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A NEW BALANCE-OF-PLANT MODEL FOR THE SASSYS-1 LMR SYSTEMS ANALYSIS CODE*

by

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Introduction

A balance-of-plant model has been added to the SASSYS-1 liquid metal reactor systems analysis code [1]. Until this addition, the only waterside component which SASSYS-1 could explicitly model was the water side of a steam generator, with the remainder of the water side represented by boundary conditions on the steam generator.

The balance-of-plant model is based on the model used for the sodium side of the plant [1]. It will handle subcooled liquid water, superheated steam, and saturated two-phase fluid. With the exception of heated flow paths in heaters, the model assumes adiabatic conditions along flow paths; this assumption simplifies the solution procedure while introducing very little error for a wide range of reactor plant problems. Only adiabatic flow is discussed in this report; see [3] for a discussion of flow through heaters.

The balance-of-plant model is explicitly coupled to the steam generator waterside model discussed in [2].

Discretization of the Balance of Plant

SASSYS-1 represents the balance of plant as a network of one-dimensional flow paths, or segments, which are joined at flow junctions called compressible volumes. Therefore, one-dimensional forms of the mass, momentum, and energy equations can be used to describe the system. The network is a discretization of the balance of plant using a non-uniform spatial mesh. The momentum equation is solved along each flow path, and the mass and energy equations are solved at each flow junction. Flow is assumed uniform throughout each flow path.

The balance of plant is a collection of several types of components. Components which primarily affect mass flow rate and pressure drop in a flow segment are best described through the momentum equation and are modelled as sections of flow segments; these sections are called flow elements. The cross-sectional area is constant throughout a given flow element. Element types include pipes, valves, check valves, and pumps. Flow segments then become strings of one or more flow elements.

Components which join two or more flow segments are best described through the mass and energy equations and are modelled as compressible volumes; these include inlet and outlet plena, piping junctions, and open heaters. Closed heaters must be described through a combination of flow elements and a compressible volume [3].

Analytical Equations

The general analytical equations are

$$\text{mass:}\quad \frac{\partial \rho}{\partial t} = -\left(\underline{\nabla} \cdot \rho \underline{u}\right) \; ; \tag{1}$$

$$\text{momentum:}\quad \frac{\partial}{\partial t}\left(\rho \underline{u}\right) = -\left[\underline{\nabla} \cdot \rho \underline{u}\underline{u}\right] - \underline{\nabla} P - \left[\underline{\nabla} \cdot \underline{\tau}\right] + \rho \underline{g} \; ; \tag{2}$$

$$\text{energy:}\quad \frac{\partial}{\partial t}\left(\rho \hat{E}\right) = -\left(\underline{\nabla} \cdot \rho \hat{E} \underline{u}\right) - \left(\underline{\nabla} \cdot \underline{q}\right) - \left[\underline{\nabla} \cdot P \underline{u}\right] - \left(\underline{\nabla} \cdot \left[\underline{\tau} \cdot \underline{u}\right]\right) \, . \tag{3}$$

These can be simplified by making the following assumptions:

1) One-dimensional flow,
2) Neglect the work done by viscous forces on a compressible volume (this is the $\underline{\nabla} \cdot \left[\underline{\tau} \cdot \underline{u}\right]$ term in Eq. 3),
3) Neglect kinetic energy and gravitation energy,
4) The viscous term in the momentum equation can be expressed as $\frac{F \rho u |u|}{2}$.

In addition, if the internal energy is expressed in terms of the enthalpy, the mass, momentum, and energy equations take the simpler forms

$$\text{mass:}\quad \frac{\partial \rho}{\partial t} = -\frac{\partial \rho u}{\partial z} \; ; \tag{4}$$

$$\text{momentum:}\quad \frac{\partial \rho u}{\partial t} = -\frac{\partial \rho u^2}{\partial z} - \frac{\partial P}{\partial z} - F\,\frac{\rho u |u|}{2} + \rho g \cos\theta \; ; \tag{5}$$

$$\text{energy:}\quad \frac{\partial \rho h}{\partial t} = -\frac{\partial \rho h u}{\partial z} - \frac{\partial q}{\partial z} + \frac{\partial P}{\partial t} \, . \tag{6}$$

The system is closed by using an equation of state of the form

$$\frac{\partial v}{\partial t} = \left(\frac{\partial v}{\partial P}\right)_h \frac{\partial P}{\partial t} + \left(\frac{\partial v}{\partial h}\right)_P \frac{\partial h}{\partial t} . \tag{7}$$

Discretized Equations

The analytical forms of the mass, momentum, and energy equations and the equation of state now need to be discretized over the compressible volumes and flow elements of the balance-of-plant nodalization. The result of the discretization will be a set of fully implicit equations which can be solved simultaneously for the changes in pressure, flow, and enthalpy in a timestep. All other quantities (e.g., densities, heat sources) will be computed explicitly.

The first step is to use the momentum equation to express the change over a timestep in the mass flow rate in each segment as a function of the changes in the segment endpoint pressures. Next, the mass and energy equations and the equation of state can be combined to express the change in pressure within a compressible volume as a function of the changes in the flows of all segments which are attached to the volume. If these two sets of equations are combined by eliminating the change in flow, the resulting matrix equation can be solved for the change in pressure in each compressible volume. The changes in flow, enthalpy, and all explicit variables can then be determined.

Discretization of the Momentum Equation

The momentum equation is discretized segment by segment. In order to represent momentum transport correctly through a segment made up of more than one flow element, the momentum equation must be integrated along the length of the segment, giving

$$\sum_k \frac{L_k}{A_k} \frac{\partial w}{\partial t} = P_{in} - P_{out} - \sum_k \frac{w^2}{A_k^2} \left(\frac{1}{\rho_{O_k}} - \frac{1}{\rho_{I_k}}\right) - \sum_k F(k) \frac{w|w|}{2\bar{\rho}_k A_k^2}$$

$$+ \bar{\rho} g \left(z_{out} - z_{in}\right) , \tag{8}$$

where the summation is over all elements in a segment. The term F is a coefficient which accounts for pressure losses due to friction, bends, area changes, and orifices or baffles. The form of Eq. 8 is valid for all element types except pumps; the convective and viscous terms have a different form for pumps, and Eq. 8 is modified accordingly when a flow segment contains a pump.

If the momentum equation in the form of Eq. 8 is now differenced over a timestep $\Delta t = t^{n+1} - t^n$ and then linearized and rearranged, the result is an expression for the change in segment flow, Δw, as a function of ΔP_{in} and ΔP_{out}, the changes in pressure at the segment inlet and outlet, respectively:

$$\Delta w = \{a_1 + a_2 + \Delta t\ (\Delta P_{in} - \Delta P_{out})\}/(a_0 - a_3)\ . \tag{9}$$

where

$$a_0 = \sum_k \frac{L_k}{A_k}\ , \tag{10}$$

$$a_1 = \Delta t[P_{in}(t) - P_{out}(t)] + \sum_k \Delta a_1(k)\ ,$$

$$\Delta a_1(k) = -\Delta t \sum_k \{\frac{w^2}{A_k^2}(\frac{1}{\rho_{0_k}} - \frac{1}{\rho_{I_k}}) - F(k)\ \frac{w|w|}{2\bar{\rho}_k A_k^2} + \bar{\rho}_k g \Delta z_k\}\ , \tag{11}$$

$$a_2 = -\sum_k (\Delta t)^2 \frac{\partial}{\partial t}(\Delta a_1(k))\ , \tag{12}$$

$$a_3 = \sum_k \frac{\partial}{\partial w}(\Delta a_1(k))\ . \tag{13}$$

Further Details on Element Component Models

The discretized momentum equation, Eq. 9, models one-dimensional flow along a flow path. This flow path model serves as the basis for the models for pipes, valves, and check valves. What distinguishes these three component models is the way in which the coefficient F is varied with time. The only contributor to F which may vary with time is the size of a flow orifice, and so any change in F is due to changes in the orifice coefficient portion of F.

The model of a pipe assumes that the orifice coefficient calculated in the steady state is valid throughout the transient. Therefore, the pipe model keeps the orifice coefficient constant at all times. The valve models, on the other hand, simulate the opening and closing of the valve by varying the orifice coefficient. The pressure drop across a valve is related to the flow through the valve by the equation

$$\Delta P = \frac{w|w|}{\rho C^2 \phi^2(y)} . \tag{14}$$

The functional relationship between the valve characteristic ϕ and the stem position y depends on the valve design and is input by the user through tables. The valve is opened or closed by varying the stem position; the user has the option of adjusting y directly or through the harmonic equation

$$m \frac{d^2 y}{dt^2} + B \frac{dy}{dt} + ky = F_1(t) , \tag{15}$$

so that y is controlled by the *driver function* $F_1(t)$, which is user-input. The orifice coefficient G_2 can then be expressed in terms of the valve calibration constant and characteristic as

$$G_2 = 2 \left(\frac{A}{C\phi}\right)^2 . \tag{16}$$

Thus, the valve aperture changes by recomputing the orifice coefficient each timestep, and the value of the coefficient is controlled by the stem position.

The check valve model works much the same as the standard valve model. However, there are a few differences. A check valve is normally either completely open or completely closed, whereas a standard valve can operate partially closed. A check valve changes between open or closed when a user-specified flow or pressure drop criterion within the valve is met. The valve then changes state by adjusting the orifice coefficient to a user-input value within a span of time that is also user-input. The criteria for opening and closing the valve, as well as the length of time the valve takes to open or close, must be set so as to avoid creating numerical instabilities in the calculation.

Because both valve models simulate valve closure by setting the orifice coefficient to a very large value, the flow through a valve is never actually zero. However, if the orifice coefficient is set sufficiently large, the resulting flow through the valve will be negligible.

Modelling pumps must be approached in a different way from that used in modelling pipes, valves, and check valves, since the convective and viscous terms are no longer simple functions of the mass flow rate. The losses represented by these terms are instead described by a set of homologous pump curves. Two types of pump curves are available in the balance-of-plant model, one using polynomial fits and one using more complex functional forms. Both options are identical to the corresponding options used in SASSYS-1 for sodium pumps.

Discretization of the Mass and Energy Equations and the Equation of State

The derivation that follows assumes perfect mixing within a compressible volume. Volumes in which liquid and vapor are separated are discussed in [3].

The mass and energy equations and the equation of state are solved at each compressible volume. Because the fluid within each volume is assumed to be perfectly mixed, the enthalpy and pressure are uniform within a volume (neglecting the pressure variations due to gravitational head). Therefore, the energy equation, Eq. 6, can be discretized in space by writing a separate energy equation for each volume ℓ,

$$V_\ell \frac{\partial(\rho_\ell h_\ell)}{\partial t} = -V_\ell \nabla(\rho u h)_\ell + Q_\ell + V_\ell \frac{\partial P_\ell}{\partial t} . \tag{17}$$

Equation 17 can be rewritten in terms of flows and masses by using the simple relationship between mass and density, $\rho V = m$, and expressing the enthalpy convection term as the rate at which enthalpy is convected into the volume. If the resulting equation is then discretized over time and the advanced time terms are linearized and second-order terms are dropped, the energy equation becomes

$$\frac{\partial m_\ell}{\partial t} h_\ell^n + m_\ell^n \frac{\partial h_\ell}{\partial t} = \sum_j h_j^{n+1} w_j^{n+1} \operatorname{sgn}(j,\ell) + Q_\ell^n + V_\ell \frac{\partial P_\ell}{\partial t} , \qquad (18)$$

where the sum is over all segments which are attached to volume ℓ. This form of the energy equation is a linear function of the volume mass, enthalpy, and pressure and of the enthalpies and flows from the segments attached to the volume. The heat source Q is assumed to be treated explicitly, and the enthalpy convection term is evaluated at the advanced time.

The mass time derivative can be eliminated from the energy equation by using the mass conservation equation, Eq. 4, written in the form

$$\frac{\partial m_\ell}{\partial t} = \sum_j w_j \operatorname{sgn}(j,\ell) . \qquad (19)$$

Substituting into Eq. 18 gives

$$m_\ell^n \frac{\partial h_\ell}{\partial t} = - h_\ell^n \sum_j w_j^{n+1} \operatorname{sgn}(j,\ell) + \sum_j h_j^{n+1} w_j^{n+1} \operatorname{sgn}(j,\ell) + Q_\ell^n + V_\ell \frac{\partial P_\ell}{\partial t} . \qquad (20)$$

There is one difficulty with the form of the energy equation shown in Eq. 20 -- the enthalpies at the interfaces between the compressible volume and the attached flow segments are treated implicitly. For the range of problems for which this model has been developed, treating these enthalpies explicitly introduces only small errors at worst. Treating them implicitly results in a solution procedure which is unnecessarily complicated and cumbersome. Therefore, it is assumed that these enthalpies can be treated explicitly. Applying this assumption to Eq. 20 and also finite differencing the time derivatives and linearizing the advanced time flows produces an energy equation of the form

$$m_\ell^n \frac{\Delta h_\ell}{\Delta t} = - h_\ell^n \sum_j \left(w_j^n + \Delta w_j\right) \operatorname{sgn}(j,\ell) + \sum_j h_j^n \left(w_j^n + \Delta w_j\right) \operatorname{sgn}(j,\ell)$$

$$+ Q_\ell^n + V_\ell \frac{\Delta P_\ell}{\Delta t} . \qquad (21)$$

Equation 21 expresses the change in volume enthalpy as a function of the change in volume pressure and the changes in flow in the attached segments. If the change in enthalpy can be eliminated, the result will be the desired equation relating the change in volume pressure to the changes in the segment flows.

The change in volume enthalpy can be eliminated from Eq. 21 by using the equation of state, Eq. 7 above, rewritten in finite differenced form as

$$- \frac{\nu_\ell^2}{V_\ell} \sum_j \left(w_j^n + \Delta w_j\right) \operatorname{sgn}(j,\ell) = \left(\frac{\partial \nu_\ell}{\partial P}\right)_h^n \frac{\Delta P_\ell}{\Delta t} + \left(\frac{\partial \nu_\ell}{\partial h}\right)_P^n \frac{\Delta h_\ell}{\Delta t} . \qquad (22)$$

Equation 22 is another expression for the change in volume enthalpy as a function of the change in volume pressure and the changes in the segment flows and so can be substituted into the energy equation, Eq. 21, to eliminate the change in volume enthalpy. By substituting density for mass and then rearranging, an equation for the change in pressure as a function of the changes in the segment flows is produced:

$$\Delta P_\ell = -\Delta t \ \{Q_\ell^n + \sum_j w_j^n \left[h_j^n - h_\ell^n + \nu_\ell^n \left(\frac{\partial h_\ell}{\partial \nu}\right)_P^n\right] \operatorname{sgn}(j,\ell)$$

$$+ \sum_j \Delta w_j \left[h_j^n - h_\ell^n + \nu_\ell^n \left(\frac{\partial h_\ell}{\partial \nu}\right)_P^n\right] \operatorname{sgn}(j,\ell)\}/$$

$$\left[V_\ell \left(1 + \frac{1}{\nu_\ell^n} \left(\frac{\partial \nu_\ell}{\partial P}\right)_h^n \left(\frac{\partial h_\ell}{\partial \nu}\right)_P^n\right)\right] . \qquad (23)$$

Solution Procedure

The segment flows can be eliminated from Eq. 23 by substituting the momentum equation of Eq. 9 to give

$$\Delta P_{\ell} = -\Delta t \{Q_{\ell}^{n} + \sum_{j} w_{j}^{n} [h_{j}^{n} - h_{\ell}^{n} + \nu_{\ell}^{n} (\frac{\partial h_{\ell}}{\partial \nu})_{P}^{n}] \operatorname{sgn}(j,\ell)$$

$$+ \sum_{j} \operatorname{sgn}(j,\ell) [h_{j}^{n} - h_{\ell}^{n} + \nu_{\ell}^{n} (\frac{\partial h_{\ell}}{\partial \nu})_{P}^{n}] \cdot [a_{1}(j) +$$

$$(a_{2}(j) + \Delta t [\Delta P_{in}(j) - \Delta P_{out}(j)])]/[a_{0}(j) - a_{3}(j)]\}/$$

$$[V_{\ell} \{1 + \frac{1}{\nu_{\ell}^{n}} (\frac{\partial \nu_{\ell}}{\partial P})_{h}^{n} (\frac{\partial h_{\ell}}{\partial \nu})_{P}^{n}\}] . \qquad (24)$$

If Eq. 24 is written for all L compressible volumes in a system, the result is a set of equations, each of the form

$$\sum_{j=1}^{L} c(I,J)\Delta P_{J} = d(I) \qquad (25)$$

These equations can be solved simultaneously for the pressure changes in all compressible volumes. Once the pressure changes are known, the change in mass flow rate in each segment can be computed from Eq. 9, and the changes in volume enthalpy can be calculated from Eq. 22. The explicit quantities, such as density and heat source, can then be updated.

Boundary Conditions

Often, it is desirable to perform calculations on only a portion of a power plant and to use boundary conditions to simulate the effect of the remainder of the plant. The balance-of-plant model has two types of boundary conditions available for this purpose: a flow boundary condition and a volume boundary condition. The flow boundary condition specifies flow and enthalpy as a function of time in a special flow segment attached to a compressible volume of the user's choice. The values of flow and enthalpy are either given by the user or are controlled by the plant control system. Multiple flow boundary conditions can be applied to the system. The volume boundary condition is modelled as a compressible volume in which pressure, enthalpy, and quality as a function of time are either user-input or controlled by the plant control system. The volume can be attached to one or more flow segments. Multiple volume boundary conditions can be designated in a network.

Coupling Between the Balance-of-Plant Model and the Steam Generator Model

It is important to note that the balance-of-plant model does not include the water side of the steam generator. Instead, the steam generator is modelled separately and is explicitly coupled to the balance-of-plant model. By so doing, the steam generator can be represented without use of the momentum equation [2], thereby significantly reducing the number of implicitly coupled momentum cells in the balance of plant. This is particularly important in a systems analysis code, where a fully-implicit solution scheme can easily result in unacceptably long running times. The omission of a momentum equation in the steam generator requires the assumptions that (1) the pressure drop across the steam generator can be neglected and that (2) the coupling between the steam generator outlet and the remainder of the balance of plant is not very strong; these assumptions are valid in a wide range of operational and accident situations in nuclear power plants. The coupling between the balance of plant and the steam generator is accomplished by having the steam generator provide a pressure at the subcooled/two-phase interface within the steam generator and a flow from the steam generator outlet into the outlet plenum, while the balance-of-plant model provides the subcooled region flow and the average steam generator pressure. This particular coupling requires some time averaging to stabilize the rate of change of the steam generator pressure, as well as limits on the rate of change of the subcooled zone flow; neither constraint affects the accuracy of the overall calculation.

Sample Problem

The plant configuration selected as a test case is diagrammed in Fig. 1. This is the high-pressure side of a balance-of-plant design for a liquid metal reactor plant. The problem encompasses that section of the plant which runs from the deaerator to the high-pressure turbine. The system contains sub-cooled liquid between the deaerator and the inlet to the once-through steam generator. The steam generator output is superheated steam which passes through a separator and finally, into the turbine. The deaerator and the turbine are both modelled as volume boundary conditions. The two heaters are represented by a simple heater model; testing of the advanced heater models is documented in [3].

When the test case is discretized, the result is the nodalization diagram of Fig. 2. The problem has now become a network of compressible volumes joined by flow segments, with the flow segments divided into one or more elements. The problem is sufficiently complex to provide a useful test of the coding, yet it is not so large as to be impractical to run many times, a necessity for a test case.

Several types of transients have been run with the sample problem of Fig. 2, among them a case in which one of the three feedwater pumps is tripped. The pump coasts down in 0.5 s. from full flow to the point at which the downstream check valve is tripped, and the check valve closes in 0.025 s., so flow is effectively zero in the branch containing the disabled pump in less than 1 s. This represents a moderately severe transient on the water side. The numerical algorithm handles this case with no difficulty, predicting a rapid decline in feedwater flow to about 78% of the original flow, as shown in Fig. 3, with a corresponding decline in steam generator pressure. The steam generator slowly boils dry, as shown in Fig. 4, with the sodium outlet temperature rising accordingly.

Other test cases which have been run include a base case null transient, a reactor trip transient, and a steam valve closure transient, in which a valve was added before the turbine on the steam side. All of these cases showed robust behavior by the algorithm similar to that displayed in the one-of-three pump trip problem.

All test cases were run on a Cray-XMP mainframe computer. In each problem, the cpu time for running SASSYS-1, including the steam generator and balance-of-plant models, was about 70% real time.

Conclusions

A viable balance-of-plant capability has been added to the SASSYS-1 code. The model has performed accurately in a variety of test cases while showing acceptably fast running times. Some additional test cases are planned for further model validation.

Nomenclature

a_0, a_1, a_2, a_3	momentum equation coefficients.
A	flow area (m^2).
B	valve damping coefficient (kg-m/s).
C	valve calibration constant (m^2).
$\hat{E}$	total energy (j/kg).
F	pressure loss coefficient.
F_1	valve driving function (kg-m/s^2).
g	gravitational constant (m/s^2).
G_2	orifice coefficient.
h	specific enthalpy (j/kg).
k	valve spring constant (kg/s^2).
L	element length (m).
m	mass (kg).
n	index on the discretized time variable.
P	pressure (pa).
q	heat flux (j/m^3).
Q	heat source (j).
$sgn(j,\ell)$	1 if the steady state flow in segment j is entering volume ℓ, -1 if the steady state flow in segment j is leaving volume ℓ.
t	time (s).
u	fluid velocity (m/s).
V	volume (m^3).
w	mass flow rate (kg/s).
z	spatial variable (m).
ν	specific volume (m^3/kg).
ϕ	valve characteristic.
ρ	density (kg/m^3).
θ	angle from vertical of a flow path
τ	viscous stress tensor (pa).

References

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3. J. Y. Ku, "SASSYS-1 Balance-of-Plant Component Models for an Integrated Plant Response," submitted to the 1989 ANS Winter Meeting.

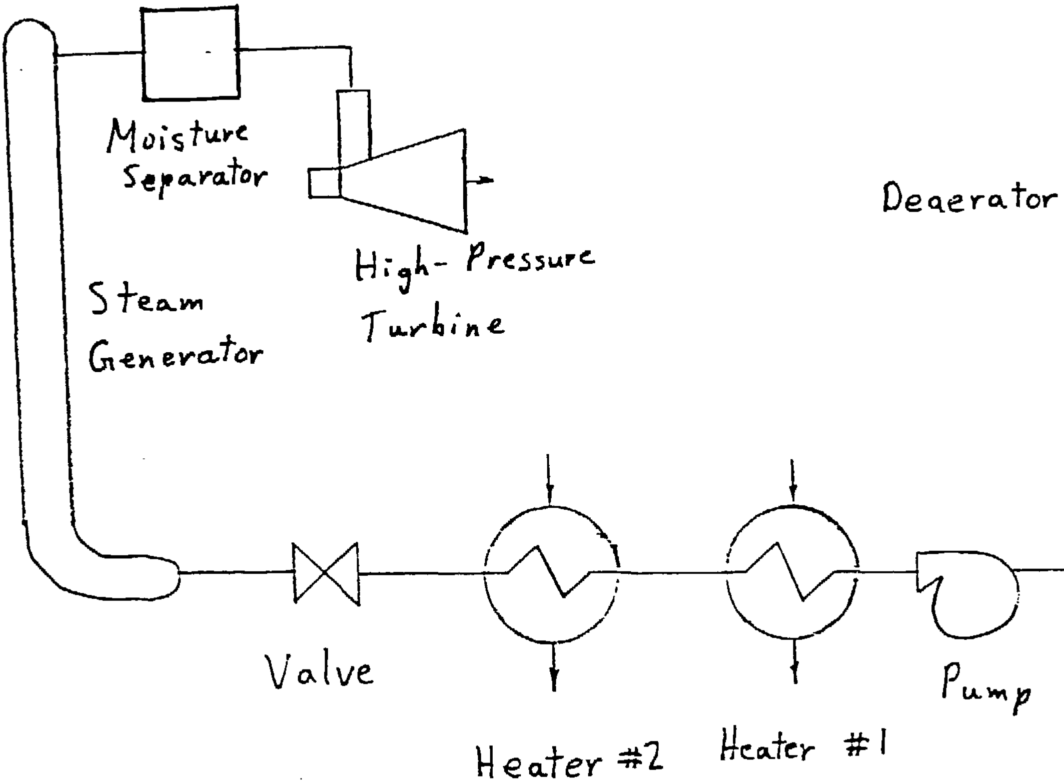

Fig. 1. Test Problem Configuration.

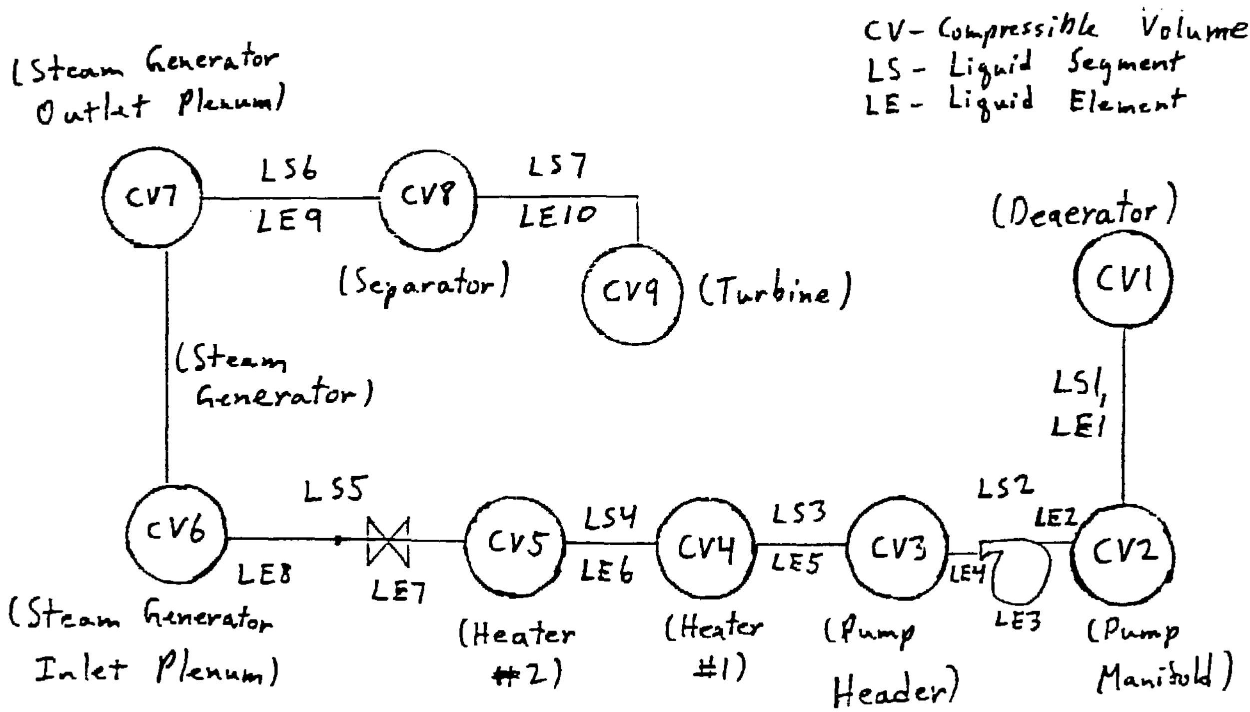

Fig. 2. Discretization of the Test Problem.

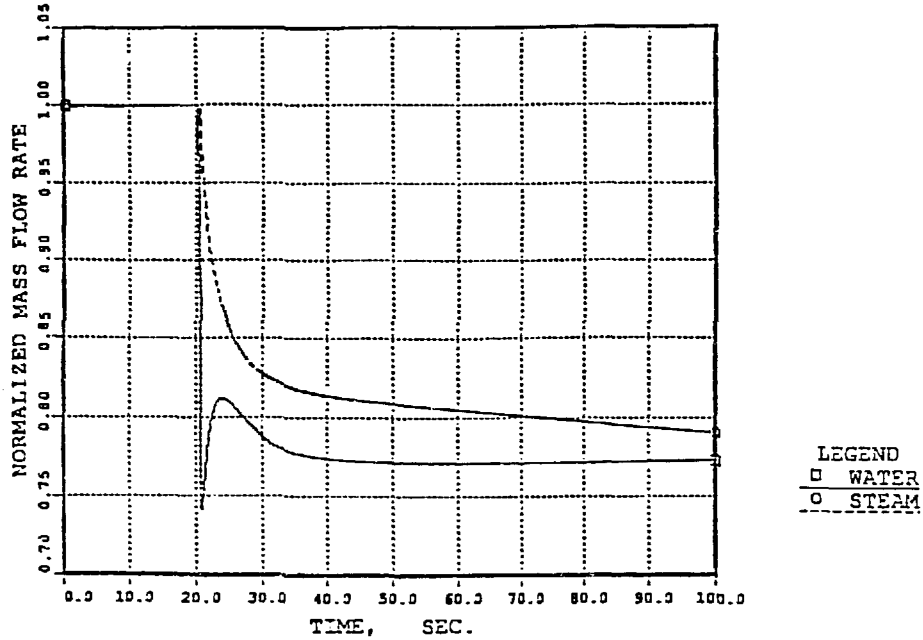

Fig. 3. Steam Generator Waterside Flows.

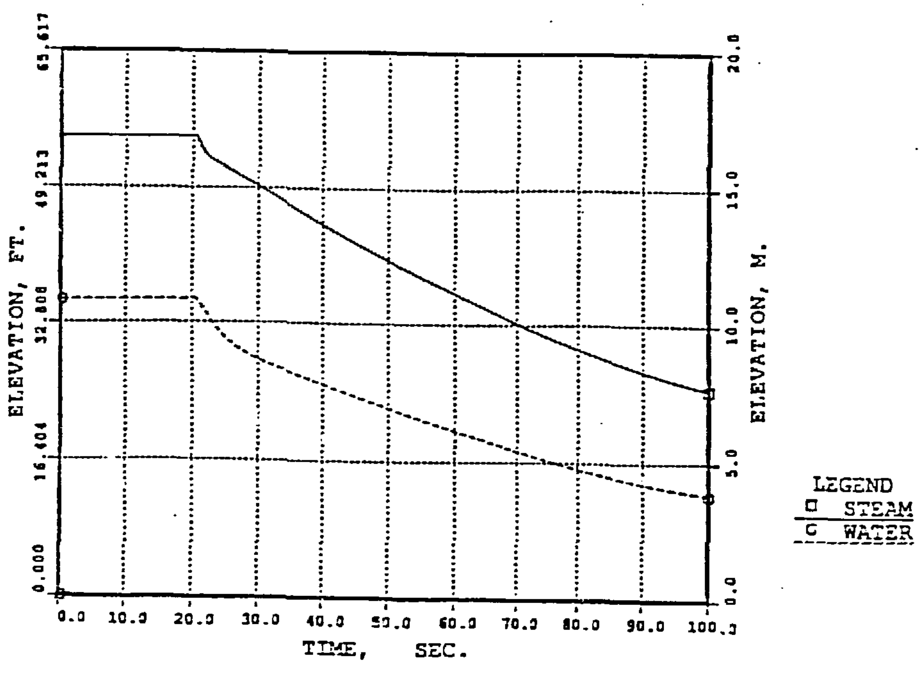

Fig. 4. Steam Generator Region Interfaces.