CHAPTER 18
PROCESS DESIGN II

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ABSTRACT

This chapter reviews the basic mass, momentum and energy equations used in typical computer codes for heat Transport System simulation. The equations are derived from first principles and the necessary approximations lead to empirical correlations. Closure is obtained by the equation of state. Steady and transient behaviour of the resulting equations, as they pertain to the Heat Transport System, are discussed.

18.1 BASIC EQUATIONS

Fluid flow and heat transfer, the phenomena of concern in the systems analysis of process systems, are governed by the conservation laws for mass, energy and momentum (reference 18.1), the constitutive laws or state equations plus numerous supporting empirical relationships.

In this section, the conservation equations, state equation and supporting empirical correlations are presented.

18.1.1 The Conservation Equations

The general integral formalism (see Appendix 18.1) involves surface integrals to depict surface fluxes. One school of thought, attended by most scientists, applied mathematicians and academics, since they usually deal with the local or microscopic approach, focuses on the conversion of the surface integrals to volume integrals using Gauss' Theorem. The volume integrals are then dropped giving the partial differential or microscopic form. This path works well when a detailed analysis is desired, such as subchannel flow in fuel fundles, moderator circulation in the calandria, etc.

The second school, which sees most favour among engineers, particularly in the chemical industry, evaluates the surface integrals as they stand without converting to volume integrals. This leads to a lumped or macroscopic approach useful for network analysis (i.e. SOPHT), distillation towers, etc.

There exist a very large number of possible derivations, each with its own advantages and disadvantages.

As more and more detail is picked up in each class of models, numerical means have to be used. In the limit of large numbers of nodes or mesh points, etc., both methods converge to the same solution.

Appendix 18.1 gives the derivation of the basic equations used in SOPHT, the system simulation code. Although the derived equations apply to a node-link network structures of the primary heat transport system, the form of the equations has general validity as will be seen in the following sections.

18.1.1.1 Conservation of Mass

Equation 6 of Appendix 18.1 states:

$$\frac{\partial M_i}{\partial t} = \sum_j \overline{\rho_j v_j} A_j \equiv \sum_j W_j \quad , \qquad 18\text{-}1$$

where the symbols as defined in the appendix are used. This states that the change in mass in a volume is the net flow into that volume.

In the steady state, of course:

$$\sum_j \overline{\rho_j V_j}\, A_j \equiv \sum_j W_j = 0 \qquad 18.2$$

For a simple circular flow loop, the mass flow rate at steady state is a constant at any point in the loop. Local area and density variations thus give rise to velocity variations around the loop.

Local velocity then is:

$$V = \frac{W}{\rho A} \qquad 18.3$$

Equation 4 of Appendix 18.1 can be easily converted to the microscopic form by using Gauss' Theorem

$$\iint_S \underset{\sim}{R}\cdot\underset{\sim}{n}\, ds = \iiint_\forall \nabla\cdot R\, d\forall$$

to give

$$\frac{\partial}{\partial t}\iiint_\forall \bar{\rho}\, d\forall = -\iiint_\forall \nabla\cdot\overline{\rho\underset{\sim}{v}}\, d\forall . \qquad 18.4$$

Dropping the volume integral:

$$\frac{\partial\bar{\rho}}{\partial t} + \nabla\cdot\overline{\rho\underset{\sim}{v}} = 0 \quad \text{or} \quad \frac{D\bar{\rho}}{Dt} = 0 . \qquad 18.5$$

This form is useful for modelling of detailed flow patterns such as in the calandria, vessels, steam generator and headers.

18.1.1.2 Conservation of Momentum

Equation 14 of Appendix 1 states:

$$\forall\frac{\partial\overline{\rho\underset{\sim}{v}}}{\partial t} = -\iint_S p\underset{\approx}{I}\cdot\underset{\sim}{n}\, ds + \iiint_\forall \left(\nabla\cdot\underset{\approx}{\tau} + \overline{\rho f}\right) d\forall \qquad 18.6$$

or

$$\forall\frac{\partial W}{\partial t} = -A_{OUT}P_{OUT} + A_{IN}P_{IN} - A\bar{\rho}\left(\frac{fL}{D} + k\right)\frac{V^2}{2g_c} + A\bar{\rho}\,\mathrm{Sin}(\Theta)\, g/g_c$$

Here, the general stress tensor is replaced by a widely accepted empirical form. This momentum equation states that the rate of change of momentum (or mass x acceleration or force) is equal to the driving pressure difference minus frictional losses plus gravity.

If circumstances requires, extra terms can be added. For instance, if a pump is present this can be considered to be an external force acting through the body force vector, $\underset{\sim}{f}$. It is usually represented as an additional head, ΔP_{pump}. Equation 18.6 would then become:

$$\forall\frac{\partial W}{\partial t} - A_{OUT}P_{OUT} + A_{IN}P_{IN} + A\,\Delta P_{PUMP} + \cdots \quad . \qquad 18.6a$$

The momentum flux terms ($A\rho V^2$) in Equation 2 of Appendix 18.1 could also be added if large area or property changes were present.

In the steady state, for a constant area pipe with no pump and no elevation change:

$$P_{IN} - P_{OUT} = \Delta P = \rho\left(\frac{fL}{D}+k\right)\frac{V^2}{2g_c} \qquad 18.6b$$
$$= \left(\frac{fL}{D}+k\right)W^2/2A^2\rho g_c \; .$$

For the microscopic form we start with equation 10 of Appendix 18.1:

$$\iiint_{\forall}\frac{\partial}{\partial t}(\rho\underset{\sim}{V})\,d\forall + \iint_s \rho\underset{\sim}{V}(\underset{\sim}{V}\cdot\underset{\sim}{n})\,ds = \iint_s \underset{\approx}{\sigma}\cdot\underset{\sim}{n}\,ds + \iiint_{\forall}\rho\underset{\sim}{f}\,d\forall . \qquad 18.7$$

Using Gauss's Theorem and dropping the volume integral:

$$\frac{\partial}{\partial t}(\overline{\rho\underset{\sim}{V}}) = \nabla\cdot\overline{\rho\underset{\sim}{V}\underset{\sim}{V}} = \nabla\cdot\underset{\approx}{\sigma} + \overline{\rho\underset{\sim}{f}} \; , \qquad 18.8$$

where $\underset{\approx}{\sigma} = -p\underset{\approx}{I} + \tau$.

Thus

$$\frac{D}{Dt}(\overline{\rho\underset{\sim}{V}}) = \frac{\partial}{\partial t}(\overline{\rho\underset{\sim}{V}}) + \nabla\cdot\overline{\rho\underset{\sim}{V}\underset{\sim}{V}} = \nabla P + \nabla\cdot\underset{\approx}{\tau} + \overline{\rho\underset{\sim}{f}} \; . \qquad 18.9$$

This is the form commonly seen in the literature, useful for microscopic modelling as per the mass conservation equation. As in Equation 18.6, $\nabla\cdot\underset{\approx}{\tau}$ is usually replaced by an empirical relation.

18.1.1.3 Conservation of Energy

Equation 38 of Appendix 18.1 states:

$$\forall\frac{\partial\overline{\rho h}}{\partial t} = \forall\frac{\partial P}{\partial t} + \sum_1\overline{W_{IN}h_{IN}} - \sum_1\overline{W_{OUT}h_{OUT}} + Q + \iiint_{\forall}\tau : \nabla\underset{\sim}{V}\,d\forall . \qquad 18\text{-}10$$

Verbally, this implies that the rate of change of enthalpy in a volume is equal to the work rate plus net flow of enthalpy into the volume plus heat work plus irreversible dissipation. In the steady state, neglecting the irreversible dissipation (turbulent heating), Equation 18.10 reduces to the familiar:

$$Q = \sum_1 W_{OUT}h_{OUT} - \sum_1 W_{IN}h_{IN} \; . \qquad 18.11$$

For a reactor or a boiler (one flow in, one flow out):

$$Q = W(h_{OUT} - h_{IN}) \equiv W C_p (T_{OUT} - T_{IN}) , \qquad 18.12$$

as we have seen in Chapter 1.

The mixture thermal energy equation of Appendix 18.1, Equation 25, can be used to generate the thermal conductance equation for solids. By setting fluid velocity to zero and converting surface integrals to volume integrals we get the microscopic form:

$$\frac{\partial}{\partial t}(\rho e) = -\nabla \cdot \underset{\sim}{q} + E , \qquad 18.13$$

where E is the internal energy generation rate term.

From thermodynamics, for solids, we have:

$$\frac{\partial (\rho e)}{\partial t} = \rho \frac{\partial e}{\partial t} = \rho C_v \frac{\partial T}{\partial t} , \qquad 18.14$$

and using Fourier's law for heat conduction:

$$\underset{\sim}{q} = -k \nabla T , \qquad 18.15$$

we have the classical form of the heat conduction equation:

$$\rho C_v \frac{\partial T}{\partial t} = \nabla \cdot k \nabla T + E \qquad 18.16$$

$$= k \nabla^2 T + E \qquad \text{for space independent k.}$$

This is useful for determining the temperature distributions in boiler tube walls, piping walls and reactor fuel pencils.

A special case of Equation 18.10 is obtained by expanding the term Q as per Equation 30 of Appendix 18.1.

$$-\iint_S \underset{\sim}{q} \cdot \underset{\sim}{n} \, ds + \iiint_{\forall} E \, d\forall \equiv Q . \qquad 18.17$$

Using Newton's Law of cooling for convection:

$$\underset{\sim}{q} \cdot \underset{\sim}{n} = h_N (T - T_s) , \qquad 18.18$$

where q.n = heat flux normal to surface, s
T = Temperature of fluid
T_s= Temperature of surface (wall)
and h_N = heat transfer coefficient.

Then Equation 18.10 becomes:

$$\forall \frac{\partial \overline{\rho h}}{\partial t} - \forall \frac{\partial P}{\partial t} \left(\equiv \forall \frac{\partial \overline{\rho e}}{\partial t} \simeq \forall \rho C_v \frac{\partial T}{\partial t} \right) \qquad 18.19$$

$$= \sum \overline{W_{IN} h_{IN}} - \sum \overline{W_{OUT} h_{OUT}} - A h_N (T - T_s) + \forall E$$

$$+ \iiint_{\forall} \underset{\approx}{\tau} : \nabla \underset{\sim}{V} \, d\forall .$$

which is useful for accounting for heat transfer between the fluid and the pipe or tube walls (ex. boiler heat transfer).

The heat transfer coefficient, h_N, is supplied through empirical relations. The turbulent heating term $\iiint_{\forall} \underset{\approx}{\tau} : \nabla \underset{\sim}{V}\, d\forall$ generally can be neglected.

18.1.2 The Equation of State (See Reference 18.2, for instance)

The equation of state determines the relationship between density and the thermodynamic pressure and temperature i.e. $\rho = \rho(p,T)$.

This relationship is extremely complicated for liquids, solids and during phase changes.

The equation of state for a <u>perfect gas</u> is

$$p = \rho RT \qquad 18.20$$

where p = pressure
R = $\mathcal{R}$/Molecular weight
T = absolute temperature
$\mathcal{R}$ = 8,314 J/kg-mol·K (universal gas constant)
ρ = density

Obviously R depends on the particular gas.

It can be shown that for an adiabatic frictionless process (isentropic) in a perfect gas we have the relationship

$$P \rho^{-k} = \text{constant} \qquad 18.21$$

where $k = C_p/C_v$.

From the conservation equations, we have three equations for each phase (mass, momentum and energy conservation) and four unknowns:

1) density, ρ or mass, $\forall\rho$

2) velocity, v, or mass flow rate, W, or momentum, mv

3) energy, e or enthalpy, $h = e + p/\rho$, or temperature, $T = f_n(e)$ or $f_n(h)$

4) pressure, P.

The fourth equation required for closure is the equation of state:

$$P = f_n(h,\rho) \text{ or } f_n(T,\rho) \quad \text{or} \quad \rho = f_n(P,T), \text{ etc} \qquad 18.22$$

Thermohynamic equilibrium is usually assumed, as in the following. For water, H_2O or D_2O, tables of properties give the required functional relationship. Often, a curve fit of the tables is used. This data is input to the computer codes and utilized in table lookup schemes or directly via the parametric curve fits.

To illustrate the process, consider a fixed volume, $\forall$, having a mass of H_2O, M, and an enthalpy, h. The volume is usually a specified geometric input. The mass flows and enthalphy have, let's assume, been calculated from the simultaneous solution of the conservation equations. The state equation gives the pressure given the mass and enthalpy for that volume, as follows.

The density, ρ, is an average for that volume, and is calculated from:

$$= M/\forall. \qquad 18.23$$

The pressure is guessed (as per the last time iteration, for instance) and the table lookup gives the associated density and specific enthalpies for liquid and vapour forms of H_2O. Since the average density is related to the specific densities by the relation:

$$\rho = \rho_l(1-\alpha) + \rho_v\alpha, \qquad 18.24$$

that is, the average density is a volume weighted sum of the specific densities. The void fraction, α, is readily calculated. Given a relationship between void fraction, α, and weight fraction (quality), x, we have the quality, x.

The mixture enthalpy, h, is given by

$$h = h_l\,(1-x) + h_v x, \qquad 18.25$$

that is, the average or mixture enthalpy is a mass weighted sum of the specific enthalpies. This mixture enthalpy is compared to the given enthalpy and the guess at the pressure is updated accordingly until convergence is reached.

To help guide the search for a compatible combination of P, h and ρ, partial derivatives, such as $\left.\frac{\partial P}{\partial h}\right|_\rho$, are, often used to calculate the next guess.

If the fluid is calculated to be subcooled, then no positive quality or void exists. A degree of freedom is gained, meaning that various combinations of ρ and h can give the same pressure. One procedure is to start with the density, use it in a table lookup to give the saturation pressure and the saturation enthalpy. Then a correction to the pressure is made to correct for the fact that the actual enthalpy is less than the saturation enthalpy. Thus:

$$P = P_{SAT_f}(\rho) + \left.\frac{\partial P}{\partial h}\right|_\rho \left[h - h_{SAT_f}(\rho)\right]. \qquad 18.26$$

A similar process is used for single phase steam, giving:

$$P = P_{SAT_g}(\rho) + \left.\frac{\partial P}{\partial h}\right|_{\rho}\left[h - h_{SAT_g}(\rho)\right]. \qquad 18.27$$

Internal energy, e (or sometimes, u), is related to enthalpy, h as follows:

$$h = e + p/\rho \qquad 18.28$$

The above calculation of pressure is sometimes performed using the variable, e or u, rather than h. The choice is one of convenience or personal preference only.

18.1.3 Empirical Correlations

As previously discussed, supporting relations are required to provide the necessary information for the conservation and state equations. The primary areas where support is needed are:

1) relationship between quality and void fractions, ie, slip velocities in two phase flow (to link the mass and energy conservation equations via the state equation).

2) the stress tensor, $\underset{\approx}{\tau}$ (effects of wall shear, turbulence, flow regime and fluid properties on momentum, or, in a word: friction).

3) heat transfer coefficients (to give the heat energy transfer for a given temperature distribution in heat exchangers, including steam generators and reactors).

4) thermodynamic properties for the equation of state.

5) flow regime maps to guide the selection of empirical correlations appropriate to the flow regime in question.

6) special component data for pumps, valves, steam drums, pressurizers, bleed or degasser condensors, etc.

7) critical heat flux information (this is not needed for the solution of the process equations but a measure of engineering limits is needed to guide the use of the solutions of the process equations as applied to process design.

The above list of correlations, large enough in its own right, is but a subset of the full list that would be required were it not for a number of key simplifying assumptions made in the derivation of the basic equations. The three major assumptions made for the primary heat transport system are:

1) one dimensional flow.

2) thermal equilibrium (except for the pressurizer under insurge).

3) one fluid model (i.e. mixture equations)

These are required because of state of the art limitations. References 18.3 to 18.14 are recommended for further reading.

18.2 Steady State Behaviour

18.2.1 Boundary Conditions

In the solution of an engineering problem represented by a system of equations, the number of equations must equal the number of unknowns for a unique solution to exist. Boundary conditions are applied to the system of equations to ensure this.

Precisely which boundary conditions are appropriate depends on what variables are specified and what variables are to be determined. For instance, in the case of liquid flow around a loop, the variables are: loop pressure drop, flow and hydraulic resistance. The system is determined by a single momentum equation and hence there should be only one unknown. In the case of designing the HTS, the flow is predetermined and the RIH to ROH pressure drop is fixed. If all the hydraulic resistances are fixed, then the system is overspecified; hence the resistance of the inlet feeders is left as a variable to enable a fine-tuning of the system to achieve steady state flow of the specified value. In actual fact, both the feeders and the pump impellers may be trimmed during commissioning of a nuclear station.

In the steady state, the power production of the reactor is fixed. The boiler pressure (hence enthalpy and temperature since the boiler drum is at saturation) is fixed since in reality it is controlled to a specified pressure. Since power transferred = flow x change in enthalpy and since the feedwater temperature is fixed (indirectly by a hold on the enthalpy of the daearator tank), the only free variable is the steam/feedwater flow. The feedwater valve is left as a variable to ensure that the system can supply the feedwater required and the deaerator tank pressure is held (as in reality) thus satisfying the momentum equation.

Since the primary and secondary system temperatures (enthalpy) and pressures are fixed, (both the ROH and the steam drum are at saturated conditions), the flows are fixed or calculated and the heat transfer is specified, then the heat transfer coefficient must be left variable in order to satisfy Newton's law of cooling by convection:

heat flux = heat transfer coefficient x temperature difference

This can be achieved by having a variable crud layer on the boiler tubes. Alternatively the ROH enthalpy could be left variable and the crud layer fixed. The balance of the plant is usually not modelled but the flows for the reheater steam supply and the drains return are known and are, hence, fixed. The pressure and enthalpy of the drains return is also known and fixed. The turbine pressure is held constant since it is considered a reservoir for steam dumping and the governor valve is usually left a variable.

Since the recirculation ratio in the boiler is defined by design, the downcomer flow is held while the downcomer link resistance is left to vary.

These boundary conditions applied to PHT simulation enable the steady state calculation to be performed. Subsequent transients based on the steady state results do not require the boundary conditions (with the exception of the deaerator tank, turbine and drains return) since any imbalance in the basic mass, energy and momentum equations manifests itself as the transient of mass, energy and momentum. Valve position is now controlled by the control systems. The controllers obtain the starting (or steady state) valve positions from the steady state results. Care must be exercised to ensure that the bias terms of the controllers are set up for the steady state positions.

Finally, when performing a steady state calculation, it should be noted that the pressure, temperature, enthalpy and flow estimates are needed as a starting point in generating a self-compatible steady state solution. The only process values which need to be specified exactly are those which are held fixed as boundary conditions.

18.3 Transient Behaviour

18.3.1 Introduction-Transient Examples

Consideration of transient bahaviour of the Primary Heat Transport System is essential since the system is composed of many interdependent items which must be modelled simultaneously for efficient design (reference 18.15). The increasing emphasis upon transient and safety related analyses dictate the use of a comprehensive model such as SOPHT, incorporating the many facets of a nuclear generating station.

It is instructive to illustrate transient behaviour by looking at a few transients using Darlington A as an example.

(a) A typical transient, say a reactor trip, starting from the steady state at 100% full power (F.P.) is caused by the rapid insertion of the shutdown control rods in the reactor core. The cause of the trip could be a real or spurious signal to the control system.

The resulting system transient is shown in Figure 18.1. The neutron power drops to decay heat levels within 2-4 seconds. The power to coolant drops more slowly due to the heat transport lag. The power drop causes a depressurization of the primary circuit and consequently a large flow from the surge tank. As the PHT system pressure drops, so does the temperature of the D_2O. This causes a boiler cooldown and depressurization as shown. Finally the governor valve closes down in an attempt to maintain the boiler pressure at the setpoint. This transient gives insight into the size of the pressurizer and pressurizer interconnect piping needed to keep the HT system pressure high enough to prevent the main pumps from cavitating.

(b) A turbine trip is initiated by a real or spurious signal, indicating an inability of the electrical generator, the electrical grid or the turbine to accept the steam power. The trip signal, in this case, closes the emergency stop valves and causes the transient shown in Figure 18.2. The trip signal automatically initiates a reactor stepback, ie, a reduction in power. The primary system pressure drops due to the power reduction while the secondary flow reduction causes an increase in boiler pressure. The surge tank provides an outsurge to relieve the pressure drop in the primary system while the boiler safety and steam discharge valves relieve the high boiler pressure. This case gives information on relief valve sizing for the secondary side as well as giving the consequences of the control philosophy used.

(c) The rapid cooldown of the boiler is initiated by tripping the reactor and opening the discharge valves. This results in a rapid depressurization, and hence cooldown of the boiler. The transient, as shown in Figure 18.3, is used in the stress analysis of the boiler, as are the other transients.

(d) An example of manoeuvring is given in Figure 18.4. Here, starting from an 80% full power steady state solution, the reactor is ramped up in power at the specified rate of 0.8%/sec as demanded via the demand power routine (DPR).

The resulting PHT pressure transient proved too much and the control scheme automatically initiated a reduction in reactor power or stepback. Such a result indicates that the rate of 0.8%/sec is too high for the system configuration studied.

(e) The final example, loss of Class IV power (the main electrical grid), represents an operational transient needed for the stress analysis of the steam generators. Figure 18.5 shows, as a function of time, two of the system parameters affecting the stress analysis. The loss of Class IV power occurs at time t = 0 and causes a loss of power to the main heat transport pumps, feedwater pumps, reheater drains pumps and turbine condenser cooling water pumps. This immediately initiates a trip signal to the turbine governor valves. The subsequent sequence of events is shown in the figure. The loss of the turbine causes a fast controlled power reduction stepback, to 60% full power. This signal clears at t = 2 seconds, but a high drum pressure causes a further slow reduction in power via a setback, and low primary flow finally trips the reactor. The use of the resulting transient in the stress analysis may lead to the conclusion that the transients are unacceptable. In that event, the action taken may be to resize the pressurizer interconnect lines.

18.3.2 Selection of Transients for Design Purposes

Generally, transients are used for a variety of reasons. Stress analysis requires the transient behaviour for all components for a prescribed set of events. These events were classified as:

A) normal,
B) upset,
C) emergency,
and
D) faulted,

in order of increasing severity and decreasing frequency of occurence.

Part of the general requirements involve protection against overpressure, covered by the Overpressure Protection Report.

For design purposes, there are a number of bounding transients that should be performed for component sizing and design checking purposes for the system in question. There are also those transients which cover systems interaction. PHT transients necessarily involve other major systems such as the PIC, secondary side, and station controls. Large simulation tools like SOPHT are specifically designed for such transients.

The following major simulations should be performed in roughly the order given to minimize the amount of design iteration necessary. Of course, many other simulations are also performed in the course of detailed design.

The following events are good events to consider for commissioning purposes and provide good code verification events.

18.3.2.1 Steady State - Initial Condition

Steady State calculations should be performed at various power levels and PHT conditions including:

1) 0% F.P. cold (100°C),
2) 0% F.P. hot (265°C),
3) 0% F.P. to 100% F.P.

These cases should be analyzed to check on the reasonableness of the main features such as:

1) pressure drop distribution around the circuit,
2) quality and void fraction distribution,
3) temperature distribution, mainly in the boiler and fuel channel,
4) gross flow and pump head,
5) feeder flows, noting redistribution as a function of power,

and

6) heat balance.

The regimes of the empirical correlations used should be checked against the calculated conditions to ensure that the correlations are being properly used. Further, the operating conditions of all components should be checked to ensure that they fall within the limits of engineering feasibility (this is a double check since a cursory check is made at the optimization stage.

One specific use of the steady state calculations is for pressurizer sizing. The mass inventory difference between the 0% F.P. hot condition and the 100% F.P. condition gives the total swell that the pressurizer must accommodate in the Bruce, Darlington and 600 MW designs. The 950 MW pressurizer is designed to accommodate the swell from 0% F.P. cold to 100% F.P. to enhance the availability of water to the PHTS in the event of a LOCA or loss of feed pumps. The mass inventory as a function of power sets the level control setpoint for the pressurizer level controller (see Figure 18.6).

Also from these steady state runs, the system void at 100% full power can be estimated. During a transient, such as a reactor trip, in which the void collapses, the maximum pressure drop can be estimated by a hand calculation using the equation of state. Given the PHTS mass as liquid (no void) and the PHTS volume, the pressure for the no void condition can be directly estimated. This gives an envelope for the PHTS pressure. This can be compared to the reactor trip transient to qualitatively verify the trip transient and to quantitatively estimate the effectiveness of the surge line and pressurizer in minimizing pressure transients. This minimum pressure is of critical importance in avoiding PHT pump cavitation.

Other special cases can be simulated. For instance, the PHT relief valves can be failed open or the D_2O feed valves can be failed open, to verify the capacities and protection capability for failures.

Pump motor failure is covered in Section 18.3.2.5.

18.3.2.2 Zero Change Transient

Before a transient can be properly simulated, a steady state or a reasonable starting condition must be established. The setpoints of the station controllers should be set so that the station is simulated to operate at the prescribed condition, 100% F.P. for instance, and that no transient results if no perturbation or event is initiated. This amounts to establishing a base case for the subsequent transient. Having achieved a zero change transient, the designer is assured that the subsequently induced transient is due to the event and not due to improper control simulation.

18.3.2.3 Reactor Trip

The reactor trip is a major transient that tests the major behaviour of the system. Trip logic and trip controllers, reactivity insertion rate and power drop rate are simulated by SOPHT and this transient permits these items to be checked by comparison to the design specifications from the control and physics groups. The relationship of the reactor trip to void collapse and pressurizer size was described in Section 18.3.2.1. The surge line (connecting the pressurizer to the main PHT piping) size is a major parameter affecting the minimum PHT pressure during a reactor trip. Also important is the steam volume above the liquid in the pressurizer and the total pressurizer volume. Adjusting these parameters to meet NPSH requirements of the PHT pumps sets the design for these components and provides the basis for subsequent transient analysis. The change in secondary side conditions (level, pressure, flows, etc.) provides an adequacy check on drum size, level setpoint, riser volume and general secondary side behaviour.

18.3.2.4 Power Manoeuvring

Manoeuvring up in reactor power is achieved by 1) demanding steam flow and allowing the boiler pressure controller (BPC) to increase reactor power to suit or 2) manually demanding a reactor power increase at a specified rate. The BPC method is a good check on controller routine. The manual manoeuvre is the most telling from a PHT point of view.

During the manual manoeuvre, the HT pressure will rise and steam bleed from the pressurizer to the degasser condenser (600 MW) or bleed condenser (Bruce and Darlington) will be required. The maximum manoeuvring rate sets a minimum requirement for the steam bleed valve capacity, tests the pressurizer level behaviour (liquid should not pass through the steam bleed valves) and checks the bleed condenser/degasser condenser capacity to accept the steam bleed. Feed and bleed action can also be compared to design expectations. Overpressure protection is another aspect to monitor. The manual manoeuvre at the maximum rate (as set by the design requirements) should not cause the PHT relief valves to open or induce a reactor trip, stepback or setback.

Manoeuvring down is enveloped by the maximum rate setback simulation. PHT performance is not an issue since a more demanding power reduction has already been considered (the reactor trip). The setback control action should be checked, however. Also, for a stepback (reduction to a lower power, usually 60% of initial power), the simulated controller should be checked to ensure proper operation. It should successfully reduce the power to a new, lower power without causing a trip. This is further discussed in Sections 18.3.2.5 and 6.

18.3.2.5 HT Pump Trip

The CANDU stations are usually required to operate with less than the full complement of pumps at a reduced power above the poison prevent power (the reactor power below which fission product poisons cause the chain reaction in the fuel to go sub-critical). Simulation of this event with a freewheeling pump rotor at, say, 70% F.P. and a braked pump rotor at, say, 60% F.P. sets the pump head-flow characteristic required. Sufficient flow must be provided to ensure protection against fuel dryout under these steady-state conditions.

The transient for this event is also important. The pump/motor set inertia is sized by this transient. The inertia should be sufficient to provide a pump flow rundown curve that gives adequate cooling to the fuel as the power runs down (due to the stepback signal which is a direct result of a pump trip). Also, the inertia should be sufficiently large to avoid a reactor trip on high pressure or low flow during the event. Overpressure is, again, a consideration. Finally, the control logic is checked by this event.

18.3.2.6 Turbine Trip

The turbine trip provides a good check on the control logic for governor valves, boiler pressure control (BPC), and boiler level control (BLC). The change in steam generator drum level provides an adequacy check on the drum size, level setpoint, secondary side riser volume and general secondary side behaviour. The subsequent reactor stepback is also tested and the turbine trip should not lead to a reactor trip. The atmospheric steam discharge valves (ASDV's) and the condenser steam discharge valves (CSDV's) are sized and their control algorithms are determined partially by this event. These valves should act to relieve the steam generator pressure rise so as to prevent the safety relief valves (SRV's) from opening since resetting the SRV's is difficult and poses an unwarranted requirement for the SRV's.

After the transient event, the CSDV's and condenser should handle the steam flow at about 60% F.P. This enables continued reactor operation at a power above the poison out level so that the station can quickly recover from spurious and short duration turbine trips.

18.3.2.7 Class IV Power Failure

Both a total and a partial loss of Class IV power further tests the design adequacy in handling an event which leads to high PHT pressure, pump failures, a reactor stepback and subsequent trip. Relief valve setpoints and relief valve sizing also play an important role for this event. Design changes are made as required, depending on the resulting transient. Usually, however, adequate protection has already been designed in by this stage as a result of the previous transients analyzed.

18.3.2.8 Crash or Rapid Cooldown

This transient poses a severe requirement for the stress analysis. Design requirements set the boiler relief capacity (usually <100% of nominal steam flow). The rapid depressurization and cooldown of the steam generator along with the simultaneous reactor trip (imposed as per the definition of the event) causes a rapid void collapse in both the secondary and primary sides. Usually, little can be done to change the design to lessen the severity of the transient except to minimize the postulated frequency of occurence and design the emergency water system (EWS) to lessen the thermal shock of makeup water injection. As with all the other events, this is a good transient to compare with actual station performance, although deliberate testing is not recommended because of the high stresses involved. It is also a good event to simulate since it tests the code's capability to handle low boiler levels.

REFERENCES

18.1 Transport Phenomena, R.B. Bird, W.E. Stewart and E.N. Lightfoot, John Wiley and sons, Inc., New York, 1960.

18.2 Fundamentals of Fluid Mechanics, Volume 1, Preliminary Edition, J. Vlachopoulos, McMaster University, 1980.

18.3 Two-Phase Flow and Heat Transfer in the Power and Process Industries, A.E. Bergles et al, Hemisphere Publishing, 1981.

18.4 Boiling Heat Transfer and Two-Phase Flow, L.S. Tong, John Wiley and Sons, Inc., 1965.

18.5 Thermohydraulics of Two-Phase Systems for Industrial Design and Nuclear Engineering, J.M. Delhaye et al, Hemisphere Publishing, 1981.

18.6 Two-Phase Flows and Heat Transfer with Application to Nuclear Reactor Design Problems, Ginoux, Hemisphere Publishing, 1981.

18.7 Two-Phase Hydrodynamics: Models and Mechanisms, S. Banerjee, Keynote Paper, ANS/ASME Topical Meeting, Nuclear Reactor Thermal-Hydraulics, Saratoga, New York - October, 1980.

18.8 Convective Boiling and Condensation, J.G. Collier, McGraw-Hill, 1972.

18.9 The Thermal-Hydraulics of a Boiling Water Nuclear Reactor, R.T. Lahey, Jr., and F.J. Moody, ANS/AEC Monograph Series on Nuclear Science and Technology, ANS, 1977.

18.10 Transport Processes in Boiling and Two-Phase Systems, Y. Hsu and R. Graham, Hemisphere Publishing, 1976.

18.11 Flow of Fluids Through Valves, Fittings and Pipe, Crane, 410-C (Engineering Units), 410-m (metric).

18.12 Piping Design and Engineering, ITT Grinnell Corp., Providence, Rhode Island, 1973.

18.13 Handbook of Hydraulic Resistance - Coefficients of Local Resistance and of Friction, National Science Foundation, Washington, D.C., 1960. (Translation from Idel'chik).

18.14 Internal Flow, A Guide to Losses in Pipe and Duct Systems, D.S. Millar, British Hydromechanics Reasearch Association, 1971.

18.15 The Thermal Hydraulic Code, SOPHT, as a Design Tool, W.J. Garland, Nuclear Systems Department, Report No. 76071, Ontario Hydro, June 1976. Also presented at the American Nuclear Society Meeting, Toronto, Ontario, June 1976.

APPENDIX 18.1

Derivation of Basic Equations in SOPHT

Nodal Structure

SOPHT represents a hydraulic network of pipes by nodes joined by links. Mass and energy changes occur at the nodes. Momentum changes occur in the lings. Thus the network is treated on a macroscopic scale requiring an integral approach to the fundamental equations. Flow details in pipes are not considered. That is, diffusion, dispersion, advection, flow regimes, flow profiles, etc., are not fundamentally accounted for but are covered by empirical correlations and averaging techniques.

Equation of Continuity

In the integral or macroscopic sense, what goes in must go out, unless it stays there or is generated or lost by phase change. Symbolically:

$$\frac{\partial}{\partial t}\iiint_{\forall}\gamma_k\rho_k\,d\forall = -\iint_s \gamma_k\rho_k\,\underset{\sim}{V}_k\cdot\underset{\sim}{n}\,ds + \iiint_{\forall}\Gamma_k\,d\forall\,, \qquad (1)$$

where t = time,
$\forall$ = volume,
ρ_k = density of phase k (1=liquid, 2=vapour),
$\iiint_{\forall}$ = volume integral,
$\iint_s$ = closed surface integral,
$\underset{\sim}{n}$ = normal vector to the surface of the volume under consideration,
$\underset{\sim}{V}$ = velocity of the fluid of density ρ thru the surface, s,
Γ_k = generation function for phase, k
γ_k = fraction of phase, k, in volume, $\forall$.

The individual densities are related as follows:

$$\begin{aligned}\bar{\rho} &= \gamma_1\rho_1 + \gamma_2\rho_2\\ &= (1-\alpha)\rho_1 + \alpha\rho_2\,,\end{aligned} \qquad (2)$$

where $\tilde{\rho}$ = the average density
and α = void fraction.

But adding both phases together, Equation (1) becomes:

$$\begin{aligned}\frac{\partial}{\partial t}\iiint_{\forall}\left[(1-\alpha)\rho_1+\alpha\rho_2\right]d\forall &= -\iint_s(1-\alpha)\rho_1\underset{\sim}{V}_1\cdot\underset{\sim}{n}\,ds + \iiint_{\forall}\Gamma_1\,d\forall\\ &\quad -\iint_s\alpha\rho_2\underset{\sim}{V}_2\cdot\underset{\sim}{n}\,ds + \iiint_{\forall}\Gamma_2\,d\forall\,.\end{aligned} \qquad (3)$$

For our case, $\Gamma_1 = -\Gamma_2$ (liquid boils or vapour condenses).

$$\therefore \quad \frac{\partial}{\partial t}\iiint_{\forall} \bar{\rho}\, d\forall = -\iint_S \overline{\rho \underset{\sim}{V}} \cdot \underset{\sim}{n}\, ds \,. \tag{4}$$

Now, $\iiint_{\forall}\bar{\rho}\, d\forall$ is the mass, M_i, in the volume, $\forall$, where i represents the node. Also, for our case, the surface integral can be written as surface integrals over the individual paths into and out of the volume or node. That is,

$$\iint_S \overline{\rho \underset{\sim}{V}} \cdot \underset{\sim}{n}\, ds = \sum_j \overline{\rho_j V_j} A_j \,, \tag{5}$$

where j represents inflow and outflow links with V_j 0 for inflow and <0 for outflow. Inherent in Equation (5) is the assumption that the integral, $\iint_S \overline{\rho \underset{\sim}{V}} \cdot \underset{\sim}{n}\, ds$ can be replaced by the simple product $\overline{\rho_j V_j} A_j$. This implies a uniform velocity and density profile across the face of the link (or pipe).

Thus we now have:

$$\frac{\partial M_i}{\partial t} = \sum_j \overline{\rho_j V_j} A_j \equiv \sum W_j \,, \tag{6}$$

where W_j is the mass flowrate.

This is the precise representation in SOPHT.

So far, then the assumptions are:

i) nodalization
ii) uniform velocity and density across the cross-section of a pipe.

These assumptions have far reaching ramifications that are not immediately obvious. No flow detail is considered as the fluid moves along a pipe. No diffusion, dispersion, advection, flow profiles or flow regimes are permitted. This is not too crude an approximation for the calculation of pressure drops and flow rates but for modelling the propagation of disturbances, this approach is inadequate as it stands.

Conservation of Momentum

In the integral sense, the rate of change of momentum is equal to the forces acting on the fluid. Thus

$$\frac{D}{Dt}\iiint_{\forall} \gamma_k \rho_k \underset{\sim}{V_k}\, d\forall = \iint_S \underset{\approx}{\sigma_k} \cdot \underset{\sim}{n}\, ds + \iiint_{\forall} \gamma_k \rho_k \underset{\sim}{f_k}\, d\forall + \iiint_{\forall} \underset{\sim}{m_k}\, d\forall \,, \tag{7}$$

where $\underset{\approx}{\sigma}$ is the stress tensor (ie., short range or surface effects
including pressure, viscosity, etc.)
$\underset{\sim}{f}$ is the long range or body force (ie., gravity)
and $\underset{\sim}{\mathfrak{m}}$ is the momentum interchange function accounting for phase
change effects.

Using Reynold's Transport theorem, which states:

$$\frac{D}{Dt}\iiint_{\forall} \Psi \, d\forall = \iiint_{\forall} \frac{\partial \Psi}{\partial t} \, d\forall + \iint_S \Psi \underset{\sim}{V} \cdot \underset{\sim}{n} \, ds \,, \tag{8}$$

equation (7) becomes:

$$\iiint_{\forall} \frac{\partial}{\partial t}(\gamma_k \rho_k \underset{\sim}{V}_k) \, d\forall + \iint_S (\gamma_k \rho_k \underset{\sim}{V}_k)(\underset{\sim}{V}_k \cdot \underset{\sim}{n}) \, ds = \iint_S \underset{\approx}{\sigma}_k \cdot \underset{\sim}{n} \, ds + \iiint_{\forall} \gamma_k \rho_k \underset{\sim}{f}_k + \iiint_{\forall} \underset{\sim}{\mathfrak{m}}_k \, d\forall \,. \tag{9}$$

Adding both phases together as per the mass balance equation, we find:

$$\iiint_{\forall} \frac{\partial}{\partial t}(\overline{\rho \underset{\sim}{V}}) \, d\forall + \iint_S \overline{\rho \underset{\sim}{V}(\underset{\sim}{V} \cdot \underset{\sim}{n})} \, ds = \iint_S \underset{\approx}{\sigma} \cdot \underset{\sim}{n} \, ds + \iiint_{\forall} \overline{\rho \underset{\sim}{f}} \, d\forall \,. \tag{10}$$

If the surface integral for the advective term is performed over the inlet and outlet areas of the pipe (link) in question, then:

$$\iint_S \overline{\rho \underset{\sim}{V}(\underset{\sim}{V} \cdot \underset{\sim}{n})} \, ds = \iint_{A_{IN}} \overline{\rho \underset{\sim}{V}(\underset{\sim}{V} \cdot \underset{\sim}{n})} \, ds + \iint_{A_{OUT}} \overline{\rho \underset{\sim}{V}(\underset{\sim}{V} \cdot \underset{\sim}{n})} \, ds \,, \tag{11}$$

where A_{IN} is the flow inlet area and A_{OUT} is the flow outlet area. If we assume the properties are constant over the areas, then:

$$\forall \frac{\partial \overline{\rho \underset{\sim}{V}}}{\partial t} - A_{IN} \overline{\rho_{IN} V_{IN} \underset{\sim}{V}_{IN}} + A_{OUT} \overline{\rho_{OUT} V_{OUT} \underset{\sim}{V}_{OUT}} = \iint_S \underset{\approx}{\sigma} \cdot \underset{\sim}{n} \, ds + \iiint_{\forall} \overline{\rho \underset{\sim}{f}} \, d\forall \,. \tag{12}$$

If we assume the properties, V, ρ & A are constant along the length of the pipe, then the second and third terms cancel.

The stress tensor, $\underset{\approx}{\sigma}$, can be split into the normal and shear components.

$$\underset{\approx}{\sigma} = -p \underset{\approx}{I} + \underset{\approx}{\tau} \,, \tag{13}$$

where p is the pressure, $\underset{\approx}{I}$ is the unity tensor and $\underset{\approx}{\tau}$ is the shear stress tensor.

Using Gauss' Theorem ($\iint_s \underset{\sim}{A}\cdot\underset{\sim}{n}\,ds = \iiint_\forall \nabla\cdot\underset{\sim}{A}\,d\forall$), Equation (11) becomes:

$$\forall\frac{\partial\overline{\rho\underset{\sim}{V}}}{\partial t} = -\iint_s p\underset{\approx}{I}\cdot\underset{\sim}{n}\,ds + \iiint_\forall(\nabla\cdot\underset{\approx}{\tau}+\overline{\rho\underset{\sim}{f}})\,d\forall \tag{14}$$

$$= -A_{OUT}\underset{\sim}{P}_{OUT} + A_{IN}\underset{\sim}{P}_{IN} - \frac{\forall\bar{\rho}}{L}\left(\frac{fL}{D}+k\right)\frac{\underset{\sim}{V}|\underset{\sim}{V}|}{2g} - LA\bar{\rho}\sin(\Theta)\,g/g_c\,,$$

where g_c is the gravitational constant and where $\nabla\cdot\underset{\approx}{\tau}+\overline{\rho\underset{\sim}{f}}$ are lumped together and avaluated by empirical correlations (the standard friction factor) plus an elevation change term (Θ is the angle w.r.t. the horizontal).

Defining the mass flow rate as $W = \overline{\rho V}A$, and L as the pipe length Equation (14) becomes:

$$\frac{\partial W}{\partial t} = \frac{A}{L}\left[(P_{IN}-P_{OUT}) - \left(\frac{fL}{D}+k\right)\frac{W^2}{2g\rho A^2}\right] + A\rho g/g_c\sin\Theta\,, \tag{15}$$

which is the form used in SOPHT.

The assumptions made for the mixture momentum equation are thus similar to those mode for the mixture mass equation and the same comments apply. One cannot hope to model such phenomena as void propagation and other two phase transient flow effects using lumped single phase equations.

Conservation of Energy

In the dintegral sense, the change in the total kinetic and internal energy of a lump of fluid is simply that due to heat transport thru the surface of volume plus internal heat sources or sinks less any work done by or on the fluid. Symbolically:

$$\frac{D}{Dt}\iiint_\forall \gamma_k\rho_k\left(e_k+\tfrac{1}{2}V_k^2\right)d\forall = -\iint_s \underset{\sim}{q}_k\cdot\underset{\sim}{n}\,ds + \iiint_\forall E_k\,d\forall \tag{16}$$

$$+ \iiint_\forall \gamma_k\rho_k\underset{\sim}{f}_k\cdot\underset{\sim}{V}\,d\forall + \iint_s(\underset{\approx}{\sigma}_k\cdot\underset{\sim}{n})\cdot\underset{\sim}{V}_k\,ds\,,$$

where e_k = internal energy of phase k
and E_k = internal heat sources and sinks of phase k.

The left hand side is the substantial derivative of the internal plus kinetic energy. The right hand side terms are, respectively:

1) surface heat flux,
2) internal sources and sinks,
3) work due to long range body forces (gravity, etc.),
4) work due to short range forces (surface tension, pressure, etc.).

Using Reynold's Transport Theorem again:

$$\iiint_{\forall}\frac{\partial}{\partial t}\left[\gamma_K\rho_K\left(e_K+\tfrac{1}{2}V_K^2\right)\right]d\forall+\iint_S\gamma_K\rho_K\left(e_K+\tfrac{1}{2}V_K^2\right)\underset{\sim}{V}_K\cdot\underset{\sim}{n}\,ds \tag{17}$$
$$=-\iint_S\underset{\sim}{q}_K\cdot\underset{\sim}{n}\,ds+\iiint_{\forall}E_K\,d\forall+\iiint_{\forall}\gamma_K\rho_K\underset{\sim}{f}_K\cdot\underset{\sim}{V}_K\,d\forall+\iiint_S\left(\underset{\approx}{\sigma}_K\cdot\underset{\sim}{n}\right)\cdot\underset{\sim}{V}_K\,ds,$$

Summing over k, the mixture equation becomes:

$$\iiint_{\forall}\frac{\partial}{\partial t}\left[\overline{\rho e}+\tfrac{1}{2}\overline{\rho V^2}\right]d\forall+\iint_S\overline{\left[\rho e+\tfrac{1}{2}\rho V^2\right]\underset{\sim}{V}}\cdot\underset{\sim}{n}\,ds \tag{18}$$
$$=-\iint_S\underset{\sim}{q}\cdot\underset{\sim}{n}\,ds+\iiint_{\forall}E\,d\forall+\iiint_{\forall}\overline{\rho\underset{\sim}{f}}\cdot\underset{\sim}{V}\,d\forall+\iint_S\left(\underset{\sim}{\sigma}\cdot\underset{\sim}{n}\right)\cdot\underset{\sim}{V}\,ds,$$

where

$$\overline{\rho e}=\gamma_1\rho_1e_1+\gamma_2\rho_2e_2\quad\text{and}\quad E=E_1+E_2.$$

Using Gauss' Theorem to change some of the surface integrals to volume integrals:

$$\iiint_{\forall}\frac{\partial}{\partial t}\left[\overline{\rho e}+\tfrac{1}{2}\overline{\rho V^2}\right]d\forall+\iint_S\overline{\rho e\underset{\sim}{V}}\cdot\underset{\sim}{n}\,ds+\iiint_{\forall}\nabla\cdot\left[\tfrac{1}{2}\overline{\rho V^2\underset{\sim}{V}}\right]d\forall \tag{19}$$
$$=-\iint_S\underset{\sim}{q}\cdot\underset{\sim}{n}\,ds+\iiint_{\forall}E\,d\forall+\iiint_{\forall}\overline{\rho\underset{\sim}{f}}\cdot\underset{\sim}{V}\,d\forall+\iiint_{\forall}\nabla\cdot\left(\overline{\underset{\approx}{\sigma}\cdot\underset{\sim}{V}}\right)d\forall.$$

Since $\underset{\approx}{\sigma}=-p\underset{\approx}{I}+\underset{\approx}{\tau}$,

$$\iiint_{\forall}\nabla\cdot\left(\overline{\underset{\approx}{\sigma}\cdot\underset{\sim}{V}}\right)d\forall=\iiint_{\forall}\left[\nabla\cdot\left(\overline{\underset{\approx}{\tau}\cdot\underset{\sim}{V}}\right)-\nabla\cdot\left(p\underset{\sim}{V}\right)\right]d\forall.$$

This is the total energy equation, composed of thermal terms and mechanical terms. We can separate the two by first generating the mechanical terms from the momentum equation (Equation 10). Forming the dot product with velocity we get:

$$\iiint_{\forall}\left[\frac{\partial}{\partial t}\overline{\left(\rho\underset{\sim}{V}\right)\cdot\underset{\sim}{V}}\,d\forall+\iiint_{\forall}\overline{\underset{\sim}{V}\cdot\left(\nabla\cdot\rho\underset{\sim}{V}\underset{\sim}{V}\right)}=\iiint_{\forall}\overline{\underset{\sim}{V}\cdot\left(\nabla\cdot\underset{\approx}{\tau}\right)}\right. \tag{20}$$
$$-\iiint_{\forall}\underset{\sim}{V}\cdot\nabla p\underset{\approx}{I}\,d\forall+\iiint_{\forall}\rho\,\overline{\underset{\sim}{f}\cdot\underset{\sim}{V}}\,d\forall.$$

Now $$\underset{\sim}{V}\cdot(\nabla\cdot\underset{\approx}{\tau}) = \nabla(\underset{\approx}{\tau}\cdot\underset{\sim}{V}) - \underset{\approx}{\tau}:\nabla\underset{\sim}{V}, \tag{21}$$

$$\underset{\sim}{V}\cdot\nabla p\underset{\approx}{I} = \nabla\cdot p\underset{\sim}{V} - p\nabla\cdot\underset{\sim}{V}, \tag{22}$$

$$\underset{\sim}{V}\cdot\frac{\partial}{\partial t}(\rho\underset{\sim}{V}) = \frac{\partial}{\partial t}\left(\frac{1}{2}\rho\underset{\sim}{V}\cdot\underset{\sim}{V}\right) = \frac{\partial}{\partial t}\left(\frac{1}{2}\rho V^2\right), \tag{23}$$

and
$$\underset{\sim}{V}\cdot(\nabla\cdot\rho\underset{\sim}{V}\underset{\sim}{V}) = \nabla\cdot\left(\frac{1}{2}\rho V^2\underset{\sim}{V}\right). \tag{24}$$

Using these identities and subtracting Equation (20) from Equation (19), we get:

$$\iiint_{\forall}\frac{\partial}{\partial t}(\overline{\rho e})\,d\forall + \iint_S\overline{\rho e\underset{\sim}{V}\cdot\underset{\sim}{n}}\,ds = -\iint_S\underset{\sim}{q}\cdot\underset{\sim}{n}\,ds + \iiint_{\forall} E + \iiint_{\forall}\overline{\underset{\approx}{\tau}:\nabla\underset{\sim}{V}}\,d\forall - \iiint_{\forall} p\nabla\cdot\underset{\sim}{V}\,d\forall. \tag{25}$$

This is the thermal form of the energy equation.

If we assume that the density and enthalpy are uniform over the node (the volume in question), then

$$\iiint_{\forall}\frac{\partial}{\partial t}(\overline{\rho e})\,d\forall = \frac{\partial U}{\partial t}, \tag{26}$$

where
$$U \equiv \forall\overline{\rho e} = LA\overline{\rho e}. \tag{27}$$

The integral of the transport term can be written over the flow surfaces:

$$\iint_S\overline{\rho e\underset{\sim}{V}\cdot\underset{\sim}{n}}\,ds = \iint_{A_1}\overline{\rho e\underset{\sim}{V}\cdot\underset{\sim}{n}}\,ds + \iint_{A_2}\overline{\rho e\underset{\sim}{V}\cdot\underset{\sim}{n}}\,ds + \cdots, \tag{28}$$

where A_1, A_2, etc., are the pipe flow cross-sectional areas. For inflow, $\underset{\sim}{V}\cdot\underset{\sim}{n}$ is negative. For outflow, $\underset{\sim}{V}\cdot\underset{\sim}{n}$ is positive. Assuming uniform velocity, enthalpy and density across the link (pipe) cross-section gives:

$$\begin{aligned}\iint_S\overline{\rho e\underset{\sim}{V}\cdot\underset{\sim}{n}}\,ds &= -\sum_{\text{IN FLOW}}\overline{\rho eV}A_i + \sum_{\text{OUT FLOW}}\overline{\rho eV}A_i \\ &= \sum\overline{W_{IN}e_{IN}} + \sum\overline{W_{OUT}e_{OUT}}.\end{aligned} \tag{29}$$

The heat flux and generation terms of the thermal energy equation can be lumped into a losely defined heat source for the volume:

$$-\iint_S\underset{\sim}{q}\cdot\underset{\sim}{n}\,ds + \iiint_{\forall}E\,d\forall \equiv Q. \tag{30}$$

Therefore, the thermal energy equation becomes:

$$\frac{\partial U}{\partial t} = \Sigma' \overline{W_{IN} e_{IN}} - \Sigma' \overline{W_{OUT} e_{OUT}} + Q + \iiint_{\forall} \underset{\approx}{\tau} : \nabla \underset{\sim}{V} \, d\forall . \tag{31}$$

The last two terms are the irreversible and reversible conversion to internal energy, respectively.

SOPHT uses:

$$\frac{\partial U'}{\partial t} = \Sigma' \overline{W_{IN} h_{IN}} - \Sigma' \overline{W_{OUT} h_{OUT}} + Q , \tag{32}$$

where $U' = \forall \overline{\rho h}$

$$h = \text{enthalpy} \equiv e + P/\rho . \tag{33}$$

By rearranging the reversible energy term of Equation (31), U and U′ can be compared. Thus, using Gauss' Theorem:

$$\iiint_{\forall} P \nabla \cdot \underset{\sim}{v} \, d\forall = \iiint_{\forall} P/\rho \, (\rho \nabla \cdot \underset{\sim}{v}) d\forall = \iint_S P/\rho \, (\rho \underset{\sim}{V} \cdot \underset{\sim}{n}) dS \tag{34}$$

(NOTE: assumes no spatial variation in P &)

Performing the surface integrals over the inflows and outflows gives:

$$\iiint_{\forall} P \nabla \cdot \underset{\sim}{v} \, d\forall = \Sigma' \overline{(P/\rho)(\rho V)} A \Big|_{IN} + \Sigma' \overline{(P/\rho)(\rho V) A} \Big|_{OUT} \tag{35}$$

$$= -\Sigma' P_{IN} \overline{W_{IN}/\rho_{IN}} + \Sigma' P_{OUT} \overline{W_{OUT}/\rho_{OUT}} ,$$

where

$$\overline{W/\rho} = \frac{\gamma_1 \rho_1 V_1 A}{\rho_1} + \frac{\gamma_2 \rho_2 V_2 A}{\rho_2} .$$

Therefore,

$$\Sigma' \overline{W_{IN} e_{IN}} - \Sigma' \overline{W_{OUT} e_{OUT}} - \iiint_{\forall} P \nabla \cdot \underset{\sim}{v} \tag{36}$$

$$= \Sigma' \overline{W_{IN} (e_{IN} + P_{IN}/\rho_{IN})} + \Sigma' \overline{W_{OUT} (e_{OUT} + P_{OUT}/\rho_{OUT})}$$

$$= \Sigma' \overline{W_{IN} h_{IN}} - \Sigma' \overline{W_{OUT} h_{OUT}} .$$

where

$$\overline{Wh} = \gamma_1 \rho_1 V_1 A h_1 + \gamma_2 \rho_2 V_2 A h_2 .$$

Equation (31) can be written as:

$$\frac{\partial u}{\partial t} = \sum \overline{W_{IN} h_{IN}} - \overline{W_{OUT} h_{OUT}} + Q + \iiint_{\forall} \underset{\approx}{\tau} : \nabla \underset{\sim}{V} \, d\forall \qquad (37)$$

$$= \frac{\partial U'}{\partial t} - \forall \frac{\partial P}{\partial t} \, . \qquad (38)$$

Therefore, comparing Equations (32) and (37), SOPHT has neglected the irreversible conversion to internal energy or turbulent heating. A lumped approximation is made to this in SOPHT by adding the pump heat to the node downstream of the pump.

The term, $\frac{\partial P}{\partial t}$, in Equation (38) is neglected in SOPHT.

At a pressure 10 MPa, which is typical for the CANDU Primary Heat Transport, this converts to 10 kJ/kg while e is approximately 1000 kJ/kg. Thus the discrepancy of the P/ρ term is negligible.

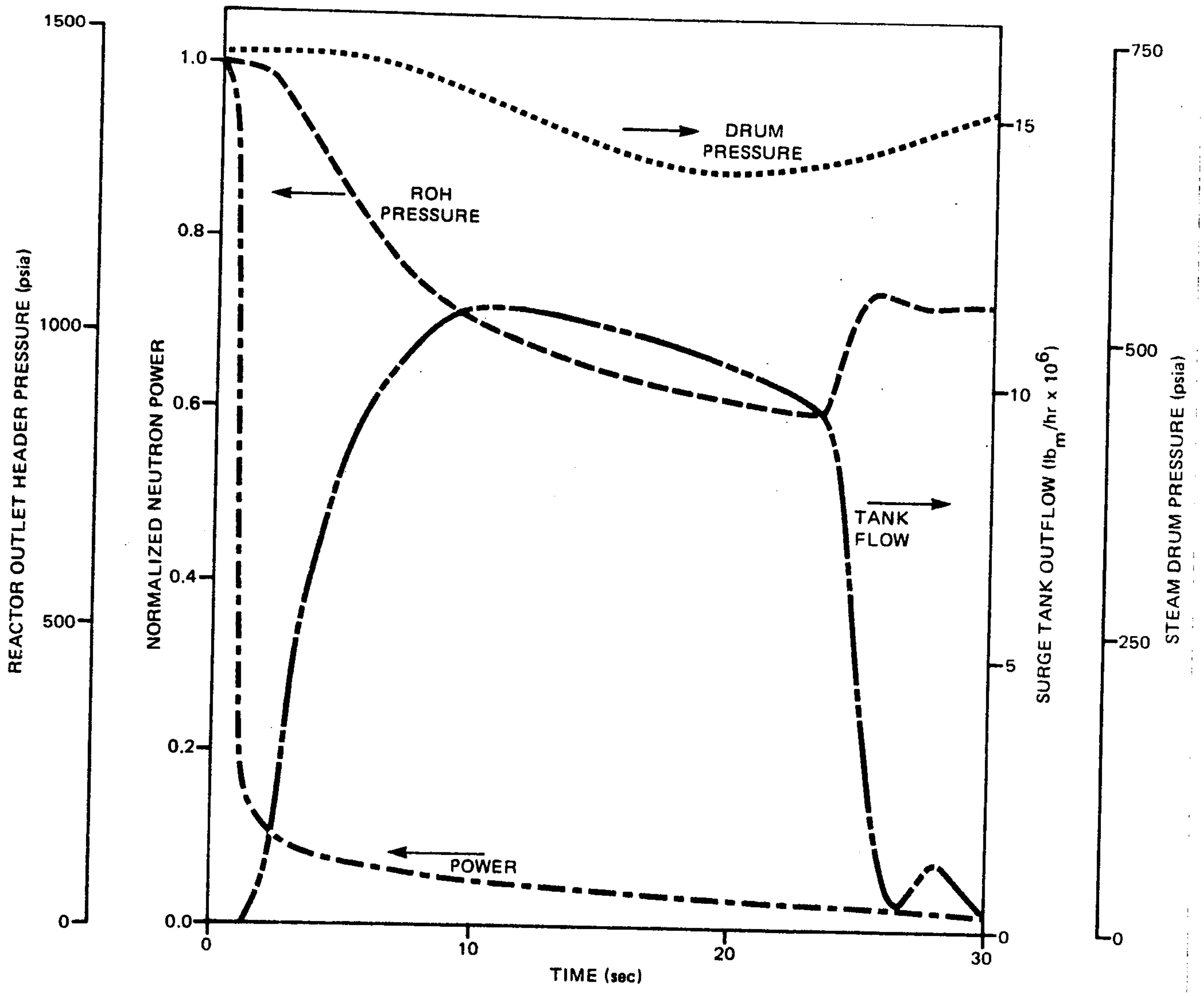

FIGURE 18.1 STEAM DRUM PRESSURE, SURGE TANK FLOW AND PRIMARY SYSTEM PRESSURE AND POWER AS A FUNCTION OF TIME FOR A REACTOR TRIP

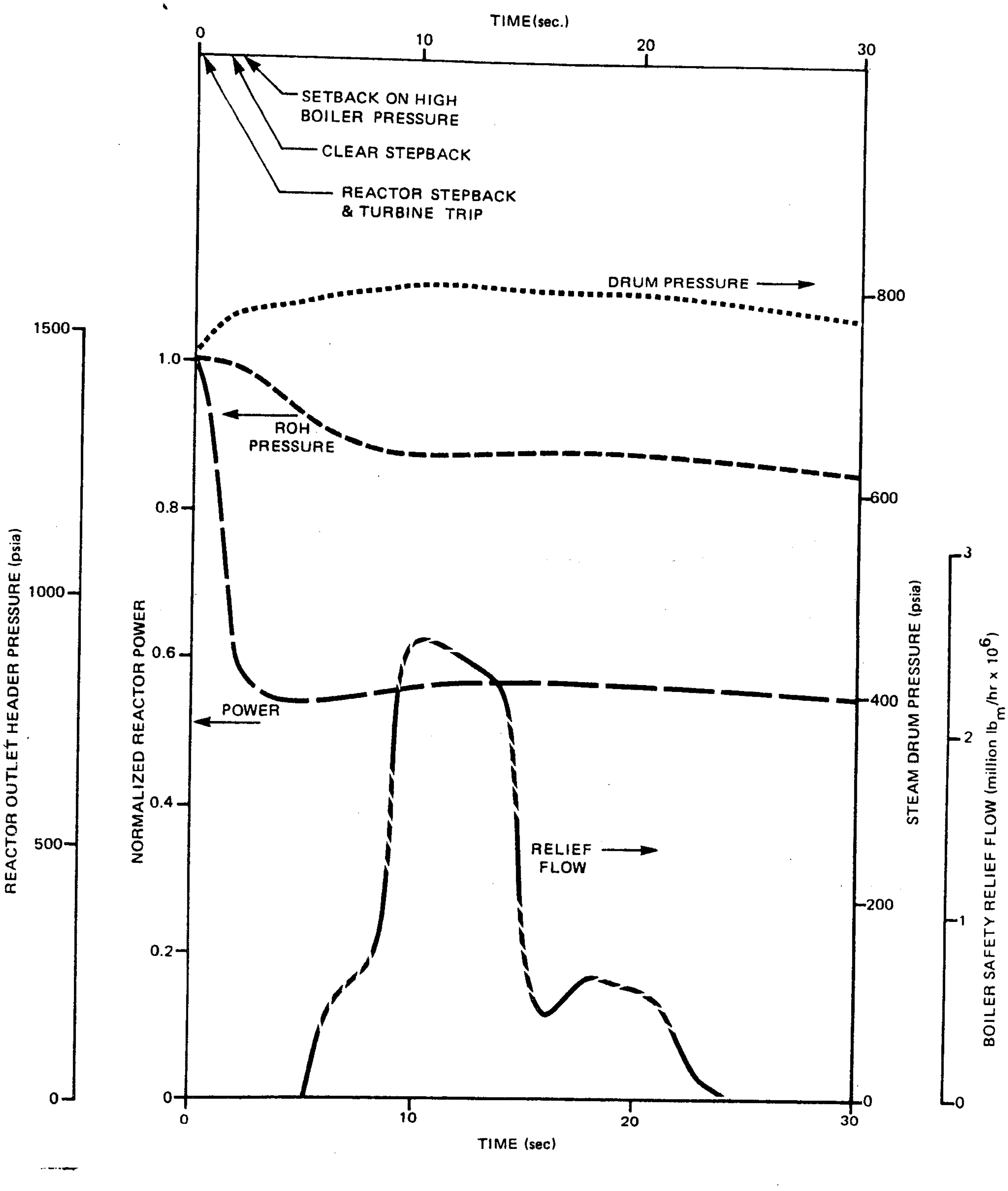

FIGURE 18.2 STEAM DRUM PRESSURE, BOILER SAFETY RELIEF FLOW AND PRIMARY SYSTEM PRESSURE AND POWER AS A FUNCTION OF TIME FOR A TURBINE TRIP

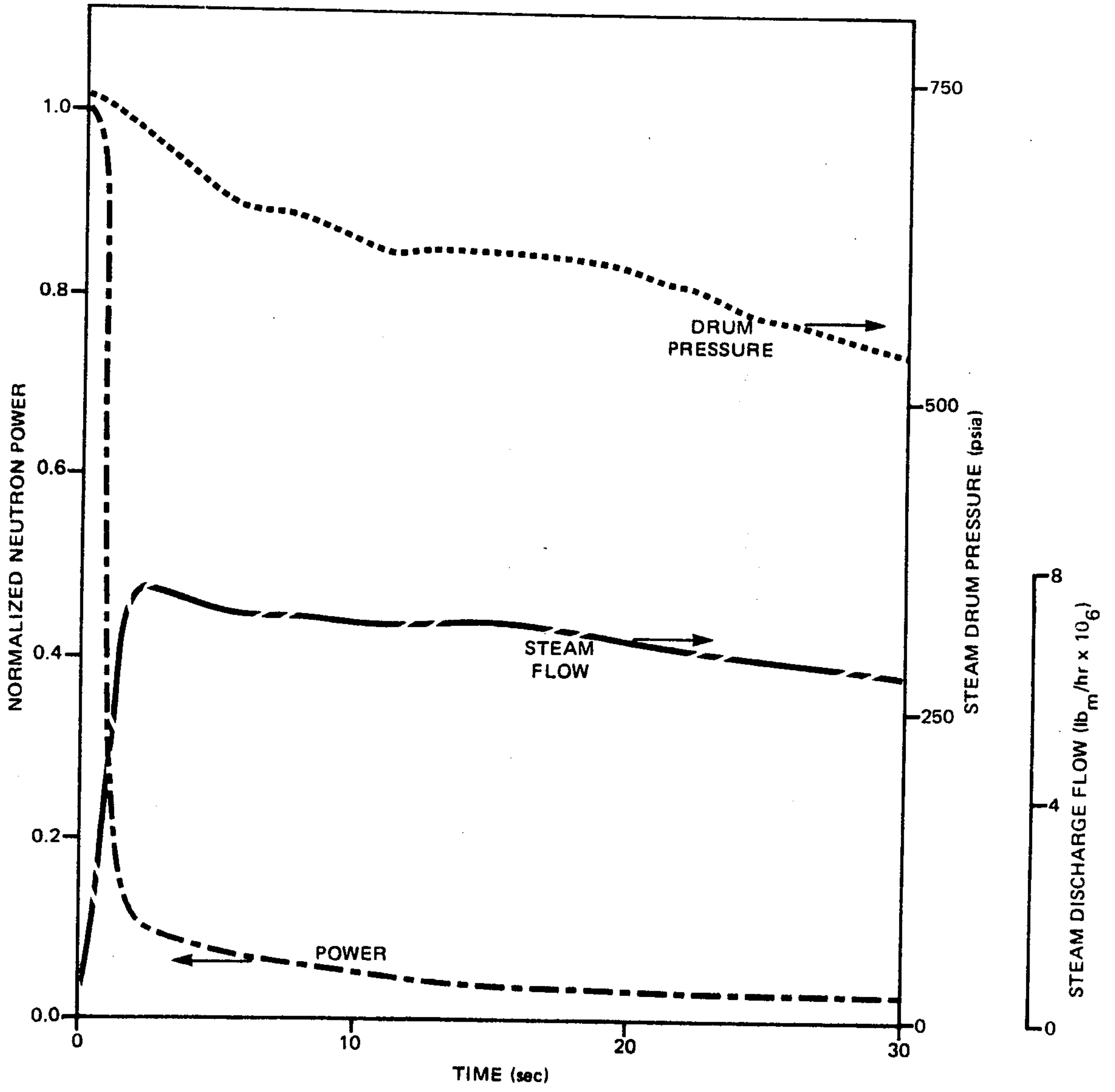

FIGURE 18.3 STEAM DRUM PRESSURE, STEAM DISCHARGE FLOW AND PRIMARY SYSTEM POWER AS A FUNCTION OF TIME FOR A RAPID COOLDOWN

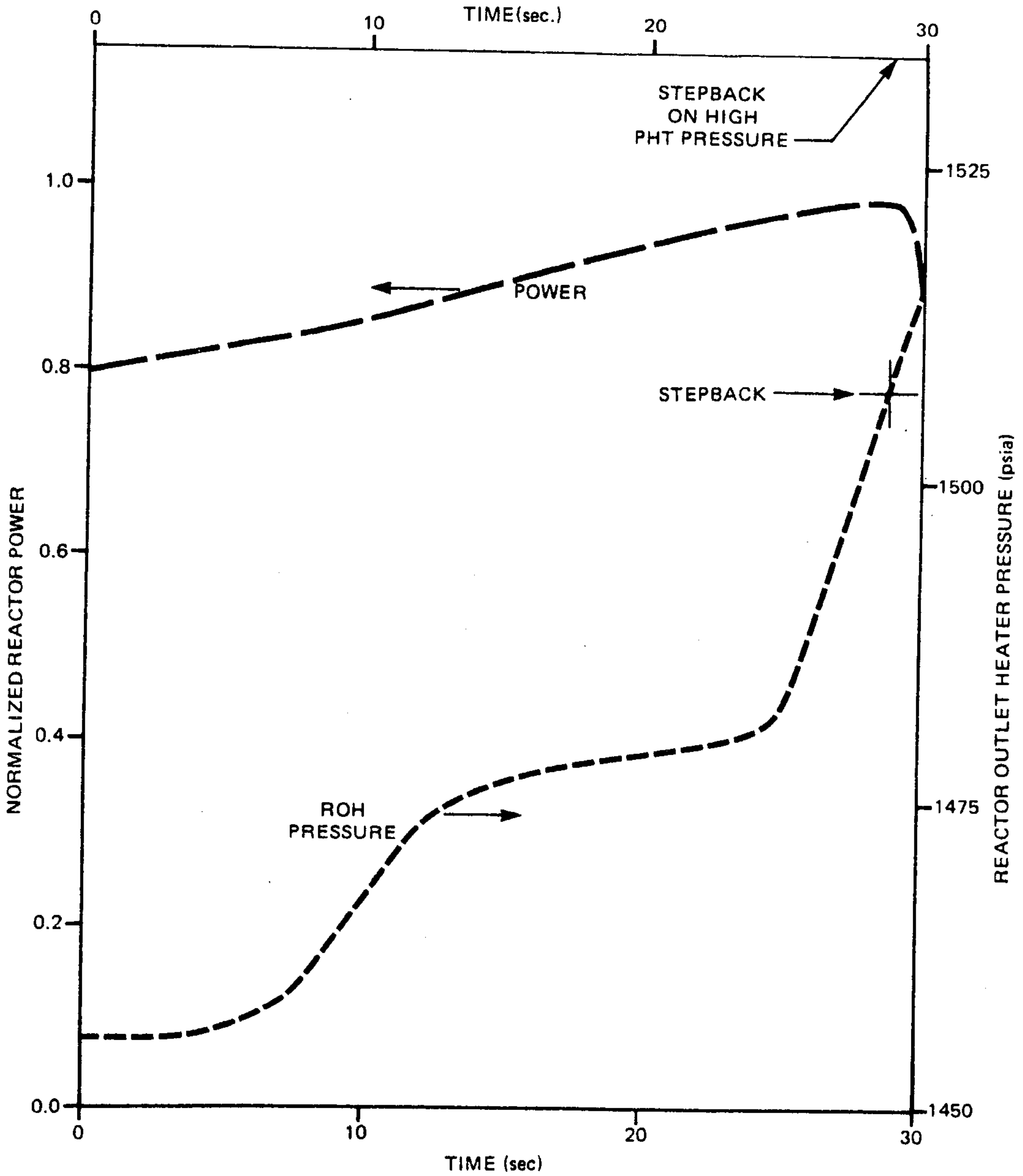

FIGURE 18.4 PRIMARY SYSTEM POWER AND PRESSURE AS A FUNCTION OF TIME FOR MANEUVERING FROM 80% TO 100% FULL POWER

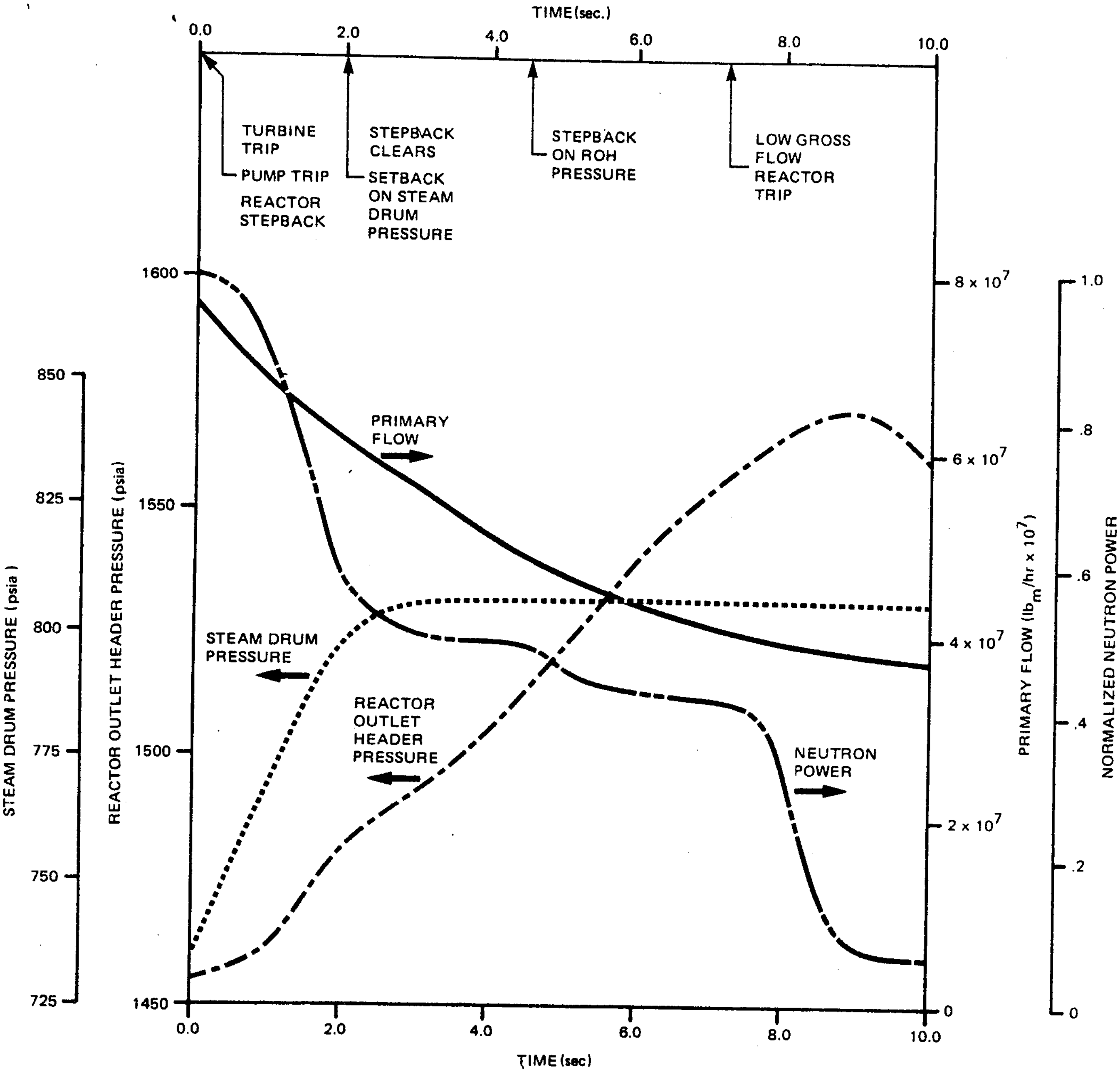

FIGURE 18.5 STEAM DRUM PRESSURE AND PRIMARY SYSTEM PRESSURE, FLOW AND POWER AS A FUNCTION OF TIME FOR A TOTAL LOSS OF CLASS IV POWER FAILURE

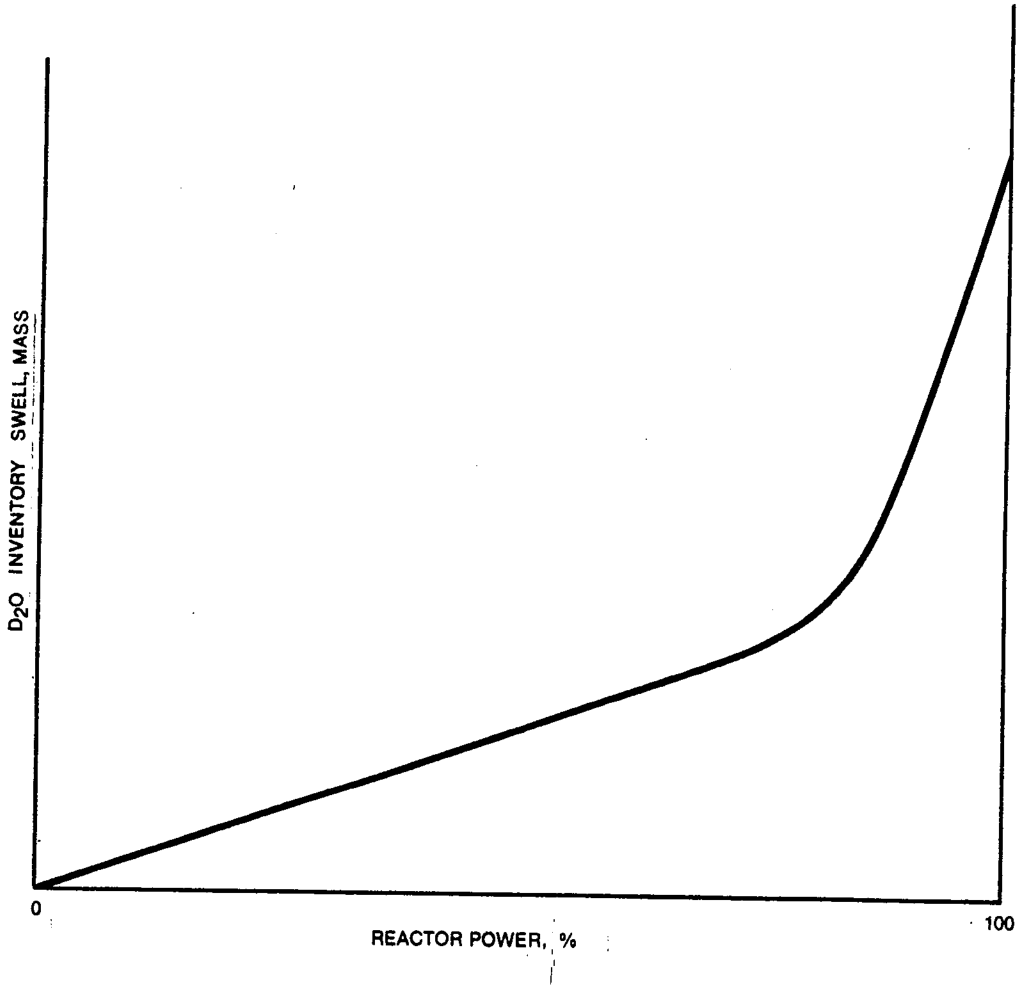

FIGURE 18-6 PHTS SWELL CURVE