THEORETICAL AND EXPERIMENTAL ANALYSIS OF THERMAL ENERGY MANAGEMENT SYSTEM OF AIR SOURCE SELF-POWERED ELECTRIC GAS GENERATOR

by

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In order to solve the problem that the auxiliary equipment of electric gas turbine can operate only by relying on external power, and realize the purpose that auxiliary equipment of electric gas turbine can operate independently without the external power grid, in this research, a management system of air source self-powered electric gas generator is proposed. Firstly, the process of the thermal energy management system of the air source self-powered electric gas generator is introduced, and the thermodynamic theory of the thermal energy management system of the air source self-powered electric gas generator is analyzed. Then, the experimental conditions of air source self-powered electric gas generator are introduced. Finally, the results of variable speed and terminal variable flow in heating condition and terminal variable flow in cooling condition of the thermal energy management system of air source self-powered electric gas generator are analyzed. The results show that whether the thermal energy management system of air source self-powered electric gas generator studied in this research is in heating or cooling conditions, both the output power of the engine and the power of the compressor increase with the increase of the rotating speed. It can be concluded from the variable flow results in heating conditions that the smaller the end flow is, the smaller the output power of the engine will be. In this way, the loss of heat transfer efficiency of the plate can be reduced as much as possible, and the users' demand for heat can be met.

Key words*: electric gas turbine, thermal energy management system thermodynamic analysis, heating cycle, cooling cycle*

Introduction

With the development of industrialization, the demand for energy is increasing, and the global energy reserve is in a state of emergency. Although China is rich in coal resources, its coal consumption capacity is also quite large [1]. According to relevant statistics, since the 1950's, China's coal consumption has accounted for 90% of all energy consumption, and coal resources play an important role in China's energy structure [2]. With the promotion of global economic integration, the energy structure of many countries in the world has changed, but energy resources are still an important factor in the energy consumption structure.

The energy structure determines that China's energy consumption is still dominated by coal, which plays a very important role in the national economy. The electricity that is most

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closely related to people's livelihood is the use of coal thermal power generation, but the use of coal has also caused certain damage to the air environment in China, causing serious pollution the air [3, 4]. Therefore, in view of China's thermal power generation peak power shortage, surplus valley power contradiction, and environmental protection needs, it is urgent to find a kind of green energy to replace thermal power generation consume electricity during peak periods. Natural gas is one of the most suitable energy sources in line with this requirement, and China's natural gas reserves are very large, so gas-fired air conditioning has achieved considerable development in China [5]. However, if gas air conditioning needs to replace electric air conditioning comprehensively, it must have high efficiency, long service life, advanced control technology, low operating cost, heat extraction, dehumidification, drying and other characteristics. In order to effectively utilize natural gas, relevant researchers have conducted a lot of researches on it and the total energy system [6]. At present, most of the total energy system adopts cogeneration technology, but the gas-fired cold and hot triple power supply is developed on the basis of cogeneration. The gas-fired cold and hot electricity tri-connection technology is a comprehensive system with the ability to heat in winter and cool in summer using natural gas to drive the generator to generate electricity and recover the excess heat. The gas-fired combined cold, hot, and electric power supply system can realize the effective use of natural gas, alleviate the problem of power supply shortage, and provide safe and reliable electricity [7]. At present, the combined cold, hot, and electric power supply of gas-fired air conditioners is unstable [8]. In order to make the gas air conditioning get better application, it is urgent to solve the problem of unstable cold, heat, and electricity equipment and lack of system integration technology.

To sum up, in order to solve the problem that auxiliary equipment of ordinary gas pump still needs to rely on external power and its advantages cannot be exerted during the period of power supply shortage, in this research, the technology of an air source self-powered electric gas generator (ASSEGG) heat pump is studied. Firstly, the ASSEGG heat pump system is introduced and the gas turbine heat management system is analyzed. Then the experiment of gas heat pump with air source is introduced. Finally, the experimental results of variable speed in heating condition, variable flow rate in heating condition, and variable speed in cooling condition are analyzed. The purpose of this study is to provide a good experimental and theoretical basis for the application of gas air conditioning in China.

Methodology

The flow of the thermal energy management system of the ASSEGG

Under the cooling condition of the system, the gas engine drives the compressor to complete the cooling cycle and provide the required cooling quantity for the user. At the same time, waste heat from the gas engine can be recovered through the cylinder liner and flue gas heat exchanger to make hot water for domestic use. Under the heating condition of the system,

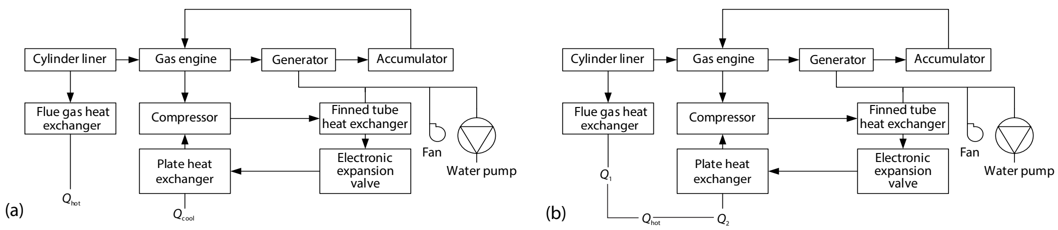

Figure 1. Schematic diagram of cooling effect and heating effect of the system; (a) the cooling process, (b) heating schematic diagram

the heat generated by the heat pump system and the waste heat recovered by the engine are both used for heating. Therefore, the gas engine heat pump system has the advantages of fast heating speed, high calorific value, and stable heating. Therefore, the system can really achieve the energy grade and grade utilization required by the whole energy system, and has higher energy efficiency. The cooling effect schematic diagram and heating effect schematic diagram of the system are shown in fig. 1.

Thermodynamic analysis of the thermal energy management system of the ASSEGG

According to the laws of thermodynamics, the ability to essentially convert to work can be regarded as efficient energy, called exergy. For stable flow system, stable flow enthalpy exergy can be expressed:

$$e_n = h - h_0 - T_0(s - s_0) \tag{1}$$

where e_n is the exergy of enthalpy of unit stable flow h – the common mass enthalpy value of evaporator outlet, h_0 – the environmental specific exergy, T_0 – the environment temperature, s – the enthalpy value of import, and s_0 – the specific entropy.

For an thermal energy management system (TEMS), its exergy balance equation can be expressed:

$$e_{\text{in}} = e_{\text{out}} + e_{\text{sh}} + \Delta e \tag{2}$$

where e_{in} and e_{out} are the exergy of input and output systems, e_{sh} – the exergy loss of the system, and Δe is the change of system exergy.

For common thermal equipment:

$$E_1 = E_{\text{pay}} - E_{\text{gain}} \tag{3}$$

where E_1 is the exergy loss, E_{pay} – the exergy paid or consumed, and E_{gain} – the benefits or exergy used.

According to the aforementioned equations, the mass-flow rate of the stable flow system is related to exergy and exergy loss, so exergy loss power can be defined:

$$W_t = E_t Q_m \tag{4}$$

where W_t is the exergy loss power, E_t – the exergy of natural gas entering the gas engine, and Q_m – the mass-flow rate of the stable flow system.

In the cooling cycle of the TEMS of the gas engine with the self-provided power source of the air source, the exergy loss power of the compressor is expressed as W_{comp}:

$$D_{\text{comp}} = T_{\text{cir}}(S_1 - S_2) \tag{5}$$

$$W_{\text{comp}} = D_{\text{comp}} M_{\text{ref}} \tag{6}$$

where D_{comp} is the exergy loss of finned tube heat exchanger, T_{cir} – the environment temperature under cooling conditions, S_1 and S_2 – are the speed of the gas engine, W_{comp} – the exergy loss power of the condenser, and M_{ref} – the mass-flow rate of the refrigerant.

Similarly, the exergy lost power of the condenser can also be expressed by eqs. (5) and (6).

Exergy loss power of throttle valve is shown:

$$D_{\text{thro}} = e_3 - e_4 = T_{\text{cir}}(S_4 - S_3) \tag{7}$$

$$W_{\text{thro}} = D_{\text{thro}} M_{\text{ref}} \tag{8}$$

where D_{thro} is the exergy loss of the throttle valve, e_3 – the exergy of outlet of finned tube heat exchanger, and e_4 – the throttle outlet exergy.

Evaporator exergy loss power:

$$D_{\text{evap}} = e_4 - e_1 - e_{Qo} = (h_4 - h_1) - T_{cir}(S_4 - S_1) - (T_{cir} - T_{in})(h_1 - h_4)/T_{\text{in}} \tag{9}$$

$$W_{\text{evap}} = D_{\text{evap}} M_{\text{ref}} \tag{10}$$

where W_{evap} is the evaporator exergy loss power, D_{evap} – the evaporator exergy loss, e_{Qo} – the system refrigeration exergy, h_4 – the enthalpy value of evaporator inlet working medium, h_1 – the enthalpy value of the working medium at the outlet of the evaporator, and T_{in} – the indoor temperature.

For different fuels, the exergy of the fuel input to the engine is different:

$$E_f = \begin{cases} H_{\text{solid}} \\ 0.975 H_{\text{liquid}} \\ 0.95 H_{\text{gas}} \end{cases} \tag{11}$$

where E_f is the exergy of natural gas entering the gas engine, H_{solid} – the high calorific value of the solid, H_{liquid} – the high calorific value of the liquid, and H_{gas} – the high calorific value of the gas.

The system in this research uses natural gas, so the fuel exergy input to the engine:

$$E_f = 0.95 H_{\text{gas}} \tag{12}$$

$$W_{\text{eng}} = D M_{\text{gas}} \tag{13}$$

Test conditions for TEMS of ASSEGG

In order to investigate the superiority of the igniter heat pump system designed in this research, three different experimental conditions of the gas engine heat pump system are carried out.

– Under the experimental condition of variable speed under heating condition, the control of each parameter is ensuring that the evaporation temperature (1 °C) and condensation temperature (46 °C) are the same at each speed, the engine speed range is adjusted to be 1000~3100 rpm, and the room temperature is 5 °C. The temperature control of the end water tank is controlled by mixing water, and the electronic expansion valve is controlled by superheat control. The superheat is maintained at 7~9 °C, the fan air volume is 20500 m^3/h, and the terminal water flow is 4.5 m^3/h; the circulating water flow of engine cooling water in the system is controlled at 3.750 m^3/h; the water flow of the defrosting hot water pipe-line is 1.80 m^3/h; the rotation ratio of the engine to the generator is 1.5; the ratio of engine to compressor is 0.7; the refrigerant used in the cooling system is R134a.
– Under the experimental condition of terminal variable flow under heating condition, the control of each parameter is: the flow of the end waterway is controlled by adjusting the ball valve of the end waterway, and the flow is, respectively 3.4 m^2/h and 4 m^2/h under the condition of each engine speed; engine speed ranges from 1300-2300 rpm; the evaporation temperature and condensation temperature of the system under variable flow conditions are shown in tab. 1.

Table 1. Evaporation temperature and condensation temperature of the system under variable flow conditions

Condensing temperature [°C]	36	46	46	45	43	43
Evaporating temperature [°C]	0	0	0	0	0	0
Rotating speed [rpm]	1300	1500	1700	1900	2100	2300

– Under the experimental condition of terminal variable flow under heating condition, the electronic expansion valve is adopted to control the superheat; the superheat is maintained at 7~9 °C, the fan air volume is 20500 m^3/h, and the terminal water flow is 7.0 m^3/h.

Results

Analysis on the results of variable speed in heating condition of the TEMS of the ASSEGG

Figure 2 shows the relationship between engine speed and power consumption, and fig. 3 shows the relationship between engine speed and fuel consumption rate. As shown in fig. 2, with the increase of engine speed, both the output power of the engine and the power of the compressor increase with the increase of engine speed. The speed has a linear relationship with the output power of the engine and the power of the compressor. When the engine speed is up to 3000 rpm, the maximum output power of the engine is 25 kW, and the maximum input shaft power of the compressor is 15 kW. At the same time, it can be concluded that the input power of the generator remains unchanged with the increase of speed, which is around 7 kW. As can be concluded from fig. 3, with the increase of engine speed, the fuel consumption rate, G_f, is in direct proportion the engine speed. However, it is too one-sided to analyze the engine's fuel consumption rate only from the perspective of fuel consumption rate, G_f. Therefore, the fuel consumption rate, G_e, is introduced in the experimental process. Since G_e includes the output power of the engine, the fuel consumption rate is analyzed through two consumption factors. According to the, G_e, curve in the figure, the fuel consumption shows a parabolic growth trend with the increase of rotating speed. The G_e is the lowest at 2000 rpm, thus indicating an economic engine speed of 2000 rpm. This result is also consistent with the specification parameters provided by the engine manufacturer. Therefore, the engine with 1 kW output at this speed has the least energy consumption and better economic efficiency.

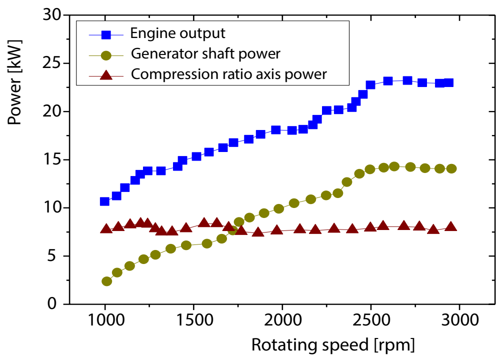

Figure 2. Diagram of speed and power

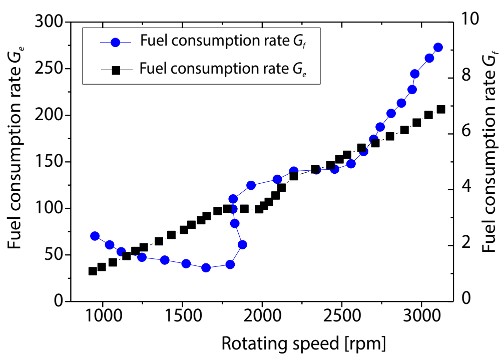

Figure 3. Diagram of the relationship between speed and rate of fuel consumption

Figure 4 shows the relationship diagram of heat transfer and waste heat recovery of the TEMS under different engine speeds. In the TEMS of the electric and gas engine studied in this research, the heat transfer of the liner heat exchanger is the least, which is because the water heat of the liner is constrained by the thermostat. The heat exchange of the main board increases with the increase of the speed, and the heat exchange of the main board at the speed of 3100 rpm is twice that of the low speed of 1000 rpm. Therefore, it can be concluded that the mainboard heat exchange and waste heat recovery of heat after the total heating heat is higher than natural heating, and the highest heat is up to 118 kW. The experimental data exceeds the designed heating capacity of 50 kW. As shown in the figure, the waste heat recovery heat changes with the engine speed. It can be concluded from the waste heat recovery that the heat changes from 13-45kW, accounting for about 32% of the total heat supply.

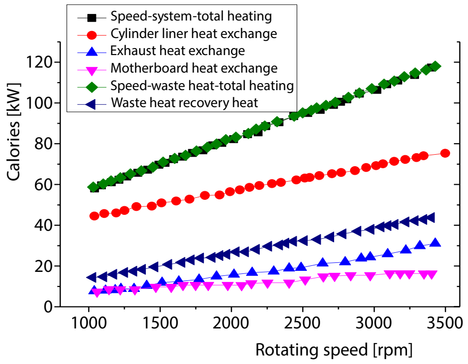

Figure 4. The relationship diagram of heat transfer and waste heat recovery of the TEMS under different engine speeds

Analysis on the terminal variable flow in heating condition of the TEMS of the ASSEGG

Table 2 shows the results of heat exchange of the main board with the change of rotating speed at different condensation temperatures. At the same rotating speed, the heat exchange of the main board at the flow rate of 4 m^3/h is greater than that at the flow rate of 3.4 m^3/h. In addition, with the increase of rotating speed, the difference of heat exchange becomes very obvious, and the difference of maximum heat exchange between different flows is close to 11% or so. This phenomenon indicates that in the actual operation process, under the premise of meeting the water exchange temperature of the main board, the water temperature in the main board will rise at the end of the water flow to achieve the purpose of large flow and small temperature difference. In this way, the loss of heat transfer efficiency of the plate can be reduced as much as possible to meet the user's demand of heat.

Table 2. The results of heat exchange of the main board with the change of rotating speed at different condensation temperatures

Rotating speed [rpm]	Heat exchange at different condensation temperatures [kW] of 4.3 m^3/h				Heat exchange at different condensation temperatures [kW] of 3.4 m^3/h			
	36 °C	46 °C	45 °C	43 °C	36 °C	46 °C	45 °C	43 °C
1300	72.4	72.7	71.1	71.5	71.3	71.5	68.2	68.1
1500	80.1	80.9	69.8	69.1	69.5	69.3	64.9	64.5
1900	52.3	52.7	52.4	52.4	52.6	52.0	50.3	50.2
2100	35.1	35.4	32.4	32.4	32.5	32.0	30.8	30.2

Table 3 shows the change results of residual heat recovery of liner water under variable flow condition. When the speed is less than 1900 rpm and the end flow rate is 3.4 m^2/h, the waste heat recovery heat at the end of the liner is greater than when the flow rate is 4 m^2/h. The main reason for this result is that the water flow at the end of the hour will make the heat transfer efficiency of the whole system lower, so that the compressor consumption of power increased.

Therefore, the engine needs to output more power to maintain the heat transfer efficiency of the whole system; when the speed is greater than 1900 rpm, the water temperature of the cylinder liner hardly changes.

Table 3. The change results of residual heat recovery of liner water under variable flow condition

Rotating speed [rpm]	Heat exchange at different condensing temperatures at different flow rates [kW] of 4.3 m³/h				Heat exchange at different condensing temperatures at different flow rates [kW] of 3.4 m³/h			
	36 °C	46 °C	45 °C	43 °C	36 °C	46 °C	45 °C	43 °C
1300	73.1	71.8	71.4	72.5	70.5	71.6	67.9	68.4
1500	80.5	81.6	69.2	68.9	69.4	69.2	64.7	64.7
1900	51.6	53.2	51.9	52.3	53.3	52.7	50.7	50.1
2100	34.1	35.2	32.6	32.3	32.1	32.4	30.6	30.5

Analysis on the terminal variable flow in heating condition of the TEMS of the ASSEGG

Figure 5 shows the relationship between engine speed and power under cooling condition, and fig. 6 shows the relationship between engine speed and fuel consumption rate. As shown in fig. 5, with the increase of engine speed, both the output power of the engine and the power of the compressor increase with the increase of engine speed. The speed has a linear relationship with the output power of the engine and the power of the compressor, while the input power of the generator remains unchanged with the increase of the speed, around 7 kW. As shown in fig. 6, the fuel consumption rate G_f and the fuel consumption rate G_e are both smaller than the fuel consumption rate under heating conditions. The main reason is that the condenser under cooling condition is air-cooled, and the condensing effect is not ideal, which makes the heat pump system work in a bad state and increases the compression power consumption of the compressor. It shows that the economic benefit is best when the fuel consumption rate G_e is 2400 rpm. The experiments of heating and cooling verify that the ratio of engine to compressor is correct. Therefore, the engine speed of 2400 rpm and the compressor speed of 1450 rpm can be taken as the setting standards in the setting of the transmission ratio.

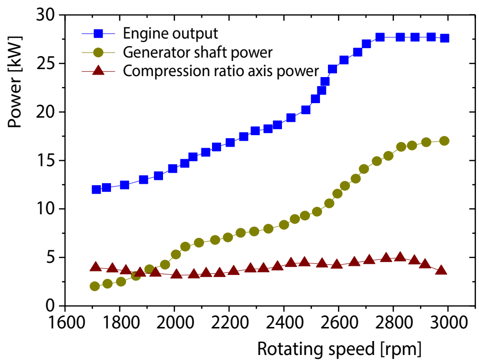

Figure 5. The relationship between engine speed and power in cooling condition

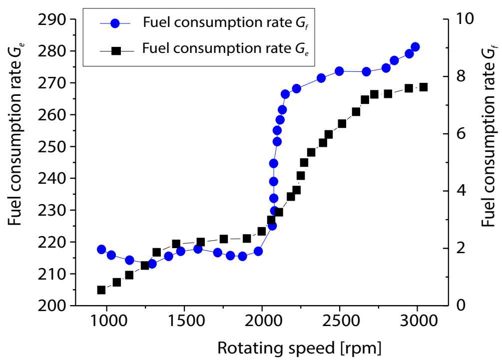

Figure 6. The relationship between engine speed and fuel consumption in cooling condition

Discussion

In this research, in order to realize the independent operation of auxiliary equipment of the TEMS of the electric gas engine without the help of external power supply, a TEMS is proposed. Its theory and experiment are analyzed. Chen and Cai [9] studied the use of gas engines and electric motors, analyzed the influence of transmission mechanism on operating characteristics, and proposed an instantaneous optimal control strategy aimed at minimizing equivalent energy consumption achieve torque distribution of continuously variable transmission. The results show that under the specified heating conditions, the compressor speeds of 1350 rpm and 1800 rpm are the switching points of the three operating modes. Besides, the engine torque and gas consumption rate are kept relatively stable, which can realize the efficient operation of the engine and improve the partial load performance of the air conditioning system. In this research, the engine speed of 2400 rpm and the compressor speed of 1450 rpm can be used as the setting standards, which can reflect the economic benefits of the gas engine. Wang *et al.* [10] studied the influence of mass-flow of high temperature cooling water on dual-loop organic Rankine cycles performance by analyzing the static performance and dynamic behavior of five typical engine conditions. The results show that the recovery of waste heat is reduced from 100-60%, and the efficiency of the combined system is improved well under engine conditions, showing good adaptability under engine conditions. In addition, increasing the mass-flow of cooling water can improve the output power of DORC system, but the effect is not obvious. It is concluded that the terminal flow can reduce the loss of heat transfer efficiency of the plate. Zhang *et al.* [2] proposed a new combined cooling, heating, and electricity generation system, in which compressed air energy storage absorbs off-peak electric energy from the grid. High pressure air is used for the combustion of biogas produced in the process of biomass gasification, and waste heat is utilized by absorption chiller and ground source heat pump. Under the simulation conditions, the round-trip efficiency and exergy efficiency are 90.06% and 31.52%, respectively, [11].

Conclusion

In order to realize the purpose that auxiliary equipment of electric gas turbine can still work without external power, the TEMS of ASSEGG is introduced. Firstly, the process of TEMS is introduced and thermodynamic theory of TEMS of ASSEGG is analyzed. Then, the test conditions of the TEMS for ASSEGG are introduced. Finally, the results of variable speed under heating condition and cooling condition are analyzed. The results show that with the increase of engine speed, both the output power of the engine and the power of the compressor increase with the increase of speed, while the input power of the generator basically remains unchanged with the increase of speed. The fuel consumption shows a parabolic growth trend with the increase of rotating speed. When the rotating speed is 2000 rpm, it has better economic efficiency. The analysis of the variable flow at the end of the system under heating condition shows that the smaller the end flow is, the smaller the output shaft power of the engine is, which can reduce the loss of heat transfer efficiency of the plate and meet the user's heat needs.

This study provides a good theoretical and experimental basis for the promotion of electric gas turbine in China. However, the research still has certain limitations, and the subsequent research needs to further study the waste heat recovery and waste heat utilization of the system in the cooling condition, so as to expand the depth of the research.

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