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— POWER TRAIN —

Hybrid Computer Simulation of a Babcock & Wilcox Nuclear Power Plant

by

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ABSTRACT

POWER TRAIN is a hybrid computer simulation of the normal and transient power operation of Babcock & Wilcox nuclear power plants. The simulation runs in real time and employs three digital computers and two analog computer consoles. POWER TRAIN includes models of all major components in both the primary and secondary loops of the plant from the reactor core to the turbine generator, including the pressurizer, steam generators, feedwater heaters, feedwater pumps, control valves, safety and relief valves, and the integrated control system.

Parameters describing plant components are input to the simulation in card image format. Pushbuttons on a "mini" control panel allow real-time operation of the simulation. Strip chart recorders are the principal output devices.

This report describes the computer facilities and the mathematical models used by POWER TRAIN. The description of the mathematical models includes modeling assumptions, equations, solution methods, limitations of the simulation, and a summary of the input requirements and output capabilities.

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1. INTRODUCTION

POWER TRAIN is a real-time, on-line hybrid computer simulation of power operation for Babcock & Wilcox nuclear power plants (177-, 205-, and 145-fuel assembly plants). The code is used to predict the performance and behavior of the major components in the nuclear steam system (NSS) for a wide range of plant conditions and operation. POWER TRAIN is designed to model as much of the power plant as is feasible, especially those components whose behavior is interrelated with that of others. This report describes the scope of the simulation, the modeling assumptions, and the modeling equations. Also given is an overview of the method of implementation, the input requirements, and the output capabilities.

The objectives of the simulation are listed below:

1. Two-loop simulation.
2. Automatic setup and checkout capability.
3. Flexibility to change system parameters, especially control system gains and setpoints.
4. Operation of the simulation (as nearly as possible) by actuating pushbuttons on or off.

The scope of the simulation, together with the automatic setup and checkout requirement, necessitated implementation techniques employing digital and hybrid (digital-analog) methods. The simulation required the resources of the two digital computers (CDC 1700 and EAI-640) and the two solid-state analog computers (EIA-680) that are part of the hybrid computation facility at the Nuclear Power Generation Division (NPGD) in Lynchburg, Virginia.

Calculational speed requirements and consideration of future machine requirements resulted in the use of a hardware floating point array processor (AP-120B) that performs calculations for the CDC-1700, and thereby greatly increases the amount of the simulation that can be programmed on the digital computers.

The requirement for flexibility in the simulation of the control system resulted in the application of Interactive Simulation Language[1], which allows a digital computer to be programmed like an analog computer and allows convenient modification of control system parameters.

The desire for basically "hands-off" operation resulted in the specification of a "mini" control panel that allows automatic plant simulation but also allows the user interactive capability for controlling those portions of the simulation that are similar to a reactor operator's role of controlling an actual plant in the manual mode. In addition, this control panel allows the user to initiate many operational transients. The mini control panel was physically constructed on one of the analog computers and consists of on-off and momentary lighted pushbuttons. The status of all the pushbuttons is scanned by the EAI-640 computer; certain signals are then forwarded to the analog computers, while others are sent to the CDC-1700 via a digital data link between the digital computers.

Figures 1-1 and 1-2 are schematic diagrams of the power plant. With respect to these diagrams, the implementation of the major components on the various computers is as follows:

1. CDC-1700, AP-120B — Reactor, control rod drive, pressurizer, primary system flows and temperatures, condensate system flows, feedwater and condensate system energy balance, and turbine extraction flows.
2. EAI-640 — Control system (ICS), protection system, atmospheric dump valves, bypass valves, and steam relief valves.
3. EAI-680, Console 1 — Steam generators and portion of steam line.
4. EAI-680, Console 2 — Feedwater system flows, remainder of steam lines, feedwater control valves, feedwater pumps, and turbine-generator.

The analog computer portions of the simulation are set up and checked out automatically using COMANCHE, a program developed by Control Data Corporation for the CDC 1700 computer, COMANCHE is a system routine (resident on the CDC 1700) which sets potentiometers on the analog consoles and then performs a "static check" of the patchboards and analog components before performing production runs. All digital computer routines are either resident in the computer system or are read in from binary paper tapes.

Figure 1-1. Primary Loop Diagram

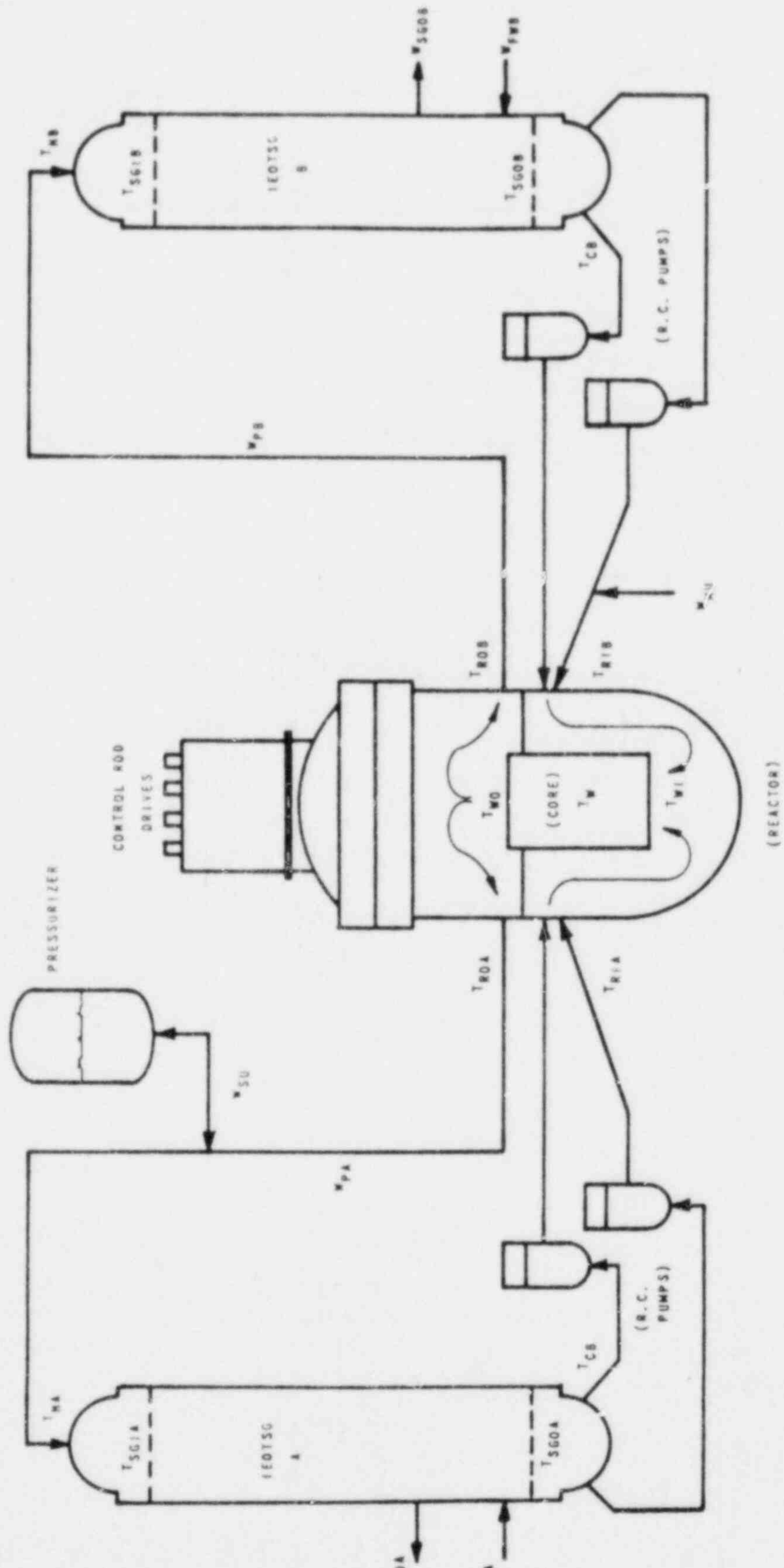
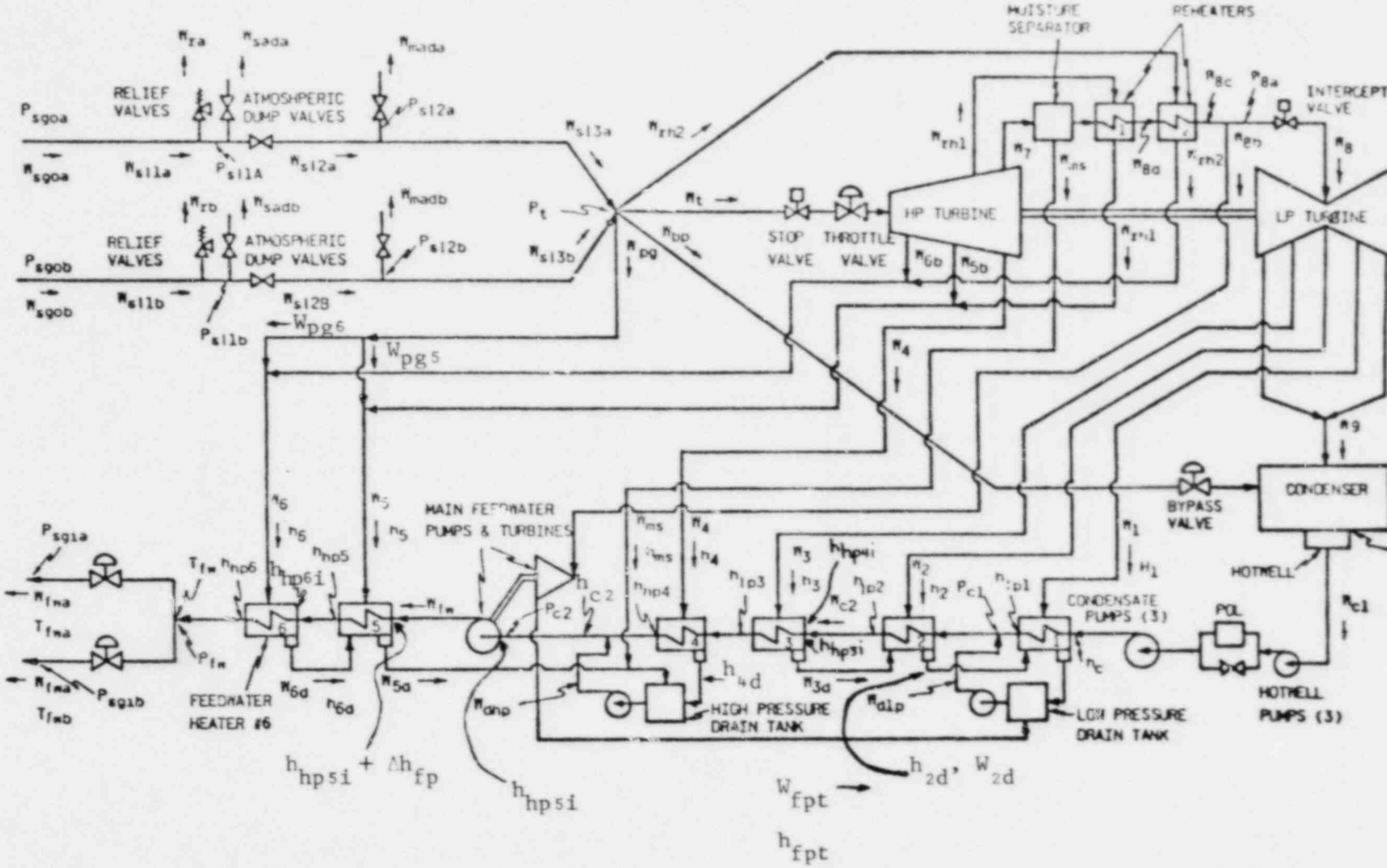


Figure 1-2. Secondary System Diagram



2. MATHEMATICAL MODEL

This section summarizes the equations employed by the POWER TRAIN simulation. Modeling assumptions are stated and the equations are presented with minimal discussion of the details of implementation. Solution methods are identified and pertinent derivations are given. Models are discussed on a component by component basis starting with components in the primary, continuing with the steam generator, and concluding with secondary components. The nomenclature for each subsection corresponds to the variable names used by the computer coding and are defined at the end of each section. This section concludes with a summary of the limitations of the simulation.

2.1. Scope of Nuclear Power Plant Simulation

The primary system of the nuclear power plant simulated in POWER TRAIN is diagrammed in Figure 1-1. The primary system model, when provided with appropriate input data, can be applied to any Babcock & Wilcox nuclear power plant (177-, 205-, or 145-fuel assembly). It comprises the reactor vessel, two steam generators, hot and cold leg piping, primary coolant pumps, the control and safety rod systems, and the pressurizer. The reactor vessel consists of inlet and outlet plenums and a reactor core with a bypass channel. Point reactor kinetics is used to model the dynamics of core power generation. The primary side of the steam generators consists of inlet and outlet plenums and a tube bundle in which the heat transfer process takes place. The pressurizer is modeled as a non-homogeneous thermodynamic system including the effects of heaters, sprays, wall condensation, and relief valves. Primary flow is modeled as a constant until on-off pushbuttons are actuated to simulate primary coolant pump failures. In those cases, primary flow is determined by flow versus time coastdown curves. All models, including the steam generators, can account for the effects of both forward and reverse flows. Reverse flow might occur if two primary pumps in one loop failed. All primary fluid is assumed to be single-phase subcooled water except in the pressurizer. Instantaneous thermal mixing is assumed to occur in all plenums. Thermal transport in the piping is modeled with a variable transport delay.

The secondary system simulation is diagrammed in Figure 1-2. Included in the steam side of the secondary system are the steam lines connecting the steam generator to the turbine, turbine throttle, bypass and dump valves, and the turbine-generator. The feedwater and condensate side of the secondary system includes the condenser, hotwell, condensate, and feedwater pumps, three low-pressure heaters with a drain tank, three high-pressure heaters with a drain tank, and feedwater control valves for each steam generator. The feedwater system model uses a single feedwater train to represent two cross-coupled parallel trains of heaters and pumps. The arrangement of components in the feedwater system differs widely from plant to plant. The arrangement modeled by POWER TRAIN and shown in Figure 1-2 is based on the Tennessee Valley Authority (TVA) Bellefonte nuclear power plant. In addition, POWER TRAIN provides (as an option) the capability to select a variation of the feedwater system based on the Sacramento Municipal Utility District (SMUD) Rancho Seco nuclear power plant.

The steam in the steam lines is assumed to be of constant enthalpy. Conservation of mass and momentum equations are used to determine the flow and pressure responses in the steam lines.

The main feedwater flow is found by solving a momentum equation assuming constant density for the various sections of the feedwater piping.

The turbine extraction and feedwater heater drain mass flows are functions of throttle flow based on the steady-state plant heat balance information provided in the input data. The enthalpy of all extraction and feedwater heater outlet flows is also a function of throttle flow based on heat balance data. The dynamic behavior of these flows and enthalpies is modeled using a single time constant in a first-order differential equation. That is, when the throttle flow changes, these quantities will apporach their new values asymptotically according to the time constant associated with each extraction flow and feedwater heater.

POWER TRAIN provides the option to model either a once-through steam generator (OTSG) or an integral economizer once-through steam generator (IEOTSG). The

OTSG requires modeling of a downcomer, a chamber in which incoming feedwater is preheated to saturation by recirculated steam before introduction to the tube bundle.

The integrated control system (ICS) is modeled extensively, including the unit load demand development subsystem, the integrated master control subsystem, the steam generator feedwater control subsystem, and the reactor control subsystem. In addition, the operator of the simulation may select, via the mini panel, either manual or automatic control of the following controllers and final control elements:

- Turbine bypass valves.
- Turbine throttle valves.
- Reactor-steam generator demand.
- Feedwater pump speed.
- Feed valve demand (A or B).
- Feed valve position (A or B).
- Reactor demand.
- Control rod position.
- Unit load demand.

Trips and abnormal conditions that may be initiated from the mini panel include he following:

- Manual reactor trip.
- Reactor coolant (RC) pump trips.
- Feedwater pump trips.
- Turbine trip.
- Loss of electrical load.
- Pressurizer heater failure.
- Enable/disable flux/flow trip of reactor.
- Enable/disable high primary pressure trip of reactor.
- Manual failure of drain and condensate pumps in feedwater train.

2.2. Reactor Kinetics and Reactivity Feedback

The point reactor kinetics equations with six groups of delayed neutrons are solved using the prompt jump approximation.

$$n = \frac{\ell^*}{\beta - \delta k_T} \sum_{i=1}^{6} \lambda_i C_i \qquad (2\text{-}1)$$

and

$$\frac{dC_i}{dt} = \frac{\beta_i}{\ell^*} n - \lambda_i C_i. \tag{2-2}$$

Reactivity feedback for fuel and moderator temperature changes are calculated using average feedback coefficients:

$$\delta k_{Dop} = \alpha_D [T_f - T_f(0)], \tag{2-3}$$

$$\delta k_{mod} = \alpha_m [T_w - T_w(0)]. \tag{2-4}$$

The effect of boron addition or dilution is modeled as a positive or negative rate of reactivity change:

$$\frac{d}{dt} \delta k_{bor} = \frac{R_{bor}}{3600} K_b, \tag{2-5}$$

$$K_b = \begin{array}{l} 0 \text{ hold} \\ +1 \text{ borate} \\ -1 \text{ deborate.} \end{array} \tag{2-6}$$

Control and safety rod reactivities are obtained from the control and safety rod models.

The total reactivity is the sum of all these contributions. A shim reactivity is calculated by the simulation to obtain zero initial total reactivity:

$$\delta k_T = \delta k_{Dop} + \delta k_{mod} - \delta k_{bor} - \delta k_r - \delta k_{sr} - \delta k_{shim}. \tag{2-7}$$

Nomenclature

C_i	Precursor concentration of delayed neutrons in group i, neutrons/cm^3
ℓ^*	Mean effective prompt neutron lifetime, s
n	Neutron density, neutrons/cm^3
R_{bor}	Boron reactivity addition (+) rate or dilution (-) rate, δk/h
T_f	Average fuel temperature, F
$T_f(0)$	Initial value of T_f, F
T_w	Average temperature of water in core, F
$T_w(0)$	Initial value of T_w, F
α_D	Doppler coefficient, δk/F

α_m	Moderator coefficient, $\delta k/F$
β	Fraction of fission neutrons which are delayed
β_i	Fraction of fission neutrons which are in delay group i
δk_{bor}	Negative reactivity due to boron addition, $-\delta k$
δk_{Dop}	Reactivity feedback due to fuel temperature, δk
δk_{mod}	Reactivity feedback due to moderator temperature, δk
δk_r	Negative reactivity due to rod insertion, $-\delta k$
δk_{shim}	Negative reactivity offset to give initial total reactivity of zero, $-\delta k$
δk_{sr}	Negative reactivity due to safety rod insertion, $-\delta k$
δk_T	Total reactivity, δk
λ_i	Decay constant for delayed neutron group i, s^{-1}

2.3. Control and Safety Rods

The control system transmits a signal to withdraw, hold, or insert control rods. The rod drive motor is simulated by two first-order differential equations for rod velocity and rod position.

$$\frac{dV_r}{dt} = (K_r V_r^* - V_r)/\tau_m, \tag{2-8}$$

$$\frac{dX_r}{dt} = V_r. \tag{2-9}$$

Control rod reactivity is calculated from linear interpolation of a rod worth versus rod position table included as part of the input data to the simulation:

$$\delta k_r = R_{rm} F_r(\%X_r)/100. \tag{2-10}$$

Safety rod reactivity is obtained from linear interpolation over a table of a safety rod worth versus time after trip. The trip signal is initiated by the reactor protection system model. The table is included in the input data.

$$\delta k_{sr} = R_{srm} F_{sr}(t_{trip})/100 \tag{2-11}$$

Nomenclature

F_r Table of percent rod reactivity versus $\%X_r$, %

F_{sr} Table of percent safety rod insertion versus t_{trip}, %

K_r 1 for rod insertion,
0 for hold,
-1 for rod withdrawal

R_{rm} Maximum total worth of control rods, $-\delta k$

R_{srm} Maximum total worth of safety rods, $-\delta k$

t_{trip} Time after signal to trip reactor, s

V_r Rod velocity, in./s

V_r^* Maximum normal rod speed, in./s

X_r Length of rod inserted, in.

$\%X_r$ Percent of rod inserted, %

δk_r Negative reactivity due to control rod insertion, $-\delta k$

δk_{sr} Negative reactivity due to safety rod insertion, $-\delta k$

τ_m Rod drive time constant, s

2.4. Core Power

The core power is assumed to be directly proportional to the neutron density (section 2.2), provided that some minimum power value (POWMIN) specified in the input data is exceeded:

$$\%Q_f = n(\text{POWERI}) \quad \text{for } \%Q_f > \text{POWMIN}. \tag{2-12}$$

Below this value fission product decay is assumed to heat the fuel according to the following model. The power from the fuel decays exponentially with a single time constant (τ_{dk}) toward a non-zero asymptotic power level (DECAY).

$$\frac{d(\%Q_f)}{dt} = -(\%Q_f - \text{DECAY})/\tau_{dk} \quad \text{for } \%Q_f \le \text{POWMIN} \tag{2-13}$$

Nomenclature

DECAY Asymptotic value of power for decay heat model, % of rated power

n Neutron density expressed as a fraction of initial neutron density, dimensionless

POWERI Initial value of core power, % of rated power

POWMIN Threshold value of POWER below which the decay heat model is assumed to operate, % of rated power

$\%Q_f$ Power being generated in core expressed as a percentage of rated power, % of rated power

τ_{dk} Decay heat time constant, s

2.5. Core Heat Transfer

POWER TRAIN assumes that the average temperatures of the fuel, cladding, and cooling water may be determined from consideration of a single reactor control volume. Conservation of energy equations are written for the fuel, cladding, and coolant, where each region is modeled as a single node:

$$(MC_p)_f \frac{d}{dt} T_f = \%Q_f - (UA)_{fc}(T_f - T_c), \tag{2-14}$$

$$(MC_p)_c \frac{d}{dt} T_c = (UA)_{fc}(T_f - T_c) - (UA)_{cw}(T_c - T_w), \tag{2-15}$$

$$M_w C_{pw} \frac{d}{dt} T_w = (UA)_{cw}(T_c - T_w) - C_{pw} W_p (T_{wo} - T_{wi}). \tag{2-16}$$

Bender's method is used to calculate core outlet temperature, given the average water temperature[2]

$$M_w C_{pw} \frac{d}{dt} T_{wo} = (UA)_{cw}(T_c - T_w) - 2C_{pw} W_p (T_{wo} - T_w). \tag{2-17}$$

The heat transfer rate between cladding and coolant is assumed to be proportional to the 0.8 power of core coolant flow,

$$(UA)_{cw} = (UA)_{cw*}(W_p/W_{p*})^{0.8}. \tag{2-18}$$

The specific heat capacity of the core coolant is a polynomial fit with P_{rc} and T_w as independent variables:

$$C_{pw} = f(P_{rc}, T_w). \tag{2-19}$$

An initial core bypass flow rate may be specified via input to establish a ratio used to calculate core flow as a fraction of total primary system flow:

$$W_p = (W_{pa} + W_{pb})\left(1 - \frac{W_{cbp*}}{W_{pa*} + W_{pb*}}\right) \tag{2-20}$$

Nomenclature

C_{pw}	Specific heat of water, Btu/lbm-°F
M_w	Mass of water in the core, lbm
$(MC_p)_c$	Cladding mass times specific heat, Btu/°F
$(MC_p)_f$	Fuel mass times specific heat, Btu/°F
P_{rc}	Primary system pressure measured at pressure tap at surge line inlet, psia
$\%Q_f$	Power being generated in core expressed as a percentage of rated power, % of rated power
T_c	Average cladding temperature, F
T_f	Average fuel temperature, F
T_w	Average water temperature, F
T_{wi}	Core inlet temperature, F
T_{wo}	Core outlet temperature, F
$(UA)_{cw}$	Cladding-to-water heat transfer rate, % of rated power/°F
$(UA)_{cw*}$	Initial cladding-to-water heat transfer rate, % of rated power/°F
$(UA)_{fc}$	Fuel-to-cladding heat transfer rate, % of rated power/°F
W_{cbp*}	Initial core bypass flow rate, lbm/s
W_p	Flow rate through core, lbm/s
W_{p*}	Initial value of W_p, lbm/s
W_{pa}	Primary flow rate through loop A, lbm/s
W_{pa*}	Initial value of W_{pa}, lbm/s
W_{pb}	Primary flow rate through loop B, lbm/s
W_{pb*}	Initial value of W_{pb}, lbm/s

2.6. Primary System Flow Rates

The primary flow system is simulated as two loops connecting the reactor vessel with two steam generators (see Figure 1-1). In an actual B&W NSS there are two cold legs, each with its own RC pump. POWER TRAIN models these two cold legs as a single cold leg with two pumps in parallel. As discussed in section 2.5, coolant flow through the reactor is modeled by dividing it into core and bypass flows.

The initial and normal primary flows (with all pumps operating) are constants specified in the input data. The capability to simulate four types of pump coastdown is provided:

- One-pump trip.
- Four-pump trip.
- Trip of two pumps in the same loop.
- Trip of two pumps, one in each loop.

Tables for flow versus time for each loop for each of the four cases are included in the input data. Reverse flow may be specified by negative values of flow. The direction of the flow will modify the calculation of loop temperatures appropriately.

2.7. Primary System Temperatures

Temperature response between any two successive points in the system is modeled by a flow-dependent mixing constant, a flow-dependent transport delay, or a set of differential equations characterizing the dynamics of the component between the two successive points.

The model used to calculate temperatures across plenums assumes that all inlet flow mixes instantaneously with fluid already in the plenum and that the outlet temperature is equal to the bulk average temperature. This model is used for the upper and lower plenums of the reactor and the steam generators. The form of the differential equation relating outlet temperature to inlet temperature is

$$\frac{dT_o}{dt} = \frac{\sum\limits_{\substack{\text{all}\\\text{inlets}}} (W_i T_i) - T_o \sum\limits_{\substack{\text{all}\\\text{outlets}}} W_o}{\rho V} \tag{2-21}$$

A flow-dependent transport delay model is used to calculate the temperature of the flow from the hot and cold legs of each loop. The expression for outlet temperature and the time delay is as follows.

$$T_o(t) = T_i(t - \tau) \tag{2-22}$$

$$\tau = \frac{V\rho}{W_i}. \tag{2-23}$$

The dynamic models used for the reactor core and steam generators are described in sections 2.5 and 2.16, respectively.

Nomenclature

T_i Temperature of fluid at inlet to plenum or section of piping, F

T_o Temperature of fluid at outlet of plenum or section of piping, F

V Volume of plenum or section of piping, ft^3

W_i Mass flow rate at inlet of plenum or section of piping, lbm/s

W_o Mass flow rate at outlet of plenum, lbm/s

ρ Density for each plenum or section of piping, input as a constant, lbm/ft^3

τ Delay time, s

2.8. Average Temperature Calculation for ICS

The integrated control system (ICS) requires average temperature for the primary system. This is computed from the hot and cold leg temperatures at the inlet and outlet of the steam generators. The control system selects either loop, and the measured average temperature is then calculated from the following differential equation which approximates the temperature measuring device characteristic:

$$\frac{dT_{avgm}}{dt} = \frac{1}{\tau_{ts}} (T_m - T_{avgm}), \tag{2-24}$$

$$T_m = \frac{1}{2} (T_{ha} + T_{ca}) \quad \text{if loop A is selected,} \tag{2-25}$$

$$T_m = \frac{1}{2} (T_{hb} + T_{cb}) \quad \text{if loop B is selected.} \tag{2-26}$$

Nomenclature

T_{avgm} Measured value of primary average temperature, F

T_{ca} Temperature of loop A cold leg, F

T_{cb} Temperature of loop B cold leg, F

T_{ha} Temperature of loop A hot leg, F

T_{hb} Temperature of loop B hot leg, F

T_m Actual value of primary average temperature, F

τ_{ts} Time constant of temperature measurement system, F

2.9. Primary System Pressure

A single pressure is used in the primary system model and is defined as the pressure that exists at the surge line connection to the loop A hot leg. The change in system pressure is calculated by considering the effects of thermal expansion of the primary fluid and net mass addition to the primary system, excluding the pressurizer. The cumulative effect of thermal expansion is considered for all subvolumes of the primary system.

$$\frac{dP_{rc}}{dt} = \frac{W_{mu} - W_{\ell d} - W_{sp} - W_{su} - \sum_i V_i \left(\frac{\partial\rho}{\partial T}\right)_P \frac{dTi}{dt}}{V_{rc}\left(\frac{\partial\rho}{\partial P}\right)_T} \tag{2-27}$$

Nomenclature

P_{rc}	Primary system pressure measured at pressure tap at surge line inlet, psia
T_i	Arithmetic mean of temperature of fluid at the inlet and outlet of subvolume i, F
V_i	Volume of subvolume i of the primary system, ft^3
V_{rc}	Total volume of the primary system excluding the pressurizer, ft^3
$W_{\ell d}$	Letdown flow rate out of the primary system, lbm/s
W_{mu}	Makeup flow rate into the primary system, lbm/s
W_{sp}	Spray flow rate into pressurizer, lbm/s
W_{su}	Surge flow rate into pressurizer, lbm/s
$(\partial\rho/\partial T)_P$	Function of temperature T_i defined in POWER TRAIN coding, lbm/ft^3-°F
$(\partial\rho/\partial P)_T$	Function of pressure P_{rc} defined in POWER TRAIN coding, lbm/ft^3-psi

2.10. Surge Line Flow

The surge line flow model assumes that the pressure drop across the surge line is due to elevation, friction, and shock losses. The following equation is solved for W_{su}:

$$P_{rc} = P_p + \Delta P_{es\ell} + K_{fs\ell} |W_{su}| W_{su}. \tag{2-28}$$

Nomenclature

$K_{fs\ell}$ Equivalent dynamic ΔP coefficient in surge line, psi/(lbm/s)^2

$\Delta P_{es\ell}$ Elevation ΔP for height difference between surge line connection and pressurizer, psi

P_p Pressurizer pressure, psia

P_{rc} Primary system pressure measured at pressure tap at surge line inlet, psia

W_{su} Surge flow rate into pressurizer, lbm/s

2.11. Pressurizer

The pressurizer is modeled with a lower liquid region, an upper liquid region, and a vapor region. Each region may have a different temperature, but all are characterized by the same pressure. The pressure is determined by the density and enthalpy of the vapor region.

The pressurizer model includes the effects of sprays, heaters, surge flows, pilot valves, relief valves, and the makeup system, whose flow is regulated by the calculated liquid level in the pressurizer.

The mass and energy transfer effects modeled include

- Mixing of upper and lower liquid regions
- Evaporation and boiling of upper region liquid to vapor
- Condensation of vapor on upper liquid region surface
- Condensation of vapor on walls
- Condensation of vapor on spray

All these effects are modeled as surface phenomena.

2.11.1. Mass Balance

During steady-state operation there is only one liquid region in the pressurizer. This liquid makes up what is called the upper region. It is assumed that during an insurge the incoming fluid will not mix instantly with the fluid already in the pressurizer. A new region, called the lower region, is created at this time. The mass of the lower region will decay exponentially as it mixes with the liquid in the upper region. Therefore, for a single insurge, two mass balances evolve that will decay into a single region as steady state is approached. An outsurge will drain the lower region before the upper region mass is affected.

Whenever there is an insurge or whenever the mass of the lower liquid region has not yet decayed to zero, the mass balance for the lower liquid region is given by

$$\frac{dM_{pw1}}{dt} = W_{su} - \lambda_{pw1} M_{pw1}. \tag{2-29}$$

Mass flow from sprays, condensation on the wall, and condensation on the sprays are all assumed to go directly to the upper liquid region. Therefore, the mass balance for the upper region is given by

$$\frac{dM_{pw2}}{dt} = \lambda_{pw1} M_{pw1} + W_{su} B_{139} \overline{B_{140}} + W_{c4} + W_{c5} + W_{c6} - W_b + W_{sp}. \tag{2-30}$$

The mass balance for the vapor region is given by

$$\frac{dM_{ps}}{dt} = W_b - W_{c4} - W_{c5} - W_{c6} - W_r - W_{pr}. \tag{2-31}$$

2.11.2. Energy Balance

The conservation of energy equation can be written for each of the three regions in the pressurizer. It is assumed that there is no heat transfer with walls or across interfaces in the lower liquid region. For an insurge the flow enters at the hot leg enthalpy, and for an outsurge the flow exits at the bulk average enthalpy of the lower liquid region. When fluid from the lower liquid region mixes with that from the upper liquid region, its enthalpy is assumed to be equal to the bulk average enthalpy of the lower liquid region.

$$\frac{dh_{pw1}}{dt} = \left[(h_{HL} B_{138} + h_{pw1} B_{139}) W_{su} - M_{pw1} \lambda_{pw1} h_{pw1} - h_{pw1} \frac{d}{dt} M_{pw1}\right] \frac{B_{140}}{M_{pw1}}. \tag{2-32}$$

Heat is added by the pressurizer heaters in the upper liquid region. Liquid is boiled off at the specific enthalpy of saturated vapor. Spray flow is assumed to be heated to saturation by the vapor and is added directly to the upper liquid region along with the condensate on the spray, all at the specific enthalpy of saturated liquid. Condensation on the vapor-liquid interface adds not only condensate but also the heat of vaporization to the upper liquid region. Condensation on the walls adds liquid to the upper liquid region at the specific enthalpy of saturated liquid. The heat of vaporization from condensation on the walls is assumed to be taken up by the wall and lost. An outsurge affects the upper liquid region only when there is no mass in the lower liquid region, in which case the outsurge occurs at the bulk average enthalpy of the upper liquid region. The insurge will be added to the lower liquid region and mix with the upper liquid region according to a mixing factor:

$$\frac{dh_{pw2}}{dt} = \Big[Q_{htr} - W_b h_g + (W_{c4} + W_{sp}) h_f + W_{c5} h_s + W_{c6} h_f + W_{su} h_{pw2} B_{139} \overline{B_{140}} + \lambda_{pw1} M_{pw1} n_{pw1} - h_{pw2} \frac{dM_{pw2}}{dt}\Big] / M_{pw2}. \tag{2-33}$$

Specific enthalpy in the vapor region is increased when there is boiling of the upper liquid region and is decreased through condensation on walls, sprays, and the surface of the upper liquid region. Energy is lost when mass is vented through the pilot and relief valves.

$$\frac{dh_s}{dt} = [W_b h_g - (W_{c4} + W_r + W_{pr} + W_{c5} + W_{c6} + \dot{M}_{ps}) h_s + V_{ps} \dot{P}] / M_{ps}. \tag{2-34}$$

By solving the conservation of mass for the vapor region mass, W_b, and substituting the result in the energy equation, a substantially simplified energy equation is obtained:

$$\frac{dh_s}{dt} = \frac{W_b (h_g - h_s)}{M_{ps}} + \frac{V_{ps}}{M_{ps}} \dot{P}. \tag{2-35}$$

2.11.3. Mass and Energy Transfer Processes

The pressurizer pressure is a function of the local density and enthalpy for a positive level in the pressurizer. Should the pressurizer empty, the pressure becomes a function of only the hot leg temperature.

Upper liquid region boiling occurs when the liquid enthalpy exceeds the saturation enthalpy corresponding to the pressurizer pressure. It is assumed that the boiling rate is a linear function of the difference between the local and saturation enthalpies.

$$W_b = K_{2,6}(h_{pw2} - h_f)B_{142} \quad (2\text{-}36)$$

The condensation rate of vapor on the spray depends on the saturation enthalpy, the spray inlet enthalpy (which is the enthalpy of the cold leg), and the spray flow rate. It is assumed that all spray flow is heated to the saturation temperature by condensation of vapor on the spray. This assumption implies that the condensation rate is such that the heat of vaporization given up is sufficient to raise the enthalpy of the spray from the inlet value to the saturation value.

$$W_{c4} = \frac{W_{sp}(h_f - h_{CL})}{h_s - h_f} \quad (2\text{-}37)$$

The condensation rate of vapor on the liquid surface and on the walls is limited by the heat of vaporization removal rate. It is assumed that this heat is removed in proportion to the ΔT between the surface and the bulk vapor temperature. The liquid surface temperature is assumed to be equal to the bulk liquid temperature of the upper liquid region. The wall temperature and the heat transfer factors (UA) are constants included in the input data.

$$W_{c5} = \overline{B_{142}}(T_{ps} - T_{pw2})[(UA)_{ps}]/(h_s - h_f) \quad (2\text{-}38)$$

$$W_{c6} = \overline{B_{143}}(T_{ps} - T_{pm2})[(UA)_{pm2}]/(h_s - h_f) \quad (2\text{-}39)$$

2.11.4. State Relations and Water Properties

To determine the state of each pressurizer region, the following additional relations are needed together with functions for water properties.

The volume of the liquid region is the sum of the volume of the two lower regions:

$$V_{pw} = \frac{M_{pw1}}{\rho_{pw1}} + \frac{M_{pw2}}{\rho_{pw2}}. \quad (2\text{-}40)$$

The volume of the steam region may then be found by

$$V_{ps} = K_{2,5} - V_{pw}. \quad (2\text{-}41)$$

The density of the steam region is

$$\rho_{ps} = M_{ps}/V_{ps}. \quad (2\text{-}42)$$

The water properties needed by the pressurizer model are expressed as polynomial fits to the ASME Steam Tables.

$$T_{sat} = f_{201}(h_f) \tag{2-43}$$

$$h_f = f_{202}(P) \tag{2-44}$$

$$h_g = f_{203}(P) \tag{2-45}$$

$$\rho_f = f_{204}(P) \tag{2-46}$$

$$h_f = f_{207}(T) \tag{2-47}$$

$$T = f_{208}(v,h) \tag{2-48}$$

$$P = f_{209}(v,h) \tag{2-49}$$

Nomenclature

Symbols beginning with B are logical variables, which have a value of either 1 or 0 to correspond with true or false. In equations, a vinculum is used over logical variables to represent the logical inverse.

B_{138}	= 1 when $W_{su} > 0$, = 0 otherwise
B_{139}	= 1 when $W_{su} \leq 0$, = 0 otherwise
B_{140}	= 1 when $M_{pw1} > 0$, = 0 otherwise
B_{142}	= 1 when $h_{pw2} > h_f$, = 0 otherwise
B_{143}	= 1 when $T_{ps} > T_{pm2}$, = 0 otherwise
e	Specific internal energy, Btu/lbm
h	Specific enthalpy, Btu/lbm
h_{CL}	Specific enthalpy of water in cold leg, Btu/lbm
h_f	Specific enthalpy of liquid at saturation, Btu/lbm
h_g	Specific enthalpy of vapor at saturation, Btu/lbm
h_{HL}	Specific enthalpy of water in hot leg, Btu/lbm

h_{pw1} Specific enthalpy of water in lower liquid region of pressurizer, Btu/lbm

h_{pw2} Specific enthalpy of water in upper liquid region of pressurizer, Btu/lbm

h_s Specific enthalpy of vapor in the steam region of pressurizer, Btu/lbm

$K_{2.5}$ Total pressurizer volume, ft^3

$K_{2.6}$ Equivalent coefficient for boil-off in pressurizer, lb/s-°F

M Total mass in a control volume, lbm

M_{ps} Total mass in steam region of pressurizer, lbm

M_{pw1} Total mass in lower liquid region of pressurizer, lbm

M_{pw2} Total mass in upper liquid region of pressurizer, lbm

p Pressure, psia

Q_{htr} Heater power, Btu/s

T_{ps} Temperature of steam region of pressurizer, F

T_{pw1} Temperature of lower liquid region of pressurizer, F

T_{pw2} Temperature of upper liquid region of pressurizer, F

$(UA)_{ps}$ Steam-to-liquid condensation heat transfer rate, Btu/s-°F

$(UA)_{pm2}$ Steam-to-wall condensation heat transfer rate, Btu/s-°F

v Specific volume, ft^3/lbm

V_{ps} Volume of steam region in the pressurizer, ft^3

V_{pw} Volume of both liquid regions in the pressurizer, ft^3

W_b Rate of boil-off from upper liquid region to steam region of pressurizer, lbm/s

W_{c4} Rate of condensation of vapor onto spray droplets in pressurizer, lbm/s

W_{c5} Rate of condensation of steam on liquid surface in pressurizer, lbm/s

W_{c6} Rate of condensation of steam on pressurizer wall, lbm/s

W_{pr} Flow rate through power-operated relief valve, lbm/s

W_r	Flow rate through Code Safety valve, lbm/s
W_{sp}	Spray flow rate, lbm/s
W_{su}	Surge flow rate, lbm/s
λ_{pwl}	Mixing factor for lower and upper liquid regions of the pressurizer, s^{-1}
ρ_{ps}	Density of steam in the steam region of pressurizer, lbm/ft^3

2.12. Pressurizer Heaters

Four banks of heaters are modeled; their total energy output is the sum of the contributions from each one. The first bank is operated proportionally, the remaining three are on-off controlled with different cut-on and cut-off temperatures and power capabilities. A 30-second time constant is associated with each heater to account for heat storage in the heater. In addition, all heaters are cut off should pressurizer level fall below the level specified by the input data.

2.13. Pressurizer Sprays

The spray valve opens and closes at a fixed velocity. The opening and closing pressure setpoints and the velocity are constants specified by the input data. Spray flow is given by the following equation, where $f(N_{rcp})$ is a function of the number of reactor coolant pumps in operation:

$$W_{sp} = K_{2.33} X_{sp} f(N_{rcp})/100 \qquad (2\text{-}50)$$

$$\frac{dX_{sp}}{dt} = \begin{cases} K_{2.42} & \text{if } P_p \geq K_{2.34} \\ 0 & \text{if } K_{2.35} < P_p < K_{2.34} \\ -K_{2.42} & \text{if } P_p \leq K_{2.35} \end{cases} \qquad (2\text{-}51)$$

Nomenclature

$f(N_{rcp})$	Function of number of reactor coolant pumps in operation, dimensionless
$K_{2.33}$	Maximum spray flow rate, lbm/s
$K_{2.34}$	Spray cut-on pressure, psia
$K_{2.35}$	Spray cut-off pressure, psia

$K_{2.42}$	Pressurizer spray valve velocity, % stroke/s
P_p	Pressurizer pressure, psia
W_{sp}	Spray flow rate, lbm/s
X_{sp}	Spray valve position limited to value between 0 and 100%, % stroke

2.14. Pressurizer Code Safety and Power-Operated Relief Valves

The Code Safety valve opens and closes immediately upon actuation. The power-operated relief valve ramps open and closed at a rate of 200% per second. The opening and closing pressures and the flow capacity of each valve are specified by the input data.

2.15. Makeup System

A level is calculated in the pressurizer according to the following equation:

$$L_p = K_{2.3}V_{pw} + K_{2.4}. \tag{2-52}$$

A proportional-integral controller monitors this pressurizer level and controls the makeup flow demand. The flow follows the demand with a first-order lag response. Flow is constrained to be less than an input constant, $K_{2.31}$.

$$\frac{dW_{mu}}{dt} = \frac{(W_{mu})_D - W_{mu}}{\tau_{mu}} \tag{2-53}$$

$$(W_{mu})_D \stackrel{\Delta}{=} K_{2.27}[K_{2.28}(K_{2.29} - L_p) + K_{2.30} \int (K_{2.29} - L_p)dt] \tag{2-54}$$

Nomenclature

$K_{2.3}$	Level to volume equivalent in cylindrical section of pressurizer, in./ft^3
$K_{2.4}$	Level equivalent of pressurizer volume below lower level sensing tap, in.
$K_{2.27}$	Conversion factor, level error to makeup flow demand, lbm/s-in.
$K_{2.28}$	Proportional gain, level error, dimensionless
$K_{2.29}$	Pressurizer level setpoint, in.
$K_{2.30}$	Integral gain, level error, s^{-1}

$K_{2,31}$ Maximum makeup flow rate, lbm/s

L_p Pressurizer level, in.

V_{pw} Volume of both liquid regions in the pressurizer, ft^3

W_{mu} Makeup flow rate, lbm/s

$(W_{mu})_D$ Makeup flow demand, lbm/s

τ_{mu} Makeup flow time constant, s

2.16. Steam Generator

Two steam generators provide the interface between the primary and secondary flow loops of the power plant by transferring heat from the primary to the secondary side. Primary water flows from the reactor into the upper plenum of the steam generators and is cooled as it flows through the steam generator tubes to the lower plenum and is pumped back to the reactor. Secondary fluid is pumped from the feedwater train into the bottom of the tube bundle and is successively heated, boiled, and superheated as it flows around the tubes until it exits the top of the tube bundle through an annulus into the steam lines.

POWER TRAIN provides the option to model either an OTSG or an IEOTSG. The term integral economizer refers to the fact that the feedwater enters the tube bundle in a subcooled state and is heated to saturation in the economizer section of the steam generator (typically, the first 8 feet of the tube bundle). The designation OTSG is used here to refer to a once-through steam generator in which feedwater is preheated to saturation by recirculated steam in the downcomer before being introduced into the tube bundle.

The model for the steam generators is described by nonlinear partial differential equations (PDE) involving both time and one spatial dimension. The boundary conditions required by the model are as follows:

1. Primary flow rate.
2. Temperature of fluid at primary inlet.
3. Secondary inlet flow rate.
4. Temperature of fluid at secondary inlet.
5. Pressure at secondary outlet.

The following modeling assumptions are made:

1. The flow areas on both primary and secondary sides are constant.
2. The primary fluid can be considered to be incompressible; hence, there is no flow variation in the channel.
3. Axial heat conduction in the tube metal is negligible when compared to that in the radial direction.
4. The secondary fluid state may be evaluated as a function of local enthalpy at an average pressure during a time step.
5. The overall dynamic behavior of the IEOTSG can be represented by an average channel.

A complete solution of the equations modeling steam generator dynamics consists of calculating the following variables as a function of time and the one spatial dimension modeled:

1. Primary side flow rate.
2. State of fluid on primary side.
3. Temperature of tube metal.
4. Secondary side flow rate.
5. State of fluid on secondary side.

The primary and secondary energy balances are coupled via the tube metal temperature distribution. The primary fluid temperature is determined as an intial value problem from flow inlet to outlet for each time step. Then the secondary fluid temperature is calculated and the metal temperature is updated. The steam generator equations are solved using the continuous space discrete time (CSDT) method, where finite differencing is used for the time derivatives and continuous integration is used for the spatial derivatives.

2.16.1. Primary Side Model

By assumption 2 above, the flow rate on the primary side of the steam generator is constant in space. This flow rate, which is calculated by the primary system flow model (see section 3.6), may be in either the normal or reverse direction. The steam generator model can accommodate either case.

The primary side fluid is assumed to always be subcooled single-phase liquid. In general, the state of the fluid can be specified by specifying its pressure and temperature. Therefore, the conservation of energy equation for the primary fluid gives us the following PDE for the primary temperature:

$$\frac{\partial T_p}{\partial t} + \frac{W_p}{\rho_p A_p} \frac{\partial}{\partial x} T_p = \frac{P_p U_{pm}}{\rho_p A_p C_p} (T_m - T_p). \qquad (2\text{-}55)$$

2.16.2. Tube Metal Model

The effect of metal conductivity is included in the values of U_{pm} and U_{sm}. By assumption 3 above, we neglect axial heat conduction along the tubes and write the following energy equation for the tube metal:

$$\frac{\partial T_m}{\partial t} = \frac{P_p U_{pm}}{\rho_m C_m A_m} (T_p - T_m) + \frac{P_s U_{sm}}{\rho_m C_m A_m} (T_s - T_m). \qquad (2\text{-}56)$$

2.16.3. Secondary Side Model

The modeling of the secondary side is complicated by the fact that the fluid changes state from subcooled liquid to a boiling mixture and finally to superheated steam as it flows from the bottom to the top of the tube bundle. By assumption 5 we will assume (for the purposes of evaluating state properties) that the pressure, P_{sg}, is the same for the entire length of the tube bundle and is equal to the pressure at the secondary outlet:

$$P_{sg} = P_{sgo}. \qquad (2\text{-}57)$$

P_{sgo} is calculated by the steam line model (see section 2.18) and is provided as a boundary condition to the steam generator model. This assumption of constant pressure along the length of the channel implies that the state of the fluid is specified when enthalpy along the channel is specified.

With respect to calculating flow and density, the secondary side of the steam generator can be divided into three regions according to the state of the secondary fluid: the subcooled region, the boiling region, and the superheated region. The boundaries of these regions may change from one time step to the next. The boundary between the subcooled and boiling regions occurs at the height where

$$h_s = h_f(P_{sg}) \quad \text{at } z = z_{sc}. \qquad (2\text{-}58)$$

The boundary between the boiling mixture and superheated vapor occurs at the height where

$$h_s = h_g(P_{sg}) \quad \text{at } z = z_b. \qquad (2\text{-}59)$$

The pressure drop across the steam generator is calculated using a pressure balance. Kinetic energy and inertial terms are assumed to be negligible. The pressure at the inlet of each steam generator is calculated for use by the feedwater system model and is not used by any other equation in the steam generator model, i.e.,

$$P_{sgi} = P_{sgo} + \Delta P_g + \Delta P_f. \tag{2-60}$$

The gravity head is the spatial integral of density,

$$\Delta P_g = \frac{g}{144 g_c} \int_o^L \rho_s dz. \tag{2-61}$$

The shock and friction losses associated with each region are modeled using a nominal pressure gradient for a reference value of flow. The pressure gradient is modeled as being proportional to the square of the actual flow. The integral of the pressure gradient is evaluated as follows:

$$\Delta P_f = \int_o^{z_{sc}} \left(\frac{\Delta P}{\Delta \ell}\right)_{sc} \frac{W_s |W_s|}{W_o^2} dz + \int_{z_{sc}}^{z_b} \left(\frac{\Delta P}{\Delta \ell}\right)_b \frac{W_s |W_s|}{W_o^2} dz + \int_{z_b}^{L} \left(\frac{\Delta P}{\Delta \ell}\right)_{sh} \frac{W_s |W_s|}{W_o^2} dz. \tag{2-62}$$

The flow on the secondary side is found by integrating the continuity equation, which is usually written in the following form:

$$\frac{1}{A_s} \frac{\partial}{\partial z} W_s + \frac{\partial}{\partial t} \rho_s = 0. \tag{2-63}$$

Since $\rho_s = \rho_s(h_s, P_{sg})$, an equivalent form of the continuity equation is

$$\frac{\partial}{\partial z} W_s = -A_s \left(\frac{\partial \rho_s}{\partial h_s}\right)_P \frac{\partial}{\partial t} h_s - A_s \left(\frac{\partial \rho_s}{\partial P_{sg}}\right)_h \frac{\partial}{\partial t} P_{sg}. \tag{2-64}$$

The derivatives of density, pressure, and enthalpy are given by the state relations, the boundary conditions, and the conservation of energy equation, respectively. With the assumption that $\partial P_{sg}/\partial t$ is small in comparison with Q_s, the energy equation becomes

$$\frac{\partial}{\partial t} \rho_s h_s + \frac{1}{A_s} \frac{\partial}{\partial z} W_s h_s - Q_s = 0. \tag{2-65}$$

Combine equation 2-65 with the conservation of mass equation 2-63 to obtain the form of the energy equation used to calculate the enthalpy along the channel:

$$\rho_s \frac{\partial}{\partial t} h_s + \frac{W_s}{A_s} \frac{\partial}{\partial z} h_s = Q_s. \qquad (2\text{-}66)$$

2.16.4. Heat Transfer Relations

Heat transfer relations are needed by the energy equation for the primary side, the tube metal, and the secondary side of the steam generator. Half of the thermal resistivity of the tube metal is combined with the primary film heat transfer coefficient, and the other half is combined with the secondary film heat transfer coefficient.

The metal-to-primary heat transfer is modeled with a single constant overall heat transfer coefficient (U_{pm} provided as input data):

$$Q_p = \frac{P_p}{A_p} U_{pm}[T_m(z,t) - T_p(z,t)]. \qquad (2\text{-}67)$$

The model for the metal-to-secondary heat transfer is more complex. The heat transfer equation is the same, but the form of the heat transfer coefficient is more complicated:

$$Q_s = \frac{P_s}{A_s} U_{sm}[T_m(z,t) - T_s(z,t)]. \qquad (2\text{-}68)$$

Modeling the metal-to-secondary heat transfer requires division of the secondary side into three regions to represent three different heat transfer processes that occur in each region: subcooled, nucleate boiling, and superheat. The boundary between these regions will change from one time step to the next. The boundary between subcooled and nucleate boiling is at the height where

$$h_s = h_g(P_{sg}). \qquad (2\text{-}69)$$

The boundary between nucleate boiling and superheat is at the height where

$$X_s \geq \text{DNBX}. \qquad (2\text{-}70)$$

DNBX is an input constant representing that quality value above which film boiling is expected to occur.

For the subcooled region the heat transfer coefficient is a weighted sum of a forced convection term and the nucleate boiling coefficient

$$U_{sm} = K_{sc}W_s(z,t) + 0.5\ U_b. \tag{2-71}$$

The nucleate boiling coefficient is based on the model developed by Thom[3] and is expressed as

$$U_{sm} = U_b \tag{2-72}$$

$$U_b = K_1\phi/(T_m - T_s) \tag{2-73}$$

$$\phi = \frac{k}{C_t}\ (T_m - T_{sat} - \Delta T_{sat}) \tag{2-74}$$

$$C_t = D_o \ell n[2\ D_o/(D_o + D_i)]/2 \tag{2-75}$$

$$\Delta T_{sat} = \frac{4.32\ \phi^{0.5}}{e^{P/1260}}\ . \tag{2-76}$$

The forced convection coefficient for the superheated region is expressed as

$$U_{sm} = K_{sh}W_s(z,t). \tag{2-77}$$

2.16.5. State Relations

The following state relations are approximated by polynomials whose coefficients are built into the simulation:

$$h_f = f(P_{sg}) \tag{2-78}$$

$$h_g = f(P_{sg}) \tag{2-79}$$

$$T_{sc} = f(h_s) \tag{2-80}$$

$$T_{sh} = f(P_{sg},h_s) \tag{2-81}$$

$$T_{sat} = f(P_{sg}) \tag{2-82}$$

$$\rho_f = f(P_{sg}) \tag{2-83}$$

$$\rho_g = f(P_{sg}) \tag{2-84}$$

$$\rho_{sc} = f(T_{sc}) \tag{2-85}$$

$$\rho_{sh} = f(T_{sc}) \tag{2-86}$$

The evaluation of density for the boiling region is based on the "drift flux" two-phase void fraction correlation proposed by N. Zuber.[4] The following equations are solved for ρ_s in the boiling region:

$$\rho_s = \rho_f - \alpha(\rho_f - \rho_g) \tag{2-87}$$

$$\alpha = \frac{X}{C_o\left(\frac{\rho_g}{\rho_f} + \frac{\rho_f - \rho_g}{\rho_f}X\right) + \frac{A_s \rho_g \bar{V}_{gj}}{W_f}} \tag{2-88}$$

$$X = \frac{h - h_f}{h_g - h_f} \tag{2-89}$$

$$C_o = 1.0 \tag{2-90}$$

$$\bar{V}_{gj} = \left(1 - \frac{\Delta X}{1 - X_c}\right) 1.41 \left[\frac{\sigma_{ggc}(\rho_f - \rho_g)}{\rho_f^2}\right]^{1/4} \tag{2-91}$$

$$\Delta X = \begin{cases} X - X_c & \text{for } X \geq X_c \\ 0 & \text{for } X < X_c \end{cases} \tag{2-92}$$

$$W_f = W_{fw} - A_s \frac{d}{dt} \int_0^{L_{sc}} (\rho_f + \rho_{sc}) dz \tag{2-93}$$

Derivatives of density with respect to pressure and enthalpy are required to solve the conservation of mass equation (2-64). In the subcooled liquid and superheated steam regions these derivatives are approximated as constants calculated from input data.

$$\left(\frac{\partial \rho_s}{\partial P_{sg}}\right)_{sc} = 0 \tag{2-94}$$

$$\left(\frac{\partial \rho_s}{\partial P_{sg}}\right)_{sh} = \text{DRHODP} \tag{2-95}$$

$$\left(\frac{\partial \rho_s}{\partial h_s}\right)_{sc} = \frac{R1 - R2}{H1 - F2} \tag{2-96}$$

$$\left(\frac{\partial \rho_s}{\partial h_s}\right)_{sh} = \frac{R4 - R5}{G2 - H4} \tag{2-97}$$

Nomenclature

DRHODP	$\partial\rho/\partial P$ at enthalpy of 1234.7, lbm/ft^3-psi
R1	Subcooled density at FW1 and P2, lbm/ft^3
R2	Saturated fluid density at T2 and P2, lbm/ft^3
F2	Saturated fluid enthalpy at P2, Btu/lbm
H1	Subcooled enthalpy at FW1 and P2, Btu/lbm
R4	Saturated gas density at T2 and P2, lbm/ft^3
R5	Superheated density at T4 and P2, lbm/ft^3
H4	Superheated enthalpy at T4 and P2, Btu/lbm
G2	Saturated gas enthalpy at P2, Btu/lbm
FW1	Minimum feedwater temperature, F
P2	Expected normal pressure, psia
T2	Saturation temperature at P2, psia
T4	Expected full power steam temperature, F

The calculation of the density derivatives is more involved in the boiling region. The expressions for each derivative can be derived by differentiating the expression for density given by equations 2-87 through 2-93. The $\partial\rho/\partial h$ for the boiling region alculated by the following set of equations:

$$\left(\frac{\partial \rho}{\partial h}\right)_P = -(\rho_f - \rho_g)\left(\frac{\partial \alpha}{\partial h}\right)_P \tag{2-98}$$

$$\left(\frac{\partial \alpha}{\partial h}\right)_P = \frac{1}{V^2}\left[V - C_o X\left(\frac{\rho_f - \rho_g}{\rho_f}\right)\right]\left(\frac{\partial X}{\partial h}\right)_P \tag{2-99}$$

$$\left(\frac{\partial X}{\partial h}\right)_P = \frac{1}{h_g - h_f} \tag{2-100}$$

$$V = C_o\left(\frac{\rho_g}{\rho_f} + \frac{\rho_f - \rho_g}{\rho_f} X\right) + \frac{A_s \rho_g \bar{V}_{gj}}{W_f}. \tag{2-101}$$

The $\partial\rho/\partial P$ for the boiling region is calculated by the following set of equations:

$$\left(\frac{\partial\rho}{\partial P}\right)_h = \frac{d\rho_f}{dP} - \alpha\,\frac{d}{dP}\,(\rho_f - \rho_g) - (\rho_f - \rho_g)\left(\frac{\partial\alpha}{\partial P}\right)_h \quad (2\text{-}102)$$

$$\left(\frac{\partial\alpha}{\partial P}\right)_h = \frac{1}{V^2}\left[\left(\frac{\partial X}{\partial P}\right)_h V - X\left(\frac{\partial V}{\partial P}\right)_h\right] \quad (2\text{-}103)$$

$$\left(\frac{\partial X}{\partial P}\right)_h = -\left(\frac{1}{h_g - h_f}\right)\left[X\,\frac{dh_g}{dP} + (1 - X)\,\frac{dh_f}{dP}\right] \quad (2\text{-}104)$$

$$\left(\frac{\partial V}{\partial P}\right)_h = C_o\left[\frac{d}{dP}\left(\frac{\rho_g}{\rho_f}\right) + \frac{\rho_f - \rho_g}{\rho_f}\left(\frac{\partial X}{\partial P}\right)_h + X\,\frac{d}{dP}\,\frac{\rho_f - \rho_g}{\rho_f}\right] + \frac{\rho_g A_s}{W_f}\left(\frac{\partial \bar{V}_{gj}}{\partial P}\right)_h \quad (2\text{-}105)$$

<u>Nomenclature</u>

A	Cross-sectional area, ft^2
C	Specific heat, Btu/lbm-°F
C_o	Distribution parameter, dimensionless
C_t	Equivalent diameter, ft
D_i	Tube inside diameter, ft
D_o	Tube outside diameter, ft
DNBX	Quality at which DNB occurs, dimensionless
g	Gravitational acceleration, ft/s^2
g_c	Gravitational conversion constant, $lbm\text{-}ft/lbf\text{-}s^2$
h	Specific enthalpy, Btu/lbm
h_f	Specific enthalpy of saturated liquid, Btu/lbm
h_g	Specific enthalpy of saturated vapor, Btu/lbm
k	Thermal conductivity of tube metal, Btu/s-ft-°F
K_{sc}	Subcooled region forced convection heat transfer factor, Btu/lbm-°F
K_{sh}	Superheated region forced convection heat transfer factor, Btu/lbm-°F
K_1	Multiplier on Thom's correlation, dimensionless
L	Length of steam generator tube bundle, ft

p	Perimeter, ft
P	Pressure, psia
ΔP_g	Gravity head, psi
ΔP_f	Shock and friction pressure drop, psi
P_{sg}	Steam generator pressure for state property evaluation, psia
P_{sgi}	Steam generator pressure at feedwater inlet of steam generator, psia
P_{sgo}	Steam generator pressure at steam outlet of steam generator, psia
$\Delta P/\Delta \ell$	Unrecoverable pressure losses per unit length, psi/ft
Q_p	Rate of heat addition to primary side of steam generator, Btu/s
Q_s	Rate of heat addition to secondary side of steam generator, Btu/s
T	Temperature, F
T_{sat}	Secondary fluid saturation temperature, F
ΔT_{sat}	Temperature differential, tube wall to secondary fluid, F
U_b	Metal-to-secondary heat transfer coefficient in boiling region, Btu/s-ft^2-°F
U_{pm}	Primary-to-metal heat transfer coefficient, Btu/s-ft^2-°F
U_{sm}	Secondary-to-metal heat transfer coefficient, Btu/s-ft^2-°F
V	Intermediate term in drift-flux model, dimensionless
$\bar{V}_{gi}$	Drift velocity of vapor, ft/s
W	Mass flow rate, lbm/s
W_f	Equivalent flow rate from subcooled to boiling region, lbm/s
W_o	References mass flow rate for secondary, lbm/s
x	Distance measured from primary fluid inlet, ft
ΔX	Difference between X_s and X_c, dimensionless
X_c	Threshold quality defined to smooth transition from boiling region to superheated region, dimensionless
z	Distance measured from bottom of tube bundle, ft
z_{sc}	Height above bottom tubesheet of interface between subcooled and boiling region, ft

z_b Height above bottom tubesheet of interface between boiling and superheat region, ft

α Void fraction, dimensionless

ρ Density, lbm/ft^3

ρ_f Density of saturated liquid, lbm/ft^3

ρ_g Density of saturated vapor, lbm/ft^3

σ Surface tension, lbf/ft

ϕ Heat flux, $Btu/s\text{-}ft^2$

Subscripts

b Boiling region

m Tube metal

p Primary side

s Secondary side

sc Subcooled region

sh Superheated region

2.17. Downcomer Model

An option is available in POWER TRAIN to simulate the steam generator with a preheating feedwater chamber. This downcomer preheats the feedwater to saturated conditions (using recirculated superheated steam) before it enters the tube bundle. Therefore, when using this option, the subcooled region is eliminated from the simulation and the following model is substituted. The following assumptions are made in deriving this model:

1. The aspiration steam maintains the downcomer at saturated conditions.
2. There is a distinct saturated fluid level.
3. Spatial momentum and transient fluid acceleration pressure differentials are negligible.
4. The time rate of change of the saturated liquid and vapor properties (specifically density and enthalpy) are negligible.

The equations that describe the dynamics of the downcomer follow.

2.17.1. Conservation of Mass

$$A_{dc} \frac{d}{dt} \left[\rho_g (L_{dc} - L_\ell) + \rho_f L_\ell\right] = W_{fw} + W_{asp} - W_f \tag{2-106}$$

where

A_{dc} = downcomer area, constant, ft^2,
L_{dc} = downcomer height, constant, ft,
L_ℓ = saturated fluid level, ft,
ρ_g = saturated vapor density, lbm/ft^3,
ρ_f = saturated fluid density, lbm/ft^3,
W_{fw} = feedwater flow rate entering downcomer, lbm/s,
W_{asp} = vapor aspiration flow rate, lbm/s,
W_f = saturated fluid flow rate leaving downcomer, lbm/s.

The continuity equation is now solved for the time rate of change of the fluid level by utilizing assumption 4 above:

$$\frac{d}{dt} (L_\ell) = \frac{1}{A_{dc}} \left(\frac{W_{fw} + W_{asp} - W_f}{\rho_f - \rho_g}\right) \tag{2-107}$$

2.17.2. Conservation of Energy

$$A_{dc} \frac{d}{dt} \left[\rho_g h_g (L_{dc} - L_\ell) + \rho_f h_f L_\ell\right] = W_{fw} h_{fw} + W_{asp} h_{asp} - W_f h_f \tag{2-108}$$

where (in addition to previously defined quantities)

h_g = saturated vapor enthalpy, Btu/lbm,
h_f = saturated fluid entha Btu/lbm,
h_{fw} = feedwater enthalpy, Btu/lbm,
h_{asp} = aspiration flow enthalpy, Btu/lbm.

This equation is now solved for the time rate of change of the fluid level by once again using assumption 4 above:

$$\frac{d}{dt} (L_\ell) = \frac{1}{A_{dc}} \left(\frac{W_{fw} h_{fw} + W_{asp} h_{asp} - W_f h_f}{\rho_f h_f - \rho_g h_g}\right). \tag{2-109}$$

Now the aspiration flow rate required to keep the downcomer at saturated conditions can be found by equating 2-107 and 2-109 and solving for W_{asp}:

$$W_{asp} = \frac{W_f (h_f - K_1) - W_{fw} (h_{fw} - K_1)}{h_{asp} - K_1} \tag{2-110}$$

where

$$K_1 = \frac{\rho_f h_f - \rho_g h_g}{\rho_f - \rho_g}. \tag{2-111}$$

2.17.3. Conservation of Momentum

The flow rate of saturated fluid (W_f) which leaves the downcomer and enters the tube bundle is found by considering the pressure differentials that exist in the tube region and downcomer. Since the downcomer is open to the tube region at the bottom (inlet orifice) and at the top (aspiration port), the sum of the following pressure differenticls must be zero:

$$\Delta P_{dcg} + \Delta P_{fwn} - \Delta P_{orf} - \Delta P_{otsg} - \Delta P_{asp} = 0 \tag{2-112}$$

where

ΔP_{dcg} = pressure differential due to fluid and vapor mass in downcomer, psi,

ΔP_{fwn} = pressure differential due to feedwater momentum force from inlet nozzle, psi,

ΔP_{orf} = pressure differential due to orifice flow rate, psi,

ΔP_{otsg} = pressure differential due to flow rate and mass in tube region, psi,

ΔP_{asp} = pressure differential due to aspiration flow rate, psi.

Neglecting spatial momentum and transient fluid acceleration effects (assumption 3),

$$\Delta P_{dcg} = \frac{g}{144 g_c}\left[\rho_f L_\ell + \rho_g (L_{dc} - L_\ell)\right], \tag{2-113}$$

$$\Delta P_{orf} = R_{orf}\left(\frac{|W_f| W_f}{\rho_f}\right), \tag{2-114}$$

$$\Delta P_{asp} = R_{asp}\left(\frac{|W_{asp}| W_{asp}}{\rho_g}\right), \tag{2-115}$$

$$\Delta P_{fwn} = R_{fdc}\left(\frac{W_{fw}^2}{\rho_{fw}}\right). \tag{2-116}$$

The tube region pressure drop (ΔP_{otsg}) is discussed in section 2.16.3 (equations 2-61 and 2-62). It is calculated from the lower tubesheet to the aspiration port and input to the downcomer model; R_{orf}, R_{asp}, and R_{fdc} above are equivalent friction, shock, and momentum coefficients determined from steady-state considerations.

Combining equations 2-112 and 2-114 and solving for the orifice flow rate (and requiring that $W_f \geq 0$),

$$W_f = \left[\frac{\rho_f}{R_{orf}} (\Delta P_{dcg} + \Delta P_{fwn} - \Delta P_{otsg} - \Delta P_{asp})\right]^{1/2}, \qquad (2\text{-}117)$$

The equations above are implemented on the digital portion of the hybrid computer. The time rate of change of fluid level is calculated using Euler integration. The remaining equations are algebraic.

2.18. Steam Lines

Steam from the two steam generators flows into the steam lines and is combined in the turbine header before flowing onward through the turbine control valves to the turbine. Only one steam line is modeled per steam generator. The partial differential equations describing the flow of steam are solved using the discrete space continuous time method. The nodalization scheme is shown in Figure 2-1.

The steam line model requires that the following boundary conditions be provided:

1. Flow out of each steam generator.
2. Positions of all dump, bypass, and control valves.

The following assumptions are made in the model:

1. No heat is lost or gained through the walls.
2. The steam is always superheated.
3. The steam enthalpy is constant.

The steam line model consists of the solutions of equations to calculate the following variables as a function of time:

1. Pressure in every node.
2. Flows into and out of every node.

Assumptions 1 and 3 eliminate the need for an energy equation. Assumptions 2 and 3 allow the equation of state to be approximated as

$$\rho(x,t) = C_1 + C_2 P(x,t) \qquad (2\text{-}118)$$

where C_1 and C_2 are constants derived from input data.

The conservation of mass equation takes the following form when this equation of state is used:

$$\frac{\partial}{\partial t} \rho + \frac{1}{A} \frac{\partial}{\partial x} W = 0. \qquad (2\text{-}119)$$

Since steam enthalpy is assumed constant, $\rho = f(P)$, and we may write

$$\frac{\partial P}{\partial t} \frac{d\rho}{dP} + \frac{1}{A} \frac{\partial}{\partial x} W = 0. \qquad (2\text{-}120)$$

The conservation of mass equations for all nodes are as follows (see Figure 2-1):

$$\frac{d}{dt} P_{sgoa} = \left(\frac{1}{ALC_2}\right)_{1a} (W_{sgoa} - W_{s\ell 1a}) \qquad (2\text{-}121)$$

$$\frac{d}{dt} P_{sgob} = \left(\frac{1}{ALC_2}\right)_{1b} (W_{sgob} - W_{s\ell 1b}) \qquad (2\text{-}122)$$

$$\frac{d}{dt} P_{s\ell 1a} = \left(\frac{1}{ALC_2}\right)_{2a} (W_{s\ell 1a} - W_{ra} - W_{sada} - W_{s\ell 2a}) \qquad (2\text{-}123)$$

$$\frac{d}{dt} P_{s\ell 1b} = \left(\frac{1}{ALC_2}\right)_{2b} (W_{s\ell 1b} - W_{rb} - W_{sadb} - W_{s\ell 2b}) \qquad (2\text{-}124)$$

$$\frac{d}{dt} P_{s\ell 2a} = \left(\frac{1}{ALC_2}\right)_{3a} (W_{s\ell 2a} - W_{mada} - W_{s\ell 3a}) \qquad (2\text{-}125)$$

$$\frac{d}{dt} P_{s\ell 2b} = \left(\frac{1}{ALC_2}\right)_{3b} (W_{s\ell 2a} - W_{madb} - W_{s\ell 3b}) \qquad (2\text{-}126)$$

$$\frac{d}{dt} P_t = \left(\frac{1}{ALC_2}\right)_t (W_{s\ell 3a} + W_{s\ell 3b} - W_t - W_{bp} - W_{rh_2} - W_{pg}) \qquad (2\text{-}127)$$

Neglecting the effect of kinetic energy change, the momentum equation is written for every junction between nodes as follows:

$$\frac{d}{dt} W_{s\ell 1a} = \left(\frac{144 g_c A}{L}\right)_{1a} (P_{sgoa} - P_{s\ell 1a}) - K_{f1a} W_{s\ell 1a} |W_{s\ell 1a}| \qquad (2\text{-}128)$$

$$\frac{d}{dt} W_{s\ell 1b} = \left(\frac{144 g_c A}{L}\right)_{1b} (P_{sgob} - P_{s_{\ell 1b}}) - K_{f1b} W_{s\ell 1b} |W_{s\ell 1b}| \qquad (2\text{-}129)$$

$$\frac{d}{dt} W_{s\ell 2a} = \left(\frac{144 g_c A}{L}\right)_{2a} (P_{s\ell 1a} - P_{s\ell 2a}) - K_{f2a} W_{s\ell 2a} |W_{s\ell 2a}| \qquad (2\text{-}130)$$

$$\frac{d}{dt} W_{s\ell 2b} = \left(\frac{144 g_c A}{L}\right)_{2b} (P_{s\ell 1b} - P_{s\ell 2b}) - K_{f2b} W_{s\ell 2b} |W_{s\ell 2b}| \qquad (2\text{-}131)$$

$$\frac{d}{dt} W_{s\ell 3a} = \left(\frac{144g_c}{L}\right)_{3a} (P_{s\ell 2a} - P_t) - K_{f3a} W_{s\ell 3a} |W_{s\ell 3a}| \quad (2\text{-}132)$$

$$\frac{d}{dt} W_{s\ell 3b} = \left(\frac{144g_c}{L}\right)_{3b} (P_{s\ell 2b} - P_t) - K_{f3b} W_{s\ell 3b} |W_{s\ell 3b}| \quad (2\text{-}133)$$

The flows through the safety relief, atmospheric dump, turbine control, and bypass valves are assumed to be proportional to the pressure at the valve. All valve positions are given to the steam line model as boundary conditions.

$$W_{ra} = K_{ra} X_{ra} P_{s\ell 1a} \quad (2\text{-}134)$$

$$W_{rb} = K_{rb} X_{rb} P_{s\ell 1b} \quad (2\text{-}135)$$

$$W_t = K_t X_t P_t \quad (2\text{-}136)$$

$$W_{bp} = K_{bp} X_{bp} P_t \quad (2\text{-}137)$$

$$W_{sada} = K_{sada} X_{sada} P_{s\ell 1a} \quad (2\text{-}138)$$

$$W_{sadb} = K_{sadb} X_{sadb} P_{s\ell 1b} \quad (2\text{-}139)$$

$$W_{mada} = K_{mada} X_{mada} P_{s\ell 2a} \quad (2\text{-}140)$$

$$W_{madb} = K_{madb} X_{madb} P_{s\ell 2b} \quad (2\text{-}141)$$

Nomenclature

A	Flow area, ft^2
$1/ALC_2$	dP/dM for steam line nodes, psi/lbm
$144g_cA/L$	Reciprocal of flow inertia for steam line nodes times gravitational conversion constant, lbm-in.^{2}/lbf-s
C_1	Density of steam at the reference pressure, lbm/ft^3
C_2	$\partial\rho/\partial P$ for steam at the reference pressure, lbm-in.^{2}/lbf-ft^3
K	Proportionality constant for valve flows, lbm-in.^{2}/lbf-s
K_f	Friction loss proportionality constant for steam line nodes, 1/lbm
P	Pressure, psia
W	Mass flow rate, lbm/s

W_{bp} Bypass valve flow rate, lbm/s

W_{pg} Pegging steam flow rate, lbm/s

W_{rh2} Reheater flow rate, lbm/s

X Valve position, %/100%

ρ Density, lbm/ft^3

Subscripts

mada, madb Main atmospheric dump valves (on/off)

ra, rb Relief valves

sada, sadb Small atmospheric dump valves (modulating)

sgia, sgib Steam generator inlet

sgoa, sgob Steam generator outlet

sℓ Steam line node

1a,1b,2a,2b Steam line node designations

3a,3b,t

2.19. Valves on Steam Lines

The response of the turbine throttle valve to the control signal supplied by the ICS is modeled according to equation 2-142:

$$\frac{dX_t}{dt} = (X_{TD} - X_t)/\tau. \tag{2-142}$$

The value of the derivative is limited to 10%/s opening velocity and 100%/s closing velocity. The throttle valve model is also used to model the turbine stop valves.

The main steam relief (safety) valves on each steam line are modeled as two banks, each bank with its own opening and closing pressure. For the purposes of simulation, all valves in a bank are combined; therefore, a total of four valves are simulated. Each valve steps fully open when the opening setpoint pressure is exceeded and steps fully closed when pressure drops below the closing setpoint pressure. The atmospheric dump valves comprise on-off and modulating valves. There is one modulating valve per steam line. The valve opens and closes according to the position demand generated by the proportional-integral (PI) valve controller. The on-off atmospheric dump valves comprise four banks per steam line. These valves step fully open and fully closed after a delay time specified by the input data. The opening and closing setpoints of all four banks assume different values under different trip conditions. For example, the setpoints after a reactor trip are different from the normal ones.

The condenser dump valves are modeled as a single bank of modulating valves. The valve position opens and closes within rate limits according to the position demanded by a PI controller. In addition, the controller setpoint changes for different trip conditions in the same manner as the setpoints for the on-off atmospheric dumps.

Nomenclature

X_t Turbine control valve position, %/100%

X_{TD} Turbine control valve position demand, %/100%

τ Turbine control valve time constant, s

2.20. Turbine and Associated Components

The turbine model includes the moisture separator and reheaters. The dynamics of the turbine model are approximated by applying time constants to steady-state heat balance data for the power plant. The heat balance data as functions of throttle flow, flow out of the second reheater, or flow into the low pressure turbine are supplied along with the time constants as part of the input data to the simulation. The extraction flows are shown in Figure 1-2. The equations that model the dynamics of these flows are given below. The B terms are logical variables having a value of either 1 or 0, depending on plant status. The F terms represent steady-state table data supplied as part of the input data. All flows are defined in Figure 1-2.

$$\frac{d}{dt} W_{6b} = \frac{1}{\tau_{6b}} \left[B_6 F_1 (W_t) - W_{6b}\right] \tag{2-143}$$

$$\frac{d}{dt} W_{5b} = \frac{1}{\tau_{5b}} \left[B_5 F_2 (W_t) - W_{5b}\right] \tag{2-144}$$

$$\frac{d}{dt} W_4 = \frac{1}{\tau_4} \left[B_4 F_3 (W_t) - W_4\right] \tag{2-145}$$

$$\frac{d}{dt} W_{rh1} = \frac{1}{\tau_{grh}} \left[B_5 F_6 (W_t) - W_{rh1}\right] \quad (2\text{-}146)$$

$$\frac{d}{dt} W_7 = \frac{1}{\tau_7} \left[W_t - B_6 F_1 (W_t) - B_5 F_2 (W_t) - B_4 F_3 (W_t) - B_5 F_6 (W_t) - W_7\right] \quad (2\text{-}147)$$

$$W_{ms} = F_1 (W_7) \quad (2\text{-}148)$$

$$\frac{d}{dt} W_{8d} = \frac{1}{\tau_{rh1}} (W_7 - W_{ms} - W_{8d}) \quad (2\text{-}149)$$

$$\frac{d}{dt} W_{8c} = \frac{1}{\tau_{rh2}} (W_{8d} - W_{8c}) \quad (2\text{-}150)$$

$$W_{rh2} = B_6 F_1 (W_{8c}) \quad (2\text{-}151)$$

$$W_{8b} = F_2 (W_{8c}) \quad (2\text{-}152)$$

$$W_{8a} = W_{8c} - W_{8b} \quad (2\text{-}153)$$

$$W_8 = B_{\ell pi} W_{8a} \quad (2\text{-}154)$$

$$\frac{d}{dt} W_3 = \frac{1}{\tau_3} \left[B_3 F_1 (W_8) - W_3\right] \quad (2\text{-}155)$$

$$\frac{d}{dt} W_2 = \frac{1}{\tau_2} \left[B_2 F_2 (W_8) - W_2\right] \quad (2\text{-}156)$$

$$\frac{d}{dt} W_1 = \frac{1}{\tau_1} \left[B_1 F_3 (W_8) - W_1\right] \quad (2\text{-}157)$$

$$\frac{d}{dt} W_9 = \frac{1}{\tau_9} \left[W_8 - B_3 F_1 (W_8) - B_2 F_2 (W_8) - B_1 F_3 (W_8) - W_9\right] \quad (2\text{-}158)$$

Logical Variables

$B_{\ell pi}$ = 0 when intercept valves are closed (2-159)

B_1 = 0 when heater 1 is failed (2-160)

B_2 = 0 when heater 2 is failed (2-161)

B_3 = 0 when heater 3 is failed (2-162)

B_4 = 0 when heater 4 is failed (2-163)

$$B_5 = 0 \text{ when heater 5 is failed} \quad (2\text{-}164)$$

$$B_6 = 0 \text{ when heater 6 is failed} \quad (2\text{-}165)$$

Tables of Flow at Steady State

$$F_1(W_t) = W_{6b} \text{ Vs } W_t \quad (2\text{-}166)$$

$$F_2(W_t) = W_{5b} \text{ Vs } W_t \quad (2\text{-}167)$$

$$F_3(W_t) = W_4 \text{ Vs } W_t \quad (2\text{-}168)$$

$$F_6(W_t) = W_{rh1} \text{ Vs } W_t \quad (2\text{-}169)$$

$$F_1(W_{8c}) = W_{rh2} \text{ Vs } W_{8c} \quad (2\text{-}170)$$

$$F_2(W_{8c}) = W_{8b} \text{ Vs } W_{8c} \quad (2\text{-}171)$$

$$F_1(W_8) = W_3 \text{ Vs } W_8 \quad (2\text{-}172)$$

$$F_2(W_8) = W_2 \text{ Vs } W_8 \quad (2\text{-}173)$$

$$F_3(W_8) = W_1 \text{ Vs } W_8 \quad (2\text{-}174)$$

2.21. Generator

The simulation of the electric power generator is based on a torque balance approach. The steam throughput of the high- and low-pressure turbines is used to determine the net mechanical steam torque; the electrical load torque is determined from the generator load angle. Electrical damping, friction, and windage torques are also included. The generator frequency is calculated from the net torque. The grid frequency is allowed to vary from normal grid frequency in step fashion if the operator so desires. The equations describing the dynamic behavior of the generator are given below.

$$M_s = F_2(W_7) + F_1(W_9) \quad (2\text{-}175)$$

$$M_e = B_{tgc}[B_t K_{tg2} + K_{tg3} \int (F_g - F_{gr})dt] \quad (2\text{-}176)$$

$$M_d = B_{tgc}[K_{tg4}(F_g - F_{gr})] \quad (2\text{-}177)$$

$$M_f = K_{tg5} F_g \quad (2\text{-}178)$$

$$M_g = M_s - M_e - M_d - M_f \quad (2\text{-}179)$$

$$\frac{d}{dt} F_g = K_{tg1} M_g \quad (2\text{-}180)$$

$$F_{gr} = F_{grn} + B_{fup}\Delta F_{gr} - B_{fdn}\Delta F_{gr} \quad (2\text{-}181)$$

Nomenclature

F_g	Generator frequency, Hz
F_{gr}	Grid frequency, Hz
ΔF_{gr}	Grid frequency step, Hz
F_{grn}	Normal grid frequency, Hz
K_{tg1}	Turbine-generator speed change rate at 100% torque, Hz/s
K_{tg2}	Normal house load as a fraction of total load, %/100%
K_{tg3}	Turbine-generator torque change per load angle change, %/deg
K_{tg4}	Turbine-generator damping constant, %/Hz
K_{tg5}	Turbine-generator friction and windage torque, %/Hz
M_d	Electrical damping torque, %/100%
M_e	Electrical load torque, %/100%
M_f	Friction and windage torque, %/100%
M_g	Net torque, %/100%
M_s	Mechanical steam torque, %/100%

Logical Variables

B_{fdown} = 1 when pushbutton is depressed to step down grid fre-frequency (2-182)

F_{fup} = 1 when pushbutton is depressed to step up grid fre-quency (2-183)

B_t = 0 when turbine is tripped (2-184)

B_{tgc} = 0 when grid breakers are open (loss of load) (2-185)

Tables

$F_2(W_7)$ = High-pressure turbine steam torque Vs W_7 (2-186)

$F_1(W_9)$ = Low-pressure turbine steam torque Vs W_9 (2-187)

2.22. Feedwater Flows and Pressures

The feedwater and condensate system includes all the equipment that connects the condenser to the steam generator. It is much easier to discuss the total system if it is divided into separate parts. Therefore, in the following discussion of the model, the feedwater system is defined as the equipment on the downstream side of the main feedwater pumps. The condensate system is that equipment from the condenser to the main feedwater pumps. The emergency feedwater system and the total secondary energy balance are also discussed separately.

The feedwater system is modeled as a single train (see Figure 1-2). The calculation of the system flows and pressures is based on the assumption of constant feedwater densities in the various sections. Because of this assumption, the conservation of momentum is the only differential equation to be solved for flows and pressures. The calculation of feedwater enthalpies and temperatures is discussed in section 2.27.

The momentum equation for constant-area flow channels, after neglecting the pressure drop due to spatial acceleration (a result of the constant-density assumption), becomes

$$\frac{1}{A}\frac{\partial}{\partial t} W + 144 g_c \frac{\partial}{\partial x} P + \frac{f/\rho}{2D_h A^2} |W| W + g\rho = 0. \qquad (2\text{-}188)$$

It is further assumed that for a particular section of pipe the friction factor f is constant, and the differential pressure in the spatial dimension can be evaluated in the same manner as for the steam lines

$$\frac{\partial}{\partial x} P = -\frac{P_o - P_i}{L} \qquad (2\text{-}189)$$

where P_o is the pressure at the downstream node, P_i is the driving pressure, and L is the length of the section of pipe. Combining equations 2-188 and 2-189 and solving for the time rate of change of the flow rate results in the following general equation for the feedwater flow rates:

$$\frac{d}{dt} W - \frac{144 g_c A}{L} (P_i - P_o - \Delta P_f - \Delta P_e). \qquad (2\text{-}190)$$

The elevation pressure drop ΔP_e is a constant, and the frictional drop ΔP_f is calculated from the following expression (note that flow is in the positive direction only; therefore $|W|W = W^2$):

$$\Delta P_f = K_f W^2. \tag{2-191}$$

The friction coefficient K_f is assumed to be constant and is determined from the initial pressures and flows so that the flow rate derivative equals zero. In lines having control valves the frictional pressure drop across the valve is a function of valve position (section 2.23). In lines with feedwater pumps the driving pressure includes the pressure head developed by the pump (section 2.24). The following equations are used to calculate the feedwater flows to each steam generator and the flow through the main feedwater pumps:

$$\frac{d}{dt} W_{fwa} = \left(\frac{144 g_c A}{L}\right)_{fwa} (P_{fw} - P_{sgia} - \Delta P_{ea} - K_{fa} W^2_{fwa} - \Delta P_{cva}), \tag{2-192}$$

$$\frac{d}{dt} W_{fwb} = \left(\frac{144 g_c A}{L}\right)_{fwb} (P_{fw} - P_{sgib} - \Delta P_{eb} - K_{fb} W^2_{fwb} - \Delta P_{cvb}), \tag{2-193}$$

$$W_{fw} = W_{fwa} + W_{fwb}. \tag{2-194}$$

The pressure (P_{fw}) at the common junction is found by solving the conservation of momentum equation for the main feedwater line and by differentiating equation 2-194:

$$\frac{d}{dt} W_{fw} = \frac{d}{dt} W_{fwa} + \frac{d}{dt} W_{fwb}, \tag{2-195}$$

$$P_{fw} = P_{c2} + \Delta P_{fp} + \Delta P_{efw} - K_{fw} W^2_{fw} - \left(\frac{1}{144 g_c A}\right)_{fw} \frac{d}{dt} W_{gw}. \tag{2-196}$$

Nomenclature

A	Cross-sectional area, ft^2
D_h	Hydraulic diameter, ft
f	Friction factor, dimensionless
g	Gravitational acceleration, ft/s^2
g_c	Gravitational conversion constant, lbm-ft/lbf-s

K_f	Proportionality constant for friction pressure drop, psi/(lbm/s)^2
K_{fa}, K_{fb}	Value of K_f for feedwater piping from the common junction to steam generators A and B, psi/(lbm/s)^2
K_{fw}	Value of K_f for main feedwater line, psi/(lbm/s)^2
L	Length of section of piping, ft
$(L/144g_cA)_{fw}$	Length over area term for the main feedwater piping, (lbm/s^2)/psi
$(144g_cA/L)_{fwa}$	Reciprocal of length over area term for the feedwater piping from the common junction to steam generator A, psi/(lbm/s^2)
$(144g_cA/L)_{fwb}$	Reciprocal of length over area term for the feedwater piping from the common junction to steam generator B, psi/(lbm/s^2)
P	Pressure, psia
P_{c2}	Pressure at feed pump inlet, psia
P_{fw}	Pressure at the common junction, psia
P_i	Inlet pressure, psia
P_o	Outlet pressure, psia
P_{sgia}, P_{sgib}	Pressures at secondary inlets to each steam generator, psia
$\Delta P_{cva}, \Delta P_{cvb}$	Pressure drop across control valves A and B, psi
ΔP_e	Elevation pressure drop or gain, psi
$\Delta P_{ea}, \Delta P_{eb}$	Elevation pressure drop in feedwater lines from the common junction to steam generators A and B, psi
ΔP_{efw}	Elevation pressure gain in main feedwater line from the feed pump inlet to the common junction, psi
ΔP_{fp}	Feed pump pressure head, psi
W	Mass flow rate, lbm/s
W_{fw}	Mass flow rate into the common junction, lbm/s
W_{fwa}, W_{fwb}	Mass flow rates from the common junction to steam generators A and B, lbm/s

2.23. Feedwater Control Valves

Simulation of the feedwater control valves involves computation of the valve position, which is then used to calculate the valve coefficient (C_V). The valve coefficient and feedwater flow through the valve are then used to calculate the pressure drop across the valve.

The C_V of a valve is defined as the flow of water at 60F (in gpm) that will result in a pressure drop of one psi across the valve:

$$\Delta P_V = \frac{\rho}{62.4}\left(\frac{Q}{C_V}\right)^2 \tag{2-197}$$

For flows given in lbm/s,

$$\Delta P_V = \frac{3228.34}{\rho}\left(\frac{W}{C_V}\right)^2 \tag{2-198}$$

The flow of feedwater to each steam generator during normal operation is controlled by two valves operating in parallel. The combined valve coefficient as a function of the percent opening of the valve is then used in equation 2-198 to find the valve pressure drop:

$$C_V = F(\%X_v). \tag{2-199}$$

The percent feedwater valve opening is calculated by a set of equations that simulate the feedwater valve motor speed and valve position. The control system supplies a feedwater flow error signal to the valve motor. If the feedwater flow needed is greater than that being supplied, the feedwater valve motor is started; this opens the valve. The valve continues to open until the flow error is brought to zero. The operation of the valve for situations where the feedwater flow required is less than that being supplied is analogous to the above except that the valve is closed. Whether the valve is opening or closing, it is limited to a maximum stroke rate regardless of the magnitude of the error. The operation of the feedwater control valves is described by the following equations (same form for both valves):

$$\frac{d}{dt} V_v = \frac{1}{\tau_v}(V_{vd} - V_v), \quad V_{max} \geq V_v \geq -V_{max} \tag{2-200}$$

$$\frac{d}{dt} X_v = V_v \tag{2-201}$$

Nomenclature

C_V	Valve coefficient, gpm/$\sqrt{psi}$
ΔP_v	Pressure drop across valve, psi
Q	Volumetric flow, gpm
V_{max}	Maximum valve opening or closing speed, %/100%-s
V_v	Valve closing velocity, %/100%-s
V_{vd}	Valve closing velocity demand, %/100%-s
W	Mass flow rate, lbm/s
$\%X_v$	Valve position, %/100%
ρ	Density, lbm/ft^3
τ_v	Valve time constant, s

2.24. Main Feedwater Pumps

The head developed by the main feedwater pumps is a function of the number of pumps running (normally two), the pump speed, the flow through each pump, and whether or not the pressure at the pump suction has dropped below the net positive suction head (NPSH) value.

The homologous pump curves are used to calculate the pump head (in psi). Pump speed (N_p), flow (W_p), and head (ΔP_p) are normalized to rated (normally 100%) values, and these normalized curves are then used to compute the head given flow and speed. Define

$$\alpha = N_p/N_r, \tag{2-202}$$

$$\beta = W_p/W_r, \tag{2-203}$$

$$\gamma = \Delta P_p/\Delta P_t. \tag{2-204}$$

Construct two tables at $N_p = N_r$ ($\alpha = 1.0$).

Table 1: $W_p \leq W_r$

$$\beta/\alpha \text{ Vs } \gamma/\alpha^2 \tag{2-205}$$

Note: $\beta/\alpha = W_p/W_r$, (2-206)

$$\gamma/\alpha^2 = \Delta P_p/\Delta P_r, \tag{2-207}$$

Table 2: $W_p \geq W_r$

$$\alpha/\beta \text{ Vs } \gamma/\beta^2 \qquad (2\text{-}208)$$

Note: $\alpha/\beta = W_r/W_p$, (2-209)

$$\gamma/\beta^2 = (\Delta P_p/\Delta P_r)/(W_r/W_p)^2 \qquad (2\text{-}210)$$

Now, given pump speed and pump flow for any operating condition, compute α and β. If $\alpha \geq \beta$, use Table 1 to find $(\gamma/\alpha^2)_1$. Then $\Delta P_p = (\gamma/\alpha^2)_1 \alpha^2 \Delta P_r$. If $\alpha < \beta$, use Table 2 to find $(\gamma/\beta^2)_2$. Then $\Delta P_p = (\gamma/\beta^2)_2 \beta^2 \Delta P_r$.

The feedwater pump speed is calculated from a fit to manufacturer's data for the pump controller and is modeled as a second-order system. The pump speed (N_p) in response to a speed demand signal (N_{pd}) received from the control system is defined by equation 2-211:

$$\frac{d^2}{dt^2} N_p = \frac{1}{\tau_{p_2}} N_{pd} - N_p - \tau_{p_1} \frac{d}{dt} N_p. \qquad (2\text{-}211)$$

If the suction pressure (P_{c2}) drops below the minimum suction pressure (NPSH), the pump speed demand (N_{pd}) in the equation above is set equal to zero.

Nomenclature

N_p, N_r	Actual and rated pump speed, rpm
N_{pd}	Pump speed demand, rpm
$\Delta P_p, \Delta P_r$	Actual and rated pump head, psi
W_p, W_r	Actual and rated pump flow, lbm/s
α	Normalized pump speed (dimensionless)
β	Normalized pump flow (dimensionless)
γ	Normalized pump head (dimensionless)
τ_{p_1}, τ_{p_2}	Pump controller time constants, s

2.25. Emergency Feedwater System

The emergency feedwater system provides for steam generator flow in the event of feedwater or steam line isolation. This would occur if all main feed pumps were tripped or if the steam line pressure fell below a setpoint pressure. When isolation occurs, block valves in the feedwater and steam lines are closed (steam line isolation occurs only if steam pressure is low) and a delay timer is started to simulate startup of the emergency feedwater pumps. After the delay time has elapsed, emergency feedwater flow is provided to the steam generators. Emergency feedwater flow control is modeled as a proportional-integral controller that acts to maintain a constant steam generator level. In the POWER TRAIN simulation any isolation signal generated causes isolation of both steam generators.

2.26. Condensate System Flows and Pressures

The condensate system is composed of all the equipment from the condenser to the main feedwater pumps. It is shown in schematic form in Figure 1-2. This section deals with the calculation of the various flow rates and pressures in the system and with the calculation of the hot well level. Refer to Figure 1-2 for definition of all flows and pressures. The flows and pressures are calculated using the same assumptions and approach as outlined in section 2.22. The main system flows are calculated from the main feedwater flow and the drain tank flows that are pumped forward.

$$W_{c2} = W_{fw} - W_{dhp} \tag{2-212}$$

$$W_{c1} = W_{c2} - W_{d\ell p} \tag{2-213}$$

The drain tank flows are calculated assuming that a constant drain tank level exists and that the flow rate differential may be approximated as a simple first-order lag. Capability is also provided to terminate drain tank flow instantly, which simulates closure of a block valve. If block valves are open,

$$\frac{d}{dt} W_{dhp} = \frac{1}{\tau_{dhp}} (W_{4d} + W_{5d} + W_{ms} - W_{dhp}) \tag{2-214}$$

and

$$\frac{d}{dt} W_{d\ell p} = \frac{1}{\tau_{d\ell p}} (W_{1d} + W_{fpt} - W_{d\ell p}). \tag{2-215}$$

If the high-pressure drain tank block valve closes,

$$\frac{d}{dt} W_{dhp} = 0, \quad W_{dhp} = 0. \tag{2-216}$$

If the low-pressure drain tank block valve closes,

$$\frac{d}{dt} W_{d\ell p} = 0, \quad W_{d\ell p} = 0. \tag{2-217}$$

Now all the system flow rates and their derivatives are defined, so it is possible to solve the momentum equation for each line to determine the pressures. From equations 2-212 and 2-213, it follows that

$$\frac{d}{dt} W_{c2} = \frac{d}{dt} W_{fw} - \frac{d}{dt} W_{dhp} \tag{2-218}$$

and

$$\frac{d}{dt} W_{c1} = \frac{d}{dt} W_{c2} - \frac{d}{dt} W_{d\ell p}. \tag{2-219}$$

Therefore,

$$P_{c1} = P_c + \Delta P_{cp} + \Delta P_{hwp} - \Delta P_{ec1} + \Delta P_{hw\ell} - K_{fc1} W_{c1}^2 - \left(\frac{L}{144 g_c A}\right)_{c1} \frac{d}{dt} W_{c1}, \tag{2-220}$$

and

$$P_{c2} = P_{c1} - \Delta P_{ec2} - K_{fc2} W_{c2}^2 - \left(\frac{L}{144 g_c A}\right)_{c2} \frac{d}{dt} W_{c2}. \tag{2-221}$$

The head developed by the condensate and hot well pumps depends on the number of pumps running, the flow rate through the pumps, and the pump head capacity curves:

$$\Delta P_{cp} = F_{cp}(W_{c1}/N_{cp}), \tag{2-222}$$

$$\Delta P_{hwp} = F_{hwp}(W_{c1}/N_{vp}). \tag{2-223}$$

The hot well level is determined from a mass balance on the condenser:

$$\frac{d}{dt} L_{hw} = \left(\frac{1}{A\rho}\right)_{hw} (W_9 + W_{bp} - W_{c1}). \tag{2-224}$$

The pressure due to the hot well level is found from the initial pressure and level, which are input, and the level calculated from equation 2-224:

$$\Delta P_{hw\ell} = \left(\frac{P_{hw\ell}}{L_{hw}}\right)_{t=0} L_{hw}. \qquad (2\text{-}225)$$

Calculation of the drain flows for each heater is discussed in the next section, which covers the energy balance for the entire feedwater and condensate system.

Nomenclature

$(1/A\rho)_{hw}$	Reciprocal of hot well area times density, ft/lbm
F_{cp}	Condensate pump head versus flow table, psi
F_{hwp}	Hot well pump head versus flow table, psi
K_{fc1}	Proportionality constant for friction pressure drop between condenser and P_{c1}, psi/(lbm/s)^2
K_{fc2}	Proportionality constant for friction pressure drop between P_{c1} and P_{c2}, psi/(lbm/s)^2
L_{hw}	Hot well level, ft
$(L/144g_cA)_{c1}$	Length over area term for piping between condenser and P_{c1}, lbm/s^2-psi
$(L/144g_cA)_{c2}$	Length over area term for piping between P_{c1} and P_{c2}, lbm/s^2-psi
ΔP_{cp}	Condensate pump head, psi
ΔP_{ec1}	Elevation pressure difference between condenser and P_{c1}, psi
ΔP_{ec2}	Elevation pressure difference between P_{c1} and P_{c2}, psi
ΔP_{hw}	Pressure difference caused by hot well level, psi
ΔP_{hwp}	Hot well pump head, psi
$(\Delta P/L_{hw})_{t=0}$	Initial value of hot well pressure difference divided by initial hot well level, psi/ft
τ_{dhp}	High-pressure drain tank time constant, s
$\tau_{d\ell p}$	Low-pressure drain tank time constant, s

2.27. Secondary System Energy Balance

The secondary system energy balance is based on plant heat balance information. Several functions of load are defined as part of the input data. Differential and algebraic equations are solved after several system simplifications are made.

As an example, the actual plant has parallel feedwater heaters. In this model they are lumped into o heater. Further, it is assumed that the high-pressure heaters (if they are not failed) will provide feedwater at the prop^r temperature, depending on load. To accomplish this, the necessary steam flow to the high-pressure heaters is calculated based on the system energy balance. This simulates the increased shell-side flow that would occur due to a lower shell-side pressure (which would result from introduction of feedwater at a lower temperature into the heater).

The basic method is to write steady-state mass and energy balances for each heater and then to assume that the transient performance of each heater can be approximated as a first-order system. The characteristic time constant of the heater is used for both the mass and energy balances. With refere to Figure 1-2, the following equations are presented (beginning at the steam generators and working back to the condenser).

2.27.1. Feedwater Temperatures and Delays

$$T_{fwa}(t) = T_{fw}[t - (M_{fwa}/W_{fwa})] \qquad (2\text{-}226)$$

$$T_{fwb}(t) = T_{fw}[t - (M_{fwb}/W_{fwb})] \qquad (2\text{-}227)$$

$$T_{fw}(t) = T'_{fw}[t - (M_{fw}/W_{fw})] \qquad (2\text{-}228)$$

$$T'_{fw} = f(h_{hp6}) \qquad (2\text{-}229)$$

2.27.2. High-Pressure Heater 6

$$h_{hp6i} = h_{hp6i}(t) = h_{hp5}[t - (M_{fws}/W_{fw})] \qquad (2\text{-}230)$$

$$h_{6d} = F_1(W_{fw}) \qquad (2\text{-}231)$$

$$W_{pg6} = [W_{fw}(h_{hp6} - h_{hp6i}) + W_{6d}h_{6d} - W_{6b}h_{6b} - W_{rh2}h_{rh2d}]/h_{pg6} \qquad (2\text{-}232)$$

If $W_{pg6} > W_{pg6m}$,

$$W_{pg6} = W_{pg6m}, \qquad (2\text{-}333)$$

$$W_6 = W_{pg6} + W_{6b} - W_{rh2}, \qquad (2\text{-}234)$$

and

$$h'_{hp6} = h_{hp6i} + (W_6 h_6 - W_{6d}h_{6d})/W_{fw}. \qquad (2\text{-}235)$$

If $W_{pg6} \le W_{pg6m}$,

$$W_6 = [W_{fw}(h_{hp6} - h_{hp6i}) + W_{6d}h_{6d}]/h_6 \tag{2-236}$$

and

$$h'_{hp6} = B_6 F_4(W_t) + \bar{B}_6 h_{hp6i}. \tag{2-237}$$

If $W_{pg6} < 0$,

$$W_{pg6} = 0 \tag{2-238}$$

and

$$W_6 + W_{6b} + W_{rh2}. \tag{2-239}$$

Finally, regardless of the value of W_{pg6},

$$\frac{d}{dt} h_{hp6} = \frac{1}{\tau_{hp6}} (h'_{hp6} - h_{hp6}) \tag{2-240}$$

and

$$\frac{d}{dt} W_{6d} = \frac{1}{\tau_{hp6}} (W_6 - W_{6d}). \tag{2-241}$$

2.27.3. High-Pressure Heater 5

$$h_{hp5i} = h_{hp5i}(t) = h_{hc2}[t - (M_{fw4}/W_{fw})] \tag{2-242}$$

$$h_{5d} = F_2(W_{fw}) \tag{2-243}$$

$$h_{rh1d} = F_1(W_{8d}) \tag{2-244}$$

$$W_{pg5} = [W_{fw}(h_{hp5} - h_{hp5i} - \Delta h_{fp}) + W_{5d}h_{5d} - W_{6d}h_{6d} - W_{5b}h_{5b} - W_{rh1}h_{rh1d}]/h_{pg5} \tag{2-245}$$

If $W_{pg5} > W_{pg5m}$,

$$W_{pg5} = W_{pg5m}, \tag{2-246}$$

$$W_5 = W_{pg5} + W_{5b} + W_{rh1}, \tag{2-247}$$

and

$$h'_{hp5} = h_{hp5i} + \Delta h_{fp} + (W_5 h_5 + W_{6d}h_{6d} - W_{5d}h_{5d})/W_{fw}. \tag{2-248}$$

If $W_{pg5} \leq W_{pg5m}$,

$$W_5 = [W_{fw}(h_{hp5} - h_{hp5i} - \Delta h_{fp}) + W_{5d}h_{5d} - W_{6d}h_{6d}]/h_5 \quad (2\text{-}249)$$

and

$$h'_{hp5} = B_5F_5(W_t) + \bar{B}_5[h_{hp5i} + \Delta h_{fp}]. \quad (2\text{-}250)$$

If $W_{pg5} < 0$,

$$W_{pg5} = 0 \quad (2\text{-}251)$$

and

$$W_5 = W_{5b} + W_{rh1}. \quad (2\text{-}252)$$

Finally, regardless of the value of W_{pg5},

$$\frac{d}{dt} h_{hp5} = \frac{1}{\tau_{hp5}} (h'_{hp5} - h_{hp5}) \quad (2\text{-}253)$$

and

$$\frac{d}{dt} W_{5d} = \frac{1}{\tau_{hp5}} (W_5 + W_{6d} - W_{5d}). \quad (2\text{-}254)$$

2.27.4. High-Pressure Heater 4

$$h_{4d} = F_1(W_{c2}) \quad (2\text{-}255)$$

$$h_{hp4i} = h_{hp4i}(t) = h_{\ell p3}[t - (M_{fw3}/W_{c2})] \quad (2\text{-}256)$$

$$h'_{hp4} = B_4[(w_{c2}h_{hp4i} + W_4h_4 - W_{4d}h_{4d})/W_{c2}] + \bar{B}_4h_{hp4i} \quad (2\text{-}257)$$

$$\frac{d}{dt} h_{hp4} = \frac{1}{\tau_{hp4}} (h'_{hp4} - h_{hp4}) \quad (2\text{-}258)$$

$$\frac{d}{dt} W_{4d} = \frac{1}{\tau_{hp4}} (W_4 - W_{4d}) \quad (2\text{-}259)$$

2.27.5. High-Pressure Heater Drain Tank

$$h_{ms} = F_3(W_7) \quad (2\text{-}260)$$

$$h'_{dhp} = \frac{W_{5d}h_{5d} + W_{4d}h_{4d} + W_{ms}h_{ms}}{W_{5d} + W_{4d} + W_{ms}} \quad (2\text{-}261)$$

$$\frac{d}{dt}\, h_{dhp} = \frac{1}{\tau_{dhp}}\, (h'_{dhp} - h_{dhp}) \qquad (2\text{-}262)$$

$$h_{hc2} = (W_{dhp} h_{dhp} + W_{c2} h_{hp4})/W_{fw} \qquad (2\text{-}263)$$

2.27.6. Low-Pressure Heater 3

$$h_{\ell p3i} = h_{\ell p3i}(t) = h_{\ell p2}[t - (M_{fw2}/W_{c3})] \qquad (2\text{-}264)$$

$$h'_{\ell p3} = B_3[W_{c2} h_{\ell p3i} + W_3 h_3 - W_{3d} h_{3d})/W_{c2}] + \bar{B}_3 h_{\ell p3i} \qquad (2\text{-}265)$$

$$h_{3d} = F_2(W_{c2}) \qquad (2\text{-}266)$$

$$\frac{d}{dt}\, h_{\ell p3} = \frac{1}{\tau_{\ell p3}}\, (h'_{\ell p3} - h_{\ell p3}) \qquad (2\text{-}267)$$

$$\frac{d}{dt}\, W_{3d} = \frac{1}{\tau_{\ell p3}}\, (W_3 - W_{3d}) \qquad (2\text{-}268)$$

2.27.7. Low-Pressure Heater 2

$$h_{\ell p2i} = h_{\ell p2i}(t) = h_{c1}[t - (M_{fw1}/W_{c2})] \qquad (2\text{-}269)$$

$$h'_{\ell p2} = B_2[(W_{c2} h_{\ell p2i} + W_2 h_2 + W_{3d} h_{3d} - W_{2d} h_{2d})/W_{c2}] + \bar{B}_2 h_{\ell p2i} \qquad (2\text{-}270)$$

$$h_{2d} = F_3(W_{c2}) \qquad (2\text{-}271)$$

$$\frac{d}{dt}\, h_{\ell p2} = \frac{1}{\tau_{\ell p2}}\, (h'_{\ell p2} - h_{\ell p2}) \qquad (2\text{-}272)$$

$$\frac{d}{dt}\, W_{2d} = \frac{1}{\tau_{\ell p2}}\, (W_2 + W_{3d} - W_{3d}) \qquad (2\text{-}273)$$

2.27.8. Low-Pressure Heater 1

$$h_{1D} = F_1(W_{c1}) \qquad (2\text{-}274)$$

$$h_c = F_2(W_{c1}) \qquad (2\text{-}275)$$

$$h'_{\ell p1} = B_1[(W_{c1} h_{c1} + W_1 h_1 + W_{2d} h_{2d} - W_{1d} h_{1d})/W_{c1}] + \bar{B}_1 h_c \qquad (2\text{-}276)$$

$$\frac{d}{dt}\, h_{\ell p1} = \frac{1}{\tau_{\ell p1}}\, (h'_{\ell p1} - h_{\ell p1}) \qquad (2\text{-}277)$$

$$\frac{d}{dt} W_{1d} = \frac{1}{\tau_{\ell f 1}} (W_1 + W_{2d} - W_{1d}) \qquad (2\text{-}278)$$

2.27.9. Low-Pressure Heater Drain Tank

$$h_{fptc} = F_7(W_t) \qquad (2\text{-}279)$$

$$\frac{d}{dt} W_{fpt} = \frac{1}{\tau_{fpt}} (W_{8b} - W_{fpt}) \qquad (2\text{-}280)$$

$$h'_{d\ell p} = \frac{W_{1d} h_{1d} + W_{fpt} h_{fptc}}{W_{1d} + W_{fpt}} \qquad (2\text{-}281)$$

$$\frac{d}{dt} h_{d\ell p} = \frac{1}{\tau_{d\ell p}} (h'_{d\ell p} - h_{d\ell p}) \qquad (2\text{-}282)$$

$$h_{c1} = (W_{d\ell p} h_{d\ell p} + W_{c1} h_{\ell p1})/W_{c2} \qquad (2\text{-}283)$$

Nomenclature

Symbols beginning with the letters P, W, h, and T correspond to pressures, mass flow rates, enthalpies, and temperatures, respectively, as shown in Figure 1-2.

ΔH_{fp}	Enthalpy rise across main feedwater pumps, Btu/lbm
M_{fw1}	Liquid mass between feedwater heaters 1 and 2, lbm
M_{fw2}	Liquid mass between feedwater heaters 2 and 3, lbm
M_{fw3}	Liquid mass between feedwater heaters 3 and 4, lbm
M_{fw4}	Liquid mass between feedwater heaters 4 and 5, lbm
M_{fw5}	Liquid mass between feedwater heaters 5 and 6, lbm
M_{fw}	Liquid mass between feedwater heater 6 and the common junction, lbm
M_{fwa}	Liquid mass between the common junction and OTSG-A, lbm
M_{fwb}	Liquid mass between the common junction and OTSG-B, lbm
W_{pg5m}, W_{pg6m}	Maximum pegging steam flow rate, lbm/s
$\tau_{\ell p1}, \tau_{\ell p2}, \tau_{\ell p3}$	Low-pressure feedwater heater time constants, s
τ_{hp4}, τ_{hp5}	High-pressure feedwater heater time constants, s

τ_{dhp} High-pressure drain tank time constant, s

τ_{dlp} Low-pressure drain tank time constant, s

Logical Variables

$B_1 = 0$ when heater 1 is failed (2-284)

$B_2 = 0$ when heater 2 is failed (2-285)

$B_3 = 0$ when heater 3 is failed (2-286)

$B_4 = 0$ when heater 4 is failed (2-287)

$B_5 = 0$ when heater 5 is failed (2-288)

Tables of Steady-State Data

$F_1(W_{c1}) = h_{1d}$ Vs W_{c1} (2-289)

$F_1(W_{c2}) = h_{4d}$ Vs W_{c2} (2-290)

$F_2(W_{c2}) = h_{3d}$ Vs W_{c2} (2-291)

$F_3(W_{c2}) = h_{2d}$ Vs W_{c2} (2-292)

$F_1(W_{fw}) = h_{6d}$ Vs W_{fw} (2-293)

$F_2(W_{fw}) = h_{5d}$ Vs W_{fw} (2-294)

$F_3(W_t) = W_4$ Vs W_t (2-295)

$F_4(W_t) = h_{hp6}$ Vs W_t (2-296)

$F_5(W_t) = h_{hp5}$ Vs W_t (2-297)

$F_7(W_t) = h_{fpt}$ Vs W_t (2-298)

$F_3(W_u) = h_{ms}$ Vs W_7 (2-299)

$F_1(W_8) = W_3$ Vs W_9 (2-300)

$F_2(W_8) = W_2$ Vs W_8 (2-301)

$F_3(W_8) = W_1$ Vs W_8 (2-302)

$F_1(W_{8d}) = h_{rh1d}$ Vs W_{8d} (2-303)

2.28. Secondary System Option

Nearly every nuclear power station has some differences in its secondary system. The system described in sections 2.26 and 2.27 can be considered typical and is based on the TVA Bellefonte design. A secondary system option is available which slightly modifies the system to represent the SMUD Rancho Seco secondary system.

The SMUD unit differs from TVA in that it does not have a low-pressure drain tank; drain flows from the feedwater pump turbine and low-pressure heater 1 are simply bypassed directly to the condenser. In addition to this, the drain flow from high-pressure heater 5 and the moisture separator flows are directed to high-pressure heater 4 rather than directly to the high-pressure drain tank. These minor differences result in the following changes to the equations presented in sections 2.26 and 2.27:

Equation 2-213

$$W_{c1} = W_{c2} \tag{2-304}$$

Equation 2-214

$$\frac{d}{dt} W_{dhp} = \frac{1}{\tau_{dhp}} (W_{4d} - W_{dhp}) \tag{2-305}$$

Equation 2-215 is eliminated and 2-217 applies all the time.

Equation 2-224

$$\frac{d}{dt} L_{hw} = \left(\frac{1}{A\rho}\right)_{hw} (W_9 + W_{bp} + W_{1d} + W_{fpt} - W_{c1}) \tag{2-306}$$

Equation 2-257

$$h'_{hp4} = B_4[(W_{c2}h_{hp4i} + W_4h_4 + W_{ms}h_{ms} + W_{5d}h_{5d} - W_{4d}h_{4d})/W_{c2}] + \bar{B}_4h_{hp4i} \tag{2-307}$$

Equation 2-259

$$\frac{d}{dt} W_{4d} = \frac{1}{\tau_{hp4}} (W_4 + W_{ms} + W_{5d} - W_{4d}) \tag{2-308}$$

Equation 2-261

$$h'_{hdp} = h_{4d} \quad (2\text{-}309)$$

Equations 2-281 and 2-282 eliminated.

Equation 2-283

$$h_{c1} = h_{\ell p1}. \quad (2\text{-}310)$$

2.29. Integrated Control System

The POWER TRAIN model is controlled by a digital simulation of the integrated control system (ICS). The ICS is modeled using a simulation language that provides a number of operational elements, such as those found in a typical ICS. The ICS model is formed by specifying the interconnection of these various elements to represent the controllers and trip modules of a typical ICS. The ICS model is extensive, including all the major subsystems

- Unit load demand subsystem.
- Integrated master control subsystem.
- Steam generator feedwater control subsystem.
- Reactor control subsystem.

The ICS model produces demand signals for the reactor and control rod drive (CRD) models, the steam generator (feedwater) model, and the turbine pressure model. The demand signals are driven by a unit load demand (ULD) signal established by the operator. The ULD station allows ramps and steps of varying degrees, automatic demand limits, and runbacks under conditions when generation is limited. The digital simulation of the ICS handles both automatic coordination of the various demand signals and manual control of the signals by the operator. The ICS simulation has the capability to automatically establish a tracking mode, whenever conditions arise that would cause unit tracking, and to perform automatic demand runback and transfer of control signals when the tracking mode is established. The model operator has full control over the ICS simulation from the analog console and can sequence and initiate the desired combination of transients by pushing the appropriate buttons and entering parameters on the keyboard. The model is designed to perform all transients without restoring to modifications in the hybrid program.

Trips and abnormal conditions that can be initiated from the mini panel include the following:

- Manual reactor trip.
- RC pump trips (all pumps, one pump, or two pumps).
- Feedwater pump trip (one or two).
- Turbine trip.
- Loss of electrical load.
- Pressurizer heater failure.
- Enable/disable flux/flow trip of reactor.
- Enable/disable high primary pressure trip of reactor.
- Manual failure of drain and condensate pumps in feedwater train.

In addition, the operator of the simulation may select, via the mini panel, either manual or automatic control of the following controllers and final control elements:

- Turbine bypass valves.
- Turbine control valves.
- Reactor-steam generator demand.
- Feed valve demand (A or B).
- Feed valve position (A or B).
- Reactor demand.
- Control rod position.
- Unit load demand.

2.30. Summary of Limitations of POWER TRAIN Simulation

The POWER TRAIN program was developed specifically for analysis of the dynamic response of the B&W nuclear power station for both normal and off-normal operating conditions. Typical types of analysis performed with POWER TRAIN (not necessarily limited to) are given below.

1. Control system optimization and evaluation of new control concepts.
2. Control system failure and system effects analysis.
3. Protection system (trip) failures and system effects analysis.
4. Verification of warranteed plant maneuvering capability.
5. System analysis and scoping studies for many anticipated transients, e.g.,

a. Turbine trip with and without reactor trip.

b. Load rejection.

c. Grid frequency upsets.

d. Loss of feedwater (complete or partial).

e. Loss of feedwater heaters.

f. Condensate system failures.

g. Spurious reactor trip.

h. Primary pump trips.

The scaling requirements of the analog portion of the simulation result in the following limits on the magnitude and rate-of-change of the variables specified below (if these limits are exceeded, the code results are not valid):

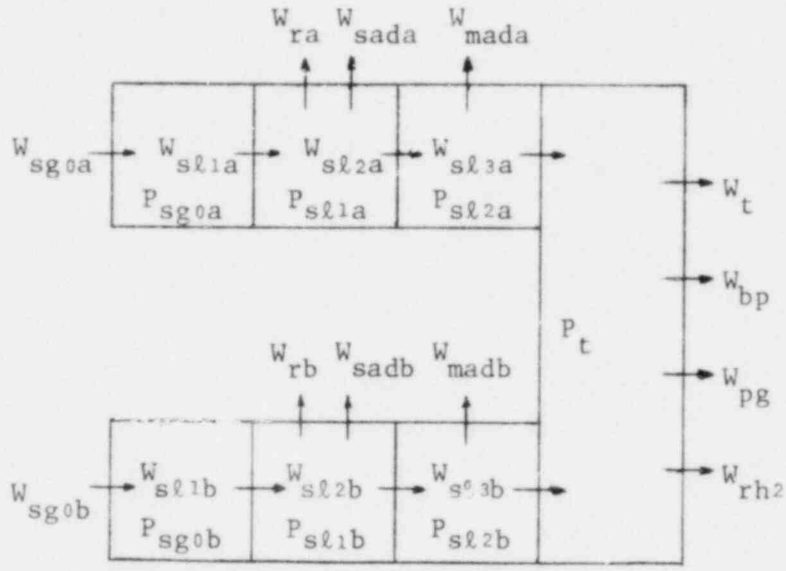
- Fluid temperatures: 350 to 650F
- Cladding and fuel temperatures: 350 to 3000 psia
- Primary flow rate: 5 to 200%
- Primary system pressure: 1500 to 3000 psia
- Reactor power level: 3 to 120%
- Secondary fluid enthalpy: 300 to 1400 Btu/lb
- Secondary fluid density: 0 to 60 lb/ft^3
- Secondary flow rate: 0 to 115%
- Secondary pressure: 500 to 1500 psia
- Secondary pressure maximum rate-of-change: 100 psia/s
- Secondary flow rate maximum rate-of-change: 100%/s

In addition to the limits above, limitations on the maximum rate of change of the enthalpy increase on the secondary side of the steam generator simulation will result in a higher than normal heat removal rate for zero secondary flow conditions. Therefore, when the steam generator is dry, caution should be exercised in the interpretation of code results. It should always be kept in mind that this version of POWER TRAIN was developed as a power range (15-100%) analysis tool and is not suitable for analysis of low power (<15%) operation (e.g., emergency feedwater control studies) despite the fact that it will operate at low power levels.

Several other code limitations apply and are listed below. These limitations are necessary because of mathematical model assumptions, scaling limitations on the analog portion of the simulation, and the time step employed.

1. The pressurizer cannot go solid or completely empty. If either condition occurs, the code results from that point are invalid.
2. Two-phase conditions in the primary system are not modeled.
3. Operation of the secondary system which results in the introduction of saturated fluid conditions in the steam lines is not valid.
4. The code is not capable of analyzing system piping breaks in the primary system.
5. No emergency safety system features (such as high-pressure injection) are included in the code capability.
6. It is not valid for analysis of fast primary system reactivity excursions (e.g., rod ejections).

Figure 2-1. Steam Line Nodalization



3. INPUT/OUTPUT

3.1. Input Data

Input data are required by each computer involved in the POWER TRAIN simulation. Input for POWER TRAIN consists of the following:

1. BLOCK DATA variables — Data for the portion of the simulation implemented on the CDC 1700 are input via a BLOCK DATA subprogram. All BLOCK DATA variables are defined in Table 3-1.

2. Tabular data — Tabular data required by the simulation are loaded onto the array processor via a program executed on the 1700. Tabular data are defined in Table 3-2.

3. Potentiometer setting data — COMANCHE, a program executed on the CDC 1700, sets all potentiometers based on user-supplied parameters. The parameters needed by COMANCHE are defined in Table 3-3.

4. ICS simulation — The ICS is simulated on the EAI-640 using the ISL simulation language. The use of ISL is defined in reference 1.

3.2. Output Data

Non-real-time output, such as a dump of input data, is printed on the teletype; real-time output is plotted on strip chart recorders. Table 3-4 lists the simulation variables the user may specify to be plotted. In addition to those variables, any variable calculated on either analog computer may be plotted by directly patching the variable to the strip chart recorder. Some commonly used analog variables are listed in Table 3-5.

Table 3-1. BLOCK DATA Variables

No.	Name	Description
1	ALPHD	Doppler coefficient, Δk/k/°F
2	ALPHM	Moderator coefficient, Δk/k/°F
3	BETA(1)	Delayed neutron fraction, group 1
4	BETA(2)	Delayed neutron fraction, group 2
5	BETA(3)	Delayed neutron fraction, group 3
6	BETA(4)	Delayed neutron fraction, group 4
7	BETA(5)	Delayed neutron fraction, group 5
8	BETA(6)	Delayed neutron fraction, group 6
9	BETA(7)	Total delayed neutron fraction
10	CLAM(1)	Decay constant, group 1, s^{-1}
11	CLAM(2)	Decay constant, group 2, s^{-1}
12	CLAM(3)	Decay constant, group 3, s^{-1}
13	CLAM(4)	Decay constant, group 4, s^{-1}
14	CLAM(5)	Decay constant, group 5, s^{-1}
15	CLAM(6)	Decay constant, group 6, s^{-1}
16	CPCM	Cladding mass × cladding specific heat, Btu/°F
17	CPFM	Fuel mass × water specific heat, Btu/°F
18	CPWM	Water mass × water specific heat, Btu/°F
19		Not used
20		Not used
21	POWERI	Initial power, %
22	RHOALP	Fluid density, SG A lower plenum, lb/ft^3
23	RHOAUP	Fluid density, SG A upper plenum, lb/ft^3
24	RHOBLP	Fluid density, SG B lower plenum, lb/ft^3
25	RHOBUP	Fluid density, SG B upper plenum, lb/ft^3
26	RHOCLA	Fluid density, cold leg A, lb/ft^3
27	RHOCLB	Fluid density, cold leg B, lb/ft^3
28	RHOHLA	Fluid density, hot leg A, lb/ft^3
29	RHOHLB	Fluid density, hot leg B, lb/ft^3
30	RHOLP	Fluid density, vessel upper plenum, lb/ft^3
31	RHOUP	Fluid density, vessel lower plenum, lb/ft^3
32	RRM	Maximum rod worth, Δk/k
33	RSRM	Maximum safety rod worth, Δk/k

Table 3-2. (Cont'd)

No.	Name	Description
34	TAUDK	Decay heat time constant, s
35	TAUM	Rod drive time constant, s
36	TAUTS	Avg temp thermal time constant, s
37	TCIN	Initial cladding temperature F,
38	TFIN	Initial fuel temperature, F
39	TWIN	Initial coolant temperature, F
40	TWIIN	Initial core inlet temperature, F
41	VCORE	Core coolant volume, ft^3
42	VCLA	Cold leg A volume, ft^3
43	VCLB	Cold leg B volume, ft^3
44	VHLA	Hot leg A volume, ft^3
45	VHLB	Hot leg B volume, ft^3
46	VLP	Vessel lower plenum volume, ft^3
47	VUP	Vessel upper plenum volume, ft^3
48	VSGA	SG A primary side volume, ft^3
49	VSGB	SG B primary side volume, ft^3
50	VSGALP	SG A lower plenum volume, ft^3
51	VSGAUP	SG A upper plenum volume, ft^3
52	VSGBLP	SG B lower plenum volume, ft^3
53	VSGBUP	SG B upper plenum volume, ft^3
54		Not used
55	WCBP	Core bypass flow rate, pps
56	WPA	Loop A flow rate, pps
57	WPB	Loop B flow rate, pps
58	XMAX	Maximum rod insertion, in.
59	XRO	Initial rod position, % withdrawn
60	VRS	Normal rod velocity, in./s
61	SGSIZE	No. of sample points in steam generators
62	K201	Rate of change of primary fluid density with pressure at constant temperature, lb/ft^3-psia; use is optional
63	K202	Not used
64	K203	Level to volume equivalent in cylindrical section of pressurizer, $in./ft^3$

Table 3-1. (Cont'd)

No.	Name	Description
65	K204	Level equivalent of volume of pressurizer below lower sensing tap, ft^3
66	K205	Total pressurizer volume, ft^3
67	K206	Equivalent coefficient for boiloff in pressurizer, lb/s-°F
68	K207	Not used
69	K208	Pressurizer wall temperature, F
70	K209	Low level cut-off for pressurizer heaters, in.
71	K210	Cut-on point for pressurizer heater 1, psia
72	K211	Proportional gain for variable output of heater 1, Btu/s-psia
73	K212	Low limit of prop. band of heater 1, psia
74	K213	Maximum capacity of heater 1, Btu/s
75	K214	Heater 2 capacity, Btu/s
76	K215	Heater 2 cut-on point, psia
77	K216	Heater 2 cut-off point, psia
78	K217	Heater 3 capacity, Btu/s
79	K218	Heater 3 cut-on point, psia
80	K219	Heater 3 cut-off point, psia
81	K220	Heater 4 capacity, Btu/s
82	K221	Heater 4 cut-on point, psia
83	K222	Pressurizer code safety valve flow rate, pps
84	K223	Static head, RC surge tap to pressurizer, psia
85	K224	Pressurizer code safety cut-on point, psia
86	K225	Pressurizer code safety cut-off point, psia
87	K226	Equivalent dynamic ΔP in surge line, $psi/(pps)^2$
88	K227	Conversion factor for converting level error to makeup flow demand, pps/in.
89	K228	Proportional gain for level error, dimensionless
90	K229	Pressurizer level setpoint, in.
91	K230	Integral gain for level error, s^{-1}
92	K231	Maximum RC makeup flow, pps
93	K232	Heater 4 cut-off point, psia
94	K233	Maximum spray flow, pps
95	K234	Spray cut-on point, psia

Table 3-1. (Cont'd)

No.	Name	Description
96	K235	Spray cut-off point, psia
97	K236	Letdown flow rate, pps
98	K237	Heater 1 cutoff point, psia
99	K238	Power-operated relief valve flow, pps
100	K239	Power-operated relief cut-on point, psia
101	K240	Power-operated relief cut-off point, psia
102	K241	Pressurizer heater time constant, s
103	K242	Pressurizer spray valve velocity, % stroke/s
104	PLAMBA	Mixing factor for pressurizer lower liquid region, s^{-1}
105		Not used
106		Not used
107	PTAUMU	Makeup flow time constant, s
108	PUAPM2	Steam-to-wall condensation heat transfer coeff × area, Btu/s-°F
109	PUAPS	Steam-to-liquid condensation heat transfer coeff × area, Btu/s-°F
110	MPW1	Initial mass in pressurizer lower liquid region, lb
111	MPW2	Initial mass in pressurizer upper liquid region, lb
112	HPW1	Initial enthalpy in pressurizer lower liquid region, Btu/lb
113	HPW2	Initial enthalpy in pressurizer upper liquid region, Btu/lb
114	MPS	Initial mass in pressurizer steam region, lb
115	HS	Initial enthalpy in pressurizer steam region, Btu/lb
116	MRC	Total mass in RC system (excluding pressurizer), lb
117	VRC	Total volume of RC system (excluding pressurizer), ft^3
118	POWMIN	Initial power level after reactor trip, % of full power
119	DECAY	Decay heat level, % of full power
120	BOR	Reactivity insertion change due to boration or deboration, $\Delta k/k/h$
121	WMAX	Maximum secondary flow for analog scaling purposes, pps
122	DCA	Equivalent cross sectional flow area for downcomer, ft^2
123	DCL	Total height of downcomer, ft
124	RASP	Friction and shock loss multiplier for steam aspiration port, psi/pps^2
125	RORF	Friction and shock loss multiplier for inlet orifice, psi/pps^2

Table 3-1. (Cont'd)

No.	Name	Description
126	RFDC	Feedwater inlet nozzle momentum multiplier, psi/pps^2
127	DELT	Time step, s
128	ALPHA	Time scaling factor, computer time/problem time, dimensionless
129	PNOM	Nominal secondary pressure for analog scaling purposes, psia
130	PMAX	Maximum deviation (+ or -) from nominal secondary pressure, psi
131	TNOM	Nominal primary or secondary temperature for analog scaling purposes, F
132	TMAX	Maximum deviation (+ or -) from nominal temperature, F
133	HNOM	Nominal enthalpy of secondary fluid for analog scaling purposes, Btu/lbm
134	HMAX	Maximum deviation (+ or -) from nominal enthalpy, Btu/lbm
135	TAU6B	Time constant for HP turbine extraction flow to heater 6, s
136	TAU5B	Time constant for HP turbine extraction flow to heater 5, s
137	TAU4	Time constant for HP turbine extraction flow to heater 4, s
138	TAUGRH	Time constant for HP turbine extraction flow to reheater 1, s
139	TAU7	Time constant for HP turbine exist flow to moisture separator, s
140	TAU3	Time constant for LP turbine extraction flow to heater 3, s
141	TAU2	Time constant for LP turbine extraction flow to heater 2, s
142	TAU1	Time constant for LP turbine extraction flow to heater 1, s
143	TAU9	Time constant for LP turbine exit flow to condenser, s
144	TAURH1	Time constant for reheater 1 exit flow, s
145	TAURH2	Time constant for reheater 2 exit flow, s
146	TAUHP6	Time constant for HP feedwater heater 6, s
147	TAUHP5	Time constant for HP feedwater heater 5, s
148	TAUHP4	Time constant for HP feedwater heater 4, s
149	TAULP3	Time constant for LP feedwater heater 3, s
150	TAULP2	Time constant for LP feedwater heater 2, s

Table 3-1. (Cont'd)

No.	Name	Description
151	TAULP1	Time constant for LP feedwater heater 1, s
152	TAUDHP	Time constant for HP drain tank flow, s
153	TAUDLP	Time constant for LP drain tank flow, s
154	TAUFPT	Time constant for feedwater pump turbine flow, s
155	H6	Enthalpy of extraction flow 6, Btu/lbm
156	H6B	Enthalpy of extraction flow 6B, Btu/lbm
157	HPG6	Enthalpy of "pegging" steam (WPG6) to HP feedwater, heater 6
158	HRH2D	Enthalpy of reheater 2 drain flow (WRH2) Btu/lbm
159	H5	Enthalpy of steam to HP feedwater heater 5, Btu/lbm
160	H5B	Enthalpy of HP turbine extraction flow W5B, Btu/lbm
161	HPG5	Enthalpy of "pegging" steam (WPG5) to HP feedwater heater 5, Btu/lbm
162	H4	Enthalpy of HP turbine extraction flow W4, Btu/lbm
163	HRH1	Enthalpy of reheater 1 flow WRH1, Btu/lbm
164	HRH2	Enthalpy of reheater 2 flow WRH2, Btu/lbm
165	H8B	Enthalpy of feedwater pump turbine flow W8B, Btu/lbm
166	H3	Enthalpy of LP turbine extraction flow W3, Btu/lbm
167	H2	Enthalpy of LP turbine extraction flow W2, Btu/lbm
168	H1	Enthalpy of LP turbine extraction flow W1, Btu/lbm
169	DHFP	Enthalpy rise across main feedwater pumps, Btu/lbm
170	WPG6M	Maximum "pegging" steam to HP feedwater heater 6, pps
171	WPG5M	Maximum "pegging" steam to HP feedwater heater 5, pps
172	MFW1	Nominal liquid mass from feedwater heater 1 to 2, lbm
173	MFW2	Nominal liquid mass from feedwater heater 2 to 3, lbm
174	MFW3	Nominal liquid mass from feedwater heater 3 to 4, lbm
175	MFW4	Nominal liquid mass from feedwater heater 4 to 5, lbm
176	MFW5	Nominal liquid mass from feedwater heater 5 to 6, lbm
177	MFW	Nominal liquid mass from feedwater heater 6 to feedwater junction A-B, lbm
178	MFWA	Nominal liquid mass from feedwater junction A-B to OTSG-A, lbm
179	MFWB	Nominal liquid mass from feedwater junction A-B to OTSG-B, lbm

Table 3-1. (Cont'd)

No.	Name	Description
180	LAC1	Equivalent length/area ratio for condensate flow WC1, ft^{-1}
181	LAC2	Equivalent length/area ratio for condensate flow WC2, ft^{-1}
182	ARHOHW	Hotwell area times density, lbm/ft
183	PC	Pressure at condenser, psia
184	PC1	Pressure at junction of LP drain tank flow and heater 1, psia
185	PC2	Pressure at junction of HP drain tank flow and heater 4, psia
186	LHW	Hotwell level, ft
187	DPEC1	Elevation pressure differential between hotwell and PC2, psi
188	DPEC2	Elevation pressure differential between PC2 and PC1, psi
189	DPHWL	Initial pressure differential due to hotwell level, psi
190	HWLMIN	Minimum hotwell level allowed, ft
191	WC1	Condensate flow, condenser to junction at PC1, pps
192	WC2	Condensate flow, junction of PC1 to PC2, pps
193	DELTNO	Number of time steps between updates of secondary system temperature delays (dimensionless)
194	DPSGMX	Maximum steam generator secondary pressure drop (psi) for analog scaling purposes
195	TFWOFF	Arbitrary feedwater temperature offset, F
196-200	DDUM(1) DDUM(5)	Dummy variables, no used

Table 3-2. Tabular Data

No.	Function	Description
1	$\%R_r = F_r(\%X_r)$	Control rod worth (%) Vs insertion (%)
2	$\%R_{sr} = F_{sr}(t)$	Safety rod worth (%) Vs time after trip (s)
3	$W_{pa} = F_{1a}(t)$	One-primary-pump trip, loop A flow rate (pps) Vs time after trip (s)
4	$W_{pb} = F_{1b}(t)$	One-primary-pump trip, loop B flow rate (pps) Vs time after trip (s)
5	$W_{pa} = F_{2a}(t)$	Four-primary-pump trip, loop A flow rate (pps) Vs time after trip (s)
6	$W_{pb} = F_{2b}(t)$	Four-primary-pump trip, loop B flow rate (pps) Vs time after trip (s)
7	$W_{pa} = F_{3a}(t)$	Two-primary-pump trip in same loop, loop A flow rate (pps) Vs time after trip (s)
8	$W_{pb} = F_{3b}(t)$	Two-primary-pump trip in same loop, loop B flow rate (pps) Vs time after trip (s)
9	$W_{pa} = F_{4a}(t)$	Two-primary-pump trip, 1/loop, loop A flow rate (pps) Vs time after trip (s)
10	$W_{pb} = F_{4b}(t)$	Two-primary-pump trip, 1/loop, loop B flow rate (pps) Vs time after trip (s)
11	$W_{6b} = F_1(W_t)$	Extraction flow W_{6b} (pps) to HP heater 6 Vs turbine throttle flow W_t (pps)
12	$W_{5b} = F_2(W_t)$	Extraction flow W_{5b} (pps) to HP heater 5 Vs turbine throttle flow W_t (pps)
13	$W_4 = F_3(W_t)$	Extraction flow W_4 (pps) to HP heater 4 Vs turbine throttle flow W_t (pps)
14	$h_{hp6} = F_4(W_t)$	Enthalpy h_{hp6} (Btu/lb) of FW leaving HP heater 6 Vs turbine throttle flow W_t (pps)
15	$h_{hp5} = F_5(W_t)$	Enthalpy h_{hp5} (Btu/lb) of FW leaving HP heater 5 Vs turbine throttle flow W_t (pps)
16	$W_{ms} = F_1(W_7)$	Moisture separator flow W_{ms} to high pressure drain tank (pps) Vs flow W_7 (pps) out of HP turbine
17	$M_{s7} = F_2(W_7)$	Torque developed M_{s7} (%) by HP turbine Vs steam flow W_7 (pps) out of HP turbine

Table 3-2. (Cont'd)

No.	Function	Description
18	$h_{ms} = F_3(W_7)$	Enthalpy (Btu/lb) of moisture separator flow W_{ms} Vs flow W_7 (pps) out of HP turbine
19	$W_3 = F_1(W_8)$	Extraction flow W_3 (pps) out of LP turbine Vs flow W_8 (pps) into LP turbine
20	$W_2 = F_2(W_8)$	Extraction flow W_2 (pps) out of LP turbine Vs flow W_8 (pps) into LP turbine
21	$W_1 = F_3(W_8)$	Extraction flow W_1 (pps) out of LP turbine Vs flow W_8 (pps) into LP turbine
22	$W_{rh2} = F_1(W_{8c})$	Live steam extraction flow W_{rh2} (pps) to reheater 2 Vs flow W_{8c} (pps) through reheater
23	$W_{8b} = F_2(W_{8c})$	Feedwater pump turbine flow W_{8b} (pps) Vs flow W_{8c} (pps) through reheater 2
24	$W_{rh1} = F_6(W_t)$	Extraction flow W_{rh1} (pps) from HP turbine to reheater 1 Vs turbine throttle flow W_t (pps)
25	$h_{rh1d} = F_1(W_{8d})$	Enthalpy h_{rh1d} (Btu/lb) of reheater 1 drain flow W_{rh1} (pps) Vs flow W_{8d} (pps) through reheater 1
26	$M_{s9} = F_1(W_9)$	Torque developed M_{s9} (%) by LP turbine Vs steam flow W_9 (pps) out of LP turbine
27	$h_{6d} = F_1(W_{fw})$	Enthalpy h_{6d} (Btu/lb) of HP feedwater heater 6 drain flow W_{6d} (pps) Vs feedwater flow W_{fw} (pps)
28	$h_{5d} = F_2(W_{fw})$	Enthalpy h_{5d} (Btu/lb) of HP feedwater heater 5 drain flow W_{6d} (pps) Vs feedwater flow W_{fw} (pps)
29	$h_{4d} = F_1(W_{c2})$	Enthalpy h_{4d} (Btu/lb) of HP feedwater heater 4 drain flow W_{4d} (pps) Vs condensate flow W_{c2} (pps)
30	$h_{3d} = F_2(W_{c2})$	Enthalpy h_{3d} (Btu/lb) of LP feedwater heater 3 drain flow W_{3d} (pps) Vs condensate flow W_{c2} (pps)
31	$h_{2d} = F_3(W_{c2})$	Enthalpy h_{2d} (Btu/lb) of LP feedwater heater 2 drain flow W_{2d} (pps) Vs condensate flow W_{c2} (pps)
32	$h_{1d} = F_1(W_{c1})$	Enthalpy h_{1d} (Btu/lb) of LP feedwater heater 1 drain flow W_{1d} (pps) Vs condensate flow W_{c1} (pps)
33	$\Delta P_{hwp} = f_2(W_{c1})$	Hotwell pump developed head (psi) Vs flow W_{c1} (pps) through pump
34	$\Delta P_{cp} = F_3(W_{c1})$	Condensate pump developed head (psi) Vs flow W_{c1} (pps) through pump

Table 3-2. (Cont'd)

No.	Function	Description
35	$h_c = F_4(W_{c1})$	Enthalpy h_c (Btu/lb) at LP heater 1 inlet Vs flow W_{c1} (pps) through heater
36	$h_{fpt} = F_7(W_t)$	Enthalpy h_{fpt} (Btu/lb) of feedwater pump turbine flow W_{fpt} (pps) Vs turbine throttle flow W_t (pps)

Table 3-3. COMANCHE Parameter Data

ALPHA	Time scale factor, computer time/problem time
ALPRIP	Reciprocal of ALPHA
AM	Steam generator metal cross-sectional area, ft^2
AP	Steam generator primary cross-sectional area, ft^2
AS	Steam generator secondary cross-sectional area, ft^2
B	Drift velocity multiplier
BETA	Spatial integration rate of steam generator, ft/s
COND	Tube metal conductivity × 100, Btu/s-F-ft
CPM	Specific heat of steam generator metal, Btu/lb-F
CPW	Specific heat of primary fluid, Btu/lb-F
CVA	Percent feed valve A coefficient at WFWA and XCVA
CVB	Percent feed valve B coefficient at WFWB and XCVB
CVSF	Scale factor (chosen to make DP = 1000 when A33, A38 are limited)
CO	Void fraction distribution coefficient
C5P1	Drift velocity times R8/B, at P1 pressure
C5P2	Drift velocity times R4/B, at P2 pressure
C5P3	Drift velocity times R9/B, at P3 pressure
DNBX	Steam quality at DNB
DPCVA	Feed valve A pressure drop at WFWA and XCVA, psia
DPCVB	Feed valve B pressure drop at WFWB and XCVB, psi
DPCVMX	Max feed valve pressure drop, psi
DPEFM	Elevation pressure gain, main feed line, psi
DPEFWA	Elevation pressure drop, feed line A, psi
DPEFWB	Elevation pressure drop, feed line B, psi

Table 3-3. (Cont'd)

DPFA	Friction pressure drop, feed line A, psi
DPFB	Friction pressure drop, feed line B, psi
DPFW	Friction pressure drop, main feed line, psi
DPSFSC	Subcooled region unrecovered delta-P, psi/ft (at WFW)
DPSFBC	Boiling region unrecovered delta-P, psi/ft (at WFW)
DPSFSH	Superheat region unrecovered delta-P, psi/ft (at WFW)
DPSMAX	Maximum SG pressure drop, psi
DPUMPR	Main feed pump rated heat at rated flow and speed, D(ρ)/DP of steam at constant enthalpy (1234.7)
DT	Major cycle time of hybrid steam generator-model
EBPLOT	Delay time for emergency feed system, s
ECV	Control valve position error for maximum velocity, %
EXP1	EXP(-P1/1260) for heat flux calculation
EXP2	EXP(-P2/1260) for heat flux calculation
EXP3	EXP(-P3/1260) for heat flux calculation
FW1	Minimum feedwater temperature, F
FW2	Normal feedwater temperature, F
FW3	Maximum feedwater temperature, F
F1	Saturated fluid enthalpy at P1 pressure, Btu/lb
F2	Saturated fluid enthalpy at P2 pressure, Btu/lb
F3	Saturated fluid enthalpy at P3 pressure, Btu/lb
G1	Saturated gas enthalpy at P1 pressure, Btu/lb
G2	Saturated gas enthalpy at P2 pressure, Btu/lb
G3	Saturated gas enthalpy at P3 pressure, Btu/lb
HFW	Feedwater enthalpy, Btu/lbm
HMAX	Scaling parameter for secondary enthalpy (range), Btu/lb
HNOM	Scaling parameter for secondary enthalpy (midpoint), Btu/lb
H1	Subcooled enthalpy at FW1 and P2, Btu/lb
H2	Subcooled enthalpy at FW2 and P2, Btu/lb
H4	Superheated enthalpy at T4 and P2, Btu/lb
L	Active length of steam generator, ft
LAFW	Length to area ratio, main feed line, 1/ft
LAFWA	Length to area ratio, feed line A, 1/ft
LAFWB	Length to area ratio, feed line B, 1/ft
LA1A	Steam line 1A length to area ratio, 1/ft

Table 3-3. (Cont'd)

LA1B	Steam line 1B length to area ratio, 1/ft
LA2A	Steam line 2A length to area ratio, 1/ft
LA2B	Steam line 2B length to area ratio, 1/ft
LA3A	Steam line 3A length to area ratio, 1/ft
LA3B	Steam line 3B length to area ratio, 1/ft
LDPSAV	Level in SG where total DP is sampled and saved, ft
MMAX	Maximum steam dump to atmosphere per SG
NPUMPF	Main feed pump final speed after trip, %
NPUMPI	Main feed pump initial speed, %
NPUMPR	Main feed pump rated speed, %
PBP	Bypass valve pressure at rated flow (WBP), psia
PC2	Normal feedpump suction pressure, psia
PC2REF	Feed pump suction pressure at which head was measured
PDMAX	Scaling parameter for PSL2A-PT and PSL2B-PT, psi
PESF	Low-pressure setpoint for containment isolation, psia
PMAD	Main atmospheric dump valve rated pressure, psia
PMAX	Scaling parameter for secondary pressure (range and midpoint), psia
PNPSH	Main feed pump trip setpoint on NPSH, psia
POWER	Neutron power, %
PP	Primary wetted perimeter, ft
PRC	Primary system pressure, psia, in pressurizer steam space
PS	Secondary wetted perimeter, ft
PSAD	Small atmospheric dump valve pressure, psia
PSGOA	Steam generator A outlet pressure, psia
PSGOB	Steam generator B outlet pressure, psia
PSL1A	Pressure at steam line node 1A, psia
PSL1B	Pressure at steam line node 1B, psia
PSL2A	Pressure at main atmospheric dump valve A, psia
PSL2B	Pressure at main atmospheric dump valve B, psia
PSV	Steam line safety valve rating pressure (103% accumulation), psia
PT	Pressure at turbine valve, psia
P1	Minimum expected pressure, psia
P2	Expected normal pressure, psia
P3	Maximum expected pressure, psia

Table 3-3. (Cont'd)

RESET1	Reset pressure of lowest bank of steam line safety valve, s
RM	Steam generator metal density, lb/ft^3
RP	Steam generator primary density, lb/ft^3
RSH1	Steam density at P1 and enthalpy at P2 and T4
RSH3	Steam density at P3 and enthalpy at P2 and T4
R1	Subcooled density at FW1 and P2, lb/ft^3
R2	Saturated fluid density at T2 and P2, lb/ft^3
R3	Saturated fluid density at T3 and P3, lb/ft^3
R4	Saturated gas density at T2 and P2, lb/ft^3
R5	Superheated density at T4 and P2, lb/ft^3
R6	Superheated density at T4 and P1, lb/ft^3
R7	Saturated fluid density at T1 and P1, lb/ft^3
R8	Saturated gas density at T1 and P1, lb/ft^3
R9	Saturated gas density at T3 and P3, lb/ft^3
SGCOMP	=1 if SG component option is used (=0 otherwise)
STROKE	Feedwater control valve stroke time to open, s
TAUCV	Feed valve motor time constant, s
TAUSV	Main steam safety valve opening and closing time constant, s
TEMPCL	Cold leg temperature, F
TEMPHL	Hot leg temperature, F
TMAX	Scaling parameter for secondary temperature (range), F
TNOM	Scaling parameter for secondary temperature (midpoint), F
TFWI	FW line isolation valve closing time, s
TSLI	Steam line isolation valve closing time, s
T1	Saturation temperature at P1 pressure, F
T2	Saturation temperature at P2 pressure, F
T3	Saturation temperature at P3 pressure, F
T4	Expected full power steam temperature, F
UPM	Primary-metal heat transfer coefficient, $Btu/s-ft^2-°F$
USC	Secondary subcooled heat transfer coefficient (at WFW), $Btu/s-ft^2-°F$
UNBMUL	MUL on Thom's coefficient
USH	Secondary superheat heat transfer coefficient (at WFW), $Btu/s-ft^2-°F$
V1A	Steam line 1A volume, ft^3
V1B	Steam line 1B volume, ft^3

Table 3-3. (Cont'd)

V2A	Steam line 2A volume, ft^3
V2B	Steam line 2B volume, ft^3
V3A	Steam line 3A volume, ft^3
V3B	Steam line 3B volume, ft^3
VT	Volume of turbine line from bypass junction, ft^3
WBP	Total condenser dump at 1050 psia (18% of 2*WFW), lb/s
WEFWA	Emergency feedwater flow, loop A maximum, lb/s
WEFWB	Emergency feedwater flow, loop B maximum, lb/s
WFW	Maximum feedwater flow into steam generator, lb/s
WMAD	Main atmospheric dump valve rated flow at PMAD (per OTSG), lb/s
WMAX	Maximum feedwater flow into steam generator, lb/s
WP	Actual primary flow through steam generator, lb/s
WPMAX	Maximum primary flow rate, lb/s
WRATED	Main feed pump rated flow per pump, lb/s
WSAD	Small atmospheric dump flow at PSAD and XSAD=100 psi
WSV	Steam line safety valve flow (112% of WFW), lb/s
WT	Turbine throttle flow at 100% power
XCVA	Feed valve A initial position, %
XCVB	Feed valve B initial position, %
XC5	Quality for start of ramp C5 to zero
XTV	Turbine valve initial position, %
ZERO	
MS	Full power steam torque, %
FREQ	Normal grid frequency, Hz
FREQDN	Negative change in grid frequency, Hz
FREQUP	Positive change in grid frequency, Hz
KTG1	T-G speed change rate, 50 Hz/9.95 s
KTG2	Normal house load, %
KTG3	T-G torque change per load angle change, 360/64.29
KTG4	T-G damping constant, percent per delta frequency
KTG5	T-G friction and windage torque per Hz, %
MWE	Normal MW electric net to grid, %

Table 3-4. Optional Output Variables

Output No.	Variable name	Description
1	WPA	Loop A RC flow, lb/s
2	WPB	Loop B RC flow, lb/s
3	MPW1	Mass in pressurizer water region 1, lb
4	MPW2	Mass in pressurizer water region 2, lb
5	HPW1	Enthalpy of pressurizer water region 1, Btu/lb
6	HPW2	Enthalpy of pressurizer water region 2, Btu/lb
7	MPS	Mass in pressurizer steam space, Btu/lb
8	HS	Enthalpy of pressurizer steam space, Btu/lb
9	PC1	Pressure of feedwater at feed heater outlet 1, psia
10	PC2	Pressure of feedwater at MFP suction, psia
11	LHW	Water level in condenser hotwell, ft
12	WC1	Feedwater flow at outlet of feed heater 1, lb/s
13	WC2	Feedwater flow upstream of MFP suction, lb/s
14	TAVEM	(TC+TH)/2 at the output of 5 s time constant, F
15	TCA	Loop A RC temperature at SG outlet, F
16	TCB	Same for loop B, F
17	THA	Loop A RC temperature at SG inlet, F
18	THB	Same for loop B, F
19	TRIA	Loop A RC temperature at reactor vessel inlet, F
20	TRIB	Same for loop B, F
21	TRO	RC temperature at reactor core outlet, F
22	TROA	RC temperature at reactor vessel loop A outlet, F
23	TROB	Same for loop B, F
24	TSGIA	Loop A RC temperature at SG tubesheet inlet, F
25	TSGIB	Same for loop B, F
26	TWI	RC temperature at reactor core inlet, F
27	TW	RC temperature at reactor core outlet, F
28	POWER	Reactor core NI power, %
29	PTHERM	Reactor core heat flux at clad surface, %
30	TF	Reactor core fuel temperature, F
31	TC	Reactor core clad temperature, F
32	TW	Reactor core average water temperature, F

Table 3-4. (Cont'd)

Output No.	Variable name	Description
33	XR	Reactor core control rod position, %
34	DELKT	Reactor core total reactivity, Δk/k
35	TSGOA	Loop A RC temperature at SG tubesheet outlet, F
36	TSGOB	Same for loop B, F
37	WP	Total RC flow, lb/s
38	WMU	RC makeup flow, lb/s
39	WSP	Pressurizer spray flow, lb/s
40	WR	Pressurizer safety valve flow, lb/s
41	WPR	Pressurizer power actuated relief valve flow, lb/s
42	LP	Pressurizer water level, in.
43	TPW1	Pressurizer water region 1 temp, F
44	TPW2	Pressurizer water region 2 temp, F
45	ROPS	Pressurizer steam space density, lb/ft^3
46	TPS	Pressurizer steam space temp, F
47	WC4	Pressurizer spray condensation rate, lb/s
48	QHTR	Pressurizer total heat output to water, Btu/s
49	PP	RC pressure in pressurizer steam space, psia
50	PRC	RC pressure at core outlet, psia
51	WB	Pressurizer vaporization rate, lb/s
52	WC5	Pressurizer interface condensation rate, lb/s
53	WC6	Pressurizer wall condensation rate, lb/s
54	WSU	RC flow through surge line, lb/s
55	DCLENA	Loop A SG downcomer length, ft
56	DCLENB	Loop B SG downcomer length, ft
57	WFWA	Feedwater flow, loop A, lb/s
58	WFWB	Feedwater flow, loop B, lb/s
59	WASPA	Loop A SG aspiration flow rate, lb/s
60	WASPB	Loop B SG aspiration flow rate, lb/s
61	WFA	SG tubesheet inlet flow, loop A, lb/s
62	WFB	SG tubesheet inlet flow, loop B, lb/s
63	DPOTA	Pressure differential between 0 and 32 ft, SG-loop A, psi
64	DPOTB	Pressure differential between 0 and 32 ft, SG-loop B, psi

Table 3-4. (Cont'd)

Output No.	Variable name	Description
65	HFWA	Enthalpy of loop A feedwater, Btu/lb
66	HFWB	Enthalpy of loop B feedwater, Btu/lb
67	PSGOA	SG outlet pressure, loop A, psia
68	PSGOB	SG outlet pressure, loop B, psia
69		Not used
70		Not used
71	HASPA	Enthalpy of aspiration steam, SG-A, Btu/lb
72	HASPB	Enthalpy of arpiration steam, SG-B, Btu/lb
73	TSGA	Average RC temperature, SG-A, F
74	TSGB	Average RC temperature, SG-B, F
75	WSGOA	Steam flow, SG-A outlet, lb/s
76	WSGOB	Steam flow, SG-B outlet, lb/s
77	TSTMA	Steam temperature, SG-A outlet, F
78	TSTMB	Steam temperature, SG-B outlet, F
79	LSCA	Subcooled length, SG-A, ft
80	LSCB	Subcooled length, SG-B, ft
81	LNBA	Nucleate boiling length, SG-A, ft
82	LNBB	Nucleate boiling length, SG-B, ft
83	QSGA	Primary-to-secondary heat transfer, SG-A, %
84	QSGB	Primary-to-secondary heat transfer, SG-B, %
85	W1	Turbine extraction flow, lb/s
86	W1D	Heater drain flow, lb/s
87	W2	Extraction flow, lb/s
88	W2D	Heater drain flow, lb/s
89	W3	Extraction flow, lb/s
90	W3D	Heater drain flow, lb/s
91	W4	Extraction flow, lb/s
92	W4D	Heater drain flow, lb/s
93	W5	Extraction flow, lb/s
94	W5D	Heater drain flow, lb/s
95	W5B	Extraction flow, lb/s
96	W6	Extraction flow, lb/s
97	W6D	Heater drain flow, lb/s

Table 3-4. (Cont'd)

Output No.	Variable name	
98	W6B	Extraction flow, lb/s
99	W7	HP turbine outlet flow, lb/s
100	W8	LP turbine inlet flow, lb/s
101	W8A	LP turbine inlet valve flow, lb/s
102	W8B	Feed pump extraction flow, lb/s
103	W8C	Reheater 2 outlet flow, lb/s
104	W8D	Reheater 1 outlet flow, lb/s
105	W9	LP turbine outlet flow, lb/s
106	WMS	Moisture separator drain flow, lb/s
107	WRH1	Extraction flow to reheater 1, lb/s
108	WRH2	Reheater 2 drain flow, lb/s
109	WPG5	Pegging flow to feed heater 5, lb/s
110	WPG6	Pegging flow to feed heater 6
111	WDLP	Flow out of LP drain tank, lb/s
112	WDHP	Flow out of HP drain tank, lb/s
113	WFPT	Flow through feed pump turbine, lb/s
114	MS7	HP turbine steam torque, %
115	MS9	LP turbine steam torque, %
116	MS	Total turbine steam torque, %
117	HC1	Feedwater enthalpy 1 heater exit, Btu/lb
118	HC2	Feedwater enthalpy MFP suction, Btu/lb
119	HLP1	Feedwater enthalpy 1 heater exit, Btu/lb
120	HLP2	Feedwater enthalpy 2 heater exit, Btu/lb
121	HLP3	Feedwater enthalpy 3 heater exit, Btu/lb
122	HHP4	Feedwater enthalpy 4 heater exit, Btu/lb
123	HHP5	Feedwater enthalpy 5 heater exit, Btu/lb
124	HHP6	Feedwater enthalpy 6 heater exit, Btu/lb
125	HDLP	Feedwater enthalpy LP drain tank exit, Btu/lb
126	HDHP	Feedwater enthalpy HP drain tank exit, Btu/lb
127	H1D	Drain enthalpy, heater 1, Btu/lb
128	H2D	Drain enthalpy, heater 2, Btu/lb
129	H3D	Drain enthalpy, heater 3, Btu/lb
130	H4D	Drain enthalpy, heater 4, Btu/lb

Table 3-4. (Cont'd)

Output No.	Variable name	Description
131	H5D	Drain enthalpy, heater 5, Btu/lb
132	H6D	Drain enthalpy, heater 6, Btu/lb
133	HMS	Moisture separator enthalpy, Btu/lb
134	HRH1D	Reheater 1 drain enthalpy, Btu/lb
135	DPHWP	Developed head, hotwell pump, psia
136	DPCP	Developed head, condensate pump, psia
137	DPFC1	Pressure losses, condenser to heater 1 exit, Btu/lb
138	DPFC2	Pressure losses, exit 1 to MFP suction, Btu/lb
139	TFW	Feedwater temperature, heater exit 6, F
140	TFWA	Feedwater temperature, SG-A inlet, F
141	TFWB	Feedwater temperature, SG-B inlet, F
142	WFW	Feedwater flow (total, A+B), lb/s
143	WT	Turbine inlet flow, lb/s

Table 3-5. Commonly Displayed Analog Variables

ICS outputs (such as ULD)

Secondary plant variables

- Throttle pressure
- Steam line safety valve flow
- Bypass valve flow
- Atmospheric dump flow
- Feed pump speed or head
- Feedwater control valve position or pressure drop
- Megawatts electric
- Throttle valve position

4. APPLICATIONS

This section will be supplied at a later date.

5. REFERENCES

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