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OPTIMAL OPERATIONAL PLANNING OF COGENERATION SYSTEMS WITH MICROTURBINE AND DESICCANT AIR-CONDITIONING UNITS

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ABSTRACT

Economic and energy-saving characteristics of cogeneration systems with microturbine and desiccant air-conditioning units are investigated on system operational planning. An optimization approach is adopted to rationally evaluate these characteristics. In this approach, on/off and rated/part load status of operation of equipment and energy flow rates are determined so as to minimize the hourly energy charge subject to satisfaction of energy demand requirements. In this optimization problem, performance characteristics of the microturbine and desiccant air-conditioning units are modeled in consideration of the influence due to ambient air temperature. Moreover, the influence due to ambient air humidity is also considered in the desiccant air-conditioning unit using the psychrometric diagram. The implementation of the numerical analysis method, discussed in this paper, to two cogeneration systems, clearly shows economic and operational benefits of using desiccant air-conditioning.

NOMENCLATURE

A : coefficient for unit conversion from J to Wh, Wh/J
a : latent heat of vaporization of water, kJ/kg
B, C : coefficients of proportion, kg/m
b : specific heat at constant pressure of air, kJ/(kg·°C)
c : specific heat of water, kJ/(kg·°C)
E : electricity, kWh/h
F : natural gas consumption, m^3/h
h : specific enthalpy, kJ/kg
J : hourly energy charge, yen/h
m : mass flow rate, kg/h
p, q, r, p', q', r' : performance characteristic values of equipment
Q : heat flow rate, kWh/h
s : sensible heat factor
t : temperature, °C
u : input energy flow rate of equipment, m^3/h, kWh/h
v : velocity of process air, m/h
x : absolute humidity, kg/kg
y : output energy flow rate of equipment, kWh/h
δ : binary variable expressing on/off status of operation
η_c : evaporative effectiveness of EC2
η_e : effectiveness on regeneration air side of SHW
η_s : effectiveness on process air side of SHW
φ : unit cost of energy charge, yen/m^3, yen/kWh
$\underline{(\ )}, \overline{(\ )}$: lower and upper limits

Equipment symbols (subscripts)
AHU : air handling unit
BG : gas-fired boiler
DCU : desiccant air-conditioning unit
DW : desiccant wheel
EC1, EC2 : evaporative coolers
EP : device for purchasing electricity
HE : heat exchanger
MT : microturbine cogeneration unit
RG : gas-fired absorption refrigerator
RX : flue gas absorption refrigerator
SHW : sensible heat wheel

Superscripts
a : auxiliary machinery
ac : auxiliary machinery in space cooling mode
ah : auxiliary machinery in space heating mode
c : cold water
ca : cold air
d : demand
dc : space cooling demand
dh : space heating demand
dw : hot water demand
h : hot water
ha : hot air
in : input
out : output
x : flue gas
xc : flue gas in space cooling mode
xh : flue gas in space heating mode

Subscripts

1 ~ 10 : air state points
buy : purchase
disp : heat disposal
E : electricity
e : regeneration air
G : natural gas
i : intake air
t : temperature
wb : wet-bulb
x : absolute humidity

INTRODUCTION

Microturbine generator units with capacities under 100 kW have recently been paid attention to as distributed power sources [1]. This is because the microturbine generator units are expected to have the advantages of low initial capital cost, high electrical generating efficiency for their small capacities, high reliability, low NOx emission and so on. Moreover, they can be used as main equipment of cogeneration systems for commercial and public purposes [2]. In using them as cogeneration units, there are some means of utilizing flue gas generated from them. One is for desiccant air-conditioning [3, 4]. Desiccant air-conditioning units can supply space cooling and heating energy directly by utilizing flue gas, and they have the advantage of high energy-saving because of dehumidification by the desiccant without subcooling energy. Some field tests for combined microturbine and desiccant air-conditioning units are performed in Japan [5]. However, economic and energy-saving characteristics of the cogeneration systems with the combined microturbine and desiccant air-conditioning units have not sufficiently been investigated.

The purpose of this paper is to construct a model in order to evaluate the economic and energy-saving characteristics on the operation of the aforementioned cogeneration systems as the first stage, and is also to investigate their characteristics through numerical studies. An optimization approach is adopted to evaluate the characteristics rationally. In the following, the configurations of microturbine cogeneration systems investigated here are first described. Second, the evaluation method based on the optimization is described. Finally, the numerical studies are carried out on systems installed in office buildings and hospitals.

SYSTEM CONFIGURATIONS

Figure 1 shows a schematic diagram and energy flow of a microturbine cogeneration system investigated here. The system has the microturbine cogeneration unit (MT), a flue gas absorption refrigerator (RX), a gas-fired absorption refrigerator (RG), a gas-fired boiler (BG), an air handling unit (AHU), a desiccant air-conditioning unit (DCU), and a device for purchasing electricity (EP). Solid, dot-dash, dot-dot-dash, dotted, broken, thick solid and thick broken lines show flows of hot water, electricity, cold air, cold water, natural gas, flue gas and hot air, respectively. Moreover, symbols for energy flow rates are listed in the Nomenclature.

In the system, electricity is supplied to users by purchasing electricity through EP from an outside electric power company and by operating MT. Electricity is also used to drive cooling towers and other auxiliary machinery. Flue gas generated from MT is used as heat sources for RX and DCU. Surplus exhaust heat is disposed of to the atmosphere. Cold air for space cooling is supplied by AHU using cold water from RX and RG as heat sinks and by DCU. Hot air for space heating is supplied by AHU using hot water from RX, RG and BG as heat sources and by DCU. Hot water for miscellaneous purposes is supplied by BG.

OPTIMAL OPERATIONAL PLANNING PROBLEM

To evaluate the economic benefit of combining the microturbine and desiccant air-conditioning units on the operation, the optimization approach is adopted. In this approach, on/off and rated/part load status of operation of equipment and energy flow rates at each junction for each energy flow are determined so as to minimize the hourly energy charge subject to satisfaction of energy demand requirements. This optimal operational planning problem

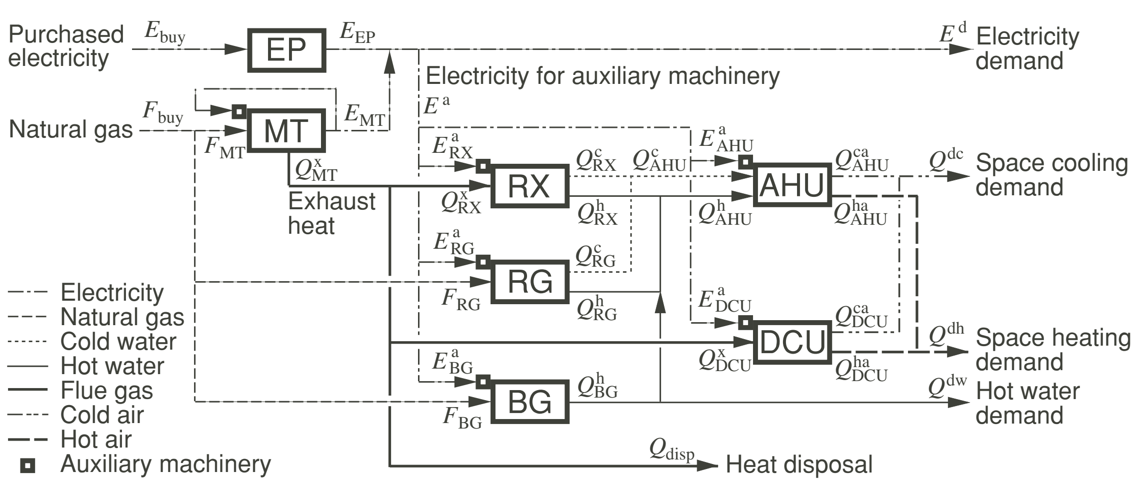

Fig. 1 Schematic diagram and energy flow of a microturbine cogeneration system

is described briefly below.

Mathematical Formulation

Performance characteristics of equipment. First, performance characteristics of pieces of equipment are considered as constraints in the optimization problem, and are formulated as relationships between input and output energy flow rates. These relationships are expressed by the following piecewise linear approximations [6]:

$$\left.\begin{array}{l} y = pu + q\delta \\ \underline{u}\delta \le u \le \overline{u}\,\delta \end{array}\right\}, \tag{1}$$

where u and y denote input and output energy flow rates, respectively; δ denotes a binary variable expressing the on/off status of operation of equipment; p and q denote performance characteristic values; and $\underline{u}$ and $\overline{u}$ denote lower and upper limits of the input energy flow rate, respectively. In Eq. (1), if $\delta = 0$, then $u = y = 0$, which expresses off status of operation. On the other hand, if $\delta = 1$, then $y = au + b$ and $\underline{u} \le u \le \overline{u}$, which expresses on status of operation.

It is supposed that coefficients p, q, $\underline{u}$ and $\overline{u}$ in Eq. (1) for RX, RG and BG are constants, and that those for MT, DCU and AHU are functions of temperature and absolute humidity of ambient air. This is because the performance characteristics of MT, DCU and AHU are influenced more greatly than those of other pieces of equipment. As examples, the performance characteristics of MT, DCU and AHU are described briefly below.

(a) Microturbine cogeneration unit (MT). The relationships between the natural gas consumption F_{MT} and the generated electricity E_{MT} or the amount of generated flue gas $Q^{\mathrm{x}}_{\mathrm{MT}}$ are expressed by the following equations:

$$\left.\begin{array}{l} E_{\mathrm{MT}} = p_{\mathrm{MT}}(t_{\mathrm{i}})F_{\mathrm{MT}} + q_{\mathrm{MT}}(t_{\mathrm{i}})\delta_{\mathrm{MT}} \\ Q^{\mathrm{x}}_{\mathrm{MT}} = p^{\mathrm{x}}_{\mathrm{MT}}(t_{\mathrm{i}})F_{\mathrm{MT}} + q^{\mathrm{x}}_{\mathrm{MT}}(t_{\mathrm{i}})\delta_{\mathrm{MT}} \\ \underline{F}_{\mathrm{MT}}(t_{\mathrm{i}})\delta_{\mathrm{MT}} \le F_{\mathrm{MT}} \le \overline{F}_{\mathrm{MT}}(t_{\mathrm{i}})\delta_{\mathrm{MT}} \end{array}\right\}, \tag{2}$$

where δ_{MT} denotes a binary variable expressing the on/off status of operation of MT; coefficients p_{MT}, q_{MT}, $p^{\mathrm{x}}_{\mathrm{MT}}$ and $q^{\mathrm{x}}_{\mathrm{MT}}$ are the performance characteristic values; $\overline{F}_{\mathrm{MT}}$ and $\underline{F}_{\mathrm{MT}}$ are the upper and lower limits on the natural gas consumption, respectively; and t_{i} denotes intake air temperature. These performance characteristic values together with the upper and lower limits are considered as functions of intake air temperature t_{i}.

(b) Desiccant air-conditioning unit (DCU). The relationships between the input energy flow rate $Q^{\mathrm{x}}_{\mathrm{DCU}}$ and the heat flow rate for space cooling air $Q^{\mathrm{ca}}_{\mathrm{DCU}}$ or the heat flow rate for space heating air $Q^{\mathrm{ha}}_{\mathrm{DCU}}$ are influenced by temperature and absolute humidity of ambient air, because DCU dehumidifies and cools or heats ambient air directly. To consider these influences, the relationships are formulated according to psychrometric processes in space cooling or heating modes of DCU. The relationship in space cooling mode is first described below, and is followed by the relationship in space heating mode.

Figure 2 shows component configuration and air flows in space cooling mode of DCU, and Fig. 3 shows the corresponding psychrometric process [3]. In Fig. 2, DCU has a desiccant wheel (DW), a sensible heat wheel (SHW), a heat exchanger (HE) and evaporative coolers (EC1 and EC2). The air stream of DCU consists mainly of a process air one expressing States ① to ④ and of a regeneration air one expressing States ⑤ to ⑨. In the process air stream, warm and wet ambient air at State ① is introduced as a process air, passes through DW, and results in hot and dry air at State ②. This temperature increase is due to the release of condensation heat of the water vapor when moisture is removed by the desiccant. This hot and dry air at State ② is cooled sensibly as it passes through SHW. The cool and dry air at State ③ is further evaporatively cooled and humidified by EC1 to air at State ④, and is then introduced to the conditioned space. On the other hand, in the regeneration air stream, the warm and humid regeneration air at State ⑤ from the conditioned space is evaporatively cooled by EC2 to air at State ⑥. The air at State ⑥ is heated sensibly as it passes through SHW. One part of the hot and humid air at State ⑦ is further heated up to the required regeneration temperature of the desiccant by the heat from the flue gas of MT through HE, and the other part bypasses HE and DW. This is because the regeneration air mass flow rate of DW is less than the process air mass flow rate if the regeneration temperature is sufficiently high. The hot and humid air at State ⑧ passes through DW in order to regenerate the desiccant. Even though this air is humid, it is sufficiently hot to remove the moisture from the desiccant. Finally, the warm and very humid air at State ⑨ is mixed with the bypass air at State ⑦, and the mixed air at State ⑩ is exhausted to the surroundings. In the aforementioned process, the input energy flow rate $Q^{\mathrm{x}}_{\mathrm{DCU}}$ and the output energy flow rate $Q^{\mathrm{ca}}_{\mathrm{DCU}}$ are expressed by the following equations:

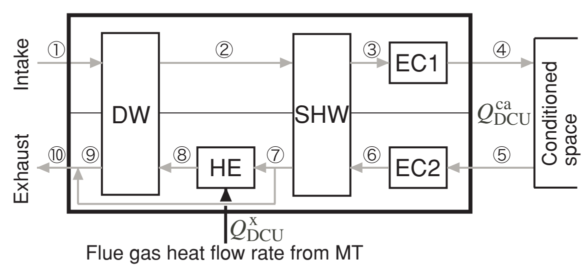

Fig. 2 Schematic diagram and air flow in space cooling mode of DCU

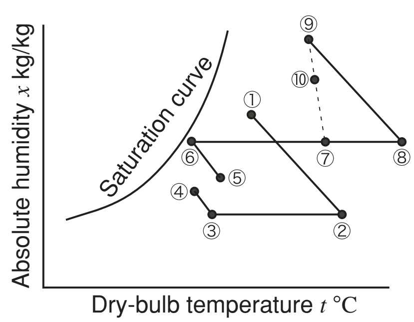

Fig. 3 Psychrometric diagram in space cooling mode of DCU

$$\left.\begin{aligned}Q_{\mathrm{DCU}}^{\mathrm{x}} &= Am_{\mathrm{e}}(h_8 - h_7)\\ Q_{\mathrm{DCU}}^{\mathrm{ca}} &= Am(h_5 - h_4)\end{aligned}\right\}, \tag{3}$$

where m and m_{e} denote the air mass flow rates of each stream and the regeneration stream passing through HE/DW, respectively; h denotes specific enthalpy; A denotes a coefficient for unit conversion from J to Wh, i.e., 1/3600; and subscript numbers correspond to air states in Figs. 2 and 3. At this point, let t and x be respectively temperature and absolute humidity, and the specific enthalpy h is expressed as follows:

$$h(t,x) = ax + bt + cxt, \tag{4}$$

where coefficients a, b and c denote latent heat of vaporization, specific heat at constant pressure of air and that of water, respectively. Therefore, Eq. (3) is rewritten by the following equations:

$$\left.\begin{aligned}Q_{\mathrm{DCU}}^{\mathrm{x}} &= Am_{\mathrm{e}}[h(t_8,x_8) - h(t_7,x_7)]\\ Q_{\mathrm{DCU}}^{\mathrm{ca}} &= Am[h(t_5,x_5) - h(t_4,x_4)]\end{aligned}\right\}. \tag{5}$$

The temperature and absolute humidity in Eq. (5) are calculated according to the psychrometric process shown in Fig. 3, and each process is formulated below.

Processes ① to ② and ⑧ to ⑨: It is supposed that the air with the required regeneration temperature t_8 is sufficiently supplied to DW, and that DW rotates at the speed most suitable for dehumidification and regeneration. Under this supposition, only process ① to ② is formulated here. Then, it can be assumed from the reference [4] that temperature t_2 and absolute humidity x_2 at State ② after dehumidification are in proportion to temperature t_1 and absolute humidity x_1 at State ①, and are in inverse proportion to process air velocity v. Therefore, t_2 and x_2 are expressed by the following equations:

$$\left.\begin{aligned}t_2 &= \frac{p_{\mathrm{t}}x_1 + q_{\mathrm{t}}t_1 + r_{\mathrm{t}}}{v} + p'_{\mathrm{t}}x_1 + q'_{\mathrm{t}}t_1 + r'_{\mathrm{t}}\\ x_2 &= \frac{p_{\mathrm{x}}x_1 + q_{\mathrm{x}}t_1 + r_{\mathrm{x}}}{v} + p'_{\mathrm{x}}x_1 + q'_{\mathrm{x}}t_1 + r'_{\mathrm{x}}\end{aligned}\right\}, \tag{6}$$

where coefficients p_{t}, q_{t}, r_{t}, $p'_{\mathrm{t}}, q'_{\mathrm{t}}, r'_{\mathrm{t}}$, p_{x}, q_{x}, r_{x}, $p'_{\mathrm{x}}, q'_{\mathrm{x}}$ and r'_{x} are dehumidification performance characteristic values. Moreover, the air mass flow rates on the process and regeneration stream sides of DW, i.e., respectively m and m_{e}, are assumed to be in proportion to the velocity v as follows:

$$\left.\begin{aligned}m &= Bv\\ m_{\mathrm{e}} &= Cv\end{aligned}\right\}, \tag{7}$$

where B and C denote coefficients of proportion.

Processes ② to ③ and ⑥ to ⑦: Let η_{s} and η_{e} be the effectiveness on process and regeneration air sides of SHW, respectively, and the relationships among temperature t_2, t_3, t_6 and t_7 are expressed by the following equations:

$$\left.\begin{aligned}\frac{t_2 - t_3}{t_2 - t_6} &= \eta_{\mathrm{s}}\\ \frac{t_7 - t_6}{t_2 - t_6} &= \eta_{\mathrm{e}}\end{aligned}\right\}. \tag{8}$$

Moreover, the relationships among absolute humidity x_2, x_3, x_6 and x_7 are expressed by the following equations:

$$\left.\begin{aligned}x_3 &= x_2\\ x_7 &= x_6\end{aligned}\right\}. \tag{9}$$

Processes ③ to ④ and ⑤ to ⑥: These processes are with constant enthalpies. Therefore, the relationships among specific enthalpies are expressed by the following equations:

$$\left.\begin{aligned}h(t_4,x_4) &= h(t_3,x_3)\\ h(t_6,x_6) &= h(t_5,x_5)\end{aligned}\right\}. \tag{10}$$

Moreover, let s and η_{c} be respectively the sensible heat factor and the evaporative effectiveness of EC2, and the following equations hold:

$$\left.\begin{aligned}\frac{h(t_5,x_4) - h(t_4,x_4)}{h(t_5,x_5) - h(t_4,x_4)} &= s\\ \frac{t_5 - t_6}{t_5 - t_{5,\mathrm{wb}}} &= \eta_{\mathrm{c}}\end{aligned}\right\}, \tag{11}$$

where subscript wb denotes wet-bulb temperature.

Process ⑦ to ⑧: The relationship in this process is expressed by the following equation:

$$x_8 = x_7. \tag{12}$$

Substituting Eqs. (4), (6)~(12) into Eq. (5) and arranging them, $Q_{\mathrm{DCU}}^{\mathrm{x}}$ and $Q_{\mathrm{DCU}}^{\mathrm{ca}}$ become functions of v, t_1, x_1, t_5, x_5 and t_8 as follows:

$$\left.\begin{aligned}Q_{\mathrm{DCU}}^{\mathrm{x}} &= p^{\mathrm{in}}(t_1,x_1,t_5,x_5,t_8)v + q^{\mathrm{in}}(t_1,x_1,t_5,x_5,t_8)\\ &\quad + r^{\mathrm{in}}(t_1,x_1,t_5,x_5,t_8)\frac{1}{v}\\ Q_{\mathrm{DCU}}^{\mathrm{ca}} &= p^{\mathrm{out}}(t_1,x_1,t_5,x_5,t_8)v + q^{\mathrm{out}}(t_1,x_1,t_5,x_5,t_8)\\ &\quad + r^{\mathrm{out}}(t_1,x_1,t_5,x_5,t_8)\frac{1}{v}\end{aligned}\right\}, \tag{13}$$

where $p^{\mathrm{in}}, q^{\mathrm{in}}, r^{\mathrm{in}}, p^{\mathrm{out}}, q^{\mathrm{out}}$ and r^{out} are performance characteristic values, which are functions of t_1, x_1 t_5, x_5 and t_8. At this point, the values of r^{in} and r^{out} are very small within the ranges of t and x considered here. Therefore, neglecting the terms with them, both $Q_{\mathrm{DCU}}^{\mathrm{x}}$ and $Q_{\mathrm{DCU}}^{\mathrm{ca}}$ become linear functions with respect to v. Then, since v can be eliminated from both functions, the relationship between $Q_{\mathrm{DCU}}^{\mathrm{x}}$ and $Q_{\mathrm{DCU}}^{\mathrm{ca}}$ is finally expressed by the following equations:

$$
\left.\begin{aligned}
Q^{\mathrm{ca}}_{\mathrm{DCU}} &= p^{\mathrm{ca}}_{\mathrm{DCU}}(t_1,x_1,t_5,x_5,t_8)Q^{\mathrm{x}}_{\mathrm{DCU}} \\
&\quad + q^{\mathrm{ca}}_{\mathrm{DCU}}(t_1,x_1,t_5,x_5,t_8)\delta^{\mathrm{ca}}_{\mathrm{DCU}} \\
Q^{\mathrm{x}}_{\mathrm{DCU}} &\geq \underline{Q}^{\mathrm{xc}}_{\mathrm{DCU}}(t_1,x_1,t_5,x_5,t_8,\underline{v})\delta^{\mathrm{ca}}_{\mathrm{DCU}} \\
Q^{\mathrm{x}}_{\mathrm{DCU}} &\leq \overline{Q}^{\mathrm{xc}}_{\mathrm{DCU}}(t_1,x_1,t_5,x_5,t_8,\overline{v})\delta^{\mathrm{ca}}_{\mathrm{DCU}}
\end{aligned}\right\}, \tag{14}
$$

where $\delta^{\mathrm{ca}}_{\mathrm{DCU}}$ denotes a binary variable expressing the on/off status of operation in space cooling mode of DCU; coefficients $p^{\mathrm{ca}}_{\mathrm{DCU}}$ and $q^{\mathrm{ca}}_{\mathrm{DCU}}$ are the performance characteristic values, which are functions of t_1, x_1 t_5, x_5 and t_8; $\overline{v}$ and $\underline{v}$ are the upper and lower limits of velocity v; and $\overline{Q}^{\mathrm{xc}}_{\mathrm{DCU}}$ and $\underline{Q}^{\mathrm{xc}}_{\mathrm{DCU}}$ are the upper and lower limits on the input energy flow rate, respectively, which are functions of t_1, x_1 t_5, x_5, t_8 and $\overline{v}$ or $\underline{v}$. If t_1, x_1 t_5, x_5, t_8, $\overline{v}$ and $\underline{v}$, i.e., ambient temperature and absolute humidity, conditioned space temperature and humidity, upper and lower bounds of process air velocity, and regeneration temperature of the desiccant, are given, $p^{\mathrm{ca}}_{\mathrm{DCU}}$, $q^{\mathrm{ca}}_{\mathrm{DCU}}$, $\overline{Q}^{\mathrm{xc}}_{\mathrm{DCU}}$ and $\underline{Q}^{\mathrm{xc}}_{\mathrm{DCU}}$ become constants. Moreover, electricity for auxiliary machinery of DCU, i.e., $E^{\mathrm{ac}}_{\mathrm{DCU}}$, is expressed as follows:

$$
E^{\mathrm{ac}}_{\mathrm{DCU}} = p^{\mathrm{ac}}_{\mathrm{DCU}}Q^{\mathrm{x}}_{\mathrm{DCU}} + q^{\mathrm{ac}}_{\mathrm{DCU}}\delta^{\mathrm{ca}}_{\mathrm{DCU}}, \tag{15}
$$

where $p^{\mathrm{ac}}_{\mathrm{DCU}}$ and $q^{\mathrm{ac}}_{\mathrm{DCU}}$ are performance characteristic values.

On the other hand, Fig. 4 shows component configuration in space heating mode of DCU, and Fig. 5 shows the corresponding psychrometric process. In the similar way to space cooling mode, the performance characteristics in space heating mode are formulated according to the psychrometric process shown in Fig. 5. The relationships between $Q^{\mathrm{x}}_{\mathrm{DCU}}$ and $Q^{\mathrm{ha}}_{\mathrm{DCU}}$ or $E^{\mathrm{ah}}_{\mathrm{DCU}}$ are

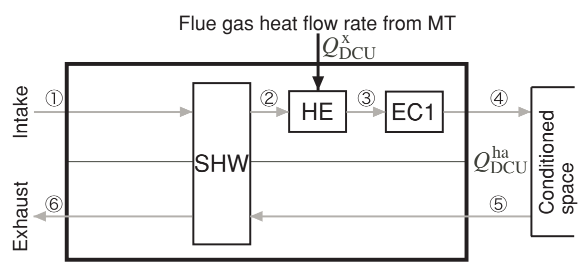

Fig. 4 Schematic diagram and air flow in space heating mode of DCU

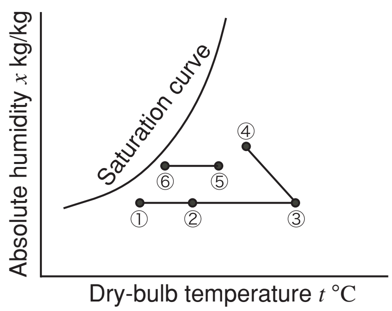

Fig. 5 Psychrometric diagram in space heating mode of DCU

finally expressed by the following equations:

$$
\left.\begin{aligned}
Q^{\mathrm{ha}}_{\mathrm{DCU}} &= p^{\mathrm{ha}}_{\mathrm{DCU}}(t_1,x_1,t_4,t_5,x_5)Q^{\mathrm{x}}_{\mathrm{DCU}} \\
&\quad + q^{\mathrm{ha}}_{\mathrm{DCU}}(t_1,x_1,t_4,t_5,x_5)\delta^{\mathrm{ha}}_{\mathrm{DCU}} \\
E^{\mathrm{ah}}_{\mathrm{DCU}} &= p^{\mathrm{ah}}_{\mathrm{DCU}}Q^{\mathrm{x}}_{\mathrm{DCU}} + q^{\mathrm{ah}}_{\mathrm{DCU}}\delta^{\mathrm{ha}}_{\mathrm{DCU}} \\
Q^{\mathrm{x}}_{\mathrm{DCU}} &\geq \underline{Q}^{\mathrm{xh}}_{\mathrm{DCU}}(t_1,x_1,t_4,t_5,x_5,\underline{v})\delta^{\mathrm{ha}}_{\mathrm{DCU}} \\
Q^{\mathrm{x}}_{\mathrm{DCU}} &\leq \overline{Q}^{\mathrm{xh}}_{\mathrm{DCU}}(t_1,x_1,t_4,t_5,x_5,\overline{v})\delta^{\mathrm{ha}}_{\mathrm{DCU}}
\end{aligned}\right\}, \tag{16}
$$

where $\delta^{\mathrm{ha}}_{\mathrm{DCU}}$ denotes a binary variable expressing the on/off status of operation in space heating mode of DCU; $p^{\mathrm{ha}}_{\mathrm{DCU}}$, $q^{\mathrm{ha}}_{\mathrm{DCU}}$, $p^{\mathrm{ah}}_{\mathrm{DCU}}$ and $q^{\mathrm{ah}}_{\mathrm{DCU}}$ are the performance characteristic values, and the former two are functions of t_1, x_1 t_4, t_5 and x_5; and $\overline{Q}^{\mathrm{xh}}_{\mathrm{DCU}}$ and $\underline{Q}^{\mathrm{xn}}_{\mathrm{DCU}}$ are the upper and lower bounds on the input energy flow rate, respectively, which are functions of t_1, x_1 t_4, t_5, x_5 and $\overline{v}$ or $\underline{v}$.

(c) Air handling unit (AHU). AHU is modeled as a variable air volume system. In the similar way to DCU, the performance characteristics are formulated according to the psychrometric process. The relationships between the cold water heat flow rate $Q^{\mathrm{c}}_{\mathrm{AHU}}$ and the heat flow rate for space cooling air $Q^{\mathrm{ca}}_{\mathrm{AHU}}$ or the electricity for auxiliary machinery $E^{\mathrm{ca}}_{\mathrm{AHU}}$ and those between the hot water heat flow rate $Q^{\mathrm{h}}_{\mathrm{AHU}}$ and the heat flow rate for space heating air $Q^{\mathrm{ha}}_{\mathrm{AHU}}$ or the electricity for auxiliary machinery $E^{\mathrm{ha}}_{\mathrm{AHU}}$ are expressed by the following equations:

$$
\left.\begin{aligned}
Q^{\mathrm{ca}}_{\mathrm{AHU}} &= p^{\mathrm{ca}}_{\mathrm{AHU}}(t_1,x_1,t_{4,\mathrm{AHU}},t_5,x_5)Q^{\mathrm{c}}_{\mathrm{AHU}} \\
E^{\mathrm{ac}}_{\mathrm{AHU}} &= p^{\mathrm{ac}}_{\mathrm{AHU}}Q^{\mathrm{c}}_{\mathrm{AHU}} \\
Q^{\mathrm{ha}}_{\mathrm{AHU}} &= p^{\mathrm{ha}}_{\mathrm{AHU}}(t_1,x_1,t_{4,\mathrm{AHU}},t_5,x_5)Q^{\mathrm{h}}_{\mathrm{AHU}} \\
E^{\mathrm{ah}}_{\mathrm{AHU}} &= p^{\mathrm{ah}}_{\mathrm{AHU}}Q^{\mathrm{h}}_{\mathrm{AHU}}
\end{aligned}\right\}, \tag{17}
$$

where $p^{\mathrm{ca}}_{\mathrm{AHU}}$, $p^{\mathrm{ha}}_{\mathrm{AHU}}$, $p^{\mathrm{ac}}_{\mathrm{AHU}}$ and $p^{\mathrm{ah}}_{\mathrm{AHU}}$ are the performance characteristic values, and the former two are functions of t_1, x_1, $t_{4,\mathrm{AHU}}$, t_5 and x_5; and $t_{4,\mathrm{AHU}}$ denotes air temperature introduced from AHU to the conditioned space, and is given as input data.

Energy balance and supply-demand relationships. Next, energy balance and supply-demand relationships are considered as constraints in the optimization problem, and are formulated by linear equations with energy flow rates at each junction for each energy flow illustrated in Fig. 1. For example, the following equations are obtained for the electricity flow:

$$
\left.\begin{aligned}
E_{\mathrm{EP}} + E_{\mathrm{MT}} &= E^{\mathrm{d}} + E^{\mathrm{a}} \\
E^{\mathrm{a}} &= E^{\mathrm{a}}_{\mathrm{RX}} + E^{\mathrm{a}}_{\mathrm{RG}} + E^{\mathrm{a}}_{\mathrm{BG}} + E^{\mathrm{a}}_{\mathrm{AHU}} + E^{\mathrm{a}}_{\mathrm{DCU}}
\end{aligned}\right\}. \tag{18}
$$

Objective function. The hourly energy charge is adopted as the objective function to be minimized from an economic viewpoint. The hourly energy charge J is expressed by the following equation:

$$
J = \varphi_{\mathrm{E}} E_{\mathrm{buy}} + \varphi_{\mathrm{G}} F_{\mathrm{buy}}, \tag{19}
$$

where φ_E and φ_G denote the unit costs of electricity and natural gas charges, respectively.

As a result, in this optimization problem, on/off and rated/part load status of operation of equipment and energy flow rates at each junction for each energy flow are determined so as to minimize the objective function (19) under constraints of the equipment performance characteristics such as Eqs. (1), (2) and (14) ~ (17) and the energy balance/supply-demand relationships such as Eq. (18).

Solution Method

The problem formulated and described in the previous section results in a mixed-integer linear programming one, and is solved by the GAMS(General Algebraic Modeling System)/Cplex solver [7], which is a commercial one combining the branch and bound method with the simplex method.

NUMERICAL STUDY

Input data

Energy demands of office buildings and hospitals with total floor areas of 1,000, 1,500 and 2,000 m² are estimated at 24 sampling times on each month of 12 representative days in a year in order to investigate the annual economic and operational benefits of using desiccant air-conditioning. The maximum electricity demands for unit area of the office buildings and hospitals are 39.6 and 33.4 W/m², respectively. As examples, Figs. 6 and 7 show hourly energy demands on February, May and August representative days for total floor area of 2,000 m² of the office building and hospital, respectively. Moreover, Fig. 8 shows temperature t_1 and absolute humidity x_1 of the ambient air on February, May and August representative days as examples. The ambient air temperature is also adopted as the intake air temperature of MT. Besides, temperature t_5, relative humidity and absolute humidity x_5 in the conditioned space are set to 26 °C, 50 % and 0.0105

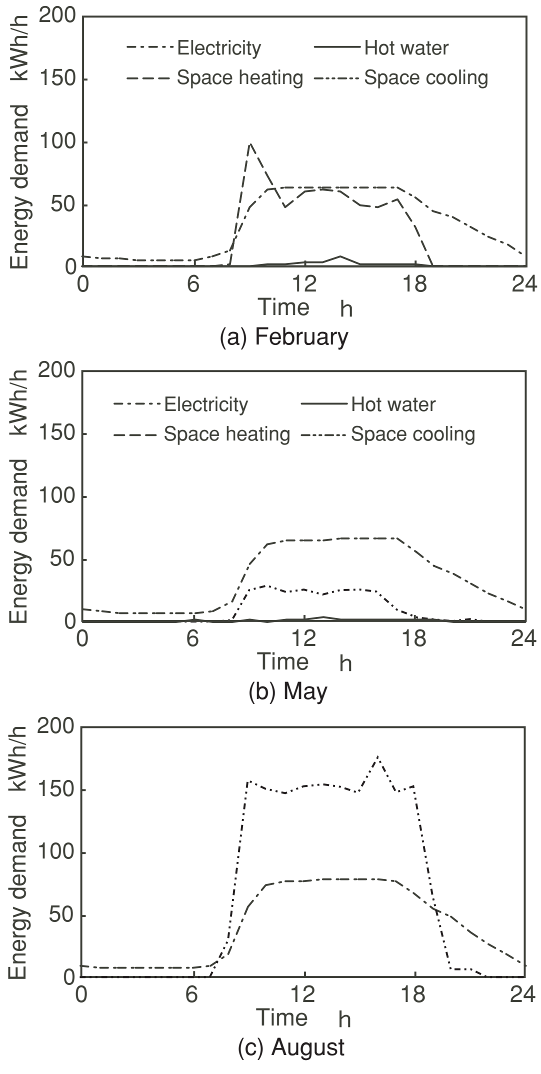

Fig. 6 Hourly energy demands (office building, 2,000m²)

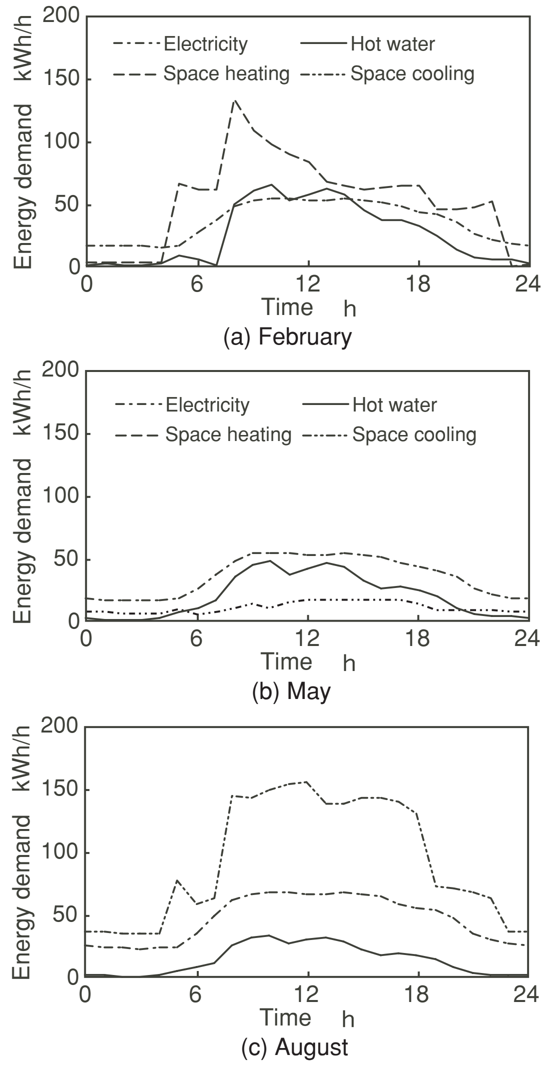

Fig. 7 Hourly energy demands (hospital, 2,000m²)

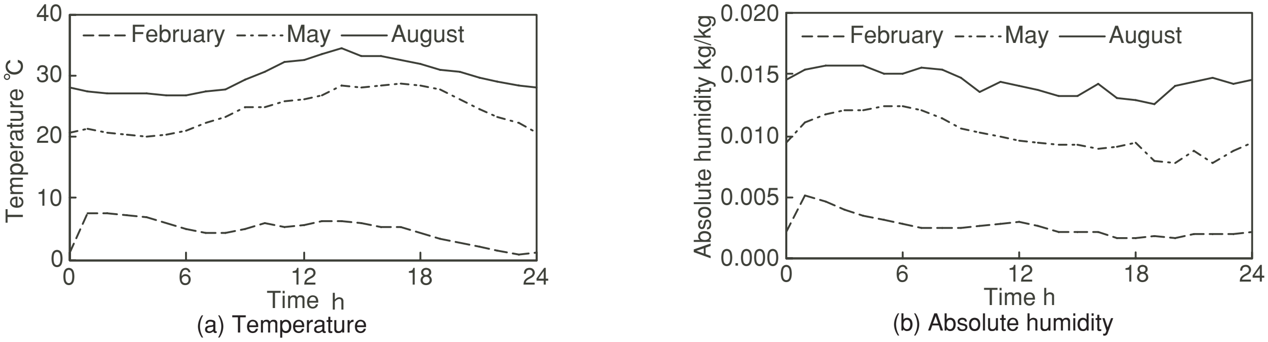

Fig. 8 Temperature and absolute humidity of ambient air

Table 1 Performance characteristic values and capacities of equipment

Equipment	Performance characteristic value (rated load)	Capacity
Microturbine cogeneration unit	Electrical generating efficiency 0.25*1,*2 Flue gas recovery efficiency 0.72*1,*2	28 kW
Flue gas absorption refrigerator	Coefficient of performance 0.56 (space cooling) Thermal efficiency 0.76 (space heating)	56 kW
Gas-fired absorption refrigerator	Coefficient of performance 1.12*1 (space cooling) Thermal efficiency 0.93*1 (space heating)	250 kW
Gas-fired boiler	Thermal efficiency 0.91*1	125 kW
Receiving device for purchased electricity	–	70 kW
Desiccant air conditioning unit	Coefficient of performance 0.73*3 (space cooling) Efficiency 0.90*4 (space heating)	62 kW*3 62 kW*4
Air handling unit	Efficiency 0.75*3 (space cooling) Efficiency 0.89*4 (space heating)	180 kW*3 180 kW*4

*1 The net thermal efficiency is calculated under the assumption that the lower heating value of natural gas is 11.55 kWh/m^3.
*2 When intake air temperature is 15 ˚C.
*3 When ambient air temperature and absolute humidity are 30 ˚C and 0.012 kg/kg, respectively.
*4 When ambient air temperature and absolute humidity are 7 ˚C and 0.005 kg/kg, respectively.

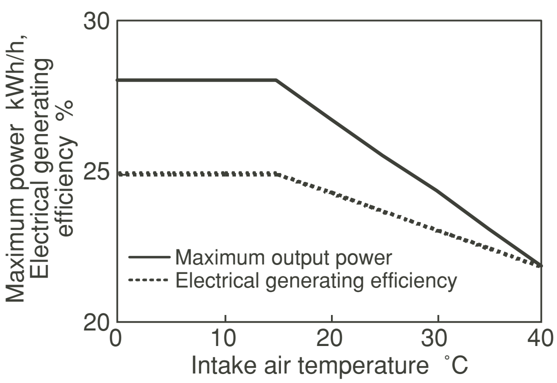

Fig. 9 Relationships between intake air temperature and maximum output power or electrical generating efficiency

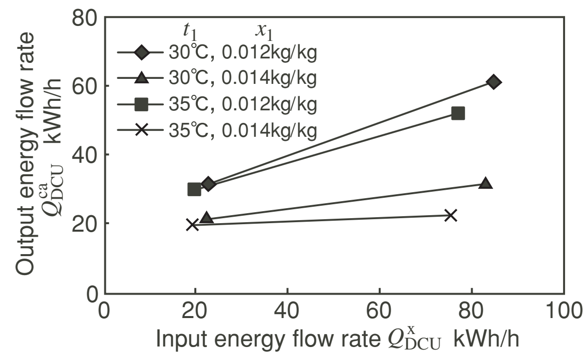

Fig. 10 Performance characteristics of DCU

Table 2 Rates for purchased electricity and natural gas

Utility	Unit cost of energy charge
Purchased electricity	14.15 (Jul. ~ Sep.) yen/kWh 12.86 (Other months) yen/kWh
Natural gas	43.13 yen/m^3

kg/kg for space cooling, respectively, and those are set to 22 °C, 40 % and 0.0067 kg/kg for space heating, respectively.

Representative values of performance characteristics and capacities of equipment are summarized in Table 1. The maximum output power and the electrical generating efficiency at the rated load status of MT are dependent greatly on the intake air temperature. Figure 9 shows the relationships between the intake air temperature and the maximum output power or the electrical generating efficiency at the rated load status. In assessing the performance characteristics of DCU shown in Table 1, coefficients p_{t}, q_{t}, $r_{\mathrm{t}}, p_{\mathrm{t}}', q_{\mathrm{t}}', r_{\mathrm{t}}'$, p_{x}, q_{x}, $r_{\mathrm{x}}, p_{\mathrm{x}}', q_{\mathrm{x}}'$ and r_{x}' are determined by regression analysis based on the dehumidifier performance in catalogs. The other parameters such as η_{s}, η_{e}, etc. are set based on references and catalog data. Moreover, as examples of the performance characteristic of DCU, the relationship between $Q_{\mathrm{DCU}}^{\mathrm{x}}$ and $Q_{\mathrm{DCU}}^{\mathrm{ca}}$ for temperature and absolute humidity are shown in Fig. 10. In this figure, the upper and lower limits of $Q_{\mathrm{DCU}}^{\mathrm{x}}$ correspond to those limits of the air velocity of DCU, respectively, as shown in Eq. (14). According to this figure, the efficiency of DCU becomes low with the increase in temperature and absolute humidity, and the efficiency at part-load status is higher than that at rated load status. The latter is reflected in the fact that the more moisture per unit air is removed by the desiccant when the air flows slowly and stays in the desiccant for a long time.

The rates for purchased electricity and natural gas adopted in this study are given in Table 2.

Under the aforementioned conditions, the influence of introducing DCU on the economic and energy-saving characteristics is investigated for the office buildings and hospitals. The economic and energy-saving characteristics are evaluated as the annual operational cost, which is the sum of hourly energy charges through a year, and the annual primary energy consumption, respectively. In evaluating the annual primary energy consumption, it is assumed that the lower heating value of natural gas is 11.55 kWh/m^3, and that the thermal efficiency of purchased electricity is 0.388.

Results and discussion

Tables 3 and 4 show respectively the annual operational cost and the annual primary energy consumption for all the cases. According to Table 3, the annual operational cost for the system with DCU is smaller than that for the system without DCU for all the cases. The reduction rates of the annual operational cost for the system with DCU to that for the system without DCU are 3~5 % and 6~10% for the office buildings and the hospitals, respectively. It is found from this result that the reduction rate of the annual operational cost for the hospital is larger than that for the office building. Moreover, for the hospital, the reduction rate of the annual operational cost becomes large with the increase in total floor area. The ranges in the output ratio of DCU to the maximum space cooling demand are 0.26~0.8, 0.2~0.6 and 0.13~0.4 for the hospitals with the total floor areas of 1,000, 1,500 and 2,000m^2, respectively. It is found from these results that the economic benefit of introducing DCU on the operation is high when the ratio of the capacity of DCU to maximum space cooling demand is less than 0.4 for the hospital with much thermal energy demand. Similar discussion is made for the annual primary energy consumption shown in Table 4. Therefore, it is found that the energy-saving benefit of introducing DCU is also high for the hospital with much thermal energy demand.

Table 3 Annual operational cost and its reduction rates

	Total floor area m^2	Annual operation cost ×10^6 yen/y		Reduction rate due to DCU %
		System with DCU	System without DCU	
Office building	1,000	2.47	2.59	4.7
	1,500	3.69	3.88	5.0
	2,000	4.95	5.15	3.9
Hospital	1,000	3.16	3.40	6.7
	1,500	4.62	5.07	8.8
	2,000	6.14	6.80	9.7

Table 4 Annual primary energy consumption and its reduction rates

	Total floor area m^2	Annual primary energy consumption GWh/y		Reduction rate due to DCU %
		System with DCU	System without DCU	
Office building	1,000	0.61	0.63	4.4
	1,500	0.89	0.94	4.8
	2,000	1.19	1.23	3.8
Hospital	1,000	0.80	0.86	6.6
	1,500	1.16	1.28	8.9
	2,000	1.53	1.69	9.9

As examples, Figs. 11 and 12 show the operational strategies on August representative day in the system with and without DCU for the hospital with the total floor area of 2,000 m^2, respectively. Comparing Fig. 11 (a) with Fig. 12 (a), electricity supply during night in the system with DCU is smaller than that in the system without DCU. This is because the electricity for auxiliary machinery becomes small by using DCU for space cooling supply. Moreover, it is found from (b) of these figures that the flue gas utilized to DCU is small. This is because the efficiency of DCU is high at part-load status as shown in Fig. 10. Besides, it is found from (d) of these figures that the cold water supply in the system with DCU is smaller than that in the system without DCU. By this result, the electricity for auxiliary machinery in the system with DCU becomes smaller than that in the system without DCU, and the electricity supply becomes small as mentioned previously.

The aforementioned results show that the economic and energy-saving benefits of the installation of DCU are high.

CONCLUSIONS

In this paper, the economic and energy-saving characteristics of the cogeneration system with the microturbine and desiccant air conditioning units have been investigated on the operation based on the optimization method. Through the numerical studies carried out on the systems installed in office buildings and hospitals, the following results have been obtained:

(a) The reduction rates of the annual operational cost for the system with DCU to that for the system without DCU are 3~5 % and 6~10% for the office buildings and the hospitals, respectively.

(b) The reduction rates of the annual primary energy consumption for the system with DCU to that for the system without DCU are 3~5% and 6~10% for the office buildings and the hospitals, respectively.

(c) The economic and energy-saving benefits of introducing DCU for hospital are larger than those for office building.

(d) By the installation of DCU, the cold water supply by RX and RG reduces. Moreover, the DCU is mainly operated at part-load status because the efficiency at part-load status is higher than that at rated load status.

In this paper, the capital costs of pieces of equipment are not taken into account because we have focused on the operational benefit of the aforementioned system as the first stage. However, it is also important to evaluate that system from the economic viewpoint including the capital costs. As the next stage, we will investigate it by extending the aforementioned method to that case.

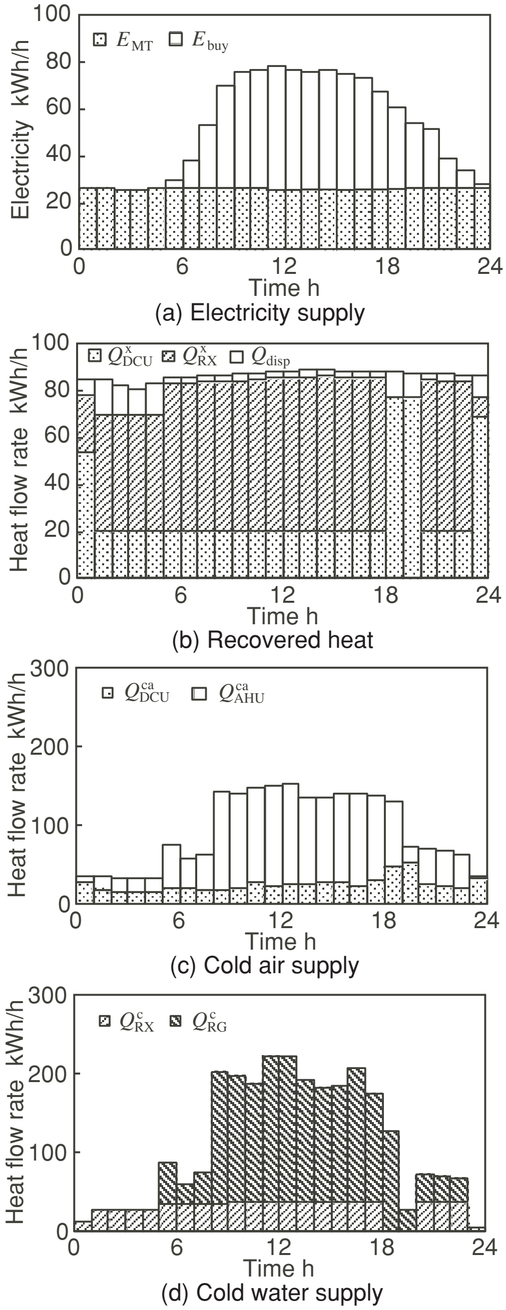

Fig. 11 Example of operational strategies for system with DCU (hospital, 2,000m2)

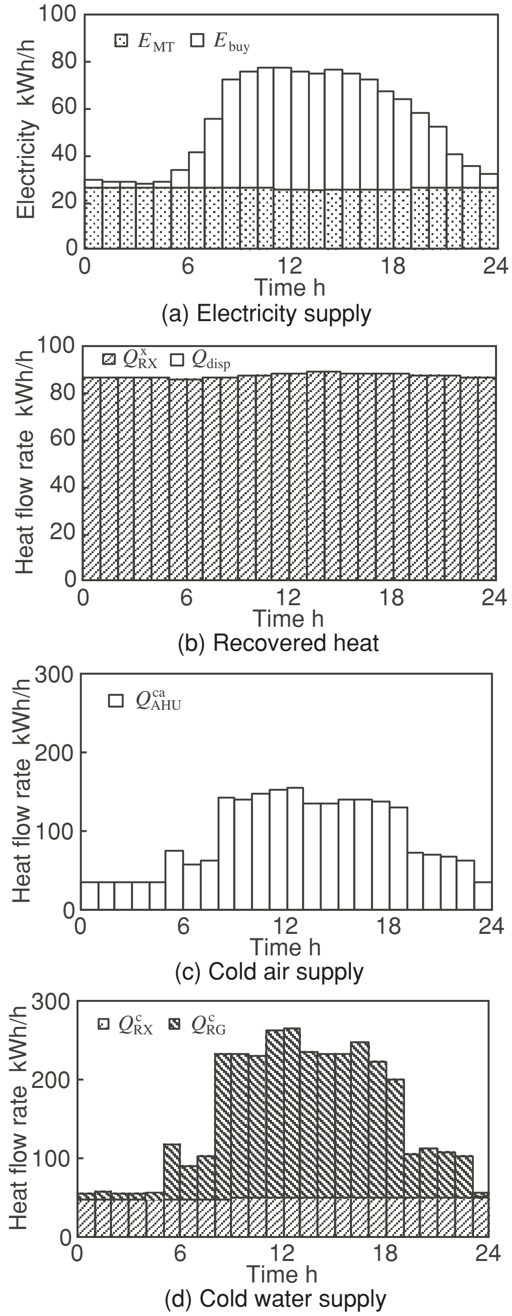

Fig. 12 Example of operational strategies for system without DCU (hospital, 2,000m2)

ACKNOWLEDGMENT

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