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Development of the Steam-Heated Natural Gas Pressure Reducing and Heating Equipment for Combined-Cycle Thermal Power Plant

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Abstract

Steam-heated natural gas pressure reducing and heating equipment which is provided from gas pipelines to combined-cycle thermal power plants has been developed and taken to a commercial plant. It consists of a heat exchanger between natural gas and steam for gas heating without hot water as a heat transfer media and a valve for gas pressure reducing. Its control system is able to cope with quick and broad changes in gas flow rates during operation. Before the commercial plant's construction, a software simulator has been developed, which calculates the dynamic behavior of the equipment. In addition, an experimental plant has been constructed, which investigates the characteristics of the equipment and estimates the validity of the simulation model and the control system. Using the simulator, the commercial plant has been well designed before it's construction, so that it's construction is on schedule and it's performance has completely satisfied it's required control specification.

1. Introduction

In the construction of thermal power plants in recent years, the natural gas-fueled combined cycle has been promoted to improve thermal efficiency and reduce SO_x, soot and CO_2 emissions. Natural gas is mainly composed of methane. The CO_2 emissions from combustion of natural gas are about a half and two-thirds of those generated from combustion of coal and oil, respectively, to obtain the same amount of heat. Since it contains no sulfur, natural gas features another advantage of no SO_x emissons[1].

Combined-cycle power generation combines gas turbine power generation utilizing the expansion energy of combustion gas with steam power generation composed of a waste heat recovery boiler and a steam turbine. With thermal efficiency higher than that of the latest large-capacity thermal power plants of the steam power generation type, the combined cycle will become a principal type of thermal power generation in the future. Combined-cycle thermal power plants have gas turbines as their main components, and their steam turbines are relatively simple and compact. For this reason, the combined-cycle thermal power plants can be easily started and stopped in a short time. In addition, several small-power unit systems can be combined to form a large-capacity plant. Output adjustment in response to demand variations can be made by changing the number of the unit systems in operation. High thermal efficiency can be achieved even when the plant is operating under low load.

As a system for controlling the pressure and temperature of natural gas supplied to a gas turbine at a natural gas-fueled combined-cycle thermal power plant, the authors developed a control system

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capable of responding to sudden and large variations in the natural gas flow rate for steam-heated natural gas pressure reducing and heating equipment composed of a shell-and-tube heat exchanger to heat natural gas with steam (hereinafter referred to as gas heater) and of a pressure reducing valve. The equipment was commercialized for use at a combined-cycle thermal power plant.

The method of heating natural gas with steam instead of hot water as intermediate heat transfer medium (hereinafter referred to as steam-heated) was a new method without no installation records up to then. Prior to commercialization of the steam-heated natural gas pressure reducing and heating equipment, a dynamic simulator was developed and used to determine the dynamic characteristics of the equipment. The performance of the simulator was also verified by a mini-plant to enable detailed control system design and evaluation. As a result, the equipment was commercialized as scheduled and provided the required control performance as originally planned.

This development project is outlined here.

2. Outline of Development

2.1 Steam-heated natural gas pressure reducing and heating equipment, and control issues

The steam-heated natural gas pressure reducing and heating equipment is schematically illustrated in **Fig. 1**. It is located between the natural gas pipeline and the gas turbine. The pressure of the natural gas supplied from the pipeline is reduced by the pressure reducing valve to the operating pressure of the gas turbine. Since its temperature is sharply lowered during this pressure reduction, the natural gas reliquefies and causes improper combustion. Given this problem, the natural gas is preheated by the gas heater and is then reduced in pressure to prevent its reliquefaction. Control of the pressure and temperature of the natural gas supplied to the gas turbine requires a control system to operate the pressure reducing valve and the steam flow rate control valve to the gas heater. To prevent the infiltration of outside air into the gas heater that may lower the heating efficiency of the gas heater and create an explosive atmosphere in the gas heater, a control function is necessary that can keep the steam pressure in the gas heater (hereinafter referred to as gas heater steam pressure) from becoming negative.

2.2 Development steps

The pressure of the natural gas supplied from the gas pipeline to the equipment is higher than the gas turbine operating pressure (30 kg/cm^{2}G) and greatly varies (70 to 36 kg/cm^{2}G) with the gas pipeline operating conditions. Its temperature also varies greatly (+29°C to −5°C) with the season. The flow rate of the natural gas supplied to the gas turbine rapidly varies over a wide range as it follows up on the plant load and as the gas turbine is frequently started and stopped. To properly control the pressure and temperature of the natural gas supplied to the gas turbine under these conditions, it is important to develop the control system by considering the dynamic characteristics of the equipment components like the gas heater, pressure reducing valve, and pipe.

Especially in the gas heater, the shell-side fluid or steam changes in phase to condensed water through heat exchange with the tube-side fluid or natural gas. It is important to grasp how much the gas heater steam drops in pressure as a result of this phase change. It is also necessary to grasp the heat exchange characteristics of the gas heater by considering the variation of the heat transfer area between the steam and the natural gas with the condensed water level. The gas heater should be taken as a system with the characteristics of a nonlinear distributed-parameter system with respect to the variation of the natural gas flow rate[2]. For example, there is a limit to handling the gas heater with the transfer function of a linear lumped-parameter system like a first-order lag plus dead-time system.

The steam-heated natural gas pressure reducing and heating equipment was developed through the following steps:

Step 1: Development of dynamic simulator and implementation of control system design environment with dynamic simulator

A new model was developed for analyzing the dynamic characteristics of the gas heater, pressure reducing valve, and pipe comprising the equipment. This development enabled not only the dynamic characteristic analysis of the gas heater as the above-mentioned nonlinear distributed-parameter system and the design of the control system with higher accuracy, but also the behavior prediction of the equipment even when the component specifications were changed in the design stage. The dynamic characteristic analysis models of the respective components were represented as blocks on

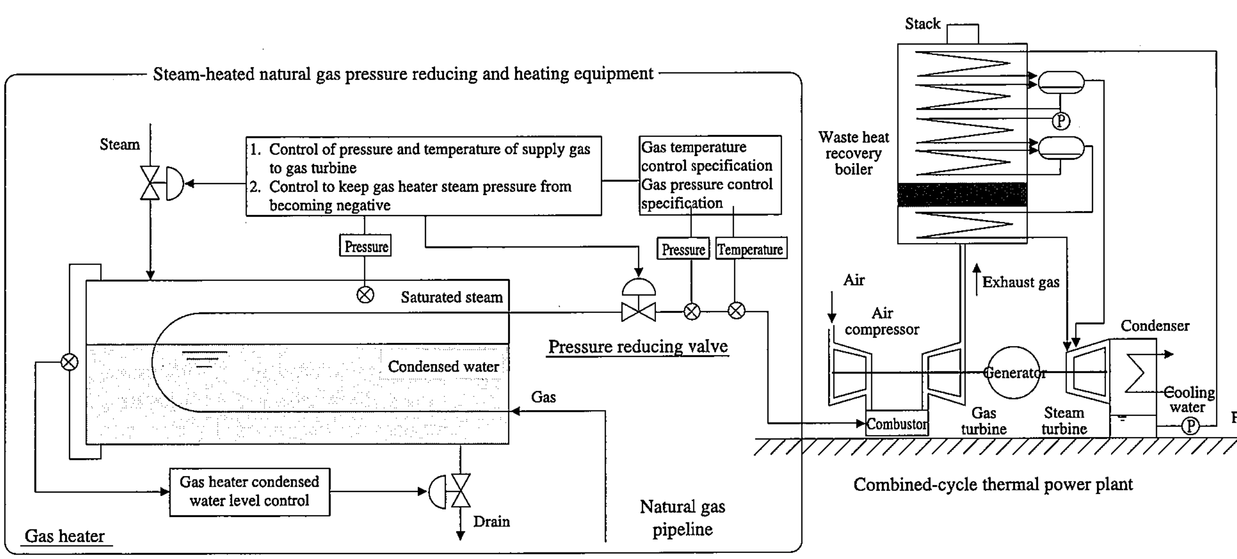

Fig. 1 Outline of steam-heated natural gas pressure reducing and

the dynamic simulator "TRAS"*1 developed by Nippon Steel Corporation (NSC) and provided with high operability and excellent expandability. This provided versatility, capability of reutilization [reutilizability], and control system design efficiency.

Step 2: Determination of characteristics of gas heater by mini-plant experimentation and validation of dynamic characteristic analysis model and control system

Various heat transfer models are available to study the heat transfer behavior of the gas heater[4]. With this equipment whose operating point greatly moves in response to large-scale load variation*2, it is necessary to clarify and grasp its characteristics experimentally. Especially, the steam-heated gas heater was a new system with no installation record then. It was thus decided to verify the reliability and validity of the equipment control system design and evaluation in the design stage before installation by experimentally validating the dynamic characteristic analysis model and the control system studied by simulator of the equipment.

Step 3: Development of control system for actual equipment by good use of dynamic simulator

Prior to installation of the actual equipment, the design and performance evaluation of the control system for the equipment were conducted by simulation with the dynamic simulator, and the specifications of the control system for the actual equipment were determined.

3. Development of Dynamic Simulator

The models for analyzing the dynamic characteristics of the gas heater, pressure-reducing valve, and pipe comprising the equipment were developed, based on physical laws, such as the mass conservation law and energy conservation law, and physical properties. Equations were formulated to take into account the dimensions of the tube bundle, heat transfer tube, and so on. This achieved versatility that can respond to changes in the fluid type and equipment specification. Two pipe models were developed for selective use. One model (pipe 1) takes external heat radiation into account, while the other model (pipe 2) takes the compressibility of the passing fluid into account. The details of the formulation procedure will be reported in another paper. Here are presented the approximate formulation procedure and results.

3.1 Nomenclature

3.1.1 Nomenclature

θ	: Temperature	(°C)
h	: Specific enthalpy	(J/kg)
u	: Internal energy	(J/kg)
p	: Pressure	(Pa)
ρ	: Density	(kg/m^3)
c	: Specific heat	(J/(kg·K))
λ	: Thermal conductivity	(W/m·K))
	Tube friction loss coefficient	(-)
w	: Heat capacity per unit length	(J/(m·K))
m	: Mass flow rate	(kg/s)
v	: Flow velocity	(m/s)
y	: Condensed water level	(m)
q	: Calorific value	(J)
S	: Flow path cross-sectional area or cross-sectional area	(m^2)
A	: Flow path cross-sectional area	(m^2)
L	: Equivalent heat transfer tube length	(m)
K	: Modified heat transfer coefficient	(W/(m^2·K))
U	: Heat transfer area per unit length	(m^2/m)
h	: Film heat transfer coefficient	(W/(m^2·K))
r_f	: Fouling factor	((m^2·K)/W)
N, F_p, x_T	: Valve parameters	(-)
C_v	: Valve flow coefficient	(-)
γ	: Ratio of specific heat	(-)
x	: Space coordinate	(m)
	Valve position	(-)
t	: Time	(s)

3.1.2 Adscripts

1	: Tube-side fluid, or inside or upstream of heat transfer tube
2	: Shell-side fluid, or outside or downstream of heat transfer tube
3	: Heat transfer tube, pipe, or inside of shell
4	: Shell or outside of shell
5	: Outside of heat insulating material
GW	: Glass wool (heat insulating material)
sur	: Outside air (side or standard)
l	: Condensed water portion (side or standard)
s	: Saturated steam portion (side or standard)
c	: Condensation
i	: Inlet
o	: Outlet
"	: Gaseous phase
'	: Liquid phase

3.2 Gas heater

The gas heater is a shell-and-tube heat exchanger as illustrated in **Fig. 2**. Consideration of the gas heater structure first focuses attention on the relationship between the condensed water level and the number of U-shaped heat transfer tubes in contact with saturated steam or condensed water as shown in **Fig. 3**. The heat transfer areas between the saturated steam and gas and between the condensed water and gas, the saturated steam volume, and the condensed water volume are calculated from the geometrical relations between the condensed water level, shell, and tube bundle arrangement.

Next, the gas heater is represented by a single U-shaped heat transfer tube whose calculated heat transfer area, saturated steam volume, and condensed water volume are equivalent to those of the gas heater. The x-axis of the space coordinate system is established in the flow direction, and the U-shaped heat transfer tube is developed in the space coordinate system as shown in **Fig. 4**. The gas heater is taken as a system where the saturated steam-gas heat exchange portion (hereinafter referred to as the saturated steam portion) and the condensed water-gas heat exchange portion (hereinafter referred to as the condensed water portion), both accompanying phase change, are connected to each other. Based on the energy conservation law and mass conservation law, a dynamic characteristic analysis model is derived for a flow-direction one-dimensional distributed-parameter system.

Assumption 1: The fluid that flows in the heat transfer tube (hereinafter referred to as the tube-side fluid) is a gas and does not change in phase.

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*1 Interactive dynamic simulator (developed by Instrument & Control R&D Div.). Dynamic systems expressed by ordinary and partial differential equations, difference equations, or algebraic equations are represented by general-purpose blocks. A dynamic simulator can be built and implemented by simply arranging these blocks on the window with the mouse and connecting them with lines[3].

*2 Large variation in gas flow rate for this equipment.

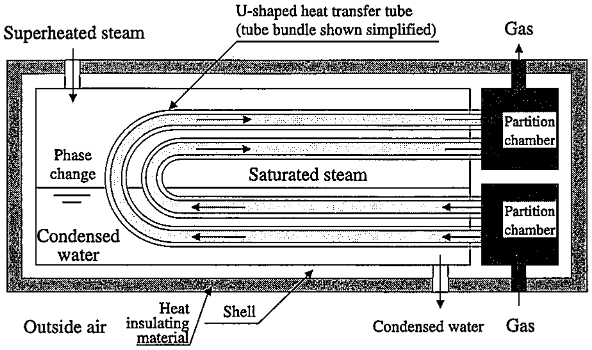

Fig. 2 Outline of gas heater

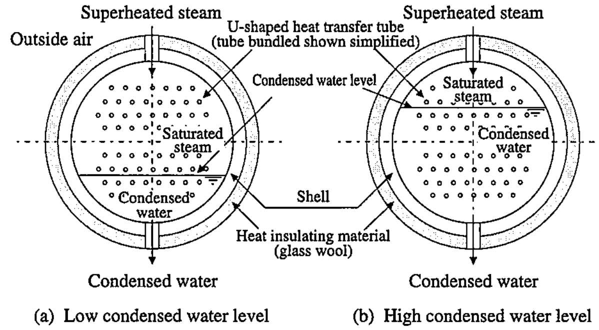

Fig.3 Condensed water level and number of U-shaped heat transfer tubes in contact with saturated steam or condensed water

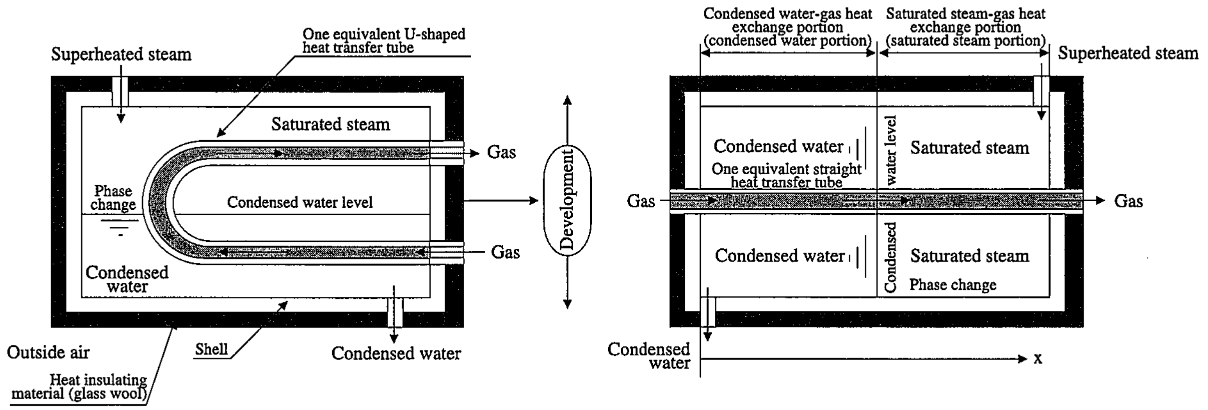

Fig.4 Coordinate system of gas heater

Assumption 2: The fluid that flows in the shell outside of the heat transfer tube (hereinafter referred to as the shell-side fluid) is steam and changes in phase to condensed water.

Assumption 3: The shell-side fluid or steam flows into the gas heater in the state of superheated steam and immediately changes in state to saturated steam.

Assumption 4: Saturated steam is handled as a lumped-parameter system. Therefore, the spatial distributions of the specific enthalpy, internal energy, temperature, pressure, and density of saturated steam are not considered. The compressibility of saturated steam is considered, however, in the sense that the changes with time in the pressure and density of saturated steam are considered.

Assumption 5: The tube-side fluid and condensed water are handled by the distributed-parameter system, but compressibility is not considered. Therefore, the spatial distribution of temperature is considered, but the spatial distributions of pressure, density and specific heat are not considered.

Assumption 6: The heat capacity of the heat transfer tube is considered and handled in the distributed-parameter system. The spatial distribution of the heat transfer tube temperature is considered one-dimensionally in the length direction, and the distribution in the thickness direction is not considered. The spatial distributions of specific heat and density are not considered.

Assumption 7: The heat capacity of the shell is considered and handled in the distributed-parameter system. The shell temperature is assumed to be spatially distributed corresponding to one dimension in the length direction, and the distribution in the thickness direction is not considered. The spatial distributions of specific heat and density are not considered.

Assumption 8: External heat radiation from the shell is considered. At this time, the heat insulating material (glass wool) is considered. Its thermal conductivity is considered, but its heat capacity is ignored.

Assumption 9: The condensed water level is considered for the flow path cross-sectional area and heat transfer area of the tube-side fluid and shell-side fluid.

Assumption 10: Calculation of the overall heat transfer coefficient considers the film heat transfer coefficient, fouling factor, and thermal conductivity of the heat transfer tube, shell and heat insulating material (glass wool). The change in the heat transfer coefficient with the flow velocity is considered. The spatial distribution of the overall heat transfer coefficient is not considered.

$$\frac{\partial \theta_{1l}}{\partial t} + v_{1l}\frac{\partial \theta_{1l}}{\partial x} = \frac{K_{1l}U_{1l}}{w_{1l}}\left(\theta_{3l} - \theta_{1l}\right) \quad (3.2.1)$$

$$\frac{\partial \theta_{2l}}{\partial t} - v_{2l}\frac{\partial \theta_{2l}}{\partial x} = \frac{\left(h'_{2s} - c_{2l}\theta_{2l}\right)m_c}{w_{2l}\cdot L} + \frac{K_{2l}U_{2l}\left(\theta_{3l} - \theta_{2l}\right) + K_{3l}U_{3l}\left(\theta_{4l} - \theta_{2l}\right)}{w_{2l}} \quad (3.2.2)$$

$$\frac{\partial \theta_{3l}}{\partial t} = \frac{K_{2l}U_{2l}(\theta_{2l}-\theta_{3l}) + K_{1l}U_{1l}(\theta_{1l}-\theta_{3l})}{w_{3l}} \tag{3.2.3}$$

$$\frac{\partial \theta_{4l}}{\partial t} = \frac{K_{3l}U_{3l}(\theta_{2l}-\theta_{4l}) + K_{4l}U_{4l}(\theta_{sur}-\theta_{4l})}{w_{4l}} \tag{3.2.4}$$

$$\frac{\partial \theta_{1s}}{\partial t} + v_{1s}\frac{\partial \theta_{1s}}{\partial x} = \frac{K_{1s}U_{1s}}{w_{1s}}(\theta_{3s}-\theta_{1s}) \tag{3.2.5}$$

$$\frac{d}{dt}(\rho''_{2s}u''_{2s}) = \frac{(h''_{2i}m_{2i} - u'_{2s}m_{2c})}{S_{2s}\cdot L} - \frac{p_{2s}\cdot m_{2o}}{\rho'_{2s}\cdot S_{2s}\cdot L} + \int_0^L \frac{K_{2s}U_{2s}(\theta_{3s}-\theta_{2s}) + K_{3s}U_{3s}(\theta_{4s}-\theta_{2s})}{S_{2s}\cdot L}dx \tag{3.2.6}$$

$$\frac{d\rho''_{2s}}{dt} = \frac{m_{2i} - m_{2c}}{S_{2s}\cdot L} \tag{3.2.7}$$

$$\frac{\partial \theta_{3s}}{\partial t} = \frac{K_{2s}U_{2s}(\theta_{2s}-\theta_{3s}) + K_{1s}U_{1s}(\theta_{1s}-\theta_{3s})}{w_{3s}} \tag{3.2.8}$$

$$\frac{\partial \theta_{4s}}{\partial t} = \frac{K_{3s}U_{3s}(\theta_{2s}-\theta_{4s}) + K_{4s}U_{4s}(\theta_{sur}-\theta_{3s})}{w_{4s}} \tag{3.2.9}$$

$$\frac{dy}{dt} = \frac{m_c - m_{2o}}{\rho_{2l}\cdot S} \tag{3.2.10}$$

$$\frac{1}{K_{1l}} = \frac{1}{h_{1l}} + r_{f1l} + \frac{1}{2\lambda_3}\cdot\frac{d_1\cdot d_2}{d_1+d_2}\cdot\ln\left(\frac{d_2}{d_1}\right) \tag{3.2.11}$$

$$\frac{1}{K_{2l}} = \frac{1}{h_{2l}} + r_{f2l} + \frac{1}{2\lambda_3}\cdot\frac{d_1\cdot d_2}{d_1+d_2}\cdot\ln\left(\frac{d_2}{d_1}\right) \tag{3.2.12}$$

$$\frac{1}{K_{3l}} = \frac{1}{h_{3l}} + r_{f3l} + \frac{1}{2\lambda_4}\cdot\frac{d_3\cdot d_4}{d_3+d_4}\cdot\ln\left(\frac{d_4}{d_3}\right) \tag{3.2.13}$$

$$\frac{1}{K_{4l}} = \frac{1}{2\lambda_4}\cdot\frac{d_3\cdot d_4}{d_3+d_4}\cdot\ln\left(\frac{d_4}{d_3}\right) + \frac{d_4}{2\lambda_{GW}}\cdot\ln\left(\frac{d_5}{d_4}\right) + \frac{1}{h_{sur}}\cdot\frac{d_4}{d_5} \tag{3.2.14}$$

$$\frac{1}{K_{1s}} = \frac{1}{h_{1s}} + r_{f1s} + \frac{1}{2\lambda_3}\cdot\frac{d_1\cdot d_2}{d_1+d_2}\cdot\ln\left(\frac{d_2}{d_1}\right) \tag{3.2.15}$$

$$\frac{1}{K_{2s}} = \frac{1}{h_{2s}} + r_{f2s} + \frac{1}{2\lambda_3}\cdot\frac{d_1\cdot d_2}{d_1+d_2}\cdot\ln\left(\frac{d_2}{d_1}\right) \tag{3.2.16}$$

$$\frac{1}{K_{3s}} = \frac{1}{h_{3s}} + r_{f3s} + \frac{1}{2\lambda_4}\cdot\frac{d_3\cdot d_4}{d_3+d_4}\cdot\ln\left(\frac{d_4}{d_3}\right) \tag{3.2.17}$$

$$\frac{1}{K_{4s}} = \frac{1}{2\lambda_4}\cdot\frac{d_3\cdot d_4}{d_3+d_4}\cdot\ln\left(\frac{d_4}{d_3}\right) + \frac{d_4}{2\lambda_{GW}}\cdot\ln\left(\frac{d_5}{d_4}\right) + \frac{1}{h_{sur}}\cdot\frac{d_4}{d_5} \tag{3.2.18}$$

$$w_{1l} = \rho_1\cdot C_{1l}\cdot S_{1l}(y) \tag{3.2.19}$$

$$w_{2l} = \rho_2\cdot C_{2l}\cdot S_{2l}(y) \tag{3.2.20}$$

$$w_{3l} = \rho_3\cdot C_3\cdot S_{3l}(y) \tag{3.2.21}$$

$$w_{4l} = \rho_4\cdot C_4\cdot S_{4l}(y) \tag{3.2.22}$$

$$w_{1s} = \rho_1\cdot C_{1s}\cdot S_{1s}(y) \tag{3.2.23}$$

$$w_{3s} = \rho_3\cdot C_3\cdot S_{3s}(y) \tag{3.2.24}$$

$$w_{4s} = \rho_4\cdot C_4\cdot S_{4s}(y) \tag{3.2.25}$$

$$\theta_{1l}(0,t) = \theta_{1i}(t) \tag{3.2.26}$$

$$\theta_{1s}(0,t) = \theta_{1l}(L,t) \tag{3.2.27}$$

$$\theta_{1o}(t) = \theta_{1s}(L,t) \tag{3.2.28}$$

$$\theta_{2l}(L,t) = \theta_{2s}(t) \tag{3.2.29}$$

$$\theta_{2o}(t) = \theta_{2l}(0,t) \tag{3.2.30}$$

Physical property data, such as the specific enthalpy of superheated steam and the specific enthalpy, internal energy and density of saturated steam, are presented in a physical property data table[5]. The CIP method*3, a technique capable of performing computation with high accuracy using a few space meshes, is adopted for numerical calculation of nonlinear hyperbolic partial differential equations like Eqs. (3.2.1), (3.2.2) and (3.2.5).

3.3 Pressure reducing valve[7]

Assumption 1: The energy conservation law that takes kinetic energy into account is a basic equation.
Assumption 2: External heat radiation is not considered.
Assumption 3: The passing fluid is assumed to be a real gas.
Assumption 4: The passing mass flow rate is given from the pressure difference between the upstream and downstream sides of the valve, valve position, and valve flow coefficient Cv.
Assumption 5: The model obtains the downstream temperature when the upstream and downstream pressures and the upstream temperature are given.

$$h_1 + \frac{v_1^2}{2} = h_2 + \frac{v_2^2}{2} \tag{3.3.1}$$

$$h_1 = h(\theta_1, p_1) \tag{3.3.2}$$

$$h_2 = h(\theta_2, p_2) \tag{3.3.3}$$

$$v_1 = \frac{m}{\rho_1\cdot A_1} \tag{3.3.4}$$

$$v_2 = \frac{m}{\rho_2\cdot A_2} \tag{3.3.5}$$

$$\rho_1 = \rho(\theta_1, p_1) \tag{3.3.6}$$

$$\rho_2 = \rho(\theta_1, p_2) \tag{3.3.7}$$

$$m = N\cdot F_p\cdot C_{V(x)}\cdot Y\cdot\sqrt{(p_1-p_2)\cdot\rho_1}, \quad Y = 1 - \frac{p_1-p_2}{3F_k\cdot x_T\cdot p_1}, \quad F_k = \frac{\gamma}{1.40} \tag{3.3.8}$$

3.4 Pipe 1 - Consideration of heat radiation outside of pipe -

Assumption 1: The model is an unsteady-state distributed-parameter model one-dimensional in the flow direction.
Assumption 2: The passing fluid is assumed to be a not-compressible fluid.
Assumption 3: External heat radiation is considered. At this time, the heat insulating material (glass wool) is considered. The thermal conductivity of the heat insulat-

*3 Cubic Interpolated Pseudo particle method[6] ing material is considered, but its heat capacity is ignored.

Assumption 4: The cross-sectional area of the pipe is assumed to be constant, irrespective of the position.

$$\frac{\partial \theta_1}{\partial t} + v_1 \frac{\partial \theta_1}{\partial x} = \frac{K_1 U_1}{w_1}\left(\theta_3 - \theta_1\right) \tag{3.4.1}$$

$$\frac{\partial \theta_3}{\partial t} = \frac{K_2 U_2 \left(\theta_{sur} - \theta_3\right) + K_1 U_1 \left(\theta_1 - \theta_3\right)}{w_3} \tag{3.4.2}$$

$$\frac{1}{K_1} = \frac{1}{h_1} + r_{f1} + \frac{1}{2\lambda_3} \cdot \frac{d_1 \cdot d_2}{d_1 + d_2} \cdot \ln\left(\frac{d_2}{d_1}\right) \tag{3.4.3}$$

$$\frac{1}{K_2} = \frac{1}{2\lambda_3} \cdot \frac{d_1 \cdot d_2}{d_1 + d_2} \cdot \ln\left(\frac{d_2}{d_1}\right) + \frac{d_2}{2\lambda_{GW}} \cdot \ln\left(\frac{d_3}{d_2}\right) + \frac{1}{h_{sur}} \cdot \frac{d_2}{d_3} \tag{3.4.4}$$

$$w_1 = \rho_1 \cdot C_1 \cdot S_1 \tag{3.4.5}$$

$$w_3 = \rho_3 \cdot C_3 \cdot S_3 \tag{3.4.6}$$

3.5 Pipe 2 - Consideration of compressibility of passing fluid -

Assumption 1: The model is an unsteady-state distributed-parameter model one-dimensional in the flow direction.

Assumption 2: The passing fluid is assumed to be a compressible fluid.

Assumption 3: External heat radiation is ignored.

Assumption 4: The pipe friction loss is considered.

Assumption 5: The cross-sectional area of the pipe is assumed to be constant, irrespective of the position.

$$\frac{\partial \rho_1}{\partial t} + \frac{\partial}{\partial x}\left(\rho_1 \cdot v_1\right) = 0 \tag{3.5.1}$$

$$\frac{\partial}{\partial t}\left(\rho_1 \cdot v_1\right) + \frac{\partial}{\partial x}\left(\rho_1 \cdot v_1^2\right) + \frac{\partial p_1}{\partial x} + \frac{\lambda_1}{2d_1}\rho_1 \cdot v_1 \cdot |v_1| = 0 \tag{3.5.2}$$

$$\frac{\partial}{\partial t}\left\{\rho_1 \cdot \left(u_1 + \frac{v_1^2}{2}\right)\right\} + \frac{\partial}{\partial x} \cdot \left\{\rho_1 \cdot \left(u_1 + \frac{v_1^2}{2}\right) \cdot v_1\right\} + \frac{\partial}{\partial x}\left(p_1 \cdot v_1\right) = p_1 \cdot \dot{q}_1 \tag{3.5.3}$$

In addition to the models described above, confluence point, branch point, and other models were developed. These models were constructed as blocks on TRAS.

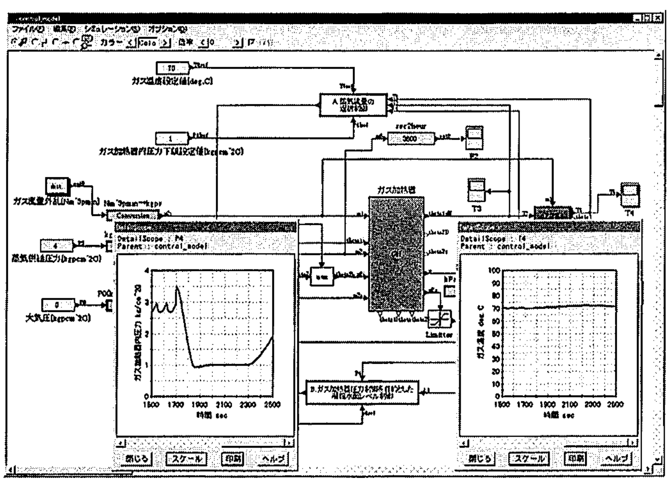

Fig. 5. Dynamic simulator "TRAS" for equipment

3.6 Dynamic simulator of equipment

The dynamic simulator of the equipment was built by setting equipment specifications, such as the fluid physical property data table, tube bundle arrangement, valve flow coefficient *Cv*, and pipe length, in the TRAS blocks of the respective components and connecting the TRAS blocks with lines as shown in **Fig. 5**. Simulation with the dynamic simulator performed the control system design and performance evaluation of the equipment, and determined the equipment control system specifications prior to the installation of the actual equipment.

4. Determination of Characteristics of Gas Heater and Validation of Dynamic Characteristic Analysis Model by Experimentation with Mini-plant

The mini-plant installed in the development project is shown in **Fig. 6**. The gas heater was a shell-and-tube heat exchanger of the same type as that of the actual gas heater, and its scale was about two-thirds of that of the actual gas heater. For experimental safety, high-pressure air was used in place of natural gas as the tube-side fluid. The experimental conditions like the flow velocity in the tube were set to make the Reynolds number and other dimensionless parameters similar to those of the actual gas heater.

The mini-plant experimentation confirmed that the film heat transfer coefficient in the heat transfer tube, a basic characteristic of the gas heater, agreed well with that calculated by the equation of Sieder and Tate, one of the convective heat transfer coefficient models, and that the gas flow velocity in the heat transfer tube was a predominant factor. The condensing heat transfer coefficient was identified from the heat balance of the gas heater on the basis of the experimental data, and was modeled. It was also found that the condensed water level was strongly correlated with the gas heater steam pressure and was an effective manipulated variable to keep the gas heater steam pressure from becoming negative.

A gas flow rate step response experiment was conducted on the gas heater alone and verified the validity of the dynamic characteristic analysis model for the gas heater as shown in **Fig. 7**. Fig. 7 shows the responses of the gas heater outlet temperature, condensed water discharge temperature, gas heater steam temperature, and gas heater steam pressure when the gas flow rate was rapidly reduced in a single step from 20 Nm^3/min (100% flow rate) to 14 Nm^3/min (70% flow rate) over the time of 700 s and was then rapidly returned back in a single step to 20 Nm^3/min over the time of 3900 s.

In Fig. 7, the measured values agree well with the calculated val-

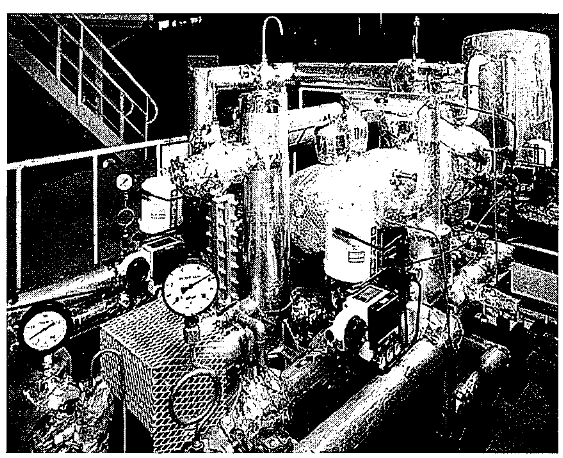

Fig. 6 Mini-plant

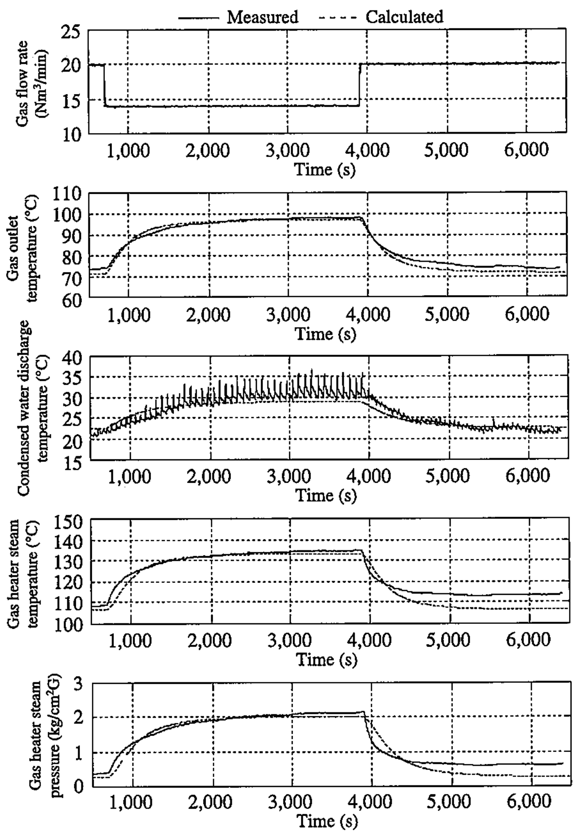

Fig. 7 Experiment to verify validity of dynamic characteristic analysis model for gas heater

ues. The validity of the dynamic characteristic analysis model for the gas heater was thus verified. The saw-toothed waves observed on the condensed water discharge temperature curve resulted from the opening and closing of the solenoid valve used as the condensed water discharge value and the operation of the hysteresis relay used in the condensed water level control system. When the solenoid valve opened, the condensed water in the gas heater was momentarily pressurized and discharged. The high-temperature condensed water near the tube bundle apart from the discharge opening passed through the discharge tube, and its temperature was measured.

5. Development of Control System

The control system to solve the control issues described in Chapter 2 was studied by using the findings about the basic characteristics of the gas heater confirmed in the mini-plant and the dynamic simulator used to verify the validity of the dynamic characteristic analysis model. The control system of the equipment was studied to see if it could properly control the gas turbine supply gas temperature and keep the gas heater steam pressure from becoming negative when the gas flow rate was suddenly reduced from 100% to about 25%, equivalent to the gas flow rate for the operation of a gas turbine at the natural gas-fueled combined-cycle thermal power plant, as shown in **Fig. 8** as the severest gas flow condition for the equipment. The gas turbine supply gas pressure was controlled by the pressure reducing valve control system.

5.1 Example of limit of control system of simple configuration

Fig. 9 shows an example of limit of a control system of such a simple configuration as to feedback control the gas turbine supply gas temperature by the flow rate of superheated steam to the gas heater and to keep constant the condensed water level in the gas heater by operating the condensed water discharge valve of the gas heater. The control system of the simple configuration shown in Fig. 9 was built on the dynamic simulator under the mini-plant conditions and was simulated. The results are shown in **Fig. 10**. Fig. 10 shows the responses of the gas temperature, steam flow rate, gas heater steam pressure, and gas heater condensed water level when the gas flow rate was suddenly reduced in a single step from 20 Nm³/min (100% flow rate) to 5 Nm³/min (25% flow rate) over 200 s. The set points of the gas temperature and gas heater condensed water level were at 70°C and 60 mm, respectively.

From Fig. 10, the following can be confirmed. Since the gas temperature sensor is apart from the gas heater in the control system of simple configuration, the controllability of the gas temperature is

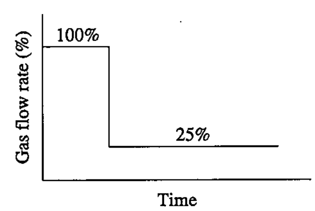

Fig. 8 Severest gas flow variation presumed for equipment

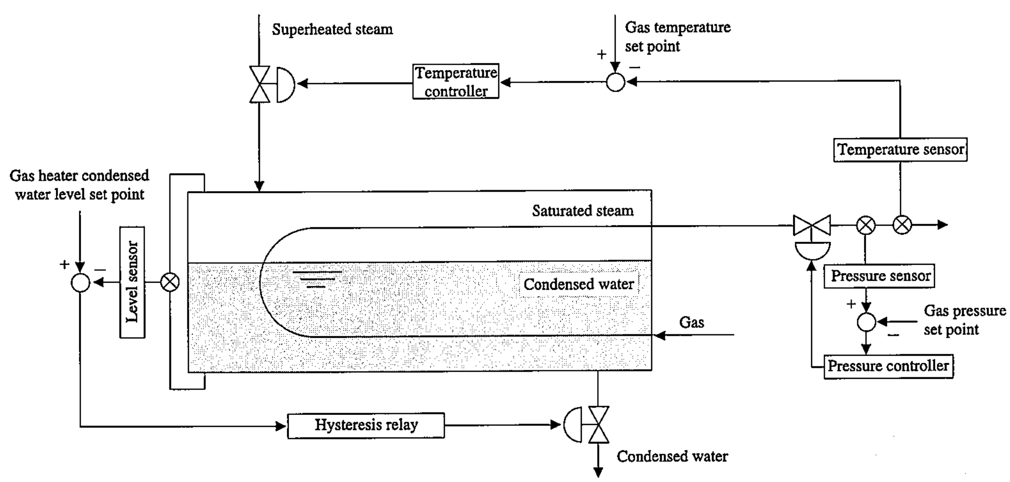

Fig. 9 Control system of simple configuration

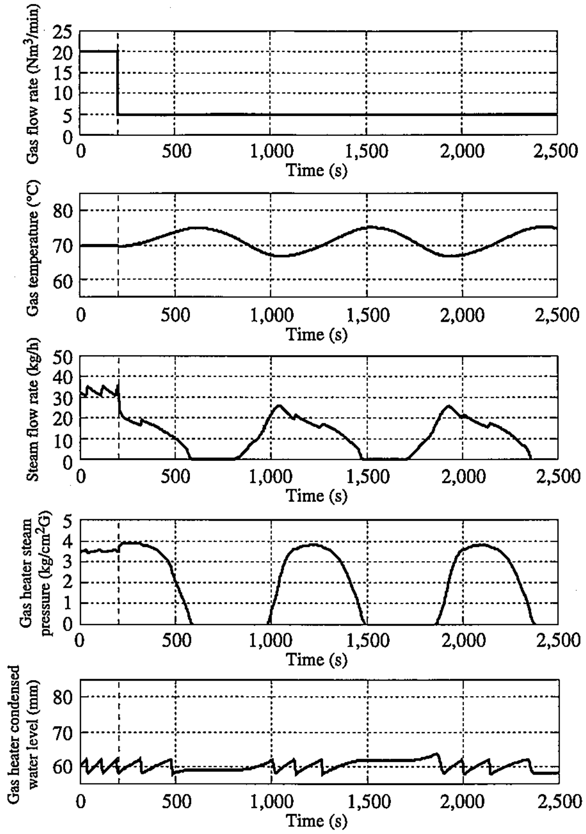

Fig. 10 Examples of limits of control system of simple configuration (as simulated under mini-plant conditions)

poor. When the controller gain is consequently set high, the steam flow rate control valve is suddenly closed as the gas temperature rises. The gas heater steam pressure becomes negative, making it impossible to solve the equipment control issues noted above. The saw-toothed waves observed on the steam flow rate and condensed water level curves in Fig. 10 mean that the dynamic simulator can reproduce the phenomena arising from the opening and closing of the solenoid valve used as condensed water discharge valve in the mini-plant and from the operation of the hysteresis relay used in the condensed water level control system.

5.2 Control system developed

To improve the controllability of the gas temperature and keep the gas heater steam pressure from becoming negative, a control system that combines steam flow rate selective control (override control) with condensed water level control for the purpose of gas heater steam pressure control was developed by using the dynamic simulator. The control system developed is schematically illustrated in **Fig. 11.**

1) Steam flow rate selective control (override control)[8]

A control scheme that automatically selects from among multiple signals the most important or appropriate variable in terms of equipment performance or safety and controls the variable is termed selective control. At this time, one signal may be selected from among multiple measurement signals and may be used as controller input, or the selection may be made on the input side. Or controllers may be provided for multiple controlled variables, and the output of one controller selected may drive the actuator (e.g., a valve), or the selection may be made on the output side.

The equipment adopted the following selective control method. As an output-side selective control, the gas temperature controller, which is the original control objective, is normally selected. Under an abnormal condition or when the gas heater steam pressure is suddenly reduced, the original gas temperature control is temporarily interrupted, and the gas heater steam pressure controller is selected to keep the gas heater steam pressure from becoming negative. This type of selective control is sometimes called override control. The detail of selective logic is omitted here. The output of the gas temperature sensor located immediately after the outlet of the gas heater and rapidly responding to the gas flow rate variation is input to the gas temperature controller to improve the controllability of the gas temperature.

2) Condensed water level control for purpose of gas heater steam pressure control

According to the results of characteristic experiment conducted in the mini-plant, it was found that the condensed water level of the gas heater is strongly correlated with the gas heater steam pressure and is an effective manipulated variable for the gas heater steam pressure. It was also confirmed that this characteristic can be reproduced by the dynamic simulator. As control system to solve the control issues of the equipment, a condensed water level control sys-

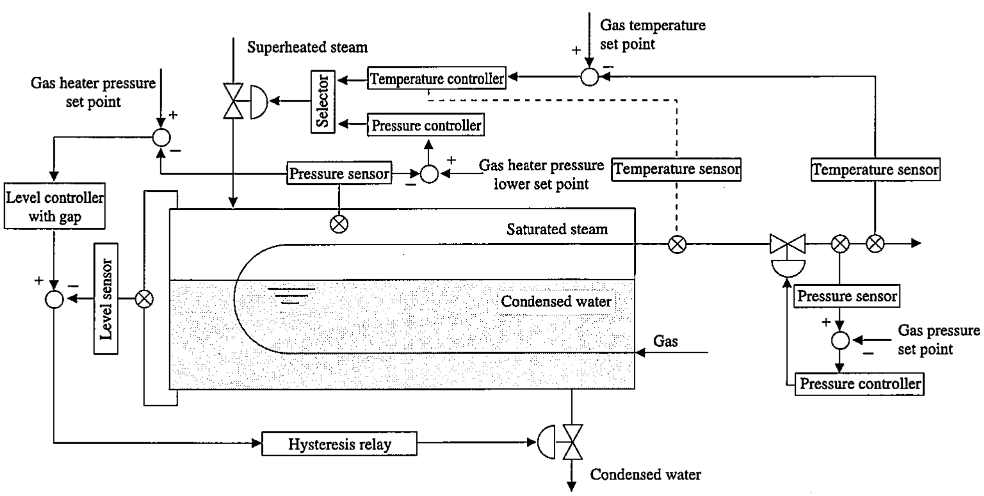

Fig. 11 Control system developed

tem was constructed for the purpose of gas heater steam pressure control.

The control system composed of steam flow rate selective control and condensed water level control for the purpose of gas heater steam pressure control solves the equipment control issues, with the two control components operating in response to a sudden gas flow rate change while complementing each other as described below.

Fig. 12 shows the responses of the gas temperature, steam flow rate, gas heater steam pressure, and gas heater condensed water level to a sudden drop in a single step of the gas flow rate from 20 Nm^3/min (100% flow rate) to 5 Nm^3/min (25% flow rate) over the time of 200 s when the control system was built on the dynamic simulator by referring to the mini-plant conditions. The set point of the gas temperature and the gas heater steam pressure upper and lower limits were at 70°C and at 3.0 and 1.0 kg/cm^2G, respectively.

Basically, the gas temperature and gas heater steam pressure are controlled according to the steam flow rate and condensed water level, respectively. Since the condensed water level is an integral system (liquid level system) with respect to the condensed water discharge rate, which is a manipulated variable, the gas heater pressure control is low in speed of response. The selective control of the steam flow rate is executed to make up for this shortcoming. When the gas heater steam pressure drops below the lower limit setting until the condensed water level rises (400 to 800 s as shown in Fig. 12), the pressure control is selected to temporarily let in the steam and to keep the gas heater steam pressure from becoming negative. When the gas heater is maintained at the lower pressure limit setting, the gas temperature control is selected to control the gas temperature at the set point. This selection logic is appropriately set to keep the gas heater steam pressure from becoming negative while maintaining the gas temperature control performance. As the condensed water level eventually rises to allow the gas heater steam pressure to be maintained at the lower limit without depending on the pressure control of the steam flow rate (as shown after 800 s in Fig. 12), the selective control of the steam flow rate positively selects the gas temperature control.

5.3 Verification of validity of control system by mini-plant

In the mini-plant where the control system shown in Fig. 11 was constructed, a control experiment was conducted to verify the validity of the control system.

Fig. 13 shows the responses of the gas temperature, steam flow rate, gas heater steam pressure, and gas heater condensed water level to a rapid step drop in the gas flow rate from 20 Nm^3/min (100% flow rate) to 5 Nm^3/min (25% flow rate) in 200 s at the mini-plant. The control parameters are the same as the simulation conditions of Fig. 12. Fig. 13 shows that the controllability of the gas temperature is improved as compared with the example of limit of the control system of simple configuration shown in Fig. 10 and that the phenomenon of the gas heater pressure going negative is constantly avoided. The validity of the developed control system was thus verified.

Figs. 12 and 13 agree well with each other. This meant that the dynamic simulator developed could reproduce the dynamic behavior of the equipment and the operation of the control system with high accuracy, thereby verifying the validity of the dynamic simulator toward the goal of designing and evaluating the control system for the actual equipment.

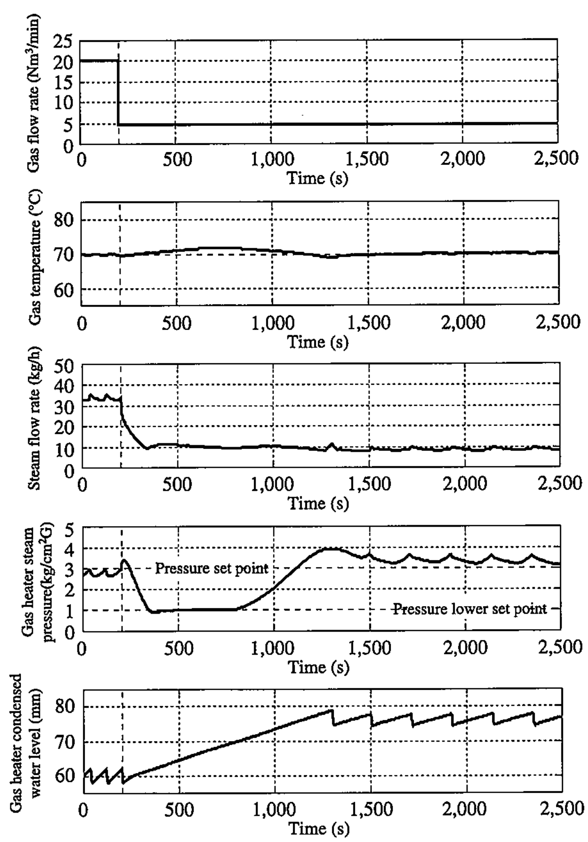

Fig. 12 Operation of control system developed (as simulated under mini-plant conditions)

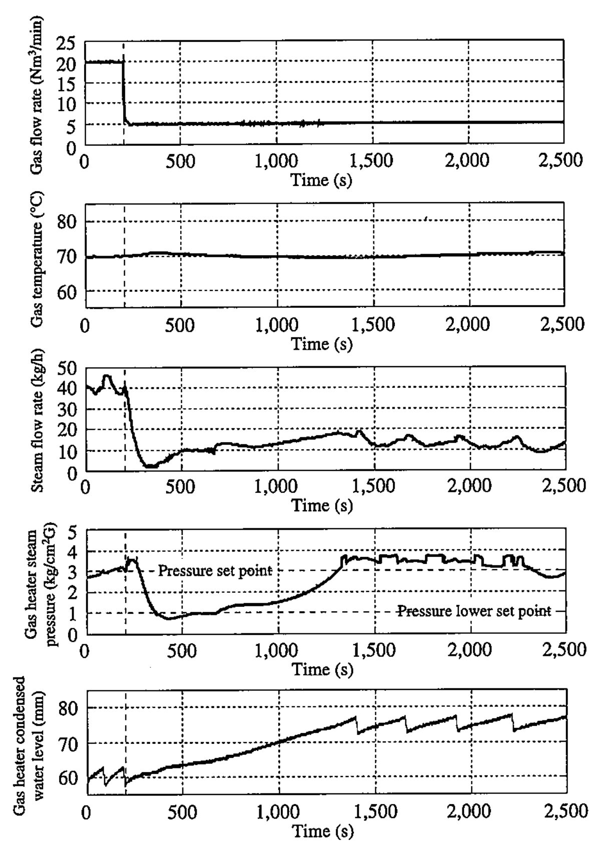

Fig. 13 Results of mini-plant experiment conducted to verify validity of control system developed

6. Commercialization and Control Results of Control System

The control system developed by using the dynamic simulator and whose validity was verified by the mini-plant experimentation was applied to the actual equipment. The dynamic simulator was put to good use in designing and evaluating the control system in the stage before the installation of the actual equipment and in rapidly adjusting the control system during site test operation. As typical examples, the control results of the control system with respect to a sudden drop in the gas flow rate are shown in **Figs. 14 and 15**.

Figs. 14 and 15 show the responses of the gas temperature and gas heater steam pressure when the gas flow rate was suddenly reduced from 80% to 20% in a single step and in 1 min, and was then gradually returned back to 80%. The calculated and measured results are given in Figs. 14 and 15, respectively.

From Fig. 15, it is evident that the control system developed effectively functioned against the sudden drop in the gas flow rate, avoided the phenomenon of the gas heater steam pressure becoming negative, and solved the control issues of the equipment. In view of its capability to successfully respond to the severest gas flow rate variation presumed for the equipment, the control system can be said to be also effective against the ordinary gas flow variation as accompanied by plant load follow-up, for example.

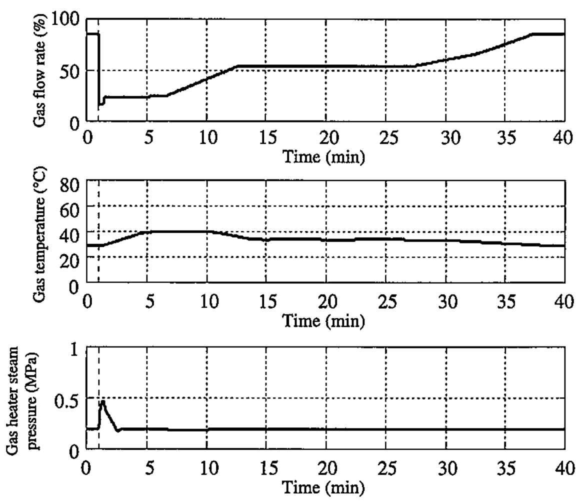

Fig. 14. Calculated results

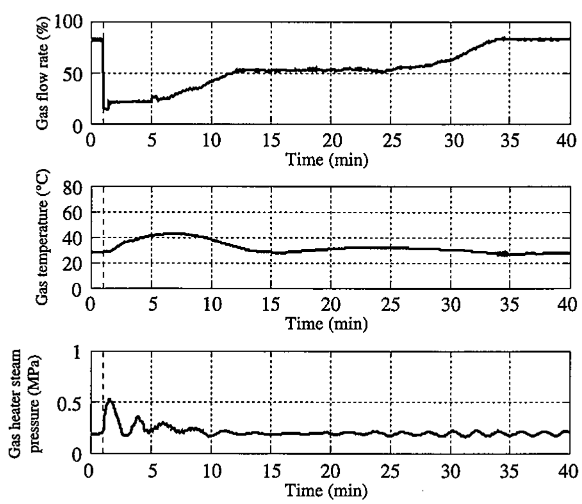

Fig. 15 Measured results

The good agreement observed between Figs. 14 and 15 also verifies the validity of the dynamic simulator. In the measured results, however, the oscillatory response of the gas heater steam pressure is measured. This is considered to have resulted from the method of estimating the saturated steam volume, including the pipe in the dynamic characteristic analysis model of the gas heater, and the assumptions made about the saturated steam as compressible fluid. These issues will have to be addressed to enhance the accuracy of the dynamic characteristic analysis model.

7. Conclusions

Steam-heated natural gas pressure reducing and heating equipment was developed as equipment for controlling the pressure and temperature of natural gas supplied by pipeline to a gas turbine at a natural gas-fueled combined-cycle thermal power plant. When developing a control system for the equipment, a dynamic simulator with high versatility and accuracy was developed. The validity of the simulator was verified by experimentation in [with] a mini-plant to enable the detailed design and evaluation of the control system before installation of the actual equipment and to allow the rapid adjustment of the control system during site test operation. As a result, the test operation and adjustment were completed as scheduled, and the control system performed as required by the actual equipment. The steam-heated natural gas pressure reducing and heating equipment came into commercial operation at the New Nagoya Thermal Power Plant of Chubu Electric Power Co., Inc., in August 1998.

8. Acknowledgments

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