Reprinted from ENGINEERING FOR POWER, *Vol. 100, No. 1, January 1978*

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Fuel Controller Design in a Once-Through Subcritical Steam Generator System

Established techniques have been applied for modeling a once-through subcritical steam generator in an oil-fired 386 MW(e) power generation system. The model was used to design a fuel controller which was implemented in the actual plant. The resulting minimum stable operating level of the system has been 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.

1 Introduction

Generation cost per megawatt for nuclear power plants is less than for conventional oil-fired units. As more nuclear power plants join in power pools, large oil-fired units have to assume new roles—load followers, rather than baseload units. For example, when the Pilgrim No. 1 nuclear unit became operational in the Boston Edison Company system, two oil-fired 386 MW, once-through subcritical units at New-Boston Station were required to operate over a larger output range and to change output more rapidly than previously. Practically, however, operation was limited to high loads because of temperature and pressure instabilities below the 50 percent load level, which would persist and grow until the operator acted to raise output or to take the unit off line. Furthermore, load rate-of-change was limited to 4 MW/min and was normally one or two megawatts per minute.

The cause of these problems had not been successfully identified. Unit No. 2, performance appeared inferior to that of Unit No. 1 and was chosen for investigation. A 14-channel magnetic tape recorder was used for data acquisition, and 20 additional physical variables were available at the station data logger. Many tests were performed to define the problem [1].[1] Fig. 1 illustrates system pressures in one experiment when the operator attempted low-load operation. Instability was terminated by raising unit power output. Accumulated test data suggested that a substantial problem existed in the fuel control system [2].

Modeling techniques and control theory were systematically applied, which led to a simple, new fuel controller and significantly better unit performance.

The overall system is described briefly in Section 2. The control objectives are given in Section 3. Section 4 details the approach employed in this study and provides discussion on the simplified controller rationale. Section 5 describes the actual implementation, with test results in Section 6, and concluding discussion in Section 7.

2 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 balance of plant is essentially of conventional reheat type except that the main feedwater pump is driven from the turbine-generator shaft through a fluid drive mechanism. Feedwater pump speed is modulated by the fluid drive to control feedwater flow or pressure.

Normal plant 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.

3 Control Objectives

Unit No. 2 control system is boiler-following [3]. Load change is accomplished by manipulating governor valves with a d-c motor operated by a raise-lower switch in the control room. Initial throttle pressure upset is brought back to set point value by the main boiler-feedpump, and feedwater flow, after some excursions, reaches a new level.

Under this mode of operation, two specific design objectives for the

[1] Numbers in brackets designate References at end of paper.

Contributed by the Power Division for publication in the JOURNAL OF ENGINEERING FOR POWER. Manuscript received at ASME Headquarters June 18, 1977.

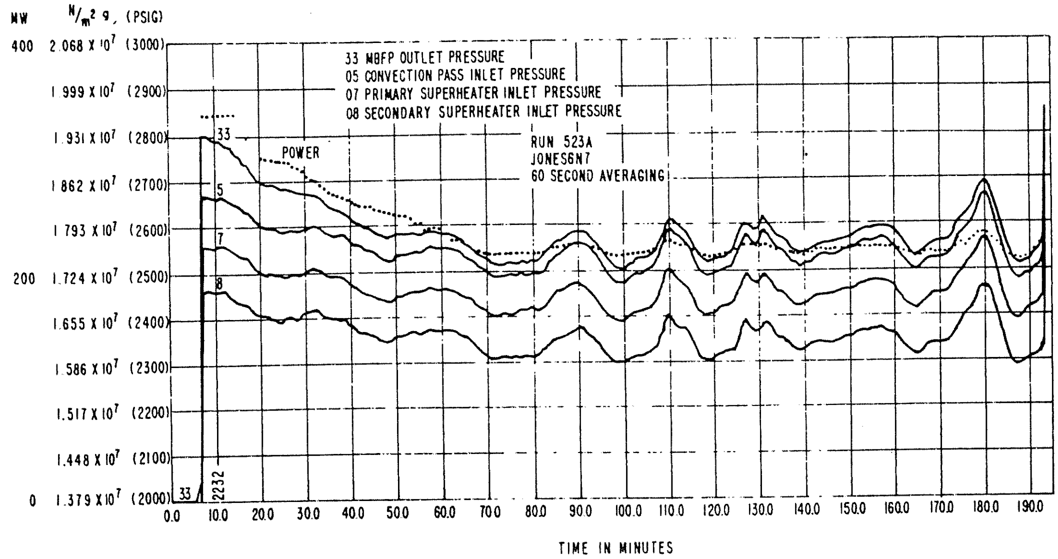

Fig. 1 Low-load instability

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 1 Plant ratings

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

Table 2 Boiler heat exchanger dimensions

fuel controller were defined. First, between 130 and 386 MW main steam temperature excursion should be bounded within ±5.6°C (±10°F) for 9 MW/min load rate-of-change. Secondly, the unit should operate stably at any load between 130 MW and 386 MW, with main steam temperature settling out to set point value 537.8°C (1000°F) in less than 30 min after the last load change is made.

4 Analysis and Controller Design

Analytical controller design requires an adequate mathematical process description or model, to provide an understanding of the process and an analytical basis for controller parameter evaluation with respect to a specified performance index.

The model must satisfy two criteria:

1 Optimum model complexity, i.e., the model should be no larger nor more complex than is necessary for the purposes at hand;

2 Numerical solutions of model equations must be readily obtained.

Analysis. The once-through subcritical steam generator was modeled using a first-principles approach on the basis of thermodynamic phase information (i.e., compressed water, two-phase mixture, and superheated steam). This concept was first introduced by Adams, et al. [4], who modeled a coal-fired once-through subcritical steam generator, and then linearized the model for analog simulation. Ray, et al. [5, 6], extended this technique to include digital simulation of the nonlinear model of such a steam generator in a gas-cooled nuclear power plant. A nonlinear model of New-Boston Unit No. 2 steam generator was derived following the same concepts and technique; it is suitable for fuel controller design but may need reformulation for other purposes (for example, study of feedwater flow oscillations).

Specifically, the infinite-dimensional distributed process in the steam generator was approximated by a finite-dimensional lumped parameter model. Steady-state model results were verified with heat balance and field test data at different load levels. The nonlinear model has the form

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

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

(See Appendix for nomenclature.) For a fuel controller designed to maintain constant main steam temperature, only two elements (main steam flow and temperature) were chosen in the **y**-vector. Model equations (1) and (2) were then linearized at seven steady-state operating points in the form

$$\delta\dot{\mathbf{x}} = \mathbf{A}\delta\mathbf{x} + \mathbf{B}\delta\mathbf{u} \tag{3}$$

$$\delta\mathbf{y} = \mathbf{C}\delta\mathbf{x} + \mathbf{D}\delta\mathbf{u} \tag{4}$$

where $\mathbf{A}(9 \times 9)$, $\mathbf{B}(9 \times 4)$, $\mathbf{C}(2 \times 9)$ and $\mathbf{D}(2 \times 4)$ are matrices of the indicated dimensions and the prefix δ signifies a small increment.

System eigenvalues of the **A**-matrix were evaluated for each linearized model and their root locus plot is shown in Fig. 2 for the 45–100 percent load range.

Two dominant eigenvalues closest to the imaginary axis monotonically decrease in magnitude with load reduction. From the linearized state equations (3) and (4), these dominant eigenvalues were closely related to main steam temperature, which signifies that main steam temperature transients are strongly influenced by energy storage in the primary and secondary superheater tube walls.

Controller Design Concepts. In the boiler following mode, changes in megawatt output are implemented by manipulating governor valves. Governor valve displacement directly effects main steam flow and pressure which, in turn, causes feedwater flow to change. Since thermal hydraulic processes in the steam generator are strongly influenced by dynamic energy balance, fuel flow must be adequately

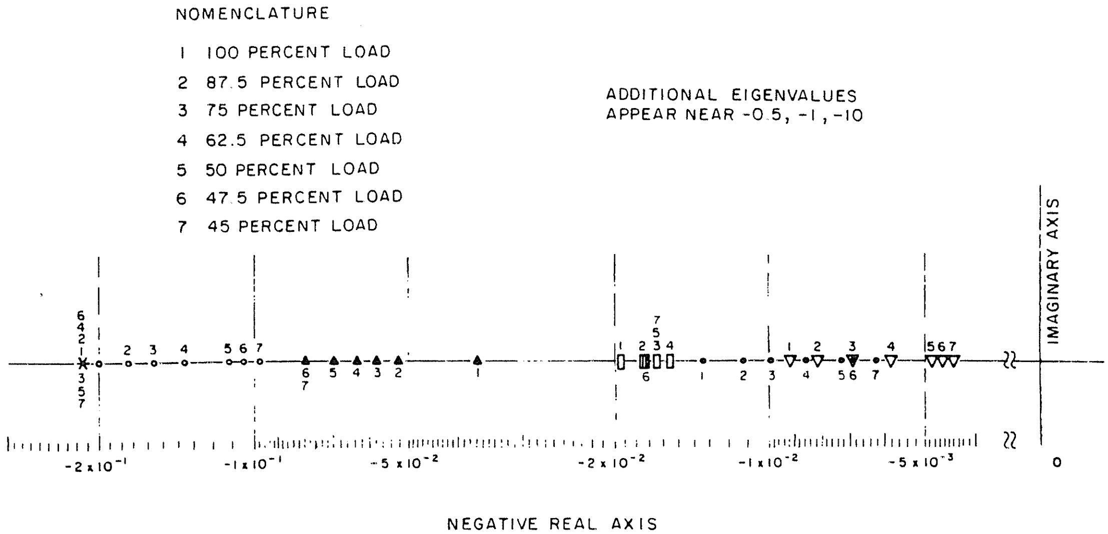

Fig. 2 Root locus plot showing variation in system eigenvalues of boiler model with load

compensated for feedwater flow variations to avoid undesirable main steam temperature deviation. Thus, feedwater flow W_{fw} is treated as an independent variable and fuel flow W_{f0} as the dependent variable. Fuel flow is also dependent on other variables (feedwater temperature, draft air flow, etc.). The task is to define fuel flow as a function of 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 [7]. It is formulated with separate static and dynamic parts.

The feedforward function acts as a fast, coarse controller. In addition, a (relatively slow) feedback trim loop is provided:

1 To accommodate for any errors in formulating the static feedforward function (i.e., to hold main steam temperature at the set point automatically under steady-state conditions), and

2 To overcome effects of unexpected disturbances (boiler tube leaks, for example).

Static Feedforward Function. The static feedforward controller calculates required steady-state fuel flow to reach the desired thermal equilibrium in the boiler. Theoretically, steady-state fuel flow depends on boiler operating conditions and can be obtained by solving equations (1) and (2) (see Appendix). In practice, however, equations (1) and (2) are very difficult to solve because **f** and **g** are nonlinear functions and there are implicit loops. An analog calculation circuit implemented to solve equations (1) and (2) would be too complex to be practical.

To circumvent this problem, a simple steady-state heat balance was employed for the boiler to derive the fuel flow demand, as illustrated in Fig. 3, assuming fixed steady-state furnace-boiler temperature distribution. Heat delivered to the furnace and boiler by feedwater, reheat attemperating water flow, draft air, HP turbine exhaust steam, and fuel oil, is exactly equal to heat removed by radiation and conduction to the environment, by draft air flow and combustion products to the stack, and by steam flow to the turbine. When input equals output, furnace and boiler temperatures approach equilibrium. Equivalently, fuel flow (W_{f0}) can be expressed as

$$W_{f0} = \frac{Q(T)}{\text{FHV}} \tag{5}$$

where $Q(T)$ is the total heat load on the furnace:

$$Q(T) = Q(\text{MSG}) + Q(\text{RHG}) + Q(\text{AR}) + Q(\text{FL}) \tag{6}$$

$Q(\text{MSG})$ = main steam generator heat load = heat carried (by steam) to HP turbine less heat carried (by feedwater) to boiler

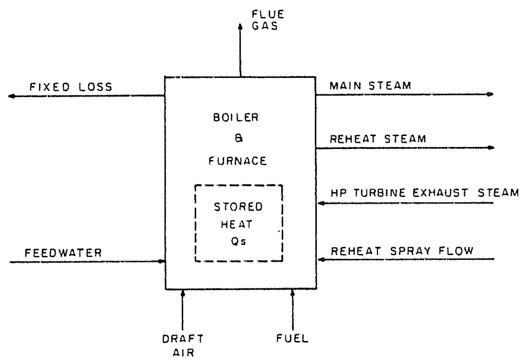

Fig. 3 Furnace-boiler heat flow

$Q(\text{RHG})$ = reheater heat load = heat carried by steam at reheater outlet less heat carried by steam at reheater inlet

$Q(\text{AR})$ = air flow heat load = heat carried out of stack by air less heat delivered to furnace by air

$Q(\text{FL})$ = heat removed from furnace-boiler by radiation and conduction to environment

FHV = fuel heating value

The four terms on the right-hand side of equation (6) are discussed separately.

1 *Main Steam Generator Heat Load—Q(MSG).* For given feedwater flow, main steam generator heat load is defined as the power required to turn feedwater into steam at set point value at the secondary superheater outlet, and is derived from the difference between superheater outlet steam enthalpy and economizer inlet feedwater enthalpy (which depends on both pressure and temperature). However, feedwater enthalpy is assumed independent of feedwater pressure because of insignificant pressure variations over the normal operating range. In addition, since steady-state throttle pressure is maintained at 1.65×10^7 N/m²g (2400 psig) by the feedpump, pressure variations in superheater outlet steam enthalpy can also be ignored. Main steam heat load is calculated by

$$\begin{aligned} Q(\text{MSG}) &= W_{fw}(h_{ms} - h_{fw}) \\ W_{fw} &= \text{feedwater flow, kg/s} \\ h_{ms} &= \text{main steam enthalpy, J/kg} \\ h_{fw} &= \text{feedwater enthalpy, J/kg} \end{aligned} \tag{7}$$

2 *Reheater Heat Load—Q(RHG).* Reheater heat load is the

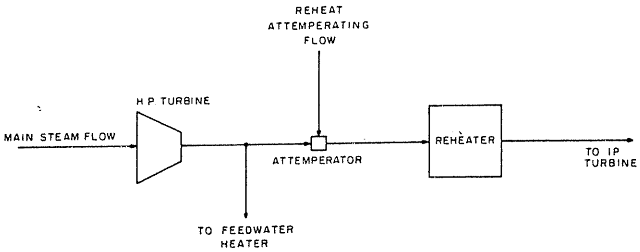

Fig. 4 Flow balance for the reheater

power required to reheat high-pressure turbine exhaust steam to set point temperature. A simplified diagram of relevant reheater flows appears in Fig. 4. Accurate calculation of reheater heat load requires measurement of steam flow through the reheater, but reheater spray water flow, steam flow to feedwater heaters, and steam flow to and from the reheater are not measured variables. Only feedwater flow is available. To avoid the expense of installing additional flow measurement instrumentation, a fixed value of 5.28×10^5 J/kg (227 Btu/lb) was assigned. Heat balance analyses and measurement data indicated values between 5.00×10^5 J/kg (215 Btu/lb) and 5.58×10^5 J/kg (240 Btu/lb) depending upon steam flow; similarly, reheater steam flow varies a few percent about an average value equal to 90 percent of measured feedwater flow. Reheater heat load is approximately

$$Q(\text{RHG}) = 0.9W_{fw}(\Delta h_{rh}) \text{ J/s}$$
$$W_{fw} = \text{feedwater flow, kg/s}$$
$$\Delta h_{rh} = \text{average enthalpy gain through reheater} \cong 5.28 \times 10^5 \text{ J/kg (227 Btu/lb)} \quad (8)$$

3 *Air Flow Heat Load—Q(AR).* Air flow heat load is defined as heat carried out of the stack by air flow less heat delivered to the furnace by air flow.

$$Q(\text{AR}) = W_{\text{air}}(\Delta h_{\text{air}})$$
$$\Delta h_{\text{air}} = \text{enthalpy gain for air J/kg}$$
$$W_{\text{air}} = \text{air flow, kg/s} \quad (9)$$

4 *Furnace and Boiler Fixed Heat Loss—Q(FL).* Heat is removed by radiation and conduction. The fixed heat loss can be derived from experimental steady-state data for fuel, feedwater, and draft air flows, using equations (5) and (6), because Q(MSG), Q(RHG), Q(AR), and W_{f0} are known from empirical steady-state data at several loads. Q(FL) can then be identified as the intercept of the following linear equation.

$$W_{f0} = \frac{1}{\text{FHV}}[Q + Q(\text{FL})] \quad (10)$$

$$Q = Q(\text{MSG}) + Q(\text{RHG}) + Q(\text{AR}) \quad (11)$$

Q(FL) derived in this manner includes the real fixed heat loss, and also residuals from the feedwater enthalpy calculation, reheater load approximation, and air flow heat load calculation.

If equations (7)–(9) and (11) are substituted into equation (10), the steady-state relation between fuel and feedwater is

$$W_{f0} = \frac{1}{\text{FHV}}\{[(h_{ms} - h_{fw}) + 0.9\Delta h_{rh}]W_{fw} + Q(\text{AR}) + Q(\text{FL})\} \quad (12)$$

Dynamic Feedforward Function. Laplace transforms of equations (3) and (4) yield the transfer function matrix $\mathbf{G}(s)$ between the input vector $\overline{\mathbf{u}}(s)$ and output vector $\overline{\mathbf{y}}(s)$.

$$\overline{\mathbf{y}}(s) = \mathbf{G}(s)\overline{\mathbf{u}}(s)$$
$$\mathbf{G}(s) = \mathbf{C}[s\mathbf{I} - \mathbf{A}]^{-1}\mathbf{B} + \mathbf{D} \quad (13)$$

The transfer functions of main steam temperature T_{ms} with respect to feedwater flow W_{fw} and fuel flow W_{f0} can be obtained from equation (13) as

$$\frac{T_{ms}}{W_{fw}}(s) = G_1(s) = K_1 \frac{\prod_{i=1}^{m_1}(1 + s/z_{1i})}{\prod_{j=1}^{n_1}(1 + s/p_{1j})} \qquad m_1 \leq n_1 \quad (14)$$

and

$$\frac{T_{ms}}{W_{f0}}(s) = G_2(s) = K_2 \frac{\prod_{i=1}^{m_2}(1 + s/z_{2i})}{\prod_{j=1}^{n_2}(1 + s/p_{2j})} \qquad m_2 \leq n_2 \quad (15)$$

The transfer function of W_{f0} with respect to W_{fw}, subject to the constraint that T_{ms} is constant, is

$$\frac{W_{f0}}{W_{fw}}(s) = -\frac{G_1(s)}{G_2(s)} = -\frac{K_1}{K_2}\frac{\prod_{i=1}^{m_1}(1 + s/z_{1i})\prod_{j=1}^{n_2}(1 + s/p_{2j})}{\prod_{i=1}^{m_2}(1 + s/z_{2i})\prod_{j=1}^{n_1}(1 + s/p_{1j})} \quad (16)$$

Since the model is of ninth order, max $(m_1, n_1, m_2, n_2) \leqslant 9$.

The steady-state gain $(-K_1/K_2)$ in equation (16) is equivalent to the coefficient of W_{fw} in equation (12), the steady-state fuel flow calculation. We define the (dynamically) compensated feedwater flow W_{dc} as

$$W_{dc}(s) = \frac{\prod_{i=1}^{m_1}(1 + s/z_{1i})\prod_{j=1}^{n_2}(1 + s/p_{2j})}{\prod_{i=1}^{m_2}(1 + s/z_{2i})\prod_{j=1}^{n_1}(1 + s/p_{1j})} W_{fw}(s) \quad (17)$$

In the static feedwater equation (12), W_{fw} is replaced by W_{dc}. Thus, fuel flow dynamics with respect to feedwater flow is included in the feedforward function.

As discussed earlier, main steam temperature is strongly influenced by two dominant system eigenvalues, and the transfer function order in equations (14) and (15) can be approximated as 2. Further, from model results, test data, and physical reasoning, there is no immediate change in T_{ms} for a given change in either W_{f0} or W_{fw}. Therefore m_1 and m_2 must be less than n_1 and n_2, respectively. Thus, max $(n_1, n_2) \leqslant 2$ and max $(m_1, m_2) \leqslant 1$ in equations (14)–(16).

For simplicity in equipment implementation and as a first step in the design of the feedforward dynamic compensator, the structures of transfer functions G_1 and G_2 in equations (14) and (15) were taken to be

$$G_1(s) = \frac{K_1}{1 + \tau_{fw}s} \quad \text{and} \quad G_2(s) = \frac{K_2}{1 + \tau_{f0}s}$$

reducing equation (17) to

$$\frac{W_{dc}}{W_{fw}}(s) = \frac{1 + \tau_{f0}s}{1 + \tau_{fw}s} \quad (18)$$

The parameters τ_{f0} and τ_{fw} are functions of plant load. Using the linearized model frequency response, τ_{f0} and τ_{fw} were identified at different load levels. To confirm the values, they were also evaluated from the nonlinear model in the time domain using a small signal perturbation technique. The results obtained by frequency-domain and time-domain identification were in close agreement.

The feedforward function was experimentally verified in the plant at different load levels with analytically determined values of τ_{f0} and τ_{fw}. Fixed values of τ_{f0} and τ_{fw} were tested for the full operating range, and found to yield satisfactory results. Actual values of τ_{f0} and τ_{fw} are 200 and 150 s, respectively; the transfer function for feedwater flow dynamic compensation is, therefore,

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

Similarly, feedwater enthalpy should be (dynamically) compensated. The transfer function between main steam temperature and feedwater enthalpy is primarily the transport delay (τ) in the steam generator tubes. Thus, the transfer function for W_{f0} with respect to h_{fw} (following a derivation similar to the feedwater flow dynamic compensator) is given by

$$\frac{W_{f0}}{h_{fw}} = -\frac{K_3(1 + \tau_{f0}s)}{e^{\tau s}}$$

This expression can be approximated as

$$\frac{W_{f0}}{h_{fw}} \cong -\frac{K_3}{1 + Ts}$$

when τ is greater than τ_{f0}.

Tests showed that T = 120 s yields satisfactory results for all loads of interest.

Feedback Controller. Main steam temperature error is an input to the proportional-integral controller. A weighted derivative of main steam temperature is added, and output of this proportional-integral-derivative controller is multiplied with the feedforward function to yield the fuel flow actuator signal drive, as shown in Fig. 5. The rationale for incorporating a multiplier is to reduce effective feedback gain as plant load decreases. As shown in Fig. 2, the dominant time-constants of the steam generation process increase as plant load decreases, which indicates that feedback action can be made stronger at high load than at low load, and controller parameters must be adjusted with load changes to ensure fast and stable operation.

Let x and y be the feedforward and feedback control signals, respectively. The signal to drive the fuel flow actuator is then given by

$$z = xy$$

Linearization of this equation at a certain operating point, (x_0, y_0) yields

$$\delta z = x_0 \delta y + y_0 \delta x \tag{19}$$

Assuming feedforward network frequency range to be higher than that of the feedback network, equation (19) is approximately

$$\delta z \cong x_0 \delta y \tag{20}$$

The feedback control signal y is the output of a standard PID device. Thus, effective controller gain associated with driving the fuel flow actuator, which is the product of true controller gain and the feedforward signal x_0, monotonically decreases with load.

Design. The frequency response plot for main steam temperature versus fuel flow was obtained from the plant model at full load and is shown in Fig. 6. With the performance criteria of gain margin of one-half and phase margin of 45 deg, the PID controller parameters were evaluated using standard frequency domain techniques for linear time-invariant single-input single-output controller design [7]. The values were used initially to operate the plant and were subsequently readjusted as operating experience increased. The Laplace transform for the PID controller is

$$0.4\left(1 + \frac{0.12}{s} + 2.93s\right)$$

5 Controller Implementation

The controller was implemented with standard control modules: two lag units, two servomultipliers, two potentiometers, four summers, one integrator, and one derivative unit (see Fig. 7).

The output of summer A1 represents the total furnace-boiler heat load. Input 1 to A1 represents the product of feedwater flow and enthalpy (see equation (7)). Since feedwater enthalpy h_{fw} of subcooled feedwater is a weak function of pressure, it is assumed to depend on feedwater temperature (T_{fw}) alone. Two piecewise linear functions were suitable for the temperature range $-17.8°C$ to $315.6°C$

$$h_{fw} = aT_{fw} + b \quad \text{and} \quad \begin{cases} a = 4.187 \times 10^3, & b = 1.254 \times 10^4 \quad \text{for} \quad -17.8°C \le T_{fw} \le 154.7°C \\ a = 4.438 \times 10^3, & b = -2.633 \times 10^4 \quad \text{for} \quad 154.7°C \le T_{fw} \le 315.6°C \end{cases}$$

Substitution in equation (12) for fuel flow yields

$$W_{f0} = \frac{1}{\text{FHV}} \times [(3.40 \times 10^6 - aT_{fw} - b + 0.9 \times 5.28 \times 10^5)\, W_{fw} + Q(AR) + Q(FL)] \tag{21}$$

in which $h_{ms} = 3.40 \times 10^6$ J/kg (1460.9 Btu/lb) for main steam conditions of 1.65×10^7 N/m²g (2400 psig) and 537.8°C (1000°F), and $\Delta h_{rh} = 5.28 \times 10^5$ J/kg (227 Btu/lb) as stated in Section 4. Lag unit L1 precedes the feedwater enthalpy servo as the dynamic compensator for feedwater enthalpy and, at steady state input 1 to A1, represents the term $-aT_{fw}W_{fw}$ in equation (21). Input 2 to A1 represents the terms $(3.40 \times 10^6 - b + 0.9 \times 5.28 \times 10^5)\, W_{fw}$ of equation (21) at steady state. The dynamic feedwater compensator is represented by the circuit between the lag unit L2 input and the summer A2 output.

Inputs 3 and 4 represent the fixed loss and air flow heat load, respectively. If equation (9) is implemented without modification, a potential control system instability arises. Draft air is proportioned to provide a desired excess oxygen value; if draft air heat load is simply summed with other heat load components, as indicated in Fig. 5, any increase in fuel flow would cause an increase in air flow, a corresponding increase in heat load, and a regenerative increase in fuel flow. To avoid this positive feedback effect the stoichiometric draft air flow value is computed from fuel flow demand and subtracted from measured air flow. The difference becomes the component of total heat load, and the stoichiometric air load is interpreted as a reduction in fuel heat content.

An air-to-fuel ratio of 14.75 for complete combustion leads to the following relation for calculating air heat load:

$$\begin{aligned} Q(\text{AR}) &= \Delta h_{\text{air}}(W_{\text{air}} - 14.75 W_{f0}) \text{ J/s} \\ \Delta h_{\text{air}} &= \text{enthalpy gain for air, J/kg} \\ W_{\text{air}} &= \text{air flow, kg/s} \\ W_{f0} &= \text{fuel oil flow, kg/s} \end{aligned} \tag{22}$$

During normal operation at New-Boston, minimum draft air flow is fixed at a value corresponding to approximately 200 MW output; below this level, draft air is held constant as fuel flow diminishes. Therefore, summer A3 output is usually close to zero above 200 MW and proportional to the term $(W_{\text{air}} - 14.75W_{f0})$ in equation (22) below 200 MW.

Summer A4 and the fuel heating value servo represent the feedback controller for fine control of main steam temperature. The position of the servo multiplier potentiometer at A1 output represents fuel-heating value. The nominal potentiometer position can be adjusted by varying the voltage at input 4 of amplifier A4. Measured main steam temperature is compared with set point value to derive ΔT_{ms}, which is summed with the integral of temperature error from I1 and the temperature derivative from unit D1. A4 output drives the fuel calibration servo.

6 Test Results and Discussions

Controller implementation and final parameter adjustment for New-Boston Station Unit 2 was completed on April 20, 1976. The unit has since operated with the revised controller. Test results show stable operation at any load between approximately 130 MW and 386 MW, and load rate-of-change exceeding 9 MW/min with main steam temperature variations within permissible limits (±5.6°C) For example, Fig. 8 illustrates load pickup on the morning of July 15, 1976, which began shortly after 6:30 AM and occurred at more than 9 MW/min. Main steam temperature, which had been 543.3°C (1010°F) on the chart, dipped to about 537.8°C (1000°F), subsequently rose to 548.9°C (1020°F), and returned to 543.3°C (1010°F), while power increased from approximately 180 MW to 330 MW. At 8:30 AM, main steam temperature set point was reset bringing main steam temperature down to 537.8°C (1000°F). Load changes during this test were initiated manually by the operator rather than by the dispatcher via a signaling system. During other tests, change rates of 12 MW/min have been demonstrated for load decreases and increases, with acceptable temperature performance.

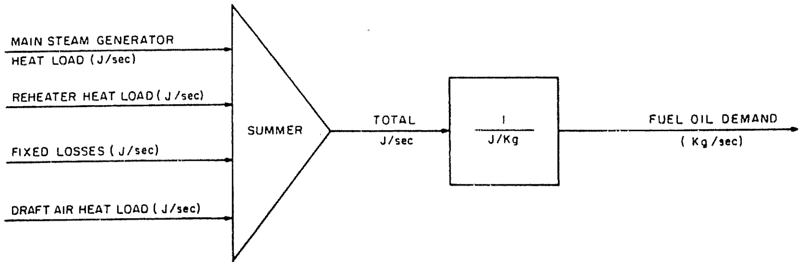

Fig. 5 **Concept for steady-state fuel oil demand calculation**

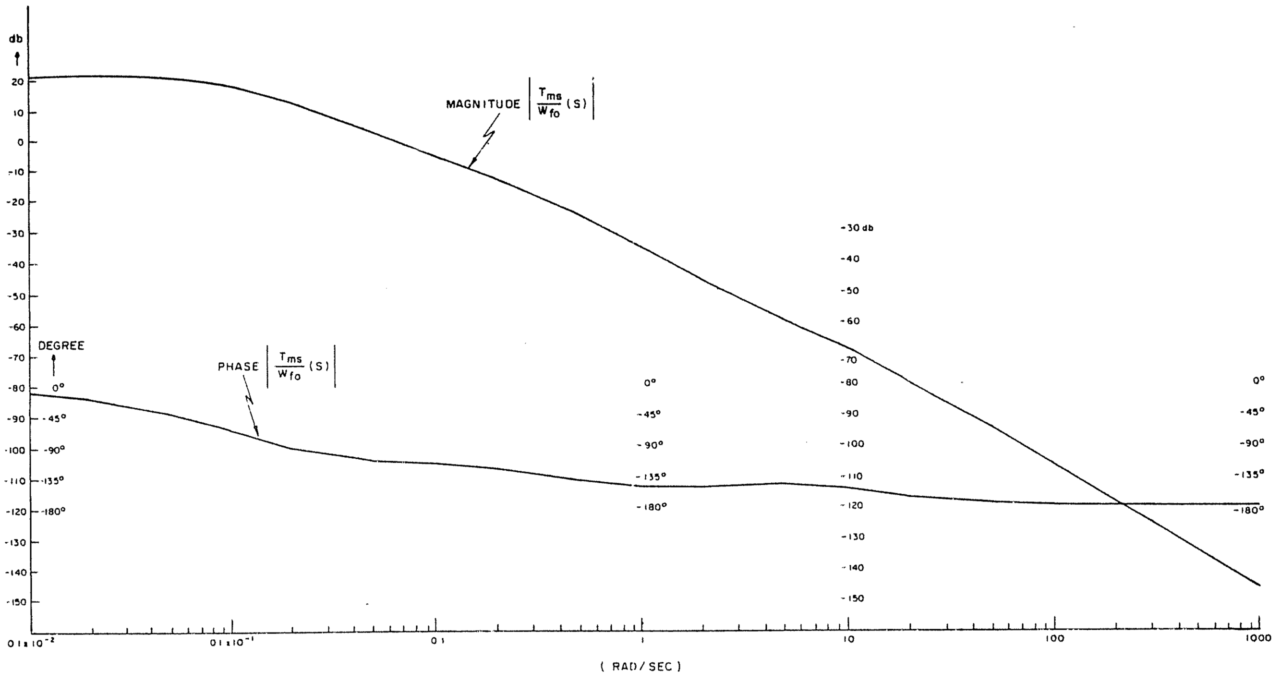

Fig. 6 **Frequency response of $(T_{ms}/W_{fo})(s)$ at 100 percent load**

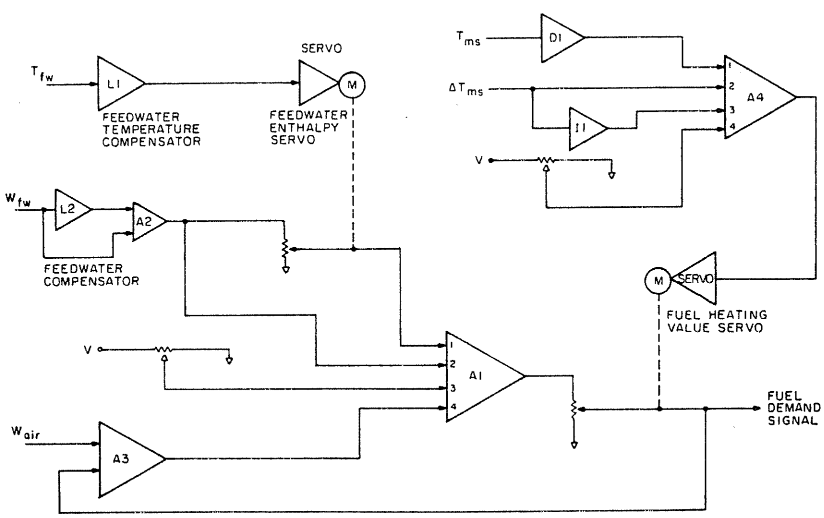

Fig. 7 **Fuel controller block diagram**

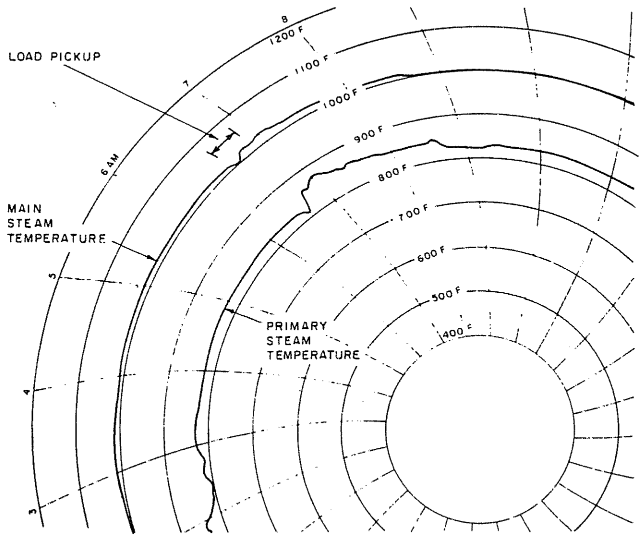

Fig. 8 150 MW load increase

A high-pressure feedwater heater leak occurred shortly after the new fuel controller was implemented. Under these conditions, the leaky heater is bypassed and main steam temperature set-point is reduced to 510°C (950°F). Prior to controller installation, load change-rate had to be decreased and main steam temperatures exhibited continuing oscillations with peak amplitude of ±2.8°C (±5°F) and a period of about 10 min.

Theoretically, the reduced enthalpy at the throttle requires small adjustment of the controller feedforward signal calculation. Practically, this was unnecessary. However, after controller installation, performance was hardly affected; 10 MW/min load changes were achieved, and low-load stability was retained.

In another test, a high-pressure heater problem 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 cause the unit to trip. With the new controller, normal operation continued and main steam temperature excursions were held within ±8.3°C (±15°F).

A boiler leak occurred during one period when the controller and unit behavior were being closely observed, and data were being recorded manually. The leak was clearly evident by an anomalous change in the fuel calibrator potentiometer. The reduction in feedwater heat load implied by feedwater escaping up the stack caused movement of the potentiometer to reduce fuel delivered to the furnace. Though this feature has not been exploited, it suggests that the controller can provide data for early warning about boiler leaks.

Tests suggested that the present load rate-of-change of the unit is not limited by the fuel controller; instead the operator usually waits for throttle pressure to return within prescribed limits before changing load again. Therefore, if throttle pressure control is improved, load rate-of-change may be increased further.

Operating personnel have gained confidence in the revised fuel controller and Boston Edison Company has automated load dispatch control so that the regional energy control center (REMVEC) remotely controls Unit 2 output megawatts. Presently, the automatic dispatch control range is 210–350 MW and load changes occur at a rate of typically 8 MW/min. The control range is being extended.

7 Conclusion

The modeling and simulation techniques were applied to a once-through subcritical steam generator in a 386 MW(e) oil-fired power generation system, and a ninth-order nonlinear model was developed and digitally simulated. The model equations (combined with an energy balance control concept), standard linear techniques, and eigenvalue analysis were employed in the design of a relatively simple fuel controller. Significant unit performance improvements in low-load stability and load rate-of-change were simultaneously achieved when the controller was implemented.

This study shows that systematic modeling and simulation can provide a basis for understanding the process, and for the design of a simple and practical controller to yield improved plant performance. The overall method of analysis presented here, although applied specifically to a once-through steam generator, is also valuable in other applications.

8 Acknowledgment

The authors acknowledge the support of Mr. William Irving and Mr. William Burton of the Boston Edison Company during the course of this study. They are grateful to New-Boston Station personnel for their cooperation in carrying out tests at the station. Thanks also go to Dr. David Berkowitz of The MITRE Corporation for editing the manuscript and clarifying some points in the paper, and to Diana Walgreen for typing the paper.

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APPENDIX

Theoretical Derivation of Steady-State Fuel Flow From the Model

The nonlinear model has the form

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

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

x is the state vector. It consists of

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 vector. It has the following elements:

u_1 = fuel flow
u_2 = feedwater flow
u_3 = feedwater enthalpy (at the inlet of economizer)

u_4 = throttle pressure

y is the output vector. Only two elements were chosen for the y-vector:

y_1 = main steam temperature

y_2 = main steam flow

To find steady-state fuel flow (u_1) to achieve 537.8°C (1000°F) of main steam temperature, for example, we set $\dot{\mathbf{x}} = \mathbf{0}$ in equation (A-1) and y_1 = 537.8°C (1000°F).

$$\left.\begin{aligned} 0 &= f_1(x_1, \ldots, x_9; u_1, \ldots, u_4) \\ &\vdots \\ 0 &= f_9(x_1, \ldots, x_9; u_1, \ldots, u_4) \\ 537.8 &= g_1(x_1, \ldots, x_9; u_1, \ldots, u_4) \\ y_2 &= g_2(x_1, \ldots, x_9; u_1, \ldots, u_4) \end{aligned}\right\} \qquad \text{(A-3)}$$

With u_2, u_3, and u_4 given, u_1 can be determined from algebraic equations (A-3) because there are eleven equations in (A-3) for eleven unknowns (i.e., $x_1, \ldots, x_9$, u_1, and y_2).