Analysis of Design and Part Load Performance of Micro Gas Turbine/Organic Rankine Cycle Combined Systems

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This study analyzes the design and part load performance of a power generation system combining a micro gas turbine (MGT) and an organic Rankine cycle (ORC). Design performances of cycles adopting several different organic fluids are analyzed and compared with performance of the steam based cycle. All of the organic fluids recover greater MGT exhaust heat than the steam cycle (much lower stack temperature), but their bottoming cycle efficiencies are lower. R123 provides higher combined cycle efficiency than steam does. The efficiencies of the combined cycle with organic fluids are maximized when the turbine exhaust heat of the MGT is fully recovered at the MGT recuperator, whereas the efficiency of the combined cycle with steam shows an almost reverse trend. Since organic fluids have much higher density than steam, they allow more compact systems. The efficiency of the combined cycle, based on a MGT with 30 percent efficiency, can reach almost 40 percent. Also, the part load operation of the combined system is analyzed. Two representative power control methods are considered and their performances are compared. The variable speed control of the MGT exhibits far better combined cycle part load efficiency than the fuel only control despite slightly lower bottoming cycle performance.

Key Words : Micro Gas Turbine, Organic Rankine Cycle, Design, Part Load, Heat Recovery

Nomenclature

A : Area [m^2]
$BTIT$: Turbine inlet temperature of the bottoming cycle [℃]
EGF : Exhaust gas flow [kg/s]
EGT : Exhaust gas temperature [℃]
h : Specific enthalpy [kJ/kg]
HRU : Heat recovery unit
LHV : Lower heating value [kJ/kg]
MGT : Micro gas turbine
$\dot{m}$: Mass flow rate [kg/s]
ORC : Organic Rankine cycle
P : Pressure [kPa]
PHT : Preheater
PR : Pressure ratio of the micro gas turbine
REC : Recuperator
$\dot{Q}$: Heat transfer rate [kW]
T : Temperature [℃]
TET : Turbine exit temperature [℃]
TIT : Turbine inlet temperature of the micro gas turbine [℃]
ΔT_{lm} : Log mean temperature difference [℃]
U : Overall heat transfer coefficient [kW/m^2℃]

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$\dot{W}$: Power [kW]
z : Exponent
η : Efficiency

Subscripts
aux : Auxiliary
BC : Bottoming cycle
bf : Bottoming cycle fluid
c : Cold fluid, compressor
CC : Combined cycle
d : Design point
$evap$: Evaporator
f : Fuel
fc : Fuel compressor
g : Gas
gen : Generator
h : Hot fluid
i : Inlet
o : Outlet
p : Pump
rec : Recuperator
t : Turbine

1. Introduction

Research and development of highly efficient and environment friendly energy systems have been active recently in various areas. In addition to exploring future systems using renewable energy sources, researchers are trying to improve conventional fossil fuel fired systems such as gas turbines. Recent worldwide interest in the distributed power generation has promoted development effort for sufficiently efficient and economically feasible small power sources. With consideration of relative technical maturity, environment friendliness and so on, one of these small power sources receiving attention is the micro gas turbine (MGT). Micro gas turbines are defined as small gas turbines, usually less than 200 kW (Rodgers et al., 2001). The maximum temperature (turbine inlet temperature, TIT) of micro gas turbines is far lower than those of large gas turbines because hot section cooling is not feasible both economically and technically. In addition, their pressure ratios are also sufficiently low. Thus, the MGT must use a recuperator to overcome its efficiency limitation. Currently available MGTs are designed with TIT of 800 to 900℃ and pressure ratio of 3 to 5, and their efficiencies are 25 to 30 percent. More efficient and power upgraded versions are still under development. In particular, designing a compact and efficient recuperator is deemed to be the most critical in improving engine performance (McDonald, 2000). Thermal efficiency of 40 percent is the current goal of micro gas turbine developers (U.S. Department of Energy, 2000). Many design improvements must be implemented to achieve 40 percent efficiency. Foremost, the turbine inlet temperature (over 1200℃, for example) must be increased by developing high temperature materials such as ceramic, which does not need cooling (McDonald and Rodgers, 2005).

Another way to improve MGT efficiency is to combine the MGT with a bottoming cycle. The concept of this cycle is similar to the conventional gas turbine/steam turbine combined cycle but differs in a couple of aspects. Since the MGT adopts the recuperator, its exhaust gas temperature is quite low. In addition, the system should be as compact as possible in order to be competitive in small power applications. Accordingly, working fluids, which are more suitable for these requirements than the steam, have been searched. Organic fluids are considered as good working media in Rankine cycle systems utilizing low grade (low temperature) heat sources (Yamamoto et al., 2001). In general, since the turbine exit state of the organic Rankine cycle remains in the superheated region, turbine blade corrosion problem is less severe. In addition, the relatively high density of organic fluids enables more compact design. In this respect, organic cycles can be widely applied to low temperature heat recovery systems. The application to heat recovery from the gas turbines exhaust (Bronicki and Schochet, 2005) is a good example. In particular, combination of the ORC with a micro gas turbine has been initiated recently (Brasz and Biederman, 2003; Zyhowski, 2003).

In this study, the design performance of MGT-ORC combined cycle has been analyzed by parametric studies. Based on a representative design condition of the current micro gas turbine, design parametric analysis of the bottoming cycle was performed for several working fluids. Design characteristics of different working fluids were compared and optimal design conditions were suggested. Design characteristics of cycles with organic fluids were compared with those of the steam cycle. Besides design efficiency, the part load performance of power generation systems must be examined, since the systems usually operate at part load conditions for a considerable part of their lifetimes. Off-design operation of the combined system was modeled and two representative methods of controlling system power (fuel only control and variable speed control) have been analyzed.

2. MGT-ORC Combined System

The layout of the combined cycle is shown in Fig. 1. The MGT operates under a recuperated cycle and the thermal energy of its exhaust gas is transferred to the bottoming Rankine cycle at the heat recovery unit (HRU). The preheater may be used when the turbine exit state remains sufficiently superheated. Organic fluids include hydrocarbons and refrigerants. In particular, some refrigerants seem suitable for ORC system because they have long been used for turbo compressor (Brasz and Biederman, 2003), which are similar to the turbine of the ORC. However, most common refrigerants for commercial refrigeration and heat pump systems do not seem quite suitable for the ORC of this study because their critical temperatures are too low so they cannot efficiently recover heat from the MGT exhaust gas. Recently, a new refrigerant (R245fa) has been developed and evaluated as having favorable characteristics (Zyhowski, 2003). In this work, R123, which has similar properties as R245fa, is adopted as the primary working fluid. Three hydrocarbons (normal butane, isobutane and propane) are analyzed as well. For comparison, the steam bottoming cycle is also analyzed. Table 1 summarizes the major properties of the working fluids examined in this study. Organic fluids have far lower critical temperatures and pressures than steam. Among them, R123 has the highest critical temperature.

Figure 2 illustrates the temperature-entropy diagrams of the steam cycle and the R123 cycle. Cycles with other organic fluids are similar to R123 cycle. The major difference between steam and organic fluids is the slope of the saturated vapor line. In contrast to steam, organic fluids have a positive slope so they become superheated at the turbine exit.

Table 1 Properties of working fluids

Working fluids	Molecular weight (kg/kmol)	Critical pressure (bar)	Critical temperature (℃)	Boiling point at 1atm (℃)
R123	152.9	36.68	186.7	27.84
Propane	44.09	42.47	96.7	−42.09
N-butane	58.12	37.96	152.0	−0.54
I-butane	58.12	36.40	134.7	−11.61
H_2O	18.02	220.6	373.8	100.00

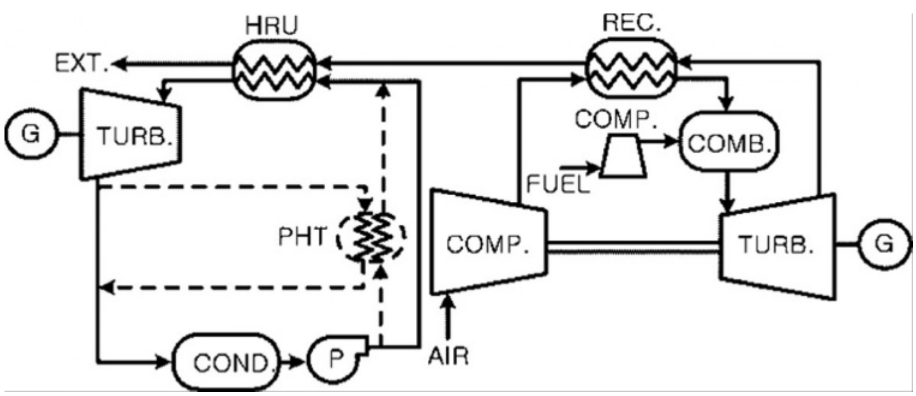
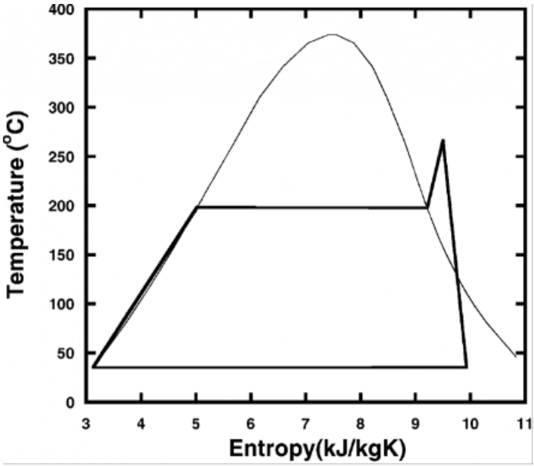
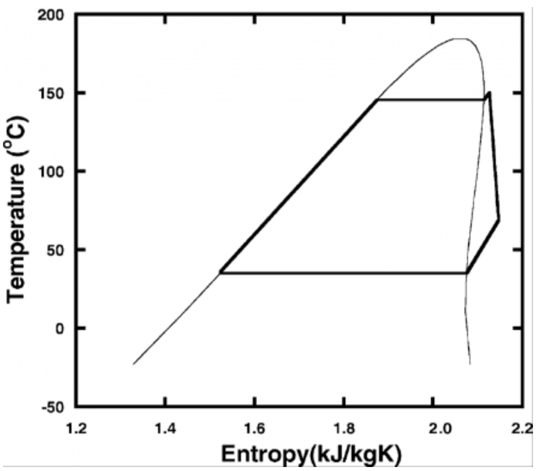


Fig. 1 Schematic of the MGT-ORC combined system



(a) Steam



(b) R123

Fig. 2 Examples of temperature-entropy diagram of the bottoming cycle

3. Design Performance

3.1 Analysis

First, the design performance of the MGT is estimated for various pressure ratios (PR) and turbine inlet temperatures (TIT). Turbine cooling is not considered. Then, based on a representative MGT design point, bottoming cycle performance is predicted for varying evaporating pressure and turbine inlet conditions. The micro gas turbine has the single shaft arrangement as shown in Fig. 1. Isentropic compressor and turbine efficiencies are used. Temperature effectiveness of the recuperator is defined as follows.

$$\eta_{rec}=\frac{T_{c,o}-T_{c,i}}{T_{h,i}-T_{c,i}} \tag{1}$$

Considering miscellaneous losses such as mechanical loss and generator loss, the power of the micro gas turbine is calculated as follows:

$$\dot{W}_{MGT}=(\dot{W}_t\cdot\eta_m-\dot{W}_c)\cdot\eta_{gen} \tag{2}$$

The heat recovery unit consists of three parts: an economizer, an evaporator and a superheater. At each heat recovery unit, the following energy balance is applied.

$$\dot{Q}=\dot{m}_g(h_{g,i}-h_{g,o})=\dot{m}_{bf}(h_{bf,o}-h_{bf,i}) \tag{3}$$

In the steam bottoming cycle, the turbine inlet temperature (BTIT) should be designed as high as possible, as in most conventional combined cycle plants. Thus, the BTIT is set to 10℃ lower than the MGT exhaust temperature. In the organic cycle, the evaporating pressure is limited to the subcritical regime. In particular, the synthetic substance R123 may be chemically unstable when the temperature is very high (Zyhowski, 2003). Therefore, the maximum BTIT of the R123 cycle is set to 150℃.

The pinch point temperature difference (minimum temperature difference between the gas and the bottoming cycle fluid) inside the HRU is set to 10℃. In case of steam, the pinch point corresponds to the evaporator inlet (saturated liquid). However, in the organic cycle, it occurs inside or at the inlet of the economizer, as will be demonstrated in section 3.2. To predict the pinch point precisely, the economizer is modeled as a multi-segment control volume.

Power and thermal efficiency of the combined cycle are described as follows:

$$\dot{W}_{CC}=\dot{W}_{MGT}+\dot{W}_{BC}-\dot{W}_{aux} \tag{4}$$

$$\dot{W}_{aux}=\dot{W}_{fc}+\dot{W}_p \tag{5}$$

$$\eta_{CC}=\frac{\dot{W}_{CC}}{\dot{m}_f LHV_f} \tag{6}$$

Table 2 summarizes the main design parameters used in the analysis. Reasonable pressure losses are assigned for all of the flow elements. Analyses

Table 2 Main design parameters for analysis

Ambient temperature	15℃
Ambient pressure	101.3 kPa
Air mass flow rate	1 kg/s
MGT turbine efficiency	0.85
MGT compressor efficiency	0.78
Fuel compressor efficiency	0.60
MGT recuperator effectiveness	0.88
Bottoming cycle turbine efficiency	0.85
Bottoming cycle condensing temperature	35℃
Mechanical efficiency	0.96
Generator efficiency	0.93

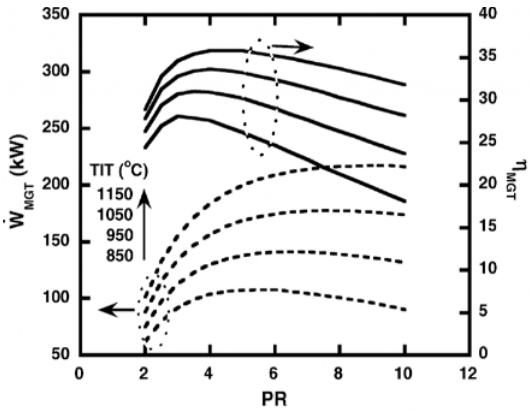


Fig. 3 Design performance of MGT

are performed using process analysis software (Aspen Technology, 2004).

3.2 Results and discussion

Figure 3 shows variations in MGT performance with turbine inlet temperature and pressure ratio. Peak efficiency varies from 27 percent to 36 percent for the TIT range from 850℃ to 1150℃. The highest efficiency available among current MGTs on the market is about 30 percent and their turbine inlet temperatures are estimated to be around or higher than 900℃. Therefore, the present results represent the current technology sufficiently, which means that the assumed parameters are close to reality. In order to analyze the MGT-ORC combined cycle based on the current representative MGT technology, the design point with TIT and PR of 950℃ and 4.5, respectively, is selected. This design point provides MGT efficiency close to 30 percent. Table 3 shows the main parameters of this reference MGT design point.

Table 3 Performance of the selected MGT design conditions

Pressure ratio	4.5
Turbine inlet temperature	950℃
Turbine exit temperature	647.4℃
Exhaust gas temperature	283.6℃
Power	131.0 kW
Efficiency	29.9%

The main design parameters to be determined in the bottoming cycle are the turbine inlet temperature (BTIT) and the evaporating pressure. In order to examine the effect of each parameter on the bottoming cycle power separately, two types of calculations are carried out. First, the effect of evaporating pressure is evaluated as shown in Fig. 4. Results for three organic fluids as well as steam are presented together. In the steam cycle, BTIT is 274℃ (10℃ less than MGT exhaust temperature), and power is maximized at an intermediate pressure. In the organic cycles, BTIT is fixed at 150℃. For a fixed BTIT, the power increases continuously with increasing pressure. Organic fluids should be designed with far higher evaporating pressures than that of the steam. At sufficiently higher evaporating pressures, R123 cycle provides larger power than the steam cycle. The larger bottoming cycle power means higher combined cycle efficiency. Secondly, the effect of increasing BTIT on the bottoming cycle power for each fluid is shown in Fig. 5. In the steam cycle, power increases with increasing BTIT. However, the trend is quite different in the organic fluids. BTIT has very little or even negative influence on the bottoming cycle power. This trend is observed in the entire pressure range for all of the organic fluids.

Tentative conclusion from these analyses is that superheating in front of the turbine is not required in the organic bottoming cycle. In addition to the slightly better efficiency, elimination of the superheater results in a more compact system

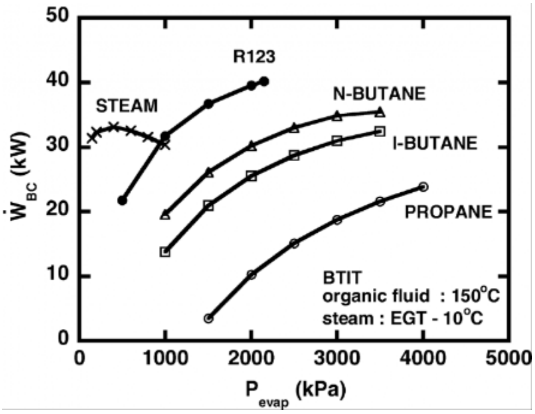


Fig. 4 Effect of evaporating pressure on the bottoming cycle performance at fixed BTIT

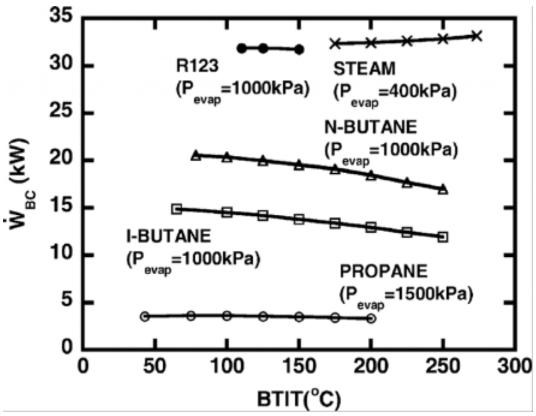


Fig. 5 Effect of BTIT on the bottoming cycle performance at a fixed evaporating pressure

as well. Therefore, the ORC cycle, with HRU exit state being saturated vapor, is analyzed and the combined cycle efficiency is shown in Fig. 6. In case of steam, turbine inlet conditions of Fig. 5 are maintained. R123 and normal butane (N-butane) provide slightly better peak efficiencies than steam does. When organic fluids are used, the bottoming cycle performance is proportional to the order of the critical temperature. R123, which has the highest critical temperature, exhibits the best efficiency. N-butane shows slightly higher efficiency than isobutene (I-butane). Other hydrocarbons with even lower critical temperatures provide far lower efficiency. For example, propane provides only 34 percent combined system efficiency at best, as shown in the figure.

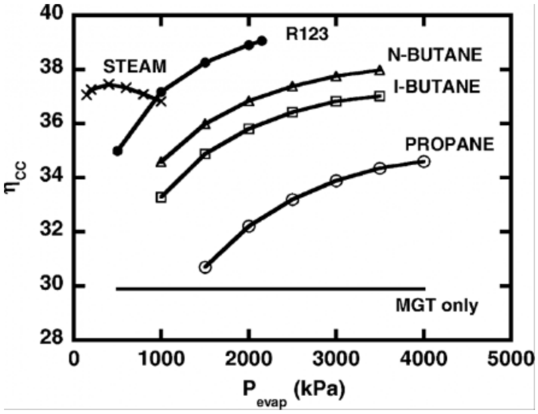
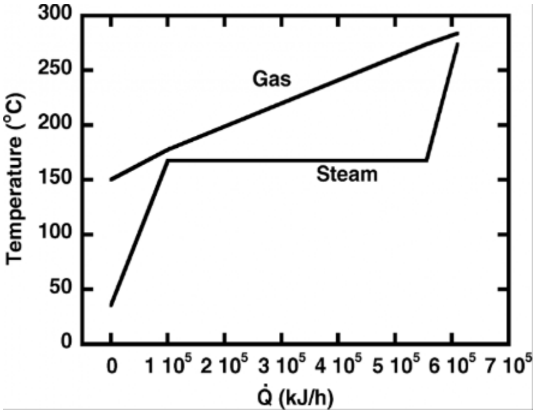


Fig. 6 Efficiency of the combined system

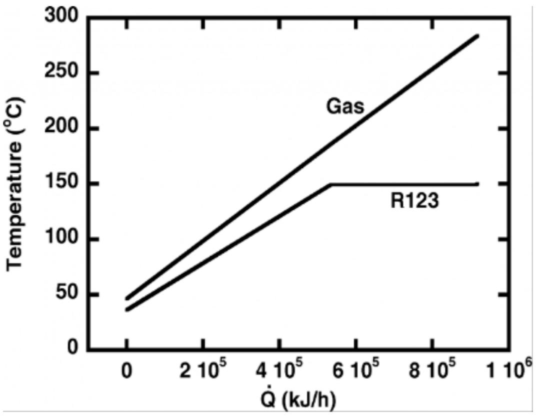
Table 4 Recovered heat and power of the bottoming cycle

Working fluids	Mass flow (kg/s)	Evaporating pressure (kPa)	Recovered heat (kW)	Power (kW)
R123	1.11	2150	255.1	40.2
Propane	0.78	4000	252.9	20.6
N-butane	0.55	3500	254.0	35.4
I-butane	0.64	3500	253.4	31.2
H_2O	0.06	400	169.5	33.1

Table 4 shows the heat recovery rate and the power of the bottoming cycle at the best efficiency condition. Fig. 7 compares temperature vs heat transfer diagrams at the HRU of R123 and steam. Since the evaporating pressure of the steam cycle is moderate, the majority of the heat recovery takes place at the evaporator. However, with the organic fluid, the evaporator recovers relatively small amount of heat because the pressure approaches the critical pressure. Moreover, since the superheater is not adopted, very large amount of heat is recovered at the economizer (liquid part of the organic fluid) and the pinch point occurs at the gas exit. Thus, the gas exhaust temperature is far lower, and the recovered heat is larger in comparison with steam cycle. The result shows that the bottoming cycle efficiency ($\dot{W}_{BC}/\dot{Q}_{HRU}$) of the organic fluid cycle is lower than that of the steam cycle. This is thermodynamically reasonable because the average temperatures of the or-



(a) Steam at 400 kPa



(b) R123 at 2150 kPa

Fig. 7 Temperature vs recovered heat diagram of HRU

Table 5 Working fluids properties at turbine exit of the bottoming cycle

Working fluids	Temp. (℃)	Degree of Superheat (℃)	Pressure (kPa)	Density (kg/m^3)
R123	64.7	29.7	130.8	7.4
Propane	36.3	1.2	1260	30.1
N-butane	56.9	21.9	337.8	7.7
I-butane	49.4	14.4	480.8	11.7
H_2O	35.6	-	5.72	0.04

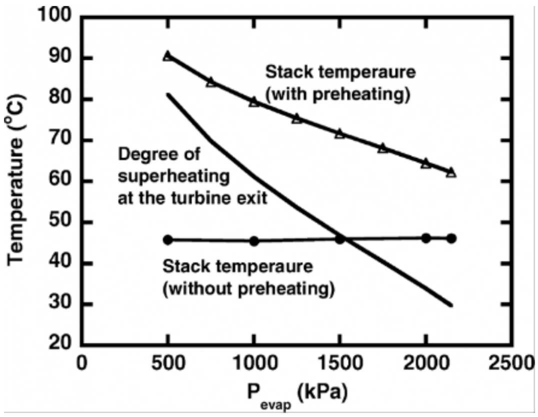


Fig. 8 Variations in degree of superheating at the turbine exit and stack temperature in case of R123

ganic fluids at the HRU are lower and moreover the heat rejection temperature at the condenser is higher (turbine exit state is superheated) compared with the steam cycle. However, in the R123 cycle, the far greater recovered heat ($\dot{Q}_{HRU}$) outweighs the lower bottoming cycle efficiency. Accordingly, the bottoming cycle power ($\dot{W}_{BC}$) is larger, and thus, the combined cycle efficiency is higher than that of the steam cycle.

Table 5 summarizes the turbine exit states and properties of all working fluids at the optimal design conditions. R123 should be designed with much larger mass flow rate than steam (see Table 4). However, since its density is much greater, a more compact bottoming cycle components can be realized. But, the lower speed of sound of R123 due to larger molecular weight causes lower flow velocity. This may somewhat reduce the advantage of the compactness. The pressure ratios of organic cycles are far less than that of the steam cycle. For example, the ratio between evaporating and condensing pressures of R123 is roughly 10, while that of the steam is about 70. This low pressure ratio is another advantage in the design of turbomachinery components such as the turbine and the pump.

Figure 8 exemplifies the degree of superheating at the turbine exit, and the effect of adopting a preheater on the stack temperature, when using R123. A higher evaporating pressure provides a larger degree of superheat at the turbine exit, which increases the thermal load of the condenser. A preheater (internal regenerative heat exchanger), as denoted by the dotted lines in Fig. 1, can be equipped to reduce the condenser load. Another merit of adopting the preheater is that the stack gas temperature is increased as shown in the figure. At the suggested design pressure of 2200 kPa, the original low stack temperature, around 45℃, is increased by at least 20℃. This may lessen the possibility of condensation of corrosive gas components in the stack gas.

The influence of the degree of recuperation of the micro gas turbine on the combined cycle performance is examined and results are shown in Fig. 9. A lower recuperator effectiveness means a lower MGT efficiency and increase of the MGT exhaust temperature. Let's first look at the result of the MGT/steam bottoming cycle. BTIT of the steam cycle increases with the reduction of effectiveness, and the corresponding optimal evaporating pressure increases. Although there exists a slight minimum of the system efficiency, reducing the effectiveness generally increases the MGT/steam combined cycle efficiency. Since the bottoming cycle is fully adapted (optimized) to the increased gas temperature, the simple gas turbine cycle is more beneficial than the fully recuperated cycle. For all the organic fluids, the bottoming cycle turbine inlet condition is still fixed at the optimal condition (maximum saturated vapor condition) suggested previously. Organic fluids exhibit an almost reverse trend, that is, the system efficiencies decrease with reduction of the MGT recuperator effectiveness. Even though the heat recovery increases due to increased gas temperature at the HRU, the bottoming cycle power does not increase because of the limited turbine inlet condition. Therefore, the organic cycle prefers full heat recuperation of the MGT for maximization of the combined system efficiency.

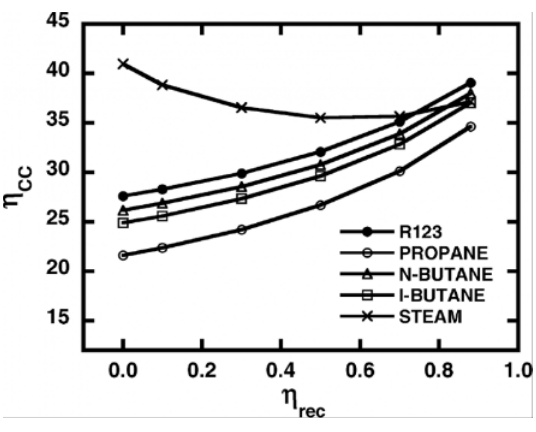


Fig. 9 Effect of MGT recuperator effectiveness on the performance of the combined system

4. Part Load Performance

4.1 Analysis

Part load analysis has been performed for R123, which is the best working fluid as examined in the last section. The system examined does not include the preheater. Two representative power control methods are considered. The simplest way of controlling the gas turbine in response to load change is to modulate the fuel supply. This method can be called fuel only control or maximum air flow control because air flow rate of the compressor is not modulated actively. Another method that can be used in the micro gas turbine is the variable speed control. The most important process in the recuperated cycle gas turbine is the exhaust heat recovery process at the recuperator. Therefore, it is important to fully utilize the heat recovery effect during the part load operation. For this purpose, the recuperator inlet temperature should be maintained as high as possible by modulating the air flow rate as well. This variable air flow control improves the part load efficiency dramatically for cycles involving heat recovery such as the combined cycle, cogeneration system and recuperated cycle (Kim et al., 2003 ; Kim and Hwang, 2006). Recently, modulation of air flow rate by varying the shaft speed has begun to be adopted in commercial micro gas turbines. This scheme provides far better part load efficiency of the recuperated gas turbine cycle than that of the fuel only control (Kim and Hwang 2006). In the fuel only control, the speed of the shaft connected to the generator should be kept constant. In the variable speed control, the engine shaft drives a high speed alternator. Then, a digital power controller (AC-DC-AC inverter) converts the varying high frequency into a constant low frequency (Rodgers et al., 2001). In this section, both the fuel only control and the variable speed control are simulated. In the variable speed control, the control target is to keep a constant recuperator inlet temperature, i.e. the turbine exit temperature, to maximize the heat recovery effect.

In order to perform the part load analysis, off-design calculation models are required for every component of the combined system. The off-design operation of the compressor is modeled by the performance map illustrated in Fig. 10. Similarly, a realistic turbine map is used to model the turbine operation. The off-design operation of the recuperator is modeled as follows:

$$\dot{Q}=UA\Delta T_{lm} \tag{7}$$

$$\frac{U}{U_d}=\left(\frac{\dot{m}}{\dot{m}_d}\right)^z \tag{8}$$

Based on the UA-LMTD method, the overall heat transfer coefficient is corrected, considering its dependence on the mass flow rate. The exponent is set 0.3 based on the examination of the performance variation of practical compact recuperators (Kim and Hwang, 2006).

In the bottoming cycle, models are required for every section of the heat recovery unit and the condenser. The HRU consists of two sections (economizer and evaporator). For each unit, energy balance between the MGT exhaust gas and the ORC fluid and an off-design heat transfer model similar to Eqs. (7) and (8) are applied. The exit state of the evaporator is always saturated vapor. The condensing temperature is assumed to be kept constant at the design value. The condenser consists of a superheated section and a condensing section. Again, energy balance between the ORC fluid and the cooling fluid (air) and relations similar to Eqs. (7) and (8) are applied to each section. The following choking condition is assumed to model the ORC turbine operation.

$$\frac{\dot{m}\sqrt{T_i}}{P_i}=\text{constant} \tag{9}$$

The design point of the micro gas turbine is the same as that in section 3 (Table 3). The ORC fluid is R123 and its design condition corresponds to those of Tables 4 and 5. Turbine inlet condition is saturated vapor at 2150 kPa, and condensing pressure is 130.8 kPa (condensing temperature, 35℃). The design efficiency of the combined system is 39 percent.

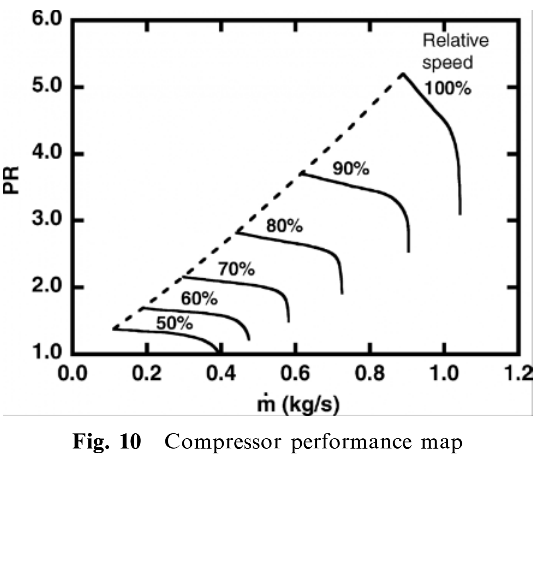
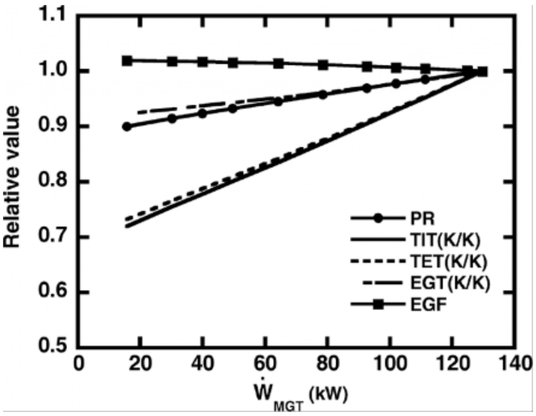


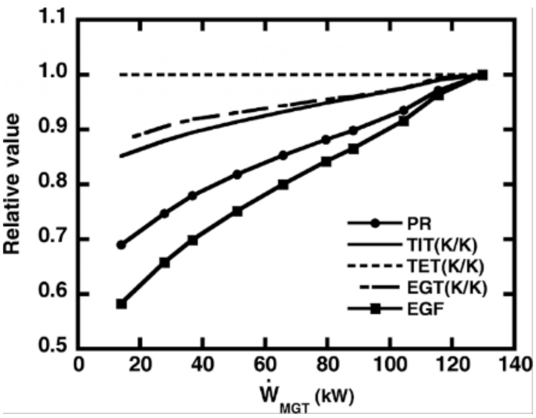
Fig. 10 Compressor performance map

4.2 Results and discussion

First, variations in the main MGT parameters are presented in Fig. 11. All the parameters are normalized by the design values. In the fuel only control, the shaft speed is fixed and pressure ratio decreases and air mass flow rate increases slightly with power reduction. On the contrary, in the variable speed control, the shaft speed decreases, and thus the air flow rate decreases with power reduction. As a result, the compressor pressure ratio of the variable speed control changes much during the part load operation. Also shown in the figures are the temperatures at the hot parts. In the fuel only control, the turbine inlet and exit temperatures decrease with power reduction rather rapidly. In the variable speed control, the turbine exit temperature is kept at the design value by reducing the mass flow rate. Accordingly, its turbine inlet temperature is also higher. Moreover, the lower recuperator inlet air temperature of the variable speed control, due to the relatively lower compressor pressure ratio, provides much larger temperature change inside the recuperator (TET-EGT), which means that the heat recovery effect at the recuperator is greater than that of the fuel only control. Fig. 12 shows efficiency of the micro gas turbine. The variable speed control provides much better part load efficiency over the simple fuel only control. This is due to the more effective utilization of the heat recovery (larger temperature difference) at the recuperator.



(a) Fuel only control



(b) Variable speed control

Fig. 11 Variations in main parameters of the MGT during part load operation

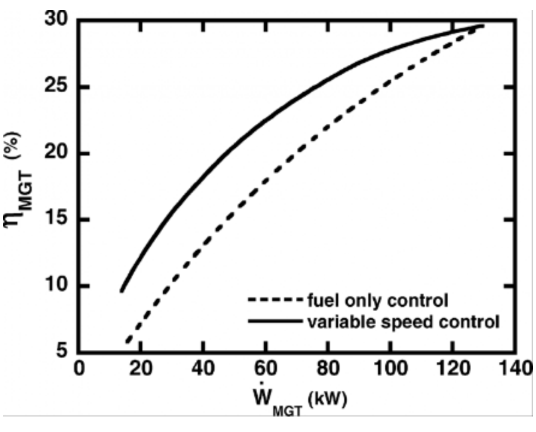
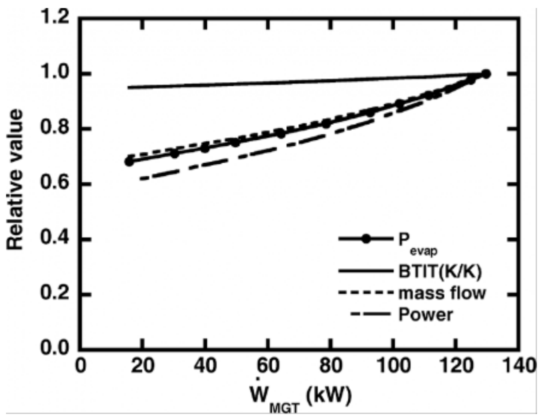
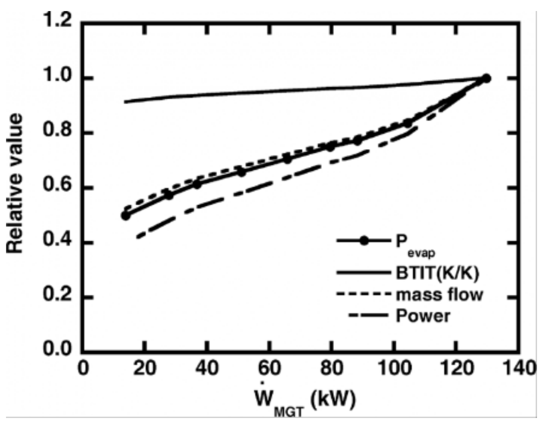


Fig. 12 Part load efficiency of MGT

Figure 13 presents the variations in the main parameters of the bottoming cycle according to MGT power for the two power control methods. All the parameters decrease with power reduction. The variable speed control causes relatively larger reductions of all parameters than the fuel only control. It is a direct result of the lower exhaust temperature and smaller gas flow rate of the micro gas turbine, which results in lower evaporating pressure and turbine inlet temperature of the bottoming cycle. The heat recovery rate at the HRU is also smaller, and thus the mass flow rate of the bottoming cycle fluid is lower. As a result, the power generated from the bottoming cycle is smaller in case of the variable speed control. This result demonstrates that the variable speed control is very effective with respect to the micro gas



(a) Fuel only control



(b) Variable speed control

Fig. 13 Variations in main parameters of the bottoming cycle during part load operation turbine efficiency itself but does not have further merit in the bottoming cycle.

Figure 14 shows the power distribution between MGT and ORC. The combined cycle power is, of course, the summation of the two parts. Since the variable speed control results in smaller ORC power at the same MGT power as shown in Fig. 13, the power fraction of MGT must be higher in order to obtain a given combined cycle power. Even though the variable speed control is not very effective with respect to the bottoming cycle performance, it is still much more advantageous than the fuel only control because it yields far better MGT performance, and furthermore, power fraction of the MGT is much higher than that of the bottoming cycle. Fig. 15 shows the part load efficiency of the combined system. As predicted, the variable speed control provides far higher part load efficiency than the fuel only control. In summary, the variable speed control is advantageous with respect to MGT performance because it provides higher heat recovery at the recuperator, but the resulting low thermal capacity of the MGT exhaust gas causes lower performance at the bottoming cycle. However, the improved MGT performance outweighs the disadvantage at the bottoming cycle, resulting in much higher combined system efficiency.

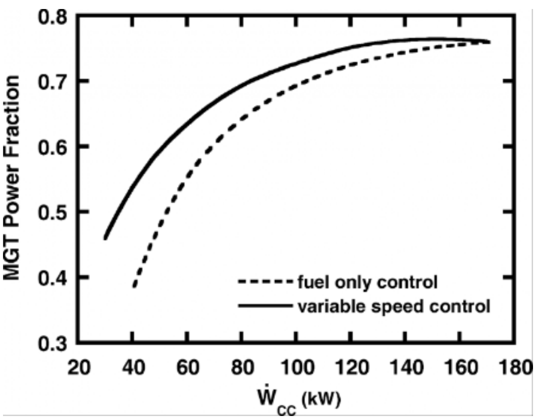


Fig. 14 Power fraction of MGT during part load operation

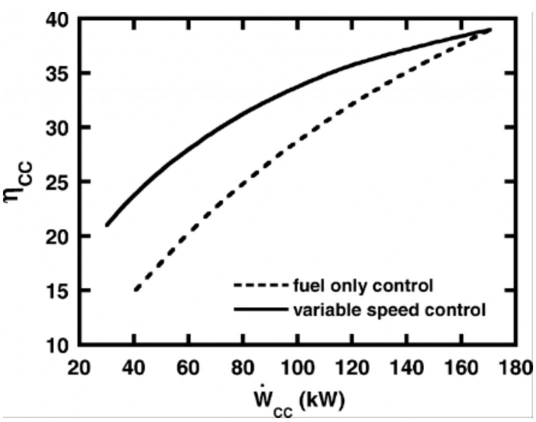


Fig. 15 Part load efficiency of the MGT-ORC combined system

5. Conclusions

The results from the analyses of the design and part load performance of micro gas turbine/organic Rankine cycle combined systems are summarized as follows.

(1) The turbine inlet state of the organic Rankine cycle does not need to be superheated and the evaporating pressure should be designed as high as possible.

(2) The amount of recovered heat of the organic cycle is much more than that of the steam cycle due to the smooth matching between gas and fluid temperature profiles at the liquid section (economizer). But the bottoming cycle efficiency of the organic fluid is low.

(3) Among several organic fluids researched, R123 produces the largest bottoming cycle power, thus providing the best combined cycle efficiency. Its efficiency is higher than that of the cycle using steam. Based on the MGT of 30 percent efficiency, the combined cycle efficiency of 39 percent can be achieved with R123.

(4) The efficiencies of the combined cycle with organic fluids are maximized when the turbine exhaust heat of the MGT is fully recovered at the MGT recuperator, whereas the efficiency of the combined cycle with steam shows an almost reverse trend.

(5) The variable speed control of the MGT results in much higher MGT part load efficiency than the fuel only control due to enhanced heat utilization at the recuperator. But, the resulting

low thermal capacity at the MGT exhaust degrades the performance of the bottling cycle. However, the improved MGT performance outweighs the disadvantage of the bottoming cycle, resulting in much higher combined system efficiency.

Acknowledgements

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