Dynamic Simulation and Optimization of Start-up Processes in Combined Cycle Power Plants*

Masakazu SHIRAKAWA**, Masashi NAKAMOTO** and Shunji HOSAKA**

This paper treats the development of a dynamic simulation model and its application to the optimization of a start-up process for a combined cycle power plant. Generally, the plant system design is complicated, and the control design is difficult to establish without dynamic simulation. The comparison of the simulation results and the plant data is reported. The obtained results demonstrate that the simulation is reliable to evaluate the plant dynamic phenomenon and available to predict the operational processes. However, not only an analytical method but also the development of a design method is important to determine the optimal operational procedure. In this paper, the dynamic simulation and the nonlinear programming are combined, and it proposes the design method that optimizes the operational parameters.

Key Words: Power Plant, Combined Cycle, Dynamic Simulation, Optimization, Optimal Operation

1. Introduction

The demand for flexible operation of a combined cycle power plant is becoming increasingly important because this plant is particularly suited for load control. Its operational characteristics have been intensively studied by the dynamic simulation(1), but it is not easy to determine the optimal operational parameters because it is necessary to iterate the dynamic simulation based on trial and error by the engineer’s intuition and experience.

Several proposals to improve start-up scheduling for a power plant have been made but each has its limitations. For instance, an adaptive-edge search operation used with Genetic Algorithms (GA) has been proposed(2). This method considers only thermal stress in the steam turbine as an operational restriction. The combined cycle power plant, however, has many operational restrictions, not only thermal stress in the steam turbine but also loading rates of the gas turbine, temperature gradients of the heat recovery steam generator and NOx emission from the plant, etc. A fuzzy expert system that uses an engineer’s experiences in fuzzy rules has been proposed(3). It requires additional cost to obtain this engineering knowledge.

This paper aims at proposing a method to determine the practical optimal operational parameters using a dynamic simulation and nonlinear programming. This method does not require a great deal of labor in order to prepare the knowledge base.

2. Nomenclature

\[ A : \text{heat transfer area [m}^2\text{]} \]
\[ c_p : \text{specific heat [kJ/(kg·°C)]} \]
\[ E : \text{Young’s modulus [N/mm}^2\text{]} \]
\[ G : \text{mass flow rate [kg/s]} \]
\[ h : \text{enthalpy [kJ/kg]} \]
\[ k : \text{heat transfer coefficient [kW/(m}^2\text{·°C)}]} \]
\[ L : \text{evaporating point [–]} \]
\[ M : \text{mass [kg]} \]
\[ P : \text{pressure [kPa]} \]
\[ Q : \text{heat flow [kW]} \]
\[ R_o : \text{outer radius [m]} \]
\[ R_i : \text{inner radius [m]} \]
\[ r : \text{radial axis [m]} \]
\[ T : \text{temperature [°C]} \]
\[ t : \text{time [s]} \]
\[ v : \text{specific volume [m}^3\text{/kg]} \]
\[ W : \text{power [kW]} \]
\[ x : \text{quality [–]} \]
\[ \alpha : \text{coefficient of linear expansion [1/°C]} \]
\[ \zeta : \text{flow coefficient [m}^2\text{]} \]
\[ \zeta(z) : \text{flow coefficient depending on valve position z [m}^2\text{]} \]
\[ \eta : \text{adiabatic efficiency of steam turbine [–]} \]

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E-mail: masakazu1.shirakawa@toshiba.co.jp
θ : rotor temperature [°C]  
λ : thermal conductivity [kW/(m·°C)]  
ν : Poisson’s ratio [–]  
ρ : density [kg/m³]  
σzz : thermal stress [N/mm²]  

Subscripts  
bo : drum blow  
c : circulation  
cd : condensation in drum  
dm : drum metal  
dv : evaporation in drum  
dw : drum holding water  
e : evaporator exit  
ev : evaporation in evaporator  
esm : evaporator saturated section metal  
eum : evaporator subcooled section metal  
f : interface  
g : gas side  
i : inlet  
m : heat exchanger metal  
o : outlet  
tb : turbine  
tbor : ideal turbine exit at 100% efficiency  
s : saturated steam  
stm : steam side  
sw : saturated water  
w : feed water  
ws : subcooled water

3. Combined Cycle Power Plant

The configuration of the combined cycle power plant is shown in Fig. 1. This plant consists of a gas turbine, a heat recovery steam generator (HRSG), a steam turbine and a generator. The gas turbine and the steam turbine drive the generator. Also, the HRSG generates steam for the steam turbine using waste heat from the gas turbine. The HRSG is the triple pressure and reheat type. The steam turbine has a high-pressure (HP) turbine, an intermediate-pressure (IP) turbine and a low-pressure (LP) turbine.

4. Simulation Model Descriptions

The dynamic simulation model consists of the process equipment model and the control system model. The control system is modeled to be equivalent to the actual control system. The first principle is adapted to model the process equipment. The following are brief descriptions of major models.

4.1 Gas turbine model

The gas turbine responds more quickly than the other equipment. The flow rate and the temperature of the exhaust gas are represented by functions of the fuel flow rate. The gas turbine duct has a time delay because of the large heat capacity. Therefore, the gas turbine model considers the heat capacity of the duct. The rotor dynamics can be disregarded in the start-up, though it is considered in a rapid response as the load rejection.

4.2 Drum and evaporator model

The HRSG is composed of heat exchangers that have a long time delay when compared to the gas turbine and the steam turbine. It has a significant effect on the starting characteristics of the entire plant. The model of the drum and evaporator is shown in Fig. 2, and the basic equations are described in Eqs. (1) to (12). The metal of the drum and evaporator has a time delay because of the large heat capacity. Therefore, the drum and evaporator model considers the heat capacity of the metal in Eqs. (3), (8) and (9).

Drum mass balance (liquid phase):

$$d(M_{dw})/dt = G_w - G_{bo} - \sum G_e$$

Drum mass balance (steam phase):

$$d(M_{s})/dt = \sum xG_e - G_{s} = G_{cd} - G_{dv}$$

Drum energy balance (liquid phase):

$$d(M_{dw}h_{dw} + M_{dm}c_{pm}T_{dm})/dt = G_{s}h_{w}$$

$$-G_{bo}h_{dw} - \sum G_{e}h_{dw} + \sum (1-x)G_{e}h_{sw}$$

$$+ G_{cd}h_{dw} = G_{s}h_{s} + M_{dw}v_{dw}(dP/dt)$$

Drum energy balance (steam phase):

$$d(M_{s}h_{s})/dt = \sum xG_{e}h_{s} - G_{s}h_{s}$$

$$-G_{cd}h_{dw} + G_{s}h_{s} + M_{dv}(dP/dt)$$

Evaporator mass balance (subcooled water phase):
Evaporator energy balance (saturated steam phase):

Total volume constant (drum):

referred to the past study (1), the heat transfer rate and the flow rate is important. Though the dynamic modeling was precise estimation of the void fraction and the circulating

Evaporator energy balance (saturated water phase):

Total volume constant (evaporator):

Experimental data of the test plant.

4.3 Heat exchanger model

The basic equations of the heat exchanger are described in Eqs. (16) to (18). The adiabatic efficiency of the steam turbine is defined as the turbine pressure ratio. The steam flow rate through the steam turbine is calculated using the constant flow coefficient. The turbine power is calculated using the enthalpy difference between the inlet and outlet of the steam turbine. No dynamic effect is evaluated in the flow and pressure relationship.

$$h_{tb} = h_{tb} - r(h_{tb} - h_{b0})$$  \hspace{1cm} (16)

$$G_{tb} = \zeta \sqrt{(P_{tb} - P_{b0})/\eta_{tb}}$$  \hspace{1cm} (17)

$$W_{tb} = G_{tb}(h_{tb} - h_{b0})$$  \hspace{1cm} (18)

The basic equations of the thermal stress are described in Eqs. (19) and (20). The thermal stress of the steam turbine rotor is obtained by the temperature distribution, which is calculated by the dynamic model of the thermal conduction that divides the rotor into several vertical cylinders.

$$\frac{\partial \theta}{\partial t} = \frac{1}{c_{p} \rho} \left( \frac{\partial^{2} \theta}{\partial r^{2}} + \frac{1}{r} \frac{\partial \theta}{\partial r} \right)$$  \hspace{1cm} (19)

$$\sigma_{zz} = \frac{E \alpha}{1 - \nu} \left( \frac{2}{R_{0}^{2} - R_{1}^{2}} \right) \int_{R_{1}}^{R_{0}} \theta_{r} dr - \theta$$  \hspace{1cm} (20)

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5. Verification of the Simulation Model

The actual plant data were measured from one-shaft, triple pressure with reheat cycle 238 MW rated capacity. The firing temperature of the gas turbine was 1 300°C class. The comparison with hot start-up (standby period is 8 hours) data is shown in Fig. 3. The comparison with cold start-up data is shown in Fig. 4. Figure (a) shows the gas turbine exhaust gas temperature and flow rate response in Figs. 3 and 4. Figures (b), (c) and (d) show the HP, IP, LP steam temperature, pressure and flow rate response in Figs. 3 and 4. The symbol lines are simulation result and the solid lines are plant data in Figs. 3 and 4. The simulation results are in good correspondence with the plant data. Accordingly, the simulation model will be available to use in the plant system and control design.

The comparison results of the start-up time from the gas turbine start to the base load operation are shown in Table 1. The differences of the start-up time are only 2 minutes (3.4%) for hot start-up and 11 minutes (5.8%) for cold start-up. The simulation results are a little faster than
the plant data, because the simulation model neglected the sequence control of some drain and vent valves to reduce the computation time.

6. Optimization of the Plant Operation

6.1 Optimization method

Nonlinear optimization problem is generally written...
as follows:

Minimize \( f(X) \) \hspace{1cm} (24)

Subject to \[
\begin{align*}
& c_i(X) = 0 \quad i = 1, 2, \ldots, m_e \\
& c_i(X) \geq 0 \quad i = m_e + 1, \ldots, m
\end{align*}
\] \hspace{1cm} (25)

where objective function \( f(X) \) and constraints \( c_i(X) \) depend on various values of parameter \( X \), \( m_e \) is number of equality constraints, and \( m \) is number of equality and inequality constraints.

The vector parameter \( X \) is represented by transpose of a matrix of all the parameters \( x_1, x_2, \ldots, x_n \) such as

\[ X = [x_1, x_2, \ldots, x_n]^T \] \hspace{1cm} (26)

where \( n \) is number of parameters.

The operational procedures of the combined cycle power plant can be formulated on the optimization problem which searches minimum starting time and/or fuel consumption during the start-up / shutdown under the various operational constraints(4).

**Parameters:** The operational parameters are control input during the plant operation (e.g., loading rate of the gas turbine).

**Objective functions:** The operational performance of the power plant can be evaluated in two ways as described here:

1. Using a function representing the plant state (e.g., starting time of the plant).
   \[ g_{t_i}(X), \quad k = 1, 2, \ldots, p_e \] \hspace{1cm} (27)

2. Using a function expressing the transition from one state to another (e.g., fuel consumption during the start-up / shutdown).
   \[ g_{x}(X) = \int_{t_1}^{t_2} g_{x}(X,t) \, dt, \quad k = p_e + 1, \ldots, p \] \hspace{1cm} (28)

where \( p_e \) is number of functions \( g_{t_i} \) in Eq. (27), and \( p \) is number of all functions \( g_x \) in Eqs. (27) and (28). For example, \( t_1 \) is time of the ignition, and \( t_2 \) is time of reached at the base load.

The objective function can be represented by a function such as

\[ f(X) = f(g_{t_1}, g_{t_2}, \ldots, g_p) \] \hspace{1cm} (29)

Functions \( g_{t_i} \) (and the process values \( g_{x_j} \)) in Eqs. (27) and (28) are determined by the dynamic simulation carried out under various values of parameter \( X \).

**Constraints:** Under the operational constraints, there are design limitation values of the equipment and environmental regulation values (e.g., the former is thermal stress in the steam turbine and the HP drum temperature gradient, the latter is NOx emission from the plant). The constraints are written as follows:

1. Using a function representing the maximum process value (e.g., maximum value of thermal stress in the steam turbine and the HP drum temperature gradient).
   \[ c_i(X) = 1 - \max_{t_1 < t < t_2} \left[ \frac{\left| g_{x_i}(X,t) \right|}{c_{x_i}} \right] \geq 0, \quad i = 1, 2, \ldots, q_e \] \hspace{1cm} (30)

2. Using a function expressing the transition from one state to another (e.g., quantity of NOx emission from the plant during the start-up / shutdown).
   \[ c_i(X) = 1 - \int_{t_1}^{t_2} g_{x_i}(X,t) \, dt / c_{x_i} \geq 0, \quad i = q_e + 1, \ldots, q \] \hspace{1cm} (31)

where \( c_{x_i} \) are the limitations, \( q_e \) is number of functions \( c_i \) in Eq. (30), and \( q \) is number of all functions \( c_i \) in Eqs. (30) and (31). The process values \( g_{x_i} \) in Eqs. (30) and (31) are determined by the dynamic simulation carried out under various values of parameter \( X \).

In order to obtain a feasible solution, each parameter is limited as follows:

\[ x_{L_j} \leq x_j \leq x_{U_j}, \quad j = 1, 2, \ldots, n \] \hspace{1cm} (32)

where \( x_{L_j} \) is the lower bound on \( x \), and \( x_{U_j} \) is the upper bound on \( x \).

**Numerical algorithm:** A calculation flowchart is shown in Fig. 5. Sequential quadratic programming (SQP) is introduced to the optimization procedure. This method is a type of nonlinear programming using the value of the gradient. The gradient of the objective function and the operational constraints are approximated by finite differences. The objective function, the operational constraints and the gradients are calculated from the dynamic simulation.

6.2 Application result

This proposal method is applied to the optimization of the hot start-up process for a multi-shaft combined cycle power plant. The firing temperature of the gas turbine is 1300°C class and total plant output is 670 MW.

The thermal stresses are generated by the rise of the exhaust gas temperature during the gas turbine loading.
The operational parameters are shown in Fig. 6, that is, the control input of eight pieces of the gas turbine ramp rate is optimized. The limitation value of each ramp rate is decided in consideration of the gas turbine’s design criteria based on the surge margin and stress limits.

\[ 0.1 \% / \text{min} \leq x_j \leq 9.0 \% / \text{min}, \quad j = 1, 2, \cdots, 8 \]

The objective function combines the starting time with fuel consumption. The operational constraints are the thermal stress in the steam turbine rotor, the HP drum temperature gradient and the NOx emission from the plant. In this case, the NOx emission had an enough margin for the limitation value.

The results of conventional heuristic approach are shown in Fig. 7. This approach has been decided using only the dynamic simulation by the engineer’s trial and error. The results of this proposed method are shown in Fig. 8. This method can be automatically obtained by optimal operational parameters. In these figures, GT is the gas turbine and ST is the steam turbine. The proposed method reduces the start-up time by 6.0 minutes (21.7%) compared to the conventional method and satisfies the operational constraints.

7. Conclusions

An optimal design method is proposed for the plant system and control design for the combined cycle power plant. The proposed method consists of the nonlinear programming and the dynamic simulation model verified by the actual plant data. It is possible to determine the optimal operational parameters by taking into account the dynamic characteristics of the plant.

References

