Modeling of the Main Component of the Bottoming Stage in an Advanced Co-Generation System on the Operational Planning*

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The advanced co-generation system (ACGS) was built at Waseda University in 1999. The ACGS consists of a gas turbine, a steam turbine, an ammonia-water mixture turbine, an ammonia absorption refrigerator and an ice storage system. These turbine systems have a cascade relation upon the utilization of energy. Meanwhile, the ammonia-water mixture turbine and the ammonia absorption refrigerator located in the bottoming stage of the ACGS to utilize the lower temperature steam. In this paper, a mathematical model of the bottoming stage of the ACGS is established according to the experimental data. And, the operational planning of the bottoming stage with three cases of the cooling demand during the nighttime is also discussed.

Key Words: Energy Saving, Refrigeration, Power Plant, Combined Cycle, Ice Storage, Operational Planning

1. Introduction

The advanced co-generation system (ACGS) was built at Waseda University. The ACGS consists of a gas turbine, a steam turbine, an ammonia absorption refrigerator, an ammonia-water mixture turbine and an ice storage system. It has three stages as described below:

- (1) Topping stage: the gas turbine cycle using town gas,
- (2) Medium stage: the steam turbine cycle using the steam produced in a heat recovery steam generator and
- (3) Bottoming stage: the two cycles consisting of an ammonia absorption refrigerator (AAR) and an ammonia-water mixture turbine (AWM/T)⁽¹⁾, using exhaust heat from the steam turbine.

Figure 1 shows a diagram of the ACGS. The energy taken out from each turbine is used for power generation and the AAR is used for the air-conditioning during the daytime in parallel with storing heat in the ice storage system to meet cooling demand during the nighttime.

A feature of a co-generation system is that simultaneous supplying of the heat and electricity can be realized while obtaining high thermal efficiency. However, each demand changes with time and the rational as well as flexible operational planning for the system is essential. In the ACGS, the combined cycle, which consists of the gas turbine and back-pressure type steam turbine, is arranged in the topping-medium stage and the output ratio of electric power and heat is relatively fixed. Then, it becomes important to plan the systems operation, which can always change the output ratio in proportion to demand, while the heat source steam is properly distributed in power generation and refrigeration system placed in the bottoming stage⁽²⁾.

In this paper, it is aimed to model a mathematical plan limited only to the bottoming stage using the optimization to establish the method, which derives the optimum operational planning of the bottoming stage.

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Fig. 4 Characteristic of ice charge performance

turbine, required when the AAR system is operated.

Also, allocation of the heat for the direct sending path and ice storing path is shown in Eq. (5).

$$X_{AAR}(k) = X_{DSP}(k) + X_{ISP}(k) \quad (5)$$

3.3 Ice storage system

To begin with, the heat balance relation of the ice storage system circumference in the heat charge/discharge mode is shown in Eq. (6).

$$S_{ISS}(k) = (1 - \eta_{LOSS})S_{ISS}(k-1) + \{X_{ISP}(k) - X_{ISS}(k)\}\Delta t \quad (6)$$

Here, $S_{ISS}(k)$ in the equation is the heat storage volume in a period k , and it is an integrated value of heat storage volume in one previous period $k-1$ and heat quantity which is stored (or discharged) in a period k . η_{LOSS} shows the heat radiation loss factor and Δt shows the sampling period.

Next, the performance diagram of the heat storage system, which can be applied to the mathematical plan model from the data attained in the experiment, is deduced in order to consider the nonlinear characteristic of the ice storage system.

Figure 4 shows the relationship between maximum heat charge, which can be supplied to the ice storage system in one period and heat stored volume in the previous period. In this figure, the result of the experiment that the heat charge drops as the heat stored volume increases, has remarkably appeared. The nonlinear characteristic of ice storage indicates this fact.

On the basis of this result, we divided the performance of charge into three parts, and each part is determined as a linear equation.

When the maximum heat supplied to the ice storage system in period k is made to be $\bar{X}_{ISP}(k)$ and the heat stored volume in the previous period is made to be $S_{ISS}(k-1)$, the relation of these two variables can be formulated in a linear expression using $Z_{ISP}^{part1}(k)$, $Z_{ISP}^{part2}(k)$ and $Z_{ISP}^{part3}(k)$. For example, in Fig. 4, $Z_{ISP}^{part1}(k)$ is a 0-1 variable, which shows 1 if the heat stored volume is under 321 [kWh] and shows 0 if it is over it. By this nonlinear characteristic, the following constraint is added to the ice charging

Fig. 5 Characteristic of ice discharge performance

mode.

$$X_{ISP}(k) \leq \bar{X}_{ISP}(k) \quad (7)$$

$$\begin{aligned} \bar{X}_{ISP}(k) = & c_{ISP}^{part1} Z_{ISP}^{part1}(k-1) - a_{ISP}^{part2} S_{ISP}^{part2}(k-1) \\ & + c_{ISP}^{part2} Z_{ISP}^{part2}(k-1) - a_{ISP}^{part3} S_{ISP}^{part3}(k-1) \\ & + c_{ISP}^{part3} Z_{ISP}^{part3}(k-1) \end{aligned} \quad (8)$$

$$S_{ISS}(k) = S_{ISP}^{part1}(k) + S_{ISP}^{part2}(k) + S_{ISP}^{part3}(k) \quad (9)$$

$$\begin{aligned} 0 \leq S_{ISP}^{part1} & \leq 321 Z_{ISP}^{part1}, \\ 321 Z_{ISP}^{part2} & \leq S_{ISP}^{part2} \leq 600 Z_{ISP}^{part2}, \\ 600 Z_{ISP}^{part3} & \leq S_{ISP}^{part3} \leq 837 Z_{ISP}^{part3} \end{aligned} \quad (10)$$

$$\begin{aligned} Z_{ISP}^{part1} & = \begin{cases} 1 & (\text{if } S_{ISS} \leq 321) \\ 0 & (\text{if } S_{ISS} \geq 321) \end{cases} \\ Z_{ISP}^{part2} & = \begin{cases} 1 & (\text{if } 321 \leq S_{ISS} \leq 600) \\ 0 & (\text{if } 321 \geq S_{ISS}, 600 \leq S_{ISS}) \end{cases} \\ Z_{ISP}^{part3} & = \begin{cases} 1 & (\text{if } 600 \leq S_{ISS}) \\ 0 & (\text{if } 600 \geq S_{ISS}) \end{cases} \end{aligned} \quad (11)$$

On the other hand, from the result of the heat discharge experiment, the aspect is proven that the discharge heat drops linearly as the heat discharge advances. Using this result, Fig. 5 shows the relationship between the possible maximum values of the discharge heat taken out from the ice storage system in one period and heat storing volume in the previous period. Here, the numerical value got from the experiment is plotted, and it has been approximated by the straight line.

Here in Fig. 5, when each $S_{ISS}(k)$ and $\bar{X}_{ISS}(k)$ was put as a transverse axis and vertical axis, the relationship is shown by following equation.

$$\begin{aligned} \bar{X}_{ISS}(k) = & a_{ISS}^{part1} S_{ISS}^{part1}(k-1) + c_{ISS}^{part1} Z_{ISS}^{part1}(k-1) \\ & + a_{ISS}^{part2} S_{ISS}^{part2}(k-1) + c_{ISS}^{part2} Z_{ISS}^{part2}(k-1) \end{aligned} \quad (12)$$

$$S_{ISS}(k) = S_{ISS}^{part1}(k) + S_{ISS}^{part2}(k) \quad (13)$$

$$0 \leq S_{ISS}^{part1} \leq 221 Z_{ISS}^{part1}, 221 Z_{ISS}^{part2} \leq S_{ISS}^{part2} \leq 837 Z_{ISS}^{part2} \quad (14)$$

$$\begin{aligned} Z_{ISS}^{part1} & = \begin{cases} 1 & (\text{if } S_{ISS} \leq 221) \\ 0 & (\text{if } S_{ISS} \geq 221) \end{cases} \\ Z_{ISS}^{part2} & = \begin{cases} 1 & (\text{if } S_{ISS} \geq 221) \\ 0 & (\text{if } S_{ISS} \leq 221) \end{cases} \end{aligned} \quad (15)$$

a_{ISS} is a slope and c_{ISS} is an intercept, here.

Taking the above non-linearity of the characteristic, the following restriction is added to the heat discharge mode.

$$X_{ISS}(k) \leq \bar{X}_{ISS}(k) \quad (16)$$

3.4 Another major constraints

A constraint on the heat source is described as follows.

$$E_{AWM}(k) + E_{AAR}(k) \leq E_{STS}(k) \quad (17)$$

However, period “ k ”, where the heat source is supplied, is limited to the 3 - 12 period and it is made to be 0 [kg/h] for the other period.

A condition of the cooling demand shows the following equation.

$$X_{DSF}(k) + X_{ISS}(k) \geq Q_{CD}(k) \quad (18)$$

$Q_{CD}(k)$ is the cooling demand of period k here, and this value becomes the input value of the model.

3.5 Objective function

The objective function $f(x)$ is aimed to maximize the sum of the output of the AWM/T during the daytime.

$$f(X) = \sum_{k=1}^{24} \{X_{AWM}(k) \Delta t\} \rightarrow \max \quad (19)$$

4. Calculation Conditions and Results

4.1 Calculation conditions

The 1800 [m²] gross floor space of the research building which installed the ACGS is the object of the air conditioning load, and the optimum heat source allocation and equipment output to the bottoming stage component in each period is derived by inputting the cooling demand data every hour during the representative August day. The cooling demand patterns during the daytime and the nighttime are shown in Fig. 6. The pattern during the daytime shows a double peak pattern having the first peak at 9:00 (term 3) and the second at 14:00 and 15:00 (terms 8 and 9). In general, a cooling demand during the daytime traces this pattern.

On the other hand, we have three cases of cooling demand during the nighttime as follows:

- (1) constant demand (case 1),
- (2) convex curve with downward-sloping (case 2) and
- (3) curve diminishing with time (case 3).

The total amount of cooling demand during the nighttime for each case is the same. Figure 6 shows a pattern of the cooling demand during the daytime and three pattern cases during the nighttime given as input data.

4.2 Calculation results

On the basis of the optimum solution derived using the optimization model, the output power of the AWM/T and the AAR in each period, and change of heat storing volume in the ice storage system, are shown below. The results show that there is an operational planning suitable for each of the three cases, which are given different night cooling demand patterns.

Fig. 6 Cooling demand pattern

Fig. 7 Output power of the AWM/T

For all the cases, it is possible to take out stabilized heat effectively with no deficiency and excess from the ice storage system during overnight and early morning when the AWM/T and the AAR are not operating. And each case concentrates the heat storage possibly in the later period in order to avoid the heat radiation loss in the heat storage period.

Figure 7 shows the output power of the AWM/T. The figure says that the AWM/T scarcely changes its output pattern among the three cases. The output pattern shows its high-power operation in the beginning and latter term and restrained-power operation in the middle. However, the large difference is found in the fourth term. That results from the ice storage pattern in the same term.

Figure 8 shows the output power of the AAR. Every pattern, except the fourth term, shows that the AAR increases its power from the beginning to the middle and decreases the power in the latter term.

The ice charging to the ice storage system is shown in Fig. 9. And Fig. 10 shows the ice stored in the ice storage system. In Fig. 9, the ice charging for each case begins in the fourth term, but its volume differs from each other. In the second case, heat charging volume exceeds about 300 kWh in the seventh period of time, and the lowering of the heat charge performance begins. Therefore, heat supply to the ice storage system from the eighth period is limited. Furthermore, because the ice is stored at almost 600 kWh in the 11th term, the heat supplied is reduced

Fig. 8 Output power of the AAR

Fig. 9 Ice charging to the ice storage system

Fig. 10 Ice stored in the ice storage system

more in the 12th term.

In case 3, the ice stored reaches the second part of ice storage volume in the 8th term. Therefore, the performance reduction of charge is found after the 9th term. In this case the total amount of ice stored doesn't exceed 600 kWh.

Although the total amount of cooling demand for each case is the same, the total amount of ice to be stored is different because of the non-linear characteristic of charge/discharge performance of the ice storage system. In case 2, the performance of the heat discharge is lower because the cooling demand is larger than the other two cases in the early morning. Then, a larger ice storage volume becomes necessary. As a result, in case 2, the total amount of stored heat is largest among the three cases.

In the meantime, in case of 3, since a night cooling

Table 1 Total output of components

	Case1	Case2	Case3
Total output power of AWM/T [kWh]	366.0	365.1	367.2
Total output power of AAR [kWh]	1738.9	1746.8	1728.8
Amount of ice stored in 12th term [kWh]	628.9	668.1	586.3
Amount of ice stored in 3rd term [kWh]	48.9	77.6	9.1

demand drops with time, the effect of heat discharge performance decrement of the ice storage system does not become a problem and the loss is small. As a result, the total amount of ice stored is also less than the other 2 cases. The total output power of the AWM/T and the AAR and the amount of ice stored are summarized in Table 1. As described above, since the loss is the least in case 3, output power of the AWM/T is largest and the AAR can be operated efficiently. The area used in the ice storage system depends on the difference of amount of ice stored between the 3rd term and 12th term. It is found that the cooling demand pattern affects the total amount of ice and the area used in the ice storage system.

5. Conclusion

The following results are obtained from this study.

(1) A mathematical model to derive the optimum operational planning of the ACGS bottoming stage was established based on the experimental data.

(2) The mathematical model is formulated as the MILP by considering the non-linear characteristics of charge/discharge of the ice storage system.

(3) It was confirmed that there is an optimum operational planning for night cooling demand, which differs according to the case due to nonlinear heat charge/discharge characteristics of the ice storage system.

(4) It was found that the pattern of night cooling demand affects the total amount of ice and the area used in the ice storage system.

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