

Analysis of an electricity-cooling cogeneration system for waste heat recovery of gaseous fuel engines

SHU GeQun^{*}, WANG Xuan, TIAN Hua, LIANG YouCai, LIU Yu & LIU Peng

State Key Laboratory of Engines, Tianjin University, Tianjin 300072, China

Received September 14, 2014; accepted December 8, 2014

Waste heat recovery (WHR) is one of the most useful ways to improve the efficiency of internal combustion engines, and an electricity-cooling cogeneration system (ECCS) based on Rankin-absorption refrigeration combined cycle for the WHR of gaseous fuel engines is proposed in the paper. This system can avoid wasting the heat in condenser so that the efficiency of the whole WHR system improves, but the condensing temperature of Rankin cycle (RC) must increase in order to use absorption refrigeration system, which leads to the decrease of RC output power. Therefore, the relationship between the profit of absorption refrigeration system and the loss of RC in this combined system is the mainly studied content in the paper. Because the energy quality of cooling and electricity are different, cooling power in absorption refrigeration is converted to corresponding electrical power consumed by electric cooling system, which is defined as equivalent electrical power. With this method, the effects of some important operation parameters on the performance of the ECCS are researched, and the equivalent efficiency, exergy efficiency and primary energy rate are compared in the paper.

gaseous fuel engines, waste heat recovery, electricity-cooling cogeneration, Rankin cycle, absorption refrigeration

Citation: Shu G Q, Wang X, Tian H, et al. Analysis of an electricity-cooling cogeneration system for waste heat recovery of gaseous fuel engines. *Sci China Tech Sci*, 2015, 58: 37–46, doi: 10.1007/s11431-014-5742-7

1 Introduction

The ICE (internal combustion engine) plays a very important role in modern industry, and consumes a lot of energy at the same time. Even modern combustion engines achieve a maximum efficiency of more than 40%, there is still about 60% of the energy in the fuel which is rejected to the environment via exhaust, charge air, jacket water and so on [1]. As energy crisis and environment pollution problem have become increasingly prominent, people are paying more and more attention to enhance the efficiency of internal combustion engines. A valuable alternative approach to improving overall energy efficiency is to capture and reclaim the “waste heat” [2].

The gaseous fuel engine is a kind of ICE that uses nature gas, coalbed methane or some other gas fuel as input energy.

With the exhaustion of petroleum resources and the increasingly serious environmental pollutions, gaseous fuel has been paid more and more attention because of its valuable features [3]. The first feature is that gaseous fuel is cleaner than petroleum. Low carbon content and clean combustion (low soot and smoke) have helped the proliferation of gaseous fuel as an alternate fuel with the introduction of ever more stringent emissions standards [4]. Another feature is that gas resources are far more abundant than oil and the shift is towards greater use of natural gas [5]. As a result, gas engines are widely used in many places like oilfield and coalmines that have a plenty of gaseous fuel resource or independent business district and so on. On the other hand, gaseous fuel engines have demonstrated a slight reduction of brake thermal efficiency when compared to diesel engines [6,7], and then more energy is wasted. Therefore, the waste heat recovery is more important for gaseous fuel engines.

^{*}Corresponding author (email: sgq@tju.edu.cn)

There is a lot of published research literature about waste heat recovery for ICEs, but only little aimed at gaseous fuel engines [8]. For the waste heat recovery of gas engines, supplying heat is the most common way [9,10]; just a very few researches use organic Rankin cycle (ORC) or water Rankin cycle (RC) [11]; Beside, some researchers applied absorption heat pump to increase the overall energy utilization rate [8]. In fact, the methods of waste heat recovery for other kinds of ICEs can be referred absolutely.

These years, the most popular way is ORC or RC and RC is usually applied to large engines. For the gas engines used to produce electricity in oilfields, coalmines or buildings, their rated power is relative large, so RC is very suitable for waste heat recovery. MAN Company uses single-pressure Rankin cycle and dual-pressure Rankin cycle to recover the waste heat of engines, and they have already produced very mature products. These two kinds of WHR system can respectively improve 4%–7% and 6%–9% of the engine output power at SMCR (specified minimum continued rate) [12]. Wartsila Company also has the similar products. Their dual-pressure RC can bring about an increase of 8.7% of output power at SMCR for large ship engines 12RT-lex96C [13].

If only RC is used to recover waste heat, a large amount of heat releases to cooling water in condenser. Although this part of energy has low quality, the amount is very large. The greatest exergy destruction in RC is also in condenser [14]. In addition, because there are many people working in the place like oilfields, coalmines or independent business district, not only electricity but also cooling is required in order to make a comfortable environment. Therefore, in our past researches, we proposed a new electricity–cooling cogeneration system (ECCS) based on Rankin–absorption refrigeration combined cycle for the WHR of large engines to avoid wasting the heat in condenser [15,16]. In this paper the ECCS is applied to recover waste heat of a gaseous fuel engine that is used for producing electricity.

However, if the absorption cooling system is used, the condensing temperature of RC must rise, which leads to the reduction of RC output power, so the relationship between the profit of absorption refrigeration and the loss of RC in this combined system is the studied content in the paper. The cooling energy in absorption refrigeration is converted to corresponding electrical power consumed by electric cooling system and it is defined as equivalent electrical power. With this method, the effects of some important operation parameters on the performance of the ECCS are researched, and the equivalent efficiency, exergy efficiency and primary energy rate are compared in the below. Finally, the advantages of the ECCS can be seen obviously.

2 System description

2.1 The gaseous fuel engine

The rated power of studied gaseous fuel engine is 1100 kW

and the corresponding output electricity power is 1000 kW. Compared to engines for ships and automobiles, the gas engine for electricity has more stable working conditions. Consequently, the WHR systems are easier to apply successfully and run steadily. The exhaust of the studied engine at the rated power is the heat source of the ECCS for waste heat recovery. The main parameters of it are in Table 1.

It can be seen from Table 1 that the actual air-fuel ratio is much greater than theoretical air-fuel ratio. This is because that lean combustion can benefit the fuel economy of gas engines [17,18]. Therefore, most of gas engines on the market adopt large air-fuel ratio. According to the actual air-fuel ratio and assume the fuel complete combustion, the volume fraction of the exhaust gas is then calculated: $N_2=74.0\%$, $CO_2=6.3\%$, $H_2O=12.7\%$ (gas), $O_2=7\%$. From this, physical properties of exhaust like specific heat can be obtained.

2.2 The ECCS description

The combined system is divided into two parts: Ranking cycle and aqua ammonia absorption refrigerating system as shown in Figure 1.

The Ranking cycle in this combined system consists of pump, preheater, evaporator, turbine and condenser. The exhaust of high temperature transfers energy via heat exchanger to the working fluid, which makes the water evaporate and become steam of high temperature and pressure. Then the steam vapor expands in a turbine to generate power, decreasing the temperature and pressure. The steam after the turbine is condensed into saturated water in the condenser and the condenser also plays a role of generator in the absorption refrigeration cycle. Finally, the saturated water is pumped to the preheater again starting a new cycle. Figure 2 is the process of RC in T-S diagram.

In the absorption refrigeration cycle, ammonia and water are used as the working pair. As mentioned above, the steam after the turbine is the heat source to distill ammonia vapor from the strong ammonia water in the generator. The ammonia vapor is rectified at first, and then condensed to be liquid ammonia through a heat exchanger. After being cooled again in the subcooler, the liquid ammonia flows through the expansion valve 1 that is used to control the flow rate of the refrigerant and drop its pressure in order to evaporate at a low pressure in evaporator, extracting heat from refrigerating medium and producing cooling capacity.

Table 1 Main parameters of engine exhaust

| Parameter | Values |
|--|--------------------------|
| Exhaust temperature | 532°C |
| Volume flow rate of intake air (standard conditions) | 1.16 m ³ /s |
| Volume flow rate of gaseous fuel (standard conditions) | 0.0784 m ³ /s |
| Actual air-fuel ratio (Volume ratio) | 14.8 |
| Theoretical air-fuel ratio (Volume ratio) | 9.52 |
| Exhaust mass flow rate (standard conditions) | 1.25 m ³ /s |

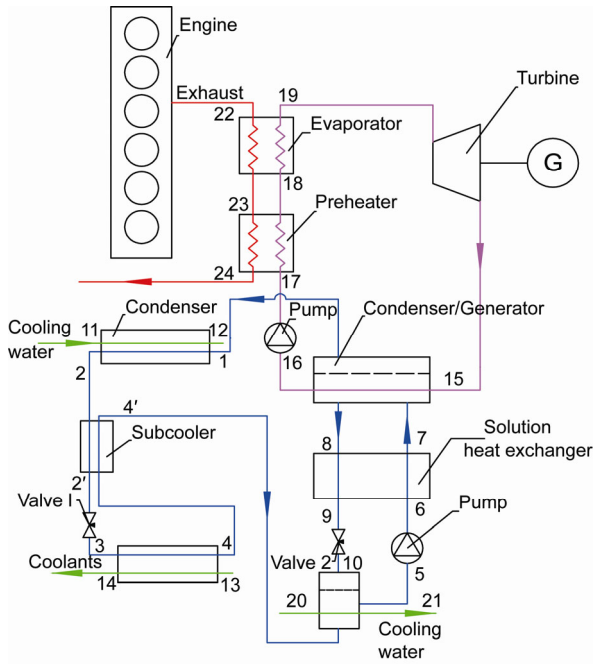


Figure 1 (Color online) Schematic diagram of the ECCS.

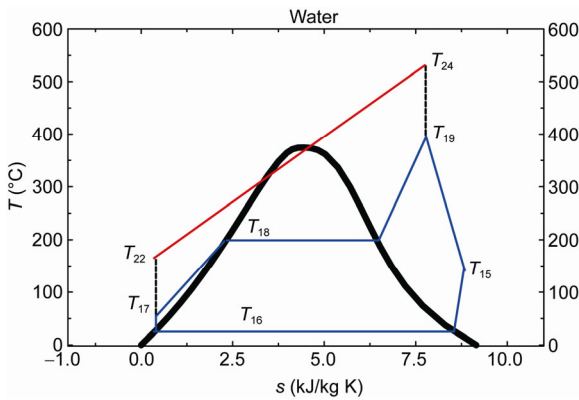


Figure 2 (Color online) T - S diagram of working fluid in Rankin cycle.

The function of subcooler is to cool the liquid ammonia further, which can increase the cooling capacity of the whole system. After that, the vapor dissolves in the left dilute solution from the generator, forming strong solution in the absorber. Since the process releases a large amount of heat, it must be taken away by cooling water to keep the low temperature in absorber so that the gaseous ammonia can dissolve in the dilute solution very well. The strong solution from the absorber is pumped into the solution heat exchanger (SHX) by pump-1, which can recover a part of energy of dilute solution to improve the thermal efficiency of cooling cycle [19], and flows into the generator finally. At this point, the cycle restarts.

2.3 The basic Rankin cycle with low condensing temperature

In order to indicate the advantages of the ECCS, the basic

condensing RC with low condensing temperature are analyzed to compare with the ECCS in the paper. The Schematic diagram of the basic Rankin cycle is shown in Figure 3. Because the heat source in generator is the steam after turbine and most of heat releases in the process of condensation, the condensing temperature of the Rankin cycle in ECCS decides the final temperature of strong solution. The final temperature should not be too low [20], otherwise the refrigeration system cannot work, especially at low evaporating temperature of absorption refrigeration system. Therefore, the RC condensing temperature in the combined system is higher than that of the basic Rankin cycle which have a condensation temperature lower than 45°C and a large output power.

3 Math model

The math model is built in MATLAB and the needed thermodynamic parameters of work fluid in different states were obtained by REFPROP. The physical property of ammonia and ammonia water solution is based on the Schulz's equation of state [21]. Before detailed analysis, typical assumptions should be first considered.

- (1) The system is operating in a steady state.
- (2) The efficiency of fluid pumps in the system is 0.8, and that of turbine is 0.8.
- (3) The final temperature of the exhaust out of the evaporator is higher than that of acid dew point.
- (4) Ammonia vapor from the generator is assumed pure before flowing into the condenser.
- (5) The refrigerant out of the condenser is saturated liquid and the refrigerant out of the evaporator is saturated vapor.
- (6) The pressure drop in pipes is ignored and all of the heat exchangers are ideal.

Main operation parameters that refer to the actual products

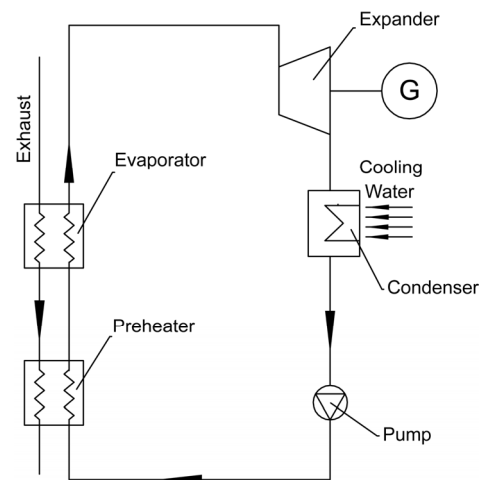


Figure 3 Schematic diagram of the basic condensing RC.

should also be set before the thermodynamic modeling [15,22], as shown in Table 2.

Based on the assumptions and the operation parameters, the working process in each part can be described as follows.

3.1 ORC

Work fluid pump (process: 16–17), the fluid pump is driven by electricity and the consumed work is

$$W_{p1} = m(h_{17} - h_{16})/\eta_{p1}, \quad (1)$$

$$\eta_{p1} = \frac{h_{17s} - h_{16}}{h_{17} - h_{16}}, \quad (2)$$

where m is the mass flow rate of work fluid, h_{17s} is the enthalpy of point 17 in isentropic expansion, η_p is the efficiency of the pump, which is assumed as 0.8. The power consumed by the pump W_p is usually not large.

Preheater (process: 17–18), it is used to heat the water to saturation temperature:

$$(T_{23} - T_{24})C_p m_g = m(h_{18} - h_{17}), \quad (3)$$

where T_{23} is the temperature of exhaust when work fluid starts to boil. m_g is the mass flow rate of exhaust. It should be noticed that the minimum temperature difference ΔT in heat exchangers, which prevents the heat transfer area from being too large, may appear at the work fluid inlet of evaporator or that of preheat. Through calculation where ΔT appears can be known. Firstly, assume ΔT at the inlet of evaporator, then $T_{23} = T_{\max} + \Delta T$, so T_{24} can be figured out by formula (3) and (4). If the calculated T_{24} minus T_{17} is less than ΔT , $T_{24} = T_{17} + \Delta T$. As a result, m and T_{23} can be figured out

as well by the two formulas. ΔT is set as 25 K in this paper.

Evaporator (process: 18–19), the saturation water is heated to superheated steam here. This is supposed to be an isobaric heating process:

$$(T_{22} - T_{23})C_p m_g = m(h_{19} - h_{18}). \quad (4)$$

Turbine (process: 19–15), non-isentropic expansion process is assumed in the turbine. A generator is connected with the turbine to convert the power into electricity and the efficiency of it is assumed as a constant value.

$$\eta_b = \frac{h_{19} - h_{15}}{h_{19} - h_{15s}}, \quad (5)$$

$$W_t = m(h_{19} - h_{15}), \quad (6)$$

$$W_{\text{net}} = W_t \eta_g - W_{p1}, \quad (7)$$

where n_b and n_g are respectively the efficiency of expander and electric generator, h_{15s} is the enthalpy by isentropic expansion, W_t is the output power of ORC system and W_{net} is net output power.

Condenser (process: 15–16), it is also the generator of ARS. The heat released in condensation process is used to heat ammonia water:

$$Q_c = m(h_{15} - h_{16}). \quad (8)$$

The thermal efficiency of ORC system η :

$$\eta = \frac{W_t - W_{p1}}{(h_{22} - h_{24})m_g}, \quad (9)$$

$$Q_{\text{in}} = (h_{22} - h_{24})m_g. \quad (10)$$

Table 2 Main parameters of operating conditions

| Parameters | Value |
|---|---------|
| Atmosphere pressure | 101 kPa |
| Atmosphere temperature | 298 K |
| cooling water temperature | 298 K |
| Efficiency of pump | 0.8 |
| Efficiency of electric generator | 0.8 |
| Efficiency of turbine | 0.8 |
| Condensing temperature of ARS | 306 K |
| Exhaust gas temperature at evaporator inlet | 805 K |
| Minimum temperature difference in the condenser of ARS | 5 K |
| Minimum temperature difference in the evaporator of ARS | 5 K |
| Minimum temperature difference in the absorber | 6 K |
| Minimum temperature difference in the pre-heater | 25 K |
| Minimum temperature difference in the evaporator of RC | 25 K |
| cooling water temperature increase in absorber | 5 K |
| cooling water temperature increase in condenser of ARS | 3 K |

3.2 Absorption cooling system

The absorption refrigeration system mainly includes eight components: generator, condenser, two expansion valves, evaporator, absorber, strong solution pump and SHX. The working process in each part can be described as follows: Generator (process: 7–18), the ammonia water solution absorbs heat from the steam after turbine in the generator which is also the condenser of RC. The energy balance can be expressed as:

$$m(h_{15} - h_{16}) = m_a h_1 + m_d h_8 - m_s h_7, \quad (11)$$

$$m_s = m_d + m_a, \quad (12)$$

$$m_s a = m_d b + m_a, \quad (13)$$

$$Q_c = Q_{\text{gen}} = m(h_{15} - h_{16}), \quad (14)$$

where m_s , m_d and m_a are respectively the mass flow rate of strong solution, dilute solution and pure ammonia. a is the concentration of strong solution and b is that of dilute solution.

Condenser (process: 1–2), in this condenser, the ammonia is condensed to liquid by cooling water. The energy balance and quality balance can be expressed as:

$$m_a h_1 - m_a h_2 = m_{cl} h_{12} - m_{cl} h_{11}, \quad (15)$$

where m_{cl} is the mass flow rate of cooling water in condenser.

Subcooler (process: 2–2'/4–4'), the subcooler can increase the cooling capacity of this system:

$$h_2 - h_{2'} = h_{4'} - h_4. \quad (16)$$

Expansion valve (process: 2'–3), the expansion valve is used to control the pressure and the mass flow rate of refrigerant. It is seen as an adiabatic process:

$$h_{2'} = h_3. \quad (17)$$

Evaporator 2 (process: 3–4), in this component, the liquid ammonia vaporizes, extracting heat from refrigerating medium and producing cooling power:

$$Q_{cool} = m_a (h_4 - h_3), \quad (18)$$

$$m_r = \frac{Q_{cool}}{h_{13} - h_{14}}, \quad (19)$$

where m_r is the mass flow rate of refrigerating medium.

Absorber (process: 4', 10–5), the ammonia vapor flows into the absorber and is absorbed by the weak solution:

$$m_a h_{4'} + m_d h_{10} + m_{c2} h_{20} = m_s h_5 + m_{c2} h_{21}. \quad (20)$$

Pump (process: 5–6), the strong solution is send to the SHX with pressure increase via a pump driven by electricity. The electricity consumed by Pump 2 can be expressed as:

$$W_{p2} = (m_s h_6 - m_s h_5) \eta_{p2}, \quad (21)$$

where η_{p2} is the efficiency of the pump.

SHX (process: 6–7, 8–9): SHX is also named economizer, which is used to improve the energy utilization rate of cooling cycle.

$$m_d h_8 - m_d h_9 = m_s h_7 - m_s h_6. \quad (22)$$

Coefficient of performance (COP) is commonly adopted as the key performance indicator of refrigeration cycle. For an absorption chiller, COP is defined as cooling capacity (Q_{cool}) divided by energy input at the generator (Q_{gen}) [21]:

$$COP = \frac{Q_{cool}}{Q_{gen}}. \quad (23)$$

3.3 The evaluation indicators of ECCS

There are many evaluation indicators of ECCS. Every indicator has its advantages and disadvantages, and they evaluate the ECCS from various aspects. Since the ECCS provides different kinds of energy, suitable evaluation indica-

tors should take account of both the quality and quantity of energy and the practicability at the same time. PER (Primary energy rate) is defined as the ratio of the energy output to the energy input [22]. In this ECCS system, it can be described as follow:

$$PER = \frac{W_{net} + Q_{cool} - W_{p2}}{Q_{in}}. \quad (24)$$

The difference of energy quality between cooling energy and electrical power is not considered in this indicator, so it is not very objective to evaluate the ECCS or CCHP (combined cooling, heat, and power). However, because of its simplicity and intuition, it has been used in many researches [23–25] and adopted as one of the widely used evaluation indicators.

Exergy efficiency is also used by many people [26,27]. According to ref. [28], exergy efficiency is generally defined as the exergy output divided by the exergy input:

$$\eta_{ex} = \frac{W_{net} + E_c - W_{p2}}{E_{in}}, \quad (25)$$

$$E_c = \left(\frac{T_0}{T_{ev}} - 1 \right) Q_{cool}, \quad (26)$$

$$E_{in} = ((h_{24} - h_{22}) - T_0 (s_{24} - s_{22})) m_g, \quad (27)$$

where E_c is the exergy associated with the cooling energy, E_{in} is the total exergy inputted in the cogeneration system, T_0 is the ambient temperature and T_{ev} is the evaporating temperature of cooling cycle. Although it distinguishes the quality of energy, this indicator is not applied very generally in engineering practice owing to the difficulty of calculation and the abstraction.

In the paper, the cooling capacity of absorption refrigeration is converted to corresponding electrical power consumed by electrical refrigeration system and it is called equivalent electrical power. In this way, the cooling energy and electricity have the same energy quality and then it is easy to compare the ECCS with basic RC. The expression formula of equivalent electrical power is

$$W_e = \frac{Q_{cool}}{COP_{ec}}, \quad (28)$$

where W_e is the equivalent electric power converted from cooling capacity. COP_{ec} is the performance coefficient of electric cooling system. So the total net output of the ECCS can be described as:

$$W_{enet} = (W_e - W_{p2}) + W_{net}, \quad (29)$$

where W_{enet} is called total equivalent electricity output. This method is intuitive and very easy to understand. Besides, it considers the difference of energy quality between cooling power and electricity. This thought also appeared in some

former researches [29,30]. Because different referenced electric cooling systems have different refrigerants and compressors, their coefficients of performance have some distinction and with the development of technology, the COP_{ec} will improve. However, for the general products of electrical refrigeration system on the present market, the differences is not very big. Therefore, it means that this method converts cooling power to electric power according to the modern average technological level of electrical refrigeration.

In the researches of the paper, the condensing temperature of the cooling system is constant as shown in Table 2. The COP_{ec} under different evaporation temperatures referred to a very common electrical refrigeration plant in China, which represents the average technological level as Table 3 [31].

It should be noticed that this method is more suitable for evaluate the ECCS used in the places where electricity and cooling are required at the same time. It has strong practicality.

4 Model validations

As there is no very suitable model to validate the whole cogeneration system, the RC and absorption refrigeration cycle are validated separately. The thermo physical properties of ammonia water were based on the Schulz's equation of state for the ammonia water and the validation of mathematical model was based on ref. [32]. The validation of RC and the absorption refrigeration cycle was based on refs. [33,34] respectively. They are shown as Tables 4 and 5. It can be seen that such a deviation is enough accurate for our investigation.

5 Results and discussion

The condensing temperature of RC is the most important parameter in the combined system and it appears a lot in this part. Besides, evaporating pressure of RC and ARS are also mentioned frequently. For simplicity, T_c , P_e , T_{ec} , represent condensing temperature of RC, evaporation pressure of RC and evaporating temperature of ARS respectively. As mentioned above, the basic condensing RC and the ECCS

Table 3 The COP_{ec} of a common electrical refrigeration plant under different evaporation temperature

| Condensing temperature (°C) | Evaporating temperature (°C) | COP_{ec} |
|-----------------------------|------------------------------|------------|
| 33 | 10 | 5.49 |
| 33 | 5 | 4.73 |
| 33 | 0 | 4.06 |
| 33 | -5 | 3.59 |
| 33 | -10 | 3.20 |

Table 4 Model validations of thermo physical properties of ammonia water

| Input parameters | | Saturation concentration of ammonia water | | Enthalpy of ammonia water (kJ/kg) | |
|------------------|-----------|---|-------|-----------------------------------|-------|
| T (°C) | P (kPa) | This paper | Ref | This paper | Ref |
| 0 | 50.7 | 0.384 | 0.365 | -231 | -238 |
| 20 | 50.7 | 0.261 | 0.249 | -90.6 | -96.6 |
| 40 | 50.7 | 0.147 | 0.148 | 56.7 | 59.1 |

Table 5 Model validations of RC and ABS

| Calculation results | Present study | Ref | Error |
|---------------------|---------------|-------|-------|
| η (RC) | 0.19 | 0.185 | 2.7% |
| W_{net} (RC) | 130.2 | 135.3 | 3.8% |
| COP | 0.66 | 0.65 | 1.5% |

are compared. For the basic RC system, T_c changes from 308 to 318 K; for the ECCS, it changes from 343 to 393 K, but the range varies at different T_{ec} within that limits, because the higher T_{ec} , the higher final generation temperature the ARS needs. All the condensing temperatures of RC in the ECCS and basic Rankine cycle are reference to the value applied in engineering practice. The net electricity output of basic condensing RC is marked as W_{bnet} .

Figure 4 describes the variation of refrigeration capacity (Q_{cool}) and net electricity output (W_{net}) of the ECCS with RC condensing temperature (T_c) at different RC evaporation pressure (P_e). The evaporation temperature (T_{ec}) of absorption cooling system is all 0°C. From this figure, it can be seen that electricity output decreases with the rising T_c nearly linearly, while refrigeration capacity increases at first slower and slower, and then drops down very slightly. Besides, W_{net} increases with P_e , while Q_{cool} is just the other way around. Refrigeration capacity and net electricity output have not only the different variation trend with T_c and P_e , but also different energy quality. Therefore, the performance of the whole ECCS cannot be seen very intuitively in Figure 4. In below content, the two forms of energy are converted to the same energy quality to evaluate the ECCS and the advantage of it can be shown obviously.

Figure 5 is the comparison between total equivalent electricity output (W_{enet}) of the ECCS and net electricity output (W_{bnet}) of the basic RC system with low condensing temperature at different T_c and T_e . T_{ec} is also 0°C in this figure. It should be noticed that the horizontal axis above is the T_c of ECCS and that below is the T_c of single condensing RC system, as well as the following figures. The figure shows that the total equivalent electricity output increases to a maximum value at first, and then drops down faster and faster. This is because when T_c is relatively low, equivalent electricity (W_e) of ARS increases faster than the loss of net electricity output (W_{net}). However, as refrigeration capacity increases slower and slower even begins to drop down, the

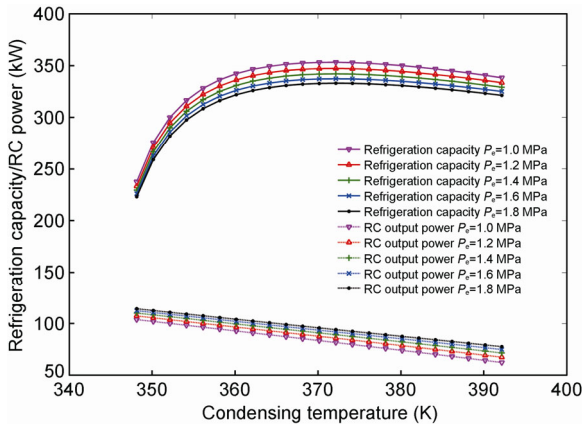


Figure 4 (Color online) The variation of Q_{cool} and W_{net} of the ECCS with RC condensing temperature at different RC evaporation temperatures.

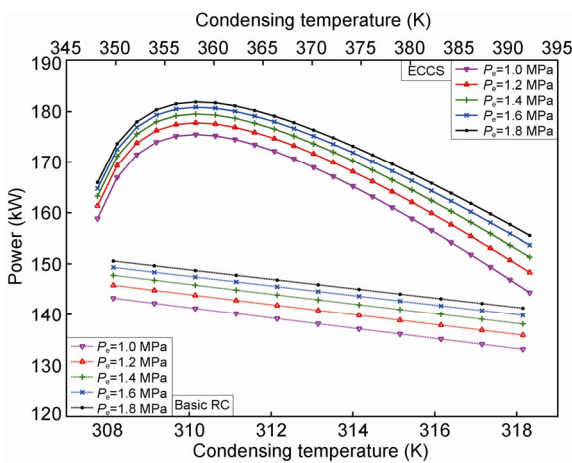


Figure 5 (Color online) The comparison between equivalent electricity of the ECCS system and electricity output of the single RC system at different T_c and T_{ec} .

reduction of W_{net} starts to exceed the augment of W_e , so W_{enet} decreases faster and faster. Consequently, there is an optimum T_c for the ECCS. From Figure 4, it can be known that Q_{cool} and W_{net} have the different variation trend with P_e , while Figure 5 shows W_{enet} increases with it. Its reason is very similar with the above. Therefore, increasing P_e can improve the performance of the whole ECCS. At the range of researched T_c and T_{ec} , W_{enet} is always greater than W_{bnet} . This indicates the better performance of ECCS than basic condensing RC. As a result, it is worth to increase the condensing temperature of the basic Rankin cycle to recover latent heat of condensation by absorption refrigeration system.

Figure 6 shows the effect of evaporating temperature of ARS on total equivalent electricity output and compares W_{enet} with W_{bnet} . With decreasing T_{ec} , the minimum T_c in ECCS increases, because the higher T_{ec} , the higher final generation temperature the ARS needs. As the figure illustrates, there is an optimum T_c of the largest W_{enet} . The largest W_{enet} becomes lower with increasing T_{ec} and it rises slightly at the same time. There are some factors for this phenome-

non: The increase of T_{ec} leads to the increase of pressure in absorber, making the strong solution denser. As consequence, the ratio of the concentration of strong solution and dilute solution gets greater, and then the mass flow rate of pure ammonia increases, which results in more cooling capacity. On the other hand, the rising T_{ec} leads to the enlargement of the COP_{ec} , which effects the calculation of W_e according to eq. (28). At last, W_{net} is not affected by T_{ec} . These factors bring out the results shown in Figure 6 together. What need remind is that even though there is an optimum T_c at every T_{ec} , considering about the actual requirement for cooling and electricity, maybe that much cooling or electricity is not required. Therefore, the actual demand should also be taken into account at the same time to choose the suitable operation parameters.

Compared to W_{bnet} , it can be known that the lower T_{ec} , the higher T_c is suitable for the ECCS and W_{enet} may be less than W_{bnet} under some operation conditions. Consequently, the operation parameters should be carefully chosen to avoid working under those bad conditions, otherwise the performance of ECCS will not be better than the basic RC with low condensing temperature.

Figure 7 shows the effect of T_c on final exhaust temperature (T_{24}) at the outlet of preheater. It can be seen that T_{24} rises with the increasing T_c . For the basic RC whose T_c varies from about 308 to 318 K, T_{24} is in the approximate range of 120–140°C, while for the ECCS whose T_c is much higher, T_{24} can reach nearly 180°C. As a result, there are more waste heat left that can be used in other aspects. For example, there is a type of absorption refrigeration equipment in the market with two kinds of different heat source like steam and exhaust gas. If the left heat in the exhaust can continue to be used in this equipment, W_{enet} will increase further.

In order to compare the energy utilization efficiency of the combined system with that of basic Rankin cycle further, the equivalent efficiency of the ECCS is defined as follows:

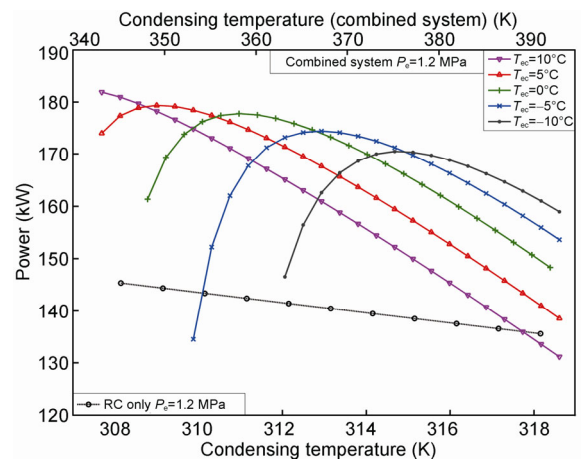


Figure 6 (Color online) The effect of evaporating temperature of ARS on total equivalent electricity output.

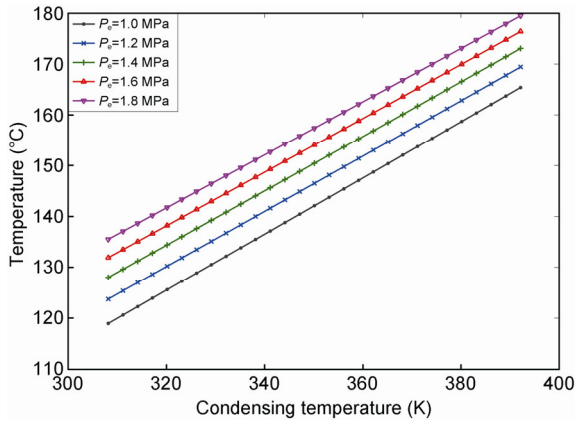


Figure 7 (Color online) The effect of T_c on final exhaust temperature (T_{24}).

$$\eta_e = \frac{W_{\text{enet}}}{Q_{\text{in}}} \quad (30)$$

Figure 8 shows the effects of T_c , T_{ec} , and P_e on η_e and compares η_e with the efficiency of basic condensing Rankin cycle. The variation trend of η_e is similar with that of W_{enet} , but the max η_e at different T_{ec} just have a very little difference. Because η_e is decided by W_{enet} and Q_{in} at the same time, although the max W_{enet} is larger at high T_{ec} , it appears at low T_c . The lower T_c , the more preheat quantity the RC needs, so Q_{in} increase as T_c drops down. These factors lead to the almost same max η_e at different T_{ec} . From Figures 5 and 8, it can be seen that increase of P_e can bring about not only more W_{enet} , but also higher η_e . Compared to the effi-

ciency of basic condensing RC, η_e is also larger in most operation conditions, so the ECCS can save more energy when electricity and cooling are required at the same time.

The exergy efficiency and primary energy rate are shown in Figures 9 and 10. Compared to Figure 8, it can be seen that equivalent efficiency and exergy efficiency have the similar variation trend but the value of exergy efficiency is greater. Both of them consider about the quality of energy: In the calculation of equivalent efficiency, cooling energy is converted to electric power according to the modern average technological level of electrical refrigeration as mentioned above, while the exergy of cooling energy is on the base of reverse Carnot cycle. In fact, if the electrical refrigeration system can work as the ideal reverse Carnot cycle, equivalent electric power converted from cooling capacity is the same with the exergy of cooling energy. This means the equivalent efficiency focuses on modern average technological level of electrical refrigeration, while the exergy efficiency considers about the limit condition. It should be noticed that in Figure 9 the exergy efficiency of basic Rankin cycle with low condensing temperature is greater than that of ECCS nearly in most situations. This is because the quality of cooling energy is much lower than electric power and the exergy of the output cooling energy is less than the reduction of output electricity power. Nevertheless, that doesn't mean the ECCS is not better, because in the places where both cooling and electricity are required, if we just produce electricity, the output electricity must be used to produce cooling, then there must be irreversible exergy loss and the exergy efficiency of the whole process will decrease.

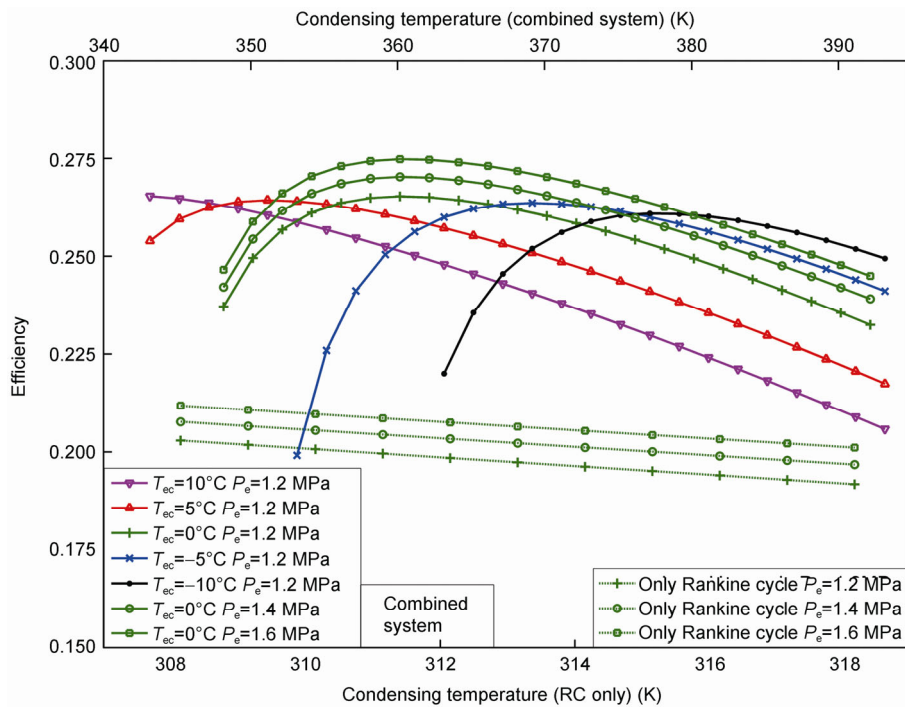


Figure 8 (Color online) The effects of T_c , T_{ec} , and P_e on η_e .

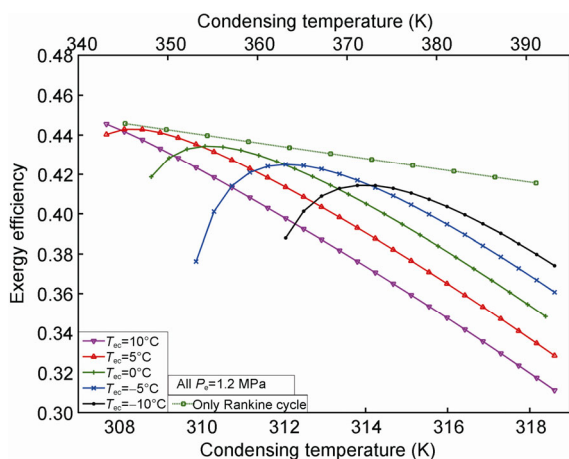


Figure 9 (Color online) The exergy efficiency of ECCS and basic Rankin cycle.

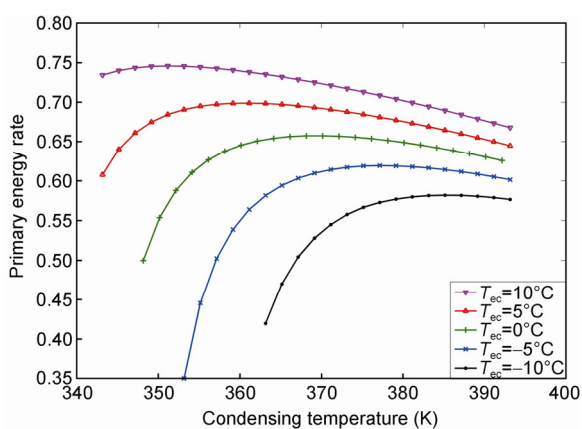


Figure 10 (Color online) The primary energy rate of ECCS.

However, the equivalent efficiency has included that irreversibility. Consequently, if cooling and electricity are required at the same time, the method of equivalent efficiency or efficiency is more suitable and more practical for engineering practice. Figures 5, 6 and 8 all show the advantages of ECCS.

The primary energy rate of ECCS is much higher than the other two kinds of efficiency. Compared to the efficiency of basic Rankine cycle with low condensing temperature in Figure 8, ECCS shows the better performance in all conditions, while Figures 8 and 6 indicate that ECCS is not always better. PER doesn't distinguish the quality of cooling and electricity, so it is not suitable to be compared with the efficiency of basic RC. In a word, it is not a very objective evaluation indicator for ECCS.

6 Conclusions

1) ECCS could obtain more than 180 kW total equivalent electricity that is about 18% of the rated electrical power of the gaseous fuel engines and have an equivalent efficiency of

more than 27.5% at most. However, the maximal output power and efficiency of basic RC is 150 kW and 21.5% respectively under the reasonable working condition. Therefore, it is worth using ECCS to recover the latent heat of condensation

2) The condensing temperature of RC in ECCS should not be too low or too high, which means there is a suitable range of T_c at every T_{cc} . Otherwise the ECCS doesn't have advantages compared to basic RC with low condensing temperature.

3) In the conditions when cooling and electricity are required at the same time, the method of equivalent efficiency or efficiency is more suitable and more practical for engineering practice than PER and the exergy efficiency,

This work was supported by the National Basic Research Program of China ("973" Project) (Gran No. 2011CB707201).

- 1 Ringler J, Seifert M, Guyotot V, et al. Rankine cycle for waste heat recovery of IC engines. In: 2009 SAE International Conference. Detroit, 2009
- 2 Shu G Q, Liang Y C, Wei H Q, et al. A review of waste heat recovery on two-stroke IC engine aboard ships. *Renew Sust Energy Rev*, 2013, 19: 385–401
- 3 Cho H M, He B Q. Spark ignition natural gas engines—a review. *Energ Convers Manage*, 2007, 48: 608–618
- 4 Imran A S, Emberson D R, Diez A. Natural gas fueled compression ignition engine performance and emissions maps with diesel and RME pilot fuels. *Appl Energ*, 2014, 124: 354–365
- 5 Ho J C, Chua K J, Chou S K. Performance study of a microturbine system for cogeneration application. *Renew Energ*, 2004, 29: 1121–1133
- 6 Papagiannakis R G, Hountalas D T. Combustion and exhaust emission characteristics of a dual fuel compression ignition engine operated with pilot diesel fuel and natural gas. *Energ Convers Manage*, 2004, 45: 2971–2987
- 7 Papagiannakis R G, Kotsiopoulos P N, Zannis T C, et al. Theoretical study of the effects of engine parameters on performance and emissions of a pilot ignited natural gas diesel engine. *Energy*, 2010, 35: 1129–1138
- 8 Li H. Application Study on Gas Engine-Based Cooling Heating and Power System. Dissertation of Doctoral Degree. Beijing: Tsinghua University, 2006
- 9 Qi T, Zhang Z Y. The application for the waste heat recovery of gas engines in oil field. *Energ Conserv*, 2006, 10: 52–54
- 10 Zhao G S. The research on the waste heat recovery of generating unit in Shi Wu power station. *Safty Health Environ*, 2011, 12: 28–30
- 11 Daniela G. Waste heat recovery from a landfill gas-fired power plant. *Renew Sust Energy Rev*, 2012, 16: 1779–1789
- 12 MAN Corporation. Technology Boosts Efficiency—WHR and TCS-PTG improve efficiency on large engines. Product manual, 2011. <http://www.entry.man.eu/de/de/index.html>
- 13 Wärtsilä Corporation. Waste heat recovery (whr): Fuel savings with less emissions. Product manual, 2007. <http://www.wartsila.com/en/Home>
- 14 Shu G Q, Liu L N, Hua T, et al. Analysis of regenerative dual-loop organic Rankine cycles (DORCs) used in engine waste heat recovery. *Energ Convers Manage*, 2013, 76: 234–243
- 15 Liang Y C, Shu G Q, Hua T, et al. Analysis of an electricity-cooling cogeneration system based on RC-ARS combined cycle aboard ship. *Energ Convers Manage*, 2013, 76: 1053–1060
- 16 Liang Y C, Shu G Q, Hua T, et al. Theoretical analysis of a novel electricity-cooling cogeneration system (ECCS) based on cascade use

- of waste heat of marine engine. *Energ Convers Manage*, 2014, 85: 888–894
- 17 Takeshi K, Kiyooki S, Hiroto N. Development of CNG fueled engine with lean burn for small size commercial van. *JSAE Review*, 2001, 22: 365–368
- 18 Badr O, Alsayed N, Manaf M. A parametric study on the lean misfiring and knocking limits of gas-fueled spark ignition engines. *PII. Appl Therm Eng*, 1998, 18: 579–594
- 19 Adewusi S A, Zubair S M. Second law based thermodynamic analysis of ammonia-water absorption systems. *Energ Convers Manage*, 2004, 45: 2355–2369
- 20 Mejbri K H, Bellagi A. Modelling of the thermodynamic properties of the water-ammonia mixture by three different approaches. *Int J Refrig*, 2006, 29: 211–218
- 21 Schulz. Equations of state for the system ammonia-water for use with computers. In: *The XIIth International Congress of Refrigeration*. Washington DC, 1971
- 22 Liu Q W. Performance studies on NH₃-H₂O absorption refrigeration HGAX cycles using low temperature exhaust heat. Dissertation of Doctoral Degree. Dalian: Dalian University of Technology, 2012
- 23 Havelsky V. Energetic efficiency of cogeneration systems for combined heat, cold and power production. *Int J Refrig*, 1999, 22: 479–485
- 24 Onovwiona H I, Ugursal V I. Residential cogeneration systems: Review of the current technology. *Renew Sust Energ Rev*, 2006, 10: 389–431
- 25 Wattana W S, Menkea C, Kamolpus D, et al. Study of operational parameters improvement of natural-gas cogeneration plant in public buildings in Thailand. *Energ Build*, 2011, 43: 925–934
- 26 Aguilar F J, Garcia M T, Trujillo E C, et al. Prediction of performance, energy savings and increase in profitability of two gas turbine steam generator cogeneration plant based on experimental data. *Energy*, 2011, 36: 742–754
- 27 Abusoglu A, Kanoglu M. Exergetic and thermoeconomic analyses of diesel engine powered cogeneration: Part 1. *Appl Therm Eng*, 2009, 29: 234–241
- 28 Kanoglu M, Dincer I. Performance assessment of cogeneration plants. *Energ Convers Manage*, 2009, 50: 76–81
- 29 Liu M, Zhang N. Proposal and analysis of a novel ammonia water cycle for power and refrigeration cogeneration. *Energy*, 2007, 32: 961–970
- 30 Xie N L. Studying of evaluation method of combined cold, heat and power system. *Build Energ Environ*, 2008, 27: 81–83
- 31 Jannelli E. Thermodynamic performance assessment of a small size CCHP (combined cooling heating and power) system with numerical models. *Energy*, 2014, 65: 240–249
- 32 Pu W. Refrigeration Technology and Equipment. Shanghai: Shanghai Jiaotong University Press, 2006
- 33 Dong J X. Exploitation of design calculation software for ammonia water absorption refrigeration system. Dissertation of Masteral Degree. Shanghai: Southeast University, 2000
- 34 Siddiqi M A, Atakan B. Alkanes as fluids in Rankine cycles in comparison to water, benzene and toluene. *Energy*, 2012, 45: 256–63
- 35 Mario D, Mateus H. Thermoeconomic assessment of an absorption refrigeration and hydrogen-fueled diesel power generator cogeneration system. *Int J Hydrogen Energ*, 2014, 39: 4590–4599