Model-Based Performance Monitoring with Dynamic Compensation for Heat Utilization Process in Distributed Energy System*

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Abstract
A model-based performance monitoring method for heat utilization processes in distributed energy systems is developed in this study. It is characterized by introducing dynamic compensation, where the response lags of heat exchangers to variations in their operating conditions are identified as first-order lag elements, and the output process variables estimated using a static input-output model are revised on the basis of these identified response lags. The estimated values of the output process variables are compared with their measured values in order to detect device failures. A numerical simulation of a heat utilization process in a gas engine cogeneration system containing a radiator with a considerable response lag reveals that the developed performance monitoring method has sufficient estimation accuracy in terms of the output process variables and ability to detect device failures, including a deterioration in the heat transfer performance of the radiator and heat exchanger, in a dynamic state.

Key words: Monitoring, Model-Based Monitoring, Fault Detection, Heat Exchanger, Cogeneration, Distributed Generation, Energy Management

1. Introduction
Recently, the utilization of distributed energy systems designed to contribute to energy conservation and economic efficiency has been extended beyond industrial and commercial applications to residential ones. However, from the viewpoint of human resources and personnel cost, it is not realistic to constantly monitor each system. Therefore, it is very important to develop a monitoring function to detect failures, including performance deteriorations, along with a diagnostic function to identify the causes of these failures. Moreover, distributed energy systems must operate appropriately in response to seasonal and hourly variations in energy demands to improve energy conservation and economic efficiency. Thus, a real-time computer-aided system that integrates energy demand prediction, optimal operational planning, and optimal control has been proposed(1). In this system, the optimal operational planning and control are carried out using the performances of the system components. However, their performances are influenced by seasonal variations and deteriorate with age. Therefore, the real-time computer-aided system also needs performance monitoring.

Several methods for the monitoring and diagnostics of energy and process systems have been proposed. Venkatasubramanian et al. classified these methods into three types: quantitative model-based methods(2), qualitative model-based methods(3), and process-history-based methods(4). Qualitative model-based methods include pattern recognition(5), artificial neural networks(6), and expert systems(7). As process-history-based methods, a principal component analysis(8) and dynamic time warping method(9) have

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mainly been reported. The advantage of these methods is that they can monitor and diagnose target systems without the need for detailed information and theoretical models. On the other hand, the quantitative model-based methods, which perform monitoring on the basis of residuals between measured variables and those calculated using theoretical models, can evaluate the performances of target systems quantitatively. They can also explain the causes of the failures that occur in system components using the theoretical models. As quantitative model-based methods, a performance monitoring method that considers variations in environmental conditions, including ambient temperature\textsuperscript{(10)(11)}, and one combined with the use of an optical torque sensor\textsuperscript{(12)} have been reported. These works targeted large-scale central energy systems; thus, variations in the operating conditions, including the load factor, were not sufficiently discussed. On the other hand, it is very important in distributed energy systems to monitor whether the systems have normal performances in response to variations in their operating conditions.

Against such a background, the authors have developed a model-based performance monitoring method for a shell-and-tube type heat exchanger, which is one of the key components of distributed energy systems\textsuperscript{(13)}. It is based on an input-output model that expresses the steady-state characteristic of the heat exchanger. It has also been shown that the developed method can be applied to performance monitoring in the dynamic state by tolerating dynamic responses within a period of a few minutes, which coincides with the settling time for the heat exchanger\textsuperscript{(14)}. However, heat utilization processes in distributed energy systems consist of multiple temperature control loops composed of heat exchangers and measurement and control devices. Moreover, the response lags of heat exchangers may become pronounced, depending on their operating conditions. Hence, this study extends the previously developed method to heat utilization processes; the extended method is characterized by introducing dynamic compensation, where the response lags of heat exchangers to variations in their operating conditions are identified as first-order lag elements, and the output process variables estimated using the input-output model are revised on the basis of these identified response lags. Furthermore, a case study based on the numerical simulation of a heat utilization process in a gas engine cogeneration system containing a radiator with a considerable response lag reveals that the developed method has sufficient estimation accuracy in terms of the output process variables and ability to detect device failures, including a deterioration in the heat transfer performance of the radiator and heat exchanger, in a dynamic state.

2. Model-based performance monitoring method with dynamic compensation

Figure 1 shows the basic concept of the model-based performance monitoring method with dynamic compensation developed in this study. It has three components: an input-output model, dynamic compensation function, and failure detection function.

2.1 Input-output model\textsuperscript{(13)}

Preliminarily, process variables are classified into input variables that express the operating conditions and output variables, which depend on the input variables. During operation, the input process variables, \( PV_{\text{in}}^M \), and output process variables, \( PV_{\text{out}}^M \), are measured in a given sampling time. They are averaged in an averaging time interval because the input and output process variables may contain measurement noises; the averaged input and output process variables in a given time, \( t \), are referred to as \( \bar{PV}_{\text{in}}^M(t) \) and \( \bar{PV}_{\text{out}}^M(t) \), respectively. Moreover, the output process variables under the normal condition, \( PV_{\text{out}}^S \), in response to \( PV_{\text{in}}^M(t) \), are estimated using an input-output model. This model consists of mass and heat balance equations in the steady state, control conditions of the system components, connection conditions, and boundary conditions that correspond to \( PV_{\text{in}}^M(t) \). The model results in the following nonlinear algebraic equations:

\[
\begin{align*}
\mathbf{f}_{10}(PV_{\text{in}}^M(t), PV_{\text{out}}^S) &= 0
\end{align*}
\]
where $f_{io}$ denotes the vector for equations expressing the input-output characteristic. $PV_{out,l}$ is derived by numerically solving Eq. (1).

2.2 Dynamic compensation

The response lags of heat exchangers are caused mainly by their heat capacities and transit times. Because the operating conditions of distributed energy systems vary widely depending on variations in the energy demands, the input-output model may not estimate the output process variables in the dynamic state correctly. Thus, the following dynamic compensation for output process variables with considerable response lags is developed.

First, the dynamic characteristics of heat exchangers are identified as first-order lag elements. A time constant that expresses the response of an output process variable to the change in an input process variable is obtained using the corresponding transfer function, which is derived by focusing on the minimal deviations from the balance point; that is, it can be calculated from the cutoff frequency, at which the gain is decreased by 3.01 dB from the gain at extremely low frequencies. In addition, the time constant is regarded as a function of the input process variables in order to express the dependence of these variables. The identification equation for the time constant that expresses the response of the $n$th output process variable to the change in the $l$th input process variable in a given time, $T_{l}^{in,n}$, is expressed as follows:

$$T_{l}^{in,n} = f(PV_{in,l}^{M}) \quad (l = 1, 2, \ldots, L; n = 1, 2, \ldots, N) \quad (2)$$

where $f$ denotes the identification equation; and $L$ and $N$ denote the numbers of input and output process variables, respectively. By using the obtained time constants, the response lags of the output process variables during the averaging time interval, $\Delta t$, are considered. If the averaging time interval is appropriately set so that the response of the target process is regarded as almost linear, the response of the focused output process variable can be expressed by the superposition of the response to the change in each input process variable. Thus, the estimated value of the $n$th output process variable in a given time, $PV_{out,l}^{E,n}$, can be expressed as follows:

$$PV_{out,l}^{E,n} = PV_{out,l-\Delta t}^{E,n} + \sum_{l=1}^{L} \left( PV_{out,l}^{S,n} - PV_{out,l-\Delta t}^{S,n} \right) \left( 1 - e^{-\frac{\Delta t}{T_{l}^{in,n}}} \right) \quad (n = 1, 2, \ldots, N) \quad (3)$$

where $PV_{out,l}^{S,n}$ denotes the $n$th output process variable estimated in a steady state and $PV_{in,l}^{M}$ denotes the $l$th input process variable. $PV_{out,l}^{E,n}$ is calculated from the input-output model under the assumption that the other input process variables do not change.

2.3 Failure detection

To perform performance monitoring, the monitoring indicator, $I_{l}^{p}$, is defined for each component of $PV_{out,l}^{M}$ and $PV_{out,l}^{E}$ estimated with the dynamic compensation or $PV_{out,l}^{S}$ estimated without the dynamic compensation.
\[ \begin{align*}
I^n &= \begin{cases} 
\frac{p^{M,n}_{\text{out},T}}{p^{M,n}_{\text{out},T}} & \text{(with dynamic compensation)} \\
\frac{p^{M,n}_{\text{out},T}}{p^{M,n}_{\text{out},T}} & \text{(without dynamic compensation)} 
\end{cases} 
(n = 1, 2, \ldots, N) 
\end{align*} \]

If one monitoring indicator continues to surpass the threshold for a given period, it is recognized that there is a device failure.

3. Heat utilization process and its simulator

The developed performance monitoring method is applied to a heat utilization process in a gas engine cogeneration system. This section describes the target heat utilization process and its simulator.

3.1 Heat utilization process

The configuration of the target heat utilization process is shown in Fig. 2. Hot water, whose heat is recovered from the exhaust heat of a gas engine (GE), flows into the high-temperature side of a water-water heat exchanger (HE) and is utilized to heat the supply water that flows into the low-temperature side of HE. The supply water flow rate varies depending on the energy demand. Surplus heat is consumed at an air-cooled radiator (RD), and the hot water is returned to GE. To maintain the temperature of the returned hot water constant, this process has temperature control loops for HE (CL1) and RD (CL2). Both CL1 and CL2 utilize three-way control valves (CV1 and CV2). These manipulate the flow rates of hot water into HE and RD.

The controlled variable in CL1 is the hot water temperature downstream of CV1, measured at a temperature indicator (TI1). When the supply water flow rate is below a rated value, the hot water temperature on the high-temperature side of HE is higher than a set point even if all the hot water flows into HE. However, if the supply water flow rate is above a rated value, the hot water temperature on the high-temperature side of HE is lower than a set point. Thus, the hot water temperature measured at TI1 is maintained at a set point by increasing the bypass water flow rate; this is based on the manipulated signal in a temperature controller (TC1). On the other hand, the controlled variable in CL2 is the hot water temperature downstream of CV2, measured by a temperature indicator (TI2). If the measured hot water temperature is higher than a set point, a temperature controller (TC2) outputs a manipulated signal to increase the flow rate of hot water into RD. RD has the same radiation capacity as the rated heat recovery capacity of GE. Therefore, GE can operate without hot water demand. When the supply water flow rate is close to the rated value, the flow rate of the hot water into RD is low because of the low surplus heat; this makes the response of the outlet hot water temperature of RD very gradual.

![Fig. 2 Configuration of target heat utilization process](image-url)
3.2 Process simulator

A simulator for the target process is constructed to simulate various device failures; this means that the simulation results are regarded as the measured process variables. Because a simulation model for a gas engine cogeneration system has already been developed\(^{15}\), it is extended to the target process. The process simulator consists of equations for the dynamic characteristics of the devices, connection conditions, and boundary conditions.

3.2.1 Modeling of device performance characteristics

For each device, the mass flow rate and temperature of hot water or air are considered as the variables to be determined, and the mass and heat balance relationships are formulated. To calculate the properties of water and air, PROPATH Ver.12.1 is employed.

(a) Gas engine (GE)

GE is modeled as a lumped parameter system. The relationship between the electric power, \( E_{GE} \), and the city gas consumption, \( F_{GE} \), is expressed by the following equation:

\[
\begin{align*}
T_{GE}^E \frac{dE_{GE}(t)}{dt} + E_{GE}(t) &= E_{SP}^{GE}(t) \\
F_{GE}(t) &= \frac{E_{SP}^{GE}(t)}{\eta_{GE}^{GE}(E_{SP}^{GE}(t))} H_U
\end{align*}
\]

(5)

where \( T_{GE}^E \) denotes the time constant expressing the response of the electric power to the change in its set point, \( E_{SP}^{GE} \), and \( \eta_{GE}^{GE} \) and \( H_U \) denote the generation efficiency and heating value of the city gas, respectively. \( \eta_{GE}^{GE} \) is regarded as a function of \( E_{SP}^{GE} \). Moreover, the heat balance for the recovery heat, \( Q_{GE} \), is formulated as follows:

\[
\begin{align*}
Q_{GE}(t) &= \eta_{GE}^Q \left( Q_{in}^{GE}(t) \right) Q_{out}^{GE}(t) \\
Q_{GE}(t) &= H_U F_{GE}(t - \tau_{GE}) \\
\rho_{GE}^{W} V_{GE} \frac{d\theta_{GE}^{out}(t)}{dt} &= g_{GE}(t) \left( \alpha_{GE}^{W}(t) - \theta_{GE}^{out}(t) \right) + Q_{GE}(t)
\end{align*}
\]

(6)

where \( \eta_{GE}^Q \) and \( Q_{GE}^H \) denote the heat recovery efficiency and heat output, respectively; \( \tau_{GE} \) denotes the dead time of the response of the outlet hot water temperature to the change in the recovery heat, \( \rho, c, V, \theta, \) and \( g \) denote the density, specific heat, volume, temperature, and mass flow rate, respectively; and the superscripts \( W, \) \( in, \) and \( out \) denote properties for water, inlet, and outlet, respectively. \( \eta_{GE}^Q \) is regarded as a function of \( Q_{GE}^H \). The heat loss to the environment is included in the exhaust heat (= \( H_U F_{GE}(t) - E_{GE}(t) - Q_{GE}(t) \)). The density and specific heat of water are calculated from the arithmetic average of the inlet and outlet hot water temperatures. Furthermore, based on Eq. (6), the time constant expressing the response of the outlet hot water temperature to the change in the recovery heat, \( T_{GE}^Q \), can be defined as follows:

\[
T_{GE}^Q(t) = \frac{\rho_{GE}^{W} V_{GE}}{\theta_{GE}^{out}(t)}
\]

(7)

(b) Water-water heat exchanger (HE)

For HE, a plate type heat exchanger is modeled as a lumped parameter system. Its heat balance relationship is formulated as follows:

\[
\begin{align*}
\rho_{HE,H}^W V_{HE,H} \frac{d\theta_{HE,H}^{out}(t)}{dt} &= g_{HE,H}(t) c_{HE,H}^W \left( \beta_{HE,H}^{W,in}(t) - \theta_{HE,H}^{out}(t) \right) \\
-K_{HE,A_{HE}}^2 \theta_{HE,H}^{out}(t) - K_{HE,A_{HE}}^1 \theta_{HE,H}^{out}(t) &= \frac{\beta_{HE,H}^{W,in}(t) + \theta_{HE,H}^{out}(t)}{2} - \theta_{HE,H}^{out}(t) \\
\rho_{HE,L}^W V_{HE,L} \frac{d\theta_{HE,L}^{out}(t)}{dt} &= g_{HE,L}(t) c_{HE,L}^W \left( \theta_{HE,L}^{W,in}(t) - \theta_{HE,L}^{out}(t) \right) \\
+ K_{HE,A_{HE}}^2 \theta_{HE,L}^{out}(t) - K_{HE,A_{HE}}^1 \theta_{HE,L}^{out}(t) &= \frac{\theta_{HE,L}^{W,in}(t) + \theta_{HE,L}^{out}(t)}{2} - \theta_{HE,L}^{out}(t)
\end{align*}
\]

(8)
where \( K, A, \) and \( \Delta \theta^L \) denote the overall heat transfer coefficient, heat transfer area, and log-mean temperature difference, respectively; the superscripts \( H \) and \( L \) denote properties for air and heat loss, respectively; and the subscripts \( H \) and \( L \) denote the high-temperature and low-temperature sides, respectively. The density and specific heat of water are calculated from the arithmetic average of the inlet and outlet hot water temperatures. In Eq. (8), the overall heat transfer coefficient, \( K_{HE} \), is calculated as follows:

\[
K_{HE} = \frac{1}{\frac{\alpha_{HE}}{A_{HE,H}} + \frac{\tau_p}{A_p} + \frac{1}{\alpha_{HE,L}}} \quad (9)
\]

where \( \alpha, \tau, \) and \( \lambda \) denote the heat transfer coefficient, thickness, and heat conductivity, respectively; and the subscript \( P \) denotes the plate. In Eq. (9), the heat transfer coefficient for the two temperature sides is calculated using the following equation (10):

\[
\frac{\alpha_{HE}D_{HE}}{\delta_{HE}W} = C_1 \left( \frac{D_{HE}G_{HE}^W}{\mu_{HE}} \right)^{C_2} \left( \frac{c_{HE,\text{H}}^W}{\rho_{HE}^W} \right)^{\frac{1}{2}} \quad (10)
\]

where \( D_{HE} \) denotes the equivalent diameter corresponding to two times the plate interval; \( G \) and \( \mu \) denote the mass speed and viscosity coefficient, respectively; and \( C_1 \) and \( C_2 \) denote the constants \( (C_1 = 0.165 \) and \( C_2 = 0.68) \). The heat conductivity, viscosity coefficient, and specific heat of water are calculated from the arithmetic average of the inlet and outlet hot water temperatures. The mass speed is calculated using the following equation (16):

\[
G_{HE}^W = \frac{\delta_{HE}^W}{b_{HE}N_{HE}^W} \quad (11)
\]

where \( b_{HE}, N_{HE}, \) and \( \delta_{HE} \) denote the width, number, and intervals of the plates, respectively. From Eqs. (9) to (11), it is found that the calculation of \( K_{HE} \) considers the changes in the water flow rate and temperature for the two temperature sides.

(c) Air-cooled radiator (RD)

For RD, a finned tube heat exchanger is modeled as a distributed parameter system divided into \( I \) equal control volumes because the response of the outlet hot water temperature may be very gradual. The air flows perpendicularly to the direction of the hot water flow. Only the response lag of the hot water is considered because the response of the air temperature with a small specific heat is extremely fast compared to that of the hot water temperature. Moreover, the following assumptions are employed: there is no difference in the water flow rates and temperature distributions among the heat transfer tubes and the overall heat transfer coefficient has no distribution in the longitudinal direction of the heat transfer tubes. The heat balance relationships on the hot water side and air side for the \( i \)th control volume are formulated as follows:

\[
\begin{align*}
\rho_{RD}^W c_{RD}^W \frac{d\theta_{RD}^W(t)}{dt} &= \sum_{i=1}^{I} \left[ g_{RD}^W(t) c_{RD}^W \left( \theta_{RD}^{W,i-1}(t) - \theta_{RD}^{W,i}(t) \right) \right] \\
&\quad - K_{RD} A_{RD} \frac{a_{RD}}{l} \left\{ \theta_{RD}^{W,i-1}(t) - \theta_{RD}^{A,in}(t) \right\} \\
&\quad + K_{RD} A_{RD} \frac{a_{RD}}{l} \left\{ \theta_{RD}^{W,i-1}(t) - \theta_{RD}^{A,out}(t) \right\} \quad (12)
\end{align*}
\]

where \( S \) and \( \Delta x \) denote the sectional area of the heat transfer tube and length of the control volume, respectively. The water and air properties in Eq. (12) are calculated from the arithmetic average of the inlet and outlet temperatures for the corresponding control volume. In Eq. (12), the overall heat transfer coefficient, \( K_{RD} \), is calculated as follows:

\[
K_{RD} = \frac{1}{A_{RD}} \left[ \frac{1}{a_{RD}^2 (A_T^2 + \phi_F A_F)} + \frac{1}{a_{RD}^2 A_T^2} \right]^{-1} \quad (13)
\]

where \( A_{RD}, A_T, A_F, \) and \( \phi_F \) denote the overall heat transfer area on the fin side, surface area of the heat transfer tubes, heat transfer area of the fins, and fin efficiency, respectively.
The heat transfer coefficient on the fin (air) side is calculated from the Fujikake equation\(^{(17)}\):
\[
\frac{a_{f,RD}^A}{\lambda_{RD}} = 2R_F \left[ 1.1 + 0.55 \left( \frac{Re_{RD}^A Pr_{RD}^A D_F}{L_F^2} \right)^{0.55} \right]
\]
where \(D_F\), \(R_F\), and \(L_F\) denote the representative length of the fin, constant determined from the fin size, and fin width, respectively. The calculation of \(D_F\) and \(R_F\) in response to \(L_F\) is based on Ref. \((17)\). The heat conductivity and Prandtl number, \(Pr_{RD}^A\), are calculated from the arithmetic average of the inlet and outlet air temperatures for RD. The Reynolds number, \(Re_{RD}^A\), is calculated from the average inflow air speed, \(D_F\), and arithmetic average of the inlet and outlet air temperatures for RD. The fin efficiency is calculated using the following Fujikake equation\(^{(17)}\):
\[
\phi_F = \tanh \left( \frac{H_F}{2} \sqrt{\frac{Z_F^2}{2} \frac{2a_{RD}^A}{\lambda_{RD}}} \right) / \tanh \left( \frac{H_F}{2} \sqrt{\frac{Z_F^2}{2} \frac{2a_{RD}^A}{\lambda_{RD}}} \right)
\]
where \(H_F\) denotes the fin height; and \(Z_F\) denotes the constant determined from the fin structure, whose calculation method is based on Ref. \((17)\). For the calculation of the heat transfer coefficient on the hot water side in Eq. \((13)\), the Dittus-Boelter equation\(^{(18)}\) is employed. As a result, the calculation of \(K_{RD}\) considers the changes in the mass flow rate and temperature of hot water and air.

**3.2.2 Connection conditions and boundary conditions**

For CVs, a three-way control valve with two inlets (inlet-1 and inlet-2) and one outlet is modeled as a lumped parameter system. The mass balance relationship is formulated as
\[
\begin{aligned}
g_{CV}^{W,\text{in1}}(t) + g_{CV}^{W,\text{in2}}(t) &= g_{CV}^{W,\text{out}}(t) \\
ge_{CV}(t)g_{CV}^{W,\text{out}}(t) &= (1 - e_{CV}(t))g_{CV}^{W,\text{out}}(t)
\end{aligned}
\]
where \(e_{CV}\) denotes the valve opening; and the superscripts \text{in1} and \text{in2} denote inlet-1 and inlet-2, respectively. By assuming the complete mixing of the inflow hot water, the heat balance relationship is formulated as follows:
\[
ge_{CV}(t)g_{CV}^{W,\text{out}}(t) = g_{CV}^{W,\text{in1}}(t) + (1 - e_{CV}(t))g_{CV}^{W,\text{in2}}(t)
\]
Moreover, the response of the valve opening to the manipulated signal is modeled as a first-order lag element, although its formulation is omitted.

**3.2.3 Pipe**

The pipes connecting the devices are modeled as a lumped parameter system because they are not very long. The heat balance relationship is formulated as follows:
\[
\frac{\rho_p W_p V_p}{d \theta_{pp}^{W,\text{out}}(t)/dt} = \theta_{pp}^{W,\text{out}}(t) c_{pp}^{W,p} (\theta_{pp}^{W,\text{in}}(t) - \theta_{pp}^{W,\text{out}}(t)) - K_{pp}^A \left[ \frac{\theta_{pp}^{W,\text{in}}(t) + \theta_{pp}^{W,\text{out}}(t)}{2} - A(t) \right]
\]
where the subscript PP denotes the piping.

**3.2.4 Other devices**

For TIs, the response of the temperature signal to a change in the measured hot water temperature is modeled as a first-order lag element. For TCs, a discrete-type feedback control algorithm employing a proportional-plus-integral action is modeled.

**3.2.2 Connection conditions and boundary conditions**

The connection conditions are considered at the connecting points of the devices; these mean to equalize the values of the corresponding variables. Furthermore, the boundary
conditions are considered at the boundary points of the target process.

3.2.3 Solution method

The aforementioned model forms nonlinear differential algebraic equations as follows:

\[
\begin{align*}
& f_{SIM}(x(t), x'(t), y(t), t) = 0 \\
& x(t_0) = x_0
\end{align*}
\]

(19)

where \( f_{SIM} \) denotes the vector for equations expressing the model; \( x \) denotes the vector for the variables with their derivatives; \( x' \) denotes the vector for the derivative of \( x \) with respect to time, \( t \); \( y \) denotes the vector for the variables without their derivatives; and \( x_0 \) denotes the vector for the initial value of \( x \) at the initial time, \( t_0 \). The equations are solved numerically using a combination of the Euler and Newton–Raphson methods.

3.3 Calculation conditions

The calculation conditions used to solve Eq. (19) are described in this subsection. The discrete time for the Euler method is 0.1 s.

3.3.1 Specifications of devices

The specifications of the target 25-kWe gas engine cogeneration system (19) are listed in Table 1. HE is made of SUS316. RD is made of aluminum, and the flow channel geometry between its fins has a triangular form. HE and RD have the same heat transfer and radiation capacities as the rated heat recovery capacity of GE, respectively. The length of all the pipes connecting the devices is assumed to be 1.3 m; the nominal pipe diameter is 40 A. The time constants expressing the responses for CV and TI are 1 s and 2 s, respectively.

Table 1 Specifications of target gas engine cogeneration system

<table>
<thead>
<tr>
<th>Components</th>
<th>Specifications</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas engine (GE)</td>
<td>Rated electric power output</td>
<td>kW 25.0</td>
</tr>
<tr>
<td></td>
<td>Rated hot water output</td>
<td>kW 39.7</td>
</tr>
<tr>
<td></td>
<td>Rated generating efficiency</td>
<td>% 33.1</td>
</tr>
<tr>
<td></td>
<td>Rated heat recovery efficiency</td>
<td>% 52.4</td>
</tr>
<tr>
<td></td>
<td>Time constant for electric power output</td>
<td>( T_{GE} ) s 1.0</td>
</tr>
<tr>
<td></td>
<td>Time constant for hot water output</td>
<td>( T_{GE} ) s 18.0</td>
</tr>
<tr>
<td></td>
<td>Dead time for input heating rate</td>
<td>( \tau_{GE} ) s 10.0</td>
</tr>
<tr>
<td></td>
<td>Flow rate of circulating hot water</td>
<td>( \dot{q}_{GE} ) kg/s 1.72</td>
</tr>
<tr>
<td>Water-water heat exchanger (HE)</td>
<td>Heat transfer area</td>
<td>( A_{HE} ) m(^2) 0.6</td>
</tr>
<tr>
<td></td>
<td>Outside surface area</td>
<td>( A_{HE} ) m(^2) ( 0.25 \times 10^{-2} )</td>
</tr>
<tr>
<td></td>
<td>Volume</td>
<td>( V_{HE} ) m(^3) ( 1.83 \times 10^{-3} )</td>
</tr>
<tr>
<td></td>
<td>Overall heat transfer coefficient for heat loss</td>
<td>( K_{HE} ) W/(m(^2)K) 0.8</td>
</tr>
<tr>
<td>Air-cooled radiator (RD)</td>
<td>Overall heat transfer area</td>
<td>( A_{RD} ) m(^2) 8.94</td>
</tr>
<tr>
<td></td>
<td>Water-side surface area of tube</td>
<td>( A_W ) m(^2) 1.64</td>
</tr>
<tr>
<td></td>
<td>Air-side surface area of tube</td>
<td>( A_F ) m(^2) 1.68</td>
</tr>
<tr>
<td></td>
<td>Surface area of fin</td>
<td>( A_F ) m(^2) 7.26</td>
</tr>
<tr>
<td></td>
<td>Total cross-section area of tube</td>
<td>( S_{RD} ) m(^2) ( 2.88 \times 10^{-3} )</td>
</tr>
<tr>
<td></td>
<td>Total number of control volume</td>
<td>( l ) 100</td>
</tr>
<tr>
<td>Pipe</td>
<td>Overall heat transfer coefficient for heat loss</td>
<td>( K_{HP} ) W/(m(^2)K) 0.8</td>
</tr>
</tbody>
</table>

Table 2 Control parameters

<table>
<thead>
<tr>
<th>Control parameters</th>
<th>TC1</th>
<th>TC2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Proportional gain</td>
<td>1/K</td>
<td>0.08</td>
</tr>
<tr>
<td>Integral time</td>
<td>s</td>
<td>10</td>
</tr>
</tbody>
</table>
3.3.2 Control conditions
The control parameters for TC1 and TC2 are listed in Table 2. The set points for the hot water temperature and control interval are 79 °C and 1 s, respectively, for both control loops.

3.3.3 Boundary conditions
The boundary points of the target process are the set point for GE electric power, mass flow rate and temperature of the supply water, and ambient temperature. GE is assumed to be operated under the rated condition because the rated power operation of the gas engine cogeneration systems for consumer use provides effective energy conservation (20). The supply water flow rate varies with the energy (hot water) demand; the supply water flow rate is expressed by normalizing with the rated flow rate of 0.212 kg/s, in which all of the circulated hot water bypasses RD. The supply water and ambient temperatures are set to be 17 °C and 16 °C, respectively, and it is assumed to be mid-season.

4. Construction of performance monitoring system
In this section, the model-based performance monitoring system for the target heat utilization process is constructed.

4.1 Definition of input and output process variables
As the input process variables, the set point of GE electric power, mass flow rate and temperature of the supply water, and ambient temperature are considered. The output process variables include the measured hot water temperatures and manipulated signals for CL1 and CL2, as well as the hot water temperature differences between the inlets and outlets of HE and RD.

4.2 Construction of input-output model
4.2.1 Formulation of device performance characteristics
The steady-state characteristics of GE, HE, RD, CVs, and pipes are formulated. The formulated equations basically coincide with the equations in the process simulator, where the time-derivative terms are omitted. RD is modeled as a lumped parameter system with a low computation load because there is little difference in the steady-state characteristics between the lumped and distributed parameter models (21).

\[
\begin{align*}
\theta_{RD}^W(t) & = \theta_{RD}^{W, \text{in}}(t) - \theta_{RD}^{W, \text{out}}(t) - K_{RD} A_{RD} \Delta \theta_{RD}(t) = 0 \\
\theta_{RD}^A(t) & = \theta_{RD}^{A, \text{in}}(t) + K_{RD} A_{RD} \Delta \theta_{RD}(t) = 0
\end{align*}
\]  
where the overall heat transfer coefficient, \( K_{RD} \), is calculated in the same manner as in the process simulator; and the water and air properties are calculated from the arithmetic average of the inlet and outlet temperatures.

4.2.2 Control conditions
Instead of formulating the static characteristic of the controllers, the controlled variables (hot water temperatures) are equalized with their set points.

4.2.3 Connection conditions and boundary conditions
As the connection conditions, the values of the corresponding variables are equalized at the connecting points of the devices. The boundary conditions are the aforementioned four input process variables.

4.2.4 Solution method
The input-output model forms a set of nonlinear algebraic equations. Solving this set of equations numerically using the Newton–Raphson method gives the output process variables in response to the measured input process variables.
4.3 Construction of dynamic compensation

4.3.1 Derivation of transfer function

This paper focuses only on the supply water flow rate as an input process variable. This is because the variations in the supply water and ambient temperatures are very gradual compared to those of the supply water flow rate, and GE is always operated under the rated power output. Furthermore, the supply water flow rate varies below the rated value so that only CL2 operates. The only output process variable employing the dynamic compensation is the hot water temperature difference between the inlet and outlet of RD, $\Delta \theta_{RD}$. This is because it was confirmed that the dynamic compensation had no effect on the other output process variables, which have quick responses.

The time constant expressing the response of $\Delta \theta_{RD}$ to the change in the supply water flow rate is first calculated from its frequency characteristic. From Fig. 2, it is found that the inlet water temperature at RD is influenced by both the change in the temperature on the high-temperature side of HE through the connecting pipe. Moreover, it is revealed that the outlet water temperature at RD is affected not only by the change in the inlet water temperature at RD but also by the control action of CL2. Based on these features, the transfer function expressing the input-output characteristic of each device is derived by conducting the Laplace transform for the minimal deviation from the balance point. Then the transfer function expressing the response of $\Delta \theta_{RD}$ to the change in the supply water flow rate is derived by integrating the transfer functions of the devices.

For HE, the transfer function, $G_{HE}$, expressing the response of the outlet hot water temperature on the high-temperature side, $\theta_{HE,H}^{W,out}$, to the change in the supply water flow rate is derived by considering the boundary condition of $g_S(t) = g_{HE,L}(t)$ for Eq. (8):

$$g_{HE,H}^{W,out}(s) = G_{HE}(s)g_S(s)$$ (21)

For the pipe between HE and RD, the transfer function, $G_{PP}$, expressing the response of the inlet hot water temperature of RD, $\theta_{RD}^{W,in}$, to the change in $\theta_{HE,H}^{W,out}$ is derived by considering the connection conditions of $\theta_{RD}^{W,out}(t) = \theta_{PP}^{W,in}(t)$ and $\theta_{RD}^{W,in}(t) = \theta_{PP}^{W,out}(t)$ for Eq. (18):

$$\theta_{RD}^{W,in}(s) = G_{PP}(s)\theta_{HE,H}^{W,out}(s)$$ (22)

The outlet hot water temperature of RD, $\theta_{RD}^{W,out}$, is influenced by both $\theta_{RD}^{W,in}$ and the inflow hot water flow rate, $g_{RD}^{W}$. Thus, the two transfer functions, $G_{RD1}$ and $G_{RD2}$, are derived by adding the time-derivation terms for the hot water temperature in Eq. (20).

$$\theta_{RD}^{W,out}(s) = G_{RD1}(s)\theta_{RD}^{W,in}(s) + G_{RD2}(s)g_{RD}^{W}(s)$$ (23)

where $G_{RD1}$ denotes the transfer function expressing the response of $\theta_{RD}^{W,out}$ to the change in $\theta_{RD}^{W,in}$, and $G_{RD2}$ denotes the transfer function expressing the response of $\theta_{RD}^{W,out}$ to the change in $g_{RD}^{W}$. For CV2, the following input-output relationship is derived by considering the connection conditions of $g_{CV}^{W,in}(t) = g_{RD}^{W}(t)$ and $g_{CV}^{W,out}(t) = g_{GE}^{W}$ in Eq. (16):

$$g_{RD}^{W}(s) = -g_{GE}^{W}\epsilon_{CV2}(s)$$ (24)

where $\epsilon_{CV2}$ denotes the valve opening of CV2. In addition, the following input-output relationship is derived by considering the connection conditions of $\theta_{CV2}^{W,in1}(t) = \theta_{RD}^{W,in}(t)$, and $\theta_{CV2}^{W,in2}(t) = \theta_{RD}^{W,out}(t)$ in Eq. (17):

$$\theta_{CV2}^{W,out}(s) = \Delta \theta_{RD}^{W}e_{CV2}(s) + \epsilon_{CV2}\theta_{RD}^{W,in}(s) + (1 - \epsilon_{CV2})\theta_{RD}^{W,out}(s)$$ (25)

where $\Delta \theta_{RD}^{W}$ denotes $\Delta \theta_{RD}$ at the balance point ($= \theta_{RD}^{W,in,0} - \theta_{RD}^{W,out,0}$); and the superscript 0 denotes the balance point. Moreover, the transfer function, $G_{CV2}$, expressing the response of the valve opening to the change in the manipulated signal, $u_{CV2}$, is derived as follows:

$$e_{CV2}(s) = G_{CV2}(s)u_{CV2}(s)$$ (26)

For TI2, the transfer function, $G_{TI2}$, expressing the response of the temperature signal, $\theta_{TI2}^{W}$,
to the change in the measured hot water, $\theta_{CV2}^{W}$, is derived as follows:

$$
\theta_{CV2}^{W}(s) = G_{CV2}(s)\theta_{CV2}^{W}(s) \quad (27)
$$

Both $G_{CV2}$ and $G_{TI2}$ are first-order lag elements. For TC2, the following equations are derived by considering the control algorithm to be a continuous type proportional-plus-integral action and the set point of the hot water temperature to be constant:

$$
e_{TC2}(s) = -\theta_{TI2}^{W}(s) \quad (28)
$$

where $G_{TC2}$ denotes the transfer function expressing the continuous type proportional-plus-integral action. The block diagram expressing the response of $\theta_{RD}^{W}$ to the change in the supply water flow rate is shown in Fig. 3. From this block diagram, the transfer functions expressing the responses of $\theta_{RD}^{W}$ to the change in the supply water flow rate are derived as follows:

$$
\theta_{RD}^{W}(s) = G_{HE}(s)G_{PP}(s) \quad (29)
$$

$$
\theta_{RD}^{W}(s) = G_{HE}(s)G_{PP}(s)\left[1 + \Delta\theta_{RD}^{W}G_{TC2}(s)G_{CV2}(s)(1 - \epsilon_{CV2})G_{RD2}(s)\right] \quad (30)
$$

From Eqs. (29) and (30), the transfer function expressing the response of $\Delta\theta_{RD}$ to the change in the supply water flow rate is derived as follows:

$$
\frac{\Delta\theta_{RD}(s)}{gs(s)} = \frac{\theta_{RD}^{W}(s) - \theta_{RD}^{W}(s)}{gs(s)} = \frac{\theta_{RD}^{W}(s)}{gs(s)} - \frac{\theta_{RD}^{W}(s)}{gs(s)} \quad (31)
$$

4.3.2 Dynamic compensation based on time constant

The frequency characteristic for the response of $\Delta\theta_{RD}$ to the change in the supply water flow rate, which is calculated using Eq. (31), is shown in Fig. 4. The normalized supply water flow rate at the balance point is 70%. The result calculated from the transfer function, which is shown as the solid lines, agrees very well with that derived from the process simulator, which is shown as the plots. This response has high-order response lags. However, the response lag that results from the transit time of the hot water in RD is seen at low frequencies, and the response lags of the other devices are seen at high frequencies.
Furthermore, the relationship between the cutoff frequency, which is shown in the plots, and the supply water flow rate is shown in Fig. 5. The cutoff frequency decreases with the increase in the supply water flow rate, because of the increase in the transit time of the hot water in RD. Thus, the approximation equation, \( f_{RD}(\theta_S) \), which is shown as the solid line, is identified as a function of the supply water flow rate. From this identification equation, the time constant expressing the response of \( \theta_{RD} \) to the change in the supply water flow rate in a given time, which is defined as \( T_{RD,t} \), is formulated on the basis of Eq. (2) as follows:

\[
T_{RD,t} = \frac{1}{2\pi f_{RD}(\theta_S^M)}
\]  

(32)

where \( \theta_S^M \) denotes the average supply water flow rate in a given time. By using this time constant, the estimated value of \( \Delta \theta_{RD} \) in a given time, to which the dynamic compensation is applied, is calculated on the basis of Eq. (3) as follows:

\[
\Delta \theta_{RD,t}^E = \Delta \theta_{RD,t}^E + \left( \Delta \theta_{RD,t}^E \theta_S^M - \Delta \theta_{RD,t}^E \theta_S^M \right) \left( 1 - \frac{1}{e^{\frac{dt}{T_{RD,t}}}} \right)
\]  

(33)

where \( \Delta \theta_{RD,t}^E \) denotes \( \Delta \theta_{RD} \) calculated in response to \( \theta_S^M \) using the input-output model.

4.4 Monitoring conditions

The sampling time interval used to measure the input and output process variables and the averaging time interval, \( dt \), are set to 1 s and 60 s, respectively, through a trial-and-error process. This is because a long averaging time interval can reduce the influence of measurement noises, but makes it more difficult to grasp the influence of the variations in the operating conditions. As a result, the estimation of the output process variables using the input-output model and dynamic compensation, and the performance monitoring are performed every 60 s. For comparison, \( \Delta \theta_{RD} \) that is estimated without the dynamic compensation is also considered. As described below, the monitoring indicators defined in Eq. (4) fluctuate in the vicinity of the normal value (1.0) even under the normal condition because of the wide range of the variations in the supply water flow rate. Therefore, the period used to detect continuous deviations in the monitoring indicators from their thresholds is set to be longer than their settling times, which coincide with about four times their time constants. From Fig. 5 and Eq. (32), it is found that the time constant for the response of \( \Delta \theta_{RD} \) to the change in the supply water flow rate, with the largest value for the time constants of the target process, is up to 30 s. Thus, the aforementioned period is set to be 5 min; the discussion of the optimal period is a topic for future work. The thresholds for the monitoring indicators used to detect device failures will be set through performance monitoring under the normal condition in Section 5. In addition, if device failures in the target process may occur suddenly, these can be detected by directly monitoring whether the process variables move beyond their upper and lower limits because considerable influences on the process variables can be immediately visible. This means that this performance monitoring focuses only on device failures with long time scales.
5. Performance monitoring under normal condition

To verify the effectiveness of the developed performance monitoring method and determine the thresholds for the monitoring indicators to detect device failures, performance monitoring is performed under the normal condition. This section focuses only on performance monitoring in the dynamic state with an irregularly fluctuating supply water flow rate, because in the steady state, the output process variables estimated by the input-output model agree very well with those calculated by the process simulator.

As a performance monitoring result, the time evolution of the supply water flow rate and the monitoring indicators for CL1 and CL2, which are calculated every 60 s, is shown in Fig. 6. Any monitoring indicator fluctuates in the vicinity of the normal value (1.0) in

![Normalized supply water flow rate](image1)

(a) Normalized supply water flow rate

![Monitoring indicators in CL1](image2)

(b) Monitoring indicators in CL1

![Monitoring indicators in CL2](image3)

(c) Monitoring indicators in CL2

Fig. 6 Monitoring result of target heat utilization process under normal condition

<table>
<thead>
<tr>
<th>Control loop</th>
<th>Output process variables</th>
<th>Maximum deviations %</th>
<th>Thresholds %</th>
</tr>
</thead>
<tbody>
<tr>
<td>CL1</td>
<td>Water temperature at TI1</td>
<td>0.050</td>
<td>0.060</td>
</tr>
<tr>
<td></td>
<td>Manipulated signal for CV1</td>
<td>0.0</td>
<td>0.2</td>
</tr>
<tr>
<td></td>
<td>Water temperature difference at HE</td>
<td>1.0</td>
<td>1.2</td>
</tr>
<tr>
<td>CL2</td>
<td>Water temperature at TI2</td>
<td>0.052</td>
<td>0.062</td>
</tr>
<tr>
<td></td>
<td>Manipulated signal for CV2</td>
<td>0.24</td>
<td>0.29</td>
</tr>
<tr>
<td></td>
<td>Water temperature difference at RD (w dynamic compensation)</td>
<td>2.9</td>
<td>3.5</td>
</tr>
<tr>
<td></td>
<td>Water temperature difference at RD (w/o dynamic compensation)</td>
<td>5.9</td>
<td>7.1</td>
</tr>
</tbody>
</table>
response to variations in the supply water flow rate. However, the fluctuation width varies depending on the output process variables. In all of the output process variables except $\Delta \theta_{RD}$, which are estimated using only the input-output model, the monitoring indicator for the hot water temperature difference between the inlet and the outlet of HE, $\Delta \theta_{HE}$, has the widest fluctuation. However, its maximum deviation is about $\pm 1\%$. For $\Delta \theta_{RD}$, the fluctuation in the monitoring indicator employing the dynamic compensation is considerably smaller than that without the dynamic compensation. This is remarkable with high supply water flow rates, where the response of $\Delta \theta_{RD}$ is very gradual because of low flow rates of the inflow hot water to RD. This result indicates the effectiveness of the developed performance monitoring method.

The thresholds for the monitoring indicators are set to be 1.2 times their maximum deviations, which are derived from Fig. 6, and are listed in Table 3. The threshold for the manipulated signal of TC1 is not considered, because it is fixed at 100% under the operating condition being considered, as described in Subsection 4.3.1.

6. Performance monitoring under device failures

6.1 Modeling of device failures

The performance monitoring is conducted under the failure conditions for RD and HE, which are the controlled objects in CL2 and CL1, respectively. In this subsection, the failures of RD and HE that are incorporated into the process simulator are modeled. The reason why the failures of RD and HE are focused on is due to the following two motivations: one is to verify the effectiveness of the developed dynamic compensation by monitoring RD; and the other is to verify the capability to distinguish the failure occurred in RD, to which the dynamic compensation is applied, from failures occurred in other devices.

As a representative failure with a long time scale for RD, the deterioration in the heat transfer performance as a result of fouling in the air passing the fins is focused on. By changing the calculating formula for the overall heat transfer coefficient of RD, $K_{RD}$, from Eq. (13) to the following equation, the decrease in the apparent heat transfer coefficient of the fin side, $a_{RD}^{AP}$, caused by the increase in the fouling factor, $R_p$, is modeled(22).

$$K_{RD} = \frac{1}{A_{RD}} \left[ \frac{1}{a_{RD}^{AP} (A_T^A + \phi_T A_P)} + \frac{1}{a_{RD}^W A_T^W} \right]^{-1}$$

$$a_{RD}^{AP} = \left[ \frac{1}{A_{RD}^{AP}} + R_p \right]^{-1}$$

$R_p$ increases linearly over time from the initial value. Moreover, as a representative failure with a long time scale for HE, the deterioration in the heat transfer performance caused by fouling in the supply water passing through the plates is focused on. By changing the calculating formula for the overall heat transfer coefficient of HE, $K_{HE}$, from Eq. (9) to the following equation, the decrease in the apparent heat transfer coefficient of the low-temperature side, $a_{HE}^{LP}$, caused by the increase in the fouling factor, $R_p$, is modeled(16).

$$K_{HE} = \left[ \frac{1}{a_{HE}^{LP} \lambda_T^P} + \frac{1}{a_{HE}^{LP} A_P^L \lambda_T^P} \right]^{-1}$$

$$a_{HE}^{LP} = \left[ \frac{1}{a_{HE}^{LP}} + R_p \right]^{-1}$$

$R_p$ increases linearly over time from the initial value.

6.2 Performance monitoring under deterioration in heat transfer performance of RD

As a performance monitoring result under the deterioration in the heat transfer performance of RD, the time evolution of the measured and estimated value of $\Delta \theta_{RD}$, and
the monitoring indicators for CL1 and CL2 is shown in Fig. 7; the variation in the supply water flow rate is the same as that in Fig. 6. $R_F$ increases linearly by $4.0 \times 10^{-6} \text{m}^2 \cdot \text{K/(W\cdot s)}$ from the broken line, and the initial value of $R_F$ is 0. First, it is difficult to use the direct monitoring of $\Delta \theta_{RD}$ to detect the performance deterioration of RD because $\Delta \theta_{RD}$ fluctuates widely depending on the variation in the supply water flow rate. However, the measured value, $\Delta \theta_{RD}^M$, obviously decreases over time as compared to the estimated values, $\Delta \theta_{RD}^E$ and $\Delta \theta_{RD}^S$. The monitoring indicators for $\Delta \theta_{RD}$ decrease over time with or without the dynamic compensation. However, the dynamic compensation can considerably suppress the fluctuation in the monitoring indicator. As a result, the performance monitoring that employs the dynamic compensation detects the failure after 94 minutes have elapsed from starting the performance deterioration (the dash-dotted line), whereas that without the dynamic compensation detects the failure after 185 minutes have elapsed (the two dot-chain line). The overall heat transfer coefficient of RD in detecting the failure with and without the dynamic compensation decrease by 8.3% and 23.0%, respectively, as compared with that under the normal condition. These results show the advantage of installing the dynamic compensation over the model-based monitoring method without it. The monitoring indicator for the manipulated signal for CV2 is also decreased, and this is detected after 185 minutes have elapsed. This is because the flow rate of the inflow hot water to RD is

![Monitoring result of target heat utilization process under RD failure condition](image-url)
increased to maintain the hot water temperature measured at TI2. Furthermore, Fig. 7(b) shows that the performance deterioration of RD has no influence on the monitoring indicators for CL1 because of the control action of CL2.

6.3 Performance monitoring under deterioration in heat transfer performance of HE

As a performance monitoring result under the deterioration in the heat transfer performance of HE, the time evolution of the measured and estimated values of $\Delta \theta_{HE}$ and the monitoring indicators for CL1 and CL2 is shown in Fig. 8. The variation in the supply water flow rate is the same as that in Fig. 6. $R_P$ increases linearly by $3.0 \times 10^{-9} \text{ m}^2\cdot\text{K/(W}\cdot\text{s})$ from the broken line, and the initial value of $R_P$ is 0. The measured value, $\Delta \theta_{HE}^M$, fluctuates widely in response to the variation in the supply water flow rate; however, it decreases over time as compared with the estimated value, $\Delta \theta_{HE}^S$. In Fig. 8(b), this is detected after 15 minutes have elapsed from starting the performance deterioration (the dash-dotted line), and the increase in the monitoring indicator for the hot water temperature measured at TI1 is also detected at the same time. For CL2, the decrease in the monitoring indicators for $\Delta \theta_{RD}$ is detected, and the detection occurs much earlier when employing the dynamic compensation. Moreover, the monitoring indicator for the manipulated signal for CV2 is decreased; thus, the monitoring indicators for CL2 have the same performance deterioration.
trend for both RD and HE. However, the combined evaluation of CL1 and CL2 makes it possible to distinguish between the RD and HE performance deteriorations because the trend for the monitoring indicators for CL1 differs between the performance deteriorations of RD and HE.

Actually, the performances of HE and RD may deteriorate with much longer time scales. Although the results are not shown here, it was confirmed that the performance monitoring that employs the dynamic compensation has an advantage in detecting the RD and HE performance deteriorations with very gradual increases in the fouling factors over that without the dynamic compensation. It was also confirmed that the developed performance monitoring can detect the occurrence of failures in the other devices, including GE, CVs, and TIs.

7. Conclusions

A model-based performance monitoring method was developed for heat utilization processes in distributed energy systems consisting of multiple temperature control loops composed of heat exchangers and measurement and control devices. This method extended the previously developed method that used a static input-output model. It is characterized by the introduction of dynamic compensation, where the response lags of heat exchangers to variations in their operating conditions are identified as first-order lag elements, and the output process variables estimated using the static input-output model are revised on the basis of these identified response lags. Furthermore, a case study was conducted through the numerical simulation for a heat utilization process in a gas engine cogeneration system containing a radiator with a considerable response lag. The results showed that the dynamic compensation considerably improves the estimation accuracy for the hot water temperature difference between the inlet and the outlet of the radiator in the dynamic state. This helps to lower the threshold of the monitoring indicator to detect device failures. Moreover, it was revealed that a deterioration in the performance of the radiator could be detected earlier by employing the dynamic compensation, and the performance deteriorations of the radiator and heat exchanger could be distinguished by monitoring the trends of the control loops. Hence, the effectiveness of the developed performance monitoring method in the dynamic state was confirmed.

As another model-based performance monitoring method, a comparison could also be considered between the measured output process variables and the results sequentially computed using a dynamic process simulation. However, this paper emphasizes the implementation of the developed performance monitoring in existing control systems. From this viewpoint, the developed performance monitoring method has the advantage of being able to suppress an increase in the computational load of a control system by the use of arithmetic processing at the averaging time interval. Moreover, in the input-output model, the number and configuration of the devices in the process can be easily changed because the models for the devices are modularized. Furthermore, the numerical analysis in this paper suggests that the dynamic compensation should only be employed for output process variables with considerable response lags; the time constants for these output process variables can be easily calculated from the frequency characteristics that are obtained using the transfer functions. Therefore, the developed performance monitoring method can be applied to other heat utilization processes.

As future works, case studies on performance monitoring will be conducted to discuss the monitoring capability in seasonal and hourly changes in the supply water and ambient temperatures, and in the operating conditions that CL1 is operated. The monitoring conditions must also be optimized. Finally, a demonstration of the developed performance monitoring method, including the identification of failed devices, will be conducted.
References


