MODELING AND SIMULATION OF A DRUM BOILER-TURBINE
POWER PLANT UNDER EMERGENCY STATE CONTROL,

by

PATRICK BENEDICT USORO

B.Eng., Ahmadu Bello University, Zaria, Nigeria
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SUBMITTED IN PARTIAL FULFILLMENT
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DEGREE OF
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MASSACHUSETTS INSTITUTE OF TECHNOLOGY

(May, 1977)

Signature redacted

Signature of Author............................................................... Department of Mechanical Engineering, May 12, 1977

Signature redacted

Certified by................................................................. Thesis Supervisor

Signature redacted

Accepted by................................................................. Chairman, Department Committee

AUG 18 1977
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Submitted to the Department of Mechanical Engineering on May 12, 1977,
in Partial Fulfillment of the Requirements for the Degree of Master of
Science in Mechanical Engineering.

ABSTRACT

A physically based forty-seventh order Digital Computer Model
has been developed to simulate the dynamics of a drum boiler-turbine
power plant under normal and emergency operating conditions. Twenty-
three state variables describe the physical processes involved while
twenty-four state variables describe the control system. The model in-
cludes all the major components and auxiliaries of the power plant and
incorporates interactions between the mechanical and electrical com-
ponents of the power system. Seven emergency type simulations have
been performed to study the behavior of the power plant under rapid
load increase/decrease, reduced voltage and frequency, and failure of
major plant equipment.

A reduced order model of the drum boiler-turbine power plant
has also been developed for use in analog computer simulation. The
simplification of the model is obtained by neglecting voltage and fre-
quency effects and lumping together a number of steam and air side
elements. The reduced model is twenty-seventh order with twelve state
variables describing the physical processes and fifteen state variables
describing the control system. Four simulations have been run to compare
the responses of the reduced model with that of the Digital Computer
Model.

Thesis Supervisor: Professor D.N. Wormley
Title: Associate Professor of Mechanical Engineering
DEDICATED TO THE ALMIGHTY GOD AND MY PARENTS
ACKNOWLEDGMENTS

I am very grateful to my Thesis Advisor, Professor D.N. Wormley, for his invaluable guidance and encouragement all through the course of this project.

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I. INTRODUCTION

An electric power system operates in a number of modes such as the Normal Mode (usual and acceptable situations) and Emergency Mode (occurrence of transients of unacceptable characteristics). The emergency mode manifests itself in slow speed frequency and voltage variations, possible power system equipment damage or even "blackout". Major power system disturbances in recent years have stimulated interest in the study of emergency state control for power system stability and reliability.

In the maneuver of a power system during emergencies, the mechanical prime movers comprising the boiler-turbine units place limitations on the performance of the overall power system. These limitations include maximum power capability limits and power rate limits which can be safely undertaken under existing conditions.

An accurate representation of the boiler-turbine unit is therefore necessary in developing and evaluating a reliable slow speed dynamics emergency state control strategy for the power system. An adequate boiler-turbine model is desired which:

(i) Can provide a physically realistic representation of the mechanical power system (boiler-turbine unit).

(ii) Incorporates interactions between the mechanical and electrical components of the power system.

(iii) Is capable of simulating a wide range of emergency situations.

(iv) Can be employed in studies and testing of emergency state control strategies.
Many boiler-turbine models are available in the literature [1,2,3,4,5,6,7,8,9,11] but these do not satisfy all the requirements stated above. Most of these models are based on certain simplifying assumptions which are valid for normal modes of operation but are inadequate for emergency situations. Such assumptions can lead to misleading results in developing and evaluating an emergency state control strategy.

A drum boiler-turbine power plant model has been developed which incorporates the characteristics required for emergency state control studies. This model is characterized by the following features:

(i) It is developed based on physical processes with the model parameters determined from geometry, material properties and manufacturers' data.

(ii) System nonlinearities are included so that the model remains valid over a wide operating range.

(iii) Steam table property fits are used to provide physically realistic thermodynamic properties over a wide operating range.

(iv) Principal plant auxiliaries such as pumps and fans are explicitly modeled along with their corresponding electrical prime movers (induction motors) and their dependence on driving voltage and frequency. These auxilliary models permit the simulation of the plant under reduced voltage and/or frequency which frequently accompany emergency conditions. This dependence of principal plant components on voltage and frequency places additional limitations on the performance of the plant. (This mechanical-electrical component interconnection is not modeled in most existing boiler turbine models).

(v) The dynamics of the feedwater and condensate side are explicitly modeled. This part of the system is usually ignored on the basis that the associated dynamics do not significantly affect the steam side. While this
ascertion is reasonably correct for operations in normal mode, simulations and experience with actual power plants show that the condensate and feedwater dynamics can affect the overall system in various ways. In fact, under certain emergency conditions it is the water side dynamics that limit the plant response; for example, it may trigger a station trip.

(vi) Though the model is intended to be representative of any typical drum boiler-turbine unit, a current operating power plant is used as a prototype for the model so that a verification of the model could be made.

With the characteristics listed above, the model developed represents an extension of some existing models [1,2,3,11] in two primary areas. First, the condensate and feedwater side dynamics have been modeled and second the electrical prime movers which run fans and pumps and their dependence upon driving voltage and frequency have been modeled.

In addition to a fully implemented "digital computer model" a simpler "standard model" has been developed for analog computer simulation. The use of the simpler 'standard model' was necessitated by the fact that the digital computer model is too large to be implemented directly in hardwired electronics. The standard model was developed in a manner similar to the digital model but based on certain simplifying assumptions which were verified through simulations of the digital model.
CHAPTER II
DESCRIPTION OF SYSTEM

The prototype unit modeled is an operating 600MW steam power plant which was recently commissioned and therefore includes features of a typical modern power plant with the associated controls.

The steam generating equipment is an oil-fired, balanced draft, controlled recirculation drum boiler capable of delivering \(4.2 \times 10^6\) lb/hr of steam at a pressure of 2600 psig and 1005°F and reheat from 625°F to 1000°F. Six recirculation pumps supply the required recirculation flow; four of these are capable of supplying sufficient flow for full load operation for seventy-two hours. Two forced-draft fans furnish the primary air and two induced-draft fans are controlled to maintain furnace pressure at a desired pre-set value. The system utilizes regenerative feedwater heating by means of both closed and open heaters; and the feedwater flow is handled by two condensate pumps and a combined booster-and main boiler feedpumps.

The turbine is a tandem compound, single reheat unit, comprising a high pressure, intermediate pressure and two double flow low pressure elements, running at a rated speed of 3600 rpm. It is designed for throttle steam conditions of 2400 psig and 1000°F, and reheat steam at 1000°F, and exhausting at 2 in. Hg absolute, with provisions for full extraction for six stages of feedwater heating. The turbine is capable of delivering up to 600 MW power.

The generator, which is directly coupled with the turbine, is a 685,600 kVa, 3 phase, 60 Hz, 22 kV, hydrogen cooled unit with 0.90 power factor.
Figure II.1 is the plant schematic showing the principal components of the system and the lines of flows of the steam and feedwater. Figure II.2 is the air-gas path schematic showing the principal components and the lines of flow in the air-gas path.

A description of the principal components of the system along with their associated state variables are summarized in Table II.1.
FIGURE II.1: PLANT SCHEMATIC
FIGURE II.2: AIR-GAS PATH SCHEMATIC
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<th>STATE VARIABLES</th>
<th>DESCRIPTION</th>
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<tr>
<td>Drum</td>
<td>Steam Density (\rho_{dr}) Water Volume (V_{dr})</td>
<td>The drum, downcomers and waterwalls constitute the recirculating loop. Saturated liquid leaves the drum to the downcomers while low quality saturated steam enters the drum from the waterwalls. Scrubbers remove liquid from the saturated steam as the steam flows out of the drum to the primary superheater. Drum water level is maintained by controlling feedwater flow. Drum steam and water are assumed to be in saturated equilibrium.</td>
</tr>
<tr>
<td>Primary Superheater</td>
<td>Density (\rho_{pso}) Enthalpy (h_{pso})</td>
<td>The first stage of superheating is accomplished by convective and radiative heat transfer with the flue gases. The primary superheater located at the top of the furnace, is the first heat exchanger to interact with the flue gases leaving the furnace. The superheated steam flows to the desuperheater spray section.</td>
</tr>
<tr>
<td>Desuperheater Spray Sect</td>
<td>Density (\rho_{sso}) Enthalpy (h_{sso})</td>
<td>The function of the desuperheater spray section is to maintain main steam temperature (throttle temperature) at its set point value. Subcooled liquid, feedwater, from the discharge of the boiler feedpumps, is sprayed in the main stream of steam. The desuperheater spray flow is controlled Complete adiabatic mixing is assumed.</td>
</tr>
<tr>
<td>Secondary Superheater</td>
<td>Density (\rho_{sso}) Enthalpy (h_{sso})</td>
<td>The final stage of superheating is accomplished by convective heat transfer with the flue gases. The secondary superheater is the second heat exchanger to interact with the flue gases leaving the furnace. Outlet temperature and pressure (main steam temperature and pressure) are maintained by desuperheater spray flow control and combustion control, respectively.</td>
</tr>
</tbody>
</table>

**TABLE II.1: MAJOR PLANT COMPONENTS**
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<th>COMPONENT</th>
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<th>DESCRIPTION</th>
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<td>COMPONENT</td>
<td>STATE VARIABLES</td>
<td>DESCRIPTION</td>
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<td>-------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------</td>
</tr>
<tr>
<td>Intermediate Pressure</td>
<td></td>
<td>The flow of steam to the intermediate pressure turbine is determined by the intercept valve area, computed from the turbine controls, and the discharge steam conditions of the re heater. The steam is assumed to be expanded adiabatically through the turbine. Steam is extracted at two locations from the intermediate pressure turbine for use in high pressure feedwater heating and supplying steam to the feedpump turbine.</td>
</tr>
<tr>
<td>Crossover Pipe</td>
<td>Density ( \rho_{\text{cro}} )</td>
<td>The crossover pipe connects the intermediate and low pressure turbines. The pipe is relatively short and its diameter large therefore no pressure drop is assumed.</td>
</tr>
<tr>
<td>Low Pressure Turbine</td>
<td></td>
<td>The flow of steam to the low pressure turbine is a function of the crossover pipe discharge steam conditions. Steam is extracted at three locations from the low pressure turbine for use in low pressured feedwater heating and supplying steam to the deaerator.</td>
</tr>
<tr>
<td>Turbines</td>
<td>Speed ( N_{\text{tr}} )</td>
<td>The state variable, turbine speed, is common to the high, intermediate, and low pressure turbines. The rated speed is 3600 rpm and is maintained by governor action of the control valves determined in the turbine control system.</td>
</tr>
<tr>
<td>Condenser</td>
<td>Power Angle ( \delta )</td>
<td>Steam from the low pressure turbine exhausts to the condenser. The variation in condenser conditions is small therefore the approximation is made of relating the condenser pressure to the inlet pressure to the low pressure turbine. Constant steam quality of the low pressure turbine discharge is assumed.</td>
</tr>
<tr>
<td>Condensate Pumps and</td>
<td>Speed ( N_{\text{cp}} )</td>
<td>Two centrifugal condensate pumps provide the flow of condensate from the condenser to the low pressure feedwater heater. The output torque of the driving induction motor is governed by its torque-slip characteristics. Combined with the pump head-flow-speed characteristics, the torque is used in the pump momentum equation to describe the dynamics of the condensate pumps.</td>
</tr>
<tr>
<td>Motors</td>
<td></td>
<td></td>
</tr>
<tr>
<td>COMPONENT</td>
<td>STATE VARIABLES</td>
<td>DESCRIPTION</td>
</tr>
<tr>
<td>--------------------------</td>
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<td>----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------</td>
</tr>
<tr>
<td>Low Pressure Feedwater</td>
<td>Enthalpy $h_{tho}$</td>
<td>Extraction steam from the low pressure turbine is used to heat the condensate and exhausts to the condenser at saturation conditions. The low pressure feedwater heaters are lumped into an equivalent low pressure feedwater heater. The condensate flow to the deaerator is regulated by the deaerator valve which is controlled to maintain deaerator water level at its set point value. Water flow for the air heating system is extracted from the condensate flow before entering the deaerator. The deaerator has three functions: deaerating, feedwater heating, and fluid storage. Condensate flow from the low pressure feedwater heater, extraction steam from the low pressure turbine, exhaust steam from the high pressure feedwater heater, and auxiliary steam returns mix in the deaerator with a net heating effect on the feedwater. Saturated liquid leaves the deaerator to the feedpumps. Deaerator water level is maintained by controlling the condensate flow. Deaerator steam and water are assumed to be in saturated equilibrium. Deaeration equipment deaerates the feedwater flow before it enters the storage tank. The boiler feedpump consists of the booster feedpump and main boiler feedpump. The feedpump turbine drives the centrifugal main boiler feedpump directly and drives the booster feedpump through gear reductions. The turbine provides the input torque used in the pump momentum equation to describe the dynamics of the pump, combined with the pump characteristics. The feedpump turbine steam flow extracted from the high pressure turbine discharge flow is controlled to maintain feedwater valve differential pressure to its set point value. Desuperheater and reheat spray flows are extracted from the pump discharge feedwater flow.</td>
</tr>
<tr>
<td>Deaerator</td>
<td>Steam Density $\rho_{des}$ Water Volume $V_{dew}$</td>
<td></td>
</tr>
<tr>
<td>Boiler Feedpump and Turbine</td>
<td>Speed $N_{fp}$</td>
<td></td>
</tr>
<tr>
<td>COMPONENT</td>
<td>STATE VARIABLES</td>
<td>DESCRIPTION</td>
</tr>
<tr>
<td>---------------------------</td>
<td>-----------------</td>
<td>-----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------</td>
</tr>
<tr>
<td>High Pressure Feedwater</td>
<td>Enthalpy</td>
<td>Extraction steam from the high and intermediate pressure turbines is used to heat the feedwater and exhausts to the deaerator at saturation conditions. The high pressure feedwater heaters are lumped into an equivalent high pressure feedwater heater. The feedwater flow is regulated by the feedwater valve.</td>
</tr>
<tr>
<td>Heaters</td>
<td>$h_{hho}$</td>
<td></td>
</tr>
<tr>
<td>Economizer</td>
<td>Enthalpy</td>
<td>The final stage of feedwater preheating is accomplished by convective heat transfer with the flue gases. The economizer, a finned tube heat exchanger, is located in the back pass and is the fourth heat exchanger to interact with the flue gases leaving the flame. The discharge pressure is drum pressure and the discharge flow is assumed to mix adiabatically with drum water in the downcomers.</td>
</tr>
<tr>
<td></td>
<td>$h_{eco}$</td>
<td></td>
</tr>
<tr>
<td>Downcomers</td>
<td></td>
<td>Six downcomers transport recirculating water from the economizer and drum to the recirculating pumps. Negligible heat loss and kinetic energy change in the downcomers and incompressible flow are assumed.</td>
</tr>
<tr>
<td>Recirculating Pumps and</td>
<td>Speed</td>
<td>Six recirculating centrifugal pumps provide the recirculating flow. Four pumps are sufficient to handle full load flows for thirty hours. The output torque of the driving induction motor is governed by its torque-slip characteristics. Combined with the pump head-flow-speed characteristics, the torque is used in the pump momentum equation to describe the dynamics of the recirculating pumps.</td>
</tr>
<tr>
<td>Motors</td>
<td>$N_{rp}$</td>
<td></td>
</tr>
<tr>
<td>Waterwall</td>
<td>Metal</td>
<td>The waterwalls form the walls of the furnace and completes the recirculating loop. The main source of steam in the boiler is generated in the waterwalls predominantly by radiation heat transfer with the hot combustion gases in the</td>
</tr>
<tr>
<td></td>
<td>Temperature</td>
<td></td>
</tr>
<tr>
<td></td>
<td>$T_{wwm}$</td>
<td></td>
</tr>
<tr>
<td>COMPONENT</td>
<td>STATE VARIABLES</td>
<td>DESCRIPTION</td>
</tr>
<tr>
<td>--------------------</td>
<td>------------------</td>
<td>---------------------------------------------------------------------------------------------------------------------------------------------</td>
</tr>
<tr>
<td>Glycol Air Heater</td>
<td></td>
<td>furnace. The saturated liquid mixture discharged from the waterwalls is assumed to be at drum pressure.</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Condensate flow extracted before the deaerator is used to provide the first stage of preheating air in the glycol air heater. A constant air temperature change is assumed. The air is drawn from the atmosphere by the forced draft fans.</td>
</tr>
<tr>
<td>Forced Draft Fans</td>
<td>Speed N&lt;sub&gt;fd&lt;/sub&gt;</td>
<td>Two centrifugal forced draft fans are used to provide sufficient air flow to the furnace for proper combustion. The output torque of the driving induction motor is governed by its torque-slip characteristics. Combined with the fan head-flow-speed characteristics, the torque is used in the fan momentum equation to describe the dynamics of the forced draft fans. The fans are equipped with inlet vanes used to control the amount of air entering the furnace. The vane positions are determined in the combustion control.</td>
</tr>
<tr>
<td>Motors</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Air Preheater</td>
<td></td>
<td>The final stage of air preheating is accomplished by convective heat transfer with the flue gases in the rotating regenerative air preheater. The air preheater is the last heat exchanger in the gas path to interact with the flue gases leaving the furnace and is located below the economizer. A constant air temperature change is assumed.</td>
</tr>
<tr>
<td>Furnace</td>
<td></td>
<td>Combustion of the fuel with preheated air from the air preheater and recirculated flue gases occurs in the furnace. Fuel, air, and gas recirculation flows are controlled, the former two used to maintain main steam pressure and the</td>
</tr>
<tr>
<td>COMPONENT</td>
<td>STATE VARIABLES</td>
<td>DESCRIPTION</td>
</tr>
<tr>
<td>----------------------------</td>
<td>-----------------</td>
<td>-------------------------------------------------------------------------------------------------------------------------------------------</td>
</tr>
<tr>
<td>Gas Path</td>
<td></td>
<td>latter used to maintain reheat steam temperature. The heat transfer process in the furnace is predominantly by radiation to the waterwalls and primary superheater. The furnace is furnished with five levels of oil guns where each level consists of four guns. The guns have tilting capabilities and are controlled to maintain reheat steam temperature. The radiation heat transfer is modified by the number of operating oil guns and the burner tilt angle. Furnace pressure is maintained at its set point valve by controlling the induced draft fan vane areas. The flue gases leaving the furnace passes in succession through the primary superheater, secondary superheater, reheater, economizer, and air preheater, transferring heat to the heat exchangers at each point. At the economizer exit flue gas is extracted and introduced into the furnace to aid in maintaining reheat steam temperature. The gas recirculation flow is controlled.</td>
</tr>
<tr>
<td>Induced Draft Fans and Motors</td>
<td>Speed $N_{id}$</td>
<td>Two centrifugal induced draft fans are used to draft the flue gases to the atmosphere. The output torque of the driving induction motor is governed by its torque-slip characteristics. Combined with the fan head-flow-speed characteristics, the torque is used in the fan momentum equation to describe the dynamics of the induced draft fans. The fans are equipped with inlet vanes used to maintain the furnace pressure at its set point value. The vane areas are determined in the boiler controls.</td>
</tr>
<tr>
<td>Stack</td>
<td></td>
<td>The stack serves the dual purposes of providing a draft and discharging the exhaust flue gases into the atmosphere.</td>
</tr>
</tbody>
</table>
CHAPTER III
MODEL DEVELOPMENT

The models presented here were developed based upon a lumped-parameter approach with describing equations governed by physical laws and constitutive relations. All major plant components and processes were modeled to satisfy the requirements for an accurate boiler-turbine model.

In spite of the large number and variety of components involved in a drum boiler-turbine unit, similarities exist in the assumptions and governing equations describing their dynamic behavior.

III.1 Assumptions

The large excursions of system variables during emergency conditions require that assumptions be carefully made in the development of a model to be used for study and evaluation of emergency state control strategies for the power system. An accurate model is desired but the governing equations describing processes are complex in their general form and it is necessary to employ a number of assumptions to reduce the complexity and ease computational requirements.

The basic assumptions are:

(1) Any property of a component at a given cross-section is suitably represented by a single effective value. For example, the temperature of steam at the inlet to the secondary superheater varies across the section but this is represented by a single effective inlet temperature. This allows work in a one-dimensional framework instead of an otherwise more complex two- or three-dimensional framework.

-25-
(ii) Transport lag time for a component is negligible compared to its characteristic time constant. This assumption permits use of the lumped parameter approach.

(iii) Water is incompressible. This assumption permits the neglect of density variation with pressure. It does not, however, allow for neglecting the variation of water density with temperature, which can be quite significant.

(iv) The variation of pv (pressure * specific volume) is negligible compared to the variation of h(enthalpy). This statement is not strictly correct as evidenced in a test calculation but it allows the complex energy equation to be reduced to a much simpler and convenient expression (At about 2400 psia and 1000°F, the change in pv accounts for about 12% of the change in internal energy, u, of steam).

(v) Inertia terms are negligible compared to friction and pressure terms in flow equations.

(vi) Friction factors defined for turbulent flows through pipes or over tube banks are constants. These are actually functions of the associated flows but for the range of flows of interest the variations were small and the average values were used.

(vii) Water and steam are in saturated equilibrium in the drum, deaerator and condenser. This assumption permits the use of only one intensive thermodynamic property to completely describe the intensive thermodynamic state of the component.

(viii) Where two or more prime mover units are in operation and are performing identical duties, they are assumed to be dynamically identical. Any units that are either broken or put out of service are treated as absent. This is most applicable to pumps and fans where two or more may be required to perform a particular duty. A group of pumps or fans are therefore described in terms of the dynamics of one of them and the number of operating pumps or fans in the group.

(ix) In the heat exchangers, the tube metal masses and water or steam masses are lumped together in an effective mass.

(x) In applying the lumped parameter approach to modeling a continuous system there is a choice of the location of the property used to represent a given component. Three
seemingly natural choices are available, namely, inlet condition, outlet condition, and the arithmetic means of the inlet and outlet conditions. These choices are commonly referred to as the forward, backward and central difference method of finite representation respectively. The forward difference method produces unstable responses, while the central difference method may lead to erroneous transient responses [22,23,3]; thus leaving the backward difference method as the suitable method and the outlet condition the representative condition.

III.2 Physical Laws

The physical laws describing the dynamics of the system include continuity, energy and momentum equations. These relationships are presented below in their operational forms as applied to a lumped component.

III.2.1 Continuity Equation. The continuity equation is a statement of mass conservation which is expressed in terms of the relationship below:

\[ \text{Rate of Change of Stored Mass} = \text{Rate of Mass Influx} - \text{Rate of Mass Efflux} \]

For a fluid flow through a lumped component, with the assumptions made above this equation reduces to the form:

\[ \frac{d \rho_{\text{out}}}{dt} = W_{\text{in}} - W_{\text{out}} \]

where:

- \( V \) = fill volume of component
- \( \rho_{\text{out}} \) = outlet fluid density
- \( W_{\text{in}} \) = rate of mass inflow
- \( W_{\text{out}} \) = rate of mass outflow
III.2.2 Energy Equation. Energy conservation is expressed in terms of the relationship below.

\[
\text{[Rate of Change of Stored Energy]} = \text{[Rate of Energy Influx]} - \text{[Rate of Energy Efflux]} + \text{[Rate of Heat Input]} - \text{[Rate of Work Output]}
\]

For a lumped component, with the assumptions made above, this equation takes the form:

\[
\frac{dh}{dt} = \frac{W_{\text{in}}}{M} h_{\text{in}} - \frac{W_{\text{out}}}{M} h_{\text{out}} + Q - W
\]

where:

- \( h_{\text{in}} \) = inlet enthalpy
- \( h_{\text{out}} \) = outlet enthalpy
- \( W_{\text{in}} \) = rate of mass inflow
- \( W_{\text{out}} \) = rate of mass outflow
- \( W \) = work done
- \( Q \) = heat input
- \( M \) = effective mass obtained by lumping tube metal mass with fluid mass

The effective mass is given by:

\[
M = \rho_e V + \frac{M S T}{\rho_m m e}
\]

where:

- \( \rho_e \) = mean fluid density
- \( h_e \) = mean fluid enthalpy
\( V = \) fill volume

\( M_m = \) metal mass

\( S_m = \) specific heat of metal

\( T_m = \) metal temperature

In the case where the fluid mass is lumped with the metal mass, the energy equation takes the form:

\[ M S_m \frac{dT_m}{dt} = Q_{in} - Q_{out} \]

And

\[ M = M_m + V \rho \frac{h}{(S_m T_m)} \]

where:

\( Q_{in} = \) heat transferred into metal

\( Q_{out} = \) heat transferred out of metal

**III.2.3 Momentum Equation.** Momentum conservation is expressed in terms of the following relationship:

\[
[\text{Rate of Change of Stored Momentum}] = [\text{Rate of Momentum Influx}] - [\text{Rate of Momentum Efflux}] + [\text{Net Force Applied}]
\]

For a fluid flow, neglecting inertia terms, this statement reduces to:

\[ P_{in} - P_{out} = f \frac{W^2}{\rho} + L \rho g/g_c \]

where:

\( P_{in} = \) inlet pressure
\[ P_{\text{out}} = \text{outlet pressure} \]
\[ f = \text{friction factor} \]
\[ W = \text{mass flow rate} \]
\[ L = \text{elevation of outlet above inlet} \]
\[ \rho = \text{density} \]
\[ g = \text{acceleration due to gravity} \]
\[ g_c = \text{unit conversion factor} \]

For pumps, fans, motors and turbines conservation of moment of momentum is considered instead and takes the form:

\[ J \frac{dN}{dt} = T_{\text{in}} - T_{\text{out}}. \]

where:

\[ J = \text{effective mass moment of inertia} \]
\[ N = \text{angular velocity} \]
\[ T_{\text{in}} = \text{input torque} \]
\[ T_{\text{out}} = \text{output torque} \]

III.3 Constitutive Relations

The constitutive relations that are explicitly employed in this model development are of three primary categories, viz;

(i) Water/steam properties relations
(ii) Pump and fan performance characteristics
(iii) Induction motor characteristics

These represent physical constraints implicit in the relationship between system properties and parameters. They are described below:
111.3.1 Water Steam Properties Relations. Given a substance in "Thermodynamic stable equilibrium state", only a limited number of the properties are independent. With the independent properties known, the thermodynamic state of the substance is completely defined and the other properties can be obtained as functions of the known independent properties [20]. For a few substances (for example, so called perfect gases) the functional relationships between the properties are expressible in simple closed form algebraic equations, but for most others (for example, water/steam), the use of tables of properties becomes necessary [13].

Direct use of the tables of properties is not very convenient for dynamic system simulations; thus, it was necessary to develop analytic approximations to the tables. This involved the use of data from the steam Table [13] and a standard nonlinear least square polynomial fit routine. In order to reduce complexity, the fits were designed to be valid only for certain ranges of interest.

The number of independent properties necessary to completely define the thermodynamic state of pure substance is prescribed by Gibbs Phase Rule [20]. For water/steam in single-phase regime, two independent intensive properties completely define the intensive thermodynamic state. Therefore, given properties $P_1$ and $P_2$, any other property, $P$, can be obtained from a functional relationship of the form:

$$P = P(P_1, P_2)$$

On the other hand, only one intensive property completely de-
fines the intensive thermodynamic state of water/steam in the two-phase regime. Given a property, \( P_1 \), any other property, \( P \), can be obtained from a functional relationship of the form:

\[
P = P(P_1)
\]

Details of the fits used for the system are presented in Appendices A and B.

III.3.2 Pump and Fan Performance Characteristics. The performance characteristics of pumps and fans are usually available in the form of graphical plots of head and efficiency relationships to flow rate for different operating speeds. These plots can be condensed into single plots of normalized head \( \left( \frac{H}{N^2} \right) \) and efficiency \( \left( \frac{n}{N^2} \right) \) against normalized flow \( \left( \frac{Q}{N} \right) \) as shown in Figure III.1 [16,17].

The functional relationships governing these characteristics are expressible in a generally accepted quadratic form as given below [16,17]:

\[
\frac{\Delta H}{N^2} = K_1 + K_2 \frac{Q}{N} + K_3 \left( \frac{Q}{N} \right)^2
\]

\[
\frac{n}{N^2} = K_4 + K_5 \frac{Q}{N} + K_6 \left( \frac{Q}{N} \right)^2
\]

The head-flow relationship for a fan, though essentially of the structure given above, is slightly modified to account for the variable vane position which is used for flow control. It was determined that the form given below suitably models the effect:
FIGURE III.1: PUMP AND FAN PERFORMANCE CHARACTERISTICS
\[
\frac{\Delta H}{A N^2} = K_1 + K_2 \frac{Q}{AN} + K_3 \left(\frac{Q}{AN}\right)^2
\]

where:

- \( A \) = normalized vane position
- \( H \) = pump or fan head
- \( N \) = pump or fan speed
- \( Q \) = volume flow rate
- \( K_1-6 \) = constants determined from least square polynomial fit of performance characteristics

Further, the characteristics may be expressed in terms of pressure and mass flow rate by making the following substitutions:

\[
\Delta H = \frac{(P_{\text{out}} - P_{\text{in}}) g_c}{\rho g}
\]

\[Q = \frac{W}{\rho}\]

where:

- \( P_{\text{in}} \) = inlet pressure
- \( P_{\text{out}} \) = exhaust pressure
- \( \rho \) = density
- \( g \) = acceleration due to gravity
- \( g_c \) = unit conversion factor

III.3.3 Induction Motor Characteristics. Induction motor torque slip characteristics is of the form shown in Fig. III.2 for different operating voltages. This relationship is suitably modeled by the expression below [21]:

-34-
FIGURE III.2: INDUCTION MOTOR TORQUE-SLIP CHARACTERISTICS
\[ T = \frac{k \cdot \frac{Y^2}{S/S_{\text{max}} + S_{\text{max}}/s}}{N_s - N} \]
\[ s = \frac{N_S - N}{N_s} \]

where:

- \( N_s \): synchronous speed = \( \frac{N_{\text{elec}}}{k_n} \)
- \( N_{\text{elec}} \): line frequency
- \( N \): actual speed of motor
- \( k_n \): number of poles/2
- \( V \): supply voltage
- \( k \): constant
- \( s \): slip
- \( S_{\text{max}} \): slip for maximum torque
- \( T \): torque

III.4 Manipulation of Equations

In their original forms, some of the governing equations are not convenient for computer simulation with regard to flow of information and causality. It was therefore necessary in certain cases to perform algebraic rearrangement to reduce the affected equations into more convenient forms. Important in this respect is the computation of flows delivered by dynamic elements such as pumps or fans. It involves the dynamics of both the pumps or fans and the associated adjoining components. A simultaneous solution of the flow (momentum) equations of the
adjoining components and the performance characteristics equations of the associated pumps or fans is used to determine the desired flow.

Another illustration of the need for manipulating governing equations is manifested in the description of the drum and deaerator dynamics. In either of these components, a simultaneous solution of the continuity and energy equations was necessary.

III.5 Digital and Standard Models

Based upon the assumptions and method described above a forty-seventh order 'Digital Computer Model' has been developed to describe the dynamic behavior of the drum boiler-turbine power plant. Twenty three state variables describe the physical processes involved while twenty four state variables describe the control system discussed below. This model was developed to exhibit the characteristics of an accurate boiler-turbine model suitable for studies and evaluation of emergency state control strategies.

A reduced order model for the boiler-turbine unit has also been developed. In this model twelve state variables describe the physical processes while fifteen state variables describe the control system.

The simplification of the reduced order model (called Standard Model) was achieved by taking the following actions:

- Lumping as many components as is realistically possible. For example, the primary superheater, secondary superheater and desuperheat spray section are lumped into a single equivalent superheater. The reheater and reheat spray section are lumped into an equivalent reheater. The intermediate pressure turbine, cross over
pipe and low pressure turbine are lumped into an equivalent low pressure turbine.

-Making use of further assumptions verified through simulations of the digital model to reduce the nonlinearities associated with the computation of heat transfers and fluid flows.

-Replacing the dynamics of pumps and fans with auxiliary circuits which model the limitations imposed on the system by their dependence on driving voltage and frequency.

-Eliminating control subloops that perform their duties very well. Thus, assumes perfect control over associated variables. For example, perfect feedwater flow, condensate flow, feedpump turbine, and furnace pressure controls are assumed.

The governing equations describing the Digital and Standard Models are presented in Appendices A and B respectively.
CHAPTER IV

CONTROL SYSTEM

The control system of the prototype plant is typical for a modern power plant. It incorporates feedforward techniques to achieve rapid response and minimize cross-interactions. Feedback controllers serve as final trims on the process sub-loops to correct for minor nonlinearities and static offsets.

The control system can be operated in three respective modes, viz:

(i) Coordinated Control Mode

(ii) Remote Coordinated Control Mode

(iii) Manual Mode

In the Coordinated Control and Remote Coordinated Control modes, the boiler-turbine controls are coordinated and the feedforward signals which drive them originate from a common source, the Load Demand Computer (LDC). The Remote Coordinated Control Mode differs from the Coordinate Control Mode only in that a change in the Load Demand Computer output is effected by pulses from a Satellite Central Control Station (REMVEC) whereas this is generated locally in the latter mode. In the Manual mode, there is no direct coordination between the boiler and turbine controls.

The control system utilizes conventional process control actions, such as, Proportional (P), Proportional plus integral (PI), and Proportional plus Integral plus Derivative (PID). In addition to
these control actions, the input-output control system relationships and dynamic actuators are also modeled. Here simplifications are made by representing them by linear transducers and simulators respectively.

IV.1 Boiler Control

The boiler control system includes ten major control loops. These are:

(i) Throttle Pressure Control
(ii) Air Flow Control
(iii) Fuel Flow Control
(iv) Furnace Pressure Control
(v) Feedwater Flow Control
(vi) Feedpump Turbine Control
(vii) Condensate Flow Control
(viii) Superheat Temperature Control
(ix) Reheat Temperature Control
(x) Gas Recirculation Control

The block diagrams of these control loops are shown in Figures IV.1 through IV.6 and descriptions of the individual control loops are discussed in Appendix C, along with associated parameter values.

IV.2 Turbine Control

An Electro-Hydraulic Control (EHC) system is employed for the turbine. The turbine is designed with the capability of operating in either full-arc or partial-arc modes. The full arc mode, is usually used
for start-ups while the partial arc mode is employed during normal plant operations. In the full-arc mode, the so-called control valves remain wide open while the stop valves are varied. The reverse happens in the partial arc mode.

The turbine control system is segmented into three units namely:

(i) Load Control Unit
(ii) Speed Control Unit
(iii) Valve Control Unit

A block diagram of the Turbine Control System is shown in Figure IV.7 and a description of the system is given in Appendix C, along with associated parameter values.
FIGURE IV.1: COMBUSTION CONTROL
Figure IV.2: Furnace Pressure and Feedpump Turbine Controls
Figure IV.3: Feedwater Flow Control
FIGURE IV.4: CONDENSATE FLOW CONTROL
FIGURE IV.5: SUPERHEAT TEMPERATURE CONTROL
FIGURE IV.6: REHEAT TEMPERATURE CONTROL
LOAD DEMAND (LDC)

POWER

OUTPUT

LOAD CONTROL UNIT

LOAD REFERENCE MOTOR

LOAD REFERENCE SIGNAL

LOAD CONTROL UNIT

LOAD REFERENCE SIGNAL

PARTIAL ARC OPEN GAS

TRIP ANTICIPATOR

OVERSPEED TRIP

SPEED SET

SPEED CONTROL UNIT

VALVE CONTROL UNIT

FIGURE IV.7: TURBINE CONTROL
CHAPTER V
SIMULATIONS AND RESULTS

The governing equations describing the digital and standard models were solved using a standard Runge-Kutta integration routine (DYSYS) available in the Joint Mechanical and Civil Engineering Computer Facility. A time step of 0.1 seconds was used. The computer program listings are attached herewith.

The models have been developed with the objective of accurately simulating a wide variety of power plant emergency conditions. Several emergency type tests have been simulated and the flexibility of the models permit a wide variety of other possible tests. These tests include but are not limited to rapid changes in line voltage and frequency, failure of the major plant components, and rapid change in power demand, which frequently characterize emergency conditions.

Seven test cases are presented below. In each case, the system was run at steady state for ten seconds before being subjected to the disturbance. It is pertinent to note that even though these tests may be extreme, they are designed to represent emergency states.

Tests 1 through 4 were designed to study the system response to rapid changes in load demand. Tests 5 and 6 simulate the system response to changes in the electrical system variables, voltage and frequency; and in particular, the effect of these changes on major plant auxiliaries driven by electrical machines, for example, pumps and fans. Also of interest and importance in studying emergency state
control strategies is the limitation imposed on the maximum plant capability (Power Output) by these conditions. Test 7 demonstrates a case of a major equipment failure.

Of importance in the course of developing a reliable model for a system is the verification of the model with respect to the actual system. Complete verification would involve comparison of both transient and steady state data for various tests. Unfortunately, at the time of writing, actual plant transient data was not available for comparable test cases. Thus, the model was verified by comparing steady state simulation results with manufacturer's steady state performance data at different load levels. These comparisons are presented in the form of tables.

In each of the tests listed below, the main steam pressure (throttle pressure) set point value was fixed (at 2415 psia). So, in effect, the system was run in the 'Boiler Following Control Mode' and not the 'Coordinated Control Mode' even though the latter mode was adequately modeled and tested. This was done to enable simulation results to be compared with available plant data. Steady state operating load levels of 100%, 77.5% and 50% were selected.

V.1 Test 1: Load Ramp from 100% to 77.5% at 15% Per Minute

The system was run at 100% load level (600MW) for 10 seconds using initial condition values corresponding to 100% load. A load command reduction at 15% per minute (90MW/min) was then applied to the Load Demand Computer (LDC) Signal for 90 seconds to ramp the load com-
mand to 77.5% (465 MW). This test was performed for the digital model and standard model respectively.

The time response of some representative variables for the plant are presented in Figures V.1 and V.2 for the Digital Model and Standard Model respectively. Notice that the turbine speed remains virtually constant despite the variation in power output. This is due to the electro-magnetic spring action of the generator, which is explicitly modeled. The main steam flow is suitably controlled so that the power output closely follows the load demand with only a slight undershoot (1%). The throttle pressure increased initially due to the closure of the governor control valves in response to the decrease in load demand. Responding to the combustion controls, the throttle pressure attempts to reach steady state with an offset error during the 90 seconds ramp but finally returns to the set point value (2415 psia) after the ramp ended.

The superheater and reheater enthalpies give reflections of the responses of the main steam and reheat temperatures respectively. Similarly, the drum and deaerator water volumes reflect drum and deaerator water levels respectively. The initial increase in the superheat temperature (enthalpy) is due to adiabatic compression of the steam caused by the closure of the governor control valves. The initial decrease in the reheat temperature (enthalpy) is due to a reverse action caused by the decrease in steam flow passing through the high pressure turbine to the reheater. These offsets in temperature generate suitable control actions which return the temperatures to their
respective set point values. The initial increases in drum and deaerator water levels (volumes) are due to decreases in main steam and feedwater flows leaving the drum and deaerator respectively. Here again, adequate control actions (feedwater flow and condensate flow controls) return the levels to their respective set point valves.

The simulation results show that the plant responds well to this change in load command with good control maintained over the variables. Also, the standard model response compares very well with the digital model response. Its response is however slightly faster due to the fact that most of the nonlinearities and time lags associated with heat transfer and flow have been ignored in the formulation of the standard model.

Steady state data obtained from the simulation are compared with plant data in Tables V.1 and V.2 for 100% and 77.5% load levels respectively. They show agreement within 5%.

V.2 Test 2: Load Ramp from 77.5% to 50% at 15% Per Minute

The system was run at 77.5% load level (465MW) for 10 seconds using initial condition values corresponding to 77.5% load. A load command reduction at 15% per minute (90MW/min) was then applied to the Load Demand Computer (LDC) signal for 110 seconds to ramp the load command to 50% (300MW). This test was performed for the Digital Model and Standard Model respectively.

The time response of representative variables for the plant are presented in Figures V.3 and V.4 for the Digital Model and Standard Model respectively. Qualitatively, these responses are similar.
to those obtained in Test 1 and the comments made under Test 1 are also appropriate. However, Test 2 differs slightly in that there are changes in main steam and reheat temperature set point values as reflected by the changes in the steady state enthalpy values. It is clear from the simulation results that the system is well behaved under this load command change and the variables are well under control.

The Standard Model response compares very well with the Digital Model both qualitatively and quantitatively. Steady state simulation data are compared with actual plant data at 50% load level in Table V.3. They show agreement within 5%, except for a few isolated variables in the condensate side where the agreement lies within 10%.

V.3 Test 3: Load Ramp from 50% to 77.5% at 15% Per Minute

In this test the system was run at 50% load level (300MW) for 10 seconds using initial condition values corresponding to 50% load. A load command increase at 15% per minute (90MW/min) was then applied to the Load Demand Computer (LDC) signal for 110 seconds to ramp the load to 77.5% (465MW). This test was performed for both the Digital Model and Standard Model respectively.

The time response of representative variables for the plant are presented in Figures V.5 and V.6 for the Digital and Standard Models respectively. Again, the turbine speed remained virtually unperturbed, the main steam flow was suitably controlled such that the power output closely follows the demand with little overshoot (1%). For reasons similar to those given under load reduction tests, the behavior of the throttle pressure, superheat and reheat temperatures
(enthalpies), and drum and deaerator water levels (volumes) follow trends opposite to those obtained for load reductions.

This is a fairly drastic test, considering the 15% per minute rate load command increase. In most power plants, this rate is limited to 10% per minute and usually less than 5% per minute during normal operations [9]. Nevertheless, the 15% per minute rate would be desirable during emergencies, as long as it lies within the safety regime of the particular plant under consideration.

The simulation data shows that under this change in load command the system is well behaved and the variables are well under control. The Standard Model compares well with the Digital Model except that the dynamics appear faster and less damped for reasons given earlier.

V.4 Test 4: Load Ramp from 77.5% to 100% at 15% Per Minute

The system was run at 77.5% load level for 10 seconds using initial condition values corresponding to 77.5% load. A load command increase at 15% per minute was then applied to the Load Demand Computer signal for 90 seconds to ramp the load to 100% load level. This test was performed for the Digital and Standard Models respectively.

Presented in Figures V.7 and V.8 are the time responses of representative plant variables for the Digital and Standard Models respectively. The data shows that under this test the system is only fairly well behaved. This is both due to the drastic nature of the test and the fact that the system was already close to its maximum
capability. Some of the control signals saturate in the course of the system transients.

As a consequence, the excursions of plant variables from set point values are relatively larger as evidence from comparing this test results with the earlier tests. Also, the power output does not follow the demand exactly and there is no overshoot.

A close look at the four tests presented above reveals the fact that the system has more difficulty increasing load than reducing load.

The Standard Model again compares fairly well with the Digital Model except for the relatively faster dynamics.

V.5 Test 5: 30% Step Drop in Voltage at 77.5% Load Level

The system was run at 77.5% load level for 10 seconds using initial condition values corresponding to 77.5% load. A 30% step reduction in line voltage was then applied to reduce the voltage from 4160V to 2912V (Most power plants would trip at this level of reduced voltage).

The test was designed to study the influence of voltage changes on the system and in particular the major plant auxiliaries. The time response of some representative plant variables are presented in Figure V.9. The data show that the components most affected by this reduction in voltage are the auxiliaries driven by electrical machines. This effect is manifested as decrease in the speeds of the pumps and fans. These reductions in speeds led to reductions of the associated flows,
for example, air flow, condensate flow and recirculating water flow.

Except for the small changes in the speeds of the Induced Draft and Forced Draft fans and recirculating and condensate pumps and minor dynamic excursions in the air and condensate sides, the boiler-turbine system continued to operate at steady state conditions. The reason for this is two-fold. First any error due to this disturbance is compensated for by adequate control action as reflected by the increase in the air flow and condensate flow control signals. Secondly, at 77.5% load level the system has sufficient excess capacity to contain a reduction in line voltage of the order of magnitude applied here. However, at load level close to 100% the system does not have sufficient excess capacity to accommodate this disturbance and unless power demand is decreased, the system may approach trip conditions.

A similar test run at 77.5% load level but with 40% step drop in voltage showed that the condensate pumps could not meet the condensate flow demand with this amount of voltage reduction. This led to an undesirable drop in the deaerator water level, which would ultimately cause a station trip. The steam side was essentially unaffected as in the 30% step reduction in voltage test and this test points out the importance of modeling the feedwater side dynamics. If the feedwater side was not modeled the 40% reduction in voltage test would indicate no appreciable changes in system output, which is an incorrect conclusion since the feedwater side is actually approaching a limiting condition.
It is important to recognize the fact that even though it is the insufficiency of the condensate pumps capability that would trigger a trip in this plant, it could as well be any of the other auxiliaries at another plant. It all depends on the characteristics of the auxiliaries and the amount of excess capacity each is designed for [10]. Actually the plant would trip automatically before this level of reduced voltage is reached.

V.6 Test 6: Frequency Drop from 60 Hz to 56 Hz in Five Seconds

Frequently, major power system disturbances are manifested as a reduction in the system frequency. These disturbances are variously caused by loss of big interconnected unit generation, sudden increase in load demand, loss of tie-lines, etc. In situations like these, the fate of the overall power system depends on the dynamics of the operating units. These test is directed towards studying the effect of frequency reduction on an operating unit and the associated unit's response.

The system was run at 77.5% load level (465MW) for 10 seconds using initial condition values corresponding to 77.5% load. A reduction of line frequency at a rate of 0.8 Hz per second was applied for 5 seconds to reduce the frequency from 60 Hz to 56 Hz.

The time response of representative variables are presented in Figure V.10. Due to the drop in frequency the governor droop action is activated. This forces the governor valves wide open in an attempt to generate more power even though the Load Demand Computer (LDC) sig-
nal yet remains unchanged. This action is intended to help alleviate the undesirable frequency deviation. The opening of the governor valve instantly increases the main steam flow and therefore the power output but simultaneously the throttle pressure decreases. Consequently, the main steam flow reaches a maximum value and begins to fall even though the governor valve remains wide open. The fall in throttle pressure causes an increase in the Boiler Master Demand (combustion control) which in turn increases the air and fuel control signals, demanding for more air and fuel. Simultaneously, the drop in frequency has an adverse effect on the pumps and fans driven by electrical machines (induction motors) which limits the maximum flows capable of being supplied by the auxiliaries. In effect, the fans can not supply sufficient air flow, to increase the pressure to set point value despite the fact that the air control signal reaches and remains at maximum value (5V) and drives the fan vanes wide open. The fuel flow is limited by the air flow to avoid waste caused by supplying more fuel than there is air available for complete combustion.

As a result, the throttle pressure set point value can not be attained and maximum power output can also not be produced despite the fact that under the conditions here it would be desirable to produce the maximum possible power. The throttle pressure settles down at 2125 psia and the power output at 537MW (the capability of the plant under normal conditions is 600MW with a throttle pressure of 2415 psia). This test demonstrates the governor valve droop action and also shows that with a 4 Hz drop in frequency the unit could produce only 90%
of its rated power output.

The time response of the pumps and fans speeds show corresponding reductions. These are translated into reductions in the associated flows. The effect of reduction of air flow on throttle pressure and power output has been discussed above. The effect of these reduced speeds on the waterside flow (condensate pump) is reflected in the initial drop in deaerator water level, followed by the sharp increase in the condensate flow control. The reduction in condensate pump speed decreases the flow supplied to the deaerator which leads to the drop in volume. However, the condensate flow control forces the deaerator valve open to compensate for the decrease in condensate pump speed. In addition, due to the limitation on the power output due to air flow, the condensate flow demand is only 90% of full load value which reduces the strain on the condensate pumps. The initial drop in drum water level is due to the sudden increase in steam outflow when the governor droop action forces the valves open. This is, however, compensated by adequate feedwater flow control action. Also, the superheat and reheat temperatures are suitably controlled by sprays and burner tilt actions.

In the test result presented here, the gas recirculation control was inactivated. An identical simulation made with the gas recirculation control on showed that the system took a much longer time to completely settle down. It may be interesting to note that problems such as this have been observed at the actual plant and the gas recirculation control is occasionally put under manual control.

-59-
V.7 Test 7: Loss of A Pair of FD and ID Fans

Apart from global emergency conditions, such as drop in system frequency described above, a given power plant may experience local emergency conditions of various magnitudes. However, these local emergencies may be globally felt depending on the magnitude of the disturbance and the size of the unit. As an example, the loss of one of two pairs of FD and ID fans is considered here. Usually, under a loss of a pair of ID and FD fans, the load command signal is forced to "run-back" to 60% load level (called Unit Run-Back). The response of the system to a Unit Run-Back test is basically similar to those presented under Test 1 and Test 2.

The system was run at 100% load level for 10 seconds using initial condition values corresponding to 100% load. The number of operating pairs of FD and ID fans was then reduced from 2 to 1, leaving the load command signal at 100% level (that is, no forced Unit Run-Back).

The time response of representative variables are presented in Figure V.11. With the loss of the pair of fans, the air flow decreased sharply causing a decrease in throttle pressure, main steam flow and therefore, power output. The combustion controls drive the air flow control signals to its maximum value (5V) but since there is only one pair of fans operating instead of two, the demand for air cannot be met and the air flow control signal remains at the maximum value. The throttle pressure keeps falling and settled at 1700 psia with slight undershoot. The maximum steady power the system can deliver
is 70% (420MW).

Also affected in this test is the drum water level which shows a general trend of an increasing level at the initial stage due to the fact that with reduced air flow, the heat input to the water-wall is reduced and the quality of steam delivered to the drum therefore low. The initial dip in the drum water level results from evaporation of drum water due to the sudden drop in drum pressure caused by the decrease in heat input. With the action of feedwater flow control, this level is ultimately brought back to the set point value. The deaerator water level and condensate flow controls essentially respond to the decrease in load level caused by the failure of the pair of fans (similar to Test 1). The superheat and reheat temperatures are also suitably controlled, but with difficulty as evidenced by the fact that the control signals saturated at their upper or lower limits during most part of the transients. Here again, the gas recirculation control was inactivated for reasons given earlier under Test 6.

The effects of this mishap on the operating fans speeds are most interesting. The FD fan speed decreases because of the extra load the single fan has to carry, as compared to what it was carrying when the two were operating together (with 2 fans, each fan supplied 615 lb/s air; but with one, the one supplies 840 lb/s). The ID fan speed increases very slightly because of the slight closure of the inlet vanes to maintain furnace pressure.

-61-
<table>
<thead>
<tr>
<th>VARIABLE</th>
<th>SYMBOL</th>
<th>UNITS</th>
<th>PLANT DATA</th>
<th>DIGITAL MODEL</th>
<th>STANDARD MODEL</th>
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<td></td>
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<td>RESULT</td>
<td>%ERROR</td>
<td>RESULT</td>
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<td>MW</td>
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<td>0.0</td>
<td>600.0</td>
</tr>
<tr>
<td>Turbine Speed</td>
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<td>Rad/sec</td>
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<td>0.0</td>
<td>377.0</td>
</tr>
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<td>Main Steam Flow</td>
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<tr>
<td>Throttle Pressure</td>
<td>Pssos</td>
<td>psia</td>
<td>2415.0</td>
<td>0.0</td>
<td>2415.0</td>
</tr>
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<td>Main Steam Temp.</td>
<td>Tssos</td>
<td>°F</td>
<td>1000.0</td>
<td>0.0</td>
<td>1000.0</td>
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<td>°F</td>
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<td>0.0</td>
<td>1000.0</td>
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<td>555.1</td>
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<td>psia</td>
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<td>2753.6</td>
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</tr>
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<td>5.3</td>
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<td>Deaerator Water Level</td>
<td>Xdew</td>
<td>in.</td>
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<td>0.0</td>
<td>—</td>
</tr>
<tr>
<td>LP Feedwater Heater Enthalpy</td>
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<td>202.5</td>
<td>1.7</td>
<td>202.5</td>
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<tr>
<td>HP Feedwater Heater Enthalpy</td>
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<td>Btu/lb</td>
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<td>1.2</td>
<td>472.3</td>
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<td>psia</td>
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### TABLE V.2

**COMPARISON OF STEADY STATE VALUES WITH PLANT DATA (77.5% LOAD)**

<table>
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<tr>
<th>VARIABLE</th>
<th>SYMBOL</th>
<th>UNITS</th>
<th>PLANT DATA</th>
<th>DIGITAL MODEL</th>
<th>STANDARD MODEL</th>
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<td></td>
<td>NAME</td>
<td></td>
<td>RESULT</td>
<td>%ERROR</td>
<td>RESULT</td>
</tr>
<tr>
<td>Power Output</td>
<td>$MW_o$</td>
<td>MW</td>
<td>465.0</td>
<td>0.0</td>
<td>465.0</td>
</tr>
<tr>
<td>Turbine Speed</td>
<td>$N_{tr}$</td>
<td>Rad/sec</td>
<td>377.0</td>
<td>0.0</td>
<td>377.0</td>
</tr>
<tr>
<td>Main Steam Flow</td>
<td>$W_{hp}$</td>
<td>lb/sec</td>
<td>831.2</td>
<td>2.1</td>
<td>824.1</td>
</tr>
<tr>
<td>Throttle Pressure</td>
<td>$P_{sso}$</td>
<td>psia</td>
<td>2415.0</td>
<td>0.0</td>
<td>2415.0</td>
</tr>
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<td>°F</td>
<td>1000.0</td>
<td>0.0</td>
<td>1000.0</td>
</tr>
<tr>
<td>Reheat Temp.</td>
<td>$T_{rho}$</td>
<td>°F</td>
<td>1000.0</td>
<td>0.0</td>
<td>1000.0</td>
</tr>
<tr>
<td>Reheat Pressure</td>
<td>$P_{rho}$</td>
<td>psia</td>
<td>419.1</td>
<td>1.8</td>
<td>413.4</td>
</tr>
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<td>Drum Pressure</td>
<td>$P_{dr}$</td>
<td>psia</td>
<td>2641.1</td>
<td>1.4</td>
<td>2607.0</td>
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<td>psia</td>
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<td>3.8</td>
<td>—</td>
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<td>Deaerator Water Level</td>
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<td>in.</td>
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<td>0.0</td>
<td>—</td>
</tr>
<tr>
<td>LP Feedwater Heater Enthalpy</td>
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<td>Btu/lb</td>
<td>190.2</td>
<td>1.8</td>
<td>194.5</td>
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<tr>
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<td>$h_{hho}$</td>
<td>Btu/lb</td>
<td>443.8</td>
<td>2.3</td>
<td>446.7</td>
</tr>
<tr>
<td>Air Flow</td>
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<td>961.3</td>
<td>1.1</td>
<td>943.9</td>
</tr>
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<td>62.6</td>
<td>1.1</td>
<td>61.3</td>
</tr>
<tr>
<td>Furnace Pressure</td>
<td>$P_{fn}$</td>
<td>psia</td>
<td>14.7</td>
<td>0.0</td>
<td>—</td>
</tr>
<tr>
<td>VARIABLE</td>
<td>SYMBOL</td>
<td>UNITS</td>
<td>PLANT DATA</td>
<td>DIGITAL MODEL</td>
<td>STANDARD MODEL</td>
</tr>
<tr>
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<td>--------</td>
<td>-------</td>
<td>------------</td>
<td>---------------</td>
<td>----------------</td>
</tr>
<tr>
<td>Power Output</td>
<td>$P_{MW}$</td>
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<td>300.0</td>
<td>300.0</td>
</tr>
<tr>
<td>Turbine Speed</td>
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<td>rad/sec</td>
<td>377.0</td>
<td>377.0</td>
<td>377.0</td>
</tr>
<tr>
<td>Main Steam Flow</td>
<td>$N_{hp}$</td>
<td>lb/sec</td>
<td>554.1</td>
<td>530.2</td>
<td>535.6</td>
</tr>
<tr>
<td>Throttle Pressure</td>
<td>$P_{ss}$</td>
<td>psia</td>
<td>2415.0</td>
<td>2415.4</td>
<td>2415.0</td>
</tr>
<tr>
<td>Main Steam Temp.</td>
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<td>°F</td>
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<td>979.5</td>
<td>981.5</td>
</tr>
<tr>
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<td>°F</td>
<td>935.0</td>
<td>956.6</td>
<td>953.3</td>
</tr>
<tr>
<td>Reheat Pressure</td>
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<td>psia</td>
<td>273.0</td>
<td>262.7</td>
<td>268.7</td>
</tr>
<tr>
<td>Drum Pressure</td>
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<td>psia</td>
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<td>2493.7</td>
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<td>in.</td>
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<td>-0.07</td>
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<td>psia</td>
<td>31.2</td>
<td>32.4</td>
<td>3.8</td>
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<tr>
<td>Deaerator Water Level</td>
<td>$X_{dew}$</td>
<td>in.</td>
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<td>-0.003</td>
<td>—</td>
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<tr>
<td>LP Feedwater Heater Enthalpy</td>
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<td>169.3</td>
<td>157.1</td>
<td>186.39</td>
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<td>HP Feedwater Heater Enthalpy</td>
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<td>Btu/lb</td>
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<tr>
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<td>652.5</td>
<td>620.2</td>
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<td>lb/sec</td>
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</tr>
<tr>
<td>Furnace Pressure</td>
<td>$P_{fn}$</td>
<td>psia</td>
<td>14.7</td>
<td>14.7</td>
<td>—</td>
</tr>
</tbody>
</table>
FIGURE V.1: TEST 1
FIGURE V.1: TEST 1 Con't.
FIGURE V.1: TEST 1 Con't.
FIGURE V.1: TEST 1 Con't.
FIGURE V.1: TEST 1 Con't.
FIGURE V.1: TEST 1 Con't.
FIGURE V.2: TEST 1
FIGURE V.2: TEST 1 Con't.
FIGURE V.2: TEST 1 Con't.
FIGURE V.2: TEST 1 Cont'.
FIGURE V.3: TEST 2
FIGURE V.3: TEST 2 Con't.
FIGURE V.3: TEST 2 Con't.
FIGURE V.3: TEST 2 Con't.
FIGURE V.3: TEST 2 Cont'.
FIGURE V.3: TEST 2 Con't.
FIGURE V.4: TEST 2
FIGURE V.4: TEST 2 Con't.
FIGURE V.4: TEST 2 Con't.
FIGURE V.5: TEST 3
FIGURE V.5: TEST 3 Con't.
FIGURE V.5: TEST 3 Con't.
FIGURE V.5: TEST 3 Cont.
FIGURE V.5: TEST 3 Con't.
FIGURE V.5: TEST 3 Con’t.
FIGURE V.6: TEST 3 Con't.
FIGURE V.6: TEST 3 CON'T.
FIGURE V.7: TEST 4
FIGURE V.7: TEST 4 Con't.
FIGURE V.7: TEST 4 Con't.
FIGURE V.7: TEST 4 Con't.
FIGURE V.7: TEST 4 Con't.
FIGURE V.7: TEST 4 Con't.
FIGURE V.8: TEST 4
FIGURE V.8: TEST 4 Con't.
FIGURE V.8:  TEST 4  Con't.
FIGURE V.8: TEST 4 Con't.
FIGURE V.9: TEST 5
FIGURE V.9: TEST 5 Con't.
FIGURE V.9: TEST 5 Con't.
FIGURE V.9: TEST 5 Con't.
FIGURE V.9: TEST 5 Con't.
FIGURE V.9: TEST 5 Con't.
FIGURE V.10: TEST 6
FIGURE V.10: TEST 6 Con't.
FIGURE V.10: TEST 6 Con't.
FIGURE V.10: TEST 6 Con't.
FIGURE V.10: TEST 6 Cont'.
FIGURE V.10: TEST 6 Con't.
FIGURE V.11: TEST 7
FIGURE V.11: TEST 7 Con't.
FIGURE V.11: TEST 7 Cont.
FIGURE V.11: TEST 7 Con't.
FIGURE V.11: TEST 7 Con't.
FIGURE V.11: TEST 7 Cont.
CHAPTER VI

CONCLUSION

A drum boiler-turbine power plant model has been presented which:

-is modeled based on physical processes and material geometry and properties and incorporates all major plant components, and so, provides a physically realistic representation of the mechanical power system

-incorporates interactions between mechanical and electrical components by modeling their dependence on voltage and frequency

-is flexible and so could simulate a wide variety of emergency situations.

-is reduced to a simpler lower order standard model and can be directly implemented in hardwired analog electronics.

A state of the arts model has thus been developed which provides a suitable working tool for emergency state control studies for the power system.

With these physically based boiler-turbine power plant models, the following deductions have been made based on computer calculations.

-A reduction in system frequency adversely affects the mechanical system capability. This effect is prominent in the mechanical equipment driven by synchronous machines; for example, pumps and fans driven by induction motors. With a 4 Hz reduction in frequency, the plant capability was reduced by 10%.

-The system is also affected by a reduction in voltage. Here again, equipment driven by electrical machines are directly affected. A 30% reduction in voltage at 77.5% load level did not cause very much of a problem, but a 40% reduction at the same load level leads to a station trip because of excessively low water level in the deaerator caused by the inability of condensate pumps, under the 40%
reduced voltage, to supply sufficient flow to meet the load demand. A 30% voltage reduction at load levels close to 100% would also cause the same problems.

While a drop in frequency affects fans most, a drop in voltage affects the condensate pumps most in the particular system under consideration. The reverse may be true for another system depending on the excess capacity and characteristics of the equipment.

A load decrease is in general much easier to accomplish than a load increase. In the process of a load increase the excursions of the system variables from set values are relatively greater than that for a corresponding load decrease. In some current operating fossil fueled power plants the load reference signal is limited to 10% per minute for generation increases and up to 133% per minute for generation decreases [9].

The limitations imposed on the mechanical power system by characteristic emergency conditions provide for valuable information in the design of an emergency state control system. Thus, these models provide a working tool for the design, testing and evaluation of Slow Speed Dynamics Emergency State Control Strategies.

The load increase characteristics of the system suggest that some effort should be directed toward developing a control system at the plant level that is capable of safely leading the plant through load increases at rates higher than the present day limits without any undue damage to the equipment.
VII. REFERENCES


APPENDIX A
DIGITAL MODEL DESCRIBING EQUATIONS AND PARAMETER VALUES

A.1 Nomenclature

In view of the large number of variables required to describe the boiler-turbine system, it was necessary to devise a systematic convention for naming the variables. Any variable names that do not conform with the said convention are listed as exceptions in Table A.1, Section E.

Associated with every variable is a name which is coded based upon the following convention:

1) A variable name contains six or less alphanumeric characters, the first of which must be an alphabet letter.

2) A variable name may include one of the prefixes listed in Table A.1, Section A, but it is not required to.

3) The first character following the prefix (if applicable) represents the primary quantity as listed in Table A.1, Section B.

4) The pair of letters following the primary quantity (if applicable) identifies the component or material as listed in Table A.1, Section C.

5) Any additional characters are used to represent conditions, materials, or processes as listed in Table A.1, Section D.

6) Closely related variables are distinguished by the use of numeric characters, which may appear anywhere in the name except that it can not be the first character. This is commonly employed in naming component parameters, and intermediate variables. A numeric character at the end of a variable name may be used to denote the order of self multiplication of the variable.
Illustrative examples of variable names used are:

- **T_{sso}**: Temperature, secondary superheater, outlet
- **W_{fl}**: Mass flow rate of fuel
- **K_{ncp}**: Constant, number of condensate pumps
- **K_{uwwms}**: Constant, heat transfer coefficient, waterwall metal to steam
- **h_{hpo}**: Enthalpy, high pressure turbine, outlet
- **M_{rhe}**: Mass of reheater, effective
- **P_{pso}**: Pressure, primary superheater, outlet
- **\rho_{drs}**: Density, drum steam
- **T_{wwge4}**: Temperature, waterwall gas, effective, raised to the fourth power
- **K_{6rp}**: Constant, 6th parameter, recirculating pump
TABLE A.1

SUMMARY OF NOMENCLATURE

Section A, Prefixes

K  ≡  A constant
C  ≡  A control system variable
K_c  ≡  A control system constant
K_{tc}  ≡  A control system time constant
Z  ≡  Intermediate variable

Section B, Primary Quantity

A,a  ≡  Area, fan vane position
E,e  ≡  efficiency
F,f  ≡  friction factor, fraction
G,g  ≡  acceleration due to gravity
G_{c},g_{c}  ≡  units conversion factor
H,h  ≡  enthalpy, heating value
J,j  ≡  moment of inertia
L,\ell  ≡  length
M,m  ≡  mass
MW  ≡  power
N,n  ≡  number, speed, frequency
P,p  ≡  pressure
Q,q  ≡  heat transfer rate
R,r  ≡  density
S,s  ≡  slip, entropy, specific heat
\[ T, t \equiv \text{torque, temperature; temperature factor} \]
\[ U, u \equiv \text{heat transfer coefficient, specific internal energy} \]
\[ V, v \equiv \text{volume, voltage} \]
\[ W, w \equiv \text{mass flow rate} \]
\[ X, x \equiv \text{burner tilt, position, length, water level} \]
\[ \eta \equiv \text{efficiency} \]
\[ \rho \equiv \text{density} \]
\[ \phi \equiv \text{function (defined later)} \]
\[ \delta \equiv \text{power angle} \]

**Section C, Components**

\[ a h \equiv \text{air heater (glycol)} \]
\[ a p \equiv \text{air preheater} \]
\[ a r \equiv \text{air} \]
\[ a t \equiv \text{atmosphere} \]
\[ b p \equiv \text{booster feed pump} \]
\[ c n \equiv \text{condenser} \]
\[ c p \equiv \text{condensate pump} \]
\[ c r \equiv \text{cross-over pipe} \]
\[ c v \equiv \text{governor control valve} \]
\[ c w \equiv \text{condensate (water)} \]
\[ d c \equiv \text{downcomer} \]
\[ d e \equiv \text{deaerator} \]
\[ d r \equiv \text{drum} \]
\[ d v \equiv \text{deaerator valve} \]
ec = economizer
fd = forced draft fan
fl = fuel
fn = furnace
fp = boiler feed pump
ft = feedpump turbine
fv = feedwater valve
fw = feedwater
gg = gun
gn = generator
gr = gas recirculation
gv = governor control valve
hp = high pressure turbine
hh = high pressure feedwater heater
id = induced draft fan
ip = intermediate pressure turbine
iv = intercept valve
lh = low pressure feedwater heater
lp = low pressure turbine
ps = primary superheater
rh = reheater
rp = recirculating pump
rw = recirculating water
ry = reheat spray
sc = steam chest
sh = superheater
ss = secondary superheater
st = stack
sy = superheat spray
sv = stop valve
tr = turbine
tv = throttle valve
vf = forced draft fan vane
vi = induced draft fan vane
ww = waterwall

**Section D, Conditions**

a = air
bd = blowdown
c = convective heat transfer
d = difference, drop, change
e = effective, average
elec = electrical
g = gas, gun
i = inlet condition, isentropic process
isen = isentropic
in = in, inlet
L = lower limit
m = metal, motor
max = maximum
o = outlet condition
\text{out} \equiv \text{out, outlet}
\text{pu} \equiv \text{per unit}
\text{r} \equiv \text{radiation, rated, ratio}
\text{s} \equiv \text{steam, supply line, seal}
\text{sr} \equiv \text{steam return}
\text{u} \equiv \text{upper limit}
\text{v} \equiv \text{valve}
\text{w} \equiv \text{water}
\text{x} \equiv \text{extraction}
\text{l} \equiv \text{inlet condition}

\text{Section E, Exceptions}

\text{Ldc} \equiv \text{load demand computer signal}
\text{q}_{\text{ww}} \equiv \text{quality of steam leaving waterwall}
\text{q}_{\text{ylpo}} \equiv \text{quality of steam leaving low pressure turbine}
\text{y}_{\text{wgr}} \equiv \text{water to gas ratio of flue gas}
\text{w}_{\text{g}} \equiv \text{flue gas flow rate}
A.2 Governing Equations

Based upon the assumptions and methods described in Chapter III, the governing equations are derived for each system component. Where necessary, specific assumptions are stated for the components. In what follows, \( \phi \) is used to denote functional relationships which are listed in Section A.5. In addition, model parameter values are listed in Table A.2.

**DRUM**

The drum water and steam are assumed to be in saturated equilibrium so that one intensive thermodynamic property is sufficient to describe the thermodynamic state of the drum. The steam density is used. The presence of both water and steam in the drum is reflected in the governing equations. Feedwater from the economizer and water-steam mixture from the waterwall flow into the drum while recirculating water leaves the drum for the downcomers and steam leaves the drum for the primary superheater and blowdown. The feedwater is assumed to mix with drum water in the downcomers. The drum equations are:

**Continuity:**

\[
\frac{d}{dt} \left[ \rho_{\text{drs}} V_{\text{drs}} + \rho_{\text{drw}} V_{\text{drw}} \right] = \dot{W}_{\text{wwo}} \left( W_{\text{rw}} - W_{\text{fw}} \right) - \dot{W}_{\text{drs}} - \dot{W}_{\text{drbd}}
\]

**Energy:**

\[
\frac{d}{dt} \left[ \rho_{\text{drs}} V_{\text{hrs}} + \rho_{\text{drw}} V_{\text{hwr}} \right] = \dot{W}_{\text{wwo}} h_{\text{wo}} - \left( W_{\text{rw}} - W_{\text{fw}} \right) h_{\text{drw}}
\]

\[
- \dot{W}_{\text{drs}} h_{\text{hrs}} - \dot{W}_{\text{drbd}} h_{\text{hrs}}
\]
Constitutive Relations:

\[ X_{drw} = K_{1xdrw} + K_{2xdrw} V_{drw} + K_{3xdrw} V_{drw}^2 \]

\[ V_{drs} = K_{vdr} - V_{drw} \]

\[ \rho_{drw} = \phi(\rho_{drs}) \]

\[ h_{drw} = \phi(\rho_{drs}) \]

\[ h_{drs} = \phi(\rho_{drs}) \]

\[ p_{drs} = \phi(\rho_{drs}) \]

\[ T_{drs} = \phi(\rho_{drs}) \]

The state variables associated with the drum are the drum steam density \( (\rho_{drs}) \) and water volume \( (V_{drw}) \). The drum water level is determined directly from the water volume based on the net water volume and does not include the effect of steam bubbles that may actually be present in the water. Except for the blowdown steam which is related to the main steam flow, other flows are determined by the dynamics of the associated components. In general, \( W_{wwo} \) is not equal to \( W_{rw} \) but if leakages and pump seal injection are ignored, the two are equal.

**DOWNCOMERS**

For the downcomers, incompressibility and negligible heat losses are assumed. Furthermore, feedwater is assumed to flow completely into the downcomers, so that the excess of the downcomers flow (recirulating flow) over the feedwater flow is furnished by the drum water. The equations for the downcomers are:
Heat Balance at Downcomer Entrance:

\[ W_{rw \ dcl} = W_{fw \ eco} + (W_{rw} - W_{fw}) \ h_{drw} \]

Momentum:

\[ P_{drs} - P_{dco} = K_{fdc} \frac{W_{rp}^2}{\rho_{dc}} - K_{ldc} \rho_{dc} \ g/g_c \]

Constitutive Relations:

\[ W_{rw} = K_{nrp} \ W_{rp} \]
\[ \rho_{dc} = \phi(P_{drs}, h_{dcl}) \]
\[ h_{dco} = \phi(P_{dco}, \rho_{dc}) \]

The flow rate \( W_{rp} \) (and therefore \( W_{rw} \)) is computed by simultaneously solving the downcomer and waterwall momentum equations with the recirculating pumps characteristics. With density and enthalpy known for both inlet and outlet, the state of water in the downcomers is essentially described.

**RECIRCULATING PUMPS**

Recirculating water from the downcomers goes through six recirculating pumps where the flow is again assumed incompressible. The dynamics of the pumps are governed by the momentum equation and energy balance. The pump performance characteristics [16] play an important role in providing a constitutive relationship between the flow rate and pressure head developed, which in turn is significant in flow computation. The driving torque for the pumps is furnished by induction motors. The pump equations are:

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Moment of Momentum:

\[ K_j \frac{dN_{rp}}{dt} = T_{rpm} - \frac{W_{rp}}{\rho_{rp}} (P_{rpo} - P_{dco}) \]

where \( \rho_{rp} = \rho_{dc} \)

Pump Performance Characteristics:

\[ \frac{P_{rpo} - P_{dco}}{\rho_{rp}} = K_{1rp} \left( \frac{W_{rp}}{\rho_{rp}} \right)^2 + K_{2rp} N_{rp} \frac{W_{rp}}{\rho_{rp}} + K_{3rp} N_{rp}^2 \]

\[ \eta_{rp} = K_{4rp} \left( \frac{W_{rp}}{\rho_{rp}} \right)^2 + K_{5rp} N_{rp} \frac{W_{rp}}{\rho_{rp}} + K_{6rp} N_{rp}^2 \]

Induction Motor Characteristics:

\[ T_{rpm} = \frac{K_{rpm} v^2}{S_{rp} + \frac{S_{rpmax}}{S_{rpmax}}} \]

\[ S_{rp} = \frac{N_{rpmax} - N_{rp}}{N_{rpmax}} \]

\( S_{rpmax} = \) slip at maximum torque

\( N_{rpmax} = \) synchronous speed of motor

\( = \frac{N_{elec}}{2} \)

Steam Table Fit:

\[ h_{rpo} = \phi(P_{rpo}, \rho_{rp}) \]
The state variable associated with the recirculating pump is the pump speed \( N_{rp} \).

**WATERWALL**

The waterwall which is between the recirculating pumps and drum completes the recirculation loop. Water from the pumps is heated in the waterwall first to saturation conditions and thereafter at saturation condition but increasing steam quality. The drum water density and saturation temperature are assumed to represent the mean density and temperature of the waterwall water. This deduction follows from the temperature profile of boiling water in vertical tubes [18]. Compressibility effects in the waterwall are neglected, since the overall steam-water mass ratio is small. Also, the waterwall water and steam are lumped with the waterwall metal by way of an effective mass. The equations describing the waterwall are:

**Momentum:**

\[
P_{rpo} - P_{drs} = K_{fww} \frac{W_{wo}}{\rho_{drw}} + K_{lww} \rho_{drw} \frac{g}{g_c}
\]

\[W_{wo} = W_{rw} + K_{wrps}\]

where \( K_{wrps} \) accounts for recirculating pump leakages and seal injection. It is small and may be neglected.

**Energy:**

\[
M_{wwme} \frac{dT_{wwm}}{dt} = Q_{wwgm} - Q_{wwmw}
\]
Heat Balance for Water:

\[ W_{wwo}(h_{wwo} - h_{rpo}) = Q_{wwmw} \]

The heat transfer from the flue gas to the waterwall is dominated by radiation while the transfer from the tube metal to water is basically dominated by boiling. \( U_{xgg} \) and \( U_{ngg} \) incorporate the effects of burner tilt and number of operating guns respectively. The explicit expressions for these quantities are given in the furnace section. The state variable associated with the waterwall is the metal tube temperature \( T_{wwm} \).

**PRIMARY SUPERHEATER**

In the steam side, in general, incompressibility is not assumed, but the inertia term in the momentum equation is considered negligible compared to the friction and pressure terms. Kinetic energy change is neglected in the energy equation and the metal mass lumped with the steam mass by a suitably defined effective mass. Furthermore, the heat transfer resistance of the metal tube is neglected.
compared with the resistance in the gas and steam sides. Consequently, the gas side thermal resistance is treated in series with the steam side thermal resistance [1,2,3,4,5,11,18]. The equations for the primary superheater are:

**Continuity:**

$$\frac{d \rho_{ps}}{dt} = \frac{W_{drs} - W_{pso}}{K_{vps}}$$

**Momentum:**

$$P_{drs} - P_{pso} = W_{pso} \frac{W^2}{\rho_{dras}}$$

**Energy:**

$$M_{pse} \frac{dh_{pse}}{dt} = W_{pso} \rho_{ps} + \frac{\rho_{ps}}{K_{mps}} \frac{K_{psm}}{h_{pse}} \frac{Q_{ps}}{T_{psm}}$$

$$Q_{ps} = Q_{psc} + Q_{psr}$$

$$Q_{psr} = K_{upsr} (T_{wwge}^4 - T_{pse}^4)$$

$$Q_{psc} = \frac{k_{ups}^0.6 K_{upsms}^0.8}{K_{ups}^0.6 + K_{upsms}^0.8} (T_{psge} - T_{pse})$$

The heat transfer coefficient for $Q_{psc}$ is obtained by a series combination of the gas and steam side heat transfer coefficients:

$$\frac{1}{K_{ups}^0.6} \frac{1}{K_{upsms}^0.8}$$
\[ W_{pse} = \frac{1}{2}(W_{drs} + W_{pso}) \]
\[ \rho_{pse} = \frac{1}{2}(\rho_{drs} + \rho_{pso}) \]
\[ h_{pse} = \frac{1}{2}(h_{drs} + h_{pso}) \]
\[ T_{psmc} = T_{pse} + \frac{Q_{pse}}{K_{upsms} \rho_{pse}^{0.8}} \]

Steam Table Fits:
\[ T_{pse} = \phi(\rho_{pse}, h_{pse}) \]
\[ P_{pso} = \phi(\rho_{pso}, h_{pso}) \]
\[ T_{pso} = \phi(\rho_{pso}, h_{pso}) \]

The primary superheater receives heat from furnace radiation and convection of the flue gas as reflected in the expression for the net heat transfer to the primary superheater steam. The state variables associated with the primary superheater are outlet steam density \( \rho_{pso} \) and enthalpy \( h_{pso} \).

**SUPERHEAT SPRAY**

Some steam from the primary superheater is extracted for auxiliary use and the rest sprayed with water as a means of regulating the temperature of the secondary superheater outlet steam. The superheat spray section is modeled simply by energy and mass balances with the pressure drop across it assumed negligible. The equations describing this effect are:
Mass Balance:

\[ \dot{w}_{ssl} = \dot{w}_{pso} - \dot{w}_{psx} + \dot{w}_{sy} \]

\[ \dot{w}_{psx} = \phi(\dot{w}_{pso}) \]

Energy Balance:

\[ \dot{w}_{ssl} h_{ssl} = (\dot{w}_{pso} - \dot{w}_{psx}) h_{pso} + \dot{w}_{sy} h_{fpo} \]

The sprayed water is feedwater extracted from the boiler feedpump discharge, and so, the enthalpy of the spray is equal to the feedpump outlet water enthalpy.

Steam Table Fits:

\[ \rho_{ssl} = \phi(P_{pso}, h_{ssl}) \]

\[ T_{ssl} = \phi(P_{pso}, h_{ssl}) \]

There is no state variable associated with the spray section.

SECONDARY SUPERHEATER

Steam from the spray section feeds into the secondary superheater for the final stage of superheating. Steam is extracted at the secondary superheater discharge for feedpump turbine drive (\( \dot{w}_{ssx} \)). The same assumptions outlined for the primary superheater are valid for the secondary superheater, and the governing equations are similar:

Continuity:

\[ \frac{d \rho_{sso}}{K_{vss}} = \dot{w}_{ssl} - \dot{w}_{sso} \]
\[ W_{ss0} = W_{cv} + W_{ssx} \]

\[ W_{ssx} = \phi(W_{cv}) \]

**Momentum:**

\[ p_{ps0} - p_{ss0} = K_{fss} \frac{W_{ssl}^2}{\rho_{ss0}} \]

**Energy:**

\[ M_{sse} \frac{dh_{ss0}}{dt} = \dot{W}_{ssl} h_{ssl} - W_{ss0} h_{ss0} + Q_{ss} \]

\[ M_{sse} = K_{vsse} \rho_{sse} + K_{msse} \frac{T_{ssme}}{h_{sse}} \]

\[ Q_{ss} = \frac{K_{ussme} W_{0.6}^0 K_{ussms} W_{0.8}^0}{K_{ussgm} W_{0.6}^g + K_{ussms} W_{0.8}^s} (T_{ssge} - T_{sse}) \]

\[ \rho_{sse} = \frac{1}{2} (\rho_{ssl} + \rho_{ss0}) \]

\[ h_{sse} = \frac{1}{2} (h_{ssl} + h_{ss0}) \]

\[ T_{ssme} = T_{sse} + \frac{Q_{ss}}{K_{ussms} W_{0.8}^s} \]

**Steam Table Fits:**

\[ T_{sse} = \phi(\rho_{sse}, h_{sse}) \]

\[ T_{ss0} = \phi(\rho_{ss0}, h_{ss0}) \]

\[ p_{ss0} = \phi(\rho_{ss0}, h_{ss0}) \]
\[ S_{SSO} = \phi(p_{SSO}, h_{SSO}) \]

The state variables associated with the secondary superheater are the outlet steam density \((p_{SSO})\) and enthalpy \((h_{SSO})\). Desired superheater outlet steam condition is maintained by regulating the temperature and pressure.

**THROTTLE AND CONTROL VALVES**

Steam from the secondary superheater is throttled through valves into the steam chest which is located at the high pressure turbine inlet. For critical flow through the turbine, the steam flow rate is related to the net normalized valve area and inlet steam condition by the expression [14,15,19]

\[ W_{cv} = K_{cv} A_{cv} \sqrt{\frac{P_{SSO}}{p_{SSO}}} \]

The particular valve in operation depends on the operating mode of the plant. That is, whether the plant is operating full arc or partial arc. During full-arc operations the control valves remain wide open while the throttle (stop) valves are varied, while in partial arc operations the throttle (stop) valves remain wide open and the control valves varied. There are two stop values and four control valves, although the total normalized area is used in the flow computation. Full-arc operation is usually employed during start-ups.

**STEAM CHEST**

The throttling process preceeding the steam chest is assumed perfect so that the change in enthalpy is negligible. With

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thorough mixing of steam in the chest assumed without any heat losses or pressure drop, the enthalpy of the steam in the chest is set equal to the secondary superheater outlet steam enthalpy. The dynamics of the steam chest are governed by the continuity equation. The flow from the steam chest to the turbine is dependent upon the steam condition [14,15]. The steam chest equations are:

**Continuity:**

\[
K_{vsc} \frac{d}{dt} \rho_{sco} = W_{cv} - W_{hp}
\]

\[
W_{hp} = K_{hp} \sqrt{P_{sco} \rho_{sco}}
\]

**Energy:**

\[
h_{sco} = h_{sso}
\]

**Steam Table Fits:**

\[
P_{sco} = \phi(\rho_{sco}, h_{sco})
\]

\[
S_{sco} = \phi(\rho_{sco}, h_{sco})
\]

The state variable associated with the steam chest is the steam density \((\rho_{sco})\).

**HIGH PRESSURE TURBINE**

Steam from the steam chest is expanded through the stages of the high pressure turbine to produce power. The expansion process in the turbine is assumed to be adiabatic and is described in terms of an isentropic efficiency [14,15,20]. The isentropic efficiency is a function of flow. With the inlet condition and one exhaust property.
known, the isentropic efficiency provides a complete determination of the exhaust condition. The flow through the turbine is assumed sonic and takes place so fast that the storage effect is negligible. There is extraction at the high pressure turbine discharge for high pressure feedwater heating and auxiliary use. The high pressure turbine equations are:

**Continuity:**

\[ W_{\text{hpo}} = W_{\text{ap}} - W_{\text{hp}} \]

\[ W_{\text{hp}} = \phi(W_{\text{hp}}) \]

**Energy:**

\[ MW_{\text{hp}} = W_{\text{hp}} (h_{\text{sc}} - h_{\text{hpo}}) \]

**Adiabatic Process:**

\[ \eta_{\text{isen}} = \phi(W_{\text{hp}}) \]

\[ h_{\text{hpo1}} = \phi(P_{\text{hpo}}, S_{\text{sc}}) \]

\[ h_{\text{hpo}} = h_{\text{hpo1}} - \eta_{\text{isen}} (h_{\text{sc}} - h_{\text{hpo1}}) \]

\[ T_{\text{hpo}} = \phi(P_{\text{hpo}}, h_{\text{hpo}}) \]

The high pressure turbine is mounted on the same shaft as the intermediate and low pressure turbines and so they have a common angular speed. The state variable associated with the turbines is the angular speed \( N_{tr} \). The dynamic equation governing the turbine speed is given under Turbine-Generator below.
REHEAT SPRAY

Exhaust steam from the high pressure turbine is sprayed with water in the reheat spray section to regulate the reheater outlet steam temperature. The analysis of the reheat spray is similar to that of the superheat spray, governed by mass and energy balance:

Mass Balance:

\[ W_{rhl} = W_{hpo} + W_{ry} \]

Energy Balance:

\[ W_{rhl} h_{rhl} = W_{hpo} h_{hpo} + W_{ry} h_{fpo} \]

Steam Table Fits:

\[ \rho_{rhl} = \phi(P_{hpo}, h_{rhl}) \]

\[ T_{rhl} = \phi(P_{hpo}, h_{rhl}) \]

The pressure drop across the reheat spray section is assumed negligible so that the outlet pressure is equal to \( P_{hpo} \).

REHEATER

The assumptions outlined for the primary superheater are also valid for the reheater and the analysis here is similar to that of the superheater:

Continuity:

\[ K_{vrh} \frac{d \rho_{rho}}{dt} = W_{rhl} - W_{ip} \]
Momentum:

\[ P_{\text{ho}} - P_{\text{ho}} = K_{\text{frh}} \frac{w^2}{\rho_{\text{ho}}} \]

Energy:

\[ M_{\text{rhe}} \frac{d}{dt} h_{\text{ho}} = W_{\text{rhl}} h_{\text{rhl}} - W_{\text{ip}} h_{\text{rho}} + Q_{\text{rh}} \]

\[ M_{\text{rhe}} = K_{\text{vrh}} \rho_{\text{rhe}} + K_{\text{mrhm}} K_{\text{srhm}} \frac{T_{\text{rhe}}}{h_{\text{rhe}}} \]

\[ Q_{\text{rh}} = \frac{K_{\text{urhgm}} w_{0.6} + K_{\text{urhms}} w_{0.8}}{K_{\text{urhgm}} w_{0.6} + K_{\text{urhms}} w_{0.8}} (T_{\text{rhe}} - T_{\text{rhe}}) \]

\[ W_{\text{rhe}} = \frac{1}{2} (W_{\text{rhl}} + W_{\text{ip}}) \]

\[ \rho_{\text{rhe}} = \frac{1}{2} (\rho_{\text{rhl}} + \rho_{\text{rho}}) \]

\[ h_{\text{rhe}} = \frac{1}{2} (h_{\text{rhl}} + h_{\text{rho}}) \]

\[ T_{\text{rhe}} = T_{\text{rhe}} + \frac{Q_{\text{rh}}}{K_{\text{urhms}} w_{0.8}} \]

Steam Table Fits:

\[ T_{\text{rhe}} = \phi(\rho_{\text{rhe}}, h_{\text{rhe}}) \]

\[ P_{\text{rho}} = \phi(\rho_{\text{rho}}, h_{\text{rho}}) \]

\[ T_{\text{rho}} = \phi(\rho_{\text{rho}}, h_{\text{rho}}) \]

\[ S_{\text{rho}} = \phi(\rho_{\text{rho}}, h_{\text{rho}}) \]
The state variables associated with the reheater are the outlet steam density \( (\rho_{\text{rho}}) \) and enthalpy \( (h_{\text{rho}}) \).

**INTERCEPT VALVES**

Flow from the reheater passes through the intercept valve before reaching the intermediate pressure turbine. As in the case of the throttle valves, the flow through the intercept valve is governed by the normalized valve area and the steam condition at the valve inlet [14,15].

\[
W_{\text{ip}} = K_{iv} A_{iv} \sqrt{P_{\text{rho}}} \rho_{\text{rho}}
\]

The intercept valve is maintained wide open in normal operation and is used only in special situations like fast valving.

**INTERMEDIATE PRESSURE TURBINE**

Reheated steam is expanded through the stages of the intermediate pressure turbine and discharged into the cross-over pipe. The flow process in the intermediate pressure turbine is assumed adiabatic and the analysis is similar to that of the high pressure turbine, based on isentropic efficiency. The flow through the intermediate pressure turbine takes place so fast that the storage effect of the turbine is negligible. Unlike the high pressure turbine, apart from steam extraction from the intermediate pressure turbine discharge, there is also interstage steam extraction for regenerative feedwater heating. The equations describing the intermediate pressure turbine are:
Continuity:

\[ W_{ipo} = W_{ip} - W_{ipx} \]

\[ W_{ipx} = \phi(W_{ip}) \]

Energy:

\[ MW_{ip} = W_{ip}(h_{rho} - h_{ipo})_{eip} \]

\( K_{eip} \) accounts for the fact that not all of \( W_{ip} \) is utilized in generating power. Some of it is extracted interstage.

Adiabatic:

\[ \eta_{isen} = \phi(W_{ip}) \]

\[ h_{ipo} = h_{ipoi} - \eta_{isen}(h_{rho} - h_{ipoi}) \]

\[ h_{cro} = h_{ipo} \]

Though \( \eta_{isen} \) is a function of \( W_{ip} \) it was found sufficient to use a constant value. No exclusive state variable is associated with the intermediate pressure turbine; the turbine angular speed \( (N_{tr}) \) is common for the three elements.

**CROSS-OVER PIPE**

The cross-over pipe connects the intermediate pressure turbine with the low pressure turbine. Heat loss and pressure drop are assumed negligible and the dynamics are governed by the continuity equation. The flow rate of steam to the low pressure is dependent
upon the cross-over pipe steam condition. The cross-over pipe steam condition depends largely on the adiabatic process in the intermediate pressure turbine as reflected in the expression for $h_{cro}$. The equations describing the cross-over pipe are.

**Continuity:**

$$K_{ycro} \frac{dp_{cro}}{dt} = W_{ipo} - W_{lp}$$

$$W_{lp} = K_{lp} \sqrt{P_{cro} \rho_{cro}}$$

**Energy:**

$$h_{cro} = h_{ipo}$$

**Steam Table Fits:**

$$P_{cro} = \phi(\rho_{cro}, h_{cro})$$

$$T_{cro} = \phi(\rho_{cro}, h_{cro})$$

The state variable associated with the cross-over pipe is the steam density ($\rho_{cro}$).

**LOW PRESSURE TURBINES**

The low pressure turbines which represent the final element of the compound turbine accepts steam from the cross-over pipe, expands it through the stages and discharges to the condenser while delivering power. There are both discharge and interstage steam extractions for regenerative feedwater heating. The low pressure tur-
bine is analyzed in a manner different from the high and intermediate pressure turbines. The discharge condition is determined by the condenser condition. Steam leaves the low pressure turbine in the two phase regime and so, knowledge of the condenser pressure defines the intensive thermodynamic state of the discharge steam and water components. To completely determine the condition of the discharge steam an additional property is required; for example, quality of steam. The equations describing the low pressure turbine are:

**Energy:**

\[ MW_{lp} = W_{lp} (h_{cro} - h_{lpo}) K_{elp} \]

\( K_{elp} \) accounts for the fact that part of \( W_{lp} \) is extracted interstage.

**Steam-Water Mixture:**

The variation in quality of the discharge steam \( (K_{qylpo}) \) is small enough to allow it to be assumed constant.

Here again, no exclusive state variable is associated with the low pressure turbine; it shares the turbine speed which is common for the three elements.

**TURBINE-GENERATOR**

The dynamics of the turbine-generator are governed by a combined energy and moment of momentum balance along with the electromagnetic spring action of the generator [21]. In modeling
the generator action, the damping effect is ignored compared to the
damping effect present in the mechanical system.

The total power produced by the turbine is the sum of the
power produced by the high, intermediate and low pressure elements,
multiplied by the overall efficiency. This power drives the generator.
The equations governing the turbine-generator dynamics are given be-
low:

\[
\begin{align*}
\text{Energy:} & \\
MW_{\text{tro}} &= (MW_{\text{hp}} + MW_{\text{ip}} + MW_{\text{lp}}) \eta_{\text{tr}} \\
\text{Moment of Momentum:} & \\
\frac{dN_{\text{tr}}}{dt} &= (MW_{\text{tro}} - MW_{\text{gn}})/N_{\text{tr}} \\
MW_{\text{gn}} &= K_{\text{mwr}} MW_{\text{gnpu}}
\end{align*}
\]

\(K_{\text{mwr}}\) is the rated power and \(MW_{\text{gnpu}}\) is the per unit generator
power output.

\text{Generator Action [21]:}

\[
\begin{align*}
MW_{\text{gnpu}} &= 2 \sin \delta \\
\frac{d\delta}{dt} &= N_{\text{tr}} - N_{\text{elec}}
\end{align*}
\]

\(N_{\text{elec}}\) is the electrical system frequency and \(\delta\) is the power
angle. By means of this action, the synchronous machine, generator,
attempts to keep the turbine speed equal to \(N_{\text{elec}}\). In effect,
an infinite bus is assumed, so that the generator accepts whatever

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power is produced by the turbine.

The inclusion of this model in the power plant model explicitly links the mechanical power unit with the rest of the power system, and makes possible a realistic simulation of the governor droop action.

The state variables associated with the turbine-generator are the turbine speed ($N_{tr}$) and power angle ($\delta$).

**CONDENSER**

The dynamics of the condenser are similar to that of the drum with only some modification to account for the condensation and associated heat transfer. The variation of condenser conditions is nevertheless small over a wide range of loads. It was found sufficient to describe the condenser condition by relating the condenser pressure to the cross-over pipe pressure. With the condenser pressure known the corresponding saturation properties are evaluated using steam table fits. The describing equations are:

$$P_{cn} = \phi(P_{cro})$$

**Steam Table Fits:**

$$\rho_{cnw} = \phi(P_{cn})$$

$$h_{cnw} = \phi(P_{cn})$$

$$h_{cns} = \phi(P_{cn})$$

$$T_{cns} = \phi(P_{cn})$$
With the representation above, the condenser is described without associated state variables.

**CONDENSATE PUMPS**

Two condensate pumps deliver water to the low pressure feedwater heater from the condensers. The analysis of the condensate pumps is analogous to that of the recirculating pumps, and the assumptions made in the section on recirculating pumps are also valid for the condensate pumps. The pumps are represented by the equations:

**Moment of Momentum:**

\[
K_{jcp} \frac{d N_{cp}}{dt} = T_{cp} - \frac{W_{cp} (P_{cpo} - P_{cn})}{n_{cp} \rho_{cp} N_{cp}}
\]

\[
\rho_{cp} = \rho_{cw}
\]

\[
W_{cp} = \frac{W_{cw}}{K_{ncp}}
\]

**Pump Performance Characteristics:**

\[
\frac{P_{cpo} - P_{cn}}{\rho_{cn}} = K_{1cp} \left( \frac{W_{cp}}{\rho_{cp}} \right)^2 + K_{2cp} N_{cp} \frac{W_{cp}}{\rho_{cp}} + K_{3cp} N_{cp}^2
\]

\[
n_{cp} = K_{4cp} \left( \frac{W_{cp}}{\rho_{cp}} \right)^2 + K_{5cp} N_{cp} \frac{W_{cp}}{\rho_{cp}} + K_{6cp} N_{cp}^2
\]
**Induction Motor Characteristics:**

\[
T_{cpl} = \frac{K_{cpm} \cdot v^2}{S_{cp} + \frac{S_{cpmax}}{S_{cpmax}}} \quad \frac{S_{cpmax}}{S_{cp}}
\]

\[
S_{cp} = \frac{N_{cpmax} - N_{cp}}{N_{cpmax}}
\]

\[
N_{cpmax} = \text{Synchronous speed of motor}
\]

\[
= \frac{N_{elec}}{2}
\]

**Steam Table Fits:**

\[
h_{cpo} = \phi(P_{cpo},\rho_{cp})
\]

The state variable associated with the condensate pump is the speed \(N_{cp}\).

**LOW PRESSURE FEEDWATER HEATER**

Condensate from the condensate pumps is heated in the low pressure feedwater heater by extraction steam from the turbines. The extraction steam leaves the heater as saturated water. The flow of water in the heater is assumed incompressible and the metal mass is lumped with the water mass as an effective mass. The feedwater heater is a closer heater. The equations for the heater are:
Momentum:

\[ P_{cpo} - P_{1ho} = K_{f1h} \frac{W_{cw}^2}{\rho_{1hl}} \]

\[ \rho_{1hl} = \rho_{cp} \]

Energy:

\[ M_{1he} \frac{dh_{1ho}}{dt} = W_{cw}(h_{cpo} - h_{1ho}) + Q_{1h} \]

\[ Q_{1h} = W_{1lhs}(h_{1lhs1} - h_{3lhs0}) + W_{2lhs}(h_{2lhs1} - h_{2lhs0}) \]

\[ W_{1lhs} = \phi(W_{1p}) \]

\[ h_{1lhs1} = \phi(h_{cro}, h_{lpo}) \]

\[ P_{3lhs} = \phi(P_{cro}, P_{cn}) \]

\[ P_{2lhs} = \phi(P_{cro}, P_{cn}) \]

\[ W_{2lhs} = \phi(W_{1p}) \]

\[ h_{2lhs1} = \phi(h_{cro}, h_{lpo}) \]

\[ M_{1he} = K_{vlh} \rho_{1he} + K_{mlhm} K_{slhm} T_{1hme} \]

\[ T_{1hme} = K_{lhm} T_{1ho} \]

\[ h_{1he} = \frac{1}{2}(h_{cpo} + h_{1ho}) \]

\[ P_{1he} = \frac{1}{2}(P_{cpo} + P_{1ho}) \]

Steam Table Fits:

\[ \rho_{1he} = \phi(h_{1he}, P_{1he}) \]

\[ h_{2lhs0} = \phi(P_{2lhs}) \]

\[ h_{3lhs0} = \phi(P_{3lhs}) \]
\[ \rho_{lho} = \phi(h_{lho}, p_{lho}) \]

\[ T_{lho} = \phi(h_{lho}, p_{lho}) \]

The state variable associated with the low pressure feedwater heater is the outlet enthalpy \( (h_{lho}) \).

**DEAERATOR FEED VALVE**

Part of the water from the low pressure feedwater heater is extracted for glycol-air heating and the rest proceeds to the deaerator through the deaerator valve. The valve is described by the following equations:

\[ w_{dvo} = w_{cw} - w_{1hx} \]

\[ p_{lho} - p_{des} = \frac{K_{dv} w^2}{\rho_{lho} A_{dv}^2} \]

\[ W_{1hx} = \phi(W_{cw}) \]

**DEAERATOR**

Water from the low pressure feedwater heater, extraction steam and auxiliary steam returns all flow into the deaerator and the feedwater is pumped out by the boiler feedpump. The analysis of the deaerator is similar to that of the drum. Essentially, the deaerator is an open heater. The equations describing its performance are:
Continuity:

\[ \frac{d}{dt} [\rho_{des} V_{des} + \rho_{dew} V_{dew}] = W_{dvo} + W_{dex} + W_{hhs} - W_{fp} + W_{des} \]

\[ V_{des} = \frac{K_{vd}}{V_{dew}} \]

Energy:

\[ \frac{d}{dt} [\rho_{des} V_{des} h_{des} + \rho_{dew} V_{dew} h_{dew}] = W_{dvo} h_{lho} + W_{dex} h_{dex} \]

\[ + W_{hhs} h_{3hhs} - W_{fp} h_{dew} \]

\[ + Q_{desr} \]

\[ W_{dex} = \phi(W_{lp}) \]

\[ h_{dex} = \phi(h_{cro}, h_{lpo}) \]

\[ X_{dew} = K_{1x} V_{dew} + K_{2x} V_{dew}^2 \]

Steam Table Fits:

\[ h_{des} = \phi(\rho_{des}) \]

\[ h_{dew} = \phi(\rho_{des}) \]

\[ \rho_{dew} = \phi(\rho_{des}) \]

\[ P_{des} = \phi(\rho_{des}) \]

\[ T_{des} = \phi(\rho_{des}) \]

The state variables associated with the deaerator are the steam density \(\rho_{des}\) and water volume \(V_{dew}\). 

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BOILER FEED PUMP AND BOOSTER PUMP

The main boiler feed pump acts in series with the booster pump. They deliver feedwater to the high pressure feedwater heater from the deaerator. The analysis of the boiler feed pumps is analogous to that of the recirculating pumps and the assumptions made for the recirculating pumps are valid for the boiler feed pumps. Instead of an induction motor, a steam turbine is employed to drive the boiler feed pump. This steam turbine obtains the steam supply from either the secondary superheater or intermediate pressure turbine extraction. The equations for the pumps are:

Moment of Momentum:

$$\frac{d}{dt} \frac{N_{fp}}{f_{pe}} = T_{fp1} - \frac{W_{fp} (P_{fpo} - P_{des})}{n_{fp} \rho_{fp} N_{fp}}$$

$$\rho_{fp} = \rho_{dew}$$

Pump Performance Characteristics:

Booster Pump:

$$\frac{P_{bpo} - P_{des}}{\rho_{fp}} = K_{lbp} \left( \frac{W_{fp}}{\rho_{fp}} \right)^2 + K_{2bp} N_{bp} \frac{W_{fp}}{\rho_{fp}} + K_{3bp} N_{bp}^2$$

$$N_{bp} = K_{nbpr} N_{fp}$$

Main Boiler Feed Pump:

$$\frac{P_{fpo} - P_{bpo}}{\rho_{fp}} = K_{lfp} \left( \frac{W_{fp}}{\rho_{fp}} \right)^2 + K_{2fp} N_{fp} \frac{W_{fp}}{\rho_{fp}} + K_{3fp} N_{fp}^2$$
Booster and Main Pump Combined:

\[ n_{fp} = K_{4fp} \left( \frac{W_{fp}}{\rho_{fp}} \right)^2 + K_{5fp} N_{fp} \frac{W_{fp}}{\rho_{fp}} + K_{6fp} N_{fp}^2 \]

Feed Pump Turbine:

\[ T_{fp} = \frac{W_{ft} (h_{ftl} - h_{fto}) K_{1}}{N_{fp}} \]

\[ h_{ss} \text{ or } h_{cro} \]

\[ h_{fto} = K h_{fto} \]

Steam Table Fits:

\[ h_{fpo} = \phi(P_{fpo}, \rho_{fp}) \]

\[ T_{fpo} = \phi(P_{fpo}, \rho_{fp}) \]

The state variable associated with the feed pump is the speed \(N_{fp}\).

**FEEDWATER VALVE**

The feedwater valve provides a means of regulating the feedwater flow. Part of the water from the boiler feed pump is extracted for superheat and reheat sprays before reaching the feedwater valve. The governing equations for the feedwater valve are similar to those for the deaerator valve:

\[ P_{fpo} - P_{fvo} = \frac{K_{fv} W_{fw}^2}{\rho_{fv} A_{fv}^2} \]
\[ \rho_{fv} = \rho_{fp} \]
\[ W_{fw} = W_{fp} - W_{sy} - W_{ry} - K_{wfx} \]
\[ h_{fvo} = h_{fpo} \]
\[ K_{wfx} = 19.7 \text{ lb/s} \]

**HIGH PRESSURE FEEDWATER HEATER**

Feedwater from the boiler feed pump is further heated in the high pressure feedwater heater by extraction steam from the turbine. This feedwater is then fed into the economizer for the final stage of heating. The metal mass is lumped with the water mass and the analysis of the heater is similar to that of the low pressure feedwater heater. The extraction steam leaves the heater as saturated water. The equations summarizing performance are:

**Momentum:**

\[ P_{fvo} - P_{hho} = \frac{K_{fhh} W_{fw}^2}{\rho_{hhl}} \]

\[ \rho_{hhl} = \rho_{fw} \]

**Energy:**

\[ M_{hhe} \frac{d}{dt} h_{hho} = W_{fw} (h_{fvo} - h_{hho}) + Q_{hh} \]

\[ M_{hhe} = K_{vhh} \rho_{hhe} + K_{mhm} K_{shh} \frac{T_{hhm}}{h_{hhe}} \]
\[ Q_{hh} = W_{1hhs} h_{hp} + W_{2hhs} h_{2hhs1} + W_{3hhs} h_{cro} - W_{hhs} h_{3hhso} \]

\[ W_{hhs} = W_{1hhs} + W_{2hhs} + W_{3hhs} = \phi(W_{hp}, W_{ip}) \]

\[ P_{3hhs} = \phi(P_{rho}, P_{cro}) \]

\[ h_{2hhs1} = \phi(h_{rho}, h_{cro}) \]

\[ h_{hhe} = \frac{1}{2} (h_{fvo} + h_{hho}) \]

\[ P_{hhe} = \frac{1}{2} (P_{fvo} + P_{hho}) \]

\[ T_{hhe} = K_{hhe} T_{hho} \]

**Steam Table Fits:**

\[ h_{3hhso} = \phi(P_{3hhs}) \]

\[ P_{hhe} = \phi(h_{hhe}, P_{hhe}) \]

\[ P_{hho} = \phi(h_{hho}, P_{hho}) \]

\[ T_{hho} = \phi(h_{hho}, P_{hho}) \]

The state variable associated with the high pressure feedwater heater is the exit water enthalpy \( h_{hho} \).

**ECONOMIZER**

The economizer which lies in the gas path allows the transfer of heat from the flue gas to the feedwater before it is fed into...
the drum. The flow is assumed incompressible and the metal mass is lumped with the water mass. The economizer equations are:

Momentum:

\[ P_{hho} - P_{drs} = K_{feco} \frac{W_{fw}^2}{\rho_{eco}} \]

Energy:

\[ M_{eco} \frac{dh_{eco}}{dt} = W_{fw} (h_{hho} - h_{eco}) + Q_{ec} \]

\[ Q_{ec} = \frac{K_{uecg} W_{g}^{0.6} + K_{uecmw} W_{fw}^{0.8}}{K_{uecg} W_{g}^{0.6} + K_{uecmw} W_{fw}^{0.8}} (T_{ecge} - T_{ece}) \]

\[ M_{eco} = K_{vec} \rho_{eco} + K_{mecm} K_{secm} \frac{T_{ecme}}{h_{eco}} \]

\[ h_{eco} = \frac{1}{2} (h_{hho} + h_{eco}) \]

\[ p_{eco} = \frac{1}{2} (p_{hho} + p_{drs}) \]

\[ T_{ecme} = T_{ece} + \frac{Q_{ec}}{K_{uecmw} W_{fw}^{0.8}} \]

Steam Table Fits:

\[ T_{ece} = \phi(h_{eco}, p_{eco}) \]

\[ \rho_{eco} = \phi(h_{eco}, p_{drs}) \]

\[ \rho_{eco} = \phi(h_{eco}, p_{drs}) \]
The state variable associated with the economizer is the exit water enthalpy \( h_{\text{eco}} \).

**AIR AND GAS SIDE**

For the gas side dynamics, compressibility effects are assumed negligible so that at any instant of time the flow into a component equals the flow out and high frequency dynamics are neglected. In addition, the net variation in local air temperature in the air path (Glycol Air Heater - Forced Draft Fan - Air Preheater) is small enough to make the assumption of constant local air temperature reasonable since the contribution of the energy associated with air is so small compared to the total heat input to the system that any error resulting from this assumption becomes insignificant. Therefore, heat balance in the air path is not explicitly shown.

On the other hand, the heat balance in the flue gas path is very important since this prescribes the energy transferred to the medium which in turn determines the power output of the plant.

The momentum equations are important for both the air and gas path since these are necessary in conjunction with fan characteristics for flow and pressure computations. Constant local gas densities are assumed. The friction coefficients used incorporate a density factor.

**GLYCOL AIR HEATER**

The first stage of preheating air from the atmosphere takes place in the glycol air heater before the air enters the forced draft
fans. The air heater equation is:

Momentum:

\[ \text{K}_{\text{pat}} - P_{\text{aho}} = K_{\text{fah}} W_{\text{ar}}^2 \]

FORCED DRAFT FANS

Two centrifugal fans supply air from the atmosphere to the furnace. The air flow is controlled by adjusting the inlet vanes. The dynamics and characteristics of the fans are similar to those of the pumps except for the extra variable which models the inlet vane position [17]. The fans are also driven by induction motors, which prescribe the input torque in the moment of momentum equation:

Moment of Momentum:

\[ \frac{d N_{\text{fd}}}{dt} = T_{\text{fd}} - \frac{W_{\text{fd}} (P_{\text{fdo}} - P_{\text{aho}})}{n_{\text{fd}} \rho_{\text{fd}} N_{\text{fd}}} \]

\[ W_{\text{fd}} = \frac{W_{\text{ar}}}{K_{\text{nfd}}} \]

\[ \rho_{\text{fd}} = 0.066 \text{ lb/ft}^3 \]

Fan Characteristics:

\[ \frac{P_{\text{fdo}} - P_{\text{aho}}}{A_{\text{vf}}} = K_{\text{1fd}} \left( \frac{W_{\text{fd}}}{A_{\text{vf}}} \right)^2 + K_{\text{2fd}} N_{\text{fd}} \frac{W_{\text{fd}}}{A_{\text{vf}}} + K_{\text{3fd}} N_{\text{fd}}^2 \]

where 'A_{\text{vf}}' models the normalized inlet vane position.
\[ n_{fd} = K_{4fd} W_{fd}^2 + K_{5fd} N_{fd} W_{fd} + K_{6fd} N_{fd}^2 \]

Density is assumed constant and is incorporated in \( K_{1fd} - K_{6fd} \).

**Induction Motor Characteristics:**

\[
T_{fdl} = \frac{K_{fdm} V^2}{S_{fd} + \frac{S_{fdmax}}{S_{fdmax}}} = \frac{N_{fdmax} - N_{fd}}{N_{fdmax}}
\]

\[ S_{fdmax} = \text{slip for maximum torque} \]

\[ N_{fdmax} = \text{synchronous speed of motor} = \frac{N_{elec}}{6} \]

The state variable associated with the forced draft fan is the speed \( N_{fd} \).

**AIR PREHEATER**

The air preheater which lies in the flue gas path transfers heat from the flue gas to the air before entering the furnace. The variation in exit air temperature is small enough to permit the assumption of constant air preheater exit air temperature. This temperature is simply related to the atmospheric temperature by the addition of constants. The air preheater equations are:

\[ T_{apao} = K_{tat} + K_{tahad} + K_{tapad} \]
Momentum:

\[ P_{fdo} - P_{fn} = K_{fap} W_{ar}^2 \]

**FURNACE**

Radiation dominates the heat transfer mechanism in the furnace. Heat is radiated both to the waterwall and primary superheater. The net heat transfer is equal to the decrease in the thermal capacity of the gas in the furnace \([IV.12]\). The furnace equations are:

**Adiabatic Flame Temperature \(T_{fnl} \)** \([20,12]\)

\[
W S_{fn} (T_{fnl} - K) = W S_{ar} (T_{ar} - K) + W S_{gr} (T_{gr} - K) + W_{f1} S_{f1} (T_{f1} - K) + W_{f1} K_{hf1} E_{f1}
\]

\[ W_g = W_{ar} + W_{f1} + W_{gr} \]

\[ T_{ar} = T_{apa0} \]

\[ K_{to} = \text{Reference temperature} = 77^\circ F \]

The recirculation gas temperature \(T_{gr} \) is close to the economizer exit gas temperature \(T_{eco} \). The specific heat of gas depends on both temperature and water-to-gas ratio \((Y_{wgr}) \) \([12]\). At temperatures of the order of the furnace temperature, the dependence on water-to-gas ratio is dominant.

\[ S_{fn} = K_{1sf} + K_{2sf} Y_{wgr} \]
Heat Balance:

\[ W_g S_{fn} (T_{fnl} - T_{wwgo}) = U_{wwgm} (T_{wwge}^4 - T_{wwm}^4) + K_{upsr} (T_{wwge}^4 - T_{pse}^4) \]

\[ T_{wwge} = \frac{1}{2} (T_{fnl} + T_{wwgo}) \]

\[ U_{wwgm} = K_{wwgm} U_{xgg} U_{ngg} \]

\[ U_{xgg} = K_{1xgg} + K_{2xgg} T \frac{1}{an} \]

\[ U_{ngg} = \left( N_{g1} K_{1ng} + N_{g2} K_{2ng} + N_{g3} K_{3ng} + N_{g4} K_{4ng} + N_{g5} K_{5ng} \right)/(N_{gg} K_{xwwe}) \]

\[ N_{gg} = N_{g1} + N_{g2} + N_{g3} + N_{g4} + N_{g5} \]

\[ U_{xgg} \text{ and } U_{ngg} \text{ incorporate the effect of burner tilt and number of operating guns respectively. The expressions for } U_{xgg} \text{ and } U_{ngg} \text{ are derived in Section A.4.} \]

**PRIMARY SUPERHEATER GAS**

For the primary superheater flue gas and flue gas in general, the specific heat is not a constant but a function of temperature and water content [12]. The description of the gas path is governed basically by momentum and energy balance:

**Momentum:**

\[ P_{fn} - P_{psgo} = K_{fpsg} W_g^2 \]
Energy:

\[ W_g \cdot S_{psg} (T_{wwgo} - T_{psgo}) = Q_{psc} \]

\( Q_{psc} \) is given in the section on primary superheater and it is the heat transferred from the flue gas to the super heater steam. The dependence of the specific heat to gas temperature and water-gas ratio is modeled using data from Ref. 12 by the following:

\[ S_{psg} = K_{osg} + k_{lsg} Y_{wgr} + (K_{sgr} + K_{sgw} Y_{wgr})(T_{wwgo} + T_{psgo}) \]

\[ T_{psge} = \frac{1}{2} (T_{wwgo} + T_{psgo}) \]

SECONDARY SUPERHEATER GAS

The description here is similar to that of the primary superheater gas:

Momentum:

\[ P_{psgo} - P_{ssgo} = K_{fssg} W_g^2 \]

Energy:

\[ W_g \cdot S_{ssg} (T_{psgo} - T_{ssgo}) = Q_{ss} \]

\( Q_{ss} \) is given in the section on secondary superheater.

\[ S_{ssg} = K_{osg} + k_{lsg} Y_{wgr} + (K_{sgr} + K_{sgw} Y_{wgr}) (T_{psgo} + T_{ssgo}) \]

\[ T_{ssge} = \frac{1}{2} (T_{psgo} + T_{ssgo}) \]
REHEATER GAS

The description here is again similar to that of the primary superheater gas:

Momentum:

\[ P_{ssgo} - P_{rhgo} = K_{frhgo} W_{g}^2 \]

Energy:

\[ W_{g} S_{rhgo} (T_{ssgo} - T_{rhgo}) = Q_{rhgo} \]

\[ S_{rhgo} = K_{osg} + K_{lsg} Y_{wgr} + (K_{sgt} + K_{sgw} Y_{wgr}) (T_{ssgo} + T_{rhgo}) \]

\[ T_{rhgo} = \frac{1}{2} (T_{ssgo} + T_{rhgo}) \]

\[ Q_{rhgo} \text{ is given in the section on re heater.} \]

ECONOMIZER GAS

This description is also similar to that of the primary superheater gas:

Momentum:

\[ P_{rhgo} - P_{ecgo} = K_{fecgo} W_{g}^2 \]

Energy:

\[ W_{g} S_{ecgo} (T_{rhgo} - T_{ecgo}) = Q_{ecgo} \]

\[ S_{ecgo} = K_{osg} + K_{lsg} Y_{wgr} + (K_{sgt} + K_{sgw} Y_{wgr}) (T_{rhgo} + T_{ecgo}) \]
\[ T_{ecge} = \frac{1}{2} (T_{rhgo} + T_{ecgo}) \]

\( Q_{rh} \) is given in the section on Economizer.

**AIR PREHEATER GAS**

At the economizer exit some gas is extracted for gas re-circulation so that the flue gas flow to the air preheater is given by:

\[ W = W_{go} - W_{gr} \]

**Momentum:**

\[ P_{ecgo} - P_{apgo} = K_{r}^{2} \frac{W_{go}^{2}}{N_{go}} \]

**INDUCED DRAFT FANS**

Two centrifugal fans draft flue gas to the atmosphere. The flow rate is controlled to maintain a constant pressure in the furnace by adjusting the inlet vane position. These fans are driven by induction motors. The equations describing the induced draft fans are similar to those for the forced draft fans. The equations summarizing performance are:

**Momentum:**

\[ K_{j}^{2} \frac{d N_{id}}{dt} = T_{idl} - \frac{W_{id} (P_{ido} - P_{apgo})}{n_{id} P_{id} N_{id}} \]

\[ W_{id} = \frac{W_{go}}{K_{nid}} \]

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\[ \rho_{id} = 0.044 \]

**Fan Characteristics:**

\[
\frac{P_{ido} - P_{apgo}}{A_{v1}} = K_{lid} \left( \frac{W_{id}}{A_{v1}} \right)^{2} + K_{2id} \frac{N_{id} W_{id}}{A_{v1}} + K_{3id} N_{id}^{2}
\]

\[ \eta_{id} = K_{4id} \frac{W_{id}^{2}}{V_{id}} + K_{5id} N_{id} W_{id} + K_{5id} N_{id}^{2} \]

where \('A_{v1}'\) models the normalized inlet vane position. Density is assumed constant and is incorporated in \(K_{lid} - K_{6id}\).

**Induction Motor Characteristics:**

\[
T_{idl} = \frac{K_{idm} V^2}{S_{id} + S_{idmax}}
\]

\[
S_{id} = \frac{N_{idmax} - N_{id}}{N_{idmax}}
\]

\(S_{idmax}\) = slip for maximum torque

\(N_{idmax}\) = synchronous speed of motor

\[
N_{elec} = \frac{N_{elec}}{4}
\]

The state variable associated with the induced draft fan is the speed (\(N_{id}\)).

**STACK**

The equation for the stack is:

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Momentum:

\[ P_{\text{id}} - K_{\text{pat}} = K_{\text{fist}} \frac{W^2}{\rho_{\text{gas}}} - K_{\text{pst}} \]

where \( K_{\text{pst}} \) is the stack draft, assumed constant, and \( K_{\text{pat}} \) is the atmospheric pressure, also assumed constant.

A.3 Manipulation of Governing Equations

Some of the equations presented above are not in the form suitable for direct digital computer simulation. Rearrangement and combination of equations are necessary. Illustrations of this technique are given below.

**DRUM AND DEAERATOR EQUATIONS**

**Drum:**

The governing equations for the drum (Section A.2) are:

\[
\frac{d}{dt} \left[ \rho_{\text{drs}} V_{\text{drs}} + \rho_{\text{drw}} V_{\text{drw}} \right] = W_{\text{wwo}} - (W_{\text{rw}} - W_{\text{fw}}) - W_{\text{drs}} - W_{\text{drbd}}
\]

\[
\frac{d}{dt} \left[ \rho_{\text{drs}} V_{\text{hrs}} + \rho_{\text{drw}} V_{\text{drw}} h_{\text{drw}} \right] = W_{\text{wwo}} h_{\text{wo}} - (W_{\text{rw}} - W_{\text{fw}}) h_{\text{drw}}
\]

\[ - W_{\text{hrs}} h_{\text{hrs}} - W_{\text{drbd}} h_{\text{drs}} \]

Put,

\[ W_{\text{wwo}} - (W_{\text{rw}} - W_{\text{fw}}) - W_{\text{drs}} - W_{\text{drbd}} = Z206 \]

and

\[ W_{\text{wwo}} h_{\text{wo}} - (W_{\text{rw}} - W_{\text{fw}}) h_{\text{drw}} - W_{\text{hrs}} h_{\text{hrs}} - W_{\text{drbd}} h_{\text{hrs}} = Z209 \]
\[
\begin{align*}
\dot{\rho}_{\text{drs}} & \frac{d V_{\text{dr}}}{d t} + V_{\text{dr}} \frac{d \rho_{\text{drs}}}{d t} + \rho_{\text{dr}} \frac{d V_{\text{dw}}}{d t} + V_{\text{dw}} \frac{d \rho_{\text{drw}}}{d t} = Z206 \ldots (1) \\
\rho_{\text{hrs}} V_{\text{hrs}} \frac{d h_{\text{hrs}}}{d t} + \rho_{\text{hrs}} h_{\text{hrs}} \frac{d V_{\text{hrs}}}{d t} + V_{\text{hrs}} \frac{d \rho_{\text{hrs}}}{d t} + \rho_{\text{drw}} V_{\text{drw}} \frac{d h_{\text{drw}}}{d t} \\
+ \rho_{\text{drw}} h_{\text{drw}} \frac{d V_{\text{drw}}}{d t} + V_{\text{drw}} h_{\text{drw}} \frac{d \rho_{\text{drw}}}{d t} = Z209 \ldots (2)
\end{align*}
\]

From the state relations (Section A.5):

\[
\begin{align*}
\rho_{\text{drw}} &= K_1 + K_2 \rho_{\text{hrs}} + K_2^2 \rho_{\text{hrs}} \\
h_{\text{drw}} &= K_4 + K_5 \rho_{\text{hrs}} + K_6^2 \rho_{\text{hrs}} \\
h_{\text{hrs}} &= K_7 + K_8 \rho_{\text{hrs}} + K_9^2 \rho_{\text{hrs}}
\end{align*}
\]

also,

\[
V_{\text{hrs}} = K_{\text{vdr}} - V_{\text{drw}}
\]

\[
\begin{align*}
\dot{\rho}_{\text{drw}} &= (K_2 + 2K_3 \rho_{\text{hrs}}) \frac{d \rho_{\text{hrs}}}{d t} = Z201 \frac{d \rho_{\text{hrs}}}{d t} \ldots (3) \\
\frac{d h_{\text{drw}}}{d t} &= (K_5 + 2K_6 \rho_{\text{hrs}}) \frac{d \rho_{\text{hrs}}}{d t} = Z202 \frac{d \rho_{\text{hrs}}}{d t} \ldots (4) \\
\frac{d \rho_{\text{hrs}}}{d t} &= (K_8 + 2K_9 \rho_{\text{hrs}}) \frac{d \rho_{\text{hrs}}}{d t} = Z203 \frac{d \rho_{\text{hrs}}}{d t} \ldots (5)
\end{align*}
\]
Substituting for Equations (3), (4), (5) and (6) in Equations (1) and (2):

From (1),

\[
Z_{204} \frac{d V_{\text{drw}}}{dt} + Z_{205} \frac{d \rho_{\text{drs}}}{dt} = Z_{206}
\]  

(7)

where:

\[
Z_{204} = \rho_{\text{drw}} - \rho_{\text{drs}}
\]

\[
Z_{205} = V_{\text{drw}} Z_{201} + V_{\text{drs}}
\]

From (2)

\[
Z_{207} \frac{d V_{\text{drw}}}{dt} + Z_{208} \frac{d \rho_{\text{drs}}}{dt} = Z_{209}
\]  

(8)

where:

\[
Z_{208} = \rho_{\text{drs}} h_{\text{drw}} - \rho_{\text{drs}} h_{\text{drs}}
\]

\[
Z_{209} = V_{\text{drs}} h_{\text{hrs}} + V_{\text{drw}} h_{\text{hrs}} Z_{201} + V_{\text{drw}} \rho_{\text{drw}} Z_{202}
\]

\[
+ V_{\text{drs}} \rho_{\text{hrs}} Z_{203}
\]

Solving Equations (7) and (8) simultaneously:

\[
\frac{d V_{\text{drw}}}{dt} = \frac{(Z_{206} * Z_{208} - Z_{205} * Z_{209})}{Z_{210}}
\]  

(9)
\[ \frac{dP_{\text{dr}}}{dt} = \frac{(Z204 \times Z209 - Z206 \times Z207)}{Z210} \] 

(10)

where:

\[ Z210 = Z204 \times Z208 - Z205 \times Z207 \]

Equations (9) and (10) represent the final form in which the drum equations are simulated.

**Deaerator:**

The deaerator equations are treated in a manner similar to the drum equations and the final form of the equations are also similar to the final drum equations. The difference exist in the parameter values and variables used.

**FLOW COMPUTATIONS**

Where pumps and fans are involved, the computation of flows requires the simultaneous solution of the momentum equations of the associated components together with the performance (head-flow) characteristics of the corresponding pumps or fans. The technique involved in this solution is illustrated for condensate, feedwater, recirculating, and air-gas flows.

**Condensate Flow:**

The equations necessary to compute condensate flow are

(Section A.2):
Condensate Pump Head/Flow Characteristics:

\[ P_{\text{cpo}} - P_{\text{cn}} = K_{1\text{cp}} \frac{W^2}{\rho_{\text{cp}}} + K_{2\text{cp}} N W \frac{W}{\rho_{\text{cp}}} + K_{3\text{cp}} \rho_{\text{cp}} N^2 \]  

(1)

Low Pressure Feedwater Heater Momentum Equation:

\[ P_{\text{cpo}} - P_{\text{Lho}} = K_{f1h} \frac{W^2}{\rho_{\text{Lho}}} \]

where:

\[ \rho_{\text{Lho}} = \rho_{\text{cp}} \]

\[ W_{\text{cw}} = K_{\text{ncp}} W_{\text{cp}} \]

\[ . . . P_{\text{cpo}} - P_{\text{Lho}} = K_{fLh} \frac{K_{2\text{ncp}}^2 W^2}{\rho_{\text{cp}}} \]  

(2)

Deaerator valve equation:

\[ P_{\text{Lho}} - P_{\text{des}} = \frac{K_{\text{dv}} W^2}{\rho_{\text{Lho}} A_{\text{dv}}^2} \]

\[ . . . P_{\text{Lho}} - P_{\text{des}} = \frac{K_{\text{dv}} K_{2\text{ncp}} W^2}{\rho_{\text{Lho}} A_{\text{dv}}^2} \]  

(3)

Equation (2) + (3) - (1), lead to:

\[ Z_6 W_{\text{cp}}^2 - Z_2 W_{\text{cp}} - Z_7 = 0 \]  

(4)

where:

\[ Z_6 = \frac{K_{\text{dv}} K_{2\text{ncp}}^2}{\rho_{\text{Lho}} A_{\text{dv}}^2} + \frac{K_{fLh} K_{2\text{ncp}}^2}{\rho_{\text{cp}}} - \frac{K_{1\text{cp}}}{\rho_{\text{cp}}} \]
\[ Z_2 = K_{2cp} N_{cp} \]

\[ Z_7 = K_{3cp} N_{cp}^2 \rho_{cp} + P_{cn} - P_{des} \]

Solving the quadratic equation (4):

\[ W_{cp} = \frac{Z_2 + \sqrt{Z_2^2 + 4Z_6Z_7}}{2Z_6} \]

This expression gives the flow through one condensate pump.

The total condensate flow is obtained by multiplying \( W_{cp} \) by the number of operating condensate pumps \( K_{ncp} \)

**Feedwater Flow:**

The equations necessary to compute feedwater flow are (Section A.2):

**Booster Feedpump Equation:**

\[ P_{bpo} - P_{des} = K_{bp} \frac{W_{fp}^2}{\rho_{fp}} + K_{2bp} N_{bp} W_{fp} + K_{3bp} \rho_{fp} N_{bp}^2 \] (1)

\[ \rho_{fp} = \rho_{dew} \]

\[ N_{bp} = K_{nbpr} N_{fp} \]

**Main Boiler Feedpump Equation:**

\[ P_{fpo} - P_{bpo} = K_{fp} \frac{W_{fp}^2}{\rho_{fp}} + K_{2fp} N_{fp} W_{fp} + K_{3fp} \rho_{fp} N_{fp}^2 \] (2)
Feedwater Valve Equations:

\[ P_{fp0} - P_{fvo} = \frac{K_{fv} W_{fw}^2}{\rho_{fv} A_{fv}^2} \]  \hspace{1cm} (3)

\[ \rho_{fv} = \rho_{fp} \]

\[ W_{fw} = W_{fp} - W_{sy} - W_{ry} - K_{wfp} \]

High Pressure Feedwater Heater Momentum Equations:

\[ \rho_{fvo} - P_{hho} = K_{fhh} \frac{W_{fw}^2}{\rho_{hhl}} \]  \hspace{1cm} (4)

\[ \rho_{hhl} = \rho_{fv} \]

Economizer Momentum Equation:

\[ P_{hho} - P_{drs} = K_{fec} \frac{W_{fw}^2}{\rho_{eco}} \]  \hspace{1cm} (5)

Equations (3) + (4) + (5) - (2) - (1) lead to:

\[ Z_{15} \frac{W_{fp}^2}{\rho_{fp}} - Z_{16} W_{fp} - Z_{17} = 0 \]  \hspace{1cm} (6)

where:

\[ Z_{15} = \frac{K_{fv}}{\rho_{fv} A_{fv}^2} + \frac{K_{fhh}}{\rho_{hhl}} + \frac{K_{fec}}{\rho_{eco}} - \frac{K_{lfp}}{\rho_{fp}} - \frac{K_{lbp}}{\rho_{fp}} \]

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\[ Z_{16} = 2(W_{sy} + W_{ry} + K_{wpfx}) \left( \frac{K_{fv}}{\rho_{fv} A_{fv}^2} + \frac{K_{fhh}}{\rho_{hh1}} + \frac{K_{pec}}{\rho_{pco}} \right) + K_{2fp}N_{fp} + K_{2bp}N_{bp} \]

\[ Z_{17} = K_{3fp} \rho_{fp} N_{fp}^2 + K_{3bp} \rho_{bp} N_{bp}^2 + P_{des} - P_{dne} \]

\[ - (W_{sy} + W_{ry} + K_{wpfx})^2 \left( \frac{K_{fv}}{\rho_{fv} A_{fv}^2} + \frac{K_{fhh}}{\rho_{hh1}} + \frac{K_{pec}}{\rho_{pco}} \right) \]

Solving the quadratic Equation (4)

\[ W_{fp} = (Z_{16} + \sqrt{Z_{16}^2 + 4Z_{15}Z_{17}}) / 2Z_{15} \]

The expression gives the flow through the boiler feedpump.

The feedwater flow is obtained by subtracting the pump extraction flows from \( W_{fp} \). Thus,

\[ W_{fw} = W_{fp} - W_{sy} - W_{ry} - K_{wpfx} \]

**Recirculating Flow:**

The equations necessary to compute the recirculating water flow are:

**Downcomer Momentum Equation:**

\[ P_{dne} - P_{dco} = K_{fpc} \frac{W_{fp}^2}{\rho_{fco}} - K_{1dc} \rho_{dc} g / g_c \]  \( \text{(1)} \)
Recirculating Pump Equation:

\[ P_{\text{rpo}} - P_{\text{dco}} = K_{\text{lrr}} \frac{W_{\text{rr}}^2}{\rho_{\text{rr}}} + K_{2\text{rp}} N_{\text{rp}} W_{\text{rp}} + K_{3\text{rp}} \rho_{\text{rp}} N_{\text{rp}}^2 \]  \hspace{1cm} (2)

\[ \rho_{\text{rr}} = \rho_{\text{dc}} \]

Waterwall Momentum Equation:

\[ P_{\text{rpo}} - P_{\text{drw}} = K_{\text{fww}} \frac{W_{\text{wwo}}^2}{\rho_{\text{drw}}} + K_{\text{lww}} \rho_{\text{drw}} g/g_c \]  \hspace{1cm} (3)

\[ W_{\text{wwo}} = W_{\text{rw}} + K_{\text{wrps}} \]

\[ W_{\text{rw}} = K_{\text{nrr}} W_{\text{rp}} \]

Equations (1) + (3) - (2) lead to:

\[ Z_8 W_{\text{rr}}^2 - Z_9 W_{\text{rp}} - Z_{10} = 0 \]  \hspace{1cm} (4)

where:

\[ Z_8 = \frac{K_{\text{fddc}}}{\rho_{\text{dc}}} + K_{\text{fww}} \frac{K_{2\text{nrp}}}{\rho_{\text{drw}}} - K_{\text{lrr}} \rho_{\text{dc}} \]

\[ Z_9 = K_{2\text{rp}} N_{\text{rp}} - 2K_{\text{nrp}} K_{\text{wrps}} K_{\text{fww}} / \rho_{\text{drw}} \]

\[ Z_{10} = K_{3\text{rp}} N_{\text{rp}}^2 \rho_{\text{dc}} + K_{\text{lwc}} \rho_{\text{dc}} / 144 - K_{\text{lww}} \rho_{\text{drw}} / 144 \]

\[ - K_{\text{wrps}}^2 K_{\text{fww}} / \rho_{\text{drw}} \]

\[ g/g_c = 144 \]
Solving the quadratic equation (4):

\[ W_{rp} = \frac{(z_4 + \sqrt{z_2^2 + 4z_8z_{10}})}{2z_8} \]

This expression gives the flow through one recirculating pump. The recirculating water flow is obtained by multiplying \( W_{rp} \) by the number of operating pumps (\( K_{n_{rp}} \)).

Air-Gas Flow:

The equations necessary to compute air and gas flows are (Section A.2):

Glycol Air Heater Momentum Equation:

\[ K_{pat} - P_{a} = K_{fah} W_{ar}^2 \]  \hspace{1cm} (1)

\[ W_{ar} = K_{nfd} W_{fd} \]

Forced Draft Fans Equation:

\[ P_{fdo} - P_{a} = \frac{K_{1fd} W_{fd}^2}{A_{vf}} + K_{2fd} N_{fd} W_{fd} + K_{3fd} A_{vf} N_{fd}^2 \]  \hspace{1cm} (2)

Air Preheater Momentum Equation:

\[ P_{fdo} - P_{fn} = K_{fap} W_{ar}^2 \]  \hspace{1cm} (3)

Primary Superheater Gas Momentum Equation:

\[ P_{fn} - P_{psgo} = K_{fpsg} W_{g}^2 \]  \hspace{1cm} (4)
\[ W_g = W_{ar} + W_{fl} + W_{gr} \]

Secondary Superheater Gas Momentum Equation:

\[ P_{psgo} - P_{ssgo} = K_{fssg} \frac{W^2}{g} \]  \hspace{1cm} (5)

Reheater Gas Momentum Equation:

\[ P_{ssgo} - P_{rhgo} = K_{frhg} \frac{W^2}{g} \]  \hspace{1cm} (6)

Economizer Gas Momentum Equation:

\[ P_{rhgo} - P_{ecgo} = K_{fecg} \frac{W^2}{g} \]  \hspace{1cm} (7)

Air Preheater Gas Momentum Equation:

\[ P_{ecgo} - P_{apgo} = K_{fapg} \frac{W^2}{go} \]  \hspace{1cm} (8)

\[ W_{go} = W_g - W_{gr} \]

Induced Draft Fan Equation:

\[ P_{ido} - P_{apgo} = \frac{K_{lid} W_{id}^2}{A_{vi}} + K_{2id} N_{id} W_{id} + K_{3id} A_{vi} N_{id}^2 \]

\[ W_{id} = \frac{W_{go}}{K_{nid}} \]

Stack Equation:

\[ P_{ido} - K_{pat} = K_{fst} \frac{W^2}{go} - K_{pst} \]  \hspace{1cm} (10)

Equations (1) - (2) + (3) + (4) + (5) + (6) + (7) + (8) - (9) + (10) lead to:

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\[ Z_{16} W_{fd}^2 + Z_{17} W_{fd} + Z_{18} = 0 \]  

(11)

where:

\[ Z_{16} = K_{fah} + K_{nfp}^2 K_{fap} - K_{lid}/A_vf \]

\[ + K_{nfd}^2 (K_{fg} + K_{fapg} + K_{fst} - K_{lid}/(A_v K_{nid}^2)) \]

\[ K_{fg} = K_{fpsg} + K_{fseg} + K_{grhg} + K_{fecn} \]

\[ Z_{17} = 2K_{nfd} [K_{fg} (W_{fl} + W_{gr}) + W_{fl} (K_{fapg} + K_{fst}) - W_{fl} K_{lid}/(A_v K_{nid}^2)] \]

\[ - K_{nfd} N_{fd} - K_{nid} N_{id} K_{nfd}/K_{nid} \]

\[ Z_{18} = K_{fg} (W_{fl} + W_{gr})^2 + W_{fl}^2 [K_{fapg} + K_{fst} - K_{lid}/(A_v K_{nid}^2)] \]

\[ - K_{nfd} N_{fd} W_f /K_{nid} - K_{3fd} N_{fd}^2 A_v - K_{3id} N_{id}^2 A - K_{psf} \]

Solving the quadratic equation (11)

\[ W_{fd} = \frac{-Z_{17} \pm \sqrt{Z_{17}^2 - 4Z_{16} Z_{18}}}{2Z_{16}} \]

This expression gives the air flow through one forced draft fan. The total air flow is obtained by multiplying \( W_{fd} \) by the number of operating FD fans. The gas flow is obtained by adding fuel and gas recirculation flows to the air flow.
HEAT TRANSFER COMPUTATION

The computation of heat transfer in the gas path involves first of all determining the exit gas temperature for the component under consideration. This temperature is determined by equating the change in gas enthalpy to the heat transfer (steady state energy balance). This technique is illustrated for the furnace and primary superheater. The secondary superheater, reheater and economizer are treated in a manner similar to the primary superheater.

Furnace:

The equations describing the heat transfer in the furnace are (Section A.2):

Heat Balance:

\[ W_{g \, fn} (T_{fnl} - T_{wwgo}) = U_{wwgm} (T_{wwge}^4 - T_{wgm}^4) \]

\[ + K_{upsr} (T_{wwge}^4 - T_{pse}^4) \]  \hspace{1cm} (1)

But,

\[ T_{wwge} = \frac{1}{2} (T_{fnl} + T_{wwgo}) \]

\[ T_{wwgo} = 2 \, T_{wwge} - T_{fnl} \]

Substituting for \( T_{wwgo} \) in Equation (1):

\[ W_{g \, fn} (T_{fnl} - 2 \, T_{wwge} + T_{fnl}) = T_{wwge}^4 (U_{wwgm} + K_{upsr}) \]

\[ - U_{wwgm} T_{wwm}^4 - K_{upsr} T_{pse}^4 \]  \hspace{1cm} -186-
\[ z_1 T_{\text{wwge}}^4 + z_2 T_{\text{wwge}} - z_3 = 0 \]  \hspace{1cm} (2)

where:

\[ z_1 = \frac{U_{\text{wwgm}}}{w_{\text{ff}}} + K_{\text{uag}} \]
\[ z_2 = 2W_g S_{\text{fn}} \]
\[ z_3 = U_{\text{wwgm}} T_{\text{wmm}}^4 + K_{\text{uag}} T_{\text{pse}}^4 + 2W_g S_{\text{fn}} T_{\text{fnl}} \]

Equation (2) is a quartic in \( T_{\text{wwge}} \) and the solution is:

\[ T_{\text{wwge}} = \frac{1}{2} (\sqrt{2z_{12} - z_{11}} - \sqrt{z_{11}}) \]  \hspace{1cm} (3)

where:

\[ z_{11} = \sqrt{z_6 + z_8} - \sqrt{z_6 - z_8} \]
\[ z_{12} = \sqrt{z_{11}^2 + 4z_5} \]

and

\[ z_4 = z_2 / z_1 \]
\[ z_5 = z_3 / z_1 \]
\[ z_6 = \frac{1}{2} z_4^2 \]
\[ z_7 = 4z_5 / 3 \]
\[ z_8 = \sqrt{z_7 + z_6^2} \]
Equation (3) gives the average furnace temperature, and with this temperature known, the outlet temperature and heat transfer can be computed.

\[
T_{wwgo} = 2T_{wwge} - T_{fnl}
\]

\[
Q_{wwgm} = U_{wwgm} (T_{wwge}^4 - T_{wwm}^4)
\]

\[
Q_{psr} = K_{upsr} (T_{wwge}^4 - T_{pse}^4)
\]

**Primary Superheater:**

The equations describing the heat transfer to the primary superheater are (Section A.2):

\[
W_{sg} (T_{wwgo} - T_{psgo}) = Q_{psc}
\]  
(1)

\[
Q_{ps} = Q_{psc} + Q_{psr}
\]

\[
Q_{psc} = U_{psgs} (T_{psge} - T_{pse})
\]

\[
U_{psgs} = \frac{K_{upsms} W_{0.6} + K_{upsms} W_{0.8}}{W_{0.6} + W_{0.8}}
\]

\[
T_{psge} = \frac{1}{2} (T_{wwgo} + T_{psgo})
\]

\[
\therefore Q_{ps} = Q_{psr} + U_{psgs} \left[ \frac{1}{2} (T_{wwgo} + T_{psgo}) - T_{psgo} \right]
\]  
(2)
Also,
\[ S_{psg} = Z_1 + Z_2 (T_{wwgo} + T_{psgo}) \]  
(3)

where:
\[ Z_1 = K_{osg} + K_{lsg} Y_{wgr} \]
\[ Z_2 = K_{sgt} + K_{sgw} Y_{wgr} \]

Substituting for \( Q_{ps} \) and \( S_{psg} \) in (1), leads to:
\[ Z_5 T_{psgo}^2 + Z_6 T_{psgo} + Z_7 = 0 \]  
(4)

where:
\[ Z_5 = W_g Z_2 \]
\[ Z_6 = \frac{1}{2} U_{psgs} + W_g Z_1 \]
\[ Z_7 = \frac{1}{2} U_{psgs} T_{wwgo} - U_{psgs} T_{pse} - Z_5 T_{wwgo}^2 - Z_1 W_g T_{wwgo} \]

Solving the quadratic equation (4)
\[ T_{psgo} = \frac{-Z_6 \pm \sqrt{Z_6^2 - 4Z_5Z_7}}{2Z_5} \]  
(5)

Equation (5) gives the primary superheater gas outlet temperature and with this temperature known the heat transfer in the primary superheater can be computed.
\[ Q_{ps} = Q_{psr} + W_g S_{psg} (T_{wwgo} - T_{psgo}) \]
A.4 Correction Factors for Burner Tilt and Non-Operating Burners

The effects of burner tilt and non-operating guns are accounted for by assuming that the effective furnace length changes with burner tilt and non-operating guns. This change in effective length of the furnace is then directly related to the effective heat transfer coefficient of the furnace. The effective length of the furnace is determined by the location of the flame ball.

**BURNER TILT**

A factor $U_{xg}$ is used to account for the effect of the burner tilt. The furnace flame ball moves with the tilt of the burners.

Assuming an effective length of furnace, $K_{xwe}$ (ft), when there is no tilt, and a movement of the flame ball through $\Delta L$ (ft) when there is a burner tilt of $X_{gg}$ radians; then the factor $U_{xg}$ is given by:

$$U_{xg} = \frac{K_{xwe} - \Delta L}{K_{xwe}} = 1 - \frac{\Delta L}{K_{xwe}}$$

from the geometry of Figure A.1, it follows that:

$$\Delta L = \frac{1}{2} \frac{D}{\tan(X_{gg})}$$

where, 'D' is the length of the furnace cross-sectional diagonal.

$$\therefore U_{xg} = 1 - \frac{D}{4K_{xwe}} \tan(X_{gg})$$

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FIGURE A.1: BURNER TILT CORRECTION FACTOR
But, \( D = 72.3 \) ft

and, \( K_{\text{xwwe}} = 64.5 \)

\[
\therefore U_{\text{ngg}} = 1 - 0.28648 \tan(X_{gg})
\]

\[
\therefore K_{1\text{ngg}} = 1.0
\]

\[
\therefore K_{2\text{ngg}} = -0.286
\]

**NON-OPERATING BURNERS (GUNS)**

The effect of non-operating guns is also accounted for in terms of the location of the flame ball. A factor \( U_{\text{ngg}} \) is used to account for this effect. This factor is determined as a ratio of the effective furnace length to the effective furnace length when all the guns in all the levels are operating. The effective length of the furnace is computed in terms of an equivalent center of mass of the operating guns.

With reference to Figure A.2, taking moments of operating guns about '0', \( U_{\text{ngg}} \) is given by:

\[
U_{\text{ngg}} = \left( N_{g1}K_{1-ngg} + N_{g2}K_{2-ngg} + N_{g3}K_{3-ngg} + N_{g4}K_{4-ngg} + N_{g5}K_{5-ngg} \right) / (N_{\text{gg}}K_{\text{xwwe}})
\]

where:

\[
N_{\text{gg}} = N_{g1} + N_{g2} + N_{g3} + N_{g4} + N_{g5}
\]

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FIGURE A.2: NON-OPERATING GUNS CORRECTION FACTOR

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$N_g^1, N_g^2, N_g^3, N_g^4, N_g^5$ are the numbers of operating guns at levels 1, 2, 3, 4, 5 respectively, and $N_g$ is the total number of operating guns.

A.5 State Relations

The thermodynamic constitutive relations used in the model are summarized in the following paragraphs:

**Drum**

\[
\rho_{d_{rw}} = 49.271 - 2.137\rho_{d_{rs}} + 0.0335\rho_{d_{rs}}^2
\]

\[
h_{d_{rw}} = 526.596 + 31.044\rho_{d_{rs}} - 0.621\rho_{d_{rs}}^2
\]

\[
h_{d_{rs}} = 1241.71 - 21.34\rho_{d_{rs}} + 0.210\rho_{d_{rs}}
\]

\[
P_{d_{rs}} = 11.188 + 500.27\rho_{d_{rs}} - 26.403\rho_{d_{rs}}^2 + 0.469\rho_{d_{rs}}^3
\]

\[
T_{d_{rs}} = 458.08 + 48.209\rho_{d_{rs}} - 3.233\rho_{d_{rs}}^2 + 0.0725\rho_{d_{rs}}^3 + 459.67
\]

**Recirculation Section: Downcomer**

\[
h_{d_{c}} = 658.45 + 11.887\rho_{d_{c}} + 3.950E-2 P_{d_{c}}
\]

\[
- 5.670E-4\rho_{d_{c}} P_{d_{c}} - 0.317\rho_{d_{c}}^2 - 3.297E-7 P_{d_{c}}^2
\]

\[
T_{d_{c}} = 163.60 + 29.199\rho_{d_{c}} + 4.268E-2 P_{d_{c}}
\]

\[
- 7.263E-4\rho_{d_{c}} P_{d_{c}} - 0.457\rho_{d_{c}}^2 + 3.183E-7 P_{d_{c}}^2
\]

**Superheater Section: Primary and Secondary Superheaters**

\[
P_{pso} (P_{sso}) = -291.36 - 964.04\rho_{p_{sso}} (\rho_{sso})
\]

\[
+ 0.218\rho_{p_{sso}} (h_{sso}) + 1.182\rho_{p_{sso}} h_{sso} (\rho_{sso} h_{sso})
\]

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\[ T_{\text{ps}0} (T_{\text{ss}0}) = -1745.1 + 129.1P_{\text{ps}0} (\rho_{\text{ss}0}) \]
\[ + 1.811h_{\text{ps}0} (h_{\text{ss}0}) = 0.0663P_{\text{ps}0} h_{\text{ps}0} (\rho_{\text{ss}0} h_{\text{ss}0}) + 459.67 \]
\[ S_{\text{ps}0} (S_{\text{ss}0}) = 1.3136 - 1.7799E-3 \rho_{\text{ps}0} (\rho_{\text{ss}0}) \]
\[ + 6.357E-4h_{\text{ps}0} (h_{\text{ss}0}) - 8.359E-2 \log(\rho_{\text{ps}0} h_{\text{ps}0})(\rho_{\text{ss}0} h_{\text{ss}0}) \]

Superheat Spray Section

\[ \rho_{\text{ss}1} = -1.903 + 1.386E - 3h_{\text{ss}1} + 6.757E-3P_{\text{ps}0} - 3.766E-6 P_{\text{ps}0} h_{\text{ss}1} \]
\[ T_{\text{ss}1} = -1654.5 + 1.7443h_{\text{ss}1} + 0.348P_{\text{ps}0} \]
\[ - 2.074E-4 P_{\text{ps}0} h_{\text{ss}1} + 459.67 \]

High Pressure Turbine

\[ \eta_{\text{isen}} = 0.589 + 2.317E-4 W_{\text{hp}} = E_{\text{hp}} \]
\[ h_{\text{hpol}} = -485.23 + 1065.28 S_{\text{ss}0} \]
\[ + 0.232 P_{\text{hpo}} \]
\[ T_{\text{hpo}} = -1639.24 + 0.119 P_{\text{hpo}} \]
\[ + 1.682 h_{\text{hpo}} + 459.67 \]

Reheat Spray

\[ \rho_{\text{rh}1} = -4.266E-2 + 3.089E-5h_{\text{rh}1} + 6.292E-3 P_{\text{hpo}} \]
\[ - 3.489E-6 P_{\text{hpo}} h_{\text{rh}1} \]
\[ T_{\text{rh}1} = -550.34 - 0.394 h_{\text{rh}1} + 1.680P_{\text{hpo}} \]
\[ -1.136E-3 P_{\text{hpo}} h_{\text{rh}1} + 9.361E-4 h_{\text{rh}1} \]
\[ -4.125E-4 P_{\text{hpo}}^2 + 2.085E-10 P_{\text{hpo}}^2 h_{\text{rh}1} + 459.67 \]
\[ -195- \]
Reheater

\[ P_{\text{rho}} = 27.061 - 1019.1 \rho_{\text{rho}} - 1.735 \times 10^{-2} h_{\text{rho}} + 1.228 \rho_{\text{rho}} h_{\text{rho}} \]

\[ T_{\text{rho}} = -2013.4 + 189.65 \rho_{\text{rho}} + 1.963 h_{\text{rho}} - 9.305 \times 10^{-2} \rho_{\text{rho}} h_{\text{rho}} + 459.67 \]

\[ S_{\text{rho}} = 1.5015 + 3.8306 \times 10^{-3} \rho_{\text{rho}} + 6.4181 \times 10^{-4} h_{\text{rho}} - 0.1094 \log(\rho_{\text{rho}} h_{\text{rho}}) \]

Crossover Pipe

\[ \eta_{\text{isen}} = 0.814 = E_{\text{ip}} \]

\[ h_{\text{ipoi}} = -1211.8 + 683.58 \rho_{\text{cro}} + 1384.39 S_{\text{rho}} \]

\[ h_{\text{ipo}} = h_{\text{ipoi}} - E_{\text{ip}} (h_{\text{rho}} - h_{\text{ipoi}}) \]

\[ h_{\text{cro}} = h_{\text{ipo}} \]

\[ P_{\text{cro}} = -381.05 + 0.2783 h_{\text{cro}} + 668.61 \rho_{\text{cro}} \]

\[ T_{\text{cro}} = -2074.92 + 2.004 h_{\text{cro}} + 71.326 \rho_{\text{cro}} + 459.67 \]

Condenser

\[ \rho_{\text{ctw}} = 62.345 - 0.289 P_{\text{cn}} \]

\[ \rho_{\text{cns}} = 2.0 \times 10^{-5} + 3.21 \times 10^{-3} P_{\text{cn}} - 3.2 \times 10^{-4} P_{\text{cn}}^2 + 8.0 \times 10^{-5} P_{\text{cn}}^3 \]

\[ h_{\text{cnw}} = -25.803 + 365.636 P_{\text{cn}} - 878.108 P_{\text{cn}}^2 + 1305.20 P_{\text{cn}}^3 - 1002.14 P_{\text{cn}}^2 + 305.01 P_{\text{cn}}^5 \]

\[ h_{\text{cns}} = 1075.87 + 29.00 P_{\text{cn}} - 196\]
\[ T_{cn} = 6.292 + 364.609 P_{cn} - 866.922 P_{cn}^2 + 1311.46 P_{cn}^3 \]
\[ - 1012.18 P_{cn}^4 + 309.494 P_{cn}^5 + 459.67 \]

\[ h_{1po} = h_{cnw} + x(h_{cns} - h_{cnw}) \]

\[ \rho_{1po} = \rho_{cnw} + x(\rho_{cns} - \rho_{cnw}) \]

**Condensate Pump**

\[ h_{cpo} = 2668.5 + 24.790 \rho_{cno} - 1.738P_{cpo} + 0.0273P_{cpo} \rho_{cno} \]
\[ - 1.075\rho_{cno}^2 + 1.027E-4P_{cpo}^2 \]

\[ T_{cpo} = -7918.5 + 366.86\rho_{cno} - 1.734P_{cpo} + 0.0271P_{cpo} \rho_{cno} \]
\[ - 3.829\rho_{cno}^2 + 1.044E-4P_{cpo}^2 + 459.67 \]

**Low Pressure Feedwater Heater**

\[ \rho_{1ho} = 62.633 + 2.309E-4P_{1ho} - 7.847E-3h_{1ho} \]
\[ + 8.453E-7P_{1ho} h_{1ho} - 1.308E-7P_{1ho}^2 - 4.417E-5 h_{1ho}^2 \]

\[ T_{1ho} = 31.363 - 3.680E-3P_{1ho} + 1.022h_{1ho} \]
\[ + 2.258E-6P_{1ho} h_{1ho} + 1.311E-6P_{1ho}^2 \]
\[ - 9.836E-5h_{1ho}^2 + 459.67 \]

**Deaerator**

\[ \rho_{dew} = 60.458 - 19.612\rho_{des} \]

\[ h_{dew} = 118.263 + 1905.65\rho_{des} - 8414.69\rho_{des}^2 + 15688.04\rho_{des}^3 \]

\[ h_{des} = 1143.50 + 224.56\rho_{des} \]
\[ -197- \]
\[ P_{\text{des}} = -1.042 + 419.172P_{\text{des}} + 131.328P_{\text{des}}^2 \]

\[ T_{\text{des}} = 150.489 + 1896.42P_{\text{des}} - 8447.88P_{\text{des}}^2 + 15757.43P_{\text{des}}^3 + 459.67 \]

**Boiler Feedpump**

\[ h_{\text{fpo}} = 9024.9 + 0.1148P_{\text{fpo}} - 90.346P_{\text{dew}} - 2.053E5/P_{\text{dew}} \]

\[ T_{\text{fpo}} = -1268.8 + 82.714P_{\text{dew}} + 0.1030P_{\text{fpo}} - 1.474E-3P_{\text{fpo}}P_{\text{dew}} \]

\[ -0.9707P_{\text{dew}}^2 - 1.801E-6P_{\text{fpo}}^2 + 459.67 \]

**Feedwater:** High Pressure Feedwater Heater and Economizer

\[ \rho_{\text{hho}}(\rho_{\text{eco}}) = 71.143 - 5.090E-3\rho_{\text{hho}}(P_{\text{drs}}) \]

\[ -3.266E-2\rho_{\text{hho}}(\rho_{\text{eco}}) + 1.169E-5\rho_{\text{hho}}(\rho_{\text{eco}}P_{\text{drs}}) \]

\[ + 6.166E-7P_{\text{hho}}^2(\rho_{\text{eca}}^2) - 3.063E-5P_{\text{hho}}^2(\rho_{\text{eca}}^2) \]

\[ -2.310E-12P_{\text{hho}}^2(\rho_{\text{eca}}^2P_{\text{drs}}^2) \]

\[ T_{\text{hho}}(T_{\text{eco}}) = 146.98 - 0.0885P_{\text{hho}}(P_{\text{drs}}) + 0.7972P_{\text{hho}}(\rho_{\text{eco}}) \]

\[ + 1.803E-4P_{\text{hho}}(\rho_{\text{eca}}P_{\text{drs}}) + 1.048E-5P_{\text{hho}}^2(P_{\text{drs}}^2) \]

\[ - 1.823E-4P_{\text{hho}}^2(\rho_{\text{eca}}^2) - 3.812E-11P_{\text{hho}}^2(\rho_{\text{eca}}^2P_{\text{drs}}^2) \]

\[ + 459.67 \]

**Low Pressure Feedwater Heater Extraction Flow**

\[ h_{31\text{hso}} = 32.5 + 43.498P_{31\text{hs}} - 7.380P_{31\text{hs}}^3 + 0.5147P_{31\text{hs}}^3 \]

\[ \rho_{31\text{hso}} = 62.345 - 0.2888P_{31\text{hs}} \]

-198-
\[ T_{3\text{hs}} = 64.059 + 43.602 P_{3\text{hs}} - 7.402 P^2_{3\text{hs}} + 0.5162 P^3_{3\text{hs}} + 459.67 \]

**High Pressure Feedwater Heater Extraction Flow**

\[ h_{3\text{hs}} = 126.874 + 4.154 P_{3\text{hs}} - 0.0422 P^2_{3\text{hs}} + 1.8E-4 P^3_{3\text{hs}} \]
\[ \rho_{3\text{hs}} = 60.532 - 0.0460 P_{3\text{hs}} \]
\[ T_{3\text{hs}} = 159.096 + 4.129 P_{3\text{hs}} - 0.0424 P^2_{3\text{hs}} + 1.8E-4 P^3_{3\text{hs}} + 459.67 \]

A.6 Steam Extraction

The relationships used in steam extraction computations are:

FLOW, PRESSURE, ENTHALPY

**Blowdown Flow to Drum**

\[ W_{\text{drbd}} = 0.3266 + 1.464E-3 W_{\text{drs}} + 8.924E-5 W_{\text{drs}}^2 \]
\[ - 5.020E-9 W_{\text{drs}}^3 \]

**Extraction to Auxiliary Plant Uses**

If \( W_{tv} < 507 \), Primary Superheater Extraction

\[ W_{\text{psx}} = 101.56 - 0.3988 W_{\text{drs}} + 4.863 - 4 W_{\text{drs}}^2 \]
\[ W_{\text{hpaux}} = 0 \]

If \( W_{tv} > 507 \), High Pressure Turbine Extraction

\[ W_{\text{psx}} = 0 \]
\[ W_{\text{hpaux}} = 17.158 + 1.08E-2 W_{\text{hp}} \]

-199-
Extraction to Boiler Feedpump Turbine

If $W_{tv} < 507$, Secondary Superheater Extraction

\[ W_{ssx} = W_{ft} \]

\[ W_{ipftx} = 0 \]

If $W_{tv} > 507$, Intermediate Pressure Turbine Extraction

\[ W_{ssx} = 0 \]

\[ W_{ipftx} = W_{ft} \]

High Pressure Turbine Extraction

To High Pressure Feedwater Heater

\[ W_{1hhs} = -7.176 + 0.0951W_{hp} - 8.664E-5W_{hp}^2 \]
\[ + 6.187E-8W_{hp}^3 \]

\[ W_{hpaux} = 17.16 - 0.0108 W_{hp} \]

\[ W_{hp} = W_{1hhs} + W_{hpaux} \]

Intermediate Pressure Turbine Extraction

To High Pressure Feedwater Heater

\[ W_{2hhs} = 1.725 + 2.727E-2W_{ip} + 2.780E-5W_{ip}^2 \]

\[ W_{3hhs} = -22.887 + 0.176W_{ip} - 1.937E-4W_{ip}^2 \]
\[ + 1.048E-7W_{ip}^3 \]

\[ P_{2hhs} = 0.450(P_{ro} + P_{cro}) \]
\[ -200- \]
\[ P_{3\text{hhs}} = 32.435 - 0.4506P_{\text{cro}} + 6.852E^{-3}P_{\text{cro}}^2 - 1.647E^{-5}P_{\text{cro}}^3 \]

\[ h_{2\text{hhs1}} = 0.503(h_{\text{rho}} + h_{\text{cro}}) \]

\[ h_{3\text{hhs}} = h_{\text{cro}} \]

**Low Pressure Turbine Extraction**

**To Low Pressure Feedwater Heater**

\[ W_{1\text{hs}} = -1.048 + 9.980E^{-2}W_{1p} - 6.825E^{-5}W_{1p}^2 + 6.14E^{-8}W_{1p}^3 \]

\[ W_{2\text{hhs}} = -6.738 + 8.488E^{-2}W_{1p} - 1.344E^{-5}W_{1p}^2 0.136 (P_{\text{cro}} + P_{\text{cn}}) \]

\[ P_{1\text{hs}} = 0.0294 (P_{\text{cro}} + P_{\text{cn}}) \]

\[ h_{1\text{hs}1} = 0.499(h_{\text{cro}} + h_{\text{lopo}}) \]

\[ h_{2\text{hs}1} = 0.458(h_{\text{cro}} + h_{\text{lopo}}) \]

**To Deaerator**

\[ W_{\text{dex}} = 4.970 + 1.757E^{-2}W_{1p} + 3.062E^{-5}W_{1p}^2 \]

\[ P_{\text{dex}} = 0.630 - 7.049E^{-2}P_{1\text{hs}1} + 6.341E^{-3}P_{1\text{hs}1}^2 - 1.298E^{-4}P_{1\text{hs}1}^3 \]

\[ h_{\text{dex}} = 0.534(h_{\text{cro}} + h_{\text{lopo}}) \]


Flows from Auxiliary Plant Uses to Deaerator

\[ W_{derp} = 15.10 \]
\[ W_{dewh} = 8.455 + 2.820E-3 \ W_{hp} - 8.58 - 7W_{hp}^2 \]
\[ W_{debd} = 0.163 + 7.319E-4 \ W_{drs} - 4.462E-6 \ W_{drs}^2 \]
\[ - 2.51E-9 \ W_{drs}^3 \]
\[ h_{derp} = 295.6 \]
\[ h_{dewh} = 179.01 + 9.663E-2 \ W_{hp} - 3.557E-5 \ W_{hp}^2 \]
\[ W_{hhs} = W_{1hhs} + W_{2hhs} + W_{3hhs} \]

If \( W_{fw} < 493 \text{ lb/s} \) then \( W_{hhs} \) flow is diverted from deaerator to the condenser.

\[ W_{desr} = W_{derp} + W_{dewh} + W_{debd} \]
\[ Q_{desr} = W_{derp} h_{derp} + W_{dewh} h_{dewh} + W_{debd} h_{des} \]

STATE RELATIONS FOR EXTRACTION FLOWS

\[ h_{1lhos} (P_{3hhso}) = 126.87 + 4.154 \ P_{1lhs} (P_{3hhso}) \]
\[ - 0.04224 \ P_{1lhs}^2 (P_{3hhso})^2 + 1.8E-4 \ P_{1lhs}^3 (P_{3hhso})^3 \]
\[ \rho_{1lhos} (P_{3hhso}) = 60.532 - 0.0460 \ P_{1lhs} (P_{3hhso}) \]
\[ T_{1lhos} (P_{3hhso}) = 159.10 + 4.129 \ P_{1lhs} (P_{3hhso}) - 0.04236 \ P_{1lhs}^2 (P_{3hhso})^2 \]
\[ + 18E-4 \ P_{1lhs}^3 (P_{3hhso})^3 + 459.67 \]
\[ -202- \]
\[ h_{21hso} (h_{31hso}) = 32.5 + 43.500 \ p_{21hs} (P_{31hs}) \]

\[ -7.380 \ p_{21hs}^2 (P_{31hs}^2) + 0.515 \ p_{21hs}^3 (P_{31hs})^2 \]

\[ \rho_{21hso} (\rho_{31hso}) = 62.35 - 0.289 \ p_{21hs} (P_{31hs}) \]

\[ T_{21hso} (T_{31hso}) = 64.06 + 43.602 \ p_{21hs} (P_{31hs}) - 7.402 \ p_{21hs}^2 (P_{31hs})^2 \]

\[ + 0.516 \ p_{21hs}^3 (P_{31hs})^3 + 459.67 \]

A.7 Model Parameter Values

The values of the parameters used to represent the local power plant are summarized in Table A.2. The parameters were evaluated using manufacturer's construction diagrams and boiler steady state data at 100% load.
<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>NOMENCLATURE</th>
<th>UNITS</th>
<th>VALUE</th>
</tr>
</thead>
<tbody>
<tr>
<td>$K_{uwgm}$</td>
<td>Waterwall: gas to metal</td>
<td>Btu/(s$^o$R$^4$)</td>
<td>3.187E-9</td>
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<tr>
<td>$K_{uwwm}$</td>
<td>Waterwall: metal to water</td>
<td>Btu/(s$^o$R$^3$)</td>
<td>173.521</td>
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<tr>
<td>$K_{upsr}$</td>
<td>Primary Superheater: radiation</td>
<td>Btu/(s$^o$R$^4$)</td>
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<td>$K_{upsr}$</td>
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<td>$K_{ussgm}$</td>
<td>Primary Superheater: metal to steam</td>
<td>Btu/(lb$^m_6s^*4^o$R)</td>
<td>2.345</td>
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<td>Secondary Superheater: gas to metal</td>
<td>Btu/(lb$^m_6s^*4^o$R)</td>
<td>3.755</td>
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<tr>
<td>$K_{ussms}$</td>
<td>Secondary Superheater: metal to steam</td>
<td>Btu/(lb$^m_8s^*2^o$R)</td>
<td>6.753</td>
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<td>Reheater: gas to metal</td>
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<td>$K_{uecgm}$</td>
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<td>Btu/(lb$^m_8s^*2^o$R)</td>
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### METAL MASSES

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<td>$K_{mwwm}$</td>
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<td>$K_{msm}$</td>
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<tr>
<td>$K_{rhm}$</td>
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**TABLE A.2: MODEL PARAMETERS**
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<th>PARAMETER</th>
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<th>UNITS</th>
<th>VALUE</th>
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<td>UNITS</td>
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<td>K_{nfd}</td>
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**INERTIAS**

**FRICITION COEFFICIENTS:** Steam and Water

**NUMBER OF OPERATING PUMPS AND FANS**
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<th>PARAMETER</th>
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<td>Primary Superheater</td>
<td>Combined into an equivalent friction coefficient, K_{fg}</td>
<td>2.638E-7</td>
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<tr>
<td>K_{fssg}</td>
<td>Secondary Superheater</td>
<td>Combined into an equivalent friction coefficient, K_{fg}</td>
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<tr>
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<td>Economizer</td>
<td>( \text{lb} s^2/(\text{in lb}^2) )</td>
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<td>K_{fapg}</td>
<td>Air Preheater</td>
<td>( \text{lb} s^2/(\text{in lb}^2) )</td>
<td>1.176E-7</td>
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<td>PARAMETER</td>
<td>NOMENCLATURE</td>
<td>UNITS</td>
<td>VALUE</td>
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<td>$K_{fst}$</td>
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<td><strong>SPECIFIC HEAT: FLUE GAS</strong></td>
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<td>$K_{sgw}$</td>
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<td>$K_{lhm}$</td>
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<td>1.05</td>
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<td><strong>BURNER TILTS AND NUMBER OF OPERATING GUNS: CORRECTION TO FURNACE HEAT TRANSFER COEFFICIENT</strong></td>
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<td>$K_{lxgg}$</td>
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<td>Level 4</td>
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<td>Slip at Max. Torque: Recirculating Pump Motor</td>
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<td>Slip at Max. Torque: Forced Draft Fan Motor</td>
<td>non-dim.</td>
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<td>NOMENCLATURE</td>
<td>UNITS</td>
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<td>$K_{nidi}$</td>
<td>Ratio: Line Frequency to Motor Synchronous Speed</td>
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<td>lbm^{0.5}ft^{1.5}in/s lbf^{0.5}</td>
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<td>$K_{eip}$</td>
<td>IP Turbine Extraction Factor</td>
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<td>$K_{elp}$</td>
<td>LP Turbine Extraction Factor</td>
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</tr>
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<td>NOMENCLATURE</td>
<td>UNIT</td>
<td>VALUE</td>
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<td>$K_{fv}$</td>
<td>Boiler Feedwater Valve Flow Coefficient</td>
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<td>4.824E-4</td>
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<td>$K_{qylypo}$</td>
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<td>$K_{tat}$</td>
<td>Constant Atmospheric Air Temperature</td>
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<td>$K_{tahad}$</td>
<td>Constant Temperature Change of Air in Air Heater</td>
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<td>$K_{tahad}$</td>
<td>Constant Temperature Change of Air in Air Preheater</td>
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<td>$S_{ar}$</td>
<td>Constant Specific Heat of Air Supplied to Furnace</td>
<td>$\frac{\text{Btu}}{\text{lb} \cdot ^\circ \text{R}}$</td>
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<td>$K_{ldc}$</td>
<td>Downcomer Length</td>
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<td>$K_{lww}$</td>
<td>Waterwall Length</td>
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<td>$g$</td>
<td>Gravitational Acceleration</td>
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<td>$K_{nbpr}$</td>
<td>Booster-Pump Feedpump Speed Ratio</td>
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<td>Drum Water Volume-Level Relationship</td>
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<td>$K_{2xdrw}$</td>
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<td>in/ft$^3$</td>
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<td>$K_{3xdrw}$</td>
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<td>in/ft$^6$</td>
<td>1.93x10$^{-6}$</td>
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<td>PARAMETER</td>
<td>NOMENCLATURE</td>
<td>UNITS</td>
<td>VALUE</td>
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<td>$K_1 \times \text{dew}$</td>
<td>Deaerator Water Volume-Level Relationship</td>
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<td>$K_2 \times \text{dew}$</td>
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<td>$K_3 \times \text{dew}$</td>
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<td>in/ft$^6$</td>
<td>2.03x$10^{-6}$</td>
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APPENDIX B

STANDARD MODEL DESCRIBING EQUATIONS
AND PARAMETER VALUES

The Standard Model as presented here assumes sufficient excess capacity that changes in voltage and frequency do not affect the performance of auxiliary drive motors. The effect of voltage and frequency will later be implemented as an add on to the standard model. The assumptions and governing equations for the standard model are presented in the sections corresponding to physical components in the boiler-turbine system. The nomenclature is as given in Appendix A, Section A.1.

B.1 Governing Equations

DRUM, DOWNCOMER AND WATERWALL

Assumptions:

i) Feedwater from the economizer mixes with drum water in the downcomers.

ii) Constant recirculation flow in the downcomer-waterwall loop. Assume sufficient excess capacity in the recirculating pumps.

iii) Perfect drum level control. Drum water volume (level) is constant. Assume sufficient excess capacity in the boiler feedpump to supply feedwater demand.

iv) Drum water and steam are in saturated condition.
The equations summarizing performance are:

**Continuity:**

Drum: \[
\frac{d}{dt} \left( \rho_{drs} V_{drs} + \rho_{drw} V_{drw} \right) = W_{dc} - W_{drs} - (W_{dc} - W_{fw})
\]

\[
V_{drw} = K_{vdrw}
\]

\[
V_{drs} = K_{vdrs}
\]

\[
W_{dc} = W_{rw}
\]

**Energy:**

Energy balance at downcomer inlet:

\[
W_{dc} h_{dcl} = W_{fw} h_{eco} + (W_{dc} - W_{fw}) h_{drw}
\]

**Waterwall:**

\[
W_{dc} (h_{wwo} - h_{dcl}) = Q_{wwnw}
\]

Drum:

\[
\frac{d}{dt} \left( \rho_{drs} V_{hrs} h_{hrs} + \rho_{drw} K_{hrw} h_{drw} \right) = W_{dc} h_{wwo} - (W_{dc} - W_{fw}) h_{drw} - W_{hrs} h_{hrs}
\]

**Steam Table Fits:**

\[
T_{drs} = \phi(\rho_{drs}) \quad \rho_{drw} = \phi(\rho_{drs})
\]

\[
h_{drs} = \phi(\rho_{drs}) \quad p_{drs} = \phi(\rho_{drs})
\]

\[
h_{drw} = \phi(\rho_{drs})
\]
The state variable associated with the drum-downcomer-waterwall loop is the drum steam density, $\rho_{drs}$. With perfect drum water level control assumed, $V_{drw}', V_{drw}$ are constants. The drum continuity and energy equations are solved simultaneously for $d\rho_{drs}/dt$ and $W_{fw}$ by first substituting for $h_{drs} = \phi(\rho_{drs})$, $h_{drw} = \phi(\rho_{drs})$ and $\rho_{drw} = \phi(\rho_{drs})$ in the left hand side.

SUPERHEATER

The primary and secondary superheaters and desuperheat spray section are lumped together and the effective mass as defined in Appendix A is assumed constant. The equations summarizing the performance of the superheater are:

**Continuity:**

$$K_{vsh} \frac{d}{dt} \rho_{sso} = W_{drs} + W_{sy} - W_{sso}$$

**Energy:**

$$K_{mshe} \frac{dh_{sso}}{dt} = W_{drs} h_{drs} + W_{sy} h_{fpo} - W_{sso} h_{sso} + Q_{sh}$$

**Momentum:**

$$P_{drs} - P_{sso} = k_{fsh} W_{drs}^2 / \rho_{drs}$$

**Steam Table Fits:**

$$P_{sso} = \phi(\rho_{sso}, h_{sso})$$

$$T_{sso} = \phi(\rho_{sso}, h_{sso})$$

$$S_{sso} = \phi(\rho_{sso}, h_{sso})$$

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The state variables associated with the superheater are the density \( \rho_{\text{SSO}} \) and enthalpy \( h_{\text{SSO}} \).

**GOVERNING STAGE**

Sonic flow is assumed through the high pressure turbine. The storage effect of the steam chest is neglected. Also the effect of temperature variation on steam flow is assumed small compared to that of pressure so that the main steam flow is a function of only pressure and the valve area and is given by:

\[
W_{\text{SSO}} = K_{cv} A_{cv} P_{\text{SSO}}
\]

\[
W_{\text{hp}} = W_{\text{SSO}} \quad \text{(Assume no extraction in superheater)}
\]

**HIGH PRESSURE TURBINE**

The process through the turbine is assumed adiabatic. The extraction steam from the high pressure turbine is assumed to be a constant fraction of the inlet steam flow, so that the outlet steam flow is also a fraction of the inlet steam flow.

The equations summarizing the performance are:

**Continuity:**

\[
W_{\text{hpo}} = K_{\text{whp}} W_{\text{hp}}
\]

**Energy:**

\[
MW_{\text{hp}} = W_{\text{hp}} (h_{\text{SSO}} - h_{\text{hpo}}) K_f
\]
Adiabatic Process:

\[\eta_{isen} = \phi(W_{hp})\]
\[h_{hpoi} = \phi(P_{hpo}, S_{ss0})\]
\[h_{hpo} = h_{hpoi} - \eta_{isen}(h_{ss0} - h_{hpoi})\]
\[T_{hpo} = \phi(P_{hpo}, h_{hpo})\]

The state variable associated with the high pressure turbine is the turbine speed \(N_{tr}\) which is common for the three elements of the compound turbine.

REHEATER

The reheater is lumped with the reheat spray section and the effective mass as described in Appendix A is assumed constant. The describing equations are:

Continuity:

\[K_{vhrh} \frac{d\rho_rh}{dt} = W_{hpo} + W_{ry} - W_{rho}\]

Energy:

\[K_{mrhe} \frac{dh_{rho}}{dt} = W_{hpo} h_{hpo} + W_{ry} h_{fpo} - W_{rho} h_{rho} + Q_{rh}\]

Momentum:

\[P_{hpo} - P_{rho} = K_{frh} W_{rho}^2 / \rho_{rho}\]

Steam Table Fits:

\[P_{rho} = \phi(\rho_{rho}, h_{rho})\]  \[T_{rho} = \phi(\rho_{rho}, h_{rho})\]
The state variables associated with the reheater are the density \( \rho_{\text{rho}} \) and enthalpy \( h_{\text{rho}} \).

**LOW PRESSURE TURBINE**

The intermediate pressure turbine, cross-over pipe and low pressure turbine are lumped together into an equivalent low pressure turbine. The low pressure turbine receives steam from the reheater and discharges to the condenser. The discharge condition is assumed fixed. The flow rate is dependent upon the reheater condition. The process through the low pressure turbine is assumed adiabatic. In computing the power output of the low pressure turbine a fractional efficiency factor is employed to account for steam extraction. The performance is described by the following equations:

\[
W_{\text{Lp}} = Kw_{\text{rho}} P_{\text{rho}}
\]

\[
W_{\text{rho}} = W_{\text{Lp}}
\]

**Energy:**

\[
M_W_{\text{Lp}} = W_{\text{Lp}} (h_{\text{rho}} - h_{\text{Lpo}}) K_{\text{eLp}} K_j
\]

**TURBINE-GENERATOR**

The description of the Turbine-Generator dynamics is similar to that given in Appendix A:

**Total Turbine Output:**

\[
M_W_{\text{tro}} = M_W_{\text{hp}} + M_W_{\text{Lp}}
\]

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Moment of Momentum:

\[
\frac{dN_{tr}}{dt} = M_{W_{tro}} - M_{W_{gn}}
\]

\[
M_{W_{gn}} = K_{w_{wtr}} M_{W_{gupu}}
\]

Generator Action:

\[
M_{W_{gupu}} = s \sin \delta
\]

\[
\frac{d\delta}{dt} = N_{tr} - N_{elec}
\]

\(N_{elec}\) is the electrical system frequency and \(\delta\) is the power angle.

The state variables associated with the turbine-generator are the turbine speed \(N_{tr}\) and power angle \(\delta\).

WATERSIDE

The low pressure feedwater heater, deaerator, high pressure feedwater heater, condensate and feed pumps are respectively modeled by replacing the heat or work inputs by equivalent enthalpy inputs. These enthalpy inputs are expressed as functions of the load level via main stream flow.

The governing equations are:

Low Pressure Feedwater Heater:

\[
\frac{dh_{lho}}{dt} = (h_{cpo} + h_{hhd} - h_{lho}) K_{tclh}
\]

\[
h_{hhd} = \phi(W_{ss1})
\]

Condensate Pump:

\[
h_{cpo} = h_{cmw} + K_{cpd}
\]
Deaerator:
\[
\frac{dh_{\text{dew}}}{dt} = \left( h_{\text{lho}} + h_{\text{ded}} - h_{\text{dew}} \right)/K_{\text{tcde}}
\]
\[h_{\text{ded}} = \phi(W_{\text{sso}})\]

Boiler Feed Pump:
\[h_{\text{fpo}} = h_{\text{dew}} + K_{\text{hfpd}}\]

High Pressure Feedwater Heater:
\[
\frac{dh_{\text{hho}}}{dt} = \left( h_{\text{fpo}} + h_{\text{hhd}} - h_{\text{hho}} \right)/K_{\text{tchh}}
\]
\[h_{\text{hhd}} = \phi(W_{\text{sso}})\]

The state variables associated with the low pressure feedwater heater, deaerator, and high pressure feedwater heater are enthalpies, \(h_{\text{lho}}\), \(h_{\text{dew}}\) and \(h_{\text{hho}}\), respectively.

ECONOMIZER

The dynamics of the economizer is governed by the energy equation:
\[
k_{\text{mece}} \frac{dh_{\text{eco}}}{dt} = W_{\text{fw}} (h_{\text{hho}} - h_{\text{eco}}) + Q_{\text{ec}}
\]

The effective mass is assumed constant and the state variable associated with the economizer is the enthalpy \(h_{\text{eco}}\).
GAS SIDE FLOW AND HEAT TRANSFER

The furnace pressure control is assumed perfect. The air flow rate is determined by the control system and it is assumed that the fans have sufficient excess capacity to meet flow demand even with changes in voltage and frequency.

Heat generated by combustion of fuel is transferred from the flue gas to steam and water. The flue gas conditions after the economizer are assumed constant. The combustion and heat transfer process in the furnace is complex but simplification is achieved by assuming a linearized radiation process with a constant heat transfer coefficient. The heat transfer rates to the superheater, reheater and economizer are obtained as fractions of the total heat available to them. The total heat available to these exchangers equals the difference in thermal capacity of flue gas between furnace outlet and economizer outlet.

FURNACE

The equations summarizing the furnace performance are:

**Combustion**: (Adiabatic flame temperature $T_{fml}$):

$$W_g K_{sfm} (T_{fml} - K_{to}) = W_{ar} K_{sar} (K_{tar} - K_{to}) + W_{gr} K_{sgr} (T_{ecgo} - K_{to})$$

$$+ W_{fl} K_{sfl} (K_{ tf1} - K_{to}) + W_{fl} K_{hf1}$$

**Linearized Radiation**:

$$Q_{wwgm} = U_{wwgm} (T_{wwge} - T_{wwm})$$

$$U_{wwgm} = K_{uwg} U_{xgg}$$

$$U_{xgg} = K_{1xgg} = K_{2xgg} \tan (X_{gg})$$

$$K_{uwg} = K_{1uw} + K_{2uw} W_{fl}$$
Heat Loss in Gas:

\[ Q_{wwgm} = \dot{W}_g S_{fn} (T_{fnl} - T_{wwgo}) \]

\[ T_{wwge} = \frac{1}{2}(T_{fnl} - T_{wwgo}) \]

**WATERWALL METAL**

The equations summarizing the waterwall performance are:

**Energy:**

\[ K_{wwme} \frac{dT_{wwm}}{dt} = Q_{wwgm} - Q_{wwmw} \]

\[ Q_{wwmw} = K_{wwmw} (T_{wwm} - T_{dres}) \]

**GAS PATH**

The total heat transferred between the furnace exit and economizer exit is shared by the superheater, reheater and economizer.

Total heat available to the superheat, reheater and economizer is given by:

\[ Q_{gp} = \dot{W}_g S (T_{wwgo} - T_{ecgo}) \]

\( T_{ecgo} \) is assumed constant, \( K_{ecgo} \).

The quantities transferred to the reheater, economizer and superheater are given by:

\[ Q_{rh} = Z_{grh} Q_{gp} \]

\[ Q_{ec} = Z_{qec} Q_{gp} \]

\[ Q_{sh} = Q_{gp} - Q_{rh} - Q_{ec} \]
and \( \frac{Z_{grh}}{Z_{gec}} \) are fractions which are functions of air and fuel flow

\[
Z_{grh} = \phi(Z_{wwg})
\]

\[
Z_{gec} = \phi(Z_{wwg})
\]

\[
Z_{wwg} = W_f + W_{ar}
\]

**CONTROL SYSTEM**

Perfect feedwater flow, condensate flow and furnace pressure controls are assumed, so these loops are not included. The remaining control loops are identical to those described in Appendix C, with the exception of the combustion control for which the output of the air controller is not vane area but air flow.

**B.2 State Relations**

**Drum:**

\[
\rho_{drw} = 49.271 - 2.137 \rho_{drs} + 0.03348 \rho_{drs}^2
\]

\[
h_{drs} = 1241.7 - 21.344 \rho_{drs} + 0.21 \rho_{drs}^2
\]

\[
h_{drw} = 526.6 + 31.044 \rho_{drs} - 0.0209 \rho_{drs}^2
\]

\[
P_{drs} = 459.41 + 358.61 \rho_{drs} - 12.04 \rho_{drs}^2
\]

\[
T_{drs} = 616.75 + 6.691 \rho_{drs} + 459.67
\]

**Superheater:**

\[
P_{sso} = -291.36 - 964.04 \rho_{sso} + 0.2178 h_{sso} + 1.1815 \rho_{sso} h_{sso}
\]

\[
T_{sso} = -1745.1 + 129.1 \rho_{sso} + 1.811 h_{sso} - 0.06631 \rho_{sso} h_{sso} + 459.7
\]

\[
S_{sso} = 0.8115 + 5.5142E-4 h_{sso} - 0.0268 \rho_{sso}
\]
High Pressure Turbine:

\[ \eta_{isen} = 0.589 + 2.317 \times 10^{-4} \, \text{Whp} \]

\[ h_{hpo} = -485.23 + 1065.28 \, S_{sso} + 0.232 \, P_{hpo} \]

\[ T_{hpo} = -1639.24 + 0.119 \, P_{hpo} + 1.682 \, h_{hpo} + 459.7 \]

Reheater:

\[ P_{\rho} = 27.06 - 1019.1 \, \rho_{\rho} - 1.735 \times 10^{-2} \, h_{\rho} + 1.228 \, \rho_{\rho} \, h_{\rho} \]

\[ T_{\rho} = 2013.4 + 189.65 \, \rho_{\rho} + 1.963 \, h_{\rho} - 9.305 \times 10^{-2} \, \rho_{\rho} \, h_{\rho} \]

Water Side:

\[ h_{Lhd} = 113.12 + 28.3 \times 10^{-3} \, \text{Whp} \]

\[ h_{ded} = 43.62 + 14.817 \times 10^{-3} \, \text{Whp} \]

\[ h_{hhd} = 136.60 + 56.2 \times 10^{-2} \, \text{Whp} \]

Gas Path:

\[ Z_{qrh} = 0.271 - 7.803 \times 10^{-6} \, Z_{wwwg} - 1.337 \times 10^{-8} \, Z_{wwwg}^2 \]

\[ Z_{qec} = 0.6912 - 8.852 \times 10^{-2} \, Z_{wwwg} + 3.971 \times 10^{-7} \, Z_{wwwg}^2 \]

B.3 Parameter Values

The parameter values for the standard model are given in Table B.1. The parameters were evaluated using manufacturer's construction diagrams and boiler steady state data at 100%.
## TABLE B.1 STANDARD MODEL PARAMETER VALUES

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>NOMENCLATURE</th>
<th>UNITS</th>
<th>VALUE</th>
</tr>
</thead>
<tbody>
<tr>
<td>$K_{vdrw}$</td>
<td>Drum water volume</td>
<td>ft$^3$</td>
<td>1170.2</td>
</tr>
<tr>
<td>$K_{vdrs}$</td>
<td>Drum steam volume</td>
<td>ft$^3$</td>
<td>788.5</td>
</tr>
<tr>
<td>$K_{vdr}$</td>
<td>Drum volume</td>
<td>ft$^3$</td>
<td>1958.7</td>
</tr>
<tr>
<td>$K_{vsh}$</td>
<td>Superheater volume</td>
<td>ft$^3$</td>
<td>5000.0</td>
</tr>
<tr>
<td>$K_{fsh}$</td>
<td>Superheater friction coefficient</td>
<td>lb$^5$s$^2$/(lb$^2$m$^2$ ft$^3$)</td>
<td>2.7E-3</td>
</tr>
<tr>
<td>$K_{mshe}$</td>
<td>Effective superheater steam mass</td>
<td>lb</td>
<td>1.693E5</td>
</tr>
<tr>
<td>$K_{cv}$</td>
<td>Control valve flow coefficient</td>
<td>lb$^5$ft$^{1.5}$ in/s lb$^0.5$ ft$^3$</td>
<td>0.459</td>
</tr>
<tr>
<td>$K_{whp}$</td>
<td>HP Turbine flow, inlet outlet</td>
<td>--</td>
<td>0.895</td>
</tr>
<tr>
<td>$K_{vrh}$</td>
<td>Reheater volume</td>
<td>ft$^3$</td>
<td>6000.0</td>
</tr>
<tr>
<td>$K_{mrhe}$</td>
<td>Reheater effective mass</td>
<td>lb</td>
<td>1.078E5</td>
</tr>
<tr>
<td>$K_{frh}$</td>
<td>Reheater friction coefficient</td>
<td>lb$^5$s$^3$/(lb$^2$m$^2$ ft$^3$)</td>
<td>4.59E-5</td>
</tr>
<tr>
<td>$K_{wrh}$</td>
<td>LP Turbine flow coefficient</td>
<td>--</td>
<td>1.784</td>
</tr>
<tr>
<td>$K_{elp}$</td>
<td>LP turbine extraction factor</td>
<td>--</td>
<td>0.84</td>
</tr>
<tr>
<td>PARAMETER</td>
<td>NOMENCLATURE</td>
<td>UNITS</td>
<td>VALUE</td>
</tr>
<tr>
<td>------------</td>
<td>---------------------------------------------------</td>
<td>---------------</td>
<td>------------</td>
</tr>
<tr>
<td>( K_{jtre} )</td>
<td>Turbine-Generator Inertia</td>
<td>lb ( \cdot ) ft ( ^2 )</td>
<td>6.25E5</td>
</tr>
<tr>
<td>( K_{mwr} )</td>
<td>Rated Power output</td>
<td>ft ( \cdot ) lb ( _m )</td>
<td>4.428E8</td>
</tr>
<tr>
<td>( K_{j} )</td>
<td>Joules constant</td>
<td>lb ( \cdot ) ft/( \text{BTU} )</td>
<td>778.</td>
</tr>
<tr>
<td>( K_{tclh} )</td>
<td>LP FW heater time constant</td>
<td>s</td>
<td>60.</td>
</tr>
<tr>
<td>( K_{hcpd} )</td>
<td>Enthalpy change across Cond. pump</td>
<td>Btu/lb</td>
<td>1.4</td>
</tr>
<tr>
<td>( K_{tcde} )</td>
<td>Time constant of generator</td>
<td>s</td>
<td>280.</td>
</tr>
<tr>
<td>( K_{hfpd} )</td>
<td>Enthalpy change across feed pump</td>
<td>Btu/lb</td>
<td>11.0</td>
</tr>
<tr>
<td>( K_{tchh} )</td>
<td>Time constant of HP FW heater</td>
<td>s</td>
<td>60.</td>
</tr>
<tr>
<td>( K_{mece} )</td>
<td>Effective mass of economizer</td>
<td>lb</td>
<td>1.784E5</td>
</tr>
<tr>
<td>( K_{to} )</td>
<td>Reference Temperature</td>
<td>°R</td>
<td>537.</td>
</tr>
<tr>
<td>( K_{sfn} )</td>
<td>Specific heat of furnace gas</td>
<td>Btu/(lb ( \cdot ) °R)</td>
<td>0.31</td>
</tr>
<tr>
<td>( K_{sar} )</td>
<td>Specific heat of air</td>
<td>Btu/(lb ( \cdot ) °R)</td>
<td>0.252</td>
</tr>
<tr>
<td>( K_{tar} )</td>
<td>Temperature of air</td>
<td>°R</td>
<td>1060.0</td>
</tr>
<tr>
<td>( K_{sgr} )</td>
<td>Specific heat of recirculation gas</td>
<td>Btu/(lb ( \cdot ) °R)</td>
<td>0.27</td>
</tr>
<tr>
<td>( K_{tece} )</td>
<td>Temperature of economizer gas</td>
<td>°R</td>
<td>1077.7</td>
</tr>
<tr>
<td>PARAMETER</td>
<td>NOMENCLATURE</td>
<td>UNITS</td>
<td>VALUE</td>
</tr>
<tr>
<td>-----------</td>
<td>--------------------------------------------------</td>
<td>--------------------</td>
<td>--------</td>
</tr>
<tr>
<td>$K_{sfl}$</td>
<td>Specific heat of fuel</td>
<td>Btu/(lb °R)</td>
<td>0.49</td>
</tr>
<tr>
<td>$K_{tfl}$</td>
<td>Temperature of fuel</td>
<td>°R</td>
<td>860</td>
</tr>
<tr>
<td>$K_{hfl}$</td>
<td>Heating value of fuel</td>
<td>Btu/lb</td>
<td>18200.</td>
</tr>
<tr>
<td>$K_{1uw}$</td>
<td>Furnace heat transfer coefficient factor</td>
<td>Btu/(s °R)</td>
<td>73.801</td>
</tr>
<tr>
<td>$K_{2uw}$</td>
<td></td>
<td>Btu/(lb °R)</td>
<td>1.4173</td>
</tr>
<tr>
<td>$K_{xgg}$</td>
<td>Burner tilt correction coefficient</td>
<td>--</td>
<td>1.0</td>
</tr>
<tr>
<td>$K_{2xgg}$</td>
<td></td>
<td>--</td>
<td>-0.2865</td>
</tr>
<tr>
<td>$K_{uw}$</td>
<td>Heat transfer coefficient waterwall to water</td>
<td>Btu/(s °R)</td>
<td>25876.4</td>
</tr>
<tr>
<td>$K_{w}$</td>
<td>Effective waterwall mass</td>
<td>lb</td>
<td>1.524E6</td>
</tr>
</tbody>
</table>
APPENDIX C

BOILER-TURBINE CONTROL

SYSTEM DESCRIPTION AND PARAMETER VALUES

C.1 General Control System

The boiler-turbine control system used in the digital computer simulation model incorporates all major control loops of a modern power plant. Control system operation and parameter values were obtained from a local power plant and if not available, were given appropriate values and verified by computer simulations [12,6].

In addition to conventional process controllers, such as P, PI, and PID controllers, models are formulated for the input-output control system relationships and for the dynamic actuators. These are modeled by linear transducers and simulators, respectively, and are discussed below. The determination of set point values are also discussed, followed by the description of the boiler control loops. The parameter values for the controller gains, time constants, and rate limits are listed in Table C.1.

LINEAR TRANSDUCER

The conversion from physical measurements into control signal voltages is accomplished by linear transducers. The linear transducer creates a control signal voltage which is a linear function of the measured input variable. Let \( x(t) \) be the time dependent variable with \( x_0 \) and \( x_m \) denoting its minimum and maximum value, respectively. Let \( c(t) \) be the control signal voltage with \( c_0 \) and \( c_m \) denoting its...
minimum and maximum voltages respectively. In the boiler and turbine control systems the minimum voltage is 1 volt and the maximum voltage is 5 volts. Thus the linear transducer is described by:

\[ c(t) = c_0 + (c_m - c_0) \frac{x(t) - x_0}{x_m - x_0} \]

Similarly the conversion from control signals into a physical variable is described by:

\[ x(t) = x_0 + (x_m - x_0) \frac{c(t) - c_0}{c_m - c_0} \]

The time dependent variables, their set point, minimum and maximum values are listed in Table C.2.

**ACTUATOR DYNAMICS SIMULATOR**

An approximation has been made in modeling the dynamics of the actuators receiving signals from the control system. The representation lumps together the effects of actuator and positioner and is called an Actuator Simulator. It provides for time delays between changes in the control signal and the resulting physical output being controlled and represents, for example, the time required to open a desuperheating spray valve to the time required to change the vane areas for air flow. Each major control loop uses an Actuator Simulator and has a time constant appropriate for the actuation method involved.
SET POINT DETERMINATION

The feedforward signal originates in the Load Demand Computer (LDC). In coordinated control and remote coordinated control the LDC output is transmitted to both the boiler and turbine controls. The signal determines the operating conditions of the boiler-turbine system.

In the boiler the LDC determines the set point values for main steam pressure. The functional relationship between set point values and the LDC signal takes on the following form:

The physical set point values and corresponding load and LDC values are listed in Table C.3.

C.2 Boiler Control Loops

COMBUSTION CONTROL: Throttle Pressure, Air Flow, Fuel Flow Controls

The LDC signal generates a pressure demand signal which is a function of megawatt demand. At low load levels the throttle pressure demand is set at a minimum level. As the load increases the throttle pressure demand increases linearly. At higher load levels the pressure demand again remains constant at its maximum value.
The throttle or main steam pressure demand signal represents the pressure set point for its associated load demand. The pressure set point is compared to the measured pressure and the pressure controller applies proportional plus integral control action to the error. The output signal is biased by a proportional frequency error signal. The output of the summer represents the Boiler Master Demand Signal.

The Boiler Master Demand Signal is transmitted to the fuel and air flow control system. The demand for fuel is the lower of two signals: 1) Boiler Master Demand or 2) total measured air flow. The latter insures that the fuel demand will never exceed the air flow. The Fuel Controller compares the total fuel flow with the fuel demand and applies proportional plus integral action on the error. The output signal is passed through an Actuator Simulator which represents the delay due to the action of the fuel actuator. The control signal is then converted to a fuel flow ($W_{fl}$) via a linear transducer.

The magnitude of the Boiler Master Demand signal transmitted to the Air Flow Controller is limited from decreasing below 25% of its full load value. The Air Flow Controller compares the air flow demand to the measured air flow and applies proportional plus integral action to the error. The output from the controller passes to the Actuator Simulator which represents the delay due to the action of the vane actuators. The control signal is converted into the Forced Draft Fan Inlet Vane Position ($A_{vf}$) to obtain the desired air flow.
FURNACE PRESSURE CONTROL

Furnace pressure is maintained at a preset level by the Furnace Pressure Controller. The Controller compares the furnace pressure set point to the measured pressure and applies proportional plus integral control action to the error. Measured air flow is used as an anticipatory bias signal. The output of the summer is transmitted to an Actuator Simulator which represents the delay due to the action of vane actuators. The output signal is converted into the Induced Draft Fan Inlet Vane Position ($A_{VI}$) which controls furnace pressure.

FEEDWATER FLOW CONTROL

Feedwater flow control is accomplished by a three element control system. First stage pressure is used as a measure of main steam flow and represents the first element. The total feedwater flow consists of the measured feedwater flow and the desuperheat and reheat spray flows, and represents the second element. Drum level is the third element. The Drum Level Controller compares the measured drum level with the drum level set point and applies proportional plus integral control action to the error to maintain the level at the required value. The output of the Drum Level Controller is added to the main steam flow signal to obtain feedwater flow demand. The Feedwater Flow Controller compares the total feedwater flow with this feedwater flow demand and applies proportional plus integral control action to balance the feedwater flow with the demand. The output of the Feedwater Flow Controller is transmitted to the Actuator Simulator which represents the delay associated with the feedwater valve actuators. The output signal is converted to feedwater valve area ($A_{fy}$)
that governs the feedwater flow rate from the feedpump.

**FEEDPUMP TURBINE CONTROL**

A constant pressure drop is maintained across the feedwater valve by varying the boiler feedpump speed. This is accomplished by comparing the required pressure drop with the measured and applying proportional plus integral control action to the error. The output of the feedpump turbine controller is transmitted to an Actuator Simulator which approximates the delay associated with the feedpump turbine that drives the pump. The output signal of the simulator is converted into extraction steam flow \( W_{ft} \) which is supplied to the feedpump turbine.

**CONDENSATE FLOW CONTROL**

The condensate flow control is a three element control system: steam flow, condensate flow, and deaerator level. The turbine reheat pressure is used as a measure of the steam flow on the low pressure side of the steam loop and represents the first element. The measured condensate flow is the second element and the deaerator water level the third element. The Deaerator Level Controller compares the measured deaerator water level with the set point and applies proportional plus integral plus derivative control action to the error to maintain the level at the required value. The output of the Deaerator Level Controller is added to the steam flow signal to obtain the condensate flow demand. The Condensate Flow Controller compares the measured condensate flow with the demand and applies proportional plus integral control action to the error to match the two. The output of the Condensate Flow Controller is transmitted to an Actuator Simulator which represents the delay due to
the action of the deaerator valve actuator. The output of the simulator is converted into deaerator level control valve area \(A_{dv}\) that controls condensate flow.

**SUPERHEAT TEMPERATURE CONTROL**

Superheat temperature control is accomplished by regulating the desuperheat spray flow between the primary and secondary superheaters. The temperature set point is a function of load as measured by the Main Steam Flow \(W_{hp}\). There is the option of manually lowering the set point by the operator. The temperature set point is compared with the measured temperature and the Superheat Temperature Controller applies proportional plus integral action to the error. Anticipated changes in load as measured by first stage pressure and in burner tilt position are added as derivative control actions. The output of the Superheat Temperature Controller and the derivative signals are sent to an Actuator Simulator which represents the delay due to the spray valve actuators. The output signal is converted into a superheat spray flow \(W_{sy}\) that maintains steam temperature exiting the secondary superheater.

**REHEAT TEMPERATURE CONTROL**

Three control schemes are available to maintain reheat temperature at set point: reheat spray, burner tilt position, and gas recirculation. The temperature set point is a function of load as represented by the steam flow. A manual set point may be used to lower the set point. The temperature set point is compared with the measured temperature and an added bias signal corresponding to an anticipated change in load (via
first stage pressure) is sent to the Reheat Temperature Controller. The Controller applies proportional plus integral action to the signal. Anticipated changes in reheat temperature is added to the output of the controller. The resulting signal is transmitted to two Actuator Simulators, one associated with the delay due to the tilt position actuators and the other associated with the delay due to the spray valve actuator. The output signals are converted to burner tilt position and spray flow respectively. The spray signal is actuated only when the burner tilt position decreases below midrange value.

The control strategy of the gas recirculation flow depends only upon the burner tilt position. The gas recirculation control contains a Track Hold Integrator that applies integral control action to the error obtained by comparing burner tilt position with its tilt position set point. The output of the Track Hold Integrator changes gas recirculation flow only when the burner tilts vary more than $\pm 5^\circ$ from midrange. In addition, the output of the Integrator is limited by the measured air flow rate. This limits the gas recirculation flow to a fraction of total air flow. The output of the limiter is transmitted to the Actuator Simulator which represents the delay due to the recirculation actuators. The output signal is converted to a gas recirculation flow ($W_{gr}$) that is taken from the flue gases at the exit of the economizer and introduced into the furnace.

The operation of the reheat temperature control is as follows: assume measured temperature is low so that burner tilts are moving up from their level position. As the burner tilt exceeds $5^\circ$ from level position gas recirculation is activated. The Track Hold Integrator begins to integrate
KC1TR=1.0
KC2TR=1.0
KTC1TR=10.0
KTC2TR=1.0
KTC3TR=1.0

LDC=3.875
JCCUNT=0
JC=0
JCNT=400

CONTINUE

---

IF (T.LT.10.0. OR. T.GT.100.0) GO TO 201
LDC=3.875+0.0125*(T-10.0)

CONTINUE

JC=JC+1
IF (JC.GE.10) GO TO 60
JC=0
PUT 6,T,WAR,WDRS,SSO,PSSO,RSSO,TRHO,TSSO,NTR,LDC,MWO,DELTA,
* ZXGG,WGR,WSY

CONTINUE

IF (T.LT.0.55) GO TO 70
JCCUNT=JCCUNT+1
IF (JCCUNT.NE.JCNT) GO TO 80
JCCUNT=0

CONTINUE

PUT 5,0,T,HECO,RDRS,TWWM,RSSO,HRHO,NTR,HLHO,HDEW,HRHO,
* CMD,C5AR,5FL,
* C2GR,C5RH,C5SY,CARD,CFLD,CGRD,CRYD,CXGD,CSYD,C6TR,C9TR,C12TR,
* DELTA,
* CBMD, WAR, WFL, WGR, TWGF, TFN1, WDRS, WSSO, WSY, WRY, WHPO, WRHO, XGG,
* PSSO, PDRS, PHEO, PRHO, TDRS, TSSO, TRHO, THPO, HHPO, HPPO, HHHO, MWHP,
* MWLP, MTRO, WPW, MWO, OGP, QWGM, OWMMW, QEC, ORH, OSH, UWWM, UXGG,
* TFN1, TWGO, MGNPU, MWTRPU, HCPO, HPPO, ZXGG

800 CONTINUE
52 CONTINUE

C
C SET CONTROLLED INPUT VALUES
C
CALL LIMCHK (CARD)
CALL LIMCHK (CFLD)
CALL LIMCHK (CGRD)
CALL LIMCHK (CWRYD)
CALL LIMCHK (CXGGD)
CALL LIMCHK (CSYD)
CALL CHECK (C12TR, KN5, KNO)
CALL XDUCER (KNU, KN5, KAVL, KAVU, C12TR, ACV)
CALL XDUCER (KCL, KCU, KWARL, KWARU, CARD, WAR)
CALL XDUCER (KCL, KCU, KWFLL, KWFLU, CFLD, WFL)
KWGR=KC2GP*WAF
CALL XDUCER (KWGPL, KWGRU, KCL, KCU, KWGR, KCWGR)
IF (CGRD.GT.KCWGR) CGRD=KCWGR
CALL XDUCER (KCL, KCU, KWGRL, KWGRU, CGRD, WGR)
CRY=CWRYD*KC1FY
CALL XDUCER (KCRYL, KCRYU, KWRYL, KWRYU, CRY, WRY)
CALL XDUCER (KCL, KCU, KXGGI, KXGGU, CXGGD, XGG)
ZXGG=XGG*57.3
CSY=CSYD*KC4SY
CALL XDUCER (KCSYL, KCSYU, KWSYL, KWSYU, CSY, WSY)

C
C * * * STATE PROPERTIES
C
CALL DRSTAT (RDRS, RDDR, HDRS, HDRW, PDRS, TDRS)
CALL SHSTAT (RSSO, HSSC, PSSC, TSSO, SSSO)
CALL RHSTAT (PRHO, HRHO, PRHO, TRHO)

C * * STFAM FLOW
KC1=1.0
PSHD=PDRS-PSSC
IF (PSHD.LT.FNO) KD1=-1.0
WDRS=KD1*SQR (ABS (PSHD)*RDRS/KPSH)
WSSO=KCV*ACV*ESSO
WHPO=KWHP*WSSC
WRHO=KWRH*PRHC

C * * TURBINE AND REHEATER
EHP=0.589+2.317E-4*WSSO
KELP=0.84
HPHO=PRHO*KFRH*WRHO*WBHC/BRHO
CALL HPSTAT (SSSO,PHPO,EHP,HSSO,HHP,HPO)
MWHP=WSSO*(HSSO-HHP)*KJ
MWLP=WRHO*(HRHO-KHLPO)*KJ*KELP
MWTRC=MWHP+MWLP
MWO=KMWX*MWTRC
MWTRPU=MWTRC/KMWR
MWGNPU=KN2*SIN (DELTA)
MWGN=MWGNPU*KMWR

C * * HEAT TRANSFER
WWWG=WAP+WFL+WGR
KUWWGM=73.801+1.4173*WFL
TFN1=KTO+(WFL*KHFL*KELP+WFL*KSFL*(KTFL-KTO)+WAR*KSAR*(KTAR-KTO)
  * WGR*KSGR*(KTECGO-KTO))/(WWWG*KSFNG)
UXGG=K1XGG+K2XGG*SIN (XGG)/COS (XGG)
UWWGM=KUWWGM*UXGG
ZPN1=WWWG*KSFNG+KNP5*UWWGM
TWNG2= (WWWG*KSFNG*ZPN1+KNP5*UWWGM*KWWGM)/ZPN1
QWWGM=UWWGM*(TWNG2-TWWM)
TWWGO=TWWGE-KNP5*QWWGM/(WWWG*KSFNG)
WWWG2=WWWG*TWWGO
QGP=WWWG*KSGPE*(TWWGO-KTECGO)
ZWWWG=WFL+WAP
ZWWWG2=ZWWWG*ZWWWG
ZQEC=0.2710-7.803E-6*ZWWWG-1.337E-8*ZWWWG2
ZQEC=0.6912-8.852E-4*ZWWWG+3.971E-7*ZWWWG2
QEC=ZQEC*QGP
QRH = ZQRH * CGP
QSH = QGP - QEC - QRA
QWWW = RWWW * (TWWW - TDRS)

C * * * DRUM
ZDR1 = KDR2 + KN2 * KDR3 * RDRS
ZDR2 = KVDR / KVDFW - KN1 + ZER1
ZDR3 = KVDRS * (HDRS + RDRS * (KDRS + KN2 * KDR6 * RDRS))
* + KVDRW * (RDRW * (KDR8 + KN2 * KDR9 * RDRS) + HDRW * ZDR1)
ZDR4 = ZDR3 - KVDRW * ZDR2 * HECO
FRDRS = (QWWW + WDRS * (HECO - HDRS)) / ZDR4
WF = KVDRW * ZDR2 / FRDRS + WDRS

HCPO = HCNW + KHCPD
HLHD = 113.12 + 28.3E-3 * WSSO
HDED = 43.62 + 14.817E-3 * WSSO
HFPO = HDEW + KHFPD
HHHD = 136.66 + 56.2E-3 * WSSO

C * * * ** CONTROLS
* * * BOILER MASTER DEMAND
KCPSSO = 4.204
KCNIE = 4.351
CALL XDUCER (KNTEL, KNTRU, KCL, KCU, NTR, CNTR)
CALL XDUCER (KPSSOL, KPSSOU, KCL, KCU, PSSO, CPSSO)
C1MD = KCPSSO - CFSSO
C2MD = C1MD * KC1MD
CALL LIMCHK (C3MD)
C4MD = C2MD * C3MD
CALL LIMCHK (C4MD)
C5MD = KC2MD * (KCNTE - CNTR)
CBMD = C4MD * C5MD
CALL LIMCHK (CBMD)

C * * * AIR FLOW CONTROL
CALL XDUCER (KWARL, KWARU, KCL, KCU, WAR, CWAR)
C2AR = CBMD
IF (C2AR .LT. KC2ARL) C2AR = KC2ARL
C3AR = C2AR - CWAR
C4AR = C3AR * KC1AR
CALL LIMCHK (C5AR)
C6AR = C5AR + C4AR
CALL LIMCHK (C6AR)

C * * FUEL FLCW CONTROL
CWFL = CFLD
C2FL = CBMD
IF (CWAR.LT.CBMD) C2FL = CWAR
C3FL = C2FL - CFIL
C4FL = C3FL * KC1FL
CALL LIMCHK (C5FL)
C6FL = C4FL + C5FL
CALL LIMCHK (C6FL)

C * * * GAS RECIRCULATION CONTROL
KCXGG = 3.0
KCGR = 0.0
IF (ABS(XGG).GT.KNP087) KCGR = 1.0
C1GR = CXGGD - KCXGG
CALL LIMCHK (C2GR)

C * * REHEAT TEMPERATURE CONTROL
KTRH = K1TRH + K2TRH * WSS0
CALL CHECK (KTRH, K4TRH, K3TRH)
CALL XDUCER (KTRHOL, KTRHOU, KCL, KCU, KTRH, KCTRHO)
CALL XDUCER (KTRHOL, KTRHOU, KCL, KCU, TRH0, CTRHO)
C3RH = KCTRHO - CTRHO
C4RH = C3RH * KC1RH
CALL LIMCHK (C5RH)
C6RH = C4RH + C5RH
CALL LIMCHK (C6RH)
CWRY = C6RH
CXGG = C6RH

C * * SUPERHEAT TEMPERATURE CONTROL
KTSS = K1TSS + K2TSS * WSS0
CALL CHECK (KTSS, K4TSS, K3TSS)
CALL XDUCER (KTSSOL, KTSSOU, KCL, KCU, KTSS, KCTSSO)
CALL XDUCER (KTSSOL,KTSSOU,KCL,KCU,TSSO,CTSSO)
C3SY=KCTSSO-CTSSO
C4SY=C3SY*KC1SY
CALL LIMCHK (C5SY)
C6SY=C4SY+C5SY
CALL LIMCHK(C6SY)
C  *  *  *  TURBINE CONTROL
C1TR=KC2TR*(KCNTR-CNTR)
CALL XDUCER (KMWTRL,KMWTRU,KNC,KN6,MWTRO,CMWTR0)
C2TR=KC1TR*(LDC-CMWTRC)
C4TR=C1TR/KCVREG
CALL CHECK (C6TR,KN1,KM1)
C7T3=C6TR+C2TR
C8TR=C7TR+LDC
CALL CHECK (C8TR,KN5,KN0)
CALL CHECK (C9TR,KN5,KN0)
C10TR=C4TR+C9TR

C  *  *  *  DYNAMIC EQUATIONS
F (1) = (WFW*(HHHO-HECO)+QEC)/KMECE
F (2) = FRDHS
F (3) = (QWWGM-QWWM)/(KMWME*KSWMM)
F (4) = (WDRS*WSY-WSSO)/KVSH
F (5) = (WDRS*HDRS+WSY*HFPO-WSSO*QSH)/KMSHF
F (6) = (WHPO*WBY-WBHO)/KVFH
F (7) = (WHPO*HHPO+WRY*HFPC-WBHC*HRHO*ORH)/KMRHE
F (8) = (MWTR0-MWGN)/(NTF*KJTFE)
F (9) = (HCPO+H1HD-H1HO)/KTCLH
F (10) = (HLHO+HDED-HDEW)/KTCDE
F (11) = (HFPO+HHHD-HHHO)/KTCCH
F (12) = C2MD/KTC1MD
F (13) = C4AB/KTC1AR
F (14) = C4FL/KTC1FL
F (15) = C1GR*KC1GF*KCGR
F (16) = C4RH/KTC1RH
F (17) = C4SY/KTC1SY
F (18) = (C6AR-CARD)/KTC2AR
SUBROUTINE DSTAT (RS, R, H, H, P, T)
    IMPLICIT REAL (K, L, M, N)
    COMMON /AREA2/ KDR1, KDR2, KDR3, KDR4, KDR5, KDR6, KDR7, KDR8, KDR9
    RS2 = RS * RS
    RW = KDR1 + KDR2 * RS + KDR3 * RS2
    HS = KDR4 + KDR5 * RS + KDR6 * RS2
    P = 459.41 + 358.61 * RS - 12.04 * RS2
    T = 616.75 + 6.691 * RS + 459.67
    RETURN
END

SUBROUTINE SHSTAT (R, H, P, T, S)
    F = 3 * H
    P = -291.36 - 964.04 * E + 0.2178 * H + 1.1815 * RH
    T = -1745.1 + 125.1 * R + 1.811 * H - 0.06631 * RH + 459.67
    S = 1.3136 - 1.7799 * E - 3 * R + 6.3573 * F - 4 * ALOG (RH)
    RETURN
END

SUBROUTINE HESTAT (S, P, E, H1, H0, T)
    COMPUTE HP TURBINE EXHAUST STEAM PROPERTIES
HI = -485.23 + 1065.28*P + 0.232*P
HO = H1 - E*(H1 - HI)
T = -1639.24 + 0.119*P + 1.682*HO
RETURN
END

SUBROUTINE RHSTAT (R, H, P, T)
  COMPUTE REHEAT STEAM PROPERTIES
  RH = R*H
  P = 27.06 - 1019.1*R - 1.735E-2*H + 1.228*RH
  T = -2013.4 + 189.65*R + 1.963*H - 9.305E-2*RH + 459.67
RETURN
END

SUBROUTINE XDUCEP (ZMIN, ZMAX, CMIN, CMAX, Z, C)
  INPUT-OUTPUT CONVERSION. PHYSICAL VARIABLE TO CONTROL SIGNAL AND VISE VERSA
  C = CMIN + (CMAX - CMIN)*(Z-ZMIN)/(ZMAX-ZMIN)
  IF (C.LT.CMIN) C = CMIN
  IF (C.GT.CMAX) C = CMAX
RETURN
END

SUBROUTINE LIMCHK (ZC)
  CHECK THAT CONTROL VARIABLE (ZC) IS WITHIN LIMITS (1.-5.)
  IF (ZC.LT.1.0) ZC = 1.0
  IF (ZC.GT.5.0) ZC = 5.0
RETURN
END

SUBROUTINE CHECK (ZC, ZMAX, ZMIN)
  CHECK THAT VARIABLE (ZC) IS WITHIN LIMITS (ZMIN-ZMAX)
  IF (ZC.LT. ZMIN) ZC = ZMIN
  IF (ZC.GT. ZMAX) ZC = ZMAX
RETURN
END
IPlot=7,8
POS=175., 175., 350., 225.
IPlot=23,8
F0S=625., 175., 350., 225.
IPlot=22,8
POS=175., 475., 350., 225.
IPlot=1,8
POS=625., 475., 350., 225.
IPlot=2,8
POS=175., 175., 350., 225.
IPlot=11,8
POS=625., 175., 350., 225.
IPlot=9,8
POS=175., 475., 350., 225.
IPlot=31,8
POS=625., 475., 350., 225.
IPlot=32,8
POS=175., 175., 350., 225.
IPlot=3,8
POS=625., 175., 350., 225.
IPlot=20,8

// END

PAT20037
PAT20038
PAT20039
PAT20040
PAT20041
PAT20042
PAT20043
PAT20044
PAT20045
PAT20046
PAT20047
PAT20048
PAT20049
PAT20050
PAT20051
PAT20052
PAT20053
PAT20054
PAT20055
PAT20056
PAT20057
PAT20058
KM1=-1.0
KN2=2.0
KN3=3.0
KN4=4.0
KM4=-4.
KN5=5.0
KM5=-5.0
KN6=6.
KN144=144.0
KMWX=1.35E-6
KMWR=4.428E8
KNTR=377.0
KCIGN=1.0

SET SYSTEM PARAMETER VALUES FROM SUBROUTINES CONST1, CONST2, CONST3
CALL CONST1
CALL CONST2
CALL CONST3
LDC=5.0
KMRL=0.111

CATVD=5.0
CAIVD=5.0

INITIALIZE VARIABLES FOR COMPUTING THEIR TIME DERIVATIVES
C2DVO=0.0
CP1ST0=4.5464
CTRHO0=3.6667
CXGG0=3.0
C2DV=0.0
CP1ST=4.5464
CTRHO=3.6667
CXGG=3.0

INITIALIZE JCOUNT, USED IN SELECTING PRINTING INTERVALS
JCOUNT=0

53 CONTINUE

COMPUTE LOAD REFERENCE MOTOR, VALVE, AND LOAD RATE LIMITS

COMPUTE TIME DERIVATIVES

FC2DV=(C2DV-C2DV0)/TSTEP

FCP1ST=(CP1ST-CP1ST0)/TSTEP

FCTRH0=(CTRHO-CTRHO0)/TSTEP

FCXGG=(CXGG-CXGG0)/TSTEP

C2DV0=C2DV

CP1ST0=CP1ST

CTRHO0=CTRHO

CXGG0=CXGG

TEST 1

IF (T.LT.10.0.OR.T.GT.100.0) GO TO 201

LDC=5.0-0.0125*(T-10.0)

201 CONTINUE

PUT 6,T,WAR,WP5O,WHIP,PSSG,TSSO,TR50,RPSO,LDC,XDRW,XDEW,XGG,

WGR,WSY,WRY,MWD,WCW,NCP,NFD,C2TR,C3TR,C4TR,C6TR,CACVD,NTR

JCOUNT=JCOUNT+1

IF (T.LT.1.05) GO TO 70

IF (JCOUNT.NE.IFIX(COUNT)) GO TO 800

JCOUNT=0

70 CONTINUE

PUT 5,0,T,NEP,HHHO,HECO,YDRW,RDS,NRP,THWM,RPSO,HPSO,RS5O,HSSO,

* RS5O,RR50,HRH0,RCRD,N50,HLHO,VDEW,RDES,NFD,NID,C35D,C5AR,

* C5FL,C3FN,C2GR,C2FT,C3FV,C7FV,C3DV,C8DV,C5RH,C5SY,CARD,CFLD,

* CFND,CGRD,CFTD,CFWD,CDWD,CXGGD,CSYD,C2TR,C4TR,CACVD,DETA,

* MWGNPU,MWTRPU,MD5O,WF50,WR50,W5RS,WPSO,WSS1,HSSO,WH5P,WIP,WLP,

* WC5,WS5,WRY,WFL,WAR,WGR,XXG,XDRW,XDEW,P5PO,P5RS,PPSO,PSSO,P5CO,

* P1ST,PH5O,PRHO,PCRD,PCPO,PDFS,PFPO,P5N,TDRS,T5SO,T5HO,TH5O,TR5O,
CONTINUE

SET CONTROLLED INPUT VALUES

CALL LIMCHK(CARD)
CALL XDUCE(KCL, KCU, KAVL, KAVU, CARD, AVF)
CALL LIMCHK(CFLD)
CALL XDUCE(KCL, KCU, KFWLL, KFLLU, CFLD, WFL)
CALL LIMCHK(CFND)
CALL XDUCE(KCL, KCU, KAVL, KAVU, CFND, AVI)
CALL LIMCHK(CGRD)
KWGR=KC2GR*WFL
CALL XDUCE(KWGR, KGRU, KCL, KCU, KWGR, KGWCR)
CALL CHECK(CGRD, KGWCR, KCL)
CALL XDUCE(KCL, KCU, KGWRL, KGRU, CGRD, WGR)
CALL LIMCHK(CFTD)
CALL XDUCE(KCL, KCU, KFWTL, KFWTU, CFTD, WFT)
CALL LIMCHK(CFWD)
CALL XDUCE(KCL, KCU, KAVL, KAVU, CFWD, AVF)
CALL LIMCHK(CDWD)
CALL XDUCE(KCL, KCU, KAVL, KAVU, CDWD, ADV)
CALL LIMCHK(CXGGD)
CRY=KCIRY*CXGGD
CALL XDUCE(KCRYL, KCRYU, KRYL, KRYU, CRY, WRY)
CALL XDUCE(KCL, KCU, KXGGGL, KXGGU, CXGGD, XGG)
ZXGG=XGG*57.3
CALL LIMCHK(CSYD)
CSY=CSYD*KCSY
CALL XDUCE(KCSYL, KCSYU, KWSYL, KWSYU, CSY, WSY)
CALL XDUCE(KNO, KN5, KAVL, KAVU, CACVD, AVG)
ATV=1.0
AIV=1.0
**STATE PROPERTIES**

CALL DRSTAT (RDRS, RDRW, HDRW, HDRS, PDRS, TDRES)
CALL DESTAT (RDES, RDEW, HDEW, HDES, PDES, TDES)
CALL SHSTAT (RPSO, HPSO, PPSO, TPSO, PPSO)
CALL SHSTAT (RSSO, HSSO, PSSO, TSSO, SSSO)
CALL SHSTAT (RSCO, HSCO, PSCO, TSCO, SCO)
CALL RHSTAT (RHRHO, HRHRO, PRHO, TRHO, SRHRO)
CALL FWSTAT (HECO, PDRS, RECO, TECO)
CALL CWSTAT (HLHO, PDES, RDVO, TDVO)

**STEAM FLOW**

PPSO=PDRS-PPSO
CALL SHFLOW (RDRS, PPSO, KFPS, WDRS)
ACV=ACV
WTV=KCV*ACV*SQRT(PPSO*RSSO)
WPSX=KNO
IF (WTV.LT.KWTV) WPSX=101.56064-0.39882*WDRS+4.8626E-4*WDRS*WDRS
PPSO=PPSO-PPSO
CALL SHFLOW (RSSO, PSSO, KFSS, WSS)
WPSO=WSS+WPSX-WSY
WHP=KHP*SQRTRSCO*PSCO)
WSSX=KNO
IF (WTV.LT.KWTV) WSSX=WFT
WSSO=WTV + WSSX
CALL HPEXT (WHPO, WHPAUX, W1HHS)
WHPO=WHPO-WHPAUX-W1HHS
WRH1=WHPO+WRY
WIV=KIP*AIW*SQRTRHRO*PRHO)
WIP=WIV

**TURBINE**

EHP=0.589+2.317E-4*WHP
PHPO=PRHO*KFRH*WIP*WIP/RRHO
CALL HPSTAT (SSSO, PHPO, EHP, HSSO, HHPO, THPO)
EIP=0.814
CALL CRSTAT (SRHO, RCRO, EIP, HRHO, HCRO, PCRO, TCRO)
WLP=KLP*SQRTRCRO*PCRO)
PCN=KOPCN+K1PCN*PCRO+K2PCN*PCRO*PCRO
OYLPO=KQYLPD
CALL CNSTAT (PCN, OYLPO, HLPO, RLPO, SLPO, TCN, RCNO, HCNO)
MWHP=WHP*(HSSO-HHPO)*KJ
KEIP=0.93
KELP=0.93
MWIP=WIP*(HRHO-HCRO)*KJ*KEIP
MWLP=WLP*(HCRO-HLPO)*KJ*KELP
MWTRQ=MWHP+MWIP+MWLP
MWO=MWTRQ*KMWX
MWGNP=KN2*SIN(DELTA)
MWGN=MWGNP*KMWR
MWTRP=MWTRQ/KMWR
CALL IPEXT (WIP, PRHO, PCRO, HRHO, HCRO, W2HHS, W3HHS, H2HHS1, P2HHS,
* P3HHS0)
* WIPFTX=WFT
IF (WSSX.GT.KNO) WIPFTX=KNO
WIP= WIP-W2HHS-W3HHS-WIPFTX
C ** WATER SIDE
CALL CWFLOW (PCN, RCNO, RNDV, PDES, NCP, ADV, KNCP, WCW, WCP, PCPO, PLHO,
* WLHX, WDVO, FCP)
CALL CPSTAT (RCNO, PCPO, HCPO, TCPO)
CALL CWSTAT (HLHO, PLHO, RLHO, TLHO)
CALL FWFLOW (WRW, WSY, RDEW, PDES, PDRS, Reco, AFV, NFP, WFP, WFW,
* WFW2, PBPO, PFPO, PHD0, PFVD, EFPO)
CALL FPSTAT (RDEW, PFPO, HFPO, TFPO)
HFV0=HFPO
CALL FWSTAT (HHHO, PHHO, RHHO, THHO)
RDC=RDRW
CALL RWFLOW (KNR, RDC, RDRW, NRP, PDRS, WRW, WRP, WWDO, PCDO, PRPO, ERP)
C ** HEAT BALANCE IN DOWNCOMERS, SUPERHEAT AND REHEAT SPRAYS
HDC1= (WFW*HECO+HDRW*(WRW-WFW))/WRW
HSS1= ((WPSO-WPSX)*HPSO + WSY*HFPO)/WSS1
HRH1= (WHPO*HHPO+WRY*HFPO)/WRH1
CALL FWSTAT (HDC1, PDRS, RDC1, TDC1)
CALL RWSTAT(RDC1, PCDO, HDCO, TDCO, SDCO)
CALL RWSTAT(RDC1, PRPO, HRPO, TRPO, SRPO)
CALL SYSTAT (HSS1, PSO, RSS1, TSS1, SSS1)
CALL RYSTAT (HRH1, PHPO, RRH1, TRH1, SRH1)
CALL LPEXT (WLP, PHC3, PCN, HCRO, HLPO, WILHS, W2LHS, W2EX, WLHST, P1LHS, P2LHS, P3LHS, H1LHS1, H2LHS1, HDE)
CALL LSSTAT (P1LHS, H1LHSO, R1LHSO, T1LHSO)
CALL LSSTAT (P2LHS, H2LHSO, R2LHSO, T2LHSO)
CALL LSSTAT (P3LHS, H3LHSO, R3LHSO, T3LHSO)
QLH = WILHS*(H1LHS1-H3LHSO) + W2LHS*(H2LHS1-H2LHSO)

WHHST = W1HHS*W2HHS+W3HHS

CALL LSSTAT (P3HHSO, H3HHSO, R3HHSO, T3HHSO)
QHH = W1HHS*HPHO+W2HHS*H2HHS1+W3HHS*HCRO-WHHST*H3HHSO
WHHS = WHHST

IF (WFW.LT.KWFW) WHHS = KNO

C * * * MEAN FLOW, TEMPERATURE, DENSITY, AND ENTHALPY
CALL AVERAG (WDRS, WPSSO, RDRS, RPSO, HDRS, HPSO, WPSE, RPSE, HPSE)
CALL AVERAG (WSS1, WSSO, RSS1, RSSO, HSS1, HSSO, HSSE, RSSE, HSSE)
CALL AVERAG (WRH1, WIP, RRHI, RRHO, HRHI, HRHO, RRHE, HRHE)
CALL AVERAG (PHHO, PDRO, HHHO, HEHO, PCRO, PLHO, PEC, HECE, PLHE)
CALL AVERAG (PFVO, PHHO, HFVO, HHHO, HCRO, HLHO, PHHE, HHHE, HLHE)
CALL LSSTAT (RPSE, HPSE, PPSE, TPSE, SPSE)
CALL LSSTAT (RSSE, HSSE, PSSE, TSSE, SSSE)
CALL RHSTAT (RRHE, HRHE, PRHE, TRHE, SRHE)
CALL FWSTAT (HECE, PECE, RECE, TECE)
CALL FWSTAT (HHHE, PHHE, RRHE, THHE)
CALL CWSTAT (HLHE, PLHE, RLHE, TLHE)

C ** AIR AND GAS FLOW
CALL ARFLOW (KNFD, KNID, NFD, NTD, HGR, WFL, AVF, AVI, WAR, WWG, WFD, WGO, WID, PAAO, PFDO, PFN, PECGO, PAPGO, PIDO, EFD, EID)

C ** AIR SIDE HEAT TRANSFER
TAHAO = KTAT+KTAHAD
TAPA = TAHAO+KTAHAD
WARD = KAFR*WFL

C ** FURNACE
UXGG = K1XGG+K2XGG*SIN(XGG)/COS(XGG)
NUMBER OF OPERATING GUNS

\( NGG = NG1 + NG2 + NG3 + NG4 + NG5 \)

\( UNGG = (NG1 \times K1NG + NG2 \times K2NG + NG3 \times K3NG + NG4 \times K4NG + NG5 \times K5NG) / (NGG \times KXWGE) \)

\( UWWGM = KUWWGM \times UXGG \times UNGG \)

\( TGR = K1TGR + K2TGR \times WFL \)

CALL FNXFER (TWW, WWG, WAR, SAR, TAPAO, WFL, SFL, TFL, KHF, EFL, WGR, SGR, TGR, UWWGM, YWGR, TPSE, KUPSR, TFFN1, TWWGE, TWWGO, QWWGM, QPSR, SWWGO)

\( OWWM = KUOWWM \times (TWWM - TDRS)^3 \)

GAS SIDE HEAT TRANSFER

\( QSSR = KNO \)

\( QRHR = KNO \)

\( QECR = KNO \)

CALL HXFER (WWW, WPSE, KUPSGM, KUPSMS, TPSE, TWWGO, QPSR, YWGR, TPSGO, QPS, TPSME, SPSSGO)

CALL HXFER (WWW, WSSE, KUSSGM, KUSSMS, TSSE, TPSGO, QSSR, YWGR, TSSGO, QSS, TSSME, SSSGO)

CALL HXFER (WWW, WRHE, KURHGM, KURHMS, TRHE, TSSGO, QRHR, YWGR, TRHGO, QHR, TRHME, SRHGO)

CALL HXFER (WWW, WFW, KUECGM, KUECMW, TECE, TRHGO, QECR, YWGR, TECGO, QEC, TECME, SECGO)

\( TLHME = KLHM \times TLHO \)

\( TTHME = KHHM \times TTHO \)

EFFECTIVE MASSES

\( MHHW = KLHHW + KVHH \times HDRH / (KSWWM \times TWWM) \)

\( MPSS = RPSF \times KVPS \)

\( MPSE = MPSS + KMPSM \times KSPSM \times TPSME / HPSE \)

\( MSSE = RSSE \times KVSS \)

\( MSSC = MSSS + KMSMM \times KSSSM \times TSSME / HSSE \)

\( MRHS = RRHE \times KVRRH \)

\( MRHE = MRHS + KMRHM \times KSRHM \times TRHME / HRHE \)

\( MECW = RECE \times KVFC \)

\( MECE = MECW + KMCECM \times KSECME / HECF \)

\( MLHW = RLHE \times KVHLH \)

\( MLHE = MLHW + KMLHM \times KSLHM \times TLHME / HLHE \)

\( MHHW = RHHE \times KVHH \)
C

MHHE=MHHW+KMHMM*KSHHM*THHME/HHHE

AIR AND GAS DENSITIES
RAHAD=0.0661
RAPG0=0.044

DRUM INTERMEDIATE VARIABLES

HDRO=HDR=HDRS-HDRW
HWWO=HRPO+QWWMM/WWWO
QYWW=(QWWMM+WRW*(HRPO-HDRD))/(WRW*HDRD)
CALL DESTMR(WDRS,WHP,WDERP,WDEWH,WDEBD,HDERP,HDEWH)
HDEBD=HDES
WDRBD=KN2*WDEBD,
WDES=WDERP+WDWH+WDDE
QDES=WDERP*HDERP+WDWH*HDEWH+WDDE*HDEBD
Z206=WFW-WRW+WWO-WDRS-WDRBD
Z209=WWW+WWO-(WRW-WFW)*HDRW-WDRS*HDRS-WDRBD*NDRS
Z226=WDO+WHHS*WDE+WDES-WF
Z229=WDO+HLH+WHMS*H3H*WDE+WDEWH+WDESWH+WDERP
CALL DRUM(KVDR,VRW,DR,WDRS,DRW,HRW,DRS,K2,K3,K5,K6,K7,K9,K10,
*Z206,Z209,F1DR,F2DR)
CALL DRUM(KVDE,VDEK,RDE,RDEW,HDE,HDES,K22,K23,K25,K26,K27,
*K29,K30,Z226,Z229,F1DE,F2DE)

PUMP AND FAN INPUT TORQUE

CALL TORQUE(NELEC,VELEC,KNRPM,NRP,LRP,SRP,SQPM,TQP,MRP)
CALL TORQUE(NELEC,VELEC,KNCPM,NCP,ECP,SCP,TQCP,MC)
CALL TORQUE(NELEC,VELEC,KNFDM,NFD,EFD,FD,SQDFM,TQFD,MC)
CALL TORQUE(NELEC,VELEC,KNIDM,NID,EID,SID,SIDM,TQID,MC)
HFT=HCR
IF(WSSX.GT.KNO).HFT1=HSSO
CALL FTPURB(WFT,HFT1,NFP,TQFP,MC)

XDEW=K1XDEW+K2XDEW+K3XDEW+VDEW+VDEW
XDRW=K1XDRW+K2XDRW+VDRW+K3XDRW+VDRW+VDRW
PHHD=PFPO-PHHD.
P1ST=KIPIST*WHP

C CONTROL SYSTEM

C * * * BOILER MASTER DEMAND
KPSSO=K1PSS+K2PSS*LDC
CALL CHECK (KPSSO,K4PSS,K3PSS)
KPSSO=K4PSS
CALL XDUCER (KPSSO,KPSSO,KCL,KCU,KPSSO,KCPSSO)
CALL XDUCER (KPSSO,KPSSO,KCL,KCU,PSSO,CPSSO)
C1MD=KCPSSO-CPSSO
C2MD=C1MD*KC1MD
CALL LIMCHK (C3MD)
C4MD=C2MD+C3MD
CALL LIMCHK (C4MD)
CALL XDUCER (KCTRL,KCTRL,KCL,KCU,CTRL,CTRL)
CALL XDUCER (KCTRL,KCTRL,KCL,KCU,KCTRL,KCTRL)
C5MD=KC2MD*(KC2TR-CNTR)
CBMD=C4MD+C5MD
CALL LIMCHK (CBMD)

C * * * AIR FLOW CONTROL
CALL XDUCER (KWRL,KWRL,KCL,KCU,WAR,CWAR)
CWFL=CWFL
C2AR=CBMD
IF (CWFL .GT. CBMD) C2AR=CWFL
IF (C2AR.LT.KC2ARL) C2AR=K2ARL
C3AR=C2AR-CWAR
C4AR=C3AR*KC4AR
CALL LIMCHK (C5AR)
C6AR=C4AR+C5AR
CALL LIMCHK (C6AR)

C * * * FUEL FLOW CONTROL
C2FL=CBMD
IF (CWAR.LT.CBMD) C2FL=CWAR
C3FL=C2FL-CWFL
C4FL=C3FL*KC4FL
CALL LIMCHK (C5FL)
C6FL=C4FL+C5FL
CALL LIMCHK(C6FL)

CALL XDUCE (KCPFNL,KPFNU,KCL,KCU,PFN,CPFN)
CALL XDUCE (KCPFNL,KPFNU,KCL,KCU,PFN,CPFN)
C1FN=KCPFNL-CPFN
C2FN=C1FN*KC1FN
CALL LIMCHK (C3FN)
C4FN=C2FN+C3FN
CALL LIMCHK(C4FN)
C5FN=C4FN*KC2FN*CWAR
CALL LIMCHK(C5FN)

C * * * GAS RECIRCULATION CONTROL
KCGR=KNO
IF (ABS(XGG).GT.KNPOBT) KCGR=KNI
C1GR=CXGGD-KCXGG
CALL LIMCHK (C2GR)
C2FT=C1FT+C2FT
CALL LIMCHK (C3FT)

CALL XDUCE (KPFVDL,KPFVDU,KCL,KCU,PFVD,CPFVD)
CALL XDUCE (KPFVDL,KPFVDU,KCL,KCU,PFVD,CPFVD)
C1FT=K1FT*(KCPFVD-CPFVD)
CALL LIMCHK (C2FT)
C3FT=C1FT+C2FT
CALL LIMCHK (C3FT)

ZWFW=WFW+WSY
CALL XDUCE (KWFWL,KWFNU,KCL,KCU,ZWFW,CWFW)
CALL XDUCE (KPISTL,KPISTU,KCL,KCU,PIST,CP1ST)
CALL XDUCE (KXDRWL,KXDRWU,KCL,KCU,XDRW,CXDRW)
CALL XDUCE (KXDRWL,KXDRWU,KCL,KCU,XDRW,CXDRW)
CALL XDUCE (KPDRL,KPDSL,KCL,KCU,PDRL,CPDRL)
C1FV=CXDRW+KC4FV*CPDRL
C2FV=KC1FV*(KCXDRW-C1FV)
CALL CHECK (C3FV,KN5,KN5)
C4FV=C2FV+C3FV
CALL CHECK (C4FV, KN5, KM5)
C5FV=KC2FV*(CP1ST-CWF)
C6FV=KC3FV*(C4FV+C5FV)
CALL LIMCHK (C7FV)
C8FV=C6FV+C7FV
CALL LIMCHK (C8FV)

CALL XDECER (KPRHOLKPRHOU,KCL,KCU,PRHO,CPRHO)
CALL XDECER (KWCHL,KWCWU,KCL,KCU,WCH,WCW)
CALL XDECER (KXDEWL,KXDEWU,KCL,KCU,XDEW,CXDEW)
CALL XDECER (KXDEWL,KXDEWU,KCL,KCU,KXDEW,KCXDEW)
C1DV=KCXDEW-CXDEW
C2DV=C1DV*(CIDV
CALL CHECK (C3DV, KN5, KM5)
C4DV=KTC2DV*FC2DV
C5DV=C2DV+C3DV+C4DV
CALL CHECK (C5DV, KN5, KM5)
C6DV=KC4DV*CPRHO-KC5DV*CWCW
C7DV=KC2DV*(C5DV+KC6DV*C6DV)
CALL LIMCHK (C8DV)
C9DV=C7DV+C8DV
CALL LIMCHK (C9DV)

CALL XDECER (KTRHOL,KTRHOU,KCL,KCU,TRHO,CTRHO)
KTRH=K1TRH*K2TRH*WHP
CALL CHECK (KTRH,K4TRH,K3TRH)
CALL XDECER (KTRHOL,KTRHOU,KCL,KCU,KTRH,KCTRHO)
C1RH=KCTRHO
C2RH=FCP1ST*KC3RH
C3RH=C1RH+C2RH-CTRHO
C4RH=C3RH*KC1RH
CALL LIMCHK (C5RH)
C6RH=C4RH+C5RH
CALL LIMCHK (C6RH)
C7RH=FCTRHO*KC2RH
C8RH=C6RH+C7RH
CALL LIMCHK(C8RH)
CWRY=C8RH
CXGG=C8RH

C * * * SUPERHEAT TEMPERATURE CONTROL

KTSS=K1TSS+K2TSS*WHP
CALL CHECK (KTSS,K4TSS,K3TSS)
CALL XDUCER (KTSSOL,KTSSOU,KCL,KCU,TSSO,CTSSO)
CALL XDUCER (KTSSOL,KTSSOU,KCL,KCU,KTSS,KCTSS)
C2SY=KTSS
C3SY=C2SY-CTSSO
C4SY=C3SY*KCLSY
CALL LIMCHK (C5SY)
C6SY=C4SY+C5SY
CALL LIMCHK(C6SY)
C7SY=FCP1ST*KC2SY
C8SY=FCXGG*KC3SY
C9SY=C6SY+C7SY+C8SY
CALL LIMCHK(C9SY)
CWSY=C9SY

C * * * TURBINE CONTROL

CALL XDUCER (KMWTWL,KMWTRU,KNO,KN6,MWTRU,CMWTRU)
CALL XDUCER (KMWTWL,KMWTRU,KNO,KN6,MWGN,CMWGN)
CNTR=KCSTR-CNTR
C1TR=KCSTR*(LDC-CMWTRU)
CALL CHECK (C2TR,KN1,KM1)
C3TR=C1TR+C2TR*LDC
CALL CHECK (C3TR,KN5,KNO)
CALL CHECK (C4TR,KN5,KNO)
C5TR=KC2TR*CNTR
C6TR=C5TR/KCVREG+C4TR
CALL CHECK (C6TR,KN5,KNO)
CALL CHECK (CACVD,KN5,KNO)

C * * * DYNAMIC EQUATIONS
C CALCULATE F OF Y
C
\[
\begin{align*}
F(1) &= (TQFP1-WFP*(PFPO-PDES)*KN144/(NFP*RDEW*EFP))/KJFPE \\
F(2) &= (TWF*(HFVO-MHVO)*QH)/MHHE \\
F(3) &= (TWF*(HHVO-HECO)*QEC)/MECE \\
F(4) &= F1DR \\
F(5) &= F2DR \\
F(6) &= (TQRP1-WRP*(PRPO-PDCO)*KN144/(NRP*RDC*ERP))/KJRP \\
F(7) &= (QWWCH-QWWM)/(MWWME*KSWWM) \\
F(8) &= (WDRS*WPSO)/KVPS \\
F(9) &= (WDRS*HDRS-WPSO*QPS)/MPSE \\
F(10) &= (WSS1-WSSO)/KVSS \\
F(11) &= (WSS1*HSS1-WSSO*HSSO+QSS)/MSSE \\
F(12) &= (WTV-WHP)/KVSC \\
F(13) &= (WRHI-WIV)/KVRH \\
F(14) &= (WRHI*HRHI-WIV*HRHO+QRH)/MRHE \\
F(15) &= (WIP*WLP)/KVCRE \\
F(16) &= (MWTR*MWGN)/(NTR*KJTRF) \\
F(17) &= (TQCP1-WCP*(PCPO-PCTN)*KN144/(NPC*RCTN*ECN))/KJCP \\
F(18) &= (QCW*(HCPO-MLHD)+QLH*KQGC)/MLHE \\
F(19) &= F1DE \\
F(20) &= F2DE \\
F(21) &= (TQFD1-WFD*(PFDO-PAHAO)*KN144/(NFD*RAHAO*EFD))/KJFD \\
F(22) &= (TQID1-WID*(PIDO-PAGDO)*KN144/(NID*RAPDO*EID))/KJID \\
F(23) &= C2MD/KTCIMD \\
F(24) &= C4AR/KTC1AR \\
F(25) &= C4FL/KTCLFL \\
F(26) &= C2FN/KTC1FN \\
F(27) &= C1GR*KC1GR*KCGR \\
F(28) &= C1FT/KTC1FT \\
F(29) &= C2FD/KTCLIF \\
F(30) &= C6FV/KTCL2F \\
F(31) &= C2DV/KTC1DV \\
F(32) &= C7DV/KTC3DV \\
F(33) &= C4RH/KTC1RH \\
F(34) &= C4SY/KTC1SY \\
F(35) &= (C6AR-CARD)/KTC2AR \\
F(36) &= (C6FL-CFLD)/KTC2FL
\end{align*}
\]
F(37) = (C5FN - CFND) / KTC2FN
F(38) = (C2GR - CGRD) / KTC1GR
F(39) = (C3FT - CFTD) / KTC2FT
F(40) = (C8FV - CFWD) / KTC3FV
F(41) = (C9DV - CDWD) / KTC4DV
F(42) = (CXXG - CXXGD) / KTC2RH
F(43) = (CSY - CSYD) / KTC2SY
F(44) = C1TR / KTC1TR
F(45) = (C4TR - C4TRD) / KTC2TR
F(46) = (C6TR - CACVD) / KTC3TR
F(47) = KClGN*(NTR - NELEC)
RETURN
END

C C C

* * * SUBROUTINES

SUBROUTINE DRSTAT (RS, RW, HW, HS, PS, TS)
C COMPUTE DRUM WATER AND STEAM PROPERTIES
RS2 = RS*RS
RS3 = RS2*RS
RW = 49.27105 - 2.13733*RS + 0.03348*RS2
HW = 526.5957 + 31.0437*RS - 0.62086*RS2
HS = 1241.713 - 21.3442*RS + 0.20998*RS2
PS = 11.1877 + 500.267*RS - 26.4031*RS2 + 0.046944*RS3
TS = 458.084 + 48.2088*RS - 3.2326*RS2 + 0.07249*RS3 + 459.67
RETURN
END

SUBROUTINE DESTAT (RS, RW, HW, HS, PS, TS)
C COMPUTE DEAERATOR WATER AND STEAM PROPERTIES
RS2 = RS*RS
RS3 = RS2*RS
RW = 60.45805 - 19.61207*RS
HW = 118.26296 + 1905.63721*RS - 8414.69018*RS2 + 15688.03769*RS3
HS = 1143.4984 + 224.556*RS
PS = -1.04181 + 419.17159*RS + 131.32803*RS2
TS=150.48933+1896.4155*RS-8447.87828*RS2+15757.43239*RS3+459.67
RETURN
END

C
SUBROUTINE SHSTAT (R,H,P,T,S)
C
COMPUTE SUPERHEATER STEAM PROPERTIES
RH=R*H
P=-291.36-964.04*H+0.21781*H+1.1815*RH
T=-1745.1+129.1*R+1.8107*H-0.066313*RH+459.67
RETURN
END

C
SUBROUTINE SYSTAT (H,P,R,T,S)
C
COMPUTE SUPERHEAT SPRAY SECTION OUTLET STEAM PROPERTIES
PH=P*H
R=-1.9033+1.3862E-3*H+6.7569E-3*P-3.7659E-6*PH
T=-1654.5+1.7443*H+0.34809*P-2.0743E-4*PH+459.67
S=0.46472+7.9581E-4*H-1.1683E-5*P-1.8658E-8*PH
RETURN
END

C
SUBROUTINE HPSTAT (S,P,E,H1,H0,T)
C
COMPUTE HP TURBINF EXHAUST STEAM PROPERTIES
H1=-485.23+1065.28*S+0.232*P
H0=H1-E*(H1-HI)
T=-1639.74+0.119*P+1.682*HO
RETURN
END

C
SUBROUTINE RHSTAT (R,H,P,T,S)
C
COMPUTE REHEAT STEAM PROPERTIES
RH=R*H
P=27.061-1014.1*R-1.7354E-2*H+1.2279*RH
S=1.5015+3.8406E-3*R+6.4181E-4*H-0.10938*ALOG(RH)
SUBROUTINE RYSTAT (H,P,R,T,S)
  COMPUTE REHEAT SPRAY SECTION OUTLET STEAM PROPERTIES
  P2=P*P
  H2=H*H
  P2H2=P2*H2
  PH=P*H
  R=-4.2661E-2+3.0892E-5*H+6.2923E-3*P-3.4891E-6*PH
  T=-550.34-0.39473*H+1.6795*P-1.1363E-3*PH+9.3611E-4*H2
  S=-0.473+2.47E-3*H-2.7302E-4*P-3.1543E-7*PH-5.6242E-7*H2
  RETURN
END

SUBROUTINE CRSTAT (SREHI,H,O,P,T)
  COMPUTE CROSS-OVER PIPE STEAM PROPERTIES
  H1=-1211.8+683.58*R+1384.39*S
  HO=HI-E*(HI-HI)
  P=-381.05+0.2783*HO+668.609*R
  T=-2074.92+2.004*HO+71.326*R+459.7
  RETURN
END

SUBROUTINE CNSTAT (P,QY,HLPO,RLPO,SLPO,T,RWHW)
  COMPUTE CONDENSER WATER AND STEAM, AND LP TURBINE EXHAUST STEAM
  PROPERTIES
  P2=P*P
  P3=P2*P
  P4=P3*P
  P5=P4*P
  RW=62.34525-0.28884*P
  RS=2.0E-5+3.21E-3*P-3.2E-4*P2+8.0E-5*P3
  Hw=-25.80341+365.63647*P-878.10799*P2+1305.19564*P3-1002.14134*P4
  RETURN
END
* +305.01148*P
HS=1075.86634+28.99664*P
T=6.2916+364.60878*P-877.92172*P2+1311.45669*P3-1012.17821*P4
* +309.49426*P5+459.67
SW=-0.05216+0.74502*P-1.84119*P2+2.15801*P3-2.12467*P4+0.64777*PS
SS=2.21936-0.58673*P+0.51817*P2-0.16668*P3
HLPO=HW+QY*(HS-HW)
RLPO=RW+QY*(RS-RW)
SLPO=SW+QY*(SS-SW)
RETURN
END

SUBROUTINE CPSTAT (R,P,H,T)
C COMPUTE CONDENSATE PUMP EXHAUST WATER PROPERTIES
P2=P*P
R2=R*R
PR=P*R
H=2668.5+24.790*R-1.7378*P+0.027264*PR-1.0745*R2+1.0266E-4*P2
T=-7918.5+366.86*R-1.7341*P+0.027146*PR-3.8292*R2+1.0439E-4*P2
* +459.67
RETURN
END

SUBROUTINE CWSTAT (H,P,R,T)
C COMPUTE CONDENSATE (WATER) PROPERTIES
P2=P*P
H2=H*H
HP=H*P
R=62.633+2.3086E-4*P -7.847E-3*H +8.4526E-7*HP-1.3075E-7*P2
* -4.4171E-5*H2
T=31.363-3.6799E-3*P+1.0224*H+2.2581E-6*HP+1.3108E-6*P2
* -9.8358E-5*H2 +459.67
RETURN
END

SUBROUTINE FPSTAT (R,P,H,T)
COMPUTE FEEDPUMP EXHAUST WATER PROPERTIES

\[
R^2 = R \times R \\
P^2 = P \times P \\
PR = P \times R \\
H = 9024.90 + 0.11479 \times P - 90.346 \times R - 2.0528 \times 10^{-5} \times R - 1.8098 \times 10^{-3} \times PR \\
T = -1268.8 + 82.714 \times R + 0.10301 \times P - 1.444 \times 10^{-3} \times PR - 0.97073 \times R^2 \\
* -1.801 \times 10^{-6} \times P^2 + 459.67 \\
\]
RETURN
END

SUBROUTINE FWSTAT (H, P, R, T)
COMPUTE FEEDWATER PROPERTIES

\[
P^2 = P \times P \\
H^2 = H \times H \\
HP = H \times P \\
H^2P^2 = H^2 \times P^2 \\
R = 71.143 - 5.0 \times P + 3.2658 \times 10^{-3} \times P + 1.1688 \times 10^{-5} \times HP + 6.1657 \times 10^{-7} \times P^2 \\
* -3.0631 \times 10^{-7} \times H^2 - 2.3095 \times 10^{-12} \times H^2P^2 \\
T = 146.98 - 0.084086 \times P + 0.79716 \times H + 1.8034 \times 10^{-4} \times HP + 1.0481 \times 10^{-5} \times P^2 \\
* -1.8231 \times 10^{-4} \times H^2 - 3.812 \times 10^{-11} \times H^2P^2 + 459.67 \\
\]
RETURN
END

SUBROUTINE RWSTAT (R, P, H, T, S)
COMPUTE RECIRCULATING WATER PROPERTIES

\[
RP = R \times P \\
R^2 = R \times R \\
P^2 = P \times P \\
T = 163.60 + 29.199 \times R + 4.2676 \times 10^{-3} \times P - 7.2625 \times 10^{-4} \times RP - 3.45712 \times R^2 \\
* + 3.1826 \times 10^{-7} \times P^2 \\
H = 658.45 + 11.887 \times R + 3.9498 \times 10^{-2} \times P - 5.6955 \times 10^{-3} \times RP - 3.1712 \times R^2 \\
* -3.2973 \times 10^{-7} \times P^2 \\
S = 7395 + 1.7257 \times 10^{-4} \times R + 2.8021 \times 10^{-5} \times P - 4.7572 \times 10^{-7} \times RP - 3.7372 \times 10^{-4} \times R^2 \\
* + 2.4354 \times 10^{-10} \times P^2 \\
\]
RETURN
END
SUBROUTINE LSSTAT (P,H,R,T)
COMPUTE FEEDWATER HEATING STEAM PROPERTIES
P2=P*P
P3=P2*P
H=126.8737+4.15377*P-0.04224*P2+1.8E-4*P3
R=60.53211-0.04603*P
T=159.09642+4.1294*P-0.04236*P2+1.8E-4*P3+459.67
RETURN
END

SUBROUTINE LWSTAT (P,H,R,T)
COMPUTE FEEDWATER HEATING STEAM PROPERTIES
P2=P*P
P3=P2*P
H=32.5+43.49757*P-7.38007*P2+0.51472*P3
R=62.34525-0.28884*P
T=64.05911+43.60199*P-7.40196*P2+0.51616*P3+459.67
RETURN
END

SUBROUTINE HPEXT (W,WAUX,WH)
COMPUTE HP TURBINE STEAM EXTRACTION
W2=W*W
W3=W2*W
WAUX=11.15813+1.0E-2*W
RETURN
END

SUBROUTINE IPEXT (W,P1,PO,H1,H0,W2X,W3X,H2X,P2X,P3X)
COMPUTE IP TURBINE STEAM EXTRACTION - FLOW AND PROPERTIES
P02=PO*PO
P03=P02*PO
HT=H1+H0
PT=P1+PO
W2=W*W
W3=W2*W
W2X=1.72456+2.72665F-2*W+2.7797E-5*W2
W3X=-22.8870+0.17584*W-1.9372E-4*W2+1.0479E-7*W3
P2X=0.44994*PT
P3X=32.43505-0.4506405*PO+6.85193E-3*PO2-1.64694E-5*PO3
H2X=0.503488*HT
RETURN
END

C
SUBROUTINE LPEXT (WLP,PCRO,PCN,HCR0,HLP0,W1LHS,W2LHS,WDEX,WLHST,
P1LHS,P2LHS,P3LHS,HILHS1,H2LHS1,HDEX)

C COMPUTE LP TURBINE STEAM EXTRACTION - FLOW AND PROPERTIES
WLP2=WLP*WLP
WLP3=WLP2*WLP
PLPT=PCRO+PCN
HLPT=HCR0+HLP0
W2LHS=-6.73762+8.48847E-2*WLP-1.3439E-5*WLP2
WDEX=4.9696+1.75669E-2*WLP+3.0622E-5*WLP2
WLHST=W1LHS+W2LHS
P1LHS=0.13571*PLPT
P2LHS=0.029372*PLPT
P3LHS=0.53009-7.1492E-2*P1LHS+6.3411E-3*P1LHS*P1LHS-
* 1.298E-4*P1LHS*P1LHS*P1LHS
H1LHS1=0.498953*HLPT
H2LHS1=0.458364*HLPT
HDEX=0.533921*HLPT
RETURN
END

C
SUBROUTINE DESTM (WDRS,WHP,WDERP,WDEWH,WDEBD,HDERP,HDEWH)

C COMPUTE DEAERATOR HEATING STEAM - FLOW AND PROPERTIES
WHP2=WHP*WHP
WDRS2=WDRS*WDRS
WDRS3=WDRS2*WDRS

PAT10757
PAT10758
PAT10759
PAT10760
PAT10761
PAT10762
PAT10763
PAT10764
PAT10765
PAT10766
PAT10767
PAT10768
PAT10769
PAT10770
PAT10771
PAT10772
PAT10773
PAT10774
PAT10775
PAT10776
PAT10777
PAT10778
PAT10779
PAT10780
PAT10781
PAT10782
PAT10783
PAT10784
PAT10785
PAT10786
PAT10787
PAT10788
PAT10789
PAT10790
PAT10791
PAT10792
WDERP=15.0975
WDEWH=8.45534+2.8201E-3*WHP-8.58E-7*WHP2
WDEBD=0.16331-7.319E-4*WDRS+4.462E-6*WDRS2-2.51E-9*WDRS3
HDERP=295.60
HDDEWH=179.01095+9.66338E-2*WHP-3.55714E-5*WHP2
RETURN
END

SUBROUTINE SHFLOW (RPDKF,W)
COMPUTF SUPERHEATER STEAM FLOW
IMPLICIT REAL (K,L,M,N)
COMMON /AREAI/ KNO,KN1,KN2,KN3,KN4,KNP5
KD1=KN1
IF (PD.GE.KNO) GO TO 920
KD1=-KN1
PD=-PD
CONTINUE
ZW=SQR(PD*R/KF)
W=ZW*KD1
RETURN
END

SUBROUTINE CWFLOW (PCN,RCNO,RDVO,PDES,NCP,ADV,KNCP,WCP,WCP,
* PCPO,PLHO,WHX,WDVO,ECP)
COMPUTE CONDENSATE (WATER) FLOW
IMPLICIT REAL (K,L,M,N)
COMMON /AREAI/ KNO,KN1,KN2,KN3,KN4,KNP5
KFLH=3.914161E-3
KDV=9.434E-3
K1CP=-1.64515E-2
K2CP=1.20115E-4
K3CP=1.25793E-4
K4CP=-8.8465E-3
K5CP=9.2870E-7
K6CP=4.34174E-7
KILHX=25.60544
K2LHX=7.32295E-2
K3LHX=-1.6317E-5
ADV2=ADV*ADV
NCP2=NCP*NCP
KNCP2=KNCP*KNCP
Z1=K1CP/RCNO
Z2=K2CP*NCP
Z3=K3CP*NCP2*RCNO
Z4=KFLH*KNCP2/RCNO
Z5=KDV*KNCP2/(ADV2*RDV)
Z6=Z4+Z5-Z1
Z7=Z3+PCN-PDES
WCP=KNP5*(SORT(Z2*Z2+KN4*Z6*Z7)+Z2)/Z6
WCW=KNCP*WCP
WCP2=WCP*WCP
WCW2=WCW*WCW
WLHX=KILHX+K2LHX*WCW+K3LHX*WCW2
WDVO=WCW-WLHX
PCP0=PCN+Z1+WCP2+Z2*WCP+Z3
PLHO=PCP0-Z4*WCW2
ECP=K4CP*WCP2/(RCNO*RCNO)+K5CP*WCP*NCP/RCNO+K6CP*NCP2
RETURN
END

SUBROUTINE FWFLOW (WRY,WSY,RDEW,PDES,PDRS,RECO,AFV,NFP,WFP,
* WFW,WFW2,PPPO,PFPO,PFVO,PVHO,PFVD,EFP)
COMPUTF FEEDWATER FLOW
IMPLICIT REAL (K,L,M,N)
COMMON /AREAl/ KNO,KN1,KN2,KN3,KN4,KNP5
K1FP=-57.3012E-3
K2FP=959.4371E-6
K3FP=203.8473E-6
K4FP=-1.735761E-3
K5FP=129.1779E-6
\[
\begin{align*}
K6FP &= 548.9264E-9 \\
K1BP &= -2.63447E-3 \\
K2BP &= 200.721E-6 \\
K3BP &= 99.9049E-6 \\
KFHH &= 4.7469E-3 \\
KFEC &= 3.878121E-3 \\
KFV &= 1.1721E-3 \\
KWFPX &= 19.6677 \\
KNBPR &= 0.333333 \\
KNBPR2 &= KNBPR \times KNBPR \\
NBP &= KNBPR \times NFP \\
NBP2 &= NBP \times NBP \\
NFP2 &= NFP \times NFP \\
AFV2 &= AFV \times AFV \\
Z1 &= KWF PX + WRY + WSY \\
Z2 &= KFV / (AFV2 \times RDEW) \\
Z3 &= KFHH / RDEW \\
Z4 &= KFEC / RECO \\
Z5 &= K1FP / RDEW \\
Z6 &= K1BP / RDEW \\
Z7 &= K2FP \times NFP \\
Z8 &= K2BP \times NBP \\
Z9 &= K3FP \times NFP2 \times RDEW \\
Z10 &= K3BP \times NBP2 \times RDEW \\
Z11 &= Z2 + Z3 + Z4 \\
Z12 &= Z5 + Z6 \\
Z13 &= Z7 + Z8 \\
Z14 &= Z9 + Z10 + PDES - PDRS \\
Z15 &= Z11 - Z12 \\
Z16 &= KN2 \times Z11 \times Z1 + Z13 \\
Z17 &= Z14 - Z11 \times Z1 \times Z1 \\
WFP &= KN5 \times \text{SORT} (Z16 \times Z16 + KN4 \times Z15 \times Z17) / Z16 \\
WFW &= WFP - Z1 \\
WFP2 &= WFP \times WFP \\
WFW2 &= WFW \times WFW \\
PPO &= PDES + Z6 \times WFP2 + Z8 \times WFP + Z10
\end{align*}
\]
SUBROUTINE RWFLOW (KNRPRDCRDRW, NRPPDRSWRWWRPWWWO,
* PDCO, PPRP, ERP)

COMPUTE RECIRCULATING WATER FLOW

IMPLICIT REAL (K,L,M,N)

COMMON /AREA1/ KNO, KN1, KN2, KN3, KN4, KN5

KN144=144.0
KLDC=137.0
KWRPS=4.3014
KIRP=-1.73366E-3
K2RP=1.64728E-4
K3RP=5.5798E-5
K4RP=-1.3391E-3
K5RP=3.45853E-4
K6RP=2.8937E-6
KFDC=381.048E-6
KFWRW=84.6537E-6
NRP2=NRP*NRP
KNRP2=KNRP*KNRP
Z1=KFDC/RDC
Z2=KLDC*RDC/KN144
Z3=K1RP/RDC
Z4=K2RP*NRP
Z5=K3RP*NRP2*RDC
Z6=KFWRW/RDRW
Z7=KLDC*RDRW/KN144
Z8=Z1-Z3+Z6*KNRP2
Z9=Z4-KN2*KWRPS
Z10=Z5+Z2-Z7-Z6*KWRPS*KWRPS

PFPO=PBPO+Z5*WFP2+Z7*WFP+Z9
PFVO=PFPG-Z2*WFW2
PHHO=PFVO-Z3*WFW2
PFVD=PFPO-PFVO
EFP=K4FP*WFP2/(RDEW*RDEW)+K5FP*WFP*NFP/RDEW+K6FP*NFP2
RETURN
END
WRP = KNP5 * (SQRT(Z9 * Z9 + KN4 * Z8 * Z10) + Z9) / Z8
WRW = KNRP * WRP
WWWO = WRW + KWRPS
WRP2 = WRP * WRP
PDCO = PDRS - Z1 * WRP2 + Z2
PRPO = PDCO + Z3 * WRP2 + Z4 * WRP + Z5
FRP = K4RP * WRP2 / (RDC * RDC) + K5RP * WRP * NRP / RDC + K6RP * NRP2
RETURN
END

SUBROUTINE ARFLOW (KNFD, KNID, NFD, NID, WGR, WFL, AVF, AVI, WAR, WWWG, * WFD, WGO, WID, PAHA0, PDFO, PFN, PECGO, PAPGO, PIDO, EFD, EID)

IMPLICIT REAL (K, L, M, N)

COMMON /AREA1/ KNO, KN1, KN2, KN3, KN4, KNP5

KPAT = 14.7
K1FD = -7.41568E-7
K2FD = 8.67456E-6
K3FD = 1.67206E-4
K4FD = -2.18247E-6
K5FD = 5.13044E-5
K6FD = -6.96841E-5
K1ID = -1.38148E-6
K2ID = 1.12227E-5
K3ID = 1.09727E-4
K4ID = -1.12212E-6
K5ID = 1.74023E-5
K6ID = 3.43528E-5
KFAH = 1.82764E-7
KFAPA = 3.968E-7
KFG = 263.7944E-9
KFAPG = 1.176409E-7
KFST = 2.109E-7
PSTD = 7.216667E-2
NFD2 = NFD * NFD
NID2 = NID * NID
KNFD2 = KNFD * KNFD
KNID2 = KNID * KNID
WFL2 = WFL * WFL
ADF2 = ADF * ADF
ADI2 = ADI * ADI
Z1 = WFL + WGR
Z2 = K1FD / AVF
Z4 = KNFD2 * KFAPA
Z5 = KFAH + Z4 - Z2
Z6 = KFAPG * KFST
Z7 = K1ID / AVI
Z9 = -Z7
Z10 = K2FD * NF0
Z11 = K2ID * NID
Z12 = K3FD * NF02 * AVF
Z13 = K3ID * NID2 * AVI
Z14 = Z12 + Z13 + PSTD
Z15 = Z9 / KNID2
Z16 = Z5 + KNFD2 * (KFG + Z6 + Z15)
Z17 = KN2 * KNF0 * (KFG * Z1 + Z6 * WFL + Z15 * WFL) - Z10 - Z11 * KNFD / KNID
Z18 = KFG * Z1 + WFL2 * (Z6 + Z15) - Z11 * WFL / KNID - Z14
WFD = KNP5 * (SQRT(Z17 * Z17 - KN4 * Z16 * Z18) - Z17) / Z16
WAR = KNFD * WFD
WWWG = WAR + Z1
WGO = WAR + WFL
WID = WGO / KNID
WFD2 = WFD * WFD
WWWG2 = WWWG * WWWG
WGO2 = WGO * WGO
WID2 = WID * WID
PAHA0 = KPAT - KFAH * WFD2
PFDO = PAHA0 + Z2 * WFD2 + Z10 * WFD + Z12
PFN = PFDO - Z4 * WFD2
PECGO = PFN - KFG * WWWG2
PAPGO = PECGO - KFAPG * WGO2
PIDO = PAPGO * Z7 * WID2 + Z11 * WID + Z13
FFD = K4FD * WFD2 + K5FD * WFD * NFD + K6FD * NFD2
EID = K4ID * WID2 + K5ID * WID * NID + K6ID * NID2
RETURN
END

C
C
C
SUBROUTINE FNXFER (TWWM, WWWG, WAR, SAR, TAPAO, WFL, SFL, TFL, KHFL, * EFL, WGR, SGR, TGR, UWWGM, YWGR, TPSF, UPSR, TFN1, TWWGE, TWWGO, QWWGM, * QPSR, SWWG0)

COMPUTE FURNACF HEAT TRANSFER

IMPLICIT REAL (K, L, M, N)
COMMON /AREA1/ KNO, KN1, KN2, KN3, KN4, KNP5

KNP33 = 0.333333
KTO = 537.0
K1SFN = 0.31
K2SFN = 0.145

TWWM2 = TWWM * TWWM
TWWM4 = TWWM2 * TWWM2
TPSE2 = TPSE * TPSE
TPSE4 = TPSE2 * TPSE2
SFN = K1SFN + K2SFN * YWGR
TFN1 = KTO + (WFL * KHFL * EFL + WFL * SFL * (TFL - KTO) + WAR * SAR * (TAPAO - KTO) * + WGR * SGR * (TGR - KTO)) / (WWWG * SFN)

Z1 = UWWGM + UPSR
Z2 = KN2 * WWWG * SFN
Z3 = UWWGM * TWWM4 + UPSR * TPSE4 + Z2 * TFN1
Z4 = Z2 / Z1
Z5 = Z3 / Z1
Z6 = KNP5 * Z4 * Z4
Z7 = KN4 * Z5 / KN3
Z8 = SQRT(Z7 * Z7 * Z7 * Z6) / Z6
Z9 = (Z6 + Z8) ** KNP33
Z10 = (Z8 - Z6) ** KNP33
Z11 = Z9 - Z10
Z12 = SQRT(Z11 + Z11 + KNP4 * Z5)
TWWGE = KNP5 * (SQRT(KN2 * Z12 - Z11) - SQRT(Z11))
SUBROUTINE HXFER (WG, WS, KUGM, KUMS, TSE, TG1, QR, YWGR, TGD, Q, TME, SG)

COMPUTE GAS-PATH HEAT TRANSFER

IMPLICIT REAL (K, L, M, N)

COMMON /AREAl/ KNO, KNI, KN2, KN3, KN4, KNP5
COMMON /AREA3/ KNP6, KNP8

KOSG = 0.2484
K1SG = 0.1428
KSGT = 10.2272E-6
KSGW = 35.0E-6
WS = ABS(WS)

UGM = KUGM * WG ** KNP6
UMS = KUMS * WS ** KNP8

UGS = UGM * UMS / (UGM + UMS)

Z1 = KOSG + K1SG * YWGR
Z2 = KSGT + KSGW * YWGR
Z3 = KNP5 * UGS
Z4 = WG * Z1
Z5 = WG * Z2
Z6 = Z3 + Z4
Z7 = Z3 * TG1 - UGS * TSE - Z5 * TG1 * TG1 - Z4 * TG1

TGD = KNP5 * (SQRT(Z6 * Z6 - KN4 * Z5 * Z7) - Z6) / Z5
SG = Z1 + Z2 * (TG1 + TGD)
QC = WG * SG * (TG1 - TGD)
Q = QC + QR
TME = TSE + QC / UMS
SG0 = Z1 + Z2 * KN2 * TG1
RETURN
SUBROUTINE AVERAG (X11,X12,X21,X22,X31,X32,XLM,X2MX3M)
COMPUTE AVERAGE VALUES
IMPLICIT REAL (K,L,M,N)
COMMON /AREAL/ KN0,KN1,KN2,KN3,KN4,KNP5
XLM=KNP5*(X11+X12)
X2M=KNP5*(X21+X22)
X3M=KNP5*(X31+X32)
RETURN
END

SUBROUTINE DRUM (KVDR, VDRW, RDRS, HDRW, HDRS, K2, K3, K5, K6, K7, 
* K9, K10, Z206, Z209, F1DR, F2DR)
COMPUTE DRUM VARIABLES
IMPLICIT REAL (K,L,M,N)
COMMON /AREAL/ KN0, KN1, KN2, KN3, KN4, KNP5
VDRS=KVDR-VDRW
Z200=KN2*RDRS
Z201=K2+K3*Z200
Z202=K5+K6*Z200+KN3*K7*RDRS*RDRS
Z203=K9+K10*Z200
Z204=RDRW-RDRS
Z205=VDRS+VDRW*Z201
Z207=RDRW*HDRW-RDRS*HDRS
Z208=VDRS*HDRS+VDRW*HDRW*Z201+VDRW*RDRW*Z202+VDRS*RDRS*Z203
Z210=Z204*Z208-Z205*Z207
F1DR=(Z206*Z208-Z205*Z209)/Z210
F2DR=(Z204*Z209-Z206*Z207)/Z210
RETURN
END

SUBROUTINE TORQUE (NELEC, VELEC, KNM, N, KM, SMAX, TQ1, MW1)
COMPUTE INDUCTION MOTOR TORQUE
IMPLICIT REAL (K,L,M,N)
NMAX=NELEC/KNM
S = (NMAX - N) / NMAX
VELEC2 = VELEC * VELEC
TQL = KM * VELEC2 / (S/SMAX + SMAX/S)
MW1 = TQI * N
RETURN
END

SUBROUTINE FPTURB (WFT, HFT1, NFP, TQFP1, MWFP1)

COMPUTE FEEDPUMP TURBINE TORQUE

IMPLICIT REAL (K, L, M, N)
COMMON / AREA1/ KNO, KN1, KN2, KN3, KN4, KN5
KJ = 778.17
HFTO = 1059.0
EFT = 1.0
MWFP1 = WFT * (HFT1 - HFTO) * EFT * KJ
TQFP1 = MWFP1 / NFP
RETURN
END

SUBROUTINE XDUCER (ZMIN, ZMAX, CMIN, CMAX, Z, C)

INPUT-OUTPUT CONVERSION. PHYSICAL VARIABLE TO CONTROL SIGNAL
AND VISE VERSA

IMPLICIT REAL (K, L, M, N)
C = CMIN + (CMAX - CMIN) * (Z - ZMIN) / (ZMAX - ZMIN)
IF (C .LT. CMIN) C = CMIN
IF (C .GT. CMAX) C = CMAX
RETURN
END

SUBROUTINE LIMCHK(ZC)

CHECK THAT CONTROL VARIABLE (ZC) IS WITHIN LIMITS (1. - 5.)

IF (ZC .LT. 1.0) ZC = 1.0
IF (ZC .LT. 5.0) ZC = 5.0
RETURN
END
SUBROUTINE CHECK (ZC, ZCMAX, ZCMIN)

CHECK THAT VARIABLE (ZC) IS WITHIN LIMITS (ZCMIN-ZCMAX)

IF (ZC.GT.ZCMAX) ZC=ZCMAX
IF (ZC.LT.ZCMIN) ZC=ZCMIN
RETURN
END

C
C SUBROUTINE CONSTI
C SET SYSTEM PARAMETER VALUES
IMPLICIT REAL (K,L,M,N)
COMMON /AREA4/ KUWWGM, KUWMMW, KUPSR, KUPSGM, KUPSMS, KUSSGM, KUSSMS,
* KURHGM, KURHMS, KUECGM, KUECMW, KMWWM, KMPSM, KMSSM, KMRHM, KMCHM, KMLHM,
* KMHHM, KSWWM, KSPSM, KSSTEM, KSRHM, KSECM, KSLHM, KSHHM, KVWM, KVPS, KVSS,
* KVRH, KVEC, KVLH, KVHH, KVDR, KVDE, KVSCE, KVCRE, KLVHM, KHHM, K2, K3, K5, K6,
* K7, K9, K10, K22, K23, K25, K26, K27, K29, K30, NELEC, VELEC, KNRPM, SRPMAK,
* KNCPM, KCPH, SPMAX, KNFDM, KFDM, SFDMAX, KNDM, SDM, MAX, KJTR,
* KJFPF, KJRPF, KJCP, KJFD, KJID, KID, KIP, KTRP, KTV, KCV, KKV, KOPCN,
* K1PCN, K2PCN, KQYLPQ, KJ, KERT, KWFW, KQGC, KPAT, KTAT, KTAHAD, KTAPAO,
* K1TGR, K2TGR, SAR, SFL, TFL, KHF, FL, EFL, SGR, YWGR, KR, KAFR, KFDC, KFW,
* KFPS, KFSS, KFRH, KFHR, KFFH, KFFC, KNRP, KNCP, KND, KNG, K2NG, K3NG, K4NG, K5NG, KXWE, KRV,

C * * * HEAT TRANSFER COEFFICIENTS
KUWWGM=3.187029E-9
KUWMMW=173.5205
KUPSR=138.576E-12
KUPSGM=1.80123
KUPSMS=2.34465
KUSSGM=3.75513
KUSSMS=6.753101
KURHGM=8.0954
KURHMS=4.47939
KUECGM=9.32379
KUECMW=8.591464

C * * * METAL MASSES
KMWWM=1063000.0
KMPSM=350000.0
KMSSM=800000.0
KMRHM=944000.0
KMECM=721000.0
KMLHM=70376.0
KMHHM=108400.0

C * * SPECIFIC HEAT OF METAL
KSWWM=0.11
KSPSM=0.11
KSSSM=0.11
KSRHM=0.11
KSECW=0.11
KSLHM=0.11
KSHHM=0.11

C * * FILL VOLUMES
C * KVEC,KVRH,KVCRE,KVSS,KVPS RESET TO NEWLY COMPUTED VALUES
KVWW=2318.61
KVPS=2000.0
KVSS=3000.0
KVRH=6000.0
KVEC=2100.0
KVLH=93.02
KVHH=114.67
KVDL=1958.72
KVDE=8029.9
KVSCE=700.
KVCRE=1220.0

C * * FEEDWATER HEATERS METAL TEMPERATURE - CONSTANTS
KLHM=1.05
KHHM=1.05

C * * DRUM AND DEAERATOR CONSTANTS
K2=-2.13733
K3=0.03348
K5=31.0437
K6=-0.62086
K7=0.0
K9 = -21.3442
K10 = 0.20998
K22 = -19.61207
K23 = 0.0
K25 = 1905.6372
K26 = -8414.69018
K27 = 15688.03769
K29 = 224.556
K30 = 0.0

C *** DRIVE MOTOR CONSTANTS
NELEC = 376.991
VELEC = 4160.0
KNRPM = 2.0
KRPM = 476.145E-6
SRPMAX = 0.05
KNCPM = 2.0
KCPM = 615.533E-6
SCPMAX = 0.05
KNFDM = 6.0
KFDM = 4.1E-3
SFDMAX = 0.05
KNIDM = 4.0
KIDM = 6.16E-3
SIDMAX = 0.05

C *** ROTOR INERTIA
KJTR = 625000.0
KFPE = 2161.7
KJRP = 576.1
KJCP = 468.0
KJFD = 181900.0
KJID = 188000.0

C *** MISCELLANEOUS
KWTV = 506.7
KHP = 15.6
KIP = 51.78706
KLP = 126.41
KTV = 32.0
KCV = 12.74214
KFV = 482.3549E-6
KOPCN = 0.26449
K1PCN = 1.38076E-4
K2PCN = 1.30065E-5
KQYLPO = 0.92632
KJ = 778.17
KETR = 0.955
KWFW = 493.03
KQGC = 1647.52
KPAT = 14.7
KTAT = 510.0
KTAHAD = 88.0
KTAPAD = 350.0
K1TGR = 929.69
K2TGR = 2.15172
SAR = 0.252
SFL = 0.490
TFL = 650.0
KHFL = 18200.0
EFL = 1.0
SGR = 0.27
YWGR = 0.0
KR = 0.357639
KAFR = 13.6507

C * * * FRICITION COEFFICIENTS
KFDC = 719.028E-6
KFWW = 159.751E-6
KFPS = 1.88E-3
KFSS = 3.162E-4
KFRH = 45.90723E-6
KFLH = 3.914168E-3
KFHN = 5.711565E-3
KFEC = 3.878121E-3

C * * * NUMBER OF OPERATING PUMPS AND FANS

PAT11261
PAT11262
PAT11263
PAT11264
PAT11265
PAT11266
PAT11267
PAT11268
PAT11269
PAT11270
PAT11271
PAT11272
PAT11273
PAT11274
PAT11275
PAT11276
PAT11277
PAT11278
PAT11279
PAT11280
PAT11281
PAT11282
PAT11283
PAT11284
PAT11285
PAT11286
PAT11287
PAT11288
PAT11289
PAT11290
PAT11291
PAT11292
PAT11293
PAT11294
PAT11295
PAT11296
BURNER TILT AND NUMBER OF OPERATING GUNS

\begin{align*}
K_{\text{NR}} &= 6.0 \\
K_{\text{PC}} &= 2.0 \\
K_{\text{FD}} &= 2.0 \\
K_{\text{ID}} &= 2.0 \\
\end{align*}

\begin{align*}
C & \quad * \quad * \quad * \\
& \quad \text{BURNER TILT AND NUMBER OF OPERATING GUNS} \\
K_{1xG} &= 1.0 \\
K_{2xG} &= -0.286483 \\
NG_1 &= 4.0 \\
NG_2 &= 4.0 \\
NG_3 &= 4.0 \\
NG_4 &= 4.0 \\
NG_5 &= 4.0 \\
K_{\text{NG}} &= 79.868 \\
K_{2NG} &= 72.0301 \\
K_{3NG} &= 64.534 \\
K_{4NG} &= 56.867 \\
K_{5NG} &= 49.200 \\
K_{XWW} &= 64.534 \\
\text{return} \\
\text{end} \\
\end{align*}

SUBROUTINE CONST?

SET CONTROL SYSTEM INPUT-OUTPUT PARAMETER VALUES

IMPLICIT REAL (K, L, M, N)

COMMON /AREA6/ K1MWD, K2MWD, K1PSS, K2PSS, K3PSS, K4PSS, KPSSOL, KPSSOU, KCL, KCU, KAVL, KAVU, KWARL, KWARU, KFWLL, KFWLU, KPFNL, KPFNU, KPFN,

KPO, KTRHL, KTRHOU, K1TRH, K2TRH, K3TRH, K4TRH, K5TRH, KTRHD, KPFDVL, KPFDU,

KPFDV, KXDRW, KXDRWU, KXDRW, KPDRSL, KPDRSU, KPISTL, KPISTU, KFWL,

KFWU, KXDEW, KXDEWU, KXDEW, KPRHL, KPRHOU, KWCW, KWCWU, KWRY, KWRYU,

KCRY, KCRYU, KTSS, K2TSS, K3TSS, K4TSS, KTSSOL, KTSSOU, KTSSO, KWSY,

KWSYU, KCSY, KCSYU, KMWTDL, KMWTRU, KMODE, KSVREG, KCVREG, KIVREG,

KXGGL, KXGGLU, K1XDRW, K2XDRW, K3XDRW, K1XDEW, K2XDEW, K3XDEW, K1PIST,

KNTRL, KNTRU, KNTRA, KSWRL, KSWRU, KWTFL, KWFITU, KCXGG

K1MWD = -110.76E6 \\
K2MWD = 110.76E6 \\
K1PSS = 81.37E0

PAT11297
PAT11298
PAT11299
PAT11300
PAT11301
PAT11302
PAT11303
PAT11304
PAT11305
PAT11306
PAT11307
PAT11308
PAT11309
PAT11310
PAT11311
PAT11312
PAT11313
PAT11314
PAT11315
PAT11316
PAT11317
PAT11318
PAT11319
PAT11320
PAT11321
PAT11322
PAT11323
PAT11324
PAT11325
PAT11326
PAT11327
PAT11328
PAT11329
PAT11330
PAT11331
PAT11332
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<tr>
<th>Parameter</th>
<th>Value</th>
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<tbody>
<tr>
<td>K2PSS</td>
<td>533.333</td>
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<tr>
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<td>815.0</td>
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<td>K4PSS</td>
<td>2415.0</td>
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<tr>
<td>KPSSGU</td>
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<td>KCL</td>
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<tr>
<td>KPFNU</td>
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<tr>
<td>KNO87</td>
<td>0.08727</td>
</tr>
<tr>
<td>KTRHOL</td>
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<tr>
<td>KTRHOU</td>
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</tr>
<tr>
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<td>KWFNL</td>
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<tr>
<td>KWFNU</td>
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<tr>
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<td>1250.0</td>
</tr>
<tr>
<td>KWFNU</td>
<td>1250.0</td>
</tr>
</tbody>
</table>
**DRUM AND DEAERATOR WATER LEVEL**

K1XDRW= -29.118
K2XURW=0.02262
K3XDRW= 1.9297E-6
K1XDEW= -29.138
K2XDEW=-9.5377E-3
SUBROUTINE CONST3

' SET CONTROL SYSTEM GAINS AND TIME CONSTANTS
IMPLICIT REAL (K,L,M,N)
COMMON /AREA7/, ADF, ADI, KTC1MD, KC1MD, KC2MD, KTC1AR, KTC2AR, KC1AR,
* KC4AR, KC2ARL, KC1FL, KTC1FL, KTC2FL, KC3FL, KTC1FN, KTC2FN, KC1FN, KC2FN,
* KC3FN, KC1GR, KTC1GR, KC2GR, KC3GR, KC1FT, KC2FT, KTC1FT, KTC2FT, KC2FV,
* KTC1FV, KTC2FV, KTC3FV, KC1FV, KC3FV, KC4FV, KC6FV, KTC1DV, KTC2DV, KC1DV,
* KTC3DV, KTC4DV, KC4DV, KC5DV, KC6DV, KTC1RH, KTC2RH, KTC3RH, KC1RH, KC2DV,
* KC2RH, KC3RH, KC4RH, KC5RH, KC1RY, KC1BT, KTC1SY, KTC2SY, KC1SY, KC2SY,
* KC3SY, KC4SY, KC5SY, KC1TR, KC2TR, KTC1TR, KTC2TR, KTC3TR,
* KTC4TR, KTC5TR, KCILD, KC2LD, KTC1LD, KMRL, KVRL, KLRL, COUNT

C * * * VALVE AND DAMPER AREAS
ADF=1.0
ADI=1.0

C * * * BOILER MASTER DEMAND
KC1MD=20.0
KC2MD=.5
KTC1MD=60.0

C * * * AIR FLOW CONTROL
KC1AR=3.0
KC2ARL=1.81656
KC4AR=1.
C * * * DEFEATER LEVEL CONTROL
  KCCM=0.1
  KCCM=1.0
  KCCM=10.0
  KCCM=20.0

C * * * FEEDWATER CONTROL
  KCCM=5.0
  KCCM=6.0
  KCCM=7.0
  KCCM=8.0

C * * * BOILER FEDPUMP CONTROL
  KCCM=1.0
  KCCM=2.0
  KCCM=3.0

C * * * GAS RECIRCULATION CONTROL
  KCCM=5.0
  KCCM=6.0
  KCCM=7.0

C * * * FURNACE PRESSURE CONTROL
  KCCM=5.0
  KCCM=6.0
  KCCM=7.0

C * * * FUEL FLOW CONTROL
  KCCM=5.0
  KCCM=6.0
KC5DV = 0.1
KC6DV = 1.0
KTC1DV = 40.0
KTC2DV = 2.0
KTC3DV = 120.0
KTC4DV = 5.0

C * * * REHEAT TEMPERATURE CONTROL
KC1RH = 5.0
KC2RH = -0.01
KC3RH = 0.01
KC4RH = 1.0
KC5RH = 1.0
KTC1RH = 60.0
KTC2RH = 5.0
KTC3RH = 5.0
KC1RY = -1.0
KC1BY = 1.0

C * * * SUPERHEAT TEMPERATURE CONTROL
KC1SY = 10.0
KC2SY = -0.01
KC3SY = -0.01
KC4SY = -1.0
KC5SY = 1.0
KTC1SY = 60.0
KTC2SY = 5.0

C * * * TURBINE CONTROL
KC1LD = 0.3
KC2LD = 0.0
KTC1LD = 1.0
KC1TR = 1.0
KC2TR = 1.0
KC1TR = 10.0
KTC2TR = 1.0
KTC3TR = 1.0
KTC4TR = 1.0
KTC5TR = 1.0
COUNT=200.0
KMRL=0.5
KVRL=0.5
KLRL=8.333E-3
RETURN
END
C
// DUP
CP USORD-LIR
FILE DAFILE 1
XB >
*PN DYSYS50
*EL USORD-LIR
*BC 300
*L 650
*LAST
STIME=0.0, TTIME=700.0, TSTEP=0.4, N=47,
Y=542.15, 478.11, 634.72, 1170.5, 9.4534, 187.19, 1159.8, 5.0129,
1247.9, 3.1331, 1460.1, 2.5600, 0.65599, 1519.0, 0.25448, 376.99, 186.56,
198.99, 6817.4, 0.13307, 61.880, 92.452, 4.5628, 4.5600, 4.5622, 4.9586,
2.4688, 4.0743, 4.9667E-3, 4.6042, 5.8393E-3, 4.5196, 3.1173, 4.4259,
4.5486, 4.5617, 4.7573, 2.4688, 4.0737, 4.6034, 4.5238, 3.1167, 4.4245,
0.0, 5.0, 5.00, 0.52362,
IPRNT=39,40,41,42,43,44,45,46,47, PRNTC=0.0,1000.40.0, &
LOOK=32,
KRVLAB=' ',
POS=175., 475., 350., 225.,
IPL0T=16, &
POS=625., 475., 350., 225.,
IPL0T=50, &
POS=175., 175., 350., 225.,
IPL0T=49, &
POS=625., 175., 350., 225.,
IPL0T=48, &
POS=175., 475., 350., 225.,
IPL0T=23, &
POS=625., 475., 350., 225.,
IPL0T=46, &
POS=175., 175., 350., 225.,
IPL0T=36, &
POS=625., 175., 350., 225.,
IPL0T=35, &
POS=175., 175., 350., 225.
TPLT=2, &
POS=625., 175., 350., 225.
TPLT=18, &

// END
```plaintext
*** THIS PROGRAM SIMULATES ***

" A DRUM BOILER-TURBINE POWER PLANT UNDER EMERGENCY STATE CONTROL "

===============================================

* STANDARD MODEL *

SUBROUTINE EOSIM
IMPLICIT REAL (K,L,M,N)
INTEGER NEWDT,N
COMMON T,TSTEP,Y(50),F(50),STIME,FTIME,NEWDT,IPWRT,N,IPR,ICD,
1 ICN,TNEXT,TBACK
COMMON /AREA2/ KDR1,KDR2,KDR3,KDR4,KDR5,KDR6,KDR7,KDR8,KDR9
EQUIVALENCE (Y(1),HECO), (Y(2),RDRS), (Y(3),TWM), (Y(4),RSSO),
* (Y(5),RSHO), (Y(6),RRHO), (Y(7),RHHO), (Y(8),NR), (Y(9),HLHO),
* (Y(10),HDEW), (Y(11),HHDW), (Y(12),C3MD), (Y(13),C5AB),
* (Y(14),C5FL), (Y(15),C3GR), (Y(16),C5RH), (Y(17),C5SY),
* (Y(18),CAPD), (Y(19),CFLD), (Y(20),CGBD), (Y(21),CWRD),
* (Y(22),CXGD), (Y(23),CSDA), (Y(24),C6TR), (Y(25),C9TR),
* (Y(26),C12TR), (Y(27),DETA), (Y(28),WF), (Y(29),PSSO),
* (Y(30),MWO), (Y(31),TSSO), (Y(32),TRHO), (Y(33),RSSH)

IP (NEWDT) 51,52,53
51 CONTINUE

* * * CONSTANS

SET CONSTANT AND SYSTEM PARAMETER VALUES

KNP5=0.5
KN0=0.0
KN1=1.0
KN2=2.0
KN3=3.0
KN4=4.0
KN5=5.0
```
KN6=6.0
KM1=-1
KM5=-5.
KN4P7=4.7
KCTGN=1.0
KMW=4.4288E8
KNTR=377.0
KTCLH=60.0
KTCD=280.0
KTCHH=60.0
HCNW=56.7
KHPD=1.4
KHPD=11.0
KETR=0.88
KMWX=1.353E-6
KPSH=2.7E-3
KCV=0.459
KWHP=0.895
KWPH=1.784
KMEE=1.784E5
KMWME=1.524E6
KSWWM=0.11
KVSH=5000.0
KMSHE=1.693E5
KFRH=4.59E-5
KHLO=1025.0
KVRH=6000.0
KMRHE=1.078E5
KEFL=1.0
KHF=18200.0
KSGPE=0.28
KSA=0.252
KTH=1060.0
KSGR=0.27
KTECG=1077.7
KSFL=0.490
KTFL=860.0
KSFNG=0.31
K1XGG=1.0
K2XGG=-0.2865
KJTRE=6.25F5
KTCFW=10.0
KUWWGM=186.19
KUWWMW=25876.4
KJ=778.17
KT0=537.0
K1TGR=929.7
K2TGR=2.1512

DRUM PARAMETERS
KVDP=1958.7
KVDRW=1170.2
KVDRS=788.5
KDR1=49.271
KDR2=-2.137
KDR3=0.03348
KDR4=1241.7
KDR5=-21.344
KDR6=0.210
KDR7=526.60
KDR8=31.044
KDR9=-0.6209

BOILER MASTER CONTROL PARAMETERS
KCL=1.0
KCU=5.0
KPSSOL=30.0
KPSSCU=3015.0
KC1MD=20.0
KC2MD=0.5
KTC1MD=60.0
AIR FLOW CONTROL PARAMETERS

KWARL=0.0
KWARU=1381.4
KC1AR=3.0
KC2ARL=1.817
KC4AR=1.0
KTC1AR=60.0
KTC2AR=5.0

FUEL FLOW CONTROL PARAMETERS

KWFLL=0.0
KWFLU=90.0
KC1FL=3.0
KC3FL=1.0
KTC1FL=60.0
KTC2FL=5.0

GAS RECIRCULATION CONTROL PARAMETERS

KNP087=0.08727
KTRHOL=1159.7
KTRHGU=1559.7
KWGBL=0.0
KWGRD=500.0
KC1GR=0.004
KTC1GR=20.0
KC2GR=0.8
KC3GR=1.0

PEHEAT TEMPERATURE CONTROL PARAMETERS

KCRL=-2.0
KCryo=-1.0
KWRL=0.0
KWRYU=50.0
KXGGL=-0.5236
KXGGU=0.5236
K1TRH=1192.5
C
C
SUPERHEAT TEMPERATURE CONTROL PARAMETERS

K2TRH=0.4126
K3TRH=1259.7
K4TRH=1459.7
KC1RH=5.0
KC4RH=1.0
KC5RH=1.0
KTC1RH=60.0
KTC2RH=5.0
KTC3RH=5.0
KC1RY=-1.0

C
C
TURBINE CONTROL PARAMETERS

KNTRL=0.0
KNTRU=450.0
KAVL=0.0
KAVU=1.0
KMWTRL=0.0
KMWTRU=531.36E6
KCVREG=0.05