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SASSYS-1 BALANCE-OF-PLANT COMPONENT MODELS FOR AN INTEGRATED PLANT RESPONSE*

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Introduction

Models of power plant heat transfer components and rotating machinery have been added to the balance-of-plant model in the SASSYS-1 liquid metal reactor systems analysis code [1]. This work is part of a continuing effort in plant network simulation based on the general mathematical models developed in [1] and [2]. The models described in this paper extend the scope of the balance-of-plant model to handle non-adiabatic conditions along flow paths. While the mass and momentum equations remain the same as in [2], the energy equation now contains a heat source term due to energy transfer across the flow boundary or to work done through a shaft. The heat source term is treated fully explicitly. In addition, the equation of state is rewritten in terms of the quality and separate parameters for each phase [3]. The models are simple enough to run quickly, yet include sufficient detail of dominant plant component characteristics to provide accurate results.

Analytical Models

Tables 1 and 2 list the various types of heat transfer and rotating machinery components. The seven types of heaters and the turbine model will be discussed below; the feedwater pump model is addressed in [2]. Simple diagrams of each component are given in Fig. 1.

Table 1. Heat Transfer Components

Open heater:	Deaerator
Closed heater:	Condenser
	Reheater
	Flashed heater
	Drain cooler
	Desuperheating heater
	Desuperheater/Drain Cooler

Table 2. Rotating Machinery Components

Feedwater Pump Turbine

<u>Heat Transfer Component Models</u>

Heaters fall into two classifications: open heaters and closed heaters. The term "open heater" refers to the fact that there is no distinction between tube and shell sides, so that hot fluid and cold fluid entering the heater mix together. As seen in Fig. 1a, an open heater is actually a closed volume containing liquid and vapor at saturation conditions. The term "closed heater" indicates that hot and cold fluids are separated between a tube side and a shell side. Heat transfer occurs across the tube without contact between hot and cold fluids. Closed heaters are shown in Figs. 1b through 1h. They consist of a closed volume, or shell side, and a tube bundle, or tube side. Flow is carried into and out of the tube bundle by pipes which lie outside the heater boundary.

The following assumptions are made in all seven heater models. Flow is incompressible on both shell and tube sides. Any two-phase fluid entering on the shell side instantaneously separates into liquid and vapor, and a new thermal equilibrium is reached immediately. The two-phase interface serves as the reference point for the saturation pressure. The momentum equation governing flow entering and exiting the shell side accounts for elevation pressure differences as gravity heads. Each phase is at a uniform temperature and enthalpy.

The tube bundle is modelled as a single tube. Mass flux and pressure drop in the tube are the same as in the tube bundle, and the mass of the metal tubing is also conserved. These constraints do not allow tube length or surface area to be conserved, and so the tube surface heat transfer area is corrected to simulate the bundle heat transfer area through the use of calibration factors which provide an effective thermal resistance for conduction heat transfer.

Details of each heater will now be described, beginning with the simplest model, the deaerator, and progressing to the more complex models.

Deaerator

A deaerator is used to remove dissolved gases from an incoming fluid. This heater is a right circular cylinder standing on end, as shown in Fig. 1a. Normally, each flow opening is located entirely within either the liquid or the vapor region; however, as the two-phase level moves during a transient, it may intersect an opening, causing two-phase fluid to flow out of the deaerator. Under these circumstances, the model assumes a slip ratio of one for the two-phase fluid and uses area weighting to compute the quality of the exiting fluid. The closed heater models treat two-phase outlet flow this same way.

The deaerator and the closed heater models must also cope with small imbalances between incoming and outgoing energies in the steady state. These imbalances are caused by small uncertainties in the correlations for thermodynamic properties. The heater models solve this problem by introducing a pseudo-heat conduction energy transfer term. The temperature gradient between the heater and the ambient conditions serves as the driving force, and the code computes a pseudo-heat transfer coefficient which will give a steady-state energy balance. The coefficient is positive if energy is accumulating and negative if energy is draining. This coefficient is then kept constant throughout the transient.

Condenser

A condenser is used to convert steam from the turbine to liquid. As shown in Fig. 1b, a condenser is a box with a tube bundle passing horizontally through it; this is essentially a deaerator with a tube bundle running through the vapor region. The tube bundle may have bends in it. The fluid on the tube side is assumed to be single phase. The tube bundle is assumed to be contained entirely in the vapor region, so a condensation heat transfer coefficient is used on the shell side if the temperature on the tube side is lower than the shell side vapor temperature, and a coefficient computed from the Dittus-Boelter equation is used if the tube side temperature is higher than that on the shell side. In either case, the heat transfer coefficient must be adjusted in the steady state so that the tube side temperature distribution is consistent with the temperatures in the remainder of the plant.

The heat transfer coefficient on the tube side is calculated using the Dittus-Boelter correlation. Heat transfer between the tube side and the shell side is assumed to take place through the mechanism of radial conduction only; axial conduction is neglected. The tube side is discretized as shown in Fig. 2, and the temperature distribution on the tube side is determined using the differenced form of the basic heat conduction equation,

$$m_{it} \frac{dh_{it}}{dt} = Q_{it} + w \left(h_{i-1,t} - h_{it}\right) , \tag{1}$$

where

$$Q_{it} = \frac{T_{im} - T_{it}}{R_{it}} . \tag{2}$$

The tube metal temperatures T_{im} are computed from a similar set of equations,

$$Q_{is} = \frac{T_s - T_{im}}{R_{is}} , \tag{3}$$

and

$$m_{im} c_m \frac{dT_{im}}{dt} = Q_{is} - Q_{it} . \tag{4}$$

These equations apply to all closed heater models.

When the shell side is hotter than the tube side, it is assumed that no local boiling occurs on the tube side and that the condensate does not form a wetted perimeter along the tube outer periphery.

Reheater

A reheater is used to improve turbine performance by reheating the moist steam as it passes between stages of the turbine. The basic features of a reheater are depicted in Fig. 1c. The shell side is assumed to be at a lower temperature than the tube side. In this type of heater, the shell side is all vapor, and the reheater appears to be little more than the vapor section of a condenser. However, there is one important difference: the fluid on the tube side changes phase from steam to liquid as it passes through the reheater. Modelling this phase transition requires use of the energy equation; however, the energy equation in the balance-of-plant formulation is solved at flow junctions (such as the shell side of a heater), not along flow paths (such as the tube side). Therefore, the reheater is modelled in the configuration shown in Fig. 1d. The shell side of Fig. 1c becomes the tube side in Fig. 1d, with steam flowing within the tube, and the tube side of Fig. 1c becomes the shell side in Fig. 1d. The shell side of Fig. 1d then easily models the phase transition which occurs in the tube side of the reheater. The reconfiguration of the reheater to Fig. 1d is done so as to conserve volume on both sides of the heater.

The model of Fig. 1d is very similar to a condenser except that the tube side is vertical instead of horizontal. This means that the tube passes through both liquid and vapor. Since the tube side vapor is heated by the shell side fluid, the heat transfer coefficient on the shell side is computed from the Dittus-Boelter correlation in the liquid region and as a condensation coefficient in the vapor region. Usually, the two-phase interface will fall within one of the nodes which discretizes the tube side; in this case, the heat transfer coefficient within the node is an area-weighted combination of the condensation coefficient and the Dittus-Boelter coefficient.

Flashed Heater

A flashed heater is needed when liquid upstream of the heater shell side is above the shell side saturation point. The flashed heater allows the liquid to flash safely upon entering the heater. The geometry of the flashed heater is given in Fig. 1e. This is a right circular cylinder lying on the side, with a u-shaped tube bundle that is partially submerged in liquid and partially surrounded by vapor. The bundle enters and leaves the shell side through one end of the cylinder, with the entrance and exit at two different elevations.

The tube-side fluid is assumed single phase, and fluid can enter at either the lower or the upper elevation. These assumptions are also made in the drain cooler, desuperheater, and desuperheater/drain cooler models to be described in subsequent sections.

If the tube side is cooler than the shell side, the heat transfer coefficient along the tube surface is computed as for the reheater. If the tube side is hotter than the shell side, the coefficient within the vapor region is calculated from the Dittus-Boelter equation, and in the liquid region, a boiling heat transfer coefficient from the water pre-DNB correlation [4],

$$h = \left(e^{P/(8.7\times10^{6})}/22.65\right)q^{0.5} , \tag{5}$$

is used.

Predicting the level of the two-phase interface is a difficult problem in a heater of this configuration for two reasons. First, the cross-sectional area parallel to the cylinder axis varies in the vertical direction (remember, the cylinder is lying on the side), and so the area as a function of vertical position must be found from a transcendental equation. A non-iterative scheme has been developed to solve this equation. Second, the volume taken up by the tube bundle must be considered when determining the two-phase level. In order to simplify the calculation, the void fraction is taken to be the ratio of the vapor volume divided by the vapor volume plus the liquid volume (rather than dividing by the total volume, which is the sum of the volumes of vapor, liquid, and tube bundle) when the interface falls between the bundle inlet and outlet. This amounts to assuming that the tube bundle is uniformly distributed throughout the region between the bundle inlet and outlet. The interface position can then be computed from the geometry and the void fraction.

Drain Cooler

When the shell side outlet liquid from a heater must be sufficiently subcooled so as to remain liquid at the lower pressure of a downstream component, a drain cooler is needed.

As shown in Fig. 1f, the drain cooler configuration is identical to that of the flashed heater with the addition of a fixed volume drain in a lower corner of the cylinder. The top of the drain extends horizontally across the cylinder perpendicular to the cylinder axis (see end view in the same figure). The drain is separated from the remainder of the shell side except for a flow inlet which brings saturated liquid from the heater into the drain. There is also a flow outlet which carries liquid out of the drain and away from the heater. Inlet and outlet flows must always be equal. Liquid within the drain is subcooled. One end of the tube bundle passes through the drain, as seen in Fig. 1f.

A one-node energy equation,

$$m_D \frac{dh_D}{dt} = Q_D + w_D \left(h_{in} - h_D\right) , \tag{6}$$

is used on the shell side along the tube within the drain. On the tube side, the multi-node treatment is retained. A heat conduction heat transfer coefficient is computed from the Dittus-Boelter equation on the shell side of the tube surface regardless of whether the tube side is hotter or colder than the shell side.

Adjustments are made separately to the drain and to the remainder of the shell side in order to achieve a steady state energy balance which is consistent with conditions in the remainder of the plant. First, the code computes a calibration factor to adjust the tube surface heat transfer area within the drain to achieve an energy balance just in the drain. Then, energy is balanced in the remainder of the heater by computing a separate tube surface heat transfer area calibration factor plus a pseudo-heat transfer coefficient to account for the uncertainties in the properties correlations.

The level of the two-phase interface is found as described for the flashed heater except now the calculation is further complicated by the need to account for the geometry of the drain.

Desuperheating Heater

A desuperheating heater is needed when the steam entering the heater shell side is highly superheated. The entering steam is initially contained in a desuperheating region, where it transfers heat to the tube side fluid rather than dissipating heat to the shell side saturated steam. Steam moving from the desuperheating region to the main section of the shell side is near saturation. The desuperheating region also protects the remainder of the heater from being damaged by highly superheated steam.

The desuperheating heater is diagrammed in Fig. 1g; it can be summed up as a drain cooler turned upside down. Instead of a drain at the bottom, it has a desuperheating region at an upper corner. The desuperheating region is filled with superheated vapor at the saturation pressure of the heater two-phase interface. Normally, the tube side is filled with a single phase fluid which is cooler than the vapor in the desuperheating region. All other aspects of the drain cooler model apply to the desuperheating heater.

Desuperheater/Drain Cooler

This heater is pictured in Fig. 1h and is a combination of the drain cooler and the desuperheating heater. All details discussed above for these two models apply also to the desuperheater/drain cooler. Two additional points to note are that 1) the mass flow rate entering the desuperheating section can differ from that entering the drain, and 2) separate calibration factors are used in the desuperheating section and in the drain for adjusting the tube surface heat transfer area to conserve energy.

Rotating Machinery Model

Turbine

A turbine is composed of many stages driving one rotor which extracts work from the flow. A series of volumes is used to model the various stages in the turbine. The stages are connected by nozzles which permit both non-choked and choked flow. Compressible flow is now very important in describing the flow behavior in the nozzles. Separate correlations based on thermodynamic conditions at the inlet are used for the nozzle flow depending on whether the flow is choked or not and whether the fluid is single phase or two phase. Turbine efficiency is based on losses to isentropic expansion and shaft work is then calculated using quasi-empirical correlations for stage efficiency. Stage efficiency is affected by many less factors like rotation loss and moisture loss. Steam at the extraction ports is treated as incompressible flow.

Integrated Test Problem

All heater models have been tested as standalone models, and the condenser, deaerator, and drain cooler have been operated as part of an integrated test problem which simulates an entire LMR plant. The configuration of the water side of the integrated test problem is diagrammed in Fig. 3a.

Heaters 1 to 3 are drain coolers. The system contains subcooled water except on the shell side of all heaters and in the line beyond the saturation point inside the steam generator. The lines which bleed two-phase fluid from the turbine to the heaters (shown as dotted lines) are represented by flow boundary conditions, as shown in Fig. 3b. This schematic depicts the nodalization diagram of the discretized system. See Ref. [2] for an explanation of the discritization algorithm. The turbine is represented by a pressure boundary condition, as seen in Fig. 3b. The water side of the once-through steam generator [5] is explicitly coupled to the rest of the balance of plant.

The test problem begins with a 20 sec null transient, and then a 12.5% step load reduction (in 3 sec) in the turbine is induced to start the transient. This is done by partially closing the valve upstream of the turbine and simultaneously decreasing the flows in the flow boundary conditions over a period of 10 sec. The test represents a moderate transient in the plant system. The system has a large overall time constant of about 20 minutes. As shown in Fig. 4a, the power is slowly approaching a new steady state by the end of the calculation at 500 sec. The steam valve and feedwater pump flows shown in Fig. 4b make some slight adjustments during the null transient, then drop due to the load reduction and attain a new steady state by the end of the calculation. The net effect of the decrease in reactor power and flow through the steam generator is to increase the steam generator outlet temperature, as shown in Fig. 4c; the inlet temperature remains constant due to the large time constant of the system. Figure 4d shows the pressure history of heater 1.

The calculation was done on a Cray-XMP mainframe computer. The CPU time for running the full SASSYS-1 code for this integrated plant model is about 10% higher than real time.

Concluding Remarks

The models described in this paper have been shown to give consistent results and to be computationally efficient. Plans for future work include introducing a multi-node energy equation on the shell side of the drain and the desuperheating region in the appropriate heater models, providing for flow reversal, and improving timestep control to enhance the efficiency of the code. A more extensive model validation effort will also be undertaken.

Acknowledgments

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Nomenclature

c	specific heat (J/kgK)
h	enthalpy (J/kg), heat transfer coefficient (J/m^2sK)
m	mass (kg)
P	Pressure (Pa)
Q	heat transfer rate (J/s)
q	heat flux (J/m^2s)
R	thermal resistance $(J/sK)^{-1}$
T	Temperature (K)
t	time (s)
w	flow rate (kg/s)

Subscripts

D	drain
i	ith node
in	inlet condition
m	metal tube
s	shell side
t	tube side

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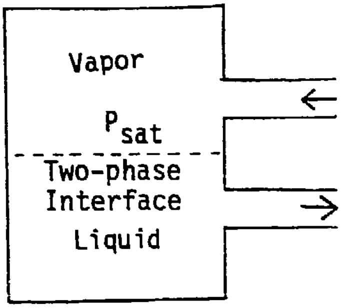

Fig. 1a. Deaerator

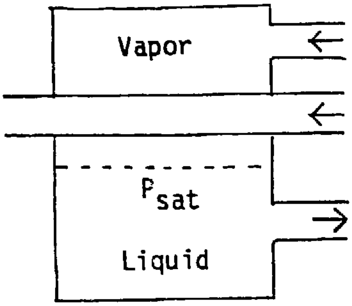

Fig. 1b. Condenser

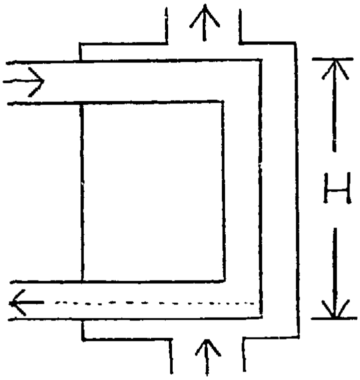

Fig. 1c. Reheater

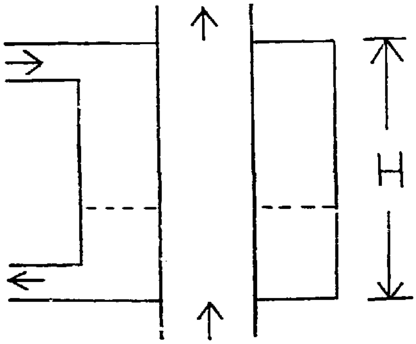

Fig. 1d. Reconfiguration of the Reheater

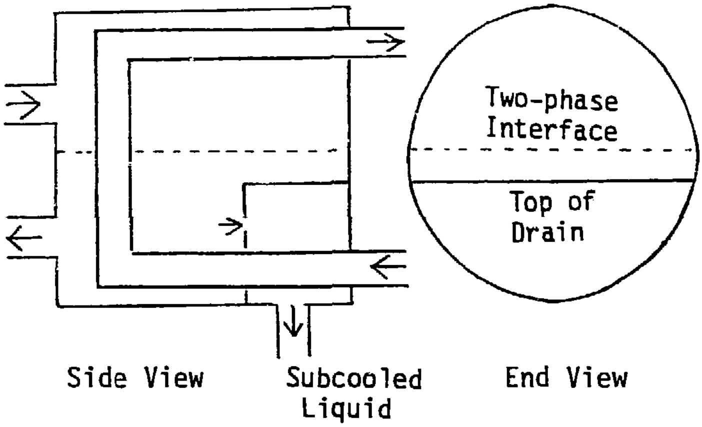

Fig. 1f. Drain Cooler

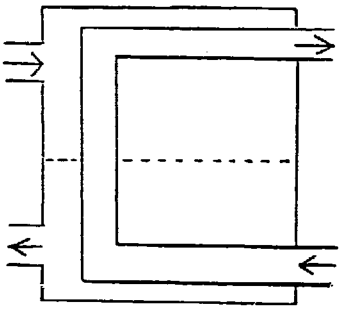

Fig. 1e. Flashed Heater

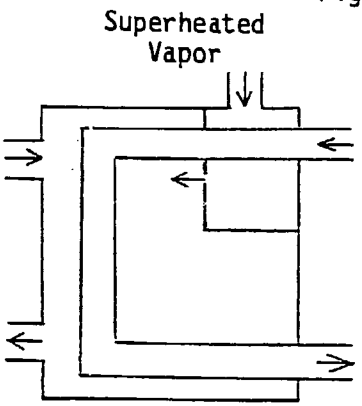

Fig. 1g. Desuperheating Heater

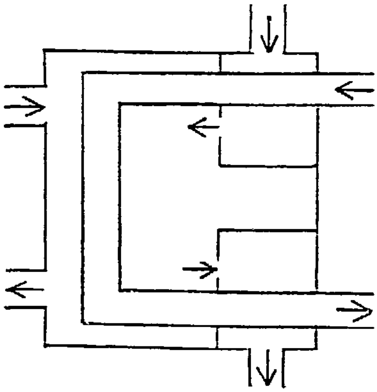

Fig. 1h. Desuperheater/ Drain Cooler

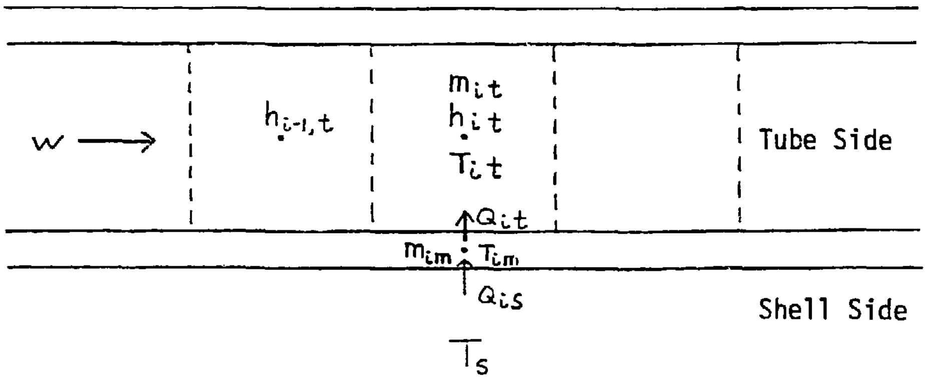

Fig. 2. Tube Side Nodalization

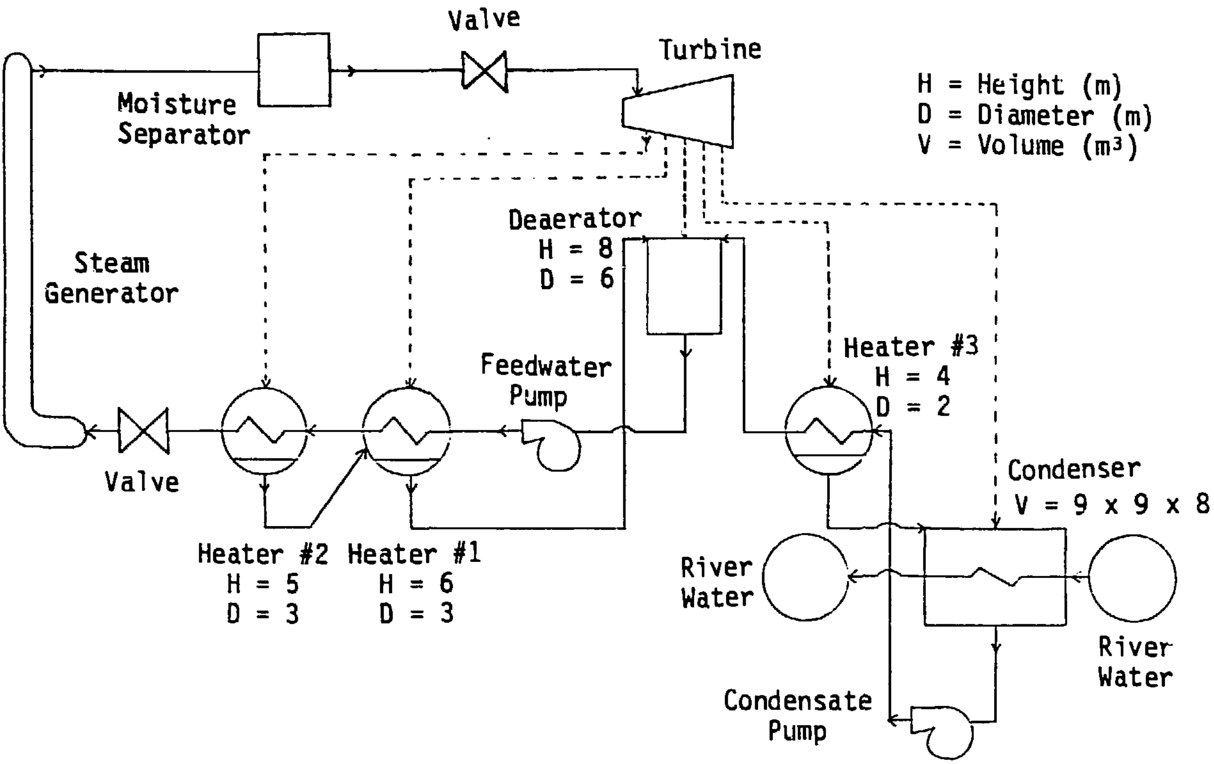

Fig. 3a. Integrated Test Problem Configuration

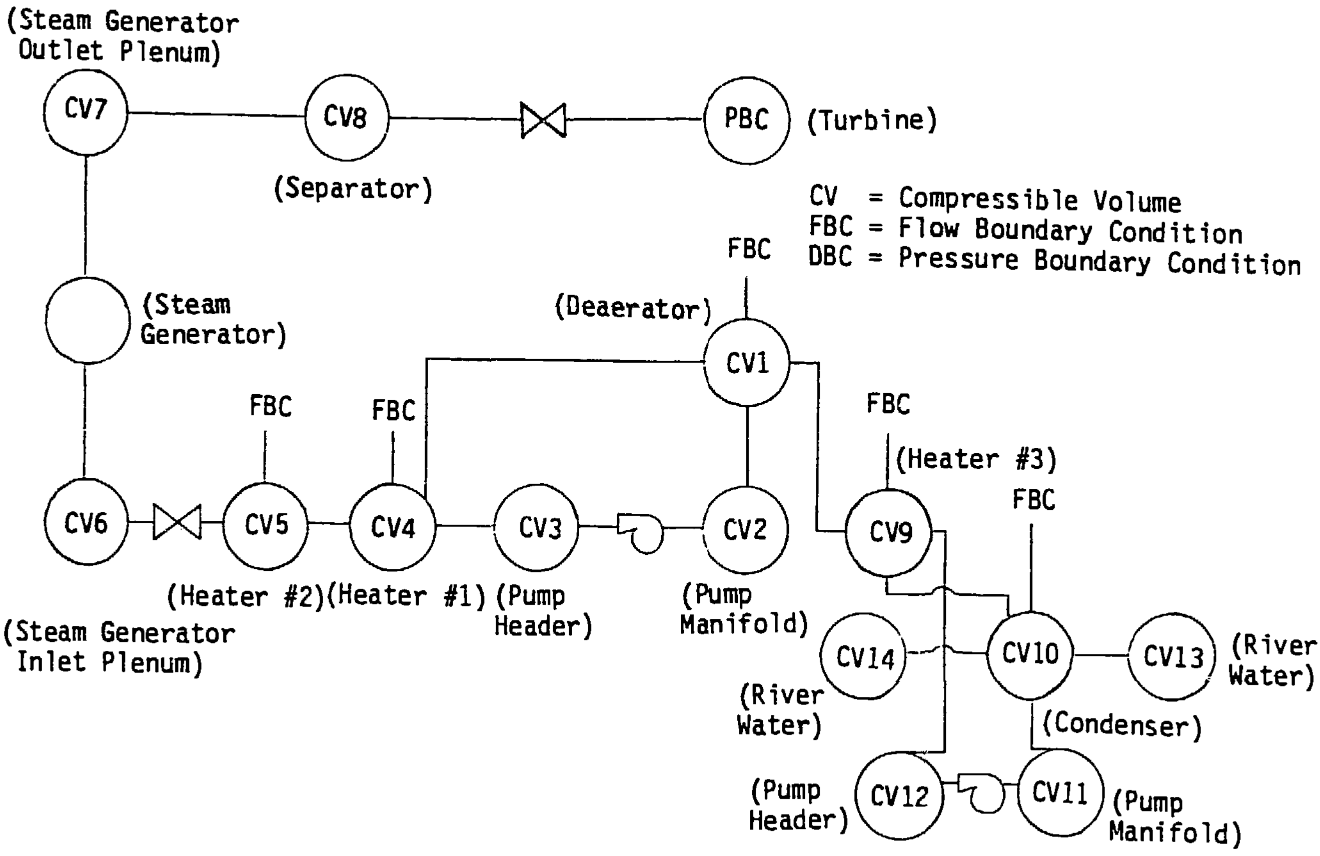

Fig. 3b. Discretization of the Test Problem

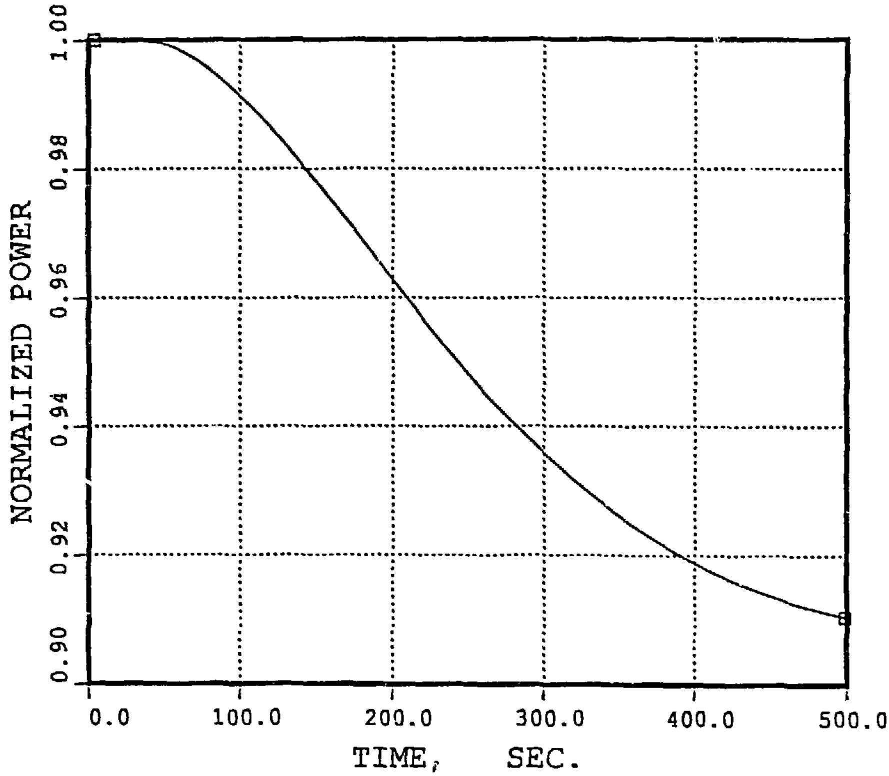

Fig. 4a. Total and Decay Power vs. Time

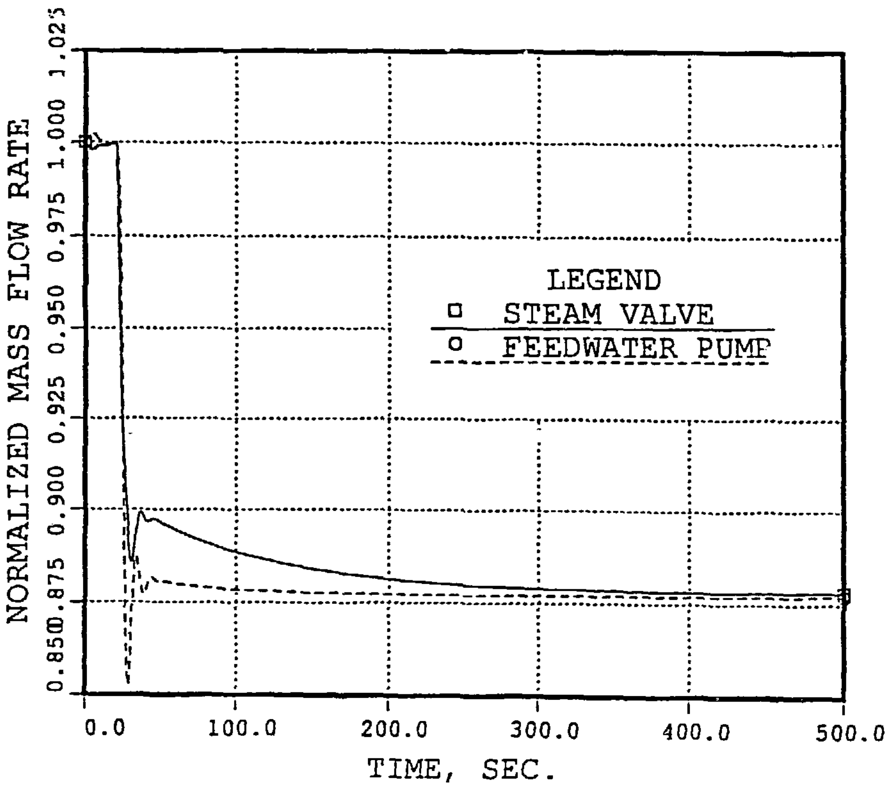

Fig. 4b. Steam Valve and Feedwater Pump Flows History

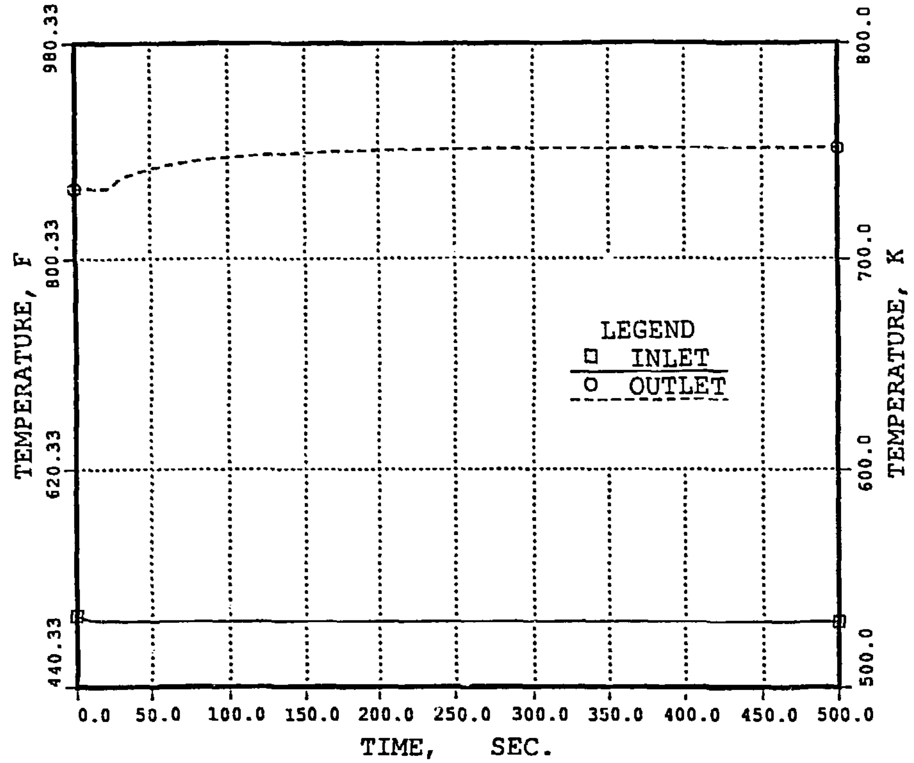

Fig. 4c. Steam Generator Water Temperatures at the Inlet and Outlet

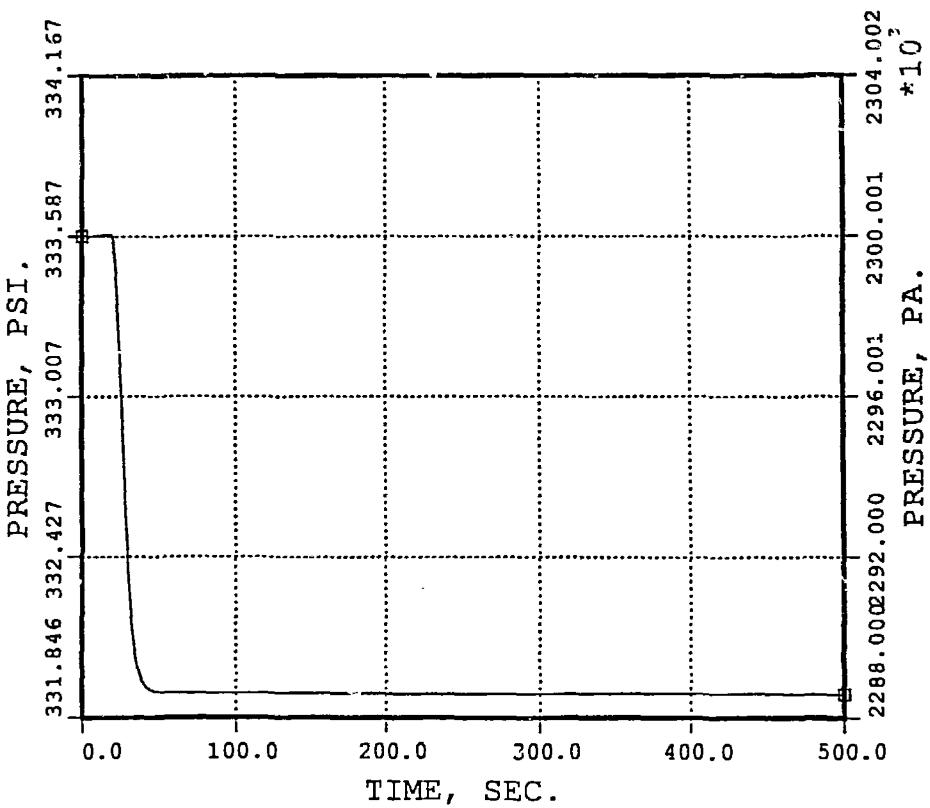

Fig. 4d. Pressure History in Heater 1