Available online at www.sciencedirect.com
ScienceDirect
DESALINATION
Desalination 207 (2007) 243–256
www.elsevier.com/locate/desal

Performance analysis of combined humidified gas turbine power generation and multi-effect thermal vapor compression desalination systems

Part 2: The evaporative gas turbine based system and some discussions

Yongqing Wang[a*], Noam Lior[b]

[a]*College of Mechanical Engineering, Jimei University, Xiamen, 361021, PR China*
Tel. +86 (137) 7992 6575; Fax: +86 (59) 2618 3503; email: wyqyjp@hit.edu.cn
[b]*Department of Mechanical Engineering and Applied Mechanics, University of Pennsylvania, Philadelphia, PA 19104-6315, USA*

Received 4 February 2006; Accepted 16 June 2006

Abstract

This is Part 2 of the paper "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". A combined power and water system based on the evaporative gas turbine (EvGT) is studied, and major features such as the fuel saving, power-to-water ratio, energy and exergy utilization, and approaches to performance improvement, are presented and discussed in comparison with STIG- and EvGT-based systems, to further reveal the characteristics of these two types of combined systems. Some of the main results of the paper are: the fuel consumption of water production in STIG-based combined system is, based on reference-cycle method, about 45% of a water-only unit, and that in an EvGT-based system, it is 31–54%; compared with the individual power-only and water-only units, the fuel savings of the two combined systems are 12%–28% and 10%–21%, respectively; a water production gain of more than 15% can be obtained by using a direct-contact gas-saline water heat exchanger to recover the stack heat; and the combined system are more flexible in its power-to-water ratio than currently used dual-purpose systems. Further studies on aspects such as operation, hardware cost, control complexity, and environmental impact, are needed to determine which configuration is more favorable in practice.

Keywords: Integrated power and water production system; Water desalination; Humidified gas turbine cycles; Multi-effect thermal vapor compression desalination; Energy and exergy analysis

*Corresponding author.

doi:10.1016/j.desal.2006.06.013

1. Introduction

The objective of this two-part paper is to study the energy, exergy, and water production performance of integrated power and water desalination systems that employ humidified gas turbines (HGT), of which the steam-injected gas turbine (STIG) and humid air turbine (HAT) or evaporative gas turbine (EvGT) systems are the most representative. Following the analysis of thermal desalination unit and STIG-based combined system in Part 1 [1], this is Part 2 of the paper, focusing on EvGT-based combined system and performance discussion and comparison of the two combined systems. The calculation conditions and assumptions, as well as the performance criteria described in Part 1 are the same in this part 2.

2. EvGT-based power and water system

2.1. System configuration of EvGT-based system

In an EvGT-based system, as shown in Fig. 1, part of the compression-generated heat of the compressed air is recovered in heat exchangers AC1 and AC2 to heat water for air humidification. The humidified air is then heated by the turbine exhaust in a high temperature regenerator (R). The exhaust gas from the regenerator is used to produce steam in a heat recovery steam generator (HRSG) for desalination. A system configuration that directly uses the exhaust gas to heat saline water was not employed, to avoid potential corrosion and scaling problems caused by the higher temperature saline water that would be generated.

As discussed in part 1 of this paper and further elaborated in this paper, the EvGT cycle promises somewhat higher efficiency than the STIG cycle in a certain range of operating conditions, here for pressure ratios somewhat below 20, but at which it produces less water for the same power production when operated as a dual-purpose system. At higher pressure ratios its efficiency is somewhat lower, but the water production higher. Beyond efficiencies and power-water production rates, the practical decision on system preference would be based to important extent on the system cost and complexity, and the EvGT requires (Fig. 1) 5 more components than the STIG: two air-water aftercoolers (*AC* and *AC*2), an air-water humidifier (*H*), a water pump, and a regenerator (*R*). While we have not made a cost comparison, it is rather obvious that the EvGT system would be somewhat more complex and expensive than the STIG.

In a conventional EvGT cycle, both the compressed air stream and the high-temperature regenerator exhaust stream are used to heat the humidification water and provide energy for humidification, while in this study, the latter is used to produce saturated steam as a heat source and vapor compression driver for multi-effect thermal vapor compression (METVC) desalination. Obviously, compared with a conventional EvGT cycle, the power output *W*, the energy efficiency of power generation η_t (*W* as a fraction of the fuel energy Q_f consumed in the combustor), and the exergy efficiency of power generation ε_e (*W* as a fraction of fuel exergy E_f), of the EvGT cycle in the combined system will decrease, as an expense of water production.

In an EvGT-based combined system, the maximal pressure and thereby the maximum temperature of the steam produced for desalination is up to the state of the exhaust gas and the design of the HRSG. Steam of different pressures, lower than the maximum value, can be obtained from the HRSG in an EvGT-based system, which is unlike the STIG-based system where the pressure of the motive steam is the same as that of the steam injected into the combustor. Our calculations show that the highest pressure can help get a maximal water production. For a specified EvGT cycle, the parameters of the gas turbine exhaust gas are fixed. Although the heat energy recovered in the HRSG by the water/steam with

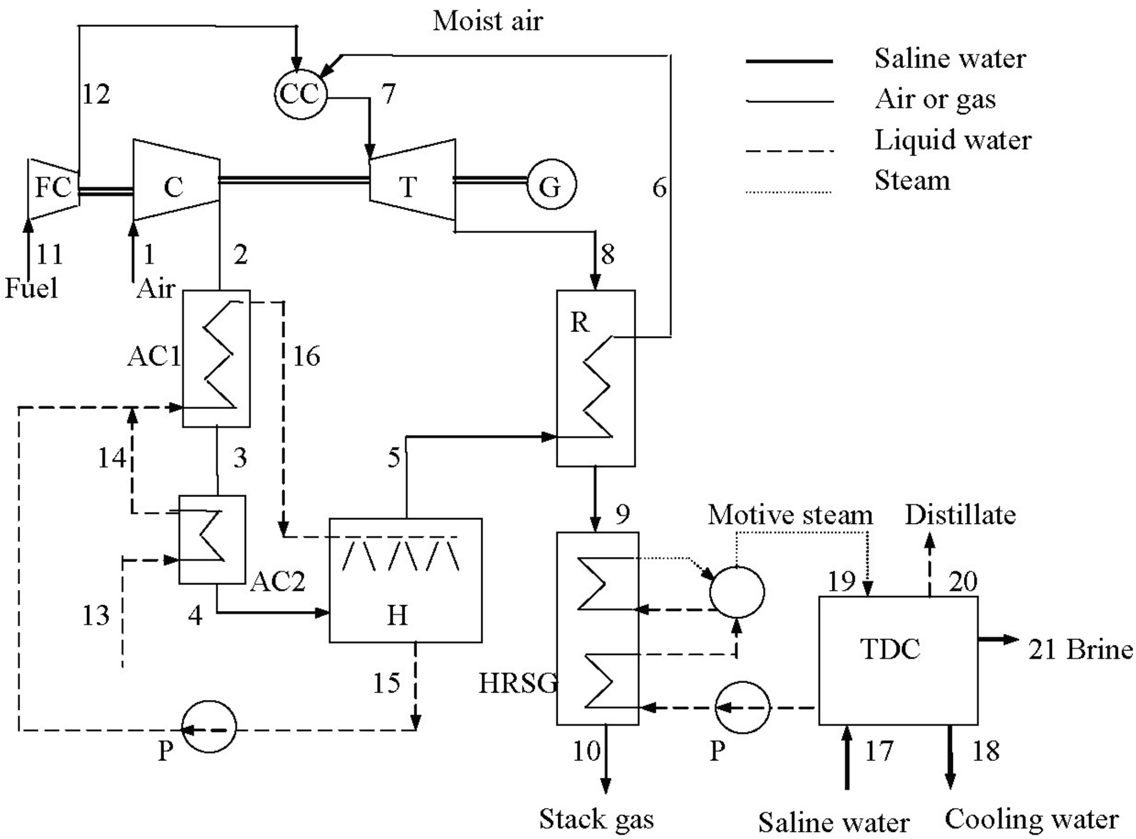

Fig. 1. EvGT-based power and water combined system. AC1, AC2, aftercoolers; C, compressor; CC, combustor; FC, fuel compressor; G, generator; H, humidifier; HRSG, heat recovery steam generator; P, pump; R, regenerator; T, turbine; TDC, thermal desalination unit.

different pressure p_m is the same, a higher p_m yields a more exergy-efficient heat transfer process. As shown in Fig. 2, for instance, the temperature difference between the gas and the water/steam is obviously smaller for p_m = 1.5 MPa than for p_m = 0.5 MPa, resulting in higher exergy recovery by the motive steam. This improved exergy utilization in the HRSG is enough to make up the increased exergy consumption in the METVC unit under high p_m (Fig. 5 in Part 1), thus resulting in an increased water production. The following analysis of the EvGT-based system is based on the maximal p_m (T_m) allowed in the HRSG. Calculations show that these maximum p_m are higher than 0.3 MPa

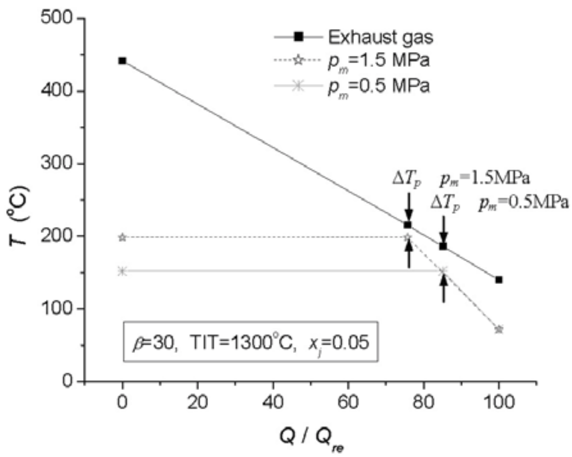

Fig. 2. *T-Q* diagram in the HRSG in the EvGT-based combined system for different p_m.

within the parameters range studied. Based on the performance analysis on METVC and MEE units performed in Part 1, a 6-effect METVC unit is also chosen for the EvGT-based system, as that for the STIG-based system.

The pressure ratio β of the compressor, the turbine inlet temperature (the "firing" temperature) TIT, and the humidification rate x_h are the most important parameters influencing the performance of the EvGT cycle, and of the combined system.

2.2. The influence of humidification rate x_h

Just as the steam injection rate x_j in the STIG-based combined system is a key parameter which determines the energy distribution between power generation and water production, the humidification rate x_h, which is the ratio of the water mass evaporated in the humidifier and the air mass through the compressor,

$$x_h = m_h / m_a \tag{1}$$

is such a key parameter in the EvGT-based systems.

Fig. 3 shows the influence of x_h on the power and water production, and Fig. 4 shows its influence on the energy and exergy utilization of the EvGT-based system. Clearly, increasing x_h causes the power output to increase, and the water production to decrease. For a specified pressure ratio β, the temperature of the compressed air from the compressor is fixed. More heat will be used to evaporate water in the humidifier if a higher x_h is wanted, leading to a lower temperature of the moist air at the outlet of the humidifier, more energy recovery in the regenerator, and less energy available for desalination (Fig. 4), and the result is higher production of power but lower of fresh water (Fig. 3).

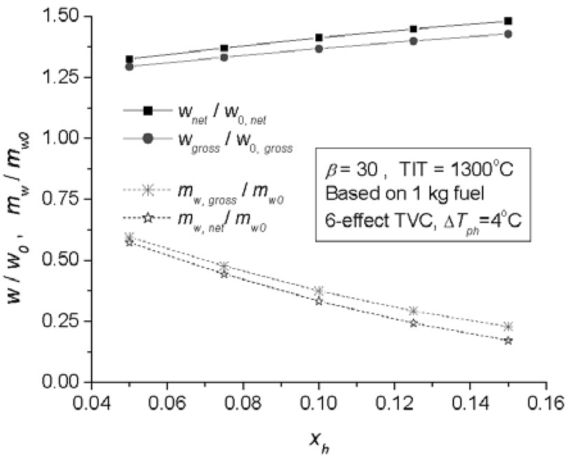

Fig. 3. Normalized power and water production of the EvGT-based system as a function of x_h.

2.3. The influence of pressure ratio β and firing temperature TIT of the cycle

Fig. 5 shows the exergy efficiency ε_e and thermal efficiency η_t of power generation, as well as the exergy recovery rate for desalination $\xi_{e,D}$, which is the exergy recovered by the motive steam as a fraction of fuel exergy E_f, and energy recovery rate for desalination $\xi_{t,D}$, which is the energy recovered by the motive steam as a

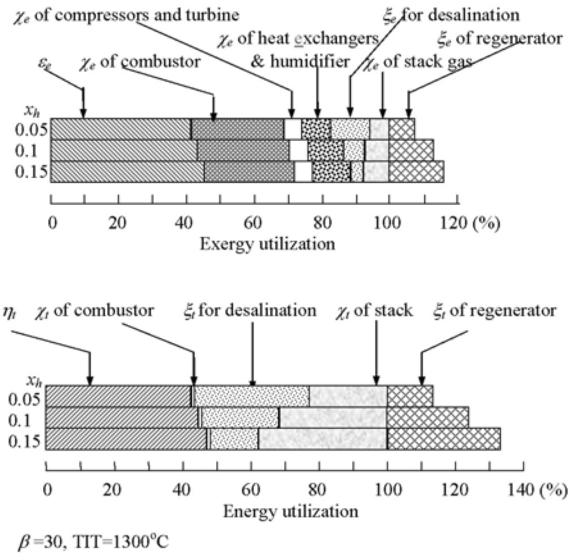

Fig. 4. Exergy and energy utilization of the EvGT-based combined system for different x_h.

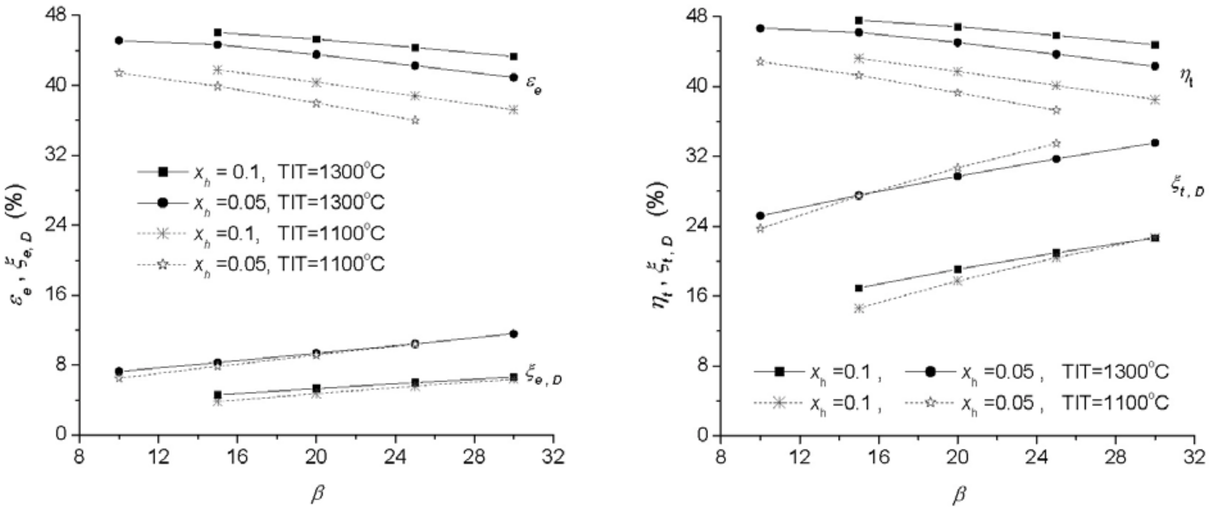

Fig. 5. Influence of β and TIT on ε_e, η_t ,$\xi_{e,D}$ and $\xi_{t,D}$ in the EvGT-based combined system. (a) Variations of ε_e and $\xi_{e,D}$ with β and TIT. (b) Variations of η_t and $\xi_{t,D}$ with β and TIT.

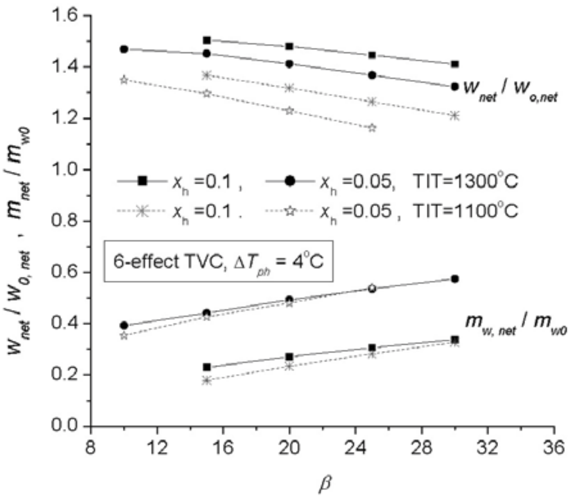

Fig. 6. Normalized power and water production of the EvGT-based combined system.

fraction of fuel energy Q_f, for different β (from 10 to 30), TIT (1100°C and 1300°C) and x_h (0.05 and 0.1) of an EvGT-based system. 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 will determine water production, as shown below, since that pumping work is only a small fraction of the desalination energy consumption. Fig. 6 shows the normalized power and water pro-duction, and Fig. 7 the exergy and energy utiliza-tion for different β.

Figs. 5 and 6 reveal that ε_e, η_t have a similar trend as the power output w, consistent with the definition of ε_e and η_t; so do $\xi_{e,D}$, $\xi_{t,D}$ and water production m_w. This behavior is the same as in the STIG-based system.

Different from the STIG-based system, a higher β lowers η_t, ε_e and w, as known in [2–4], while it raises $\xi_{t,D}$, $\xi_{e,D}$ and m_w in the EvGT-based systems. A higher β results in a higher air temperature at the compressor outlet, and then, for a specified x_h, a higher humid air temperature at the humidifier outlet, leaving less heat and exergy for recovery by the humid air in the regenerator, and more available for desalination (Fig. 7), and the result is lower power output and higher water production.

A higher TIT is beneficial to the power generation of the EvGT-based system, as determined by thermodynamic principles [2–4]. Different

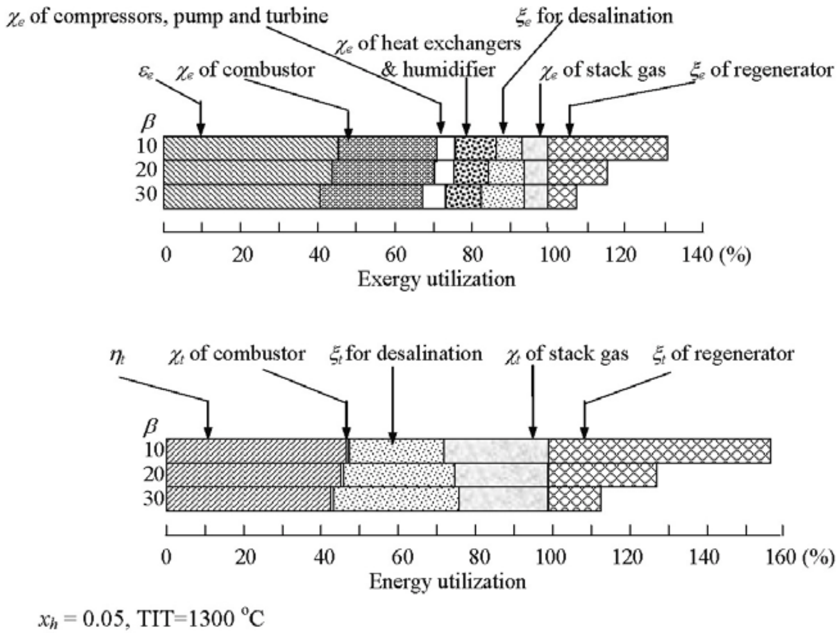

Fig. 7. Exergy and energy utilization of the EvGT-based combined system for different β.

from the STIG-based system in which a higher TIT also leads to a distinct increase in water production 1 kg/s fuel consumed, in the EvGT-based system the TIT does not have such an obvious positive influence on m_w, and even contrarily, for higher β the m_w of the EvGT-based system with a higher TIT may equal or even be lower than that with a lower TIT, as shown in Fig. 6. This means that in the EvGT-based system, for 1 kg/s fuel consumed, increase of the TIT will indeed increase the power output, but the water production may increase a little, remain the same, or even decrease. One can see from Fig. 8 that shows the exergy and energy utilization of an EvGT-based system for β = 25, x_h = 0.05 and TIT = 1300°C and 1100°C, that more fuel energy, and correspondingly almost the same amount of fuel exergy, is recovered for desalination when TIT = 1100°C than at TIT = 1300°C. This can also be seen clearly from Fig. 5. The reason is that when changing TIT from 1300°C to 1100°C, the air mass that 1 kg/s fuel can heat in the combustor will, obviously, increase, from 45.7 kg/s to 55.0 kg/s in the sample cases, leading to an increased mass flow of exhaust gas, from 49.0 kg/s to 58.8 kg/s, through the HRSG. The energy and exergy that can be recovered for desalination in the EvGT-based system depend mainly on the pressure, temperature and mass flow of the exhaust gas entering the HRSG, with the latter two being the main factors (since the pressure of the exhaust gas in the EvGT-based system is always close to that of the surroundings). In the sample cases, based on 1 kg/s fuel consumed, the mass and temperature of the exhaust gas are 49.0 kg/s and 148°C respectively for TIT = 1300°C, and 58.8 kg/s and 117°C for TIT = 1100°C. For TIT = 1100°C, the decreased exhaust temperature causes decreased specific enthalpy and exergy, while the increased exhaust mass raises the total amount of energy/exergy that could be recovered, resulting in a higher $\xi_{t,D}$ and an almost equal $\xi_{e,D}$ compared with that when TIT = 1300°C. It is the exergy, not the energy, that determines the water production, so almost the same amount of water

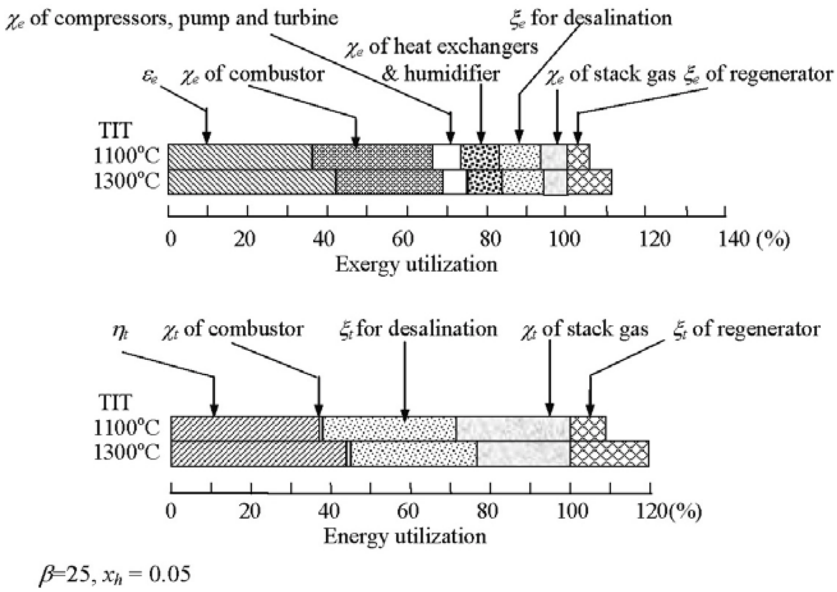

Fig. 8. Exergy and energy utilization of the EvGT-based combined system under different TIT.

is obtained in the two cases (Fig. 6). It is noteworthy that in both STIG- and EvGT-based systems, changing β, TIT or x_j/x_h for 1 kg/s fuel consumed, change both the temperature and mass of the exhaust gas for producing steam for desalination, but at all the conditions except the above-discussed ones, the temperature, not the mass flow of the exhaust gas, is the key factor influencing the amount of the energy/exergy recovery. Only the influence of the temperature is therefore mentioned in all the other conditions except the above-discussed ones.

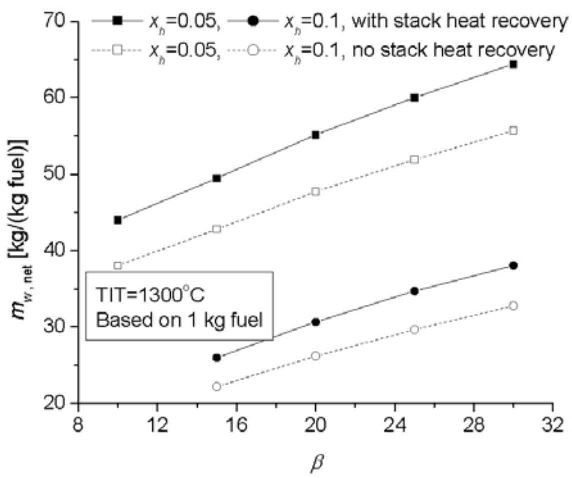

Fig. 9. Net water production of EvGT-based combined system with and without stack heat recovery.

2.4. Energy recovery from the stack gas

A direct-contact gas-saline water heat exchanger is also considered in the EvGT-based system for recovering the stack gas energy. Part of the saline water from the end condenser of the METVC unit is fed to the direct-contact heat exchanger, heated to 63°C by the stack gas, and then, after mixing with the saline water from the end condenser according to the mass flow and temperature required by each effect, used as the feed of a 6-effect METVC unit, in which no preheaters are used because the feed saline water has been preheated by the stack gas. The net water production with and without stack heat recovery are both shown in Fig. 9, and the result again indicates a distinct positive effect of stack heat recovery.

3. Discussions on STIG- and EvGT-based combined systems

3.1. Energy saving of the STIG- and EvGT-based combined systems

Fig. 10 shows the optimal thermal efficiencies $\eta_{t,opt}$ and the corresponding steam injection rate x_j and humidification rate x_h of the conventional STIG cycle and EvGT cycle, for TIT = 1300°C for different pressure ratios β. To get the optimal efficiency, all the steam produced in the HRSG in the STIG cycle is injected into the combustor at the highest temperature allowed by the working condition of the HRSG, and all the parameters in the EvGT cycle are optimized by taking thermal efficiency as the objective function. We can see from the figure that, within the parameter range studied, $\eta_{t,opt}$ of the STIG cycle increases substantially with β, while that of the EvGT cycle is not so sensitive to β. The STIG cycle has higher injection rate, from 0.195 to 0.306, compared with 0.16 to 0.22 for the EvGT cycle. This paper will focus on the characteristics of the combined systems, and the readers are referred to references [2–4] to get detailed performance analysis and comparison on the two power-only cycles.

As mentioned above, water production in both STIG- and EvGT-based systems is at the expense of work-production efficiency. When β =30 and TIT = 1300°C, for instance, $\eta_{t,opt}$ of the power-only STIG cycle is 52.8% with a x_j of 0.195 [which means that there is no water production, and actually net water consumption of 7 kg/(kg fuel)]. Reducing x_j to 0.05, and using the extra energy to produce steam for the METVC unit, reduces η_t to 44.85%, with a net water production of 74.7 kg/(kg fuel).

It is noteworthy that the value and the trend of η_t of the STIG cycle and the EvGT cycle in Fig. 10 are different from those in Fig. 13 in Part 1 and Fig. 5 in this part, because the former is based on the optimized x_j or x_h which can help get an optimal efficiency, while the latter is based on a specified x_j, say 0.05 or 0.1.

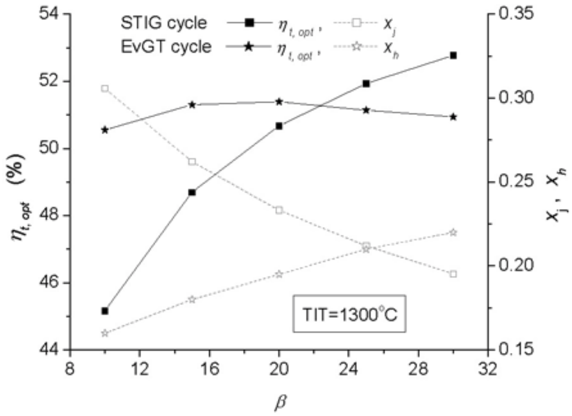

Fig. 10. Performance of conventional power-only STIG and EvGT cycles.

The fuel energy allocation between power and water in a dual-purpose system can be made in several ways [5,6], including the lost work method, exergy method, reference cycle method, etc. The reference cycle method [5] is used in this paper to calculate the energy consumption of water production, but differently: instead of taking a specified efficiency as the reference, the energy allocation here is based on the optimal efficiency of a power-only STIG or EvGT cycle having the same pressure ratio and firing temperature as the corresponding combined power/ water production system. In an optimal power-only cycle, m_f kg fuel is needed to produce the work W:

$$m_f = W/(q \cdot \eta_{t,opt}) \tag{2}$$

where q is low heat value of fuel. Since part of the fuel energy is used to produce water in the combined systems, which leads to a decreased efficiency η_t, more fuel, m'_f, is needed to generate the same amount of work:

$$m'_f = W/(q \cdot \eta_t) \tag{3}$$

The extra fuel needed can be considered as the fuel consumption of water production:

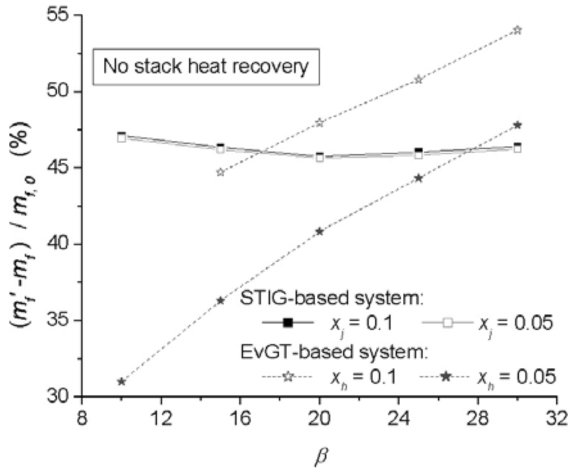

Fig. 11. Normalized fuel consumption of water production based on reference cycle method.

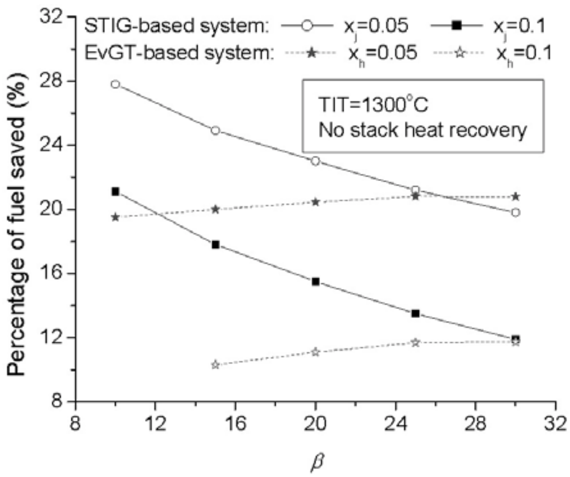

Fig. 12. Fuel saved compared with single-purpose power and water production units.

$$m'_f - m_f = m'_f\left[1-(\eta_t/\eta_{t,opt})\right] \quad (4)$$

Based on this fuel allocation method, the fuel exergy consumption per kg produced fresh water, not including the desalination pumping work, is calculated and shown in Fig. 11. These values are normalized by the specific fuel consumption $m_{f,0}$, 6.059×10^{-3} kg/(kg distillate), of a 6-effect METVC unit run by 2.5 MPa saturated steam from a boiler with an efficiency of 0.9. For the STIG-based system, the fuel consumption is about 45% of that of the water-only unit, and for the EvGT-based system, it is 31%–54%, when x_j and x_h change from 0.05 to 0.1.

The energy saving of combined systems can also be illustrated from another angle. Compare the dual purpose system to two separate units, one producing just power at optimal efficiency (Fig. 10), and one METVC unit run by 2.5 MPa saturated steam from a boiler producing just water, producing the same amount of net power and fresh water as the dual-purpose system at the same pressure ratio and firing temperature. The results are shown in Fig. 12. Thanks to the combination, the fuel saving of the STIG-based system is 19.8%–27.8% and 11.9%–21.1% when x_j = 0.05 and 0.1 respectively, and that of the EvGT-based system is 19.5%–20.8% and 10.3%–11.8% when x_h = 0.05 and 0.1 respectively.

The results of the above two methods indicate substantial improvements in fuel utilization of STIG- and EvGT-based combined systems, although with different rate and trend.

3.2. Power-water ratio of STIG- and EvGT-based combined systems

Fig. 13 shows the power and water ratio R_{pw} of the two combined systems. Within the parameter range studied, the trend of R_{pw} of the combined systems is the same as that of the η_t or ε_e of the two cycles. Higher β and higher x_j are beneficial for STIG cycle to get a higher η_t and ε_e (Figs. 11 and 13 in Part 1), and then tend to a higher R_{pw} (Fig. 13), and to EvGT cycle, lower β and higher x_h are helpful to an increased η_t, ε_e and R_{pw} (Figs. 3, 5 and 13).

Compared with the typical designed dual-purpose system [7], including steam turbine-MSF plant with a R_{pw} of 4–19, gas turbine-MSF plant with a R_{pw} of 6–13, and combined cycle-MSF plant with a R_{pw} of 9–18, STIG-based and EvGT-

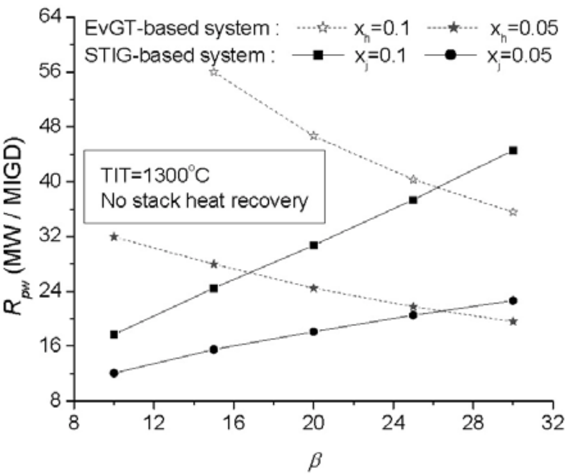

Fig. 13. Power and water ratio of combined power and water systems.

based system are seen to have higher R_{pw}, 12–44 and 20–56 respectively, when x_j and x_h change from 0.05 to 0.1, mainly because of the higher efficiency of the base cycles.

One advantage of the combined systems studied in this paper is the great design and operational flexibility of water and power production, because R_{pw} can be regulated in a wide range by changing x_j or x_h (Fig. 13). For instance, for a STIG-based system with β of 20 and TIT of 1300°C, varying x_j from 0 to 0.1, R_{pw} changes from 11.6 to 30.7.

3.3. Energy and exergy utilization of STIG- and EvGT-based combined systems

To make the exergy and energy utilization of STIG- and EvGT-based combined systems and ways to improve them clearer, the exergy and energy flow diagrams of the two systems for β = 20, TIT = 1300°C, and x_j or x_h = 0.1 are shown in Figs. 14–17.

As well known, exergy and energy analyses give different results. For instance, in the STIG-based system, the exergy loss rate, χ_e, in the combustor is 29.5%, ranking as the highest among the exergy losses of the components, while the energy loss rate, χ_t, is only 1%; χ_e in the stack is 6.5%, while χ_t is 30.3%, ranking as the highest among the energy losses of the components; χ_e of the compressors, turbine and HRSG are 1.9%, 3.1% and 6.9% respectively, while the χ_t magnitudes are negligible. Detailed exergy and energy utilization as well as performance improvement approaches, such as intercooling, reheating, recuperation, etc, of STIG and EvGT cycles can be found in many references, and a summary can be found in [8]. Here we focus only on the interface of the power and water production subsystems.

In a STIG-based system, the exergy loss in the HRSG is large, the second among the exergy losses of the components, because of the large heat-transfer temperature difference between the exhaust gas and water/steam, especially in the water evaporation process (Fig. 2). Coming directly from the gas turbine, the temperature of the exhaust gas used to produce steam in the HRSG is usually high, 630°C in the case shown in Figs. 14 and 15, while the evaporation temperature of water is much lower, about 224°C (2.5 MPa) in the said case, resulting in a large heat-transfer irreversibility, and thus a large exergy loss. Obviously, increasing the steam pressure and correspondingly the evaporation temperature can help reduce the exergy loss (Fig. 2), at the expense of an increase of the heat-transfer area owing to the reduced heat-transfer temperature difference. In a conventional STIG system, however, no benefit can be obtained by the production of higher pressure steam, because the steam must be throttled to a pressure close to that of the combustor before its injection into the combustor, and thus that extra exergy gained by the steam in the HRSG would be uselessly lost in the throttling process. Moreover, such high pressure steam may cause unstable operation of the steam jet ejector. Even if the operation would be stable, the gain in water production would not be significant because the performance ratio, PR, of

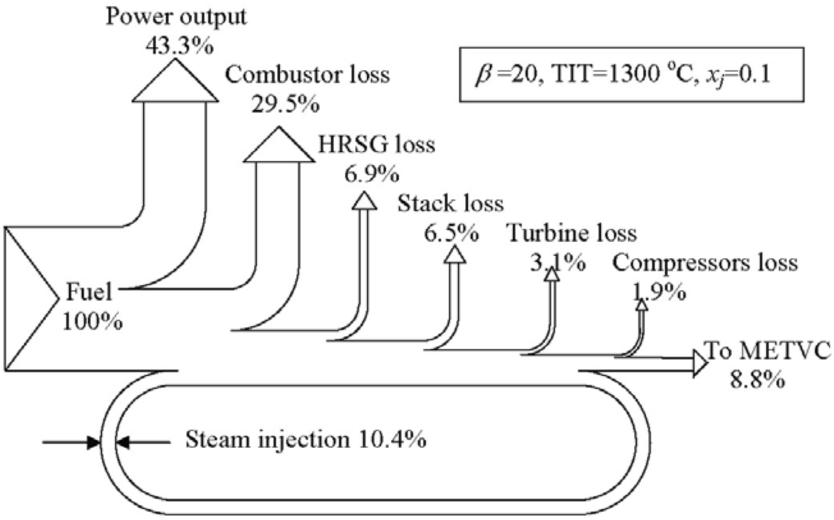

Fig. 14. Exergy flow diagram of the STIG-METVC system.

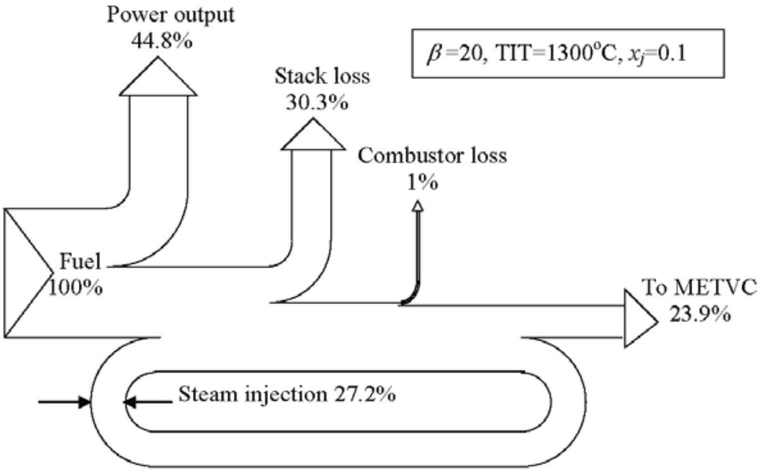

Fig. 15. Energy flow diagram of the STIG-METVC system.

the METVC unit has a diminishing rate of an increase as p_m is raised (Fig. 5 in Part 1). For instance, for the 6-effect METVC unit used in this paper, *PR* increases from 8.9 to 9.2 when p_m is raised from 1.5 MPa to 2.5 MPa, but only from 9.2 to 9.3 when p_m is raised further from 2.5 MPa to 3.5 MPa. Obviously, using steam at a pressure of, say, 10 MPa, would thus not raise the water production rate much when compared with that at 3.5 MPa.

The exergy analysis results point to several ways for performance improvement of the STIG-based system by more efficient use of the high pressure steam that could be produced in the HRSG. For instance, (1) that steam can be expanded in a steam turbine to the pressure needed for supplying the heat for the desalination and thus produce power in that intermediate process. From the exergy viewpoint, expanding the steam to a suitable pressure, e.g. 0.034MPa, to

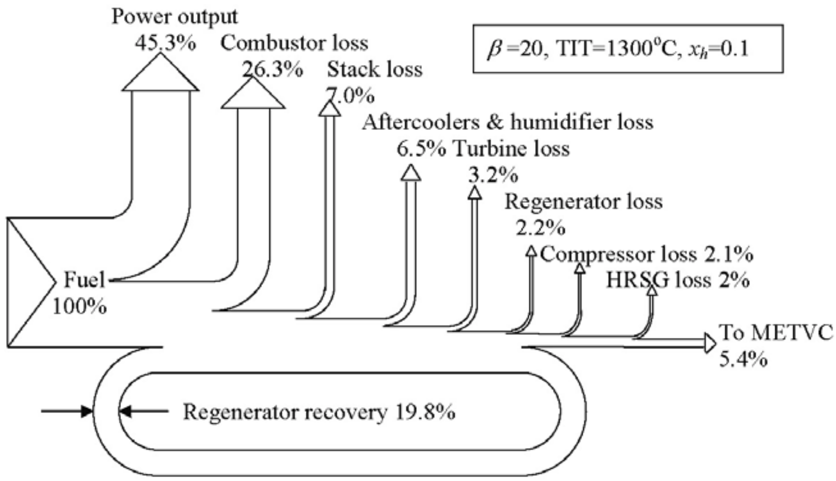

Fig. 16. Exergy flow diagram of the EvGT-METVC system.

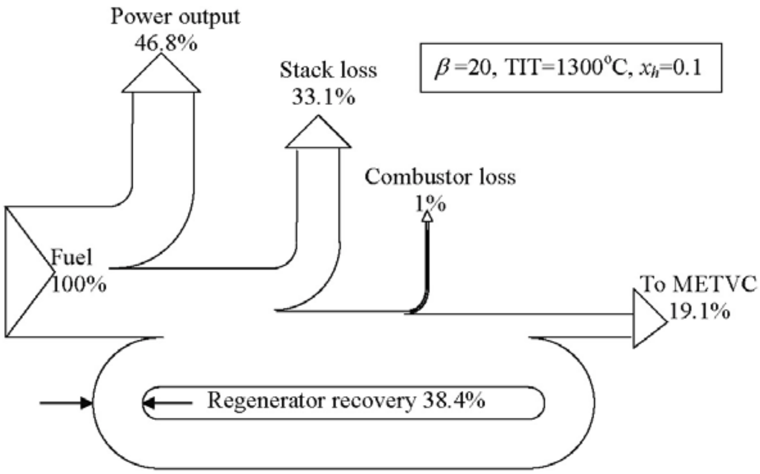

Fig. 17. Energy flow diagram of the EvGT-METVC system.

run an MEE unit is more efficient than expanding it to a relatively high pressure, e.g. 2 MPa, to run a METVC unit, mainly because the steam jet ejector is not an efficient compressor (ejectors are commonly used in industrial processes anyway because they are simple devices without moving parts, and of low maintenance and cost), (2) the steam can be expanded in a steam turbine to the lowest pressure allowed by the environmental coolant to produce more power, which can then be used for running a reverse osmosis (RO) desalination unit, or a mechanical vapor compression one in which the thermal vapor compressor (i.e. the steam jet ejector in a METVC unit) is replaced by a mechanical compressor; mechanical compressors are much more efficient than ejectors. Detailed study of these and other improvement options are under way.

In an EvGT-based system within the parameters range studied in this paper, the pressure of the steam produced in the HRSG is lower than 2 MPa when the minimum pinch point temperature difference of the HRSG $\Delta T_p = 15\,^\circ$C because of the relatively low temperature (310 °C in the case shown in Figs. 16 and 17) of the exhaust gas from the regenerator in which part of its exergy has been recovered by the moist air. Higher p_m and then higher exergy utilization efficiency of HRSG and higher water production could be obtained by further decreasing ΔT_p through increasing the heat transfer area in the HRSG, but because that ΔT_p is already very low, little room is left for improvement. Although p_m in an EvGT-based system is usually much lower than that in a STIG-based system, it is still much higher than the 0.034 MPa required for the MEE unit in this paper, so the approaches proposed for improving the STIG-based system can also be used in the EvGT-based one.

It is noteworthy that in both combined systems the highest energy losses are in the stack, 30.3% and 33.1%, respectively, for the sample cases shown in Figs. 14–17, and the exergy losses are much lower, but still 6.5% and 7%. The large latent heat of condensation of the water vapor in the flue gas is one of the main reasons for the high energy/exergy losses. Three methods were discussed in Part 1 to recover this low-level heat, and addition of a gas-saline water direct contact heat exchanger to preheat the feed saline water by using the stack gas was found to provide a distinct gain in water production for the same fuel input as that without stack heat recovery. More pumping work will, however, be consumed because (1) more saline water and produced fresh water need to be pumped, (2) the pressure of part of the saline water needs to be raised to a value higher than that without stack heat recovery so as to make it possible to be sprayed into the exhaust gas in the heat exchanger. Further study is needed to evaluate the advantages and disadvantages of this modification.

This paper focuses only on the thermodynamic performance of the STIG- and EvGT-based combined systems. Further study of other aspects such as operation, size, cost and environmental impact, is necessary to determine which configuration is more favorable.

4. Conclusions

Combined power and water systems based on the STIG cycle and on the EvGT cycle were configured, modeled, and analyzed in detail in this two-part paper. Both energy and exergy performance criteria were employed. The main results are:

1. Great synergy, reflected in energy savings, is accomplished by combining the humidified power cycle with the METVC desalination unit. The fuel consumption of water production in a STIG-based combined system is, based on a reference-cycle method, about 45% of a water-only unit, and that in an EvGT-based system it is 31–54%. Compared with the individual power-only and water-only unit, the fuel savings of the two combined system are 11.9%–27.8% and 10.3%–20.8%, respectively.
2. In a STIG-based system, higher pressure ratios are advantageous to power generation, but disadvantageous to water production, and the opposite is true for the EvGT-based system. In both systems, higher injection or humidification rate can result in a higher power/water ratio.
3. The steam injection rate in the STIG-based system and the humidification rate in the EvGT-based system have strong influence on the power and water production of the two systems, providing greater flexibility for design and operation.
4. A distinctly positive effect is that a water production gain of more than 15% can be obtained by using a direct-contact saline water-gas heat exchanger to recover the stack heat.
5. Compared with the currently used typical dual-purpose systems, these two combined systems have higher power-to-water ratios because of the higher efficiency of the base cycles.
6. The exergy analysis gave guidance for some potentially promising methods for further efficiency improvements.

5. Symbols

E — Exergy, kJ
m — Mass, kg
p_m — Pressure of motive steam, MPa
PR — Performance ratio of thermal desalination
q — Low heat value of fuel, kJ/kg
Q — Energy, kJ
R_{pw} — Power-to-water ratio, MW/MIGD
T_m — Temperature of motive steam
TIT — Turbine inlet temperature (or called firing temperature), °C
w — Specific work, kJ/kg
W — Power output, kJ
x_j — Steam injection rate in STIG cycle
x_h — Humidification rate in EvGT cycle

Greek

β — Pressure ratio of gas turbine cycle
ε_e — Exergy efficiency of power generation, %
η_t — Thermal efficiency of power generation, %
χ_t — Heat loss rate, %
χ_e — Exergy loss rate, %
ξ_t — Heat recovery rate, %
ξ_e — Exergy recovery rate, %
ΔT_p — Pinch point temperature difference of HRSG, °C
ΔT_{ph} — Temperature rise of saline water in preheaters of METVC unit, °C

Subscripts

a — Air
D — Desalination
f — Fuel
l — Loss
net — Net production
r — Recovery
w — Fresh water
opt — Optimal
0 — Reference parameter for power and water production

Acknowledgement

The authors gratefully acknowledge the support of the State Scholarship Fund of China to the first author.

References

[1] Y. Wang and N. Lior, 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, Desalination, 196 (2006) 84–104.
[2] P. Chiesa, G. Lozza and S. Consonni, An assessment of the thermodynamic performance of mixed gas-steam cycles, Part A: Intercooled and steam-injected cycle; part B: Water-injected and HAT cycles, ASME J. Eng. Gas Turbine Power, 117 (1995) 489–498; 499–508.
[3] Y. Wang, A. Chen and J. Yan, Performance comparison of combined cycle, STIG cycle, HAT cycle and other relevant cycles, J. Eng. Thermophysics, (1998) 545–548 [in Chinese].
[4] M.A. Bartlett and M.O. Westermark, A study of humidified gas turbine for short-term realization in midsized power generation Part I: Nonintercooled cycle analysis, ASME J. Eng. Gas Turbine Power, 127 (2005) 91–99.
[5] N.M. Wade, Energy and cost allocation in dual-purpose power and desalination plants, Desalination, 123 (1999) 115–125.
[6] M.A.K. Al-Sofi and M.M. Srouji, Fuel allocation in dual-purpose plants, Desalination, 100 (1995) 65–70.
[7] A.M. El-Nasgar, Cogeneration for power and desalination — state of the art review, Desalination, 134 (2001) 7–28.
[8] M. Jonsson and J. Yan, Humidified gas turbines — a review of proposed and implemented cycles, Energy, 30 (2005) 1013–1078.