1970

A Digital Simulation of a Once-Through Supercritical Steam Generator.

Alfred Julius Flechsig

Louisiana State University and Agricultural & Mechanical College

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A DIGITAL SIMULATION OF A ONCE-THROUGH SUPERCritical STEAM GENERATOR

A Dissertation

Submitted to the Graduate Faculty of the Louisiana State University and Agricultural and Mechanical College in partial fulfillment of the requirements for the degree of Doctor of Philosophy in The Department of Electrical Engineering

by Alfred Julius Flechsig, Jr. B. S., Washington State University, 1957 M. S., Washington State University, 1959 January, 1970
PLEASE NOTE:

Not original copy. Several pages have blurred, light and indistinct type. Filmed as received.

UNIVERSITY MICROFILMS.
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<tr>
<td>A</td>
<td>Area of Cross Section</td>
</tr>
<tr>
<td>$C_{pw}$</td>
<td>Specific Heat of Tubes</td>
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<tr>
<td>g</td>
<td>Gravitational Acceleration $\frac{ft}{sec^2}$</td>
</tr>
<tr>
<td>$h_i$</td>
<td>Inlet Steam Enthalpy, btu/lb-°F</td>
</tr>
<tr>
<td>$h_o$</td>
<td>Outlet Steam Enthalpy, btu/lb</td>
</tr>
<tr>
<td>$K_p$</td>
<td>Friction Term, $sec^2/lb - in^2 = psi (lb/sec)^2$</td>
</tr>
<tr>
<td>L</td>
<td>Length of Section, ft</td>
</tr>
<tr>
<td>m</td>
<td>Weight of Steam in Tubes, lb</td>
</tr>
<tr>
<td>$m_w$</td>
<td>Weight of tubes, lb</td>
</tr>
<tr>
<td>$P_i$</td>
<td>Inlet Steam Pressure, psi</td>
</tr>
<tr>
<td>$P_o$</td>
<td>Outlet Steam Pressure, psi</td>
</tr>
<tr>
<td>Q</td>
<td>Heat Flux from Tube Walls, btu/sec</td>
</tr>
<tr>
<td>$Q_w$</td>
<td>Heat Flux into the walls, btu/sec</td>
</tr>
<tr>
<td>t</td>
<td>Time</td>
</tr>
<tr>
<td>$T_b$</td>
<td>Average Bulk Temperature, °F</td>
</tr>
<tr>
<td>$T_f$</td>
<td>Average Film Temperature, °F</td>
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<tr>
<td>$T_g$</td>
<td>Temperature of gas, °F</td>
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<td>$T_i$</td>
<td>Inlet Steam Temperature, °F</td>
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<td>$T_o$</td>
<td>Outlet Steam Temperature, °F</td>
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<td>$T_w$</td>
<td>Tube Wall Temperature, °F</td>
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<tr>
<td>$u$</td>
<td>Internal Energy, btu/lb</td>
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<td>$U_f$</td>
<td>Convection Film Conductance</td>
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<tr>
<td>$U_g$</td>
<td>Convection Film Conductance, btu/ft$^2$-sec-°F</td>
</tr>
<tr>
<td>$v$</td>
<td>Specific Volume, ft$^3$/lb</td>
</tr>
<tr>
<td>$V$</td>
<td>Velocity, ft/sec</td>
</tr>
<tr>
<td>$W$</td>
<td>Steam Flow Rate, lb/sec</td>
</tr>
<tr>
<td>$Z$</td>
<td>Potential Energy, ft-lb/lb</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>Emissivity</td>
</tr>
<tr>
<td>$\mu$</td>
<td>Viscosity, lb/hr-ft</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density, lb/ft$^3$</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>Stephen Boltzmann Constant = $1.73 \times 10^{-9}$ btu/ft$^2$-hr T$^4$</td>
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ABSTRACT

The purpose of this dissertation is to give a comprehensive description of a digital simulation of a once-through supercritical steam generator with its associated control system. The simulated plant is rated at 560 MW, 3,980,000 lb/hr steam flow and a superheater outlet pressure of 3600 psig and temperature of 1005°F. It has one stage of reheat to 1005°F. The paper includes an extensive bibliography, which references many papers relating to the subject of simulation of steam generators.

A lumped parameter system was utilized in the development of the simulation equations. Twenty-three lumps are included. Twelve of these lumps consider the fluid flow dynamics and the heat transfer characteristics through the walls in a dynamic fashion while the remaining 11 lumps in the model are considered in a heat balanced fashion.

The problem was solved in a nonlinear fashion and the method of solution of the process differential equations used is the Runge-Kutta third order method. For the control system equations the method of integration used is the trapezoidal rule.

The fundamental equations that are used in the process are considered and developed, followed by a complete description of the process itself. The various component parts of the supercritical once-through steam generator are described. They are the economizer, furnace, superheater, steam leads, re heater, air heater, boiler feed pump, feedwater heater, turbines, turbine valves, and spray valves. The heat cycle of the plant is given with actual points plotted on the T-S and H-S diagrams.

Relatively large excursions from operating points are allowed since the simulation computes the properties of the steam at each iterative cycle. The
temperature is computed as a function of pressure and enthalpy for all points in the cycle and a discussion of its accuracy in the regions of the heat cycle is given.

In the computer simulation there is provision made to include the change in density in the superheater region of the steam generator. The change in density is computed by the use of the subroutine which is valid in the region where the specific volume is above .1603 ft$^3$/lb. A discussion is included in the paper of its accuracy in the region under consideration. The properties of the combustion gas and the method by which it is included in the simulation is given in detail.

A complete description is given of the control system of the simulated plant and its interaction with the process equations in the digital simulator. The control system is of the integrated mode type. The plant that is being simulated has an analog type control system. The simulation is a complete digital simulation and, hence, the control system as well as the process, is represented with a digital simulation. The equations are solved using numerical techniques. Iterations are made to combine the control and process system together. The digital simulator may be run either with an open loop control system (control system not connected to the process) or with the control system operating. Hence, either open loop operation (manual control on the units) is available or closed loop control (automatic operation) of the simulated plant is available. A complete discussion of the operation of the digital simulator is given. There are 17 simulated tests illustrated. Fourteen are open loop tests and 3 are closed loop tests using the control system. Open loop tests were made on the unit that was simulated. A description of the tests made on the actual plant is included.

The Lagrangian interpolation method for the simulation of the boiler feed pump and the complete digital computer program are included for reference.
CHAPTER 1

INTRODUCTION

Since 1920, the steam turbine has been the primary prime mover in generating stations. At that time most units used steam with conditions 200 psig, 500°F and machine ratings from 5 to 30 megawatts. By 1940, the conditions of steam were increased to 1200 psig and 950°F with a maximum rating of 160 MW for a single shaft unit rated at 1800 rpm.

In 1957, the first supercritical unit with two stages of reheat began commercial service. The steam conditions were 4500 psig and 1150°F/1050°F/1000°F with a generator rating of 125 MW.

The plant that is being simulated in this study, Michoud, unit number three, owned and operated by the New Orleans Public Service, Inc., is a once-through, supercritical steam generator. The plant is seen in Figure 1.1. It has the following characteristics:

- Rated Output 560 megawatts
- Rated Steam Flow 3,980,000 lbs./hr.
- Superheater Outlet Pressure 3600 PSIG
- Temperature 1005°F
- Reheat Temperature 1005°F

It is a gas fired unit and has a 3600 RPM tandem-compound turbine driving a 685 MVA generator with an output voltage of 24,000 volts.

The boiler process is based on the Benson once-through design and does not have a steam drum. The feedwater enters the economizer and is heated continuously up to the final outlet of the superheater.

There have been several excellent papers published dealing with various aspects of the simulation of steam-generating systems. In Table 1.1 is a
Fig. 1.1 Michoud steam electric plant, unit number 3
representative list of past cases of such simulation. It is seen from the table that the simulation efforts may be divided roughly into three categories, electronic analog, pneumatic analog, and digital.

The pneumatic analog simulator has been the most successful to date in the training of operators to run a thermal-electric generating plant. In essence this method of simulation has taken actual pneumatic valves and logic elements that would be found on the real system and coupled these to an actual, though scaled down in size, synchronous generator. This generator would normally be driven by a steam turbine but in the simulators it is driven by a direct current motor. While this type of simulator has been very successful, it has the disadvantage that once it is designed and built, it represents one particular thermal-electric generating unit and its characteristics are not easily changed and, in fact, usually are not changed. The effect of the inflexible nature of the system is to allow only operator training for one particular station and no means to study the control problems of a plant before the plant is built.

In order to circumvent the various drawbacks of the "direct analog" simulators, successful attempts have been made to represent the thermal-electric plant with mathematical relationships and to subsequently solve these equations using either electronic-analog or digital computers. These are listed also in Table 1.1. The papers of J. K. Dillard and J. L. Everett; F. T. Thompson; D. J. Ahner, C. E. Dyer, F. P. deMello, and V. C. Summer; F. P. deMello; B. L. Littman and T. S. Chen; and J. Adams, D. R. Clark, J. R. Louis and J. P. Spanbauer are of particular importance since they illustrate in more detail the procedure that is involved in making such a simulation study. Basically the methods used by these men and their associated study teams were similar, but the details of the solution procedure and the technique they employed to solve the resulting "model" system were different.

The general approach taken in a simulation study such as has been carried out consists of first, developing a feel for the overall system; second, setting up the general equations necessary to describe the various processes involved; third, determining and programming the logic necessary for interconnecting
### TABLE 1.1

**LIST OF PAST CASES OF SIMULATION**

<table>
<thead>
<tr>
<th>Author</th>
<th>Title</th>
<th>Process</th>
<th>Technique</th>
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</thead>
<tbody>
<tr>
<td>Whitesell, Bowles (American Oil)</td>
<td>Train Power Station Operation by Analog Jan., 1965</td>
<td>Power station operation</td>
<td>Digital, 120 linear D.E., lumped parameter</td>
</tr>
<tr>
<td>Dillard, (S). Everett (Phil. Elec.)</td>
<td>Simulation of the Steam Power Plant, 1961</td>
<td>Drum type boiler, turbine</td>
<td>Analog, air elec., general</td>
</tr>
<tr>
<td>Gardner, C. M. (SCE)</td>
<td>Simulated Controls Aid Operator Training Oct., 1965</td>
<td>(Bailey &amp; SCE) Power Plant Operation</td>
<td>Analog, air général</td>
</tr>
<tr>
<td>Howard, E. D. (SCE)</td>
<td>Boiler Simulator Control Circuit Designs Simulator Trains Plant Operators</td>
<td>Feedwater and Combustion Control</td>
<td>Analog, air (Pneumatic) general</td>
</tr>
<tr>
<td>Garrett, R. T.</td>
<td>A Simulated Boiler and Control Board for Operating Procedures Training</td>
<td>(Con. Ed. N.Y. by Curtis Wright) Boiler, Boiler load control</td>
<td>Analog EDA Digital to get transient response</td>
</tr>
<tr>
<td>Author</td>
<td>Title</td>
<td>Process</td>
<td>Technique</td>
</tr>
<tr>
<td>---------------------</td>
<td>----------------------------------------------------------------------</td>
<td>------------------------------------------------------------------------</td>
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</tr>
<tr>
<td>Dyer, C. E.</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(General Electric)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Summer, V. C.</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(S. C. Elec. and gas)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Thompson, F. T.</td>
<td>Plant Dynamics and Control Analysis</td>
<td>Linearized boiler turbine-a reheater superheater nonlinear DE-b</td>
<td>Analog EDA-a</td>
</tr>
<tr>
<td>(Westinghouse)</td>
<td></td>
<td></td>
<td>Digital-b</td>
</tr>
<tr>
<td>DeMello, F. P.</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(General Electric)</td>
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</table>
and operating the simulation; fourth, solving the equations developed using the computing equipment available and fifth, producing useful simulated output.

Since the dynamic behavior of the physical apparatus and processes should be represented by a set of nonlinear, partial differential equations, (these are equations where the dependent variables and their differential coefficients are greater than first degree), the formal solution is generally impossible and a straightforward control-system synthesis technique of a nonlinear system has been limited to low order systems with only a very few controlled variables.

Because of these limitations, most of the study groups have only approximated the nonlinear power plant with a linear model. The procedure one would follow to do this is to first develop the nonlinear, ordinary differential equations which would apply and then to linearize these equations about a steady-state operating point. The obvious limitation to this procedure is, of course, the fact that the use of these linearized equations is restricted to the chosen operating point and to small excursions about it.

Also the variables used in the basic partial differential equations governing the fluid flow and heat transfer are generally functions of all three space dimensions, as well as time. It has been found that the temperature of the outlet of a long heated tube may be determined for simple configurations, if the problem variables are only a function of time and the tube length. For complex configurations, however, the solution is not practical without the additional use of some device such as "space" or "time" lumping. This gives rise to the familiar lumped parameter representation of a distributed parameter system. Most of the simulations, as well as this one, have incorporated the space lump scheme. Schmidt & Clark describe the use of the time lump in their paper.

Historically the first record of a commercially operating plant designed to be of the supercritical type was cited in the article by P. W. Swain who was writing in Power in 1923. These two early papers that describe the Benson Super Pressure Plant pointed out the economic reasons for going to a supercritical unit and they are the same ones that are presently used in the description of the present day modern supercritical units. That is: (1) a large
saving in fuel costs, (2) a reduction in initial cost, (3) substantial saving in floor space, (4) substantial saving in weight.

The plant that was described almost 50 years ago was an experimental test plant that was erected in Rugby, England with an equivalent rating of 1 megawatt. The steam was to be generated at a pressure above 3200 psia and then throttled with a throttle valve to 1500 psia before being superheated and then entering the high pressure turbine at 1500 psia and 788°F.

The cycle is very similar to the cycle to be described for the modern supercritical plants and the basis under which the present plants operate is identically the same. The water is compressed using a boiler feed pump to a rather high pressure above the critical pressure and then water is heated and turned into steam at this high pressure. There is no boiling that takes place but rather the water changes to steam as it passes through the critical region. Since boiling does not occur in the normal sense there is a marked saving in the heat that is not required to evaporate the water which is the primary reason for the economic savings in the operation of the plant.

As mentioned earlier the simulation that is being used is a lumped representation of the plant. The 23 lumps that are represented are shown in Figure 1.2 which is the fluid and gas flow paths for the supercritical once-through unit. The fluid leaves the boiler feed pump at a high pressure and enters the high pressure feedwater heater and then passes into the main part of the boiler starting with the economizer and through the various lower and upper wall passes of the furnace. During these final passes in the furnace the fluid changes from a liquid form to a vapor form (steam) and passes into lump 9, the primary superheater and on to the finishing superheater through the steam leads to the throttle valve, lump 16, where it is throttled before passing into the high pressure turbine. After exhausting from the high pressure turbine the steam is reheated and passed again through the intermediate and low pressure turbine from which it exhausts to the condenser. The fluid then is pumped by the condensate pump from the hot well to the low pressure feedwater heater to complete its cycle.
Fig. 1.2 Fluid and gas flow paths for supercritical once-through unit.
Note that there are two levels of extraction, one from the high pressure turbine and one from the intermediate and low pressure turbine as indicated by the flows WE1 and WE2 respectively. These extraction flows of steam are used to heat the feedwater before it passes into the furnace and the condensate from this extracted steam is recirculated as shown in the diagram.

The combustion gas flow follows the path from the air heater where the air comes from the atmosphere and is then combined with the fuel which, in this case of the simulated plant, is natural gas where it is burned as is described in a later section. The combustion process gives off heat to the various portions of the furnace through the radiation and convection passes. After passing through the finishing superheater the combustion gas passes to one of two different paths indicated by WG(2) and WG(3). The damper, block 22, determines what proportion of the total combustion gas flows over either the primary superheater or over the reheater. The gases are then recombined and passed over the economizer and go out to the stack through the air heater. The detail of the heat transfer and other equations will be discussed in later sections throughout the paper.

In Figure 1.3 a side elevation of the steam generator is presented and in it the various parts of the steam generator are seen labeled and can be recognized in relation to the total fluid and gas flow paths as was described in relation to Figure 1.2. Specifically the front of the steam generator is given to the left of the figure where the furnace is labeled. Pass number 1 is the floor pass which consists of tubes through which the water flows and is heated by radiation. Pass 2 and part of pass 5 are front wall passes. Pass 3 and part of pass 5 are side wall passes. Pass 4 is a rear wall pass.

The natural gas comes through the burners, is combined with the air in the furnace, and combustion takes place. The hot combustion gases then flow over the platen superheater indicated at the top of the front of the furnace and then pass over the finishing superheater. From there they go to the rear of the steam generator which is the right of the Figure 1.3 and the damper then directs the proportion of the combustion gas flow over either the reheater or
Fig. 1.3 Side elevation of the steam generator
the primary superheater sections. The combustion gas then flows over the economizer out through the air heater and to the stack where it is exhausted.

In order to get a feeling for the various properties of the fluid as it passes through the system an approximate profile is indicated in Figure 1.4 for the pressures, enthalpies, and temperatures from the economizer to the output of the steam leads which is the input to the throttle valve. Here is seen that the input condition of the fluid to the economizer is at approximately 500°F with a pressure of approximately 4100 psia while after transition through the various parts of the steam generator it exits the steam leads in preparation to enter the turbine at approximately 3500 psia with a temperature of approximately 1000°F. The detailed heat cycle will be seen in a later section.

In addition to the simulation of the process as has been indicated the simulation includes also the portions of the control system that are required to control the main functions of the unit. This is basically the feedwater flow control, the reheat steam temperature control, a firing rate control, and a fuel flow control. The two simulated parts, that of the control system and the process have been combined to form an integral part such that the complete steam generator can be simulated as will be described throughout the paper.

As has been mentioned earlier most of the simulations that had been done have treated the equations in a linearized form.

The uniqueness of this presentation is that the equations themselves are not linearized but are treated and solved as much as possible in their non-linearized forms. In particular, throughout the simulation the values of the state properties of the water or steam, that is the working fluid, are computed at each time interval and the assumptions are not made as is often made that the properties themselves do not change. This then does not restrict the operation to a specific operating point in the sense that if variations occur there are provisions in the simulation to make new computations for the steam properties. This is a very important feature in the supercritical plant due to the fact that the process, itself, as is seen in the heat cycle, passes directly through the
Fig. 1.4 Initial condition temperature, pressure, and enthalpy profile

critical region where the properties of the working fluid change rather dramatically with rather small variations in independent variables.

Since this simulation considers both the process and the control system in such detail as will be described in future sections the simulation is unique. In addition, during the plant testing, a large number of variables were recorded and plotted which will be useful for further studies.

The basic purpose of the research has been to create a simulation package that will be useable for performance studies on the supercritical unit and for other studies. It is desired to present the simulation in a manner that is sufficiently documented such that the reader can follow each step in the simulation and make modifications or additions where the reader might so desire.
CHAPTER 2

THE FUNDAMENTAL EQUATIONS USED IN THE PROCESS

In the introduction, general comments were made regarding the entire simulation and its relative complexity. The purpose of this chapter is to take the general statements and equations and translate them into forms that are suitable for the subsequent digital simulation.

As will be seen in detail later, the simulation is divided into two distinct areas. They are the simulated control system and the simulated process. The equations that are used in the control system simulation are based on an analysis of the existing control system in the simulated plant and are described in detail in Chapter 7. Hence this chapter is devoted to an investigation of the equations that are used in the process simulation.

The basic equations that must be considered for the process simulations are the following:

A. Mass balance
B. Energy balance
C. Momentum
D. Heat transfer
E. Equations of state of the working substance

The mass balance equation is essentially the continuity equation. For a general compressible fluid, the vector form of the continuity equation is

\[ \frac{\partial \rho}{\partial t} + \nabla : (\rho \vec{V}) = 0 \]  

(2.1)

where \( \rho \) = density \( \text{lb} \text{ ft}^{-3} \)

\( t \) = time (sec)

\( \vec{V} \) = velocity \( \text{ft sec}^{-1} \)
The mass flow rate of fluid at any point in a tube may be expressed as

\[ W = \rho VA \frac{lb}{sec} \]  

(2.2)

where \( A \) = cross section of tube (ft\(^2\))

This relation may be observed with the following illustration. Think of a cube of fluid with density \( \rho \), moving with a velocity \( V \) through an area \( A \).

Then the mass \((\rho A)\) dl moves past a point in \( dt \) seconds so that the "mass flow rate" \( W \) is,

\[ W = \rho A \frac{dl}{dt} = \rho AV \frac{lb}{sec} \]

Now consider the mass flow rate as \( W_i \) at the inlet of the elementary volume described above, and \( W_o \) at the outlet. Then \( W_i - W_o \) is the total mass change per unit time.

Since the mass is given by,

\[ m = \iiint \rho \, dv \]  

(1b)  

(2.3)

where \( v \) = volume (ft\(^3\))

the rate of change (increase) of mass is,

\[ \frac{\partial m}{\partial t} = \frac{\partial}{\partial t} \iiint \rho \, dv \]  

(2.4)
Now considering the area of the tubes fixed, the volume is $AL$ and

$$\frac{\partial m}{\partial t} = \frac{\partial}{\partial t} \int_{0}^{L} A \rho \, dl$$  \hspace{1cm} (2.5)

Now assuming that within the elemental volume the spatial variation in density is negligible. Equation (2.5) becomes

$$\frac{\partial m}{\partial t} = \frac{\partial}{\partial t} \left( A \rho L \right) = A \frac{\partial (\rho L)}{\partial t}$$  \hspace{1cm} (2.6)

Hence

$$W_i - W_o = A \frac{d}{dt} (\rho L) \frac{lb}{sec}$$  \hspace{1cm} (2.7)

The energy balance equation stems from the first law of thermodynamics, which in turn is derived from the law of conservation of energy. First, look at the application of the energy balance equation to an open system, under a steady flow. Figure 2.1 below represents the thermodynamic system.

![Fig. 2.1 Open thermodynamic system](image_url)
The energies associated with the system are as follows:

<table>
<thead>
<tr>
<th>Energy Type</th>
<th>Expression</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Normalized kinetic energy</td>
<td>$\frac{V^2}{2g}$</td>
<td>ft lb/lb</td>
</tr>
<tr>
<td>Potential energy</td>
<td>$Z$</td>
<td>ft lb/lb</td>
</tr>
<tr>
<td>Flow work</td>
<td>$Pv$</td>
<td>ft lb/1b</td>
</tr>
<tr>
<td>Internal energy</td>
<td>$u$</td>
<td>btu/1b</td>
</tr>
<tr>
<td>Work done on the fluid</td>
<td>$W_K$</td>
<td>ft lb/1b</td>
</tr>
<tr>
<td>Heat added to system</td>
<td>$Q'$</td>
<td>ft lb/1b</td>
</tr>
</tbody>
</table>

where $V = \text{velocity}$

$v = \text{specific volume}$

These various energies may be put together into the energy balance equation (2. 8) whereby the energy in all forms entering a system must equal the energy in all forms leaving the system.

Note that (# of ft-lb) = (# of btu) 778

\[
\frac{V_1^2}{2g778} + \frac{P_1 v_1}{778} + \frac{Z_1}{778} + \frac{W_K}{778} + Q' = \\
\frac{V_2^2}{2g778} + \frac{P_2 v_2}{778} + \frac{Z_2}{778} 
\]

(2.8)

By defining enthalpy as being equal to the sum of the flow work and intrinsic energy of the fluid ($h = \frac{Pv}{778} \text{ btu/1b} + u \frac{\text{btu}}{1\text{b}}$), equation (2.8) becomes

\[
\frac{V_1^2}{2g778} + \frac{Z_1}{778} + \frac{W_K}{778} + Q' = \\
\frac{V_2^2}{2g778} + h_2 + \frac{Z_2}{778} \text{ btu/1b} 
\]

(2.9)

When the system energy is considered under dynamic conditions, another form of the energy equation (first law of thermodynamics) should be used.

For the boiler sections, the internal energy corresponding to thermal energy is of much larger magnitude than the kinetic and potential energy terms.
Hence, for this simulation the kinetic and potential terms are neglected. The shaft work is also zero in the boiler sections so that the only terms affecting the energy balance are the heat flux and internal energy terms. The mass flow rate, as defined earlier, is \( W \text{ lb/sec} \). When this is multiplied by the enthalpy an equivalent power equation results. That is, the energy equation becomes a power equation, where power in this case has units, btu/sec.

\[
\frac{\partial t}{\partial t} (\rho \text{ave} \ h \text{ave}) = \frac{W_i \ h_i - W_o \ h_o}{v} + \frac{Q' \ W \text{ ave btu}}{v \ sec-ft^3} \tag{2.10}
\]

where the heat flux to the volume from the combustion gas is

\[ Q' \ W \text{ ave} = Q \frac{\text{btu}}{\text{sec}} \]

The average heat flux density change, with respect to time, is given by equation (2.10). The average values indicated in equation (2.10) are spatial averages.

\[
\frac{\partial t}{\partial t} (\rho \text{ave} \ h \text{ave}) = \rho \text{ave} \frac{\partial h \text{ave}}{\partial t} + \left( \frac{\partial \rho \text{ave}}{\partial t} \right) \ h \text{ave}
\]

Now assuming that the change in density with respect to time is negligible, that is \( \frac{\partial \rho}{\partial t} = 0 \)

Hence equation (2.10) may be rewritten

\[
v \left\{ \rho \text{ave} \frac{dh \text{ave}}{dt} \right\} = W_i \ h_i - W_o \ h_o + \frac{Q \text{ btu}}{\text{sec}} \tag{2.11}
\]

but \( v \rho \text{ ave} = \text{mass} = m \text{ (lb)} \), hence

\[
\frac{mdh \text{ave}}{dt} = W_i \ h_i - W_o \ h_o + Q \left( \frac{\text{btu}}{\text{sec}} \right) \tag{2.12}
\]

Now if the assumption is made that \( \frac{\partial m}{\partial x} = 0 \), that is the mass change of the fluid along the tube is zero, then
One final assumption needs to be made. That the time rate of change of the output enthalpy is equal to the time rate of change of the average enthalpy in the elemental volume. Then

\[
\frac{m \, dh_{\text{ave}}}{dt} = W (h_i - h_0) + Q
\]  

Equation (2.13)

Equation (2.14) is used in the subsequent simulation of the boiler.

The heat storage capability of the tube wall must be considered and is also an energy balance equation. In this relation, the net gain in heat flux contributes to a changing wall temperature. The equation below gives the relation.

\[
\frac{m \, dh_w}{dt} = Q - W (h_0 - h_i) \frac{\text{btu}}{\text{sec}}
\]  

Equation (2.14)

The momentum equation stems from the pressure distribution across the system and it is this momentum which assures the flow of fluid in the circuit. In the simulation it was assumed that the elevation term and the acceleration term of the pressure drop equation were negligible compared to the friction term.

The momentum equation is given as (see reference 1)
\[ P_i - P_o = \frac{f L \pi D W^2}{115} \left( \frac{2}{g_o \rho A} \right)^3 + \frac{g L \rho}{g_o (144)} + \frac{1}{g_o (144) A} \frac{d(WL)}{dt} \]  

Hence

\[ P_i - P_o = K p W^2 \quad \text{(psia)} \]  

Equation (2.17) is used in the simulation.

The heat transfer from the gas to the metal is composed of a radiation and a convection term and is given by the following equation

\[ Q_w = \sigma \varepsilon A (T_g^4 - T_w^4) + U_g A (T_g - T_w) \]  

where \( U_g \) = convection film conductance, \[ \left\{ \frac{\text{btu}}{\text{ft}^2 \cdot \text{sec} \cdot \text{°F}} \right\} \]

The method of computing the heat flux from the combustion gas to the metal is given in Chapter 5. This value of \( Q_w \) is then utilized in the process solution in equation (2.15)

The heat transfer equation representing the heat flow by conduction from the metal tube wall to the working fluid is

\[ Q = U_f A (T_w - T) \]  

where

\[ U_f = \frac{0.023 C_{pw}^{0.4} K_{0.6} T_b^{0.8}}{d_i^{0.2} \mu^{0.4} T_f^{0.8}} \left\{ \frac{W}{A} \right\}^{0.8} \]

\( T_w = \) wall temperature

\( T = \) fluid temperature

For the simulation the convection film conductance term is taken as a constant except for the fluid flow. With the assumption that the output fluid
temperature of the lump, \( T_o \), is the average fluid temperature as in equation (2.14), the equation becomes

\[
Q = K q W^{0.8} (T_w - T_o)
\]

(2.20)

The state equations are used in the form \( T = f(p, h) \). These equations describe the properties of the working fluid and are essentially a model of the steam tables. The detail of the method by which they were simulated is given in Chapter 4. An important feature of this simulation is the fact that the properties of the working fluid are obtained by recalculation at each time increment of the integration.

This gives five equations that must be solved simultaneously for each lump. Two of the equations are first order differential equations and two of the equations are nonlinear algebraic equations. The last equation represents the properties of steam, and is highly nonlinear.
CHAPTER 3

DESCRIPTION OF THE PROCESS

3.1 ECONOMIZER, FURNACE, SUPERHEATER, STEAM LEADS, REHEATER, AIR HEATER

The boiler, or as it is sometimes called in this simulation, the steam generator, is the device from which the thermal energy is taken from the burned fuel, in this case natural gas, and then transferred into the water which is circulating in the primary cycle of the plant. Within the boiler the pressurized water is converted into steam before it passes on to the super heater. This is probably the most significant portion of the entire generator.

Figure 3.1, below, illustrates the basic fundamentals in getting energy from the fuel to the furnace passes, convective heating surfaces and into the water. The feedwater enters the steam generator in the primary plant cycle while the fuel enters the combustion area of the furnace and is burned. There is appropriate control of the unit to permit satisfactory output of the steam and still maintain adequate plant operation. Heat is transferred into the various heat absorbing surfaces to the working fluid and finally the steam leaves the steam generator to be used in the turbine to convert the energy available into power.

The design of the steam generator is based on economic and the thermodynamic principles. During the past decades in which the development of the steam generator has been carried out there have been many improvements to increase the efficiencies of these particular units. The units may be classified basically into three different categories. The first category being shell type boiler, the second being fire tube type boiler and the third being a water tube type boiler. Each of these individual categories can then be subdivided into classes which would distinguish themselves as being particular designs.
Fig. 3.1 Basic boiler

The shell type boiler basically is one in which water is heated much as it is on the stove in a teakettle in which a kettle of water, a closed shell, is heated. The steam then arises out of the top of the shell and is carried off through the steam outlet valve to drive a turbine. This is one of the earlier designs, of course.

The fire tube boiler is one in which the fire actually goes through the tubes with the water being on the outside of the tubes.

The more modern boiler is now considered to be the water tube type boiler in which the working fluid, water in this case, passes through the boiler tubes and the hot combustion gases pass over the outside of the tubes. The water tube type boilers can be subclassified into straight tube boilers and further into those with drums or without drums. The use of the water tube boiler has allowed greater efficiency in the units. Reportedly as high as 90% boiler efficiencies have been obtained with this particular type of unit. The boiler efficiency at the simulated plant is estimated to be about 85%.

As has been mentioned in an earlier section, the plant which is being simulated is the once-through supercritical boiler and it has no drum. In recent years the furnaces have been integrated to include a unitized concept where the boiler, the superheater, the furnace and the economizer are treated
as one complete unit rather than considering the furnace and other units as individual pieces. In addition, it is now standard with the large size units, such as a simulated plant, to have one boiler for one turbine combination. This is due to the very large economic incentive to have very large electrical generators based on the fact that investment in labor costs and in operation is much decreased when the size of the units is increased.

Generally speaking the larger units have a relatively high steam pressure and temperature and use feedwater heating and reheaters. All of these concepts are used on a simulated plant and are described in the present section.

The high steam pressure means that there is a high saturation temperature of the water and this gives rise to the relatively low temperature difference between the boiling point and the steam utilized in the turbine. A practical limit is placed on the materials which limits the maximum temperature which the steam can have when it enters the turbine and when it is in the superheater tubes before it gets to the turbine. Basically this smaller temperature difference is caused by the need for the higher temperature in which to get the water boiling when the water is under the higher pressure. Thus, the high steam temperature required gives design impetus to a high initial temperature in the unit and then also means that the steam must be reheated to a relatively high temperature again when the steam is used again in the intermediate and the low pressure turbines.

The feedwater heating, which is obtained by the extraction of steam from the intermediate, low pressure, and high pressure turbine gives rise to an increase in the combustion gas temperature which leaves the economizer. This is because the economizer follows the feedwater heaters in the working fluid cycle and must be designed for proper heat transfer from the combustion gas to the working fluid. Due to the fact that this increased combustion gas temperature leaving the economizer would give rise to an uneconomical operation, air heaters are then used which lower the final combustion gas temperature before the gases leave the stack into the atmosphere. In addition to this the air heater has the advantage that it also increases the temperatures of the
hot atmospheric air that is used for combustion. Hence, when all of these different considerations are taken into account the overall efficiency can be increased, even though some parts of the plant efficiency are reduced.

It is then seen that as the steam pressures are increased, as is the case with this supercritical unit which is being simulated, the steam temperatures must also be increased, and then this gives rise also to a proportionally greater amount of superheating surface and less amount of boiler surface than may have been used in previous style units.

As mentioned earlier in the discussion the principle of operation of the once-through supercritical boiler such as is used in the simulated plant is based on the Benson principle. In this case the working fluid is passed through the unit one time and is recirculated. It passes in sequence from the economizer to the furnace and the superheater, to the turbines, the re heater, the low pressure turbine sections, condenser and then the re heaters, boiler feed pump and then the high pressure feedwater heaters. See Figure 1.2. In this process, the working fluid, compressed water in this case, absorbs the heat in the various furnace sections through radiant heating and in the other passes through convection heating principles. As it absorbs this heat, the fluid is turned into steam and then it leaves the various sections reasonably near the desired steam temperature so as to not adversely affect the materials of which the unit is built.

There is no specific internal recirculation as is the case with the drum type boiler and as a matter of fact this is the reason then that a drum is not required to separate the water from the steam. Because the unit is operating at supercritical pressure (that is, above 3206.2 psia) the water forms a mixture with the steam and in the transition period it is not clear how the boiling process takes place. The water simply makes the transition from liquid form into the vapor form (steam). The actual cycle through which the fluid passes will be described in some detail in a later section and hence will not be described at this time.
For the best efficiency in a steam generator, it would be desirable to use the highest available temperature of the source that is available. In the steam generator itself, the transfer of heat takes away from the available energy that is released in the furnace. The combustion process takes place at approximately 2800°F; however, the heat that is transferred to the water takes place at a temperature less than 705°F and in terms of a vapor the heat transfer will take place at temperatures less than 1100°F. Since there are limits, as mentioned earlier, on the steel that is used to hold the water and have the water circulate through the unit, temperatures that are in excess of 1100°F are prohibited for all practicality. Hence, there is a large loss in the available energy that can actually be utilized and therefore the efficiency of the units is cut down somewhat from what it might be under an entirely theoretical standpoint.

Figure 3.2, below, illustrates the basic principles of one of the two most common types of steam boilers. That is of the steam drum with natural circulation. Viewing Figure 3.2 it is seen that the water goes down through an unheated downcomer and then out into the furnace, possibly being parts of the furnace wall which may be used for water cooling of the metal exposed to the radiative heating of the flame. Heat transfer takes place in the heated riser in which the steam bubbles are formed and are mixed with the water where they rise into the steam drum and are allowed to escape to the top of the drum which is then a steam mixture. The steam then is allowed to escape from the steam outlet after appropriate measures are taken to reduce any water content in the steam to a low enough level such that the rest of the steam generator is not adversely affected. For example, if there is a too high moisture content in the steam, the turbine blades might become pitted, or the superheater sections might become excessively corroded due to dissolved salts that are in the water droplets which are in the steam and provide operational difficulties in terms of burned out tubes.
The natural circulation principle, shown in Figure 3.2 has been used in boilers which utilize pressure up to about 2,600 psia, and the velocity of the water is generally considered high enough to give effective heat transfer without burning the tubes.

Figure 3.3 illustrates the once-through type of steam generator which is the type used in the simulated plant. In Figure 3.3 it is seen that the heated feedwater from the feedwater pump passes directly through the economizer and through the furnace of the boiler where radiant heat is transferred to the fluid.
The working fluid then passes through the convectively heated surfaces of the furnace and superheaters, then leaves without any recirculation as superheated steam which is then admitted into the turbine. In this type unit there is a forced circulation rather than a natural circulation and the forced circulation is thereby controlled so that the tube temperatures are maintained at values which are below their thermal limits.

Since the simulated plant is of the integrated type; that is, the complete design is integrated into one particular unit, the following items in the plant are going to be considered together: the boiler (steam generator), the economizer, the superheater, the steam leads and the reheater.

The combustion gases which leave the boiler convection heating surfaces, the finishing and primary superheating surfaces, still have a relatively high temperature which is considerably higher than the steam saturation temperature. In order to make the unit economical some of this energy must be recovered and, in this type of unit, it is economical to recover this heat by the use of the economizer. In the case of the economizer, the gases pass over the tube surfaces in exactly the same manner as they did over the other surfaces and, hence, it is still considered to be a water tube type of heat recovery area. The boiler feedwater, which has just come out of the high pressure feedwater heater, passes into this economizer which is placed in this unit in the rear part of the furnace. See Figure 1.3. The feedwater then picks up the heat by virtue of convection heat transfer before it passes into the boiler itself.

The feedwater temperature entering the economizer in this unit is approximately 500°F, which is above the dew point of the boiler combustion gases. If the combustion gas was allowed to condense it would cause corrosion of the tubes.

There are two basic types of superheaters available. One is the convection superheater and the other is the radiant superheater. In the simulated plant, a combination of both is being used. In the original designs of steam generators the use of the superheater was desired in order to increase the final steam temperature before entry into the turbine. The convection type unit was used
because the furnace gas temperatures, where the superheaters were placed, were relatively low. With this type unit which used only convection superheating, it was found that as the steam output increased, more water was flowing through the boiler part of the furnace and hence there is less heat pick up in the furnace walls themselves by the radiant heat transfer and, therefore, there was more heat left over in the combustion gases to be transferred to the superheated steam. Hence, when the boiler output increased, the steam temperature increased due to the increased heat which was given to the superheater.

To alleviate this problem, the radiation and convection superheater in current designs are placed in series. The result is a fairly uniform final steam temperature, such that the steam which enters the turbine enters at a fairly uniform temperature at varying loads simply by virtue of the amount of heating surface in the superheater which is allotted to the radiant and to the convection heating surfaces. This is illustrated in Figure 3.4. One of the major advantages of using superheating is to reduce the amount of moisture content in the steam as it enters the turbine and this not only decreases the erosion effect of the water in the steam, but it also increases the engine efficiency of the turbine.

![Radiant-convection superheater combination](image)

**Fig. 3.4 Combination radiant and convection superheater effects**

The purpose of the reheater is to take the steam which is exhausted from the high pressure turbine and to increase its heat content, by virtue of essentially a superheating effect, back up to an elevated temperature in order to
allow it to pass properly through the intermediate and low pressure turbine stages of the unit. The simulated plant has one stage of reheat and the purpose of the reheater is exactly the same as the superheater which is ahead of the high pressure turbine. The reheater is located in the rear of the furnace (see Figure 1.3) and combustion gas passes over the reheating surface with the steam passing through the tubes and the heat transferred to it.

In addition to the elements of the boiler which have been mentioned thus far there is an air heater which takes the last amount of heat from the combustion gases before they are exhausted to the atmosphere. The fresh air is preheated before it is allowed into the combustion area where the fuel is burned with the air. In the simulation, the air heater is not considered; however, it seems appropriate to include a brief description of the air heater at this point. The type of air heater which is used in the simulated unit is a Ljungstrom continuous regenerative type preheater. It was developed in Sweden, in the early 1920's.

The air heater consists of a large housing divided into two outer compartments and one middle compartment in which the heating surface is contained in a slowly moving rotor. The outside compartments are divided by partitions which confine the hot gas to one side of the apparatus while the air heated is on the other side. For each revolution of the rotor there is a complete cycle exchange in which the heat from the hot gas is absorbed by the regenerator method of heating surface and the heat is then given up as the rotor moves into the path of the air which is to be heated. This is illustrated schematically in Figure 3.5 below. The top picture of Figure 3.6 shows the side view of the air heater while the bottom picture shows the air heater under construction.

The steam leads are considered in this simulation since the plant is operating at the very high pressures of steam (above 3500 psia) and consequently contains a large mass of metal. Since the leads are so long and massive there is a definite change in the enthalpy as it leaves the superheater, before it gets to the turbines. This is then considered in the steam leads the same way as it
is considered in the other lumps, except in this case the heat transfer is out of
this lump instead of into the lump. The water treatment facility is shown in
Figure 3.7. The steam leads are shown in Figure 3.8.

The combustion gases are moved with a forced draft fan. There are two
fans which are controlled by the control system, however a constant fuel to air
ratio has been assumed, as mentioned before, and although this aspect will be
considered, it will not be an operating part of the computer program.

The condenser of the simulated plant is not being included in the
simulation. The condenser takes the steam from the low pressure turbine and
simply condenses it to water by virtue of passing cooling water from the
nearby river into and through the tubes and by passing the steam over the out-
side of the tubes where the steam then is condensed into water and the water is
then passed into the condensate pump and is recirculated through the system
again. It should be noted that there is a complete division between the cooling
water of the condenser and the working fluid which is being cooled.

In the following section some of the plant constants that are useful in the
simulation are discussed, computed, and tabulated. The dimensions of the
various parts of the plant such as the tube sizes, (outside diameters, inside
diameters, lengths) and the computation of the corresponding volumes are
computed. The volumes are computed for the economizer, the furnace
passes 1, 2, 3, 4, 5, 6, 7, the primary superheater, the finishing superheater,
Fig. 3.6 Air heater
Fig. 3.7
Water treatment facility

Fig. 3.8 Steam leads
and the reheater. The interior volume is also computed for the steam leads, which is lump 11.

The volumes are required in the simulation in order to consider the mass change, or the density change of the working fluid so that this aspect could be included in the simulation. Two checks were given on this computation of the volume. One was obtained from the boiler manufacturer where the volumes were computed approximately by them; the other was taken from the construction diagrams in which the dimensions of the various tubes were given. A summary of the dimensions is given in Table 3.1.

The final column of Table 3.1 gives the equivalent tube volumes in cubic feet. The computation of the volume for the steam leads was obtained from the main steam piping diagram and was computed in the same way as the other volumes.

For an example of the computation involved for a typical lump consider the primary superheater, lump 9. The tube length is 140 feet; there are 513 tubes in parallel with an average outside diameter of 2.25 inches and a wall thickness of 0.4 inch.

\[
R = \frac{OD}{2} - w.t.
\]

where \( R \) = radius in (inches)

OD = Outside diameter of tubes (inches)

w. t. = Wall thickness of tubes (inches)

For this case, \( R = \frac{2.25}{2} - 0.4 = 0.725 \) in. The area of a tube \( \text{in ft.}^2 \) is given by

\[
A = \frac{\pi R^2}{144} \text{ (ft}^2\text{).}
\]

For this case \( A = \frac{\pi}{144} (0.725)^2 = 0.01142 \text{ ft.}^2 \). The volume of the tubes is then

\[
V = AI = (0.01142)(140) \text{ ft.}^3 = 1.605 \text{ ft.}^3.
\]

For 513 parallel tubes this gives an equivalent tube volume of 823. \text{ft.}^3.

The dimensions of the steam leads are given in Table 3.2. These dimensions were then used to compute the volume for Table 3.1. Table 3.3 gives the mass of the working fluid in the steam generator that is used in the
<table>
<thead>
<tr>
<th>Description</th>
<th>Lump No.</th>
<th>Tube Length (ft.)</th>
<th>Tube Area (ft²)</th>
<th>Number of Tubes</th>
<th>Equiv. Area (ft²)</th>
<th>Tube Equiv. Volume (ft³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Economizer</td>
<td>1</td>
<td>136</td>
<td>0.0182</td>
<td>171</td>
<td>3.12</td>
<td>1170</td>
</tr>
<tr>
<td></td>
<td></td>
<td>312</td>
<td>0.01405</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pass 1</td>
<td>2</td>
<td>50</td>
<td>0.00386</td>
<td>430</td>
<td>1.67</td>
<td>83.3</td>
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<td></td>
<td>2</td>
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<td>0.00386</td>
<td>493</td>
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<td>3</td>
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<td>463</td>
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<td>125.</td>
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<td>1006</td>
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<td></td>
<td>6</td>
<td>Variable¹</td>
<td>Variable</td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td>7</td>
<td>36</td>
<td>0.00984</td>
<td>258</td>
<td>2.54</td>
<td>305.4</td>
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<td></td>
<td></td>
<td>42</td>
<td>0.0198</td>
<td></td>
<td></td>
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<tr>
<td>Primary Superheater</td>
<td>9</td>
<td>140</td>
<td>0.01142</td>
<td>513</td>
<td>5.86</td>
<td>823.</td>
</tr>
<tr>
<td>Finish Superheater</td>
<td>10</td>
<td>66</td>
<td>0.0156</td>
<td>336</td>
<td>5.25</td>
<td>596.</td>
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<td></td>
<td>108</td>
<td>0.00692</td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>Steam Leads</td>
<td>11</td>
<td>Variable²</td>
<td>Variable</td>
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<td></td>
<td>309.</td>
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<tr>
<td>Reheater</td>
<td>12</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>3261³</td>
</tr>
</tbody>
</table>

¹Six parallel paths
²See table 2
³Value obtained from Manufacturer
simulation. The volumes of the various lumps that are given correspond to the manufacturer's data, and are comparable to the previously computed volumes as shown in Table 3.1.

**TABLE 3.2**

STEAM LEAD DIMENSION

<table>
<thead>
<tr>
<th>Section</th>
<th>Outside Diameter (inches)</th>
<th>Mean Wall Thickness (inches)</th>
<th>Approximate Length (ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>22</td>
<td>4.54</td>
<td>34.9</td>
</tr>
<tr>
<td>2</td>
<td>22</td>
<td>4.54</td>
<td>34.95</td>
</tr>
<tr>
<td>3</td>
<td>23</td>
<td>4.75</td>
<td>197.48</td>
</tr>
<tr>
<td>4</td>
<td>22</td>
<td>4.54</td>
<td>28.4</td>
</tr>
<tr>
<td>5</td>
<td>22</td>
<td>4.54</td>
<td>24.75</td>
</tr>
</tbody>
</table>

The values of the specific volume for the corresponding temperatures and pressures are obtained from the ASME Steam Tables. The values of the working fluid mass that are computed are then used in the subsequent simulation to aid in the determination of the dynamic response of the system. In addition to the mass of the working fluid that is required for the simulation, other design data that is also needed is that of the metal weights of the various lumps that are being considered. These metal weights are also included in Table 3.3. The values that are obtained for the metal weights are given by the manufacturing data or they could be obtained by computation of the masses of metals involved in the tubes as outlined in Table 3.1 provided that the density of the metal was known.

The mass of the working fluid is given in column 7 of Table 3.3 and the metal weights are given in column 8 of Table 3.3. In terms of the computer simulation the mass of the working fluid is labeled M and the metal weight is labeled MW. The specific heat of the metal is required for the simulation and it is approximately .169 and is assumed to be .169 for all lumps, although it
<table>
<thead>
<tr>
<th>Lump Description</th>
<th>Lump No.</th>
<th>Average Temperature (°F)</th>
<th>Output Pressure (PSIA)</th>
<th>v</th>
<th>Working Fluid Mass (lb)</th>
<th>Metal Weight (lb)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Economizer</td>
<td>1</td>
<td>530</td>
<td>4075</td>
<td>.0204</td>
<td>942.2</td>
<td>46,150.</td>
</tr>
<tr>
<td>Pass 1</td>
<td>2</td>
<td>575</td>
<td>4040</td>
<td>.0215</td>
<td>134.7</td>
<td>6,620.</td>
</tr>
<tr>
<td>2</td>
<td>3</td>
<td>635</td>
<td>3990</td>
<td>.0238</td>
<td>295.0</td>
<td>12,400.</td>
</tr>
<tr>
<td>3</td>
<td>4</td>
<td>690</td>
<td>3935</td>
<td>.0278</td>
<td>131.2</td>
<td>4,720.</td>
</tr>
<tr>
<td>4</td>
<td>5</td>
<td>725</td>
<td>3865</td>
<td>.0370</td>
<td>152.</td>
<td>4,105.</td>
</tr>
<tr>
<td>5</td>
<td>6</td>
<td>740</td>
<td>3840</td>
<td>.0613</td>
<td>184.1</td>
<td>3,005.</td>
</tr>
<tr>
<td>6</td>
<td>7</td>
<td>750</td>
<td>3825</td>
<td>.0775</td>
<td>418.3</td>
<td>5,400.</td>
</tr>
<tr>
<td>7</td>
<td>8</td>
<td>760</td>
<td>3750</td>
<td>.0942</td>
<td>315.7</td>
<td>3,350.</td>
</tr>
<tr>
<td>Primary Superheater</td>
<td>9</td>
<td>780</td>
<td>3680</td>
<td>.1131</td>
<td>756.</td>
<td>6,690.</td>
</tr>
<tr>
<td>Finish Superheater</td>
<td>10</td>
<td>900</td>
<td>3600</td>
<td>.1697</td>
<td>851.4</td>
<td>5,020.</td>
</tr>
<tr>
<td>Steam Leads</td>
<td>11</td>
<td>1000</td>
<td>3510</td>
<td>.2052</td>
<td>309.</td>
<td>1,505</td>
</tr>
<tr>
<td>Reheater</td>
<td>12</td>
<td>797</td>
<td>644</td>
<td>1.09</td>
<td>3261.</td>
<td>3,000</td>
</tr>
</tbody>
</table>
is realized that this number is variable and dependent upon the type of material used. As the pressure and temperatures change in the boiler itself, the designer's choice of material would change and hence the specific heat of the metal would change but not considerably from this value. If the exact value of the specific heats were known then they could be substituted for this value of specific heat for the various lumps. The computer label for specific heat of metal is given as CPW.

In initializing the process variables it is seen in the computer program that there is a computer variable entitled MWCPW(I) = MW(I) * CPW(I) which groups the product of the mass of the metal, which has been illustrated in Table 3.3, and the specific heat of the metal, which is given as 0.169.

Also, in this same computer DO loop is the computation of the pressure constant and the heat flux constant. These two constants need to be computed from the equations which are described in an earlier section. The pressure drop may be computed for the various sections with the use of the pressure constant. The heat flux constant is used in the equation for determining the heat flux that enters into the working fluid. The pressure constant, KP, is used since the difference in the pressure between input and output of a particular lump is proportional to the flow rate squared. The value of the pressure constant is taken as the ratio of the difference of the pressures divided by flow squared. The value of the KP(I) is computed for the conditions in which the unit is operating at full load, or 560 megawatts, where the corresponding flow is given as 1,105 lb/sec. of steam or water flow. These constants are used for all operating conditions but could be adjusted if it were desired to consider different conditions. It is therefore assumed that the resistance due to the flow does not depend on the amount of load, but is independent of the flow.

In a similar fashion, the heat flux constant KQ(I), as it is used in the computer program, is also computed based on the full load, steady state values. The value of the heat flux into the fluid is proportional to this constant times the difference in the wall temperature and the lump output temperature times the flow rate to the 0.8 power. Hence, the value of this constant KQ(I)
can then be computed by considering the steady state conditions of the heat flux, flow, and temperature difference at a particular lump under a balanced condition. The value of \( KQ(I) \) is obtained from the equation below.

\[
KQ(I) = \frac{QW(1,I)}{(1105.)^{0.8} \Delta T}
\]

where \( QW(1,I) = \) heat flux into lump I \( (\text{btu/sec}) \)

\( 1105 = \) flow of fluid \( (\text{lb/sec}) \)

\( \Delta T = \) temperature difference across wall \( (^\circ F) \)
3.2 BOILER FEED PUMP

The boiler feed pump is the device which pumps the working fluid and, in this case, increases its pressure to a rather high pressure, a supercritical pressure, of 4300 psia at the output of the pump which forces the fluid to circulate throughout the complete once-through cycle. The boiler feed pumps are centrifugal pumps with a rating of 13,970 hp each. The boiler feed pump is driven by the boiler feed pump turbine which is a steam turbine that is fed from the 12th stage of the intermediate pressure turbine. The losses in the pump are not being considered in this simulation. The turbine which drives the boiler feed pump is not in line with the main turbine which drives the generators, but is a separate unit.

The position of the boiler feed pump in the complete cycle is indicated in the Figure 1.2. The boiler feed pump is a very important part of the operation of the system in the sense that the boiler feed pump is used to control the amount of feedwater flow that is allowed through the complete cycle. As will be seen in the control section, the boiler feed pump speed is controlled by virtue of controlling the speed on the turbine which drives the boiler feed pump and as the speed of the pump is controlled by the actions of the control system, the output flow of the boiler feed pump is allowed to vary. The output flow will be seen later to vary as a function, not only of the speed of the boiler feed pump, but also of the output pressure of the boiler feed pump as well as the suction pressure of the boiler feed pump. The suction pressure of the boiler feed pump is considered to be a constant value and this will be explained as the simulation of the boiler feed pump unfolds.

The boiler feed pump is one of the major power users of the auxiliaries in the plant itself and the amount will depend on the pressure output that is desired. As an example a 2,400 psia feed pump uses approximately 2.5% of the gross plant output and one that operates at about 5,500 psia requires up to about 6% of the plant output. Since the operating pressure of the boiler feed pumps in this simulated plant are of the order of 4,300 psia, it would be estimated that the power requirement is around 5% of the rated plant output.
For supercritical plants the speeds of the boiler feed pump are relatively high and in this case a full load speed of 5,300 rpm is used. This particular plant utilizes two, half capacity boiler feed pumps and both are operated for loads from approximately 30 to 100% and then the amount of flow in each is varied according to the feedwater flow that is demanded of the unit in order to supply the required steam output.

Two pictures of one of the turbine driven boiler feed pumps are shown in Figure 3.9.

The general energy equation may be written as

\[
\frac{V_1^2}{2g778} + \frac{P_1 V_1}{778} + u_1 + \frac{Z_1}{778} + \frac{W_K}{778} + Q = \frac{V_2^2}{2g778} + \frac{P_2 V_2}{778} + u_2 + \frac{Z_2}{778}
\]  

(3.1)

The change in elevation head, \( Z_1 - Z_2 = \Delta Z = 0 \). The change in the velocity of the fluid from the inlet to the pump to the outlet of the pump is zero.

\[
\frac{V_1^2}{2g778} - \frac{V_2^2}{2g778} = 0
\]

The heat transferred to the atmosphere or from the atmosphere to the pump is zero.

\( Q = 0 \)

The compressed water is considered to have negligible change in density or specific volume in the pump (that is, it is treated as an incompressible fluid) \( v_1 = v_2 \). Also, the internal energy, \( u \), which is the energy associated with the kinetic energy of the molecules and the forces between them; is generally assumed to be constant in an incompressible fluid pump.

Hence equation (3.1) becomes

\[
\frac{P_1 v}{778} + \frac{W_K}{778} = \frac{P_2 v}{778}
\]

\[
W_K = \left( \frac{P_2 - P_1}{10^5} \right) \left( \frac{ft \cdot lb}{lb} \right)
\]

(3.2)
Fig. 3.9 Turbine driven boiler feed pump
In this case $P_2$ is the boiler feedpump discharge pressure and $P_1$ is the pressure of the inlet, and the total work input to the fluid represents the static head, in ft, that the pump produces.

In this simulation, the primary dependent variable is considered to be the feedwater flow and the primary independent variables that are considered are the boiler feed pump speed, output pressure, and a megawatt demand signal converted to feedwater demand. There were two sources from which the information for this simulation was obtained. The first and primary source is the operating characteristics of the boiler feed pump itself which is comprised of the system head curves. The system head curves give a relation between discharge pressure of the boiler feed pump in psia versus the boiler feed pump flow in thousands of pounds per hour and a third dimension which is the speed of the boiler feed pumps. As mentioned before there are two boiler feed pumps and, for the loads which are being considered, both pumps are considered to be in operation. However, the lowest flow that could be obtained on the system head curves with two pumps operating corresponded to 2,900,000 pounds per hour, which is approximately 74% of rated output. Now, in actuality, both pumps are being operated in the simulated plant down to approximately 40% of rated power and rated flow output. For this reason, the speed characteristics had to be extrapolated down and are indicated in Table 3.4, which indicates the cardinal points of the boiler feed pump characteristics.

The second source of information for the simulation of the boiler feed pump was the actual test run which was made at the simulated plant (this is test run #9) which was a ramp change in load from 500 megawatts to 250 megawatts. This test will be described in a later section in detail.

The information which was required from this particular test for the boiler feed pump simulation was the relationship between the feedwater flow and the power output in megawatts. A plot of this data is made in Figure 3.10 below. From the curve, it is seen that there exists a linear relationship relating the feedwater flow and megawatt output. The computer labels on that curve are FWRE for feedwater flow and POWER for megawatts. The equation below
TABLE 3.4
CARDINAL POINTS OF BOILER FEED PUMP VARIABLES

<table>
<thead>
<tr>
<th>Discharge Pressure (psia)</th>
<th>Speed (rpm)</th>
<th>Feedwater Flow (K lb/hr.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>PBFP</td>
<td>RPM</td>
<td>FWRE</td>
</tr>
<tr>
<td>3860.</td>
<td>4600.</td>
<td>2590.</td>
</tr>
<tr>
<td>3920.</td>
<td>4700.</td>
<td>2835.</td>
</tr>
<tr>
<td>3995.</td>
<td>4800.</td>
<td>3085.</td>
</tr>
<tr>
<td>4065.</td>
<td>4900.</td>
<td>3330.</td>
</tr>
<tr>
<td>4135.</td>
<td>5000.</td>
<td>3540.</td>
</tr>
<tr>
<td>4215.</td>
<td>5200.</td>
<td>3780.</td>
</tr>
<tr>
<td>4280.</td>
<td>5300.</td>
<td>3980.</td>
</tr>
</tbody>
</table>

Fig. 3.10 Feedwater flow - MW
which relates these variables is

\[ \text{FWRE} = 8.3 \times \text{POWER} - 660 \]  

(3.3)

As mentioned earlier, the system head curves of the boiler feed pump which are obtained from manufacturing data included the three different variables, but it was not a useable curve that could be used in equation form; hence, an equation, or some other relationship had to be found which would enable the computation of the discharge pressure of the boiler feed pump for functions of the other two variables; in particular, the equivalent speed of the boiler feed pump in rpm and the boiler feed pump flow in thousands of pounds per hour. Note that it is the equivalent boiler feed pump speed that is considered since the two boiler feed pumps are considered to be actually one equivalent boiler feed pump in the simulation.

In the preliminary studies for the simulation, there were three different techniques which were used to represent the boiler feed pump characteristic curves. The three are specifically: (1) linear approximation which consisted of straight line segments connected between the cardinal points of the three variables in three space, (2) a least square approximation fit to the points using a parabola, (3) a multiple Lagrangian interpolation relationship.

The method which worked first, and which gave satisfactory results, was the linearized method in which the boiler feed pump characteristic curve was broken into six straight line segments over the operating range and the equation for a straight line was written for the individual regions between the cardinal points. Then for the values of the independent variables the particular straight line segment was obtained by the appropriate logic in the computer program and an equation of the line was then used to obtain the proper value of the feed-water flow. The multiple Lagrangian interpolation technique is included in Appendix A for the information of the reader. It could be easily connected into the computer program of the simulated plant by simply removing the portion of the boiler feed pump simulation that is presently in the program and inserting the Lagrangian interpolation program into this particular section.
Figure 3.11 illustrates the straight line segments which are used in the simulation for the boiler feed pump characteristic curve.

This figure shows a plot of the feedwater flow, FWRE versus PBFP, pressure output of the boiler feed pump which is equivalent to PO(1,15), with the third variable indicated on the graph being the speed of the equivalent boiler feed pump.

Consider the last straight line segment of the linearized boiler feed pump system head curve. The following computer variables are defined in the section following computer statement number 452 in block 9 of the computer program. (See the computer program listing in Table B-1.)

\[ A = \text{RPM}_2 - \text{RPM}_1 = 5300 - 5200 = 100 \text{ rpm} \]
\[ B = \text{FWRE}_2 - \text{FWRE}_1 = 3980 - 3780 = 200 \text{ Klb/hr} \]
\[ C = \text{PBFP}_2 - \text{PBFP}_1 = 4280 - 4215 = 65 \text{ psia} \]

The equation of a straight line in space in rectangular (Cartesian) coordinates in the two point form is given by,\(^{85}\)
Now let $X = \text{RPM}$ speed of the equivalent boiler feed pump (rpm)

$Y = \text{FWRE}$ feedwater flow ($\text{K-lb/hr}$)

$Z = \text{PBFP}$ boiler feed pump outlet pressure (psia)

Then

$A = X_2 - X_1$

$B = Y_2 - Y_1$

$C = Z_2 - Z_1$

Equation (3.4) becomes

\[
\frac{X - X_1}{A} = \frac{Y - Y_1}{B} = \frac{Z - Z_1}{C}
\]  

(3.5)

Solving the first and second terms for $Y$,

\[
Y = \frac{B}{A}(X - X_1) + Y_1
\]  

(3.6)

Solving the second and third terms for $Y$,

\[
Y = \frac{B}{C}(Z - Z_1) + Y_1
\]  

(3.7)

Adding equations (3.6) and (3.7) gives,

\[
2Y = \frac{B}{A}(X - X_1) + \frac{B}{C}(Z - Z_1) + 2Y_1
\]  

(3.8)

Solving for $Y$ yields

\[
Y = Y_1 - \frac{B}{2}\left(\frac{X_1}{A} + \frac{Z_1}{C}\right) + \frac{B}{2}\left(\frac{X}{A} + \frac{Z}{C}\right)
\]  

(3.9)

Substituting the computer variables yields, equation (3.10) which is used subsequently in the computer programs.

\[
\text{FWRE} = \text{FWRE1} - \frac{B}{2}\left(\frac{\text{RPM1}}{A} + \frac{\text{PBFP1}}{C}\right) + \frac{B}{2}\left(\frac{\text{RPM}}{A} + \frac{\text{PBFP}}{C}\right)
\]  

(3.10)
Continuing with the example case, equation (3.10) becomes

\[
FWRE = 3780 - \frac{200}{2} \left( \frac{5200 + 4215}{100 + 65} \right) + \frac{200}{2} \left( \frac{RPM + PBFP}{100 + 65} \right) 
\]

\[
FWRE = 3780 - \left( \frac{5200 + 4215}{100 + 65} \right) + \left( \frac{RPM + PBFP}{100 + 65} \right)
\] (3.11)

Now from Figure 3.11 it is seen that \(FWRE_1 = SP_1\), that is, the lower point of the 2 points in the line segment is called "set point 1", \(SP_1 = 3780\).

The distance, \(X\), above \(SP_1\), in terms of feedwater flow is thus

\[X = FWRE - SP_1\] (3.12)

In a similar manner set point 2, \(SP_2\), is equal to the lower point in terms of boiler feed pump pressure. From Figure 3.11 it is seen that,

\[SP_2 = 4215\]

Then the actual value of the boiler feed pump outlet pressure may be obtained from the following equation,

\[PBFP = 3500 + (SP_2 - 3500) + \frac{C}{B} (X) \text{ (psia)} \] (3.13)

The first term on the right hand side, 3500.0 psia, represents the output pressure of the throttle valve and is the bias by which the boiler feed pump system head curves are based. While this quantity is a dependent variable in the process equation and is in fact solved for, its variation in equation (3.13) resulted in values of feedwater flow in equation (3.11) that were not consistent with the operation and hence this bias was held constant at the value indicated which is steady state throttle pressure for all values of load.

Continuing with the example, consider full load conditions, in which \(X = 200\). Substituting in equation (3.13) yields,

\[PBFP = 3000 + (4215 - 3500) + \frac{65}{200} (200) \]

\[PBFP = 4215 + 65 = 4280 \text{ psia} \]

Now, since the boiler feed pump speed is adjusted by sensing an error between actual speed and desired speed to maintain the proper flow (as will be described
in the control section) it will be at 5300 rpm initially.

Hence equation 3.13 becomes

\[
FWRE = 3780 - \left( \frac{5200 + 4215 \cdot \frac{100}{65}}{5300 + 4280 \cdot \frac{100}{65}} \right)
\]

\[
FWRE = 3780 + 100 + \frac{100}{65} (4280 - 4215) = 3980 \text{ Klb/hr}
\]

This is the feedwater flow that corresponds to the stated conditions.

As mentioned earlier another approach to the boiler feed pump simulation is included in Appendix A.
3.3 FEEDWATER HEATERS

In order to make a more complete simulation, it was decided to include the effect of heating the feedwater before it is fed into the furnace. The simulated plant has both low pressure feedwater heaters and high pressure feedwater heaters. The terms low and high pressure refer to the pressure of the working fluid that is being heated. At this point in the cycle the working fluid is compressed water.

The feedwater heaters are of the closed cycle type. That is, there is steam extracted from various points in the heat cycle and is passed through tubes in the heater, over which the feedwater is flowing. The feedwater is thus heated by convective heat transfer from the higher temperature steam. The term closed cycle means that the condensate of the extracted steam used for heating is put back again into the main feedwater stream.

There are five stages of low pressure feedwater heating and two stages of high pressure feedwater heating where the heating steam is extracted from the turbines. Specifically, the low pressure feedwater heaters (in the order in which feedwater is heated) are the 18th stage heater, 17th, 16th, 14th, and 12th stage heater. The number of the stage refers to the turbine stage at which the steam is extracted. The high pressure feedwater heaters are heated by the 10th and 7th stage extraction flows.

The stages of the turbines refer to the various combination of rows of blades, fixed and movable, that form an integral part of the total turbine. They are numbered progressively from the inlet of the high pressure turbine down through the low pressure turbine. Stages 1 through 7 comprise the high pressure turbine, stages 8 through 12 comprise the intermediate pressure turbine, and stages 14 through 19 comprise the two low pressure turbines. Pictures of three of the feedwater heaters are shown in Figure 3.12.

The amount of extraction flow from each stage of the turbines is dependent on the existing pressure of the stage, the size of the orifice from which the extraction steam flows and the steam conditions in the extraction line. While these values do fluctuate somewhat during load changes, it is assumed for the
Fig. 3.12 Low pressure feedwater heater and high pressure feedwater heater
purpose of this simulation that the value of the extraction flow and the condition of the steam extracted is constant throughout the simulated tests. Furthermore, the two high pressure feedwater heaters are grouped into one equivalent high pressure feedwater heater with the steam conditions of the heater being a value that meets the heat balance requirements. This value will be computed subsequently. In Figure 1.2 this is shown as lump 14. In a similar manner, the five low pressure feedwater heaters are grouped into one equivalent low pressure feedwater heater. It is shown schematically in Figure 1.2 as lump 13.

Since the extraction flows WE1 and WE2 are considered constant, they are determined in the following way. In computer block 19, the value of the reheater steam flow is computed as in the following equation

\[ W(12) = \frac{3480}{3980} W(11) \]  \hspace{1cm} (3.14) \]

where

- \( W(11) \) = steam lead steam flow (lb/sec)
- \( W(12) \) = reheater steam flow (lb/sec), lump 12

The ratio in equation (3.14) is obtained from the steady state full load ratio of these two quantities. In the transient condition, this ratio will be altered nonlinearly to a change in density of the steam in these two lumps but is being neglected in this phase of the simulation.

The first extraction flow, that which is used for high pressure feedwater heating, is then determined from the equation

\[ WE1 = W(11) - W(12) \]  \hspace{1cm} (3.15) \]

where

- \( WE1 \) = high pressure feedwater extraction flow (lb/sec)

Since the intermediate and low pressure turbines are combined for this analysis, the average value of the flow through them is used to compute their combined power output, as is described in the turbine section. Hence, the average flow is taken as the steady state, full load ratio of \( W(11) \), steam lead
steam flow, as in the determination of the reheater steam flow.

\[
WC = \left(\frac{3.2}{3.98}\right) W(11) \tag{3.16}
\]

where

\( WC = \) average steam flow through intermediate and low pressure turbine

This ratio is estimated based on the value of input steam flow to the intermediate pressure turbine which equals 3,480,000 lb/hr and the input steam flow to the low pressure turbine of 2,940,000 lb/hr at full load. The value chosen was 3,200,000 lb/hr of equivalent steam flow through the combined turbine.

The second extraction flow is thus obtained from a knowledge of the reheater flow and the computed equivalent steam flow as in the following equation:

\[
WE_2 = W(12) - WC \tag{3.17}
\]

where

\( WE_2 = \) low pressure feedwater extraction flow (lb/sec)

A block diagram is shown in Figure 3.13 which illustrates the way in which the steam and condensate are handled in the feedwater heaters. The heat flux input to the high and low pressure feedwater heaters (lumps 13 and 14) is then computed. In order to compute the net heat flux that is utilized to heat the water, the input steam and output condensate conditions of the extracted flow must be known in addition to the previously computed extraction flows.

From the block diagram, Figure 3.13, it is noted that the condensate of the second extraction flow, \( WE_2 \), is readmitted to the stream just before the condensate pump. At this point the enthalpy from the heat balance is \( H_{019} = 69.1 \text{ btu/lb} \). This is also the same condition which exists for the feedwater into the low pressure feedwater heater. This value is assumed constant.

The conditions of the second extraction flow is assumed to be

\( HE_2 = 1272.2 \text{ btu/lb} \). Thus, the heat input to lump 13 becomes

\[
Q_{W(1,13)} = (WE_2) (HE_2) \text{ btu/sec} \tag{3.18}
\]
The heat output of the condensate of the extraction flow is given by the following equation

\[ Q_{H13} = (WE2) (HO19) \text{ btu/sec} \quad (3.19) \]

The difference between equations (3.18) and (3.19) gives the amount of heat added to the feedwater in the low pressure feedwater heater (lump 13)

\[ DQ_{13} = Q_{W(1,13)} - (WE2) (HO19) \text{ btu/sec} \]

The heat content of the feedwater entering lump 13 is given by equation

\[ Q_{I13} = W(12)HO19 \text{ btu/sec} \quad (3.20) \]

The heat content of the feedwater as it leaves the feedwater heater is then given by the following equation

\[ Q_{O13} = Q_{I13} + DQ_{13} \text{ btu/sec} \quad (3.21) \]

The condition of the feedwater as it leaves lump 13, the low pressure feedwater heater can then be computed as the heat content divided by the feedwater flow.

\[ HO_{13} = Q_{O13}/W(12) \quad (3.22) \]
where

\[ H_{013} = \text{the enthalpy of the feedwater leaving the low pressure feedwater heater} \]

With the assumption that there is no appreciable change in the enthalpy of the feedwater as it passes through the boiler feed pump, the input enthalpy of the high pressure feedwater heater is then given by the value just computed for \( H_{013} \).

\[ Q_{I14} = W(11)H_{013} \] \hspace{1cm} (3.23)

where

\[ Q_{I14} = \text{heat content of the feedwater entering lump 14} \]

The heat content of the condensate of the first extraction is also given by \( H_{013} \) since that is where the condensate is readmitted to the feedwater stream. This heat content is labeled \( Q_{H14} \).

\[ Q_{H14} = (WE1)H_{013} \] \hspace{1cm} (3.24)

The condition of the first extraction flow as it leaves the turbine is computed by heat balance to be equal to 1400 btu/lb.

Thus the heat input to lump 14 is given by the following equation:

\[ Q_{W(1,14)} = (WE1)(HE1) \, \text{btu/sec} \] \hspace{1cm} (3.25)

The amount of heat added to lump 14, \( DQ_{14} \), is the difference between equation (3.25) and (3.24).

\[ DQ_{14} = Q_{W(1,14)} - (WE1)(HO13) \] \hspace{1cm} (3.26)

The heat content, \( Q_{O14} \), of the feedwater that leaves the high pressure feedwater heater, is then given by the following equation:

\[ Q_{O14} = Q_{I14} + DQ_{14} \] \hspace{1cm} (3.27)

Finally, the enthalpy of the compressed water leaving lump 14, the high pressure feedwater heater, is labeled \( H_{014} \) and is the ratio of the heat content of the water to the flow.
\[ H_{O14} = \frac{Q_{O14}}{W(1)} \text{ btu/lb} \quad (3.28) \]

At steady state, full load, the heat balance gives a value of 487.5 btu/lb for the output enthalpy of lump 14 which is, as seen in Figure 1.2, the input enthalpy of lump 1, the economizer.

\[ H_{I(1,1)} = H_{O14} \quad (3.29) \]
3.4 TURBINES

As mentioned in the introduction, the simulated plant has a 3600 rpm tandem-compound turbine with an equivalent rating of 560 megawatts. On the single shaft are the high pressure, intermediate pressure and two low pressure turbines.

In Chapter 4 the heat cycle of the plant is discussed in detail and it will be seen from the temperature-entropy diagram that the steam expands through the turbines approximately isentropically. That is, there is very little increase in entropy in the steam conditions as it passes from entrance to exhaust of the turbine. Entropy is a property of steam that is useful in energy determination of heat engines. The change in entropy in moving from one state to another is defined as:

\[ S_2 - S_1 = \int_{1}^{2} \frac{d'Q}{T} \]  

(3.30)

where

\[ S = \text{entropy (btu/ib - }^\circ\text{F)} \]

\[ d'Q = \text{heat added or subtracted (btu)} \]

\[ T = \text{temperature of the state } ^\circ\text{F} \]

For a simple system with heat added at a temperature above the threshold level, the available energy for useful work is equal to the heat added minus the product of the entropy change and the absolute temperature of the threshold.

\[ Q \text{ available} = Q \text{ added} - T_T \left( S_2 - S_1 \right) \left( \frac{\text{btu}}{\text{lb}} \right) \]  

(3.31)

where

\[ T_T = \text{Threshold temperature} \]

This is shown in Figure 3.14 below

When the steam leaves the boiler it passes through the steam leads that connect the boiler with the turbine valves as described earlier. The steam leaves the valves and passes into a steam chest from which it passes into the
first stage of the turbine through specially designed nozzles. The steam has a particular state at entrance to the turbine and its state changes as it passes successively through the stages of the turbine.

In the specifications of the plant it was noted that the turbine configuration was a tandem compound, single reheat unit. This is illustrated in Figure 3.15.

The state property of the steam that is most useful for the simulation of the turbines is the enthalpy (btu/lb). Recall that the enthalpy of a substance is equal to \( h \).

\[
h = E + PV \left( \frac{\text{btu}}{\text{lb}} \right)
\]  

(3.32)

where

\( E = \text{intrinsic energy of the system} \left( \frac{\text{btu}}{\text{lb}} \right) \)
Thus the steam has this equivalent energy as it crosses a boundary in addition to other forms of energy, specifically kinetic and potential energy.

The steam expands in the nozzles down to a state at exhaust pressure. The steam leaves the nozzles at a high velocity and is directed onto the blades which are mounted on the shaft. The velocity of the steam is higher than the blade velocity and hence the steam exerts a force on the blade and work is done on the blade. This work is transmitted through the wheel and into the shaft and then to the rotating d.c. field of the synchronous generator where electromagnetic restraining torque is produced by virtue of the interaction of the direct current (d.c.) produced magnetic field and the armature produced magnetic field that is a function of the electrical load on the generator.

In an ideal nozzle the steam expands isentropically. All the available energy (enthalpy) of the steam is converted to kinetic energy. Actually there is turbulence and friction which cause a gain in entropy such that the enthalpy of the steam is higher than the ideal value. This is illustrated in Figure 3.16 with the quantity known as "nozzle reheat" indicated.

![Figure 3.16 H-S diagram for nozzle process](image-url)
A similar phenomena occurs in the various stages of the turbines and is called blade reheat. Since all the blade passages are not at all times filled with steam, there is a "fanning" loss. Also there is friction since steam surrounds the rotating wheel. These are the so-called rotational losses. In addition there is steam leakage through the seals which causes a throttling effect as described in the throttle valve section. These combined effects are called stage reheat.

The several stages are then combined to yield the total "reheat" of the turbine. Note that it was assumed that the enthalpy was converted to kinetic energy and that the steam had no kinetic energy at entrance or exit from the turbine.

The effect is shown in Figure 3.17

![H-S diagram for turbine stage](image)

Fig. 3.17 H-S diagram for turbine stage

It is the reheat phenomena that causes the turbine stage efficiency to be less than 100%.

The multi-stage turbine such as the one that is under consideration in this study accumulates this reheat effect for each stage. In this simulation the individual stages of the turbine are not studied, however, the cumulative effect for the high pressure turbine and the combined intermediate and low pressure turbine is shown on the H-S diagram in Figure 3.18.
The exact values of enthalpy that are given in Figure 3.18 are obtained from the simulation itself. Initial values were used from the operation data obtained in test run 2 which is described in a later section. The values of enthalpy were obtained from the steam table corresponding to pressure and temperature values of the required spatial points. The turbine outlet conditions are assumed to remain constantly at the values obtained while the input conditions vary as the process moves from one equilibrium point to another. This will occur when there is a power demand change or individual parameters are varied. Hence for a given value of flow, computed from the boiler feed pump, taking into consideration the density change that occurs in the transient solution, the value of power output of the turbines can be computed.

Since the system must operate in equilibrium and accurate values of the turbine reheat factor and engine efficiency are not known, they are estimated in this simulation. The term reheat factor is the ratio of the sum of the isentropic enthalpy drops for the turbine stages (high pressure and combined intermediate and low pressure turbine) to the isentropic enthalpy drop for the turbine. This is shown in Figure 3.19.
The reheat factor is assumed to be 1.08 for both turbines. The engine efficiency of the high pressure turbine is assumed to be 90% ($\eta_1 = 0.90$).

With the same reheat factor for the low and intermediate pressure turbine, its efficiency is computed to be 94.1% ($\eta_2 = 0.941$) in order that the system balance at 560 mw, full load.

First consider the high pressure turbine.

The work per pound of steam is given by the following equation.

$$ W_1 = \text{(reheat factor)} \times (\Delta H)\eta_1 $$

(3.33)

where

$\Delta H = \text{the difference between input and output enthalpy.}$

In terms of computer variables this is equation (3.34)

$$ W_1 = (RHF) \times (HO(1,11) - HO(1,17))\eta_1 $$

(3.34)

where

$$ HO (1,17) = 1276 \left(\frac{\text{btu}}{\text{lb}}\right) $$

The output power of the turbine in hp is
$HP1 = (W_1) \frac{(W(11)3600)}{2545}$ \hspace{1cm} (3.35)

where

$W(11) = \text{turbine flow (lb/sec)}; 2545 \text{btu/lb} = 1 \text{ hp}$

The equivalent output of the turbine in megawatts is given by equation (3.36). \hspace{1cm} (746 \text{ watts/hp})

$\text{POWER1} = \frac{(RHF) \eta_1 \frac{746}{(HO(1,11) - HO(1,17)) W(11)3600}}{(2545)10^6}$ \hspace{1cm} (3.36)

Combining the constants together yield

$K1 = \frac{RHF(\eta_1)746}{2545} = \frac{(1.08)(0.9)746}{2545} = 0.285$

Hence equation (3.36) comes

$\text{POWER1} = 0.285 (HO(1,11) - 1276.) W(11) 3.6 \times 10^{-3} \text{ MW}$ \hspace{1cm} (3.37)

In a similar way the power equation is obtained for the combined intermediate and low pressure turbines.

The work per pound of steam in the intermediate and low pressure turbine (turbine 2) is given by equation (3.38)

$W2 = (RHF2) (\Delta H2) (\eta_2) (\text{btu/lb})$ \hspace{1cm} (3.38)

where

$RHF2 = \text{reheat factor of turbine 2}$

$\Delta H2 = \text{enthalpy drop across the turbine}$

$\eta_2 = \text{engine efficiency}$

In terms of the process variables, with $HO(1,18)$, the turbine outlet enthalpy, equal to 1113, btu/lb, $RHF = 1.08$ and $\eta_2 = 0.941$ as described earlier. Hence,

$W2 = ((RHF2) \eta_2 (HO(1,12) - HO(1,18))) \text{ btu/lb}$ \hspace{1cm} (3.39)
The output power of turbine 2 in hp is determined from equation (3.40). The value of the equivalent flow is \( (3.2/3.98) \, W(11) \) as is discussed in the section on feedwater heaters. This equivalent turbine flow results from the extraction flow.

\[
HP_2 = \frac{W_2 (W(11)3.2/3.98)3600}{2545} \, \text{hp} \tag{3.40}
\]

The equivalent output of the turbine in megawatts is given by equation (3.41)

\[
POWER_2 = \frac{(RHF_2) \eta_2 746 (HO(1,12) - HO(1,18))3.2 \, W(11)3600}{(2545)(3.98)(10^6)} \, \text{MW} \tag{3.41}
\]

Combining the constants together yield

\[
K_2 = \frac{(RHF_2) \eta_2 746 (3.2)}{2545(3.98)} = \frac{(1.08)(0.941)(746)(3.2)}{2545(3.98)} = 0.24
\]

Equation (3.41) thus becomes

\[
POWER_2 = 0.24(HO(1,12) - 1113.)W(11)3.6 \times 10^{-3} \, \text{MW} \tag{3.42}
\]

The total power output of the turbines is then the sum of the power from turbine 1 and turbine 2.

\[
POWER = POWER_1 + POWER_2 \, \text{MW} \tag{3.43}
\]

where

\[
POWER = \text{total simulated generated power in megawatts.}
\]

The turbine control cabinet and high pressure turbine is shown in Figure 3.20. The low pressure turbine and generator are shown in Figure 3.21.
Fig. 3.20. Turbine Control and High Pressure Turbine.

Fig. 3.21. Low Pressure Turbine and Generator.
3.5 TURBINE VALVES

The turbine valves act in a way to throttle the steam as it enters the high pressure turbine. The throttling process occurs when a gas, in this case steam, flows through an orifice. The throttling process simply obstructs the fluid flow. This is an entirely dissipating process which is somewhat like applying brakes on an automobile. An example of a throttle is a partly opened valve in a fluid or gas flow line, such as one would find in a faucet. When the gas is at a temperature approximately equal to the surroundings or, as is the case in this simulation, when the passage is well insulated from the surroundings, the heat transfer may be neglected. In this case, the initial and the final enthalpies, as seen at the inlet and outlet of the valve, are approximately equal. That is, the energy balance equation indicates that \( \Delta H = 0 \). Even though this is the case, there is still a significant change in the properties of the working fluid as it passes through the valve. In particular, there is a pressure drop across the valve. Referring to the general energy equation (3.1). The initial and final velocities equal (that is \( V_1 = V_2 \)) and with the elevation term neglected (i.e., potential energy is considered negligible) and with no net heat transfer or work done on the fluid in the valve, the following equation results:

\[
\frac{u_2 + \frac{P_2 v_2}{778}}{778} = \frac{P_1 v_1}{778} + u_1
\]

but these are equivalent to the enthalpies, \( H_1 \) and \( H_2 \), and, as mentioned, the enthalpy from input to output remains unchanged.

The final conditions of the steam as it passes through the throttle valve are obtained easily from the temperature–entropy diagram.

The temperature, \( T_3 \), shown in Figure 3.22 represents the condenser temperature, the recovery temperature, of the heat cycle. It is noted that due to the throttling process the cross hatch area which represents work on the T-S diagram is actually lost due to the throttling process. Hence, no matter what is done in the design there is no way of making the steam in the throttled
condition, (that is, point 2 on the diagram) even for an ideal steam engine, yield the same amount of work as would have been obtained had the steam been delivered at the conditions of point 1, the input to the throttle valve. There will always be an amount of work or energy lost corresponding to the cross hatched area. This may also be illustrated by the use of the H-S diagram as is shown in Fig. 3.23.
In Figure 3.23, above, is shown the throttling effect on the H-S diagram. If the input to the throttle valve is considered as point 1, the output of the throttle valve, point 2, and the expansion from the constant pressure, \( P_2 \) to \( P_3 \) is being made, it is seen that the available energy is equivalent to \( H_{32} \) on the diagram. This is shown, again, in the section on the turbines relating to the power output of the turbines. However, if throttling had not taken place then by expanding from a constant pressure, \( P_1 \), to the constant pressure, \( P_3 \), the process would have had a change of enthalpy corresponding to \( H_{41} \), and consequently a greater output power, or an equivalent amount of work which is done is greater than that which could have been obtained using the throttle valve for controlling the flow of steam through the turbine. The basic purpose of using the throttle valve for a controlling device is to control the steam turbine's speed and this is done as indicated by controlling the amount of steam that is allowed into the turbine. This flow is varied by varying the area of the passage through which the steam must flow in order to get to the turbine. The force which is developed in the turbine must be exactly equal to the force which is required to drive the synchronous generator at synchronous speed, which in this case is 3,600 rpm. The so-called resistance to the flow of steam, which is given by the valve, decreases the rate at which the steam flows into the turbine and, hence, is brought to a lower initial pressure as seen by the previous figures. This is a very convenient and effective method for obtaining the speed control which has the obvious disadvantage that the energy which is obtained is smaller as a result of this throttling process, as mentioned in relation to Figures 3.22 and 3.23.

The type of turbine that is used in the simulated plant is an impulse turbine which has a limited number of nozzles. The speed control is obtained by the automatic adjustment of valves ahead of the nozzles. As the valves are closed there is a decrease in the weight of the fluid which enters the turbine which tends to reduce the force exerted and reduces the inherent losses in the throttling process. The device which automatically opens and closes the valves is
called a governor and, in actuality, it opens and closes several steam admission valves in sequence. The system uses lift rods, or cams, to open the individual nozzle valves. Since there are four valves in the simulated plant, there are four valve position steps which are carried out. For the lowest loads one valve is opened, and as the loads are increased another valve, and then another valve is opened in sequence. The valve characteristics which relate the power output to the steam flow are not exactly linear, but are considered so when considered as valve steps.

The turbine stop valves and throttle valves and linkages are shown in Figure 3.24 and Figure 3.25.

The governors which control the steam turbines may be operated either manually or automatically. The governors have a drooping speed characteristic which is often set around 5% droop compared with the no load speed. A speed load characteristic is shown in Figure 3.26. The normal operating point for this unit, when operating at a full load of 560 megawatts, is 3,600 rpm, and since the synchronous generator has two poles this corresponds to a frequency of 60 hertz. Since the generator of the simulated plant is connected to many other similar generators in parallel to comprise the electric power system, and all machines on the system must operate at the same frequency, the speed must be regulated by the governor to maintain this particular speed. The characteristic could be shifted either up and down or right and left by virtue of the governor action. First consider a shift up from position A to position B. This would give rise to an increased load taken by the machine, since the intersection with the synchronous speed of 3,600 rpm and the characteristic B would be an increase load to $P_B$. If the governors had reduced the steam going into the turbine, the machine would tend to slow down, but could not slow down since it was tied synchronously to the other machines of the system, and as the characteristic moved to the C characteristic the power output of the simulated plant would drop from 560 megawatts to $P_C$ megawatts.

As has been mentioned, the valve simply varies the amount of restriction which it introduces into the line supplying the steam to the turbine. The
Fig. 3.24. Turbine Stop Valves and Throttle Valves.

Fig. 3.25. Throttle Valve Linkage.
restriction of the flow can be defined in terms of groups of relationships between the various flows, pressures, and the valve position itself. If the pressure were plotted as the abscissa with the flow as the ordinant, and then different curves were plotted with the parameter of variation being the valve position, plots would be obtained which were analogous to the families of plate characteristics of vacuum tubes. The simulated valve approximates an ideal orifice which has a linear relationship between the flow rate and the area of opening if the pressure drop across the valve is held at a constant value. However, the relation between the flow and the pressure is parabolic if the area of the valve is held constant.

The approximating equation that is used for the simulation is equation (3.45). (See p. 184, ref. 89)

\[ W = K_1(a)\sqrt{2\Delta p/\rho} \]  

(3.45)

where

\[
\begin{align*}
W &= \text{flow of working fluid (lb/sec)} \\
(a) &= \text{area of valve opening (ft}^2) \\
K_1 &= \text{discharge coefficient} \\
\Delta P &= \text{Pressure drop across the valve (psia)}
\end{align*}
\]
\( \rho = \text{density (lb/ft}^3) \)

In terms of the computer variables, the flow used is the flow out of lump 11, the steam leads, \( W(11) \). The pressure drop is given by,

\[ \Delta P = P_O(1,11) - P_O(1,16) \]

where it is noted that the throttle valve is lump 16 in Figure 1.2.

The density used is that of lump 11.

There are four steam admission valves. One has an inside diameter of 8" while the other three have inside diameters of 7" each. The area of the valve opening is considered as a composite of these valves and is computed using a product of the throttle valve position and the total area. The value of the throttle valve position is obtained from the control system and will be described in detail there, however, it should be noted here that for full load the throttle valve position, labeled TVPXR, equals 1.0 and will be less than 1.0 for loads less than 560 MW.

The valve area is given by

\[ AA = TVPXR \left\{ \pi \left( \frac{8}{12} \right)^2 + 3 \left( \frac{7}{12} \right)^2 \right\} \text{ ft.}^2 \quad (3.46) \]

The value of the discharge constant, \( K \), in equation (3.45) is considered constant for all simulated loads and is obtained from data at steady state full load. At full load, \( \Delta P = 100 \text{ psia}, \ W(11) = 1105 \text{ lb/sec.}, \ RHO(11) = 5 \text{ lb/ft}^3 \).

Substituting these values into equation (3.45) yield

\[ W(11)^2 = K_1^2 (AA)^2 2 \left[ P_O(1,11) - P_O(1,16) \right] RHO(11) \quad (3.47) \]

Solving for \( 2K_1^2 = KV \),

\[ KV = \frac{W(11)^2}{(AA)^2 \left[ P_O(1,11) - P_O(1,16) \right] RHO(11)} \quad (3.48) \]

\[ KV = \frac{(1105)^2}{(1.15)^2 \left( \frac{5}{100} \right)} = 1846.541 \]
In the computer simulation the output pressure of the valve is desired, hence solving equation (3.47) for \( P_{O(1,16)} \),

\[
P_{O(1,16)} = P_{O(1,11)} - \frac{W(11)^2}{(1846.541)RHO(11)(AA)^2}
\]  

(3.49)

This equation is used in block 19 of the computer program.
3.6 SPRAY VALVES

The simulated plant has a spray flow valve that admits water into the steam in order to obtain a quick, but non-permanent change in the output temperature of the superheater. The amount of spray flow is determined by the deviation of the final superheater temperature from a predetermined set point as is discussed in the control section. There is a signal sent from the control system to the process indicating an increase or decrease of spray flow that is required for the temperature difference. In the plant operation the spray flow reacted somewhat sporadically as is seen in the discussion of the test runs. For the simulation, the maximum value of spray flow considered was 100,000 lb/hr which is equivalent to 27.8 lb/sec of flow. This flow is a very small percentage of the full load steam flow of 1105 lb/sec and offers a negligible increase in flow and mass of the superheater steam flow.

The value of spray flow in lb/hr is taken as the value of the spray valve position times 1000, where the spray valve position varies from 0 to 100.

The spray flow is obtained by extracting water from the feedwater flow at the entrance to the economizer and then spraying the water into the superheated steam between the primary and finishing superheater, that is between lump 9 and lump 10.

The heat content, $Q_{10}$, of the working fluid as it moves into lump 10, without the spray flow is given by the following equation.

$$Q_{10} = W(9) H(1, 10) \text{ btu/sec} \quad (3.50)$$

where

$W(9)$ is the steam flow output of lump 9. (lb/sec)

$H(1, 10)$ equals $H(1, 9)$ in this case. However, if spray flow is injected into the superheated steam at the output of the primary superheater, the enthalpy of the steam is changed since the spray water vaporizes upon contact with the high temperature steam.
The heat absorbed increases the temperature of the spray water and by the vaporization changes the internal energy of the steam which is represented by a drop in enthalpy of the working fluid as it enters lump 10.

Since the spray flow, WS, (lb/sec) is given for a particular operating condition, the heat extracted from the working fluid is to be determined.

Since the fluid of the spray is at supercritical pressure, that is above 3206.2 psia, the value of the heat of vaporization is negligible, and thus the only contributing factor to a decrease in enthalpy of the main stream flow is proportional to the amount of heat required to raise the temperature of the spray water from its value of approximately 582° to TO1(1, 9).

The value of HI(1,1) is computed from the feedwater heater section and is available after the first iteration. The value of PI(1,1) is not however available but may be calculated from a knowledge of the steady state full load pressure drop across lump 1 which is 25 psia. The value of the pressure PI(1,1) is given by equation (3.51)

\[ PI(1,1) = PO(1,1) - \frac{W(1)(25)}{1105} \text{ psia} \] (3.51)

Then TI(1,1) is computed by use of the computer subroutine TSSPH that is described later. Basically, \( T = f(p, H) \) and when the subroutine is called, it returns with the temperature corresponding to the \( p \) and \( H \) supplied to it.

The specific heat of a substance is numerically equal to the number of btu required to raise the temperature of 1 pound of substance 1°F.

The specific heat of water is \( \frac{1 \text{ btu}}{\text{lb} \cdot ^\circ F} \) at 1 atm. but at the approximate operating temperature and pressure of the spray water, 4000 psi, 600°F, the specific heat is \( 1.29 \text{ btu/} \text{lb} \cdot ^\circ F \). 50

The specific heat, \( C_p \), of the steam at its approximate operating values (800°F, 3700 psia) output of lump 9 is \( \frac{1.6 \text{ btu}}{\text{lb} \cdot ^\circ F} \). 50

For the simulation, \( C_p = 1.44 \frac{\text{btu}}{\text{lb} \cdot ^\circ F} \) was chosen and used in the following equations to represent the heat absorbed by the water when spray was added.
\[ Q_{O9} = W(9) \cdot H(1,9) \frac{\text{btu}}{\text{sec}} \]  

(3.52)

where

\( Q_{O9} = \text{heat content of working fluid leaving lump 9} \)

The heat absorbed by the spray water is:

\[ D_{QS} = (WS) (1.44) (T_O(1,9) - T_I(1,1)) \]  

(3.53)

Then the heat content of the steam entering lump 10 is given by equation (3.54)

\[ Q_{I10} = Q_{O9} - D_{QS} \]  

(3.54)

The enthalpy of the steam entering lump 10 is then,

\[ H_I(1,10) = Q_{I10}/W(9) \]  

(3.55)

Thus the enthalpy is adjusted by the spray flow to give the corrective action needed to force the finishing superheater outlet temperature to the set-point desired.
CHAPTER 4

PLANT HEAT CYCLE AND THE PROPERTIES OF THE WORKING FLUID

The properties which are needed to study the problems of thermodynamics are temperature, pressure, volume, internal energy, enthalpy, and entropy. If the portion of internal energy due to gravity, observable motion, surface tension, electricity and magnetism are separately accounted for, a pure substance has, in general, only two independent properties. For example, if the pressure and specific volume are fixed then all other properties become fixed.

Thus we can express any property of a substance as a function of two other properties and lines may be plotted of a constant value for any one property on coordinates of two other properties. The equations which so represent the state of the system are appropriately called "The State Equations of the System".

For the purposes of the present simulation, the independent variables which have been chosen are pressure and enthalpy. As has been indicated earlier and will be seen further in the discussion of the computer program, frequently it is the temperature of a particular point in the heat cycle which is desired. It is obtained by knowing what the current values of the pressure and enthalpy are at that particular point in the cycle at that particular time.

Before discussing the details associated with the methods of the computation of the state properties, it is appropriate to discuss what the concept of the heat cycle does in terms of the plant operation and to describe here the detailed heat cycle which is involved for the once-through supercritical steam generator.
There are three different graphs which are used to illustrate what the heat cycle is like. The graphs are, the temperature-entropy graph, the temperature-enthalpy graph, and the enthalpy-entropy graph. The last graph, that is the enthalpy-entropy graph, is commonly called the Mollier diagram. By inspection of these graphs, a rather quick estimation of the type of cycle that is used in a particular plant may be judged and may be compared with other familiar cycles. Every point on one of these diagrams determines the conditions of the working fluid which in this case is either steam or water; that is, for example, on the Mollier, the H-S diagram, for every pair of H and S there is a corresponding definite pair of values such as P and V which can be read directly from the chart.

As mentioned in an earlier section, the area beneath the curve of a closed cycle on the T-S diagram represented a particular amount of heat that was put into the particular process and represented, therefore, a form of energy. However, note that in the H-S diagram, as was mentioned in relation to the explanation of the turbines, there is a distinct advantage that the heat quantities are represented by simply measuring the distances between two respective enthalpies, and these may be measured very easily with a ruler. Consider the rough block diagram representing the fluid flow path of the unit, as shown in Figure 4.1. Starting with the output of the boiler feed pump, point a, passing through the high pressure feedwater heater, to the furnace, to the superheater, through the turbine, (the output of the turbine marked at point e), through the reheater, through the turbine and back through the condenser to the low pressure feedwater heater. This is the closed cycle that is used in the simulated plant.

The cycle steps may be traced through in a sequential fashion and they are shown schematically in Figures 4.2 and 4.3, where Figure 4.2 is a representation of the heat cycle on the T-S diagram and Figure 4.3 is a representation of the heat cycle on the H-S diagram, or Mollier chart. The various steps in the process are indicated, as mentioned, with the letters between the various blocks.
During the passage of the fluid from d to e through the high pressure turbine stage, the steam decreases in temperature, pressure and enthalpy, and increases in volume and there is a slight increase in the entropy, although generally this expansion has been considered to be isentropic; that is, with a constant entropy. As noticed from the figure, the exhaust steam has a slightly superheated value. As the fluid passes between e and f, it is reheated at an approximate constant pressure in the reheater. From point f to point g there is again the expansion of the steam through the intermediate and low pressure turbines of the simulated plant, where again the exhaust steam to the condenser is decreased in pressure, temperature and enthalpy and increased in the volume and entropy. Then from this point in the cycle, the steam passes to the condenser from g to h where there is an isobaric (that is, a constant pressure) lowering of the temperature, such that the condensation occurs. Between points h and i, the feedwater is heated at approximately a constant pressure and the temperature is raised from the condenser temperature to a higher temperature illustrated in the T-S diagram as point i. The amount of reheating feedwater heating is a function of the plant design.
Fig. 4.2 Heat cycle T-S diagram

Fig. 4.3 Heat cycle H-S diagram
Between i and a the liquid is passed through a feedwater pump in which there is a significant increase in the pressure of the liquid to the boiler pressure, which in this case is approximately 4,200 psia. The pressure is obtained by putting mechanical energy into the feed pump. Besides the increase in the pressure at this point, there is only a very slight change in the enthalpy, entropy, temperature and volume and, hence, the state of the fluid at i and a are almost coincidental on the two diagrams. Between points a and b, the high pressure feedwater is again heated in much the same fashion as the low pressure feedwater was heated; that is, by extracted steam from the high pressure turbine and the intermediate and low pressure turbines. The value of the pressure changes only slightly in the feedwater heating while the temperature is raised significantly, as indicated on the T-S diagram. At point b, the fluid enters the economizer portion of the furnace and is then heated by the convection heating and, as it moves up to point c, which is the input to the superheater, with this supercritical cycle, it is always passing above the saturation line, as indicated in both the T-S and the H-S diagrams.

In a conventional unit the boiling would occur along a constant temperature line, but the details of that will not be given further attention in this consideration. With the supercritical unit, as has been mentioned earlier, there is no exact division point between the steam and the water. There is essentially a composite mixture and a very non-distinct point at which the water changes into steam as it is heated. Point c occurs at approximately 660°F. Between point c and point d, the steam is then superheated at an approximate constant pressure and is then ready at point d to enter the high pressure turbine and to pass through the cycle again as described.

With the type of turbines where extraction steam heats the feedwater, for each pound of steam which enters the turbine not all of the steam goes through the condenser. This is due to the fact that some of the steam is used in the feedwater heaters. This tends to raise the efficiency of the unit as is indicated by the cross-hatched area below line g-x on Figure 4.2 which indicates the heat rejected by the condenser. The net heat into the steam is represented by
the area on the T-S diagram under the curve between b and d and e and f. Again there is some of the area there which is not cross-hatched due to the fact that not all of the steam passes through from the high pressure to the low pressure turbine due to the extraction flow used for the feedwater heater. As mentioned earlier the reheating effect makes the performance of the unit better which justifies the expense of the additional piping and the reheater in the steam generator, as does the feedwater heating units. After considering the T-S and H-S diagrams in a general way, look now at Figures 4.4 and 4.5 which are the T-S diagram and H-S diagram respectively for the units in which the various points are plotted as accurately as could be obtained from the data for the simulated plant.

In the previous figures, the heat cycle of the once-through supercritical steam generator is illustrated. In terms of the properties of the steam the simulation must consider where these points of the heat cycle move under the various conditions that are imposed upon the model in order to simulate the actual plant. This would entail either knowing exactly where on the heat cycle each particular point in question is located or of being able to compute the representative points by virtue of excursions which are taken from the steady state operating point.

In order to find the properties of the water or steam in the process, there are basically two approaches which may be taken. The first approach would be that of tabulating within the computer simulation itself the values of the various properties needed to solve the equations of the process. That would require a tabulation of all the values wherever the heat cycle may move to on the T-S or H-S diagram. The traditional way of handling the state equations as has been indicated in most of the publications has been by considering only a linearized model of the plant and allowing the dependent variables to take small excursions from fixed operating points. Then by assuming that the properties of the working fluid do not change rapidly (which is an incorrect assumption in the critical region) with small changes in independent variables only the steady state values of the properties needed to be stored for fairly accurate results.
Fig. 4.4. Heat Cycle; Simulated Plant, T-S Diagram
Fig. 4.5: Heat Cycle, Simulated Plant, H-S Diagram
It was decided that a better approach to the problem would be to try to allow for a very large variation in the variables. This meant since, as seen from the previous figure, the heat cycle of the plant covered most of the regions of the temperature-entropy diagram, that almost all of the steam tables would have to be represented on the digital computer. These steam tables could be represented on the digital computer by either tabulation or by representing the steam table by an appropriate equation.

The regions of the steam table which must be uniquely defined are the superheated steam, compressed liquid, wet vapor and the so-called critical region of superheated steam. Note that the heat cycle of the supercritical boiler passes directly through the critical region. This region around the critical point, 705.47°F, 3,208.23 psia, has rapidly changing properties and, at the time that the preliminary research was done, the equations were not available to represent points with specific volumes less than 0.1603 cu. ft./lb. So, in the earlier research which was done in this area, that particular region was tabulated. In the superheated steam region, equations described in a paper by H. C. Schnackel\(^7\) were utilized. Note from the heat cycle figures that this portion covers most of the heat cycle.

The rest of the heat cycle that needed to be simulated was in the compressed liquid region, which according to Schnackel would also have to be tabulated. Hence, in the earlier phases of the research a complete tabulation of these regions was made on the digital computer with provisions made for interpolating where necessary. However, the recently published book by McClintock and Silvestri\(^8\) has described the formulations and iterative procedures for the calculation of steam in the critical region, and also in the compressed liquid regions. Hence, the procedures were adapted to fit the existing program and the subroutine was developed using the equations described in the aforementioned book and is given in Appendix B attached to the main computer program. The subroutine is TSSPH. It is to be noted that the entry into the subroutine is made with pressure and enthalpy as the independent variables.
In addition to the computation of temperature as a function of pressure and enthalpy, it was also desired in the simulation to consider the effect of the changing density of the steam in lumps 9, 10, and 11, which are the superheater and steam lead sections of the simulation. The desire for this change in density consideration was prompted by the desire to include the dynamics associated with the continuity equation to make the simulation more complete. Within the compressed liquid region, that is, before the water gets to the critical point, it is assumed that the density change is negligible, but starting in lump 9, in the superheater area, the density change is then considered. With this transition point being used, the equations for the specific volume as a function of pressure and temperature are computed. The subroutine that is used to compute the specific volume is also given at the end of the main computer program in Appendix B. It is based on the paper listed in the bibliography by Steltz and Silvestri. The specific volume is given in that subroutine with the computer variable \( VNU = (VV1 + VV2 + VV3 + VV4 - VV5) \) where

\[
\begin{align*}
VV1 &= R/PP(TAU) \\
VV2 &= B0 \\
VV3 &= (B0)^2 (TAU)^2 (DELTA) PP \\
VV4 &= EPSILON (B0)^4 [TAU(PP)]^3 \\
VV5 &= RH02 (B0)^13 (TAU(PP)^{12})
\end{align*}
\]

The other variables mentioned in the equation above are defined in the computer program in the appendix.

The programs themselves as were used were checked to see if the accuracy was within a tolerable amount for the simulation. Table 4.1 shows the various points tested and their corresponding enthalpies and entropies as were obtained from the steam tables for the corresponding points shown in the earlier figures of the heat cycles.

Using the program TPHOM, Table 4.2 illustrates the percentage error of the calculated temperature as compared with the actual temperature from the
### TABLE 4.1

TEST POINTS OF HEAT CYCLE

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<th>ENTROPY (BTU/LB)</th>
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<td>TEMP ACT (DEG F)</td>
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steam tables, and the error is tolerable, generally less than 5% and in many cases closer. It should be recognized that for many of the points in the heat cycle where interpolation had to be made it was difficult to obtain an actual temperature which was completely accurate. In Table 4.2 the heat cycle is made into four sections which are illustrated in the figure representing the heat cycle. Also, in Table 4.3 a similar analysis is made illustrating the relative accuracy of the specific volume computations.

By use of these equations, which represent the steam tables, at each point within the iteration in the computer program, the values of the temperatures corresponding to pressures and enthalpies; and specific volumes corresponding to pressures and temperatures are computed in the process at each time interval, which then adds to the accuracy of the simulation of the plant.
TABLE 4.3
TEST POINTS OF HEAT CYCLE WITH SPVOL3

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CHAPTER 5

THE PROPERTIES OF THE COMBUSTION GAS
AND ITS RELATION TO THE SIMULATION

There are several types of fuel available for the production of heat which is in turn transferred to the working fluid. In the United States the primary sources of energy for steam-electric generation are fuel oil, natural gas, and coal. In addition to these sources, a fourth energy source, nuclear fuel, is also becoming increasingly more popular as it is economically competitive in more applications and has the esthetic advantage that there is no associated air pollution.

The primary fuel for the simulated boiler is natural gas with fuel oil serving as the secondary fuel. Fuel oil is used only in emergency or during periods of cold weather when the consumer demand for natural gas is at its peak and the electric utility can not be served adequately.

The burners that are used in the plant under consideration are combination type burners and do not have to be changed when the fuel is changed from natural gas to oil. The plant operator has a selector switch on the control console that allows him to switch from one fuel to another in each pair of burners very quickly and, in fact, both fuels (natural gas and oil) may be burned in the furnace simultaneously (using different pairs of burners of course). The controller and control valve for the natural gas entering the plant are shown in Figure 5.1. A row of burners for the fuel is shown in Figure 5.2.

Since natural gas is used the majority of the time, it is assumed in this study that only natural gas will be used and there is no provision in the simulation for other fuels to be burned. If oil were to be considered, it would be handled in a manner similar to natural gas.

The natural gas that is supplied to the simulated plant is of the same composition as that which flows through New Orleans gate number 4.
Fig. 5.1 Controller and control valve for natural gas to plant

Fig. 5.2 Row of fuel burners
This fuel was analyzed by a laboratory and the resulting analysis is given below and in Table 5.1.

The gas sample was from meter 1-06-01-042, which had a line pressure of 225 psig and a line temperature of 58°F.

The heat value of the fuel, natural gas, was determined by a calorimeter reading performed in May, 1969. The result was (for 60°F, dry condition at 14.9 psia) \( \frac{\text{BTU}}{\text{ft}^3} \) 1071. which is used in the subsequent work to determine the amount of heat liberated by the combustion process.

In addition to the fuel constituents listed in Table 5.1, the natural gas contained 0.04 grains/100 cu. ft. of sulfur; a calculated value of 0.01 g.p.m. of NGPA charcoal; and had a specific gravity, referenced to air, of 0.5887 when the conditions were 60°F, 14.735 psia. Gas analyses from one specific gas field do not remain constant, but vary slowly with time.

The specific heat of the combustion gas, \( C_{pg} \), is required in order to determine the heat leaving the furnace. The value of the specific heat is a function of the gas composition and its operating temperature. The method by which the specific heat of the combustion gas is obtained is presented below.

The fuel composition is recorded in columns 1, 2, and 3 of Table 5.1 as mentioned previously.

The combustion constants are obtained for 100% total air in units of mole per mole combustible. These constants are recorded in columns 4 and 5 of Table 5.1.

The amount of the oxygen and dry air required for the complete combustion of the various constituent gases of the natural gas is then computed and given in columns 6 and 7 of Table 5.1 in units of moles per 100 moles of fuel. For example, consider the constituent methane, \( \text{CH}_4 \), since there are 95.23 moles of \( \text{CH}_4 \) per 100 moles of fuel the \( \text{O}_2 \) required is \( 2.0(95.23) = 190.46 \) moles. The dry air required is \( 9.53(95.23) = 906 \) moles per 100 moles of fuel.
### TABLE 5.1

**FUEL COMPOSITION AND COMBUSTION REQUIREMENTS**

<table>
<thead>
<tr>
<th>Fuel Constituent</th>
<th>Chem. Formula</th>
<th>Moles</th>
<th>Comb. Constants</th>
<th>Gases Req'd. For Combustion</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>O2 Air</td>
<td>O2 Air, Dry Moles/100 Moles Fuel</td>
</tr>
<tr>
<td>Oxygen</td>
<td>O2</td>
<td>Nil</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>Carbon Dioxide</td>
<td>CO2</td>
<td>0.31</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>N2</td>
<td>0.32</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>Methane</td>
<td>CH₄</td>
<td>95.23</td>
<td>2.0 9.53</td>
<td>190.46 906.00</td>
</tr>
<tr>
<td>Ethane</td>
<td>C₂H₆</td>
<td>2.99</td>
<td>3.5 16.68</td>
<td>10.45 49.90</td>
</tr>
<tr>
<td>Propane</td>
<td>C₃H₈</td>
<td>0.72</td>
<td>5.0 23.82</td>
<td>3.60 17.10</td>
</tr>
<tr>
<td>Iso-Butane</td>
<td>C₄H₁₀</td>
<td>0.14</td>
<td>6.5 30.97</td>
<td>0.91 4.34</td>
</tr>
<tr>
<td>N-Butane</td>
<td>C₄H₁₀</td>
<td>0.15</td>
<td>6.5 30.97</td>
<td>0.975 4.64</td>
</tr>
<tr>
<td>Pentane</td>
<td>C₅H₁₂</td>
<td>0.14</td>
<td>8.0 38.11</td>
<td>1.12 5.34</td>
</tr>
<tr>
<td>Helium</td>
<td>He</td>
<td>Nil</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>TOTAL</td>
<td></td>
<td>100.00</td>
<td></td>
<td>207.515 987.32</td>
</tr>
</tbody>
</table>

### TABLE 5.2

**MOLE COMPOSITION OF FUEL**

<table>
<thead>
<tr>
<th>Constituent</th>
<th>Formula</th>
<th>Moles of C per 100 Moles Fuel</th>
<th>Moles of H₂ per 100 Moles Fuel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Methane</td>
<td>CH₄</td>
<td>95.23</td>
<td>190.46</td>
</tr>
<tr>
<td>Ethane</td>
<td>C₂H₆</td>
<td>5.98</td>
<td>9.00</td>
</tr>
<tr>
<td>Propane</td>
<td>C₃H₈</td>
<td>2.16</td>
<td>2.89</td>
</tr>
<tr>
<td>Iso-Butane</td>
<td>C₄H₁₀</td>
<td>.56</td>
<td>0.70</td>
</tr>
<tr>
<td>N-Butane</td>
<td>C₄H₁₀</td>
<td>.60</td>
<td>0.75</td>
</tr>
<tr>
<td>Pentane</td>
<td>C₅H₁₂</td>
<td>.70</td>
<td>0.84</td>
</tr>
<tr>
<td>TOTAL</td>
<td></td>
<td>105.23</td>
<td>204.64</td>
</tr>
</tbody>
</table>
From the results of these computations it is noted that for 100 moles of fuel, 207.515 moles of O\textsubscript{2} are required.

Table 5.2 lists the mole composition of the fuel. The hydrocarbon constituents are listed in columns 1 and 2. The number of moles of carbon, C, per 100 moles of fuel is listed in column 3. It is obtained as follows: for 100 moles of fuel there are 95.23 moles of methane, CH\textsubscript{4}, and hence 95.23 moles of carbon. For 100 moles of fuel there are 2.99 moles of ethane, C\textsubscript{2}H\textsubscript{6}, and since there are 2 carbon atoms in each molecule of ethane, there are 2(2.99) = 5.98 moles of carbon for each 2.99 moles of ethane or 100 moles of fuel.

Column 4 of Table 5.2 is obtained in a similar fashion. For example, there are 0.14 moles of iso-butane per 100 moles of fuel. Since there are 10 atoms of hydrogen in each molecule of iso-butane and 2 atoms of hydrogen in molecular hydrogen, there are 10(0.14)/2 = 0.70 moles of H\textsubscript{2} per moles of fuel.

Hence for 100 moles of fuel, there are 204.64 moles of H\textsubscript{2} and 105.23 moles of C.

Next the analysis turns to the determination of the flue (combustion) gas composition. The constituents of the fuel are burned and yield new products of combustion.

Carbon burns and yields carbon dioxide, CO\textsubscript{2}. Since 1 mole of carbon yields 1 mole of carbon dioxide upon combustion, and there is 1 carbon atom in each CO\textsubscript{2} molecule, the number of moles of carbon per 100 moles of fuel also equals the number of moles of carbon dioxide formed. Hence from the foregoing discussion, and Table 5.2, 105.23 moles of CO\textsubscript{2} are formed per 100 moles of fuel.

The natural gas had 0.31 moles CO\textsubscript{2} per 100 moles fuel as given in Table 5.1 and there was negligible sulfur content which when burned forms SO\textsubscript{2}.
Thus the total $CO_2 + SO_2$ in the flue gas is $105.23 + 31 = 105.54$ moles per 100 moles of fuel.

The nitrogen, $N_2$, in the combustion gas is obtained from column 3 of Table 5.1 and is 0.32 moles per 100 moles of fuel.

Next the amount of oxygen required for the computed amount of $CO_2$ that is obtained by burning the carbon is taken as 105.23 since there are 2 atoms of oxygen in each molecule of $CO_2$ and there are 2 atoms of oxygen in $O_2$.

To determine the amount of oxygen required for the $H_2O$ formed by the burning of $H_2$, look at the resultant sum in column 4 of Table 5.2 where it is noted that 204.64 moles of $H_2$ exist per 100 moles of fuel and when the $H_2$ burns $H_2O$ is formed. Since there is 1 atom of oxygen per molecule of $H_2O$ and there are 2 atoms of oxygen in $O_2$, there is $204.64/2 = 102.32$ moles of $O_2$ required to burn the $H_2$.

Hence the total oxygen, $O_2$, required is $105.23 + 102.32 = 207.55$ moles per 100 moles of fuel.

The total nitrogen, $N_2$, supplied is equal to $3.76 (O_2$ required) or $3.76(207.55) = 782$ moles $N_2$ per 100 moles of fuel.

The total dry air supplied for the combustion process is thus $207.55 + 782 = 989.55$ moles dry air per 100 moles of fuel.

The water, $H_2O$, in the air supplied for the combustion process is computed using a standard multiplying factor of $0.0212$ for air at $80^\circ F$ and 60% relative humidity. Hence the $H_2O$ in the air is $0.0212(989.55) = 21$.

This water is combined with the water, $H_2O$, formed by burning $H_2$ which was 204.64 moles per 100 moles of fuel to yield the total $H_2O$ as $204.64 + 21 = 225.64$ moles of $H_2O$ per 100 moles of fuel. It is assumed that no $C$ burned to yield $CO$. 
From these values of the composition of the wet flue gas the total amount may be calculated as shown in Table 5.3.

### TABLE 5.3

**TOTAL WET FLUE GAS**

<table>
<thead>
<tr>
<th>Constituent</th>
<th>Moles</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\text{CO}_2 + \text{SO}_2$</td>
<td>105.54</td>
</tr>
<tr>
<td>$\text{O}_2$</td>
<td>0.782</td>
</tr>
<tr>
<td>$\text{N}_2$</td>
<td>225.64</td>
</tr>
<tr>
<td>$\text{H}_2\text{O}$</td>
<td>0.11</td>
</tr>
<tr>
<td>$\text{CO}$</td>
<td>0.18</td>
</tr>
</tbody>
</table>

1113.18

The percent moisture in the combustion gas is given by the following equation, with data from Table 5.3.

\[
\% \text{ moisture} = \frac{225.64}{1113.18} \times 100 = 20.3\%
\]

Now using the curve below from the reference. 92

![Graph showing specific heat of combustion gas](image)

$C_{pg} = \text{Mean specific heat of combustion gas (Btu/lb-°F)}$

$t_1 + t_2 = \text{Sum of entering and leaving temp., °F}$

Fig. 5.3 Approximate mean specific heat of combustion gas.
\[ T_1 = 2155^\circ F \text{ = value of furnace exit temperature given} \]

\[ T_2 = 70^\circ F \text{ = value of atmospheric temperature assumed} \]

\[ T_1 + T_2 = 2225^\circ F \]

The value of \( C_{pg} = 0.335 \frac{\text{btu}}{\text{lb}^\circ F} \text{ from Fig. 5.3} \]

The value of specific heat used in the simulation is \( 0.34 \frac{\text{btu}}{\text{lb}^\circ F} \)

Later in the discussion, it will be shown that based on a heat balance, the furnace exit temperature will actually be used as \( 2190^\circ F \) rather than \( 2155^\circ F \).

The preceding analysis of the combustion gas allowed the computation of the specific heat of the gas. This is required in order to determine the portion of the total heat liberated by the combustion process that leaves the furnace. It is assumed that within the furnace itself, where the tube walls are exposed to the free flame, the heat is transferred entirely by radiation. Whereas in the sections of the boiler which are shielded visually from the actual flame caused by the combustion process, the heat transfer is assumed to be entirely by convection.

There is only one lump in this simulation which is a combination of the two modes of heat transfer. That is lump 10, the platen and finishing superheater. The reason it is heated as a combination is due to its placement, as seen in the side elevation of the plant, Figure 1.3. The platen portion of this superheater overhangs and is in view of the flame and hence is subject to radiative heating.

Table 5.4 below summarizes the heating that is done by the combustion gases and also by other heat sources throughout the steam generator.

The amount of heat that is transferred from one substance to another is determined by a variety of factors, but in a most general way may be described as heat flowing from a relatively warmer substance to a relatively colder substance.

The analysis of the flame in combustion and specifically the determination of its temperature and the subsequent determination of the temperature of the combustion gases by rigorous methods is difficult, inaccurate and beyond the
<table>
<thead>
<tr>
<th>Description</th>
<th>Lump No.</th>
<th>Heating Mode</th>
<th>Heat Transfer From</th>
<th>Heat Transfer To</th>
</tr>
</thead>
<tbody>
<tr>
<td>Economizer</td>
<td>1</td>
<td>Convection</td>
<td>Combustion gas</td>
<td>Compressed water</td>
</tr>
<tr>
<td>Pass 1</td>
<td>2</td>
<td>Radiation</td>
<td>Combustion gas</td>
<td>Compressed water</td>
</tr>
<tr>
<td>2</td>
<td>3</td>
<td>Radiation</td>
<td>Combustion gas</td>
<td>Compressed water</td>
</tr>
<tr>
<td>3</td>
<td>4</td>
<td>Radiation</td>
<td>Combustion gas</td>
<td>Compressed water</td>
</tr>
<tr>
<td>4</td>
<td>5</td>
<td>Radiation</td>
<td>Combustion gas</td>
<td>Compressed water</td>
</tr>
<tr>
<td>5</td>
<td>6</td>
<td>Radiation</td>
<td>Combustion gas</td>
<td>Compressed water</td>
</tr>
<tr>
<td>6</td>
<td>7</td>
<td>Convection</td>
<td>Combustion gas</td>
<td>Compressed water</td>
</tr>
<tr>
<td>7</td>
<td>8</td>
<td>Convection</td>
<td>Combustion gas</td>
<td>Compressed water</td>
</tr>
<tr>
<td>Primary Superheater</td>
<td>9</td>
<td>Convection</td>
<td>Combustion gas</td>
<td>Superheated steam</td>
</tr>
<tr>
<td>Platen and Finishing, S. II.</td>
<td>10</td>
<td>Radiation and Convection</td>
<td>Combustion gas</td>
<td>Superheated steam</td>
</tr>
<tr>
<td>Steam Leads</td>
<td>11</td>
<td>Convection</td>
<td>Superheated steam</td>
<td>Air</td>
</tr>
<tr>
<td>Reheater</td>
<td>12</td>
<td>Convection</td>
<td>Combustion gas</td>
<td>Superheated steam</td>
</tr>
<tr>
<td>Low Pressure Feedwater Heater</td>
<td>13</td>
<td>Convection</td>
<td>Superheated steam</td>
<td>Compressed water</td>
</tr>
<tr>
<td>Hi Pressure Feedwater Heater</td>
<td>14</td>
<td>Convection</td>
<td>Superheated steam</td>
<td>Compressed water</td>
</tr>
<tr>
<td>Air Heater</td>
<td>21</td>
<td>Convection</td>
<td>Combustion gas</td>
<td>Air</td>
</tr>
</tbody>
</table>
scope of this work. Since the determination of the combustion gas temperature would lead to the most obvious solution of the heat flux into the working fluid, an alternate solution had to be obtained.

Data was obtained that related the heat flux from the combustion gases at full load operation to the various lumps of the steam generator.

Table 5.5 below is that data. The table is divided into two parts, radiative and convective heat transfer, so note that lump 10-a is convective and lump 10-b is radiative heat transfer.

From Table 5.5 it is seen that at full load, (i.e., with rated conditions) the total heat transferred by the combustion gas to the lumps given is $4581.5 \times 10^6 \text{ btu/hr}$. While the air heater also utilizes heat taken from the combustion gases and transferred to the atmospheric air that is used for combustion, it is assumed in this study that it has a negligible effect on the process simulation. Economically, of course, it is important to preheat the air in this way, prior to combustion.

The next question that naturally arises is; how is the heat distributed under different conditions of load and more particularly during the transient periods? In answer to the first part of the question, it is assumed that the same distribution ratio of heat flux would exist on any load as would exist at full load. This assumption is based on the premise that in a steady-state condition the temperature profile in space of the combustion gases would be essentially the same for any load, as will be the temperature profile of the working fluid. Then since the heat transfer is proportional to the temperature difference, the ratio of the heat flux into one lump to the total heat transferred by the combustion gas will be constant.

In regard to the last part of the above question, the transient effect of the heat flux would be determined by the density and velocity of the combustion gas. Basically, the problem is that after changing the firing rate, in changing the amount of fuel burned, there will be a time delay before the various lumps notice the change. That is, the lumps further along the combustion gas flow path (see Figure 1.2) will not experience a change as rapidly as lumps early in
## TABLE 5.5

HEAT FLUX AT RATED CONDITIONS

<table>
<thead>
<tr>
<th>Description</th>
<th>Lump No.</th>
<th>Heat Flux (btu/hr.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Economizer</td>
<td>1</td>
<td>$247 \times 10^6$</td>
</tr>
<tr>
<td>Pass 6</td>
<td>7</td>
<td>$319 \times 10^6$</td>
</tr>
<tr>
<td>Pass 7</td>
<td>8</td>
<td>$159 \times 10^6$</td>
</tr>
<tr>
<td>Primary Superheater</td>
<td>9</td>
<td>$398 \times 10^6$</td>
</tr>
<tr>
<td>Finishing Superheater</td>
<td>10-a</td>
<td>$481 \times 10^6$</td>
</tr>
<tr>
<td>Reheater</td>
<td>12</td>
<td>$867 \times 10^6$</td>
</tr>
</tbody>
</table>

Total Convective Heat Transfer $2471 \times 10^6$

| Pass 1                        | 2        | $199.5 \times 10^6$ |
| Pass 2                        | 3        | $391 \times 10^6$   |
| Pass 3                        | 4        | $359 \times 10^6$   |
| Pass 4                        | 5        | $398 \times 10^6$   |
| Pass 5                        | 6        | $319 \times 10^6$   |
| Finishing Superheater (Platen)| 10-b     | $444 \times 10^6$   |

Total Radiative Heat Transfer $2110.5 \times 10^6$

Total Heat Transferred by Combustion Gas $4581.5 \times 10^6$
the combustion gas flow path. The average density of the combustion gas is much smaller than the average density of the working fluid and it moves at much higher velocities. The velocity of boiler circulation is less than 700 ft/min, steam in superheater tubes is 2,000–5,000 ft/min while the combustion gas has a velocity of 3,000–6,000 ft/min. (See pp. 8–10, reference 92)

It is then reasonable, as was done in this simulation, to assume that the dynamic effect of the combustion gas could be neglected. Thus when the firing rate is increased the increase in heat flux will be observed at each lump simultaneously.

Referring back to the previous analysis of the fuel, it was found that the heat value of the natural gas used in the simulated plant was 1071 btu/ft$^3$. The total value of heat liberated in the combustion process is thus obtained by knowing the fuel flow. Equation (5.1) below gives this relation.

$$Q_G = \frac{(W_{NG}) (H_C)}{3600} \quad (5.1)$$

where

- $Q_G$ = Total heat liberated (btu/sec)
- $W_{NG}$ = Flow of natural gas (ft$^3$/hr)
- $H_C$ = Heat value of fuel (btu/ft$^3$)

The full load (560 MW) value of fuel flow is $5 \times 10^6$ ft$^3$/hr so the heat liberated by combustion at full load is $Q_G = (5 \times 10^6) (1071) = 5.36 \times 10^9$ btu/hr or $Q_G = 5.36 \times 10^9 / 3.6 \times 10^3 = 1.49 \times 10^6$ btu/sec.

Comparing this result with that indicated in Table 5.5, it is observed that there is an excess amount of heat generated over the heat that is actually transmitted to the water and the steam. This may be accounted for in the heat lost due to leakage from the furnace, heat exhausted to the atmosphere through the stack, heat that is used to heat the incoming air in the air heater and heat transferred from the furnace and other equipment to the atmosphere by conduction.
The difference in the total heat liberated and that absorbed by the process as simulated is called excess heat. Using the values just cited, the excess heat at full load is given by the following equation.

$$QE = QG - 4581.5 \times 10^6 \text{ btu/hr}$$  \hspace{1cm} (5.2)

$$QE = (5360 - 4581.5) \times 10^6 \text{ btu/hr} = 778.5 \times 10^6 \text{ btu/hr}$$

At this point note that the thermal efficiency, $\eta$, equals the heat flux utilized divided by the heat flux generated and in this case becomes,

$$\eta = \frac{4581.5}{5360} \times 100 = 85.3\%$$

Also, another useful computed quantity is the heat rate and could be computed from the results of the simulation.

$$HR = \frac{QG}{POWER}$$  \hspace{1cm} (5.3)

where

- $HR$ = Heat rate (btu/hr-kw)
- $QG$ = Total heat liberated (btu/hr)
- POWER = Output power

At full load $HR = \frac{4581.5 \times 10^3}{560} = 8190$ btu/hr-kw. The lower the heat rate, the better (more efficient) the unit.

From the discussion earlier, it is evident that the total heat liberated by combustion equals the sum of radiative, convective, and excess heat. This is shown in equation (5.4).

$$QG = QR + QC + QE$$  \hspace{1cm} (5.4)

Let $QC + QE = QCT$ where $QCT$ represents that portion of the heat flux liberated by combustion that the combustion gases still contain when they leave the radiative portion of the furnace.

The heat content, as it is called, of the combustion gas leaving the furnace may be computed using an equation that utilizes a knowledge of the combustion
gas mass flow and the furnace exit temperature in addition to the value of the specific heat that has already been computed.

The equation (see p. 6-28, reference 30) is given as

$$Q_{CT} = \frac{W_G(1) \cdot (C_{PG}) \cdot (T_G-70)}{3600} \quad (5.5)$$

where

- $Q_{CT}$ = Heat content of gas leaving furnace $\text{ btu } \text{sec}$
- $W_G(1)$ = Combustion gas flow (lb/hr)
- $C_{PG}$ = Specific heat of combustion gas (btu/lb-$^\circ$F)
- $T_G$ = Temperature of combustion gas at furnace exit ($^\circ$F)

From design data it is known that the mass flow of the combustion gas at rated conditions for the simulated unit is 4,509,000 lb/hr. That is, $W_G(1) = 4,509,000 \text{ lb/hr} = (4,509,000/3600) \text{ lb/sec} = 1252 \text{ lb/sec}$.

As mentioned earlier in this discussion of the combustion process the determination of the temperature of the combustion gases was not done in this simulation.

The value of the temperature of combustion gas at furnace exit ($T_G$) was obtained from the manufacturer as a representative value for full load conditions, and since the flame temperature remains essentially constant and the air flow is controlled in conjunction with the fuel flow, it is further assumed that $T_G$ (furnace exit temperature) remains constant for all loads simulated.

The value of $T_G$ obtained was $2155^\circ$F, but this value had to be adjusted slightly in order to provide for a proper heat balance. As will be seen later, the problem of tuning the control system with the process is more critical than one would at first imagine.

The method of computation of the corrected value of $T_G$ is given in the following discussion.

From equation (5.4)

$$Q_{C+Q} = Q_{CT} = Q_G - Q_R \quad (5.6)$$
From Table 5.5, QR is given as $2110.5 \times 10^6$ btu/hr at full load. Equation (5.1) gives the relation for QG and just following that equation it is seen that $QG = 5360 \times 10^6$ btu/hr at full load. Hence equation (5.5) becomes

$$QCT = QC+QE = QG-QR = (5360.2110.5) \times 10^6 \text{ btu/hr}$$

$$QCT = 3249.5 \times 10^6 \text{ btu/hr} = WG(1) (CPG) (TG-70.)$$

Since $WG(1) = 4.509 \times 10^6 \text{ lb/hr}$ and $CPG = 0.34 \text{ btu/lb}^\circ\text{F}$ from earlier discussions, the value of TG may be computed. Hence

$$3249.5 \times 10^6 \text{ btu/hr} = (4.509 \times 10^6 \text{ lb/hr}) (0.34 \text{ btu/lb}^\circ\text{F}) (TG-70.) \circ\text{F}$$

From which, $TG = 2190. \circ\text{F}$. This value is subsequently used throughout the simulation study for all loads.

The excess heat, QE, leaving the furnace has been computed earlier for full load conditions as $778.5 \times 10^6 \text{ btu/hr}$. Since the heat flux is directly proportional to the total combustion gas mass flow, the value of excess heat at other loads may be computed from the equation below.

$$QE = \frac{WG(1)}{4.509 \times 10^6 \text{ lb/hr}} \frac{778.5 \times 10^6 \text{ btu/hr}}{3600\text{ sec}}$$

$$QE = \frac{(WG(1))(778.5)}{(4.509)(3600)} \text{ btu/sec}$$

where $WG(1)$ is the combustion gas mass flow (lb/hr).

Hence the value of the convective heat transfer may be obtained for any load. In the computer program, it is in block 18, which is given as in equation (5.7)

$$QC = QCT - \frac{(WG(1))(778.5)}{(4.509)(3600.)} \quad (5.7)$$

The value of the combustion gas mass flow is obtained for this simulation from the value of required fuel flow. The variable name for fuel flow is WNG.
and its units are ft$^3$/hr as mentioned earlier. The fuel to air ratio is assumed constant so that with a change in the fuel flow, a corresponding change occurs in the air flow to the furnace. Hence when the fuel is burned, yielding combustion gas, there is a direct relationship between the amount of combustion gas and the amount of fuel burned.

This relationship may be derived as follows. From the actual plant data obtained during a test run, the values of measured combustion gas flow and fuel flow were recorded as a function of the load of the unit. The combustion gas flow was measured at an orifice in the stack by three recorders, whose outputs were sent to the logging computer and the weighted average taken as the correct value. Due to the variable parameters, the measurement is considered to be only an approximate value.

The simulated plant has a digital computer that is used for data, logging, various computations and in addition has backup control system capability. It has the capability of logging approximately 600 plant variables that are either measured directly or are internally computed. Each of these has a computer name that is a combination pneumatic name and code number. For example, the measured combustion gas flow is labeled BFOOX, the generated power is GEO3A and the flow of natural gas (fuel flow) is FFO6X. Table 10.1 gives a complete listing. The plot below in Figure 5.4 relates measured combustion gas flow with generated power as obtained from test run number 9.

By inspection of the test data in Figure 5.4, it is observed that the relation is linear and the equation relating the variables is given by equation (5.8)

$$\text{AWG} = \frac{120,000}{13} \text{POWER} + 30,000 \text{ lb/hr} \quad (5.8)$$

where

$$\text{AWG} = \text{Combustion gas flow}$$

Also on the same figure is the relationship obtained between the natural gas (fuel flow) and the generated power. It too is linear, and with the same slope as the curve relating AWG-POWER. Its relationship is given by equation (5.9)
\[
\text{WNG} = \frac{120,000}{13} \text{POWER} - 170,000 \text{ft}^3/\text{hr} \tag{5.9}
\]

Solving for \(\text{POWER}\) in equation (5.8)

\[
\text{POWER} = (\text{AWG} - 30,000) \frac{13}{120,000} \tag{5.10}
\]

Substituting this value into equation (5.9) results in

\[
\text{WNG} = \left(\frac{120,000}{13}\right) \left(\frac{13}{120,000}\right) (\text{AWG} - 30,000) - 170,000 \text{ft}^3/\text{hr} \tag{5.11}
\]

\[
\text{WNG} = \text{AWG} - 200,000
\]

\[
\text{AWG} = \text{WNG} + 200,000 \text{lb/hr} \tag{5.12}
\]

As an example consider the fuel flow at rated load (560 mw) where

\[
\text{WNG} = 5,000,000 \text{ ft}^3/\text{hr}.
\]

Then from equation (5.12), \(\text{AWG} = 5.200,000 \text{ lb/hr}.

Now recall that this is only an approximate value of the combustion gas flow. Assuming that there is no loss of combustion gas along the furnace path, then the actual flow is equal to \(\text{WG}(1)\) as given earlier as 4,509,000 lb/hr. So
the measured value is higher than the actual value. Hence assuming linearity, 
WG(1) may be obtained from the following equation

$$WG(1) = (AWG) \left( \frac{4.509}{5.20} \right)$$

(5.13)

Combining equation (5.12) and (5.13) the relationship between fuel flow and combustion gas flow is obtained.

$$WG(1) = (WNG + 200,000.) \left( \frac{4.509}{5.20} \right) \text{ lb/hr}$$

(5.14)

In Figure 5.5 below, a block diagram summarizing the determination of the convection, radiative, and excess heat as described is given.

---

**Fig. 5.5 Block diagram of heat flux computation**

The portion of the heat flux passing into any one lump at a specific load is thus found by first determining the amount of convective (QC) and radiative (QR) heat flux that is produced by the particular fuel and air flow and then the proper ratio is selected based on the steady-state heat balance. Table 5.6 below illustrates this, using full load data obtained from a heat balance.
<table>
<thead>
<tr>
<th>Description</th>
<th>'Lump' No (I)</th>
<th>Ratio</th>
<th>QW (1,I) (btu/sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Economizer</td>
<td>1</td>
<td>(\frac{247}{2471} = 0.1)</td>
<td>QW (1,1) = (0.1) QC</td>
</tr>
<tr>
<td>Pass 1</td>
<td>2</td>
<td>(\frac{199.5}{2110.5} = 0.0945)</td>
<td>QW (1,2) = (0.0945) QR</td>
</tr>
<tr>
<td>Pass 2</td>
<td>3</td>
<td>(\frac{391}{2110.5} = 0.185)</td>
<td>QW (1,3) = (0.185) QR</td>
</tr>
<tr>
<td>Pass 3</td>
<td>4</td>
<td>(\frac{359}{2110.5} = 0.17)</td>
<td>QW (1,4) = (0.17) QR</td>
</tr>
<tr>
<td>Pass 4</td>
<td>5</td>
<td>(\frac{398}{2110.5} = 0.1885)</td>
<td>QW (1,5) = (0.1885) QR</td>
</tr>
<tr>
<td>Pass 5</td>
<td>6</td>
<td>(\frac{319}{2110.5} = 0.151)</td>
<td>QW (1,6) = (0.151) QR</td>
</tr>
<tr>
<td>Pass 6</td>
<td>7</td>
<td>(\frac{319}{2471} = 0.129)</td>
<td>QW (1,7) = (0.129) QC</td>
</tr>
<tr>
<td>Pass 7</td>
<td>8</td>
<td>(\frac{159}{2471} = 0.0645)</td>
<td>QW (1,8) = (0.0645) QC</td>
</tr>
<tr>
<td>Primary Superheater</td>
<td>9</td>
<td>(\frac{398}{2471} = 0.161)</td>
<td>QW (1,9) = (0.161) QC</td>
</tr>
<tr>
<td>Platen and finishing SH</td>
<td>10</td>
<td>(\frac{481}{2471} + \frac{444}{2110.5} = 0.195 + 0.21)</td>
<td>QW (1,10) = (0.195) QC + (0.21) QR</td>
</tr>
<tr>
<td>Reheater</td>
<td>12</td>
<td>(\frac{867}{2471} = 0.35)</td>
<td>QW (1,12) = (0.35) QC</td>
</tr>
</tbody>
</table>
Table 5.5 it is found that the total radiative heat flux (QR) equals $2110.5 \times 10^6$ btu/hr and the total convective heat flux (QC) equals $2471.1 \times 10^6$ btu/hr.

In this simulation, the control of the flow of combustion gases flowing over the primary superheater and reheater is considered. There is a damper in the upper part of the rear of the boiler that can be adjusted to direct the combustion gases over the two heating surfaces with different ratios. This damper is adjusted by the control system whereby a particular demand signal is fed to the damper actuator and an appropriate ratio is obtained. This actuating signal will be described in more detail in the chapter dealing with the control system.

The amount of damper action depends on the load (i.e., generated power) and its relation to the demanded load. The control system signal that ultimately directs the damper is obtained in the Reheat Steam Temperature Control section of the control system. Its label is DDEC and is described in detail later.

The action of the dampers provide a diversion of the main stream of combustion gas flow, WG(1), into two separate combustion gas flows. As seen from Figure 1.2, part of the combustion gases flow over the rear heating surfaces of the rear part of the furnace, which is the primary superheater. This combustion gas flow is labeled WG(2). The remainder of the combustion gas flows over the reheater which is located in the front of the rear portion of the furnace. Its heating combustion gas is labeled WG(3). See also Figure 1.3.

There is no exact relationship known for the damper action or the relation of the damper action to the combustion gas flow in the two paths. From operating experience it is known however that for higher loads, near the full load operating point, there is relatively more combustion gas flow over the primary superheater, as compared with that flowing over the reheater, than there is at lower loads. The converse is also true at low load levels, below 50% load. This is illustrated in Figure 5.6 below.

From Table 5.6 it is seen that at full load the heat flux into the reheater, lump 12, is equal to 0.35 times the convective heat input to the system.
Likewise the heat flux into the primary superheater is 0.161 times the convective heat transferred.

\[ Q_{W(1,9)} = 0.161Q_C \]  \hspace{1cm} (5.16)

Figure 5.7, below, illustrates the effect of the damper action. \( R_{12} \), marked on the diagram, is developed subsequently and is a ratio that is greater than 1 for loads between no load and full load. At full load \( R_{12} = 1 \) in order to satisfy the heat balance requirement.

Writing the equations for the combustion gas flow from the information given in Figure 5.6 yields:

\[ W_{G(2)} = W_{G(1)} \left( 0.48 + \frac{0.04}{560} \right) DDEC \]  \hspace{1cm} (5.17)

where

\[ DDEC = \text{adjusted load demand} \]
\[ W_{G(1)} = \text{total combustion gas flow}. \]
Figure 5.7 Damper action for heat flux

\[ \begin{align*}
\text{WG}(2) &= \text{combustion gas flow over primary superheater} \\
\text{WG}(3) &= \text{combustion gas flow over reheater}
\end{align*} \]

\[ \text{WG}(3) = \text{WG}(1) \left(0.52 - \frac{0.04}{560}\right) \text{DDEC} \]  
(5.18)

where

\[ \text{WG}(3) = \text{combustion gas flow over reheater} \]

The ratio of the combustion gas flow of the reheater to that of the primary superheater gives a relationship by which the damper action can be simulated. If this ratio is normalized to full load demand, 560 mw, the ratio R12 is obtained.
\[
R_{12} = \frac{WG(3)}{WG(2) R_{12}'} = \frac{WG(1) \left(0.52 - \left(\frac{0.04}{560.}\right)DDEC\right)}{WG(1) \left(0.48 + \frac{0.04}{560.} DDEC\right) R_{12}'}
\]

where

\[
R_{12}' = R_{12}\bigg|_{DDEC = 560 \text{ mw}} = \frac{0.48}{0.52}
\]

\[
R_{12} = \left(\frac{0.52 - 0.04 DDEC/560.}{0.48+0.04 DDEC/560.}\right) \left(0.48/0.52\right)
\]

(5.19)

The heat flux equation for lump 12, the reheater, becomes

\[
Q_{W(1,12)} = (R_{12}) (0.35)Q_{C}
\]

(5.20)

From Table 5.6 it is seen that there is another constraint on the heat flux equations, namely, that

\[
Q_{W(1,9)} + Q_{W(1,12)} = (0.161+0.35)Q_{C}
\]

\[
Q_{W(1,9)} + Q_{W(1,12)} = 0.511Q_{C}
\]

Hence the heat flux for the primary superheater is given by the following equation

\[
Q_{W(1,9)} = (0.511Q_{C}) - Q_{W(1,12)}
\]

(5.21)

If more exact information is available, the above equations may be adjusted for the simulation of the dampers.
CHAPTER 6
TECHNIQUES USED FOR THE SOLUTION
OF THE PROCESS EQUATIONS

As described earlier, in a previous section, the fundamental equations describing the process may be expressed as an ordinary differential equation of the first order. One of these equations relates the enthalpy to time and the other is the relationship between the wall temperature and time. The enthalpies and wall temperatures are the dependent variables with the time being the independent variable. In the solution of the process equations, the equations are considered to be of the initial value type. That is to say, the values of the independent variables are assumed to be known at the beginning of the solution time.

This type problem may be stated as follows: given a particular point, \((t_0, h_0)\) and a differential equation, \(\frac{dh}{dt} = f(t, h)\), find the unknown function, \(h(t)\), which passes through the initially known point, \(t_0, h_0\), and has the derivative, \(\frac{dh}{dt} = f(t, h)\). The desired solution, \(h(t)\), is the curve which passes through the point, \((t_0, h_0)\), with a derivative \(\frac{dh}{dt} = f(t, h)\). This value of the derivative is equivalent to the slope of the solution curve at any point, \((t, h)\), on the curve.

The reason for using a numerical solution for the solution of these differential equations is due to the fact that the equations are very difficult, or impossible, to solve analytically. While there are a number of different methods available for the numerical solution of the differential equations and the various methods have varying degrees of sophistication and accuracy, only two methods have been investigated for this process. Specifically, the Euler method and the Runge–Kutta method, have been utilized. The program, as it is
presently set up, uses the Runge-Kutta, the third order method, and a brief description of that method follows. Consider equation (6.1):

$$\frac{dh}{dt} = f(t, h)$$  \hspace{1cm} (6.1)

Referring to Figure 6.1 which represents the solution of the equation, the points, \( t_0 \) and \( H_{o} \), are noted on the figure. In this case, consider that equation (6.1) is satisfied and the initial conditions are given as \( H = H_{o} \) at \( T = t_0 \) and it is desired to find the value of \( H \) when \( T = t_0 + \text{DTIME} \). The initial conditions are given by the point, \( P1 \), in the figure. The length, \( K1 \), can be found from a knowledge of the equation (6.1), as is illustrated below for the particular variables under consideration.

Consider the actual equation, as represented earlier in the process. This is equation (6.2).
\[
\frac{dh}{dt} = \frac{1}{m} (Q - W(h_0 - h_1)) \quad (6.2)
\]

The value of \( K_1 \) is shown on the figure to be a very rough approximation of the actual functional value at a time, \( t_0 + \text{DTIME} \). In this case, \( K_1 \) is given by equation (6.3):

\[
K_1 = \text{DTIME} \{ f(t_0, H_0) \} \quad (6.3)
\]

The value of the estimate is equal to \( H_0 + K_1 \) and is a poor approximation of the desired function. Hence, a new estimate, \( K_2 \), is found using the following equation:

\[
K_2 = \text{DTIME} \{ f(t_0 + \frac{1}{2} \text{DTIME}, H_0 + \frac{1}{2} K_1) \} \quad (6.4)
\]

There are three curves shown in the figure, curve 1, curve 2, and curve 3, with only curve 1 being a member of the family of curves representing a solution of equation (6.1) which actually passes through the initial value point. The point \( P_2 \) lies on the tangent to curve 1 which has a tangent at point \( P_1 \) and the value of the \( H \) coordinate of \( P_2 \) corresponds to the \( H_0 + (1/2) K_1 \).

It is recognized that this point, \( P_2 \), is not on the curve \( P_1 \), but is on a different curve of the family of curves. It would have been desirable to have been on curve 1, but at this point in the solution the \( H \) value or the value of the independent variable is not known at the value of \( T = t_0 + (1/2) \text{DTIME} \). A tangent to the new curve is then taken at the point, \( P_2 \), and a line, which is parallel to this line, is then constructed passing through the initial point, \( P_1 \). This line then intersects the vertical line at \( t_0 + \text{DTIME} \), at a value of \( H_0 + K_2 \), as shown in Figure 6.1.

The method of Runge-Kutta utilizes an average value of the gradient at the three points indicated, \( P_1 \), \( P_2 \), and \( P_3 \), of the lines \( L_1 \), \( L_2 \), and \( L_3 \) marked on Figure 6.1. This average is an average of the three slopes, one at the initial point, one at the center, and one at the end of the time interval. The third
slope is taken from a curve above curve 1 (that is, curve 3) and the other slope was taken from curve 2 which appears in Figure 6.1 below curve 1. Since the point \((t_o + \Delta T, H_o + K_2)\) is indicated nearly on the curve 1 and \(H_o = H_o + K_1\) is below curve 1, as indicated with this family of curves, then \(H_o = H_o + K_2 + (K_2 - K_1)\) should be above curve 1. This is due to the fact that \(K_2 - K_1\) is a positive number. Hence for point 3, \(H_o = H_o + 2K_2 - K_1\) and point 3 is taken as coordinate \((t_o + \Delta T, H_o + 2K_2 - K_1)\). The tangent to curve 3 at \(P_3\) gives the line \(L_3\), then the value of \(K_3\) is obtained as illustrated in Figure 6.1 by drawing a line parallel to \(L_3\) through \(P_1\).

\[
K_3 = (\Delta T) f (t_o + \Delta T, H_o + 2K_2 - K_1)
\] 

Equation (6.5) gives the value for \(K_3\). Generally, the value of \(H_o\) can then be computed from the initial value \(H_o\) + a weighted average of these values of \(K\). This is derived analytically by taking an equation such as equation (6.6) and expanding the functions \(K_1/\Delta T\), \(K_2/\Delta T\), \(K_3/\Delta T\) in a Taylor series about the point \(t_o\), \(h_o\) and then substituting these values back into equation (6.6) getting a number of coefficients as functions of \(a's\) and \(b's\).

\[
h_1 = h_o + a_1 K_1 + a_2 K_2 + a_3 K_3
\] 

where

\[
K_1 = (\Delta t) f (t_o, h_o)
\]

\[
K_2 = (\Delta t) f (t_o + b \Delta t, h_o + b_2 K_1)
\]

\[
K_3 = (\Delta t) f (t_o + b_3 \Delta t, h_o + b_4 K_2 + b_5 K_1)
\]

Then the Taylor series expansion, shown in equation (6.7) is compared term by term with equation (6.6) and the constants are solved for equation 6.7.
\[ h_1 = h_0 + \frac{dh}{dt} \bigg|_0 \Delta t + \frac{d^2h}{dt^2} \bigg|_0 \frac{\Delta t^2}{2} + \frac{d^3h}{dt^3} \bigg|_0 \frac{\Delta t^3}{6} + \ldots \]  \hspace{1cm} (6.7)

where

\[ \frac{dh}{dt} \bigg|_0 = f(t_0, h_0) \]

\[ \frac{d^2h}{dt^2} \bigg|_0 = \left[ \frac{\partial f}{\partial t} \bigg|_0 + \frac{\partial f}{\partial h} \bigg|_0 \right] f(t_0, h_0) \]

\[ \frac{d^3h}{dt^3} \bigg|_0 = \left[ \frac{\partial^2 t}{\partial t^2} \bigg|_0 + \frac{2\partial^2 t}{\partial t \partial h} \bigg|_0 \right] f(t_0, h_0) + \left[ \frac{\partial^2 t}{\partial h^2} \bigg|_0 \right] (f(t_0, h_0))^2 + \frac{\partial f}{\partial h} \bigg|_0 \left[ \frac{d^2h}{dt^2} \right] \]

The result is given as equation (6.8)

\[ HO = HO_0 + \frac{1}{6}(K_1 + 4K_2 + K_3) \]  \hspace{1cm} (6.8)

where

\[ K_1 = \text{DTIME} \left[ f(t_0, HO_0) \right] \]

\[ K_2 = \text{DTIME} \left[ f(t_0 + \frac{1}{2} \text{DTIME}, HO_0 + \frac{1}{2} K_1) \right] \]

\[ K_3 = \text{DTIME} \left[ f(t_0 + \text{DTIME}, HO_0 + 2K_2 - K_1) \right] \]

Thus equation (6.8) is the third order Runge-Kutta formula.

The labels of the constants used in the computer program are DTWR, mnemonically representing ATW, Runge-Kutta method, and DHOR, representing \( \Delta HO \) Runge-Kutta method. The method of solution is indicated in the block diagram in Figure 6.2 where it is seen that the loop is completed 3 times in order to compute the three separate constants required for the Runge-Kutta third order method. First, the subroutine, TSSPH, is called to obtain the output temperature of a lump corresponding to the pressures and enthalpies at a particular point in question. Then, the value of heat flow into the water or steam of the lump is computed which is a function of the wall temperatures and output temperatures. Then the respective constants, DTWR and DHOR, are
DO 40 
J1 = 1, 3
Call TSSPH
TO(J1, N)
Q(J1, N) = f(TW, TO(J1, N))

1

DTWR(J1) → DHOR(J1) → IF
(J1 > 3)
J1 < 3

TW(2, N) = TW(1, N) + DTWR(1)/2

2

TW(3, N) = TW(1, N) + 2*DTWR(2) - DTWR(1)

3

HO(3, N) = HO(1, N) + 2*DHOR(2) - DHOR(1)

4

INCHRO = INCHRO + (D(1) + 4D(2) + D(3))/6

5

HO(1, N) = HO(1, N) + INCHRO

6

HI(1, N+1) = MO(1, N)

TW(1, N) = TW(1, N) + (D(1) + 4D(2) + D(3))/6

Fig. 6.2 Block diagram of Runge-Kutta solution procedure
computed. Following this, the logic leads to the determination of the ordinate values corresponding to the points P2 and P3 in Figure 6.1. These are represented by TW(2,N) and TW(3,N) and HO (2, N) and HO (3, N).

It should be noted that the detail of the computation is illustrated in the computer program listed in a latter section. The new value of wall temperature for each lump is then computed using equation (6.9).

\[
TW (1, N) = TW(1, N) + \frac{1}{6} \left( (DTWR(1) + (4) DTWR(2) + DTWR(3) \right) \tag{6.9}
\]

The incremental value of enthalpy is computed as given in equation (6.10)

\[
INCRHO = INCRHO + \frac{DHOR(1) + (4) DHOR(2) + DHOR(3)}{6} \tag{6.10}
\]

Since the simulation is done using the enthalpy increment whereby the initial enthalpy into the system is computed as described in the section on feedwater heaters using the heat balance computation and then the succeeding values of enthalpies are changed according to the figure shown below which is Figure 6.3.

![Fig. 6.3 Enthalpy increment](image)

After the enthalpy increment is computed, the output enthalpy of a respective lump may be computed using equation (6.11)

\[
HO(1, N) = HO(1, N) + INCRHO \tag{6.11}
\]
In a way of comparison, the two methods used in the process, as mentioned earlier, were the Euler method and the Runge-Kutta method of solution of differential equations. Both methods yielded fairly comparable results with the proper choice of step size. With the Euler method the step size should be approximately 0.1 second in order for a stable result to be obtained, while, with the Runge-Kutta third order method, the time step could be approximately 0.5 second.

Both of these types of methods are considered one step methods rather than multi-step methods. The one step method of numerical solution is one that uses an algorithm as demonstrated earlier, where only one point is required and the step size, ΔT, is known. These methods have the advantage that they are self-starting methods whereas the multi-step methods require the step size ΔT and more than one point, and hence, in order to start these methods for initial value problems, a method utilizing a one step method must be used such as the Runge-Kutta method which has been described.

The advantages of using a multi-step numerical method for the solution of differential equations would be that there might be less chance for propagation of error in the solution and also that there could possibly be a more efficient use of the programming technique to utilize less computer time and less computer storage. However, these other methods were not investigated in this study. Other references could be used such as listed in the bibliography for further information relative to these more sophisticated methods of solution.

In the control system part of the simulation, it was noted that there were many integrators utilized in the simulation of that part of the plant. In that section, the method of integration which was used was one utilizing the trapezoidal rule, and with that process, accurate results could be obtained in the integration only by using a smaller time interval than was used with the Runge-Kutta technique. In the case of this simulation, a time step of 0.01 second was used in relation to the control system, and the trapezoidal rule was used as follows.
Consider for example the integrator from block 7 in the control system which gives the function DEMW2. Equation (6.12) is the method by which the integration is carried out.

$$\text{DEMW2} = \text{DEMW2} + \left( \frac{B1 + B1A}{2} \right) T_1 \left( \frac{1}{60} \right)$$  \hspace{1cm} (6.12)

$B1A = B1$

Figure 6.4, below, illustrates the basic technique in the integration that is involved with equation (6.12)

Fig. 6.4 Trapazoidal method of integration
CHAPTER 7

DESCRIPTION OF THE CONTROL SYSTEM

Basically there are two distinct systems in the simulation of the once-through supercritical steam generator. These two systems are classified into: (a) the physical process of generating the steam and using this generated steam for production of power, which is the desired purpose of the plant, and (b) the control system, which must function to control the process in a manner in which the plant can operate in a safe fashion and with a desirable level of performance.

The description of the process has been given in the previous sections and it is the purpose of this section to consider in detail the structure of the control system for the simulated plant, and to investigate the manner in which it was simulated for the digital computer. The approach to the analysis and simulation that was used, was to obtain the circuit descriptions of the various control phases and to simulate these controllers in individual blocks in order to preserve the separate functions that each block performs. The various blocks which will be considered in sequence are block #7 of the computer block diagram, which is the unit load demand development; block #8, which is the integrated boiler turbine master; block #9, which is the feedwater flow control; block #10, which is the firing rate calibration; block #11, the spray valve control system; block #12, the fuel flow control; block #13, the air flow calibration and air flow control; and finally, block #14, which is the reheat steam temperature control circuit. Each of these blocks serves as a link between the desired control and the process itself, and as will be seen later, there are certain links between the control system and the process itself, such as to make the control and the process work simultaneously in order to give the satisfactory operation for the plant.
In the consideration of a control system, the controllers which are used are commonly classified with four types according to the response which they give to an error signal. The response to a sudden change in an input error signal may be classified as:

1. Proportional
2. Reset
3. Rate action
4. On-off

The proportional action yields an output which is proportional to the error. The reset action yields an output which is proportional to the integral of the error which appears as a reset action of the set point. In terms of the subsequent discussion this will be referred to by a certain number of repeats per minute which will be described later. The rate action yields an output which is a function of the rate of change of the error.

The fourth classification of control mentioned is on-off control. This control is the type of control of which a familiar example would be that of a refrigerator or air compressor where the unit is either full-on or full-off and the response of the control follows this value and maintains an approximate steady state value, depending upon the controller action. There is a sustained oscillation of the controlled variable, such as temperature, which is characteristic of the on-off control device. The oscillation in the on-off controller is due to the natural set points at which the device is turned on or off by the control system.

In the development of control systems, the next logical approach is to continuously relate the controller action to the controlled variable or function and the easiest way to do this is to take a control signal that is directly proportional to the controlled variable. This control is then called proportional control. The definition of proportional control is that in which any change in the measured value of the controlled variable causes a control action to be taken that is proportional to the deviation from the set point.
In terms of using the proportional control, as mentioned earlier, an error signal is obtained which is the difference in the desired value of the controlled variable and the actual value of the controlled variable. Let this difference be called \( e \), which is the applied control variable and let the response be \( R \). Then an equation can be written which relates the control variable to the process. The equation is (7.1)

\[
R = - \alpha (e)
\]

(7.1)

where

- \( R \) is the controller response
- \( e \) is the deviation in the process variables between their set points and actual values
- \( \alpha \) is the proportionality ratio

This is illustrated in Figure 7.1.

![Fig. 7.1 Block diagram of proportional control](image)

The process will remain in a correct stable condition if it is adjusted so that the actual value is exactly equal to the set point value and yields 0 error, \( e \) and, hence, 0 proportional control, \( R \), at all times when the process is balanced. If the conditions of the plant or process change for that particular setting of the process regulator, which is \( PR \), the actual value, \( T_1 \), as seen in Figure 7.1, will deviate from the set point value and give rise to a continuous offset value, and hence as the process changes an adjustment must be made to
the set point or to the measured value from the process to give the right value for the process variable.

Describing the offset again, Figure 7.2 shows the proportional control offset. Consider the process set point, $T_o$, which takes a step down, at time, $t_1$, to a value $T_0'$, then the comparable process variable, $T_1$, will have a response as shown in Figure 7.2.

![Fig. 7.2 Proportional control offset](image)

The process regulator will have a particular position corresponding to the initial set point value and the proportional control unit will move the process regulator position, as shown in Figure 7.3. In this case, the process variable, $T_1$, is going down and let it be assumed that this requires the process regulator position to be of a higher value in the steady state. In order for the process regulator position to be at a higher value than it was originally, there must be a steady state error created, for if there were not a steady state error, then the process regulator position would return to its initial value (call this PR\(_1\)). In order to maintain the new process regulator position, PR\(_2\), there is an offset required, as shown in Figure 7.2.

In order to overcome the problem of the offset the integral control can be used. In the case of the integral control, the process regulator position is not
directly related by a fixed correlation to the actuating signal, or the error, \( e \) in this case. If an error does not exist between the set point value and the controlled variable of the process, no corrective action is made. As long as an error is available at the input to the integral controller, the process regulator is changing its position. Hence, it is impossible for the controlled variable, \( T \), to come to a steady state value at any other point than the desired set point since the process regulator position, governed by \( R \), will continue to change as long as an error input to the controller exists. Hence there can be no offset as was the case with the proportional control device. The relationship between the variables of the controller are given in equation (7.2).

\[
\frac{dR}{dt} = - \beta \cdot (e)
\]  

(7.2)

where

\( \beta = \) the integral control factor

Integrating equation (7.2) yields equation (7.3) which is

\[
R = - \beta \int e \, dt
\]

(7.3)

Hence, the correction given to the process is proportional to the integral of the deviation between the controlled variable and its set point value and hence the
name of the control is applicable, which is the integral controller. Figure 7.4 illustrates the block diagram of the integral controller.

![Block diagram of integral control](image)

Fig. 7.4 Block diagram of integral control

In Figure 7.5, the integral control action is seen.

![Integral control action](image)

Fig. 7.5 Integral control action
The system variables are the set point, $T_0$, which illustrates a step change at the time $t$. The process variable which is being controlled follows this according to the process equation and is indicated on the figure by $T_1$. The error that is shown is the difference between $T_0$ and $T_1$, and the process regulator position is shown which starts at its initial value and then is changed according to the integral of the error and multiplied by a constant. The process regulator goes to a new process position which is $PR_2$, from the original process regulator position, $PR_1$. It is seen that the offset which was obtained with the proportional control is not occurring in this control because the process regulator moves to its desired position to accommodate the change in the set point.

The next item to consider is the rate action, or the so-called derivative control. In the proportional type controllers the value of the process regulator position is a function, or is dependent exactly on the measured value of the controlled variable, which was considered to be $T_1$. With the rate controller and integral controller this dependency does not exist. The correction term depends on the rate or integral of the deviation. Since this type of controller does not use or require a set point it must be used in conjunction with other types of controllers. The equation which describes the rate controller is given as equation (7.4)

$$R = \gamma \frac{de}{dt}$$  \hspace{1cm} (7.4)

In Figure 7.6, the block diagram of the rate control device is shown. Figure 7.7 illustrates the rate controller response.

With the basic control elements, as described, they can be combined to provide a wide variety of different controlling schemes. Of particular interest is the proportional plus integral control which is used in the simulated plant and will be described at this point. Since there are limitations on the value of $\beta$ the integral control factor, it turns out that the correction of the process
Correction

$-\gamma \frac{d}{dt}$

Fig. 7.6 Block diagram of rate control

Fig. 7.7 Rate control action
regulator position is not rapidly available with simply integral control. Even though the controlled condition is finally brought back to the desired value, after a disturbance as described earlier, there may be a rather considerable delay in attaining this. This is a disadvantage in applications where there are large disturbances, or disturbances come frequently. For this reason, the integral control is used in conjunction with the proportional control in the simulated plant.

The component controls, that is the proportional control and the integral control, may be combined in the controlling instrument, in a separate instrument or in the process regulating unit itself. In any case, the integral control will continue to give a changing correction to the process regulator position which tends to bring the deviation between the set point and the control variable of the process to zero. Hence, the offset that was described in relation to the proportional control and, as a matter of fact, would be produced by the proportional control is no longer possible with the combined proportion plus integral control. The use of the proportional plus integral control is seen in block 8, the integrated boiler turbine master; block 12, the fuel flow control; block 13, the air flow control; and block 14, the reheat steam temperature control.

The primary variables which are controlled with the control system are given in the following list.

- Throttle valve position
- Feedwater flow
- Spray valve position
- Fuel flow
- Equivalent combustion gas damper position
- Air Flow*

The list of process variables which are fed back to the control system are:

- Throttle pressure  PO(1,16)
- Power output of the unit
- Output temperature of the primary superheater
Primary output temperature of the finishing superheater

Output temperature of the reheater

$O_2^*$

* These variables are described in the control system, however, since there is an assumed constant fuel to air ratio the actual control of the inlet vanes on the fans which provide the air for the combustion process is actually controlled in the simulation by virtue of the amount of fuel that is desired; hence, this signal is not actually sent to the process system.

There are traditionally two distinct types of control systems that are used in connection with the control of boilers, or steam generators. These two modes of operation are the turbine following mode and the boiler following mode. The turbine following mode involves the use of control valves to regulate the boiler pressure. The demand for a load change is fed directly to the boiler through its feedwater pumping and firing rate control. When the energy level of the boiler changes there is a subsequent change in the process variables such as pressure, temperature and enthalpy. The pressure control senses a change and changes the position of the turbine valve to adapt to the load change.

As mentioned earlier, the simulated plant has a once-through boiler which can respond very quickly when the load is changed due to the fact that it has little storage capacity to limit its action. This is due in turn to the fact that the water side of the boiler is directly connected to the steam side of the boiler and hence, there is no cushioning effect such as exists in the case of the drum type boiler where a continuous boiling takes place in the water and steam lines and superheaters are distinct entities in themselves. The response then of a once-through boiler is fast, but is still much slower than the pressure control action on the turbine valves. Hence this turbine following mode of operation is slow but is stable. Figure 7.8 illustrates the basic idea of the turbine following mode of control systems.

The majority of conventional drum type boilers are operated in what is known as the boiler following mode where changes in the power output of the unit are initiated by the turbine control valves. There is immediate response
to the action of the turbine valves with the mass flow rate of the steam changing immediately. Consequently, there is an insufficient quantity of steam available from the furnace and appropriate control signals must be given to the boiler to adjust its operation to catch up with the steam production that is required. In addition, the boiler must over, or under, pump and fire to account for the energy that has been borrowed from, or deposited in the boiler while reaching the new load level. The response of this type of control system is fast, but pressure and temperature deviations may be excessive when stored energy changes are large. The boiler control must be responsive to these changes and stable in order to withstand this mode of operation. The boiler following mode is illustrated in Figure 7.9

Considering the advantages and the disadvantages of the two modes of control; that is, the turbine following mode and the boiler following mode, and realizing the need for varying degrees of compromise between the desire for fast response to load changes and the desire for boiler safety and satisfactory
control of the steam conditions in the system, a logical extension of boiler control philosophy led to the development of a control system that combined the boiler following and turbine following schemes. The result was labeled "the integrated control system" which in essence takes the load demand signal and applies it to both the turbine and the boiler in parallel, but with the final valve position set by the pressure control. When this system is properly applied and adjusted it provides the advantages of a rapid initial load response associated with the boiler following system and the steady load increase and stability associated with the turbine following system.

In the integrated control system, the maximum use is made of the available boiler thermal storage. As will be seen in the subsequent discussion of computer blocks 8, 9, and 10, over or under feedwater pumping and firing rate transients are proportional to the generation error. Therefore, when the generation demand level is obtained this action is reduced to zero. For pressure control, the pressure deviation is programmed during transients from a function of the megawatt error, E1, and the available boiler thermal storage that is
transmitted through the control system. The basic idea of the integrated control system is illustrated in Figure 7.10.

Figure 7.10 Integrated control system

Figure 7.11 shows the integrated control system block diagram and indicates the various blocks and shows their interconnection in relation to one another.

The initiating signal is a load demand signal that is obtained from the automatic dispatching equipment for the entire power system. This dispatching signal is sent to the individual units throughout the entire power system. The particular level of load that is desired for the simulated plant comes in over a communication link in the form of pulses which have a variable width. These pulses, when integrated for a specific length of time, determine the power
level change which the unit should accomodate. The pulses are given to either raise or lower the generation level of the particular unit.

This load demand signal is then transferred through the control system to the unit load demand development block, shown as block 7. In the simulation,
the load demand signal is simply simulated as a step change as desired. After appropriate conditioning the signal leaves the unit load demand development as computer variable DEMW2 and passes into block 8, the integrated boiler turbine master block.

The variables which come from the process into block 8 are POWER and throttle pressure. The output of block 8 to the process is the throttle valve position, TVPXR, and the output to the other control system blocks are computer variable DEMW4.

This variable, DEMW4, goes to three distinct blocks of the control system. Specifically, the feedwater flow control, block 9, the firing rate calibration, block 10, and the reheat steam temperature control, block 14. The output of the feedwater flow control block to the process is the computer variable FWRE, which is feedwater flow.

The signal from block 8 passes also to the reheat steam temperature control and its output to the process is the computer variable, R12, which is a signal which indicates to the process the proportion of the combustion gases that should be passed over either the primary superheater or the reheater and in actuality the control moves dampers which divert the combustion gases over one surface or the other in a varying proportion. This has been described earlier in a previous section.

The signal from the integrated boiler turbine master also passes to the firing rate calibration, block 10. Input signals to this block from the process are computer variables TO(1, 9) and TO(1, 10) which are the output temperatures of the primary and finishing superheaters, respectively.

A signal from the firing rate calibration, as computer variable DERER1 passes into block 11, the spray valve control. The output of the spray valve control passes to the process in terms of the computer variable SVP, which is the spray valve position.

Also coming out of block 10 is a signal labeled FRDEMC which passes to the fuel flow control, block 12, and the air flow calibration and control, block 13. The output of the fuel flow control is a computer variable WNGTH
which indicates the amount of natural gas that is flowing into the unit to be burned in the combustion process.

The output signal from the air flow calibration and control goes to the process in computer variable AFLOW. As mentioned earlier, this variable does not actually go to the process in the simulation since a constant fuel to air ratio was assumed.

There are two different types of diagrams which are very useful in the detailed description of the control system. Since the simulated plant has an analog control system, it was desired to include the block diagram form of the control system and in addition to include the analog computer presentation for the control system. The first that will be presented will be the analog computer control configuration, and this will be discussed in detail. Each block which has been mentioned in connection with Figure 7.11 will be seen to have many different signals and with various trimmings, limits, proportional gains, resets and other conditioning, as mentioned earlier. Hence the interconnections between the various blocks will be found to be more numerous than are shown in the simplified block diagram just discussed.

Following a detailed description of each block, as outlined, the other form of control system representation will be presented. It will be seen that it will have the same essential features as the block diagram has that is described, but it gives a different point of view to the descriptions.

As has already been observed there are a great number of different variables involved in a simulation of this magnitude, and the two distinct types of variables will be listed in separate tables. Since the immediate discussion deals with the control system, the variables which are used in the control system are listed below in Table 7.1. In another section a similar list of variables is given that deals primarily with the process system. In these two tables, the variables are described and their appropriate units are given. These units are those which correspond to the real process. It is noted that in the actual analog control system, voltages are used to represent the process variables (voltage levels between +10v and -10v are utilized). Hence, when
<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A Parameter for equation of boiler feed pump(rpm)</td>
<td></td>
</tr>
<tr>
<td>AFC Gate provides for the air flow control to be on (1.) or off (0.) the system</td>
<td></td>
</tr>
<tr>
<td>AFI Integrated air flow error (K-lb/hr)</td>
<td></td>
</tr>
<tr>
<td>AFLow Air flow (K-lb/hr)</td>
<td></td>
</tr>
<tr>
<td>AFD Air flow demand (K-lb/hr)</td>
<td></td>
</tr>
<tr>
<td>AL Amplified principle megawatt error (psia)</td>
<td></td>
</tr>
<tr>
<td>AL1 Converted calibrated megawatt error (psia)</td>
<td></td>
</tr>
<tr>
<td>AL1U Calibrated megawatt error (MW)</td>
<td></td>
</tr>
<tr>
<td>B Parameter for equation of boiler feed pump (K-lb/hr)</td>
<td></td>
</tr>
<tr>
<td>BB Amplified megawatt error (MW)</td>
<td></td>
</tr>
<tr>
<td>B1 Limited amplified megawatt error (MW)</td>
<td></td>
</tr>
<tr>
<td>B1A Dummy variable</td>
<td></td>
</tr>
<tr>
<td>C Parameter for equation of boiler feed pump (psia)</td>
<td></td>
</tr>
<tr>
<td>CS Gate provides for the control system to be on or off the system</td>
<td></td>
</tr>
<tr>
<td>C10 Amplified error (E25) (°F)</td>
<td></td>
</tr>
<tr>
<td>DDEC Adjusted boiler demand in reheat steam temperature control (MW)</td>
<td></td>
</tr>
<tr>
<td>DEMW1 External megawatt demand (MW)</td>
<td></td>
</tr>
<tr>
<td>DEMW2 Megawatt demand corrected (MW)</td>
<td></td>
</tr>
<tr>
<td>DEMW2A Calibrated megawatt demand (MW)</td>
<td></td>
</tr>
<tr>
<td>DEMW4 TRIM megawatt demand (MW)</td>
<td></td>
</tr>
<tr>
<td>DERER1 Primary and final superheater steam temperature error (°F)</td>
<td></td>
</tr>
<tr>
<td>DERER2 Calibrated primary and final superheater steam temperature and megawatt error (°F)</td>
<td></td>
</tr>
<tr>
<td>DERERT Calibrated primary and final superheater steam temperature error (°F)</td>
<td></td>
</tr>
<tr>
<td>DERT Derivative with respect to time of primary superheater steam temperature (°F/sec)</td>
<td></td>
</tr>
<tr>
<td>E Megawatt error (MW)</td>
<td></td>
</tr>
<tr>
<td>E1 Principle megawatt error (MW)</td>
<td></td>
</tr>
<tr>
<td>E11 Integrated megawatt error (MW-sec)</td>
<td></td>
</tr>
<tr>
<td>E5 Delayed feedwater flow error (K-lb/hr)</td>
<td></td>
</tr>
<tr>
<td>E5A Amplified delay feedwater flow error (K-lb/hr)</td>
<td></td>
</tr>
<tr>
<td>E5AA Dummy variable</td>
<td></td>
</tr>
<tr>
<td>E6 Feedwater flow error (K-lb/hr)</td>
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TABLE 7.1 Continued

<table>
<thead>
<tr>
<th>Code</th>
<th>Description</th>
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</thead>
<tbody>
<tr>
<td>E6A</td>
<td>Dummy variable</td>
</tr>
<tr>
<td>E13</td>
<td>Final superheater steam temperature error (°F)</td>
</tr>
<tr>
<td>E13G</td>
<td>Amplified final superheater steam temperature error (°F)</td>
</tr>
<tr>
<td>E13GA</td>
<td>Dummy variable</td>
</tr>
<tr>
<td>E13GI</td>
<td>Integrated final superheater steam temperature (°F-hr)</td>
</tr>
<tr>
<td>E20</td>
<td>Fuel flow error (K-ft(^3)/hr)</td>
</tr>
<tr>
<td>E20A</td>
<td>Dummy variable</td>
</tr>
<tr>
<td>E25</td>
<td>Error between final superheater steam temperature and reheat steam temperature (°F)</td>
</tr>
<tr>
<td>E25A</td>
<td>Dummy variable</td>
</tr>
<tr>
<td>E50</td>
<td>Air flow error (K-lb/hr)</td>
</tr>
<tr>
<td>E50A</td>
<td>Dummy variable</td>
</tr>
<tr>
<td>EGO</td>
<td>Air flow regulator (K-lb/hr)</td>
</tr>
<tr>
<td>E60A</td>
<td>Dummy variable</td>
</tr>
<tr>
<td>ESP</td>
<td>Spray valve position error (p. u.)</td>
</tr>
<tr>
<td>ESPA</td>
<td>Dummy variable</td>
</tr>
<tr>
<td>ET8</td>
<td>Dummy variable</td>
</tr>
<tr>
<td>E20FA</td>
<td>Dummy variable</td>
</tr>
<tr>
<td>EV15</td>
<td>Throttle valve position error (p. u.)</td>
</tr>
<tr>
<td>EV15A</td>
<td>Dummy variable</td>
</tr>
<tr>
<td>FFC</td>
<td>Gate provides for the fuel flow control to be on or off the system</td>
</tr>
<tr>
<td>FFDM</td>
<td>Fuel flow demand (K-ft(^3)/hr)</td>
</tr>
<tr>
<td>FLFU1</td>
<td>Integrated fuel flow error (K-ft(^3)/hr)</td>
</tr>
<tr>
<td>FRDEM2</td>
<td>Same as DEMW4</td>
</tr>
<tr>
<td>FRDEM3</td>
<td>Adjusted boiler demand in firing rate calibration loop (MW)</td>
</tr>
<tr>
<td>FRDEMC</td>
<td>Calibrated firing rate calibration demand (MW)</td>
</tr>
<tr>
<td>FRTRIM</td>
<td>Calibrated air flow demand (MW)</td>
</tr>
<tr>
<td>FWC</td>
<td>Gate provides for the feedwater flow control to be on or off the system</td>
</tr>
<tr>
<td>FWCD</td>
<td>Feedwater flow demand (K-lb/hr)</td>
</tr>
<tr>
<td>FWD1</td>
<td>Delayed feedwater flow error (K-lb/hr)</td>
</tr>
<tr>
<td>FWD2</td>
<td>Integrated feedwater flow error (rpm)</td>
</tr>
<tr>
<td>FWRE</td>
<td>Feedwater flow (K-lb/hr)</td>
</tr>
<tr>
<td>FWRE1</td>
<td>Parameter for equation of boiler feed pump (K-lb/hr)</td>
</tr>
<tr>
<td>FWVF</td>
<td>Boiler feed pump speed error (rpm)</td>
</tr>
<tr>
<td>Variable</td>
<td>Description</td>
</tr>
<tr>
<td>----------</td>
<td>-------------</td>
</tr>
<tr>
<td>FWVFA</td>
<td>Dummy variable</td>
</tr>
<tr>
<td>IBTM</td>
<td>Gate provides for the integrated boiler turbine master to be on or off the system</td>
</tr>
<tr>
<td>O2</td>
<td>Oxygen (constant)</td>
</tr>
<tr>
<td>O2SP</td>
<td>Oxygen set point (constant)</td>
</tr>
<tr>
<td>OXE</td>
<td>Oxygen error (constant)</td>
</tr>
<tr>
<td>OXEA</td>
<td>Dummy variable</td>
</tr>
<tr>
<td>OXEI</td>
<td>Integrated oxygen (constant)</td>
</tr>
<tr>
<td>P50</td>
<td>Amplified air flow error (K-lb/hr)</td>
</tr>
<tr>
<td>PBFP</td>
<td>Boiler feed pump pressure (psia)</td>
</tr>
<tr>
<td>PBFP1</td>
<td>Parameter for equation of boiler feed pump (psia)</td>
</tr>
<tr>
<td>PC</td>
<td>Amplified throttle pressure error (psia)</td>
</tr>
<tr>
<td>PE</td>
<td>Throttle pressure error (psia)</td>
</tr>
<tr>
<td>PER</td>
<td>Converted throttle pressure error (psia)</td>
</tr>
<tr>
<td>PEROXE</td>
<td>Percent oxygen error</td>
</tr>
<tr>
<td>PRO5</td>
<td>Fuel flow error amplified (K-ft$^3$/hr)</td>
</tr>
<tr>
<td>PSPC</td>
<td>Adjusted throttle set point (psia)</td>
</tr>
<tr>
<td>PSPE</td>
<td>Throttle pressure error (psia)</td>
</tr>
<tr>
<td>PSPEA</td>
<td>Dummy variable</td>
</tr>
<tr>
<td>R12</td>
<td>Damper control signal (p.u.)</td>
</tr>
<tr>
<td>RHSTC</td>
<td>Gate provides for the reheat steam temperature control (damper control) to be on or off the system</td>
</tr>
<tr>
<td>RPM</td>
<td>Speed of boiler feed pump (rpm)</td>
</tr>
<tr>
<td>RPM1</td>
<td>Parameter for equation of boiler feed pump (rpm)</td>
</tr>
<tr>
<td>SAF</td>
<td>Adjusted air flow error (K-lb/hr)</td>
</tr>
<tr>
<td>SCI</td>
<td>Integrated error (E25) (°F-sec)</td>
</tr>
<tr>
<td>SP1</td>
<td>Parameter for equation of boiler feed pump (K-lb/hr)</td>
</tr>
<tr>
<td>SP2</td>
<td>Parameter for equation of boiler feed pump (psia)</td>
</tr>
<tr>
<td>SSC1</td>
<td>Adjusted error (E25) (MW)</td>
</tr>
<tr>
<td>SUFL</td>
<td>Adjusted fuel flow error, applied to controller (K-ft$^3$/hr)</td>
</tr>
<tr>
<td>SVC</td>
<td>Gate, provides for the spray valve control to be on or off the system</td>
</tr>
<tr>
<td>SVP</td>
<td>Negative spray valve position (p.u.)</td>
</tr>
<tr>
<td>SVP1</td>
<td>Limited spray valve position (p.u.)</td>
</tr>
<tr>
<td>T1</td>
<td>Increment of time for control system variables computation (sec)</td>
</tr>
</tbody>
</table>
TABLE 7.1 Continued

<table>
<thead>
<tr>
<th>Term</th>
<th>Description</th>
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</thead>
<tbody>
<tr>
<td>T2</td>
<td>Increment of time for interconnection of control system variables with process variables (sec)</td>
</tr>
<tr>
<td>TOE19</td>
<td>Dummy variable (used in Bl. 4, 10 and 17)</td>
</tr>
<tr>
<td>TRIM1</td>
<td>Linear function of integrated megawatt error (p. u.)</td>
</tr>
<tr>
<td>TRIM2</td>
<td>Linear function of net load (p. u.)</td>
</tr>
<tr>
<td>TRIM3</td>
<td>Linear function of boiler demand (p. u.)</td>
</tr>
<tr>
<td>TRIM4</td>
<td>Linear function of final superheater steam temperature (p. u.)</td>
</tr>
<tr>
<td>TRIM5</td>
<td>Linear function of oxygen error (p. u.)</td>
</tr>
<tr>
<td>TVPX</td>
<td>Throttle valve position amplified (p. u.)</td>
</tr>
<tr>
<td>TVPXR</td>
<td>Corrected throttle valve position (p. u.)</td>
</tr>
<tr>
<td>VIC</td>
<td>Integrated throttle pressure error (psia-sec)</td>
</tr>
<tr>
<td>VPS</td>
<td>Modified throttle pressure error (p. u.)</td>
</tr>
<tr>
<td>WNGTH</td>
<td>Fuel flow (K-ft³/hr)</td>
</tr>
<tr>
<td>WSTH</td>
<td>Spray flow (K-lb/hr)</td>
</tr>
<tr>
<td>X</td>
<td>Parameter for equation of boiler feed pump (K-lb/hr)</td>
</tr>
</tbody>
</table>

Comparing actual schematic diagrams of the control system with those of the simulated system there is not found a one to one correspondence between all the blocks.

The plant is controlled from a central control room shown in Figure 7.12. The control panel for the simulated plant is shown in Figure 7.13.

The control cabinets housing the electronic analog controller are shown in Figure 7.14.

As mentioned earlier, the electronic analog computer sends control signals to the various controlled process variables. One control signal goes to the forced draft fan inlet vanes to adjust the amount of air that is forced through the unit in accordance with proper combustion of the fuel. The control linkage and vanes are shown in Figure 7.15.

A view of the controller cabinet, linkage and forced draft fans are shown in Figure 7.16 and Figure 7.17.

In this simulation an attempt has been made to simulate the same devices and to use values of constants which are used in the actual plant. Several
Fig. 7.12 Plant control room

Fig. 7.13 Control panel for unit number 3
Fig. 7.14 Analog control of elements.

Fig. 7.15 Control linkage operating intake vanes on forced draft fan.
Fig. 7.16 Controller and linkage for forced draft fans

Fig. 7.17 Forced draft fans
parts have been simplified considerably, and when information was not available certain assumptions were made in regard to the control system.

The control system which is used in the plant is of the analog type and, as mentioned, analog diagrams will be used to illustrate the method of the digital simulation. As mentioned earlier, the setting of the integrators in units of repeats per minute is to be described here. The basic time increment in the simulated control system, $\Delta T$, is labeled $T_1$ for the computer program.

Where this value of $\Delta T$ is used in the calculation of the integrals it is also used for other computations where the $\Delta T$ is present, and will be, of course, the same time. This is due to the fact that internal loops are not used in the computation of the control variables. Each control system variable is computed at the end of a time interval equal to $T_1$ after which the time is increased by this interval and the complete process is repeated until the desired time of computations is reached. The method by which the speed of integration is varied is by multiplying $T_1$ by some constant. This, of course, is equivalent to changing the input to the integrator. If this time interval is not changed in the integrations, the resulting settings, repeats per minute, will be at 60 repeats per minute for a constant input. For a slower, or faster, integration rate, the time increment, $\Delta T$, should be multiplied by the factor $X$ divided by 60, where $X$ is the number of repeats per minute desired. The division by 60 converts the setting to seconds, which is the unit of time used in the simulation. As an example of the concept of repeats per minute, consider a constant input of one per unit that is integrated for one minute.

$$
\int_{t=0}^{t=1 \text{ min}} E_i \, dt = 60 \text{ sec} \quad \frac{\text{d}T}{\text{d}t} = k \quad t = 0 \quad \frac{\text{d}T}{\text{d}t} = k \frac{T}{60} = K 60 \\
t = 0 \quad 0 \quad t = 0
$$

Now, if the integral had been multiplied by $\frac{3}{60}$ or $\frac{3 \text{d}T}{60}$ had been used then the resulting value of output voltage would be $(3/60) (K) (60-0) = 3K$. Hence, for this example with 3 repeats per minute and one per unit input to the integrator there is three per unit output after 1 minute of integration.
The first block which will be considered in detail is block 7, the unit load demand development, which is shown in Figure 7.18. As has been mentioned earlier the pulse sent from the load dispatcher is integrated with respect to time to obtain the actual value of change in demand. If the change in demand is applied manually it is not necessary to integrate. In either case, after the new demand has been established to the control system the change will appear as a step function. In the simulation, the step input is given with DEMW1. The complete step change desired in megawatts is not applied to the unit, but rather a signal which is gradually increasing in the form of a modified ramp function is applied.

This signal conditioning is carried out by obtaining a megawatt error, E, between the demanded megawatts and the existing megawatt demand that is presently going to the unit. This megawatt error is multiplied by a gain of 3 which results in a faster response for small changes in load. Then the signal is limited to a value of 35 megawatts with the limiter and then is multiplied by an adjustable percentage gain, G2, which is presently set at 1, but can be set at other values to permit a reduction in the rate of change of the demanded signal.

![Diagram](image-url)
The output of the gain, G2, which is B1, is then integrated with a rate of one repeat per minute and this signal, which is then DEMW2, is sent to block 8, the integrated boiler turbine master.

If a step change is introduced to the control system that is greater than 35 megawatts there will be a ramp change in the output signal until the difference in the output signal and the desired demand is less than 35 megawatts, at which time the demand signal will follow an exponential curve as shown in Figure 7.19. If the change in the demand was 35 megawatts or greater, the output rate of change would be 35 megawatts per minute. Since this is higher than normally tolerated by the operating guidelines, an adjustment of gain G2 may be made as desired to limit this rate.

![Exponential curve](image)

**Fig. 7.19** Output diagram of unit load demand development

Consider next the integrated boiler turbine master, which is block 8 of the computer program, and is shown in detail in Figure 7.20. This block receives the load demand, DEMW2, which is the primary input signal and converts it into: (a) a boiler demand, DEMW4, and (b) a turbine demand which is sent to the process system in the form of a throttle valve position setting, TVPXR.

As mentioned earlier, the turbine demand is to provide the desired generation in a relatively short time. The turbine is used to provide the initial response since the boiler cannot change the steam output as quickly and still
Fig. 7.20 Integrated boiler turbine master, block 8
keep the throttle pressure constant. The boiler demand is fed to the process in terms of feedwater flow, fuel flow, and air flow signals. The important process variables which are fed into this system are the actual power generated, POWER, and the actual throttle pressure, PO(1,16).

Looking first at the turbine demand section of block 8, the megawatt error, E1, is obtained by taking the difference in the actual power generated from the process and the demanded power from block 7. This error is multiplied by a gain, G3, to give a variable, AL, which is then limited to plus and minus 21 megawatts. This variable is then modified through the action of a multiplier where a trimmed value, which, in this case, is labeled TRIM2, is multiplied by AL to give the output signal, AL1U.

The value of the trimming signals throughout the simulated control system and the actual control system are obtained by experience and experiment. The values of the trims are generally functions of the power output and in each case an equation is written which relates the trimming value to the variable under consideration.

In this case, the TRIM2 is a value which lies between 1 and 2.5. This number is then multiplied by AL which is the megawatt error limited. When the load on the unit is less than or equal to 175 megawatts, a value of 1 is used for TRIM2, and when the load on the unit would exceed 700 megawatts a value of 2.5 is used for TRIM2. Between these two extremes, the value of TRIM2 is given in equation (7.5).

\[
TRIM2 = \left( \frac{1.5}{525} \right) (POWER) + 0.5
\]  

(7.5)

The value of AL1U is then multiplied by a gain G5. This resulting signal, the calibrated megawatt error, is AL1. AL1 = AL1U (G5)

Now G5 is given as 50/7 and this value is computed in the following manner. The control system operates on certain voltage levels, and the voltage levels that are used in this control system are from -10 volts to +10 volts. The ranges which are used correspond to 0 megawatts for -10 volts, and 700 megawatts for +10 volts, hence, the megawatt error rate is equal to \((700/20) = 35\)
megawatts per volt. In a similar fashion, the pressure range varies from 0 psia corresponding to -10 volts and 5,000 psia corresponding to +10 volts. Hence, the pressure error rate is 5,000 psia divided by 20 volts or 250 psia/volt. Hence, the value used for $G_5$ = the pressure error rate divided by the megawatt error rate which equals 250 psia/volt divided by 35 megawatts per volt which equals 50/7 psia/megawatt. Hence, when $A_{L1U}$ in units of megawatts is multiplied by 50/7 in units of psia/megawatt the resultant value, $A_{L1}$, is in the proper units of psia to be added to the throttle pressure set point.

It is seen also that the gain, $G_6$, going to the boiler demand is given, which is simply the inverse of this value; in other words $G_6 = 7/50$ megawatts/psia and it will be seen that the pressure error is transferred to the proper units of megawatts when added to the modified power demand signal before it is passed on to the other blocks.

The value, $A_{L1}$, is then converted to an equivalent throttle pressure demand by adding it to the throttle pressure set point which, for the simulated unit, is taken as 3,500 psia. The resulting value, labeled PSPC, is then compared with and the difference taken between it and the actual throttle pressure labeled PO(1,16) from the process. The resulting variable, PSPE, is then used as the input of a proportional plus integrator unit, or a proportional plus reset unit. The proportional plus integral takes the throttle pressure error and multiplies it by a gain, $G_4$, which is equal to 5 and, at the same time in parallel, integrates it with an integration setting of 3 repeats per minute. The sum of these two operations is taken yielding the variable VPS, which is the mnemonic name for valve position setting.

The position of the throttle valve was simulated using an integrator with a unity feedback loop as is shown by the variable TVPX in Figure 7.20. A steady state value of TVPX was chosen as for 1,000 units. This corresponds to a full load position. Using a smaller value than this for the steady state led to unstable operation of the system due to the fact that the throttle pressure deviation was relatively large compared to some of the initial values that were tested, such as 1, which is the value that is used for the process variable.
TVPXR, which represents the actual throttle position, as has been discussed in that section. The integrator with output TVPX, has an initial condition corresponding to the full load condition of 1,000 in order that a zero error signal is given to the simulated position demand integrator. The value of G7 is seen to be \( \frac{1}{1000} \) in order to provide the proper value for TVPXR, the actual valve opening in per unit. This variable then is sent to the process block 17.

Now consider the other path in the integrated boiler turbine master, the boiler demand path. This path ultimately sends a signal to the firing rate calibration, feedwater control, and reheat steam temperature control. The demand from the unit load demand development is again modified by multiplying it by TRIM1, another trimming value. This trim is obtained from the megawatt error between the actual power output of the unit and the demanded power from the unit. This error, \( E_1 \), is integrated at a rate of \( \frac{1}{5} \) repeats per minute and is limited to \( \pm 350 \) megawatts and is then converted to TRIM1 with the following equation.

\[
TRIM1 = \left( \frac{0.2}{350} E_1 \right) + 1
\]  

(7.6)

Now this trim multiplied times DEMW2 yields DEMW2A. Figure 7.21 illustrates the effect of TRIM1 as a function of the integrated megawatt error.

![Diagram of TRIM1 - integrated MW Error](image)
The megawatt demand, DEMW2A, is then converted into boiler demand by adding to it any throttle pressure deviation, PER, which is converted, as mentioned earlier, from pressure units to megawatt units by gain G6. The megawatt error is reduced to 0 during any steady state operation.

The next block to be considered is the feedwater flow control, block 9 and is shown in Figure 7.22. This block responds primarily to the boiler demand for water flow. The response is coordinated with the firing rate, that is fuel flow, to minimize the interaction between the steam pressure control and the steam temperature control. This is accomplished by adding to the boiler demand, DEMW4, from the block 8, a feedwater trim, labeled DERER2, from the firing rate calibration block 10. This trim is seen to be a function of load error, E1, primary superheater, outlet temperature, TO(1,9) and the final superheater outlet temperature error, which is a set point temperature minus TO(1,10). A trimmed feedwater demand is then obtained by use of the equation shown in the boiler feed pump section which related water flow to megawatts. This equation modified by the trim is given by equation (7.7).

\[ \text{FWCD} = \left( \frac{456}{55} \times (\text{DEMW4}) - (\text{DERER2}) \times 0.005 \right) - 660 \]  

(7.7)

Now, notice that 456 divided by 55 is 8.3 which is the slope used in the boiler feed pump section. The trimmed feedwater demand FWCD is then delayed exponentially in order to coordinate rate of change with the rest of the system. This value of FWCD is the resulting feedwater flow after steady state conditions are restored, after a change.

Figure 7.23 illustrates the exponential delay unit that is designed for use in other blocks in a similar way. In the case of the firing rate calibration the delay needed was very small in practice so it was taken off the circuit.

The delayed value of feedwater demand, FWD1, is then compared with the actual flow, FWRE, of the boiler feed pump which is described in the boiler feed pump section and this yields the feedwater error, E6. This error is then integrated to produce a new speed error for the boiler feed pumps which will
Fig. 7.22 Feedwater flow control, block 9
then adjust the characteristics of the pump to provide the proper value for the system operation.

Block 10 is shown in Figure 7.24 and is the firing rate calibration section. Here the demand for fuel is taken from the integrated boiler turbine master as DEMW4, and will eventually leave the firing rate calibration block to be passed on to the fuel flow control and the air flow calibration and control blocks as FRDEMC, a calibrated firing rate calibration demand.

The firing rate calibration block modifies the relationship between the firing rate and the feedwater flow in order to maintain the desired steam temperature output of the finishing superheater. The ratio of feedwater flow to firing rate is modified to produce a permanent correction of the steam temperature. The control serves the purpose of providing the steady state calibration of the firing rate demand with respect to the boiler demand and it provides the required transient firing rate corrections in order to balance the energy in the steam generator, for the load under consideration.

The temperature output of the finishing superheater, TO(1,10), is subtracted from the temperature set point for this device, which in this simulation is considered to be 1,017.391. This difference is multiplied by the gain, G9, which is considered to be 1, and this yields the computer variable, E13G. This resultant error signal is passed to two different paths. In the first case the error, E13G, is taken and from it is subtracted the derivative, or rate of change, of the primary superheater output temperature, which is TO(1,9), with the resulting signal DERER1. This signal is then multiplied by a trimming function, TRIM3, which is a linear function of the boiler demand and is biased
Fig. 7.24 Firing rate calibration, block 10
to obtain a new value of error which will be equal to the error times 1.0 if the
load demand is 0, and will be the error times 2.5 if the load demand is 700
megawatts. This equation is given as equation (7.8).

\[ \text{TRIM3} = \left( \frac{1.5}{700} \right) (\text{DEMWT}) + 1 \]  \hspace{1cm} (7.8)

The calibrated error sum is then given by computer variable DERERT and
this is equation (7.9).

\[ \text{DERERT} = \text{DERER1} \times (\text{TRIM3}) \]  \hspace{1cm} (7.9)

This calibrated error, DERERT, is added to the load error from block 8. The
load error is E1 and the resulting variable is DERER2. This variable, which
is a calibrated temperature error, is added to the boiler demand to obtain a
correction in the firing rate demand for transient conditions. At a steady state
value these errors are 0 and hence, would not change the firing rate demand
signal.

The signal, DERER2, is also fed to the feedwater flow control, block 9,
and was described in that section. That gave an interaction between the firing
rate and the feedwater flow control. The practical result of this is that when
the firing rate demand is causing the temperatures of the superheated steam to
be above their normal values, a negative steam temperature error will be sub-
tracted from the feedwater flow demand which will produce an increase in the
water flow as is seen in equation (7.7) of the discussion given of block 9. Also,
if the temperatures are lower than the set point values, the feedwater control
will cause a decrease in feedwater, as seen by equation (7.7) and the firing
rate will increase as required until the two blocks, the fuel flow and feedwater
flow, adjust their respective process variables to obtain the set value of temp-
erature.

The boiler demand, FRDEM2, is then added to the total errors and the
signal which results is FRDEM3. This is modified or calibrated by multiplying
it by a function of the limited integrated temperature error of the finishing
superheater. This trimming signal is TRIM4 and is given by equation (7.10).
Following the block diagram through to TRIM4, it is seen that the temperature error signal, E13G, is integrated at a rate of 0.5 repeats per minute yielding the signal E13GI. This value is limited to ±120°F and is then converted to a linear function with a range from 0.9 to 1.1 using equation (7.10) which is:

\[
TRIM4 = \frac{0.1}{120} (E13GI) + 1.0
\]  

(7.10)

The signal, FRDEMC, which is a calibrated firing rate calibration demand, is obtained in equation (7.11) which is:

\[
FRDEMC = FRDEM3 \times (TRIM4)
\]  

(7.11)

It is this signal which is then sent to the fuel flow control and the air flow calibration and control blocks.

The spray valve control is illustrated in Figure 7.25. This section, as has been described earlier, serves to give an immediate, but not permanent, change to the output temperature of the finishing superheater. Since its purpose is to sense the temperature errors and help control the steam temperature errors as the steam enters the turbine, it logically derives its input signal from that portion of the control system where this comparison between the actual temperature and the desired, or set point temperature has been measured, and this has been just described in the firing rate calibration section, block 10.

The signal to the spray valve control is taken off the point where DERER1 is available which is the temperature error signal. This temperature error is utilized in the simulation by passing it through an integrator with unity feedback whose output represents the value of the spray valve position. Consider the example of a finishing superheater output temperature which is above the set point value. The value of E13G is then a negative quantity which will yield a negative quantity for DERER1 from the firing rate calibration to the spray valve control system. This then yields a negative error for ESP which causes the integrator, which is set at 5 repeats per minute to decrease. Hence, the spray valve position variable, SVP, would decrease. A computer variable,
Fig. 7.25 Spray valve control, block 11

Fig. 7.26 Fuel flow control, block 12
SVP1, which equals -SVP is considered the simulated value for the spray valve position and its value will increase giving a larger amount of flow than the initial set point. Its magnitude is limited to between 0 and 100, and as mentioned in the process this corresponds to thousands of pounds of spray water flow. This variable is then passed to the process, block 19.

The next control block which is considered is the fuel flow control. This is block 12 and is shown in Figure 7.26. The fuel flow control produces the appropriate fuel flow corresponding to the desired plant output and uses the calibrated boiler demand, FRDEMC, as obtained from the firing rate calibration block. For a certain demanded power output, there is required a certain value of fuel flow. This relationship is given in equation (7.12) where

$$\text{FFDM} = \left[ \frac{120}{13} \right] (\text{FRDEMC})^{-170}$$

(7.12)

This equation is obtained from the data which was obtained in the actual plant tests, run #9, as has been described in the combustion section and also in detail in a latter section dealing with the test runs. In the combustion section, it was shown that the flow of natural gas, WNG, was linearly proportional to the power output of the unit, specifically,

$$\text{WNG} = \left( \frac{120,000}{13} \right) (\text{POWER})^{170,000 \text{ cu.ft/hr.}}$$

(7.13)

The control system variable which corresponds to the flow of natural gas is FFDM and the control variable corresponding to power, in this particular case, is FRDEMC. Since the control system is simulated using units of thousands of cubic feet per hour, rather than cubic feet per hour, the equation given in the combustion section is divided by 1000 with the result given as equation (7.12). The actual value of WNGTH after the steady state condition is reached is the same as FFDM and is obtained from this equation. This demand signal for fuel flow is then taken and from it is subtracted the simulated fuel flow to obtain the fuel flow error. The fuel flow error, E20, is then multiplied by the gain G10 and, in parallel, integrated at a rate of 3 repeats per minute to
yield the resulting value \( SUFL = PR05 + FLFU1 \). This output variable is then used as the input to the simulated fuel control valve which is an integrator with a reset value of 1 repeat per minute with unity feedback as seen in Figure 7.26. The output of this integrator is then the simulated fuel flow in thousands of cubic feet per hour and is converted to the process variable, WNG, by multiplying \( WNGTH \times G11 \) where \( G1 = 1000 \) and then this value, which is WNG, or fuel flow, is then passed to the process equations, block 17, where it is used to compute the heat fluxes into the various lumps.

The air flow calibration and air flow control, block 13, is shown in Figure 7.27. The air flow calibration adds an error to the firing rate demand which comes from block 10, the firing rate calibration block. This error which is introduced to the demand is done so through the TRIM5, which will be later seen in equation (7.15). The air flow calibration and air flow control system are included in this discussion for completeness, however, as mentioned earlier, the resulting variable which is simulated air flow in the unit, is not actually used since it has been assumed there is a constant fuel to air ratio.

The TRIM5, which is multiplied times the calibrated firing rate demand signal from block 10 is obtained as follows. The oxygen, \( O_2 \), in the combustion gas which is leaving the stack is subtracted from the oxygen set point yielding the computer variable \( OXE \), which is oxygen error. This oxygen error is integrated at a reset rate of 60 repeats per minute, yielding the variable \( OXEI \). The percent oxygen error which is in essence the excess \( O_2 \) of the plant is obtained by taking the ratio of \( OXEI \) to oxygen set point times 100. That equation is

\[
\text{PEROXE} = \frac{OXE1}{O2SP} \times 100
\]  

(7.14)

This value of percent excess oxygen is limited to \( \pm 5 \). In the actual plant, it should be realized that it always should be positive since it would not be desirable to have a negative excess oxygen or incomplete combustion would occur.
Fig. 7.27  Air flow calibration and air flow control, block 13
This limited oxygen error is then related to TRIM5 through equation (7.15) which is

\[ \text{TRIM5} = \left( -\frac{0.2}{5} \right) (\text{PEROXE}) + 1 \]  

(7.15)

This TRIM5 is then multiplied by FRDEMC to yield the firing rate calibration variable, FRTRIM. The demanded air flow, computer variable AFLOWD, is related to the adjusted megawatt demand, FRTRIM, through the equation (7.16) which is

\[ \text{AFLOWD} = \left( \frac{120}{13} \right) (\text{FRTRIM}) + 30 \]  

(7.16)

These variables are the same variables which were described in the combustion section in the discussion of air flow and equation (7.16) is obtained from the actual plant data which was obtained from test run 9 and is noted here to be in units of thousands of pounds per hour rather than the pounds per hour as was given in an earlier section. The air flow demand is then taken, and from it is subtracted the simulated air flow yielding the air flow error, E50, which is then applied to a proportional plus integral controller with a gain, G12 = 5, and an integral reset rate of 4 repeats per minute. This unit yields an output variable SAF which is then used as the input to the simulated air flow device which is an integrator with a 4 repeat per minute reset rate with unity feedback to yield the computer variable AFLOW. This variable is then converted into the process units with the following equation:

\[ \text{AFLO} = \text{AFLOW} (1000) \]  

(7.17)

This process variable then could be connected to the process in a manner similar to the other blocks if the process used this variable, but since a constant fuel air ratio is used only the fuel flow is necessary to simulate the combustion equations.

The reheat steam temperature control is block 14 and is illustrated in Figure 7.28. The purpose of the reheat steam temperature control is to send a signal to the process which will indicate a damper setting so that the
combustion gases may be apportioned between the primary superheater and the reheater, as has been described in the process section. Even though the primary superheater and reheater are receiving the ultimate combustion gas changes, a temperature difference is taken in this control section between the reheater output steam temperature, $TO(1,12)$, and the finishing superheater output temperature, $TO(1,10)$. The temperature error is,

$$E_{25} = TO(1,12) - TO(1,10)$$  \hspace{1cm} (7.18)

This temperature error is applied to a proportional plus integral unit with a gain $G_{13} = 2$ and an integrator with a reset rate of $1/3$ repeats per minute.

![Diagram](image)

Fig. 7.28 Reheat steam temperature control, block 14

The resulting value of signal, $SSCI$, is then added to the megawatt demand which is obtained from block 8, the integrated boiler turbine master, with a resulting signal of $DDEC$.

The signal sent to the process is a variable $R_{12}$ and its equation is related to this adjusted megawatt demand signal and was given earlier in the process section.
The following set of figures represent the block diagram of the control system using a control type, rather than analog type, of configuration. Rather than going through a detailed description of any one of the block diagrams, using the different notations, it is suggested that the reader compare, for example, the firing rate calibration, block 10, using the analog diagrams with the control system diagrams which are shown in the following figures and the variables on each are labeled so it is easy to follow the signals and make a comparison with the two systems. It should be noted, however, that for example in block 8, the integrated boiler turbine master, the actual turbine control valves are shown in a box labeled f(x) with VPS, the control variable entering this box and in the lower part of this figure a more detailed drawing is given which shows a representation of the turbine control valves. Similar diagrams to this would be obtained in the other sections where a DC motor is used to drive a particular valve or vane, such as the inlet vanes of the fans which blow the air into the unit for combustion purposes. Block 7 is shown in Figure 7.29, block 8 is shown in Figure 7.30, block 9 is shown in Figure 7.31, block 10 is shown in Figure 7.32, block 11 is shown in Figure 7.33, block 12 is shown in Figure 7.34, block 13 is shown in Figure 7.35 and block 14 is shown in Figure 7.36.

The analog type and control type symbols used in the control system diagram are shown in Table 7.3 and Table 7.2 respectively.
Figure 7.29 Unit load demand development, block 7.
Figure 7.30 Integrated boiler turbine master, block 8.
Figure 7.31 Feedwater flow control, block 9.

Figure 7.32 Firing rate calibration, block 10.
Figure 7.33 Superheater spray control valve, block 11.

Figure 7.34 Fuel flow control, block 12.
Figure 7.35 Air flow calibration, block 13.

Figure 7.36 Reheat steam temperature control, block 14.
**TABLE 7.2**

CONTROL TYPE SYMBOLS USED IN THE PLANT'S CONTROL SYSTEM DIAGRAM

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>o</td>
<td>Input or constant value.</td>
</tr>
<tr>
<td>H</td>
<td>Manual input.</td>
</tr>
<tr>
<td>A</td>
<td>Amplifier.</td>
</tr>
<tr>
<td>T</td>
<td>Transfer relay.</td>
</tr>
<tr>
<td>S</td>
<td>Servo-motor operated potentiometer.</td>
</tr>
<tr>
<td>Σ</td>
<td>Summer.</td>
</tr>
<tr>
<td>Δ</td>
<td>Error, difference between two signals.</td>
</tr>
<tr>
<td>%</td>
<td>Gain controller.</td>
</tr>
<tr>
<td>x</td>
<td>Static multiplier, provides variable gain.</td>
</tr>
<tr>
<td>F(x)</td>
<td>Limiter.</td>
</tr>
<tr>
<td>f(t)</td>
<td>Function of variable input, used to control process.</td>
</tr>
<tr>
<td>d/dt</td>
<td>Time delay function.</td>
</tr>
<tr>
<td>d/dt</td>
<td>Time derivative.</td>
</tr>
<tr>
<td></td>
<td>Converts step into ramp function.</td>
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### TABLE 7.2 CONTINUED

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</tr>
<tr>
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<td>Proportional plus integral controller.</td>
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### TABLE 7.3

**ANALOG TYPE SYMBOLS USED IN THE SIMULATION OF CONTROL SYSTEM**

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</tr>
<tr>
<td><img src="amplifier.png" alt="Amplifier, gain or percentage" /></td>
<td>Amplifier, gain or percentage.</td>
</tr>
<tr>
<td><img src="summer.png" alt="Summer" /></td>
<td>Summer.</td>
</tr>
<tr>
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</tr>
<tr>
<td><img src="equation.png" alt="Equation; output is a function of input" /></td>
<td>Equation; output is a function of input.</td>
</tr>
<tr>
<td><img src="time_derivative.png" alt="Time derivative" /></td>
<td>Time derivative.</td>
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CHAPTER 8

THE COMPLETE SIMULATION OF THE POWER PLANT
WITH THE PROCESS AND THE CONTROL
SYSTEM INTERCONNECTED

The discussion to this point has been dealing with the details of an individual part of the complete digital simulation of the supercritical steam generator with its associated control system. At this point it is appropriate to combine the parts that have been described in earlier chapters into a compact package. This may be done by looking at the complete simulation in the form of the digital computer program.

The basic block diagram which illustrates how the problem is solved is given in Figure 8.1. The digital simulator takes the particular problem and initializes the variables to the appropriate values. It then determines whether or not the control system is included in or excluded from the particular simulated problem. If the control system is to be included, it then computes the values of the control variables which are TVPXR, FWRE, PBFP, SVP1, WNG, and DDEC.

After computing these controlled variables the problem solution then follows and the process equations are solved for the values of the output pressure, temperature and enthalpy of the various lumps that have been described and are shown in Figure 1.2 and for the power output of the unit and the heat input to the various lumps. Then if a printing is desired, the output at that time increment is printed and the problem is then reiterated until the programmed solution time has been elapsed.

In order to facilitate the inspection of the subsequent flow chart and computer program, Table 8.1 below gives a complete list of the computer variables used in the process simulation. It is noted that the control system
variables also used in the problem are given in Table 7.1. In Figure 8.2 the complete flow chart of the digital simulator is given. References will be made to Figure 8.2 in this chapter. In addition to this, a complete copy of the computer program for the digital simulator is given in Table B-1 in the Appendix B.

In Table B-1 it is seen that the individual block numbers are given so that a reference can be made between the computer program and the block diagram as is given in Figure 8.2. Since most of the equations that have been used in the digital simulator have been described in previous sections, there will be no attempt at this point to go into the detail of the equations. Hence, only a brief description of each block in the digital simulator will be given.

Fig. 8.1 Block diagram for problem solution
### TABLE 8.1
COMPUTER VARIABLES USED IN THE PROCESS SIMULATION

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>SPVOL constant</td>
</tr>
<tr>
<td>AA</td>
<td>Throttle valve area (ft(^2))</td>
</tr>
<tr>
<td>ALPHA</td>
<td>SPVOL constant</td>
</tr>
<tr>
<td>AWG</td>
<td>Combustion gas flow (lbs/hr)</td>
</tr>
<tr>
<td>AO1−A10</td>
<td>Constants used in the subroutine TSSPH. Used when H is greater than H(_\text{MIN}).</td>
</tr>
<tr>
<td>B</td>
<td>SPVOL constant</td>
</tr>
<tr>
<td>BO</td>
<td>SPVOL variable</td>
</tr>
<tr>
<td>C</td>
<td>SPVOL variable</td>
</tr>
<tr>
<td>CO1−C16</td>
<td>Constants for the subroutine TSSPH when H is less than H(_\text{MIN}).</td>
</tr>
<tr>
<td>CPG</td>
<td>Specific heat of the combustion gas (btu/lb-°F)</td>
</tr>
<tr>
<td>CPW(14)</td>
<td>Specific heat of the metal (btu/lb°F)</td>
</tr>
<tr>
<td>D</td>
<td>SPVOL constant</td>
</tr>
<tr>
<td>DDEC</td>
<td>Adjusted boiler demand in reheat steam temperature control (megawatts)*</td>
</tr>
<tr>
<td>DELT</td>
<td>Temperature adjustment in subroutine TSSPH (°F)</td>
</tr>
<tr>
<td>DEMW1</td>
<td>External megawatt demand (megawatts)*</td>
</tr>
<tr>
<td>DHOR(3)</td>
<td>Runge-Kutta integration constant (btu/ID)</td>
</tr>
<tr>
<td>DQ13</td>
<td>Heat added to the feedwater in the low pressure feedwater heater (btu/sec)</td>
</tr>
<tr>
<td>DQ14</td>
<td>Heat added to the feedwater in the high pressure feedwater heater (btu/sec)</td>
</tr>
<tr>
<td>DQS</td>
<td>Heat absorbed by the spray water (btu/sec)</td>
</tr>
<tr>
<td>DRHO(15)</td>
<td>Change in the density of the working fluid (lbs/ft(^3))</td>
</tr>
<tr>
<td>DTIME</td>
<td>Time increment used in the process integration (sec)</td>
</tr>
<tr>
<td>DTWR(3)</td>
<td>Runge-Kutta integration constant for wall temperature (°F)</td>
</tr>
<tr>
<td>E</td>
<td>SPVOL constant</td>
</tr>
<tr>
<td>EPSI</td>
<td>SPVOL constant</td>
</tr>
<tr>
<td>F</td>
<td>SPVOL constant</td>
</tr>
<tr>
<td>G</td>
<td>SPVOL constant</td>
</tr>
<tr>
<td>H</td>
<td>Enthalpy used in the subroutine TSSPH (btu/lb)</td>
</tr>
<tr>
<td>HC</td>
<td>Heat value of the fuel (btu/ft(^3))</td>
</tr>
<tr>
<td>HE2</td>
<td>Enthalpy of the 2nd extraction flow (btu/lb)</td>
</tr>
</tbody>
</table>
TABLE 8.1 Continued

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>HH</td>
<td>SPVOL constant</td>
</tr>
<tr>
<td>HI(3,21)</td>
<td>Input enthalpy of the lump (btu/lb)</td>
</tr>
<tr>
<td>HMIN</td>
<td>Enthalpy corresponding to 662°F in the compressed liquid region (btu/lb)</td>
</tr>
<tr>
<td>HO(3,21)</td>
<td>Output enthalpy of the lump (btu/lb)</td>
</tr>
<tr>
<td>HO13</td>
<td>Output enthalpy of the low pressure feedwater heater (btu/lb)</td>
</tr>
<tr>
<td>HO14</td>
<td>Output enthalpy of the high pressure feedwater heater (btu/lb)</td>
</tr>
<tr>
<td>HO19</td>
<td>Output enthalpy of the working fluid from the condenser (btu/lb)</td>
</tr>
<tr>
<td>HPFWH</td>
<td>High pressure feedwater heater**</td>
</tr>
<tr>
<td>INCRHO</td>
<td>Incremental enthalpy of a lump added during the time increment (btu/lb)</td>
</tr>
<tr>
<td>J</td>
<td>Number of furnace lumps considered for initial DO loop.</td>
</tr>
<tr>
<td>KP(14)</td>
<td>Pressure constant (psi/lbs-sec²)</td>
</tr>
<tr>
<td>KQ(14)</td>
<td>Heat transfer constant</td>
</tr>
<tr>
<td>LPFWH</td>
<td>Low pressure feedwater heater**</td>
</tr>
<tr>
<td>M(14)</td>
<td>Mass of the working fluid in lump (lbs)</td>
</tr>
<tr>
<td>MW(14)</td>
<td>Mass of metal of lump (lbs)</td>
</tr>
<tr>
<td>MWCPW(14)</td>
<td>MW times CPW</td>
</tr>
<tr>
<td>OLT</td>
<td>Gate, provides for open loop control tests</td>
</tr>
<tr>
<td>P</td>
<td>Pressure used in subroutine TSSPH (psia)</td>
</tr>
<tr>
<td>PBFP</td>
<td>Boiler feed pump output pressure (psia)</td>
</tr>
<tr>
<td>PI(1,21)</td>
<td>Input pressure to lump (psia)*</td>
</tr>
<tr>
<td>PO(1,21)</td>
<td>Output pressure of lump (psia)</td>
</tr>
<tr>
<td>POA(15)</td>
<td>Average pressure in lump (psia)</td>
</tr>
<tr>
<td>POWER</td>
<td>Total power of the turbines (megawatts)</td>
</tr>
<tr>
<td>POWER1</td>
<td>Equivalent power of the high pressure turbine (megawatts)</td>
</tr>
<tr>
<td>POWER2</td>
<td>Equivalent power of the low pressure and intermediate pressure turbines (megawatts)</td>
</tr>
<tr>
<td>PP</td>
<td>SPVOL constant (psia)</td>
</tr>
<tr>
<td>PR</td>
<td>Gate, provides for the process to be included in (1.) or excluded from (0.) the simulation</td>
</tr>
<tr>
<td>PRD</td>
<td>Gate, provides for the variable density to be included in (1.) or excluded from (0.) this system</td>
</tr>
<tr>
<td>Q(3,21)</td>
<td>Heat transfer to working fluid (btu/sec)</td>
</tr>
<tr>
<td>QC</td>
<td>Convective heat transfer (btu/sec)</td>
</tr>
<tr>
<td>QCT</td>
<td>Total convective heat plus the excess heat leaving the furnace (btu/sec)</td>
</tr>
<tr>
<td>QE</td>
<td>Excess heat leaving the furnace (btu/sec)</td>
</tr>
<tr>
<td>QG</td>
<td>Total heat liberated by the combustion process (btu/sec)</td>
</tr>
</tbody>
</table>
TABLE 8.1 Continued

QH13  Heat output of the extraction flow (2) condensate (btu/sec)
QH14  Heat output of the extraction flow (1) condensate (btu/sec)
QI10  Heat content of the steam entering lump 10 (btu/sec)
QI13  Heat content of the feedwater entering the low pressure feedwater heater (btu/sec)
QI14  Heat content of the feedwater entering the high pressure feedwater heater (btu/sec)

Q09  Heat content of the working fluid leaving lump 9 (btu/sec)
QO13  Heat content of the water leaving the low pressure feedwater heater (btu/sec)

QO14  Heat content of the compressed water leaving the high pressure feedwater heater (btu/sec)
QR  Radiative portion of the heat transferred (btu/sec)
QW(1,21) Heat transferred from the combustion gas to the lump (btu/sec)

R  SPVOL constant
RATIO1  Ratio of the total heat input to the water flow
RHO(15)  Density of the working fluid (lbs/ft$^3$)
RHON(15)  Dummy variable for density
RHOSV  Density computed in SPVOL

RUN  Run number used in the logic of the program
R12  A damper control signal (p. u.)*
SPVOL  Subroutine used to compute specific volume and density

T  Temperature used in subroutine TSSPH
TAU  SPVOL constant
T1  Increment of time for control system variables computation (sec)*
T2  Increment of time for interconnection of control system variables with process variables (sec)*
TG  Temperature of the combustion gas at furnace exit (°F)

TI(1,21)  Input temperature of a lump (°F)
TIME  Time in the process (sec)
TO(3,21)  Output temperature of a lump (°F)
TOA(15)  Average temperature of a lump(°F)
TPRINT  Time of the print of the output (sec)

TSSPH  Subroutine for the computation of the temperature
TT  SPVOL constant
TVPXR  Corrected throttle valve position (p. u.)*
TW(3,21)  Wall temperature (°F)
TABLE 8.1 Continued

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>V(15)</td>
<td>Volume of the equivalent lump (cu. ft.)</td>
</tr>
<tr>
<td>VK1, VK2</td>
<td>SPVOL constant</td>
</tr>
<tr>
<td>VNU</td>
<td>Specific volume (cu. ft. /lb.) computed in SPVOL</td>
</tr>
<tr>
<td>VJ</td>
<td>SPVOL constant</td>
</tr>
<tr>
<td>VV1-VV5</td>
<td>SPVOL variable</td>
</tr>
<tr>
<td>W(20)</td>
<td>Working fluid mass flow rate (lbs/sec)</td>
</tr>
<tr>
<td>WC</td>
<td>Average steam flow through the intermediate and low pressure turbine (lbs/sec)</td>
</tr>
<tr>
<td>WE1</td>
<td>High pressure feedwater extraction flow (lbs/sec)</td>
</tr>
<tr>
<td>WE2</td>
<td>Low pressure feedwater extraction flow (lbs/sec)</td>
</tr>
<tr>
<td>WG(3)</td>
<td>Combustion gas flow (lbs/hr)</td>
</tr>
<tr>
<td>WNG</td>
<td>Flow of natural gas (cu. ft. /hr.)</td>
</tr>
<tr>
<td>WS</td>
<td>Spray flow (lbs/sec)</td>
</tr>
<tr>
<td>WSTH</td>
<td>Spray flow (K-lb/hr)*</td>
</tr>
<tr>
<td>WRSS</td>
<td>Steady state reheater mass flow (lbs/sec)</td>
</tr>
<tr>
<td>XP(14)</td>
<td>Pressure drop across a lump</td>
</tr>
<tr>
<td>XQ(14)</td>
<td>Temperature difference across the wall of the lump (°F)</td>
</tr>
<tr>
<td>Y</td>
<td>SPVOL constant</td>
</tr>
</tbody>
</table>

* Also listed in control system variables
** Abbreviation used in comment statements

Blocks 1, 2, 3 and 4 as shown in Figure 8.2 are used in the simulator to initialize the problem variables. In Block 5.1 the computer variable, OLT, is used for a gate to determine whether or not the control system should be included in, or excluded from, the problem solution. If OLT equals 1 this indicates that the control system should be run under open loop conditions and hence, the problem flow is transferred to Block 15 at this point. However, the control system should be included, the problem flow is passed on to Block 5.2. In Block 5.2 and 5.3, the scheduled power is checked to see if it is within the range of the simulated plant. If it is not within range of the simulated plant, appropriate alarms are given. If it is within the range of the simulated plant the solution flow is then passed on to Block 6 where the headings are printed on the output.
The computer Blocks 7, 8, 9, 10, 11, 12, 13 and 14 have been described in detail in Chapter 7. These blocks are the computer representations of the control system. Block 7 is the unit load demand development. Block 8 is the integrated boiler turbine master. Block 9 is the feedwater flow control. Block 10 is the firing rate calibration. Block 11 is the spray valve control system. Block 12 is the fuel flow control. Block 13 is the air flow calibration and air flow control and Block 14 is the reheat steam temperature control. An analysis of the block diagram for these blocks would parallel exactly the analysis given in Chapter 7 where the analog diagrams were investigated in detail. The reader is then referred to Chapter 7 for a detailed description of these blocks.

In Block 15, the time is incremented and there is an interface between the control system and the process. As will be recalled from Block 5.1 there is an entrance to Block 15 and if this path is followed, the logic in Block 15.5 determines which particular run is being computed and sets the control variables for the appropriate run. If the control system was not excluded then the problem flows directly from Block 14 into Block 15 and standard time increments are then implemented before going to the solution of the process equations.

In Block 17 the variables that are used in both the control and the process are adjusted in order to match the units properly for the two subsystems. At this point also some preliminary calculations are made before entering the process solution. In Block 18 the heat flux to the various lumps is computed. This was described in detail in Chapter 5.

In Block 19 the fundamental process equations, as are described in Chapter 2, are solved. The techniques that are used for the solution of the differential equations describing the process have been described in Chapter 6. This is implemented for Lumps 1 through 11 in Blocks 19.1 through Block 19.26. In Blocks 19.27 through Block 19.31 the process equations for the reheater are solved.
The one fundamental equation that has not been discussed previously in detail for the computer program is the continuity equation. At the present time a subroutine exists for computing the specific volume or density only in the superheater and not in the critical region. In the furnace, that is in Lumps 1 through 8 of the simulated plant, the density is assumed to be constant so that the mass flow into a particular lump is equal to the mass flow leaving a particular lump. In Lumps 9, 10, and 11, however, the transient change in density of the working fluid may be considered. As is shown in the block diagram with Lumps 19.5 through 19.11, the solutions of the process include the density change by including the continuity equation with the other process equations. In order to include these equations the computer variable PRD is set equal to 1. If the lump under consideration is 9, 10 or 11, the solution follows through the blocks that allow computation of the density and the flow change. The form of the continuity equation that is used in the solution, as seen earlier in Chapter 2, is given in equation (8.1).

\[ W_i - W_o = Ad \left( \frac{\rho L}{dt} \right) \]  

(8.1)

Since the length and area of a particular lump will remain constant in the simulation, equation (8.1) becomes

\[ \frac{d\rho}{dt} = \frac{1}{AL} (W_i - W_o) = \frac{1}{V} (W_i - W_o) \]  

(8.2)

It is assumed that the density of a particular lump is the average density and will be computed by taking an average of the input and output pressures of a lump and an average of the input and output temperatures of the corresponding lump and then computing the density using the digital computer subroutine SPVOL.

From equation (8.2) above

\[ \Delta \rho = \Delta t \left( \frac{1}{V} (W_i - W_o) \right) \]  

(8.3)
The value of $W_1$ is known and the value of $W_0$ is desired.

The initial values of $\text{RHO}$ for each lump studied must be given; call these values $\text{RHO}(N)$, $N=9, 10, 11$. The change in density for a time increment is given by the presently computed value of $\text{RHO}$ subtracted from the previously computed value of $\text{RHO}$ for the previous time interval. Then, if the density had changed during the time increment, the new flow out of the lump may be found by rewriting equation (8.3) as equation (8.4) where the computer variables are used.

$$W(N) = W(N-1) - \frac{V(N) \cdot (\text{DRHO}(N))}{\text{DTIME}}$$  \hspace{1cm} (8.4)

In Block 20 the turbine equations are used to determine the simulated power output for the unit. These equations were described in Section 3.4. In Block 21 the solution of the enthalpy output of the high pressure feedwater heater is obtained. This enthalpy output of the high pressure feedwater heater then becomes the initial input enthalpy to the economizer which then closes the loop on the complete simulation. In Block 21 the equations are solved which take the extracted flow from the intermediate and low pressure turbine and heat the low pressure feedwater heater and the extraction flow from the high pressure turbine to heat the high pressure feedwater heater. A heat balance is obtained in order to compute the value of the enthalpy output for the high pressure feedwater heater which is then the enthalpy input to the boiler itself. These equations have been described in Section 3.1.

In Block 22 the printing of the output variables that are desired are effected and, in addition, the logic by which different runs can be made is included. Following the final block in the computer program is the data required for the simulation which is, in the order of its appearance, the number of lumps iteratively solved, the pressure drop used to compute the pressure constant, the temperature drop used to compute the heat flux constant, the mass of the water, the mass of the metal, the specific heat of the metal, the initial enthalpy input, the enthalpy outputs, the initial wall temperatures, and the initial heat flux inputs from the furnace to the water.
Following the input data required for the simulated plant, are the two subroutines that have been described in Chapter 4. The two subroutines are: (a) TSSPH, which computes the temperature as a function of pressure and enthalpy for the regions through which the heat cycle of the plant pass and (b) SPVOL, which computes the specific volume or density of the superheated steam which is used in connection with the continuity equation described in this chapter.
Fig. 8.2 Digital simulator flow chart
Fig. 8.2 Continued
Fig. 8.2 Continued
Determine Throttle Valve Position

IF IBTM = 0,

TVPRX = 1.0

TVPRX = TVPX/1000.

Compute Feedwater Flow Demand

IF FWCD > 1200.

Assign Parameters For Line Segment

IF FWRE ≤ 2835.

Assign Parameters For Line Segment

IF FWRE ≤ 3085.

Fig. 8.2 Continued
Fig. 8.2 Continued
Integrate Feedwater Error; Obtain FDW2
Obtain New Value of RPM

\[ X = \text{FWRE} - \text{SP1} \]

IF \( X \leq B \)

IF \( FWC = 0.0 \)

Obtain New Value of PBFP

IF \( FWC \neq 0.0 \)

PBFP = 4280

FWRE = 3980.

Obtain New Value of FWRE

Set Firing Rate Calibration Demand Input

Fig. 8.2 Continued
10.2 Obtain Final Superheater Steam Temp. Error: $E_{13G}$

10.3 Calculate Primary Superheater Steam Temp. $d/dt$: $DERT$

10.4

- $DERER_1 = E_{13G} - DERT$
- $TRIM_3 = (1.5/700) \times DEMW_4 + 1.0$

10.5

- $FRDEM_3 = FRDEM_2 + E_1 + TRIM_3 \times DERER_1$

10.5a

- Integrate $E_{13G} = E_{13GI}$

10.6

- IF $\text{ABS}(E_{13GI}) \leq 120$
  - No: Go to 39
  - Yes: Go to 10.7

10.7

- $TRIM_4 = E_{13GI} \times (.1/120.) + 1.0$

10.8

- IF $E_{13GI} \leq 120$
  - No: Go to 39
  - Yes: Go to 10.9

10.9

- $E_{13GI} = 120$

10.10

- $E_{13GI} = -120.$

Fig. 8.2 Continued
10.11 FRDEMC = FRDEM3*TRIM4

11.1 ESP = DERER1 - SVP

11.2 Integrate To Obtain SVP
SVP1 = -SVP

11.3 IF SVP1 ≤ 100, Yes
SVP1 = 100

11.4 No

11.5 IF SVC = 0.0, Yes
WSTH = 0.0

11.6 No

11.7

12.1 Calculate Fuel Flow Demand FFDM

12.2 Obtain Fuel Flow Error: E20

12.3 Obtain Integral + Proportion of Fuel Flow Error

12.4 Obtain Fuel Flow Value WNGTH

12.5 IF FFC = 0.0, Yes

12.6 WNG = WNGTH*10.0

12.7 No

13.1 Obtain Integrated Oxygen Error: OXE1

Fig. 8.2 Continued
Fig. 8.2 Continued
13.12 Obtain New Air Flow Value

13.13 IF AFC = 0

13.14 Set Steady State Value of Air Flow

13.15 Compute New Value of Air Flow

14.1 Obtain Difference Between Superheater Steam Temp. and Reheater Steam Temp.

14.2 Obtain Integral + Proportion of Temperature Error

14.3 Add Result to Adjusted MW Demand, DEMW4

14.4 IF RHSTC = 0.0

14.5 Compute New Value of R12

14.6 R12 = 1.0

15.1 Compute Time and Time Increment

15.2 IF TIME > T2

15.3 T2 = T2 + DTIME

Fig. 8.2 Continued
17.1 Compute Feed - Water Flow
\[ W(N) = \frac{FWRE}{3.6} \]

17.2 Compute Area of Throttle Valve

17.3 Compute Combustion Gas Flow

17.4 Relate PI(1, 1) to Controlled BFP

17.5 \[ WS = \frac{WSTH}{3.6} \]

18.1 Compute Total Convective Heat + Heat to Stack

18.2 Compute Heat Liberated by the Combustion Process

18.3 Compute Radiant Heat Transferred

18.4 Compute the Convective Heat Transferred

Fig. 8.2 Continued
18.5 Compute Heat Transferred to Each Heat Absorbing Surface

19.1 DO 50 N = 1, J

19.2 Compute Pressure For Each Lump

19.3 DO 40 J1 = 1, 3

19.4 Call TSSPH Compute TO (J1, N)

19.5 IF PRD = 0.0

19.6 IF J1 = 1.

19.7 IF N ≥ 9

19.8 IF N = 12

19.9 Compute Average Temp. and Pressure of Lumps 9, 10, 11

19.10 Call SPVOL Compute RHO(N)

19.11 Compute Density and Flow Change

Fig. 8.2 Continued
19.12 \[ Q(J1, N) = f(TW, TO(J1, N)) \]

19.13 Compute Wall Temp. Change: DTWR(J1)

19.14 Compute Enthalpy Change: DHOR(J1)

19.15 IF J1 ≥ 3.
   Yes → 72
   No → 70

19.16 Compute TW(2, N)

19.17 Compute TW(3, N)

19.18 Compute HO(2, N)

19.19 Compute HO(3, N)

19.20 Compute INCHRO

19.21 \[ HO(1, N) = HO(1, N) + \text{INCRHO} \]

Fig. 8.2 Continued
19.22 IF N = 9

Yes

19.24 Compute Heat Content of Steam at Finishing Superheater Input

No

19.23 HI(1, N+1) = HO(1, N)

19.25 HI(1, 10) = QI10/W(9)

19.26

74

Compute TW(1, N)

75

19.27 N = 12

19.28 Compute Throttle Pressure: PO(1, 16)

19.29 Compute Reheater Steam Flow

19.30 Initialize Reheater PI(1, 12; HI(1, 12)

19.31 Compute HO(1, 12), TW(1, 12), TO(1, 12), PO(1, 12)

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77

Fig. 8.2 Continued
20.1 Compute POWER1 and POWER2

20.2 Compute Total Power Output

21.1 Compute Extraction Flow: WE1

21.2 Compute Average Flow Through Intermediate and Low Pressure Turbines: WC

78

21.3 Compute Extraction Flow: WE2

79

21.4 Compute Heat Added to Water in LPFWH

21.5 Compute Heat Content of Water Leaving LPFWH

80

21.6 Compute Enthalpy of Water Leaving LPFWH

21.7 Compute Heat Added to Water in HPFWH

82

21.8 Compute Heat Content of Water Leaving HPFWH

21.9 Compute Enthalpy of Compressed Water Leaving HPFWH: HI(1,1)

83

22.1 IF TIME > TPRINT

No

22.3 Print Output Increment TPRINT

Yes

22.2 IF OLT < 0.0

Yes

84

No

86

Fig. 8.2 Continued
Set Controlled Variables For Run S-1

Set Controlled Variables For Run S-2

STOP

TIME = TIME + DTIME

IF

RUN = RUN - 2.

Fig. 8.2 Continued
CHAPTER 9

DESCRIPTION OF THE SIMULATED TESTS
USING THE DIGITAL SIMULATOR

In the preceding chapter the details of the digital simulator were described. This chapter illustrates the use of the simulator in the study of several simulated conditions and describes the method of the operation of the simulator.

As has been mentioned earlier, the digital simulator is divided into two basic sections; the control system and the process system.

There are basically two modes of operation available; control system interconnected with the process and operating, or held at a steady value.

Any one of the control system primary variables may be included in, or taken out of the system by a series of "gate" functions listed below:

- IBTM Integrated boiler turbine master
- FWC Feedwater flow control
- SVC Spray valve control
- FFC Fuel flow control
- AFC Air flow control
- RHSTC Reheat steam temperature control

If, for example, it is desired to observe the operation of the simulated plant with the fuel flow control on manual (i.e., off automatic) the computer variable FFC is set equal to 0 instead of 1.0, in Block 4 of the computer program. With the current program, this would cause the fuel flow (computer variable WNG) to be set equal to 5,000,000 ft$^3/hr$, which is the proper full load value for this variable. If it would be desired subsequently to change this variable, it could be adjusted in Block 12.
For a steady state simulated run with all control variables at their respective full load values, the gate variable CS is set equal to zero, (see Block 4). If the other gate variables, listed earlier are then set equal to CS, as given in the program, the appropriate steady state values are picked up in the program.

The simulator computer program also has capability of including or excluding the density change in Lumps 9, 10, and 11. Its gate variable is PRD, and it operates in the same way as the gates first mentioned. In Block 19 a logical decision is made to determine whether the program includes, \( PRD = 1 \), or excludes, \( PRD = 0 \), the density change in the solution to the process equations.

As mentioned in the previous chapter the data for the particular plant under study is included in the program. If another plant were to be simulated, the appropriate data would have to be obtained, as outlined, and substituted for the existing data.

The program is run by inserting an appropriate power demand card. The computer variable changed is DEMW1, which is the scheduled power. This may be changed in Block 4 or, as is presently being done, in Block 22. The reason for making the step change in DEMW1 later in the program is to allow the system to settle down to an approximate steady state value before subjecting the system to a disturbance. This feature was required in order to alleviate spending a very large amount of time in the fine tuning process to get the system in exact balance.

A typical run would consist of a 600 second problem run at steady state, 560MW, and then a \(-10\)MW step change in DEMW1. The problem would then settle down within the next 600 seconds.

In Table 9.1 the simulated tests that have been made on the model are presented. It is seen that there are 14 simulated test runs made under open control system loop (i.e., off automatic) and 3 simulated test runs that had the control system controlling the process.
### TABLE 9.1
SIMULATED TEST RUNS

<table>
<thead>
<tr>
<th>RUN</th>
<th>PROCESS VARIABLE CHANGED</th>
</tr>
</thead>
<tbody>
<tr>
<td>S-1</td>
<td><strong>STEP INCREASE IN FIRING RATE AT ( \approx 560 \text{ MW} )</strong></td>
</tr>
<tr>
<td>S-2</td>
<td><strong>STEP DECREASE IN FIRING RATE AT ( \approx 560 \text{ MW} )</strong></td>
</tr>
<tr>
<td>S-3</td>
<td><strong>STEP DECREASE IN FEEDWATER FLOW AT ( \approx 560 \text{ MW} )</strong></td>
</tr>
<tr>
<td>S-4</td>
<td><strong>STEP INCREASE IN FEEDWATER FLOW AT ( \approx 560 \text{ MW} )</strong></td>
</tr>
<tr>
<td>S-5</td>
<td><strong>STEP INCREASE IN SPRAY FLOW AT ( \approx 560 \text{ MW} )</strong></td>
</tr>
<tr>
<td>S-6</td>
<td><strong>STEP DECREASE IN SPRAY FLOW AT ( \approx 560 \text{ MW} )</strong></td>
</tr>
<tr>
<td>S-7</td>
<td><strong>STEP DECREASE IN THROTTLE VALVE POSITION AT ( \approx 560 \text{ MW} )</strong></td>
</tr>
<tr>
<td>S-8</td>
<td><strong>STEP INCREASE IN THROTTLE VALVE POSITION AT ( \approx 560 \text{ MW} )</strong></td>
</tr>
<tr>
<td>S-9</td>
<td><strong>STEP DECREASE IN FIRING RATE AT ( \approx 280 \text{ MW} )</strong></td>
</tr>
<tr>
<td>S-10</td>
<td><strong>STEP INCREASE IN FIRING RATE AT ( \approx 280 \text{ MW} )</strong></td>
</tr>
<tr>
<td>S-11</td>
<td><strong>STEP DECREASE IN FEEDWATER FLOW AT ( \approx 280 \text{ MW} )</strong></td>
</tr>
<tr>
<td>S-12</td>
<td><strong>STEP INCREASE IN FEEDWATER FLOW AT ( \approx 280 \text{ MW} )</strong></td>
</tr>
<tr>
<td>S-13</td>
<td><strong>STEP INCREASE IN THROTTLE VALVE POSITION AT ( \approx 280 \text{ MW} )</strong></td>
</tr>
<tr>
<td>S-14</td>
<td><strong>STEP DECREASE IN THROTTLE VALVE POSITION AT ( \approx 280 \text{ MW} )</strong></td>
</tr>
<tr>
<td>S-15</td>
<td><strong>RAMP CHANGE IN LOAD FROM 560 \text{ MW} \rightarrow 280 \text{ MW}</strong></td>
</tr>
<tr>
<td>S-16</td>
<td><strong>STEP CHANGE IN SCHEDULED LOAD FROM 560 \rightarrow 550 \text{ MW}</strong></td>
</tr>
<tr>
<td>S-17</td>
<td><strong>STEP CHANGE IN SCHEDULED LOAD FROM 550 \rightarrow 560 \text{ MW}</strong></td>
</tr>
</tbody>
</table>
Table 9.2 shows the change in the simulated variables for the 17 runs. In column 2 is listed the corresponding test that was actually performed on the plant. These tests will be described in Chapter 10. Column 3 of Table 9.1 lists the actual variations in independent variables while columns 4 and 5 give the simulated independent variables with their deviations respectively.

In Table 9.3 the values set for the remaining controlled variables are given for the 14 test runs. In Table 9.4 the variables that were tabulated in the simulated tests are given. It should be noted that while only 8 variables were chosen for listing in the output, many more could be listed if there was a need to investigate them. The choice of the output variables is determined by the programmer in Block 6 and 22.

The tabulated variables for the closed control system loop tests are given in Table 9.5.

The results of the 17 tests made on the digital simulator are recorded in the graphs in Figure 9.1 to Figure 9.17. Figure 9.1 illustrates the results of test S-1 and Figure 9.17 contains the information concerning the output of test S-17. The results of other tests run correspond, having corresponding figure numbers.

The results of the tests are similar to the actual tests as will be seen in Chapter 10.

Alarms are programmed into the simulator corresponding to scheduled loads that are either too high or too low. Two test runs, S-18 and S-19 were made to illustrate the alarms. Run S-18 is for 200 MW and S-19 is for 700 MW scheduled load. The computer output for these runs is shown in Figure 9.18. If other alarms corresponding to other undesired conditions were desired, they could be included in the program.

A sample output of the digital simulator is given in Table 9.6.
<table>
<thead>
<tr>
<th>Run</th>
<th>Actual Plant Test</th>
<th>Actual Change</th>
<th>Computer Independent Variable Changed</th>
<th>Simulated Change</th>
<th>Variable Change From</th>
<th>To</th>
</tr>
</thead>
<tbody>
<tr>
<td>S-1</td>
<td>A1</td>
<td>+4.5 %</td>
<td>WNG</td>
<td>+5 %</td>
<td>5.0x10^6</td>
<td>5.25x10^6 Kft^3/hr</td>
</tr>
<tr>
<td>S-2</td>
<td>A2</td>
<td>-4.8 %</td>
<td>WNG</td>
<td>-5 %</td>
<td>5.25x10^6</td>
<td>5.0x10^6 Kft^3/hr</td>
</tr>
<tr>
<td>S-3</td>
<td>A3</td>
<td>-3.5 %</td>
<td>FWRE</td>
<td>-5 %</td>
<td>3980</td>
<td>3780 Klb/hr</td>
</tr>
<tr>
<td>S-4</td>
<td>A4</td>
<td>+3.3 %</td>
<td>FWRE</td>
<td>+5 %</td>
<td>3980</td>
<td>3980 Klb/hr</td>
</tr>
<tr>
<td>S-5</td>
<td>A6</td>
<td>+104.4 Klb/hr</td>
<td>WSTH</td>
<td>+100 Klb/hr</td>
<td>0</td>
<td>100 Klb/hr</td>
</tr>
<tr>
<td>S-6</td>
<td>A5</td>
<td>-81.1 Klb/hr</td>
<td>WSTH</td>
<td>-100 Klb/hr</td>
<td>100</td>
<td>0 Klb/hr</td>
</tr>
<tr>
<td>S-7</td>
<td>A7</td>
<td>-1 %</td>
<td>TVPXR</td>
<td>-2 %</td>
<td>1</td>
<td>0.98 P.U.</td>
</tr>
<tr>
<td>S-8</td>
<td>A8</td>
<td>+2 %</td>
<td>TVPXR</td>
<td>+2 %</td>
<td>0.98</td>
<td>1 P.U.</td>
</tr>
<tr>
<td>S-9</td>
<td>A10</td>
<td>-7 %</td>
<td>WNG</td>
<td>-5 %</td>
<td>2.415x10^6</td>
<td>2.294x10^6 Kft^3/hr</td>
</tr>
<tr>
<td>S-10</td>
<td>A11</td>
<td>+7.3 %</td>
<td>WNG</td>
<td>+5 %</td>
<td>2.294x10^6</td>
<td>2.415x10^6 Kft^3/hr</td>
</tr>
<tr>
<td>S-11</td>
<td>A12</td>
<td>-6 %</td>
<td>FWRE</td>
<td>-5 %</td>
<td>1660</td>
<td>1575 Klb/hr</td>
</tr>
<tr>
<td>S-12</td>
<td>A13</td>
<td>+4.2 %</td>
<td>FWRE</td>
<td>+5 %</td>
<td>1575</td>
<td>1660 Klb/hr</td>
</tr>
<tr>
<td>S-13</td>
<td>A14</td>
<td>+1 %</td>
<td>TVPXR</td>
<td>+2 %</td>
<td>1.452</td>
<td>1.481 P.U.</td>
</tr>
<tr>
<td>S-14</td>
<td>-</td>
<td>-</td>
<td>TVPXR</td>
<td>-2 %</td>
<td>1.481</td>
<td>1.461 P.U.</td>
</tr>
<tr>
<td>S-15</td>
<td>A9</td>
<td>-250 MW</td>
<td>DEMW1</td>
<td>-280 MW</td>
<td>560</td>
<td>280 MW</td>
</tr>
<tr>
<td>S-16</td>
<td>-</td>
<td>-</td>
<td>DEMW1</td>
<td>-179 %</td>
<td>560</td>
<td>550 MW</td>
</tr>
<tr>
<td>S-17</td>
<td>-</td>
<td>-</td>
<td>DEMW1</td>
<td>+182 %</td>
<td>550</td>
<td>560 MW</td>
</tr>
</tbody>
</table>
### Table 9.3

**Variables for Open Loop Tests**

<table>
<thead>
<tr>
<th>Run</th>
<th>WNG ((\times 10^6))</th>
<th>FWRE</th>
<th>PBFP</th>
<th>WSTH</th>
<th>TVPXR</th>
<th>R12</th>
<th>DEMW1</th>
<th>RUN Computer Variable</th>
</tr>
</thead>
<tbody>
<tr>
<td>S-1</td>
<td>5.25</td>
<td>3980</td>
<td>4280</td>
<td>0</td>
<td>1</td>
<td>1</td>
<td>560</td>
<td>2</td>
</tr>
<tr>
<td>S-2</td>
<td>5.0</td>
<td>3980</td>
<td>4280</td>
<td>0</td>
<td>1</td>
<td>1</td>
<td>560</td>
<td>3</td>
</tr>
<tr>
<td>S-3</td>
<td>5.0</td>
<td>3780</td>
<td>4215</td>
<td>0</td>
<td>1</td>
<td>1</td>
<td>560</td>
<td>2</td>
</tr>
<tr>
<td>S-4</td>
<td>5.0</td>
<td>3980</td>
<td>4280</td>
<td>0</td>
<td>1</td>
<td>1</td>
<td>560</td>
<td>3</td>
</tr>
<tr>
<td>S-5</td>
<td>5.0</td>
<td>3980</td>
<td>4280</td>
<td>100</td>
<td>1</td>
<td>1</td>
<td>560</td>
<td>2</td>
</tr>
<tr>
<td>S-6</td>
<td>5.0</td>
<td>3980</td>
<td>4280</td>
<td>0</td>
<td>1</td>
<td>1</td>
<td>560</td>
<td>3</td>
</tr>
<tr>
<td>S-7</td>
<td>5.0</td>
<td>3980</td>
<td>4280</td>
<td>0</td>
<td>0.98</td>
<td>1</td>
<td>560</td>
<td>2</td>
</tr>
<tr>
<td>S-8</td>
<td>5.0</td>
<td>3980</td>
<td>4280</td>
<td>0</td>
<td>1.0</td>
<td>1</td>
<td>560</td>
<td>3</td>
</tr>
<tr>
<td>S-9</td>
<td>2.294</td>
<td>1660</td>
<td>3680</td>
<td>0</td>
<td>1.452</td>
<td>1.08</td>
<td>280</td>
<td>3</td>
</tr>
<tr>
<td>S-10</td>
<td>2.415</td>
<td>1660</td>
<td>3680</td>
<td>0</td>
<td>1.452</td>
<td>1.08</td>
<td>280</td>
<td>4</td>
</tr>
<tr>
<td>S-11</td>
<td>2.415</td>
<td>1575</td>
<td>3665</td>
<td>0</td>
<td>1.452</td>
<td>1.08</td>
<td>280</td>
<td>3</td>
</tr>
<tr>
<td>S-12</td>
<td>2.415</td>
<td>1660</td>
<td>3665</td>
<td>0</td>
<td>1.452</td>
<td>1.08</td>
<td>280</td>
<td>4</td>
</tr>
<tr>
<td>S-13</td>
<td>2.415</td>
<td>1660</td>
<td>3650</td>
<td>0</td>
<td>1.481</td>
<td>1.08</td>
<td>280</td>
<td>3</td>
</tr>
<tr>
<td>S-14</td>
<td>2.415</td>
<td>1660</td>
<td>3650</td>
<td>0</td>
<td>1.481</td>
<td>1.08</td>
<td>280</td>
<td>4</td>
</tr>
</tbody>
</table>
#### TABLE 9.4
VARIABLES TABULATED FOR OPEN LOOP TESTS

<table>
<thead>
<tr>
<th>Run S-1</th>
<th>Run S-2</th>
<th>Run S-3</th>
<th>Run S-4</th>
<th>Run S-5</th>
<th>Run S-6</th>
<th>Run S-7</th>
</tr>
</thead>
<tbody>
<tr>
<td>WNG POWER</td>
<td>WNG POWER</td>
<td>PBFP POWER</td>
<td>PBFP POWER</td>
<td>WSTH POWER</td>
<td>WSTH POWER</td>
<td>TVPXR POWER</td>
</tr>
<tr>
<td>FWRE</td>
<td>FWRE</td>
<td>FWRE</td>
<td>FWRE</td>
<td>TO(1, 9)</td>
<td>TO(1, 9)</td>
<td>FWRE</td>
</tr>
<tr>
<td>TO(1, 7)</td>
<td>TO(1, 7)</td>
<td>TO(1, 7)</td>
<td>TO(1, 7)</td>
<td>TO(1, 10)</td>
<td>TO(1, 10)</td>
<td>TO(1, 7)</td>
</tr>
<tr>
<td>PO(1, 7)</td>
<td>PO(1, 7)</td>
<td>PO(1, 7)</td>
<td>PO(1, 7)</td>
<td>PO(1, 9)</td>
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<td>TO(1, 11)</td>
<td>PO(1, 10)</td>
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<td>PO(1, 11)</td>
<td>PO(1, 11)</td>
<td>HO(1, 9)</td>
<td>HO(1, 9)</td>
<td>PO(1, 11)</td>
</tr>
<tr>
<td>TO(1, 12)</td>
<td>TO(1, 12)</td>
<td>TO(1, 12)</td>
<td>TO(1, 12)</td>
<td>HI(1, 10)</td>
<td>HI(1, 10)</td>
<td>TO(1, 12)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Run S-8</th>
<th>Run S-9</th>
<th>Run S-10</th>
<th>Run S-11</th>
<th>Run S-12</th>
<th>Run S-13</th>
<th>Run S-14</th>
</tr>
</thead>
<tbody>
<tr>
<td>TVPXR POWER</td>
<td>WNG POWER</td>
<td>WNG POWER</td>
<td>PBFP POWER</td>
<td>PBFP POWER</td>
<td>TVPXR POWER</td>
<td>TVPXR POWER</td>
</tr>
<tr>
<td>FWRE</td>
<td>FWRE</td>
<td>FWRE</td>
<td>FWRE</td>
<td>FWRE</td>
<td>FWRE</td>
<td>FWRE</td>
</tr>
<tr>
<td>TO(1, 7)</td>
<td>TO(1, 7)</td>
<td>TO(1, 7)</td>
<td>TO(1, 7)</td>
<td>TO(1, 7)</td>
<td>TO(1, 7)</td>
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</tr>
<tr>
<td>PO(1, 7)</td>
<td>PO(1, 7)</td>
<td>PO(1, 7)</td>
<td>PO(1, 7)</td>
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<td>TO(1, 11)</td>
<td>TO(1, 11)</td>
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<tr>
<td>PO(1, 11)</td>
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<td>PO(1, 11)</td>
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</tr>
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</table>
TABLE 9.5

VARIABLES TABULATED FOR CLOSED LOOP TESTS

<table>
<thead>
<tr>
<th></th>
<th>Run S-15</th>
<th>Run S-16</th>
<th>Run S-17</th>
</tr>
</thead>
<tbody>
<tr>
<td>POWER</td>
<td>POWER</td>
<td>POWER</td>
<td>POWER</td>
</tr>
<tr>
<td>PO(1,16)</td>
<td>PO(1,16)</td>
<td>PO(1,16)</td>
<td></td>
</tr>
<tr>
<td>FWRE</td>
<td>FWRE</td>
<td>FWRE</td>
<td></td>
</tr>
<tr>
<td>WNG</td>
<td>WNG</td>
<td>WNG</td>
<td></td>
</tr>
</tbody>
</table>
The simulated test runs were divided into three separate categories. This division was arbitrarily made in accordance with the program logic that was used to obtain the results that are illustrated in the graphs of Figure 9.1 through Figure 9.17.

Simulated test runs S-1 through S-8 were tests made at full load, that is 560 MW and were run as one group of tests. Simulated test runs S-9 through S-15 were made at one-half load, 280 MW and constituted the second major division of the simulated tests. Simulated test runs S-16 and S-17 were closed loop control system tests that were made at 560 MW.

In each of the simulated test runs, the system was subjected first to a scheduled load of 560 MW, which is incidently the base operating point. The control system was included in the first portion of all runs so that there would be less chance for transient conditions to occur that would be caused by parameters other than the controlled parameter. The reason that this was necessary was due to the fact that the simulated system was not tuned perfectly and some values of initial set points were not exactly at the balanced condition. As the digital simulator was being developed, many severe imbalances occurred and resulted in an unstable operation. The steady state operation period was selected to be 240 seconds (4 minutes) which is not quite long enough for the system to settle down. However, if a full 10 minutes were allowed for each simulated run the length of the computer runs would have been excessive. After the 4 minute steady state run the values of power, temperature, pressure, and feedwater flow were approximately normal with the fuel flow higher than normal.

For the eight simulated test runs at 560 MW, the computer variable OLT was changed from 0. to 1., after 240 seconds of operation, indicating that the open loop control tests had begun. Then, for the next 600 seconds (10 minutes), the process was allowed to change dynamically with no control system interaction with the process. That is, for example, if the superheater output temperature were to increase above the set point value, the control system
would ordinarily react by sensing the temperature deviation and adjusting the firing rate accordingly. Under the open loop condition, no such adjustment is made.

In order to conserve the handling time for the operator of the digital simulator, the program was arranged so that two test runs could be completed for each time that the program was submitted. Thus, appropriate logic was included in the computer program such that after 600 seconds of problem time, a new simulated test run was started. This is illustrated by inspection of the sample computer output given in Table 9.6.

The computer logic that makes provision for the different runs, is seen in Block 4, Block 15, and Block 22 of the computer program given in Appendix B, in Table B-1.

For the simulated tests that were made at 280 MW, provision was made to take the simulated unit from 560 MW to 280 MW under closed loop control system. This was accomplished by introducing a -280 MW scheduled load change after the steady state period of 240 seconds. The computer variable OLT was kept at 0, then while the system underwent the change from 560 MW to 280 MW. It took 450 seconds to reach 280 MW, with the current limits set on the unit load demand development, Block 7. Again, as with the case of the steady state run, the system had not actually settled down to 280 MW when that portion of the run was halted. As will be seen upon later inspection of the graphs, this effect influenced the results.

At this point in the run, that is 240 + 450 = 690 seconds, the two open loop simulated test runs were made as described earlier.

Run S-15 constituted the ramp change in load from 560 MW to 280 MW that was used to get the system from 560 MW to 280 MW.

Simulated test run S-1, which is comparable to the actual plant test A-1, corresponded to a 5% step increase in firing rate. As seen from Table 9.2, the computer variable that was changed was WNG, the natural gas fuel flow. It was changed from 5000 to 5250 K-ft³/hr. The other variables that are transferred from the control system to the process were set at their appropriate values for
that operating point and listed in Table 9.3. It is noted that the feedwater flow was held constant, as was attempted in the comparable actual plant test. The results of the test are shown in Figure 9.1. For all the graphs, the abscissa is Time (Min.) and the ordinates vary according to the test that was made. Many variables could be tabulated and plotted but only 4 or 5 representative functions were chosen to be plotted for each test run. If others would be desired, they could be easily obtained by appropriate programming. For the step increase in firing rate the power, and temperature increased as shown in the figure.

In test run S-2 the firing rate was decreased 5%. The controlled computer variable that is adjusted is WNG. It was reduced from 5250 to 5000 K-ft³/hr. The other variables that link the control system and process were maintained at the same level as in test S-1. With the reduction in the fuel flow, the power and temperature outputs of the superheater and reheater drop as shown in Figure 9.2.

Test run S-3 represents a decrease in feedwater flow. The results of the run are illustrated in Figure 9.3. The controlled computer variable is FWRE which is decreased from 3980 to 3780 K-lb/hr, a 5% decrease. An associated decrease in the output pressure of the boiler feed pump is included, as seen in Table 9.3. With a reduction in the feedwater flow the furnace output temperature and reheater output temperature increased, since the firing rate remained constant and the working fluid moved slower.

Figure 9.4 illustrates the results of test run S-4. In that test FWRE was increased from 3780 to 3980 K-lb/hr with the other control variables listed in Table 9.3. In this case the temperature decreased. The power decreased in the simulation since it is computed from the enthalpy change and the input enthalpy decreased corresponding to the temperature decrease.

In simulated test run S-5 the spray flow was increased from 0 to 100 K-lb/hr. Recall that the purpose of the spray flow is to reduce the superheater output temperature. This effect is noted in the results plotted in
Figure 9.5. In the graph the difference in the enthalpy before and after the spray is noted. The power decrease is due to the lowering of the enthalpy input to the turbine.

Test run S-6 is the companion to run S-5. Here the spray flow (computer variable WSTH) is decreased from 100 to 0 K-lb/hr. The model then indicates a subsequent rise in the superheater output temperature as anticipated. The other variables that were tabulated for this test are shown in Table 9.4.

Simulated test run S-7 illustrates the effect of a 2% reduction in the throttle valve position. The expected power decrease normally associated with a reduction in throttle valve position did not occur with the model. Instead, the power increased slightly. This was due to the fact that the feedwater flow was held constant and the density was not changed in the superheater. Hence, the power, which is a function of feedwater flow and enthalpy, followed the enthalpy trend. The enthalpy increased as is illustrated in the temperature graph of Figure 9.7 and a knowledge of the pressure variation.

The way the model is arranged, this particular controlled variation, that is, change in throttle valve position (TVPXR), was not similar to the actual tests. This may be accounted for by investigation of the turbine valve equation. The pressure output of the throttle valve is computed as a function of the throttle valve position, but this variable is used only in conjunction with the control system, as seen in Chapter 7, to regulate the feedwater flow. Since the control system is considered to be on manual, i.e., open loop, the throttle valve position (TVPXR) does not actually affect the process power. This discrepancy could be alleviated by programming the simulator to adjust the feedwater flow in conjunction with a throttle valve position change.

This lack of influence of the throttle valve position on the process is further illustrated by inspection of the results of simulated test S-8. The results of the test are shown in Figure 9.8 and it is recognized that a steady state condition has been reached. As can be seen from Table 9.3, test run S-7 and test run S-8 are run together and the process simply settled out to a steady state condition.
Simulated test run S-9 corresponds to a step decrease in firing rate of 5%. WNG is reduced from 2415 to 2294 K-ft$^3$/hr. This test is comparable to the actual test A-10. Here the load is reduced from 560 MW to 280 MW before the step change is introduced as was described earlier. The results are somewhat under the influence of the ramp change since the system was not allowed to attain a steady 280 MW before the step change occurred. However, the power and superheater output temperature reacted as in the actual tests. The results are shown in Figure 9.9.

The opposite effect is seen in the results of simulated test run S-10, as presented in the graphs in Figure 9.10. The controlled variable, WNG, is here increased 5%, from 2294 to 2415 K-ft$^3$/hr. The reheater output temperature increased in response to the increased firing rate.

In the test runs, S-9 through S-14, it is noted in Table 9.3 that the value of R12 is changed from 1.0 to 1.08. This is in accordance with the value of this combustion variable after the load had been reduced from 560 MW to 280 MW. This value was obtained by inspection of the output of simulated test run S-15, and, consequently, had to be run after run S-15.

In addition to the change in R12, the value of FWRE shown in column 3 of Table 9.3 for test run S-9 and S-10 was changed in the plotted run from 1660 to 1906.87 K-lb/hr. This was due to the fact that when the ramp from 560 MW to 280 MW was made, the feedwater flow came to this value rather than 1660 K-lb/hr. Also, the value of the throttle valve position (TVPXR) at 280 MW was 1.452, as obtained from run S-15.

In simulated test run S-11 the controlled variable was FWRE. Again, a 5% step decrease was initiated. The feedwater flow was reduced from 1906.87 K-lb/hr to 1811.53 K-lb/hr, while the output pressure of the boiler feed pump was held at the value of 3692 psia, its value at the end of the ramp to 280MW. With the decrease in the feedwater flow, the superheater and reheater temperatures increased as shown in Figure 9.11. Also, the power increased as in run S-3.
The following run, S-12, illustrated the effect of a 5% increase in feedwater flow. FWRE was changed from 1811.53 to 1902.53 K-lb/hr. The pressure of the output of the boiler feed pump was again held at 3692. psia. The results of the test are just the opposite of those in the preceding run and are shown in Figure 9.12. The temperatures follow those obtained in the actual test runs, which are described in Chapter 10.

Test runs S-13 and S-14 gave the response of the simulated system to a step change in the throttle valve position, TVPXR. As mentioned in relation to test runs S-7 and S-8, the method by which the simulated throttle valve change was introduced did not lead to results that were consistent with actual tests. This is seen in Figures 9.13 and 9.14 where the results of the tests are plotted. It is seen that while the variables did change, they were changing due to the continued influence on the process by the ramp change in scheduled load. This ramp from 560 MW to 280 MW had not settled out before the change was introduced into the system.

The results of test run S-15 are given in Figure 9.15. It is the ramp change in scheduled load from 560 MW to 280 MW with the control system operating. It is comparable to the actual test A-9.

Two simulated test runs were made with the control system operating with the simulated unit. Those runs, S-16 and S-17, involved a step change of -10 and +10 MW, respectively, after the steady state run. Under the influence of the control system, the simulated unit responded quickly to the change and the temperature and pressure deviation from the set point values was small. The results are shown in Figures 9.16 and 9.17.
Temperature (°F) | Temperature (°F) | Power (MW) | Feedwater flow (K-lb/hr)

Fig. 9.3 Step decrease in feedwater flow
Fig. 9.5: Step increase in spray flow

Temperature (°F) | Enthalpy (Btu/lb) | Power (MW) | Spray flow (K-lb/hr)

TO (1, 10) | HI (1, 10) | HO (1, 9) | Power

Time (Min.) | 2 | 2 | 2 | 2
| 4 | 4 | 4 | 4
| 6 | 6 | 6 | 6
| 8 | 8 | 8 | 8
| 10 | 10 | 10 | 10

HSH
Temperature (°F)

Power (MW)

Feedwater flow (K-lb/hr)

Fig. 9.11 Step decrease in feedwater flow
Fig. 9.12 Step increase in feedwater flow
FIG. 9.13 Step increase in throttle valve position

Temperature (°F)

TO (1.12)

Temperature (°F)

TO (1.11)

Power (MW)

Valve position

TVPr

1.45

1.48

1.5
Fig. 9. Step decrease in throttle valve position

Temperature (°F)

Power (MW)

Valve position
Fig. 9.16 Step change in scheduled load, 560 to 550 MW

Fuel flow (K-\text{ft}^3/\text{hr})

Temperature (°F)

Feedwater flow (K-lb/hr)

Pressure (psia)

Power (MW)

4700

5300

1000

1025

3860

3980

3550

3600

550

545

560
Fig. 9.17 Step change in scheduled load, 550-560 MW
SIMULATION OF A SUPERCRITICAL STEAM GENERATOR
WITH THE CONTROL SYSTEM
23 LUMPS, RUNGE-KUTTA INTEGRATION
PRINT THE NEW VALUE OF DEMWIT200.

WARNING... WARNING... WARNING...

WARNING... WARNING... WARNING...
FLOW DEMAND IS BELOW MINIMUM VALUE ACCEPTABLE
TRIP UNIT... TRIP UNIT... TRIP UNIT...

PROGRAM STOP AT 1770

USED 9.07 UNITS

SCCS   23:01   09/06/69

SIMULATION OF A SUPERCRITICAL STEAM GENERATOR
WITH THE CONTROL SYSTEM
23 LUMPS, RUNGE-KUTTA INTEGRATION
PRINT THE NEW VALUE OF DEMWIT700.

WARNING... WARNING... WARNING...

FLOW DEMAND IS ABOVE MAXIMUM VALUE ACCEPTABLE
TRIP UNIT... TRIP UNIT... TRIP UNIT...

PROGRAM STOP AT 2530

USED 9.07 UNITS

SCCS   23:02   09/06/69

Fig. 9.18 Computer output illustrating alarms
TABLE 9.6
SAMPLE OUTPUT FROM DIGITAL SIMULATOR
RUN S-1 AND RUN S-2

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**STEP INCREASE IN FIRING RATE INTRODUCED HFRE , RUN S-1**

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CHAPTER 10

DESCRIPTION OF THE ACTUAL PLANT TESTS AND THEIR COMPARISON WITH THE SIMULATED TESTS

The testing of a unit the size of the simulated plant, which is rated at 560 megawatts, is not a simple task. It is not an experiment such as one would perform in a laboratory that may have run several times before. There was considerable planning involved such as obtaining the various clearances in order to operate the plant in a way that was different than the scheduled operation. It is not desirable, under any circumstances, to subject a plant of this importance to operational changes which could even remotely cause the unit to trip from the line and thereby cause a disruption in the power flow to the grid.

The simulated unit is a base loaded unit for the system and hence could be operated only at certain loads during certain parts of the year. For example, during the summer months when the power system must generate its peak load in this area due to the hot weather and the high density of air conditioning loads, there would be no opportunity to test the unit at reduced loads. Also, scheduled maintenance for the unit had to be considered when making the plans for the test. After consultation with the operational personnel, it was determined that the best time in the year to test the plant would be in the spring just after its maintenance period and before other units were taken off the line so there would be a ready reserve in the system in case of an emergency. It was desired to test the unit at its full operational load and also at a reduced load. Since the unit would normally operate at its full operational load during the day time peaking periods, and since this unit is base loaded, full load conditions would be from 8 a.m. until sometime after 8 p.m. After that time, it would be possible to reduce the load on the unit to approximately half load.
The clearances were made to use the unit under these test conditions which will be prescribed later and a date was set for the testing. The tests were authorized and scheduled to be run on April 10, 1969.

It had been hoped to get a rather large step variation in the controlled variables of the plant; however, it was determined that a large variation in the controlled variables under an open loop condition would give rise to variable changes that would exceed the limits which the unit is capable of handling without causing an automatic trip out of the unit, which, of course, as has been mentioned, is completely out of the question. Hence, the changes in the controlled variables which could be allowed had to be limited to 5% or less, depending upon how the plant process functioned when the variable was changed. Most of the tests were to be run in an open loop configuration in order to determine what the response of the system was under these conditions. Only one test was run with the control system completely included.

In the ramp change from approximately full load to approximately half load, the control system was included in order that the variable changes within the unit would not exceed the tolerable values.

Within the simulated plant, there are approximately 500 test points from which data may be taken. Some of these points are measured parameters, such as throttle pressure, while some of the parameters are actually computed by use of the digital logging computer which measures other values and then uses an appropriate equation or averaging technique to obtain another variable of interest within the system. Two of these variables are spray flow and reheat steam flow.

There are four ways in which the information about the operation of the plant can be displayed to an operator or other people who would be interested in the operation data of the plant. The first is a direct readout of the various instruments, gauges, and charts which are available on the operator's control panel. These recorders are very important for the daily, minute-by-minute, operation of the plant, but are not particularly useful in terms of obtaining a large quantity of data for analysis at a later time. Due to the relatively low
sampling rate which could be obtained with this technique, only a very slow variable variation could be recorded.

Second, there were six available trend recorders which could be used. These actually consisted of two instruments with three pens in each instrument. Each pen had an adjustable gain and zero and upper maximum setting points. These trend recorders received their information by a logic selection through the digital computer which was used for logging; hence, any one of the 500 variables could be chosen for the trend recorder values.

A limitation of the trend recorder was that the gears which are used to drive these particular units had speeds which were relatively slow and hence the limitation, mentioned in the previous recording technique is also true here in that the variables which were recorded could only be observed if they had a very slow rate of change; therefore, fast transients could not be observed or recorded.

Third, there was a Friden flexowriter coupled to the CDC digital computer which is used for logging, and with the proper programming of the digital computer a list could be set up for the flexowriter which could accommodate 38 of the 500 available variables on one typewriter. In addition, another typewriter could be coupled to give an additional data logging capability. On those units, the variables are logged at a time interval of approximately one minute, hence, here again, if there would be a rapid transient, the effect might be lost by the relatively low sampling rate.

Most of the plant time constants are of the order of 7 minutes. This output proved to be the best and easiest method of getting the information for plotting the curves as will be seen in a later table.

If, for some reason, the typewriter malfunctioned during the test runs, the digital computer monitoring this output would sense the malfunction and would automatically start a high speed paper tape punch which would record the malfunction. The paper tape would then be fed through a flexowriter equipped with a paper tape reader and the output would be recorded in a typed form.
Fourth, for the standard output of the CDC logging computer there was a series of lists which were available. Each list could accommodate 8 variables and each list could be scanned at a different scanning rate. At the start of the preliminary test runs, 5 lists were formed with variable scan rates. List 1 had a 10 second rate, with list 2 to 5 having a 20 second scan rate. The variables were arranged in an order which was easily identifiable, in that they were set up in such a manner that they followed sequentially around the fluid flow path of the unit.

After the first preliminary test run was made it was determined that the lists had a priority within the computer, such that only the first list could be completed each time with the scan rates which were originally chosen and with the number of variables which were desired. Hence, the lists were rearranged into a more workable form and shortened so that they could be accommodated with the scan rates desired in order to observe transients in particular variables of interest.

After some of the variables were taken off the CDC lists, they were added to the flexowriter list at the one minute scan rate. A list of the most important recorded plant variables is given in Table 10.1. The three columns there represent, first, the computer variable mnemonic name, second, the plant function and third, the units. An example of this is the throttle pressure; the mnemonic name given it in the computer is TP00X. It is a mathematically weighted average of three measured quantities and is a pressure.

Suffix A indicates direct analog measurement

- X indicates mathematical weighted average of 3 A values
- E indicates mathematical calculation based on X and/or A quantities

After association of the computer variable name with the plant function, it becomes relatively easy to familiarize the readout variables with the associated quantity.

In Table 10.2 is given the lists of variables for the test runs from the CDC computer. There were finally three lists used and they are given in the table, with the computer variable name only being listed. This list can be cross-referenced with Table 10.1.
### TABLE 10.1

**LIST OF RECORDED PLANT VARIABLES**

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The output from the logging computer was in the form of a strip of paper which had a high speed printer printing the points as they were logged and stored in the computer. A sample output from the computer is included in Table 10.3. The plant time is given and, after the plant time, the list number and its associated variable name, quantity and units are given. While in the table, the three lists appear side by side as they come out of the computer on a narrow sheet of paper in a sequential fashion. In Table 10.4 is a sample output from the flexo-writer and on it it is seen that the time is given and the headings of the appropriate variables followed by their values. No units are given in this printout.
TABLE 10.2
LISTS OF VARIABLES FOR TEST
RUNS FROM CDC COMPUTER

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<tr>
<th>List 1</th>
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<th>List 3</th>
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</thead>
<tbody>
<tr>
<td>GEO3A</td>
<td>BC10A</td>
<td>WT20X</td>
</tr>
<tr>
<td>TPOOX</td>
<td>PP20A</td>
<td>BT10X</td>
</tr>
<tr>
<td>WF20X</td>
<td>BP10X</td>
<td>BT70X</td>
</tr>
<tr>
<td>FF05X</td>
<td>TP06A</td>
<td>BT74A</td>
</tr>
<tr>
<td>BT85X</td>
<td>HP10A</td>
<td>TTOOA</td>
</tr>
<tr>
<td>BFOOX</td>
<td>TP10A</td>
<td>TT10A</td>
</tr>
<tr>
<td>WP70A</td>
<td>PP03A</td>
<td>TT36A</td>
</tr>
<tr>
<td>TS17A</td>
<td>PTOOA</td>
<td>BT04X</td>
</tr>
</tbody>
</table>

In Figure 10.1 is the sample of the trend recorder strip chart. As mentioned earlier, the trend recorders would accomodate up to six of the 500 variables available. A list of those variables chosen and their associated properties is given in Table 10.5.

Because of the importance of associating the variables measured with the process, the fluid and gas flow paths figure which was included in an earlier section is repeated here with the additional notes on the figure of the computer variables and their relative position in the process. This is Figure 10.2.

During the course of the tests, sixteen test runs were made of which two runs were considered to be trial runs to adapt those involved in the tests with the procedure and, in particular, to determine the best use of the various forms of the output which was available for recording the variables of interest. A list of the 14 test runs is given in Table 10.6. As indicated by inspection of Table 10.6 there were three fundamental variables which were controlled
### TABLE 10.3

**CDC DIGITAL COMPUTER PRINT OUT OF PLANT VARIABLES**

<table>
<thead>
<tr>
<th></th>
<th>1831 57 REPEAT RDOT</th>
<th>9 HP10A 548 PSI G</th>
</tr>
</thead>
<tbody>
<tr>
<td>LIST</td>
<td></td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>TS17A 92 PRCT</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>HP70A 83.2 KLBH</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>BLOOX 4885 KLBH</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>BT85X 1025 DEGF</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>FF05X 4510 MSCFH</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>WJ20X 3457 KLBH</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>TMOX 3428 PSI G</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>GE03A 496.4 MW</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>4706A 2041 PSI G</th>
<th>3 BT10X 3627 PSI G</th>
</tr>
</thead>
<tbody>
<tr>
<td>LIST</td>
<td></td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>BT04X 83 DEGF</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>TJ36A 117 DEGF</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>TJ10A 1011 DEGF</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>TT00A 1015 DEGF</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>BT74A 792 DEGF</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>BT70X 803 DEGF</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>BT10X 760 DEGF</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>WT20X 476.2 DEGF</td>
<td></td>
</tr>
</tbody>
</table>
TABLE 10.4
SAMPLE FLEXOWRITER OUTPUT FROM PLANT COMPUTER
1 TIME
1 2322 00

GE03A
249.6

TPOOX
3536.

WF20X
1665.

FF05X
2373.

BT65X
1016.

BFOOX
2367.

WP70A
0.0

TS17A
55.

BCIOA
2.0

PP20A
3543.

BPlOX
3525.

TP06A
1003.

HPIOA
2S3.

TPIOA
253.

PP05A
547.

P

T : : A V.720V.
3:6. 414.4

1 TIME
1 2329 05

GE03A
250.2

TPOOX
3529.

WF20X
1671.

FF05X
2206.

BT85X
1016.

BFOOX
2179.

WP70A
0.0

TS17A

BC10A
2.0

PP20A

HPIOA

TPIOA

FP03A

PT::-

354/,.

BP10X
3539.

TP06A

55.

10C6.

234.

253.

547.

305.

•,rr?:<
414.1

1 2331.04

247.4

3479.

1668.

2224.

1007.

2153.

0.0

55.

2.2

3488.

3494.

993.

231.

255.

;47_

3C5.

413.6

1 2332 04

24S.1

3454.

1658.

2215.

1003.

2148.

0.0

55.

2.1

3463.

3472.

995.

230.

255.

55C.

5:5.

413.2

75:.

'-..

1 2333 05

245.9

3437.

1676.

2203.

999.

2157.

0.0

55.

2.1

3445.

3446.

993.

279.

254.

545.

3C5.

412.5

7tD.

7S£.

1 2334 04

246.4

3425.

1664.

2202.

"94.

2158.

0.0

55.

2.1

3433.

3434.

991.

279.

254.

547.

3C4.

412.7

~>C "

733.

1 2335 05

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3425.

1660.

2202.

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2.1

3431.

3423.

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279.

254.

54:.

304.

412.6

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-ii'-

1 2336 04

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3411.

1657.

2206.

984.

2177.

0.0

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2.1

3419.

3423.

993.

2S0.

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549.

304.

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• J :

1 2337 04

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1678.

2198.

978.

2162.

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55.

2.1

3409.

3412.

996.

281.

257.

546.

412.9

751.

77:.

1 2338 04

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1684.

2202.

972.

2146.

0;0

55.

2.0

3397.

3402.

995.

282.

258.

552.

305.

412.9

75C.

765.

1 2339 05" 247.8

3373.

1660.

2207.

965.

2128.

0.0

55.

2.0

3378.

3589.

999.

284.

259.

543.

3:5.

413.1

749.

754.

1 TIME

577:'.

GE03A
247.7

TPOOX

WF20X

FF05X

BT85X

BFOOX

UP70A

TS17A

BCIOA

PP20A

BPlOX

TPIOA

PPC3A

P702A

WT2CX

3T1C/.

3349.

1671.

2212.

959.

2160.

0.0

55.

2.0

3355.

3366.

TP06A
10G4.

HPIOA

1 2340 05

'

236.

259.

549.

3G5.

413.5

747.

7

1 2342 04

246.7

3312.

1650.

2202.

949.

2141.

0.0

55.

2,0

3316.

3331.

1001.

286.

260.

553.

3C6.

413.5

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•57.

.1 2343 04

245.8

3292.

1659.

2203.

945.

2148.

0.0

55.

2.1

3299.

3313.

996.

284.

25«.

544.

306.

413.5

741.

755.

I 2344 06

244.8

3274.

1631.

2207.

942.

2156.

0.0

"55.

2.0

3233.

3293.

*96.

2C5.

259.

355.

3C6.

03.1

741.

1 Z2*

1 2345 04

243.9

3251.

1654.

2205.

940.

2150.

0.0

55.

2.1

3259.

3275.

989.

233.

253.

556.

506.

-.12.:

• - *?•

"32.

1 2346 05

241.8

3233.

1673.

2207.

938.

2138.

0.0

55.

990.

282.

257.

554.

306.

412.5

733.

75D.

240.5

3212.

1685.

2300.

938.

2353.

0.0

55.

2 . 1 . 3241.
2.1 3223.

3257.

1 2347 04

3243.

984.

280.

254.

545.

305.

412.1

736.

749.

j 2348 05

240.7

3227.

1672.

2340.

938.

2437.

0.0

55.

2.5

3233.

3235.

979.

279.

255.

55U

505.

411.5

735.

743.

1 2349 05

241.2

3254.

1694.

2340.

940.

2480.

0.0

55.

3.2

3264.

3252.

988.

280.

255.

549^

305.

411.5

733.

749.

1 2350 0 4

243.8

3273.

1664.

2337.

940.

2472.

0.0

55.

3.2

3280.

3277.

993.

282.

258.

550.

305.

411.5

733.

745.
BT7CX

61.

I TIME

GE03A

TPOOX

UF20X

FF05X

BT85X

BFOOX

WP70A

TS17A

BCIOA

PP20A

BPlOX

TP06A

HPIOA

TPIOA

PP03A

PTOOA

WT20X

3710X

1 2351 05

245.5

3293.

1690.

2338.

936.

2443.

0.0

55.

3.1

3303.

3299.

994.

285.

261.

549..

3C5.

412.3

734.

746.

L2353 04

249.4

3306.

16J2.

2366.

931.

2444.

0.0

55.

2.9

3321.

3319.

1011.

2?C.

255.

549.

3C6.

413.5

?3i.

747.
to
to


Changing power from 250 to 500 MW

Main Steam Temperature

Reheat Steam Temperature

Fig. 10.1 Sample output from trend recorder
Fig. 10.2 Fluid and gas flow paths with computer variables for supercritical once-through unit
independently under open loop testing procedures. Those four variables were a firing rate change, a feedwater flow change, a turbine valve position change, which corresponds to a load change, and a spray flow change. Each of these variables was given a step increase and a step decrease at the approximate full capacity of the machine and at approximately one-half capacity of the unit. In addition to these runs, test run #9 was utilized as a closed loop control system run in which the load was changed in an approximate ramp from approximately 500 megawatts to approximately 250 megawatts. In each of the test runs there were many variables for which data was gathered and these variables can be seen from the partial list of tabulated data in Table 10.7.

In test run #1, the independent controlled variable was the fuel flow. This variable label is FF05X and was given a step increase of approximately 4-1/2% by increasing the fuel rate from 4,565 to 4,775 thousands of cubic feet per hour. The increase was obtained by manually sending signals to the fuel gas valve controller from the control room.

In the operation of the plant there was no way to give an exact step increase to a particular variable since there were toggle switches which sent signals to
TABLE 10.6

TEST RUNS OF SIMULATED UNIT

<table>
<thead>
<tr>
<th>RUN</th>
<th>PROCESS VARIABLE CHANGES</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>STEP INCREASE IN FIRING RATE AT ≈ 500MW</td>
</tr>
<tr>
<td>2</td>
<td>STEP DECREASE IN FIRING RATE AT ≈ 500MW</td>
</tr>
<tr>
<td>3</td>
<td>STEP DECREASE IN FEEDWATER FLOW ≈ 500MW</td>
</tr>
<tr>
<td>4</td>
<td>STEP INCREASE IN FEEDWATER FLOW ≈ 500MW</td>
</tr>
<tr>
<td>5</td>
<td>SPRAY FLOW DECREASE FROM 81.1→0 KLB/HR AT ≈ 500MW</td>
</tr>
<tr>
<td>6</td>
<td>SPRAY FLOW INCREASE FROM 0→104.4 KLB/HR AT ≈ 500MW</td>
</tr>
<tr>
<td>7</td>
<td>STEP DECREASE TB VALVE POSITION AT ≈ 500MW</td>
</tr>
<tr>
<td>8</td>
<td>STEP INCREASE TB VALVE POSITION AT ≈ 500MW</td>
</tr>
<tr>
<td>9</td>
<td>RAMP CHANGE IN LOAD 500→250 MW</td>
</tr>
<tr>
<td>10</td>
<td>STEP DECREASE IN FIRING RATE AT ≈ 250MW</td>
</tr>
<tr>
<td>11</td>
<td>STEP INCREASE IN FIRING RATE AT ≈ 250MW</td>
</tr>
<tr>
<td>12</td>
<td>STEP DECREASE IN FEEDWATER FLOW AT ≈ 250MW</td>
</tr>
<tr>
<td>13</td>
<td>STEP INCREASE IN FEEDWATER FLOW AT ≈ 250MW</td>
</tr>
<tr>
<td>14</td>
<td>STEP INCREASE IN THROTTLE VALVE POSITION AT ≈ 250MW</td>
</tr>
</tbody>
</table>

the DC motors which controlled the valves in question and, until the control variable was measured and a read out was available, it was not known how large a step had been obtained. Throughout the tests, then, there were times when it was necessary to readjust a variable, while a test was in progress, in order to keep another dependent variable from reaching a limit position which would cause a trip of the unit due to unsafe operating practices which could not be tolerated by the system to which the unit was connected. In Table 10.8 the lists of the variable plots for each test run of the simulated unit was given. In this case, considering test run #1, it is noted that the first 4 variables listed under column run #1, are FF05X, GE03A, WF20X, and PP20A. Now, these four variables are plotted on the first page of Figure 10.3. On the second page of Figure 10.3 are given the variables WT20X, BT10X, BP10X, TT00A, and in a similar fashion the subsequent pages of Figure 10.3 represent the rest of the list under run #1 of Table 10.8. Hence, the use of Table 10.8 and the
### TABLE 10.7

**PARTIAL LIST OF TABULATED DATA FROM TEST RUN (1)**

<table>
<thead>
<tr>
<th>Actual Time</th>
<th>Plotting Time (Min)</th>
<th>Generator Power GEO03A (MW)</th>
<th>Throttle Pressure TPOOX (PSIG)</th>
<th>Feedwater Flow WF20X (kIb/hr)</th>
<th>Fuel Gas Flow FF05X (kIb/hr)</th>
<th>Air Flow BFOOX (kIb/hr)</th>
</tr>
</thead>
<tbody>
<tr>
<td>16:56:23</td>
<td>0</td>
<td>504.1</td>
<td>3508</td>
<td>3618</td>
<td>4538</td>
<td>4937</td>
</tr>
<tr>
<td>16:57:23</td>
<td>1</td>
<td>507.0</td>
<td>3526</td>
<td>3664</td>
<td>4714</td>
<td>5092</td>
</tr>
<tr>
<td>16:58:21</td>
<td>2</td>
<td>508.8</td>
<td>3536</td>
<td>3627</td>
<td>4717</td>
<td>5078</td>
</tr>
<tr>
<td>16:59:19</td>
<td>3</td>
<td>510.7</td>
<td>3557</td>
<td>3619</td>
<td>4726</td>
<td>4975</td>
</tr>
<tr>
<td>17:00:25</td>
<td>4</td>
<td>512.8</td>
<td>3555</td>
<td>3591</td>
<td>4733</td>
<td>5069</td>
</tr>
<tr>
<td>17:01:21</td>
<td>5</td>
<td>512.2</td>
<td>3562</td>
<td>3589</td>
<td>4753</td>
<td>5069</td>
</tr>
<tr>
<td>17:02:23</td>
<td>6</td>
<td>513.5</td>
<td>3555</td>
<td>3592</td>
<td>4789</td>
<td>5061</td>
</tr>
<tr>
<td>17:03:22</td>
<td>7</td>
<td>513.7</td>
<td>3542</td>
<td>3602</td>
<td>4791</td>
<td>5068</td>
</tr>
<tr>
<td>17:04:19</td>
<td>8</td>
<td>514.1</td>
<td>3562</td>
<td>3606</td>
<td>4754</td>
<td>5066</td>
</tr>
<tr>
<td>17:05:22</td>
<td>9</td>
<td>514.8</td>
<td>3577</td>
<td>3600</td>
<td>4756</td>
<td>5062</td>
</tr>
<tr>
<td>17:06:24</td>
<td>10</td>
<td>516.9</td>
<td>3564</td>
<td>3602</td>
<td>4747</td>
<td>5089</td>
</tr>
<tr>
<td>17:07:23</td>
<td>11</td>
<td>518.0</td>
<td>3566</td>
<td>3606</td>
<td>4736</td>
<td>5103</td>
</tr>
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<td>17:08:26</td>
<td>12</td>
<td>518.0</td>
<td>3566</td>
<td>3596</td>
<td>4800</td>
<td>5080</td>
</tr>
<tr>
<td>17:09:20</td>
<td>13</td>
<td>521.0</td>
<td>3562</td>
<td>3594</td>
<td>4798</td>
<td>5059</td>
</tr>
<tr>
<td>17:10:19</td>
<td>14</td>
<td>522.9</td>
<td>3571</td>
<td>3604</td>
<td>4746</td>
<td>5063</td>
</tr>
<tr>
<td>17:11:21</td>
<td>15</td>
<td>524.5</td>
<td>3575</td>
<td>3603</td>
<td>4748</td>
<td>5105</td>
</tr>
<tr>
<td>17:12:22</td>
<td>16</td>
<td>521.3</td>
<td>3582</td>
<td>3603</td>
<td>4749</td>
<td>5092</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Actual Time</th>
<th>Plotting Time (Min)</th>
<th>HP EWH Out Press PP20A (PSIG)</th>
<th>Furn Exit Press BP10X (PSIG)</th>
<th>1 ST STG Press TPO6A (PSIG)</th>
<th>RH INPUT Press HP10A (PSIG)</th>
<th>RH OUT Press TP10A (PSIG)</th>
</tr>
</thead>
<tbody>
<tr>
<td>16:56:23</td>
<td>0</td>
<td>3848</td>
<td>3675</td>
<td>2104</td>
<td>565</td>
<td>530</td>
</tr>
<tr>
<td>16:57:32</td>
<td>1</td>
<td>3869</td>
<td>3675</td>
<td>2104</td>
<td>565</td>
<td>532</td>
</tr>
<tr>
<td>16:58:30</td>
<td>2</td>
<td>3890</td>
<td>3711</td>
<td>2110</td>
<td>570</td>
<td>534</td>
</tr>
<tr>
<td>16:59:21</td>
<td>3</td>
<td>3896</td>
<td>3719</td>
<td>2123</td>
<td>570</td>
<td>535</td>
</tr>
<tr>
<td>17:00:49</td>
<td>4</td>
<td>3905</td>
<td>3726</td>
<td>2122</td>
<td>570</td>
<td>535</td>
</tr>
<tr>
<td>17:01:23</td>
<td>5</td>
<td>3906</td>
<td>3726</td>
<td>2122</td>
<td>570</td>
<td>533</td>
</tr>
<tr>
<td>17:02:38</td>
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<td>3734</td>
<td>2121</td>
<td>569</td>
<td>534</td>
</tr>
<tr>
<td>17:03:30</td>
<td>7</td>
<td>3921</td>
<td>3741</td>
<td>2125</td>
<td>568</td>
<td>532</td>
</tr>
<tr>
<td>17:04:34</td>
<td>8</td>
<td>3926</td>
<td>3743</td>
<td>2129</td>
<td>568</td>
<td>531</td>
</tr>
<tr>
<td>17:05:14</td>
<td>9</td>
<td>3928</td>
<td>3748</td>
<td>2125</td>
<td>567</td>
<td>532</td>
</tr>
<tr>
<td>17:06:26</td>
<td>10</td>
<td>3943</td>
<td>3726</td>
<td>2134</td>
<td>569</td>
<td>534</td>
</tr>
<tr>
<td>17:07:25</td>
<td>11</td>
<td>3947</td>
<td>3768</td>
<td>2136</td>
<td>569</td>
<td>535</td>
</tr>
<tr>
<td>17:08:38</td>
<td>12</td>
<td>3945</td>
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### TABLE 10.8

**LISTS OF VARIABLE PLOTS FOR TEST RUN OF SIMULATED UNIT**

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subsequent figures which give the plots for the various runs make it easy to find a particular variable which one would like to investigate for a particular test.

At first, in a preliminary run, the feedwater flow was held on automatic control, but since this would not maintain constant feedwater flow it was decided to put the feedwater flow on manual control and then to try to hold it manually with appropriate pulses to the boiler feed pump at the level of feedwater flow which it had been operating at the beginning of the test run. This was to insure that the effects which would be seen throughout the system would be the effects of the fuel flow changes and not due to a change in feedwater flow caused by a subsequent pressure change which would have resulted from the fuel flow change. This interconnection between the pressure, pump speed and feedwater flow was described in an earlier section.

It is thus noted that the feedwater flow, WF20X, is somewhat erratic during the step change, and just following the step change. This was due to the manual control which was used to try to hold the feedwater flow at a constant value. Of particular interest in response to the step increase of fuel flow is that the output power increased. The high pressure feedwater heater output pressure, as well as the other pressures which were monitored during the test run, increased which was caused by the increased steaming conditions which were the result of the increase in fuel input. This is also true of the temperatures and, in particular, notice the furnace exit temperature, BT10X, and the throttle steam temperature. The test run was 22 minutes long from the step input to the termination of the test, at which time the system was fairly stable and then preparations were made for the second test.

In test run #2, there was a step decrease in the firing rate, where the fuel flow was changed from approximately 4,708 thousands of cubic feet per hour to approximately 4,481 thousands of cubic feet per hour which corresponds to a negative step of approximately 4.8%. The feedwater flow was held fairly constant as is seen from its curve in Figure 10.4. As expected, this case was exactly the reverse of the results obtained in test number 1, and in this case
the generated power decreased in response to the negative step change in the fuel flow. The various pressures monitored through this system decreased and the temperatures decreased.

In test run #3, there was a step decrease in the feedwater flow following the attainment of steady state from the previous tests. The feedwater was decreased from approximately 3,580 thousands of pounds per hour to approximately 3,453 thousands of pounds per hour which corresponds to an approximate 3.5% decrease in the feedwater flow. An approximate step was obtained as indicated in Figure 10.5. The decrease in the water flow was made by manually adjusting the control switch which controlled the boiler feed pump motor speed controller. The decrease in feedwater flow should cause an increase in temperatures throughout the system under open loop control conditions, and this was experienced as seen from Figure 10.5.

The pressures follow, generally, the pressure curve of the high pressure feedwater heater output pressure which was expected, but it was fairly erratic in its resulting form with a roller coaster effect; that is, a drop down at 6 minutes, but back up again at 12 minutes of plant time. This must be attributed to the change in the output pressure of the boiler feed pump primarily, as the controller action on the boiler feed pump to reduce its flow caused a new operating point on the characteristic curves. (As has been expected throughout the simulation, the pressure variation is the part of the plant simulation which appears to be the most troublesome to simulate properly and this is indicated by the erratic nature of the pressure in this particular run.)

In run #4, there was a step increase in the feedwater flow from approximately 3,476 to 3,595 thousands of pounds per hour which was handled in the same way as the previous run. This corresponds to an increase of approximately 3.3% in the feedwater flow. The approximate step change is shown in Figure 10.6 for this run. It is noted that the furnace exit temperature and other temperatures in the system generally decrease as expected due to the increased feedwater flow and, hence, with the same amount of firing rate
there is more water to heat and it does not get as hot as it would have if there had not been as much water.

Again, the pressures generally follow the curve of the high pressure feedwater heater output pressure which is representative of the output pressure of the boiler feed pump itself. It, again, is rather erratic. It is noted in this run that the reheat input pressure which corresponds to the output pressure of the high pressure turbine has a steady exponential increase as does the reheater output pressure.

Test run #5 used a step decrease in the superheater spray flow from 81.1 to 0 thousands of pounds per hour at a load corresponding to about 500 megawatts. The curves obtained from the plant data are shown in Figure 10.7. The spray flow was changed by simply turning off the spray valve by remote control from the control panel. The valve position corresponding to the 81.1 thousands of pounds per hour was approximately 50% open, and the spray flow step is indicated in the first figure of Figure 10.7. The firing rate was held constant as well as the turbine valve position. The feedwater flow was adjusted to keep it at a constant value. It will be noted that the feedwater flow and gas flow did vary slightly during these runs. The expected change would be in comparison with the superheater temperature before and after the spray valve which are indicated in the second and third graphs of Figure 10.7 and the difference is noted, as expected after the spray was shut off, that the temperature after the spray increased relatively to the superheater temperature before the spray. However, the change was not very large in this case.

It should be noted that the steam temperature portion of the integrated boiler turbine master of the control system was on manual and, hence, there was not correcting action of the firing rate for a change in the temperature.

Test run #6 is a step increase in the spray flow from the previously obtained zero spray flow to full spray flow on which corresponded in this case to 104.4 thousands of pounds per hour. This approximate step increase is indicated in Figure 10.8 and again the temperatures before and after the spray valve between the primary and finishing superheaters changed, but only a slight
deviation was detected. The general trend, as indicated in graph 3, was for the immediate response of the spray to decrease the temperature after the spray and this was obtained. As was mentioned in the earlier discussions, there would not be a permanent change and this was substantiated with this test run. The important variable, the throttle steam temperature, gave the deviation as expected. When the spray was added, the temperature went down.

Test run #7 corresponds to a step decrease in the turbine valve position at 500 megawatts. In this test, the turbine valve setting, whose variable is TS17A, was decreased approximately 1%. This decrease of turbine valve setting is comparable to decreasing the load on the unit. However, even a 1% decrease in a turbine valve setting gave rise in this test to an approximate 23 megawatt drop in power which was too large to be tolerated for the unit due to excessive pressure and temperature deviations. Hence, the throttle valve had to be readjusted which is indicated by the generated output power curve given in Figure 10.9. Upon investigation of the equation which relates the quantity PO(1,16), which is the output pressure of the throttle valve and its relationship with the throttle valve position, it is seen that the simulated value and the actual plant value is correct. As it turns out, there was not a measurement of throttle valve output pressure during the test but the first stage throttle pressure is indicated on graph 4 of Figure 10.9. For a corresponding decrease in the throttle valve position, which was the case in this run, there would be a decrease in the output pressure of the throttle valve which is indicated in this run #7 as a decrease in the first stage pressure. This is then as would be expected. The increase in the throttle pressure, as shown in the test, which is the pressure at the input to the valve as is shown in the placement of the variables on Figure 10.2. This input to the throttle valve pressure increased and this is reflected by the feedwater flow decrease as seen in Figure 10.9. There was an increase in the throttle pressure because there was less feedwater flow, hence, there was a smaller resistance drop due to the flow through the rest of the system.
Test run #8 corresponds to a step increase in the turbine valve position. The turbine valve was increased from an opening corresponding to approximately 90% to 92% opening with a step shown in Figure 10.10. This step change in turbine valve position corresponds to a load change from 493.2 megawatts to 508.5 megawatts or a +15.3 megawatt change. This corresponds to approximately 3.1% change in power output of the unit. The results of this test are just the opposite of the previous test; specifically, the output to the throttle valve pressure increased rather than decreased as it did earlier in test run #7, and the throttle valve pressure, that is the pressure at the input to the throttle valve, decreased in this run.

Test run #9 was a run, as mentioned earlier, which was carried out under closed control loop operation and was done under the close guidance of the central dispatcher at Pine Bluff, Arkansas. This test run was a ramp change in load from approximately 500 megawatts to 250 megawatts. The results of the run are given in Figure 10.11. Several of these graphs of Figure 10.11 have already been discussed in the previous sections where they were used to determine specific relationships between the variables.

Test run #10 was a step decrease in firing rate at a load of approximately 250 megawatts. The firing rate was decreased from approximately 2,370 to 2,205 thousands of cubic feet per hour of natural gas flow which corresponded to an approximate 7% decrease in the firing rate. The results of this run are comparable to that of run #2. Figure 10.12 shows the results.

Test run #11 was a step increase in firing rate from approximately 2,206 to 2,370 thousands of cubic feet per hour of natural gas flow which corresponded to approximately a 7.3% increase in the firing rate. Figure 10.13 shows the results of that test run and they are comparable to test run #1.

Test run #12 corresponds to a step decrease in the feedwater flow at a generator rating of approximately 250 megawatts. The results of this test are given in Figure 10.14. There was a decrease from 1,670 to 1,569 thousands
of pounds per hour in the feedwater flow which corresponds to approximately a 6% decrease. The results are comparable to those in test run #3, which were run at approximately 500 megawatts.

Test run #13 corresponds to a step increase in the feedwater flow from approximately 1,580 to 1,647 thousands of pounds per hour or an approximate 4.2% increase in the feedwater flow. The results of this test are given in Figure 10.15 and the results there are comparable to those obtained in test run #4.

Test run #14 was a step increase in the throttle valve position of approximately 1% and the results of this test are shown in Figure 10.16 which are comparable to the results of a similar step increase in turbine valve position as indicated for run #8.

The changes in the spray flow were not made formally into test runs for the reduced load conditions due to the fact that very little spray water entered the superheater with the spray valve full on.

In addition to this series of formal tests which were made at the plant under study, there were several other sessions with operating personnel and management in which various plant constants and gains of the control system, repeat rates, and various readouts were obtained from the CDC computer in order to determine such items as tube metal temperatures and steam chest pressures as were needed throughout the simulation.
Fig. 10.3 Step increase in firing rate
Fig. 10.3 Continued
Fig. 10.3 Continued
Pie: 10.4 Step decrease in firing rate

Time (min.)
Fig. 10.4 Continued

- RH OUT PRESS (PSIG)
- RH INPUT TEMP (PSIG)
- RH INPUT TEMP (DEG F)
- THROTTLE PRESS (PSIG)
Fig. 10.5 Step decrease in feedwater flow
Fig. 10.5 Continued
Fig. 10.5 Continued
Fig. 10.6 Step increase in feedwater flow
Fig. 10.6 Continued
Fig. 10.6 Continued
Fig. 10.7 Spray flow decrease from 81.1 to 0 Klb/hr
Fig. 10.7 Continued

THROTTLE PRESS (PSIG)

THROT STM TEMP (DEG F)

FURN EXIT PRESS (PSIG)

FURN EXIT TEMP (DEG F)
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Fig. 10.7 Continued
Fig. 10.8 Spray flow increase from 0 - 104.4 Klb/hr.

- Generator Power (MW)
- SH Temp A-Spray (Deg F)
- SH Temp B-Spray (Deg F)
- SH Spray Flow (Klb/hr)
Fig. 10.8 Continued
RH OUT TEMP (DEG F)

RH OUT PRESS (PSIG)

RH INPUT TEMP (DEG F)

RH INPUT PRESS (PSIG)
Fig. 10. 9 Step decrease TB valve position
Fig. 10.9 Continued
Fig. 10.9 Continued
Fig. 10.9 Continued
Fig. 10.10. Step increase TB valve position
Fig. 10.10 Continued
Fig. 10.11 Ramp change load 500 → 250 MW
Fig. 10.11 Continued

SH SPRAY FLOW (KLB/HR)

SH TEMP A-SPRAY (DEG F)

SH TEMP B-SPRAY (DEG F)

RH OUT TEMP (DEG F)
Fig. 10.11 Continued
Fig. 10.12 Step decrease in firing rate @ 250 MW
Fig. 10.12 Continued
Fig. 10.12 Continued
Fig. 10.13 Step increase in firing rate @ 250 MW
Fig. 10.13 Continued
THROTTLE PRESS (PSIG)

THROT STM TEMP (DEG F)

FURN EXIT PRESS (PSIG)

FURN EXIT TEMP (DEG F)

Fig. 10.14. Continued
Fig. 10.14 Continued
Fig. 10.14 Continued
Fig. 10. 15 Step Increase in Feedwater Flow @ 250 MW

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Note: The diagram shows the changes in various parameters over time.
Fig. 10.15 Continued
Fig. 10.15 Continued
RH INPUT TEMP
(DEG F)

SH TEMP A-SPRAY
(DEG F)
Fig. 10.16 Step increase in throttle valve position @ 250 MW
Fig. 10.16 Continued
Fig. 10.16 Continued
Fig. 10.16 Continued
CHAPTER 11

CONCLUSION

The research that has been done in the development of the digital simulator for a once-through supercritical steam generator has satisfied its purpose. The goal of the study was to investigate thoroughly the operational aspects of the steam-electric generating unit and to develop a simulator that could be used to study the operation of the unit.

The various aspects of the simulated plant were described in detail and mathematical equations were developed that could be used in the digital simulator computer program.

The unit that was simulated was tested under actual operating conditions as was described in detail in Chapter 10. Many additional variables were included and plotted in relation to the 14 actual test runs performed so that the reader would have an opportunity to compare these tests with similar tests of other units.

In Chapter 9 the results of the 17 simulated test runs that were performed using the digital simulator were recorded. The simulator tests were described there and it is noted that simulated tests comparable to actual tests were performed (see Table 9.2). By comparison of the results of the actual and simulated series of tests, a general agreement is found.

The discrepancies in the actual and simulated tests may be attributed to the following facts. In the actual tests, a step change could not be made but instead, a manual adjustment was made that approximated a step change. Also, after an adjustment was made, the operation dictated a change in the controlled parameters which then caused a secondary effect that caused the output results to be difficult to interpret. In the simulated tests, the time constants of the process under open loop conditions were relative long compared to the
10 minute run times and some of the variables that were recorded were still under the influence of the previous test conditions when the step change was initiated. As mentioned in Chapter 7, the feedwater flow control was simulated under the assumption that the throttle pressure was constant in order to maintain solution stability when the process and control system were interconnected. In addition, the recorded simulated test runs were made with the assumption that the density of the working fluid was constant.

The computer program for the entire digital simulator is given in Appendix B and as has been described in Chapter 8, provision for the density change has been included. If the density change was included in the simulated test runs where the throttle valve position was changed, the computed power would have been comparable to the actual runs. In the simulated and actual test runs, it is noted that feedwater flow is held constant for this run. The computed power is a function of the enthalpy and flow and with the flow held constant the power follows the trend of the enthalpy.

The operation of the digital simulator is described in Chapter 8. Modifications could be made to it to suit the particular application desired. If other units were to be simulated, the same procedures could be followed that have been described in detail in this thesis, with the proper values substituted for the simulated plant constants.

Some sophistication of the alarm system could be included in the digital simulator in order to warn the operator that an undesired operating condition had developed. This could be included in the computer program with the addition of logic statements.

If it would be desired to actually train or re-train power plant operators to operate units, such as the simulated unit, under abnormal as well as normal conditions, an analog-digital interface could be created that would link the digital computer simulator with an operator's control panel.

The digital simulation of the once-through supercritical steam generator can be used in its present form to study the operation of the unit with, or without, the associated control system included.
An engineer or operator could study the operation of the unit with one or more of the control subsystems included in the system. Modifications of the control system could be made on the basis of the results of studies made using the simulator. Also, alternative unit operation modes could be developed by investigating the behavior of the unit under simulated conditions.
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APPENDIX A

LAGRANGE INTERPOLATION

A method for the determination of the feedwater flow as a function of boiler feed pump speed and output pressure above throttle pressure by the Lagrangian multiple interpolation technique is herein described. In this discussion equally spaced intervals between the independent variables is assumed. The basic idea of the interpolation technique is to replace the functional relationship between the feedwater flow and the boiler feed pump speed and pressure by a polynomial. In this case a three point interpolation is used which gives rise to a polynomial of 2nd degree.

Consider first interpolation only with respect to pump speed. Let X represent the speed. The computer variable that corresponds to this speed is RPML.

Let WL represent the feedwater flow. This corresponds in the computer program variable FWREL. For the three points, consider WL₁, WL₂, WL₃, corresponding to the three values of speed, X₁, X₂, X₃. A polynomial of the nth degree (there are n + 1 points chosen such that for 3 points n = 2) that passes through the points is given by the equation below.

\[
F(X) = \frac{(X-X_2)(X-X_3)}{(X_1-X_2)(X_1-X_3)} WL_1 + \frac{(X-X_1)(X-X_3)}{(X_2-X_1)(X_2-X_3)} WL_2 + \frac{(X-X_1)(X-X_2)}{(X_3-X_1)(X_3-X_2)} WL_3
\]

(A.1)
For \( n = 2 \), let

\[
P(X) = (X-X_1)(X-X_2)(X-X_3) = \prod_{j=1}^{n+1} (X-X_j) \tag{A.2}
\]

\[
P_i(X) = (X-X_i)^{-1} P(X) = (X-X_i)^{-1} \prod_{j=1}^{n+1} (X-X_j) \tag{A.3}
\]

Equation (A.1) becomes

\[
F(X) = \frac{P_1(X)}{P_1(X)} WL_1 + \frac{P_2(X)}{P_2(X)} WL_2 + \frac{P_3(X)}{P_3(X)} WL_3 \tag{A.4}
\]

Now if the point desired is on one of the cardinal data points the other two terms in equation (A.4) vanish, that is \( P_m(X) = 0 \) if \( m \neq n \) so that

\[
F(X_m) = \frac{P_m(X_m)}{P_m(X_m)} WL_m = WL_m \quad (m = 1, 2, 3, \ldots)
\]

and thus the polynomial of degree \( n \) (2 in this case) passes through the three given points.

Since there are three (odd number) points and they are arranged in an order \( X_{-r}, \ldots, X_{-1}, X_0, X_1, \ldots, X_r \) where

\[ n + 1 = \text{number of points} \]
\[ 2r + 1 = \text{number of points} \]
\[ n + 1 = 2r + 1 = 3 \]
\[ \therefore r = 1 \]

For interpolation using three points the following equation (see p. 147, ref. 71) represents the Lagrange interpolation coefficients for the case of \( r = 1 \).
\[ A_i(\Delta X) = (-1)^{i+1} \frac{1}{(1+i)!} \left( \prod_{j=1, j \neq i}^{1} (\Delta X - j) \right) \]  

(A.5)

Where \( \Delta X \) = difference term

For \( i = -1, 0, 1 \)

Consider the equally spaced interval such that the coordinate

\[ X_i = X_0 + i\Delta X \]

\[ X_{i-1} = X_0 + (-1)\Delta X \]

\[ X_0 = X_0 + (0)\Delta X \]

\[ X_1 = X_0 + (1)\Delta X \]

Since 0 subscripts are not allowed in the computer let \( K = I - 2 \), so when \( K = -1, I = 1; K = 0, I = 2; K = 1, I = 3. \)

Now equation (A.4) is invariant under a linear transformation. That is, \( X \) may be replaced by

\[ X = (\Delta X) \cdot (\Delta X) + X_0 \]

and \( X_i = \Delta X \cdot X_i + X_0 \)

where \( X_0 \) and \( \Delta X \) are constant.

From equation (A.3)

\[ P_i(X) = P_i(\Delta X + X_0) = (\Delta X + X_0)^{-1} \left( \Delta X + X_0 \right)^{-1} \left( \Delta X + X_0 \right)^{-1} \]

\[ \left( \Delta X + X_0 \right) \ldots \left( \Delta X + X_0 \right) \left( \Delta X + X_0 \right) \]

\[ P_i(X) = \left( \Delta X \cdot P_i \right)^{-1} \left( \Delta X \cdot P_i \right)^{-1} \left( \Delta X \cdot P_i \right)^{-1} \left( \Delta X \cdot P_i \right)^{-1} \left( \Delta X \cdot P_i \right)^{-1} \left( \Delta X \cdot P_i \right)^{-1} \left( \Delta X \cdot P_i \right)^{-1} \left( \Delta X \cdot P_i \right)^{-1} \left( \Delta X \cdot P_i \right)^{-1} \left( \Delta X \cdot P_i \right)^{-1} \]

\[ P_i(X) = \left( \Delta X \right)^{i-1} \left( \Delta X \right)^{i-1} \left( \Delta X \right)^{i-1} \left( \Delta X \right)^{i-1} \]
\( P_i(X) = \Delta X^{-1} \Delta X^3 (PX-PX_1)^{-1} (PX-PX_2)(PX-PX_3) \)

\[ P_i(X) = \Delta X^2 P_i(PX) \quad (A. 6) \]

Also from equation (A. 3) \( P_i(X_i) = \frac{(X_i-X_1)(X_i-X_2)(X_i-X_3)}{(X_i-X_1)} \) (i = 1, 2, 3)

That is \( P_i(X_i) = (X_i-X_1)(X_i-X_2)\ldots(X_i-X_{i-1})(X_i-X_{i+1})\ldots(X_i-X_{n+1}) \)

\[ (A. 7) \]

so \( P_i(X_i) = (PX_i \Delta X+X_0-(PX_i \Delta X+X_0))(PX_i \Delta X+X_0-(PX_{i+1} \Delta X+X_0))\ldots \)

\[ (A. 8) \]

\[ (A. 9) \]

From equation (A. 7) then

\[ P_i(X_i) = (\Delta X)^n P_i(PX_i) \quad (A. 8) \]

Now, since \( n = 2 \), equation (A. 8) becomes

\[ P_i(X_i) = (\Delta X)^2 P_i(PX_i) \quad (A. 9) \]

and equation (A. 4) becomes
\[ F(X) = F(PX \cdot \Delta X + X_0) = \frac{P_1(X_1)}{P_1(PX)} WL_1 + \frac{P_2(X_2)}{P_2(PX)} WL_2 + \frac{P_3(X_3)}{P_3(PX)} WL_3 \]

Using equations (A.6) and (A.9)

\[ F(X) = F(PX \cdot \Delta X + X_0) = \frac{(\Delta X)^2}{(\Delta X)^2} \left[ \frac{P_1(PX)}{P_1(PX_1)} WL_1 + \frac{P_2(PX)}{P_2(PX_2)} WL_2 + \frac{P_3(PX)}{P_3(PX_3)} WL_3 \right] \]

(A.10)

Hence, since the right hand side of equation (A.10) is \( F(PX) \),

\[ F(X) = F(PX) \]

Then since \( X = \Delta X \cdot PX + X_0 \)

\[ PX = \frac{X - X_0}{\Delta X} \]

For our case, since a multiple interpolation is involved let

\[ PX = \frac{RPML - XI}{\Delta X} \]

Where \( XI \) = center interpolation point

\( \Delta X \) = spacing interval

\( RPML \) = desired speed

and also let the difference term for the pressure be

\[ PR = \frac{PBFPL - RJ}{\Delta R} \]

where \( PBFPL \) = desired pressure

\( RJ \) = center interpolation point

\( \Delta R \) = pressure interval between cardinal points
Rewriting equation (A.5) the following equation is obtained

\[ A_1(PX) = \frac{(-1)^{1-i}}{(l+i)! (1-i)!} \prod_{j=-1}^{1-i} (PX-j) \]  
\[ \text{for } j \neq i \]  
\[ \text{(A.11)} \]

For \( i = -1, 0, 1 \), the equation above becomes,

\[ A_{-1}(PX) = \frac{(-1)^2}{0! 2!} P(PX-1) \]

\[ A_0(PX) = \frac{(-1)^1}{1! 1!} (PX+1)(PX-1) \]

\[ A_1(PX) = \frac{(-1)^0}{2! 0!} (PX+1)PX \]

Now note that \((-1)^{1-i} = (-1)^{1-i+2} = (-1)^{i+1}\)

and \( PX(PX^2 - 1) = PX(PX+1)(PX-1) \)

so that when \( K = -1 \)

\[ \frac{PX(PX^2 - 1)}{PX-K} = \frac{PX(PX+1)(PX-1)}{(PX+1)} = PX(PX-1) \]

when \( K = 0 \)

\[ \frac{PX(PX^2 - 1)}{PX-K} = \frac{PX(PX+1)(PX-1)}{PX} = (PX+1)(PX-1) \]

and when \( K = +1 \)

\[ \frac{PX(PX^2 - 1)}{PX-K} = \frac{PX(PX+1)(PX-1)}{(PX-1)} = PX(PX+1) \]
so the interpolation coefficient in equation (A. 5) can be written as follows

\[ A_{K_3}(P_X) = \frac{(-1)^{K+1} P_X(P_X^2 - 1)}{(1+K)! (1-K)! (P_X-K)} \quad (K=0, \pm 1) \]  

Then using the interpolation coefficients the flow equation becomes

\[ W_L(X) = A_{-1}(P_X)W_{L_{-1}} + A_0(P_X)W_0 + A_{+1}(P_X)W_{L_1} \]

When using \( K \) as the subscript variable or in terms of the subscript \( I \) the equation would become

\[ W_L(X) = A_1(P_X)W_{L_1} + A_2(X)W_L + A_3(P_X)W_{L_3} \]  

(A.13)

Now when the pressure interpolation is also considered, it becomes necessary to add another subscript on the flow variable, that is, consider \( W_{L_{m,n}} \). As will be seen in the computer program the interpolation with respect to speed (RPML) is done for three different values of pressure (PBFPL), specifically, two cardinal points below the desired point and one cardinal point above the desired point.

Let the interpolation coefficients be labeled \( A_X(I) \) and \( A_R(I) \), \( I = 1, 2, 3 \), for the speed and pressure respectively and for a particular value of \( P_X \) and \( P_R \).

Table A-1 below illustrates the parameters used for interpolation.

**Feedwater flow** = \( f(\text{speed}, \text{pressure}) \)

\[ W_L = f(X, R) \]

Consider an example point where the desired point is RPML = 4400 rpm, PBFPL = 450 psia above the throttle pressure bias of 3500 psia. From Figure A-1 and Table A-1 it is seen that the desired point is just above the cardinal coordinate point \( W_L(3, 4) \). The computer program
### TABLE A-1

**PARAMETER TABLE FOR INTERPOLATION**

<table>
<thead>
<tr>
<th>Speed X RPM</th>
<th>Pressure R PBFPL</th>
<th>R₁</th>
<th>R₂</th>
<th>R₃</th>
<th>R₄</th>
<th>R₅</th>
<th>...</th>
<th>R₉</th>
</tr>
</thead>
<tbody>
<tr>
<td>X₁</td>
<td>WL(1,1)</td>
<td>WL(1,2)</td>
<td>WL(1,3)</td>
<td>WL(1,4)</td>
<td>WL(1,5)</td>
<td>...</td>
<td>WL(1,9)</td>
<td></td>
</tr>
<tr>
<td>X₂</td>
<td>WL(2,1)</td>
<td>WL(2,2)</td>
<td>WL(2,3)</td>
<td>WL(2,4)</td>
<td>WL(2,5)</td>
<td>...</td>
<td></td>
<td></td>
</tr>
<tr>
<td>X₃</td>
<td>WL(3,1)</td>
<td>WL(3,2)</td>
<td>WL(3,3)</td>
<td>WL(3,4)</td>
<td>WL(3,5)</td>
<td>...</td>
<td></td>
<td></td>
</tr>
<tr>
<td>X₄</td>
<td>WL(4,1)</td>
<td>WL(4,2)</td>
<td>WL(4,3)</td>
<td>WL(4,4)</td>
<td>WL(4,5)</td>
<td>...</td>
<td></td>
<td></td>
</tr>
<tr>
<td>X₅</td>
<td>WL(5,1)</td>
<td>WL(5,2)</td>
<td>WL(5,3)</td>
<td>WL(5,4)</td>
<td>WL(5,5)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>X₆</td>
<td>WL(6,1)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>X₇</td>
<td>WL(7,1)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>X₈</td>
<td>WL(8,1)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>X₉</td>
<td>WL(9,1)</td>
<td>...</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>WL(9,9)</td>
<td></td>
</tr>
</tbody>
</table>

**Fig. A-1** Data for boiler feed pump Lagrangian interpolation
computes the coordinates of the cardinal point, XII = 3 and RJ = 4.
This corresponds to a speed of 4300 rpm and a pressure of 430 psia
above the throttle pressure which is verified on the computer output,
Figure A-2. This output shows XI = 4300 and RJ = 430 where XI and
RJ have been defined earlier. In this example the values of PX and PR
are computed as follows,

\[
P_X = \frac{RPML-XI}{\Delta X} = \frac{4400-4300}{200} = \frac{100}{200} = 0.5
\]

\[
PR = \frac{PBFP-L-J}{\Delta R} = \frac{450-430}{120} = \frac{20}{120} = .1667
\]

Next compute the factorials needed for the determination of the
Lagrangian interpolation coefficients.

Note K = 0, ±1

In the computer program K = I-2 such that I = 1 corresponds to
K = -1, I = 2 corresponds to K = 0, and I = 3 corresponds to K = +1.

Check to see if the PX or the PR are zero which is the case if
the desired value of speed and pressure respectively minus the
"base point" values are on the desired point. If it is on the base
point then set the value of the interpolation coefficients equal to the
following, AX(1) = 0, AX(2) = 1, AX(3) = 0.

For example if the point desired is on the X coordinate given as
a base point, say X = 3, and between R = 4 and R = 5. (See Figure
A-2 for this example.)

\[
WL(X, 2) = AX(1) \ WL(2, 3) + AX(2) \ WL(3, 3) + AX(3) \ WL(4, 3) \quad (A.14)
\]

\[
WL(X, 3) = AX(1) \ WL(2, 4) + AX(2) \ WL(3, 4) + AX(3) \ WL(4, 4) \quad (A.15)
\]

\[
WL(X, 4) = AX(1) \ WL(2, 5) + AX(2) \ WL(3, 5) + AX(3) \ WL(4, 5) \quad (A.16)
\]
LAGRANGIAN INTERPOLATION

BOILER FEED PUMP

PRINT THE NEW VALUES OF SPEED + PRESSURE
4400.0, 450.0

<table>
<thead>
<tr>
<th>XII</th>
<th>RJ</th>
<th>XI</th>
<th>RJ</th>
<th>PX</th>
<th>PR</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.0</td>
<td>4.0</td>
<td>4300.0</td>
<td>430.0</td>
<td>5000</td>
<td>1667</td>
</tr>
</tbody>
</table>

AX(1) AX(2) AX(3) AR(1) AR(2) AR(3)
-12500 .75000 .37500 -.06944 .97222 .09722

SPEED = 4400.00 PRESSURE = 450.00
BOILER FEED PUMP FLOW = 1609.7917

PRINT THE NEW VALUES OF SPEED + PRESSURE
4300.0, 500.0

<table>
<thead>
<tr>
<th>XII</th>
<th>RJ</th>
<th>XI</th>
<th>RJ</th>
<th>PX</th>
<th>PR</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.0</td>
<td>4.0</td>
<td>4300.0</td>
<td>430.0</td>
<td>0000</td>
<td>5833</td>
</tr>
</tbody>
</table>

AX(1) AX(2) AX(3) AR(1) AR(2) AR(3)
00000 1.00000 .0000 .12153 .65972 .46181

SPEED = 4300.00 PRESSURE = 500.00
BOILER FEED PUMP FLOW = 1069.0278

PRINT THE NEW VALUES OF SPEED + PRESSURE
5300.0, 790.0

<table>
<thead>
<tr>
<th>XII</th>
<th>RJ</th>
<th>XI</th>
<th>RJ</th>
<th>PX</th>
<th>PR</th>
</tr>
</thead>
<tbody>
<tr>
<td>8.0</td>
<td>7.0</td>
<td>5300.0</td>
<td>790.0</td>
<td>0000</td>
<td>0000</td>
</tr>
</tbody>
</table>

AX(1) AX(2) AX(3) AR(1) AR(2) AR(3)
00000 1.00000 .0000 .0000 1.00000 .0000

SPEED = 5300.00 PRESSURE = 790.00
BOILER FEED PUMP FLOW = 3980.00

PRINT THE NEW VALUES OF SPEED + PRESSURE
5300.0, 950.0

PRESSURE IS ABOVE MAX. ACCEPTABLE VALUE
TRIP UNIT... TRIP UNIT...

Fig. A-2 Computer output of Lagrangian interpolation
where AX(1) = 0
AX(2) = 1
AX(3) = 0

If the point were not on a step point with respect to speed (i.e., RPML ≠ 4300) then the Lagrangian coefficients would be calculated in the usual manner as seen by equation (A.12). (See Figure A-2 for the results of this case also.)

Again, if the desired point is on the X cardinal point,

\[ WL(X, 2) = WL(3, 3) \]
\[ WL(X, 3) = WL(3, 4) \]
\[ WL(X, 4) = WL(3, 5) \]

Then interpolating with respect to the R variable, PBFPL, use the three quantities just computed in equations (A.14), (A.15), and (A.16). Compute the interpolation coefficients, AR(I), I = 1, 2, 3 using the following equations.

From equation (A.12), and since \( K = I-2 \) and \( K+1 = I-1 \), then
\[ PX-K = PX-(I-2) = PX-I+2. \]
Now let PROD = \( (1+K)!/(1-K)! \)

Then
\[ AX(I) = \frac{(-1)^{I-1}PX(PX^2-1)}{(PROD)(PX-I+2)} \]
(A.17)

Then next, interpolate with respect to R, (i.e., pressure)

\[ AR(I) = \frac{(-1)^{I-1}PR(PR^2-1)}{(PROD)(PR-I+2)} \]
(A.18)

Thus the interpolated feedwater flow becomes,
\[ WL(X, R) = AR(1)WL(X, 2) + AR(2)WL(X, 3) + AR(3)WL(X, 4) \]
(A.19)
In the computer program since an $X$ is not a suitable subscript for one quantity of storage, the location $WL(10, L)$ is used for the $X$ interpolated values. In this case $WL(10, 2)$, $WL(10, 3)$ and $WL(10, 4)$ are used. Then $WL(10, 10)$ is used for the final interpolated value.

Now referring to the computer program NO, NOO, and NPP are the subscripts to find the correct values of flow by which to multiply the interpolation constants.

$RJJ$ is the subscript of the pressure point below the desired point, in the example it is 4.

$RJJ-1$ is desired as the first subscript and then $RJJ$, and $RJJ+1$, the two points below the desired point and one point above the desired point are needed next.

$NO = RJJ-1$ is the lowest subscript needed for interpolation to get the desired point for pressure.

$NOO = NO+2$ is the highest subscript needed for interpolation to get the desired point for pressure.

$NP = XII-1$ is the lowest subscript needed for interpolation to get the desired point for speed.

In this example $NO = 4-1 = 3$

$NOO = 5$

$XII = 3$

$NP = 2$

Hence the right 9 points are used as seen in equations (A.14), (A.15), and (A.16) and Table A-1.

The interpolated values with respect to speed of flow are $WL(10, RJJ-1)$, $WL(10, RJJ)$ and $WL(10, RJJ+1)$. The 10 is a dummy subscript as mentioned earlier.

Then the interpolation with respect to pressure $(R)$ gives the
desired point properly interpolated. It is

\[ \text{WL}(10,10) = \text{AR}(1) \times \text{WL}(10, \text{NO}) + \text{AR}(2) \times \text{WL}(10, \text{NO}+1) + \text{AR}(3) \times \text{WL}(10, \text{NO}+2) \]

By setting \( \text{FWREL} = \text{WL}(10,10) \), the boiler feedpump discharge flow in units of thousands of pounds per hour, is obtained and could then be used in the process equations as was mentioned in the section where the boiler feedpump was discussed.

A sample output of the program is given previously, corresponding to the example mentioned earlier. It is shown in Figure A-2. The flow chart and program are given in Figure A-3 and Table A-2, respectively.
Fig. A-3 Flow chart for Lagrangian interpolation
Fig. A-3 Continued
TABLE A-2

COMPUTER PROGRAM FOR LAGRANGIAN INTERPOLATION

100C LAGRANGIAN INTERPOLATION
105 PRINT "LAGRANGIAN INTERPOLATION"
110 PRINT "" BOILER FEED PUMP"
115 PRINT "" DIMENSION AX(10), AR", "RPM", "PBFP", "WL(10,10)
120 PRINT "" READ(RPM(K),K=1,9)
125 READ(PBFP(K),K=1,9)
130 READ(WL(J,K),J=1,9),K=1,9)
135 READ,DRPML,DPBFPPL
140 PRINT ""
145 PRINT "PRINT THE NEW VALUES OF SPEED + PRESSURE"
150 INPUT,RPML,PBFPFL
155 PRINT,
160 IF(RPML<4100.) THEN,12,11,11
165 12 CONTINUE
170 PRINT "SPEED IS BELOW MIN. ACCEPTABLE VALUE"
180 PRINT "TRIP UNIT... TRIP UNIT..."
185 STOP
190 11 CONTINUE
195 IF(PBFPFL<190.) THEN,14,13,13
200 14 CONTINUE
210 PRINT "PRESSURE IS BELOW MIN. ACCEPTABLE VALUE"
220 PRINT "TRIP UNIT... TRIP UNIT..."
225 STOP
230 13 CONTINUE
235 IF(RPML>5300.) THEN,15,15,16
240 15 CONTINUE
250 PRINT "SPEED IS ABOVE MAX. ACCEPTABLE VALUE"
260 PRINT "TRIP UNIT... TRIP UNIT..."
265 STOP
270 14 CONTINUE
275 IF(PBFPFL>910.) THEN,17,17,18
280 17 CONTINUE
290 PRINT "PRESSURE IS ABOVE MAX. ACCEPTABLE VALUE"
300 PRINT "TRIP UNIT... TRIP UNIT..."
305 STOP
310 17 CONTINUE
315 XI=AIN(T(RPML-3800.)/DRPML)
320 RJJ=AIN(T(PBFPFL+50.)/DPBFPPL)
325 XI=AIN(T(RPML-33000.)/DRPML)*200+3900
330 RJJ=AIN(T(PBFPFL-700.)/DPBFPPL)*120+700
340 PX=(RPML-XI)/DRPML
345 PR=(PBFPFL-RJ)/DPBFPPL
350 PRINT "" XI RJJ XI RJ
355 PX PR
360 PRINT 95,XI1,RJJ,XI,RJ,PX,PR
365 95 FORMAT(1X,4F10.1,2F10.4)
370 DO 50 I=1,3
380 PROD=1.0
385 50 NN=I-1
390 PROD=1.0
395 PRINT 580 PROD
TABLE A-2 CONTINUED

590 IF(NN) 41, 41, 40
600 41 NN=1
610 40, MM=3-1
620 IF(MM) 31, 31, 30
630 31 MM=1
640 30 NFAC=1
650 DO 51 IN=1, NN
660 NFAC=NFAC*IN
670 51 CONTINUE
680 MFA=1
690 DO 52 IJ=1, MM
700 MFAC=MFAC*IJ
710 52 CONTINUE
720 PROD=NFAC*MFAC
730 IF(PR) 112, 112, 111
740 112 AX(1)=0.0; AX(2)=1.0; AX(3)=0.0
750 GO TO 450
760 111 CONTINUE
770 AX(I)=((-1)**(I-1)*PX*(PX/PX-1)/(PROD*(PX-I+2)))
780 450 CONTINUE
790 IF(PR) 212, 212, 211
800 212 AR(1)=0.0; AR(2)=1.0; AR(3)=0.0
810 GO TO 50
820 211 CONTINUE
830 AR(I)=((-1)**(I-1)*PR*(PR-PR-1)/(PROD*(PR-I+2)))
840 50 CONTINUE
850 PRINT,
860 PRINT,"AX(1) AX(2) AX(3) AR(1) AR(2)
861 +5X, 5HAR(2), 5X, 5HAR(3))
880 PRINT 96, (AX(I), I=1, 3), (AR(I)), I=1, 3)
890 96 FORMAT(6F10.5),
900 NO=RJ-1
910 NOO=NO+2
920 NP=XII-1
930 DO 70 L=NO, NOO
940 WL(10, L)=AX(1)*WL(NP, L)+AX(2)*WL(NP+1, L)+AX(3)*WL(NP+2, L)
950 70 CONTINUE
960 WL(10, 10)=AR(1)*WL(10, NO)+AR(2)*WL(10, NO+1)+AR(3)*
970 +WL(10, NO+2)
980 FWREL=WL(10, 10)
990 PRINT,
1000 PRINT,"SPEED =", RPM, "PRESSURE =", PBFPFL
1010 PRINT, "BOILER FEED PUMP FLOW =", FWREL
1020 PRINT,
1030 GO TO 1
1040 END
1050 STOP
1070 70., 90., 910., 1030.
| 1080 | 1450, 2270, 2590, 3510, 4200, 4700, 5030, 5360, 6090, 1090, 400, 1170, 1240, 2610, 3280, 3990, 4490, 4850, 5210, 1100, 170, 890, 1610, 2330, 3050, 3780, 4280, 4670, 5060, 1110, -260, 510, 1280, 2050, 2320, 3570, 4030, 4490, 4900, 1120, -710, 100, 910, 1720, 2530, 3350, 3850, 4330, 4750, 1130, -1140, -300, 540, 1380, 2220, 3130, 3670, 4150, 4630, 1140, -1620, -740, 140, 1020, 1900, 2890, 3450, 3980, 4490, 1150, -2400, -1380, -360, 660, 1680, 2650, 3230, 3790, 4350, 1160, -2000, -1600, -600, 0, 900, 2050, 2750, 3330, 3840, 1170, 200, 120 |
APPENDIX B

TABLE B-1

COMPUTER PROGRAM FOR DIGITAL SIMULATOR

C BLOCK 1
C SIMULATION OF A SUPERCRITICAL STEAM GENERATOR
C WITH THE CONTROL SYSTEM
C 23 LUMPS, RUNGE-KUTTA INTEGRATION
C

WRITE(6,900)
900 FORMAT(32X,35HSIMULATION OF A SUPERCRITICAL STEAM)
WRITE(6,901)
901 FORMAT(33X,33HGENERATOR WITH THE CONTROL SYSTEM)
WRITE(6,902)
902 FORMAT(33X,33H23 LUMPS, RUNGE-KUTTA INTEGRATION/)

C BLOCK 2
C
COMMON H,P,T,RHOSV
DIMENSION WG(3),W(2C),POA(15),TOA(15)
DIMENSION V(15),RHON(15),RHO(15),DRHO(15)
DIMENSION PI(1,21),PO(1,21),QW(1,21),HI(3,21),HO(3,21)
,TO(3,21),TW(3,21),Q(3,21),DHOR(3),DTWR(3),XP(14),X0(1
14),CPW(14),TI(1,21)
REAL KP(14),KO(14),M(14),MWCPW(14),MW(14),IATM,INCRHO
C BLOCK 3
C INITIALIZE THE PROCESS VARIABLES
C
RUN=1.
PR=0.0
PRD=PR
QLT=0.0
PO(1,16)=3500.
CQ=.34
TG=2190.
HC=1071.
WNG=5.E6
DO 31 I=1,J
MWCPW(I)=MW(I)*CPW(I)
KP(I)=XP(I)/((1105.*1105.*)
31 KO(I)=QW(I,1)/((1105.***.8)*XQ(I))
TIME=.1
WPSS=1105.*3480./3980.
POWER=560.

350
TABLE B-1 CONTINUED

$$\text{KP}(12) = \frac{\text{XP}(12)}{\text{WRSS}}$$
$$\text{KP}(12) = \frac{\text{XP}(12)}{(\text{WRSS}^2 + R) \times \text{XQ}(12)}$$
$$\text{MKTPW}(12) = \text{MW}(12) \times \text{CPW}(12)$$
$$\text{TO}(1,9) = 950$$
$$\text{TO}(1,10) = 1015.391$$
$$\text{TO}(1,12) = 1015.391$$

C RHON(N) IS A DUMMY VARIABLE CORRESPONDING TO STEADY STATE DENSITY

RHON(9) = 10.11451
RHON(10) = 6.0964563
RHON(11) = 4.932273
RHON(12) = 4.9568375
RHON(11) = 5.0
V(9) = 823.
V(10) = 596.
V(11) = 309.
V(12) = 2669.9

C CHANGE THESE VALUES OF PO TO STEADY STATE

PO(1,9) = 3790.
PO(1,10) = 3710.
PO(1,11) = 3600.
PO(1,12) = 644.
HS = 300.

C-----------------------------
C BLOCK 4
C INITIALIZE THE CONTROL SYSTEM VARIABLES
C
CS = 1.
IBTM = CS
FWC = CS
SVC = CS
FFC = CS
AFC = 0.
RHSTC = CS

C GIVE NEW VALUE OF MW DEMAND
C DEMW1 GOES HERE

DEMW1 = 560.
DEMW2 = 560.
R1A = 0.0
E1I = 0.
ETB = 0.0
VIC = 1000.
PSPEA = 0.0
TVPX = 1000.
FV15A = 0.0
DERER2 = 0.
FWRE = 3980.
TABLE B-1 CONTINUED

FWA1 = 3980
FSPA = 0.0
FWA2 = 5300
FWVFA = 0.0
TOE19 = 95.0
E3GA = 0.0
SVP = 0.0
FWA = 0.0
FWTH = 5000
FLFL1 = 5000
F2PA = 0.0
F2PA = 0.0

C 02 IS NOT USED
P2 = 5200
C OXYGEN SET POINT IS TO BE CALCULATED
P2SP = 5200
P2ISA = 0.0
P2ISA = 0.0
AEM = 5200
AF1 = 5200
E5OA = 0.0
E6OA = 0.0
SCI = 0.0
F25A = 0.0

12 CONTINUE
IF(RUN - 5.0) 13, 14, 14
14 STOP
13 CONTINUE
TIME = 0.0
T1 = 0.0
T2 = 0.0
TPRINT = 0.0

C-----------------------------------------
C BLOCK 5
C
C IF(ALT) THEN 1004, 590, 1004
C 1004 IS IN BLOCK 15
580 CONTINUE
IF(DEMW1 - 225) 405, 401, 401
401 IF(DEMW1 - 680) 402, 402, 480
402 CONTINUE
C-----------------------------------------
C BLOCK 6
C PRINT HEADING
TABLE B-2 CONTINUED

C WRITE(6,439)
635 FORMAT(25X,4H1MF,4X,5HTVXR,6X,4HFWRF,5X,7HTN(1,7),12X,4HTN(1,11),3X,4HTN(1,12))
   WRITE(6,1010)
1010 FORMAT(44X,5HPWR,F4X,7HPN(1,7),2X,8HPN(1,11))
C ------------------------------------------------
C BLOCK 7
C UNIT LOAD DEMAND DEVELOPMENT
C
C 650 F=DEMW1-DEMW2
B=3.*E
   IF(ABS(B)-35.)610,610,615
615 IF(B-35.)620,620,625
620 R=-35.*
   GO TO 610
675 R=35.*
610 B1=R*1.
   REMW1=DFMW2+((B1+B1A)/2.)*T1*(1./60.)
   B1A=91
C ------------------------------------------------
C BLOCK 8
C INTEGRATED BOILER TURBINE MASTER
C
C F1=DEMW2-POWER
F11=F1+(E1+E1)/2.)*T1*(1./300.)
ET1=F1
   IF(F11)665,670,665
665 IF(ABS(E11)-350.)670,670,674
674 IF(E11-350.)676,676,680
676 E11=-350.
   GOTO 670
680 E11=350.
670 CONTINUE
   TR1M1=(*2/350.)*F11+1.
   DEMW2A=DEMW2*TR1M1
   AL=E1*1.0
   IF(ABS(AL)-21.)600,600,605
605 IF(AL-21.)606,606,607
606 AL=-21.
   GOTO 600
607 AL=21.
600 CONTINUE
   IF(POWER-175.)691,692,692
692 IF(POWER-700.)693,693,694
693 TRM2=(*1.0/525.)*POWER+.5
   GOTO 695
TABLE B-1 CONTINUED

691 TRIM2=1.0
GOTO 695
694 TRIM2=2.5
695 CONTINUE
AL IU=AL.*TRIM2
C TEMPORARY AL IU W/0 TRIMMING
AL IU=AL.*1.0
AI=AL IU*(50./7.)
PF=PO(1,16)-3500.
PFR=PF*(7./50.)
DFMW4=DFMW2A+PFP
PSPC=3500.*+AI.1
PSPF=PSPC-P7(1,16)
PC=PSPF*5.
VIC=VIC+((PSPF+PSPFA)/2.)*T1*(1./12.)
PSPEA=PSPE
VPS=PC+VIC
FV15=VPS-TV PX
TV PX=TV PX+((FV15+EV15A)/2.)*T1*(1./12.)
EV15A=FV15
IF(INTM.0)811,810,811
811 TV PX=TV PX/1000.
GO TO 812
810 TV PX=1.
812 CONTINUE
C----------------------------------
C BLOCK 9
C FLFD WATER FLOW CONTROL
C
FWCD=(456./55.1*P(DFMW4-DFRER.005)-660.
IF(FWCD-1200.0)405,400
405 WRIFE(6,700)
700 FORMAT(1HO,6X,3(I11H WARNING...10X))
WRIFE(6,701)
701 FORMAT(4OH0FLOW DEMAND IS BELOW MINIMUM ACCEPTABLE)
WRITE(6,711)
711 FORMAT(6H VALUE)
WRITE(6,702)
702 FORMAT(43H0TRIP UNIT... TRIP UNIT... TRIP UNIT...)
STOP
406 CONTINUE
A=100.
B=245.0
C=60.0
SP1=2590.0
SP2=3860.0
<table>
<thead>
<tr>
<th>Table B-1 Continued</th>
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<tbody>
<tr>
<td>RPM1 = 4600</td>
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<tr>
<td>FWF1 = 2590</td>
</tr>
<tr>
<td>PRFP1 = 3860</td>
</tr>
<tr>
<td>GO TO 560</td>
</tr>
<tr>
<td>410 IF (FWPE - 3080)</td>
</tr>
<tr>
<td>412 CONTINUE</td>
</tr>
<tr>
<td>A = 100.0</td>
</tr>
<tr>
<td>B = 255.0</td>
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<td>C = 75.0</td>
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<td>SP1 = 2835.0</td>
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</tr>
<tr>
<td>FWF1 = 2835.0</td>
</tr>
<tr>
<td>PRFP1 = 3920.0</td>
</tr>
<tr>
<td>GO TO 560</td>
</tr>
<tr>
<td>420 IF (FWRF - 3330)</td>
</tr>
<tr>
<td>422 CONTINUE</td>
</tr>
<tr>
<td>A = 100.0</td>
</tr>
<tr>
<td>B = 245.0</td>
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<td>C = 70.0</td>
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<tr>
<td>SP1 = 3085.0</td>
</tr>
<tr>
<td>SP2 = 3995.0</td>
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<tr>
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<tr>
<td>FWRF1 = 3085</td>
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<tr>
<td>PRFP1 = 3995.0</td>
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<tr>
<td>GO TO 560</td>
</tr>
<tr>
<td>430 IF (FWRF - 3540)</td>
</tr>
<tr>
<td>432 CONTINUE</td>
</tr>
<tr>
<td>A = 100.0</td>
</tr>
<tr>
<td>B = 210.0</td>
</tr>
<tr>
<td>C = 70.0</td>
</tr>
<tr>
<td>SP1 = 3330.0</td>
</tr>
<tr>
<td>SP2 = 4065.0</td>
</tr>
<tr>
<td>RPM1 = 4900</td>
</tr>
<tr>
<td>FWRF1 = 3330</td>
</tr>
<tr>
<td>PRFP1 = 4065.0</td>
</tr>
<tr>
<td>GO TO 560</td>
</tr>
<tr>
<td>440 IF (FWRF - 3780)</td>
</tr>
<tr>
<td>442 CONTINUE</td>
</tr>
<tr>
<td>A = 200.0</td>
</tr>
<tr>
<td>B = 240.0</td>
</tr>
<tr>
<td>C = 80.0</td>
</tr>
<tr>
<td>SP1 = 3540.0</td>
</tr>
<tr>
<td>SP2 = 4135.0</td>
</tr>
<tr>
<td>RPM1 = 5000</td>
</tr>
<tr>
<td>FWRF1 = 3540</td>
</tr>
<tr>
<td>PRFP1 = 4135.0</td>
</tr>
</tbody>
</table>
TABLE B-1 CONTINUED

GO TO 560
450 IF( FWR < 5000.) 452, 452, 490
452 CONTINUE

A = 100.
B = 200.
C = 65.
SP1 = 3790.
RPM1 = 5200.
FWRE1 = 3780.
PBFP1 = 4215.
SP2 = 4215.
IF( FWR > SP1) 442, 560, 560

CONTINUE

E5 = FWCD - FWDF1
E5A = E5 / 15.
FWDF1 = FWDF1 + ((E5A + E5AA) / 2.) * T1
E5AA = E5A
E6 = FWDF1 - FWRF
FWDF2 = FWDF2 + ((E6 + E6A) / 2.) * T1 * (1. / 15.)
E6A = E6
FWVF = FWDF2 - RPM
RP'M = RPM + ((FWVF + FWVFA) / 2.) * T1
FWVFA = FWVF
X = FWRF - SP1
IF( X < 814) 488, 488, 489

489 X = B
488 CONTINUE

IF( FWC ) 813, 814, 813
813 CONTINUE

PBFP = 3500.* (SP2 - 3500.) + (C / R) * X
GO TO 815
814 PBFP = 4280.
815 CONTINUE

IF( FWC ) 805, 804, 805
805 FWRE = FWRE1 - (B / 2.) * (RPM1 / A + PBFP1 / C) + (R / 2.)*(RPM/A + PBFP / 10)
GO TO 806
804 FWRE = 3980.
806 GO TO 733

480 WRITE(6,700)
480 WRITE(6,703)
703 FORMAT(4CH0FLOW DEMAND IS ABOVE MAXIMUM ACCEPTABLE)
703 WRITE(6,712)
712 FORMAT(6H VALUE)
STOP
733 CONTINUE
TABLE B-1 CONTINUED

C BLOCK 10
C FIRING RATE CALIBRATION
C
FREDM2=DEMW4
F13=1C15.301-T0(1,10)
E13G=E13*1.
DERT=(T0(1,9)-T0F19)/DTIMF
DERE1=E13G-DERT
TRIM3=(1.5/70O.)*DEMW4+1.
DEREPT=DERE1*TRIM3
DEREP2=DEREPT+F1
FREDM3=FREDM2+DEREP2
E13GI=E13GI+(F13G+F13GA)/2.)*T1*(1./120.)
E13GA=E13G
IF(ABS(F13GI)-120.1790,790,791
791 IF(E13GI-120.1792,792,793
792 E13GI=-120.
793 GO TO 790
790 TRIM4=E13GI*(.1/120.)+1.
FREDMC=FREDM3*TRIM4
C BLOCK 11
C SPRAY VALVE CONTROL
C
ESP=DERE1-SVP
SVP=SVP+((ESP+FSPA)/2.)*T1*(1./12.)
SVP1=-SVP
ESPA=ESP
IF(SVP1-100.)798,798,799
799 SVP1=-100.
798 CONTINUE
IF(SVC)817,816,817
C WSTH= SPRAY FLOW(THOUSANDS OF POUNDS PER HOUR)
817 WSTH=SVP1
GO TO 818
816 WSTH=0.
818 CONTINUE
C BLOCK 12
C FUEL FLOW CONTROL
C
FFDM=(120./13.)*FREDMC-170.
E20=FFDM-WNGTH
PRO5=E20*.3
FLFU1=FLFU1+(F20+E20A)/2.)*T1*(1./20.)
\[ E20A = F20C \]
\[ S1FL = F1FL1 + PP1I \]
\[ F20F = S1FL - WNGTH \]
\[ WNGTH = WNGTH + ( (E20F + E20FA) / 2. ) \times T1 \times ( 1. / 60. ) \]
\[ E20FA = E2CF \]
\[ IF ( FFC ) R01, R00, R01 \]

801  
\[ WNG = WNGTH \times 1000 \]
GO TO 804

804  
\[ WNG = 5. \times E6 \]
803 CONTINUE

C-------------------------------
C BLOCK 13
C AIR FLOW CALIBRATION AND AIR FLOW CONTROL
C
\[ OXF = O2 - O2SP \]
\[ OXE1 = OXF + ( ( OXE + OXE ) / 2. ) \times T1 \times ( 1. / 1. ) \]
\[ OXE = OXE \]
\[ PEROXF = ( OXE1 \times 100. ) / 02SP \]
\[ IF ( ABS ( PEROXF ) = 5. ) 515, 515, 516 \]

516  
\[ IF ( PEROXE = 5. ) 517, 517, 518 \]

517  
\[ PEROXF = - 5. \]
GO TO 515

518  
\[ PEROXE = 5. \]

515 CONTINUE

\[ TRIM5 = (- 2. / 5. ) \times PEROXF + 1. \]
\[ FRTRIM = FRDFMC \times TRIM5 \]
\[ AFL0W = ( 120. / 13. ) \times FRTRIM + 30. \]
\[ E50 = AFL0W - AFL0W \]
\[ P50 = E50 \times 5 \]
\[ AFI = AFI + ( ( E50 + E50A ) / 2. ) \times T1 \times ( 1. / 15. ) \]
\[ E50A = E50 \]
\[ SAF = P50 + AFI \]
\[ AFL0W = AFL0W + ( ( E60 + E6CA ) / 2. ) \times T1 \times ( 1. / 15. ) \]
\[ E60 = E60 \]
\[ AFL0W = AFL0W + 1000 \]
\[ IF ( AFC ) R20, R19, R20 \]

820  
\[ WRITE ( 6, 527 ) \]

527  
FORMAT ( 37H AIR FLOW CONTROL CONNECTS TO PROCESS )

819 CONTINUE

C-------------------------------
C BLOCK 14
C REHEAT STEAM TEMPERATURE CONTROL ( DAMPER CONTROL )
C
\[ E25 = TO ( 1, 12 ) - TO ( 1, 10 ) \]
\[ C1 = E25 \times 2. \]
\[ SCI = SCI + ( ( E25A + E25 ) / 2. ) \times T1 \times ( 1. / 180. ) \]
\[ E25A = E25 \]
TABLE B-1 CONTINUED

\[ SSCI = SC + SCI \]
\[ PDEC = SSCI + PDEMA \]
\[ IF(RHSTC) I09, 807, 808 \]

\[ R12 = (0.52 - DOC * 0.04/56) / (0.48 + DOC * 0.04/56) \times (0.48/52) \]
\[ GO TO 809 \]

\[ R12 = 1. \] AT 560 MM ONLY
\[ 807 \]
\[ R12 = 1. \]

\[ 809 \] CONTINUE

C BLOCK 15
C TIME INCREMENT AND INTERFACE BETWEEN CONTROL SYSTEM
C AND PROCESS

\[ \text{TIME} = \text{TIME} + T1 \]
\[ T1 = 0.1 \]
\[ \text{IF}((\text{TIME} - T2) > 560, 510, 510 \]

\[ \text{TIME} = \text{TIME} + \text{DTIME} \]

C VARIABLES FOR OPEN LOOP TESTS FOLLOW

\[ \text{RUN NO.} \quad 513 \quad 514 \]
\[ \text{IF (RUN = 3.0)} \]
\[ \text{WRITE}(6,1008) \]
\[ \text{WRITE}(6,1008) \]
\[ \text{FORMAT}((12HSTOP AT 1007) \]

\[ 1007 \]
\[ \text{WRITE}(6,1008) \]
\[ \text{WRITE}(6,1008) \]

\[ \text{FORMAT}((12HSTOP AT 1007) \]

\[ 1005 \]
\[ \text{FWPE} = 2.41E6 \]
\[ \text{FWPE} = 1906.87 \]
\[ \text{PRFP} = 3650. \]
\[ \text{WSRH} = 0.0 \]
\[ \text{TVVXP} = 1.481 \]
\[ \text{R12} = 1.08 \]
\[ \text{DFMW1} = 280. \]
\[ \text{GO TO 500} \]

\[ 1006 \]
\[ \text{WNG} = 2.41E6 \]
\[ \text{FWPE} = 1906.87 \]
\[ \text{PRFP} = 3650. \]
\[ \text{WSRH} = 0.0 \]
\[ \text{TVVXP} = 1.461 \]
\[ \text{R12} = 1.09 \]
\[ \text{DFMW1} = 280. \]
\[ \text{GO TO 500} \]

\[ 510 \]
\[ \text{GO TO 500} \]

C BLOCK 16
C ERROR CHECK

\[ 500 \]
\[ \text{IF}((\text{ABS}(E5) > 0.0005) \]
\[ 570 \]
\[ \text{IF}((\text{ABS}(E6) > 0.0005) \]

C-------------------------
TABLE B-1 CONTINUED

C BLOCK 17
C INTERCONNECTION OF CONTROL VARIABLES WITH PROCFSS
C
490 DO 92 N=1,11
  W(N)=FWRE/3.6
92 CONTINUE
TNE19=TO(1,9)
AA=TVPXR*(3.14159/4.)*(8./12.)*(8./12.)*3.*(7./12.)*(17./12.)
AWG=WNG+200000.
WG(1)=AWG*(4.5099/5.2)
WG(2)=WG(1)*(.4+.04/560.)*DDEC
WG(3)=WG(1)*(.52-.04/560.)*DDEC
PI(1,1)=PBF1
C WS IS SPRAY FLOW IN POUNDS PER SECOND
C CHANGE WS STADY STATE TO 50 LB/3.6 SEC
WS=WSTH/3.6

C BLOCK 18
C HFAT FLUX COMPUTATIONS
C
OCT=QC+QF
C TOTAL CONVEXTIF HEAT+EXCESS HEAT LEAVING FURNACE
C
OCT=WG(1)*CPG*(TG-7C.)/3600.
C QG IS THE TOTAL HFAT LIBERATED BY THE COMBUSTION
C PROCFSS
QG=WNG*HC/3600.
C QR=RADIATIVE PORTION OF THE HEAT TRANSFERRED
QR=QG-OCT
C QC IS THE CONVEXTIVE HEAT TRANSFERRED
QC=OCT-(WG(1)*778.5/(4.509*3600.))
QW(1,1)=.1*QC
QW(1,2)=.0945*QR
QW(1,3)=.185*QR
QW(1,4)=.17*QR
QW(1,5)=.185*QR
QW(1,6)=.151*QR
QW(1,7)=.129*QC
QW(1,8)=.0645*QC
QW(1,9)=.161*QC
QW(1,10)=.195*QC+.21*QR
QW(1,12)=.35*QC
QW(1,12)=.35*QC+.12
QW(1,9)=.511*QC-QW(1,12)
TABLE B-1 CONTINUED

C BLOCK 19
C SOLUTION OF THE PROCESS EQUATIONS
C
INCRHO = 0
DO 50 N = 1, J
50 CONTINUE
DO(1,N) = PI(1,N) - KP(N)*W(N)*W(N)
PI(1,N+1) = DO(1,N)
DO 40 J = 1, 3
P = DO(J,N)
H = HO(J,N)
CALL TSSPH
T(1,N) = T
IF(PR3)94, 91, 94
94 CONTINUE
C TO SKIP COMPUTING RHO THREE TIMES
IF(J-1)91, 93, 91
93 CONTINUE
IF(N-9)91, 87, 87
C COMPUTE RHO(N)
87 IF(N-12)88, 91, 88
88 CONTINUE
DOA(N) = (DO(1,N) + DO(1,N-1))/2.
TOA(N) = (TO(1,N) + TO(1,N-1))/2.
90 P = DOA(N)
T = TOA(N)
CALL SPVOL
RHO(N) = RHOSV
DPHO(N) = RHO(N) - PHON(N)
PHON(N) = RHO(N)
W(N) = W(N-1) - V(N)*DRHO(N)/DTIME
91 CONTINUE
O(J1,N) = KO(N)*X(N)*8*(TW(J1,N) - TO(J1,N))
DTWP(J1) = DTIME*QW(1,N) - Q(J1,N) / MWCPW(N)
DHOR(J1) = DTIME*{W(N)*HI(1,N) - HO(J1,N)} + Q(J1,N)/M(N)
IF(J1-3)80, 70, 70
80 TW(2,N) = TW(1,N) + DTWP(1)/2.
TW(3,N) = TW(1,N) + 2.*DTWR(2) - DTWR(1)
HO(2,N) = HO(1,N) + DHOR(1)/2.
40 HO(3,N) = HO(1,N) + 2.*DHOR(2) - DHOR(1)
70 INCRHO = INCRHO + (DHOR(1)*4.*DHOR(2) + DHOR(3))/6.
HO(1,N) = HO(1,N) + INCRHO
IF(N-9)85, 84, 85
C ADD SPRAY FLOW ENTHALPY INCREMENT TO HO(1,9)
84 QO = W(9)*HO(1,9)
PI(1,1) = PO(1,1) - (W(1)*25.)/1105.
TABLE B-1 CONTINUED

P = PI(1,1)
H = HI(1,1)
CALL TSSPH
T0(1,1) = T
Q0 = Q0 - DOS
HI(1,10) = Q10/W(9)
GO TO 86

85 CONTINUE
HI(1,N+1) = HO(1,N)

86 CONTINUE
TW(1,N) = TW(1,N) + ((DTWR(1) + 4 * DTWR(2) + DTWR(3)) / 6)

50 CONTINUE

C COMPUTE PROCESS VARIABLES FOR LUMP 12

52 N = 12
P0(1,16) = P0(1,11) - (W(11) * W(11)) / ((1846.541 * PHI(11) * AA * AA))
W(12) = (3480 / 3990) * W(11)
PI(1,12) = 668.
HI(1,12) = 1295.

C CORRESPONDING TI(1,12) = 596.044
INCRHO = 0.
PO(1,N) = PI(1,N) - KP(N) * W(N) * W(N)
PI(1,N+1) = PO(1,N)
DO 73 J1 = 1, 3
P = PO(1,N)
H = HO(J1,N)
CALL TSSPH
T0(J1,N) = T
Q(J1,N) = Q0(N) * W(N) ** 8 * (TW(J1,N) - TO(J1,N))
DTWR(J1) = DTIME * (QW(1,N) - O(J1,N)) / MWCPW(N)
DHOR(J1) = DTIME * (W(N) * (HI(1,N) - HO(J1,N)) + Q(J1,N)) / M(N)
IF(J1 = 3) 74, 75, 75
74 TW(2,N) = TW(1,N) + DTWR(1) / 2.
TW(3,N) = TW(1,N) + 2 * DTWP(2) - DTWR(1)
HO(2,N) = HO(1,N) + DHOR(1) / 2.
73 HO(3,N) = HO(1,N) + 2 * DHOR(2) - DHOR(1)
75 INCRHO = INCRHO + ((DHOR(1) + 4 * DHOR(2) + DHOR(3)) / 6)
HO(1,N) = HO(1,N) + INCRHO
HI(1,N+1) = HO(1,N)
TW(1,N) = TW(1,N) + ((DTWR(1) + 4 * DTWR(2) + DTWR(3)) / 6)

C TO(1,10) IS ADJUSTED TEMPORARILY
TO(1,10) = TO(1,10) - 8.89

C BLOCK 20
C TURBINE EQUATIONS
C
TABLE B-1 CONTINUED

\[ W(11) = W(12) \times (30AC./34AC.) \]

\[ PWFR1 = 0.285 \times (H0(1,11) - 1276) \times W(11) \times 3.6E-3 \]

\[ PWFR2 = 0.240 \times (H0(1,12) - 1113) \times W(11) \times 3.6E-3 \]

C

THE TOTAL POWER IN MEGAWATTS IS POWER

\[ \text{POWER} = \text{POWER}_1 + \text{POWER}_2 \]

C

BLOCK 21

C

LOW AND HIGH PRESSURE FEEDWATER HEATERS

C

COMPUTE THE EXTRACTION FLOW FROM THE HIGH PRESSURE TURBINE

C

\[ WF1 = W(11) - W(12) \]

C

THE AVERAGE FLOW THROUGH THE IPT AND LPT IS WC

\[ WC = (3.2/3.03) \times W(11) \]

C

COMPUTE THE EXTRACTION FLOW FROM THE INTERMEDIATE AND LOW PRESSURE TURBINES. (LB./SEC.)

\[ WF2 = W(12) - WC \times 0.85 \]

C

HE1 = 1400*

C

OW(1,14) = WE1*HF1

C

HF2 = 1400*

C

OW(1,13) = WE2*HE2

C

ASSUME THE INPUT CONDITIONS INTO THE LPFWH ARE

C

CONSTANT

C

H019 = 69.1

C

Q113 = W(12)*H019

C

Q13 = OW(1,13) - WF2*H019

C

Q013 = Q113 + Q13

C

Q013 IS THE HEAT CONTENT OF THE WATER LEAVING LPFWH (BTU/SEC.)

C

H013 IS THE ENTHALPY OF THE WATER LEAVING LPFWH

C

Q114 = W(11)*H013

C

Q14 = Q013 + Q114

C

Q014 = Q013 + Q14

C

Q014 IS THE HEAT CONTENT OF THE WATER LEAVING LPFWH

C

H014 IS THE ENTHALPY OF THE WATER ENTERING THF

C

H014 IS THE HEAT CONTENT OF THE WATER ENTERING THF

C

HPFWH

C

Q114 = W(11)*H013

C

Q14 = Q014 + Q113

C

Q014 IS THE HEAT CONTENT OF THE COMPRESSED WATER LEAVING HPFWH

C

ASSUME THAT ENTHALPY CHANGE ACROSS THE BFP IS

C

CONSTANT

C

H014 IS THE ENTHALPY OF THE COMPRESSED WATER LEAVING THE HPFWH

C

H014 = Q014/W(11)

C

H1(1,1) = H014

C

H1(1,1) SHOULD BE 487.5 AT 560 MW, STEADY STATE

C

BLOCK 22
TABLE B-1 CONTINUED

C PRINT OUTPUT

IF (TIME = TPRINT) 1014, 63, 63
63 WRITE (6, 15) TIME, TVAXP, FWRF, T0(1, 7), T0(1, 11), T0(1, 12)
15 FORMAT (2CX, F10.1, 5F10.2)
WRITE (6, 1000) POWFR, P0(1, 7), P0(1, 11)
1000 FORMAT (40X, 3F10.2/
TPRINT = TPRINT + 10.
1014 IF (RUN = 1.0) 72, 72, 1003
1013 IF (OLT) 72, 72, 1011
72 IF (TIME = 240.) 81, 81, 82
81 GO TO 650
82 DF

91 GO TO 650
92 DF

1004 IF (TIME = 600.) 1004, 1012, 1012
1012 RUN = RUN + 10.
1011 IF (TIME = 600.) 1004, 1012, 1012
1003 IF (RUN = 3.) 1022, 1021, 1021
1022 IF (TIME = 450.) 81, 81, 1020
1020 RUN = RUN + 10.
83 CONTINUE

DATA J/11/
DATA XQ/10*40., -40., 40., 2*0./
DATA M/46150., 6670., 12400., 4720., 4105., 3075., 5400., 335
10., 6690., 5020., 1505., 3000., 2*0./
DATA MW/857000., 176500., 382000., 172000., 199000., 241000
1., 546000., 413000., 782000., 932000., 312000., 978000., 2*0.
1/
DATA CPW/12*., 169., 2*0./
DATA HI/487.5, 32*0., 1285., 29*0./
DATA HO/549.05, 2*0., 599.13, 2*0., 697.15, 2*0., 787.43, 2*0.
1., 887.4, 2*0., 967.66, 2*0., 1047.9, 2*0., 1088.07, 2*0., 1188
2.4, 2*0., 1422.32, 2*0., 1415.85, 2*0., 1533.604, 29*0.
DATA TW/592.66, 2*0., 633.94, 2*0., 659.24, 2*0., 730.58, 2*0.
1., 768.35, 2*0., 782.01, 2*0., 792.66, 2*0., 796.46, 2*0., 829.
291, 2*0., 1070.76, 2*0., 973.17, 2*0., 1066., 29*0./
TABLE B-1 CONTINUED

```
DATA 0W/68500, 55400, 104500, 99800, 110500, 88700, 108700, 44400, 110500, 257000, -3320, 241000, 9
STOP
END
```

```
COMPUTATION OF THE STEAM TABLE DATA

SUBROUTINE TSSPH
COMMON H,P,T,RHOSV
HMIN=7C8.5-.0033*P
IF(H-HMIN) 5, 6, 6
5 COL=26.168337
  C02=1.1506135
  C03=-1.1481565F-3
  C04=3.64562836F-6
  C05=-5.3719405F-9
  C06=2.429964F-12
  C07=-7.7482837F-5
  C08=6.5137189F-7
  C09=-2.152367F-9
  C10=3.1233032E-12
  C11=-1.601857OF-15
  C12=6.969206F-15
  C13=-8.0459901F-12
  C14=2.896191E-14
  C15=-4.198743E-17
  C16=1.9534042E-20
  T=C01+C02*H+C03*H**2+C04*H**3+C05*H**4+C06*H**5
  1+P*(C07*H+C08*H**2+C09*H**3+C10*H**4+C11*H**5)
  1+P**2*(C12*H+C13*H**2+C14*H**3+C15*H**4+C16*H**5)
GOT02
6 DFLT=461.-.035*P-.39*H
  IF(P-1000.)7,7,8
8 IF(H-1280.)9,7,7
7 A01=-1.0659659E4
  A02=2.0110965F1
  A03=-1.250954F-2
  A04=2.9274992F-6
  A05=4.9815820
  A06=-7.7618275F-6
  A07=2.4391612F-10
  A08=-9.8147341E-3
  A09=6.582439CF-6
  A10=-1.4747938F-9
GOT010
9 A01=-4.5298646E3
  A02=15.358850
```
TABLE B-1 CONTINUED

\[ AC_3 = -1.5655537 \times 10^{-2} \]
\[ A_{04} = 5.2687849 \times 10^{-6} \]
\[ A_{05} = 4.4185386 \times 10^{-1} \]
\[ A_{06} = -9.1654905 \times 10^{-6} \]
\[ A_{07} = 2.7540766 \times 10^{-10} \]
\[ A_{08} = -1.1541553 \times 10^{-3} \]
\[ A_{09} = 1.2384560 \times 10^{-8} \]
\[ A_{10} = -4.1724604 \times 10^{-10} \]

\[ T = A_{01} + A_{02} H + A_{03} H^2 + A_{04} H^3 + A_{05} P + A_{06} P^2 + A_{07} P^3 \]
\[ 1 + P \left( A_{08} H + A_{09} H^2 + A_{10} H^3 \right) \]

RETURN
END

C-------------------------------
C
C SUBROUTINE SPVOL
COMMON H,P,T,RHOSV
C THIS PROGRAM Computes the Specific Volume of Superheated Steam
C
COMPUTE VV1
PP=P/14.696
TT=(5./9.)*(T-32.)
VK1=273.16
P=4.5504
TAU=1./(VK1+TT)
VV1=R/(PP*TAU)

C COMPUTE VV2
A=1.89
B=2.641.62
C=10.
D=80870.
ALPHA=B*C*(D*TAU**2.)
BO=A-TAU*ALPHA
VV2=B0
BC=-BO

C COMPUTE VV3
E=82.546
F=1.6246E5
DELTA=E-F*TAU
VV3=((BO*TAU)**2.)*DELTA*PP

C COMPUTE VV4
G=0.21828
H=1.2697E5
EPSI=G-H*(TAU**2.)
VV4=EPSI*(BO*TAU)**2.*DELTA*PP

C COMPUTE VV5
VJ=3.635E-4
VK2=6.768E-8
TABLE B-1 CONTINUED

\begin{align*}
R_{HJ2} &= VJ - VV2 * (\Omega * (10.0 ** 3.0)) ** 24.0 \\
VV5 &= RH02 * (BO ** 13.0) * (\Omega ** PP) ** 12.0 \\
VV5 &= -VV5 \\
C \text{ \hspace{1cm} COMPUTE SPECIFIC VOLUME, VNU} \\
Y &= 0.160185 \\
VNU &= (VV1 + VV2 + VV3 + VV4 - VV5) * Y \\
C \text{ \hspace{1cm} DFNSITY=1./SPECIFIC VOLUME = RHO SV} \\
RHO SV &= 1.0 / VNU \\
\text{RETURN} \\
\text{END} 
\end{align*}
VITA

Alfred J. Flechsig, Jr. was born on October 16, 1935, in Tacoma, Washington. He is the son of Alfred J. and Josephine M. Flechsig and has one sister, Patricia. He attended grade school in Seattle, Washington; Portland, Oregon; and Auburn, Washington. He graduated from Auburn Senior High School in 1953. He then attended Washington State University and studied electrical engineering, graduating with a B. S. degree in 1957. He was married to the former Nancy L. Fletcher in the summer of 1957 and then attended graduate school at Washington State University. He received the Master of Science in Electrical Engineering Degree in 1959. He then spent one year of graduate study at the University of Wisconsin and following that, returned to Washington State University to begin a teaching career in the Department of Electrical Engineering where he is presently an assistant professor. In addition to teaching during the academic year he worked in industry during the summer months. The industrial experience included positions with the U. S. Army Corps of Engineers; Boeing Airplane Company; Pacific Power and Light Company; U. S. Bureau of Reclamation and the Department of Water and Power, City of Los Angeles. In 1967 he began his doctoral program at Louisiana State University and has been studying there full-time until the present. He is the father of two girls, Becky Jo, born July 6, 1958, and Jennifer Lynn, born December 6, 1960.
EXAMINATION AND THESIS REPORT

Candidate: Alfred Julius Flechsig, Jr.

Major Field: Electrical Engineering

Title of Thesis: A Digital Simulation of a Once-Through Supercritical Steam Generator

Approved:

[Signatures]

Major Professor and Chairman

Dean of the Graduate School

EXAMINING COMMITTEE:

[Signatures]

Date of Examination:

September 18, 1969