	heat recovery
i	stage number
in	inlet
l	liquid phase
loss	temperature drop across the demister
r	heat recovery in *HEX1*
sw	feed seawater
v	vapor

Author details

Qian Chen[1*], Muhammad Burhan[1], Muhammad Wakil Shahzard[2], Raid Alrowais[1], Doskhan Ybyraiymkul[1], Faheem Hassan Akhtar[1], Yong Li[3] and Kim Choon Ng[1]

1 Water Desalination and Reuse Center, King Abdullah University of Science and Technology, Thuwal, Saudi Arabia

2 Northumbria University, Newcastle upon Tyne, United Kingdom

3 Institute of Refrigeration and Cryogenics, Shanghai Jiao Tong University, Shanghai, China

*Address all correspondence to: chen_qian@u.nus.edu

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