Enhancing Micro Gas Turbine Performance by Inlet Air Cooling Using Ejector Refrigeration

マイクロガスタービン（MGT）は有望な分散型発電システムだが、外気温が上がると発電効率が落ちる。この欠点を克服する冷却方法を検証した。

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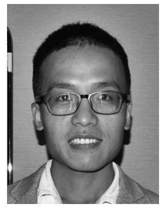

Abstract

Micro gas turbine (MGT) is attracting a lot of interest in the distributed generation (DG) market. But there is a significant limitation on the application of MGTs; the performance of MGT is strongly sensitive to ambient conditions, and its output and generating efficiency will drop with the increase of the ambient temperature. This paper presents a novel inlet air cooling method by ejector refrigeration driven by waste gas heat from the MGT, enhancing the performance of MGT under high temperature condition. First, the influence of inlet air temperature on the micro gas turbine is analyzed. The experimental results show power output decrease 0.47kW/K. A thermodynamic simulation model of the recuperated gas turbine at full condition is developed with Matlab/Simulink software. Heat recovered from the exhaust gas drives the ejector refrigeration to cool the inlet air, the performance of the ejector refrigeration is obtained through numerical calculation under different conditions. Cooling effect comparison between evaporative and refrigerated cooling methods was carried out under different meteorological conditions. Results show this novel inlet air cooling method through jet refrigeration driven by waste heat has obvious advantage under high temperature and heavy humidity environment condition.

Keywords micro gas turbine, Matlab/Simulink, inlet air cooling, ejector refrigeration

1. Introduction

Micro gas turbine (MGT) is relatively new technology that is attracting a lot of interest in distributed energy system (DES) with the electrical output ranging from 25kW~500kW[1-4]. As a prime mover in DES, even if MGTs perform low electrical generation efficiency compared with reciprocating engines (ICEs), MGTs offer a large number of advantages like, short construction period, fast response, high power density, low environment compact, low operation and maintenance costs and multi fuel capability. The use of micro turbine is considered a very attractive option in cogenerations system, to meet both electrical and thermal energy needs of residential and non-residential buildings. The performance of MGT largely depends on environment; high ambient temperature limits the air mass intake and thus leads to the reduction of power output and electrical efficiency, compared with the ISO conditions of 1.01 bar pressure, 288K, 60RH[5]. Typically, a IK rise in the ambient temperature will drop 0.5%~0.9% power output on medium/large GTs[6-9]. The electrical power output of MGTs is shown to larger decrease with ambient temperature at a rate of about 1.22%/K[10-12], due to a reduction of both air density and volumetric flow. Inlet air cooling techniques have been studied and applied to reduce its influence, several methods are available[13]: (i) wetted media evaporative cooling (ii) high-pressure fogging (iii) refrigeration cooling (iv) absorption chiller cooling. Evaporative cooling methods are most effectively applied in hot and dry areas to cool the ambient temperature near to the wet-bulb temperature. Refrigeration cooling methods usually use the mechanical chiller, which are relatively simple and reliable in design and operation but require large electric power. Absorption cooling can recover energy from the GTs exhausts but are complex systems requiring expertise in design, operation and maintenance. Compared with traditional vapor compression refrigeration system, jet refrigeration has a good advantage of simplicity in construction, installation and maintenance. Moreover, ejector refrigeration system can be driven by low grade heat without consuming mechanical energy[14].

In this study, it is proposed to cool the intake air through the jet refrigeration driven by waste gas heat from the micro gas turbine. The paper is organized as follows: the MGT C30 Simulink model under analysis is presented and the effect of temperature on its performance is discussed; reports the design of the jet refrigeration system and the description of the refrigeration performance; illustrates the applied results of this cooling method; finally reports the concluding remark.

2. Description of the Inlet Air Cooling System

The schematic of this novel MGT inlet air cooling system is presented in Fig.1. This system consists of two parts: jet refrigeration system driven by heat recovered from the fuel gas and micro gas turbine C30 system. Air is compressed in compressor and passes through the regenerator. The fuel is injected into the combustion chamber and ignition occurs. The fuel gas passes through the turbine blades and produce about 28 kW power. The hot fuel gas increase the refrigerant temperature and transform it into saturated steam to drive the jet refrigeration system to work. The inlet air pass through the refrigeration evaporator part as air cooler before gas turbine intake reducing temperature and enhancing the performance of MGT under high ambient temperature condition.

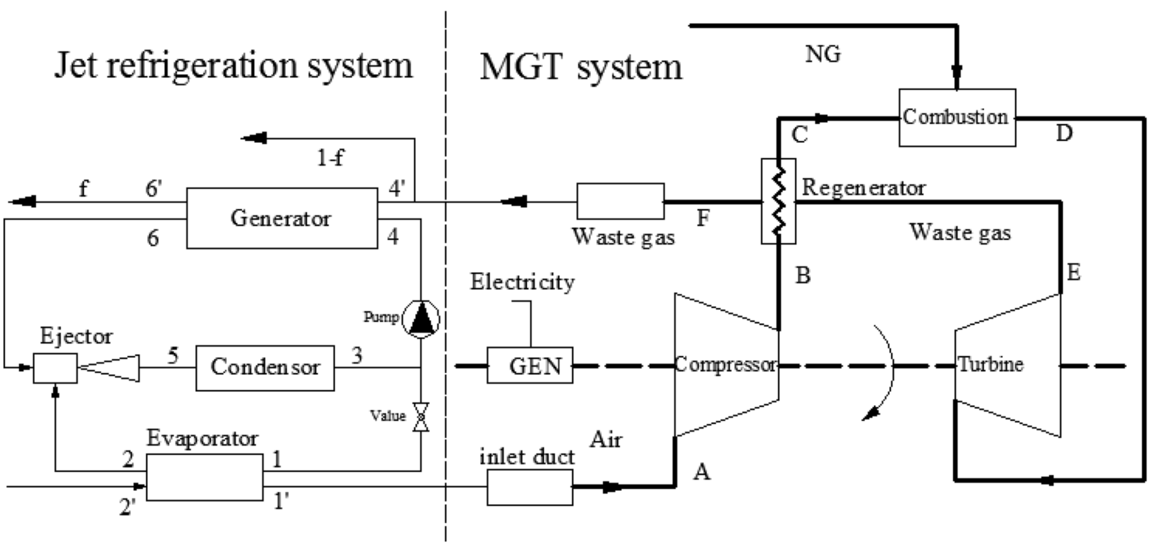

Fig.1 Schematic of ejector refrigeration inlet air cooling driven by waste heat from MGT

3. The MGT system

3.1 MGT Plant C30

The machine chosen for the analysis was a Capstone C30 based on a regenerative Brayton cycle. This system was divided into five main parts: compressor, regenerator, combustion, turbine, rotor. Each component is described individually.

Table 1 Parameters of MGT C30

Fuel	Natural gas
Power	28±2kW
Electrical Efficiency	26%
Exhaust Temperature	275°C
Exhaust Gas Flow	0.31kg/s
Net Heat Rate LHV	13.8 MJ/kWh

Nominal full power performance at ISO conditions: 59˚F, 14.696 psia, 60% RH

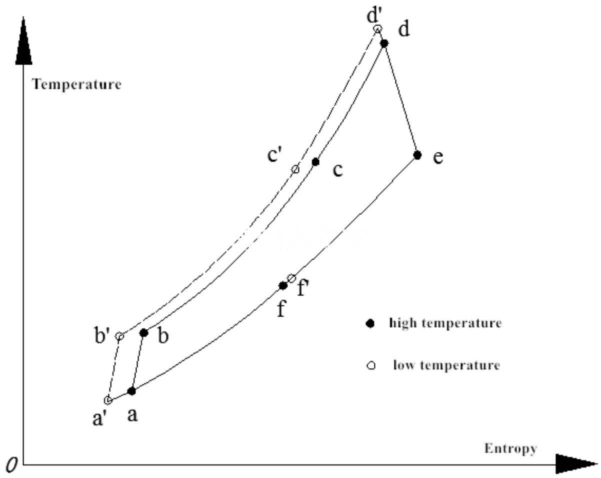

Fig.2 T-s diagram of micro gas turbine under different ambient temperature

The effects on the entropy-temperature diagram of the micro gas turbine utilizing a regenerator is presented in Fig 2. Air at ambient conditions is drawn into the compressor, where its temperature and pressure are raised (*a-b*). The high-pressure air leaving the compressor can be heated by transferring heat to it from the hot exhaust gases in regenerator (*b-c*). The air proceeds into the combustion chamber, where the fuel is burned at constant pressure (*c-d*). The resulting high-temperature gases then enter the turbine, where they expand to the atmospheric pressure while producing power (*d-e*). The exhaust gases leaving the turbine are thrown out (*e-f*). As ambient temperature T_a decreases the cycle (*a'-b'-c'-d'-e-f'*) is shown in Fig 2, the MGT power output increases.

3.2 Gas Turbine Model Analysis and simulation

The micro gas turbine C30 was developed with Matlab/Simulink software. The each component's mathematical model and solving procedure are described as follows.

(1) Compressor

The centrifugal compressor in design condition was modelled using the nominal pressure ratio β_c, the air mass flow rate m_a, inlet air temperature T_a, air adiabatic exponent k_a, air specific heat capacity $c_{p,a}$,=1.004 $kJ/(kg{\cdot}k)$. The temperature T_b at the out of the compressor and the power consumption P_c can be caculated by means Eqs (1) and (2)

$$T_b = T_a \beta_c^{\frac{k_a - 1}{\eta_{y,c} k_a}} \quad (1)$$

$$P_c = m_a c_{p,a} (T_b - T_a) \quad (2)$$

Some modifications are needed to describe the off-design behavior of the compressor, the following performance formulas for compressor are proposed.

$$\bar{\beta}_c = \frac{\beta_c}{\beta_{c,0}} \qquad \bar{m}_a = \frac{\dfrac{m_a\sqrt{T_1}}{p_1}}{\dfrac{m_{a,0}\sqrt{T_{1,0}}}{p_{1,0}}} \quad (3)$$

$$\bar{n}_c = \frac{\dfrac{n}{\sqrt{T_1}}}{\dfrac{n_0}{\sqrt{T_{1,0}}}} \qquad \bar{\eta}_{y,c} = \frac{\eta_{y,c}}{\eta_{y,c,0}} \quad (4)$$

The air mass flow can be calculated by equations:

$$\bar{\beta} = c_1 \bar{m}_a^2 + c_2 \bar{m}_a + c_3 \qquad (5)$$

$$\bar{\eta}_{y,c} = \left[1 - c_4 (1-\bar{n})^2 \right] \frac{n_c}{\bar{m}_a} \left(2 - \frac{n_c}{\bar{m}_a}\right) \qquad (6)$$

where

$$c_1 = \frac{n_c}{p(1-\frac{m}{\bar{n}_c}) + \bar{n}_c(\bar{n}_c - m)^2} \qquad (7)$$

$$c_2 = \frac{p - 2m n_c}{p(1-\frac{m}{\bar{n}_c}) + \bar{n}_c(\bar{n}_c - m)^2} \qquad (8)$$

$$c_3 = -\frac{pm\bar{n}_c - m^2 \bar{n}_c^3}{p(1-\frac{m}{\bar{n}_c}) + \bar{n}_c(\bar{n}_c - m)^2} \qquad (9)$$

$$c_4 = 0.3 \qquad (10)$$

Typical value for compressor in design condition β_c=4.0, inlet air temperature $T_{1,0}$ =288.15K, air mass flow $m_{a,0}$=0.31kg/s, compressor efficiency $\eta_{y,c,0}$ =0.80, p=1.8, m=1.8[15,16].

(2) Turbine

The parameters of radial turbine are: inlet pressure P_d, inlet temperaature T_d, turbine pressure ratio β_t, gas mass flow m_f.

Turbine pressure ratio :

$$\beta_t = \eta_p \beta \qquad (11)$$

Power output of turbine:

$$w_t = m_f c_{p,f} (T_d - T_{ge}) \qquad (12)$$

Outlet temperature of turbine:

$$T_{ge} = \frac{T_d}{\beta_t^{\frac{\eta_{y,t}(k_f - 1)}{k}}} \qquad (13)$$

η_p is the pneumatic efficiency of the system, which takes into account the pressure drops due to combustion and regenerator, η_t is the hydraulic efficiency. Typical data value are assumed $c_{p,f}$=1.091$kJ/(kg{\cdot}k)$

Some modifications are needed to describe the off-design behavior of the turbine, the following performance formulas for turbine are proposed.

$$\bar{m}_f = \frac{\frac{m_f\sqrt{T_d}}{p_d}}{\frac{m_{f,0}\sqrt{T_{d,0}}}{P_{d,0}}} = \sqrt{1.4 - 0.4\frac{n}{n_0}}\,\frac{p_{d,0}}{p_d}\sqrt{\frac{\beta_t^2 - f}{\beta_{t,0}^2 - f}} \qquad (14)$$

$$\bar{\eta}_{y,t} = \left[1 - s(1-\bar{n}_t)^2 \right] \frac{\bar{n}_t}{\bar{m}_f}\left(2 - \frac{\bar{n}_t}{\bar{m}_f}\right), \quad \bar{n}_t = \frac{\frac{n}{\sqrt{T_d}}}{\frac{n_0}{\sqrt{T_{d,0}}}} \qquad (15)$$

Commonly f=4.0, s=0.3[15,16].

(3) Combustion chamber

The parameters of interest for the combustion chamber are: air inlet temperature T_c, fuel mass flow m_b,air and fuel mass ratio α, energy conservation are used to caculate the outlet temperature T_d.

$$T_d = T_c + \frac{\eta_b LHV_b}{(1+\alpha)c_{p,f}} \qquad (16)$$

$$\alpha = m_a / m_b \qquad (17)$$

$$w_b = m_b LHV_b \qquad (18)$$

(4) Regenerator

The inputs for this modeling block are: inlet exhaust gas temperature T_{ge}, inlet air temperature T_b, ε_{re} is the efficiency of the regenerator, T_c is the outlet temperature of the air exhaust from the regenerator.

$$T_c = \varepsilon_{re}(T_{ge} - T_b) + T_b \qquad (19)$$

(5) Rotor

The electrical power output and efficiency can be caculated by means of equation (21) and (22). P_t, P_c and P_m are the mechanical power produced by turbine, the mechanical power required by compressor and power deliverd to shaft, and $\eta_{el,PG}$, η_{el} are the power generator efficiency and electrical efficiency respectively.

$$P_m = P_t - P_c \qquad (20)$$

$$P_m = \frac{P_{el}}{\eta_{el,PG}} \qquad (21)$$

$$\eta_{el,g} = \frac{P_{el}}{P_b} \times 100\% \quad (14) \qquad (22)$$

Table2 List of data input for the model

Parameter	U.m.	Value	Source
β	-	4.0	Datasheet
η_c	-	0.80	0.80[17] 0.82[15-16]
η_t	-	0.82	0.83[17] 0.85[15-16]
η_b	-	0.99	0.99[15-18]
η_p	-	0.89	0.89[15,19]
$\eta_{el,PG}$	-	0.86	0.85[15]0.87[16]
ε_{re}	-	0.82	Datasheet
α	kg/kg	135.30	Datasheet
m_b	kg/s	2.24×10-3	Datasheet
V	m³/h	12.0	Datasheet
LHV_b	MJ/kg	49.24	Datasheet

Condition 101.03kPa, 293.15k, natural gas density 0.6733kg/m^3, Composition of the natural gas : N_2-2.1579, O_2-0.4094, C_1-94.0917, C_2-1.5142, CO_2-0.0431, C_3-0.6353, i-C_4-0.2876, n-C_4-0.3753, i-C_5-0.1651, n-C_5-0.1914, i-C_6-0.0373, n-C_6-0.0193, i-C_7-0.0240, n-C_7-0.0065,ΣC_8 -0.0408.

Air specific heat capacity [32]

$$c_{p_a} = \frac{8.314}{28.97}(3.653 - 1.337\times10^{-3} T_{av} + 3.294\times10^{-6} T_{av}^2 - 1.913\times10^{-9} T_{av}^3 + 2.763\times10^{-13} T_{av}^4)$$

A Simulink model was developed with the thermodynamic cycle process and parameters of each component. The model shown in Fig.3 consists of compressor, turbine, regenerator, combustion and generator. The aim is to show the effect of inlet air temperature on C30 power output by real time input from 0 to 45 represent the simulation inlet air temperature value.

Simulation performance results for the inlet air temperature slow increase for the MGT C30 are represented in Fig.4-5, the comparison of the silmulation data and experimental results under different inlet air temperature, Fig6.

Fig.6 reports the electrical power output as a function of the inlet air temperature. The blue dots refer to the experimental data carried out under different ambient conditions. The black line refers to the simulation results. It is clearly visible in the data series showing power output decrease trend with the increasing inlet air temperature. Experiment data shows that about 0.47kW power output drop for every 1 K temperature increase. Simulation data has a similar trend of 0.23kW/K. Experimental data shows inlet air temperature has a greater effect on power output than simulation condition. Micro gas turbine C30 output decreases sharply under the high ambient temperature, power can only reach 18.71kW at the inlet air temperature of 38.50 °C, about 35.50% reduction compared with the ISO condition.

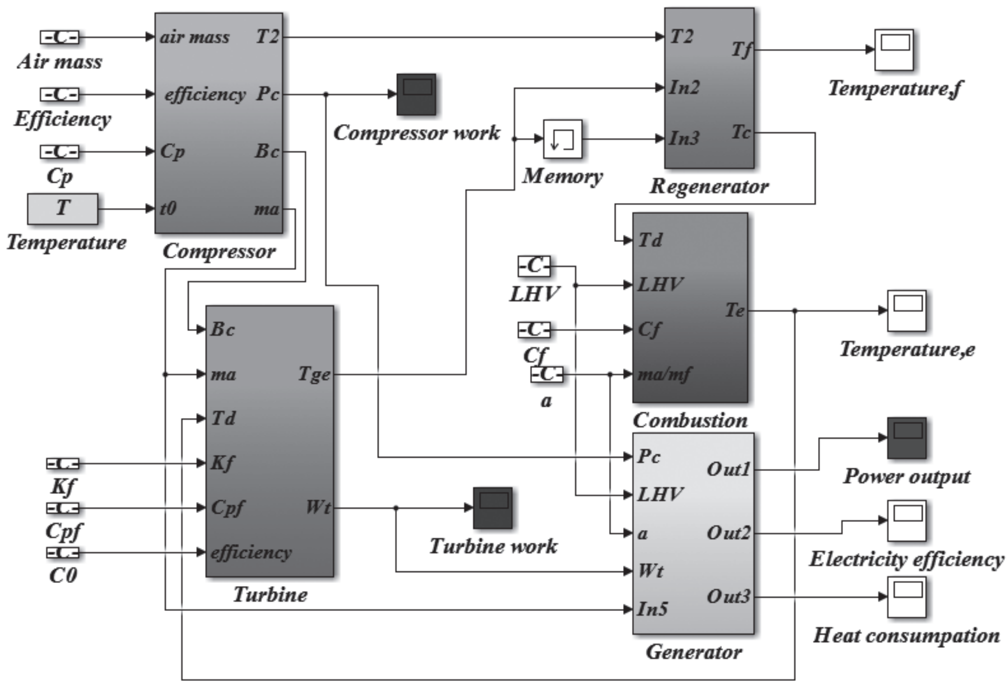

Fig.3 Simulation model of MGT C30 built by Simulink

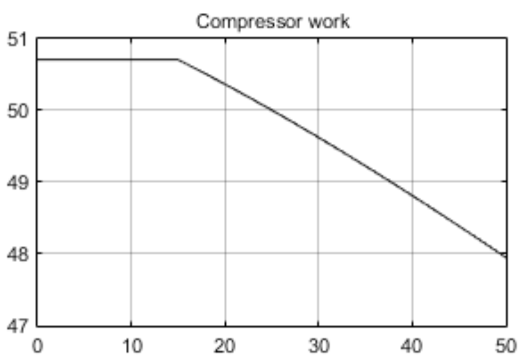

Fig.4 C30 Compressor simulation power consumpation

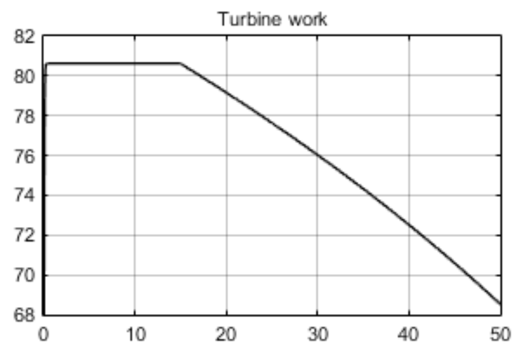

Fig.5 C30 turbine power output simulation

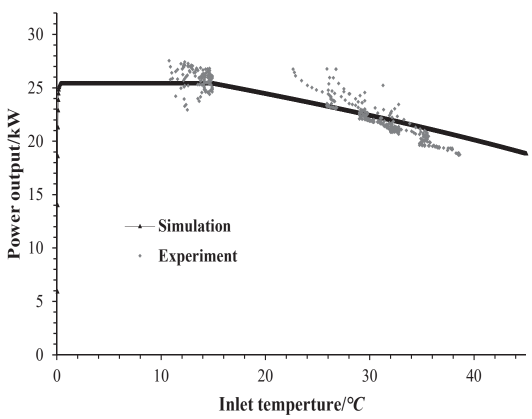

Fig.6 Comparison of inlet air temperature influence on micro gas turbine performance by simulation and experiment data

4. The Ejector Cooling System

There are three main types of heat-driven cooling technologies: absorption, adsorption and ejector refrigeration. Jet refrigeration has a good advantage of simplicity in construction, installation and maintenance, which can be driven by low grade heat source such as solar and waste industrial heat and enables the reduction of the mechanical work requirement. [19,20].

4.1 Description of the Ejector Refrigeration Cycle

A ejector refrigeration system employs an ejector to fulfill the function of a compressor, which can be seen in Fig.7. This mainly consists of four parts; primary nozzle, suction chamber, mix chamber and diffuser. The high pressure hot refrigerant gas from the generator enters the primary nozzle, expands, accelerates and reaches the supersonic state. At the exit of the nozzle, the high velocity gas enters the suction chamber, creating an area of low pressure at the secondary entrance of the ejector into which the fluid from the evaporator is entrained. The two fluid begin to mix in the mixing chamber undergoes a shock and increase the static pressure and become subsonic. Then the mix fluid flows into the diffuser with an additional pressure lift, the flow finally discharges into the condenser and becomes fluid state.

An ejector refrigeration cycle is a thermo-compressor cycle, the compressor effect is achieved using low grade heat source supplied to generator. Fig.8 shows a schematic diagram of the jet refrigeration system, consisting of ejector, condenser, evaporator, generator and a circulation pump. As is shown in Fig.8 the mixed fluid discharged from ejector becomes condenser fluid by rejecting heat to environment. One part of the mixed fluid enter the evaporator after passing through the throttle value, where it evaporates and produce the refrigerating effect (3-1-2). The other part is lifted to the generator via the pump and be vaporized by the delivered heat energy then the high pressure vapor enter into the ejector again (3-4-5). The two parts of fluid mix in the ejector, complete the cycle.

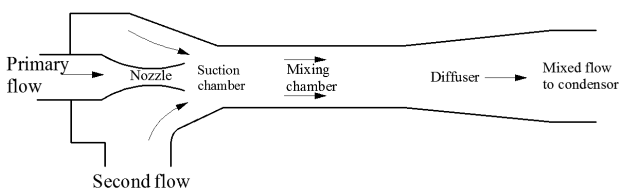

Fig.7 Schematic view of ejector structure

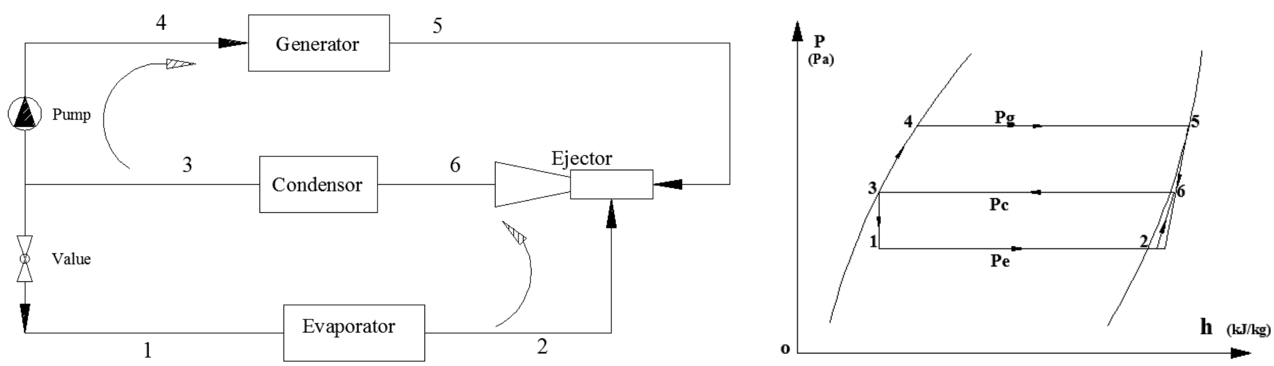

Fig.8 Ejector refrigeration cooling cycle

4.2 Mathematical Modeling and Performance Criteria

The coefficient of performance (COP) can be defined as the ratio of the refrigerating capacity to the heat supply to the generator

The cooling capacity obtained at the evaporator is determined as follows:

$$Q_e = m_e(h_2 - h_1) \quad (23)$$

The heat input to the generator:

$$Q_g = m_g(h_5 - h_4) \quad (24)$$

Performance of the ejector is evaluated by the entrainment ratio, defined as mass ratio of the secondary to primary flow rates:

$$\omega = \frac{m_e}{m_g} \quad (25)$$

The coefficient of performance of the jet cycle is calculated as the thermal ratio of cooling capacity over generator capacity:

$$\text{COP} = \frac{Q_e}{Q_g + W_p} \quad (26)$$

Neglecting the pump work, the COP of the refrigeration cycle is:

$$\text{COP} = \frac{Q_e}{Q_g} \quad (27)$$

The COP can be defined as:

$$\text{COP} = \omega \frac{h_2 - h_1}{h_5 - h_4} \quad (28)$$

The ejector performance simulation is carried out based on the one-dimensional constant pressure model. Huang[22,23] postulated a one dimensional model to analyze the ejector performance, which shows a good agreement with the experimental data. R141b shows a good performance compared with most refrigerant, which have been certified by many researchers[22,23,24,25], in addition the condenser pressure of R141b is closely to the ambient condition and we selected R141b as the work refrigerant. To obtain the system COP, a computational procedure based on C++ with iteration process was developed to calculate the entrainment ratio ω and the refrigerant properties were taken from the NIST database[26]. In the resent analysis, efficiencies of primary nozzle, secondary nozzle and diffuser were selected as η_p=095, η_s=085, η_d=085 suggested[22].

Fig.9 (a)-(b) show the effects of evaporator temperature on entrainment ratio ω and COP under different generator temperature with a condenser temperature 30 °C. An increase in the T_e leads to a rise in both ω and COP. This is because refrigerant R141b saturated evaporation pressure increases with the corresponding evaporator temperature increasing, leads to the increase of the mixed flow pressure in ejector mix chamber. The rise of pressure difference between the mixing chamber and the diffuser outlet, leading to a higher entrainment ratio ω and improving the coefficiency of refrigeration performance.

Fig.10 (a)-(b) show the effects of condenser temperature on entrainment ratio ω and COP under different evaporator temperature with a generator temperature 90 °C. An increase in the T_c leads to a decrease in both ω and COP. This is because refrigerant R141b saturated condenser pressure increases with the corresponding condenser temperature increasing. The decreasing pressure difference between the mixing chamber and the diffuser outlet, leads to a decrease both ω and the coefficiency of refrigeration performance.

5. Case Study

To get a better understanding of the inlet air process, one often refers to the psychometric chart. Fig.11 illustrates the various psychometric processes applicable to an inlet air cooling unit[7,27]. Evaporative

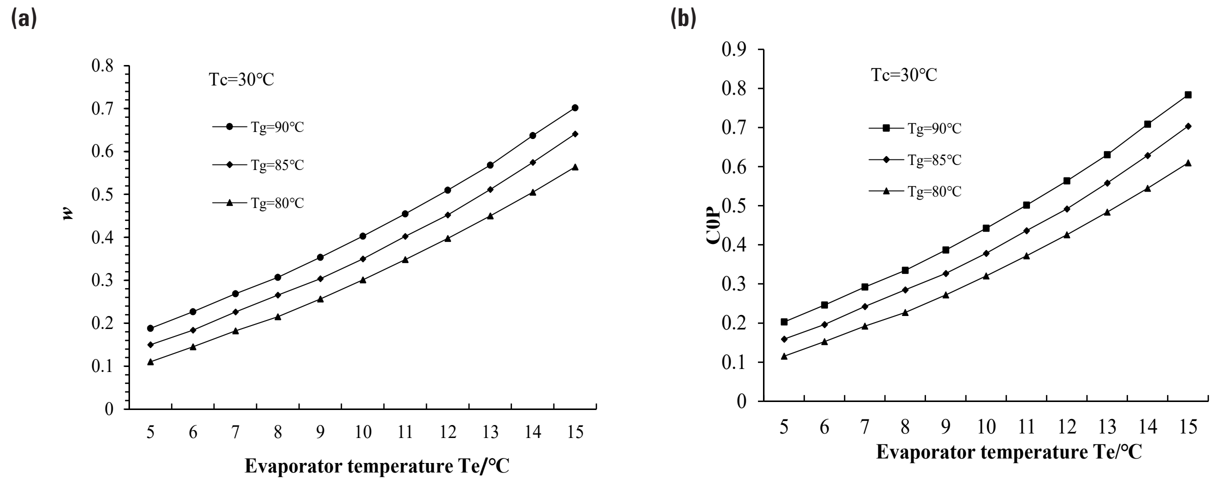

Fig.9 Effect of evaporator temperature on the system performance for Tc=30°C

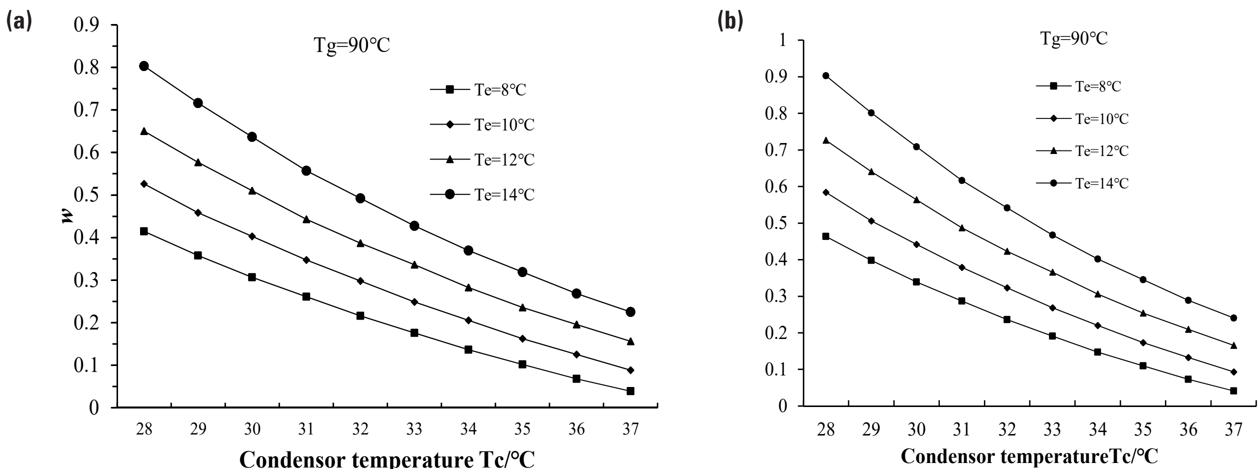

Fig.10 Effect of condenser temperature on the system performance for Tg=90°C

cooling is based on the evaporation of water in the intake air of the gas turbine. As water evaporates, the latent heat of evaporation is absorbed from the surrounding air. As a result, the air is cooled undergoes the constant enthalpy humidification process. Evaporative cooling is most suited to hot dry areas as it uses the latent heat of vaporization to cool ambient temperature from the dry-bulb to the wet-bulb temperature. Refrigeration cooling can reach a cooling temperature below the wet bulb, first the air temperature drops while the relative humidity continues to rise until its dew point temperature is reached (a-b). A further cooling process(b-c) continues removing the latent heat of the condensation of the water in the air until it reaches the desired temperature point c.

The total refrigerating cooling load includes the sen-

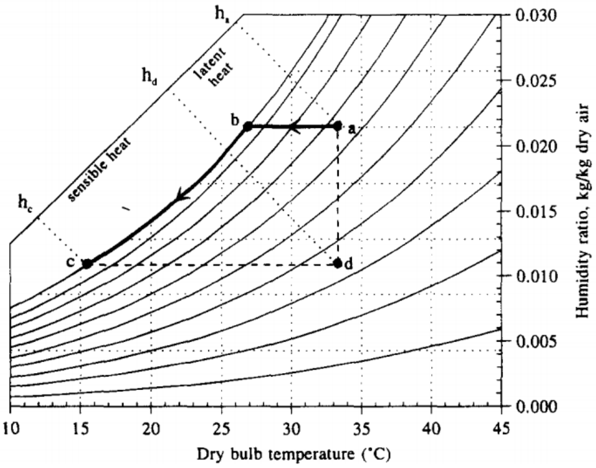

Fig.11 The air cooling process on psychometric chart

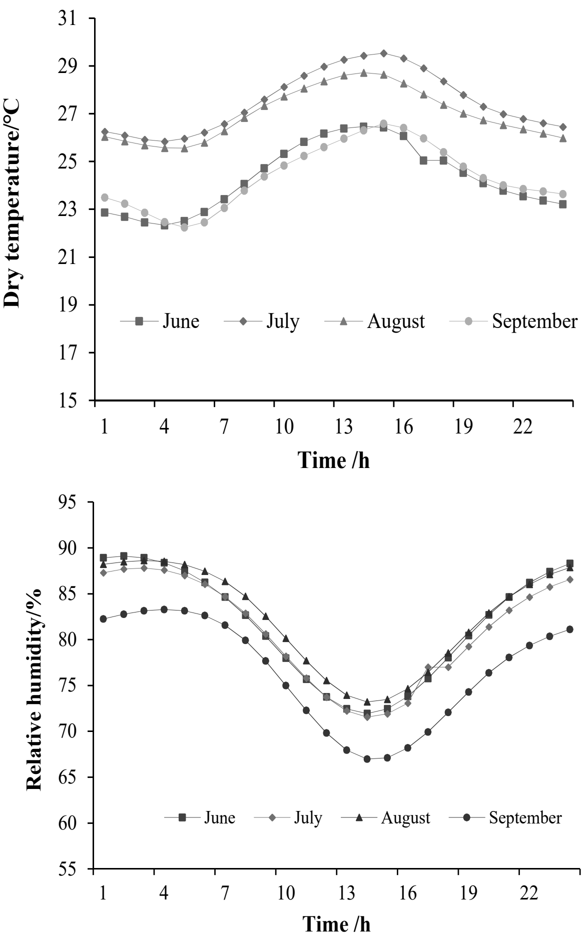

Fig.12 Hourly meteorological parameters from June to Septermber in Shanghai

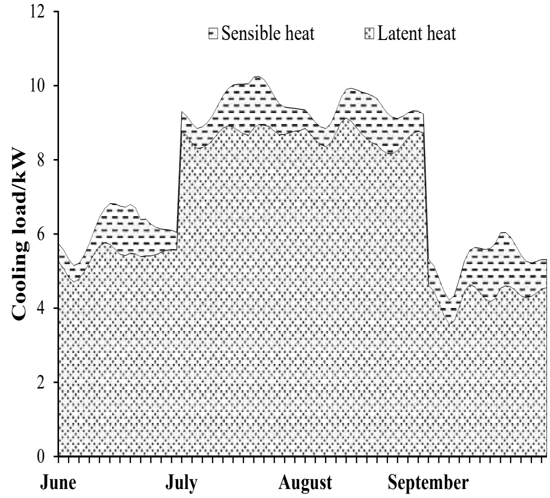

Fig.13 Cooling load variation from June to September

sible heat of air and the heat required to be removed, to condense the moisture contained in the air:

$$Q_t = m_a(Q_{sensible} + Q_{latent}) \quad (29)$$

$$Q_{latent} = h_a - h_d \quad (30)$$

$$Q_{latent} = h_a - h_d \quad (31)$$

h_a, h_c and h_d are the enthalpy of air at points a, c and d respectively.

The cooling load calculation procedure is as follows: the climate data [28] in Shanghai City of average day of each month (June-September) are calculated. Twenty-four pairs of data (temperature and relative humidity) represent an average day, Fig.12. Cool the inlet air temperature down to 18 °C, the cooling loads composed of sensible and latent heat in different months are presented in Fig 13. It can be clearly seen that the contribution of latent heat is comparable to sensible heat, especially in August. This is because the high relative humidity of the air in Shanghai during hot months.

Evaporative cooler effectiveness is given by:

$$E = \frac{T_{1DB} - T_{2DB}}{T_{1DB} - T_{2WB}} \quad (32)$$

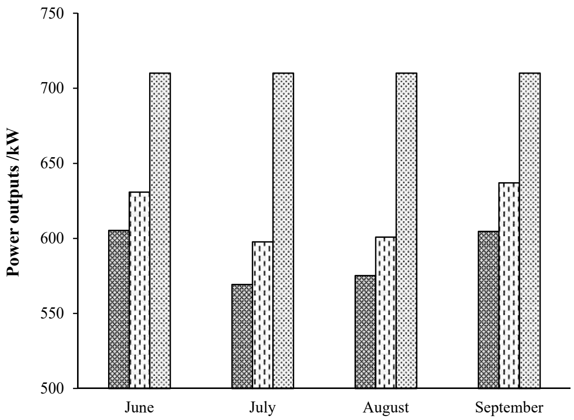

Fig.14 Comparison power augment performance with different cooling methods

Where T_1 is inlet air temperature, T_2 is exit temperature of evaporative cooler; *DB* is dry bulb, *WB* is wet bulb. A typical value for evaporative cooling effectiveness *E* is 85–90%[29,30,31]. The inlet air temperature drop assuming an effectiveness of 0.9, is given by

$$\Delta T_{db} = 0.9 \cdot (T_{1,DB} - T_{2,WB}) \quad (33)$$

Cooling load of MGT C30 variation from June to September can be seen in Fig.13. The average daily maximum hourly cooling load 10.23 kW occurs at 3:00 PM in July with average dry temperature 30°C. Recovering part heat from the waste gas driving the ejector refrigeration to work can really cover the cooling load under the design condition of evaporative temperature 10°C, generator temperature 85°C, the calculation *COP*=0.378. As the inlet air temperature increase, the experimental results analysis show that the power output of C30 drop 0.47 kW/K. Assuming the evaporative effective is 0.9, Fig.14 shows the technical results obtained with the application of evaporative and ejector refrigerating cooling methods.

Evaporative cooling method generally has a good effect during September due to the relatively low temperature and humidity weather condition and the total power augment can increase 5.4%. The power augment become slightly with the increasing of temperature and humidity with evaporative cooling. However, jet refrigeration cooling method shows a good inlet air cooling performance, the total power monthly augment capacity can reach 24.8%, 23.5% in July and August respectively. According to the results, the ejector refrigerating cooling method has a good advantage over evaporative cooling method on gas turbine power augment under higher temperature and humid weather condition.

6. Conclusion

A thermodynamic analysis of MGT C30 jet refrigeration inlet air cooling driven by recovering waste gas heat systems has been carried out. The results of the calculation suggest the following comments:

The performance of MGT C30 is particularly sensible to the ambient inlet air temperature whose increase determines a significant loss 0.47 kW in terms of experimental performance, even higher than that of large sized GTs as documented in several works. Modeling and simulation of C30 was developed with Matlab/Simulink, the power outputs of turbine reduce much greater compared with the work consumption of compressor, leading to the decrease of net power output, simulation results have a good agreement with experimental data.

The effect of some main parameters on ejector refrigeration performance with refrigerant R141b was investigated: COP increased with both increasing generator and evaporator temperature, and decreased with condenser temperature. Heat recovered from the exhaust gas drives the ejector refrigeration to cooling the inlet air, the performance of the ejector refrigeration is obtained through numerical calculation under different conditions, which can really cover the inlet air cooling load under design condition.

Cooling effect comparison between evaporative and refrigerated cooling methods was carried out in Shanghai city. Ejector refrigerating cooling method driven by waste gas has obvious advantage under high temperature and heavy humidity environment condition.

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