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— POWER TRAIN —

GENERAL HYBRID SIMULATION FOR REACTOR COOLANT
AND SECONDARY SYSTEM TRANSIENT RESPONSE

by

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POWER TRAIN -- General Hybrid Simulation for Reactor
Coolant and Secondary System Transient Response

P. T. Christy, V. J. Galan

Key Words: POWER TRAIN, Digital Analog, PWR,
Primary Loop, Secondary Loop,
Integrated Control System

ABSTRACT

POWER TRAIN is a real-time hybrid computer simulation of a PWR nuclear power plant developed by Babcock & Wilcox's Nuclear Power Generation Division. It is used to predict the performance and behavior of major components in the primary and secondary loops over a wide range of plant conditions and operations. It is also used for detailed study of the integrated control system, which is accurately modeled.

Three basic methods of equation solution are performed in the POWER TRAIN model. The first is the classic method of solving time-dependent, space-independent ordinary differential equations on an analog computer with the appropriate choice of scale factors. The second solution method is the continuous space, discrete time (CSDT) method of solving partial differential equations in both space and time. The third method is the simulation of various algebraic and differential equations of the system on the digital computer complement of the hybrid facility.

General parameters for all the various components in the system are input in card form along with an initialization program, COMANCHE, which sets up the proper steady-state solution on the hybrid computer. This input data is a list of basic setpoints, pressures, flows, temperatures, volumes, masses, and the like, that specify the overall plant configuration.

POWER TRAIN solves the time-dependent neutron kinetics equations, the reactor and steam generator heat transfer, the pressure in the pressurizer, and the temperature response of the primary and secondary system. It includes options for any of the reactor trip signals, as well as programmable primary flow coastdown.

The program is characterized by on-line, real-time solutions to the equations and flexibility in accepting changes in plant parameters. Output is usually on a strip chart recorder.

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1. INTRODUCTION

The POWER TRAIN model (PTM) is a real-time, on-line hybrid computer code developed for the simulation of the B&W nuclear steam system (NSS). POWER TRAIN is used to predict the performance and behavior of the major components in the NSS for a wide range of plant conditions and operation.

The PTM is designed to model as much of the NSS as is feasible, especially those components whose behavior is interrelated to other components. For this reason, virtually every major component around the primary loop is modeled extensively.

The reactor is modeled in the PTM by standard methods. The space-independent differential equations for nuclear kinetics are solved with the appropriate delay group terms. The reactivity feedback in the reactor consists of moderator and Doppler (fuel) contributions plus the reactivity addition of the control rods. Any other effects, such as long-term xenon effects or other poisons, are lumped together in a shim reactivity term and are not operated upon. Since the program uses point kinetics, the flux shape is assumed to be constant with time.

The heat production in the core is simulated by a fuel pin model of the thermal time constants of the various materials in the core; fuel, gap, cladding, and moderator are the major components to be considered.

The primary flow loop of the NSS is modeled on the PTM with the exception of the primary pumps. The primary flow can be changed, but it is simply a specified variable rather than a calculated one. The primary loop model provides the mechanism for transporting the temperature distribution between the reactor and the steam generator. Transport delays in the hot and cold legs and thermal mixing in the inlet and outlet plenums are provided by this model. Since this model is the only connection between reactor and steam generator, correct dynamic modeling of these delays and time constants is essential. The temperature distribution around this model can be sampled at regular points for control or information.

The pressurizer on the PTM is an open-loop model which accepts a surge flow and produces a system pressure. The model contains simulation of the spray flow, the heaters, and the pilot and relief valves. There is a control system on the model to regulate these valves and heaters so that pressure is maintained within certain control limits. The only input is the flow through the surge line that connects the pressurizer with the hot leg piping. The surge is obtained by calculating the expansion and contraction of the primary fluid as the system changes temperature.

The steam generator is a once-through (OTSG) type. The primary flow passes through a straight tube bundle down the steam generator, while the secondary fluid flows up on the outside of the tubes. This design produces superheated steam at the generator outlet ports, and plant performance is based on the ability to provide superheated steam under all conceivable situations of plant generation. The steam generator is modeled as a counter-current heat exchanger with single-phase primary flow and two-phase secondary flow. The completed model provides many channels of information: primary outlet temperature, steam temperature, secondary region lengths, and ten full spatial variables, which describe various thermodynamic properties in the steam generator along the axis of both flows. The steam generator model is a "black box" model which solves the equations of the counter-current heat exchanger with only a limited number of external inputs: primary inlet temperature and flow, secondary (feedwater) temperature and flow, and secondary pressure. The dynamic behavior of the steam generator is well represented in the solution of the equations that describe it. The behavior has been checked against several reference programs and the model is calibrated to yield comparable results. In this way the correct primary outlet and steam temperatures are obtained for the appropriate flows, and the lengths of the various regions in the secondary side are adjusted to the lengths expected. The steam generator model provides the only coupling from hot leg to cold leg, from feedwater to steam, and from primary to secondary. It is essential that the steam generator be properly calibrated to yield the correct response, so that the functional relations between these various quantities are correct.

The feedwater model contains simulation of control valves, feedwater heater, a feed pump, and the feedwater line length and mass needed

to connect these to the steam generator inlet nozzles. Two types of feedwater systems are currently being modeled: (1) a fixed-speed feedwater pump with a valve set by the control system to produce the required feedwater flow and (2) a variable-speed feedwater pump and control valve set by the control system to maintain a constant pressure drop across the feedwater valve. The feedwater system is set up as a single line feeding one steam generator; there is no distinction between various parallel pumps and pipes.

The steam system is modeled with various valves for the pressure relief system and the bypass and dump system with the correct number of valves, operational characteristics, rated flow for each valve, and closure times. The steam line itself is a nodal model with conservation equations written between nodes. This simulates the spatial effects of the lengths of line and produces the correct dynamic response of pressures and flows to various upsets in the line. The end of the line goes to the turbine control valve, which regulates pressure and flow to the turbine header. This valve position is regulated by the control system to produce the required generation for various plant conditions.

The PTM is controlled by a digital simulation of the integrated control system (ICS) concept applied to the major control parameters and components in the NSS. The ICS model produces demand signals for the reactor and rod drive models, the steam generator (feedwater) model, and the turbine pressure model. The demand signals are driven by a unit load demand (ULD) signal that is 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 the automatic coordination of the various demand signals and the manual control of the signals by the operator. The ICS simulation has the capability of automatically establishing tracking mode whenever such conditions arise that would cause unit tracking, and of performing automatic demand runback and transfer of control signals when tracking mode is established. The model operator has full control over the ICS simulation from the analog console and can sequence and initiate whatever combination of transients are desired by pushing the appropriate buttons and entering parameters on the keyboard. The model is designed to perform all transients without resorting to modifications of the patching.

The starting point for operation of the model is always 100% power condition. The lower limit of operation of the model is about the 5 to 8% power level. From the initial 100% power level, the operator can start sequencing transients to maneuver to other demand conditions. With very few exceptions (reactor trip, total loss of feedwater flow), it is not necessary to restore the whole model to 100% power conditions after each transient.

The basic set of normal transients run on the model are ramps, steps, and steady-state power levels over various ranges with the control system in a fully automatic mode so that the overall plant response can be observed. The control system can be selectively disabled or manually controlled to perform parametric or failure modes and effects studies. The basic set of upset transients run on the model includes step load reduction resulting from turbine trip or electrical load rejection, reactor trip, change of reactor coolant flow, rod withdrawal, control rod drop, loss of station power, loss of feedwater flow, and loss of feedwater heater. The basic set of emergency transients run on POWER TRAIN includes rapid depressurization of the secondary system and stuck open turbine bypass or safety valve.

2. METHODS

This section summarizes the solution techniques and the development of equations that make up the POWER TRAIN model. The equations are presented in the unscaled form although some of the intermediate algebraic steps are eliminated in the interest of conciseness.

There are two basic forms of equations used on the PTM but three forms of solutions. The first equation form has time as a continuous independent variable and no spatial considerations; i.e., the equation describes some interaction at one point in space. Examples of this type of equation are the reactor kinetics equations, the fuel pin model, and the feedwater system. Although several points in space might be considered (as in the feedwater model where pumps, valves, and heaters are modeled as discrete interactions), the basic equation lumps these points together and has no spatial effect in its form. Such an equation form is called "discrete space-continuous time," or DSCT; several points in space are modeled continuously in time and the spatial effects are not present.

The other equation form in the PTM is more complex because both space and time are independent variables in the solution of the equations. These are the "continuous space-discrete time" (or CSDT) equations that are solved in the hybrid steam generator model by the technique of the same name. The equations are simply the partial differential equations (PDE) of a one-dimensional steam generator which expresses its dynamics in both space and time.

The first method of equation solution discussed is the CSDT method as it is applied to the CSDT equations of the steam generator. The task is to solve the equations by integrating continuously in space while differencing discretely in time. The final form of the equations shows them with the spatial term converted to computer time units through the scaling factor β . The first part of the task is clear: scale the

equation, patch it on the analog computer, and integrate as a function of computer time. The spatial effects of the model will emerge directly through the scaling factor β . Unfortunately, this is difficult to arrange because of the various inputs to the spatial equations with all combinations of superscripts that difference the terms forward and backward in time with reference to the time step of the equation solution. What is required is to have all these various terms available with their proper time steps and coefficients, so that they can be combined simultaneously to form the inputs to the spatial equations of the steam generator. In general, two kinds of inputs are required — those with the same time step as the equation and those with time steps skewed from the equation. The inputs with the same time step are handled trivially; they are connected directly from the component where they are being simultaneously solved. Inputs with other time steps imply a memory capability to retain these terms from one time step to another. This is the function of the digital computer: to store the various terms from various time steps and to play them back at the appropriate time step when required. This, then is the general method of the CSDT technique.

There are several points to remember about the CSDT method:

1. The various inputs to equations are functions of space (read that as "computer time with β ") and must be "memorized" by the digital computer in functional form.
2. The synchronization and orientation of these various inputs must agree. There can be no spatial skewing of one signal to the next, the initial and boundary conditions of all must start and stop at the same place, and the signals must all proceed in the same direction; i.e., if the solution moves up the steam generator, then all the inputs must move up also and be synchronized in space with each other.
3. It is sometimes necessary to difference signals further to bring about correct temporal synchronization of inputs. For example, an input at the $i + 1$ time step can be obtained by a weighted combination of signals at the $i + 1/2$ and $i - 1/2$ time steps. These differenced signals are indicated by asterisks in the equations and are done by a suitable algorithm in the digital computer.

4. While time extends indefinitely forward and backward toward infinity, this is not the case for space; in fact, for the CSDT equations of the steam generator, space extends only as far as the length of the tubes in the active area of the LEOTSG. This indicates a repetitive form of operation in which we solve the equations for each successive time step over the same length of the steam generator, incrementing the discrete time representation by the time step every time.

The second equation form applied to the equations of the POWER TRAIN model is the use of the digital computer to solve equations that are expressed continuously in time. The timing arrangement of the PTM allows a short period of time between each spatial solution (or "space sweep") to perform these digital calculations. Spatial solutions are performed every half time step (i , $i + 1/2$, $i + 1$, ...) so a digital calculation can also be done every half time step. The basic idea is to take a continuous time equation, which is amenable to digital solution, determine that the "stepping" along the time axis will not affect the solution, and solve it discretely on the digital computer. Thus, it must be determined first what equation types can be solved on the digital computer. Algebraic, logic, control, and a very few differential equations are candidates for digital solution. Of these, only the ones with slowly varying time components can be considered because of the dynamic degradation of the solution of a rapidly changing variable on a digital computer. Several examples of equations that have been successfully modeled on the digital computer are the reactor kinetics delay groups, the reduced control system for POWER TRAIN, and the delays and mixing effects on temperature propagation through the primary system. Examples of equations that cannot be solved are the CSDT steam generator equations, which require real-time integration of complicated waveforms, and the steam line valves, which operate too rapidly for accurate digital simulation.

By modeling on the digital computer the equations that can be accurately modeled by a digital solution, the equipment load on the analog computer is substantially reduced, and there is more room for improved analog models. Various problems have been encountered with digital modeling; however, one consistent problem is loss of accuracy when performing integer mode calculations on the digital computer. This is especially

severe when modeling equations with very slow dynamic behavior as the nuclear kinetics delay group models. The accumulated round-off and truncation errors will yield incorrect answers. Various techniques have been developed to do double-precision arithmetic, quasi-floating-point calculations, and fractional accumulation to better solve this class of problems.

One class of equations that has worked extremely well on the digital computer is the logic and control that simulates the ICS on the PTM. Instead of cumbersome analog equivalents for the ICS diagrams, programming is done on the digital computer, where it can be performed very efficiently. Demand signals, runbacks, tripped/reset conditions, error signals, deadbands, limit sensors, and many other logic and control structures can be programmed quickly and flexibly on the digital computer.

The third method of equation solution is the classic analog solution scheme. The equations solved in this manner are those which are too complicated or too fast to be done on the digital computer. The CSDT equations are all solved directly on the analog, while the digital is dedicated to signal recording and playback of the various spatial profiles. This is a true hybrid solution of equations. Other equations that involve either rapid integration or are too fast for digital solutions are done in this manner. Some of the circuits perform analog integration on a digitally produced derivative; such an approach greatly simplifies some of the models used. Certain circuits perform at such high rates that they cannot be done digitally, e.g., the steam line bypass system. Another class of circuits acts as linear interpolators to buffer the signals sent back to the analog computer on the digital-to-analog converters.

3. MATHEMATICAL MODEL

The order of the following equations is the same as the direction of the flow of energy: from the reactor to the steam generator to the turbine. The nomenclature for each subsection is placed at the end of that subsection.

3.1. Reactor Kinetics Model

The space-independent reactor kinetics equations are as follows:

$$\frac{d\Phi}{dt} = \frac{\delta k}{L^*} \Phi - \frac{\beta}{L^*} \Phi + \sum_{i=1}^6 \frac{\beta_i}{L^*} \gamma_i \quad (3.1-1)$$

$$\frac{d\gamma_i}{dt} = \lambda_i (\Phi - \gamma_i) \quad (3.1-2)$$

$$\gamma_i = \frac{L^* \lambda_i}{\beta_i} C_i \quad (3.1-3)$$

where

$$\beta = \sum_{i=1}^6 \beta_i, \quad (3.1-4)$$

$$\delta k = \delta k_{\text{mod}} + \delta k_{\text{dop}} + \delta k_{\text{rod}} + \delta k_{\text{shim}}, \quad (3.1-5)$$

$$\delta k_{\text{mod}} = \alpha_{\text{mod}} (T_m - T_{\text{mo}}), \quad (3.1-6)$$

$$\delta k_{\text{dop}} = \alpha_{\text{dop}} (T_f - T_{\text{fo}}). \quad (3.1-7)$$

Nomenclature

- Φ = neutron flux ratio to initial value
- L^* = effective neutron lifetime, s
- C_i = yield of delayed neutron group i

λ_i = decay constant of delayed neutron group i, s⁻¹
 β_i = conc. fraction of delayed neutron group i
 δk_{shim} = shim reactivity, $\delta k/k$
 δk_{rod} = reactivity of control rod, $\delta k/k$
 α_{mod} = moderator coeff of reactivity, $(\delta k/k)/^{\circ}\text{F}$
 T_m = average moderator temp, F
 T_{mo} = initial average moderator temp, F
 α_{dop} = Doppler coeff of reactivity, $(\delta k/k)/^{\circ}\text{F}$
 T_f = average fuel temp, F
 T_{fo} = initial average fuel temp, F

3.2. Control Rod Drive

The control rod velocity is represented by a first-order time constant simulation of a bipolar error signal applied to the rod drive motor.

$$\frac{dV}{dt} = \frac{1}{\tau_{\text{rd}}} (V_m S - V) \quad (3.2-1)$$

The control rod position is obtained by integrating the velocity:

$$X = \int_0^t V dt + X_0. \quad (3.2-2)$$

The control rod reactivity is obtained by interpolating from a curve with this position:

$$\delta k_{\text{rod}} = f(X); \quad (3.2-3)$$

$$\delta k_{\text{shim}} = f(X_0). \quad (3.2-4)$$

A reactor trip is simulated by accelerating the rods into the core in free fall. The reactivity is determined from the rod position.

$$X = X_t - \frac{1}{2} g t^2 \quad (3.2-5)$$

Nomenclature

V = control rod velocity, in./min
 V_m = maximum control rod velocity, in./min
 S = normalized bipolar error signal (+1, 0, -1)
 τ_{rd} = time constant of rod drive motor, s
 X = control rod position, in. withdrawn
 X_o = initial control rod position, in. withdrawn
 f = control rod reactivity vs position function
 X_t = control rod position at instant of trip, in.
 g = gravitational acceleration constant
 t = time after rod trip, s

3.3. Fuel Pin

The average fuel pin is described by a two-region model consisting of fuel and water. The heat transfer equation expressing the temperature change in the fuel is as follows:

$$\frac{d\bar{T}_f}{dt} = \frac{UA}{M_f C_{pf}} (\bar{T}_w - \bar{T}_f) + \frac{b}{M_f C_{pf}} \phi \quad (3.3-1)$$

and in the water:

$$\frac{d\bar{T}_w}{dt} = \frac{W_{rc}}{M_w} (T_{wi} - T_{wo}) + \frac{UA}{M_w C_{pw}} (\bar{T}_f - \bar{T}_w) \quad (3.3-2)$$

where $T_{wo} = 2\bar{T}_w - T_{wi}$. (3.3-3)

Nomenclature

$\bar{T}_f$ = average fuel temp, F
 $\bar{T}_w$ = average water temp, F
 T_{wi} = water temp at core inlet, F
 T_{wo} = water temp at core outlet, F
 M_f = mass of fuel in core, lb
 M_w = mass of water in core, lb
 W_{rc} = mass flow rate of coolant through core, lb/s

U = overall core heat transfer coeff, Btu/s-ft²-°F
 A = overall core heat transfer area, ft²
 b = ratio of heat rate to neutrons, Btu/s-n
 Φ = neutron production in core, n
 C_{pf} = specific heat of fuel, Btu/lb-°F
 C_{pw} = specific heat of water, Btu/lb-°F

3.4. Reactor Coolant Pumps

The primary flow is a variable specified either directly by the operator or as a function of pump status. In the latter case, the terminal flow value for any number of pumps running is part of the internal data of the programs, and the model will increase or decrease the flow on a variable time constant curve as the operator changes the pump status from one condition to another.

The temperature rise (about 0.5F) across the reactor coolant pumps is assumed to be constant for all cases.

3.5. Primary Flow System

The primary flow system is simulated by a complete closed-loop, variable transport delay system with eight access points to the temperature distribution. These eight access points are as follows:

1. Reactor (core) outlet.
2. Hot leg inlet (reactor vessel outlet nozzle).
3. Hot leg outlet (steam generator inlet nozzle).
4. Steam generator inlet (upper tubesheet).
5. Steam generator outlet (lower tubesheet).
6. Cold leg inlet (steam generator outlet nozzle).
7. Cold leg outlet (reactor vessel inlet nozzle).
8. Reactor (core) inlet.

The temperature transfer between any two successive points around the loop is either a first-order time constant, a pure transport delay, or an algebraic relation depending on the dynamics of the particular region. The specific relations are as follows:

$$\text{Nodes } 1,2: \quad \frac{dT_2}{dt} = \frac{1}{\tau_{12}} (T_1 - T_2) \quad (3.5-1)$$

$$\begin{array}{ll} \text{Nodes} & T_3 = T_2(t - \tau_{23}) \\ 2,3: & \end{array} \quad (3.5-2)$$

$$\begin{array}{ll} \text{Nodes} & \frac{dT_4}{dt} = \frac{1}{\tau_{34}} (T_3 - T_4) \\ 3,4: & \end{array} \quad (3.5-3)$$

$$\begin{array}{ll} \text{Nodes} & T_5 = \text{steam generator model outlet temp} \\ 4,5: & \end{array} \quad (3.5-4)$$

$$\begin{array}{ll} \text{Nodes} & \frac{dT_6}{dt} = \frac{1}{\tau_{56}} (T_5 - T_6) \\ 5,6: & \end{array} \quad (3.5-5)$$

$$\begin{array}{ll} \text{Nodes} & T_7 = T_6(t - \tau_{67}) \\ 6,7: & \end{array} \quad (3.5-6)$$

$$\begin{array}{ll} \text{Nodes} & \frac{dT_8}{dt} = \frac{1}{\tau_{78}} (T_7 - T_8) \\ 7,8: & \end{array} \quad (3.5-7)$$

$$\begin{array}{ll} \text{Nodes} & T_1 = 2\bar{T}_w - T_8 \\ 8,1: & \end{array} \quad (3.5-8)$$

Nomenclature

T_n = temp at point n around primary loop, F

τ_{nm} = first-order time constant or transport delay between points n and m, s

t = time, s

$\bar{T}_w$ = average water temp from fuel pin, F

3.6. Pressurizer/Surge

The pressurizer model computes the pressure with one set of equations for increasing pressure and another set for decreasing pressure. Saturated conditions are assumed at all times throughout the pressurizer. The forcing function is a surge rate with safety valves, heaters, and spray to restrain the transient within acceptable limits. The thermodynamics of the pressurizer are based on the following linear fits to quantities in the steam tables:

$$\gamma_1 = a_1 + b_1 P_z \quad (3.6-1)$$

$$\gamma_2 = a_2 + b_2 P_z \quad (3.6-2)$$

$$P_z = a_3 + b_3 v_g \quad (3.6-3)$$

$$v_f = a_4 + b_4 P_z \quad (3.6-4)$$

$$\gamma_3 = a_5 + b_5 P_z \quad (3.6-5)$$

$$h = a_6 + b_6 P_z \quad (3.6-6)$$

3.6.1. Pressure Decrease Model

The pressure decrease model calculates pressures from consideration of the total system heat.

$$Q_T = M_f h_f + M_g h_g \quad (3.6-7)$$

The total mass and total volume are defined as

$$M_T = M_f + M_g \quad (3.6-8)$$

and

$$V_T = V_f + V_g \quad (3.6-9)$$

Using the thermodynamic property

$$h_g = h_f + h_{fg}, \quad (3.6-10)$$

equation 3.6-7 can be rewritten as

$$\begin{aligned} Q_T &= M_f h_f + M_g (h_f + h_{fg}) \\ &= M_T h_f + M_g h_{fg}. \end{aligned} \quad (3.6-11)$$

Apply the identity $(v_g - v_f)/(v_g - v_f)$ to the last term of equation 3.6-11 along with the following relations:

$$M_g = V_g / v_g; \quad (3.6-12)$$

$$M_f = V_f / v_f \quad (3.6-13)$$

produces the equation

$$Q_T = M_T h_f + \frac{V_g}{v_g} h_{fg} \left(\frac{v_g}{v_g - v_f} - \frac{v_f}{v_g - v_f} \right) \quad (3.6-14)$$

$$Q_T = M_T h_f + h_{fg} \left(\frac{V_g}{v_g - v_f} - \frac{V_g}{v_g} \frac{v_f}{v_g - v_f} \right) \quad (3.6-15)$$

Add a zero to equation 3.6-15 in the form of

$$h_{fg} \left(\frac{V_f}{v_g - v_f} - \frac{V_f}{v_g - v_f} \right)$$

and regroup terms.

$$Q_T = M_T h_f + h_{fg} \left[\frac{V_g + V_f}{v_g - v_f} - \left(\frac{V_g}{v_g} + \frac{V_f}{v_f} \right) \frac{v_f}{v_g - v_f} \right] \quad (3.6-16)$$

Apply equations 3.6-8 and 3.6-9 to equation 3.6-16:

$$Q_T = M_T h_f + h_{fg} \left(\frac{V_T}{v_g - v_f} - M_T \frac{v_f}{v_g - v_f} \right). \quad (3.6-17)$$

Equation 3.6-10 is also valid for specific volumes:

$$v_{fg} = v_g - v_f. \quad (3.6-18)$$

Use this to further reduce equation 3.6-17:

$$Q_T = V_T \frac{h_{fg}}{v_{fg}} + M_T \left(h_f - v_f \frac{h_{fg}}{v_{fg}} \right). \quad (3.6-19)$$

Two functions will be defined to reduce this equation further:

$$\gamma_1 = \frac{h_{fg}}{v_{fg}} = a_1 + b_1 P_z \quad (3.6-1)$$

$$\gamma_2 = h_f - \frac{h_{fg}}{v_{fg}} v_f = a_2 + b_2 P_z \quad (3.6-2)$$

These are two of the six steam table functions required in this model.

Employ them to reduce further the form of the heat equation:

$$Q_T = \gamma_1 V_T + \gamma_2 M_T \quad (3.6-20)$$

Write a new equation for heat balance in the pressurizer model:

$$\begin{aligned} \dot{Q}_T = (W_{\text{surge}} - W_{\text{spray}})h_{\text{surge}} + W_{\text{h spray}} + \dot{Q}_{\text{ht}} \\ - W_{\text{h relief}} - W_{\text{h pilot}} \end{aligned} \quad (3.6-21)$$

Substitute the correct steam fits into equation 3.6-20, differentiate, and set equal to equation 3.6-21.

$$\begin{aligned} b_1 V_T \dot{P}_z + (a_2 + b_2 P_z) \dot{M}_T + b_2 M_T \dot{P}_z = (W_{\text{surge}} - W_{\text{spray}})h_{\text{surge}} \\ + W_{\text{h spray}} + \dot{Q}_{\text{ht}} - W_{\text{h relief}} - W_{\text{h pilot}} \end{aligned} \quad (3.6-22)$$

Rearrange to isolate the $\dot{P}_z$ term:

$$\begin{aligned} \dot{P}_z = \left[(W_{\text{surge}} - W_{\text{spray}})h_{\text{surge}} - (a_2 + b_2 P_z) \dot{M}_T + W_{\text{h spray}} \right. \\ \left. - W_{\text{h relief}} - W_{\text{h pilot}} + \dot{Q}_{\text{ht}} \right] / b_1 V_T + b_2 M_T \end{aligned} \quad (3.6-23)$$

Because most mass change is through surge, the surge enthalpy is related to pressure by the last steam table fit.

$$\begin{aligned} \dot{P}_z = \left[(W_{\text{surge}} - W_{\text{spray}})(a_6 + b_6 P_z) - (a_2 + b_2 P_z) \dot{M}_T \right. \\ \left. + W_{\text{h spray}} - W_{\text{h relief}} - W_{\text{h pilot}} + \dot{Q}_{\text{ht}} \right] / b_1 V_T + b_2 M_T \end{aligned} \quad (3.6-24)$$

The change of total mass is simply the sum of flows into the system:

$$\frac{dM_T}{dt} = W_{\text{surge}} - W_{\text{relief}} - W_{\text{pilot}} \quad (3.6-25)$$

3.6.2. Pressure Increase Model

The pressure increase model is developed from a function relating the pressure and the specific volume,

$$P_z = a_3 + b_3 v_g \quad (3.6-3)$$

Rearrange equation 3.6-12 and substitute in the equation above

$$P_z = a_3 + b_3 \frac{V_g}{M_g} \quad (3.6-26)$$

Differentiate this to form an expression for $\dot{P}_z$:

$$\dot{P}_z = b_3 \frac{(\dot{V}_g M_g - \dot{M}_g V_g)}{M_g^2} \quad (3.6-27)$$

Rearrange equation 3.6-26 and use it to reduce equation 3.6-27:

$$\dot{P}_z = \frac{b_3 \dot{V}_g M_g}{M_g M_g} - \frac{b_3 \dot{M}_g V_g}{M_g M_g} \quad (3.6-28)$$

$$\dot{P}_z = \frac{b_3 \dot{V}_g}{M_g} - (P_z - a_3) \frac{\dot{M}_g}{M_g} \quad (3.6-29)$$

$$\dot{P}_z = \frac{a_3 \dot{M}_g + b_3 \dot{V}_g - P_z \dot{M}_g}{M_g} \quad (3.6-30)$$

Since the total volume of the pressurizer is constant, the change in the gas volume must be the same as the change in fluid volume, or surge.

$$\dot{V}_g = -W_{\text{surge}} v_f \quad (3.6-31)$$

Use this to further reduce the pressure equation:

$$\dot{P}_z = \frac{a_3 \dot{M}_g - b_3 W_{\text{surge}} v_f - P_z \dot{M}_g}{M_g} \quad (3.6-32)$$

Use equation 3.6-4 to make v_f a function of pressure.

$$\dot{P}_z = \frac{a_3 \dot{M}_g - b_3 W_{\text{surge}} (a_4 + b_4 P_z) - P_z \dot{M}_g}{M_g} \quad (3.6-33)$$

$$\dot{P}_z = \frac{(a_3 - P_z) \dot{M}_g - b_3 (a_4 + b_4 P_z) W_{\text{surge}}}{M_g} \quad (3.6-34)$$

The change in gas mass is obtained by mass balancing the flows in this region:

$$\dot{M}_g = -W_{\text{relief}} - W_{\text{pilot}} - W_{\text{cond}} \quad (3.6-35)$$

The energy of the steam condensing is equal to the energy of the spray coming to saturation conditions.

$$W_{\text{cond}}(h_g - h_f) = W_{\text{spray}}(h_f - h_{\text{spray}}) \quad (3.6-36)$$

$$W_{\text{cond}} = W_{\text{spray}}(h_f - h_{\text{spray}}) / h_{fg} \quad (3.6-37)$$

Use equation 3.6-5 to establish a function for use in equation 3.6-37:

$$\gamma_3 = (h_f - h_{\text{spray}}) / h_{fg} = a_5 + b_5 P_z. \quad (3.6-38)$$

This results in the final form of the mass balance in the gas region:

$$\frac{dM_g}{dt} = -W_{\text{relief}} - W_{\text{pilot}} - W_{\text{spray}}(a_5 + b_5 P_z). \quad (3.6-39)$$

The selection of pressure increase or pressure decrease is made logically by determining the status of the model through the following set of conditions:

If $|P_z - P_o| \leq P_a$, do not change model.

If $P_z < P_b$, use pressure decrease model until $P_z > P_o$.

If $\dot{P}_z - \epsilon > 0$, use pressure increase model.

If $\dot{P}_z + \epsilon < 0$, use pressure decrease model.

3.6.3. Surge Flow

The surge flow is obtained from the average system temperature, which is a volume-weighted measurement of the temperatures around the primary loop.

$$\bar{T}_{\text{sys}} = \sum_{i=1}^8 \frac{V_i}{V_T} \frac{T_{i+1} - T_i}{2} \quad (3.6-40)$$

This average system temperature is linearly related to the total mass of the system by considering the change in coolant density with temperature and the total system volume.

$$M_{\text{sys}} = bV_T \bar{T}_{\text{sys}}. \quad (3.6-41)$$

The change in the mass of the system is the surge to the pressurizer; it is obtained by differentiating this temperature.

$$W_{\text{surge}} = \dot{M}_{\text{sys}} = bV_T \frac{d\bar{T}_{\text{sys}}}{dt} \quad (3.6-42)$$

Nomenclature

P_z = pressurizer pressure, psia

a_n, b_n = constants for linear fits to steam tables

$\gamma_1 = h_{fg}/v_{fg}$, fluid-to-gas enthalpy over fluid-to-gas specific volume

$\gamma_2 = h_f - v_f(h_{fg}/v_{fg})$, fluid enthalpy minus fluid specific volume times fluid-to-gas enthalpy over fluid-to-gas specific volume

Q_T = total heat of pressurizer, Btu

M_f = fluid mass in pressurizer, lb

M_g = gas mass in pressurizer, lb

M_T = total mass in pressurizer, lb

h_f = saturated fluid enthalpy, Btu/lb

h_g = saturated gas enthalpy, Btu/lb

V_f = fluid volume in pressurizer, ft³

V_g = gas volume in pressurizer, ft³

V_T = total volume in pressurizer, ft³

v_f = saturated fluid specific volume, ft³/lb

v_g = saturated gas specific volume, ft³/lb

W = flows to pressurizer from various sources, lb/s

h = enthalpy of various surges, Btu/lb

$\dot{Q}_{ht}$ = heat rate of pressurizer heater banks, Btu/s

$\gamma_3 = (h_f - h_{\text{spray}})/h_{fg}$, saturated fluid enthalpy minus spray line enthalpy over fluid-to-gas enthalpy

P_a = small pressure deadband around normal pressure setpoints, psi

P_o = normal pressurizer pressure setpoint, psia

P_b = low pressure setpoint for selecting pressure decrease model, psia
 ϵ = small offset to provide hysteresis to selection of pressure model
 T_i = temp at the i th temperature node around the primary loop, F
 V_i = volume of primary loop region between two successive temperature nodes, ft^3
 V_T = total active volume of primary loop, ft^3
 $\bar{T}_{\text{sys}}$ = average system temp, F
 M_{sys} = mass of primary system, lb
 $b = \delta\rho/\delta T$, change in density with temp, $\text{lb}/\text{ft}^3\text{-}^\circ\text{F}$

3.7. OTSG Primary Side

The equations on the primary side of the OTSG are derived from the partial differential equations of a concentric counter-flow heat exchanger. The following assumptions are made to simplify the modelling of the heat exchanger:

1. No longitudinal heat conduction in the tube walls.
2. No longitudinal mixing of the fluids.
3. Perfect radial mixing of the fluids.
4. Constant tube wall temperature in the radial direction.

The primary temperature can be obtained by writing a conservation-of-energy equation:

$$M_p C_{pp} \frac{\partial T_p}{\partial t} = W_p C_{pp} (T_{pi} - T_{po}) - U_{pm} P L (T_p - T_m). \quad (3.7-1)$$

This equation equates the change in primary temperature with respect to time to the difference in the inlet and outlet temperatures and the loss across the tube heat transfer area to the secondary side. The primary mass is related to other parameters by

$$M_p = \rho_p A_p L. \quad (3.7-2)$$

Use this to reduce the primary temperature equation,

$$\frac{\partial T_p}{\partial t} = \frac{W_p}{\rho_p A_p L} (T_{pi} - T_{po}) - \frac{U_{pm} P}{\rho_p A_p C_{pp}} (T_p - T_m). \quad (3.7-3)$$

The difference in the two end temperatures over the length can be reduced to a partial in a limit.

$$\lim_{L \rightarrow 0} \frac{T_{pi} - T_{po}}{L} = \frac{-\Delta T_p}{\Delta L} = \frac{-\partial T_p}{\partial x} \quad (3.7-4)$$

Hence, the consideration of end point conditions leads to the change in temperature with respect to length. Note the change of sign since the difference is made in the wrong direction of primary flow.

$$\frac{\partial T_p}{\partial t} = -\frac{W_p}{\rho_p A_p} \frac{\partial T_p}{\partial x} + \frac{U_{pm} P}{\rho_p A_p C_{pp}} (T_m - T_p) \quad (3.7-5)$$

Rearrange to isolate the spatial term:

$$\frac{\partial T_p}{\partial x} = \left[-\rho_p A_p \frac{\partial T_p}{\partial t} + \frac{U_{pm} P}{C_{pp}} (T_m - T_p) \right] \frac{1}{W_p} \quad (3.7-6)$$

The temporal partial is now re-expressed by differencing in the time domain. The following is the notation for time and temperature used for the differencing:

$$t^i = i\Delta t; \quad T(x, t) = T(x, i\Delta t) = T^i(x). \quad (3.7-7)$$

Change the terms in equation 3.7-6 to this notation:

$$\frac{\partial T_p^{i+1}}{\partial t} \leftarrow \frac{T_p^{i+1} - T_p^i}{\Delta t}. \quad (3.7-8)$$

$$\frac{\partial T_p^{i+1}}{\partial x} = \frac{-\rho_p A_p \left(\frac{T_p^{i+1} - T_p^i}{\Delta t} \right) + \frac{U_{pm} P}{C_{pp}} (T_m^{i+1} - T_p^{i+1})}{W_p} \quad (3.7-9)$$

$$\frac{\partial T_p^{i+1}}{\partial x} = \frac{\frac{\rho_p A_p}{\Delta t} T_p^i + \frac{U_{pm} P}{C_{pp}} T_m^{i+1} - \left(\frac{\rho_p A_p}{\Delta t} + \frac{U_{pm} P}{C_{pp}} \right) T_p^{i+1}}{W_p} \quad (3.7-10)$$

This is the final form of the primary equation before it is transformed into the computer time domain. The metal temperature will now be developed and brought to this point, after which they will be handled together.

The metal temperature is obtained by writing a conservation-of-energy equation:

$$\rho_m C_{pm} A L \frac{\partial T_m}{\partial t} = U_{pm} L P_p (T_p - T_m) - U_{sm} L P_s (T_m - T_s) \quad (3.7-11)$$

Divide through by $\rho_m C_{pm} A L$ and rearrange:

$$\frac{\partial T_m}{\partial t} = \frac{1}{\rho_m C_{pm} A L} [U_{pm} P_p T_p + U_{sm} P_s T_s - (U_{pm} P_p + U_{sm} P_s) T_m]. \quad (3.7-12)$$

Apply the time differencing scheme as before --

$$\frac{\partial T_m^{i+1}}{\partial t} \leftarrow \frac{T_m^{i+1} - T_m^i}{\Delta t}; \quad (3.7-13)$$

then, with algebraic manipulation of the resulting equation,

$$\begin{aligned} T_m^{i+1} = & \left(1 + \frac{\rho_m C_{pm} A L}{U_{pm} P_p \Delta t} \right)^{-1} T_p^{i+1} + \frac{\frac{P_s \Delta t}{\rho_m C_{pm} A L}}{1 + \frac{U_{pm} P_p \Delta t}{\rho_m C_{pm} A L}} U_{sm} T_s^{i+1} \\ & - \frac{\frac{P_s \Delta t}{\rho_m C_{pm} A L}}{1 + \frac{U_{pm} P_p \Delta t}{\rho_m C_{pm} A L}} U_{sm} T_m^{i+1} + \left(1 + \frac{U_{pm} P_p \Delta t}{\rho_m C_{pm} A L} \right)^{-1} T_m^i \end{aligned} \quad (3.7-14)$$

Both primary and metal temperatures have now been put in their final difference form. The conversion from partial differential form with space dependence to time dependence is done by scaling space to be proportional to computer time.

$$\beta = \frac{\partial x}{\partial \tau}. \quad (3.7-15)$$

The scaling of computer time to real time should be included in the general case, but for all current development of the model it is assumed to be unity. Hence, computer time units and real time units are equal. Use equation 3.7-15 to remove the spatial dependence from both primary and metal temperature equations 3.7-10 and 3.7-14.

$$\frac{\partial T_p^{i+1}}{\partial \tau} = \frac{1}{W_p} \left[\frac{\beta \rho_p A_p}{\Delta \tau} T_p^i + \frac{\beta U_{pm} P_p}{C_{pp}} T_m^{i+1} - \left(\frac{\beta \rho_p A_p}{\Delta \tau} + \frac{\beta U_{pm} P_p}{\Delta \tau C_{pp}} \right)^{-1} T_p^{i+1} \right] \quad (3.7-16)$$

$$T_m^{i+1} = \left(1 + \frac{\rho_m C_{pm} A_m}{U_{pm} P_p \Delta \tau} \right)^{-1} T_p^{i+1} + \frac{\frac{P_s \Delta \tau}{\rho_m C_{pm} A_m}}{1 + \frac{\rho_m C_{pm} A_m}{U_{sm} P_s \Delta \tau}} U_{sm}^{i+1} T_s^{i+1} - \frac{\frac{P_s \Delta \tau}{\rho_m C_{pm} A_m}}{1 + \frac{\rho_m C_{pm} A_m}{U_{sm} P_s \Delta \tau}} U_{sm}^{i+1} T_m^{i+1} + \left(1 + \frac{U_{pm} P_p \Delta \tau}{\rho_m C_{pm} A_m} \right)^{-1} T_m^i \quad (3.7-17)$$

Equations 3.7-16 and 3.7-17 are the final forms of the spatial equations that describe the response of the primary and metal temperatures along the one-dimensional steam generator.

There are two supplementary equations which must be provided for shifting the time step of the secondary terms in the metal temperature. The general form of the two equations is

$$S^{i+1} = \left(1 - \frac{1}{\theta} \right) S^{i-1/2} + \frac{1}{\theta} S^{i+1/2}. \quad (3.7-18)$$

A discussion of the value of θ to be used is included in the report on hybrid computer techniques referenced by this report. The limits on θ are 0.5 to 1, and the final value is a compromise between stability and accuracy. The eventual value was picked to be between these limits and to be easily programmed.

$$\theta = \frac{2}{3} = 0.666\bar{6} \quad (3.7-19)$$

$$S^{i+1} = \frac{3}{2} S^{i+1/2} - \frac{1}{2} S^{i-1/2} \quad (3.7-20)$$

The two terms differenced by this equation are

$$U_{sm}^{i+1*} = \frac{3}{2} U_{sm}^{i+1/2} - \frac{1}{2} U_{sm}^{i-1/2} \quad (3.7-21)$$

$$T_s^{i+1*} = \frac{3}{2} T_s^{i+1/2} - \frac{1}{2} T_s^{i-1/2} \quad (3.7-22)$$

Nomenclature

- $T_p = T_p(x,t)$; primary coolant temp, F
- $T_m = T_m(x,t)$; IEOTSG tube (metal) temp, F
- $T_s = T_s(x,t)$; secondary coolant temp, F
- T_{pi} = primary coolant inlet temp, F
- T_{po} = primary coolant outlet temp, F
- U_{pm} = primary-to-metal heat transfer coeff, Btu/s-ft²-°F
- U_{sm} = secondary-to-metal heat transfer coeff, Btu/s-ft²-°F
- C_{pp} = specific heat of primary coolant, Btu/lb-°F
- C_{pm} = specific heat of IEOTSG metal, Btu/lb-°F
- P_p = wetted perimeter of primary coolant, ft
- P_s = wetted perimeter of secondary coolant, ft
- L = length of steam generator active region, ft
- W_p = primary coolant flow through IEOTSG, lb/s
- A_p = primary coolant flow cross-sectional area, ft²
- A_m = IEOTSG metal cross-sectional area, ft²
- ρ_p = primary coolant density, lb/ft³
- ρ_m = IEOTSG metal density, lb/ft³
- t = real time, seconds
- τ = computer time, seconds
- x = real space, ft

3.8. OTSG Secondary Side

The basic equation on the secondary side establishes conservation for one-dimensional flow.

$$\frac{\partial G}{\partial z} = - \frac{\partial \rho}{\partial t} \quad (3.8-1)$$

This partial differential equation is transformed from space to computer time dependence by the same variable change as the primary equation.

$$\frac{\partial G}{\partial \tau} = \beta \frac{\partial \rho}{\partial \tau} \quad (3.8-2)$$

The change of sign is obtained because the flow on the secondary side is in the opposite direction to the flow on the primary side. This introduces a negative sign in the one-dimensional coordinate notation used on the steam generator.

The change in density with respect to time is done by differencing from one time step to the next.

$$\frac{\partial \rho^{i+1/2}}{\partial \tau} = \frac{\rho^{i+1/2} - \rho^{i-1/2}}{\Delta \tau} \quad (3.8-3)$$

Note the half time steps used in the notation, which skew the secondary equations with the primary.

$$\frac{\partial G^{i+1/2}}{\partial \tau} = \frac{\beta}{\Delta \tau} \rho^{i+1/2} - \frac{\beta}{\Delta \tau} \rho^{i-1/2} \quad (3.8-4)$$

This is the form of the equation for the mass flux. It relates the spatial change in mass flux to the change from one time step to the next of the density.

The energy on the IEOTSG is related by the equation

$$\frac{\partial E}{\partial t} = - \frac{\partial H}{\partial z} + Q. \quad (3.8-5)$$

This is reinforced by

$$H = Gh. \quad (3.8-6)$$

The change of energy density with time is again determined by differencing from one time step to the next.

$$\frac{\partial E^{i+1/2}}{\partial t} = \frac{E^{i+1/2} - E^{i-1/2}}{\Delta \tau} \quad (3.8-7)$$

Note that the use of two time variables poses no contradiction since, for our purposes, computer time and real time are identical.

The Q term in equation 3.8-5 is the heat transferred per unit volume from the tubes to the secondary; it is expressed by temperature differences and heat transfer areas.

$$Q = \frac{P}{A_v} U_{sm} (T_m - T_s) \quad (3.8-8)$$

Combining equations 3.8-7 and 3.8-8 with equation 3.8-5 yields

$$\frac{\partial H^{i+1/2}}{\partial z} = - \left[\frac{E^{i+1/2} - E^{i-1/2}}{\Delta \tau} \right] + \frac{P}{A_v} U_{sm}^{i+1/2} (T_m^{i+1/2} - T_s^{i+1/2}) \quad (3.8-9)$$

The equation has been put in time-difference form with the correct superscript indicating its time relation one half time step skewed from the primary. The equation is changed from space to computer time as the others by a scale factor operation.

$$\begin{aligned} \frac{\partial H^{i+1/2}}{\partial \tau} &= \frac{\beta}{\Delta \tau} E^{i-1/2} - \frac{\beta}{\Delta \tau} E^{i+1/2} + \frac{\beta P}{A_v} U_{sm}^{i+1/2} T_m^{i+1/2} \\ &\quad - \frac{\beta P}{A_v} U_{sm}^{i+1/2} T_s^{i+1/2} \end{aligned} \quad (3.8-10)$$

The calculation of the energy density is done by equation 3.8-11:

$$E^{i+1/2} = \rho^{i+1/2} h^{i+1/2} \quad (3.8-11)$$

The final form of equation 3.8-10 is obtained by recombining the various terms with their scale factors. While this is not the place for such a discussion, it can be summarized as forming such a correct combination of terms, so that the sums and differences of scaling factors cancel out to yield the final equation form.

The quality in the boiling region is calculated from the enthalpies with the following equation:

$$\chi = \frac{h - h_f}{h_g - h_f} \quad (3.8-12)$$

This is used as a monitor on the conditions in the boiling region and signals when film boiling is entered in the last part of the region.

Another indicator of steam generator conditions is a Q term only in the boiling region. The per-unit-volume heat transfer rate is integrated over the length of the boiling region to produce the total heat transferred through the boiling region.

$$Q_B = \int_0^{L_B} Q dz = \int_0^{\tau_B} \beta Q d\tau \quad (3.8-13)$$

$$Q_B = \int_0^{\tau_B} \frac{\beta P}{A_V} U_{sm} (T_m - T_s) d\tau \quad (3.8-14)$$

This value is the total heat to the boiling region.

Nomenclature

- $G = G(x,t)$; mass flux of secondary fluid, lb/s-ft²
- $\rho = \rho(x,t)$; density of secondary fluid, lb/ft³
- $E = E(x,t)$; energy density of secondary fluid, Btu/ft³
- $H = H(x,t)$; energy flux of secondary fluid, Btu/s-ft²
- $h = h(x,t)$; enthalpy of secondary fluid, Btu/lb
- $Q = Q(x,t)$; heat transferred per unit volume, Btu/s-ft³
- $U_{sm} = U_{sm}(x,t)$; heat transfer coeff, Btu/s-ft²-°F
- $T_m = T_m(x,t)$; metal temp, F
- $T_s = T_s(x,t)$; secondary temp, F
- $\chi = \chi(x,t)$; quality of boiling secondary fluid
- β = space-to-time scaling factor, ft/s
- P_s = wetted perimeter of secondary fluid, ft
- A_V = flow area of secondary fluid, ft²
- $z = L-x$; minus space, ft
- x = real space, ft

t = real time, seconds
 τ = computer time, seconds
 h_f = enthalpy at saturated fluid conditions, Btu/lb
 h_g = enthalpy at saturated gas conditions, Btu/lb

3.9. Secondary Pressure

Steam pressure is defined at a point and is a function of time only. The point is the saturated vapor point (100% quality). This point moves as the gas boundary moves within the steam generator channel.

Conservation of mass gives the basic equation for change of density at the point:

$$\dot{\rho}_g = \frac{1}{A_V L_B} (W_i - W_o) \quad (3.9-1)$$

The gas boundary is always at saturated conditions, so the steam properties are functions of pressure.

$$F(P_g) = \frac{P_g}{\rho_g h_g} = b_1 + m_1 P_g \quad (3.9-2)$$

Experimental fits of this property with the steam table have shown that it can be adequately represented by a linear relation as indicated in this equation. The equation can be rearranged to yield gas boundary pressure:

$$P_g = F(P_g) \rho_g h_g \quad (3.9-3)$$

This feedback equation uses ρ_g from equation 3.9-1 and h_g from another steam table fit to produce P_g from a steam table fit.

Nomenclature

P_g = pressure at saturated gas point of secondary, psia
 ρ_g = density at saturated gas point of secondary, lb/ft³
 h_g = enthalpy at saturated gas point of secondary, Btu/lb
 A_V = flow area of secondary fluid, ft²
 L_B = length of boiling region, ft
 W_i = flow into boiling region, lb/s
 W_o = flow out of boiling region, lb/s
 b_1, m_1 = constants used for steam table fits

3.10. Secondary Steam Tables

The behavior of various thermodynamic properties on the secondary side of the steam generator is determined by linear algebraic equations which match these properties to one another through the ASME steam tables.

In general, the steam tables fit spatial functions to spatial functions and point functions to point functions. The following is a list of all the equations:

$$T_b(x,t) = a_1 P_g(t) + b_1 \quad (3.10-1)$$

$$h_f(t) = a_2 P_g(t) + b_2 \quad (3.10-2)$$

$$h_g(t) = a_3 P_g(t) + b_3 \quad (3.10-3)$$

$$T_{sc}(x,t) = a_4 h(x,t) + b_4 P_g(t) + c_4 \quad (3.10-4)$$

$$T_{sh}(x,t) = a_5 h(x,t) + b_5 P_g(t) + c_5 \quad (3.10-5)$$

$$\rho_{sc}(x,t) = a_6 T_{sc}(x,t) + b_6 \quad (3.10-6)$$

$$\rho_{sh}(x,t) = a_7 T_{sh}(x,t) + b_7 P_g(t) + c_7 \quad (3.10-7)$$

$$h_{fw}(t) = a_8 T_{fw}(t) + b_8 \quad (3.10-8)$$

The violations to the general rule that like functions are fit to like functions cause no problem. In equation 3.10-1 the temperature throughout the boiling region is a constant; hence, it can be fit correctly by a point pressure model. Other equations that contain a point pressure term behave similarly.

Note that several of the equations are linked together — equation 3.10-6 is solved from the output of equation 3.10-4. This makes it necessary that all equations be solved in a continuous manner, so that the thermodynamic properties are transferred instantaneously between equations.

The three equations for temperature are switched in at the appropriate points in the steam generator for the region in which they are valid. The regions are determined by comparing the enthalpy, h , with the saturated fluid and gas enthalpies in equations 3.10-2 and 3.10-3.

Boiling Region Density

The boiling region density derivation starts with an expression of thermodynamic properties in the boiling region.

$$h = h_f + \chi(h_g - h_f) \quad (3.10-9)$$

$$v = v_f + \chi(v_g - v_f) \quad (3.10-10)$$

Since density is the inverse of specific volume, the two equations can be solved together to yield an expression for inverse specific volume or density.

$$\rho_b = \frac{1}{v} = \frac{\frac{h_g - h_f}{v_g - v_f}}{v_f \frac{h_g - h_f}{v_g - v_f} - h_f + h} \quad (3.10-11)$$

This equation relates the density in the boiling region to the enthalpy and various combinations of saturated fluid and gas specific volumes and enthalpies. Two steam table fits are defined:

$$\frac{h_g - h_f}{v_g - v_f} = a_9 + m_9 P_g \quad (3.10-12)$$

$$v_f \frac{h_g - h_f}{v_g - v_f} - h_f = a_{10} + m_{10} P_g \quad (3.10-13)$$

The final form of the equation is more complex than the first eight steam table equations, but it is fit to the tables in the same way.

$$\rho_b(x,t) = \frac{a_9 + m_9 P_g(t)}{a_{10} + m_{10} P_g(t) + h(x,t)} \quad (3.10-14)$$

Once again there is the discrepancy of using a spatial and a point function to determine a new spatial function.

Nomenclature

- ρ_b = density in the boiling region, lb/ft³
 h = enthalpy of secondary fluid, Btu/lb
 T_{fw} = temp of feedwater, F
 T_{sc} = temp in subcooled region, F
 T_b = temp in boiling region, F
 T_{sh} = temp in superheat region, F
 ρ_{sc} = density in subcooled region, lb/ft³
 ρ_{sh} = density in superheat region, lb/ft³
 h_f = saturated fluid enthalpy, Btu/lb
 h_g = saturated gas enthalpy, Btu/lb
 h_{fw} = enthalpy of feedwater, Btu/lb

3.11. Feedwater System

The feedwater system consists of a main feedwater pump, a control valve, feedwater heaters, and a control system. Two types of feedwater systems have been modeled: a variable-speed feedwater pump with constant-pressure-drop feedwater valve and a constant-speed feedwater pump with variable-pressure-drop feedwater valve.

The feedwater pressure balance is

$$P_{sgi} = P_d - \Delta P_f - \Delta P_{st} - \Delta P_v - \Delta P_{ta} \quad (3.11-1)$$

where

$$\Delta P_{ta} = \frac{L \dot{W}_{fw}}{144gA}, \quad (3.11-2)$$

$$\Delta P_f = K_2 W_{fw}^2, \quad (3.11-3)$$

and

$$\Delta P_v = \left(\frac{W_{fw}}{C_v} \right)^2. \quad (3.11-4)$$

Rearrange equation 3.11-1 to isolate the derivative.

$$P_{sgi} = P_d - K_2 W_{fw}^2 - \Delta P_{st} - \left(\frac{W_{fw}}{C_v} \right)^2 - \frac{L \dot{W}_{fw}}{144gA} \quad (3.11-5)$$

$$\frac{dW_{fw}}{dt} = \frac{144gA}{L} \left[P_d - K_2 W_{fw}^2 - \Delta P_{st} - \left(\frac{W_{fw}}{C_v} \right)^2 - P_{sgi} \right] \quad (3.11-6)$$

The main feed pump discharge pressure and flow as functions of speed are obtained from the head capacity curve —

$$P_d = f_1(W_{fw}) \quad (3.11-7)$$

The friction loss term is a square function of the initial friction loss in the line.

$$K_2 = \frac{\Delta P_{fw(o)}}{W_{fw(o)}^2} ; \text{ initial 100\% pressure and flow.} \quad (3.11-8)$$

The static pressure loss is

$$\Delta P_{st} = \text{constant.} \quad (3.11-9)$$

The valve loss is a function of flow and valve coefficient. The valve coefficient is usually entered as a vendor-supplied function of valve stroke. This allows for "shaping" the valve characteristics at various flow values.

$$C_v = f_2(X_{cv}) \quad (3.11-10)$$

The steam generator inlet pressure is usually computed from the steam pressure by a friction loss term.

$$P_{sgi} = P_g + K_3 W_{fw}^2 \quad (3.11-11)$$

where

$$K_3 = \frac{P_{sgi(o)} - P_{g(o)}}{W_{fw(o)}^2} ; \text{ initial 100\% pressure and flow.} \quad (3.11-12)$$

The steam generator outlet pressure is calculated by a similar formula:

$$P_{sgo} = P_g - K_4 W_{st}^2 \quad (3.11-13)$$

where

$$K_4 = \frac{P_{g(o)} - P_{sgo(o)}}{W_{st(o)}^2} \quad (3.11-14)$$

The feedwater valve coefficient is a functional interpolation from the feedwater valve stroke. The stroke is obtained by control action on an error signal for the feedwater demand.

$$\frac{dX_{cv}}{dt} = \frac{X_{cv(o)} + C_1 E_{fwd} + C_2 \int_0^t E_{fwd} dt - X_{cv}}{\tau_{cv}} \quad (3.11-15)$$

The modeling of a variable-speed feedwater model is very similar; the difference is that the pump pressure is obtained from a family of curves that relate pump head to both flow and speed. This is formulated as

$$\Delta P_p = w^2 f_3(W_{fw}/w) \quad (3.11-16)$$

where f_3 is the function that is the family of curves relating $\Delta P_p/w^2$ versus W/w . After the pressure across the pump is obtained, the discharge pressure is obtained by

$$P_d = \Delta P_p + P_s \quad (3.11-17)$$

where

$$P_s = f_4(W_{fw}). \quad (3.11-18)$$

Pump speed is regulated by a controller on feedwater valve pressure error:

$$\frac{dw}{dt} = \frac{w(o) + C_3(\Delta P_v - \Delta P_{v,set}) + C_4 \int_0^t (\Delta P_v - \Delta P_{v,set}) dt - w}{\tau_{pump}} \quad (3.11-19)$$

Nomenclature

- P_g = steam generator gas boundary pressure, psia
- P_{sgi} = steam generator inlet pressure, psia
- P_{sgo} = steam generator outlet pressure, psia
- P_d = main feed pump discharge pressure, psia
- ΔP_f = feedwater line friction loss, psi
- ΔP_{st} = feedwater line static loss, psi
- ΔP_v = feedwater line control valve loss, psi
- ΔP_{ta} = feedwater line transient acceleration loss, psi
- ΔP_p = main feed pump pressure gain, psi
- w = main feed pump impeller speed, rpm
- P_s = main feed pump suction pressure, psia
- L = feedwater line active length, ft
- A = feedwater line active area, ft²

Nomenclature

K_{tv}	= turbine valve coeff, lb/s-psia
K_2	= sensitivity of pressure sensor in controller
K_3	= valve stroke rate/pressure applied, %/s/psia
K_1	= pressure error/megawatt error, %/%
P_t	= turbine pressure, psia
P_{tset}	= turbine set pressure, psia
MW_d	= megawatt demand, %
MW_g	= megawatts generated, %
W_t	= turbine flow, lb/s
W_{st}	= steam line flow, lb/s
P_g	= steam generator gas boundary pressure, psia
K_{gt}	= friction loss from gas boundary to turbine, psi/(lb/s) ²
P_v	= pressure $\times$ specific volume product, psia $\times$ ft ³ /lb
W_{sgo}	= steam generator outlet flow, lb/s
g_c	= gravitational constant, lbf-ft/lbf-s ²
P_{sgo}	= pressure at steam generator outlet, psia
K_{sgo}	= friction loss from P_g to P_{sgo} , psi/(lb/s) ²
W_{st11}	= steam flow between P_{sgo} and P_{st1} , lb/s
K_{st11}	= friction loss from P_{sgo} to P_{st1} , psi/(lb/s) ²
W_{st12}	= steam flow between P_{st1} and P_t , lb/s
K_{st12}	= friction factor from P_{st1} to P_t , psi/(lb/s) ²
P_{st1}	= pressure at midpoint of steam line valves, psia
L/A_n	= length/area ratio for region n, 1/ft
V_n	= volume of region n, ft ³

3.13. Turbine Bypass System/Safety Valves

The expression for the turbine bypass system and the steam safety valves is

$$W_{bp} = P_3 K_{bp} \quad (3.13-1)$$

$$W_{sv} = P_3 K_{sv} \quad (3.13-2)$$

These equations assume critical flow across the bypass and safety valves.

The bypass valve flow coefficient is a function of valve stroke, and valve stroke as a function of time is established by the control system.

$$K_{bp} = f(S_{bp}) \quad (3.13-3)$$

The steam safety valve flow coefficient is a function of pressure and is determined from vendor data.

The number and size of the bypass and safety valves vary depending on contract requirements. The number and capacity of bypass valves are simulated for each contract, and the capacity of the safety valves is varied for contract requirements.

Nomenclature

- W_{bp} = turbine bypass flow, lb/s
- W_{sv} = steam safety valve flow, lb/s
- P_3 = steam line pressure at node 3, psi
- K_{bp} = bypass valve flow coefficient
- K_{sv} = steam safety valve flow coefficient
- S_{bp} = stroke of bypass valves, %

3.14. Turbine-Generator Model

The turbine-generator model develops a generation term, which is produced by considering the quantity and quality of fluid flowing through the control valves to the turbine.

$$MW_g = k \times W_t \times H_{stm} \times \eta \quad (3.14-1)$$

The flow term comes from the equation in the steam system that uses pressure and control valve coefficient. The efficiency term is interpolated from a function relating it to turbine flow:

$$\eta = f_1(W_t) \quad (3.14-2)$$

The enthalpy term comes from rearranging a steam table equation which produced steam temperature from enthalpy.

$$T = a_5 h + b_5 P + c_5 \quad (3.14-3)$$

Isolate the enthalpy term

$$h = \frac{T - b_5 P - c_5}{a_5} \quad (3.14-4)$$

Solve this equation at the steam generator outlet conditions, i.e., $T = T_{stm}$, $P = P_{sgo}$.

$$H_{stm} = \frac{T_{stm} - b_5 P_{sgo} - c_5}{a_5} \quad (3.14-5)$$

This equation is solved at the conditions above. The constant term in the equation is used to normalize the values of the various terms to calculate a megawatt generation in convenient units.

The generation is determined from steam production parameters. The demand is set by the unit load demand (ULD) controller. A megawatt error term is obtained from these two.

$$MW_e = MW_d - MW_g \quad (3.14-6)$$

This term is applied to the turbine control system.

Nomenclature

- MW_d = generation demand, MW
- MW_g = generation production, MW
- MW_e = generation error, MW
- W_t = turbine flow, lb/s
- H_{stm} = enthalpy at steam generator outlet
- η = efficiency of turbine-generator
- f_1 = function relating efficiency to flow
- T = temperature, F

T_{stm} = steam outlet temperature, F
 a_5, b_5, c_5 = steam table constants for superheat region
 P_{sgo} = steam generator outlet pressure, psia

4. INPUT/OUTPUT

This section describes pertinent input information required and the output information obtained for a typical transient. A 10% down step of the plant demand from full power (100%) level to the 90% level is the reference transient. The relevant input data to describe the plant modeled are presented as a direct listing of the data deck submitted to the hybrid initialization program. Sample tracings of important system parameters that are recorded on the strip chart (eight-channel) recorder are reproduced along with the proper scaling factors to convert directly to engineering units.

4.1. Input Data

ALPHA0=.1044	10000 * THE ABSOLUTE DELTA K COEFFICIENT.
ALPHA0=.000	1000 * THE ABSOLUTE DELTA K COEFFICIENT.
AM=7.0137	STEAM GENERATOR METAL CROSS SECTION AREA, FT ² .
AP=27.074	STEAM GENERATOR PRIMARY FLOW CROSS SECTION AREA, FT ² .
ASHD=22.000	FLOW AREA OF STEAM GENERATOR SHROUD, FT ² .
ASTL1=20.40	FLOW AREA OF STEAM LINE FROM PSGO TO PSTL FT2
ASTL2=17.22	FLOW AREA OF STEAM LINE FROM PSTL TO PT FT2
AV=40.760	STEAM GENERATOR SECONDARY FLOW CROSS SECTION AREA, FT ² .
BETA=631.515	SPATIAL INTEGRATION RATE OF STEAM GENERATOR, FT/SEC.
BETA1=.000208	CONCENTRATION FRACTION OF DELAYED NEUTRON GROUP NO. 1.
BETA2=.001410	CONCENTRATION FRACTION OF DELAYED NEUTRON GROUP NO. 2.
BETA3=.001394	CONCENTRATION FRACTION OF DELAYED NEUTRON GROUP NO. 3.
BETA4=.002701	CONCENTRATION FRACTION OF DELAYED NEUTRON GROUP NO. 4.
BETA5=.000028	CONCENTRATION FRACTION OF DELAYED NEUTRON GROUP NO. 5.
BETA6=.000315	CONCENTRATION FRACTION OF DELAYED NEUTRON GROUP NO. 6.
BETAT=.000380	TOTAL DELAYED NEUTRON FRACTION FOR SIX GROUPS.
CHI=95.0	STEAM QUALITY ASSUMED AT START OF FILM BOILING, PCT.
CPF=.005113	SPECIFIC HEAT OF FUEL IN THE REACTOR, BTU/LB-F.
CPM=.121	SPECIFIC HEAT OF METAL IN STEAM GENERATOR, BTU/LB-F.
CPW=1.32500	SPECIFIC HEAT OF PRIMARY FLUID, BTU/LB-F.
CPLMX=5.0	DEADEND OF REACTOR CROSSLIMITS, PERCENT.
CPLMF=5.0	DEADEND OF FEEDWATER CROSSLIMITS, PERCENT.
DPVSFT=0.0	FEEDWATER CONTROL VALVE PRESSURE LOSS SET POINT, PSI.
DT=.2	MAJOR CYCLE TIME OF HYBRID STEAM GENERATOR MODEL, SEC.
DYNMIC=65.7	DYNAMIC FLOW LOSSES OF FEEDWATER SYSTEM, PSI.
E=2.71828	BASE OF THE NATURAL LOGARITHMS.
F1=526.719	SATURATED FLUID ENTHALPY AT T1 AND P1, BTU/LB.
F2=544.867	SATURATED FLUID ENTHALPY AT T2 AND P2, BTU/LB.
F3=571.871	SATURATED FLUID ENTHALPY AT T3 AND P3, BTU/LB.
F4=642.520	SATURATED FLUID ENTHALPY AT P4 PRESSURE, BTU/LB.
F5=696.659	SATURATED FLUID ENTHALPY AT P5 PRESSURE, BTU/LB.
F6=763.920	SATURATED FLUID ENTHALPY AT P6 PRESSURE, BTU/LB.

FW1=390.0	MINIMUM ALLOWABLE FEEDWATER TEMPERATURE, F.
FW2=457.0	EXPECTED NORMAL FEEDWATER TEMPERATURE, F.
FW3=465.0	MAXIMUM ALLOWABLE FEEDWATER TEMPERATURE, F.
G1=1196.39	SATURATED GAS ENTHALPY AT T1 AND P1, BTU/LB.
G2=1192.38	SATURATED GAS ENTHALPY AT T2 AND P2, BTU/LB.
G3=1184.81	SATURATED GAS ENTHALPY AT T3 AND P3, BTU/LB.
G4=1155.53	SATURATED GAS ENTHALPY AT T4 PRESSURE, BTU/LB.
G5=1121.28	SATURATED GAS ENTHALPY AT P5 PRESSURE, BTU/LB.
G6=1062.97	SATURATED GAS ENTHALPY AT P6 PRESSURE, BTU/LB.
G7=1093.32	SATURATED GAS ENTHALPY AT P7 PRESSURE, BTU/LB.
G8=1112.27	SATURATED GAS ENTHALPY AT P8 PRESSURE, BTU/LB.
GC=32.17	GRAVITATIONAL CONSTANT, LB-FT/LB-SEC**2.
H1=365.3	SUBCOOLED ENTHALPY AT FW1 AND P2, BTU/LB.
H2=440.521	SUBCOOLED ENTHALPY AT FW2 AND P2, BTU/LB.
H3=447.397	SUBCOOLED ENTHALPY AT FW3 AND P2, BTU/LB.
H4=1243.42	SUPERHEATED ENTHALPY AT T4 AND P2, BTU/LB.
H5=1228.54	SUPERHEATED ENTHALPY AT T5 AND P2, BTU/LB.
H6=566.520	SUBCOOLED ENTHALPY AT TEMPL AND P5, BTU/LB.
IOT=-0.1	NEGATIVE OF THE TIME STEP FOR DIGITAL CALCULATIONS, SEC.
INTG=4.0	INTEGRAL GAIN FOR TBP, PERCENT STROKE/PSI DEGREE/SECOND
K2=1.0	TURBINE VALVE, PRESSURE ERROR/PRESSURE MEASURED.
K5=1.0	FEEDWATER VALVE, PERCENT STROKE/PERCENT ERROR.
KATCI=.05	AVERAGE TEMPERATURE CONTROL INTEGRAL GAIN, FCT/DEGF/SEC.
KATCF=2.0	AVERAGE TEMPERATURE CONTROL PROPORTIONAL GAIN, PCT/DEGF.
KLSTAP=.0346	1000 * THE LENTHON LIFETIME, SEC.
KMWEI=.05	MEGAWATT ERROR FEEDBACK INTEGRAL GAIN, FCT/SEC/FCT.
KMWEF=0.0	MEGAWATT ERROR FEEDBACK PROPORTIONAL GAIN, FCT/FCT.
KPMPSI=0.0	FEEDWATER PUMP SPEED CONTROL INTEGRAL GAIN, FCT/SEC/FCT.
KPMPSF=0.0	FEEDWATER PUMP SPEED CONTROL PROPORTIONAL GAIN, PCT/FCT.
KXCVI=.033	FEEDWATER CONTROL VALVE INTEGRAL GAIN, FCT/FCT/SEC.
KXCVF=1.0	FEEDWATER CONTROL VALVE PROPORTIONAL GAIN, PCT/FCT.
LAFW=440.873	TOTAL OF LENGTHS/AREAS FOR FEEDWATER SYSTEM, 1/FT.
L=52.1	ACTIVE LENGTH OF STEAM GENERATOR, FT.
LAMDA1=.0125	DECAY CONSTANT OF DELAYED NEUTRON GROUP NO. 1, 1/SEC.
LAMDA2=.0308	DECAY CONSTANT OF DELAYED NEUTRON GROUP NO. 2, 1/SEC.
LAMDA3=.114	DECAY CONSTANT OF DELAYED NEUTRON GROUP NO. 3, 1/SEC.
LAMDA4=.337	DECAY CONSTANT OF DELAYED NEUTRON GROUP NO. 4, 1/SEC.
LAMDA5=1.13	DECAY CONSTANT OF DELAYED NEUTRON GROUP NO. 5, 1/SEC.
LAMDA6=3.19	DECAY CONSTANT OF DELAYED NEUTRON GROUP NO. 6, 1/SEC.
LSG=40.0	LENGTH OF SUPERHEAT REGION, PLUS OUTLET PLENUM, FT.
LSHD=40.0	LENGTH OF STEAM GENERATOR SHROUD, FT.
LSTL1=304.0	LENGTH OF STEAM LINE FROM PSCJ TO ESTL, FT.
LSTL2=301.3	LENGTH OF STEAM LINE FROM ESTL TO P1, FT.
MF=297145.	MASS OF FUEL IN REACTOR CORE, LPM.
MG=6476.4	INITIAL GAS MASS IN THE PRESSURIZER, LB.
MINUS=-.9999	DEFINITION OF MINUS REFERENCE FOR HANDSET LIMITERS.
MT=51420.4	TOTAL INITIAL MASS IN THE PRESSURIZER, LB.
MW=45975.5	MASS OF WATER IN REACTOR CORE, LPM.
MWELIM=13.0	BIPOLAR LIMIT ON MEGAWATT ERROR TO TURBINE, FCT.
MZEPG=-0.0	DEFINITION OF MINUS ZERO FOR HANDSET LIMITERS.
NOOTYH=.00055	TIME BETWEEN SPATIAL NODES IN HYPR1 DEUTSG, SEC.
P1=900.0	MINIMUM EXPECTED PRESSURE AT STEAM INTERFACE, PSIA.
P2=1015.0	EXPECTED NORMAL PRESSURE AT STEAM INTERFACE, PSIA.
P3=1200.0	MAXIMUM EXPECTED PRESSURE AT STEAM INTERFACE, PSIA.
P4=1750.0	MINIMUM PRESSURIZER PRESSURE, PSIA.
P5=2210.0	EXPECTED PRESSURIZER PRESSURE, PSIA.
P6=2750.0	MAXIMUM PRESSURIZER PRESSURE, PSIA.
P7=2515.0	PRESSURIZER RELIEF VALVE SET PRESSURE, PSIA.
P8=2310.0	PRESSURIZER FILL VALVE SET PRESSURE, PSIA.
P9=2260.0	PRESSURIZER DRAIN VALVE SET PRESSURE, PSIA.
P10=2210.0	PRESSURIZER HEATER BANK SET PRESSURE, PSIA.
PCOND=0.0	TURBINE CONDENSER PRESSURE, PSIA.
PI=3.14159	UNIVERSAL CONSTANT.

PLUS=.9339	DEFINITION OF PLUS REFERENCE FOR HANDSET LIMITERS.
PN1=1025.0	BYPASS CONTROL VALVE 1 NORMAL SETPOINT, PSIA.
PN2=1025.0	BYPASS CONTROL VALVE 2 NORMAL SETPOINT, PSIA.
PN3=1025.0	BYPASS CONTROL VALVE 3 NORMAL SETPOINT, PSIA.
PN4=1025.0	BYPASS CONTROL VALVE 4 NORMAL SETPOINT, PSIA.
PP=2333.2	STEAM GENERATOR PRIMARY SIDE WETTED PERIMETER, FT.
PROPO=0.4	PROPORTIONAL GAIN FOR TOP, PERCENT STROKE/PSI ERROR
PRT1=1125.0	BYPASS CONTROL VALVE 1 REACTOR TRIP SETPOINT, PSIA.
PRT2=1125.0	BYPASS CONTROL VALVE 2 REACTOR TRIP SETPOINT, PSIA.
PRT3=1125.0	BYPASS CONTROL VALVE 3 REACTOR TRIP SETPOINT, PSIA.
PRT4=1125.0	BYPASS CONTROL VALVE 4 REACTOR TRIP SETPOINT, PSIA.
PS=2618.0	STEAM GENERATOR SECONDARY SIDE WETTED PERIMETER, FT.
PSGI=1050.0	STEAM GENERATOR INLET PRESSURE, PSIA.
PSGO=1000.0	STEAM GENERATOR OUTLET PRESSURE, PSIA.
PSVS=1180.0	ATMOSPHERIC SAFETY VALVE SET PRESSURE, PSIA.
PSVO=1223.0	ATMOSPHERIC SAFETY VALVE OPENING PRESSURE, PSIA.
PSVR=1123.0	ATMOSPHERIC SAFETY VALVE RESET PRESSURE, PSIA.
PSTL=983.65	PRESSURE OF STEAM LINE AT CONDENSER BYPASS LINE, PSIA.
FTSET=975.0	TURBINE TRIP SETPOINT, PSIA.
PTT1=975.0	BYPASS CONTROL VALVE 1 SETPOINT FOR TURBINE TRIP, PSIA.
PTT2=975.0	BYPASS CONTROL VALVE 2 SETPOINT FOR TURBINE TRIP, PSIA.
PTT3=975.0	BYPASS CONTROL VALVE 3 SETPOINT FOR TURBINE TRIP, PSIA.
PTT4=975.0	BYPASS CONTROL VALVE 4 SETPOINT FOR TURBINE TRIP, PSIA.
PZERO=0.0	DEFINITION OF PLUS ZERO FOR HANDSET LIMITERS.
Q=3720.0	NET THERMAL RATING OF REACTOR CORE POWER, MW.
QHEAT=1462.52	PRESSURIZED HEATER BACK TOTAL ENERGY OUTPUT, BTU/SEC.
QPUMPS=20	POWER ADDITION TO SYSTEM FROM FOUR REACTOR COOLANT PUMPS
R1=54.3232	SUBCOOLED DENSITY AT P1 AND P2, LB/FT ³ .
R2=46.1997	SATURATED FLUID DENSITY AT T2 AND P2, LB/FT ³ .
R3=44.8038	SATURATED FLUID DENSITY AT T3 AND P3, LB/FT ³ .
R4=2.27935	SATURATED GAS DENSITY AT T2 AND P2, LB/FT ³ .
R5=2.90424	SUPERHEATED DENSITY AT T4 AND P2, LB/FT ³ .
R6=1.71890	SUPERHEATED DENSITY AT T4 AND P1, LB/FT ³ .
R7=47.1019	SATURATED FLUID DENSITY AT T1 AND P1, LB/FT ³ .
R8=1.94636	SATURATED GAS DENSITY AT T1 AND P1, LB/FT ³ .
R9=2.75901	SATURATED GAS DENSITY AT T3 AND P3, LB/FT ³ .
RAMP=10.0	TIME TO RAMP THE SETPOINT FROM NORMAL TO TX TRIP, SEC.
RATEPV=200.0	PRESSURIZED RELIEF VALVE OPENING RATE, PERCENT/SEC.
ROTRVL=243.25	TOTAL ROD TRAVEL OF COMBINED ROD GROUPS, INCHES.
RM=525.3	DENSITY OF THE METAL IN THE STEAM GENERATOR, LB/FT ³ .
RP=43.256	AVERAGE DENSITY OF PRIMARY FLUID, LB/FT ³ .
STATIC=7.1	STATIC ELEVATION LOSSES OF FEEDWATER SYSTEM, PSI.
STROKE=45.0	TURBINE VALVE STROKE TIME FOR STROKE PSI ERROR, SEC.
STKPR=10.0	PRESSURE GRAB FOR VALVE CLOSURE IN STROKE TIME, PSI.
SURGE=4000.0	MAXIMUM EXPECTED PRESSURIZER SURGE FLOW, LB/SEC.
S=4000.0	SET EQUAL TO SURGE PARAMETER, LB/SEC.
SVCLCL=.921825	SPECIFIC VOLUME OF COLD LEG FOR TEMCL, FT ³ /LB.
SVCLHL=.924595	SPECIFIC VOLUME OF HOT LEG FOR TEMHL, FT ³ /LB.
SVCLFW=.31947	SPECIFIC VOLUME OF FEEDWATER FLOW FOR FW2, FT ³ /LB.
T1=531.955	SATURATION TEMPERATURE AT P1 PRESSURE, F.
T2=546.333	SATURATION TEMPERATURE AT P2 PRESSURE, F.
T3=567.137	SATURATION TEMPERATURE AT P3 PRESSURE, F.
T4=595.4	EXPECTED FULL POWER STEAM TEMPERATURE, F.
T5=579.741	SUPERHEATED TEMPERATURE AT P1 AND P4, F.
T6=532.114	SUBCOOLED TEMPERATURE AT P3 AND P1, F.
TAUCV=3.0	TIME CONSTANT OF FEEDWATER CONTROL VALVE, SEC.
TAUFW=3.0	DUMMY VARIABLE (TO OF FW PUMP, IF PUMP IS VARIABLE SPEED)
TAUIE=3.0	TIME CONSTANT OF ATC INTEGRAL BLEED CIRCUIT, SEC.
TAURC=15.0	TIME CONSTANT OF REACTOR COOLANT PUMP COASTDOWN, SEC.
TAURD=.25	TIME CONSTANT OF CONTROL ROD DRIVE MOTOR, SEC.
TAUSTH=.5	TIME CONSTANT OF STEAM TEMPERATURE RESPONSE, SEC.
TAUTC=3.0	TIME CONSTANT OF PRIMARY LOCK THE LOCKOUTS, SEC.

TEMPHIL=624.59	100 PERCENT POWER HOT LEG TEMPERATURE, F.
TEMPCL=564.77	100 PERCENT POWER COLD LEG TEMPERATURE, F.
TEMPRI=565.25	100 PERCENT POWER REACTOR INLET TEMPERATURE, F.
TFAK=1340.	AVERAGE FULL TEMPERATURE AT 100 PERCENT POWER, F.
THEAT=30.0	PRESSURIZER HEATER RACK TIME CONSTANT, SEC.
TFILOT=.25	PRESSURIZER PILOT VALVE TIME CONSTANT, SEC.
TSPRAY=3.0	TIME CONSTANT OF PRESSURIZER SPRAY VALVE, SEC.
TWBAF=594.32	100 PERCENT POWER REACTOR AVERAGE COOLANT TEMP., F.
UPM=.8452	PRIMARY-METAL HEAT TRANSFER COEFFICIENT, BTU/SEC-FT ² -F.
UB=0.8750	SECONDARY BOILING HEAT TRANSFER COEFFICIENT.
USC=.1873	SECONDARY SUBCOOLED HEAT TRANSFER COEFFICIENT.
USH=.0120	SECONDARY SUPERHEATED HEAT TRANSFER COEFFICIENT.
V1=.024498	SATURATED FLUID SPECIFIC VOLUME AT P4 PRESSURE, FT ³ /LB.
V2=.026748	SATURATED FLUID SPECIFIC VOLUME AT P5 PRESSURE, FT ³ /LB.
V3=.030791	SATURATED FLUID SPECIFIC VOLUME AT P6 PRESSURE, FT ³ /LB.
V4=.227118	SATURATED GAS SPECIFIC VOLUME AT P4 PRESSURE, FT ³ /LB.
V5=.161536	SATURATED GAS SPECIFIC VOLUME AT P5 PRESSURE, FT ³ /LB.
V6=.107471	SATURATED GAS SPECIFIC VOLUME AT P6 PRESSURE, FT ³ /LB.
V7=.021231	SATURATED FLUID SPECIFIC VOLUME AT P1 PRESSURE, FT ³ /LB.
V8=.021645	SATURATED FLUID SPECIFIC VOLUME AT P2 PRESSURE, FT ³ /LB.
V9=.022320	SATURATED FLUID SPECIFIC VOLUME AT P3 PRESSURE, FT ³ /LB.
V10=.500912	SATURATED GAS SPECIFIC VOLUME AT P1 PRESSURE, FT ³ /LB.
V11=.438606	SATURATED GAS SPECIFIC VOLUME AT P2 PRESSURE, FT ³ /LB.
V12=.362443	SATURATED GAS SPECIFIC VOLUME AT P3 PRESSURE, FT ³ /LB.
VCLRXO=1601.34	ACTIVE PRIMARY FLOW VOLUME, REACTOR OUTLET PLENUM, FT ³ .
VOLHL=1500.25	ACTIVE PRIMARY FLOW VOLUME, BOTH HOT LEGS, FT ³ .
VOLSGI=529.29	ACTIVE PRIMARY FLOW VOLUME, 2 IDENTICAL INLET PLENUMS, FT ³ .
VOLSGO=2985.50	ACTIVE PRIMARY FLOW VOLUME, 2 IDENTICAL TUBE BUNDLES, FT ³ .
VOLSGO=557.06	ACTIVE PRIMARY FLOW VOLUME, 2 IDENTICAL OUTLET PLENUMS, FT ³ .
VOLCL=1313.63	ACTIVE PRIMARY FLOW VOLUME, ALL FOUR COLD LEGS, FT ³ .
VCLRXI=1623.10	ACTIVE PRIMARY FLOW VOLUME, REACTOR INLET PLENUM, FT ³ .
VOLRX=826.67	ACTIVE PRIMARY FLOW VOLUME, REACTOR CORE, FT ³ .
VOLT=10936.8	TOTAL ACTIVE PRIMARY FLOW VOLUME, FT ³ .
VCLFW=700.77	ACTIVE FEEDWATER FLOW VOLUME, HEATERS TO GENERATOR, FT ³ .
VT=2250.0	TOTAL VOLUME OF THE PRESSURIZER, FT ³ .
WFW=2198.0	FULL FEEDWATER FLOW PER IDENTICAL, LB/SEC.
WPILOT=41.67	PRESSURIZER PILOT VALVE FULL FLOW RATE, LB/SEC.
WPELEF=277.78	PRESSURIZER RELIEF VALVE FULL FLOW RATE, LB/SEC.
WST=2198.0	FULL STEAM FLOW PER IDENTICAL, LB/SEC.
WSPRAY=28.07	PRESSURIZER SPRAY VALVE FULL FLOW RATE, LB/SEC.
WT=41875.0	TOTAL PRIMARY FLOW THROUGH COLE, LB/SEC.
XCR0=91.43	INITIAL POSITION OF COMBINED ROD GROUPS, PCT WITHDRAWN.
XCV=95.0	NORMAL FULL FLOW FEEDWATER VALVE STROKE, PCT.
XHEAT=0.0	PRESSURIZER HEATER DEAD BAND OF SET PRESSURE, PERCENT.
XPILOT=1.08	PRESSURIZER PILOT DEAD BAND OF SET PRESSURE, PERCENT.
XRELEF=2.5	PRESSURIZER RELIEF DEAD BAND OF SET PRESSURE, PERCENT.
XSPRAY=2.21	PRESSURIZER SPRAY DEAD BAND OF SET PRESSURE, PERCENT.
XCAP1=25.00	BYPASS CONTROL VALVE 1 FLOW CAPACITY AT P1SET, PCT.
XCAP2=25.00	BYPASS CONTROL VALVE 2 FLOW CAPACITY AT P1SET, PCT.
XCAP3=25.00	BYPASS CONTROL VALVE 3 FLOW CAPACITY AT P1SET, PCT.
XCAP4=25.00	BYPASS CONTROL VALVE 4 FLOW CAPACITY AT P1SET, PCT.
XSAFE=112.0	ATMOSPHERE SAFETY VALVE RELIEF CAPACITY AT P5G0, PCT.

4.2. Output Data

These data (for 10% demand step response — 100 to 90%) are shown in graph form (analog output) on the following pages.

Primary
Variables

Secondary
Variables