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Combined Thermal Efficiency Evaluation and Dynamic Characteristics of Micro Gas Turbine Centered Co-generation System

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ABSTRACT

Recently, research and development of micro co-generation system is conducted actively. However, few studies were reported on the combined thermal efficiency based on the actual data measured all through the year. In this paper, firstly, we report actual data based combined thermal efficiency evaluation of micro gas turbine centered micro co-generation system especially focused on ambient temperature effect. During the measurement we found temperature fluctuations of heat transfer medium under the condition that the ambient temperature is less than 28 degree Celsius. Then, secondly, we report the analysis of this phenomenon by formulating the relation of heat balance taking account of nonlinearity of valve characteristics.

NOMENCLATURE

Q	: energy [kj]
T	: temperature [℃]
T_C	: Time Constant [s]
T_L	: Time Lag [s]
a	: coefficient of performance of boiler and absorption type refrigerator
b	: constant [kjs/℃]
c	: constant [kj/℃]
d	: initial temperature [℃]
t	: time [sec]
Φ	: function describing the operation of three-way valve for controlling boiler
Ψ	: function describing the operation of three-way valve for controlling absorption type refrigerator

Subscript

boi	: boiler
ch, 1	: chilled water
co, 2	: cool water
g	: exhaust gas
hm	: heat medium
loss	: heat loss
r	: absorption type refrigerator

INTRODUCTION

Recent deregulation of electric supply system in Japan is promoting active research and development of small size cogeneration systems (CGS). Although lots of studies on planning and design of CGS are reported, combined thermal efficiency evaluation based on actually measured data especially focused on the ambient temperature effect has not yet studied.

Author's group performed combined thermal efficiency measurements all through the year at J/POWER Chigasaki R&D center where target system is composed by oil-fired 28kW class Micro Gas Turbine (MGT), small size boiler for recovering exhaust heat and small size absorption type refrigerator. We measured temperature fluctuations of heat medium, chilled water and cool water as well as flow rate, generated electric power and fuel consumption rate. During the measurement we found interesting temperature fluctuating phenomena in the following two cases; the first case is that where the ambient temperature is lower than expected and the second case is that MGT is operated at electrical load over 25kW.

At the latter part of this paper, we made mathematical model for explaining such temperature fluctuations and proposed a countermeasure for attenuating such phenomena.

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EXPERIMENTAL APPRATUS

In Fig.1, system layout of CGS system under study is shown where main components are oil fuel tank, pump, MGT, fan for cooling MGT package, boiler, absorption type refrigerator, fan coil and cooling tower. Between the boiler and the refrigerator, two three-way valves for controlling absorption type refrigerator and boiler are equipped and corresponding functions of which are to shut down heat supply to the refrigerator when the temperature of chilled water becomes below 6 degree Celsius and to shut down heat supply from the boiler when the temperature of the heat medium goes over 95 degree Celsius, respectively.

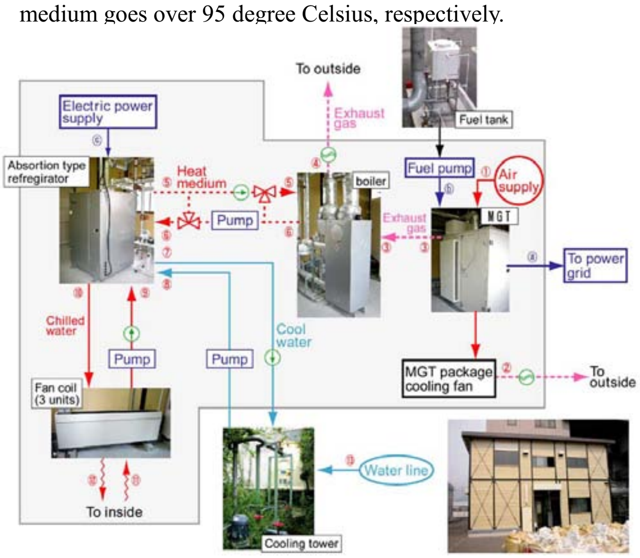

Fig.1 CGS System

EXPERIMENTAL RESULTS

We measured temperature, flow rate of heat medium, generated electric power and fuel consumption rate by changing electric power load from 5 to 25kW by 5kW at summer season (July), intermediate season (September) and winter season (February). In the following, we will show typical recorded temperature data of outside air, MGT inlet, Chilled water (to and from refrigerator), Cool water (to and from refrigerator), Heat medium (to and from refrigerator) and Exhaust gas (from boiler and MGT).

Intermediate season

Temperature fluctuation data measured from September 26 to 29, 2001 are shown in Fig. 2 and 3, where electric power load are 20 and 25kW, respectively.

In Fig.2, small amplitude temperature fluctuations of heat medium and chilled water are found where the three-way valve for controlling refrigerator starts in bypass operation. In this case, ambient temperature is 26.4 degree Celsius which is lower than the designed temperature, therefore, the heat supply to the refrigerator by heat medium is shut down by the three-way valve.

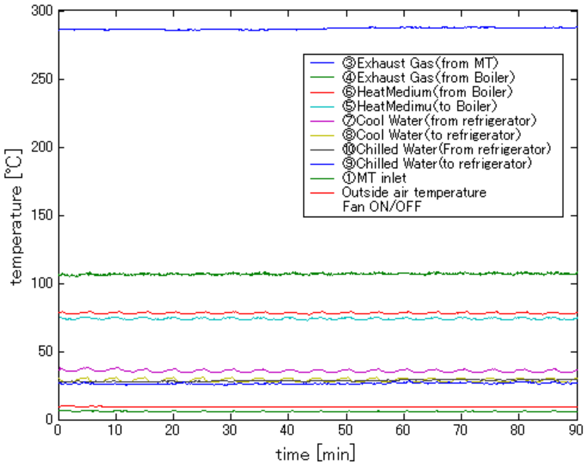

Fig.2 MGT 20kW（Intermediate season, Cooling）

In Fig.3, large amplitude temperature fluctuation is observed in exhaust gas from boiler together with heat medium and chilled water temperature fluctuations where the three-way valves for controlling boiler also set in bypass operation in addition to one for controlling the refrigerator. In this figure, temperature fluctuations of cool water is also found because cooling fan on-off operation is linked to the operation of the three-way valve for controlling refrigerator.

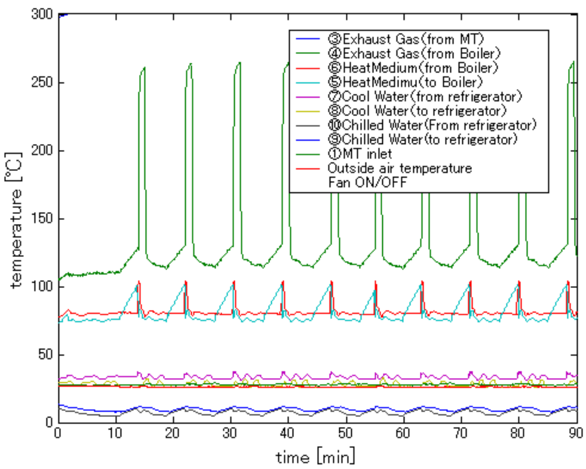

Fig.3 MGT 25kW（Intermediate season, Cooling）

Summer season

In Figs.4-6, observed temperature fluctuations in summer season measured in late July 2002 are shown where electric power load are 10, 15 and 20kW, respectively.

In.Fig.4, ambient temperature is 28.1 degree Celsius which is almost the same ambient temperature in intermediate season.

Observed data shows similar fluctuations as the case of intermediate season. In Figs. 5 and 6, no temperature fluctuation is observed because ambient temperature in these cases is 32 degree Celsius which is over designed temperature. Hence, no bypass operation of the three-way control valve is set in. Here, we have to comment on the fact that intake air temperature in summer season is high, therefore, maximum electric power output increases up to 20kW even if catalogue electric power of MGT is 28kW.

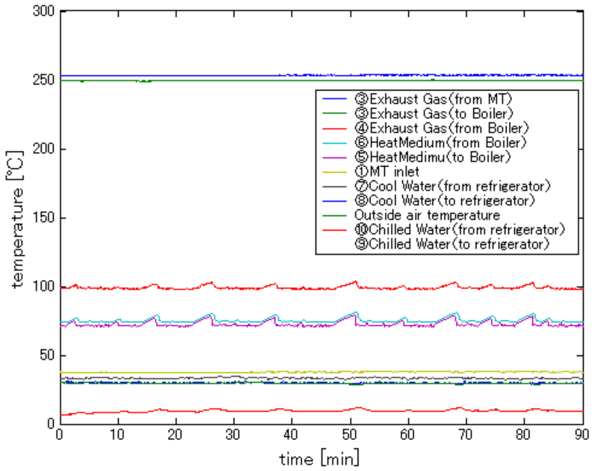

Fig.4 MGT 10kW（Summer season, Cooling）

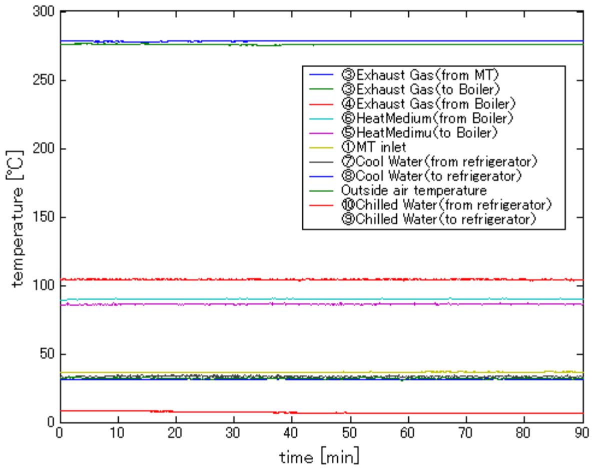

Fig.5 MGT 15kW（Summer season, Cooling）

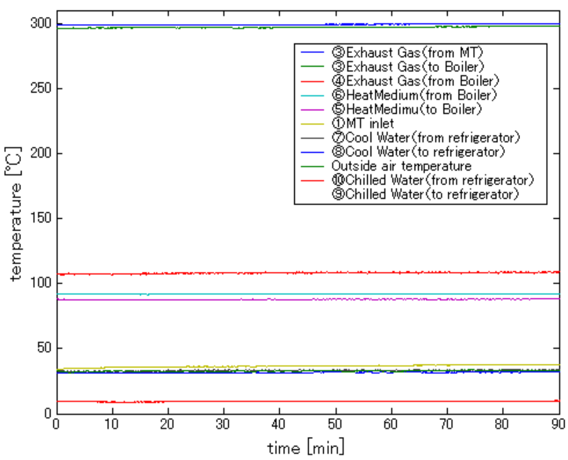

Fig.6 MGT 20kW（Summer season, Cooling）

Winter season

Fig. 7 and 8 show temperature fluctuating data in winter season measured in mid January 2002 where electric power load are 20 and 25kW, respectively and exhaust gas is used for heating. From Fig.7, we can find that no fluctuating phenomena are observed. However, in Fig.8, large amplitude temperature fluctuations in exhaust gas from the boiler and heat medium are observed at larger electric power load. In this case, bypass operation for the three-way valve for controlling boiler set in.

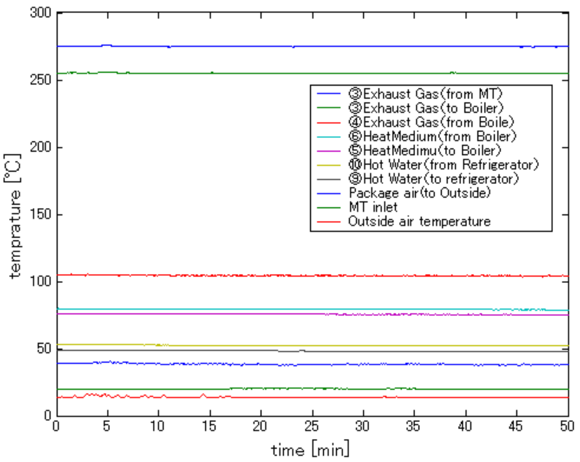

Fig. 7 MGT 20kW (Cold season, Heating)

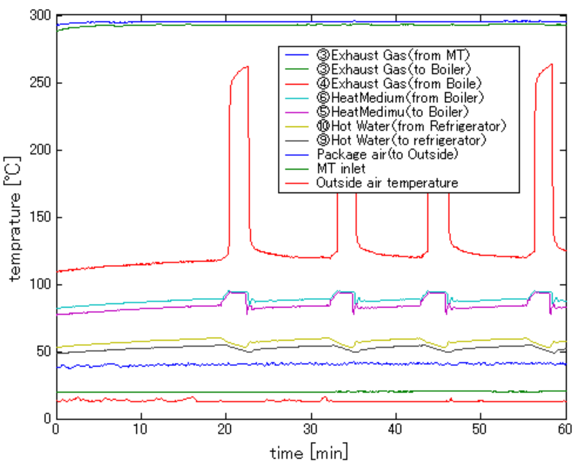

Fig.8 MGT 23kW (Cold season, Heating)

In the experiments, temperature measurement was done continuously for two hours with each electric power load and fuel consumption rate and flow rate were measured every thirty minutes.

CGS COMBINED EFFICIENCY

Based on the measured data, we evaluated CGS combined efficiency as a function of MGT generating power corresponding to three seasons. The definition of CGS combined efficiency is formulated by the following relation,

CGS combined efficiency = (MGT generating power + utilized amount of heat from exhaust gas) / (heat generated from supplied fuel)

Fig.9 shows evaluated result of CGS combined efficiency. As one can easily expect, CGS combined efficiency takes maximum in cold season and minimum in intermediate season. An interesting finding in this figure is that CGS combined efficiency in winter season takes maximum in the case of 15kW. Then, we calculated loss contribution rate of each element constituting CGS system based on exergy analysis to investigate the reason.

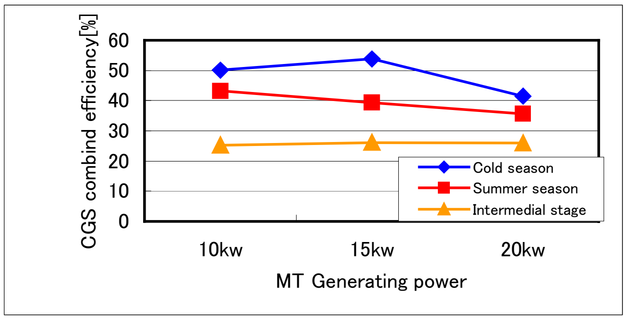

Fig.9 CGS combined efficiency

In Fig.10, loss contribution rate of MGT, boiler, refrigerator and the pipe connecting MGT and the boiler is shown as a function of MGT generating electric power. Loss contribution rate of MGT is large compared with the other elements and it takes minimum at 15kW, which is the reason why CGS combined efficiency takes maximum at 15kW in winter season.

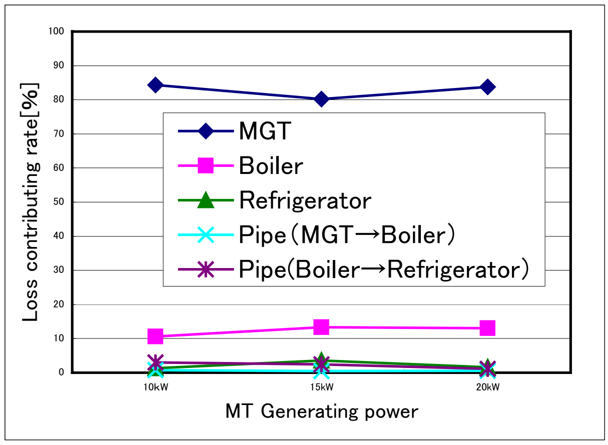

Fig.10 Exergy loss contribution rate in winter season

ANALYSIS OF DYNAMIC CHARACTERISTICS

In the following, the mechanism of temperature fluctuations is analyzed. First, we will construct mathematical model taking account of nonlinearity included in the system and secondly, we perform simulation with temperature fluctuations of heat medium, chilled water and cool water.

Mathematical modeling of thermal system

In Fig.11, heat balance with heat medium, chilled water and cool water is shown. Exhaust gas energy Qg from MGT is supplied to heat medium at the boiler and this energy is supplied to the refrigerator. Chilled water is cooled by making use of the energy Qhm supplied from heat medium inside the refrigerator and is heated by ambient temperature by fan coils. Cool water deprives energy once the energy Qhm is supplied from heat medium to the refrigerator and the deprived energy is dumped at a coolig tower.

Depending on the temperature of heat medium and chilled water, on-off state of three-way valves is determined. Here, we have to pay special attention that on-off state of a fan for the cooling tower is controlled by the same signal controlling the three-way valve for

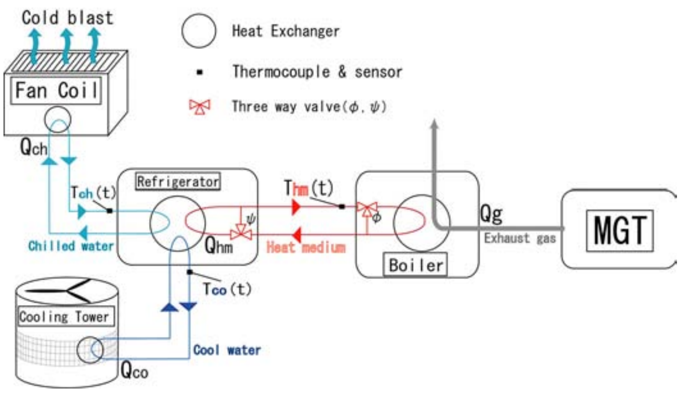

Fig.11 Diagram of thermal system

the refrigerator. Then, equations governing heat balance with these three loops in Fig.11 can be described as follows together with functions,

$$\phi a_b Q_g - \psi Q_{hm} - Q_{loss} = b_{hm}\dot{T}_{hm} + c_{hm}T_{hm} \quad \cdots\cdots(1)$$

$$\psi a_{r1}Q_{hm} + Q_{ch} = -b_{ch}\dot{T}_{ch} - c_{ch}T_{ch} \quad \cdots\cdots(2)$$

$$\psi a_{r2}Q_{hm} - Q_{co} = b_{co}\dot{T}_{co} + c_{co}T_{co} \quad \cdots\cdots(3)$$

$$\phi = \begin{cases} 1 & T_{hm}(t) < 95 \\ 0 & T_{hm}(t) > 95 \end{cases}$$

$$\psi = \begin{cases} 1 & T_{ch}(t) > 6 \\ 0 & T_{ch}(t) < 6 \end{cases}$$

where ϕ , ψ denote switching function for heat medium and chilled water, respectively.

In Fig.12, we represent above equations as a form of block diagram.

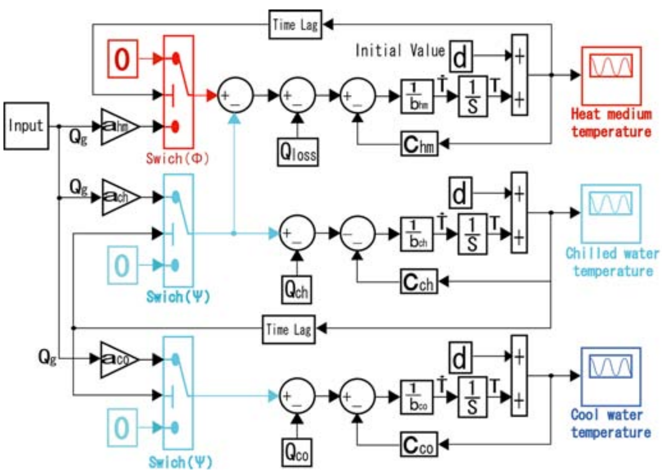

Fig.12 Block Diagram

Step Input: Qg

Output:

Heat Medium Temperature

Chilled Water Temperature

Cool Water Temperature

Switch(ϕ) : three-way valve for controlling boiler

Switch(ψ) : three-way valve for controlling refrigerator

Time Lag : 50 ~ 60 sec

Solving Eq. (1) ~ (3) yields the following output equation, where d denotes initial temperature for each loops.

$$T_{hm}(t) = \frac{\mathbb{Q}_{hm1}}{c_{hm}} - \frac{(\mathbb{Q}_{hm1} - c_{hm}d_{hm})}{c_{hm}}\exp(-\frac{c_{hm}}{b_{hm}}t)、 T_{hm}(0) = d_{hm} \quad (4)$$

$$T_{ch}(t) = -\frac{\mathbb{Q}_{ch}}{c_{ch}} + \frac{(\mathbb{Q}_{ch} + c_{ch}d_{ch})}{c_{ch}}\exp(-\frac{c_{ch}}{b_{ch}}t)、 T_{ch}(0) = d_{ch} \quad (5)$$

$$T_{co}(t) = \frac{\mathbb{Q}_{co}}{c_{co}} - \frac{(\mathbb{Q}_{co} - c_{co}d_{co})}{c_{co}}\exp(-\frac{c_{co}}{b_{co}}t)、 T_{co}(0) = d_{co} \quad (6)$$

In Fig. 13, the definition of time lag is shown where the case with three-way valve controlling refrigerator is taken as an example. When the temperature of chilled water decreases down to 6 degree Celsius, bypass of heat medium is expected to be in operation to maintain the temperature over 6 degree Celsius. However, as is shown schematically in Fig. 13, actual temperature is 5.4 degree Celsius at 20 kW electric power load. Time lag is defined by the time which the temperature decreases from 6 to 5.4 degree Celsius. Here we should notice that time lag contributed by a mechanical switch (2 sec) is also included in our calculation.

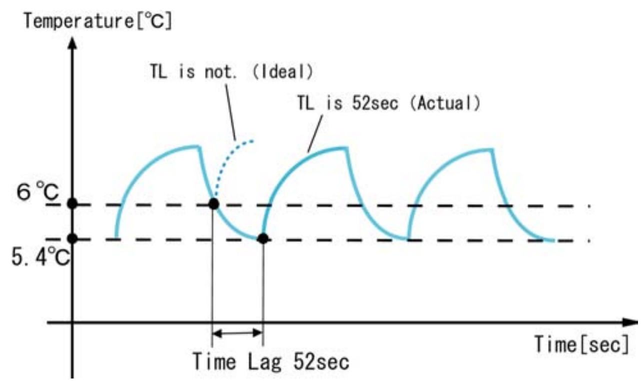

Fig.13 Time lag (three-way valve for refrigerator)

<u>Simulated results</u> (Case I: Intermediate season: ϕ =1)

When we start simulation, initial value, terminal value and the time from the starting state to the terminal state are necessary. To evaluate these values, as a preparatory work, we calculate coefficients a, b, c, d as follows,

①Solving equations (1) – (3) and obtain T(t).

②Determining Qg from exhaust gas energy.

③Determining terminal value of Q/c from experimental data.

④Determining c from ② and ③.

⑤Determining initial value d from experimental data.

⑥Determining the time from the starting state to the terminal state Tt based on experimental data.

⑦Substituting Q, c, d, Tt, Thm (experimentally determined terminal value) and obtain b.

Then, taking the algorithm of three-way valve into account, we can evaluate the temperature fluctuation phenomena. In this case, since the temperature of heat medium is always less than 95 degree Celsius, heat medium passes through the loop all the time i.e. bypass for the boiler side does not occur which means ϕ =1.

In Fig.14, calculated results in the case of 20kW electric power load are shown in contrast with experimental data where oscillatory behavior of heat medium, cool water and chilled water is reproduced by the combination of the temperature increasing stage and decreasing stage. One can find that the calculated results represent the waveform experimentally observed pretty well.

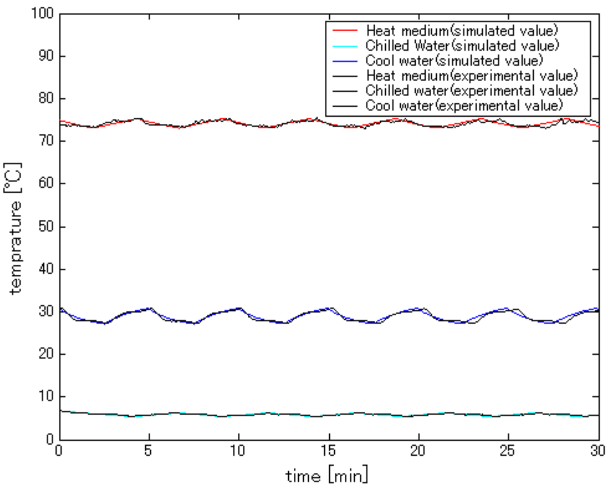

Fig.14 Simulation results (ϕ =1)

<u>Simulated results</u> (Case II: Intermediate season: ϕ =0)

In the following, we try to simulate the case with 25kW electric power load where three-way valves for both boiler side and refrigerator side are in operation. Even though the modeling is more complicated than case I, as the same manner as the case I, we finally can simulate the fluctuating temperature of heat medium, cool water and chilled water as shown in Fig.15 – 17 where ①, ②, ③, ④ show initial values.

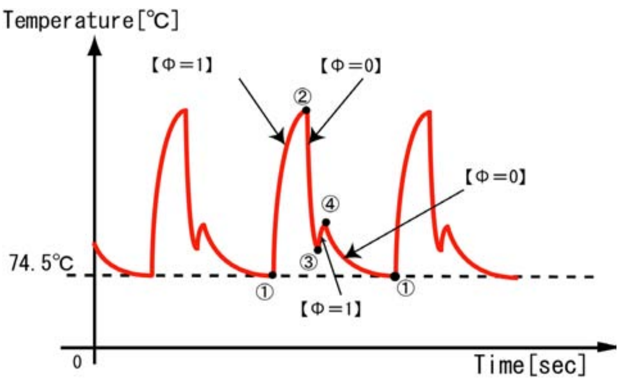

Fig.15 Simulation results of heat medium temperature

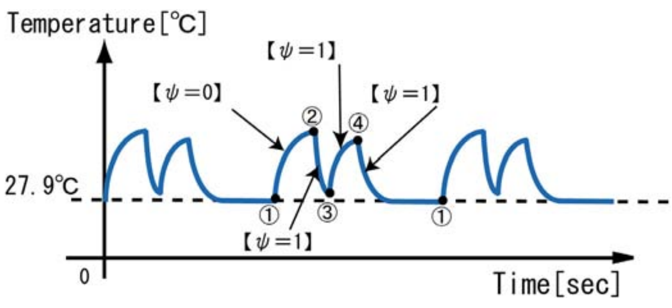

Fig.16 Simulation results of cool water temperature

Simulation results are shown in Fig.18 in contrast with measured data. We can find that the simulation results represent actual data very well and the scenario for the temperature fluctuations we proposed in this paper is verified.

In Fig.19, we will discuss a little bit on the time delay due to the boiler side three-way valve. When the temperature of heat medium increases up to 95 degree Celsius, bypass operation of heat medium is anticipated to maintain the temperature under 95 degree Celsius.

However, as is shown schematically in Fig. 19, in reality, temperature rise up around 101.1 degree Celsius at 25 kW electric power load. Time lag in this case is defined by the time which the temperature increases from 95 to 101.1 degree Celsius. Here, once again, we should notice that time lag contributed by a mechanical switch (2 sec) is also included in the calculation.

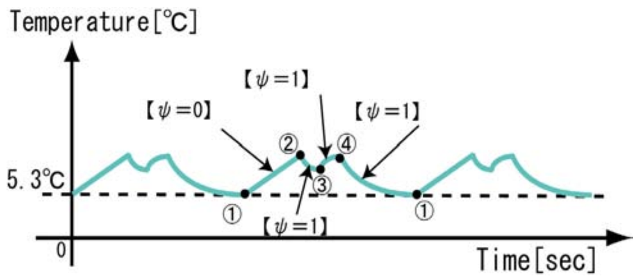

Fig.17 Simulation results of chilled water temperature

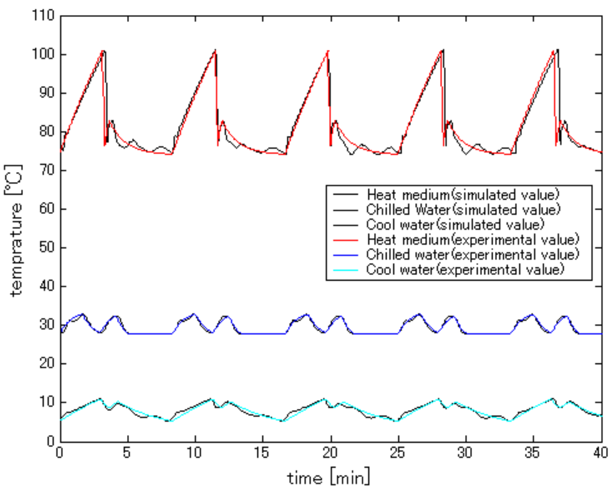

Fig.18 Simulation results (ϕ =0)

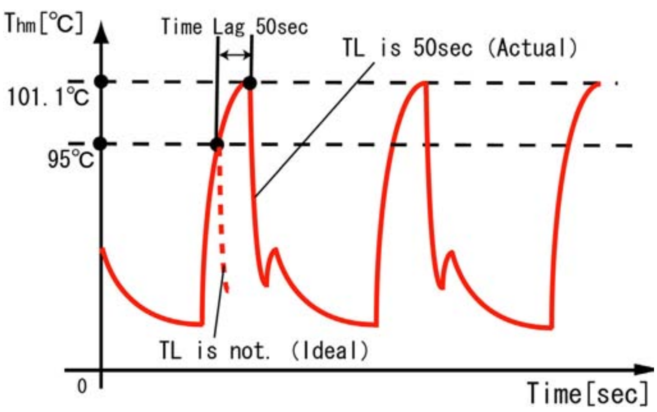

Fig.19 Time Lag (Boiler side valve)

COUNTERMEASURE

In the above simulations, we successfully identified the mechanism of temperature fluctuations. Now, we will propose a countermeasure to mitigate these fluctuations.

In the following, taking Case I as an example, we will explain the mechanism of countermeasure. As is already introduced in Fig. 13, there exists 52 seconds time delay (T_L) when the refrigerator side thee-way valve operates which causes over cooling of chilled water. Therefore, in order to avoid temperature fluctuations, we have to erase T_L. Then, we examined from the standpoint of non-dimensional T_L (T_L/T_C) evaluated as follows,

$$T_L/T_C = T_L \times (c/b) \tag{7}$$

where T_C denotes time constant found in Eq.(6) and is b/c.

Assuming b equals to constant, we can reduce non-dimensional time delay T_L/T_C by reducing c. However, if the value of c decreases, T_C becomes larger which means it will take longer time to settle down as is shown in Fig.20.

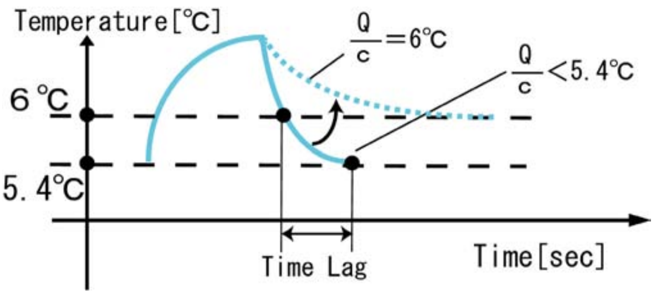

Fig.20 Ideal limit value

Calculation examples on the ideal value of c for eliminating time lag are shown in Table 1 where we studied the effect of ambient temperature on non-dimensional T_L and time constant T_C. From this table, we can find that temperature fluctuations can be avoided by decreasing the value c. As an actual countermeasure to realize such a condition, increasing the amount of the heat exchange at fan coil is recommended.

Table 1 Ideal value of c for eliminating time lag

ambient temperature	C	TL×c/b	TC(=b/c)
24. 7℃	0. 39→0. 350	0. 372→0. 334	161. 3→179. 7
26. 4℃	1. 33→0. 670	0. 114→0. 057	454. 5→902. 2
27. 5℃	0. 84→0. 583	0. 103→0. 071	496. 7→714. 8

CONCLUSIONS

In this paper, we discussed combined thermal efficiency evaluation of micro gas turbine centered micro co-generation system especially focused on ambient temperature effect. During the measurement, we found temperature fluctuations of heat transfer medium under the condition that the ambient temperature is less than 28 degree Celsius. Then, we analyzed the phenomena by formulating the relation of heat balance of the loops taking account of nonlinear valve characteristics and successfully identified the mechanism. Finally, we proposed a countermeasure to mitigate the fluctuation based on the physical considerations.

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