Design of a Practical Controller for a Commercial Scale Fossil Power Plant

Asok Ray[1]
Mechanical Engineering Department,
Carnegie-Mellon University,
Pittsburgh, Pa. 15213

D. A. Berkowitz
JAYCOR,
Woburn, Ma. 01801

To design an improved fuel controller for an operating, 386 MW(e), oil-fired power generation system, a mathematical model of the plant was developed from fundamental principles to predict thermal-hydraulic transients. This controller was successfully implemented in the actual plant by replacing a portion of the original control system. It was shown that the resulting minimum operating level of the system could be reduced from 220 MW(e) to flash tank level of 130 MW(e), and the customary load rate of change during normal operation improved from approximately 2 MW/min to 9 MW/min. Operation of the plant on automatic dispatch was subsequently demonstrated.

Introduction

Due to the rising fuel oil costs, economic dispatch favors operating large oil-fired power plants as load-followers and nuclear plants as baseload units. For example, in the Boston Edison Company system, two oil-fired 386 MW(e) once-through subcritical units were required to operate over a wider load range and to change output more rapidly following start-up of a new nuclear plant. However, with the original control scheme, the lower load limit was restricted to 220 MW(e) because of temperature and pressure instabilities. Fig. 1 shows growing throttle pressure and temperature oscillations following a load reduction; output power had to be raised to regain stability, or the unit would be shut down. Furthermore, load rate-of-change was normally limited to 2 MW/min.

To identify the cause of the problems and to formulate an appropriate control algorithm, a dynamic model of the once-through subcritical steam generator and accessories was derived using established techniques [1, 2]. The model was verified with plant data collected by a multichannel tape recorder and data acquisition system [3]. A similar modeling methodology has been applied to Cromby No. 2, a drum-type unit, by the Philadelphia Electric Company, whose experience was beneficial in proceeding with the work reported here[4].

Simulation studies and test data suggested that low load instability at New Boston Unit No. 2 was due to the fuel control system. Two specific design objectives for an improved fuel controller were defined: (1) the unit should operate stably at any load between 130 MW and 386 MW, with main steam temperature settling to set point value of 537.8°C (1000°F) less than 30 min after load change; and (2) main steam temperature excursions should be bounded within ± 5.6°C (± 10°F) for 9 MW/min load rate-of change between 130 MW and 386 MW.

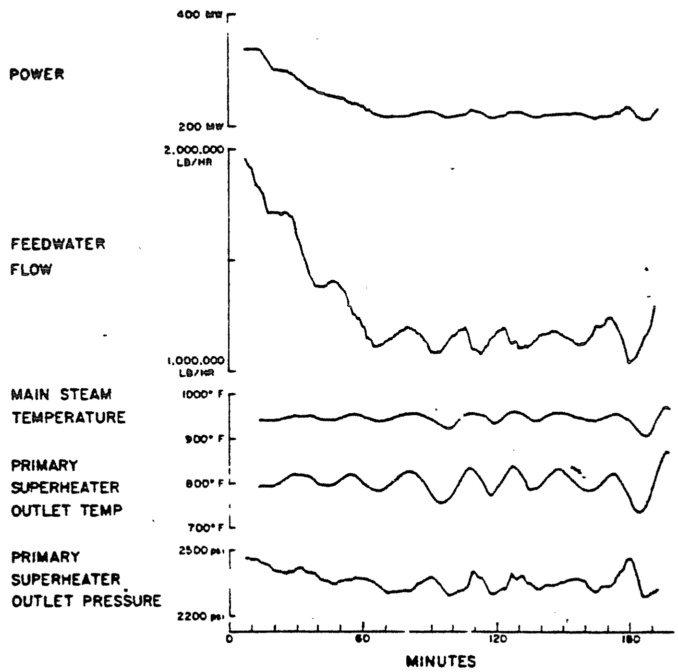

Fig. 1 Temperature and pressure instability at low load

Design of an optimal linear regulator for New Boston No. 2, following the approach of McDonald and Kwatny [5] would have implied complete replacement of the existing controller, which was beyond the authorized scope of effort. Instead, since fuel control was identified as the major problem area, a frequency domain approach was used leading to successful design of a simple, practical fuel controller replacing only a portion of the original control system. Minimum load reduction to 130 MWe was demonstrated and load rate-of change during normal operation increased to 9 MW/min. Unit No. 2 at New Boston Station has operated

[1] The authors were located at the MITRE Corporation, Bedford, Mass., when the work reported here was carried out.

Contributed by the Dynamic Systems and Control Division for publication in the JOURNAL OF DYNAMIC SYSTEMS, MEASUREMENT, AND CONTROL. Manuscript received at ASME Headquarters, September 17, 1979.

Table 1 Plant ratings

Electrical power	386 MW	
Throttle steam pressure	1.655×10^7 N/m^2 g	(2400 psig)
Throttle steam temperature	537.8 C	(1000 F)
Reheat steam temperature	537.8 C	(1000 F)

Table 2 Boiler heat exchanger dimensions

Exchanger	Surface area	Water holding capacity
Furnace	1,717 m^2	51,040 kg
Primary superheater	11,890	111,100
Secondary superheater	1,505	18,720
Reheat superheater	3,756	60,050
Economizer	5,481	57,960

with the revised control system. It was subsequently demonstrated that the plant could be automatically dispatched from the regional control center.

System Description

Plant ratings for New-Boston Unit No. 2 are given in Table 1. It incorporates a once-through subcritical Babcock & Wilcox steam generator. Heat exchanger dimensions are given in Table 2. The furnace has 24 guns for oil firing and two forced draft fans. Two gas recirculating fans, drawing flue gas just ahead of the air preheaters, discharge to recirculating ducts below the burners and to tempering ducts near the top of the furnace radiation section; the ratio of flue gas distribution between recirculating and tempering ducts can be adjusted manually or automatically. The main feedwater pump is driven from the turbine generator shaft through a fluid drive mechanism; pump speed is modulated by the fluid drive to control feedwater flow or pressure.

Normal design operating range is from 100 percent to 33 percent rated feedwater flow rate. Below 33 percent, a portion of main steam is diverted through the flash tank.

Mathematical Model

A mathematical description or model of the process provides an understanding of its dynamics and forms the basis for plant controller design and evaluation. The model must satisfy two criteria:

1. The model should be no larger nor more complex than is necessary for the purposes at hand;
2. Numerical solutions of the model equations must be readily obtained.

The once-through subcritical steam generator, which is the most significant plant component, was modelled using a first-principles approach based on thermodynamic phase information (i.e., compressed water, two-phase mixture, and superheated steam corresponding to economizer, evaporator, and superheater, respectively). This concept was first introduced by Adams, et al. [1], who modelled a coal-fired once-through subcritical steam generator, then linearized the model for analog simulation. Ray and Bowman extended this technique for digital simulation of such a steam generator in a gas-cooled nuclear power plant [2]. A nonlinear dynamic model of the New Boston Unit No. 2 steam generator was derived from the same concept, but with some modifications. A model solution diagram showing input and output variables is given in Fig. 2. The primary modeling objective was to represent main steam temperature dynamics for the purpose of analytical design of a fuel controller.

Locations of saturated water and steam phase boundaries vary with time. Although rate-of-change of these boundary

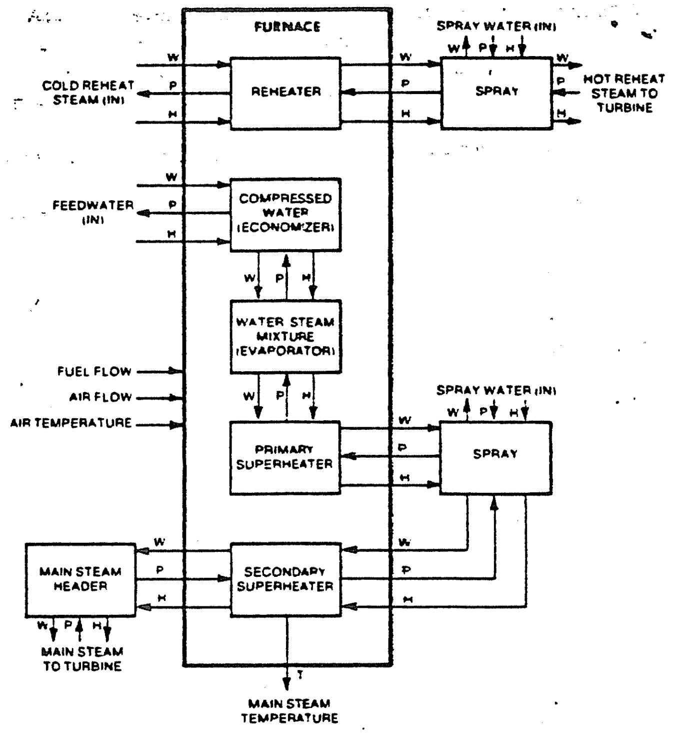

Fig. 2 Steam generator model solution diagram

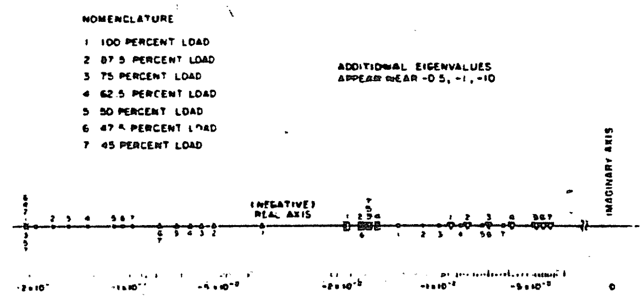

Fig. 3 Root locus plot showing variations in system eigenvalues of boiler model with load

locations may be high under transient conditions, the range of variation is relatively narrow. Fractional changes in lengths of compressed water and superheated steam regions are less than five percent within the load region of interest. Thus the nonlinear model was structured with fixed thermodynamic phase boundaries. For linearization at different load levels, lengths of each phase region were entered as parameters in the nonlinear model. Further, since steam temperature is strongly influenced by heat transfer rate, the superheater region was modelled in primary and secondary superheater sections. When the nonlinear model was linearized at different operating points, model parameters for the linearized models were adjusted such that model results would agree with steady-state plant data.

The nonlinear, time-invariant, continuous-time model has the form

$$\dot{x} = f(x;u) \tag{1}$$

$$y = g(x;u) \tag{2}$$

x is the state vector, with the elements:

x_1 = secondary superheater average metal temperature
x_2 = secondary superheater average steam enthalpy
x_3 = secondary superheater average steam density

x_4 = primary superheater average metal temperature
x_5 = primary superheater average steam/water enthalpy
x_6 = primary superheater average steam/water density
x_7 = evaporator (two-phase section) average metal temperature
x_8 = economizer (subcooled water section) average metal temperature
x_9 = economizer (subcooled water section) average water enthalpy

u is the input (control) vector, whose elements are:

u_1 = fuel flow
u_2 = feedwater flow
u_3 = feedwater enthalpy (at the economizer inlet)
u_4 = throttle pressure

y is the output vector. To design a fuel contrtoller for maintaining constant main steam temperature, only two elements were chosen as output variables:

y_1 = main steam temperature
y_2 = main steam flow

Model equations (1) and (2) were linearized at seven steady-state operating points in the 45-100 percent load range. System eigenvalues (i.e. eigenvalues of the A-matrix) were evaluated for the seven linearized models; their root plot is shown in Fig. 3. Two dominant eigenvalues (closest to the imaginary axis) monotonically decrease in magnitude with load reduction. By examination of the linearized models, main steam temperature was found to be closely associated with these two eigenvalues which are strongly influenced by energy storage in the primary and secondary superheater tube walls.

The original control system was modelled and incorporated in the plant models. The linearized closed loop system yielded a pair of system eigenvalues very close to the imaginary axis. With decreasing load, these eigenvalues monotonically drifted to the right-half plane. Closed loop system stability improved with certain changes in fuel controller parameters, but at the cost of response time. It was felt that a revised fuel control scheme was required to satisfy the design objective.

Controller Design Concepts

In the boiler following mode, changes in electrical output power are made by manipulating governor valves whose displacement directly influences main steam flow and pressure, which, in turn, cause feedwater flow to change. Thermal-hydraulic processes in the steam generator are dependent on both feedwater and fuel flow. Following a load change, fuel flow must be compensated for feedwater flow variations to avoid undesireable deviations in main steam temperature. Thus, feedwater flow W_{fw} is treated as an independent variable. Fuel flow is also dependent on other variables (such as feedwater temperature, draft air flow, etc.). The control design task is to structure the dependence of fuel flow on several independent variables (feedwater flow being the most significant) such that main steam temperature does not deviate beyond specified limits for permissible variations of the independent variables. This analytical relationship is known as the feedforward function [6], which acts as a fast, coarse controller. In addition, a relatively slow feedback loop is provided to compensate for modeling errors and spurious noise, i.e., to hold main steam temperature at the set point under steady-state conditions.

After the controller was designed in the above manner, small signal stability of the closed loop system (process with feedforward and feedback control) was verified at several operating points within a wide range of load levels.

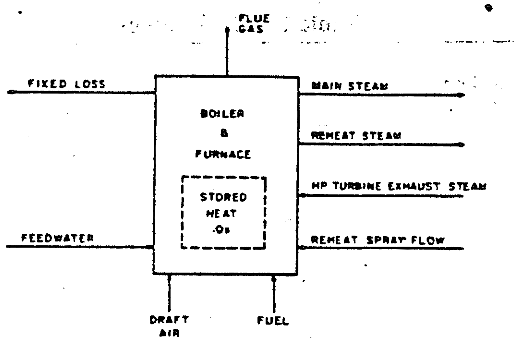

Fig. 4 Furnace-boiler heat balance scheme

Feedforward Function. The feedforward function has been constructed in two parts: static and dynamic. The static part calculates steady-state fuel flow required for thermal equilibrium in the steam generator. To find the steady-state fuel flow, u_1, for achieving a desired value of main steam temperature T_d, set $\dot{x} = \mathbf{0}$ and $y_1 = T_d$ in equations (1) and (2), respectively, which results in eleven equations.

$$\left.\begin{array}{ll} 0 & = f_1(x_1, \ldots, x_9; u_1, \ldots, u_4) \\ \cdot & \quad \cdot \\ \cdot & \quad \cdot \\ \cdot & \quad \cdot \\ 0 & = f_9(x_1, \ldots, x_9; u_1, \ldots, u_4) \\ T_d & = g_1(x_1, \ldots, x_9; u_1, \ldots, u_4) \\ y_2 & = g_2(x_1, \ldots, x_9; u_1, \ldots, u_4) \end{array}\right\} \quad (3)$$

With u_2, u_3, u_4 given, equations (3) could be solved for the eleven unknowns $u_1, y_2, x_1, x_2, x_3, x_4, x_5, x_6, x_7, x_8$, and x_9. In practice, however, equations (3) are very difficult to solve in closed form because f and g are nonlinear functions that involve implicit loops. Analog solution would also be too complex to be practical. To circumvent this problem, steady-state fuel flow demand was obtained by a simple energy balance in the furnace and steam generator as shown in Fig. 4. Heat delivered by feedwater, reheat attemperating water, high pressure turbine exhaust steam, draft air and fuel matches heat removed by flue gas, steam and fixed loss to the environment, under thermal equilibrium. If, at steady-state, reheater load is assumed proportional to feedwater flow, and main steam enthalpy has the desired constant value, then fuel flow W_{fo} can be expressed in terms of feedwater flow W_{fw} and temperature T_{fw} in the following form:

$$W_{fo} = K[(a + bT_{fw})W_{fw} + c] \quad (4)$$

a, b, and c are constants whose physical meaning is discussed in reference [7].

For the dynamic portion of the feed forward function, main steam temperature transfer functions with respect to feedwater flow, W_{fw}, and fuel flow, W_{fo}, can be obtained from a linearized version of equations (1) and (2) as,

$$\frac{T_{ms}}{W_{fw}}(s) = G_1(s) = K_1 \frac{\prod_{i=1}^{m_1}(1 + s/z_{1i})}{\prod_{j=1}^{n_1}(1 + s/p_{1j})}, \quad m_1 \le n_1 \quad (5)$$

and

$$\frac{T_{ms}}{W_{fo}}(s) = G_2(s) = K_2 \frac{\prod_{i=1}^{m_2}(1 + s/z_{2i})}{\prod_{j=1}^{n_2}(1 + s/p_{2j})}, \quad m_2 \le n_2 \quad (6)$$

The transfer function of W_{fo} with respect to W_{fw}, subject to the constraint that T_{ms} is constant, is

$$\frac{W_{fo}}{W_{fw}}(s) = -\frac{G_1(s)}{G_2(s)} = -\frac{K_1}{K_2}\frac{\prod_{i=1}^{m_1}(1+s/z_{1i})\prod_{j=1}^{n_2}(1+s/p_{2j})}{\prod_{i=1}^{m_2}(1+s/z_{2i})\prod_{j=1}^{n_1}(1+s/p_{1j})} \tag{7}$$

Since the model is ninth order, the numer of any transfer function poles cannot exceed 9. Therefore, max $(m_1, m_2, n_1, n_2) \leq 9$.

The coefficient of W_{fw} in the linearized form of equation (4) is equal to the steady-state gain $(-K_1/K_2)$ of equation (7) if equations (1) and (2) are linearized at the same point as equation (4). We define dynamically compensated feedwater flow, W_{dc} as

$$W_{dc}(s) = \frac{\prod_{i=1}^{m_1}(1+s/z_{1i})\prod_{j=1}^{n_2}(1+s/p_{2j})}{\prod_{i=1}^{m_2}(1+s/z_{2i})\prod_{j=1}^{n_1}(1+s/p_{1j})}W_{fw}(s) \tag{8}$$

Parameters z and p are functions of the load conditions at which equations (1) and (2) are linearized. Replacing W_{fw} by W_{dc} in equation (4) includes fuel flow dynamics with respect to feedwater flow in the feedforward function.

As discussed earlier, main steam temperature is strongly influenced by two dominant eigenvalues (see Fig. 3). Therefore, the transfer function order in equations (5) and (6) can be approximated by 2 or less. Further, test data and model results show no immediate change in T_{ms} following a step change in either W_{fo} or W_{fw}, which means that m_1 and m_2 must be less than n_1 and n_2, respectively. Thus, max$(n_1, n_2) \leq 2$ and max$(m_1, m_2) \leq 1$ in equation (8), and an analog simulation of the transfer function can be constructed by at most three cascaded lag units. As a first step in the feedforward dynamic compensator design, the two transfer functions $G_1(s)$ and $G_2(s)$ in equations (5) and (6) are each approximated by a single pole representation, resulting in the simplified structures,

$$G_1(s) = \frac{K_1}{1 + \tau_{fw}s} \text{ and } G_2(s) = \frac{K_2}{1 + \tau_{fo}s}$$

which reduce equation (8) to

$$\frac{W_{dc}}{W_{fw}}(s) = \frac{1+\tau_{fo}s}{1+\tau_{fw}s} \tag{9}$$

τ_{fo} and τ_{fw} are functions of plant load. The transfer function in equation (9) can be constructed by a single lag unit, simplifying equipment installation.

Values of τ_{fo} and τ_{fw} were identified at several load levels using the linearized model frequency responses. They were confirmed by values determined from the nonlinear model in the time domain using a small signal perturbation technique. Thus, results obtained by frequency-domain and time-domain identification were in fairly close agreement.

The feedforward function with analytically determined values of τ_{fo} and τ_{fw} was verified experimentally at different load levels. Results indicated that the dynamic compensator structure in equation (9) was adequate for stabilizing initial oscillations occurring shortly after disturbance. Slower transients and steady-state drift are overcome by feedback control (discussed in the next section).

Analytically determined values of τ_{fo} and τ_{fw} were found to be 150 and 90 seconds, respectively, at full load, and to increase monotonically as load was reduced. Although τ_{fo} and τ_{fw} can be expressed as functions of load, they are not readily convertible to analog circuit representation. If the functional dependence of τ_{fo} and τ_{fw} on plant load could be avoided, equipment implementation would be simpler, resulting in improved reliability. Fixed values of 200 and 150 seconds, respectively, for τ_{fo} and τ_{fw} were found to be acceptable over the full operating range in a series of tests in the actual plant; temperature transients were within design objectives. The transfer function for the feedwater flow dynamic compensator (which was implemented in the plant) is, therefore,

$$\frac{W_{dc}}{W_{fw}}(s) = \frac{1 + 200s}{1 + 150s} \tag{10}$$

Feedwater temperature T_{fw} is also compensated dynamically. The transfer function between main steam and feedwater temperatures is primarily the result of a transport delay, τ_d, in the steam generator tubes. Following a derivation similar to the feedwater flow dynamic compensator, the transfer function for W_{fo} with respect to T_{fw} is approximately

$$\frac{W_{fo}}{T_{fw}}(s) = \frac{K_3(1 + \tau_{fo}s)}{\exp(\tau_d s)} \tag{11}$$

For equipment simplification, equation (11) was further approximated by the relation:

$$\frac{W_{fo}}{T_{fw}}(s) = \frac{K_3}{1 + \tau s} \tag{12}$$

With $\tau = 120$ s, in-plant tests were conducted at different load levels in which a disturbance in T_{fw} was applied by bypassing the high pressure feedwater heaters. The results were satisfactory. The transfer function for the feedwater temperature dynamic compensator (which was implemented in the plant), is therefore

$$\frac{T_{dc}}{T_{fw}}(s) = \frac{1}{1 + 120s} \tag{13}$$

Using equations (10) and (13) in time domain form, the feedforward function of equation (4) is modified to include dynamic compensation.

$$W_{fo} = K[(a + bT_{dc})W_{dc} + c] \tag{14}$$

Feedback Controller. A conventional proportional-integral-derivative (P-I-D) controller was used for feedback control of main steam temperature. The error signal input to the proportional-integral (P-I) controller is the difference between set point and measured value of main steam temperature. A weighted derivative of main steam temperature is added to the P-I controller output to obtain the P-I-D function. The signal required to drive the fuel flow actuator is generated by multiplying the P-I-D controller output with the dynamically compensated feedforward signal from equation (14). A multiplier is incorporated to reduce effective feedback gain as plant load decreases. As shown in Fig. 3, the dominant time-constants (i.e., inverse of the dominant eigenvalues) increase as plant load decreases, indicating that feedback can be made stronger at high load than at low load. Controller parameters must be adjusted with load change to ensure fast and stable operation. In this case, effective controller gain associated with fuel flow actuator drive, which is the product of true controller gain and feedwater signal, monotonically decreases with load.

To design the feedback controller, a Bode plot of main

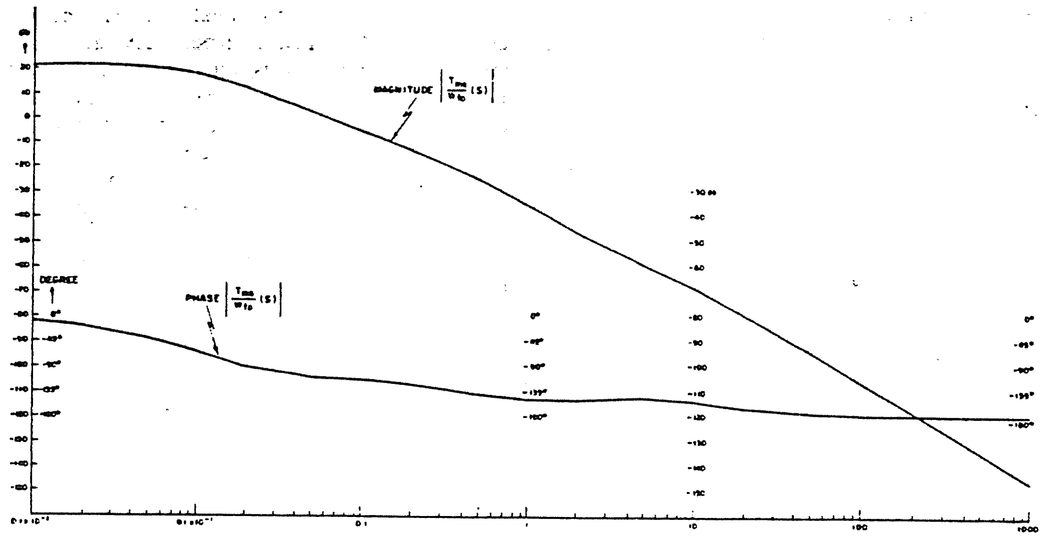

Fig. 5 Frequency response of T_{ms}/W_{fo} *(s)* at 100 percent load

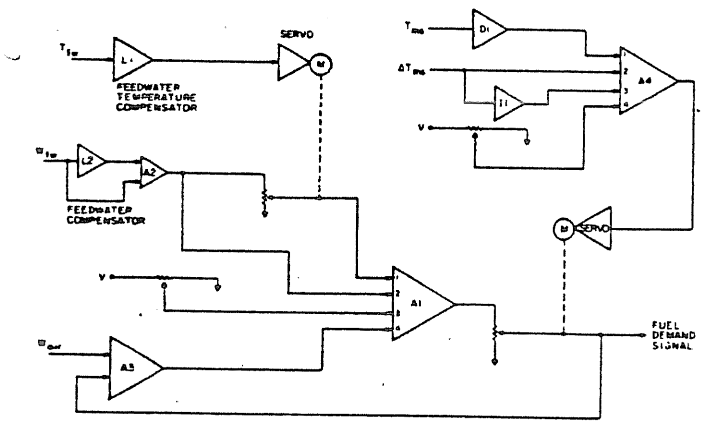

Fig. 6 Fuel controller block diagram

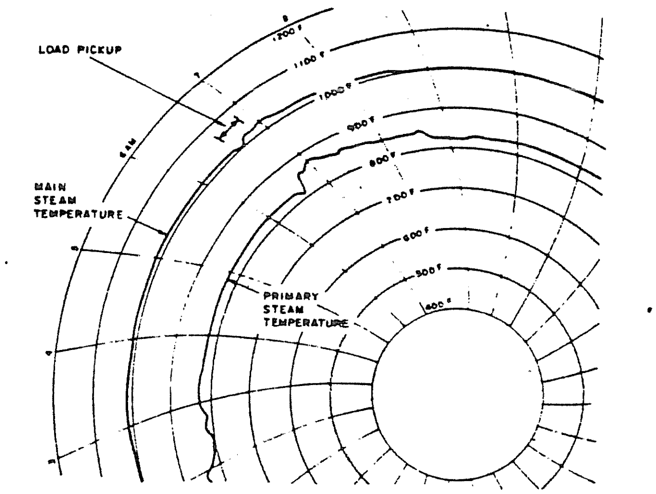

Fig. 7 Typical load escalation profile with the revised fuel controller

steam temperature versus fuel flow was generated from the plant model at full load and is shown in Fig. 5. With performance criteria of gain margin = 6 db and phase margin = 45°, P-I-D controller parameters were evaluated using standard frequency-domain design techniques for linear time-invariant single-input single-output systems [8]. Analytically evaluated parameters were subsequently readjusted as operating experience increased. The Laplace transform for P-I-D controller impulse response is

$$0.4\left(1 + \frac{1}{\tau_i s} + \tau_d s\right) \tag{15}$$

where $\tau_i = 500$ s and $\tau_d = 176$ s.

Controller Implementation

The revised fuel controller was implemented with existing spare components, installed in an available control panel module, and integrated with the overall plant control system. It required two lag units, two servomultipliers,, two potentiometers, four sum units, one integrator and one derivative unit as shown in Fig. 6.

Lag unit L1 is the dynamic compensator for feedwater temperature (see equation (13)); lag unit L2 and sum unit A2 are for feedwater flow (see equation (10)). Sum unit A1 output represents the fuel flow signal generated by the feedforward function (see equation (14)). Input 1 at A1 is proportional to the product of feedwater flow and temperature; input 2 is proportional to feedwater flow, only. Inputs 3 and 4 represent the fixed term of equation (14), which includes air flow heat load. Sum unit A3 output is proportional to the term $(W_{\text{air}} - \xi W_{fo})$ where ξ is the air-fuel ratio. For plant load above 200 MW, air flow is proportional to fuel flow, and the output of A3 is practically zero; for lower loads, air flow is held constant corresponding to 200 MW although fuel flow diminishes. The output of sum unit A4 is the feedback control signal which is multiplied with the A1 output to yield the fuel flow actuator drive signal. Main steam temperature error ΔT_{ms} is summed with its own integral from I1 and the weighted derivative of T_{ms} from D1. The nominal servomultiplier position at A1 output can be adjusted by varying input 4 to A4.

Results and Discussion

The fuel controller, described above, was installed at New Boston Unit No. 2. Test results showed stable operation in the load range of approximately 130 MW to 386 MW, and load rate-of-change exceeding 9 MW/min with main steam temperature variations within permissible limits (± 5.6°C). Fig. 7 shows a typical load pick up at more than 9 MW/min.

Main steam temperature, which had been 543.3°C (1010°F), subsequently rose to 548.9°C (1020°F), and then returned to 543.3°C (1010°F), while power increased from approximately 180 MW to 330 MW. Later on, main steam temperature set point was adjusted bringing the main steam temperature down to 537.8°C (1000°F). In this case, load change was accomplished manually by the operator. In other tests, load rate-of-change up to 12 MW/min on automatic dispatch has been accomplished with permissible temperature fluctuation for both load increase and decrease.

A high pressure feedwater heater leak occurred shortly after the new fuel controller was installed. The leaky heater was bypassed and main steam temperature set point reduced to 510°C (950°F). Under this operating condition, a 10 MW/min load rate-of-change was achieved and low load stability was retained with no adjustment of control parameters. During another test, a high pressure heater failure caused a 33.3°C (60°F) feedwater temperature change in 5 min while the unit was operating at a fixed load. Previously, this type of disturbance could trip the unit. With the new controller, operation continued and main steam temperature excursions were held within ± 8.3°C (± 15°F).

The revised fuel controller was utilized for a period of 1½ years during which time automatic dispatch of New Boston No. 2 was successfully demonstrated. Although controlled load reduction to 130 MWe was achieved, minimum load was held to 190 MWe because of transient spiking of feedwater flow at lower loads unrelated to fuel control, but not previously observed when minimum load was limited to 220 MWe. At present, the revised controller is temporarily out of service (and the plant manually dispatched), because performance of the electro-mechanical servo-multipliers (see Fig. 6) was unacceptably poor resulting in maintenance problems. Contactless, solid state components have now been acquired, and methodical readaptation of the fuel controller is underway.

Conclusions

A mathematical model for an oil-fired 386 MW(e) once-through subcritical steam power plant was formulated using established techniques. The model was used to design a relatively simple fuel controller which was installed in the plant replacing a part of the original control system. Significant improvement in stability and load rate-of-change was achieved, and it was possible to put the plant on automatic dispatch.

The study shows that first-principles modeling and linear control theory can be applied for designing simple and practical controllers to improve system performance. The overall method of analysis presented here, although specifically related to a once-through subcritical steam generator, is also applicable to other processes.

Acknowledgment

The authors acknowledge the participation of Dr. Y. Lin and Mr. R. S. Nielsen in this project, and are grateful to New Boston Station personnel for their cooperation in conducting tests at the station.

References

1 Adams, J., Clark, D. R., Louis, J. R. and Spanbauer, J. P., "Mathematical Modeling of Once-Through Boiler Dynamics," *IEEE Trans., Power Apparatus and Systems*, Vol. 84, No. 2, Feb. 1965, pp. 146-156.

2 Ray, A. and Bowman, H. F., "A Nonlinear Dynamic Model of a Once-Through Subcritical Steam Generator," ASME JOURNAL OF DYNAMIC SYSTEMS, MEASUREMENTS, AND CONTROL, Vol. 98, No. 3, Sept. 1976, pp. 332-339.

3 Berkowitz, D. A. and Nielsen, R. S., "Techniques for Improving Existing Plants," Presented at 96th Winter Annual Meeting (Automatic Control, Section 6 - Panel on Dynamics and Control of Large Thermal Power Stations), Nov. 30 - Dec. 5, 1975, Houston, Tex.

4 Kwatny, H. G., McDonald, J. P., and Spare, J. H., "A Nonlinear Model for Reheat Boiler-Turbine-Generator Systems," Parts I & II, *Proceedings of 12th Joint Automatic Control Conference*, 1971, pp. 219-236.

5 McDonald, J. P. and Kwatny, H. G., "Design and Analysis of Boiler-Turbine-Generator Controls Using Optimal Linear Regulator Theory," *IEEE Trans. Automatic Control*, Vol. AC-18, June 1973, pp. 202-209.

6 Shinky, F. G., *Process Control Systems*, McGraw-Hill, New York, 1979.

7 Lin, Y., Nielsen, R. S., and Ray, A., "Fuel Controller Design in a Once-through Subcritical Steam Generator System," ASME *Journal of Engineering for Power*, Vol. 100, No. 1, Jan. 1978. pp. 189-196.

8 Rosenbrock, H. H., *Computer-Aided Control System Design*, Academic Press, London, 1974.