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Applied Performance Research of a Cogeneration Arrangement with Proposed Efficiency Well-Balance Method*

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Abstract

According to the third conference of parties (COP3), Japan has set a target of reducing greenhouse gas emissions by 6% by the year 2010. Cogeneration system is a recently potent method which its environmental benefits, through the highly efficient utilization of fuel that is related to reduction emissions. The particular purpose of this paper is to support the selection of cogeneration technologies by acquiring the optimal useful thermal energy and electrical power from system with well-efficiency balance method and fuel saving approaching method. When a micro gas turbine (MGT) is operated under ambient condition, the discharged hot gases from the MGT may be expanded at its exhaust stage and cooled by an exhaust heat exchanger which composes with a single stage absorption heat exchanger. The performance and annual total fuel saving amount of cogeneration plant will be investigated and compared with separated production of heat and power system. Eventually, this cogeneration plant will be reduced the fuel consumption rate in operation that will be also reduced the emissions and fuel cost when the system will gain highly efficiency of thermal energy and electrical power.

Key words: CGS, Micro Gas Turbine, Heat Exchanger, Annual Total Efficiency

1. Introduction

Use for fuels is one of the particular roles throughout the century and is greatly related the environmental issues such as air pollution and global climate change. Technologies of the fuels utilization in production sectors of electrical power and thermal heat energy are extremely begun to come to the fore. Cogeneration is often considered as a means for utilizing biogas, yet the total performance of this approach in practice remains to clarify adequately. A micro gas turbine cogeneration system (CGS) at Kitami City Sewage Treatment Center in Japan is adopted as a basis for simulations, experimental data and catalog data on the liberation of simulation crisis [(1), (2)]. The purpose of the analysis is to simulate the optimal performance benefit of exhaust thermal heat, electrical power and heat recovery in the CGS installed for utilization of biogas by sewage treatment facilities is examined. In particular, determination of those performances can support to obtain the maximum supply from the CGS and a well decision for choosing technology in any sector. Fuel saving amount can be also expected to realize the reduction fuel consumption in long term operation period and it is also related the reduction pollutants in the environment. The temperature of air at the compressor inlet stage of the MGT is employed as an important parameter determining the performance of the MGT. And then, the influence of the mass flow rate in exhaust stage of the MGT can be realized with the constituent quality for each portion. The proposed efficiency well-balanced method can accentuate the performance

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analysis and support to realize that system is operated with sufficient efficiency or not. Acquisition from analysis performance simulation of the CGS could be expressed in the following and it can be supported to develop the cumulative the CGS plants.

Nomenclature

COP : coefficient of performance
c_p : specific heat at constant pressure, kJ/kg K
C_R : capacity ratio of the EHE
$FESI$: fuel energy saving index
h : enthalpy, kJ/kg
$I_{m,C}$: Irreversibility of air compressor, kW
$I_{m,T}$: Irreversibility of turbine, kW
LHV : lower heating value, MJ/m^3
m_1 : exhaust mass flow rate, kg/s
m_c : cold fluid mass flow rate, kg/s
m_{coa} : correction mass flow rate, kg/s
m_f : fuel mass flow rate, kg/s
m_h : hot fluid mass flow rate, kg/s
m_w : water mass flow rate, kg/s
N_c : compressor work-done, kW
n_{cor} : correction speed of the MGT, rpm
n_r : rated speed of the MGT, rpm
N_t : turbine work-done, kW
NTU : number of transfer unit
P_e : electrical power, kW
Q_c : sum of heat quantity from condenser and absorber, kW
Q_e : cooling capacity, kW
Q_g : heat medium capacity, kW
Q_h : heating capacity, kW
$Q_{th,ehr}$: exhaust heat recovery, kW
$Q_{th,exe}$: exhaust thermal heat, kW
$Q_{th,f}$: fuel thermal heat, kW
$Q_{th,loss}$: exhaust heat loss, kW
r : exhaust heat to power ratio
SQ_e : standard cooling capacity, kW
SQ_h : standard heating capacity, kW
t_3 : exhaust temperature, ˚C
t_5 : outlet temperature of the EHE, ˚C
t_{8e} : inlet temperature for cooling cycle, ˚C
t_{8h} : inlet temperature for heating cycle, ˚C

Greek symbols

α : correction factor of AHE
ε : effectiveness of the EHE
η : efficiency

Subscripts

ccf : cooling capacity factor
e : cooling cycle
h : heating cycle
hcf : heating capacity factor
$hmfc$: heat medium flow correction
m : mechanical type

2. System configuration

The configuration of a CGS with an absorption heat exchanger (AHE) for analyzing the performance of system is illustrated in Fig. 1. In particular, useful thermal energy is achieved for using an exhaust heat exchanger (EHE), one shell pass-two tube exchanger that is unavailable to determine in actual plant design. The hot gas is expanded in the MGT exhaust stage and cooled in the EHE. The cold side of the heat exchanger is fed with water coming from heat medium of the AHE which has working fluid as LiBr and refrigerant as water. Standard heat rejection from cooling tower is 60.7kW and it is usually used to get cooling capacity when the AHE is operated in refrigeration cycle. Plenty produced amount of biogas from anaerobic digester is a particular combustion fuel for prime mover in the CGS. It is compressed by a gas compressor that includes a fuel drying device as a liquid trap. Ambient air enters a generator and passes systematically through a centrifugal compressor, a recuperator, an annular combustion chamber, and a radial turbine. The compressor pressure ratio is assumed equal to 3.5 and rotating components of the MGT are mounted on a single shaft supported by air bearings. The generator is cooled by the inlet air, and simultaneously produces both electrical power and exhaust heat energy. Electricity demand of sewage treatment center is supplied by operating the MGT generator and by purchasing electricity from an electrical power company.

The cold water inlet temperature of the EHE, t_4 value is here assumed equal to 80˚C and minimum temperature difference between the cold water inlet and outlet temperature is equal to 8 ~ 10˚C that is agreeable to measuring data of actual experiment. In order to

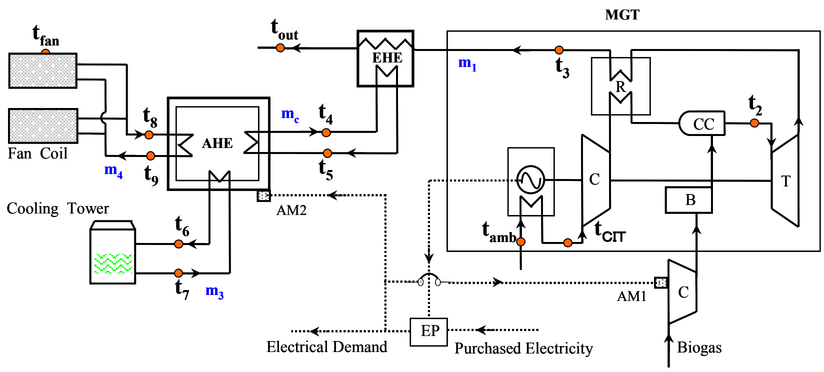

Fig. 1 Configuration of the CGS

Table 1 Design parameters and performances

Ambient temperature, t_{amb}	-13 ~ 37 ˚C
Ambient pressure	101.3 kPa
Turbine inlet temperature	850 ˚C
Compressor & turbine efficiency	0.76
Combustion efficiency	0.99
Recuperator efficiency	0.74
Mechanical efficiency, η_m	0.97
Rated speed of MGT, n_r	96,000 rpm
Rated output power of MGT, p_e	30 ±2 kW
Electrical efficiency, η_e	26 ± 2 %
Compressor pressure ratio	3.5
NOx emission from MGT	<9 @ 15 % O_2
Effectiveness of EHE, ε	0.80
Cold water inlet temperature of EHE, t_4	80 ˚C
Cold water mass flow rate of EHE, m_c	1.616 kg/s
Correction factor of the mean temperature difference, F	0.965
Capacity ratio of EHE, C_R	0.055 ~ 0.067
Cooling outlet temperature of AHE, t_{9h}	7 ˚C
Heating outlet temperature of AHE, t_{9e}	55 ˚C
Standard cooling capacity, SQ_e	25 kW
Standard heating capacity, SQ_h	35.7 kW
Standard heat rejection to cooling tower	60.7 kW
Standard heat medium input capacity	35.7 kW

obtain the optimal heat recovery, effectiveness of the EHE is assumed equal to 80% and correction factor of the mean temperature difference is equal to 0.965. According to the actual experiment data, the exhaust gas and the cold water mass flow rate are assumed equal to 0.34~0.4kg/s and 1.616kg/s, respectively. Exhaust gas mass flow rate could be obtained in simulation procedure by solving with rated speed of 96,000rpm and relative temperatures. The corresponding design parameters and performances of the CGS are listed in Table 1.

3. The evaluation methods

To evaluate the electrical power and thermal part load, the CGS plant design is divided into subsections for analytical approached purpose. In order to achieve this purpose, it could be separated for each approaching method of the MGT, the EHE and the AHE.

The general behavior of the CGS is determined by ambient temperature is a parameter that has a substantial influence of the efficiency of the MGT operation. Generally, controlling on compressor inlet temperature obviously influences to cycle efficiency and exhaust thermal energy [(3) – (5)].The efficiency of electricity generation was found in the experiments to increase remarkably at low ambient temperature when exhaust thermal energy is decreased. With regard to the MGT actual operation data, temperature difference between the ambient temperature (t_{amb}) and the compressor inlet temperature (t_{CIT}) values are typically 10 to 13˚C that could be called generator cooling effect. Figure 2 shows the influence of t_{amb} on electrical power or electrical efficiency at the rated load of the MGT when electrical efficiency, η_e value is equal to 0.26 at 25˚C of t_{CIT}.

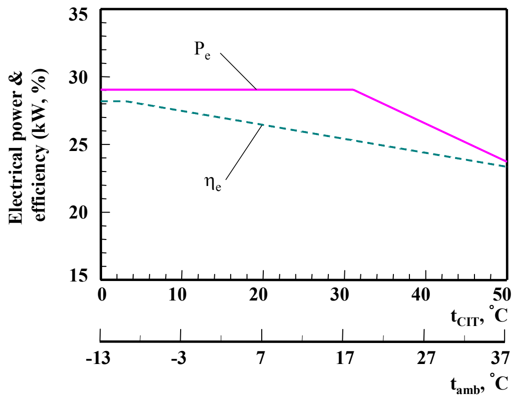

Fig. 2 Influence of ambient air to the MGT

This electrical power, P_e values are based on measuring data and catalogs data of the MGT. Although P_e value is maintained as a constant value in operation till 32˚C of t_{CIT}, η_e is strongly decreased with certain slope angle of horizontal temperature axis when the t_{amb} is increased in environment. As mention for explanation of operating system with designer desires, there could be many control systems for P_e and cycle efficiency. In the present analysis, η_e value could be concentrated to analyze in the below definitions of η_e which displays with fuel thermal energy, $Q_{th,f}$ value related lower heating value of biogas. If P_e value is obtained from measuring or manufacturer supported data, η_e may be evaluated as:

$$\eta_e = \frac{P_e}{Q_{th,f}} \qquad (1)$$

Simulated η_e values are smoothly agreed to catalog data of electrical efficiency. During the operation, there are two kinds of auxiliary machines in the CGS which operate by utilizing some portion of electrical power from the MGT generator. It is easy to achieve electrical demand for auxiliary machines from manufacturing data of biogas compressor and the AHE. Maximum motor horse power of biogas compressor is equal to 3.5kW at 1000rpm and maximum working pressure of 600kPa condition. The AHE associates three types of pump where are installed in beginning of solution heat exchanger, heat medium and cooling tower. The total rated horse power of the AHE is equal to 4.5kW in summer time and 3.5kW in winter time. It should be noted that, the electrical demand of auxiliary machines is equal to 7kW and it is equal to 24% of the MGT output power. P_e value, in the range of 29 to 23kW, could be found in operation at respected ambient temperature. The useful electrical power for other purpose is only 75% of the MGT output power or the range of 22~18kW. Otherwise, if t_{amb} value is chosen at the fixed point of 25˚C in the figure 2, the value of useful electrical power or 22kW could be utilized for other purpose of the CGS.

LHV of biogas is assumed equal to 21.5MJ/m^3 and input energy of the MGT, $Q_{th,f}$ value could be obtained by calculating with this value. If fuel mass flow rate in actual plant, m_f value is equal to 24~30Nm^3/h, $Q_{th,f}$ value may be evaluated from the following equation:

$$Q_{th,f} = m_f \times LHV_{biogas} \tag{2}$$

It should be noted that $Q_{th,f}$ is only input energy of the CGS and it could be also used throughout simulation of thermal efficiency balance.

Irreversibility of mechanical type may be one of the common tools for measuring the mechanical efficiency of the MGT. If symbol $I_{m,T}$ and $I_{m,C}$ is represent to irreversibility of turbine and air compressor, they could be defined as:

$$I_{m,T} = (1-\eta_m)N_t \tag{3}$$

$$I_{m,C} = (1-\eta_m)\frac{N_c}{\eta_m} \tag{4}$$

Furthermore, the other necessary parameters for energy efficiency balance method could be considered. The effectiveness of the recuperator in the MGT is found to be within the range of 50 to 80% assuming a combustion chamber outlet temperature of 850˚C. If t_3 value is obtained in a range of 200 to 400˚C after solving Brayton cycle analysis (neglecting the gas composition in exhaust flue gas), the enthalpy of exhaust gas, h value may be evaluated as:

$$h(t) = \left[65.275+239.566\left(\frac{t}{1000}\right)+2.923\left(\frac{t}{1000}\right)^2+36.76\left(\frac{t}{1000}\right)^3-16.70\left(\frac{t}{1000}\right)^4\right]\times 4.184 \tag{5}$$

If t_{amb} value is normally changed under atmospheric conditions of respected region and revolving speed (n_r), corrected revolving speed, n_{cor} value may be defined as:

$$n_{cor} = \sqrt{\frac{t_N}{t_{amb}}} \times n_r \tag{6}$$

Corrected revolving speed greatly influences to mass flow rate of the MGT, which may be expressed as follows:

$$m_{coa} = (4.2227\times10^{-11} n_{cor}^{\;2}) - (2.5051\times10^{-7} n_{cor}) + 0.025 \tag{7}$$

The exhaust heat energy is calculated from enthalpy change in exhaust stage and compressor inlet stage. If m_1 value is linked to the MGT exhaust temperature with correction factor of mass flow rate for exhaust stage that is assumed to 0.96, exhaust heat energy ($Q_{th,exe}$) with m_1 value may be defined as:

$$Q_{th,exe}/m_1 = \left[h(t_3) - h(t_{CIT})\right] \tag{8}$$

After achieving the necessary parameters of prime mover, exhaust heat energy efficiency can be solved in the following general formula. The energy efficiency is one of the common measuring tools for the CGS system and $\eta_{th,exe}$ value may be defined as:

$$\eta_{th,exe} = \frac{Q_{th,exe}}{Q_{th,f}} \tag{9}$$

On consideration for the EHE here, it is possible to use the ε-NTU-C_R relation in heat recovery system. Several researchers have proposed optimization schemes for exhaust heat energy and heat recovery from the viewpoint of energy demand and reduction of greenhouses gas emissions[(6)–(8)]. In this paper, exhaust heat and exhaust heat recovery is optimized to obtain the total heat supply from the CGS process. The relative thermal size, capacity ratio of the hot gas and water fluids is obviously an important parameter and it is possible to achieve. When capacity ratio, C_R value is equal to 0.055 to 0.067, number of transfer unit, NTU value is equal to 1.767 to 1.769. As mentioned, the hot gas mass flow rate, m_1 value is equal to previous analysis results of the MGT and the cold water mass flow rate, m_w value is shown table1. If assumed ε value is equal to 0.8, C_R value may be defined as a ratio of minimum capacity $(mC_p)_{min}$ and maximum capacity $(mC_p)_{max}$.

The concept of exchanging heat effectiveness, ε and the associated NTU was suggested by Nusselt and the relationship among the NTU, ε and C_R values are also considered to evaluate the performance of thermal heat recovery. If the EHE could be a one-shell and two-tube pass heat exchanger, the relationship may be defined as follow:

$$NTU = -(1+C_R^2)^{-0.5} \ln\left[\frac{2/\varepsilon-1-C_R-(1+C_R^2)^{0.5}}{2/\varepsilon-1-C_R+(1+C_R^2)^{0.5}}\right] \tag{10}$$

One of the definitions of heat exchanger effectiveness is also supported to solve the cold and hot terminal temperatures. It could be four terminal temperatures as input and output temperature of cold and hot fluid. If capacity rate of cold fluid or m_cC_{pc} is greater than m_hC_{ph}, outlet temperature of hot fluid (t_{out}) may be found as:

$$t_{out} = t_3 - \left[\varepsilon\,(t_3 - t_4)\right] \tag{11}$$

Moreover, two relations from the overall heat balance are concerned to solve the performance of the EHE. If input temperature of hot and cold fluid, t_3 and t_4 value could be known in observation, the overall heat transfer or heat recovery, $Q_{th,ehr}$ may be defined as:

$$Q_{th,ehr} = m_cC_{pc}(t_5 - t_4) = m_hC_{ph}(t_3 - t_{out}) \tag{12}$$

If m_1 value is counted in above equation, $Q_{th,her}/m_1$ value could be obtained that can be also compared with performance analysis results of prime mover. After achieving the heat recovery, its efficiency ($\eta_{th,ehr}$) value may be found as:

$$\eta_{th,ehr} = \frac{Q_{th,ehr}}{Q_{th,f}} \tag{13}$$

The amount of heat losses from the EHE is calculated from enthalpy change with respect to outlet temperature of exhaust air, t_{out} and t_{CIT}. If m_1 value could be used in loss definition, exhaust heat loss ($Q_{th,loss}$) may be defined as:

$$Q_{th,loss}/m_1 = \left[h(t_{out}) - h(t_{CIT})\right] \tag{14}$$

If $Q_{th,f}$ value is instead of m_1 value in above equation, heat loss efficiency may be obtained in heat recovery operation.

For the compendious concept for the AHE here considers, there are several ways to approach the performance analysis of the AHE. Figure 3 shows simple block diagram for the AHE. One of the general purposes or to obtain the maximized heat transfer from the AHE, the zero-order model determination could be concerned in the process that is composed with complex part of two fluids in vapor or liquid condition. Simulation is also based on catalog data of manufacturer (that is mentioned in table 1) and control characteristics. Herold and Radermacher could descript a detailed approached method of the zero-order model [(9)]. One of the particular facts is easy to solve and to understand the produced predictions of performance trends. It could be that the internal losses are ignored in this model. The system is required an overall energy balance that the total three heat transfer must be zero. If heat medium input quantity and cooling capacity of the AHE are

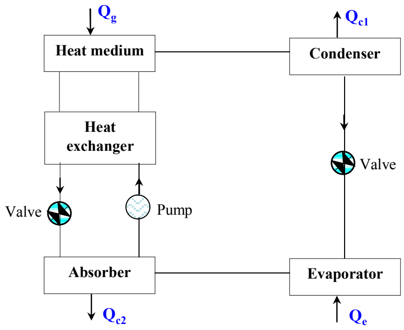

Fig. 3 Schematic diagram of single effect absorption cycle

denoted as Q_g and Q_e respectively and this system is assumed as steady state and ignored environmental heat losses, the sum of heat transfer from condenser and absorber (Q_c) may be defined as:

$$Q_c = Q_g + Q_e \tag{15}$$

If the AHE is operated in heating cycle operation, Q_c value must be zero or it is assumed that no heat transfer in condenser and absorber in system and it could directly depend on standard heating capacity, SQ_h of working fluid. If cooling capacity factor and heat medium flow correction, α_{ccf} and α_{hmfc} value could be obtained from manufacturer supported data, the cooling capacity (Q_e) value may be defined with standard cooling capacity (SQ_e) as following:

$$Q_e = \alpha_{ccf} \times \alpha_{hmfc} \times SQ_e \tag{16}$$

SQe value, standard control outlet temperatures and its mass flow rate are mentioned in table 1. If heating capacity factor, α_{hcf} value could be found in manufacturer supported data, the heating capacity, Q_h value may be defined as:

$$Q_h = \alpha_{hcf} \times \alpha_{hmfc} \times SQ_h \tag{17}$$

After achieving those Q_e or Q_h value, temperature difference between input and output stage of evaporator, generator and cooling tower could be found in following general formula.

$$\Delta t_i = \frac{Q_i}{m_i C_{pi}} \tag{18}$$

Symbol i meant that could be respected region, flow rate and condition of the AHE. The coefficient of performance, COP of the AHE is also one of the important measuring tools for the AHE that could be explained in refrigeration or heating cycle. If COP in refrigeration cycle is denoted in COP_e, it may be defined as:

$$COP_e = \frac{\textit{Useful heat moved or obtained}}{\textit{Energy requirement to drive process}} \tag{19}$$

With regard to relationship between refrigeration cycle and heating cycle of the AHE, COP for heating cycle, COP_h may be defined as:

$$COP_h = COP_e + 1 \tag{20}$$

4. The performance analysis of the CGS

When the thermal load part of the MGT is required to consider, Brayton open-cycle analysis allows the temperatures, specific heat capacities at constant pressure and other parameters of the MGT to be calculated. Figure 4 shows the simulated necessary parameters or exhaust gas temperature (t_3), outlet temperature of the EHE (t_5), inlet temperature of the AHE (t_8), (i.e. subscript 'e' and 'h' is denoted as cooling and heating cycle respectively) and mass flow rate of MGT, m_1 value with t_{CIT}. Although the exhaust temperature is increased with respect to t_{amb} or t_{CIT}, speed and m_1 value of the MGT are obviously reduced. The standard air mass flow rate in manufacturer supported data is 0.31kg/s, that is 0.061kg/s lower than simulated m_1 at t_{amb} value of 15˚C. It is found that simulated t_3 and m_1 value are 245 to 298˚C, and 0.41 to 0.35kg/s respectively. With regarding to exhaust heat energy definition, exhaust heat could be decreased in operation with respect to m_1 value when t_3 value is increasing at high value of t_{CIT}. This is because high t_{amb} or t_{CIT} makes the low air density, and resultant of low mass flow rate into the MGT reduces exhaust heat and its efficiency.

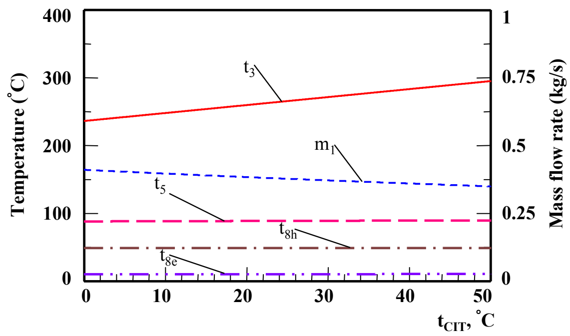

Fig. 4 Temperatures and m_1 verses t_{CIT}

When a further consideration is required, t_5 value of outlet stage in the EHE operation is slightly increased reaching its standard temperature difference for a value of 10˚C that directly depend on the standard medium input data of the AHE. The other input temperature of the AHE, t_{8e} and t_{8h} at evaporator stage could be obtained equal to 10.5 and 48˚C respectively. According to the configuration figure of the CGS, if t_{9e} and t_{9h} values are measured in the actual experiment or obtained from catalog data, temperature difference value of evaporator for each cycle could be calculated equal to 3.54 and 6˚C respectively. It is possible to control the environmental temperature of the specific room for the CGS. It should be known that this AHE could be only operated with standard temperature difference of heat medium and standard output temperature of evaporator.

To consider the irreversibility due to mechanical efficiency, Fig. 5 shows irreversibility of turbine and air compressor with respect to the t_{CIT}. The irreversibility of turbine, $I_{m,T}$ value is gradually increasing to reach its maximum value of 2.7689kW and $I_{m,C}$ value is also grown up to 1.8411kW. It should be noted that irreversibility of turbine is nearly 78% larger than air compressor in working process with variable t_{CIT}.

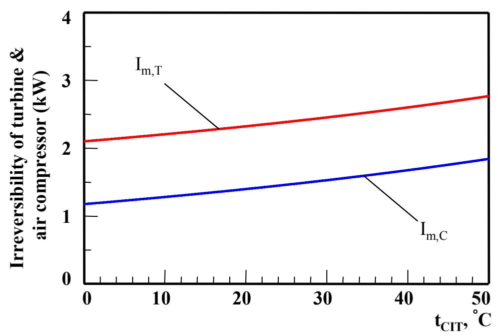

Fig. 5 Irreversibility of turbine and air compressor

Moreover, heat energies could be compared as ratios with electrical power output. The dependence of heat to power ratio on t_{CIT} is shown in Fig. 6 (a). The simulated exhaust heat to power ratio, $Q_{th,exe}/P_e$ or r value is gradually decreasing with the increase of temperature, and the opposite tendency is shown over t_{CIT} value of 32˚C. The lowest ratio value is equal to 2.6 and highest value is equal to 3 at t_{CIT} value of 50˚C that is a good agreement with the experimental data. Its ratio values could be lower than steam turbine standard ratio and larger than open cycle gas turbine standard ratio. The curve characteristic of ratio r value is occurred in simulation procedure. It may be caused as different slope between exhaust thermal energy and electrical power. It is meant that the drop of P_e output value is larger than drop of simulated $Q_{th,exe}$ value in t_{CIT} range of 32 to 50°C. This fact can be shown that electrical power is more sufficient in low t_{CIT} level up to 32°C. At the optimal t_{CIT} range, good electrical efficiency is obtained with acceptable levels of exhaust heat energy.

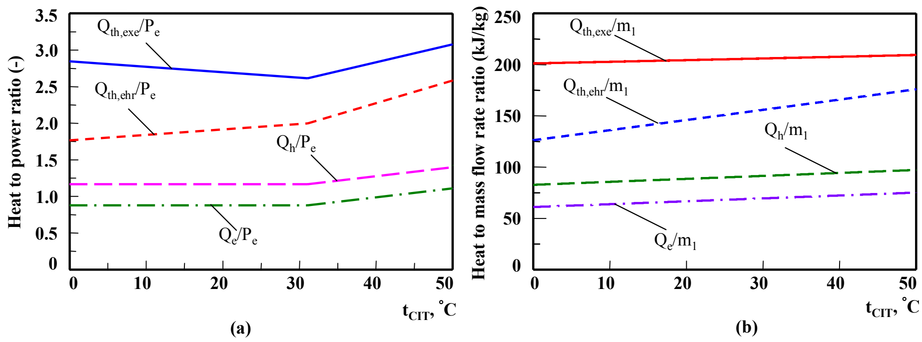

Fig. 6 Heat to power ratios and heat energy with m_1 verse t_{CIT}

To conclude the performance analysis of heat recovery system with the EHE, $Q_{th,ehr}/P_e$ value gradually goes up from 1.72 to 2.6. And then, Q_h and Q_e with P_e value is also reported in figure and those data also point the suitable t_{CIT} range. It should be noted that the system performance could be occurred with sufficient performance within t_{CIT} value of 25 to 35°C. Those data also have a good agreement with the experimental data.

Furthermore, heat energy ratio with m_1 is shown at right hand side in Fig. 6 (b). $Q_{th,exe}/m_1$ value is slightly moving up to 210kJ/kg at t_{CIT} of 50°C. It should be note that m_1 value obviously influences to $Q_{th,exe}$ of the MGT exhaust stage. With regard to ε of the EHE, $Q_{th,ehr}/m_1$ value is obviously increased up to 180kJ/kg at t_{CIT} of 50°C. Q_h and Q_e values are increased in the separated operation cycles that could be defined as refrigeration cycle or heating cycle of the AHE. As mentioned and known, when t_{CIT} value is 0°C, $Q_{th,ehr}$ value is equal to 60% of exhaust thermal heat and gradually increases up to 82% at t_{CIT} of 50°C. This fact also points to the suitable possible t_{CIT} range. The possible percentage of Q_h and Q_e value could be calculated and compared with the exhaust thermal heat. It should be noted that the performance of heating cycle is more sufficient than of cooling cycle in the AHE.

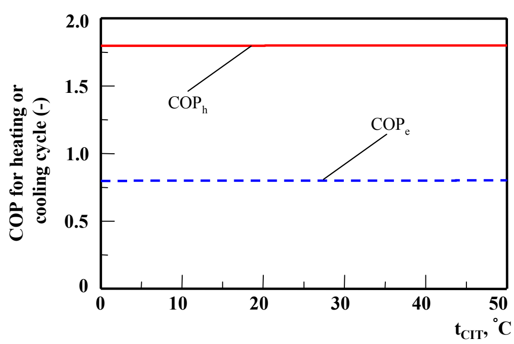

Fig. 7 *COP* value for each case of the AHE

In order to clarify the performance of the AHE, Fig. 7 shows the coefficient of performance for each case. With regard to the standard catalog data, it could be found that great value of *COP* in the AHE operation and it could be nearly constant value. If heat recovery, $Q_{th,ehr}$ value is directly applied into *COP* defined equation, (ignored to standard input heat medium of the AHE) it will be obviously changed. It should be noted that operation with standard values could be achieved a sufficient performance of the AHE than operation with heat recovery. Otherwise, the CGS still remains the useful thermal energy in heat recovery process.

5. The fuel energy saving index and annual performance of the CGS

As mentioned and known, the fuel energy saving index (*FESI*) is one of the particular characteristics for selecting a CGS for various purposes and sectors. If total energy efficiency, η_t value is the summation of η_e and $\eta_{th,exe}$ which those are found in the above performance simulation, *FESI* may be evaluated in following definition;

$$FESI = 1 - \frac{1+r}{r\eta_t\left(\frac{1}{r\eta_e^*}+\frac{1}{\eta_{th}^*}\right)} \tag{21}$$

The reference electrical and thermal η_e^* heat efficiencies (η_e^* and η_{th}^* values) are assumed equal to 0.37 and 0.85 [10]. *FESI* is also utilized as a directed measuring tool of saved fuel consumption in the CGS and it can be explained that a CGS substitutes separate systems of electrical power and thermal heat production with the reference efficiencies η_e^* and η_{th}^* values with respect.

Figure 8 shows influence of heat to power ratio and compressor inlet temperature to *FESI* value. It is possible to obtain two character of *FESI* value in the figure 8a with regard to previous mentioned r value. In the range of 0 to 31˚C of t_{CIT}, upper dotted line is appeared to go down for achieving the value of 0.32 when r value is equal to 2.62. Sequentially, a visible line is also appeared to go down for reaching the minimum value of *FESI*. To concern about the suitable t_{CIT} range or t_{CIT} value of 25 to 35˚C, the figure 8b can show that how *FESI* is working with t_{CIT} and its character in the process. It should be noted that the CGS can reduce the total fuel energy consumption by 33 to 31% in the above suitable t_{CIT} range. It is also can shown that the present CGS has higher efficiency than the efficiency of separate production of heat and electrical power. Otherwise, the CGS can also yield a reduction in emissions of prime mover by using fuel more efficiently.

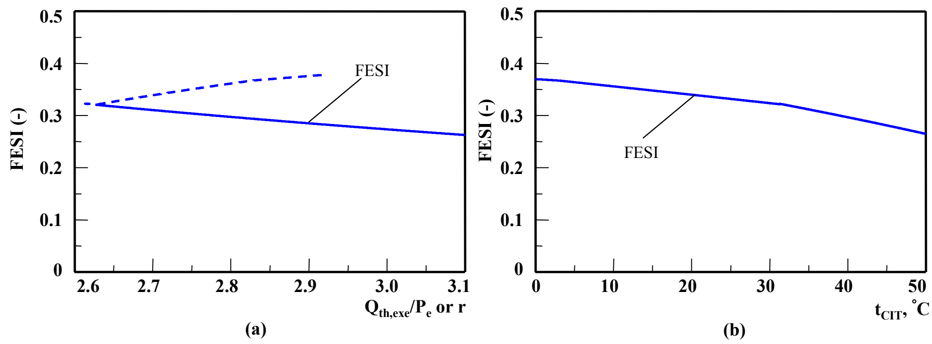

Fig. 8 Influence of r and t_{CIT} to *FESI* value

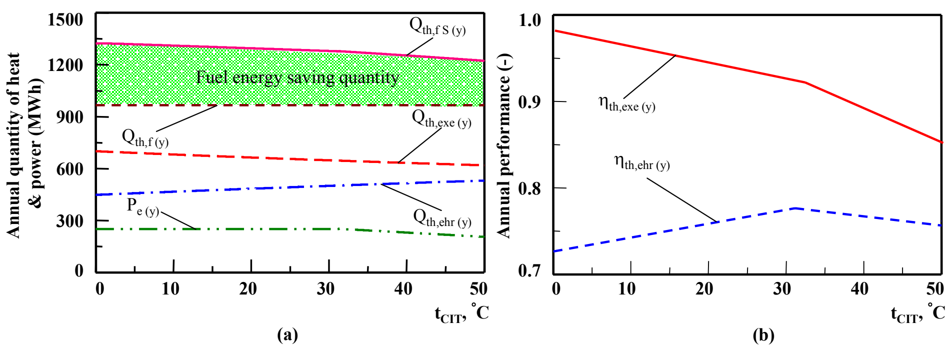

Fig. 9 Annual performances of the CGS

Figure 9 shows possible performance of this CGS for a whole year. (Operating day of the CGS is assumed equal to 360day) $Q_{th,exe}$, $Q_{th,ehr}$ and P_e for a whole year are expressed in Fig. 9 (a).The subscript y and fs is denoted as a year and separated fuel consumption of separated production of P_e and $Q_{th,exe}$. High light region, shaped area in the Fig. 9 (a) represents the possible annual fuel energy saving quantity. Its value is equal to 358 ~ 256MWh and $Q_{th,fs\ (y)}$ value is goes down to reach minimum value of 1222MWh with respect to *FESI*. It should be noted that this CGS can be operated in some kind of sector which its demand is nearly equal to this supply. Moreover, the Fig 9 (b) shows the annual total efficiencies of exhaust thermal heat and heat recovery, $\eta_{th,exe\ (y)}$ and $\eta_{th,ehr\ (y)}$ with t_{CIT}. They are possible to contemplate the influence of performances on a year. $\eta_{th,exe\ (y)}$ value is smoothly moving down till t_{CIT} of 31˚C and then obviously happened with regard to P_e value. $\eta_{th,ehr\ (y)}$ value is obviously increasing to reach the maximum heat recovery of the system till above same t_{CIT} temperature and then goes down. It should be noted that those facts can support to choice the operation temperature of the CGS. It can be also noted that although the heat recovery thermal energy can be increased with the operation in high t_{CIT} range, annual total heat recovery efficiency will be obtained the opposite characteristic in the system beyond t_{CIT} of 31˚C.

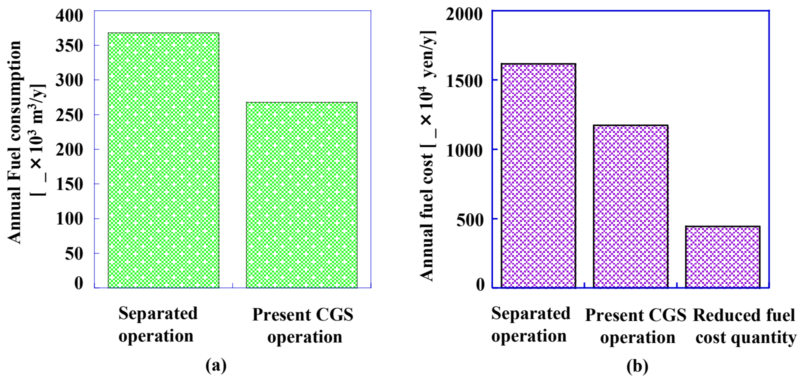

Fig. 10 Annual fuel consumption and cost of fuel

As possible to observe the annual fuel cost for this CGS, it is also supported to consumer need and clarify the system benefit. Figure 10 shows annual fuel consumption and cost of fuel for two systems as the CGS and separate produced electrical power and heat system. Annual fuel consumption for two systems is shown in Fig. 10 (a). The fuel is assumed as natural gas and its unit cost is collected equal to 44yen/m^3 (i.e. it is obtained from KITAGAS home page, that price could be changed on location, type of fuel and usage amount of fuel) As mentioned in Fig. 10 (b), annual fuel cost of separated operation is nearly 1lakh m^3/y greater than fuel cost of the present CGS operation. It should be expected that reducing cost is equal to over 4million yen/y at the fixed t_{CIT} of 25˚C.

6. Conclusion

In this paper, performance for use and energy saving characteristic of the CGS has been successfully investigated under various inlet temperature and other conditions by examining the related operation of the MGT in cogeneration arrangement, the maximum heat recovery and cooling or heating quantity of the AHE. Moreover, a proposed efficiency well-balanced method has been carried out on the operation installed in existing Sewage Treatment Facility.

The mass flow rate of the MGT exhaust stage obviously influences to thermal energy of the CGS and could be contemplated that the CGS operates with sufficient efficiency or not. Actually, the MGT operation mass flow rate is obviously smaller than the gas turbine operation and it could be supported to consider the advanced combined cogeneration systems. Some of system is need to increase both the mass flow rate and exhaust temperature as the important roles in the operation.

It could be found that remarkable electrical power and acceptable thermal exhaust heat energy can be obtained when the compressor inlet temperature is equal to the range of 25 to 35˚C. The maximum heat recovery energy can be also found in above procedure and the annual total heat recovery efficiency can also reveal the suitable inlet temperature range. This CGS can be operated in some kind of sector which its annual demand is equal to 400 ~500MWh. Furthermore, it could be need the multiple installation of micro gas turbine cogeneration system for large business.

This system can reduce the total energy consumption rate by at least 30% of the fuel energy consumption of the separate produced power and heat energy system. It is meant that related emissions and fuel cost are also reduced in system by this ratio when the system gains maximum exhaust heat recovery. It could be obviously influenced to environmental issues by utilization of fuel effectively.

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