Model-based Performance Monitoring of Heat Exchange Process in Transient State^{*}

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Abstract

A model-based performance monitoring method for a heat exchange process, which is one of the key components of distributed energy supply systems, is developed. This performance monitoring, which utilizes a static input-output model of the process, first estimates the output process variables and heat exchange performance in response to variations in measured input process variables. The estimated output process variables are then compared with the measured ones. In this study, the effectiveness of the performance monitoring method in a transient state with irregular variations in water and steam flow rates is verified through a numerical simulation. The result shows sufficiency of the estimation accuracy of the output process variables and heat exchange performance and the capability of detecting the deterioration in the heat exchange performance in a transient state.

Key words: Monitoring, Model-based Monitoring, Diagnostics, Heat Exchanger, Distributed Energy System, Heat Exchange Performance

1. Introduction

Recently, distributed energy supply systems, which are expected to contribute to energy conservation, have been extended to not only industrial and commercial applications but also residential ones because of the relaxation of regulations for their operation and maintenance. However, it is not realistic from the viewpoint of human resources and personnel cost for system management personnel to constantly monitor many systems. Therefore, the development of a self-monitoring method that detects failures and identifies the causes of these failures during the operation of these systems is very important. Moreover, distributed energy supply systems require adequate operation in response to variations in energy demands in order to improve their energy conservation effects. For meeting this objective, an advanced management system with the ability to predict energy demands for operational planning, carry out optimal operational planning on the basis of predicted energy demands, and achieve optimal control in response to the derived operational planning has been developed⁽¹⁾. In this advanced management system, the optimal operational planning and optimal control are carried out by using performances of system components; however, their performances are influenced by seasonal variations and deterioration with age. Therefore, in this advanced management system, performance monitoring is necessary to update the performance data.

Several methods for monitoring energy supply and process systems have been proposed. Venkatasubramanian et al. classified the methods into three types: quantitative model-based methods⁽²⁾, qualitative model-based methods⁽³⁾, and process-history-based methods⁽⁴⁾. In the qualitative model-based methods, monitoring is carried out using the qualitative knowledge of target systems, which includes pattern recognitions⁽⁵⁾, artificial neural networks⁽⁶⁾, and expert systems⁽⁷⁾. The process-history-based methods carry out monitoring using previous operation data; studies using the identification model⁽⁸⁾ and a principal component analysis⁽⁹⁾ have been reported. The advantage of the qualitative model-based and process-history-based methods is that they can monitor target systems

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without any detailed information about these systems and theoretical models; however, the performances cannot be quantitatively evaluated for optimal operational planning and optimal control. The quantitative model-based methods, which conduct monitoring on the basis of residuals between measured variables and those calculated using mathematical models, can evaluate the performances of target systems quantitatively. A model-based performance monitoring method for combined cycle power plants has already been proposed⁽¹⁰⁾; however, variations in the operating conditions of the plants have not been sufficiently discussed. In distributed energy supply systems, it is very important to monitor whether systems have normal performances in response to variations in their operating conditions, which depend on variations in energy demands.

The objective of our study is to develop a model-based performance monitoring method for distributed energy supply systems that is useful to variations in their operating conditions. As the first phase of the study, a heat exchange process between steam and water, which is one of the key components of distributed energy supply systems, is focused on. A previous study has revealed the effectiveness of performance monitoring using a static input-output model through experiments performed not only under static operating conditions but also during deterioration in the heat exchange performance owing to the air accumulation in the shell of the heat exchanger⁽¹¹⁾. In this study, the effectiveness of model-based performance monitoring in a transient state is verified through a numerical simulation, because distributed energy supply systems are often in a transient state owing to variations in their operating conditions. The monitoring result for irregular variations in water and steam flow rates shows sufficiency of the estimation accuracy of the heat exchange performance and capability to detect the deterioration in the heat exchange performance in a transient state.

2. Nomenclatures

A	heat transfer area	m^2
c	specific heat	$J/(kgK)$
d	diameter	m
G	mass flow rate	kg/s
h'	specific enthalpy of saturated liquid	J/kg
h''	specific enthalpy of saturated vapor	J/kg
HEP	vector for heat exchange performance	
P	pressure	kPa
PV	vector for process variables	
S	sectional area	m^2
t	time	s
U	tube surface area per unit length	m^2/m
V	volume	m^3
W	water mass inside shell	kg
Δx	control volume length	m
α	heat transfer coefficient	$W/(m^2K)$
δ	performance deterioration rate	$\%/s$
λ	heat conductivity	$W/(mK)$
θ	temperature	K
ρ	density	kg/m^3
Superscript		
E	estimated value	M measured value
Subscript		
a	ambient air	C condensate
H	header	i inside
in	inlet	o outside

out	outlet	S	steam
SH	shell	T	tube
W	water		

3. Heat exchange process⁽¹²⁾

Figure 1 shows the general view of a heat exchange process using a shell-and-tube-type heat exchanger; this is the target process in this study. Dry saturated steam generated in the boiler flows into the shell of the heat exchanger through a flow control valve. In the shell, steam condenses completely because of the level control of the evacuated condensate in the drain pot. Water pumped up from a water pool is heated by passing through tubes and is then returned to the pool; the water temperature in the pool is kept approximately constant by the cooling tower.

Figure 1 also shows the measurement and control instruments in this process. For water, the inlet and outlet temperatures, inlet pressure, and flow rate are measured; for steam, the upstream temperature and pressure of the flow control valve, flow rate, pressure inside the shell, and condensate level in the drain pot are measured. The steam flow rate, upstream steam pressure of the flow control valve, condensate level, and water flow rate are controlled to operate the process. Both the control and measuring intervals are 1 s.

Fig.1 General view of a heat exchange process

4. Numerical simulation model for heat exchange process

In this study, a numerical simulation model for the target heat exchange process is first constructed in order to verify the effectiveness of our model-based performance monitoring in a transient state. This model makes it possible to simulate both process responses in a given transient state and various types of process failures.

4.1 Modeling

In this numerical simulation model, the steam inside the shell is considered to be under the saturated state and treated as a lumped parameter system, whereas water and the tubes are treated as a distributed parameter system, which has N control volumes, because the retention time and the temperature distribution are not negligible. Moreover, the following conditions are assumed for the modeling:

- There is no temperature gradient in the shell.
- The saturation temperature corresponding to the pressure inside the shell is regarded as the steam condensing temperature in the shell.
- The condensate is immediately evacuated from the shell.
- There are no streamwise mixture and radial temperature and flow velocity distributions for the water inside the tubes.
- The pressure loss of the water inside the tubes is ignored.

- There is no difference in the water flow rate and the temperature distribution among the tubes.
- Heat transfer coefficients inside and outside the tube have no distribution in the longitudinal direction of the tubes.
- There is no enthalpy loss of steam at the flow control valve.

(a) Water inside tube

The heat balance for the water inside M tubes in the n th control volume is expressed by the following differential equation:

$$\rho_W c_W M S_i \Delta x \frac{d\theta_{W,n}(t)}{dt} + G_W c_W \{\theta_{W,n}(t) - \theta_{W,n-1}(t)\} = \alpha_i M U_i \Delta x \{\theta_{Ti,n}(t) - \theta_{W,n}(t)\} \quad (1)$$

$(n=1,2,\dots,N)$

where the heat transfer coefficient inside the tube α_i is calculated using the Sieder-Tate equation for the turbulent heat transfer inside tubes⁽¹³⁾.

(b) Tube

The heat balance for M tubes in the n th control volume is expressed by the following differential equation:

$$\rho_T c_T M S_T \Delta x \frac{d\theta_{T,n}(t)}{dt} = \alpha_o M U_o \Delta x \{\theta_C(t) - \theta_{To,n}(t)\} - \alpha_i M U_i \Delta x \{\theta_{Ti,n}(t) - \theta_{W,n}(t)\} \quad (2)$$

$(n=1,2,\dots,N)$

where the heat transfer coefficient outside the tube α_o is calculated using the Nusselt equation for the film condensation heat transfer on the surface of horizontal smooth tubes⁽¹⁴⁾. The representative tube wall temperature $\theta_{T,n}$ in Eq. (2) is calculated from the following algebraic equations:

$$\theta_{Ti,n}(t) = \frac{\theta_{T,n}(t) + C_i \theta_{W,n}(t)}{1 + C_i} \quad (n=1,2,\dots,N) \quad (3)$$

$$\theta_{To,n}(t) = \frac{\theta_{T,n}(t) - C_o \theta_C(t)}{1 - C_o} \quad (n=1,2,\dots,N) \quad (4)$$

where dimensionless coefficients C_i and C_o are expressed by the following algebraic equations considering the heat conductivity in the tube.

$$C_i = \left(\frac{d_i}{d_o - d_i} - \frac{1}{\ln(d_o/d_i)} \right) \frac{d_i}{2} \frac{\alpha_i}{\lambda_T} \ln \left(\frac{d_o}{d_i} \right) \quad (5)$$

$$C_o = \left(\frac{d_o}{d_o - d_i} - \frac{1}{\ln(d_o/d_i)} \right) \frac{d_o}{2} \frac{\alpha_o}{\lambda_T} \ln \left(\frac{d_o}{d_i} \right) \quad (6)$$

(c) Steam inside shell

The heat balance for the steam inside the shell is expressed by the following differential equation:

$$\frac{d}{dt} \{h''(\theta_C) \mathcal{W}_S + h'(\theta_C) \mathcal{W}_C\} = G_S \{h''(P_S) - h'(\theta_C)\} - \alpha_o M U_o \Delta x \sum_{n=1}^N \{\theta_C(t) - \theta_{To,n}(t)\} - \alpha_{SHi} A_{SHi} \{\theta_C(t) - \theta_{SH}(t)\} \quad (7)$$

where the heat transfer coefficient inside the shell α_{SHi} is calculated using the Nusselt equation for the laminar film condensation heat transfer inside horizontal tubes⁽¹⁵⁾.

(d) Outlet header for water

Water heated by passing through the tubes converges in the outlet header and then flows out from the heat exchanger. This is regarded as a complete mixing process and formulated as a first-order lag element.

$$\rho_W c_W V_H \frac{d\theta_{Wout}(t)}{dt} = G_W c_W \{\theta_{W,N}(t) - \theta_{Wout}(t)\} \quad (8)$$

(e) Shell

A portion of the condensed latent heat of steam is radiated to ambient air through the shell. This is expressed by the following differential equation:

$$\rho_{SH} c_{SH} V_{SH} \frac{d\theta_{SH}(t)}{dt} = \alpha_{SHi} A_{SHi} \{\theta_C(t) - \theta_{SH}(t)\} - \alpha_{SHo} A_{SHo} \{\theta_{SH}(t) - \theta_a(t)\} \quad (9)$$

where the heat transfer coefficient outside the shell α_{SHo} is identified through experiments.

4.2 Numerical solution and calculation condition

The aforementioned Eqs. (1)–(9) and the boundary conditions, which are the steam flow rate G_S , upstream steam pressure of the flow control valve P_S (hereafter referred to as the steam pressure), water flow rate G_W , and water inlet temperature θ_{Win} , form a set of nonlinear differential algebraic equations. This set of equations is solved by using a combination of the Euler and Newton-Raphson methods.

The specifications of the heat exchange process are listed in Table 1. The properties for water and steam are calculated by using PROPATH Ver. 12.1. The discrete time for the Euler method is 0.25 s.

Table 1 Specifications of heat exchange process

Item			Specification
Number of control volumes	N		20
Number of tubes	M		4
Heat transfer area	A	m^2	0.728
Inside surface area of shell	A_{SHi}	m^2	0.905
Outside surface area of shell	A_{SHo}	m^2	0.946
Specific heat of shell wall	c_{SH}	$J/(kg \cdot K)$	481
Specific heat of tube	c_T	$J/(kg \cdot K)$	396
Inside diameter of tube	d_i	m	1.66×10^{-2}
Outside diameter of tube	d_o	m	1.91×10^{-2}
Volume of water header	V_H	m^3	3.78×10^{-3}
Volume of shell wall	V_{SH}	m^3	4.19×10^{-2}
Heat transfer coefficient	α_{SHo}	$W/(m^2 \cdot K)$	61.1
Density of shell wall	ρ_{SH}	kg/m^3	7 860
Density of tube	ρ_T	kg/m^3	8 950
Heat conductivity of tube	λ_T	$W/(m \cdot K)$	385

4.3 Model validation

For the model validation, the simulation result of the process response to the changes in the steam and water flow rates is compared with the experimental result⁽¹¹⁾. Figure 2(a) shows the experimental result of the change in the steam flow rate from 152 kg/h to 182 kg/h and the two-stage change in the water flow rate from 1500 kg/h to 1700 kg/h per 100 kg/h; the steam pressure is 0.6 MPa, and the water inlet temperature and pressure are 35 °C and 0.4 MPa, respectively. The changes in the flow rates are brought about by changing the set point values in the controllers; however, the change in the water flow rate is gradual,

Fig. 2 Comparison result between experiment and simulation in changes in water and steam flow rates

because the proportional gain in the controller is small in order to prevent the influence of the flow noise. Figure 2(b) shows the response of the water outlet temperature and the temperature inside the shell; the simulation result of both the temperatures agrees very well with the experimental result. Hence, in the following section, the variables of the simulation result are regarded as the measured process variables.

5. Model-based performance monitoring⁽¹¹⁾

5.1 Outline of model-based performance monitoring

Failures of distributed energy supply systems are generally classified into two types: accidental ones and gradual ones with a long time scale. Accidental failures, which include tube bursts and steam spouts, immediately have a considerable influence on process variables; hence, they can be detected by directly monitoring the process variables over their upper and lower limits on an ordinary measurement and control system. On the other hand, gradual failures, which affect the optimal operation planning and control, cannot be detected only by monitoring the process variables. Therefore, this study focuses on the monitoring for the deterioration in heat exchange performance, which is a representative example of gradual failures.

Figure 3 shows a schematic of our model-based performance monitoring. Preliminarily, process variables are classified into input and output variables. The output process variables under normal operating conditions without failures PV_{out}^E are estimated in response to variations in the measured input process variables PV_{in}^M using a static input-output model; in order to prevent the influence of measurement noise, the measured input process variables are averaged before the estimation of the output process variables. The static input-output model requires the measured output process variables only to estimate the heat discharge to ambient air. In addition, the heat exchange performance HEP^E is estimated using the estimated output process variables. Then, the measured output process variables PV_{out}^M and measured heat exchange performance HEP^M , which are also averaged, are compared with their estimated values. This method can monitor whether the process exhibits normal performance in response to changes in its operating condition.

Fig. 3 Schematic of model-based performance monitoring

5.2 Definition of input and output process variables

To evaluate the heat exchange performance, the following process variables are required: steam flow rate and pressure, temperature inside the shell, water flow rate, and water inlet and outlet temperatures. In this study, the steam flow rate, steam pressure, water flow rate, and water inlet temperature are considered as the input process variables. The output process variables, namely, the water outlet temperature and the temperature inside the shell, are estimated.

5.3 Summary of static input-output model

The static input-output model is summarized here; its details are described in Ref. (11). The static input-output model mainly consists of a heat balance equation and temperature

relational equation. The heat balance equation is formulated in a lumped parameter system. The temperature relational equation among the water inlet and outlet temperatures and the temperature inside the shell is derived by solving the local heat balance equation of the process. Moreover, in order to consider the change in the heat exchange performance owing to variations in the operating conditions, the overall heat transfer coefficient is estimated; the heat transfer coefficients inside and outside the tube, which are required to estimate the overall heat transfer coefficient, are calculated using the Sieder-Tate equation⁽¹³⁾ and the Nusselt equation⁽¹⁴⁾, respectively, as in the numerical simulation model. Furthermore, the performance correction factor is introduced to compensate for the model errors.

The static input-output model and the boundary conditions, which are the measured input process variables, form a set of nonlinear algebraic equations. By solving this set of equations using the Newton-Raphson method, the output process variables are estimated.

5.4 Monitoring conditions

This model-based performance monitoring focuses on not only the output process variables but also the heat exchange performance; this implies that the overall heat transfer coefficient is calculated using the water-side exchanged heat rate and the heat transfer coefficients inside and outside the tube. Although the overall heat transfer coefficient is generally used for the evaluation of the heat exchange performance in a steady state, this study employs it for the monitoring the performance during operation. The calculation procedure for the estimated and measured overall heat transfer coefficients is described in Ref. (11). The comparison of the estimated and the measured values of the output process variables and heat exchange performance is carried out after averaging these values; thus, the monitoring interval coincides with the averaging time interval (1 min in this study).

6. Performance monitoring in transient state

The effectiveness of model-based performance monitoring in a transient state is verified through a numerical simulation. First, the process performance under the normal operating conditions without failures is monitored. Then, the capability to detect the deterioration in the heat exchange performance is evaluated.

6.1 Performance monitoring under normal operating conditions

6.1.1 Performance monitoring in the case of irregular variations in flow rates

As a performance monitoring result in a transient state where the water and steam flow rates change irregularly, Fig. 4 shows the comparison between the simulation for the output process variables and heat exchange performance as alternatives of their measured values and the estimation derived using the static input-output model. As shown in Fig. 4(a), three types of transient states are focused on: separate variations in the water and steam flow rates and their simultaneous variation. Noise signals are superimposed on the water and steam flow rates to conduct a realistic simulation. The steam pressure is 0.6 MPa, and the water inlet temperature and pressure are 35 °C and 0.4 MPa, respectively. The operating conditions of this process are restricted by the following two reasons: (1) The water outlet temperature must be below 100 °C to supply hot water at atmospheric pressure. (2) The temperature inside the shell must be above 100 °C to prevent the development of negative pressure in the shell. The irregular variations in the water and steam flow rates shown in Fig. 4(a) meet these operating restrictions.

Figure 4(b) shows that the estimated output process variables agree very well with the simulated ones in a transient state. This is attributed to the fact that the estimated output process variables exhibit lag characteristics to changes in the water and steam flow rates, because the average processing for the measured input process variables acts as a high-cut filter. The time constant in the process response shown in Fig. 2 is around 1 min and roughly equal to the averaging time interval.

Fig. 4 Simulation and estimation for process response in case of irregular variations in water and steam flow rates

As shown in Fig. 4(c), the overall heat transfer coefficient varies greatly with the variation in the water flow rate, because the variation in the water flow rate directly causes the variation in the water-side exchanged heat rate. However, the variation in the overall heat transfer coefficient to the steam flow rate is small, because the variations in the water-side exchanged heat rate to the steam flow rate are gradual. Moreover, the variation in the simulated overall heat transfer coefficient is wider than that in the estimated one for any flow rate variation. This is caused by the slight difference in the time constant between the simulated water outlet temperature and the simulated temperature inside the shell; although both the time constants seem to be almost equal as shown in Fig. 2, the actual time constant for the water outlet temperature is slightly longer than that for the temperature inside the shell because of the lag in the mixing process at the outlet header.

The ratio of the simulated output process variables and overall heat transfer coefficient to their estimated values is evaluated; hereafter, this ratio is referred to as the monitoring indicator. Figure 5 shows the monitoring indicators derived from the result in Fig. 4. All the monitoring indicators are the largest in the case of variations in both the water and steam flow rates and the smallest in the case of a variation in only the steam flow rate. For the water outlet temperature and the temperature inside the shell, the deviation of the monitoring indicators from the normal value (1.0) is very small. On the other hand, the deviation for the overall heat transfer coefficient is up to 3%; however, the large deviation does not last for a long time.

Fig. 5 Monitoring indicators in case of irregular variations in water and steam flow rates

(a) Minimum variation period : 1 min

(b) Minimum variation period : 2 min

(c) Minimum variation period : 5 min

(d) Minimum variation period : 10 min

Fig. 6 Influence of minimum variation period for water and steam flow rates on monitoring indicators

6.1.2 Influence of minimum variation period of flow rates on monitoring indicators

Although the process performance under the normal operating condition in the case of irregular variations in the water and steam flow rates can be monitored using the static input-output model, the monitoring capability may be influenced by the variation period of the flow rates. Therefore, the process performance in the case of irregular variations in the flow rates with different minimum periods is monitored. Figure 6 shows the monitoring indicators in the case of irregular variations in the water and steam flow rates, whose minimum period is 1, 2, 5, and 10 min; the irregular variations in the water and steam flow rates in Figs. 6(a)–(c) are derived by superposing the irregular variations with a minimum period of 1, 2, and 5 min on the irregular variation with the minimum period of 10 min (Fig. 6(d)). The result for the minimum variation period of 2 min coincides with that shown in Fig. 5.

In the case of any irregular variations in the flow rates, the deviation of the monitoring indicators from the normal value for the water outlet temperature and the temperature inside the shell is very small, whereas the deviation for the overall heat transfer coefficient is large in the case of the minimum variation periods of 2 and 5 min. Because the process response can follow the variations in the flow rates when the minimum period is longer than the settling times, which are around 5 min as shown in Fig. 2, the deviation in the minimum variation period of 10 min is small. Moreover, the deviation in the minimum variation period of 1 min is small because the irregular variations in the flow rates with the minimum period are attenuated by the average processing. As a result, it seems that Fig. 6(a) with the minimum variation period of 1 min coincides with Fig. 6(d) with the minimum variation period of 10 min. Therefore, by tolerating transiently large deviations of the monitoring indicators, the process performance in a transient state can be monitored using the static input-output model.

6.2 Detection of deterioration in heat exchange performance in transient state

6.2.1 Modeling of deterioration in heat exchange performance

Causes of deterioration in the heat exchange performance in this process include air accumulation in the shell⁽¹¹⁾, scale adhesion outside the tubes⁽¹⁶⁾, and blockage in the tubes⁽¹⁷⁾. As a typical example, the scale adhesion outside the tubes is focused on.

The deterioration in the heat exchange performance owing to the scale adhesion outside the tubes may progress nonlinearly with time; however, it is assumed that the heat transfer coefficient outside the tube reduces at a constant rate owing to the scale adhesion. That is, the heat transfer coefficient outside the tube α_o in eqs. (2) and (7) for the numerical simulation model is reduced at a constant rate δ compared to the heat transfer coefficient outside the tube α_o^M , which is calculated from the simulated process variables.

$$\alpha_o = \left\{ 1 - \frac{\delta}{100}(t - t_0) \right\} \alpha_o^M \quad (10)$$

where t_0 denotes the start time of the reduction in the heat transfer coefficient outside the tube.

6.2.2 Detection of deterioration in heat exchange performance

As a performance monitoring result in the case of the deterioration in the heat exchange performance, Fig. 7 shows the comparison between the simulated and the estimated values of the output process variables, overall heat transfer coefficient, and monitoring indicator in the case of irregular variations in the water and steam flow rates with a minimum period of 2 min where the deviation of the monitoring indicator from the normal value is the largest in the former analysis. The reduction rate in Eq. (10) is 0.004 %/s. The heat transfer coefficient outside the tube begins to decrease at the positions indicated by the broken line in the figures. Because the simulated output process variables and overall heat transfer coefficient vary widely with the irregular variations in the flow rates as shown in Figs. 7(b) and 7(c), their direct monitoring cannot detect the deterioration in the heat exchange performance. However, the comparison of the simulation with the estimation, it is revealed that in the

Fig. 7 Simulation and estimation for process with deterioration in heat exchange performance in case of irregular variations in water and steam flow rates

case of the deterioration in the heat exchange performance, the simulated water outlet temperature and simulated temperature inside the shell slightly decreases and increases as compared to the estimated values, respectively. As a result, the simulated overall heat transfer coefficient becomes smaller than the estimated one. Moreover, Fig. 7(d) shows that the monitoring indicators under the normal operating conditions fluctuate in the vicinity of the normal value (1.0). The standard deviations for the overall heat transfer coefficient and the temperature inside the shell under normal operating conditions are 0.0205 and 0.00204, respectively. After a lapse of 90 min from the beginning of the performance deterioration, the monitoring indicators for the overall heat transfer coefficient and the temperature inside the shell decrease by 7% and increase by 0.7%, respectively. Therefore, there is no difference in the sensitivity to performance deterioration between them. If the 5-min continuation of deviation in the monitoring indicator of more than 5% from the normal value is regarded as the deterioration in the heat exchange performance, the deterioration is detected at the dash-dotted line in Fig. 7(d).

In actual operation, the deterioration in the heat exchange performance may often progress with a time scale that is considerably longer than that in this analysis; however, it is confirmed that the deterioration in the heat exchange performance with the reduction rates for the considerably smaller heat transfer coefficient outside the tube than that in this analysis can be detected under the aforementioned detection conditions.

6.2.3 Identification of cause of performance deterioration

The static input-output model can also estimate the heat transfer coefficients inside and outside the tube⁽¹¹⁾. In order to evaluate the capability to identify the cause of the performance deterioration, Figure 8 shows the comparison between the simulated and the estimated values of the two heat transfer coefficients derived from the result in Fig. 7 and the result for the malfunction of the orifice flow meter for the water. In the latter result, the measured water flow rate increases at a constant rate (10%/h) as compared to the water flow rate in the simulation owing to the increase in the differential pressure at the orifice; this increase is caused by the decrease in the discharge coefficient of the orifice owing to the contaminant adhesion.

Under normal operating conditions before the beginning of the deterioration in the heat exchange performance and the malfunction of the orifice flow meter, the estimated heat transfer coefficients agree very well with the simulated ones. The heat transfer coefficient inside the tube widely varies, while the variation in the heat transfer coefficient outside the tube is small. This is because the heat transfer coefficient inside the tube is a function of the water flow rate, whereas the heat transfer coefficient outside the tube is a function of the water outlet temperature and the temperatures inside the shell, which have small variations and lag characteristics with respect to the flow rates.

Fig. 8 Simulation and estimation for heat transfer coefficients under deterioration in heat exchange performance and malfunction of orifice flow meter

In the case of the deterioration in the heat exchange performance, the estimated heat transfer coefficient inside the tube still agrees with the simulated one. This is because the heat conductivity and coefficient of viscosity for water and the Nusselt number, Prandtl number, and Reynolds number, which are necessary to calculate the heat transfer coefficient inside the tube, are hardly affected by the change in the process variables owing to the reduction in the heat transfer coefficient outside the tube. The simulated heat transfer coefficient outside the tube is reduced as compared to the estimated one because the difference between the temperature inside the shell and the liquid film temperature on the surface of the tube, which must be calculated on the basis of Ref. (11), becomes larger. On the other hand, in the case of the malfunction of the orifice flow meter, the estimated heat transfer coefficient inside the tube still agrees with the simulated one, while the estimated heat transfer coefficient outside the tube is reduced as compared to the simulated one. This is because the liquid film temperature on the surface of the tube increases because of the increase in the estimated temperature inside the shell.

Consequently, the difference in the comparison result for the heat transfer coefficients inside and outside the tube enables us to distinguish between the deterioration in the heat exchange performance and the malfunction of the orifice flow meter.

7. Conclusions

The effectiveness of the model-based performance monitoring method for the heat exchange process between steam and water in a transient state is verified through a numerical simulation in the case of irregular variations in the water and steam flow rates. In this monitoring, the output process variables and heat exchange performance in response to variations in the measured input process variables are first estimated by using the static input-output model. Then, the estimated output process variables and heat exchange performance are compared with their measured values. The obtained results are summarized as follows:

- The output process variables and heat exchange performance vary widely in transient state; however, the process performance can be estimated by using the static input-output model.
- The deterioration in the heat exchange performance owing to the reduction in the heat transfer coefficient outside the tube can be detected in transient state.
- The estimation of the heat transfer coefficients inside and outside the tube enables us to distinguish between the deterioration in the heat exchange performance and the malfunction of the orifice flow meter.

These results indicate the effectiveness of this model-based performance monitoring method in a transient state.

In this study, the tolerance of the transient process response to the variations in the flow rates with the period of a few minutes, which coincides with the settling time for the process response, enables us to monitor the process performance in a transient state. Hence, an important problem to be dealt with in the future is to monitor the performance of a large capacity process, which has a large settling time compared to that of the process considered in this study. Furthermore, in a future study, we intend to demonstrate the effectiveness of this performance monitoring method for other failures.

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