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### Performance analysis of combined humidified gas turbine power generation and multi-effect thermal vapor compression desalination systems — Part 1: The desalination unit and its combination with a steam-injected gas turbine power system

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### Abstract

Humidified gas turbines (HGT) have been identified as a promising way of producing power. The use of the steaminjected gas turbine (STIG) HGT cycle in a combined power and water desalination system was analyzed using energy and exergy performance criteria. A brief description and rationale of the background of HGT cycles and dual-purpose power and water systems is given. A thermal desalination unit was modeled and analyzed, and the results led to the selection of a multi-effect thermal vapor compression (METVC) unit for producing fresh water from seawater for both general use and humidification; then the performance of a STIG-based combined system was investigated. The analysis performed improved the understanding of the combined STIG power and water desalination process and of ways to improve and optimize it. Some specific conclusions are that: (1) a METVC desalination system is preferred to a multieffect evaporation one when the pressure of the motive steam is high enough,  $>\sim$ 3 bar, to run a steam jet ejector; (2) the steam injection rate in the STIG cycle has a strong effect on water and power production, offering good flexibility for design and operation; (3) higher pressure ratios and higher steam injection rates in the STIG cycle increase power generation, but decrease water production gain can be obtained by recovering the stack gas energy. The results indicate that such dual-purpose systems have good synergy, not only in fuel utilization, but also in operation and design flexibility.

*Keywords:* Integrated power and water system; Humidified gas turbine; Multi-effect thermal vapor compression desalination; Energy and exergy analyses

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### 1. Introduction

Humidified gas turbines (HGT) that use gaswater mixtures as the working fluid have been identified as a promising way to generate power, and their incorporation into a dual-purpose system producing both power and fresh water by desalination was investigated, especially regarding some synergies between the power generation and water desalination processes. Compared with combined cycles, the main features of HGT include high efficiency, high specific power output, reduced specific investment cost, reduced NO<sub>x</sub> emission and improved part-load performance [1,2]. There are several configurations of HGT cycles, with the steam-injected gas turbine (STIG) cycle, humid air turbine (HAT) or evaporative gas turbine (EvGT) cycle being the most representative.

In a STIG plant, steam is produced in a heat recovery steam generator (HRSG) using the exhaust heat of the gas turbine stack, and is then injected into the gas turbine combustion chamber. In a HAT or EvGT cycle, compressed air from the compressor is humidified with hot water in a humidifier that recovers the low-grade heat in the system, and is then heated by the exhaust gas from the gas turbine before entering the combustor. Different from HRSG in which steam is produced at a certain saturation temperature, the humidifier is an air-water direct-contact component in which water evaporates nonisothermally. With the potential of reaching an efficiency over 60%, the HAT and EvGT cycles have been considered as strong future competitors to the combined cycle.

Large water consumption is a major disadvantage of HGT, which restricts the use of the plant, especially in water-short areas. A LM5000 STIG<sup>™</sup> plant commercialized by General Electric, for instance, consumes about 29 t/d water per MW power output (1450 t/d water with a power output of 50.7 MW) when running under a full STIG pattern [3]. Two categories of solutions are proposed to solve the problem: one is to recover the steam in the flue gas for reuse, and another is to produce useful water by water desalination.

Introducing a condenser to recover the water vapor in the flue gas and reuse after treatment has been widely studied [3–6]. Owing to the combustion reaction of hydrogen and oxygen in the combustor, the flue gas contains more water than consumed. For instance, for each kg of air entering the compressor and 0.15 kg steam injected into the combustor, a CH<sub>4</sub>-fueled STIG cycle with a pressure ratio of 10 and firing temperature of 1300 °C contains 0.2176 kg water vapor in the flue gas. Both theoretical and experimental results showed that self-sufficient water recovery was possible [3–6]; as a further consideration, the heat gained in the process by the condenser coolant could be used in district heating [7].

As to using desalination for producing the water for humidification, Cerri [8] proposed a multi-stage flash (MSF) unit driven by exhaust gas heat of turbines for producing demineralized water from seawater for a STIG cycle; a small reverse osmosis (RO) unit was suggested [9] to produce water for steam injection, and the calculations predicted a minor effect of the addition of this system on the final operating cost of the power plant. The benefit of the integration of HGT with desalination is not limited to production of water for the cycle only. Just as in other power plants, the existence of the lowtemperature heat in HGT cycles provides a favorable condition for power and water cogeneration, and the second-law based energy utilization model of cogeneration systems indicates an advantageous performance over single-purpose units [10].

In some water-short areas, large quantities of low-grade thermal energy, of which the temperature is usually lower than 130°C, are needed to run thermal desalination units for producing fresh water from saline water. It is a great waste of exergy to provide such low-grade heat for desalination by burning fuel in a boiler. Combining power plants with desalination units produces a great synergy between power generation and water production. Such combined power and water production systems are usually called dualpurpose plants [11–13]. Most of the dual-purpose systems operating in the world are the combination of steam turbines with thermal desalination units; but in recent years, there has been an obvious interest in moving to gas turbines or combined cycle-based systems [14,15] owing to their increasing installed capacity.

The synergy of the combination of power and water production has significant energy, economy and environment benefits. For example, coupling MSF with a steam turbine plant showed a 44.4% energy saving of water production, from 40 kWh/m<sup>3</sup> equivalent work for a water-only system to 22.3 kWh/m<sup>3</sup> for water–power cogeneration [16], and a 44.7% water cost reduction from \$2.66/m<sup>3</sup> to \$1.47/m<sup>3</sup> [17]. A life-cycle assessment showed that the environmental load of thermal desalination technologies was reduced about 75% when operating in a hybrid plant based on a combined cycle [18].

A combination of HGT (STIG) and desalination was studied [19], based on a single-shaft gas turbine of 38.3 MWe, in which 10 kg/s steam produced in the HRSG was injected into the combustor, and the balance was used to run two steam jet ejectors and then two multi-effect distillation desalination units. A direct-type gas– seawater heat exchanger was introduced to recover further the exhaust heat of the flue.

The objective of this two-part paper was to study the energy, exergy, and water production performance of integrated power and water systems that use HGT as the prime mover. To understand the performance and parameter selection of water production, a thermal desalination unit was modeled and analyzed, and a combined system based on a STIG plant was investigated. In a separate paper, Part 2, an EvGT-based combined system was studied, and the results discussed to clarify further the performance of the two combined systems.

# 2. Multi-effect thermal vapor compression (METVC) desalination

### 2.1. Desalination processes

MSF, multi-effect evaporation (MEE), thermal vapor compression (TVC) and RO are four commonly used desalination processes. The first three are heat-driven, but also consume pumping work. Some of the advantages of thermal desalination are its lower sensitivity to the salinity and quality of the feedwater, and its ability to produce much higher quality distillate which can also be used for applications requiring water purer than needed for drinking, such as for boiler feed [19]. RO is driven only by mechanical work. Compared with current commercial thermal desalination, RO is much more energy-efficient, more compact, and more flexible in design and operation because of its modular structure and simper and quicker start-up/shut-down characteristics [13,16]. RO has been increasing its market share in recent years, yet thermal desalination still dominates the seawater desalination market. Being driven by low-temperature heat, thermal desalination is suitable to be combined with power generation or other industrial processes to improve the total energy efficiency.

MEE and METVC desalination units with a top brine temperature (TBT) lower than 70°C have attracted attention in recent years [20]. Compared with the most widely used MSF desalination, MEE has the advantages of lower corrosion and scaling rates, lower capital cost, longer operation life and less pumping power consumption [21]. When moderate-pressure (around 3 bar or higher [16]) steam is available, it can be used effectively for entraining and compressing the vapor produced in the last effect of



Fig. 1. Schematic diagram of a four-effect TVC system. ECON, end condenser; EVA1–EVA4, evaporators; FLA1–FLA4, flashing boxes; H1–H3, preheaters; SJE, steam jet ejector; WST, water storage tank.

the MEE plant by using a steam jet ejector. Compared with a stand-alone MEE plant, such a METVC arrangement needs less cooling water, and thereby lower pumping power and pretreatment costs [22].

Fig. 1 schematically illustrates a four-effect (EVA1-EVA4 with associated FLA1-FLA4) TVC unit with seawater preheaters (H1, H2, H3), which is apparently the combination of a steam jet ejector and a conventional MEE unit. Run by the motive steam (1), the steam jet ejector (SJE) entrains and compresses part of the water vapor (2) produced in the last effect (EVA4). The steam (3), which is called heating steam, leaves the SJE and condenses in EVA1 providing energy for seawater evaporation. Part of the condensate (5) returns to the boiler or HRSG, and the remaining part (6) is introduced into the associated flashing box (FLA1) where a small amount of vapor (7) flashes off because of a pressure drop. The vapor (8) evaporated from the seawater in EVA1 passes through the preheater (H1) to preheat the feed seawater (9), and is then routed into the second effect (EVA2) together with the flashing vapor (7) from FLA1, serving as the heat source in EVA2. The balanced brine (12) from EVA1 flows into the second effect (EVA2) and produces vapor by flashing. This process is repeated for all effects until the last one. Part of the vapor (2) formed in the last effect is entrained by the steam jet ejector, and the remainder (13) is introduced into the end condenser (ECON) where it releases its latent heat by heating seawater (15). Part of the heated seawater (16) is used as the feed of the METVC unit, and the balance (17) is rejected back to the sea.

### 2.2. Performance criteria

To understand the water production and parameter selection of the METVC unit used in this paper, the performance of the METVC was studied first and compared with the performance of a MEE unit using the following criteria.

1. Performance ratio (PR) — the ratio between the mass of the produced fresh water  $m_w$  to that of the consumed motive steam  $m_m$ :

$$PR = m_w / m_m \tag{1}$$

2. Specific heat transfer area (a) — the heat transfer area needed to produce 1 kg fresh water:

$$a = A/m_{w} \tag{2}$$

where A is the total heat transfer area of the desalination unit, composed of the area of the effects  $A_{ef}$  and the condensation area of the end condenser  $A_{econ}$ .

3. Specific exergy consumption  $(e_c)$  — the exergy consumed for producing 1 kg fresh water:

$$e_{c} = \frac{m_{m}(e_{m} - e_{hc})}{m_{w}}$$

$$= \frac{m_{m}[(h_{m} - h_{hc}) - T_{o}(s_{m} - s_{hc})]}{m_{w}}$$
(3)

where  $e_m$  is the specific exergy of the motive steam, and  $e_{hc}$  that of the condensate (stream 5 in Fig. 1) of the motive steam flowing out of the desalination unit;  $T_0$  is the temperature of the surroundings.

4. Exergy efficiency of the desalination unit  $(\varepsilon_D)$ :

$$\varepsilon_D = \frac{m_w w_{\min}}{m_m (e_m - e_{hc}) + W_P} \tag{4}$$

where  $W_P$  is the pumping work consumed by desalination, and  $w_{\min}$  the minimum work needed in a reversible separation process for producing 1 kg of fresh water. The calculation of the minimum work is given in Appendix A.

5. Exergy efficiency ( $\varepsilon_E$ ) and exergy loss rate

 $(\chi_E)$  of the steam jet ejector — The steam jet ejector is the basic component in the METVC unit that distinguishes METVC from MEE. Two indices are used to evaluate the performance of the SJE, its exergy efficiency  $\varepsilon_E$  and exergy loss rate  $\chi_E$ :

$$\boldsymbol{\varepsilon}_{E} = \frac{\boldsymbol{m}_{en} \left[ \left( \boldsymbol{h}_{h} - \boldsymbol{h}_{en} \right) - \boldsymbol{T}_{o} \left( \boldsymbol{s}_{h} - \boldsymbol{s}_{en} \right) \right]}{\boldsymbol{m}_{m} \left[ \left( \boldsymbol{h}_{m} - \boldsymbol{h}_{h} \right) - \boldsymbol{T}_{o} \left( \boldsymbol{s}_{m} - \boldsymbol{s}_{h} \right) \right]} \tag{5}$$

$$\chi_{E} = \frac{T_{0} (m_{h} s_{h} - m_{m} s_{m} - m_{en} s_{en})}{m_{m} (e_{m} - e_{hc})}$$

$$= \frac{T_{0} [m_{m} (s_{h} - s_{m}) + m_{en} (s_{h} - s_{en})]}{m_{m} [(h_{m} - h_{hc}) - T_{0} (s_{m} - s_{hc})]}$$
(6)

The numerator and the denominator in Eq. (5) represent the exergy the entrained steam gains and the motive steam loses in the steam jet ejector process, respectively, and those in Eq. (6) represent the exergy destruction in the steam jet ejector and the exergy provided for the METVC unit.

### 2.3. Modeling and simulation

The modeling and simulation were made using the Engineering Equation Solver (EES) software [23]. The properties of seawater and brine are taken from Husain [11]. The heat transfer coefficients of evaporation and condensation, the boiling point elevation of brine, as well as the non-equilibrium allowance of flashing evaporation in the flashing box were taken from El-Dessouky and Hisham [24]. The performance of the steam jet ejector was taken from Power [25]. In the modeling and simulation, the distillate produced in each effect was considered to be salt free, and, in accordance with industrial practice, each evaporator had the same heat transfer area [24]. Table 1 shows the calculation conditions. More detailed information on MEE and METVC can be found in many publications (cf. [24,26]).

#### Table 1 Calculation conditions for MEE and METVC

Temp. of seawater $T_{sw}$ , °C Temp. of rejected cooling seawater	30 5° lower than
$T_{\text{cool}}, ^{\circ}\text{C}$	$T_{con}$
TBT, °C	69
Salinity of seawater $X_{sw}$ , ppm	36,000
Salinity of rejected brine $X_{br}$ , ppm	70,000
State of motive steam	Saturated
Condensation temp. of heating steam $T_{hc}$ , °C	72
Pressure of heating steam $p_h$ , MPa	0.03397 <sup>a</sup>
Min. condensation temp. $T_{con}$ in end condenser, °C	40

<sup>a</sup>Saturated temperature is 72°.

Using the model developed by the authors, PR, specific area of effects  $a_{ef}$ , which is the ratio of  $A_{ef}$  and  $m_{w}$ , and  $e_{c}$  of four cases are calculated under the same conditions as those previously given [26–28], and the results are shown in Table 2.  $a_{ef}$  accounts for the main part of a, in a six-effect METVC unit studied below; for instance, the typical value of  $a_{ef}/a$  is around 94%. It was observed that the model predictions compare well with the data in the literature (Table 2). and the relative differences are within 3%, except  $a_{ef}$ . The larger difference of  $a_{ef}$  is due to the different correlation of heat-transfer coefficients and the calculation model used in this paper and the literature [26]. The heat transfer coefficient correlations used are shown in Appendix B; they were reported by their authors to have been validated through comparison against other correlations and available experiments [24].

A parametric analysis was carried out to investigate three important factors: compression ratio (*CR*), pressure of the motive steam ( $p_m$ ), and seawater preheating on the performance of the METVC unit.

### 2.4. Influence of the compression ratio (CR)

The compression ratio *CR*, which is the pressure ratio of the heating steam entering the first

effect and the entrained steam from the last effect

$$CR = p_h / p_{en} \tag{7}$$

and the expansion ratio *ER*, which is the pressure ratio of the motive and entrained steam,

$$ER = p_m / p_{en} \tag{8}$$

are two parameters which determine the performance of the steam jet ejector and then of the METVC unit, under the calculation conditions shown in Table 1. When the pressure of the heating steam  $p_h$  is specified (0.034 MPa in this paper), a certain *CR* corresponds to a certain range of operating temperatures of the MEE in the METVC unit, since a change of *CR* means that there is a change of the pressure and temperature of the entrained steam, therefore, the operation pressure and temperature of the last effect. If  $p_m$  is also given, *ER* and then the performance of the METVC unit can be calculated.

Fig. 2 shows the influence of *CR* on *PR* and *a* when  $p_m$  is fixed. Fig. 3 shows the variations of  $e_c$ ,  $\varepsilon_E$  and  $\chi_E$  with *CR*, and Fig. 4 shows the variation of  $\varepsilon_D$  of METVC. For a specified number of effects *n*, increasing *CR* causes the decrease of the parameters *a*, *PR* and  $\varepsilon_D$ , and an increase of  $e_c$ . An increase in *CR* implies a lower



Fig. 2. Dependence of *PR* and *a* of METVC on *CR* and *n*.

Calculation condition		Case 1 [26]	Case 2 [26]	Case 3 [27]	Case 4 [28]	
n		6	6	4	4	
$T_{hc}$ °C		65	65	62	62.7	
$T_{hr}^{nc}$ °C		42.8	42.8	46.8	48.4	
TBT, °C		61.8	61.8	58.8	_	
$P_m$ , MPa		2.5	2.5	2.5	2.4	
$T_{\rm cool}, ^{\circ}{\rm C}$		40	40	40	44	
$T_{sw}^{\circ}$ °C		30	30	_	33	
$\Delta T_{nh}$ , °C		3.8	No preheaters	No preheaters	No preheaters	
$m_m/m_{en}$		1.36	1.36	0.86	0.8	
$X_{sw}$ , ppm		$47,800^{a}$	$47,800^{a}$	47,800 <sup>a</sup>	47,800	
$X_{br}$ , ppm		71,500 <sup>a</sup>	71,500 <sup>a</sup>	71,500 <sup>a</sup>	71,500	
PR	Reference	10.05 [26]	8.87 [26]	7.65 [27]	8.61 [28]	
	Model	10.11	9.10	7.73	8.49	
$a_{ab}$ m <sup>2</sup> /(kg/s)	Reference	326.2 [26]	341 [26]	_	_	
	Model	353.4	354.8	347	363.7	
e . kJ/kg	Reference	87.91 [26]	99.26 [26]	_	_	
,8	Model	89.44	99.3	117.09	106.05	

 Table 2

 Comparison of model predictions against available data for METVC units

<sup>a</sup>Assumed values based on the context of the references.



Fig. 3. Dependence of  $e_c$ ,  $\varepsilon_E$  and  $\chi_E$  on *CR* and *n*.



Fig. 4. Dependence of  $\varepsilon_D$  of METVC on *CR* and *n*.

 $p_{en}$  (Eq. 7) and thus a lower  $T_{en}$ , leading to an increased temperature difference between the first and the last effect and thus an increased temperature difference across each effect. This obviously results in a decrease of the needed heat transfer area. At the same time, the increased

temperature difference increases the heat transfer irreversibility in each effect, causing a higher  $e_c$ and lower *PR* and  $\varepsilon_D$ . For the same reason, for a specified *CR*, a higher number of effects *n* results in higher *PR*, *a* and  $\varepsilon_D$ , but a lower  $e_c$ . It is noteworthy that the exergy efficiency of the METVC unit is very low, <3.5%, as shown in Fig. 4, indicating great potential for performance improvement of the desalination process.

As well know, the steam jet ejector is not an exergy-efficient component. From Fig. 3, with a motive pressure of 0.5 MPa and *CR* between 1.8 and 4.5,  $\varepsilon_E$  is only 16–25%, and about 45% of the exergy provided for desalination is lost in the SJE.

### 2.5. Influence of the motive steam pressure $(p_m)$

As analyzed in Section 2.4, specification of the CR and  $p_h$  determines  $p_{en}$ , and thus the performance of the steam jet ejector and of the entire METVC unit can be determined given  $p_m$ . Fig. 5 illustrates the effect of  $p_m$  on the performance of the METVC unit. It is clear that PR increases at a diminishing rate with the increase of  $p_m$ . When the steam jet ejector is run by higher-pressure motive steam, it draws more vapor from the last effect, thus increasing the amount of the heating steam, consequently the amount of product water, resulting in an increased PR. The increase rate of PR diminishes because (1) the increase rate of the exergy of the motive steam provided for desalination diminishes as  $p_m$  increases, as determined by the properties of the saturated steam; and (2) the increased exergy destruction in the steam jet ejector and the evaporators at higher  $p_m$  [26] cause the specific exergy consumption  $e_c$  to increase (Fig. 5).

Calculations also show that  $p_m$  has only a slight influence on *a* (Fig. 5). The reason is that the higher enthalpy of the motive steam at the higher  $p_m$  results in an increased heating steam temperature  $T_h$ , which increases the temperature difference for the sensible heat transfer in the first effect, and consequently slightly reduces  $a_{ef}$ . At the same time, higher-pressure motive steam increases the amount of vapor entrained from the last effect, and thus reduces the amount of vapor



Fig. 5. Impact of  $p_m$  on *PR*, *a*, and  $e_c$  of METVC.



Fig. 6. Performance of the MEE unit.

that needs to be condensed in the end condenser, leading to a slightly reduced  $a_{econ}$ .

Fig. 6 shows the performance of a conventional MEE unit operated by saturated steam at different conditions: 72 °C (0.034 MPa), 151.8 °C (0.5 MPa) and 242.6 °C (3.5 MPa). Only one area curve is shown due to the small influence of  $p_m$ ( $T_m$ ) on *a*. Compared with the data in Figs. 2 and 5, one can see that, although the exergy efficiency of the steam jet ejector  $\varepsilon_E$  is very low (Fig. 3), the METVC unit has a significant advantage over MEE running with the same heat source when  $p_m$ is high enough to run the steam jet ejector. For



Fig. 7. Exergy efficiency of METVC and MEE.

instance, a seven-effect MEE unit run by saturated steam at 3.5 MPa has a PR of 5.5 for a specific area of 310 m<sup>2</sup>/kg; however, a six-effect METVC unit with a CR of 3.5 run by the same heat source has a PR of 8.6 for a specific area of  $301 \text{ m}^2/\text{kg}$ . Different from the MEE unit in which no work is produced, mechanical work is performed in the METVC unit by the motive steam in the steam jet ejector process by compressing the entrained steam. This is more efficient exergy-wise because the steam is available at a pressure significantly higher than needed for supplying heat to the MEE; instead of wastefully throttling it to the heat supply pressure, the ejector is used for vapor compression in the METVC process.

Fig. 7 shows the exergy efficiency,  $\varepsilon_D$ , of the six-effect METVC and seven-effect MEE units mentioned above for different  $p_m$ . It is revealed that a higher  $p_m$  leads to a lower  $\varepsilon_D$  for both METVC and MEE units due to the increased exergy consumption of water production  $e_c$  (Figs. 5 and 6); run by the same heat source, METVC has a higher  $\varepsilon_D$  than MEE due to the work process in the steam jet ejector in METVC.

Table 1 shows the calculation conditions used

for METVC and MEE, including the TBT of 69°C. Assuming that the minimal heat transfer temperature difference between the heating steam and seawater is 3°C, the temperature of the heating steam is 72°C, for which the saturation pressure is 0.034 MPa. The irreversibility in the steam jet ejector process and the heat transfer process between heating steam and seawater is reduced as the temperature of the steam provided for desalination reaches closer to 72°C and its pressure to 0.034 MPa, thus resulting in higher  $\varepsilon_{D}$ . Thus, under the same calculation conditions, the MEE unit run at 72°C (0.034 MPa) saturated steam has a higher  $\varepsilon_D$  than the METVC or MEE units run by higher-temperature/pressure steam, although the latter has higher PR; this higher PR is at the expense of higher  $e_c$ . This can be seen clearly from Figs. 5-7. For instance, the seveneffect MEE run at 72°C (0.034 MPa) saturated steam has a  $\varepsilon_D$  of 3.8%, *PR* of 5.5 and  $e_c$  of 51 kJ/(kg distillate), while the six-effect METVC run at motive steam of 240.6°C (3.5 MPa) has a  $\varepsilon_D$  of 1.9%, PR of 8.6 and  $e_c$  of 109 kJ/(kg distillate). Consequently, when the pressure/ temperature of the steam provided for desalination is higher than 0.034 MPa/72°C, a more exergy-efficient way of utilizing it is to lower its temperature/pressure to 72°C/0.034MPa by producing work, and only then use as the heat source of MEE.

### 2.6. Influence of seawater preheating

The influence of feed seawater preheating in the preheaters (H1, H2, H3 in Fig. 2) is shown in Fig. 8. High preheating  $\Delta T_{ph}$ , which is the temperature rise of the feed seawater in each preheater, results in a high *PR* with slightly increased specific area *a* compared with a METVC unit without preheaters. In MEE or METVC units, the feed seawater is first heated to boiling temperature (the seawater heating process), and is then boiled to produce vapor (the



Fig. 8. Impact of preheating on PR and a.

seawater evaporation process), by absorbing the condensation heat of the vapor from the former effect (heating steam is used for the first effect). Seawater preheating helps reduce the temperature difference between the heated seawater and the condensing vapor in each effect except the last effect, leading to a more exergy-efficient heat transfer process, which increases the water production but also increases the specific area needed in the seawater heating process. Since the seawater evaporation process is the main heat transfer process in the effects, and the evaporation area is the dominant factor of the total area, the increased area in the seawater heating process has a minor influence on a.

# 2.7. Parameter selection and the performance of the METVC unit

In a STIG-based combined system, the pressure of the saturated steam for desalination is the same as that of the injected steam, 0.5 MPa higher than the operation pressure of the combustor in this paper. In an EvGT-based system (to be discussed in Part 2), the steam pressure is higher than 0.3 MPa. Consequently, an METVC unit is preferred to MEE, both in STIG- and EvGT-based systems, based on the analysis in Section 2.5.



Fig. 9. PR of a six-effect TVC unit.

In an MEYVC unit,  $p_m$ , which is determined by the heat source (boiler, power plant or industrial process) has a great influence on PR, especially when  $p_m < 1.5$  MPa (Fig. 5), but little influence on the specific area a, while the compression ratio CR and the number of effects nhave significant influence on both PR and a. The specific area a, important in influencing equipment cost, is determined therefore mainly by CR and n under the calculation conditions shown in Table 1. In lieu of detailed economic and technological optimization studies needed for parameter selection, we have chosen here, after referring to the information from commercial METVC units previously described [20], is a sixeffect TVC unit with a CR of 3.5, which has a specific area a of about 301 m<sup>2</sup>/(kg/s), changing only slightly with  $p_m$  (Fig. 5). The *PR* of the unit with and without preheaters is shown in Fig. 9.

# **3.** Calculation conditions and performance criteria for the analysis of combined power and desalination systems

### 3.1. Calculation conditions and assumptions

The main calculation conditions and assumptions are summarized in Table 3. The commercial

Table 3				
Main conditions	and assumptions	for the modeling	of HGT-based s	systems

Inlet air conditions	25°C, 1 atm air with 100% relative humidity
Dead state of exergy analysis	25°C, 1a tm air with 100% relative humidity
Fuel	CH <sub>4</sub>
Isentropic efficiency of compressor	0.88
Isentropic efficiency of turbine	0.9
Pressure drop of compressor inlet	1% of inlet pressure
Pressure drop in combustor	3% of inlet pressure
Pressure drop in stack	1% of inlet pressure
Pressure drop in HRSG	3% of inlet pressure
Pressure drop in gas-gas heat exchanger	3% of inlet pressure
Pressure drop in water-gas heat exchanger	2% of inlet pressure
Pressure drop in direct-contact gas-water heat exchanger	2% of inlet pressure
Combustion chamber efficiency	0.99
Minimum pinch point temperature difference of HRSG, °C	15
Minimum temperature difference at hot side of HRSG, °C	50
Minimum exhaust temperature leaving the HRSG, °C	140
Minimum driving force of enthalpy difference in humidifier, kJ/kg	25
Temperature effectiveness of gas-gas heat exchanger	0.82
Pressure of steam injected in STIG cycle	0.5 MPa higher than the combustor pressure

Aspen Plus code [29] was used to carry out the simulation of the HGT cycle.

This study focused on the power and water production of the combined systems. A detailed performance analysis and comparison of the power-only cycles is available elsewhere [30–32].

The computerized models of the HGT-based combined systems were validated by (1) examining the physical sensibility of the calculation results for each component and the entire cycle, (2) allowing a relative error tolerance of only  $10^{-4}$  in the Aspen Plus program, (3) comparison with available results (Table 2), (4) examining the relative errors in the mass and energy balance of the computerized model of the METVC unit where the former was found to be  $<10^{-5}$  and the latter  $<10^{-13}$ . It would have been good to validate the results by using experimental data, but the only case we found was a conceptual design of a STIG–METVC system [19] with insufficient information to run our model.

### 3.2. Performance criteria

The performance criteria used to analyze the energy and exergy utilization and the water and power production of the combined systems, which are very helpful in helping understand the process and ways by which it may be improved, are described below.

1. Thermal efficiency  $\eta_t$  and exergy efficiency  $\varepsilon_e$  of the cycles:

$$\eta_t = \frac{W}{Q_f} = \frac{W}{m_f \cdot q_f} \tag{9}$$

$$\boldsymbol{\varepsilon}_{\boldsymbol{e}} = \frac{W}{E_f} = \frac{W}{\boldsymbol{m}_f \cdot \boldsymbol{e}_f} \tag{10}$$

where W is the net power output from the gas turbine plant,  $m_f$  is the mass of fuel input to the combustor, and  $q_f$  and  $e_f$  are the low heat value and the specific exergy of fuel, respectively.

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2. Heat loss ratio  $\chi_t$  and exergy loss ratio  $\chi_e$  of each component. The heat loss ratio  $\chi_t$  is the heat loss  $Q_t$  in each component as a fraction of the input fuel energy  $Q_{f^3}$  and exergy loss rate  $\chi_e$  the exergy loss  $E_t$  to the fuel exergy  $E_t$ :

$$\chi_t = \frac{Q_l}{Q_f} = \frac{Q_l}{m_f \cdot q_f} \tag{11}$$

$$\chi_e = \frac{E_l}{E_f} = \frac{E_l}{m_f \cdot e_f}$$
(12)

3. Heat recovery rate  $\xi_t$  and exergy recovery rate  $\xi_e$ . In the two combined systems studied, part of the exhaust gas energy is recovered by the injection steam in the HRSG in the STIG-based system or by the humid air in the regenerator in the EvGT-based system, and is then returned to the combustor; part of the energy is recovered by the motive steam used as the heat source for desalination. Obviously, improved heat recovery improves the process efficiency, and the criteria used are  $\xi_t$ , the heat recovered  $Q_r$  as a fraction of the input fuel energy  $Q_f$ , and  $\xi_e$  the exergy recovered  $E_r$  as a fraction of the input fuel exergy  $E_f$ :

$$\xi_t = \frac{Q_r}{Q_f} = \frac{Q_r}{m_f \cdot q_f}$$
(13)

$$\xi_e = \frac{E_r}{E_f} = \frac{E_r}{m_f \cdot e_f} \tag{14}$$

4. Power-to-water ratio  $R_{pw}$ . Power-to-water ratio  $R_{pw}$  is the ratio of the generated power w to the mass of the produced water  $m_w$ , which is a key factor in determining the performance of the combined power and water system.

$$R_{pw} = w/m_w \tag{15}$$

### Table 4 Reference system performance for normalization

Cycle pattern	Simple gas turbine cycle
Desalination unit	Six-effect TVC, $CR = 3.5$ , $\Delta T_{ph} = 4^{\circ}C$
Pressure ratio, $\beta$	10
Firing temperature (TIT), °C	1300
Pressure of motive steam for desalination, $p_m$ , MPa	1.5
Gross power output, $w_{0,\text{gross}}$ kJ/(kg fuel)	16,390
Net power output <sup>a</sup> , $w_{0,net}$ kJ/(kg fuel)	15,692
Water production, $m_{w0}$ , kg/(kg fuel)	97

<sup>a</sup>By taking the pumping work for water production as 7.2 kJ/(kg distillate) [16].

MW/MIGD is often used as the unit of  $R_{pw}$ (1 MIGD = 52.662 kg/s).

5. Normalized power and water production. To exhibit more clearly the sensitivity of power and water production in the combined systems,  $m_w$  and w are normalized by the simple-cycle water production  $m_{w0}$  and power output  $w_0$  shown in Table 4. The flowsheet of the combined system based on the simple cycle is the same as that of the STIG-based system shown in Fig. 10, except that no steam is injected into the combustor in the simple cycle.

### 4. STIG-based power and water system

### 4.1. System configuration of STIG-based combined system

Fig. 10 schematically shows a STIG-based integrated power and water production system in which part of the saturated steam produced in the HRSG is used to operate a thermal desalination unit (TDC), and the balance is superheated and then injected into the combustor (CC) to enhance power output. A system configuration that uses



Fig. 10. STIG-based power and water combined system. C, compressor; CC, combustor; FC, fuel compressor; G, generator; HRSG, heat recovery steam generator; P, pump; T, turbine; TDU, thermal desalination unit.

the exhaust gas to heat seawater was not used to avoid potential corrosion and scaling problems caused by the higher temperature seawater that would be generated.

In a conventional STIG cycle, all the heat of the gas turbine exhaust is used to produce steam for injection, and the optimum performance can be obtained when both the temperature and the mass of the steam reach the maximum value allowed by the technical condition of the HRSG. If part of the energy is used to produce water, the efficiencies  $\eta_t$  and  $\varepsilon_e$  of the cycle, and thereby the power output, decrease. The power output is even lower when the pumping work consumed by the desalination unit is also considered. Water production is at the expense of the reduction of produced power, which is a common characteristic of integrated power and water systems.

The pressure ratio  $\beta$  of the compressor, the turbine inlet temperature (the "firing" tempera-

ture, TIT), and the steam injection rate  $x_j$  are the most important parameters influencing the performance of the STIG cycle and the combined system.

#### 4.2. Influence of steam injection rate $x_i$

In a STIG-based system at a specified  $\beta$  and TIT, the mass ratio of the injected steam to the compressed air,  $x_{i}$ ,

$$x_j = m_j / m_a \tag{16}$$

is the dominant factor influencing  $R_{pw}$  because it determines the energy and exergy distribution of the exhaust gas in the HRSG between the injection steam and motive steam, and thus the fuel energy and exergy distribution between the power and water production.

Fig. 11 shows the influence of  $x_j$  and Fig. 12 is an example of energy and exergy utilization under different  $x_j$ . Increasing  $x_j$  increases the amount of the thermal energy and exergy of the gas turbine exhaust gas that is used to produce steam for injection, thus reducing the amount available for desalination (Fig. 12). This results in higher power output and lower water production (Fig. 11). Increasing  $x_j$  obviously also increases the water consumption, thus reducing the net water production even further (Fig. 11). At a certain  $x_j$ , the net water production  $m_{w,net}$  is zero (Fig. 11), which is called "self-water production".

Self-water production is a way to solve the problem of injection water supply for a poweronly STIG plant. A saline water desalination system including a seawater pretreatment unit, desalination unit and brine feed and drain pipelines, etc, is required, and part of the steam, about 10% in the case shown in Fig. 11, produced in the HRSG is needed to serve as the heat source for this desalination. The premise of this course is that saline water is available. Adding a condenser after the HRSG is another way to provide water for injection, as previously discussed [3–6]. In the



Fig. 11. Normalized power and water production of the STIG-based system as a function of  $x_i$ .



Fig. 12. Exergy and energy utilization of the STIG-based combined system for different  $x_i$ ,  $\beta = 10$ ; TIT = 1300°C.

case shown in Fig. 11, the flue gas should be cooled to 53 °C to recover the injected water. A condenser, a coolant circulation and cooling system, and a condensate treatment unit are needed in this situation. The method of choice is determined by an economic analysis for the concrete condition of the power plant.



Fig. 13. Influence of  $\beta$  and TIT on  $\varepsilon_{e}$ ,  $\eta_{t}$ ,  $\xi_{e,D}$  and  $\xi_{t,D}$  in the STIG-based combined system. (a) Variations of  $\varepsilon_{e}$  and  $\xi_{e,D}$  with  $\beta$  and TIT; (b) Variations of  $\eta_{t}$  and  $\xi_{t,D}$  with  $\beta$  and TIT.

# 4.3. Influence of pressure ratio $\beta$ and tubine inlet temperature (TIT) of the cycle

Fig. 13 shows the exergy efficiency  $\varepsilon_e$  and thermal efficiency  $\eta_t$  of the STIG cycle, as well as the exergy recovery rate  $\xi_{e,D}$  and energy recovery rate  $\xi_{t,D}$  for desalination, for different  $\beta$  (from 10 to 30), TIT (1100°C and 1300°C) and  $x_i$  (0.05



Fig. 14. Normalized power and water production of the STIG-based combined system.



Fig. 15. Exergy and energy utilization of the STIG-based combined system for different  $\beta$ . TIT = 1300 °C;  $x_j = 0.05$ .

and 0.1). We can see the percentage of fuel energy/exergy converted into power and consumed by desalination. Although  $\xi_{e,D}$  and  $\xi_{t,D}$  only represent the thermal energy/exergy provided for desalination, they determine water production, as shown below, since that pumping work is only a



Fig. 16. Heat exchanger T-Q diagram of the HRSG for different TIT.

small fraction of the desalination energy consumption. Fig. 14 shows the normalized power and water production, and Fig. 15 the exergy and energy utilization for different  $\beta$ .

These figures reveal that  $\varepsilon_e$  and  $\eta_t$  have a similar trend, as the power output *w*, consistent with the definition of  $\varepsilon_e$  and  $\eta_t$  by Eqs. (9) and (10). Within the range of  $\beta$  studied, for a fixed  $x_j$ , a higher  $\beta$  increases  $\varepsilon_e$ ,  $\eta_t$  and *w*, as known [30]. Increasing TIT, as well known, also improves power production efficiency.

Figs.13 and 14 also show that  $\xi_{e,D}$  and  $\xi_{t,D}$  and the water production  $m_w$  have a similar trend, consistent with the definition of  $\xi_{e,D}$  and  $\xi_{t,D}$  by Eqs. (11) and (12). Opposite to the influence of  $\beta$ on power,  $m_w$  decreases with the increase of  $\beta$ because the temperature of the exhaust gas from the gas turbine is lower for higher  $\beta$ ; therefore, less energy and exergy can be recovered for desalination (Fig. 15).

A higher TIT is not only beneficial to the power generation in the STIG-based system but also to the water production rate (Fig. 14) because the increased turbine outlet temperature makes more energy and exergy available for desalination. It is noteworthy that  $\xi_{e,D}$ ,  $\xi_{t,D}$  and  $m_w$  drop more sharply with  $\beta$  at TIT = 1100°C than at TIT = 1300°C, which is determined by the heat transfer process in the HRSG. As shown in Fig. 16, when TIT = 1300°C and  $\beta$  = 30, the exhaust gas can be cooled to the minimal temperature we allowed, 140°C, but when TIT = 1100°C, the pinch point temperature  $\Delta T_p$  of the HRSG restricts the heat exchange process. To meet the minimum  $\Delta T_p$  we allowed, the exhaust temperature leaving the HRSG must in this case be increased to 167.7°C, causing a sharper decrease of  $\xi_e$  and  $\xi_p$  and thereby of  $m_w$ .

### 4.4. Energy recovery from the stack gas

Fig. 12 shows that the energy rejected to the environment rises significantly with the increase of  $x_j$  due to the increased steam fraction in the flue gas and the corresponding large latent heat of water. For example, the heat and exergy losses with the stack gas are 19.3% and 5.2%, respectively, when  $\beta = 10$ , TIT = 1300 °C and  $x_j = 0.05$ , and 34.9% and 7.1%, respectively, when  $x_j = 0.15$ .

It is difficult to use the heat of the rejected stack gas because of its low temperature. The gas contains water vapor, and thus, when cooled, loses first its sensible heat until the temperature drops to the water vapor condensation level, at which point the latent heat of condensation is released too. From Fig. 17, which shows the heat released in the process of cooling the stack gas from  $140^{\circ}C$  to  $40^{\circ}C$ , we can see that when  $x_i = 0.05$ , more than 60% of the total heat is released in the temperature range of 57.5°C to 40°C during the condensation process of the water vapor in the flue, and when  $x_i = 0.15$ , about 80% of the heat is released in the temperature range of 68.5°C to 40°C. It is hence impossible to use this heat to produce steam for water desalination.

There are three ways to use the stack gas heat. The first one is for heating service water, and at



Fig. 17. Heat release process of the stack gas leaving the HRSG.

the same time the water in the stack gas is recovered by condensation. The second is to preheat the seawater (part of stream 16 in Fig. 1) feed to the desalination unit to recover the heat. and at the same time recover the vapor in the stack gas. A water-gas indirect contact heat exchanger is needed in both cases, and if the recovered water is to be used, it should be treated to separate it from undesirable flue gas components. The third way is by using a directcontact gas-seawater heat exchanger to heat seawater for the desalination unit, but only if the flue gas does not contain species that would be harmful if present in the product water or that may impair plant performance. The second way is the most energy-profitable for the STIG-based combined system because heating the seawater can help increase the production of fresh water, and the reuse of the vapor in the stack gas results in higher net water production. The heat exchanger must be designed to withstand the presence of both corrosive fluids. The third way, in which the gas-seawater direct-contact heat exchanger is very similar to the long commercialized wet-gas desulfurization system, is the simplest and thus seems to be technically feasible [19]. This third



Fig. 18. Net water production of STIG-based combined system with and without the direct-contact heat exchanger for stack gas heat recovery.

way is chosen in this paper to recover the stack heat.

Part of the seawater from the end condenser is fed to the direct-contact heat exchanger, heated to  $63 \,^{\circ}$ C, and then, after mixing with the seawater from the end condenser according to the mass flow and temperature required by each effect, used as the feed of a six-effect METVC unit in which no preheaters are used because the feed seawater has been preheated by the stack gas. The net water production with and without stack heat recovery are shown in Fig. 18. The effect of heat recovery is significant, and the results indicate that more than a 16% water production gain can be obtained.

### **5.** Conclusions

The analysis performed improves the understanding of the combined STIG power and water desalination process and of ways to improve and optimize it. Some specific conclusions are that: (1) a METVC desalination system is preferred to a MEE one when the pressure of the motive steam is high enough,  $> \sim 3$  bar, to run a steam jet ejector, as in the two combined systems studied; (2) the steam injection rate in the STIG cycle has a strong effect on water and power production, offering good flexibility for design and operation; (3) higher pressure ratios and higher steam injection rates of the STIG cycle increase power generation, but decrease water production rates, and higher turbine inlet temperatures increase both power and water production; (4) a distinct water production gain can be obtained by recovering the stack gas energy.

### 6. Symbols

- *a* Specific heat transfer area,  $m^2/(kg/s)$
- A Heat transfer area,  $m^2$
- c Specific heat, kJ/( kg K)
- *CR* Compression ratio
- e Specific exergy, kJ/kg
- E Exergy, kJ
- *e<sub>c</sub>* Specific exergy consumption, kJ/(kg distillate)
- *ER* Expansion ratio
- g Specific free enthalpy, kJ/kg
- h Specific enthalpy, kJ/kg
- *m* Mass, kg
- M Molar mass, kg/kmol
- *n* Number of effects
- *p* Pressure, MPa
- *PR* Performance ratio
- q Low heat value, kJ/kg
- Q Heat energy, kJ
- R Gas constant, kJ/(kg K)
- $R_{pw}$  Power-to-water ratio, MW/MIGD
- s Specific entropy, kJ/(kgK)
- $S^0$  Absolute entropy at 25 °C and 1 atm, kJ/(kmol K)
- T Temperature, °C or K
- $T_c$  Condensation temperature, °C
- $T_e$  Evaporation temperature, °C
- TIT Turbine inlet temperature (firing temperature), °C
- TBT Top brine temperature, °C

- $U_{c}$ - Heat-transfer coefficient of condensation,  $kW/(m^2 \circ C)$ - Heat-transfer coefficient of evapora- $U_{\rho}$
- tion, kW/ (m<sup>2</sup>  $^{\circ}$ C)
- Specific work, kJ/kg W
- Power, kJ W
- Steam injection rate in STIG cycle  $X_i$
- Humidification rate in EvGT cycle  $X_h$
- Salinity of saline water, ppm X
- Mass fraction v
- Molar fraction z

### Greek

β		Pressure ratio of gas turbine cycle
$\mathbf{\epsilon}_{e}$		Exergy efficiency of thermal cycle,
		°⁄0
$\mathbf{\epsilon}_{E}$	—	Exergy efficiency of steam jet ejec-
		tor, %
$\mathbf{\epsilon}_D$		Exergy efficiency of thermal desali-
		nation unit, %
$\eta_t$		Thermal efficiency of thermal cycle,
		%

- Heat loss rate, % χt
- Exergy loss rate, % X
- Exergy loss rate of steam jet ejector,  $\chi_E$ %
- Heat recovery rate, %
- Exergy recovery rate, %
- $\xi_t$  $\xi_e$  $\Delta h_f^0$ — Formation of enthalpy at 25°C and 1 atm, kJ/ kmol
- $\Delta T_p$ — Pinch point temperature difference of HRSG, °C

 $\Delta T_{ph}$ - Temperature rise of seawater in preheater, °C

### **Subscripts**

a	— Air
br	<ul> <li>Rejected brine</li> </ul>
con	— Condensate at end condenser
cool	<ul> <li>Rejected cooling seawater</li> </ul>
D	— Desalination
Ε	<ul> <li>Steam jet ejector</li> </ul>

econ		End condenser
ef		Effects in METVC or MEE unit
en	—	Entrained steam
f	—	Fuel
h	—	Heating steam
hc	—	Condensate of heating steam
l		Loss
т		Motive steam
min		Minimum
net		Net production
0		Surroundings
r	_	Recovery
saline	_	Saline water
SW		Seawater
w		Fresh water

0 - Reference parameter for water and power production

### **Superscripts**

0 — Standard reference state in chemical thermodynamics

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## Appendix A: Calculation of the minimal work needed in seawater desalination process

In a reversible process, the separation of salt and fresh water from seawater occurs at constant temperature T and pressure p equal to that of the surroundings, 30°C and 1 atm in this paper. The minimal work required for the separation of water from the saline solution is the difference in the Gibbs energy [33]:

$$w_{\min} = \left( m_{br} g_{br} + m_{w} g_{w} - m_{sw} g_{sw} \right) / m_{w}$$
(A1)

where subscripts *br*, *w* and *sw* represent rejected brine, produced fresh water and feed seawater respectively, and *g* is specific Gibbs energy,

$$g = h - Ts \tag{A2}$$

In the desalination process, saline water, including rejected brine and feed seawater, can be considered as an ideal solution (at least as a first approximation). To simplify calculation further, the salt in the saline water is considered to be only NaCl. If the mass fractions of NaCl and water in saline water are  $y_{\text{NaCl}}$  and  $y_{\text{w}}$ , and mole fractions  $z_{\text{NaCl}}$  and  $z_{\text{w}}$ , are then based on the properties of ideal solution [34],

$$h_{\text{saline}}(T,p) = y_{\text{NaCl}}h_{\text{NaCl}}(T,p) + y_{w}h_{w}(T,p) \text{ (A3)}$$

$$s_{\text{saline}}(T,p) = y_{\text{NaCl}}s_{\text{NaCl}}(T,p) + y_{w}s_{w}(T,p)$$

$$-R_{\text{NaCl}}\ln z_{\text{NaCl}} - R_{w}\ln z_{w} \text{ (A4)}$$

where  $R_{\text{NaCl}}$  and  $R_{\text{w}}$  are the gas constants for NaCl and water, respectively.

To ensure that different components in the separation process have the same reference standard,  $T^0 = 298.15$  K and  $p^0 = 1$  atm are taken as the reference state, and [35]:

$$h_{\text{NaCl}}(T^0, p^0) = \frac{\Delta H_{f,\text{NaCl}}^0}{M_{\text{NaCl}}} = 7032.85 \text{ kJ/kg}$$
 (A5)

$$h_{\rm w}(T^0, p^0) = \frac{\Delta H_{f,{\rm w}}^0}{M_{\rm w}} = 390.38 \text{ kJ/kg}$$
 (A6)

$$s_{\text{NaCl}}(T^0, p^0) = \frac{S_{\text{NaCl}}^0}{M_{\text{NaCl}}} = 1.2385 \text{ kJ} \cdot \text{K}$$
 (A7)

$$s_{\rm w}(T^0, p^0) = \frac{S_{\rm w}^0}{M_{\rm w}} = 3.8801 \text{ kJ}\cdot\text{K}$$
 (A8)

where  $\Delta H_f^0$  and  $S^0$  are enthalpy of formation and absolute entropy at  $T^0$  and  $p^0$ , and M is the molar mass. Then

$$h_{\text{NaCl}}(T,p) = h_{\text{NaCl}}(T^{0},p^{0})$$

$$+ \left[h_{\text{NaCl}}(T,p) - h_{\text{NaCl}}(T^{0},p^{0})\right] \qquad (A9)$$

$$= h_{\text{NaCl}}(T^{0},p^{0}) + \Delta h_{\text{NaCl}}$$

$$h_{w}(T,p) = h_{w}(T^{0},p^{0}) + [h_{w}(T,p) - h_{w}(T^{0},p^{0})]$$
$$= h_{w}(T^{0},p^{0}) + \Delta h_{w}$$
(A10)

$$s_{\text{NaCl}}(T,p) = s_{\text{NaCl}}(T^{0},p^{0})$$
$$+ \left[s_{\text{NaCl}}(T,p) - s_{\text{NaCl}}(T^{0},p^{0})\right] \qquad (A11)$$
$$= s_{\text{NaCl}}(T^{0},p^{0}) + \Delta s_{\text{NaCl}}$$

$$s_{w}(T,p) = s_{w}(T^{0},p^{0}) + [s_{w}(T,p) - s_{w}(T^{0},p^{0})]$$
$$= s_{w}(T^{0},p^{0}) + \Delta s_{w}$$
(A12)

Since NaCl and water are incompressible,

$$\Delta h = c \left( T - T^0 \right) \tag{A13}$$

$$\Delta s = c \ln \left( T / T^0 \right) \tag{A14}$$

where c is the specific heat.

Appendix B: Heat-transfer coefficient correlations used in the METVC and MEE models [24]

$$U_e = 1.9394 + 1.40562 \times 10^{-3} T_e$$

$$- 2.07525 \times 10^{-5} T_e^2 + 2.3186 \times 10^{-6} T_e^3$$
(B1)

$$U_c = 1.6175 + 1.537 \times 10^{-4} T_c$$

$$+ 1.825 \times 10^{-4} T_c^2 - 8.026 \times 10^{-8} T_c^3$$
(B2)

where  $U_e$  and  $U_c$  are heat-transfer coefficients of evaporation and condensation [in kW/(m<sup>2</sup>°C)], respectively, and  $T_e$  and  $T_c$  (in °C) the temperatures of evaporation and condensation.

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