

## **The SAS4A/SASSYS-1 Safety Analysis Code System**

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**Nuclear Engineering Division**

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## **The SAS4A/SASSYS-1 Safety Analysis Code System**

### **Chapter 5:**

### **Primary and Intermediate Loop Thermal Hydraulics Module**

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## NOMENCLATURE

Symbol	Description	Units
$A$	Coolant flow area	$m^2$
$A_{CP}, A_{CI}$	Primary and intermediate coolant flow areas in the intermediate heat exchanger	$m^2$
$A_k$	Coolant flow area of element $k$ in a liquid segment	$m^2$
$A_w$	Wall area of a compressible volume	$m^2$
$A_{inter}$	Area of liquid-gas interface	$m^2$
$a_0(i)$		$1/m$
$a_1(i)$	Coefficients for liquid segment $i$ .	$1/m\text{-s}$
$a_2(i)$		$1/m\text{-s}$
$a_3(i)$		$1/m$
$a_1(j), \dots, a_4(j)$	Coefficients in the simultaneous equations for the temperature changes at node $j$ in the intermediate heat exchanger	$J/m^2\text{-K}$
$\Delta a_0(k)$		$1/m$
$\Delta a_1(k)$	Contributions from element $k$ in liquid segment $i$ to the	$1/m\text{-s}$
$\Delta a_2(k)$	coefficients $a_0(i), \dots, a_3(i)$	$1/m\text{-s}$
$\Delta a_3(k)$		$1/m$
$b_0(j)$	Coefficients in the pressure change expression for	Pa
$b_1(j)$	Compressible volume $j$	$kg/m\text{-s}$
$b_2(j)$		$kg/m\text{-s}\text{-K}$
$c_c, c_l$	Specific heat of coolant or liquid	$J/kg\text{-K}$
$c_w$	Specific heat of compressible volume or pipe wall	$J/kg\text{-K}$
$\bar{c}_p, \bar{c}_l$	Spatially averaged primary and intermediate coolant heat capacities in the intermediate heat exchanger	$J/kg\text{-K}$
$\bar{c}$	Spatially averaged coolant heat capacity in the bypass	$J/kg\text{-K}$
$C_1, C_2, C_3$	User-supplied correlation coefficients in film heat-transfer calculation	--
$c_1(j), \dots, c_6(j)$	Coefficients in the simultaneous equations for the temperature changes at node $j$ in the intermediate heat exchanger	$J/m\text{-K}$
$C_0(L, ic)$	Coefficients in the expression for core channel	$kg/s^2$
$C_1(L, ic)$	flow estimate in channel $ic$ at end $L$	$m$
$C_2(L, ic)$	$L = 1$ for inlet, $2$ for outlet	$m$

<b>Symbol</b>	<b>Description</b>	<b>Units</b>
$C_3(L,ic)$		1/kg
$c_{ij}$	Matrix in cover-gas pressure change equation	---
$c(I,J)$	Matrix in compressible volume pressure change equation for liquid flow	kg
$\Delta c(I,J)$	Contribution to the matrix $c(i,j)$	kg
$D_1,D_2,D_3$	Right side of simultaneous equation solution in the cover gas, intermediate heat exchanger, and bypass channel treatments	W/m
$D,D_h$	Hydraulic diameters	m
DRACS	Direct reactor auxiliary cooling system	
$d_A,d_B,d_D$	Reflector thicknesses in the bypass channel	m
$d_{SH},d_{TU}$	Shell and tube thicknesses in the intermediate heat exchanger	m
$d_1$	Coefficients in coolant-wall temperature	J/m-K
$d_2$	Calculations	J/m <sup>2</sup> -K
$d_5$		m-K/J
$d_j$	Right side of cover-gas pressure change matrix equation	Pa
$d(j)$	Right side of liquid flow pressure change matrix equation	kg <sup>2</sup> /m-s <sup>2</sup>
$\Delta d(j)$	Contribution to $d(j)$	kg <sup>2</sup> /m-s <sup>2</sup>
$E_{src}$	Heat source	W/m <sup>3</sup>
$e_1(j),\dots,e_{10}(j)$	Coefficients in the simultaneous equations for the temperature changes at node $j$ in the intermediate heat exchanger	
$e_0(L,ic)$	Coefficients in the expression for core channel	kg/s
$e_1(L,ic)$	flow estimate in channel $ic$ at end $L$	m-s
$e_2(L,ic)$		m-s
$F_{ij}$	Cover-gas flow rate from compressible volume $i$ to compressible volume $j$	kg/s
$F_{0ij}$		kg/s
$F_{1ij}$	Coefficients in the expression for $F_{ij}$	m-s
$F_{2ij}$		m-s
$f$	Moody friction factor	--
$f_r$	Fraction of a coolant node at pipe inlet	--

Symbol	Description	Units
$f_1(j), \dots, f_8(j)$	Coefficients in the simultaneous equations for the temperature changes at node $j$ in the intermediate heat exchanger	
$f_1, f_2$	Fractions of reactor power	--
$G$	Specific flow rate	kg/m <sup>2</sup> -s
$GH$	Gravity head	m
$G_2$	Pressure drop parameter	--
$g$	Acceleration due to gravity	m/s <sup>2</sup>
$H$	Head	m
$H_r$	Pump reference head	m
$H_{SP}, H_{PT}, H_{TI}$	Overall heat-transfer coefficients for the intermediate heat exchanger	W/m <sup>2</sup> -K
$H_{AB}, H_{BC}, H_{CD}$	Overall heat-transfer coefficients between regions in the bypass channel	W/m <sup>2</sup> -K
$H_{snk}A_{snk}$	Heat transfer coefficient times area	W/K
$(hA)_{snk}$	Heat transfer coefficient times area	W/K
$H_D$	Steam drum total enthalpy	J
$h_D$	Specific enthalpy	J/kg
$h_f$	Liquid specific enthalpy	J/kg
$h_g$	Vapor specific enthalpy	J/kg
$h_c, h_{CP}, h_{CI}$	Coolant film coefficients	W/m <sup>2</sup> -K
$h_w, h_{wc}$	Wall and wall-to-coolant heat-transfer coefficients	W/m <sup>2</sup> -K
$h_{FS}, h_{f_{FT}}$	Fouling factors in the intermediate heat exchanger	W/m <sup>2</sup> -K
$IHX$	Intermediate heat exchanger	
$K_{cold}$	A cold leg pressure loss coefficient	s <sup>2</sup> /m
$k_c$	Coolant thermal conductivity	W/m-K
$k_{SH}, k_{TU}$	Shell and tube thermal conductivities in the intermediate heat exchanger	W/m-K
$L, L_i$	Length	m
$\Delta L(N)$	Length-integrated volumetric heat source	W/m <sup>2</sup>
$L_k$	Length of element $k$ in a liquid flow segment	m
$L_n$	Length of a node at pipe inlet	m
$L_B/D_B$	Length-to-diameter ratio per bend in an element	--
$M$	Mach number for cover gas flow	--
$M_D$	Steam drum total mass	kg

<b>Symbol</b>	<b>Description</b>	<b>Units</b>
$M_w$	Pipe wall mass per unit length	kg/m
$\delta_m$	Change in liquid mass	kg
$\delta m_i$	Change in cover gas mass in compressible volume $i$ during a sub-interval of time	kg
$\Delta m, \Delta m_l$	Change in liquid mass	kg
$\Delta m_{ji}$	Mass of cover gas flowing from compressible volume $j$ to $i$ during a sub-interval of time	kg
$m(i)$	Cover-gas mass in compressible volume $i$	kg
$m$	Number of evaporators in parallel	--
$m_g$	Gas mass in a compressible volume	kg
$m_r$	Liquid mass in a compressible volume at a reference pressure	kg
$\Delta m$	Change in liquid mass in a compressible volume during a time sub-interval	kg
$N_B$	Number of bends in an element	--
$P, P_{er}, P_A, P_D, P_{SP}, P_{TI}$	Perimeters	m
$P_s$	Power shape for vertical section	--
$p, p_l$	Liquid pressure in a compressible volume	Pa
$p_d, p_{d1}, p_{d2}$	Pump head coefficients	
$p_r$	Reference liquid pressure in a compressible volume	Pa
$\Delta p$	Change in liquid pressure in a compressible volume during time sub-interval	Pa
$p(i), p_g$	Cover-gas pressure in compressible volume $i$	Pa
$P_1, P_2$	Power shape normalization for Regions 1 and 2	
$p(JIN), p(JX)$	Inlet and outlet plenum pressures in core flow estimate	Pa
$\Delta p_i$	Change in cover-gas pressure in compressible volume $i$ during a sub-interval of time	Pa
$p_{in}(i), p_{out}(i)$	Pressures at the inlet and outlet of liquid segment $i$	Pa
$\delta p$	Change in liquid pressure	Pa
$\Delta p_{fr}(i)$	Frictional pressure loss in liquid segment $i$	Pa
$\Delta p_{w2}(i)$	Pressure loss proportional to the square of the flow in liquid segment $i$	Pa
$\Delta p_v(i)$	Valve pressure loss in liquid segment $i$	Pa
$\Delta p_{gr}(i)$	Gravity head in liquid segment $i$	Pa
$\Delta p_p(i)$	Pump head in liquid segment $i$	Pa

<b>Symbol</b>	<b>Description</b>	<b>Units</b>
$P_{TOT}Q_{MULT}$	Reactor power for a time step	W
$P_s(j)$	Power shape by nodes	--
$P1, P3$	Power shape normalization factors	--
$Q_A, Q_B, Q_D$	Neutron and decay heating sources in reflectors	W/m <sup>3</sup>
$Q_{MULT}$	Power multiplication factor	--
$q, Q$	Heat flow from compressible volume walls	W/m <sup>2</sup>
$Q_i^{Na}$	Region length-integrated sodium heat source w	W/m <sup>2</sup>
$Q_i$	Length-integrated volumetric heat source	W/m <sup>2</sup>
$Q$	Heat removal rate	--
$R$	Universal gas constant	J/kg-K
$Re$	Reynolds number	--
$Ri$	Richardson number	--
$S$	Slant height ratio of the tube-side in the intermediate heat exchanger	--
$s$	Pump speed	1/s
$T$	Pump torque	N-m
$T, T_g$	Cover-gas temperature	K
$T, T_l$	Liquid temperature	K
$T_c, T_w$	Coolant and wall temperatures	K
$T_f$	Friction torque loss	N-m
$T_m, T_p$	Motor torque, pump torque	N-m
$T_{SH}, T_{TU}$	Shell and tube node temperatures in the intermediate heat exchanger	K
$\bar{T}_{CP}, \bar{T}_{CI}$	Spatially averaged coolant temperatures in the intermediate heat exchange	K
$T(i)$	Cover-gas temperature in compressible volume i	K
$T_{in}(j), T_{out}(j)$	Temperature of the incoming and outgoing fluid at compressible volume j	K
$T_{ji}$	Temperature of the cover-gas flowing from compressible volume j to i	K
$T_B$	Temperature of the flow from a bypass channel to the outlet plenum	K
$T_C$	Temperature of the flow from a core channel to the outlet plenum	K
$T_{snk}$	Temperature of the heat sink	K

<b>Symbol</b>	<b>Description</b>	<b>Units</b>
$T_{Na,i}$	Sodium temperature in the $i$ -th region	K
$T_{out}$	Outlet temperature in outlet plenum	K
$T_{out}(t)$	Steam generator outlet temperature	K
$\delta T$	Change in liquid temperature	K
$\Delta T$	Liquid temperature change during a time sub-interval	K
$\Delta T_w$	Wall temperature change during a time sub-interval	K
$\Delta T_{SH}, \Delta T_{TU}$	Shell and tube node temperature changes during a time sub-interval	K
$\Delta T_{CP}, \Delta T_{CI}$	Coolant node temperature changes during a time a sub-interval	K
$\Delta t$	Time-step size	s
$\Delta t_s, \delta t$	Sub-interval of a time step	s
$U$	Overall heat transfer coefficient	W/m <sup>2</sup> -K
$u$	Flow velocity	m/s
$V(i), V_g$	Cover-gas volume in compressible volume $i$	m <sup>3</sup>
$V_D$	Steam drum volume	m <sup>3</sup>
$v_D$	Steam drum specific volume	m <sup>3</sup> /kg
$v_j$	Jet velocity at hot-cold interface	m/s
$v$	Liquid velocity	m/s
$V, V_l$	Liquid volume in a compressible volume	m <sup>3</sup>
$V_r$	Liquid volume in a compressible volume at a reference pressure	m <sup>3</sup>
$\Delta V_g$	Change in cover gas volume	m <sup>3</sup>
$\Delta V_l$	Change in liquid volume	m <sup>3</sup>
$w$	Pump flow rate	kg/s
$w(i)$	Liquid flow rate in segment $i$	kg/s
$\bar{w}(i)$	Time-averaged liquid flow rate in segment $i$	kg/s
$\bar{w}_{in}(i),$ $\bar{w}_{out}(i)$	Time-averaged liquid flow rate into and out from compressible volume $j$	kg/s
$\bar{w}_B$	Average flow rate from a bypass channel into the outlet plenum	kg/s
$\bar{w}_C$	Average flow rate from a core channel into the outlet plenum	kg/s
$w_c(L, ic)$	Core flow rate in channel $ic$ at end $L$ at the beginning of the timestep	kg/s

<b>Symbol</b>	<b>Description</b>	<b>Units</b>
$\Delta W(i)$	Change in the liquid flow rate in segment $i$ during a time sub-interval	kg/s
$Z, Z_r, Z_{ref}$	Reference elevation	m
$Z_i, Z_{inter}$	Liquid-gas interface elevation	m
$Z_{in}, Z_{out}$	Height of element inlet and outlet	m
$Z_{PLENL}$	Lower plenum elevation	m
$Z_{PLENU}$	Upper plenum elevation	m
$Z_{IHX}$	Reference height of the thermal center of the intermediate heat exchanger	m
$\Delta z, \Delta z(j)$	Height of the $j$ -th node	m
$\alpha_P$	Volume pressure expansion coefficient	1/Pa
$\alpha_T$	Volume temperature expansion coefficient	1/K
$\alpha_{1, \dots, \alpha_4}$	Tri-diagonal matrix coefficients in the cover gas, intermediate heat exchanger, and bypass channel treatments	W/m-K
$\beta_P$	Liquid sodium compressibility	1/Pa
$\beta_T$	Liquid sodium thermal expansion coefficient	1/K
$\beta_{1, \dots, \beta_4}$	Tri-diagonal matrix coefficients in the cover gas, intermediate heat exchanger, and bypass channel treatments	w/m-K
$\gamma$	Ratio of specific heat at constant pressure to that at constant volume	--
$\gamma_{N1}, \gamma_{N3}$	Fraction of neutron power	--
$\gamma_{D1}, \gamma_{D3}$	Fraction of decay power	--
$\gamma(i)$	Factor in the degree of implicitness in liquid segment $i$	--
$\gamma_c(L, ic)$	Factor in the degree of implicitness in core channel $ic$ at end $L$	--
$\delta_{ij}$	Kronecker delta symbol	--
$\varepsilon, \varepsilon(k)$	Pipe roughness factor	m
$\varepsilon_i$	Factor in equation for cover-gas flow in compressible volume $i$	--
$\varepsilon_v$	Cover-gas volume fraction	--
$\xi$	Ratio of shutoff head to design head	--
$\rho, \rho_l$	Liquid density	kg/m <sup>3</sup>
$\bar{\rho}_P, \bar{\rho}_I$	Spatially averaged primary and intermediate coolant densities in the intermediate heat exchanger	kg/m <sup>3</sup>

<b>Symbol</b>	<b>Description</b>	<b>Units</b>
$\rho_i, \rho_0$	Fluid density at element inlet and outlet	kg/m <sup>3</sup>
$(\rho c)_{SH}, (\rho c)_{TU}$	Shell and tube densities times specific heats in the intermediate heat exchanger	J/m <sup>3</sup> -K
$(\rho c)_A, (\rho c)_B, (\rho c)_D$	Densities times specific heats in the bypass channel	J/m <sup>3</sup> -K
$\tau$	Cover-gas time constant	s
$\tau_m, \tau_p$	Motor torque, pump torque	N-m
$\theta_1(i), \theta_2(i)$	Degree of implicitness in the time differencing in liquid segment $i$	--
$\theta_{2c}(L, ic)$	Degree of implicitness in the time differencing in core channel $ic$ at end $L$	--

Subscripts 3 and 4 appended to quantities refer to the values of those quantities at the beginning and at the end, respectively, of a time interval.

## PRIMARY AND INTERMEDIATE LOOP THERMAL HYDRAULICS MODULE

### 5.1 Introduction

The PRIMAR-4 module computes coolant pressures, flow rates, and temperatures in the primary and intermediate heat transport loops. This module is designed for analysis of a wide range of transients, from fast unprotected LOF or TOP cases to slow operational transients or natural circulation shut-down heat-removal cases. Also, an arbitrary arrangement of components in either a loop-type or a pool-type system can be treated. Semi-implicit and fully implicit numerical schemes have been developed to handle the full range of transients efficiently. This chapter describes the physical models and the numerical algorithms used in PRIMAR-4. As indicated in Fig. 5.1-1, PRIMAR-4 couples with the driver and blanket subassembly coolant dynamics calculations in the pre-voiding thermal hydraulics module TSCL0, and the boiling module TSBOIL. Also, PRIMAR-4 couples with the detailed steam generator model described in Chapter 7, and with the control system described in Chapter 6.

The PRIMAR-4 module contains both a simple PRIMAR-1 type option and the more detailed PRIMAR-4 treatment. The PRIMAR-1 option supplies only the minimum quantities of information required to drive the subassembly coolant dynamics calculations. It is often used when the more detailed treatment is not required. This minimum information consists of the inlet and outlet plenum pressures, the subassembly inlet temperatures, and the outlet plenum temperature in the event of flow reversal. All of these quantities are computed as functions of time by PRIMAR-1 from user-supplied information independently of what is happening in the core or in the rest of the primary loop.

The detailed multi-loop PRIMAR-4 treatment models heat transfer and coolant flow in the inlet and outlet coolant plenums, the pipes, pumps, and valves in the primary and intermediate loops, the intermediate heat exchangers, the steam generators, and the pool in a pool system. It utilizes a modular approach. The user specifies the properties of various components and arranges them in an arbitrary manner. Each type of component is treated in a separate section of the code. Components can be added or existing component treatments can be modified without impacting the rest of the code.

The detailed PRIMAR-4 treatment includes both a steady-state initialization sub-module and a transient sub-module. The main parts of the transient sub-module are a hydraulic section in which liquid coolant flow rates and pressures are calculated, a liquid temperature section, and a gas section that calculates gas flow between cover gasses. For a given time step, the liquid flow rates and pressures are calculated first, taking into account pressure and flow rate changes during the time step, but ignoring temperature changes and gas flow changes during the time step. This not only simplifies the calculations, but it makes explicit use of the fact that the incompressible single-phase liquid flow rates are more sensitive to pressure changes than they are to temperature and gas flow changes. Then, with the flow rates known, the liquid temperatures are calculated for the time step. Finally, with the liquid pressures, flow

rates, and temperatures known, the gas flow rates and pressures are calculated, and the liquid pressures are adjusted to account for changes in the cover-gas pressures.

A number of significant changes have been made recently to the PRIMAR-4 treatment in SAS4A and SASSYS-1. The detailed steam generator model has been replaced by a new, faster running version. Also, a Balance-of-Plant treatment of the water system beyond the steam generator has been added. The new steam generator and the Balance-of-Plant treatment are described in Chapter 7. Another change is that multiple inlet plena and outlet plena are now allowed. Previously, all core channels connected to the same inlet plenum and to the same outlet plenum. This made modeling of EBR-II difficult, since EBR-II has two separate inlet plena.

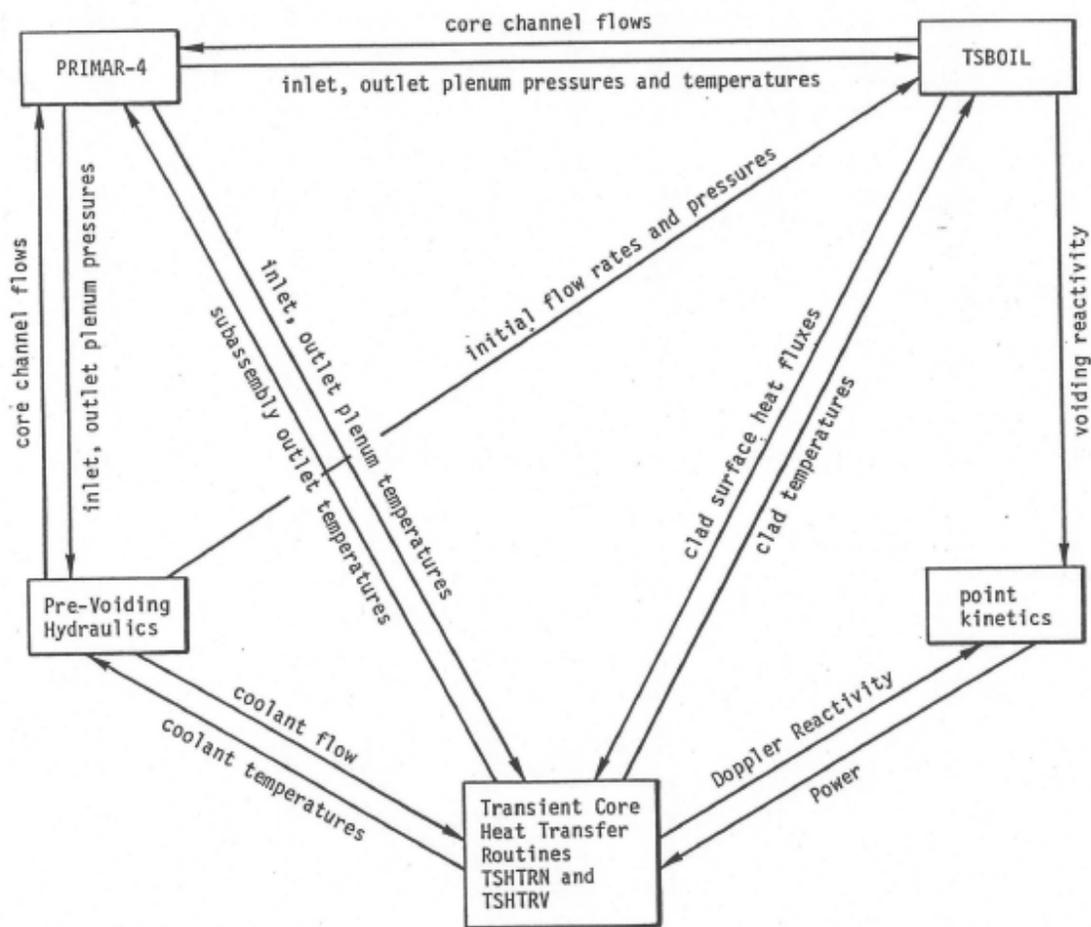


Figure 5.1-1. Interaction between PRIMAR-4 and Other Modules

## 5.2 Hydraulic Calculations

### 5.2.1 Compressible Volumes and Liquid Segments

The primary and intermediate loop thermal hydraulics calculations use a generalized geometry, as indicated in Fig. 5.2-1. A number of compressible volumes are connected by liquid or gas segments, and each liquid segment can contain one or more elements. The treatment allows compressible volumes and segments to be connected in an arbitrary manner. Table 5.2-1 lists the types of compressible volumes used in PRIMAR-4. Compressible volumes are characterized by pressure, volume, mass, and temperature. They can accumulate liquid or gas by compressing the cover gas or the liquid, and it is the pressure in the compressible volumes that drives the flows through the liquid and gas segments. Table 5.2-2 lists the types of elements that can make up a liquid segment. Liquid flow elements are characterized by incompressible single-phase flow, with the possible exception of the core element. The core subassemblies, which are treated by the coolant dynamics modules, are a special case and are discussed in Section 5.2.2.

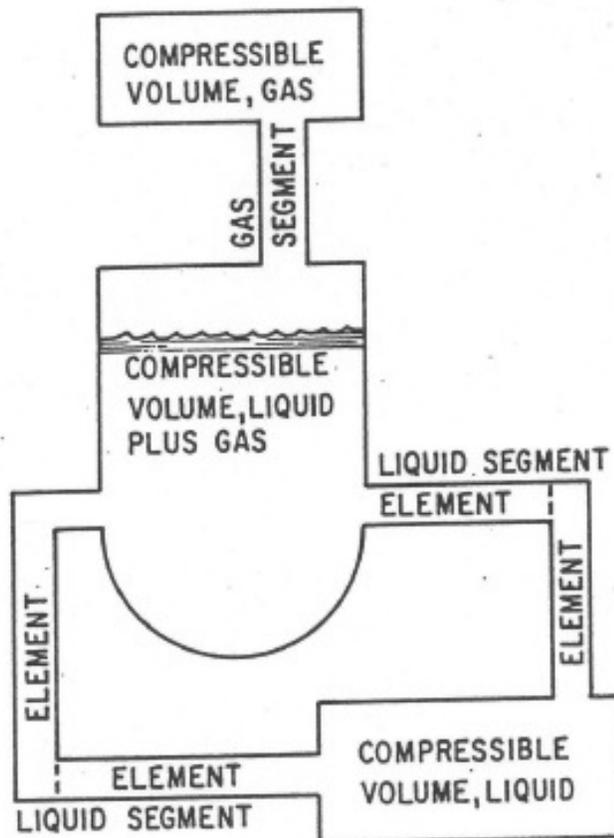


Figure 5.2-1. PRIMAR-4 Generalized Geometry

Table 5.2-1. Compressible Volume Types Used in PRIMAR-4

Type Number	Description
1	Inlet plenum
2	Compressible liquid volume, no cover gas
3	Closed outlet plenum, no cover gas
4	Almost incompressible liquid junction, no cover gas
5	Pipe rupture source
6	Pipe rupture sink, guard vessel
7	Outlet plenum with cover gas
8	Pool
9	Pump bowl and cover gas
10	Expansion tank with cover gas
11	Compressible gas volume, no liquid

Table 5.2-2. Liquid Flow Element Types Used in PRIMAR-4

Type Number	Description
1	Core subassemblies, SAS channels
2	Core bypass assemblies
3	Pipe
4	Check valve
5	Pump impeller
6	IHX, shell side
7	IHX, tube side
8	Steam generator, sodium side
9	DRACS heat exchanger, tube side
10	DRACS heat exchanger, shell side
11	Valve
12	Air dump heat exchanger, sodium side
13	Annular element

The hydraulic equations for the primary and intermediate heat-transport loops are solved by a semi-implicit or fully implicit time differencing scheme in which the pressures and flows for all connected compressible volumes and segments are solved for simultaneously. By linearizing the equations for each time step, a semi-implicit or fully implicit solution can be obtained without resorting to iteration techniques. Linearized semi-implicit or fully implicit methods are most useful for long transients in

which temperatures and flows change slowly, since in such cases accurate results can be obtained with large time steps so long as the step sizes are small enough that changes during a step are small.

Three equations are used in calculating the pressures in the compressible volumes and the flow rates in the connecting liquid segments. They are the momentum equation for incompressible single-phase flow in a segment, an expression for the average flow rate in a segment during a time step, and an expression for the change in pressure in a compressible volume as a result of flow into it and out from it during a time step. Each of these equations is taken up in turn.

### 5.2.1.1 Momentum Equation

The momentum equation for a single-phase incompressible liquid is taken as

$$\frac{1}{A} \frac{\partial w}{\partial t} + \frac{1}{A^2} \frac{\partial}{\partial z} \left( \frac{w^2}{\rho} \right) + \frac{\partial p}{\partial z} + \left( \frac{\partial p}{\partial z} \right)_{loss} = 0 \quad (5.2-1)$$

where

$w$  = the mass flow rate

$A$  = the flow area

$\rho$  = the density of the liquid

$\frac{\partial p}{\partial z}$  = the pressure gradient driving the flow

$\left( \frac{\partial p}{\partial z} \right)_{loss}$  = the pressure drop from all of the loss terms

When Eq. 5.2-1 is integrated over a segment containing several elements, it can be written as the basic equation for the flow in segment  $i$ :

$$\sum_k \frac{L_k}{A_k} \frac{dw(i)}{dt} = p_{in}(i) - p_{out}(i) - \Delta p_{fr}(i) - \Delta p_{w2}(i) - \Delta p_v(i) - \Delta p_{gr}(i) + \Delta p_p(i) \quad (5.2-2)$$

Here,  $dw(i)/dt$  is the time rate of change of the mass flow rate through segment  $i$ . The summation is over the elements in segment  $i$ , and  $L_k$  and  $A_k$  are the length and flow area of element  $k$ . The term  $p_{in}(i)$  is the pressure at the inlet to segment  $i$ , which is the pressure in the compressible volume at the inlet of segment  $i$ . The term  $p_{out}(i)$  is the pressure at the outlet of segment  $i$ , or in the compressible volume at the outlet of segment  $i$ . The term  $\Delta p_{fr}(i)$  is the frictional pressure change for the segment  $i$ , and with the minus sign in Eq. 5.2-2, it is a loss term. However, this frictional loss term for the

segment is actually the sum of similar loss terms for each element in the segment. The same things can be said about the remaining terms in Eq. 5.2-2. Together with their respective signs,  $\Delta p_{w2}(i)$  is any orifice or bend pressure drop proportional to the square of the mass flow rate,  $\Delta p_v(i)$  is any valve pressure drop,  $\Delta p_{gr}(i)$  is the gravity-head pressure drop, and  $\Delta p_p(i)$  is the pump-head pressure increase from all of the pumps in segment  $i$ .

Equation 5.2-2 has the form:

$$\sum_k \frac{L_k}{A_k} \frac{dw(i)}{dt} = f(w, t) \quad (5.2-3)$$

which can be written in finite difference form

$$\sum_k \frac{L_k}{A_k} \frac{\Delta w(i)}{\Delta t} = \theta_1(i) f_i(w, t) + \theta_2(i) f_i(w + \Delta w, t + \Delta t) \quad (5.2-4)$$

where  $\theta_1 + \theta_2 = 1$ . The parameters  $\theta_1$  and  $\theta_2$  determine the degree of implicitness of the solution. For a fully explicit solution,  $\theta_1 = 1$  and  $\theta_2 = 0$ . For a fully implicit solution  $\theta_1 = 0$  and  $\theta_2 = 1$ . The degree of implicitness is discussed in Section 5.2.4 and in Appendix 3.1 in Chapter 3.

The linearization consists in making the approximation that

$$f(w + \Delta w, t + \Delta t) = f(w, t) + \Delta t \frac{\partial f}{\partial t} + \Delta w \frac{\partial f}{\partial w} \quad (5.2-5)$$

so the flow equation becomes

$$\sum_k \frac{L_k}{A_k} \Delta w(i) = \Delta t \left\{ f(w, t) + \theta_2(i) \left[ \Delta t \frac{\partial f}{\partial t} + \Delta w \frac{\partial f}{\partial w} \right] \right\} \quad (5.2-6)$$

which takes the form

$$a_0(i) \Delta w(i) = a_1(i) + \theta_2(i) \{ a_2(i) + \Delta t [\Delta p(ji) - \Delta p(jo)] + a_3(i) \Delta w(i) \} \quad (5.2-7)$$

or

$$\Delta w(i) = \frac{a_1(i) + \theta_2(i) \{ a_2(i) + \Delta t [\Delta p(ji) - \Delta p(jo)] \}}{a_0(i) - \theta_2(i) a_3(i)} \quad (5.2-8)$$

where  $ji$  and  $jo$  are the compressible volumes at the inlet and outlet of the liquid segment.

In general, the  $a$ 's are sums of contributions from each element,  $k$ , in the segment. The terms are

$$a_0 = \sum_k \Delta a_0(k) \quad (5.2-9)$$

$$\Delta a_0(k) = \frac{L_k}{A_k} \quad (5.2-10)$$

$$a_1 = \Delta t [p(ji, t) - p(jo, t)] + \sum_k \Delta a_1(k) \quad (5.2-11)$$

$$\Delta a_1(k) = [-\Delta p_{fr}(k, t) - \Delta p_{gr}(k, t) - \Delta p_{w2}(k, t) - \Delta p_v(k, t) + \Delta p_p(k, t)] \Delta t \quad (5.2-12)$$

$$a_2 = \sum_k \Delta a_2(k) \quad (5.2-13)$$

$$\begin{aligned} \Delta a_2(k) = \Delta t^2 \frac{\partial}{\partial t} [\Delta p_p(k) - \Delta p_{fr}(k) - \Delta p_{w2}(k) \\ - \Delta p_v(k) - \Delta p_{gr}(k)] \end{aligned} \quad (5.2-14)$$

$$a_3 = \sum_k \Delta a_3(k) \quad (5.2-15)$$

and

$$\Delta a_3(k) = \Delta t \frac{\partial}{\partial w} [\Delta p_p(k) - \Delta p_{fr}(k) - \Delta p_{w2}(k) - \Delta p_v(k) - \Delta p_{gr}(k)] \quad (5.2-16)$$

In the above equations, it should be recognized that  $\Delta$  is used in three different ways. First  $\Delta t$  is the time step,  $\Delta w(i)$  is the change in the mass flow rate in the liquid segment  $i$  during the time step, and  $\Delta p(ij)$  and  $\Delta p(jo)$  are the changes in pressures in the compressible volumes at the inlet and outlet ends of liquid segment  $i$  during the time step. Second, the  $\Delta p$ 's in Eq. 5.2-2 represent pressure differences, increases or decreases, along liquid segment  $i$ . And third, as seen in Eqs. 5.2-9, 5.2-11, 5.2-13, and 5.2-15, the  $\Delta a$ 's are incremental contributions from each of the elements to the  $a$ 's for the whole segment.

The denominator in Eq. 5.2-8 should never be zero because  $a_3$  is negative or zero, since friction increases with increased flow and pump head decreases with increased flow.

### 5.2.1.2 Average Flow Rate

The second main equation used in the hydraulics calculation is an expression for the average mass flow rate in a liquid segment during a time step. The average mass flow rate  $w(i)$  for segment  $i$  is taken as a simple average of the flow rate at the beginning of the time step and that at the end of the time step.

$$\bar{w}(i) = [w(i, t) + w(i, t + \Delta t)] / 2 \quad (5.2-17)$$

Linearization consists in expanding  $w(i, t + \Delta t)$  to two terms and finite differencing the time derivative to give

$$\bar{w}(i) = w(i, t) + \Delta w(i) / 2 \quad (5.2-18)$$

Equation 5.2-18 relates the average mass flow rate in a segment during a time step to the change in the mass flow rate during that time step.

### 5.2.1.3 Compressible Volume Pressure Changes

The third main equation in the hydraulics calculations is an expression for the change in pressure in a compressible volume during a time step. The pressure in a compressible volume can be affected in several ways. Liquid can flow in or out through the segments connecting the compressible volumes. The entering liquid may be at a higher or lower temperature than that already there, and the liquid flowing out removes liquid at the compressible volume temperature. In addition to the changes related to the liquid flows, the compressible volumes can be heated or cooled externally, or liquid can be added or withdrawn by an external agent. The flow of cover gas into or out of the compressible volume will also affect the pressure. The cover gas flows are treated separately, as in Section 5.6.

The pressure in a compressible volume is assumed to vary linearly with changes in the mass or temperature of the liquid. Therefore, the change  $\Delta p(j)$  in pressure in the compressible volume  $j$  during a time step is taken as a linear approximation in the average mass flow rates into and out from that compressible volume as:

$$\begin{aligned} \Delta p(j) = & b_0(j) + b_1(j) \left[ \sum \bar{w}_{in}(j) - \sum \bar{w}_{out}(j) \right] \\ & + b_2(j) \left[ \sum \bar{w}_{in}(j) T_{in} - \sum \bar{w}_{out}(j) T_{out}(j) \right] \end{aligned} \quad (5.2-19)$$

Here  $\sum \bar{w}_{in}(j)$  is the sum of the average mass flow rates into compressible volume  $j$  from all of the attached liquid segments flowing into it. It should be noticed that Eq. 5.2-

18 is the expression for the average mass flow rate in a segment, and if that segment flows into compressible volume  $j$ , then it is included with all of the other segment contributions to compressible volume  $j$ . Similarly,  $\sum \bar{w}_{out}(j)$  is the sum of the average mass flow rates out of the compressible volume  $j$  from all of the attached liquid segments flowing out. The last two sums in Eq. 5.2-19 are the same as the ones just described except that all of the mass flow rates are multiplied by the temperatures of the flows: the average mass flow rates flowing into the compressible volume are multiplied by the temperatures in the respective segments, whereas the average mass flow rates flowing out are each multiplied by the temperature in the compressible volume. The coefficients  $b_1(j)$  and  $b_2(j)$  include the time-step size and are computed for each type of compressible volume. The remaining term  $b_0(j)$  also contains the time step size and can be used to account for the effects of heat transfer to the compressible volume liquid from the compressible volume wall or from other components.

### 5.2.2 Estimated Core Flow

The liquid segments representing the core channels are a special case. In principle, the coolant mass flow rates in the core channels could be calculated simultaneously along with the other segments in the primary loop. But after the onset of boiling in the core, this would unduly complicate the boiling model. Instead, an estimate of the core mass flow rates is made from information stored by the core channel coolant dynamics routines during the previous time step, and this estimated flow is included in the primary loop calculation. After the primary and intermediate loop hydraulics calculations have been done, the core channel coolant dynamics routines compute the actual channel flows for each channel independently, using the newly calculated inlet and outlet plenum pressures as boundary conditions. Then, the differences between the estimated core flow and the actual computed core flow for a time step is used to adjust the coolant masses in the inlet and outlet plenums before the start of the calculations for the next time step.

The mass flow rate for core channel  $ic$  at the end  $L$ , where  $L=1$  is the inlet and  $L=2$  is the outlet, is estimated by the momentum equation

$$\begin{aligned} \frac{dw_c(L,ic)}{dt} = & C_0(L,ic) + C_1(L,ic)p(JIN) + C_2(L,ic)p(JX) \\ & + C_3(L,ic)w_c(L,ic) | w_c(L,ic) | \end{aligned} \quad (5.2-20)$$

where  $p(JIN)$  is the pressure in the compressible volume representing the inlet plenum and  $p(JX)$  is the pressure in the compressible volume representing the outlet plenum. The rate of change of the mass flow rate is taken as proportional to the inlet and outlet pressures and proportional to the square of the mass flow rate. The coefficients  $C_0$ ,  $C_1$ ,  $C_2$ , and  $C_3$  are the information stored during the previous time step, and are described in Section 3.11.2 of Chapter 3. The channel mass flow rate  $w_c(L,ic)$  is evaluated at the beginning of the time step.

Equation 5.2-20 can be written in finite difference form as

$$\begin{aligned}
 \Delta w_c(L, ic) = & \frac{\Delta t}{1 - 2\theta_{2c}(L, ic)C_3(L, ic)w_c(L, ic)\Delta t} \{C_0(L, ic) \\
 & + C_1(L, ic)[p(JIN, t) + \theta_{2c}(L, ic)\Delta p(JIN)] \\
 & + C_2(L, ic)[p(JX, t) + \theta_{2c}(L, ic)\Delta p(JX)] \\
 & + C_3(L, ic)w(L, ic)|w_c(L, ic)| \}
 \end{aligned}
 \tag{5.2-21}$$

where  $\theta_{2c}$  is the degree of implicitness for core channel  $ic$ . Equation 5.2-21 plays the role for each core channel that Eq. 5.2-8 does for all the other liquid segments in the primary loop.

Before boiling begins in the core channels, the differences between the estimated and the actual core flows are very small, largely because Eq. 5.2-20 is equivalent to the equation used by the pre-boiling core channel coolant dynamics routines, except that the coefficients in Eq. 5.2-20 do not account for the effects of coolant temperature changes during the current time step. After boiling begins, rapid changes in vapor pressures cause rapid changes in the inlet plenum pressure, necessitating a decrease in the time step size.

### 5.2.3 Method of Solution

Equations 5.2-8, 5.2-18, 5.2-19, and 5.2-21 constitute a set of simultaneous equations for changes in the mass flow rates  $\Delta w(i)$  in the liquid segments and for changes in the pressures  $\Delta p(j)$  in the compressible volumes during a time step. Eliminating the  $\Delta w$ 's, these equations can be written as a single matrix equation for the  $\Delta p$ 's:

$$\sum_J c(I, J)\Delta p(J) = d(I)
 \tag{5.2-22}$$

The coefficients  $c(I, J)$  and  $d(I)$  are sums of contributions  $\Delta c(I, J)$  and  $\Delta d(I)$  from each segment. From segment  $i$ , in which the flow is from compressible volume I to compressible volume J, the contribution to  $c(I, J)$  is

$$\Delta c(I, J) = \frac{-\theta_2(i)[b_1(J) + b_2(J)T_{out}(i)]\Delta t}{2[a_0(i) - a_3(i)\theta_2(i)]}
 \tag{5.2-23}$$

and the contribution to  $c(J, I)$  is

$$\Delta c(J,I) = \frac{-\theta_2(i)[b_1(I)+b_2(I)T_{in}(i)]\Delta t}{2[a_0(i)-a_3(i)\theta_2(i)]} \quad (5.2-24)$$

The contribution to  $d(J)$  is

$$\Delta d(J) = w(i,t) + \frac{a_1(i)+\theta_2(i)a_2(i)}{2[a_0(i)-\theta_2(i)a_3(i)]} [b_1(J)+b_2(J)T_{out}(i)] \quad (5.2-25)$$

and the contribution to  $d(I)$  is

$$\Delta d(I) = -w(i,t) - \frac{a_1(i)+\theta_2(i)a_2(i)}{2[a_0(i)-\theta_2(i)a_3(i)]} [b_1(I)+b_2(I)T_{in}(i)] \quad (5.2-26)$$

The contributions to diagonal terms are

$$\Delta c(I,I) = \frac{\theta_2(i)[b_1(I)+b_2(I)T_{in}(i)]\Delta t}{2[a_0(i)-\theta_2(i)a_3(i)]} \quad (5.2-27)$$

and

$$\Delta c(J,J) = \frac{\theta_2(i)[b_1(J)+b_2(J)T_{out}(i)]\Delta t}{2[a_0(i)-\theta_2(i)a_3(i)]} \quad (5.2-28)$$

Also,  $b_0(J)$  is added to  $d(J)$  and 1.0 is added to the diagonal terms in  $c$ . In these equations  $T_{in}(i)$  and  $T_{out}(i)$  are the temperatures at the inlet and outlet of the liquid segment.

The contributions to the coefficients  $c(L,J)$  and  $d(I)$  from the segments representing the core channel flows are obtained from Eq. 5.2-21, which is re-written as

$$\Delta w_c(L,ic) = e_0(L,ic) + e_1(L,ic)\Delta p(JIN) + e_2(L,ic)\Delta p(JX) \quad (5.2-29)$$

where

$$e_0(L,ic) = \frac{\Delta t}{d_n} \left[ C_0(L,ic) + C_1(L,ic)p(JIN,t) + C_2(L,ic)p(JX,t) + C_3(L,ic)w_c(L,ic) |w_c(L,ic)| \right] \quad (5.2-30)$$

$$e_1(L, ic) = \theta_{2c}(L, ic) \Delta t C_1(L, ic) / d_n \quad (5.2-31)$$

$$e_2(L, ic) = \theta_{2c}(L, ic) \Delta t C_2(L, ic) / d_n \quad (5.2-32)$$

and

$$d_n = 1 - 2\theta_{2c}(L, ic) C_3(L, ic) | w_c(L, ic) | \Delta t \quad (5.2-33)$$

The contribution to  $d(JIN)$  is then

$$\Delta d(JIN) = -[b_1(JIN) + b_2(JIN) T_{in}(ic)] [w_c(1, ic) + e_0(1, ic) / 2] \quad (5.2-34)$$

also,

$$\Delta d(JX) = [b_1(JX) + b_2(JX) T_{out}(ic)] [w_c(2, ic) + e_0(2, ic) / 2] \quad (5.2-35)$$

$$\Delta c(JIN, JIN) = [b_1(JIN) + b_2(JIN) T_{in}(ic)] e_1(1, ic) / 2 \quad (5.2-36)$$

$$\Delta c(JX, JX) = -[b_1(JIN) + b_2(JX) T_{out}(ic)] e_2(2, ic) / 2 \quad (5.2-37)$$

$$\Delta c(JIN, JX) = [b_1(JIN) + b_2(JIN) T_{in}(ic)] e_2(1, ic) / 2 \quad (5.2-38)$$

and

$$\Delta c(JX, JIN) = [b_1(JX) + b_2(JX) T_{out}(ic)] e_1(2, ic) / 2 \quad (5.2-39)$$

The contributions to the coefficients  $c(I, J)$  and  $d(J)$  from all the liquid segments in a loop have been made at this point, and Eq. 5.2-22 is solved by Gaussian elimination to yield the pressure changes in all the compressible volumes in the loop during the time step. With the pressure changes now known, Eq. 5.2-8 is solved for the mass flow-rate changes in the liquid segments in the loop, and Eq. 5.2-21 is solved for the estimated channel flow changes during the time step. The pressure changes and the mass flow rate changes are then added to the values of the pressures and flow rates at the beginning of the time step to obtain the respective values at the end of the time step.

The above procedure is carried out separately for the primary loops and for the intermediate loops. The core channel flow segments are included only in the primary loops.

### 5.2.4 Degree of Implicitness

As mentioned in Section 5.2.1, the parameters  $\theta_1$  and  $\theta_2$  determine the degree of implicitness of the calculation. For small time steps, a semi-implicit treatment with  $\theta_1 = \theta_2 = .5$  is most accurate. For large time steps, a fully implicit calculation with  $\theta_1 = 0$  and  $\theta_2 = 1$  is more accurate and numerically more stable. As discussed in Appendix 3.1, the degree of implicitness is computed separately for each liquid segment,  $i$ , as

$$\theta_2(j) = \frac{a + b\gamma(i) + \gamma(i)^2}{2a + c\gamma(i) + \gamma(i)^2} \quad (5.2-40)$$

where

$$\gamma(i) = -\frac{a_3(i)}{a_0(i)} \quad (5.2-41)$$

$$a = 6.12992$$

$$b = 2.66054$$

and

$$c = 3.56284$$

Then

$$\theta_1(i) = 1 - \theta_2(i) \quad (5.2-42)$$

Note that  $a_3(i)$  is always negative and is proportional to  $\Delta t$ , the time-step size. Equations 5.2-40 and 5.2-41 give the results that  $\theta_2$  approaches 0.5 for small time steps,  $\theta_2$  approaches 1.0 for large time steps, and  $\theta_2$  makes a smooth transition between 0.5 and 1.0 for intermediate-sized time steps.

Liquid segments attached to an almost compressible liquid junction are a special case. Because such compressible volumes are much smaller than other compressible volumes, their time constants for changes of all kinds are short; and because of it, these liquid segments are treated with a fully implicit ( $\theta_2 = 1.0$ ) flow calculation regardless of time-step size.

The degree of implicitness used in the calculated estimated channel flow for channel  $ic$  is

$$\theta_{2c}(L, ic) = \frac{a + b\gamma_c(L, ic) + \gamma_c(L, ic)^2}{2a + c\gamma_c(L, ic) + \gamma_c(L, ic)^2} \quad (5.2-43)$$

where  $\gamma_c$  is given by

$$\gamma_c = \frac{2\Delta t}{w_c(L,ic) C_3(L,ic)} \quad (5.2-44)$$

### 5.3 Specific Components: Contributions to Flow and Pressure Equations

In this section, we shall consider the contributions to the mass flow-rate and pressure equations from the various components in the primary and intermediate loops. Each compressible volume contributes a  $b_0$ , a  $b_1$ , and a  $b_2$  in Eq. 5.2-19, and each element in each segment connecting two compressible volumes contributes a  $\Delta a_0$ , a  $\Delta a_1$ , a  $\Delta a_2$ , and a  $\Delta a_3$  in Eqs. 5.2-10, 5.2-12, 5.2-14, and 5.2-16. We shall consider compressible volumes without cover gas, compressible volumes with cover gas, pipes, intermediate heat exchangers, and pumps.

#### 5.3.1 Compressible Volumes Without Cover Gas

A compressible volume with no cover gas is treated as a compressible liquid in an expandable container. The volume  $V$  is assumed to vary linearly with pressure  $p$  and temperature  $T$ :

$$V = V_r [1 + \alpha_p (p - p_r) + \alpha_T (T - T_r)] \quad (5.3-1)$$

where  $V_r$  is the volume at a reference pressure  $p_r$  and reference temperature  $T_r$ . Also the sodium density  $\rho$  is assumed to vary linearly with  $p$  and  $T$ :

$$\rho = \rho_r [1 + \beta_p (p - p_r) + \beta_T (T - T_r)] \quad (5.3-2)$$

where

$$\alpha_p = \text{the volume pressure expansion coefficient, } \frac{1}{V} \frac{\partial V}{\partial p}$$

$$\alpha_T = \text{the volume thermal expansion coefficient, } \frac{1}{V} \frac{\partial V}{\partial T}$$

$$\beta_p = \text{the sodium compressibility, } \frac{1}{\rho} \frac{\partial \rho}{\partial p}$$

$$\beta_T = \text{the sodium thermal expansion coefficient, } \frac{1}{\rho} \frac{\partial \rho}{\partial T}$$

The mass of the liquid in the compressible volume is

$$m = \rho V \quad (5.3-3)$$

Using Eqs. 5.3-1 and 5.3-2 in Eq. 5.3-3 and dropping second-order terms gives

$$m = m_r \left[ 1 + (\alpha_p + \beta_p)(p - p_r) + (\alpha_T + \beta_T)(T - T_r) \right] \quad (5.3-4)$$

which can be rewritten as

$$\delta p = \frac{\delta m / m_r - (\alpha_T + \beta_T) \delta T}{\alpha_p + \beta_p} \quad (5.3-5)$$

where

$$\delta m = m - m_r$$

$$\delta p = p - p_r$$

$$\delta T = T - T_r$$

Equation 5.3-5 is a general relationship for the pressure change in a compressible volume with no cover gas as a result of mass and temperature changes.

To obtain expressions for  $b_0(j)$ ,  $b_1(j)$ , and  $b_2(j)$ , which characterize this compressible volume  $j$ , we apply conservation of mass and conservation of energy principles to the volume with flow in and flow out during a time step. Conservation of mass gives

$$\Delta m = \Delta t \left[ \sum \bar{w}_{in} - \sum \bar{w}_{out} \right] \quad (5.3-6)$$

where  $\Delta m$  is the change in the liquid mass in the compressible volume during the time step  $\Delta t$ ,  $\sum \bar{w}_{in}$  is the sum of the average mass flow rates into the compressible volume during  $\Delta t$ , and  $\sum \bar{w}_{out}$  is the sum of the average mass flow rates out during  $\Delta t$ .

Conservation of energy gives

$$(m_3 + \Delta m)(T_3 - \Delta T) = m_3 T_3 + \Delta t \left[ \sum \bar{w}_{in} T_{in} - \sum \bar{w}_{out} T_{out} \right] + \frac{Q}{c_\ell} \Delta t \quad (5.3-7)$$

where  $m_3$  and  $T_3$  are the mass and temperature of the liquid in the compressible volume at the beginning of the time step,  $\Delta T$  is the change in temperature of the liquid in the compressible volume during the time step,  $\sum \bar{w}_{in} T_{in}$  is the sum of the average mass flow

rates into the volume multiplied by the incoming temperature,  $\sum \bar{w}_{out} T_{out}$  is the sum of the average mass flow rates out of the volume multiplied by the outgoing temperature,  $Q$  is the heat flow rate from the compressible volume walls and from other components in contact with the compressible volume liquid, and  $c_l$  is the heat capacity of the liquid in the compressible volume. Equation 5.3-7 expresses the fact that the energy in the liquid in the compressible volume at the end of the time step is the sum of the energy present at the beginning of the time step, the excess of the energy flowing in over that flowing out during the time step, and the energy contributed to the liquid from the walls of the compressible volume during the time step.

Solving Eq. 5.3-7 for the change in the liquid temperature during the time step, gives

$$\Delta T = \frac{-T_3 \Delta m + \Delta t \left[ \sum \bar{w}_{in} T_{in} - \sum \bar{w}_{out} T_{out} \right] + \frac{Q}{c_l} \Delta t}{m_3 + \Delta m} \quad (5.3-8)$$

To first order, the  $m_3 + \Delta m$  in the denominator is approximated as  $m_3$ . Inserting Eqs. 5.3-6 and 5.3-8 into Eq. 5.3-5 gives

$$\Delta p = \left[ \sum \bar{w}_{in} - \sum \bar{w}_{out} \right] \frac{\Delta t}{\alpha_p + \beta_p} \left[ \frac{1}{m_r} + \frac{(\alpha_T + \beta_T) T_3}{m_3} \right] - \frac{(\alpha_T + \beta_T) \Delta t}{(\alpha_p + \beta_p) m_3} \left[ \sum \bar{w}_{in} T_{in} - \sum \bar{w}_{out} T_{out} + \frac{Q}{c_l} \right] \quad (5.3-9)$$

Comparison of this equation with Eq. 5.2-19 shows that for the compressible volume with no cover gas

$$b_0 = - \frac{(\alpha_T + \beta_T) Q \Delta t}{(\alpha_p + \beta_p) m_3 c_l} \quad (5.3-10)$$

$$b_1 = \frac{\Delta t}{\alpha_p + \beta_p} \left[ \frac{1}{m_r} + \frac{(\alpha_T + \beta_T) T_3}{m_3} \right] \quad (5.3-11)$$

and

$$b_2 = - \frac{(\alpha_T + \beta_T) \Delta t}{(\alpha_p + \beta_p) m_3} \quad (5.3-12)$$

The  $Q \Delta t$  term in Eq. 5.3-10 is calculated in a manner similar to that described in Section 5.4.4, except that the value for  $b_0$  is calculated before the temperatures at the end of the step are calculated, so  $q$  is calculated on the basis of temperatures at the

beginning of the time step. In the present version of the code, the reference mass  $m_r$  and the reference temperature  $T_r$  are taken as the mass and temperature at the beginning of the time step.

### 5.3.2 Compressible Volumes With Cover Gas

A compressible volume with cover gas is treated in a fashion similar to that without cover gas except that the compression of the liquid is neglected compared with the compression of the gas, so that all of the expansion or compression is attributed entirely to the gas. The cover gas is assumed to expand or compress adiabatically, and an increase in the gas volume is equal to the decrease in the liquid volume. The accompanying rise or fall in the level of the gas-liquid interface is taken as the volume change divided by the area of the compressible volume. In addition, the conservation of mass and the conservation of energy principles are observed. The conservation of mass is applied by taking the increase in the mass of the liquid in a compressible volume as the difference between the liquid flowing in and that flowing out during a time step, and the conservation of energy is taken as the increase in the mass of the liquid in the compressible volume times its temperature as the difference between the mass times temperature flowing in and that flowing out during a time step.

The liquid pressure at an elevation,  $z_r$ , in the compressible volume is given by

$$p_\ell = p_g + \rho_\ell g (z_i - z_r) \quad (5.3-13)$$

where

$p_\ell$  = the pressure in the liquid

$p_g$  = the pressure of the cover gas above the liquid

$\rho_\ell$  = the liquid density

$g$  = the acceleration of gravity

$z_i$  = the height of the liquid gas interface

$z_r$  = the reference height for the compressible volume

and the change in the liquid pressure in the compressible volume is obtained by taking differentials of Eq. 5.3-13:

$$\Delta p_\ell = \Delta p_g + g(z_i - z_r)\Delta\rho_\ell + \rho_\ell g \Delta z_i \quad (5.3-14)$$

Adiabatic compression of the cover gas is taken as

$$p_g V_g^\gamma = \text{const} \quad (5.3-15)$$

where

$V_g$  = the volume of the cover gas

$\gamma$  = the ratio of the specific heat at constant pressure to that at constant volume for the cover gas

and in differential form becomes:

$$\frac{\Delta p_g}{p_g} + \lambda \frac{\Delta V_g}{V_g} = 0 \quad (5.3-16)$$

The conservation of liquid mass for a compressible volume gives

$$\Delta m_\ell = \Delta t \left[ \sum \bar{w}_{in} - \sum \bar{w}_{out} \right] \quad (5.3-17)$$

where

$\sum m_\ell$  = the liquid mass increase in the compressible volume during the time step

$\Delta t$  = the time-step size

$\sum \bar{w}_{in}$  = the sum of the average liquid mass flow rates into the compressible volumes during the time step

$\sum \bar{w}_{out}$  = the sum of the average liquid mass flow rates out from the compressible volume during the time step.

The conservation of energy for a compressible volume yields

$$(m_\ell + \Delta m_\ell)(T_\ell + \Delta T_\ell) = m_\ell T_\ell + \Delta t \left[ \sum \bar{w}_{in} T_{in} - \sum \bar{w}_{out} T_{out} \right] \quad (5.3-18)$$

where

$m_\ell$  = the liquid mass at the beginning of the time step

$\Delta T_\ell$  = the increase in liquid temperature in the compressible volume during the time step

$\sum \bar{w}_{in} T_{in}$  = the sum of the average liquid mass flow rates times temperatures entering the compressible volume during the time step

$\sum \bar{w}_{out} T_{out}$  = sum of the average liquid mass flow rates times temperatures leaving the compressible volume during the time step.

In the present version of the code,  $\Delta m_l$  is neglected in comparison with  $m$ , giving

$$\Delta T_\ell = \frac{\Delta t}{m_\ell} \left[ \sum \bar{w}_{in} T_{in} - \sum \bar{w}_{out} T_{out} \right] \quad (5.3-19)$$

In addition to the above equations, we take

$$\Delta V_g = -\Delta V_\ell \quad (5.3-20)$$

$$V_\ell = m_\ell / \rho_\ell \quad (5.3-21)$$

$$\Delta z_i = \Delta V_\ell / A \quad (5.3-22)$$

where

$V_\ell$  = the volume of the liquid in the compressible volume at the beginning of a time step

$m_\ell$  = the mass of the liquid in the compressible volume at the beginning of a time step

$\rho_\ell$  = the density of the liquid

$A$  = the area of the compressible volume.

Differencing Eq. 5.3-21, we have

$$\Delta V_\ell = \frac{\Delta m_\ell}{\rho_\ell} - \frac{m_\ell}{\rho_\ell^2} \Delta \rho_\ell \quad (5.3-23)$$

and taking

$$\Delta \rho_\ell = \frac{\partial \rho}{\partial T} \Delta T_\ell \quad (5.3-24)$$

we can write the following expression for the change in the liquid pressure during a time step as

$$\Delta p_\ell = \left( \gamma \frac{p_g}{V_g} + \frac{\rho_\ell g}{A} \right) \left( \frac{\Delta m_\ell}{\rho_\ell} - \frac{m_\ell}{\rho_\ell^2} \frac{\partial \rho}{\partial T} \Delta T_\ell \right) + g(z_i - z_r) \frac{\partial \rho}{\partial T} \Delta T_\ell \quad (5.3-25)$$

Inserting Eq. 5.3-17 for  $\Delta m_\ell$  and Eq. 5.3-19 for  $\Delta T_\ell$  and then comparing with Eq. 5.2-19, we see that the values of the  $b$ 's for a compressible volume with a cover gas are

$$b_0 = 0 \quad (5.3-26)$$

$$b_1 = \Delta t \left\{ \frac{\gamma P_g}{V_g \rho} + \frac{g}{a} + T \frac{\partial \rho}{\partial T} \left[ \frac{g(z_r - z_i)}{m_\ell} + \frac{1}{\rho} \left( \frac{\gamma p_g}{V_g \rho} + \frac{g}{A} \right) \right] \right\} \quad (5.3-27)$$

$$b_2 = \Delta T \frac{\partial \rho}{\partial T} \left[ \frac{g(z_r - z_i)}{m_\ell} - \frac{1}{\rho} \left( \frac{\gamma p_g}{V_g \rho} + \frac{g}{A} \right) \right] \quad (5.3-28)$$

### 5.3.3 Pipes and Intermediate Heat Exchangers

In considering the flow through a pipe or through an intermediate heat exchanger, several factors are taken into consideration. The contribution of the element to  $a_o$  in Eq. 5.2-9 is taken as

$$\Delta a_o = L / A \quad (5.3-29)$$

where

$L$  = the length of the element

$A$  = the flow area of the element

The pressure drop contribution of the element to  $a_1$  in Eq. 5.2-11 is composed of a number of terms. One of these terms is the frictional pressure drop, which is written as

$$\Delta p_{fr} = f \frac{L}{D_h} \frac{\rho v |v|}{2} = f \frac{L}{D_h} \frac{w |w|}{2 \rho A^2} \quad (5.3-30)$$

where

$\Delta p_{fr}$  = the frictional pressure drop

$f$  = the Moody friction factor

$L$  = the length of the element

$D_h$  = the hydraulic diameter of the element

$\rho$  = the density of the liquid

$v$  = the liquid velocity

$w$  = the liquid mass flow rate

$A$  = the flow area of the element

The Moody friction factor  $f$  for turbulent flow in pipes [5-2] is taken as

$$f = C_1 \left[ 1 + \left( C_2 \frac{\varepsilon}{D_h} + \frac{C_3}{\text{Re}} \right)^{C_4} \right] \quad (5.3-31)$$

where

$$C_1 = 0.0055$$

$$C_2 = 20,000$$

$$C_3 = 1.0 \times 10^6$$

$$C_4 = 1/3$$

$\varepsilon$  = the user-supplied roughness of the element

$\text{Re}$  = the Reynolds number

For laminar flow, the friction factor  $f$  is taken as

$$f = 64 / \text{Re} \quad (5.3-32)$$

The Reynolds number in either case is

$$\text{Re} = \frac{D_h |w|}{A\mu} \quad (5.3-33)$$

where  $\mu$  is the viscosity of the fluid.

A second term is the pressure drop caused by bends in the flow path, and these are modeled as an additional frictional drop term, written as

$$\Delta p = f \frac{L_B}{D_B} N_B \frac{w|w|}{2\rho A^2} \quad (5.3-34)$$

where

$L_B/D_B$ =a user-supplied input number for an effective length-to-diameter ratio per bend

$N_B$ =the number of bends in the element

A third term is the pressure drop, also proportional to the square of the mass flow rate, to account for baffles or restrictive orifices which cause greater pressure drops than would be accounted for by friction or bends above. This term is taken as

$$\Delta p = G_2 \frac{w|w|}{2\rho A^2} \quad (5.3-35)$$

where  $G_2$  is a user-supplied input number.

A fourth item is the second term in Eq. 5.2-1, which is proportional to the square of the mass flow rate, and is due to the difference in the fluid densities at the inlet and outlet ends of the element. It has the form

$$\Delta p = w^2 \left( \frac{1}{\rho_o} - \frac{1}{\rho_i} \right) \quad (5.3-36)$$

where

$\rho_o$ =the fluid density at the element outlet

$\rho_i$ =the fluid density at the element inlet

A final term is the gravity-head term for the element, taken as

$$\Delta p = g(z_{out} - z_{in})\bar{\rho} \quad (5.3-37)$$

where

$z_{out}$ =the height of the element outlet

$z_{in}$ =the height of the element inlet

$g$ =the acceleration of gravity

$\bar{\rho}$  =the average of the inlet and outlet fluid densities

Taking the above five terms together, the contribution from the element to  $a_1$  is

$$\begin{aligned} \Delta a_1 = -\Delta t \left[ f \left( \frac{L}{D_b} + \frac{L_B}{D_B} N_B \right) \frac{w|w|}{2\rho A^2} + G_2 \frac{w|w|}{2\rho A^2} \right. \\ \left. + w^2 \left( \frac{1}{\rho_0} + \frac{1}{\rho_1} \right) + g(z_{out} - z_{in}) \bar{\rho} \right] \end{aligned} \quad (5.3-38)$$

The contribution of the element to  $a_2$ , which is the derivative of the pressure drops with respect to time, as shown in Eq. 5.2-14, is zero, since the friction factors and the geometry are assumed not to change independently with time. Hence

$$\Delta a_2 = 0 \quad (5.3-39)$$

The contribution of the element to  $a_3$ , however, which is the derivative of the pressure drops with respect to the mass flow rate, as shown in Eq. 5.2-16, is obtained by differentiation. For laminar flow, differentiating Eq. 5.3-32 gives

$$\frac{\partial f}{\partial w} = -f/w \quad (5.3-40)$$

For turbulent flow, differentiating Eq. 5.3-31 gives

$$\frac{\partial f}{\partial w} = -\frac{C_1 C_3 C_4}{\text{Re}|w|} \left( C_2 \frac{\varepsilon}{D_h} + \frac{C_3}{\text{Re}} \right)^{C_4-1} \quad (5.3-41)$$

Taking the derivative with respect to  $w$  of each term in Eq. 5.3-38, we have for the contribution to  $a_3$

$$\begin{aligned} \Delta a_3 = -\Delta t \left\{ \left( \frac{L}{D_b} + \frac{L_B}{D_B} N_B \right) \frac{1}{2\rho A^2} \left[ 2f|w| + w|w| \frac{\partial f}{\partial w} \right] \right. \\ \left. + G_2 \frac{|w|}{\rho A^2} + 2w \left( \frac{1}{\rho_0} + \frac{1}{\rho_1} \right) \right\} \end{aligned} \quad (5.3-42)$$

### 5.3.4 Pumps

There are three pump choices available in the PRIMAR-4 module: a user-supplied table look-up, a centrifugal pump, and an electromagnetic pump. There are a number of options for the centrifugal and electromagnetic pumps.

### 5.3.4.1 Table Look-Up

The user supplies a table of relative pump head vs. time, and the code interpolates linearly between points in the table. The pump head as a function of time is given by

$$H(t) = H_r f(t) \quad (5.3-43)$$

where  $f(t)$  is the user-supplied table, with  $f(t = 0) = 1.0$ , and  $H_r$  is the reference pump head for the particular pump. The contribution to the coefficients in Eqs. 5.2-10, 5.2-12, 5.2-14, and 5.2-16 are

$$\Delta a_0 = L / A \quad (5.3-44)$$

$$\Delta a_1 = \Delta t [H(t_3) - GH] \quad (5.3-45)$$

$$\Delta a_2 = \Delta t [H(t_4) - H(t_3)] \quad (5.3-46)$$

$$\Delta a_3 = 0 \quad (5.3-47)$$

where

$L$  = the length of the pump element

$A$  = the element flow area

$\Delta t$  = the time interval

$t_3, t_4$  = the times at the beginning and end of the time interval

$GH$  = the gravity head for the pump element

### 5.3.4.2 Centrifugal Pumps

#### 5.3.4.2.1 Option 1

In this option, the pump head and the pump torque are represented either by constant values or by polynomial fits, depending upon the pump flow rates or upon the pump speeds. The flow rates and speeds, and also the pump head and torque, are expressed as multiples of rated quantities, which are user-supplied. The constant values and the coefficients in the polynomial fits are also user-supplied.

The pump head and torque are expressed as functions of the parameter  $\chi$ , which is

$$\chi = \bar{w} / \bar{s} \quad (5.3-48)$$

with

$$\bar{w} = w / w_R \quad (5.3-49)$$

$$\bar{s} = s / s_R \quad (5.3-50)$$

where

$w$  = the pump flow rate

$w_R$  = the rated pump flow rate

$s$  = the pump speed

$s_R$  = the rated pump speed

### Pump Head

For positive pump speeds the pump head is taken as

$$\Delta p_p = H_R s^2 (A_1 + A_2 \chi + A_3 \chi^2 + A_4 \chi^3 + A_5 \chi^4), \quad \text{if } \chi < A_6 \quad (5.3-51)$$

$$\Delta p_p = H_R A_7 \bar{w}^2, \quad \text{if } |\chi| > A_6 \text{ and } \bar{w} \geq A_{18} \quad (5.3-52)$$

$$\Delta p_p = -H_R A_8 \bar{w}^2, \quad \text{if } |\chi| > A_6 \text{ and } \bar{w} < -A_{18} \quad (5.3-53)$$

$$\Delta p_p = H_R A_7 \bar{w} A_{18} \quad \text{if } |\chi| > A_6 \text{ and } A_{18} > \bar{w} \geq 0 \quad (5.3-54)$$

$$\Delta p_p = -H_R A_8 \bar{w} A_{18} \quad \text{if } |\chi| > A_6 \text{ and } 0 > \bar{w} \geq -A_{18} \quad (5.3-55)$$

where

$\Delta p_p$  = the pump head

$H_R$  = the reference pump head

$A_1, \dots, A_8$  = user-supplied coefficients

For negative pump speeds, or flow reversal, the pump head is taken as

$$\begin{aligned}\Delta p_p &= H_R A_7 d_1^2 \text{ if } d_1 \geq 0 \\ &= H_R A_8 d_1^2 \text{ if } d_1 < 0\end{aligned}\tag{5.3-56}$$

with

$$d_1 = \bar{w} - A_{10} \bar{s},$$

where

$A_{10}$  = user-supplied coefficient

### Pump Torque

Next, for positive pump speeds, the pump torque is

$$\bar{\tau} = \bar{s}^2 (A_{11} + A_{12}\chi + A_{13}\chi^2 + A_{14}\chi^3 + A_{15}\chi^4), \text{ if } \left| \frac{\bar{w}}{\bar{s}} \right| \leq A_6\tag{5.3-57}$$

$$\bar{\tau} = A_{16} \bar{w}^2, \text{ if } \left| \frac{\bar{w}}{\bar{s}} \right| > A_6, \bar{w} \geq 0\tag{5.3-58}$$

$$\bar{\tau} = A_{17} \bar{w}^2, \text{ if } \left| \frac{\bar{w}}{\bar{s}} \right| > A_6, \bar{w} < 0\tag{5.3-59}$$

with

$$\bar{\tau} = \frac{\tau}{\tau_R}\tag{5.3-60}$$

where

$\tau$ =the pump torque

$\tau_R$ =the reference pump torque

$\chi, \bar{w}, \bar{s}$ =the same as for the pump head

$A_{11}, \dots, A_{15}$ =user-supplied coefficient

For negative pump speeds, the pump torque is taken as

$$\bar{\tau} = A_{16} (\bar{w} - A_{20} \bar{s})^2, \text{ if } \bar{w} \geq A_{20}\tag{5.3-61}$$

$$= A_{17} (\bar{w} - A_{20} \bar{s})^2, \quad \text{if } \bar{w} < A_{20} \quad (5.3-62)$$

### Pump Speed

The pump speed and pump torque are related by the equation is

$$I \frac{ds}{dt} = \tau_m - \tau_p - k_1 w \quad (5.3-63)$$

where

$I$  = the moment of inertia of the pump and motor

$s$  = the pump speed

$\tau_m$  = the motor torque

$\tau_p$  = the pump torque

$k_1$  = constant of proportionality

$w$  = the pump flow rate

The term  $-k_1 w$  is a drag term proportional to the flow rate. The motor torque  $\tau_m$  is a user-supplied table look-up.

### Rotor Lock-Up

$$\Delta a_o = L_p / A_p \quad (5.3-64)$$

$$\Delta a_1 = \Delta p_p - \rho_l g (z_{in} - z_{out}) \quad (5.3-65)$$

$$\begin{aligned} \Delta a_2 &= (\Delta t)^2 \frac{\partial}{\partial t} \Delta p_p \\ &= (\Delta t)^2 \frac{\partial \Delta p_p}{\partial s} \frac{\partial s}{\partial t} \end{aligned} \quad (5.3-66)$$

$$\Delta a_3 = \Delta t \frac{\partial}{\partial w} \Delta p_p \quad (5.3-67)$$

If the normalized pump speed,  $\bar{s}$ , falls below the value  $A_9$ , and if  $A_9 > 0$ , then the rotor locks; and the speed is held at zero after that.

### Flow Rate Coefficients

The contribution to the  $a$ 's in Eqs. 5.2-10, 5.2-12, 5.2-14 and 5.2-16 are then as follows:

$L_p$ =the length of the pump element

$A_p$ =the flow area of the pump element

$\Delta p_p$ =the pump head

$\rho_\ell$ =the liquid density

$g$ =the acceleration of gravity

$z_{in}$ =the inlet elevation of the pump element

$z_{out}$ =the outlet elevation of the pump element

$\frac{\partial \Delta p_p}{\partial t}$ ,  $\frac{\partial s}{\partial t}$ ,  $\frac{\partial \Delta p_p}{\partial w}$  =the partial derivatives of pump head and pump speed described above under the various conditions of speed and flow

#### 5.3.4.2.2 Option 2

This pump option makes use of the homologous pump curves and should permit operation of the pump in all four quadrants of pump operations--normal pump mode, energy dissipation mode, turbine mode, and reverse pump mode. The actual curves used for the pump head and torque are, at present, built into the model using data statements and are the three-region curve fit taken from a memo by J. F. Koenig [5-1].

The pump head and torque values are obtained from

$$H = H_r (\bar{w}^2 + \bar{s}^2) \sum_{i=0}^6 C_{ji} x^i \quad (5.3-68)$$

$$T = T_r (\bar{w}^2 + \bar{s}^2) \sum_{i=0}^8 D_{ji} x^i \quad (5.3-69)$$

where

$H$ = pump head

$T$ =pump torque

$w$  = flow rate

$s$  = pump speed

$x = \pi + \tan^{-1} (\bar{w} / \bar{s})$

and the bar denotes normalization with respect to reference quantities (i.e.,  $\bar{w} = w/w_r$ ) and the subscript  $r$  denotes reference quantities. The coefficients  $C_{ji}$  and  $D_{ji}$  represent the three-region curve fit with

$$\begin{aligned} j &= 1, & 0 \leq x \leq 3.14159 \\ &= 2, & 3.14159 < x \leq 4.7124 \\ &= 3, & 4.7124 < x \end{aligned}$$

The value for  $H_r$  is an input while  $T_r$  is obtained from

$$T_r = H_r w_r / (s_r \eta \rho) \quad (5.3-70)$$

where  $\eta$  is the pump efficiency at reference conditions and  $\rho$  is the fluid density.

At low flow conditions with  $|\bar{w}| < w_c$  where  $w_c$  represents a cutoff value specified by the user, the pump head is calculated as a linear interpolation between the pump head at zero flow and at the  $w_c$  flow:

$$H(w, s) = (H(\bar{w}_c, \bar{s}) \bar{w} + H(0, \bar{s}) (w_c - \bar{w})) / w_c \quad (5.3-71)$$

The value  $w_c$  is specified as an input value in APMPHD (13,IPMP) in input block 18. The frictional torque losses,  $T_f$ , are taken as a three-region quadratic fit over  $\bar{s}$

$$T_f = T_r (A_{j1} + A_{j2} \bar{s} + A_{j3} \bar{s} |\bar{s}|) \quad (5.3-72)$$

where

$$\begin{aligned} j &= 1, & \bar{s} < s_1 \\ &= 2, & s_1 < \bar{s} < s_2 \\ &= 3, & s_2 < \bar{s}. \end{aligned}$$

The nine coefficients,  $A_{ji}$ , are specified as input values in APMPHD (L,IPMP) in input block 18. The cutoff speeds  $s_1$  and  $s_2$  are also input values in APMPHD (L, IPMP).

For the steady-state, the subroutine SSPUMP makes use of a root finder to locate the pump speed necessary for a specified pump head. It is assumed that steady-state will be at a flow level above  $w_c$ .

The subroutine PUMPFL includes the following calculations for  $\Delta a_0$ ,  $\Delta a_1$ ,  $\Delta a_2$  and  $\Delta a_3$ , with

$$\Delta a_0 = L_K / A_K \quad (5.3-73)$$

$$\Delta a_1 = Hg + H \quad (5.3-74)$$

$$\Delta a_2 = \Delta t^2 \frac{\partial}{\partial t} H \quad (5.3-75)$$

$$\Delta a_3 = \Delta t \frac{\partial}{\partial w} H \quad (5.3-76)$$

where  $L$ ,  $A$ ,  $H_g$ ,  $H$  and  $\Delta t$  represent the length, area, gravity head, pump head and time interval, respectively. The terms  $\Delta a_0$  and  $\Delta a_1$  are calculated as in the centrifugal pump option 1;  $\Delta a_2$  is found as follows:

$$\Delta a_2 = \Delta t^2 \frac{\partial \bar{s}}{\partial t} \frac{\partial H}{\partial s} \quad (5.3-77)$$

where

$$\frac{\partial \bar{s}}{\partial t} = (T_m - T_p - T_f) / (s_r I) \quad (5.3-78)$$

$T_m$  = motor torque

$I$  = pump inertia

and from Eqs. 5.3-67 and 5.3-70

$$\frac{\partial H}{\partial \bar{s}} = 2H_r \bar{s} \sum_{i=0}^6 c_{ji} x^i - H_r \bar{w} \sum_{i=1}^6 c_{ji} i x^{i-1} \quad (5.3-79)$$

for  $|\bar{w}| > w_c$  and

$$\frac{\partial H}{\partial \bar{s}} = \left[ \frac{\partial H}{\partial \bar{s}}(w_c, \bar{s}) \bar{w} + \frac{\partial H(0, \bar{s})}{\partial \bar{s}} (w_c - \bar{w}) \right] / w_c \quad (5.3-80)$$

for  $|\bar{w}| < w_c$ .

$\Delta a_3$  is found using 5.3-76 and

$$\frac{\partial H}{\partial \bar{w}} = 2H_r \bar{w} \sum_{i=0}^6 c_{ji} x^i + H_r \bar{s} \sum_{i=1}^6 C_{ji} i x^{i-1} \quad (5.3-81)$$

$$\frac{\partial H}{\partial w} = \frac{1}{w_r} \frac{\partial H}{\partial \bar{w}}$$

for  $|\bar{w}| > w_c$  and from

$$\frac{\partial H}{\partial \bar{w}} = (H(w_c, \bar{s}) - H(0, \bar{s})) / w_c \quad (5.3-82)$$

for  $|\bar{w}| < w_c$ , and

The subroutine PUMPFN includes the pump torque curves in integrating Eq. 5.3-78 to determine  $s$ . Approximating Eq. 5.3-78 as

$$I \frac{\Delta s}{\Delta t} = T_m - 1/2(T_3 + T_4) - 1/2(T_{f3} + T_{f4}) \quad (5.3-83)$$

Note where the subscripts 3 and 4 denote the beginning and ending times of the  $\Delta t$  interval and approximating

$$T_4 = T_3 + \frac{\partial T}{\partial s} \Delta s + \frac{\partial T}{\partial w} \Delta w \quad (5.3-84)$$

$$T_{f4} = T_{f3} + \frac{\partial T_f}{\partial s} \Delta s \quad (5.3-85)$$

in Eq. 5.3-83,  $\Delta s$  may be obtained as

$$\Delta s = \frac{T_m - T_3 - 1/2 \frac{\partial T}{\partial w} \Delta w - T_{f3}}{\left[ I / \Delta t + 1/2 \frac{\partial T}{\partial s} + 1/2 \frac{\partial T_f}{\partial s} \right]} \quad (5.3-86)$$

#### 5.3.4.2.2.1 Locked Rotor

If  $\Delta s < A_{20}$  where  $A_{20}$  is the input variable APMPHD (20, IPMP), and if  $\bar{w} < A_{19}$  then the rotor locks and the pump speed is set to zero and held at zero. For a locked rotor the pump head is calculated as

$$H = \begin{cases} -T_r A_{12} w & |w| \geq A_{13} \\ -T_r A_{12} A_{13} \bar{w} & \text{if } |\bar{w}| < A_{13} \end{cases} \quad (5.3-86b)$$

where  $A_{12}$  and  $A_{13}$  are input in the APMPHD(K, IPMP) array.

#### 5.3.4.2.2.2 Pump Cavitation

In a centrifugal pump, cavitation will occur if the applied net positive suction head (ANPSH), or pump inlet pressure, is less than the required net positive suction head (RNPSH). This can occur if there is a break in the outlet pipe from the pump, leading to an increase in the pump flow rate and an increase in the RNPSH. If pump cavitation occurs, there will be a drop in pump head. Although there is reasonably good RNPSH data for many pumps, there is little data on pump behavior in the cavitation range, mainly because cavitation tends to destroy a pump, so no one wants to operate in that range. On the other hand, transient pump cavitation can occur for a short time after a pipe rupture, and the resulting decrease in pump head can have a significant impact on the flow out the rupture and on the flow rate through the core. The homologous pump model in PRIMAR-4 tests for pump cavitation and reduces the pump head if cavitation occurs.

The required net positive suction head is calculated as

$$RNPSH = \bar{s}^2 (C_{c0} + C_{c1} Z + C_{c2} Z^2 + C_{c3} Z^3 + C_{c4} Z^4) \quad (5.3-87)$$

where

$$Z = \bar{w} / \bar{s} \quad (5.3-88)$$

Then, if  $RNPSH > ANPSH$ , the cavitated head,  $H_c$ , is calculated as

$$H_c = f_c H \quad (5.3-89)$$

where  $H$  is the normal head, calculated as described in the previous sections, and the cavitation factor  $f_c$  is calculated as

$$f_c = \frac{1}{1 + 1000x^2} \quad (5.3-90)$$

where

$$x = \frac{RNPSH - ANPSH}{ANPSH} \quad (5.3-91)$$

The cavitation factor of Eq. 5.3-90 is somewhat arbitrary; but, as previously mentioned, there is little data for behavior in the cavitation range, and this expression results in a rapid drop in pump head as the pump goes farther into the cavitation range. With this treatment, the expressions used in Eq. 5.3-76 and 5.3-77 for  $\Delta a_2$  and  $\Delta a_3$  are

$$\frac{\partial H_c}{\partial \bar{s}} = f_c \left[ \frac{\partial H}{\partial \bar{s}} - 2000 xH \frac{\partial x}{\partial \bar{s}} \right] \quad (5.3-92)$$

$$\frac{\partial H_c}{\partial \bar{w}} = f_c \left[ \frac{\partial H}{\partial \bar{w}} - 2000 xH \frac{\partial x}{\partial \bar{w}} \right] \quad (5.3-93)$$

#### 5.3.4.2.3 EBR-II Pump Model

The third centrifugal pump option in PRIMAR-4 is essentially the pump model used in the NATDEMO code [5-5] for EBR-II. By adjusting the coefficients in the pump characteristics curves, this model could probably also be used for other reactors. This model is somewhat simpler than the option 2 model, in that the EBR-II pump model is mainly applicable to the first quadrant (positive flow, positive pump speed). This model can be used for negative flow, but it is probably not applicable to negative pump speed. Also, for the EBR-II pump model, the user specifies the pump speed as a function of time; whereas in options 1 and 2, the code calculated the pump speed.

In this option, the pump head is calculated from

$$\bar{H} = b_1 \bar{s}^2 + b_2 \bar{s} \bar{w} + b_3 \frac{\bar{w}}{|\bar{w}|} |\bar{w}|^{b_4} \quad (5.3-94)$$

where

$$\bar{H} = H / H_r \quad (5.3-95)$$

$$\bar{s} = s / s_r \quad (5.3-96)$$

$$\bar{w} = w / w_r \quad (5.3-97)$$

$H$ =pump head

$s$ =pump speed

and

$w$ = coolant flow rate

The  $r$  subscript refers to the rated value. The coefficients  $b_1$  and  $b_2$  are constant, but the values of  $b_3$  and  $b_4$  depend on the speed and flow rate:

$$b_3 = b_{3m} \text{ if } \bar{s} > \varepsilon_m \bar{w} \quad (5.3-98a)$$

$$b_3 = b_{3t} \text{ if } \bar{s} \leq \varepsilon_m \bar{w} \text{ and } w \geq \bar{w}_t \quad (5.3-98b)$$

$$b_3 = b_{3\ell} \text{ if } \bar{s} \leq \varepsilon_m \bar{w} \text{ and } w < \bar{w}_t \quad (5.3-98c)$$

$$b_4 = b_{4t} \text{ if } \bar{s} > \varepsilon_m \bar{w} \text{ or } w \geq \bar{w}_t \quad (5.3-99a)$$

$$b_4 = b_{4\ell} \text{ if } \bar{s} \leq \varepsilon_m \bar{w} \text{ and } w < \bar{w}_t \quad (5.3-99b)$$

Equations 5.3-98 and 5.3-99 are based on the idea of using a stopped-rotor pressure drop if  $\bar{s} < \varepsilon_m \bar{w}$ . Also, the stopped-rotor pressure drop is a laminar value if  $\bar{w} < \bar{w}_t$ , or a turbulent value if  $\bar{w} \geq \bar{w}_t$ . The default values for the coefficients are:

$$b_1 = 1.1740 \quad (5.3-100a)$$

$$b_2 = .0818 \quad (5.3-100b)$$

$$b_{3m} = -.2558 \quad (5.3-100c)$$

$$b_{3t} = -.5923 \quad (5.3-100d)$$

$$b_{3\ell} = -.0471 \quad (5.3-100e)$$

$$b_{4t} = 1.9 \quad (5.3-100f)$$

$$b_{4\ell} = 1.0 \quad (5.3-100g)$$

$$\bar{w}_t = .06 \quad (5.3-100h)$$

$$\varepsilon_m = .55 \quad (5.3-100i)$$

$$H_r = 358,530 \text{ Pa} \quad (5.3-100j)$$

$$w_r = 250.2 \text{ kg/s} \quad (5.3-100k)$$

and

$$s_r = 14.5 \text{ revolutions / s} = 870 \text{ rpm} \quad (5.3-100l)$$

This correlation produces a discontinuity in pump head at the switch from spinning rotor to stopped rotor. With the default values, the pump head is continuous at the transition from turbulent to laminar flow.

For the steady-state initialization in SSPUMP, the initial normalized pump head,  $\bar{H}_0$  and flow rate,  $\bar{w}_0$ , are known, and Eq. 5.3-94 is solved to find  $\bar{s}_0$ , the initial steady-state pump speed:

$$\bar{s}_0 = \frac{-b_2 \bar{w}_0 + \left[ b_2^2 \bar{w}_0^2 - 4b_1 (b_3 \bar{w}_0^{b_4} - \bar{H}_0) \right]^{1/2}}{2b_1} \quad (5.3-101)$$

It is assumed that  $\bar{s}_0 > \varepsilon_m \bar{w}_0$ , and the values used for  $b_3$  and  $b_4$  in SSPUMP are  $b_{3m} b_{4t}$ . The user supplies a table of normalized pump speed vs. time, normalized to the initial steady-state speed. The code then interpolates linearly from this table to get pump speeds for the transient calculation.

In PUMPFL where  $\Delta a_0$ ,  $\Delta a_1$ ,  $\Delta a_2$ , and  $\Delta a_3$  are calculated,  $\Delta a_0$  and  $\Delta a_1$ , are calculated using Eq. 5.3-64 and 5.3-65 or 5.3-73 and 5.3-74, as in the other pump options. For this option

$$\Delta a_2 = \Delta t^2 \frac{dH}{ds} \frac{ds}{dt} \quad (5.3-102)$$

with

$$\frac{dH}{ds} = \frac{(2b_1 \bar{s} + b_2 \bar{w}) H_r}{S_r} \quad (5.3-103)$$

and  $ds/dt$  obtained from the user-specified table of normalized pump speed vs. time:

$$\frac{ds}{dt} = \frac{s(t + \Delta t) - s(t)}{\Delta t} \quad (5.3-104)$$

also,

$$\Delta a_3 = \Delta t \frac{dH}{dw} = \Delta t H_r \left[ b_2 \bar{s} + b_3 b_4 \left( \frac{\bar{w}}{|w|} \right) |w|^{b_4-1} \right] \quad (5.3-105)$$

#### 5.3.4.2.4 Pump Head vs. Flow Option

One pump option in SASSYS-1 allows the user to specify the normalized pump head as a function only of normalized flow rate. In this option, the normalization is to the initial steady-state values. The user supplies a table of normalized pump head vs. normalized flow rate, and the code uses linear interpolation between the entries in the table.

For this option,  $\Delta a_0$  and  $\Delta a_1$  for the pump are calculated in the same way as for the other pump options. Then

$$\Delta a_2 = 0 \quad (5.3-106)$$

and

$$\Delta a_3 = \Delta t \frac{dH}{dW} \quad (5.3-107)$$

with  $dH/dw$  being evaluated numerically from the table of  $\bar{H}$  vs.  $\bar{w}$ :

$$\frac{dH}{dw} = \frac{H(w + .01 w_0) - H(w)}{.01 w_0} \quad (5.3-108)$$

where  $w_0$  is the initial steady state flow rate,  $w$  is the current flow rate, and  $H$  is the pump head.

#### 5.3.4.3 Electromagnetic Pumps

The electromagnetic pump is modeled as a pipe element. The expression for the pump head for the electromagnetic pump is

$$\Delta p_p = H_s(t) \left( 1 - v_{Na} / v_{sy} \right) \quad (5.3-109)$$

with

$$v_{Na} = \frac{w}{\rho A} \quad (5.3-110)$$

where

$\Delta p_p$  = the pump head

$H_s(t)$  = the pump stall head

$v_{Na}$  = the sodium velocity at the beginning of the time interval

$w$  = the pipe flow rate

$A$  = the pipe area

$\rho$  = the sodium density

$v_{sy}$  = user-supplied synchronous magnetic field velocity

The quantity

$$H_s(t) / H_s(o) \quad (5.3-111)$$

is a user-supplied table of normalized stall head as a function of time.

The contributions to the  $a$ 's in Eqs. 5.2-10 through 5.2-16 for the electromagnetic pump are

$$\Delta a_o = L_p / A_p \quad (5.3-112)$$

$$\Delta a_1 = \Delta t \Delta p_p(t_3) \quad (5.3-113)$$

$$\Delta a_2 = \Delta t [H_s(t_4) - H_s(t_3)] (1 - v_{Na} / v_{sy}) \quad (5.3-114)$$

$$\Delta a_3 = -\Delta t H_s(t_3) v_{Na} \quad (5.3-115)$$

where

$L_p$  = the length of the pump element

$A_p$  = the flow area of the pump element

$\Delta p_p(t_3)$  = the pump head evaluated at the beginning of the time interval

$\Delta t$ =the time interval

$H_s(t_3)$ =the stall head at the beginning of the time interval

$H_s(t_4)$ =the stall head at the end of the time interval

#### 5.3.4.4 LMR EM Pump - Motor Option

A current LMR design contains motor-generators on the primary pumps to extend the coastdown times of these EM pumps. Since this combination of EM pumps and motor-generators can not be modeled well with previous SASSYS-1 pump options, a new pump option was included in SASSYS-1 to handle this configuration. The new option uses a model similar to that used by General Electric in their ARIES-P code. It is based on head and efficiency data obtained by GE and transmitted to ANL (5-29).

In the current LMR design a synchronous motor is running all of the time during normal operation, but the power to the EM pump does not go through the motor as long as the normal pump power is available. If normal pump power is lost, then the motor becomes a generator, and a switch is thrown automatically to supply voltage from the motor-generator to the pump. The pump coastdown rate is then determined by the inertia of the motor. The motor is designed such that it will initially supply 60% of nominal voltage to the pump. Thus, when normal power is lost and the motor-generator power is switched on there is a sudden drop in pump head and flow followed by a gradual coastdown.

#### Pump Head

The pump head is correlated with an expression of the form

$$\bar{H} = (\bar{V} / \bar{f})^{3.5} h_n (\bar{W} / \bar{f}) - L_f \bar{w}^2 \quad (5.3-116)$$

where

$$\bar{H} = H / H_r \quad (5.3-117)$$

$$\bar{V} = -V / V_r \quad (5.3-118)$$

$$\bar{f} = f / f_r \quad (5.3-119)$$

$$\bar{w} = w / w_r \quad (5.3-120)$$

$H$ =pump head

$H_r$ =rated head

$V$ =pump voltage

$V_r$ =rated voltage

$f$ =frequency

$f_r$ =rated frequency

$w$ =mass flow rate

$w_r$ =rated mass flow rate

$L_f$ =friction loss coefficient

and  $h_n$  is a head curve correlated as

$$h_n \left( \bar{w} / \bar{f} \right) = \sum_{j=1}^5 a_j \left( \bar{w} / \bar{f} \right)^{j-1} \quad (5.3-121)$$

with the coefficients  $a_j$  determined by a least-squares fit to the data.

### Pump Efficiency

The pump efficiency,  $\varepsilon_f$ , is correlated as

$$\varepsilon_f = F(\bar{V}) G(\bar{w} / \bar{f}) \varepsilon_{fr} \quad (5.3-122)$$

with

$$F(\bar{V}) = \sum_{j=1}^7 b_j \bar{V}^{j-1} \quad (5.3-123)$$

$$G(\bar{w} / \bar{f}) = \begin{cases} .01 & \text{if } \bar{w} / \bar{f} \geq 5 \\ \sum_{j=1}^9 c_j (\bar{w} / \bar{f})^{j-1} & \text{if } \bar{w} / \bar{f} \leq 5 \end{cases} \quad (5.3-124)$$

and  $\varepsilon_{fr}$  = rated efficiency

### Pump Voltage

Before the cut-over to the motor-generator, the pump voltage is assumed to be constant at its rated value. Also, the frequency is constant at its rated value. Immediately after cut-over, the voltage drops to a fraction,  $V_{fr}$ , of its initial value. Then the voltage is proportional to the square of the frequency, so after cut-over the voltage is

$$\bar{V} = V_{fr} \bar{f}^2 \quad (5.3-125)$$

where

$$V_{fr} = 0.6 \quad (5.3-126)$$

### Motor Speed

The equation for the motor speed,  $s$ , is

$$\frac{ds}{dt} = -\frac{(\tau_p + \tau_\ell)}{I} \quad (5.3-127)$$

where

$\tau_p$ =pump torque

$\tau_\ell$ =friction loss

$I$ =moment of inertia

Note that the motor speed and the pump frequency are the same:

$$f = s \quad (5.3-128)$$

The pump torque is given by

$$\tau_p = \frac{Hw}{\tau_f \rho s} \quad (5.3-129)$$

where

$\rho$ =liquid density

The friction loss in the motor is assumed to have the form

$$\tau_\ell = \tau_r L_m s \quad (5.3-130)$$

where  $L_m$  is a loss coefficient and  $\tau_r$ , the rated torque is given by equation 14 with rated values used for all terms.

### Correlations to Pump Data

Reference 5-29 transmitted the data taken for the pump. This data has been fit by a least-squares fitting program to give the parameters listed in Table 5.3-1 for use in equations 5.3-121, 5.3-123, and 5.3-124. Also a value of .07592 is used for  $L_f$  in

equation 5.3-116. Table 5.3-2 lists the data from Ref. 5-29, as well as the head and efficiency values calculated using the correlations of equations 5.3-116, 5.3-121, 5.3-122, and 5.3-124 with the coefficients in Table 5.3-1. Also, some of this data is plotted in Figures 5.3-1 and 5.3-2. It can be seen that the correlations match the data well except for a couple of points.

Table 5.3-1. Correlation Coefficients for Use with the LMR Pumps

<b>j</b>	<b>a<sub>j</sub></b>	<b>b<sub>j</sub></b>	<b>c<sub>j</sub></b>
1	1.133	-.148	0.
2	.996	7.110	-51.235
3	-2.498	-15.972	684.934
4	6.056	9.942	-3483.628
5	-4.611	12.024	9119.690
6	-	-18.536	-13449.761
7	-	6.577	11279.948
8	-	-	-5014.503
9	-	-	915.555

Table 5.3-2. LMR Pump Head and Efficiency

<b>Normalized Voltage</b>	<b>Normalized Frequency</b>	<b>Normalized Flow</b>	<b>Normalized Measured</b>	<b>Head Fit</b>	<b>Efficiency</b>	
					Measured	Fit
1.000	1.000	1.148	0.042	0.041	0.115	0.133
1.000	1.000	1.124	0.250	0.242	0.505	0.469
1.000	1.000	1.086	0.500	0.524	0.781	0.807
1.000	1.000	1.000	1.000	1.000	1.000	0.997
1.000	1.000	0.800	1.483	1.494	0.928	0.915
1.000	1.000	0.600	1.508	1.514	0.710	0.709
1.000	1.000	0.400	1.399	1.389	0.469	0.468
1.000	1.000	0.200	1.265	1.270	0.229	0.229
1.000	1.000	0.0	1.134	1.133	0.0	0.0
0.627	0.778	0.886	0.042	0.042	0.183	0.256
0.627	0.778	0.862	0.167	0.158	0.551	0.600
0.627	0.778	0.824	0.333	0.314	0.817	0.879
0.627	0.778	0.762	0.581	0.500	0.970	0.932
0.628	0.778	0.762	0.581	0.505	0.970	0.932

---

0.312	0.472	0.532	0.042	0.052	0.318	0.381
0.312	0.472	0.520	0.083	0.097	0.529	0.608
0.312	0.472	0.495	0.167	0.178	0.773	0.848
0.312	0.472	0.476	0.227	0.226	0.863	0.884
1.202	1.111	1.143	1.306	1.144	1.008	1.009
0.929	0.944	0.952	0.905	0.912	0.998	1.003
0.820	0.833	0.857	0.734	0.835	0.986	0.981
0.519	0.667	0.667	0.444	0.415	0.954	0.932
0.437	0.556	0.571	0.327	0.383	0.917	0.920
0.251	0.500	0.476	0.227	0.096	0.855	0.789
0.219	0.389	0.381	0.145	0.143	0.757	0.757
0.153	0.278	0.286	0.082	0.111	0.567	0.602
0.071	0.222	0.190	0.037	0.025	0.268	0.255
0.027	0.111	0.095	0.009	0.010	0.026	0.031

---

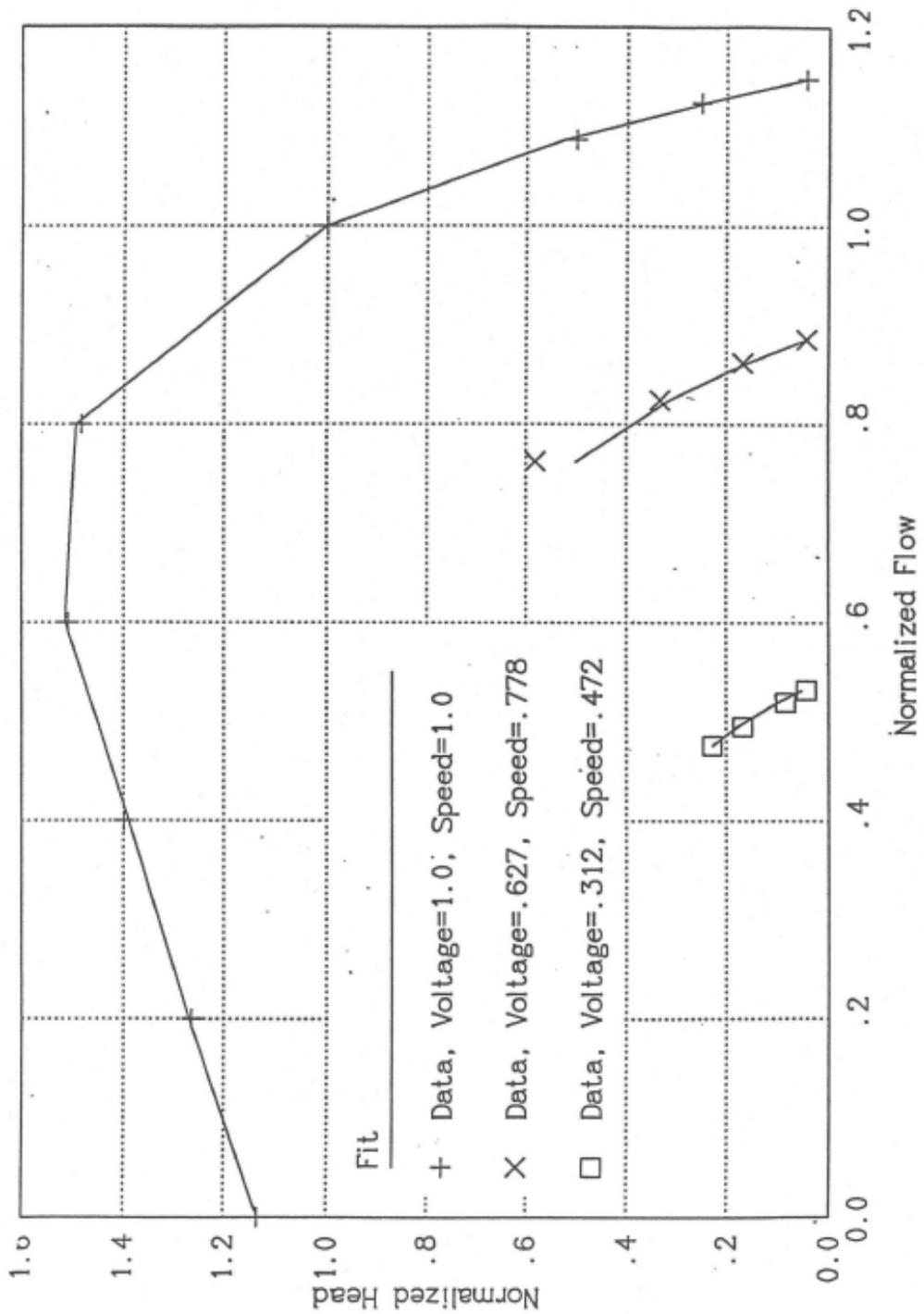


Figure 5.3-1. Pump Head, Data and Fits

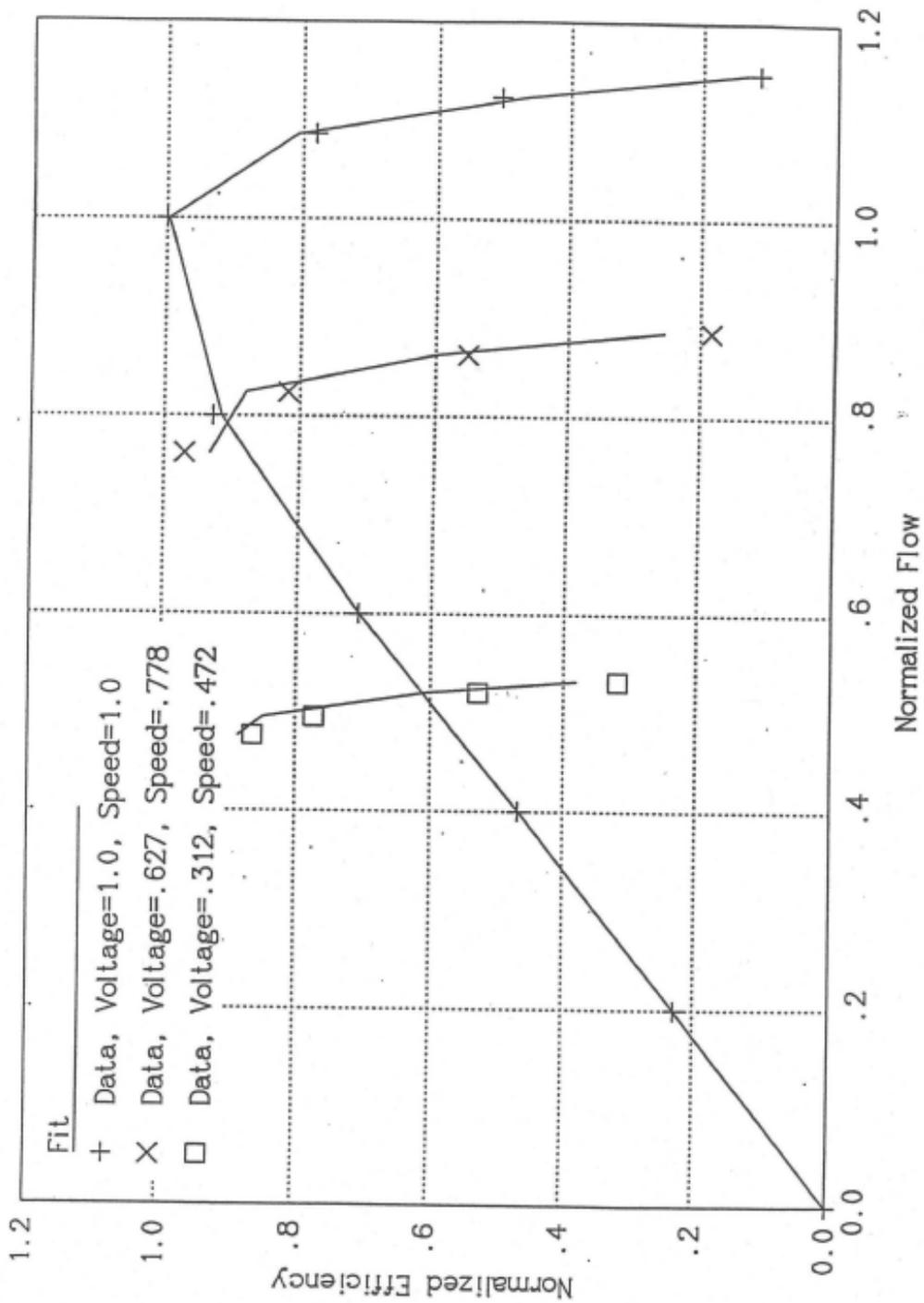


Figure 5.3-2. Pump Efficiency, Data and Fits

### 5.3.5 Valves

SASSYS-1 contains a simple valve treatment in which the user can specify the valve pressure loss coefficient as a function of time. The pressure drop,  $\Delta p_v$ , through a valve is calculated as

$$\Delta p_r(t) = G_2(t) \frac{w|w|}{2\rho A^2} \quad (5.3-131)$$

where

$w$  = coolant flow rate through the valve

$\rho$  = coolant density

$A$  = nominal valve flow area

$G_2$  = valve coefficient

The user supplies a table of  $G_2$  as a function of time. The table is used to obtain an average value of  $G_2$  for each step of the transient. With this treatment, a valve is never entirely shut; although  $G_2$  can be made very large so that the flow through the valve becomes very small.

Some care must be taken in setting the input for a valve. The table of  $G_2$  vs.  $t$  goes in the DTMPBTB and TMPMTB input arrays, but in addition  $G_2(t=0)$  goes in the input variable G2PRDR (ELL) for the element IELL corresponding to the valve. During the steady-state initialization, G2PRDR may be adjusted by the code if necessary to achieve a steady-state pressure balance. At the end of the steady-state initialization, if G2PRDR does not match the table entry for  $t=0$ , then a constant value is added to all of the entries in the DTMPBTB table for the valve so that the table does match G2PRDR at  $t=0$ . Up to eight valves can be used. Each valve can use its own separate valve coefficient table. It is also possible to use the same table for two or more valves but this option must be used with care. All valves using the same table should have the same steady-state valve coefficients and the same time dependence. Also, if a valve table is adjusted to achieve a steady-state pressure balance for one valve, the adjustment may not be appropriate for other valves using the same table.

### 5.3.6 Check Valves

SASSYS-1 contains a simple check valve treatment in which the user can specify the valve orifice coefficient as a function of coolant flow rate. Every time step an orifice coefficient ( $G_2$  in Eq. 5.3-35) is re-set using a user-supplied table of orifice coefficient vs. normalized flow rate, with a flow rate normalized to its steady-state value. Other than the variable orifice coefficient, a check valve is treated the same as a pipe, as described in section 5.3.3.

## 5.4 Liquid Temperature Calculations

In Section 5.2, we described how the liquid pressures and liquid mass flow rates are calculated during a time step, ignoring liquid temperature changes and gas mass flow rate changes during the same time step. In this section, we shall describe how liquid temperatures are calculated during a time step while still ignoring gas mass flow rate changes during the time step.

Currently PRIMAR-4 contains liquid temperature calculations for a number of types of components. These include pipes, IHXs, steam generators, bypass channels, a general one-node perfect mixing model for compressible volumes, and a thermal stratification model for the outlet plenum. For the IHX two models are available: a moderately detailed model of both the tube and the shell sides, and a simple table look-up model for the primary side only. Similarly there are two steam generator models available: a moderately detailed model for both the sodium and the water sides, and a simple table look-up model for the sodium side only. Currently, valves, check valves, and pump impellers are treated as pipes when liquid temperatures are calculated.

For temperature calculations, the elements in a liquid segment are combined into temperature groups. Each temperature group contains one or more consecutive elements. All of the elements in the temperature group are treated with the same type of liquid temperature calculation. When the moderately detailed IHX model is used, the tube-side element and the shell-side element are in the same temperature group, since temperatures in both sides are computed at the same time. Also, the Lagrangian calculations used for pipe temperatures are more efficient and accurate if a number of connected pipe segments are strung together into a single temperature group, rather than calculating temperatures for each pipe segment separately.

### 5.4.1 Pipe Temperatures

The pipe temperature model is a slug flow model with heat transfer to the pipe walls, as indicated in Fig. 5.4-1. The coolant in a pipe is divided into a number of moving nodes or slugs. The node boundaries move with the coolant flow. All nodes in a pipe have equal volumes except for the first and last nodes. The inlet node size starts at zero and grows as the flow continues until it reaches the size of the other nodes. At that point a new node is started at the inlet. Similarly, the outlet node shrinks and eventually is removed when its volume reaches zero. The temperature in a coolant node changes only due to heat transfer to the pipe wall. There is one wall node for each coolant node. One radial node is used in the pipe wall. Heat transfer from the outside of the pipe wall is described in Section 5.4.7 on component-to-component heat transfer. Wall nodes do not move, so the wall node in contact with a given coolant node changes periodically as the coolant node boundaries pass wall nodes.

All of the elements in a pipe temperature group are handled at the same time as if they made a single long pipe. The use of equal coolant volumes for each node determines the locations of the wall nodes. If the region represented by a wall node spans the boundary between two elements, then weighted averages are used to obtain the coolant flow area,  $A_c$ , wall perimeter,  $P_{er}$ , wall mass,  $M_w$ , wall heat capacity,  $c_w$ , and

wall heat-transfer coefficient,  $h_w$ , for the node. The averaging is done so as to conserve coolant volume and wall mass times heat capacity.

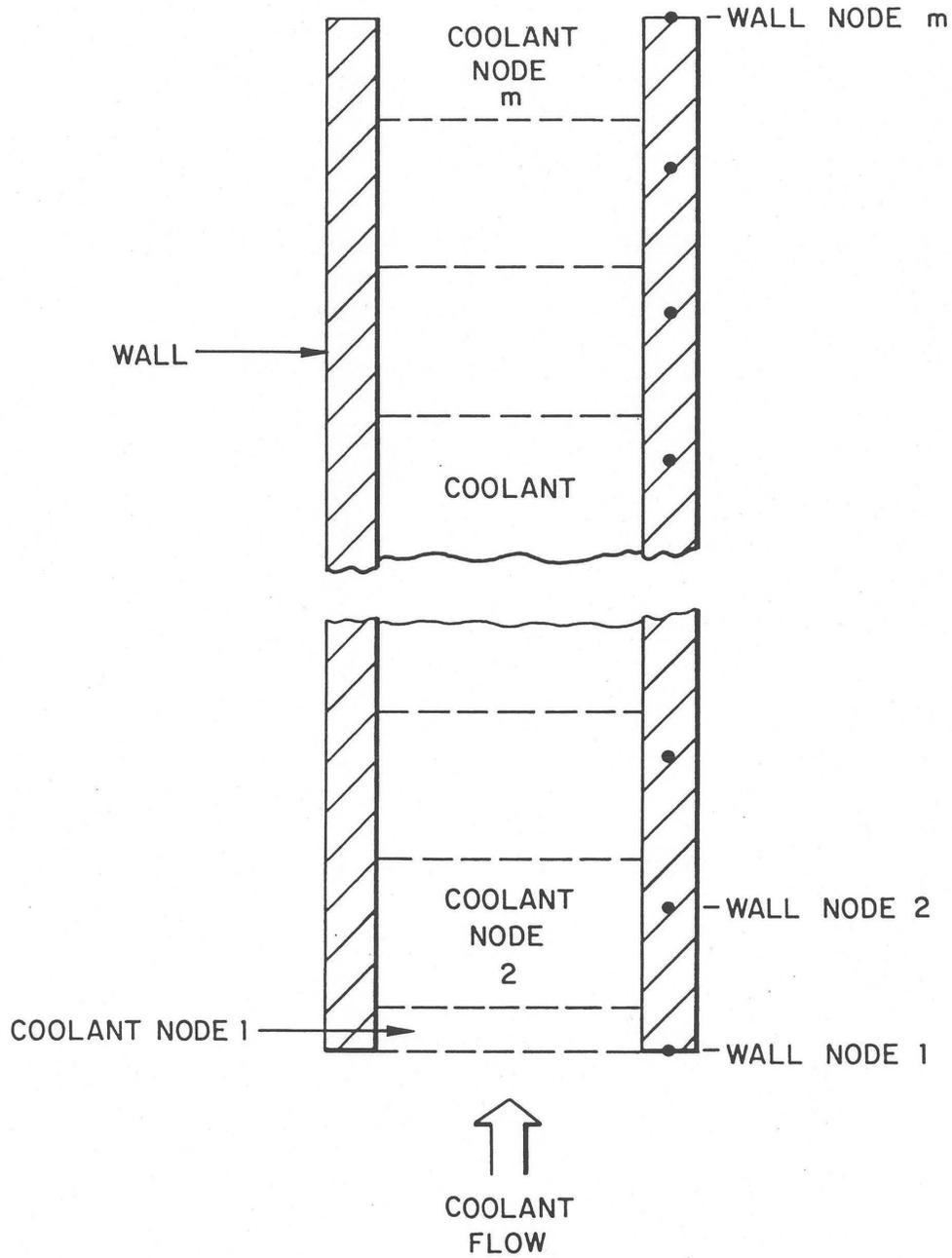


Figure 5.4-1. Pipe Temperature Calculations

The primary loop time step is divided into sub-intervals for the pipe temperatures calculations. The coolant slug is ejected from the end, a new slug is formed at the inlet, and the node indexes for intermediate slugs are increased by one. In subsequent sub-intervals, the coolant moves exactly one node per sub-interval until the end of the primary loop time step is approached. Usually, the coolant does not move exactly an integral number of nodes in a primary loop step, so in the last sub-interval the coolant usually moves only a fraction of a node.

For any node except the inlet node, the heat-transfer equation used for the coolant is:

$$\rho_c d_c A_c = \frac{\partial T_c}{\partial t} = P_{er} h_{wc} (T_w - T_c) \quad (5.4-1)$$

and that for the wall is

$$M_w c_w = \frac{\partial T_w}{\partial t} = P_{er} h_{wc} (T_c - T_w) + (hA)_{snk} (T_{snk} - T_w) \quad (5.4-2)$$

where

$T_c$  = coolant temperature

$T_w$  = wall temperature

$\rho_c$  = coolant density

$c_c$  = coolant specific heat

$h_{wc}$  = heat-transfer coefficient between the wall and the coolant

$M_w$  = wall mass per unit length

$c_w$  = specific heat of the wall

$T_{snk}$  = temperature of a heat sink outside the wall

$(hA)_{snk}$  = heat transfer coefficient times area per unit length for heat transfer to air or liquid sodium outside the pipe wall

The heat-transfer coefficient  $h_{wc}$  contains a coolant film coefficient,  $h_c$ , in series with a wall heat-transfer coefficient,  $h_w$ :

$$\frac{1}{h_{wc}} = \frac{1}{h_c} + \frac{1}{h_w} \quad (5.4-3)$$

or

$$h_{wc} = \frac{h_c h_w}{h_c + h_w} \quad (5.4-4)$$

The film coefficient is calculated as

$$h_c = \frac{k_c}{D_h} \left[ C_1 \left( \frac{D_h |w| c_c}{A_c k_c} \right)^{c_2} + C_3 \right] \quad (5.4-5)$$

$C_1$ ,  $C_2$  and  $C_3$  = user-supplied correlation coefficients

$D_h$  = pipe hydraulic diameter

$w$  = coolant flow rate

$k_c$  = thermal conductivity of the coolant

$c_c$  = specific heat of the coolant

The wall heat-transfer coefficient represents heat transfer from the interior of the wall to the surface in contact with the coolant.

Finite differencing of Eqs. 5.4-1 and 5.4-2 gives

$$\rho_c c_c A_c \frac{(T_{c6} - T_{c5})}{\delta t} = \frac{P_{er} h_{wc}}{2} (T_{w6} - T_{c6} + T_{w5} - T_{c5}) \quad (5.4-6)$$

and

$$M_w c_w \frac{(T_{w6} - T_{w5})}{\delta t} = \frac{P_{er} h_{wc}}{2} (T_{c6} - T_{w6} + T_{c5} - T_{w5}) + (hA)_{snk} [T_{snk} - (T_{w6} + T_{w5})/2] \quad (5.4-7)$$

where

$\delta t$  = sub-interval time-step size

$T_{c5}$  = coolant temperature at beginning of the sub-interval

$T_{c6}$  = coolant temperature at end of the sub-interval

$T_{w5}$ ,  $T_{w6}$  = wall temperatures at the beginning and end of the sub-interval

Simultaneous solution of these equations gives

$$T_{c6} = \frac{[d_2 - d_1 d_5 (M_w c_w + d_3)] T_{c5} + 2M_w c_w d_1 d_5 T_{w5} + 2d_1 d_3 d_5 T_{snk}}{d_2 + d_1 d_5 (M_w c_w + d_3)} \quad (5.4-8)$$

and

$$T_{w6} = d_5 (M_w c_w - d_1 - d_3) T_{w5} + d_1 d_5 (T_{c6} + T_{c5}) + 2d_3 d_5 T_{snk} \quad (5.4-9)$$

where

$$d_1 = \frac{\delta t}{2} h_{wc} P_{cr} \quad (5.4-10)$$

$$d_2 = \rho_c c_c A_c$$

$$d_3 = \frac{\delta t}{2} (hA)_{snk}$$

and

$$d_5 = \frac{1}{d_1 + M_w c_w + d_3} \quad (5.4-12)$$

For the inlet node, the wall temperature calculation is the same as that used for the other nodes, but the coolant temperature calculation is different, since new coolant is being added to the node. The basic equation used for the coolant temperature is

$$L_n \rho_c c_c A_c \frac{\partial}{\partial t} (f_r T_c) = L_n \rho_c c_c A_c \bar{T}_{in} \frac{\partial f_r}{\partial t} + L_n h_{wc} P_{er} f_r (T_w - T_c) \quad (5.4-13)$$

where

$L_n$  = length of a full node at the inlet

$f_r$  = fraction of a full node at the inlet

$f_r L_n$  = current length of the inlet node

$\bar{T}_{in}$  = pipe inlet temperature

After finite differencing, this equation becomes

$$\begin{aligned}
 & \rho_c c_c A_c f_{r6} T_{c6} - \rho_c c_c A_c f_{r5} T_{c5} \\
 & = \rho_c c_c A_c \bar{T}_{in} (f_{r6} - f_{r5}) \\
 & + \frac{\delta t}{2} h_{wc} P_{er} [f_{r5} (T_{w5} - T_{c5}) + f_{r6} (T_{w6} - T_{c6})]
 \end{aligned} \tag{5.4-14}$$

where

$f_{r5} = f_r$  at beginning of step

$f_{r6} = f_r$  at end of step.

The simultaneous solution of Eqs. 5.4-14 and 5.4-7 gives

$$\begin{aligned}
 T_{c6} = & \left\{ (d_2 f_{r5} - d_1 f_{r5} + d_1^2 d_5 f_{r6}) T_{c5} \right. \\
 & + d_1 [f_{r5} + f_{r6} d_5 (M_w c_w - d_1 - d_3)] T_{w5} \\
 & + d_2 (f_{r6} - f_{r5}) \bar{T}_{in} + 2 f_{r6} d_1 d_3 d_5 T_{snk} \left. \right\} / \\
 & f_{r6} [d_2 + d_1 d_5 (M_w c_w + d_3)]
 \end{aligned} \tag{5.4-15}$$

and Eq. 5.4-9 is again used for the wall temperature.

If flow in the pipe reverses direction, then the temperature calculations are the same, except that the outlet node becomes the inlet node and the inlet node becomes the outlet node.

#### 5.4.1.1 Eulerian Calculations

The slug flow pipe temperature model described above is a LaGrangian treatment that avoids the spurious numerical diffusion that results from typical Eulerian treatments. On the other hand, there are situations in which this treatment requires significantly more computing time than a Eulerian treatment would; and in many of these situations the effects of numerical diffusion would be small, so there is little gain from the time consuming LaGrangian treatment. The Eulerian computation time per subinterval is comparable to the LaGrangian computation time per subinterval, but the LaGrangian time step subinterval size is limited by the restriction that the coolant can not be allowed to move more than one node per subinterval, whereas no such restriction applies in the Eulerian case. Therefore, there is no advantage to a Eulerian treatment if the coolant flow rates are small and the time step sizes are small; but a Eulerian treatment is much faster than a LaGrangian treatment if the coolant flow rate is high and time steps are large enough that the coolant moves many nodes per time step. Therefore, a Eulerian speed-up option has been added to the pipe temperature calculations in the code.

The Eulerian speed-up option can be especially useful in the null transient used for steady-state initialization when component-to-component heat transfer is used. In this case, the temperature time constants are often large, requiring a long null transient to

obtain converged temperatures. The temperature solution is numerically stable for large time steps, so one would use a large time step size in the null transient to reduce computing time; but much of the benefit from a large time step size is nullified if a LaGrangian pipe temperature calculation limits the subinterval size to a small value. Also, the numerical diffusion from a Eulerian solution is small or non-existent in a steady-state pipe temperature result, so there is no reason not to use the Eulerian treatment in the null transient.

There are three options for using the Eulerian speed-up. The default option is to always use only the LaGrangian treatment. The second option is to use the Eulerian speed-up in the steady-state null transient but not in the regular transient. The third option is to use the Eulerian speed-up both in the null transient and in the regular transient. In any case, the LaGrangian calculation is used for small time steps in which the coolant will move less than two nodes. If the Eulerian speed-up is being used for a large time step, then first a LaGrangian subinterval is used to move the coolant to the next node boundary. Next, a Eulerian calculation is used to move the coolant the maximum whole number of nodes that will fit within the time step. Finally, a LaGrangian subinterval is used to finish the time step and move the coolant many fraction of a node remaining. The Eulerian part of the calculation is described in Appendix 5.6.

#### 5.4.1.2 Annular Element Temperatures

An annular element is treated the same as a pipe except that an annular element has two walls in contact with the coolant instead of one. The annular element was added to SASSYS-1 in order to model the coolant flow in an RVACS/RACS system in which a relatively thin annulus of sodium flows between the vessel wall and an inner liner. Significant heat transfer occurs between the sodium in the annulus and both the vessel wall and the inner liner.

For the annular element, the heat transfer equation used for the coolant is

$$\rho_c c_c A_c \frac{\partial T_c}{\partial t} = P_{era} h_{wca} (T_{wa} - T_c) + P_{erb} h_{wcb} (T_{wb} - T_c) \quad (5.4-15a)$$

where  $P_{era}$ ,  $h_{wca}$ , and  $T_{wa}$  refer to wall a, and  $P_{erb}$ ,  $h_{wcb}$ , and  $T_{wb}$  refer to wall b. Equations 5.4-2 and 5.4-7 are still applicable to each wall. Finite differencing of Eq. 5.4-15a gives

$$\begin{aligned} \rho_c c_c A_c \frac{(T_{c6} - T_{c5})}{\partial t} = & \frac{P_{era} h_{wca}}{2} (T_{wa6} - T_{c6} + T_{wa5} - T_{c5}) \\ & + \frac{P_{erb} h_{wcb}}{2} (T_{wb6} - T_{c6} + T_{wb5} - T_{c5}) \end{aligned} \quad (5.4-15b)$$

Simultaneous solution of the finite difference equations for  $T_c$ ,  $T_{wa}$ , and  $T_{wb}$  gives

$$\begin{aligned}
 T_{c6} = & \left\{ \left[ d_2 - d_{1a} d_{5a} (M_{wa} c_{wa} + d_{3a}) - d_{1b} d_{5b} (M_{wb} c_{wb} + d_{3b}) \right] T_{c5} \right. \\
 & + 2d_{1a} d_{5a} (M_{wa} c_{wa} T_{wa5} + d_{3a} T_{snka}) \\
 & \left. + 2d_{1b} d_{5a} (M_{wb} c_{wb} T_{wa5} + d_{3a} T_{snkb}) \right\} \\
 & / \left[ d_2 + d_{1a} d_{5a} (M_{wa} c_{wa} + d_{3a}) + d_{1b} d_{5b} (M_{wb} c_{wb} + d_{3b}) \right]
 \end{aligned} \tag{5.4-15c}$$

The solutions for  $T_{wa}$  and  $T_{wb}$  are the same as Eqn. 5.4-9. For the inlet node, simultaneous solution of the finite difference equations gives

$$\begin{aligned}
 T_{c6} = & \left\{ \left[ d_2 f_5 - d_{1a} (f_5 - f_6 d_{1a} d_{5a}) - d_{1b} (f_5 - f_6 d_{1b} d_{5b}) \right] T_{c5} \right. \\
 & + d_{1a} \left[ f_5 + f_6 d_{5a} (M_{wa} c_{wa} - d_{1a} - d_{3a}) \right] T_{wa5} \\
 & + d_{1b} \left[ f_5 + f_6 d_{5b} (M_{wb} c_{wb} - d_{1b} - d_{3b}) \right] T_{wb5} \\
 & + d_2 \bar{T}_{in} (f_6 - t_5) + 2f_6 \left[ d_{1a} d_{3a} d_{5a} T_{snka} \right. \\
 & \left. + d_{1b} d_{3b} d_{5b} T_{snkb} \right] \left. \right\} / f_6 \left[ d_2 + d_{1a} d_{5a} (M_{wa} c_{wa} + d_{3a}) \right. \\
 & \left. + d_{1b} d_{5b} (M_{wb} c_{wb} + d_{3b}) \right]
 \end{aligned} \tag{5.4-15d}$$

## 5.4.2 Intermediate Heat Exchangers: Detailed Option

### 5.4.2.1 Introduction

The intermediate heat exchanger is characterized by a shell, a primary coolant channel, a tube, and an intermediate coolant channel. Under normal conditions the coolant flows down through the primary channel and up through the intermediate channel. Flow reversal is included, so that the flow can be either way in either channel. Also, a slant-height parameter is used to permit the flow path through the tube side to be longer than the flow path through the shell side, and a fouling-factor is included to allow for reduced heat transfer in the primary and intermediate flow channels.

The intermediate heat exchanger is modeled as shown in Fig. 5.4-2. The shell, primary coolant channel, tube, and intermediate coolant channel are divided into between one and 62 vertical sections. The temperatures in the coolants are calculated at the interfaces of the vertical sections, whereas the shell and tube temperatures are calculated at the centers of the vertical sections. The flow rates and temperatures at the beginning of the time step as well as the inlet temperatures at the end of the time step are taken from COMMON blocks. A set of heat transfer equations is set up and solved for the temperatures at the end of the time step. In addition, gravity heads for both the primary and intermediate flow channels are calculated using the resulting temperature distributions. The final values are then stored in COMMON blocks.

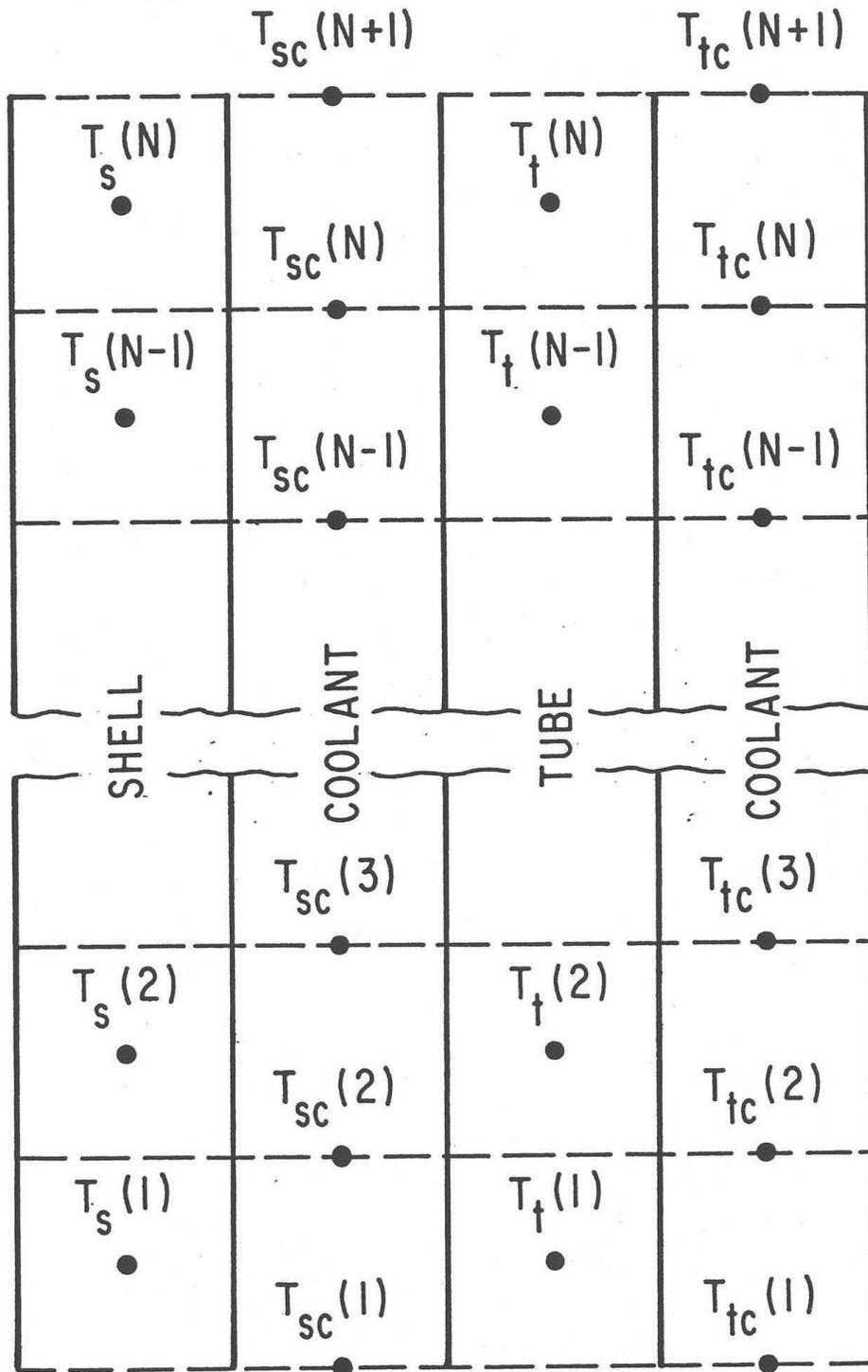


Figure 5.4-2. Intermediate Heat Exchanger Schematic

### 5.4.2.2 Basic Equations

In the configuration shown in Fig. 5.4-2, a heat balance equation is written for every vertical section of the heat exchanger. Thermal conduction is ignored vertically. The outside surface of the shell can be in thermal contact with a heat sink representing other components or air, as discussed in Section 5.4.7. The heat balance for a vertical section of the shell, which is in thermal contact with the adjacent section of the shell-side liquid, is

$$(\rho c)_{SH} \Delta z P_s d_{SH} \frac{\partial T_{SH}}{\partial t} = \Delta z P_s H_s (\bar{T}_{CS} - T_{SH}) + \Delta z (hA)_{snk} (T_{snk} - T_{SH}) \quad (5.4-16)$$

with

$$\bar{T}_{CS} = \frac{1}{2} [T_{CS}(j) + T_{CS}(j+1)] \quad (5.4-17)$$

$$\frac{1}{H_s} = \frac{1}{h_{CS}} + \frac{d_{SH}}{2k_{SH}} + \frac{1}{h_{FS}} \quad (5.4-18)$$

$$h_{CS} = \frac{\bar{k}_{CS}}{D_h} \left[ C_1 \left( \frac{D_h |w_s| \bar{c}_{CS}}{A_{CS} \bar{k}_{CS}} \right)^{C_2} + C_3 \right] \quad (5.4-19)$$

where

$(\rho c)_{SH}$  = the shell density times specific heat

$\Delta z$  = the height of the shell section

$P_s$  = the perimeter between the shell and coolant

$d_{SH}$  = the shell thickness

$T_{SH}$  = the temperature of the shell section

$T_{snk}$  = the temperature of a heat sink outside the shell

$(hA)_{snk}$  = the heat transfer coefficient times area per unit height for heat transfer to the sink

$T_{CS}(j+1)$  = the shell-side coolant temperature at the lower end of the vertical section

$T_{CS}(j+1)$ =the shell-side coolant temperature at the upper end of the vertical section

$k_{SH}$ =the shell thermal conductivity

$h_{FS}$ =user-supplied shell-side coolant channel fouling factor

$\bar{k}_{CS}$  =the primary coolant thermal conductivity at the temperature  $\bar{T}_{CS}$

$\bar{c}_{CS}$  =the primary coolant specific heat at the temperature  $\bar{T}_{CS}$

$|w_s|$  =the absolute value of the shell-side coolant mass flow rate

$D_h$ =the shell-side coolant channel hydraulic diameter

$A_{CS}$ =the shell-side coolant flow area

$C_1, C_2, C_3$ =user-supplied correlation coefficients

The shell density times specific heat and shell thermal conductivity are constants independent of temperature, whereas the liquid coolant thermal conductivity and specific heat are evaluated at the average coolant temperature of a vertical section. The fouling factor is a one-parameter effective film coefficient modeling of the heat transfer in a fouled heat exchanger.

The heat balance for a vertical section of the shell-side coolant, which is in thermal contact with the adjacent sections of the shell and of the tube, is

$$\begin{aligned} A_{CS} \bar{\rho}_{CS} \bar{c}_{CS} \Delta z \frac{\partial \bar{T}_{CS}}{\partial t} + A_{CS} \Delta z \frac{w_s \bar{c}_{CS}}{A_{CS}} \frac{\partial T_{CS}}{\partial z} \\ = \Delta z P_S H_S (T_{SH} - \bar{T}_{CS}) + \Delta z P_{ST} H_{ST} S (T_{TU} - \bar{T}_{CT}) \end{aligned} \quad (5.4-20)$$

with

$$\frac{1}{H_{ST}} = \frac{1}{h_{CS}} + \frac{d_{TU}}{2k_{TU}} + \frac{1}{h_{FS}} \quad (5.4-21)$$

where

$\bar{\rho}_{CS}$ =the shell-side coolant density evaluated at the temperature  $\bar{T}_{CS}$

$w_s$ =the shell-side coolant mass flow rate

$P_{ST}$ =the perimeter between the shell-side coolant and the tube

$S$ =user-supplied slant height ratio of the tube vertical section

$T_{TU}$ =the temperature of the tube vertical section

The remaining symbols are the same as defined for the shell.

The slant height ratio enables the user to model the tube-side of the heat exchanger as a coil imbedded in the primary coolant. If no slant height is entered in the input, the code sets the slant height to one.

The heat balance for a vertical section of the tube, which is in thermal contact with the adjacent sections of the primary and intermediate coolants is

$$\begin{aligned} (\rho c)_{TU} \Delta z S \frac{1}{2} (P_{ST} + P_{TT}) d_{TU} \frac{\partial T_{TU}}{\partial t} \\ = \Delta z S P_{ST} H_{ST} (\bar{T}_{CS} - T_{TU}) + \Delta z S P_{TT} H_{TT} (\bar{T}_{CT} - T_{TU}) \end{aligned} \quad (5.4-22)$$

with

$$\bar{T}_{CT} = \frac{1}{2} [T_{CT}(j) + T_{CT}(j+1)] \quad (5.4-23)$$

$$\frac{1}{H_{TT}} = \frac{1}{h_{CT}} + \frac{d_{TU}}{2k_{TU}} + \frac{1}{h_{FT}} \quad (5.4-24)$$

$$h_{CR} = \frac{\bar{k}_{CT}}{D_h} \left[ C_1 \left( \frac{D_h |w_T| \bar{c}_{CT}}{A_{CT} k_{CT}} \right)^{C_2} + C_3 \right] \quad (5.4-25)$$

where

$(\rho c)_{TU}$ =the tube density times specific heat

$\Delta z$ =the height of the tube section

$d_{TU}$ =the tube thickness

$T_{CT}(j)$ =the tube-side coolant temperature at the lower end of the vertical section

$T_{CT}(j+1)$ =the tube-side coolant temperature at the upper end of the vertical section

$k_{TU}$ =the tube thermal conductivity

$h_{FT}$ =user-supplied intermediate coolant channel fouling factor

$\bar{k}_{CT}$  = the tube-side coolant thermal conductivity at the temperature  $\bar{T}_{CT}$

$\bar{c}_{CT}$  = the tube-side coolant specific heat at the temperature  $\bar{T}_{CT}$

$|w_T|$  = the absolute value of the tube-side coolant mass flow rate

$D_h$  = the tube-side coolant channel hydraulic diameter

$A_{CT}$  = the tube-side coolant flow area

$C_1, C_2, C_3$  = user-supplied correlation coefficients

The tube density times specific heat and tube thermal conductivity are constants independent of temperature, whereas the tube-side coolant thermal conductivity and specific heat are evaluated at the average tube-side coolant temperature of the vertical section. The fouling factor is similar to that for the shell side.

The heat balance for a vertical section of the tube-side coolant, which is in thermal contact with only the adjacent section of the tube, is

$$\begin{aligned} A_{CT} \bar{\rho}_{CT} \bar{c}_{CT} \Delta z S \frac{\partial \bar{T}_{CT}}{\partial t} + A_{CT} \Delta z S \frac{w_T \bar{c}_{CT}}{A_{CT} S} \frac{1}{S} \frac{\partial T_{CT}}{\partial z} \\ = \Delta z S P_{TT} H_{TT} (T_{TU} - \bar{T}_{CT}) \end{aligned} \quad (5.4-26)$$

where

$\bar{\rho}_{CT}$  = tube-side coolant density evaluated at the temperature

$w_T$  = the intermediate coolant mass flow rate

The remaining symbols are the same as already defined in this section.

After the primary and intermediate coolant temperatures have been calculated, as described below, the gravity heads for both coolants are calculated by summing terms like the following for each primary and each intermediate coolant section:

$$GH = \bar{\rho} g \Delta z \quad (5.4-27)$$

where

$GH$  = the gravity head for each coolant section

$\bar{\rho}$  = the average coolant density for a coolant section

$g$  = the acceleration of gravity

$\Delta z$  = the height of the coolant section

### 5.4.2.3 Finite Difference Equations

If both the shell-side and tube-side coolants did flow through the intermediate heat exchanger in the same direction, the solution would be fairly simple. A set of four simultaneous equations could be solved for each vertical section, and the solutions for all sections could be found by starting at the inlet end and matching the length of the heat exchanger to the exit end. With the flows in opposite directions, however, two approaches are available: iteration or solving a large matrix. Iteration entails guessing one inlet temperature, solving successive sets of 4-by-4 matrices down the length of the IHX, comparing outlet temperatures, and repeating the calculation until outlet temperatures matched. On the other hand, solving a large matrix may involve inverting a 62-by-62 matrix. The iteration method is chosen for the steady-state initialization because that is done only once, and solving the large matrix is chosen for the transient calculation because that is done for each time step.

Equations 5.4-16, 5.4-20, 5.4-22, and 5.4-26 are converted into difference equations, and the temperature changes during the time step are solved for. Time derivatives are replaced by

$$\frac{\partial T}{\partial t} = \frac{\Delta T}{\Delta t} ; \Delta T = T_4 - T_3 \quad (5.4-28)$$

where  $T_3$  and  $T_4$  are the temperatures at the beginning and at the end of the time interval  $\Delta t$ .  $T_1$  and  $T_2$  usually denote the beginning and end of a PRIMAR time step, and any subdivision of that time step is denoted by 3 and 4. Space derivatives are taken in the direction of flow. If the flow is down,

$$\frac{\partial T}{\partial z} = \frac{T(j) - T(j+1)}{\Delta z} \quad (5.4-29)$$

and if the flow is up,

$$\frac{\partial T}{\partial z} = \frac{T(j+1) - T(j)}{\Delta z} \quad (5.4-30)$$

where  $T(j)$  and  $T(j+1)$  are the temperatures evaluated at the two interfaces of the vertical section of height  $\Delta z$ . The degree of implicitness is introduced by replacing  $T$  with

$$T = \theta_1 T_3 + \theta_2 T_4 \quad (5.4-31)$$

where  $\theta_1 + \theta_2 = 1$ , and is described in Section 5.2.4 and in Appendix 2.1 in Chapter 2, and  $T_3$  and  $T_4$  are the temperatures at the beginning and end of the time interval  $\Delta t$ .

After making the above substitutions, the equations for the temperature changes during  $\Delta t$  for the  $j$ -th vertical section of the IHX for either direction of flow in either coolant channel can be written as:

For the shell,

$$a_1(j)\Delta T_{SH}(j)+a_2(j)\Delta T_{CS}(j)+a_3(j)\Delta T_{CS}(j+1)=a_4(j) \quad (5.4-32)$$

For the shell-side coolant,

$$\begin{aligned} e_1(j)\Delta T_{CS}(j)+e_2(j)\Delta T_{SH}(j-1)+e_3(j)\Delta T_{SH}(j) \\ +e_4(j)\Delta T_{TU}(j-1)+e_5(j)\Delta T_{TU}(j)+e_6(j)\Delta T_{CS}(j-1) \\ +e_7(j)\Delta T_{CS}(j+1)=e_8(j)-e_9(j)\Delta T_{CT}(j-1)-e_{10}(j)\Delta T_{CT}(j) \end{aligned} \quad (5.4-33)$$

For the tube,

$$\begin{aligned} c_1(j)\Delta T_{TU}(j)+c_2(j)\Delta T_{CS}(j)+c_3(j)\Delta T_{CS}(j+1) \\ +c_4(j)\Delta T_{CT}(j)+c_5(j)\Delta T_{CT}(j+1)=c_6(j) \end{aligned} \quad (5.4-34)$$

And for the tube-side coolant,

$$\begin{aligned} f_1(j)\Delta T_{CT}(j)+f_2(j)\Delta T_{TU}(j-1)+f_3(j)\Delta T_{TU}(j) \\ +f_4(j)\Delta T_{CT}(j-1)+f_5(j)\Delta T_{CT}(j+1) \\ =f_6(j)-f_7(j)\Delta T_{CS}(j)-f_8(j)\Delta T_{CS}(j+1) \end{aligned} \quad (5.4-35)$$

The expression for each coefficient in terms of the quantities defined in Section 5.4.2.2 are given in Appendix 5.1.

#### 5.4.2.4 Solution

The solution of Eqs. 5.4-32 through 5.4-35 is carried out by a Gaussian elimination scheme using the zeros present in the matrix. The solution algorithm is given in Appendix 5.2. The solution yields the temperature changes throughout the IHX during the interval  $\Delta t$ , and these values are added to the corresponding temperatures at the beginning of the time interval and the results are stored in COMMON blocks.

#### 5.4.2.5 Steady-State Temperatures

The steady-state temperature distributions in the intermediate heat exchanger are obtained from Eqs. 5.4-16, 5.4-20, 5.4-22, and 5.4-26 by setting the time-derivative terms to zero and noting that the temperatures at the beginning and at the end of a time interval are the same. Because in the steady-state solution an adiabatic boundary condition is used on the outside of the shell and because of our neglect of any axial heat transfer, the steady-state shell temperatures are the same as the average shell-side coolant temperatures for a vertical section. As a result, only a 3-by-3 matrix equation must be solved for each vertical section in the iterative solution mentioned at the beginning of Section 5.4.2.3. Also only normal coolant flow, down in the primary and up in the intermediate, is considered. Finally, once the equilibrium temperature

distributions have been determined, the gravity heads for the primary and intermediate flow channels are computed as indicated in Eq. 5.4-27.

The 3-by-3 matrix that must be solved for the j-th vertical section is:

$$\begin{pmatrix} \alpha_1 + \beta_1 & -\beta_1 & 0 \\ -\beta_1 & \alpha_2 + \beta_1 + \beta_2 & -\beta_2 \\ 0 & -\beta_2 & \alpha_3 + \beta_2 + \beta_3 \end{pmatrix} \begin{pmatrix} T_{CS}(j) \\ T_{TU}(j) \\ T_{CT}(j) \end{pmatrix} = \begin{pmatrix} D_1 \\ D_2 \\ D_3 \end{pmatrix} \quad (5.4-36)$$

with

$$\alpha_1 = \frac{1}{2} \frac{w_s \bar{c}_{CS}(j)}{\Delta z(j) S} - \beta_1$$

$$\alpha_2 = \beta_1 + \beta_2$$

$$\alpha_3 = -\frac{1}{2} \frac{w_T \bar{c}_{CT}(j)}{\Delta z(j) S} - \beta_2$$

$$\beta_1 = \frac{1}{2} P_{ST} H_{ST}(j)$$

$$\beta_2 = \frac{1}{2} P_{TT} H_{TT}(j)$$

$$\beta_3 = 0$$

$$D_1 = \alpha_1 T_{C2}(j+1)$$

$$D_2 = \beta_1 T_{CS}(j+1) + \beta_2 T_{CT}(j+1)$$

$$D_3 = \alpha_3 T_{CT}(j+1)$$

where

$w_s, w_T$  = the positive steady-state flow rates for the shell and tube sides

The other symbols have the same meanings as in Section 5.4.2.2.

The matrix equation 5.4-36 is solved by standard tri-diagonal inversion in subroutine INVTR3.

### 5.4.3 Bypass Channels

#### 5.4.3.1 Introduction

The purpose of the bypass channel is to model components, like control rods and radial shields, that do not need the detailed treatment of a SAS channel. The bypass channel is then considered a part of the primary loop and is not included in the SAS channel core treatment.

The bypass channel is shown schematically in Fig. 5.4-3. It is modeled as two reflectors, A and B, a coolant channel, C, and a duct wall, D. The bypass channel is divided into from 1 to 7 vertical sections, with temperatures of the two reflectors and duct wall taken at the centers of each vertical section and the coolant temperatures taken at the interfaces of each vertical section. The outside surface of reflector A is taken as adiabatic, and heat conduction in the axial direction is neglected. The outside surface of the duct wall can be in contact with a heat sink that represents neighboring subassemblies, as discussed in Section 5.4.1.

Heat sources are included in each of the two regions of reflectors A and B and also in the duct wall, taken as region 3. A vertical power shape for the heat source can be assigned to the sections in the three regions. Two heat sources are included, a neutron heat source and a decay heat source. The neutron heat source arises from fissions caused by neutrons, and is a fraction of the reactor power. The decay heat source arises from fission product decay which produces no neutrons.

A heat balance equation is written for each vertical section of the bypass channel, and the temperature changes during a time step are computed. The choice of the degree of implicitness used in the solution is discussed in Section 5.2.4 and in Appendix 2.1 in Chapter 2. The temperatures at the beginning of a time step are known, and the coolant mass flow rates at the beginning and at the end of a time step are known from the hydraulics calculations described in Section 5.2. Also, the heat sources during the time step are known. As a result, a set of four simultaneous equations is solved for each vertical section in turn for the length of the bypass channel. In the case of flow reversal, the series of solutions is carried out along the bypass channel in the opposite direction. The temperature changes during the time step are added to the corresponding temperatures at the beginning of the time step, and the final values are then stored in COMMON blocks.

#### 5.4.3.2 Basic Equations

A vertical section of reflector A, with the outside surface assumed adiabatic and with axial thermal conduction ignored, is in thermal contact with only the adjacent section of reflector B. The heat balance equation for the section in A is:

$$(\rho c)_A \Delta z P_A d_A \frac{\partial T_A}{\partial t} = \Delta z P_A H_{AB} (T_B - T_A) + \Delta z P_A d_A Q_A \quad (5.4-37)$$

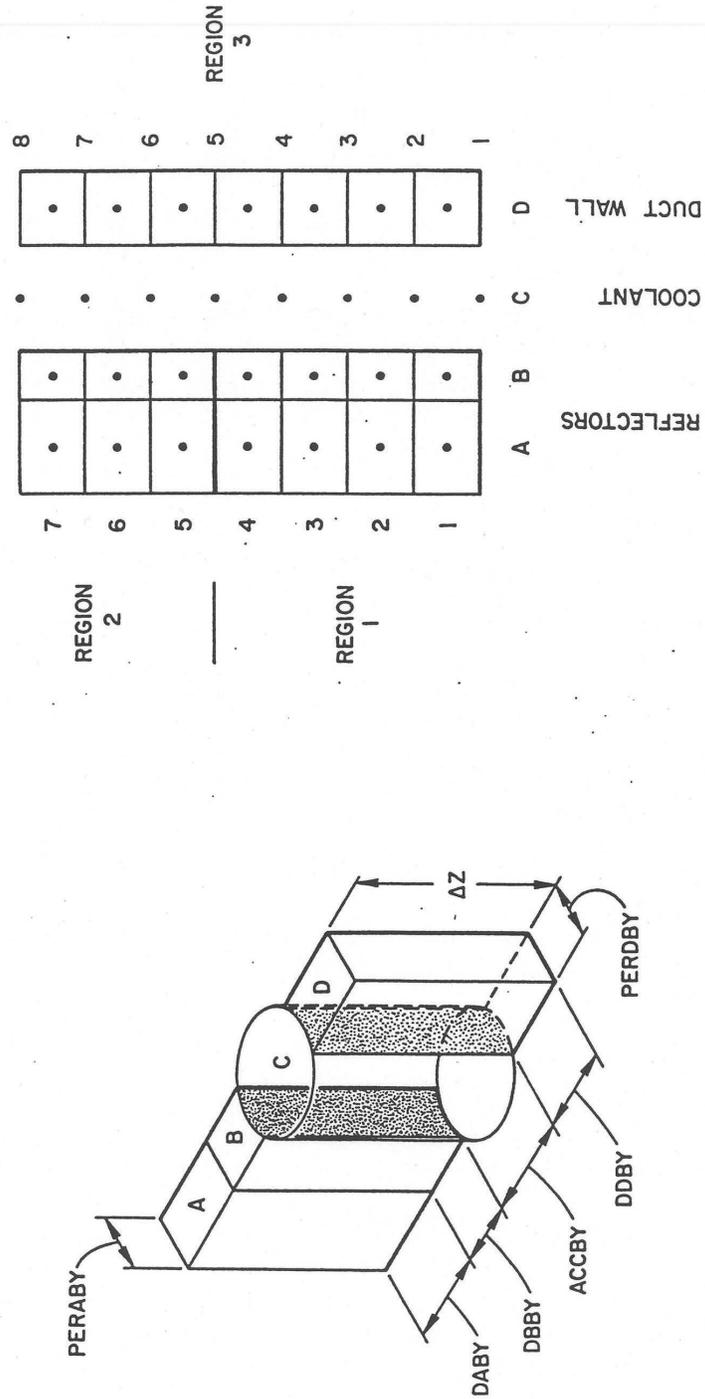


Figure 5.4-3. Bypass Channel Schematic

with

$$\frac{1}{H_{AB}} = \frac{d_A}{2k_A} + \frac{d_B}{2k_B} \quad (5.4-38)$$

$$Q_A = \frac{P_{TOT} Q_{MULT}}{\Delta z} f_1 \frac{P_s}{P_1} (\gamma_{N1} + \gamma_{D1}), \quad \text{Region1} \quad (5.4-39)$$

$$Q_A = \frac{P_{TOT} Q_{MULT}}{\Delta z} f_2 \frac{P_s}{P_2} (\gamma_{N2} + \gamma_{D2}), \quad \text{Region2} \quad (5.4-40)$$

where

$(\rho c)_A$  = the reflector A density times specific heat

$\Delta z$  = the height of the vertical section

$P_A$  = the perimeter between reflectors A and B and between B and C

$d_A$  = the thickness of reflector A

$T_A$  = the temperature of the vertical section of reflector A

$T_B$  = the temperature of the vertical section of reflector B

$d_B$  = the thickness of reflector B

$k_A$  = the thermal conductivity of reflector A

$k_B$  = the thermal conductivity of reflector B

$Q_A$  = the heat source in the vertical section of reflector A

$P_{TOT} Q_{MULT}$  = the reactor power for this time step

$f_1$  = the fraction of the reactor power distributed in reflector A in Region 1 of the bypass channel

$f_2$  = the fraction of the reactor power distributed in reflector A in Region 2 of the bypass channel

$P_s$  = the power shape assigned to the vertical sections

$P_1$  = the power shape normalization for Region 1

$\gamma_{N1}$  = the fraction of the reactor power in Region 1 attributed to neutron heating

$\gamma_{D1}$  = the fraction of the reactor power in Region 1 attributed to decay heating

$P_2$  = the power shape normalization for Region 2

$\gamma_{N2}$  = the fraction of the reactor power in Region 2 attributed to neutron heating

$\gamma_{D2}$  = the fraction of the reactor power in Region 2 attributed to decay heating

The density times specific heat for reflector A and the thermal conductivities for reflectors A and B are assumed to be independent of temperature. The reactor power for the current time step,  $P_{TOT}Q_{MULT}$ , is described in Chapter 4. The power shape,  $P_s$ , for each vertical section is an input value, and the code calculates a normalization factor for each region in the bypass channel. The decay heat produced in a section of the reflector or duct wall is taken, in the current version of the code, to be a fraction of the reactor power, just as is the neutron heating. These fractions, the  $\gamma$ 's, are input numbers, as are  $P_s$ ,  $f_1$ , and  $f_2$ .

The heat balance for a vertical section of reflector B, which is in thermal contact with the adjacent sections of reflector A and of the coolant, is

$$\begin{aligned} (\rho c)_B \Delta z P_A d_B \frac{\partial T_b}{\partial t} = & \Delta z P_A H_{AB} (T_A - T_B) \\ & + \Delta z P_A H_{BC} (\bar{T}_C - T_B) \\ & + \Delta z P_A d_B Q_B \end{aligned} \quad (5.4-41)$$

with

$$\frac{1}{H_{BC}} = \frac{1}{h_C} + \frac{d_B}{2k_B} \quad (5.4-42)$$

$$h_C = \frac{\bar{k}_c}{D_h} \left[ C_1 \left( \frac{D_h |w_c| c_C}{A_C \bar{k}_c} \right)^{C_2} + C_3 \right] \quad (5.4-43)$$

$$\bar{T}_c = 1/2 [T_c(j) + T_c(j+1)] \quad (5.4-44)$$

$$Q_B = \frac{P_{TOT} Q_{MULT}}{\Delta z} (1 - f_1) \frac{P_s}{P_1} (\gamma_{N1} + \gamma_{D1}), \quad \text{Region1} \quad (5.4-45)$$

$$Q_B = \frac{P_{TOT} Q_{MULT}}{\Delta z} (1 - f_2) \frac{P_s}{P_2} (\gamma_{N2} + \gamma_{D2}), \quad \text{Region2} \quad (5.4-46)$$

where

$(\rho c)_B$  = the reflector B density times specific heat

$Q_B$  = the heat source in the vertical section of reflector B

$T_c(j)$  = the coolant temperature at the lower end of the vertical section

$T_c(j+1)$  = the coolant temperature at the upper end of the vertical section

$\bar{k}_c$  = the coolant thermal conductivity at the temperature  $\bar{T}_c$

$\bar{c}_c$  = the coolant specific heat at the temperature  $\bar{T}_c$

$|w|$  = the absolute value of the coolant mass flow rate

$D_k$  = the coolant channel hydraulic diameter

$A_c$  = the coolant flow area

$C_1, C_2, C_3$  = user-supplied correlation coefficients

The remaining symbols are the same as defined above.

The heat balance for a vertical section of the coolant, which is in thermal contact with the adjacent sections of reflector B and the duct wall D, is

$$A_c \Delta z \bar{\rho}_c \bar{c}_c \frac{\partial \bar{T}_c}{\partial t} + A_c \Delta z \frac{\partial}{\partial z} \left( \frac{w \bar{c}_c}{A_c} \bar{T}_c \right) = \Delta z P_A H_{BC} (T_B - \bar{T}_c) + \Delta z P_D H_{CD} (T_D - \bar{T}_c) \quad (5.4-47)$$

with

$$\frac{1}{H_{CD}} = \frac{1}{h_c} + \frac{d_D}{2k_D} \quad (5.4-48)$$

where

$\bar{\rho}_C$  = the coolant density evaluated at temperature  $\bar{T}_C$

$w$  = the coolant mass flow rate

$P_D$  = the perimeter between the coolant and the duct wall

$d_D$  = the thickness of the duct wall

$k_D$  = the thermal conductivity of the duct wall

$T_D$  = the temperature of the vertical section of the duct wall

The thermal conductivity of the duct wall is assumed to be independent of temperature. The remaining symbols are the same as already defined.

The heat balance for a vertical section of the duct wall, which, is in thermal contact with the coolant and an external heat sink is

$$(\rho c)_D \Delta z P_D d_D \frac{\partial T_D}{\partial t} = \Delta z P_D H_{CD} (\bar{T}_C - T_D) + \Delta z P_D d_D Q_D + \Delta z (hA)_{snk} (T_{snk} - T_D) \quad (5.4-49)$$

with

$$Q_D = \frac{P_{TOT} Q_{MULT}}{\Delta z} \frac{P_s}{P_3} (\gamma_{N3} + \gamma_{D3}) \quad (5.4-50)$$

where

$(\rho c)_D$  = the duct wall density times specific heat

$Q_D$  = the heat source in the vertical section of the duct wall

$P_3$  = the power shape normalization for the duct wall

$\gamma_{N3}$  = the fraction of the reactor power in the duct wall attributed to neutron heating

$\gamma_{D3}$  = the fraction of the reactor power in the duct wall attributed to decay heating

$T_{snk}$  = the temperature of the heat sink outside the duct wall

$(hA)_{snk}$  = the heat transfer coefficient times area per unit height for heat transfer to the sink

The remaining symbols are the same as already defined.

### 5.4.3.3 Finite Difference Equations and Solution

The four equations 5.4-37, 5.4-41, 5.4-47, and 5.4-49 are a set of simultaneous equations that can be solved for one vertical section of the bypass channel. If the coolant flow is upward (normal flow), then the coolant temperature at the bottom of the lowest vertical section is known, and the coolant temperature at the top of the vertical section as well as the reflector and duct wall temperatures are solved for. This process is repeated up the bypass channel for each vertical section. If, on the other hand, the coolant flow is downward (reversed flow), the coolant temperature at the top of the uppermost vertical section is known, and the coolant temperature at the bottom of the section as well as the reflector and duct wall temperatures are solved for. This process is then repeated down the bypass channel for each vertical section.

First, however, the above four equations are converted to finite difference equations and the temperatures at the end of the time step are solved for. The differencing is accomplished by replacing time derivatives by

$$\frac{\partial T}{\partial t} = \frac{\Delta T}{\Delta t} ; \Delta T = T_4 - T_3 \quad (5.4-51)$$

where  $T_3$  and  $T_4$  are the temperatures at the beginning and at the end of the time interval  $\Delta t$ . The temperatures  $T_1$  and  $T_2$  usually denote the beginning and end of a PRIMAR time step, and any subdivision of that time step is denoted by 3 and 4. Space derivatives are taken in the direction of flow. If the flow is up,

$$\frac{\partial T}{\partial z} = \frac{T(j+1) - T(j)}{\Delta z} \quad (5.4-52)$$

and if the flow is down,

$$\frac{\partial T}{\partial z} = \frac{T(j) - T(j+1)}{\Delta z} \quad (5.4-53)$$

where  $T(j)$  and  $T(j+1)$  are the temperatures evaluated at the two interfaces of the vertical section of height  $\Delta z$ . The degree of implicitness is introduced by replacing  $T$  with

$$T = \theta_1 T_3 + \theta_2 T_4 \quad (5.4-54)$$

where  $\theta_1 + \theta_2 = 1$ , as described in Section 5.2.4 and in Appendix 2.1 in Chapter 2, and  $T_3$  and  $T_4$  are the temperatures at the beginning and at the end of the time interval  $\Delta t$ .

After making the above substitutions, the equations for the temperatures at the end of the time interval  $\Delta t$  for the  $j$ -th vertical section of the bypass channel can be written as:

$$\begin{pmatrix} \alpha_1 + \beta_1 & -\beta_1 & 0 & 0 \\ -\beta_1 & \alpha_2 + \beta_1 + \beta_2 & -\beta_2 & 0 \\ 0 & -\beta_2 & \alpha_3 + \beta_2 + \beta_3 & -\beta_3 \\ 0 & 0 & -\beta_3 & \alpha_4 + \beta_3 + \beta_4 \end{pmatrix} \begin{pmatrix} T_{A4} \\ T_{B4} \\ \bar{T}_{C4} \\ T_{D4} \end{pmatrix} + \begin{pmatrix} D_1 \\ D_2 \\ D_3 \\ D_4 \end{pmatrix} \quad (5.4-55)$$

with

$$\alpha_1 = (\rho c)_A P_A d_A \quad (5.4-56)$$

$$\alpha_2 = (\rho c)_B P_B d_B \quad (5.4-57)$$

$$\alpha_3 = A_c \bar{\rho}(j) \bar{c}(j) + \frac{2\bar{c}(j)}{\Delta z(j)} \theta_2 \Delta t |w_4| \quad (5.4-58)$$

$$\alpha_4 = (\rho c)_D P_D d_D \quad (5.4-59)$$

$$\beta_1 = P_A H_{AB} \Delta t \theta_2 \quad (5.4-60)$$

$$\beta_2 = P_A H_{BC} \Delta t \theta_2 \quad (5.4-61)$$

$$\beta_3 = P_A H_{DC} \Delta t \theta_2 \quad (5.4-62)$$

$$\beta_4 = (hA)_{snk} \Delta t \theta_2 \quad (5.4-63)$$

$$D_1 = [(\rho c)_A P_A d_A - P_A H_{AB} \Delta t \theta_1] T_{A3} + P_A H_{AB} \Delta t \theta_1 T_{B3} + \Delta t Q_A \quad (5.4-64)$$

$$D_2 = P_A H_{AB} \Delta t \theta_1 T_{A3} + [(\rho c)_B P_B d_B - P_A H_{AB} \Delta t \theta_1 - P_A H_{BC} \Delta t \theta_1] T_{B3} + P_A H_{AB} \Delta t \theta_1 \bar{T}_{C3} + \Delta t Q_B \quad (5.4-65)$$

$$\begin{aligned}
 D_3 = & P_A H_{DC} \Delta t \theta_1 T_{B3} \\
 & + \left[ A_c \bar{\rho}(j) \bar{c}(j) - \frac{2\bar{c}(j)}{\Delta z(j)} \theta_1 \Delta t |w_3| - P_A H_{BC} \Delta t \theta_1 - P_D H_{DC} \Delta t \theta_1 \right] \bar{T}_{C3} \\
 & + P_D H_{DC} \Delta t \theta_1 T_{D3} + \frac{2\bar{c}(j)}{\Delta z(j)} \Delta t \left[ \theta_1 |w_3| T_{C3} + \theta_2 |w_4| T_{C4} \right]
 \end{aligned} \quad (5.4-66)$$

$$\begin{aligned}
 D_4 = & P_D H_{DC} \Delta t \theta_1 \bar{T}_{C3} \\
 & + \left[ (\rho c)_D P_D d_D - P_D H_{DC} \Delta t \theta_1 - (hA)_{snk} \Delta t \theta_1 \right] T_{D3} + \Delta t Q_D \\
 & + (hA)_{snk} \Delta t T_{snk}
 \end{aligned} \quad (5.4-67)$$

The matrix Eq. 5.4-55 is solved by standard tri-diagonal inversion in subroutine INVRT3, and the node temperatures of reflectors A and B and duct wall D and the interface temperatures of the coolant are stored in COMMON blocks. In addition, the gravity head of the bypass element is calculated, using the new coolant temperatures, and also stored in COMMON blocks.

#### 5.4.3.4 Steady-State Bypass Channels

The steady-state bypass temperature calculations, performed in subroutine SSBYPS, differ from the transient calculations in that the time derivative terms in Eqs. 5.4-37, 5.4-41, 5.4-47, and 5.4-49 are set to zero and in that the steady-state power input into the reflectors and duct wall for each vertical section must be conducted to the coolant in that section. The coolant temperatures are calculated at the interfaces of each vertical section, marching up the bypass channel and iterating over the temperature-dependent thermal properties. From these coolant temperatures, the node temperatures of the reflectors and duct wall are then determined. In addition, the gravity head is calculated from the average coolant density for the whole channel, and the power normalization factors are calculated for all three regions.

#### 5.4.4 Compressible Volumes

The liquid temperature for a compressible volume other than the outlet plenum is computed as a one-point perfect mixing model. If cover gas is present, its contribution to the liquid temperature is ignored because of the small heat capacity of the gas, and the calculation is carried out as though no gas were present. In Eq. 5.3-8 of Section 5.3.1 for the liquid pressure calculation in a compressible volume, the heat flow from the walls was ignored. Here in the liquid temperature calculation it is included in the energy balance equation.

For the temperature calculation, the heat flow is taken as

$$Q\Delta t = \Delta t \left[ \theta_{1w} (T_{w3} - T_3) + \theta_{2w} (T_{w4} - T_4) \right] h_w A_w + E_{src} \quad (5.4-68)$$

where

$T_w$  = the compressible volume wall temperature

$h_w$  = the effective heat-transfer coefficient

$A_w$  = the compressible volume wall area

$\theta_{2w}$  = the degree of implicitness

$T_3$  = the liquid temperature at the beginning of the subinterval

$T_4$  = the liquid temperature at the end of the sub-interval

$E_{src}$  = heat flow to the liquid from other components

Also, the wall temperature is determined by

$$m_w c_w (T_{w4} + T_{w3}) = -Q\Delta t + \Delta t H_{snk} A_{snk} (T_{snk} - \theta_{1w} T_{w3} - \theta_{2w} T_{w4}) \quad (5.4-69)$$

where

$m_w$  = the mass of the compressible volume wall

$c_w$  = the heat capacity of the compressible volume wall

$T_{snk}$  = the temperature of a heat sink representing heat loss to sodium or air outside the compressible volume, as discussed in Section 5.4.7

$H_{snk}$  = the heat transfer coefficient from the compressible volume wall to the heat sink

Combining Eqs. 5.4-68 and 5.4-69 gives

$$T_{w4} = \left\{ T_{w3} [m_w c_w - \theta_{1w} \Delta t (h_w A_w + H_{snk} A_{snk})] + \Delta t h_w A_w (\theta_{1w} T_3 + \theta_{2w} T_4) + \Delta t H_{snk} A_{snk} T_{snk} \right\} / [m_w c_w + \theta_{2w} \Delta t (h_w A_w + H_{snk} A_{snk})] \quad (5.4-70)$$

for the wall temperature at the end of the time interval.

The  $\sum \bar{w}_{out} T_{out}$  term in Eq. 5.3-8 involves a compressible volume liquid temperature averaged over the time step. For the hydraulic calculations in Section 5.2, this term is evaluated using the temperature at the beginning of the step as the average; but for the compressible volume temperature calculations this term is evaluated as

$$\sum \bar{w}_{out} T_{out} = \sum_i \theta_1(i) |w_3(i)| T_3 + \theta_2(i) |w_4| T_4 \quad (5.4-71)$$

where the summation is over all liquid segments,  $i$ , in which the flow is out of the compressible volume. Then

$$\Delta t \left\{ \sum \bar{w}_{in} T_{in} - \sum \bar{w}_{out} T_{out} \right\} = S_3 + S_4 T_4 \quad (5.4-72)$$

where

$$S_3 = \Delta t \sum_k \left[ \theta_1(k) |w_3(k)| T_{3ex}(k) + \theta_2(k) |w_4(k)| T_{4ex}(k) \right] - \Delta t \sum_i \theta_1(k) |w_3(k)| T_3 \quad (5.4-73)$$

$$S_4 = -\Delta t \sum_i \theta_2(i) |w_4(i)| \quad (5.4-74)$$

$T_{3ex}$ ,  $T_{4ex}$  = the temperature of the liquid leaving liquid segment  $k$  at the beginning and end of the time semiinterval

In Eq. 5.4-73 and Eq. 5.4-74 the  $k$  summations are over all liquid segments in which the flow is out of the compressible volume. Combining Eqs. 5.3-8, 5.4-70, and 5.4-72 gives

$$T_4 = \left\{ m_3 T_3 + S_3 + \frac{d_1 d_3 d_6 (T_{snk} - \theta_1 T_3)}{c_\ell} + \frac{E_{src}}{c_\ell} + \frac{d_1 d_2 d_5 (T_{w3} - \theta_{1w} T_3)}{c_\ell} \right\} / \left[ m_4 - S_4 + \frac{d_1 d_3}{c_\ell} (d_2 + \theta_{2w} d_6) \right] \quad (5.4-75)$$

where

$$d_1 = \Delta t h_w A_w \quad (5.4-76)$$

$$d_2 = m_w c_w \quad (5.4-77)$$

$$d_6 = \Delta t H_{snk} A_{snk} \quad (5.4-78)$$

$$d_5 = \frac{1}{d_2 + \theta_{2w} d_1 + \theta_{2w} d_6} \quad (5.4-79)$$

$$d_3 = \theta_{2w} d_5 \quad (5.4-80)$$

$E_{src}$  = a heat source due heat transfer other components the volume liquid, as section 5.4.7.

In the code, first Eq. 5.4-75 is solved for  $T_4$ ; and then Eq. 5.4-70 is solved for  $T_{w4}$ .

### 5.4.5 Table Look-Up for Steam Generators or Intermediate Heat Exchangers

Table look-ups for steam generators or intermediate heat exchangers are used when the behavior of these components is known from other sources or when the details of the temperature distributions within the components are not important for the case at hand. The user supplies a table of normalized temperature drops vs. time and also a table of elevation of the thermal center vs. time. The thermal center of the steam generator or intermediate heat exchanger is the point above which the circulating fluid is assumed to have the density corresponding to the inlet temperature and below which it is assumed to have the density corresponding to the outlet temperature.

The outlet temperature of the steam generator or intermediate heat exchanger is calculated from the inlet temperature by

$$T_{out}(t) = T(t) + f(t)[T_{out}(t=0) - T_{in}(t=0)] \quad (5.4-81)$$

where  $f(t)$  is obtained by linear interpolation from the user-supplied table of temperature drops. Also

$$f(t=0) = 1.0 \quad (5.4-82)$$

For the table look-up steam generator there is an option in the code for the user to specify the steam generator outlet temperature directly instead of specifying the temperature drop. For this option the user supplies a table of steam generator outlet temperature vs. time.

The gravity head is calculated from

$$\Delta p_{gr}(t) = \rho_{in}(t)g[z_c(t) - z_{in}] + \rho_{out}(t)g[z_{out} - z_c(t)] \quad (5.4-83)$$

where

$\rho_{in}(t), \rho_{out}(t)$  = the densities of the inlet and outlet fluids at the inlet and outlet temperatures

$z_c(t)$  = the user-supplied table of the elevation of the thermal center as a function of time

$z_{in}, z_{out}$  = the inlet and outlet elevations

$g$  = the acceleration of gravity

#### 5.4.6 Component-to-Component Heat Transfer

SASSYS-1 accounts for heat transfer between components or from a component to a constant temperature heat sink which might represent the air. The code accounts for heat transfer from a core channel duct wall to a core channel duct wall or to a constant temperature heat sink. Also, it accounts for heat transfer from a pipe wall, a bypass channel duct wall, an IHX shell, or a compressible volume wall to the liquid in a compressible volume, to a RVACS (Radiant Vessel Auxiliary Cooling System), or to a constant temperature heat sink. This heat transfer is usually a small effect at normal operating conditions, but it is often important at decay heat power levels. Currently, component-to-component heat transfer is only accounted for in the transient calculations; a null transient, as described in Section 5.9.5, must be run to account for steady-state component-to-component heat transfer.

The component-to-component heat transfer is controlled by user input. The user specifies which components are in contact with which other components, as well as specifying the heat transfer coefficients and areas involved in component to component heat transfer. If the user does not list a component in the component-to-component heat transfer tables, then an adiabatic boundary condition is used on the outside of the component.

The core channel-to-core channel heat transfer is described in Chapter 2. The implementation of component-to-component heat transfer in PRIMAR-4 was carried out in two main steps:

1. Terms  $T_{snk}$  and  $(hA)_{snk}$  or  $H_{snk} A_{snk}$  were added to the temperature equations for the annular element, the pipe wall (Eq. 5.4-2), the bypass channel wall (Eq. 5.4-4a), the IHX shell (Eq. 5.4-16), and the compressible volume wall (Eq. 5.4-69) to account for heat flow to adjacent components. Also, a heat source term,  $E_{src}$ , was added to Eq. 5.4-68 for the temperature of the liquid in a compressible volume.
2. The  $T_{snk}$  terms are re-calculated for each time step of the transient to correspond to the current temperatures of neighboring components. Also, contributions to the  $E_{src}$  term are added for each participating component for each time step.

If a pipe wall is in contact with the liquid in a compressible volume I, then  $T_{snk}$  for the pipe wall temperature calculation is set equal to the compressible volume liquid temperature at the beginning of each PRIMAR time sub-internal, and  $(hA)_{snk}$  is set to a user-specified value. The contribution to  $E_{src}$  from node  $N$  of the pipe wall to the compressible volume liquid is calculated as

$$\Delta E_{src}(I) = [T_w(N) - T_{snk}] (hA)_{snk} \Delta L(N) \quad (5.4-84)$$

where

$T_w(N) =$  the pipe wall temperature at node  $N$

and

$\Delta L(N) =$  the length of node  $N$ .

Each node in the liquid element representing the pipe adds a similar contribution to  $E_{src}(I)$ . Also, if a bypass channel wall, IHX shell, or the wall of another compressible volume is in contact with the liquid in compressible volume I, it also adds similar contributions to  $E_{src}(I)$ .

If a constant temperature heat sink is used for a component, then the user supplies the values for  $T_{snk}$  and  $(hA)_{snk}$  or  $H_{snk} A_{snk}$ .

The RVACS model supplies  $T_{snk}$  and  $H_{snk} A_{snk}$  for components involved in the RVACS, and component temperatures are supplied to the RVACS routine.

### 5.4.7 RVACS/RACS Models

Two RVACS (Reactor Vessel Auxiliary Cooling System) or RACS (Reactor Air Cooling System) Models have been included in SASSYS: a simple model in which the user supplies the relevant information on air-side performance, and a more detailed model in which air temperatures and flow rates are calculated by the code. The sodium-side treatment and the representation of the reactor vessel wall are the same in both models. The vessel wall is represented by a combination of compressible volume walls and the walls of pipes and annular elements. The component-to-component heat transfer capability described in Section 5.4.7 above is used to remove heat from the reactor vessel walls, and the two models are used to set  $T_{snk}$ ,  $H_{snk}$ , and  $A_{snk}$ , for the wall.

#### 5.4.7.1 Air Side Treatment, Simple Model

For the simple model the representation of the RVACS stops at the reactor vessel wall. The user supplies a single air temperature and a table of effective heat transfer coefficient vs. vessel wall temperature. For each axial node in the reactor vessel wall, the code then sets the sink temperature,  $T_{snk}$ , to the air temperature. The sink heat transfer coefficient,  $h_{snk}$ , is re-calculated each time step based on the current vessel wall temperature at the node.

#### 5.4.7.2 Detailed Air Side Model

Figure 5.4-4 shows the general RVACS model and noding scheme. Wall temperature nodes are used for the reactor vessel, guard vessel, finned shell (inside and outside), outer wall, and a constant temperature deep in the concrete or the ground. Air nodes are included for the down-comer and for the up-flow section. An air inlet section and an outlet stack are also included. Either vertical or non-vertical sections of vessel wall can be treated.

The reactor vessel wall can be made up of a number of SASSYS components, including a hot pool wall, a cold pool wall, a pipe wall, or an annular flow element wall.

The annular flow element is a new type of liquid flow element that has recently been added to SASSYS-1, mainly for modeling the RVACS and for modeling multi-dimensional effects in pools.

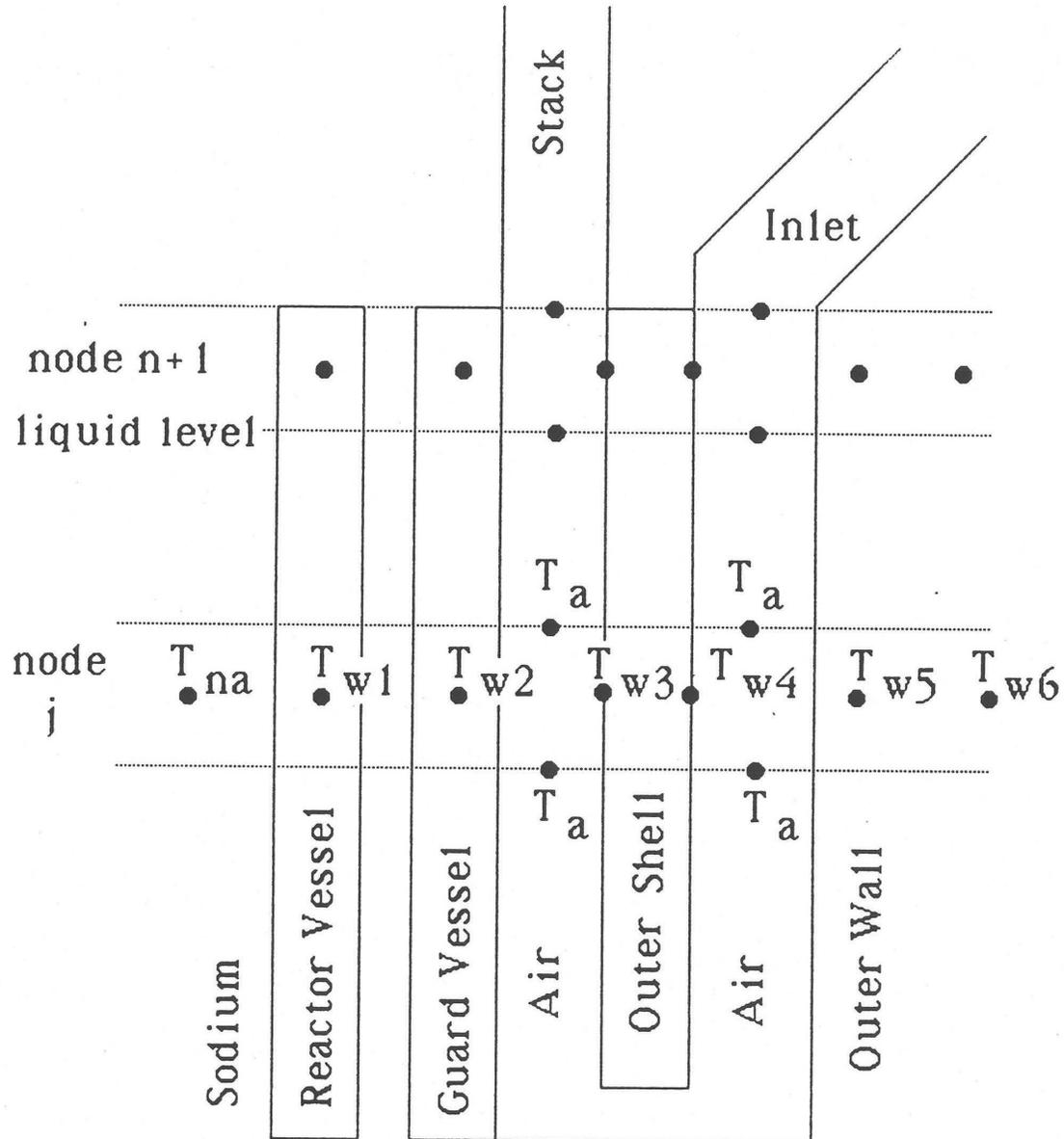


Figure 5.4-4. SASSYS-1 Model of an RVACS/RACS System

Both radiative and convective heat transfer from the reactor vessel to the guard vessel are modeled. Radiation from the guard vessel to the finned shell as well as convective heat transfer to air from the guard vessel and the finned shell are treated.

The finned shell is assumed to be insulated, but less-than-perfect insulation can be treated. Temperatures on both sides of the insulation are calculated. Heat transfer from the inside of the to the outside, and from the outside of the finned shell to the incoming air, is treated. Radiation from the finned shell to the outer wall is treated as well as convective heat transfer from the outer wall to the incoming air. Also, conduction from the outer wall to a constant temperature node deep in the concrete or in the ground is handled. With perfect insulation on the finned shell and little heat transfer into the concrete, the calculation of air and wall temperatures on the down-comer side would be unnecessary; the air temperature at the bottom of the up-flow side could be set to the outside air inlet temperature. The model with heat transfer to the incoming air allows consideration of the impact on RVACS-RACS performance of limited insulation between the downcomer and the up-flow side.

As indicated in Fig. 5.4-4, one axial node is included above the liquid level in the vessel. This node is included to account for heat transfer through the finned shell between incoming and outgoing air. For this node, the heat transfer coefficient between the reactor vessel and the guard vessel is set to zero.

#### 5.4.7.2.1 Basic Equations

##### A. Wall Temperatures

##### 1. Reactor Vessel

The basic equation for the reactor vessel wall temperature is

$$m_{w1} C_{w1} \frac{dT_{w1}}{dt} = h_{NW1} A_{NW1} (T_N - T_{w1}) + h_{w12} A_{w12} (T_{w2} - T_{w1}) \quad (5.4-85)$$

where

$m_{w1}$  = reactor vessel wall mass per unit length

$C_{w1}$  = reactor vessel specific heat

$T_{w1}$  = reactor vessel wall temperature

$t$  = time

$h_{NW1}$  = heat transfer coefficient between the sodium inside the vessel and the vessel wall

$A_{NW1}$  = inner perimeter, or heat transfer area per unit length, of the vessel wall

$T_N$  = sodium temperature

$h_{w12}$  = heat transfer coefficient between the reactor vessel and the guard vessel, including both radiation and convection terms

$A_{w12}$  = outer perimeter of the reactor vessel

$T_{w2}$  = guard vessel temperature

## 2. Guard Vessel

$$m_{w2} C_{w2} \frac{dT_{w2}}{dt} = h_{w12} A_{w12} (T_{w1} - T_{w2}) + h_{w23} A_{w23} (T_{w3} - T_{w2}) + h_{w2a} A_{w2a} (T_{a1} - T_{w2}) \quad (5.4-86)$$

where

$m_{w2}$  = guard vessel mass per unit length

$C_{w2}$  = specific heat of the guard vessel

$h_{w23}$  = radiation heat transfer coefficient between the guard vessel and the finned shell

$A_{w23}$  = outer perimeter of the guard vessel

$T_{w3}$  = temperature of the finned shell inner node

$T_{a1}$  = air temperature in the up-flow section

$h_{w2a}$  = heat transfer coefficient between the guard vessel and air

$A_{w2a}$  = perimeter between the guard vessel and the air

## 3. Finned shell, Inner Node

$$m_{w3} C_{w3} \frac{dT_{w3}}{dt} = h_{w34} A_{w34} (T_{w4} - T_{w3}) + h_{w23} A_{w23} (T_{w2} - T_{w3}) + h_{w3a} A_{w3a} (T_{a1} - T_{w3}) \quad (5.4-87)$$

$m_{w3}$  = mass per unit length for the inside of the finned shell

$C_{w3}$  = specific heat of the inside of the finned shell

$T_{w3}$  = finned shell inner node temperature

$h_{w34}$  = heat transfer coefficient between the inside and outside of the finned shell

$A_{w34}$ = perimeter of the finned shell

$h_{w3a}$ = heat transfer coefficient between the finned shell and the upflowing air

$A_{w3a}$ = perimeter between the finned shell and the upflowing air

$T_{w4}$ = finned shell outer node temperature

#### 4. Finned Shell, Outer Node

$$m_{w4} C_{w4} \frac{dT_{w4}}{dt} = h_{w45} A_{w45} (T_{w5} - T_{w4}) + h_{w34} A_{w4} (T_{w3} - T_{w4}) + h_{w4a} A_{w4a} (T_{a2} - T_{w4}) \quad (5.4-88)$$

$m_{w4}$ = mass per unit length for the outside of the finned shell

$C_{w4}$ = specific heat of the outside of the finned shell

$T_{w5}$ = concrete wall inner node temperature

$h_{w45}$ = heat transfer coefficient between the finned shell and the concrete

$A_{w45}$ = finned shell outer perimeter

$T_{a2}$ = temperature of the down-flowing air

$h_{w4a}$ = heat transfer coefficient between the finned shell outer surface and the down-flowing air

$A_{w4a}$ = perimeter between the finned shell outer surface and the down-flowing air

#### 5. Cavity Wall, Inner Node

$$m_{w5} C_{w5} \frac{dT_{w5}}{dt} = h_{w56} A_{w56} (T_{w6} - T_{w5}) + h_{w45} A_{w45} (T_{w4} - T_{w5}) + h_{w5a} A_{w5a} (T_{a2} - T_{w5}) \quad (5.4-89)$$

$m_{w5}$ = mass per unit length for the inner cavity wall node

$C_{w5}$ = specific heat of the inside of the cavity wall

$T_{w6}$ = constant temperature deep in the concrete or ground

$h_{w56}$  = heat transfer coefficient between the cavity wall surface node and the constant temperature heat sink

$A_{w56}$  = cavity wall perimeter

$h_{w5a}$  = heat transfer coefficient from the cavity wall to the down-flowing air

## B. Air Temperature

The air is treated with a quasi-static approximation, neglecting the time derivative of the air temperature and density.

### 1. Up-Flowing Air Between the Guard Vessel and the Finned Shell

$$w_a C_a \frac{dT_{a1}}{dz} = h_{wa2} A_{wa2} (T_{w2}(z) - T_{a1}(z)) + h_{wa3} A_{wa3} [T_{w3}(z) - T_{a1}(z)] \quad (5.4-90)$$

where

$w_a$  = air mass flow rate, kg/sec.

$C_a$  = air specific heat

### 2. Down-Flowing Air Between the Finned Shell and the Concrete Wall

$$W_a C_a \frac{dT_{a2}}{dz} = h_{wa4} A_{wa4} (T_{w4}(z) - T_{a2}(z)) + h_{wa5} A_{wa5} [T_{w5}(z) - T_{a2}(z)] \quad (5.4-91)$$

## C. Air Flow Rate

As shown in Fig. 5.4-4, the air flow path is modeled as an inlet section, a downflow section between the outer surface of the finned shell and the cavity wall, an upflow section between the guard vessel and the inner surface of the finned shell, and an outlet stack. The air temperature in the inlet section is assumed to be a constant value, equal to the external air temperature. In the stack, the air temperature is assumed to equal the value at the outlet from the guard vessel-finned shell region. Between the inlet section and the stack, the air temperature is calculated on a node-by-node basis.

The inertia of the air is ignored, and the air flow rate is calculated by balancing the air gravity head with the loss terms.

$$\Delta p_{gr} = \Delta p_{loss} \quad (5.4-92)$$

The gravity head,  $\Delta p_{gr}$ , is calculated as

$$\Delta p_{gr} = \int \rho g dz = g \left\{ -\rho_{in} \Delta z_{in} + \rho_{stack} \Delta z_{stack} - \sum_j \bar{\rho}_{a2}(j) \Delta z_j \right. \\ \left. + \sum_j \bar{\rho}_{a1}(j) \Delta z_j - \rho_{in} (z_{st} - z_{in}) \right\} \quad (5.4-93)$$

where

$g$  = acceleration of gravity

$\rho_{in}$  = air density in the inlet section

$\rho_{stack}$  = air density in the stack

$\bar{\rho}_{a1}(j)$  = average air density in node  $j$  for upflow air

$\Delta z_j$  = height of node  $j$

$\Delta z_{in}$  = elevation gain in the inlet

$\Delta z_{stack}$  = height of the stack

$z_{st}$  = elevation of the stack outlet

and

$z_{in}$  = inlet elevation

The loss term,  $\Delta p_{loss}$ , is calculated as

$$\Delta p_{loss} = \sum \frac{w_a^2}{2\rho_a A_a^2} \left( f \frac{L}{D_h} + k_{or} \right) \quad (5.4-94)$$

where

$w_a$  = air flow rate

$A_a$  = air flow area

$f$  = friction factor

$k_{or}$  = orifice coefficient

$L$  = length of the section

$D_h$  = hydraulic diameter

The summation is over the inlet section, the stack, and each of the nodes between.

The friction factor is calculated as

$$f = \begin{cases} A_{fr} Re^b & \text{if } Re \geq Re_t \\ \frac{A_{fl}}{Re} & \text{if } Re < Re_t \end{cases} \quad (5.4-95)$$

where

$$Re = \text{Reynoldsnumber} = \frac{D_h w_a}{\mu_a A_a} \quad (5.4-96)$$

$\mu_a$  = viscosity

$A_{fr}$  = user-supplied turbulent friction factor coefficient

$b$  = user-supplied coefficient, and

$A_{fl}$  = user-supplied laminar friction factor coefficient

The value of  $Re_t$ , the Reynolds number for the transition from turbulent to laminar, is calculated by the code to make the friction factor continuous at the transition point.

$$A_{fr} Re_t^b = \frac{A_{fl}}{Re_t} \quad (5.4-97)$$

Or

$$Re_t = \left( \frac{A_{fl}}{A_{fr}} \right)^{\frac{1}{1+b}} \quad (5.4-98)$$

## D. Heat Transfer Coefficients

### 1. Reactor Vessel (RV) to Guard Vessel (GV)

$$\frac{1}{h_{w12}} = \frac{R_{w1}}{2} + \frac{1}{h_{cv12} + \varepsilon_{12} \sigma (T_{w1} + T_{w2}) (T_{w1}^2 + T_{w2}^2)} + \frac{R_{w2}}{2} \quad (5.4-99)$$

The approximation is made that  $R_{w1}$  and  $R_{w2}$  are lumped in with  $h_{cv12}$ , so

$$h_{w12} = h_{cv12} + \varepsilon_{12} \sigma (T_{w1} + T_{w2}) (T_{w1}^2 + T_{w2}^2) \quad (5.4-100)$$

where

$$\varepsilon_{12} = \frac{1}{\frac{1}{\varepsilon_{RV}} + \frac{1}{\varepsilon_{GVI}} - 1} \quad (5.4-101)$$

and

$h_{cv12}$  = user-supplied convective heat transfer coefficient, RV to GV

$\varepsilon_{RV}$  = emissivity of the reactor vessel wall

$\varepsilon_{GVI}$  = emissivity of the guard vessel inner surface

$\sigma$  = Stefan-Boltzmann Constant

$$R_{w1} = G_{RV} / k_{RV} \quad (5.4-102)$$

$$R_{w2} = G_{GV} / k_{GV} \quad (5.4-103)$$

$G_{RV}$  = thickness of the reactor vessel

$G_{GV}$  = thickness of the guard vessel

$k_{GV}$  = thermal conductivity of guard vessel

$k_{RV}$  = thermal conductivity of the reactor vessel

## 2. GV to Finned Shell (FS)

$$\frac{1}{h_{w23}} = \frac{R_{w2}}{2} + \frac{1}{\varepsilon_{23} \sigma (T_{w2} + T_{w3}) (T_{w2}^2 + T_{w3}^2)} \quad (5.4-104)$$

$R_{w2}$  is neglected, so

$$h_{w23} = \varepsilon_{23} \sigma (T_{w2} + T_{w3}) (T_{w2}^2 + T_{w3}^2) \quad (5.4-105)$$

$$\varepsilon_{23} = \frac{1}{\frac{1}{\varepsilon_{GVO}} + \frac{1}{\varepsilon_{FSI}} - 1} \quad (5.4-106)$$

$\varepsilon_{GVO}$  = emissivity of the guard vessel outer surface

$\varepsilon_{FSI}$  = emissivity of the finned shell inner surface

### 3. Finned Shell

$h_{w34}$  = a constant, user-supplied, conduction coefficient

### 4. Outer Wall

$$\frac{1}{h_{w45}} = R_{w5} + \frac{1}{\epsilon_{45} \sigma (T_{w4} + T_{w5})(T_{w4}^2 + T_{w5}^2)} \quad (5.4-107)$$

or

$$h_{w45} = \frac{\epsilon_{45} \sigma (T_{w4} + T_{w5})(T_{w4}^2 + T_{w5}^2)}{1 + R_{w5} \epsilon_{45} \sigma (T_{w4} + T_{w5})(T_{w4}^2 + T_{w5}^2)} \quad (5.4-108)$$

where

$$\epsilon_{45} = \frac{1}{\frac{1}{\epsilon_{FSO}} + \frac{1}{\epsilon_{OW}} - 1} \quad (5.4-109)$$

$R_{w5}$  = thermal resistance in the outer wall to the location of  $T_{w5}$

$h_{w56}$  = a constant, user supplied, conduction coefficient

$\epsilon_{FSO}$  = emissivity of the finned shell outer surface

$\epsilon_{OW}$  = emissivity of the outer wall

### 5. Air

$$\frac{1}{h_{w2a}} = \frac{R_{w2}}{2} + \frac{1}{h_{a1}} \quad (5.4-110)$$

again  $R_{w2a}$  is neglected, so

$$h_{w2a} = h_{a1} \quad (5.4-111)$$

where the air heat transfer coefficient is

$$h_{a1} = \frac{ka}{D_{hal}} N_{u1} \quad (5.4-112)$$

$k_a$  = air thermal conductivity

$D_{ha1}$  = hydraulic diameter between the guard vessel and the finned shell

$N_{u1}$  = Nusselt number

$$N_{u1} = \begin{cases} C_1 R_{e1}^{C_2} P_{r1}^{0.4} & \text{if } Re_1 \geq Re_t \\ C_3 & \text{if } Re_1 < Re_t \end{cases} \quad (5.4-113)$$

$$Re_1 = \frac{D_{ha1} w_a}{\rho_{a1} \mu_{a1}} \quad (5.4-114)$$

$w_a$  = air flow rate

$\rho_{a1}$  = air density

$\mu_{a1}$  = air viscosity

$P_{r1}$  = Prandtl number

$C_1, C_2, C_3$  = user supplied correlation coefficients

$C_1 \sim .023$

$C_2 \sim 0.8$

$C_3 \sim 3-8$

$Re_t$  = Reynolds number for transition from turbulent to laminar heat transfer

Also,

$$h_{w3a} = h_{a1} \quad (5.4-115)$$

$$h_{w4a} = h_{a2} \quad (5.4-116)$$

$$\frac{1}{h_{w5a}} = R_{w5} + \frac{1}{h_{a2}} \quad (5.4-117)$$

or

$$h_{w5a} = \frac{h_{a2}}{1 + R_{w5} h_{a2}} \quad (5.4-118)$$

$$h_{w5a} = \frac{k_{a2}}{D_{haz}} N_{u2} \quad (5.4-119)$$

## E. Air Properties

Correlations used for air thermal properties are:

Density

$$\rho_a = .0003985 + 352.25/T + 207.89/T^2 \quad \text{kg} / \text{m}^3 \quad (5.4-120)$$

where

$T$  = air temperature ( $k$ )

Specific Heat

$$C_{pa} = 972.93 + .086802T + 7.8654 \times 10^{-5} T^2 \quad \text{J} / \text{kg} - \text{K} \quad (5.4-121)$$

Prandtl Number:

$$P_r^{0.2} = .89887 - 1.1906 \times 10^{-4} T + 8.6681 \times 10^{-8} T^2 \quad (5.4-122)$$

Viscosity

$$\mu = 3.872 \times 10^{-6} + 5.332 \times 10^{-8} T - 1.531 \times 10^{-11} T^2 \quad \text{kg} / \text{m} - \text{s} \quad (5.4-123)$$

$$\mu^{0.2} = .09628 + 6.358 \times 10^{-5} T - 2.633 \times 10^{-8} T^2 \quad (5.4-124)$$

Thermal Conductivity:

$$k = 2.085 \times 10^{-3} + 8.820 \times 10^{-5} T - 2.304 \times 10^{-8} T^2 \quad \text{W} / \text{m} - \text{K} \quad (5.4-125)$$

These correlations agree with the tables of dry air properties on page 522 of Ref. 5-6 over the range from 255.4 K (0°F) to 1088.7 K (1500°F) to within .2% for  $\rho_a$ , .34% for  $C_{pa}$ , .03% for  $P_r^{0.2}$ , 1% for  $\mu$ , .4% for  $\mu^{0.2}$ , and 2.7% for  $k$ .

### 5.4.7.2.2 Finite Difference Solution

#### A. Finite Difference Equations

## 1. Reactor Vessel

The coupling between the RVACS/RACS model and the rest of SASSYS takes place at the reactor vessel wall. Heat transfer from the vessel wall to the guard vessel is treated using the component-to-component heat transfer capability of the code. Equation 5.4-85 becomes

$$m_{w1} C_{w1} \frac{dT_{w1}}{dt} = h_{Nw1} A_{Nw1} (T_N - T_{w1}) + h_{snk} A_{snk} (T_{snk} - T_{w1}) \quad (5.4-126)$$

where  $h_{snk}$ ,  $A_{snk}$ , and  $T_{snk}$  are the sink heat transfer coefficient, area, and temperature used in the component-to-component heat transfer treatment. At the beginning of each time step these values are re-set as

$$h_{snk} = h_{w12} \quad (5.4-127)$$

$$A_{snk} = A_{w12} \Delta z_j \quad (5.4-128)$$

$$T_{snk} = T_{w2} \quad (5.4-129)$$

for node  $j$ . The vessel wall is modeled as a combination of compressible volume walls, pipe walls, and annular element walls; and the thermal treatments for these components solve Eq. 5.4-126.

## 2. Air Temperature

A treatment similar to the log-mean temperature difference treatment is used for the air. Fully implicit time differencing is used in the sense that the values used for  $T_{w2}$  and  $T_{w3}$  in Eq. 5.4-90 are the values at the end of the time step. Also, it is assumed that  $T_{w2}$  and  $T_{w3}$  are constant across a node. For node  $j$ , which extends from  $z_j$  to  $z_{j+1}$ , the solution of Eq. 5.4-90 then becomes

$$T_{a1}(z') = T'_{a1} + (T_{a1}(0) - T'_{a1}) \varepsilon^{-\lambda_1 z'} \quad (5.4-130)$$

where

$$z' = z - z_j \quad (5.4-131)$$

$$T'_{a1} = f_2 T_{w3j} + f_3 T_{w2j} \quad (5.4-132)$$

$$f_2 = \frac{h_{w2a} A_{w2a}}{h_{w2a} A_{w2a} + h_{w3a} A_{w3a}} \quad (5.4-133)$$

$$f_3 = 1 - f_2 \quad (5.4-134)$$

and

$$\lambda_1 = \frac{h_{w2a} A_{w2a} + h_{w3a} A_{w3a}}{w_a C_a} \quad (5.4-135)$$

The heat flow from the guard vessel to the air in node  $j$  is then

$$q_{2aj} = \int_{z'=0}^{\Delta z_j} h_{w2a} A_{w2a} [T_{w2j} - T_a(z')] dz' \quad (5.4-136)$$

where

$$\Delta z_j = z_{j+1} - Z_j \quad (5.4-137)$$

Combining Eq. 5.4-130 and Eq. 5.4-136 gives

$$q_{2aj} = h_{w2a} A_{w2a} \left\{ \Delta z_j (T_{w2j} - T'_{a1}) + \frac{[T'_{a1} - T_{a1}(0)](1 - e^{-\lambda_1 \Delta z_j})}{\lambda_1} \right\} \quad (5.4-138)$$

The wall temperatures are solved for simultaneously with the air temperatures, so one uses

$$T_{w2j}(t + \Delta t) = T_{w2j}(t) + \Delta T_{w2j} \quad (5.4-139)$$

and

$$T_{w3j}(t + \Delta t) = T_{w3j}(t) + \Delta T_{w3j} \quad (5.4-140)$$

where the time step size is  $\Delta t$ . Then Eq. 5.4-138 has the form

$$a_{2aj} = b_o + b_2 \Delta T_{w2j} + b_3 \Delta T_{w3j} \quad (5.4-141)$$

where

$$b_0 = h_{w2a} A_{w2a} \left\{ \Delta z_j [T_{w2j}(t) - T'_{a1}(t)] + \frac{[T'_{a1}(t) - T_{a1}(0)]}{\lambda_1} (1 - e^{-\lambda_1 \Delta z_j}) \right\} \quad (5.4-142)$$

$$b_2 = h_{w2a} A_{w2a} \left\{ \Delta z_j (1 - f_2) + \frac{f_2}{\lambda_1} (1 - e^{-\lambda_1 \Delta z_j}) \right\} \quad (5.4-143)$$

and

$$b_3 = h_{w2a} A_{w2a} \left\{ \Delta z_j (1 - f_3) - \frac{f_3}{\lambda_1} (1 - e^{-\lambda_1 \Delta z_j}) \right\} \quad (5.4-144)$$

Similarly,

$$q_{3aj} = C_o + C_2 \Delta T_{w2j} + C_3 \Delta T_{w3j} \quad (5.4-145)$$

$$q_{4aj} = d_o + d_4 \Delta T_{w4j} + d_5 \Delta T_{w5j} \quad (5.4-146)$$

and

$$q_{5aj} = e_o + e_4 \Delta T_{w4j} + e_5 \Delta T_{w5j} \quad (5.4-147)$$

where

$$C_o = h_{w3a} A_{w3a} \left\{ \Delta z_j [T_{w3j}(t) - T'_{a1}(t)] + \frac{T'_{a1}(t) - T_{a1}(0)}{\lambda_1} (1 - e^{-\lambda_1 \Delta z_j}) \right\} \quad (5.4-148)$$

$$C_2 = -h_{w3a} A_{w3a} \left\{ \Delta z_j f_2 - \frac{f_2}{\lambda_1} (1 - e^{-\lambda_1 \Delta z_j}) \right\} \quad (5.4-149)$$

$$C_3 = -h_{w3a} A_{w3a} \left\{ \Delta z_j (1 - f_3) + \frac{f_3}{\lambda_1} (1 - e^{-\lambda_1 \Delta z_j}) \right\} \quad (5.4-150)$$

$$d_o = h_{w4a} A_{w4a} \left\{ \Delta z_j [T_{w4j}(t) - T'_{a2}(t)] + \frac{[T'_{a2}(t) - T_{a2}(\Delta z)] (1 - e^{-\lambda_2 \Delta z_j})}{\lambda_2} \right\} \quad (5.4-151)$$

$$d_4 = h_{w4a} A_{w4a} \left\{ \Delta z_j (1 - f_4) + \frac{f_4}{\lambda_2} (1 - e^{-\lambda_2 \Delta z_j}) \right\} \quad (5.4-152)$$

$$d_5 = -h_{w4a} A_{w4a} \left\{ \Delta z_j f_5 - \frac{f_5}{\lambda_2} (1 - e^{-\lambda_2 \Delta z_j}) \right\} \quad (5.4-153)$$

$$e_0 = h_{w5a} A_{w5a} \left\{ \Delta z_j [T_{w5}(t) - T'_{a2}(t)] + \frac{[T'_{a2}(t) - T_{a2}(\Delta z)](1 - e^{-\lambda_2 \Delta z_j})}{\lambda_2} \right\} \quad (5.4-154)$$

$$e_4 = -h_{w5a} A_{w5a} \left\{ \Delta z_j f_4 - \frac{f_4}{\lambda_2} (1 - e^{-\lambda_2 \Delta z}) \right\} \quad (5.4-155)$$

$$e_5 = h_{w5a} A_{w5a} \left\{ \Delta z_j (1 - f_5) + \frac{f_5}{\lambda_2} (1 - e^{-\lambda_2 \Delta z}) \right\} \quad (5.4-156)$$

$$T'_{a2} = f_4 T_{w4j} + f_5 T_{w5j} \quad (5.4-157)$$

$$f_4 = \frac{h_{w4a} A_{w4a}}{h_{w4a} A_{w4a} + h_{w5a} A_{w5a}} \quad (5.4-158)$$

and

$$f_5 = 1 - f_4 \quad (5.4-159)$$

### 3. Guard Vessel

Fully implicit time differencing is used for the wall temperature. Equation 5.4-86 becomes

$$\begin{aligned} \Delta z_j m_{w2} C_{w2} \Delta \frac{T_{w2}}{\Delta t} = & h_{w12} A_{w12} [T_{w1}(t + \Delta t) - T_{w2}(t) - \Delta T_{w2}] \Delta z_j \\ & + h_{w23} A_{w23} [T_{w3}(t) + \Delta T_{w3} - T_{w2}(t) - \Delta T_{w2}] \Delta z_j - q_{2aj} \end{aligned} \quad (5.4-160)$$

or

$$\alpha_2 \Delta T_{w2} + \alpha_3 \Delta T_{w3} = \alpha_0 \quad (5.4-161)$$

where

$$\alpha_2 = 1 + \frac{h_{w12} A_{w12} \Delta t}{m_{m2} C_{w2}} + \frac{h_{h23} A_{w23} \Delta t}{m_{w2} C_{w2}} + \frac{b_2}{\Delta z_j} \frac{\Delta t}{m_{w2} C_{w2}} \quad (5.4-162)$$

$$\alpha_3 = -\frac{h_{w23} A_{w23} \Delta t}{m_{w2} C_{w2}} + \frac{b_3}{m_{w2} C_{w2}} \frac{\Delta t}{m_{w2} C_{w2}} \quad (5.4-163)$$

and

$$\alpha_0 = \frac{\Delta t}{m_{m2} C_{w2}} \left\{ h_{w12} A_{w12} [T_{w1}(t + \Delta t) - T_{w2}(t)] \right. \\ \left. + h_{w23} A_{w23} [T_{w3}(t) - T_{w2}(t)] - \frac{b_o}{\Delta z_j} \right\} \quad (5.4-164)$$

#### 4. Finned Shell, Inner Node

Equation 5.4-87 becomes

$$\beta_3 \Delta T_{w2} + \beta_3 \Delta T_{w3} + \beta_4 \Delta T_{w4} = \beta_0 \quad (5.4-165)$$

where

$$\beta_0 = \frac{\Delta t}{m_{m3} C_{w3}} \left\{ h_{w34} A_{w34} [T_{w4}(t) - T_{w3}(t)] \right. \\ \left. + h_{w23} A_{w23} [T_{w2}(t) - T_{w3}(t)] - \frac{C_o}{\Delta z_j} \right\} \quad (5.4-166)$$

$$\beta_2 = \frac{\Delta t}{m_{w3} C_{w3}} \left( \frac{C_3}{\Delta z_j} - h_{w23} A_{w23} \right) \quad (5.4-167)$$

$$\beta_3 = 1 + \frac{\Delta t}{m_{w3} C_{w3}} \left( h_{w34} A_{w34} + h_{w23} A_{w23} + \frac{C_3}{\Delta z_j} \right) \quad (5.4-168)$$

$$\beta_4 = -\frac{\Delta t h_{w34} A_{w34}}{m_{w3} C_{w3}} \quad (5.4-169)$$

### 5. Finned Shell, Outer Node

Equation 5.4-88 becomes

$$\gamma_3 \Delta T_{w3} + \gamma_4 \Delta T_{w4} + \gamma_5 \Delta T_{w5} = \gamma_0 \quad (5.4-170)$$

where

$$\gamma_0 = \frac{\Delta t}{m_{m4} C_{w4}} \left\{ h_{w45} A_{w45} [T_{w5}(t) - T_{w4}(t)] + h_{m34} A_{w34} [T_{w3}(t) - T_{w4}(t)] - \frac{d_o}{\Delta z_j} \right\} \quad (5.4-171)$$

$$\gamma_3 = -\frac{\Delta t}{m_{w4} C_{w4}} h_{w34} A_{w34} \quad (5.4-172)$$

$$\gamma_4 = 1 + \frac{\Delta t}{m_{m4} C_{w4}} \left( \frac{d_4}{\Delta z_j} + h_{w34} + A_{w34} + h_{w45} A_{w45} \right) \quad (5.4-173)$$

and

$$\gamma_5 = -\frac{\Delta t}{m_{m4} C_{w4}} \left( \frac{d_5}{\Delta z_j} - h_{w45} A_{w45} \right) \quad (5.4-174)$$

### 6. Cavity Wall, Inner Node

Equation 5.4-89 becomes

$$\xi_4 \Delta T_{w4} + \xi_5 \Delta T_{w5} = \xi_0 \quad (5.4-175)$$

where

$$\xi_0 = \frac{\Delta t}{m_{w5} C_{w5}} \left\{ h_{w45} A_{w45} [T_{w4}(t) - T_{w5}(t)] + h_{w56} A_{w56} [T_{w56}(t) - T_{w5}(t)] - \frac{e_o}{\Delta z_j} \right\} \quad (5.4-176)$$

$$\xi_4 = \frac{\Delta t}{m_{w5} C_{w5}} \left( \frac{e_4}{\Delta z_j} - h_{w45} A_{w45} \right) \quad (5.4-177)$$

and

$$\xi_5 = 1 + \frac{\Delta t}{m_{w5} C_{w5}} \left( \frac{e_5}{\Delta z_j} + h_{w45} A_{w45} + h_{w56} A_{w56} \right) \quad (5.4-178)$$

## 7. Cavity Wall, Outer Node

The sink temperature is assumed to be constant:

$$T_{w6} = \text{constant}$$

### B. Solution of Finite Difference Equations

#### 1. Simultaneous Solution of Equations

##### a. Down-Flowing Air, Transient Solution

The finned shell outer node and cavity wall inner node temperatures are solved for simultaneously, using an estimated value for the finned shell inner node temperature. Solving Eq. 5.4-170 and 5.4-175 simultaneously gives

$$\Delta T_{w4} = \frac{\gamma_o \xi_5 - \gamma_5 \xi_o}{\gamma_4 \xi_5 - \gamma_5 \xi_4} \quad (5.4-179)$$

Then Eq. 5.4-175 can be solved for  $\Delta T_{w5}$ .

##### b. Up-Flowing Air, Transient Solution

The guard vessel and finned shell inner node temperatures are solved for simultaneously, using an estimate value for the finned shell outer node temperature. Solving Eq. 5.4-161 and 5.4-160 simultaneously gives

$$\Delta T_{w3} = \frac{\alpha_3 \beta_o - \alpha_o \beta_3}{\alpha_3 \beta_2 - \alpha_2 \beta_3} \quad (5.4-180)$$

Then Eq. 5.4-180 is solved for  $\Delta T_{w3}$ .

## 2. Solution Method

No steady-state solution for the RVACS/RACS has been coded; the initial steady-state results are obtained by running a null transient. First the air and wall temperatures are set to the air inlet temperature. Then the null transient is run to set the initial steady-state temperatures and flow rate. Finally the regular transient is run. During the null transient the core channel calculations are bypassed, the sodium flow rates are held constant, the inlet and outlet plenum temperatures are held constant, and temperatures and air flow rates are calculated for the rest of the sodium and the RVACS/RACS. The routines used in the null transient are the same as those used in the regular transient.

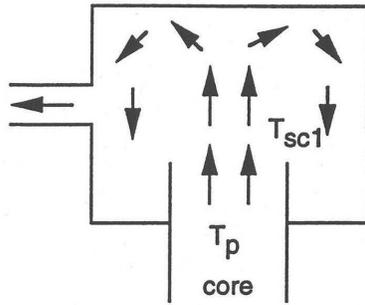
An iteration on air flow rate is used in the transient solution. An air flow rate is assumed, temperatures are calculated for this air flow rate, the air gravity head is calculated, and the air pressure loss is calculated. Then the gravity head is compared with the pressure loss; and if the two do not balance, another air flow rate is tried. The iteration on air flow rate continues until a balance is achieved. The temperature calculation starts at the inlet and works down the inlet side then back up the air upflow side. The heat transfer coefficients and the heat transfer across the finned shell are calculated based on conditions at the beginning of the time step. The rest of the calculation is fully implicit in its time differencing.

### 5.4.8 Stratified Volume Model

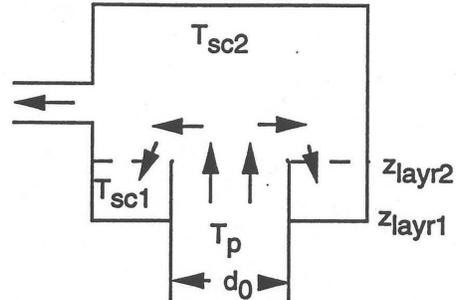
In addition to the uniform mixing compressible volume model described in Section 5.4.4, PRIMAR-4 contains a stratified temperature model for the liquid in a compressible volume. This stratified model can be used for an outlet plenum and/or for a pool. This model borrows from the PLENUM-2A model [5-7] of Howard and Lorenz, but the PRIMAR-4 model has been extended beyond the capabilities of the PLENUM-2A model. Borrowed from PLENUM-2A is the concept of a small number of distinct temperature regions in the coolant, separated by horizontal interfaces. Also, borrowed are the concept of distinct stages in the calculation, a plume height correlation, and a correlation for interface rise due to entrainment of a hot layer into a cooler plume rising from the core outlet. One extension of the PRIMAR-4 model is the provision for handling up transients as well as down transients: PLENUM-2A will only handle transients in which the core outlet temperature is cooler than the plenum temperature, whereas the PRIMAR-4 model will also handle transients in which the core outlet temperature is hotter than the plenum temperature. Another extension is the option in the SAS4A model to handle a horizontal discharge from an IHX into a cold pool: PLENUM-2A will only handle a vertical discharge from the core into an outlet plenum. The SAS4A model handles up to three regions and five stages, whereas PLENUM-2A considers only two regions and three stages. Also, the PRIMAR-4 model treats thermal conduction across the interface between regions, and this model includes detailed multi-node wall temperature calculations.

Figure 5.4-5 shows the various stages and cases considered in this stratified model. At the start of a transient in which the core outlet temperature is dropping, the plume in

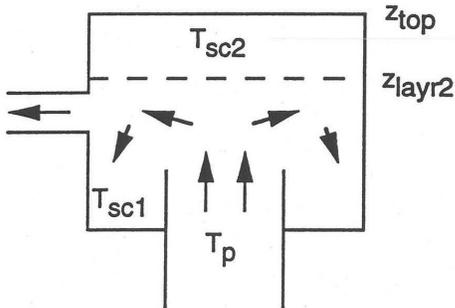
the outlet plenum goes to the top of the plenum; and the outlet plenum is fully mixed, giving stage 1. As the core outlet temperature and velocity drop, the plume no longer reaches the top of the plenum. This leads to the start of stage 2 in which the outlet coolant goes to layer 1. In stage 2 the layer boundary is at the elevation of the core outlet. After enough cool liquid has entered layer 1 to fill one quarter of its volume, stage 3, case 3.1 begins. In this case, the plume coolant still goes to layer 1, but the interface between layers rises as liquid is added to layer 1. In this case, the plume also entrains hot liquid from the interface into layer 1. If the core outlet temperature at the start of the transient becomes hotter than the outlet plenum temperature, then stage 3, case 3.2 is entered. In this case, the core outlet coolant goes to a top hot layer, entraining cool outlet plenum liquid as it passes through. The three layer cases of stages 4 and 5 can occur in the later stages of a transient if the core outlet temperature starts out rising and later falls, or if the core outlet temperature starts out falling and later rises. If the coolant inlet into the volume is horizontal, as in the discharge of an IHX into a cold pool, then only stages 1, 3, and 5 are used.



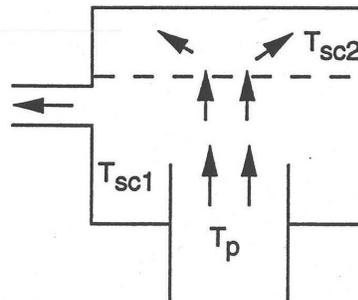
Stage 1, fully mixed, 1 layer



Stage 2, 2 layers, interface at the core outlet, filling the bottom with cool liquid



Stage 3, case 3.1, 2 layers, interface moving  $T_p < (T_{sc1} + T_{sc2})/2$ , entrainment at the interface, plume height  $< z_{top}$



Stage 3, case 3.2, 2 layers, interface moving,  $T_p > (T_{sc1} + T_{sc2})/2$ , hot outlet coolant goes to upper layer, entrains from the lower layer as it passes through

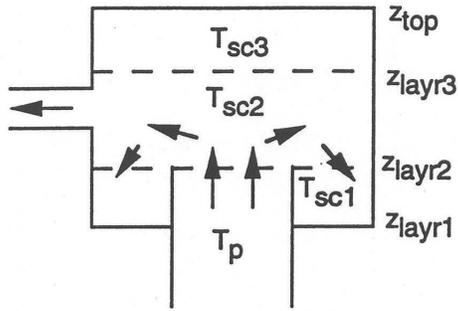
$T_p$  = plume temperature = core outlet temperature

$T_{sci}$  = temperature in layer  $i$

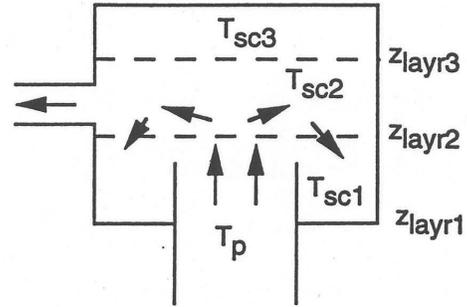
$z_{top}$  = elevation at top of plenum

$z_{layri}$  = interface elevation at bottom of layer  $i$

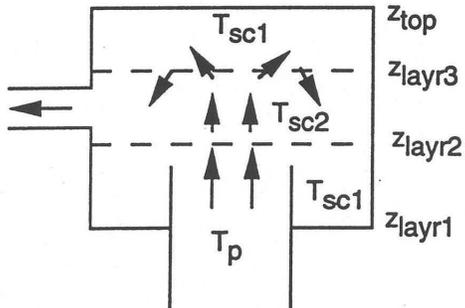
Figure 5.4-5. Stratified Volume Stages



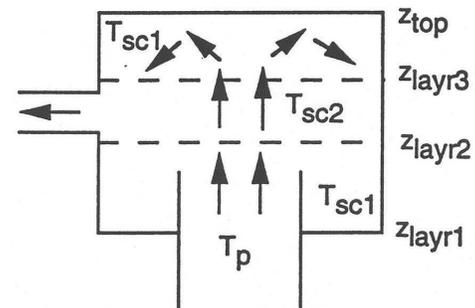
Stage 4, 3 layers, interface at the core outlet, filling the bottom with cool liquid, plume height <  $z_{layer3}$



Stage 5, case 5.1, 3 layers, interfaces moving, cool plume liquid goes to layer 1 entrainment from layer 2 at  $z_{layer2}$   
 $T_p < (T_{sc1} + T_{sc2})/2$



Strage 5, case 5.2, 3 layers, interfaces moving, plume passes through layer 1 to layer 2, entraining from layer 1 as it passes through, also entraining from layer 3 at  $z_{layer3}$   
 $T_{sc2} + T_{sc3} > 2T_p > T_{sc1} + T_{sc2}$



Stage 5, case 5.3, 3 layers, interfaces moving, plume passes through layers 1 and 2, goes to layer 3, entraining from layers 1 and 2 as it passes through  
 $T_p > (T_{sc2} + T_{sc3})/2$

Figure 5.4-5. Stratified Volume Stages (Continued)

The jet height or plume height is calculated from an equation given by Yang [5-8]:

$$h_{jet} = 1.0484 F_r^{.785} \quad (5.4-181)$$

where

$h_{jet}$ =height of the jet or the plane

$$F_r = \text{Froudenumber} = \frac{v_o^2 \rho_{plume}}{g r_o^2 (\rho_{plume} - \rho_{plenum})} \quad (5.4-182)$$

$v_o$ = core exit velocity

$\rho_{plume}$ = density of the plume

$\rho_{plenum}$ = density of the plenum

$r_o$ = core effective radius

$g$ = acceleration of gravity

For entrainment at an interface, Howard and Lorenz give

$$w_{ent} = .2\pi \rho_{plume} V_j d_j F_f^{-1.1} \quad (5.4-183)$$

where

$v_j$ = plume average velocity at the interface and

$d_j$ = plume effective diameter at the interface

$w_{ent}$ = entrainment rate (kg/s).

The values of  $v_j$  and  $d_j$  depend on elevation and on whether the interface occurs within the zone of flow establishment or in the zone of established flow. The elevation change,  $z_o$ , from the core outlet to the top of the zone of flow establishment is

$$z_o = \frac{r_o}{.111} \quad (5.4-184)$$

For  $z < z_o$ , or the zone of flow establishment:

$$\frac{v_j}{v_o} = \frac{.25 + .02095(z/d_o) + .003969(z/d_o)^2}{[1/2 + .1052(z/d_o)]^2} \quad (5.4-185)$$

and

$$\frac{d_j}{d_o} = 1 + .2104(z/d_o) \quad (5.4-186)$$

For  $z > z_o$ , or the zone of established flow,

$$\frac{v_j}{v_o} = \frac{2.018}{z/d_o} \quad (5.4-187)$$

and

$$\frac{d_j}{d_o} = .8649(z/d_o) \quad (5.4-188)$$

For the wall temperatures, multi-node treatments are used. The vertical wall around the outside of the outlet plenum or pool is treated with a number of vertical nodes. Each vertical node contains a number of lateral nodes, with coolant in contact with the first node. There is also an option to have another coolant compressible volume in contact with the last lateral node to account for heat transfer from a hot outlet plenum to a cold pool. The model has an option for a horizontal wall at the top or bottom of the plenum. This wall is handled with a 1-D multinode treatment. Again, the first node is in contact with the plenum liquid, and the last node can be in contact with the coolant in another compressible volume.

The input for this model is as follows:

Table 5.4-1. Input for the Stratified Volume Model

<b>Block</b>	<b>Location</b>	<b>Name</b>	<b>Meaning</b>
3	1313	NSTRCV	Number of stratified compressible volumes
3	1314-1316	ICVSTR(ICVST)	1 for vertical coolant inlet, as in an outlet plenum 2 for a horizontal coolant inlet
3	1320-1322	NUMWAL(ICVST)	Number of wall sections
3	1323-1325	IFSTWL(ICVST)	Wall number of the first wall section

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3	1326-1334	IWLHRZ(IW)	0 for a vertical wall 1 for a horizontal wall at the top of a CV 2 for a horizontal wall at the bottom of a CV
3	1335-1343	NVNDWL(IW)	Number of vertical nodes in a vertical wall. NVNDWL = 1 for a horizontal wall
3	1344-1352	NLNDWL(IW)	Number of lateral nodes in a wall section, Max. = 8
			Note: $\text{sum}(\text{NVNDWL} * \text{NLNDWL}) \leq 300$ .
3	1353-1361	ICV2WL(IW)	Number of the CV in contact with the outer side of the wall section. = 0 for an adiabatic outer boundary. If $\text{ICV2WL} > 38$ , $\text{ICV2WL}$ = the temperature of a constant temperature heat sink.
3	1362	IDBSTR	Debug flag for stratified temperature model = 0, no debug prints = 1, final results only = 5, everything
3	1363	ISTDBS	PRIMAR time step when stratified debug starts.
3	1364	ISTSTP	Stop the run at PRIMAR step ISTSTP. Not used if $\text{ISTSTP} = 0$ or $\text{NSTRCV} = 0$ .
3	1365	IFT16	Write out stratified CV output of file 16 every 1FT16 PRIMAR steps. No output if $\text{IFT16} = 0$ .
18	5008	RCORE	Core radius for use in Froude number.
18	5009-5017	HCSTWL(IW)	Coolant heat transfer coefficient at the inner surface of the wall section.
18	5027-5035	ASTWL(IW)	Area of the wall section
18	5036-5107	HINVWL(I,IW)	Thickness/thermal conductivity of node I in the wall section $I = 1 - 8$
18	5108-5179	XMCSTW(I,IW)	Mass x heat capacity of node I in the wall.

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18	5180-5182	ZINST(ICVST)	z of inlet, used only for a vertical inlet. Otherwise ZOUTEL(IELL) is used.
18	5183-5185	VOLBLI(ICVST)	Volume below the inlet, not used if there is a cover gas in the CV. In this case, the information is obtained from other input for the CV.
18	5186-5188	EPSTST(ICVST)	Minimum temperature difference for switching stages.
18	5189-5191	XLENTTR(ICVST)	Entrainment length. A hot plume with a flow rate $W_h$ , rising through a cool layer of thickness $dz$ , will entrain cool liquid at a rate $(dz/XLENTTR) \times W_h$ .

---

## 5.5 Steam Generator

In addition to the simple table look-up steam generator model described in Section 5.4.6, SASSYS-1 contains a detailed steam generator model. The simple table look-up model is intended for cases in which the steam generator behavior has little impact on the results of a transient. The detailed model is intended for cases in which changes in the steam generator heat removal rate and/or gravity head have a significant impact on the results. Also, the detailed steam generator is required when the balance of plant model is used. The detailed steam generator model is described in Chapter 7.

## 5.6 Air Dump Heat Exchanger

SASSYS-1 contains a simple quasi-static sodium-to-air heat exchanger model. This model is intended mainly for use with a DRACS (Direct Reactor Auxiliary Cooling System), as shown in Fig. 5.6-1; but it could also be used for the air dump heat exchangers on the FFTF reactor. Currently, only sodium can be used in the DRACS loop, but soon an option to use *NaK* in the DRACS loop will be added to the code.

The model can handle either forced convection or natural circulation air flow. For a forced convection case, the user supplies the air flow as a function of time. For a natural circulation case, the user supplies a table of the stack inlet loss coefficient vs. time. The stack inlet loss coefficient represents the shutter opening, which may change during the transient.

A log mean temperature model is used for both the sodium side and the air side of the heat exchanger. Heat capacity of the tube structural material is ignored.

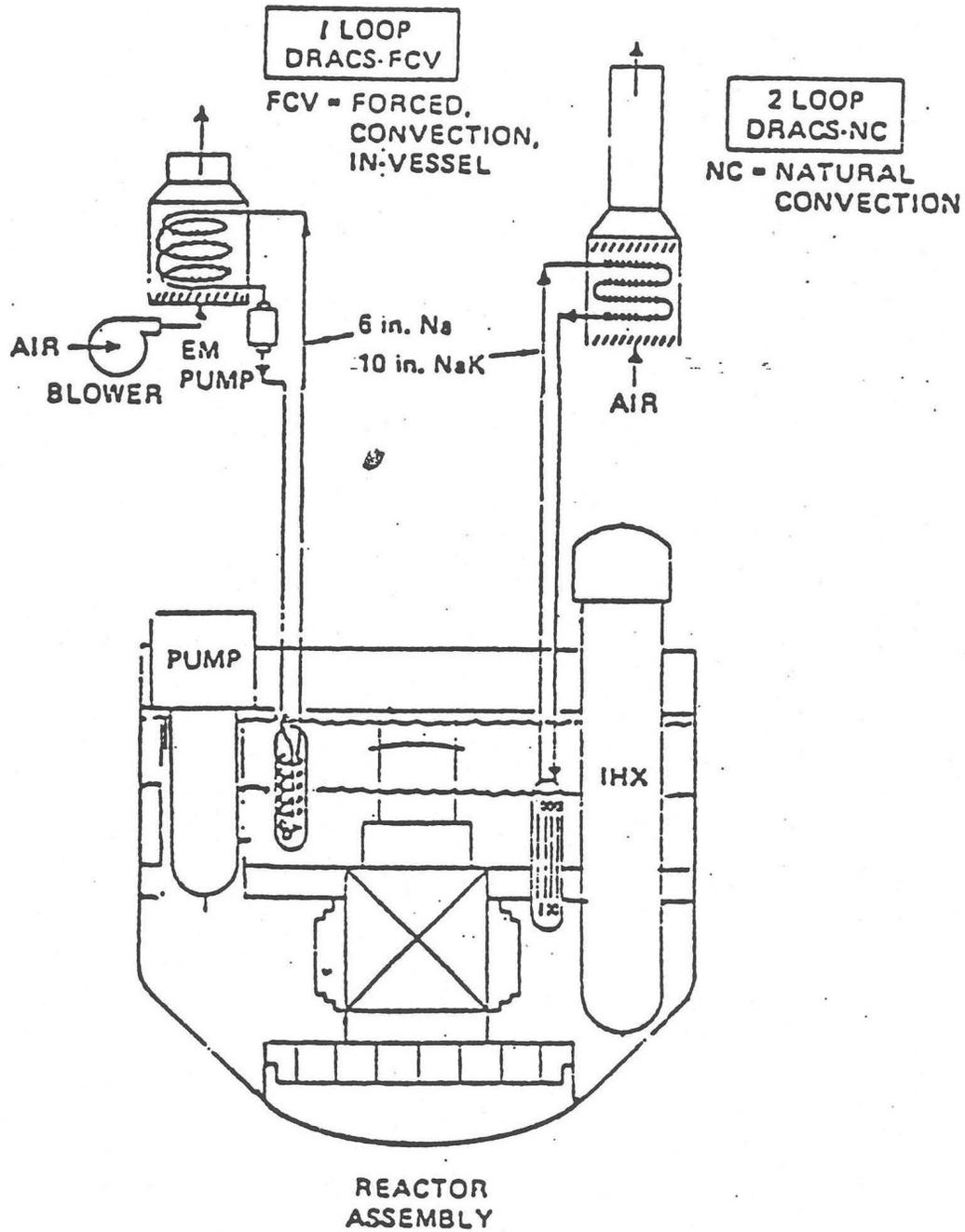


Figure 5.6-1. Shutdown Heat Removal System for a Large Pool Plant

Equations used in the model are as follow: For the coolant, we ignore storage of energy within the heat exchanger and assume specific heat is independent of temperature so that an energy balance gives

$$Q = w_{Na} C_{Na} (T_{Nat} - T_{Nab}) = w_a C_a (T_{at} - T_{ab}), \quad (5.6-1)$$

where

$Q$  = heat removal rate,

$w$  = mass flowrate,

$C$  = specific heat,

$T$  = temperature,

$a$  = air side subscript,

$Na$  = liquid metal side subscript,

$t$  = subscript for the top of the heat exchanger,

$b$  = subscript for the bottom of the heat exchanger.

The momentum equation for the air side is from Appendix 5.4

$$w^2 = \frac{(\rho_c - \rho_h) g \ell A_g^2}{\frac{K_{SI} \left( \frac{A_g}{A_{SI}} \right)^2}{2\rho_c} + \frac{K_{HX} \left( \frac{A_g}{A_{HX}} \right)^2}{2\rho_c} + \frac{K_{SO} + K_R}{2\rho_h}} \quad (5.6-2)$$

where

$w$  = air mass flowrate,

$A$  = area,

$\rho_c$  = inlet air density,

$\rho_h$  = outlet air density,

$K$  = loss coefficient,

$\ell$  = height of stack,

$SI$  = stack inlet subscript,

$SO$  = stack outlet subscript,

$R$ = riser subscript

$g$ = acceleration of gravity.

The overall heat transfer coefficient evaluated at the tube outside surface (air side) is

$$U = \left[ \frac{r_o}{r_i h_i} + \frac{r_o}{k} \ln \frac{r_o}{r_i} + \frac{1}{h_o} \right]^{-1}, \quad (5.6-3)$$

where

$h_i$ = tube inside heat transfer coefficient,

$h_o$ = tube outside heat transfer coefficient,

$r_i$ = tube inner radius,

$r_o$ = tube outer radius,

$k$ = thermal conductivity.

In this implementation of the model, the heat transfer coefficients  $h_i$  and  $h_o$  are treated as constants and are input by the user. Strictly, however, their values change with fluid temperature and flowrate. Typically,  $h_o$  dominates Eq. (5.6-3) and because the air side is a complex geometry of pins and tubes, its value is not accurately known. Therefore, assuming a constant value for  $h_o$  that is credible should not introduce appreciably more error than the uncertainty that already exists because of its complex geometry.

The basic equations for temperatures are

$$W_{Na} C_{Na} \frac{dT_{Na}}{dz} + \frac{UA}{L} (T_a - T_{Na}) = 0 \quad (5.6-4a)$$

and

$$-W_a C_a \frac{dT_a}{dz} + \frac{UA}{L} (T_{Na} - T_a) = 0 \quad (5.6-4b)$$

where

$T_{Na}$ = sodium temperature

$T_a$ = air temperature

$z$ = elevation

$W_{Na}$  = sodium flow rate

$W_a$  = air flow rate

$C_{Na}$  = sodium heat capacity

$C_a$  = air heat capacity

$U$  = heat transfer coefficient

$A$  = total heat transfer area, and

$L$  = height of the heat transfer section

The solution of Eqs. 5.6-4a and 5.6-4b of the form

$$T_{Na} = T_a = ae^{\alpha z} \quad (5.6-4c)$$

where  $a$  is a constant, and

$$\alpha = \frac{(W_a C_a + W_{Na} C_{Na})UA / L}{W_a C_a W_{Na} C_{Na}} \quad (5.6-4d)$$

$T_{Nab}$  = sodium outlet temperature, and

$T_{ab}$  = air inlet temperature

The sodium temperature then has the form

$$T_{Na} = b + ce^{\alpha z} \quad (5.6-4e)$$

The gravity head is

$$\Delta p_{gr} = \int_0^L \rho_{Na}(z) dz \quad (5.6-4f)$$

where  $\rho_{Na}$  is the sodium density and  $z = 0$  at the bottom of the heat transfer section. The thermal center,  $z_{tc}$ , is defined by

$$\Delta p_{gr} = \rho_{Nat} (L - z_{tc}) + \rho_{Nab} z_{tc} \quad (5.6-4g)$$

where

$\rho_{Nat}$  = sodium density at the top and

$\rho_{Nab}$  = sodium density at the bottom

If the density varies linearly with temperature, then an equivalent equation for the thermal center is

$$T_{Nat} (L - z_{tc}) + T_{Nab} z_{tc} = \int_0^L T_{Na}(z) dz \quad (5.6-4h)$$

where  $T_{Nat}$  is the sodium temperature at the top of the heat transfer section. Carrying out the integration of Eq. 5.6-4e in Eq. 5.6-4h gives

$$z_{tc} = L \left[ 1 / (1 - e^{-\alpha L}) - 1 / \alpha L \right] \quad (5.6-4i)$$

For the steady-state initialization, conservation of energy gives

$$W_{Na} C_{Na} (T_{Nat} - T_{Nab}) = W_a C_a (T_{at} - T_{ab}) \quad (5.6-4j)$$

Also, Eq. 5.6-4c can give

$$\frac{T_{Nat} - T_{at}}{T_{Nab} - T_{ab}} = x = \exp \left( UA \left( \frac{1}{W_{Na} C_{Na}} - \frac{1}{W_a C_a} \right) \right) \quad (5.6-4k)$$

combining Eqs. 5.6-4j and 5.6-4k gives

$$T_{Nab} = \frac{x W_a C_a}{x W_a C_a - W_{Na} C_{Na}} \left[ T_{ab} \left( 1 - \frac{1}{x} \right) + \frac{T_{Nat}}{x} \left( 1 - \frac{W_{Na} C_{Na}}{W_a C_a} \right) \right] \quad (5.6-4l)$$

The above equations are solved by subroutine NAKAIR. SASSYS-1 couples to this routine as follows: Values for liquid metal flowrate at the heat exchanger inlet, the temperature at this point and the inlet air temperature are passed as computed by SASSYS-1 to subroutine NAKAIR. This routine solves the above equations for the temperature drop on the liquid metal side of the heat exchanger and returns this value to SASSYS-1 which uses it to advance to the next time step.

The subroutine is also applicable to the case where the air flowrate is forced. In this case, the known air mass flowrate is used in place of Eq. (5.6-2).

### Steady-State Solution

A capability has been added to the SASSYS-1 air dump heat exchanger (DHX) model to provide a steady-state initialization procedure that will calculate the air flow rate required to remove the steady-state power delivered to the heat exchanger. What is done with the steady-state air flow rate in the DHX depends on whether the forced air flow option or the natural circulation air flow option is used. For forced flow, the user-

supplied table of air-flow rate vs. time is renormalized by the code to give the correct steady-state value. For natural circulation air flow, the code will calculate and print the inlet orifice coefficient required to give the calculated steady-state air flow rate. The code does not re-set the inlet orifice coefficient to this calculated value, since this is not always what the user would want.

A special case occurs when the user sets the steady-state sodium flow rate through the DHX to zero, indicating that the steady-state heat removal is negligible. In this case, the steady-state air flow rate is set to zero and the thermal center is set to the geometrical center.

For the steady-state initialization of the DHX, the sodium and air inlet temperatures,  $T_{Nat}$  and  $T_{ab}$ , are known. Also, the sodium flow rate,  $W_{Na}$ , is known. Furthermore, the correct value for the sodium outlet temperature,  $T_{Nab}$ , is known from initializing the rest of the system. Therefore, the task of the DHX initialization routine is to find the air flow rate which gives the correct value of  $T_{Nab}$  in Eqn. 5.6-4m. This is carried out by iterating on air flow until the correct sodium outlet temperature is found.

## 5.7 Cover Gas Flow and Pressure Calculations

### 5.7.1 Introduction

After the hydraulics calculations for the liquid flows and pressures have been completed for a time step, then the liquid temperatures are calculated, and, after they have been completed, the cover gas flows and pressures are computed. The cover gas treatment is a fully implicit calculation, ignoring the inertial effects of the gas, and modeling the flow as quasi-static and isothermal. It may be thought of as a modification of the liquid flow treatment in which the left side of Eq. 5.2-1 is set to zero and  $\theta_1$  and  $\theta_2$  set to zero and one, respectively, in Eq. 5.2-17.

The gas process may be thought of in the following way. Only compressible volumes containing cover gas are considered. Some may also contain liquid and some only gas. For those containing liquid, the liquid level has risen or dropped, as determined by the hydraulics calculations earlier in the time step, and the cover gas has been compressed or expanded by this action. The compression or expansion is assumed to be adiabatic, and new equilibrium conditions are calculated. Next, heat transfer between the gas and the liquid is computed, assuming that the gas temperature approaches that of the liquid without affecting the liquid temperature. Finally, taking all compressible volumes with cover gas, the gas flow between compressible volumes through connecting pipes is computed, and the temperatures, pressures, and masses of the gases in the compressible columns are adjusted appropriately.

### 5.7.2 Basic Equations

In the compressible volumes with cover gas and liquid, the hydraulics routines provide the rise or fall in the liquid level during the time step, thereby giving the change in the cover gas volume during the time step. Let  $V_{03}$  be the volume of the cover gas before the liquid level change and  $V_3$  after the level change. The process is assumed to

be adiabatic, and the pressure  $p'_{03}$  after the change in terms of the pressure  $p_{03}$  before is given by

$$p'_{03} = p_{03} (V_{03} / V_3)^\gamma \quad (5.7-1)$$

and the accompanying new gas temperature  $T_g$  is given by the ideal gas law as:

$$T_g = p'_{03} V_3 / m_3 R \quad (5.7-2)$$

where

$\gamma$ = the ratio of the specific heat at constant pressure to that at constant volume for the gas

$m_3$ = the cover gas mass at the beginning of the time step

$R$ = the universal gas constant

Heat transfer between the cover gas and the liquid is also accounted for. Since the heat capacity of the gas is much less than that of the liquid, it is assumed that the gas temperature approaches the liquid temperature with a time constant  $\tau$  without affecting the liquid temperature. The time constant  $\tau$  is a user-supplied input quantity. The adjusted gas temperature  $T_3$  is taken as

$$T_3 = T_g \frac{\tau}{\tau + \Delta t} + T_l \frac{\Delta t}{\tau + \Delta t} \quad (5.7-3)$$

where

$T_l$ = the temperature of the liquid

$\Delta t$ = the time step

The new gas temperature means a new gas pressure  $p_3$ , which is computed again by the ideal gas law as

$$p_3 = m_3 R T_3 / V_3 \quad (5.7-4)$$

With  $p_3$ ,  $V_3$ ,  $T_3$ , and  $m_3$  now known for compressible volumes with or without liquid present, we turn to the gas flow through the pipes between compressible volumes. The difference in pressure between two connected compressible volumes causes gas flow from one to the other, and the amount of gas delivered to or withdrawn from a compressible volume in turn modifies the gas pressure in it. As a result, the pressure changes in all of the compressible volumes are solved for simultaneously.

The gas mass flow rate,  $F_{ij}$ , from compressible volume  $i$  to compressible volume  $j$  is approximated at the end of the time step as

$$F_{ij} = F_{oij} + F_{1ij} \Delta p(j) + F_{2ij} \Delta p(i) \quad (5.7-5)$$

Here  $F_{oij}$  is the gas mass flow rate at the beginning of the time step from compressible volume  $i$  to compressible volume  $j$ . This flow rate is modified by the pressure changes in the compressible volumes  $i$  and  $j$  as a result of the flow during the time step. The coefficients of the pressure changes are taken as

$$F_{1ij} = -F_{oij} / [p(i) - p(j)] \quad (5.7-6)$$

$$F_{2ij} = F_{oij} / [p(i) - p(j)] \quad (5.7-7)$$

where  $p(i) - p(j)$  is the pressure difference between compressible volumes  $i$  and  $j$  at the beginning of the time step. Also

$$\Delta p(i) = p_4(i) - p_3(i) \quad (5.7-8)$$

where

$p_3(i)$  = the gas pressure in compressible volume  $i$  at the beginning of the time step

$p_4(i)$  = the gas pressure in compressible volume  $i$  at the end of the time step

The flow equation for isothermal unchoked flow of an ideal gas is given by Shapiro [5-3] as:

$$f \frac{L}{D_h} = \frac{1 - [p(i)/p(j)]^2}{\gamma M^2} - \ln[p(i)/p(j)]^2 \quad (5.7-9)$$

where

$f$  = the Moody friction factor

$L$  = the length of the pipe

$D_h$  = the hydraulic diameter of the pipe

$p(i)$  = the inlet pressure

$p(j)$  = the outlet pressure

$\gamma$  = the ratio of the specific heat at constant pressure to that at constant volume for the gas

$M$  = the Mach number

The Moody [5-2] friction factor is given by:

$$f = 0.0055 \left[ 1 + \left( 20000 \varepsilon / D + 10^6 / \text{Re} \right)^{1/3} \right] \quad (5.7-10)$$

where

$\varepsilon$  = the pipe roughness

$D$  = the pipe diameter

$\text{Re}$  = the Reynolds number

In addition the Reynolds number can be written as

$$\text{Re} = \frac{F_{oij} D}{A \mu} \quad (5.7-11)$$

where

$A$  = the pipe area

$\mu$  = the viscosity of the gas

and the Mach number is related to the gas mass flow rate  $F_{oij}$  by

$$\gamma M^2 = \left( \frac{F_{oij}}{A} \right)^2 \frac{RT}{\gamma} \quad (5.7-12)$$

For turbulent flow, the gas mass flow rate is determined by an iterative process. With an initial built-in guessed value for  $f$ , Eqs. 5.7-9 and 5.7-12 determine a value for  $F_{oij}$ , which is used in Eq. 5.7-10 to calculate a new value for  $f$ , and the iteration is continued until consistency is achieved. For laminar flow,  $F_{oij}$  is determined from Eqs. 5.7-9 and 5.7-12 using the laminar flow value for  $f$  of

$$f = 64 / \text{Re} \quad (5.7-13)$$

### 5.7.3 Solution

The results of the gas flow between compressible volumes, where a compressible volume may be connected by gas segments to several other compressible volumes, can be written as:

$$p_4(i) = p_3(i) (1 + \varepsilon_i) \quad (5.7-14)$$

$$m_4(i) = m_3(i) + \delta m_i \quad (5.7-15)$$

$$T_4(i) = \frac{p_4(i) V_4(i)}{m_4(i) R} \quad (5.7-16)$$

where

$$\varepsilon_i = \frac{\gamma}{m_3(i) T_3(i)} \sum_j T_{ji} \Delta m_{ji} \quad (5.7-17)$$

$$\delta m_i = m_4(i) - m_3(i) = \sum_j \Delta m_{ji} \quad (5.7-18)$$

$$\Delta m_i = \Delta t [F_{oji} + F_{1ji} \Delta p(i) + F_{2ji} \Delta p(j)] \quad (5.7-19)$$

and  $T_{ji}$  is the temperature of the gas flowing from compressible volume  $j$  to compressible volume  $i$ . Combining Eqs. 5.7-14, 5.7-8, 5.7-17, and 5.7-19 yields a matrix equation of the form

$$\begin{pmatrix} c_{11} & c_{12} & \dots \\ c_{21} & c_{22} & \dots \\ \dots & \dots & \dots \\ \dots & \dots & \dots \end{pmatrix} \begin{pmatrix} \Delta p(1) \\ \Delta p(2) \\ \dots \\ \dots \end{pmatrix} = \begin{pmatrix} d_1 \\ d_2 \\ \dots \\ \dots \end{pmatrix} \quad (5.7-20)$$

where

$$c_{ij} = \delta_{ij} \left[ 1 - \frac{p_3(i) \gamma \Delta t}{m_3(i) T_3(i)} \sum_k T_{ki} F_{1ki} \right] - \frac{p_3(i) \gamma \Delta t}{m_3(i) T_3(i)} T_{ji} F_{2ji} \quad (5.7-21)$$

and

$$d_i = \frac{p_3(i)\gamma\Delta t}{m_3(i)T_3(i)} \sum_j T_{ji} F_{oji} \quad (5.7-22)$$

Here  $\delta_{ij}$  is the Kronecker delta.

The matrix equation 5.7-20 is solved by Gaussian elimination for the  $\Delta p(i)$ . These are added to the pressures at the beginning of the time step to obtain the corresponding pressures at the end of the time step. The cover gas mass at the end of the time step  $m_4(i)$  in each compressible volume is computed using Eqs. 5.7-18 and 5.4-19, and finally the gas temperatures at the end of the time step  $T_4(i)$  are computed for each compressible volume using the equation

$$T_4(i) = \frac{p_4(i)V_4(i)}{m_4(i)R} \quad (5.7-23)$$

All of the final values are then stored in COMMON blocks.

A solution algorithm for the entire cover gas treatment is given in Appendix 5.3.

## 5.8 Overall Solution and Time Step Control

### 5.8.1 Order of Calculation in PRIMAR

The computational sequence used in a time step in the primary and intermediate loops is as follows:

1. Adjust the inlet and outlet plenum pressures for any errors between the estimated and the calculated core flows in the last step.
2. Calculate  $b_0$ ,  $b_1$ , and  $b_2$  for the compressible volumes in the primary loop.
3. Calculate the contributions to  $a_0$ ,  $a_1$ ,  $a_2$ , and  $a_3$  for each element in a liquid segment and sum them for all of the liquid segments in the primary loop.
4. Calculate contributions to  $c_{ij}$  and  $d_j$  from all of the segments in the primary loop.
5. Add the contributions to  $c_{ij}$  and  $d_j$  from the estimated core flow.
6. Solve for  $\Delta p$ .
7. Calculate  $\Delta w$ .
8. Repeat steps 2, 3, 4, 6 and 7 for the intermediate loops, if any are present.
9. Repeat steps 2, 3, 4, 6 and 7 for the DRACS loops, if any are present.
10. Calculate the liquid temperatures.
11. Recalculate the compressible volume pressures with the new liquid temperatures.
12. Calculate the cover gas flows and the cover gas pressures.

In this sequence of events, the liquid flow hydraulics calculations are done first, followed by the liquid temperature calculations and then by the gas flow and temperature calculations. In order to reduce the sizes of the matrix equations that must be solved, the hydraulics calculations are done for the primary loop first, then for the intermediate loops, and finally for the DRACS loops, if any are present. Adiabatic compression of the cover gases is accounted for during the initial hydraulics calculations, but heat transfer to the gas and gas flows through connecting pipes are not accounted for until the gas calculations at the end of the computational sequence for a time step.

### 5.8.2 PRIMAR Time Step Size

The initial PRIMAR time-step size is entered as an input quantity. Then the time-step size is determined by the coupling with the coolant dynamics calculations for the core subassemblies. PRIMAR is called to calculate a new time step before the core coolant dynamics routines are called and, as a result, part of the PRIMAR calculation consists in estimating the new core flows based on information supplied by the coolant dynamics routines. Before the start of voiding, the core flows can be estimated accurately in PRIMAR because the single-phase subassembly flow calculations are relatively simple. Consequently, the time step can be fairly large before the start of voiding. After the start of voiding, the core flows are strongly influenced by the rapidly varying pressures in the voiding region, and the PRIMAR time-step size must be cut back to 20 milliseconds or less.

If PRIMAR predicts a large change in the inlet plenum pressure during a time step, the step size must be reduced and the quantities determined in the time step recalculated. Alternatively, if the pressure, temperature, or flow in the primary or intermediate loop change rapidly, but do not have a large effect on the inlet or outlet plenum pressures and temperatures, then the PRIMAR time step is subdivided into smaller intervals for the PRIMAR calculations without requiring additional coolant dynamics calculations in the core subassemblies.

### 5.9 Steady-State Initialization

The steady-state initialization for PRIMAR-4 is based on finding a steady-state thermal hydraulic solution that is consistent with the user-supplied input for parameters such as reactor power, flow, coolant inlet temperature, and coolant outlet plenum pressure. The core channels are initialized first, leading to the temperatures and pressures in the inlet and outlet coolant plenums. Then the initialization works outward through the primary heat-transport loops, the *IHX*'s, the intermediate heat-transport loops, and finally, the steam generators.

The primary-loop temperature initialization consists mainly of setting the temperatures of all elements and compressible volumes in the hot side of the loop to the coolant outlet plenum temperature, and setting the components in the cold side of the loop to the inlet plenum temperature. Intermediate heat exchangers and bypass channels are separate cases, since the temperatures change from one end to the other end of these elements. If necessary, the liquid segment flows are adjusted so that the

liquid flow into each compressible volume balances the flow out. After the primary-loop temperatures are set, the pressure drops through all elements are calculated, and the pump heads are set to the values necessary to achieve a pressure balance.

After the main primary-loop initialization calculations, the *IHX* primary and intermediate side flow rates are known, and the primary side inlet and outlet temperatures are known. Then, an iteration procedure is used to find intermediate side inlet and outlet temperatures that provide a consistent steady-state solution. The detailed node-by-node temperature distributions within the *IHX* are also calculated in this process.

After the *IHX* temperature distributions have been calculated, the temperatures in the intermediate loop between the *IHX* outlet and the steam generator inlet are set to the *IHX* outlet temperature, and the temperatures between the steam generator outlet and the *IHX* inlet are set to the *IHX* inlet temperature. Pressure drops are calculated for each element; and pressures at each point are calculated, starting from a user-specified *IHX* inlet pressure. As in the primary loop, the intermediate loop pump head is set to the value necessary to achieve a pressure balance.

After the intermediate loop initialization, the steam generators are initialized, using the known sodium side inlet and outlet temperatures and flow rate.

The steady-state gas flow rates are all set to zero, and cover-gas pressures are initially set to user-supplied values.

### 5.9.1 Core Channels, Inlet Plenum, and Outlet Plenum

Since the core channel calculations are described in detail in Chapter 3, they are only summarized here. The user supplies the steady-state values for the coolant outlet plenum pressures, the coolant inlet temperatures, the coolant flow for each channel, and the power for each channel, as well as geometric and hydraulic information. The code then calculates the coolant outlet temperature and an inlet pressure for each channel, as well as temperature and pressure distributions within the channels. In general, each channel will have a different inlet pressure, so for each inlet plenum the channel with the highest inlet pressure is picked as the reference, and the orifice coefficients at the bottom of the first reflector zone in each other channel are adjusted to give a common inlet pressure. This common inlet pressure determines the coolant inlet plenum pressure.

For each outlet plenum, the average steady-state temperature is a mixed mean of the outlet temperatures of all the core channels and all the bypass channels emptying into the plenum. The core model provides steady-state channel outlet temperatures, but the loop model is designed to begin with known temperatures in the inlet and outlet plena of the core channels and from these calculate temperatures throughout the remainder of the plant, and so outlet temperatures for the bypass channels are not available at the point at which the outlet plenum temperatures need to be calculated. The code works around this as follows. To a particular outlet plenum *ICV* are attached *N* channels and *M* bypass channels. The mixed-mean outlet temperature,  $T_{mm}$ , of the core channels is given by

$$T_{mm}(ICV) = \frac{\sum_{ic=1}^N W_{J(ic)} T_{exp}(J(ic))}{\sum_{ic=1}^N W_{J(ic)}} \quad (5.9-1)$$

where  $T_{exp}(J(ic))$  is the steady-state outlet temperature for core channel  $J(ic)$  and  $w_j(ic)$  is the steady-state flow through channel  $J(ic)$ . The model assumes that the difference between  $T_{mm}(ICV)$  and the average inlet temperature of the  $N$  core channels,  $T_{in}(ICV)$ , is proportional to the ratio of power to flow in the channels, or

$$T_{mm}(ICV) - T_{in}(ICV) = K \frac{f_{pr}(ICV)}{f_{fl}(ICV)}, \quad (5.9-2)$$

where  $K$  is the constant of proportionality,  $f_{pr}(ICV)$  is the ratio of the power in the core channels attached to outlet plenum  $ICV$  to the power in all channels (core + bypass) attached to plenum  $ICV$ , and  $f_{fl}(ICV)$  is the ratio of the flow in the core channels attached to outlet plenum  $ICV$  to the flow in all channels (core plus bypass) attached to plenum  $ICV$ . The model also assumes that the difference between the average temperature of plenum  $ICV$ ,  $T_{out}(ICV)$ , and the average inlet temperature of the  $N$  core channels and the  $M$  bypass channels,  $T_{in,total}(ICV)$ , is proportional to the ratio of power to flow and that the constant of proportionality is the same as for the core channels alone, so that

$$T_{out}(ICV) - T_{in,total}(ICV) = K. \quad (5.9-3)$$

The average inlet temperatures are calculated as

$$T_{in}(ICV) = \frac{\sum_{ic=1}^N W_{J(ic)} C_{pJ(ic)} T_{in,channel}(J(ic))}{\sum_{ic=1}^N W_{J(ic)} C_{pJ(ic)}} \quad (5.9-4)$$

and

$$T_{in,total}(ICV) = \frac{\sum_{ic=1}^N W_{J(ic)} C_{pJ(ic)} T_{in,channel}(J(ic)) + \sum_{jc=1}^M W_{K(jc)} C_{pK(jc)} T_{in,bypass}(K(jc))}{\sum_{ic=1}^N W_{J(ic)} C_{pJ(ic)} + \sum_{jc=1}^M W_{K(jc)} C_{pK(jc)}}, \quad (5.9-5)$$

where  $C_p$  is the specific heat of the fluid at the entrance to a channel. The inlet temperatures and the flows are input by the user, and the specific heats are computed

from the equation of state. The fractions  $f_{pr}$  and  $f_{fl}$  are calculated from user-input power and flow information, and  $T_{mm}$  is computed from Eq. (5.9-1), so the only unknown quantities in Eqs. (5.9-2) and (5.9-3) are  $K$  and  $T_{out}$ . Eliminating  $K$  between the two equations gives

$$T_{out}(ICV) - T_{in,total}(ICV) = \frac{T_{mm}(ICV) - T_{in}(ICV)}{\frac{f_{pr}(ICV)}{f_{fl}(ICV)}}, \quad (5.9-6)$$

or

$$T_{out}(ICV) = T_{mm}(ICV) \frac{f_{fl}(ICV)}{f_{pr}(ICV)} + T_{in,total}(ICV) - T_{in}(ICV) \frac{f_{fl}(ICV)}{f_{pr}(ICV)}. \quad (5.9-7)$$

### 5.9.2 Primary Heat Transport System

After the core channel initialization, and after the temperatures and pressures have been set for the inlet and outlet plenums, the rest of the primary heat-transport loop is initialized. This involves setting temperatures for all compressible volumes and liquid elements, liquid pressures and liquid-cover gas interface heights for compressible volumes, and flow rates for all liquid segments. Flow rates for gas segments are initialized to zero, and the user specifies the initial cover-gas pressure for each compressible volume.

A two-pass procedure is used for initializing the primary heat-transport system other than core channels, the inlet plenums, and the outlet plenums. In the first pass, the temperatures are set for all compressible volumes and elements, and the gravity head and pressure drop for each element other than a pump is set. Also the liquid segment flows are adjusted, if necessary, to make the flow into each compressible volume equal the flow out. In the second pass, the compressible volume pressures and pump heads are set.

For each pass through the primary heat-transport system, a table is constructed containing the compressible volume numbers of the compressible volumes whose temperatures or liquid pressures are already known. The inlet and outlet plenums are the first entries in this table. Compressible volumes are treated one at a time in the order in which they occur in the table, and new entries are added to the table as temperatures or pressures of new compressible volumes are found.

For the temperatures pass, the first step in the treatment of a compressible volume is to adjust the flows in the liquid segments attached to the compressible volume so that the flow in matches the flow out. The user supplies the steady-state flow rates for each liquid segment, but if the user-supplied values are not consistent then the code adjusts some of them to obtain consistency. If the pressure drops in a liquid segment have already been set, then a flag is set for that segment; and its flow can no longer be changed, since changing flows would require re-calculating the pressure drops.

Therefore, any flow adjustment is made for liquid segments whose flows have not been fixed. A fixed net flow,  $w_{fixed}$ , is calculated by summing the flows of all fixed segments attached to the compressible volume, with a positive sign if the flow is into the compressible volume, or a negative sign if the flow is out of the compressible volume. Similarly, an adjustable net flow,  $w_{adj}$ , is calculated by summing over the remainder of the liquid segments attached to the compressible volume. Then a factor,  $f_w$ , is calculated as

$$f_w = -\frac{w_{fixed}}{w_{adj}} \quad (5.9-8)$$

and the flow for each unfixed liquid segment attached to the compressible volume is multiplied by  $f_w$ .

The next step in the treatment of a compressible volume consists of following each liquid segment attached to the compressible volume and setting the temperatures for all of the elements until either the other end of the liquid segment is reached or a stop is made at an IHX or at the element representing the core channels. Initially, the compressible volume temperature is used for setting the temperatures of the elements in a segment; but if a bypass channel is encountered, then the bypass channel initialization routine is called, and the bypass outlet temperature is used for setting the temperatures of any additional elements in the segment. The steady-state gravity head and pressure drop for an element are set as soon as the temperature is set. The liquid segment containing the core channels is ignored, since the core channels have already been initialized. The initialization procedure used for an IHX requires knowing both the primary side inlet temperature and the primary side outlet temperature; so if an IHX is encountered before the temperatures of the compressible volumes at both ends of the liquid segment are known, then the remainder of this segment is skipped until later, when it will be approached from the other end.

If a liquid segment is followed to the end without encountering the core channels or an IHX, then the compressible volume at the other end of the segment is checked. If its temperature has not already been set, then it is now set to the current temperature, and the new compressible volume is added to the compressible volume table. In this manner, all of the elements, liquid segments, and compressible volumes in the primary loop will eventually be initialized if they are all inter-connected, and if two IHX's are not connected in series. If two or more IHX's are connected in series, then the user must specify the steady-state temperature drops across all but one of them. Similarly, if two or more pumps are connected in series, then the user must specify the steady-state pump heads for all but one of them.

For the second pass through the primary heat-transport system, the compressible volume table is started over, with the inlet and outlet plenums again providing the first entries in the table. Now the table contains only compressible volumes whose pressures are known. In this pass, each liquid segment attached to a compressible volume is again treated. The pressure drops and gravity heads for all of the elements, other than pumps, in a liquid segment are summed. If the liquid segment contains a

pump, and if the pressure has already been determined for the compressible volumes at both ends of the liquid segment, then the pump head is set to the value needed to balance the pressures. If the segment contains a pump, and if the pressure in one of the compressible volumes at the end of the liquid segment has not been set yet, then this liquid segment is skipped until later, when the compressible volume pressure has been set. If there are no pumps in the liquid segment, and if the pressure in the compressible volume at one end of the liquid segments has not yet been set, then the pressure in this compressible volume is set, based on the pressure in the compressible volume at the other end of the liquid segment and the sum of the pressure drops and gravity heads through the segment. This compressible volume is then added to the table. If there is no pump in the liquid segment, and if the pressures in the compressible volumes have already been set, then the orifice-type pressure-drop coefficient for the first element in the liquid segment is adjusted to give the proper pressure balance.

During the first pass, after the primary side inlet and outlet temperatures and the primary side flow rate have been determined, an IHX initialization routine is called to initialize the primary and intermediate side IHX temperatures and gravity heads. This routine is described in Section 5.9.4.3.

### 5.9.3 Intermediate Heat Transport System

The steady-state initialization of the intermediate heat-transport system is similar to that of the primary heat-transport system, except that the intermediate loop treatment starts from the IHX and works out. The IHX intermediate side inlet and outlet temperatures have been determined by the IHX initialization routines. Also, since the IHX flow rates were used in the IHX routines, the flow rate through the segment containing the intermediate side of the IHX is fixed at the user-supplied value for this segment. The pressure calculations start from a user-supplied IHX inlet pressure.

Three passes through the intermediate loop compressible volumes are made. In the first pass, the liquid segment flows are adjusted so that the flow into each compressible volume equals the flow out. This is exactly the same method used for the primary loop, with sums over fixed and adjustable flows and modifications of the adjustable flows. In the second pass, temperatures, gravity heads, and pressure drops are set for each element except for pumps. In the third pass, pump heads and compressible volume pressures are set. The second and third passes use the same type of compressible volume tables used for the primary loop. These passes use the same kind of calculations as the corresponding primary loop passes, except that the calculations start in the middle of a segment at the IHX, and the first compressible volumes added to the table are those at the ends of the IHX segment.

One difference between the primary and intermediate loop initialization procedures is that, in the intermediate loop, it is necessary to consider the possibility of two steam generators connected in series, which is a situation that occurs when an evaporator-superheater combination is used. For such a case, the user specifies the fraction of the total sodium-side steady-state temperature drop that occurs across the evaporator.

### 5.9.4 Individual Components

For each element, other than the core channels, of each liquid segment in the primary and intermediate heat-transport systems, a steady-state initialization routine is called. These routines compute pressure drops, except for pumps, and gravity heads. Temperature distributions within the component are also calculated for IHXs and bypass channels. Also, routines are called to initialize the temperatures, liquid pressures, gas pressures, and liquid-gas interface elevations of compressible volumes. These routines for individual components are described below.

#### 5.9.4.1 Pressure Drop, Subroutine PRES DR

For all elements other than core channels and pumps, subroutine PRES DR is called to compute the pressure drop due to friction, bends, orifices, and density change. The pressure drop for element  $k$  in segment  $i$  is calculated as

$$\Delta p_{el}(k) = w(i) \left\{ \frac{f |w(i)|}{2\bar{\rho} A_k^2} \left[ \frac{L_k}{D_h(k)} + N_b(k) \left( \frac{L}{D} \right)_b \right] + w(i) \left[ \frac{1}{\rho_{out}(k)} - \frac{1}{\rho_{in}(k)} \right] + \frac{|w(i)|}{2\bar{\rho} A_k^2} G_{2pr}(k) \right\} \quad (5.9-9)$$

The first term is the friction, as enhanced by bends.  $N_b$  is the number of bends in the element, and  $(L/D)_b$  is the effective length/hydraulic diameter per bend. The second term is an acceleration term due to density changes. In this term,  $\rho_{in}$  and  $\rho_{out}$  are the sodium densities at the inlet and outlet of the element. The last term represents orifice-type pressure drops proportional to the square of the flow rate, and  $G_{2pr}$  is the dimensionless loss coefficient for this term. The average density is the average of  $\rho_{in}$  and  $\rho_{out}$ .

The Moody correlation [5-2] is used for the friction factor for Reynolds numbers,  $Re$ , above 1082, whereas a laminar friction factor is used for lower Reynolds numbers:

$$f = \begin{cases} .0055 \left[ 1 + \left( 20000 \frac{\varepsilon(k)}{D_n(k)} + \frac{10^6}{Re} \right)^{\frac{1}{3}} \right] & \text{if } Re \geq 1082 \\ \frac{64}{Re} & \text{if } Re < 1082 \end{cases} \quad (5.9-10)$$

where  $\varepsilon(k)$  is the surface roughness. The value of 1082 was picked for the transition between laminar and turbulent friction factors because at this value both correlations give the same result if  $\varepsilon/D_h$  is small.

#### 5.9.4.2 Pipe

For a pipe, the pressure drop is computed as indicated in section 5.9.4.1 above. The gravity head is calculated as

$$\Delta p_{gr}(k) = \rho_k [z_0(k) - z_i(k)] g \quad (5.9-11)$$

where  $k$  is the element number,  $\rho_k$  is the sodium density,  $z_i$  and  $z_0$  are the inlet and outlet elevations, and  $g$  is the acceleration of gravity.

### 5.9.4.3 IHX

The steady-state initialization of an IHX depends on whether the detailed IHX model or the simple table loop-up model is used. In either case, the pressure drop is calculated in subroutine PRES DR, as described in Section 5.9.4.1. The temperature calculations for the detailed model are described in Section 5.4.2.5. For the simple table look-up model, the gravity head in both the steady-state and the transient is calculated using Eq. 5.4-83.

### 5.9.4.4 Bypass Channel

The steady-state temperature solution for the bypass channel is described in section 5.4.3. The pressure drop is calculated in subroutine PRES DR, as described in Section 5.9.4.1.

### 5.9.4.5 Steam Generator

The table look-up steam generator model is treated the same as the table look-up IHX model. Subroutine PRES DR, as described in Section 5.9.4.1, calculates the pressure drop, and the gravity head is calculated using Eq. 5.4-83. The detailed steam generator model is described in Chapter 7.

### 5.9.4.6 Compressible Volumes

If compressible volume  $i$  contains liquid, then the steady-state liquid pressure,  $p_\ell(i)$ , and temperature,  $T_\ell(i)$ , are determined by the procedure described in Section 5.9.2 or Section 5.9.3. Then the initial liquid volume,  $V_\ell(i)$ , is

$$V_\ell = \begin{cases} V_{tot}(i) & \text{if there is no gas} \\ V_{tot}(i) - V_{go}(i) & \text{if there is gas} \end{cases} \quad (5.9-12a-b)$$

where  $V_{tot}(i)$  is the total volume, and  $V_{go}(i)$  is the initial gas volume. The values of  $V_{tot}(i)$  and  $V_{go}(i)$  are specified by the user, whereas  $V_\ell(i)$  is computed by the code. The liquid mass,  $m_\ell(i)$ , is then computed as

$$m_\ell(i) = \rho_\ell(i) V_\ell(i) \quad (5.9-13)$$

where  $\rho_\ell$  is the liquid density. The wall temperature is set equal to the liquid temperature.

If there is only liquid in the compressible volume, then the reference pressure,  $p_r(i)$ , is set equal to the steady-state pressure  $p_\ell(i)$ ; and the reference temperature,  $T_r(i)$ , is set equal to the steady-state liquid temperature,  $T_\ell(i)$ .

The liquid pressure within a compressible volume varies with elevation due to the gravity head. The pressure  $p_\ell(i)$  is defined at the reference elevation  $z_\ell(i)$ , so the pressure at any elevation  $z$ , within the liquid is given by

$$p(z) = p_\ell(i) + (z_\ell(i) - z)\rho_\ell g \quad (5.9-14)$$

where  $g$  is the acceleration of gravity. The steady-state liquid-gas interface elevation,  $z_i(i)$ , is the elevation at which the user-supplied gas pressure,  $p_g(i)$ , is equal to the liquid pressure:

$$z_i(i) = z_\ell(i) + \frac{p_\ell(i) - p_g(i)}{\rho_\ell(i)g} \quad (5.9-15)$$

Also, for use in the transient calculations, a minimum elevation,  $z_m(i)$ , is defined as

$$z_m(i) = z_i(i) - \frac{V_\ell(i)}{A_i(i)} \quad (5.9-16)$$

where  $A_i(i)$  is the liquid-gas interface area. The minimum elevation,  $z_m(i)$ , is the "bottom" of the compressible volume; if the liquid-gas interface elevation drops to  $z_m$ , then there will be no liquid left in the compressible volume.

If the compressible volume contains only gas, then the gas volume,  $V_g(i)$ , is equal to the total volume,  $V_{tot}(i)$ . The initial gas and wall temperatures are set to a user-supplied value.

In any compressible volume containing gas, the initial gas mass is calculated as

$$m_g(i) = \frac{p_g(i)V_g(i)}{R_g T_g(i)} \quad (5.9-17)$$

where  $R_g$  is the gas constant.

#### 5.9.4.7 Pumps

The steady-state pump head, flow, and temperature for each pump are determined by the procedure in Section 5.9.2 or Section 5.9.3. Then, the gravity head is calculated using Eq. 5.9-11. If a centrifugal pump option is used then the pump speed required to obtain the required steady-state pump head with the specified flow is obtained by iteration, as described in Section 5.9.4.2.

An electromagnetic pump is treated as a loss-less pump in series with a pipe. The pump head is calculated as

$$\Delta p_p(t) = H_{so} F_p(t) \left[ 1 - \frac{w}{\rho A V_s} \right] - \Delta p_{pipe} \quad (5.9-18)$$

where  $H_{so}$  is the steady-state stall head,  $F_p(t)$  is a user-supplied function for the time dependence of the stall head,  $V_s$  is the synchronous velocity of the magnetic field, and  $\Delta p_{pipe}$  is the pipe-type pressure drop, given by Eq. 5.9-9. For the steady-state solution  $F_p(t)$  is 1.0. After the steady-state pump head and flow are known, the steady-state stall head is obtained from Eq. 5.9-12 as

$$H_{so} = \frac{\Delta p_p(t=0) + \Delta p_{pipe}}{1 - \frac{w}{\rho A V_s}} \quad (5.9-19)$$

### 5.9.5 Null Transient for Steady-State Initialization

In order to eliminate a spurious reactivity insertion at the start of SASSYS-1/SAS4A transient calculations due to the neglect of component-to-component heat transfer and RVACS heat removal in the PRIMAR-4 steady-state initialization, a new capability to bring the PRIMAR-4 temperatures into equilibrium before the start of the transient calculations has been added to the code. This new capability requires small modifications to the code and uses a relatively small amount of computer time to provide significantly better initial conditions at the start of the transient calculations.

The current PRIMAR-4 steady-state initialization does not account for component-to-component heat transfer and RVACS heat removal, although these effects are accounted for during the transient calculations. In the current LMR designs, component-to-component heat transfer and RVACS heat removal have a significant impact on vessel wall temperatures; and changes in vessel wall temperatures can lead to large reactivity changes. Therefore, at the start of a transient calculation the vessel wall temperatures will change as they approach a new state in equilibrium with component-to-component heat transfer and RVACS heat removal; and these temperature changes will add a significant amount of spurious reactivity.

Re-writing the PRIMAR-4 steady-state initialization routines to provide a direct equilibrium temperature solution including arbitrary component-to-component heat transfer would require re-formulating the whole steady-state initialization algorithm and would require completely re-writing many of the routines. Also, a direct equilibrium solution could require a large amount of computer time. Therefore, a simpler approach was taken. The current PRIMAR-4 steady-state initialization routines are used to provide coolant flow rates and a starting point for the temperatures. Then a null transient is run to set the temperatures before the start of the regular transient.

During the null transient the transient heat transfer routines are used; and component-component heat transfer is accounted for, as well as RVACS heat removal.

### Null Transient

During the null transient, only the PRIMAR-4 transient heat transfer routines are called, and everything except the PRIMAR-4 temperatures is held constant. Also, the inlet plenum temperature is held constant. A user-specified time step size is used for the null transient. The user also specifies the number of steps to be used in the null transient and specifies a convergence criterion. The null transient is run until either the specified number of steps have been run or until the maximum fractional change from step to step in component-to-component heat transfer or RVACS heat removal is less than the specified convergence criterion. After the completion of the null transient, the reactivity components are set to zero before the start of the regular transient calculation.

During the null transient, the heat removal from steam generators and table look-up IHXs is reduced by an amount equal to the RVACS heat removal so that the total heat removal rate matches the total power generation rate.

There are a number of advantages to this null transient capability. It allows the temperatures to converge toward the correct equilibrium values, and it eliminates spurious reactivity insertion during the regular transient due to non-equilibrium initial temperatures for the vessel wall. Also, the null transient does not require much computing time. Each time step is fast because all core channel calculations and PRIMAR-4 flow and pressure calculations are bypassed. The time steps can be large, without any stability limitations due to flow rate calculations, although some heat transfer time step stability limits still apply.

The null transient approach has some limitations. One limitation is that temperature changes during the null transient change gravity heads. The PRIMAR-4 steady-state initialization adjusts pump heads and orifice coefficients so that the pressures and flows are in equilibrium at the start of the null transient. Currently, no additional adjustments are made at the end of the null transient; so at the beginning of the regular transient the liquid pressures and flows might not be exactly in equilibrium. At nominal flow rates, the gravity heads are a small part of the over-all pressure balances; so moderate temperature changes during the null transient will usually have only a very small impact on the pressure balances at the start of the regular transient. A second limitation is that it can take a long time for the null transient to converge to an equilibrium situation if there are long thermal time constants in the system. This limitation can usually be gotten around by using fictitiously low heat capacities for some components during the null transient and then changing the heat capacities to their proper values on a restart before the start of the regular transient.

### Time Step Limitations

Most of the temperature calculations in PRIMAR-4 are semi-implicit or fully implicit, and there are no time step stability limits imposed by most of these calculations, but in the component-to-component heat transfer calculations there is at least one place where explicit forward differencing limits the time step size. For

component-to-component heat transfer from a source with temperature  $T_s$  to a compressible volume liquid, the equation solved is of the form

$$M_\ell C_\ell \frac{dT_\ell}{dt} = hA(T_s - T_\ell) + \text{other terms} \quad (5.9-20)$$

where  $M_\ell$ ,  $C_\ell$ , and  $T_\ell$  are the mass, specific heat, and temperature of the liquid, and  $hA$  is the heat transfer coefficient times area for the component-to-component heat transfer. The evaluation of the  $hA(T_s - T_\ell)$  term is partly explicit, so for stability the maximum time step that can be used is of the order of  $\tau$ , where

$$\tau = \frac{M_\ell C_\ell}{hA} \quad (5.9-21)$$

For typical cases, this maximum time step is quite large (tens or hundreds or thousands of seconds), but in some cases it can be small enough to be a significant limitation.

### 5.10 PRIMAR-1 Option

PRIMAR-4 contains both a simple PRIMAR-1 type option and the more detailed PRIMAR-4 treatment. The PRIMAR-1 option supplies only the minimum quantities required to drive the subassembly coolant dynamics module. It is often used when the more detailed treatment is not required. The minimum information consists of the inlet and outlet plenum pressures, the subassembly inlet temperatures, and the outlet reentry temperatures. All of these quantities are supplied as a function of time by PRIMAR-1 from user-supplied information, independent of what is happening in the core or in the rest of the primary loop.

The inlet pressure  $p_{in}(t)$  is calculated as

$$p_{in}(t) = p_x + p_{gr} + \Delta p_p(t) \quad (5.10-1)$$

where  $p_x$  is the user-specified exit plenum pressure,  $p_{gr}$  is the gravity head, and  $\Delta p_p(t)$  is the pump head. The gravity head is calculated as

$$p_{gr} = [\rho_{HOT} (z_{PU} - z_{IHX}) + \rho_{COLD} (z_{IHX} - z_{PL})]g \quad (5.10-2)$$

where  $\rho_{HOT}$  is the average steady-state outlet density,  $\rho_{COLD}$  is the average steady-state inlet density,  $z_{PU}$  is the reference height at which the outlet plenum pressure is calculated,  $z_{PL}$  is the reference height at which the inlet plenum pressure is calculated, and  $z_{IHX}$  is the reference height of the thermal center of the intermediate heat exchanger. The quantity  $p_{gr}$  is calculated in subroutine SSPRIM as PINGR and placed in a COMMON block.

The pump head is calculated as

$$\Delta p_p(t) = \Delta p_o f_p(t) \quad (5.10-3)$$

where  $\Delta p_o$  is the steady-state pump head, taken as

$$\Delta p_o = p_{in}(t=0) - p_{gr} - p_x \quad (5.10-4)$$

and  $f_p(t)$  is the user-supplied time dependence of the pump head, which should be normalized so that

$$f_p(t=0) = 1.0 \quad (5.10-5)$$

There are two options for specifying  $f_p(t)$ . One option is for the user to supply a table of  $f_p$  as a function of time. With this option, the code interpolates linearly between table entries. The other option is for the user to supply the three coefficients  $p_d$ ,  $p_{d1}$ ,  $p_{d2}$  for use in the equation

$$f_p(t) = \exp\left[-(p_d t + p_{d1} t^2 + p_{d2} t^3)\right] \quad (5.10-6)$$

In addition, the time derivatives of the inlet and exit plenum pressures are calculated as

$$\frac{dp_x}{dt} = 0.0 \quad (5.10-7)$$

and

$$\frac{dp_{in}}{dt} = \frac{p_2 - p_1}{\Delta t} \quad (5.10-8)$$

where  $p_1$  and  $p_2$  are the pump head values at the beginning and end of the time interval  $\Delta t$ .

## 5.11 Interaction With Other SAS4A/SASSYS-1 Models

### 5.11.1 Information Received by PRIMAR from Other Models

PRIMAR-1 receives no information from other routines. It is driven entirely by the input regardless of what happens elsewhere in the code. Two kinds of input can be supplied: either a pressure drop as a function of time or the parameters for an exponentially decreasing pressure drop. These are described in Section 5.10 and mentioned in the input listing in Chapter 3.

PRIMAR-4 receives  $C_0, C_1, C_2$ , and  $C_3$  and also the inlet and outlet temperatures for each core channel from the coolant dynamics routines. These are the variables

described in Section 5.2.2. The coolant dynamics routines also supply PRIMAR with the net mass flow and the net mass flow times temperature from all of the core channels to each outlet plenum, as well as the net mass flow and the net mass flow times temperature into all of the core channels from each inlet plenum during the last PRIMAR time step. The PRIMAR model then adjusts the inlet and outlet plenum mass, pressure, and cover-gas interface to account for differences between the estimated channel flows (PRIMAR) and the computed channel flows (coolant dynamics routines) for the last PRIMAR time step. Symbolically,

$$\Delta m = \int w_c dt - \int w_e dt$$

where

$\Delta m$  = the mass adjustment

$w_c$  = the total calculated mass flow rate from all the core channels

$w_e$  = the total estimated mass flow rate from all the core channels

The integration is over the time step. Then

$$m_\ell = m + \Delta m$$

$$V_\ell = m_\ell / \rho_\ell$$

$$z_{inter} = z_{ref} + V_\ell / A_{inter}$$

where

$m_\ell$  = the adjusted liquid mass in the plenum

$m$  = the liquid mass in the plenum before adjustment

$V_\ell$  = the liquid volume in the plenum

$\rho_\ell$  = the liquid density in the plenum

$z_{inter}$  = the liquid interface height in the plenum

$z_{ref}$  = the reference height input for the plenum

$A_{inter}$  = the area of the liquid interface input for the plenum

The coefficients  $C_0$ ,  $C_1$ ,  $C_2$ , and  $C_3$ , supplied to the PRIMAR model by the coolant dynamics routines, are used to estimate the rate of change in the mass flow rate for each channel, as indicated in Eq. 5.2-20 and written more simply for a particular channel as

$$\frac{dw}{dt} = C_0 - C_1 p + C_2 p_x + C_3 w |w|$$

where the symbols have the same meaning as in Eq. 5.2-20. Before boiling begins in a core channel,  $dw/dt$  depends on  $p_{in} - p_x$ , and  $C_1$  is set equal to  $-C_2$ . After boiling starts, the inlet flow is independent of  $p_x$ , and depends on  $p_{in} - p_b$ , where  $p_b$  is the bubble pressure. Also after boiling starts, the outlet flow is independent of  $p_{in}$ , and depends on  $p_b - p_x$ . The bubble pressure  $p_b$  is included in  $C_0$ . In both the preboiling and post-boiling cases,  $C_0$  also includes the gravity head contribution. The last term in Eq. 5.11-1 accounts for friction and orifice pressure drops, and any pressure discrepancies on the right side of Eq. 5.11-1 are also included in  $C_0$ .

### 5.11.2 Information Supplied by PRIMAR to Other Models

PRIMAR supplies information to the pre-voiding coolant dynamics routines, the boiling model, LEVITATE and PLUTO-2. Both PRIMAR-1 and PRIMAR-4 supply the same information, but it is arrived at differently by the two options. The information supplied is

$p_{in}(t_{P1})$  = the inlet plenum pressure at the beginning of the PRIMAR time step

$p_{out}(t_{P1})$  = the outlet plenum pressure at the beginning of the PRIMAR time step

$dp_{in}/dt, dp_{out}/dt$  = the time derivatives of the inlet and outlet plenum pressures

$T_{in} T_{out}$  = the inlet and outlet plenum temperatures

At any time  $t$  during the PRIMAR time step, the inlet and outlet plenum pressures are taken as

$$p_{in}(t) = p_{in}(t_{P1}) + (t - t_{P1}) dp_{in}/dt$$

and

$$p_{out}(t) = p_{out}(t_{P1}) + (t - t_{P1}) dp_{out}/dt$$

The plenum pressures  $p_{in}$  and  $p_{out}$  that PRIMAR supplies to the coolant routines are at the plenum reference heights  $Z_{PLENL}$  and  $Z_{PLENU}$ , and the coolant routines compute the core channel inlet and outlet pressures from these by the formula:

$$p(z_{ci}) = p_{in} + \rho_{in} g(z_{PLENU} - z_{ci})$$

and

$$p(z_{co}) = p_{out} + \rho_{out} g(z_{PLENU} - z_{co})$$

It should be noticed that in the PRIMAR-4 input (Block 18, location 1575) a reference height is entered for each compressible volume, including the compressible volumes that are the inlet and outlet plenums. In addition, inlet and outlet plenum reference heights,  $Z_{PLENL}$  and  $Z_{PLENU}$ , are also entered as input (Block 14, locations 87 and 88 or Block 18, locations 4761-4798). If these two sets of reference are different, the

PRIMAR-4 code calculates the appropriate adjustments and passes the inlet and outlet plenum pressures at  $Z_{PLENL}$  and  $Z_{PLENU}$  to the coolant routines.

The inlet and outlet plenum temperatures are discussed in Section 3.3.6 of Chapter 3.

## 5.12 Subroutine Descriptions and Flow Charts

### 5.12.1 Subroutine Listing and Description

The subroutines used in the PRIMAR-4 module are listed in Table 5.12-1. The driver for the transient subroutines is PRIMAR, and the driver for the steady-state subroutines is SSPRM4.

### 5.12.2 PRIMAR-4 Module Flow Chart

The main subroutines in the PRIMAR-4 module are linked as shown in Fig. 5.12-1.

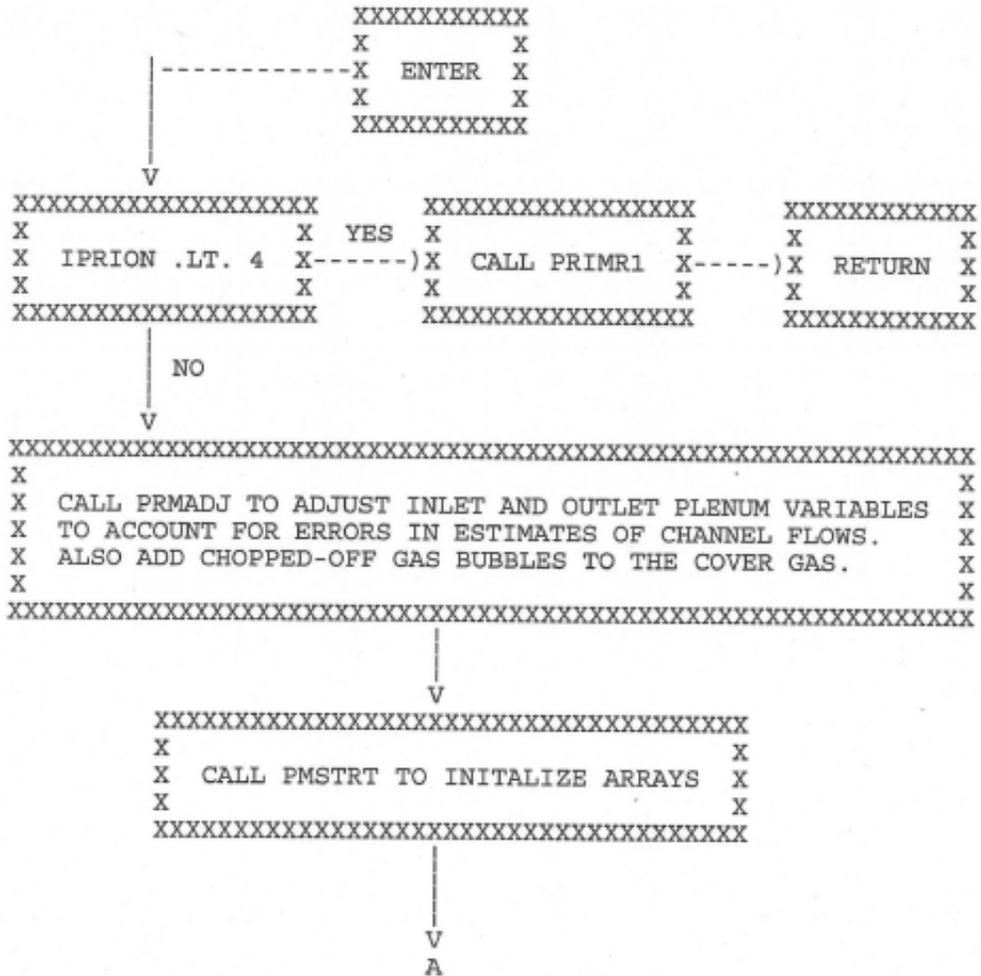


Figure 5.12-1. PRIMAR-4 Module Flow Chart

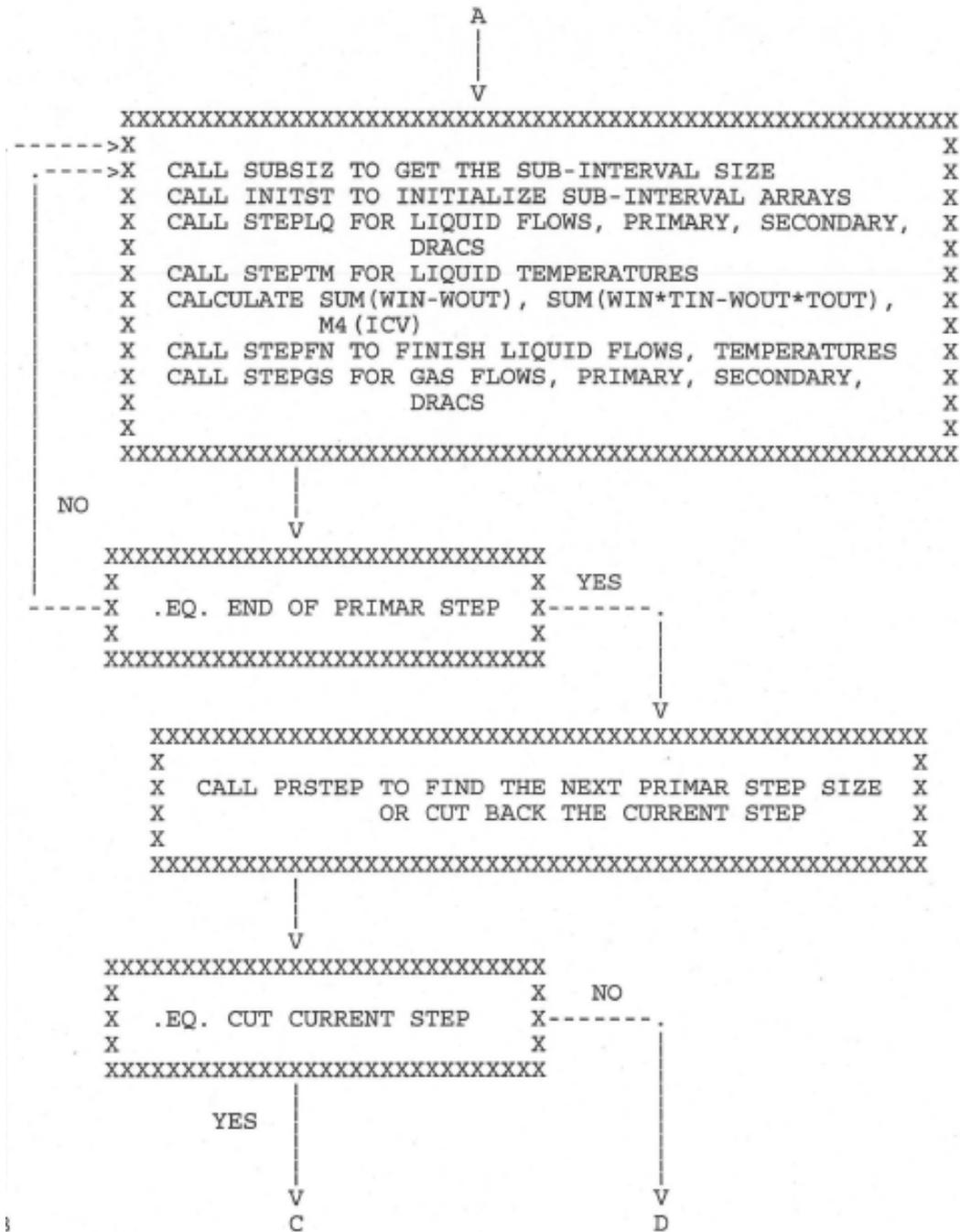


Figure 5.12-1. PRIMAR-4 Module Flow Chart (Cont'd)



Table 5.12-1. PRIMAR-4 Subroutines

<b>Routine</b>	<b>Description</b>
ANETMP	Annular element temperatures
AVGVAL	Average motor torque over a time step for centrifugal pump
BYPSTM	Temperatures of the coolant, reflectors, and duct wall for a bypass channel; called after the flow is known
CKVLFL	Check valve flow coefficients
CMCOPY	Moves blocks of data around in memory
COREFL	Estimates core channel flow
CPCPHT	Component to component heat transfer
CUTBAK	Re-initializes variables when the PRIMAR time step is cut back
CVTEMP	Liquid temperatures for a compressible volume, using a one-point perfect mixing model
DRACSF	Shell-side flow coefficients for DRACS heat exchanger
DRACTF	Tube-side flow coefficients for DRACS heat exchanger
GRVHED	Gravity head
IHXSHF	Flow coefficients for IHX shell-side
IHXTBF	Flow coefficients for IHX tube-side
INCPVF	Pressure coefficients for an almost incompressible liquid volume with no cover gas
INITST	Initializes arrays for a sub-interval
INPLNF	Inlet plenum pressure coefficients
LIQCV	Pressure coefficients for a compressible volume with no cover gas
LIQFIN	Compressible volume pressure at end of sub-interval, with liquid only and no cover gas
LQGSFN	Compressible volume pressure at end of sub-interval, with liquid plus cover gas
OUTNGF	Pressure coefficients for a compressible outlet plenum, no cover gas
OUTPLF	Pressure coefficients for an outlet plenum with cover gas
PIPEFL	Pressure coefficients for pipe flow with friction
PIPTMP	Pipe temperatures using Lagrangian mesh
PMPBLF	Pressure coefficients for a pump bowl with cover gas
PMSTRT	Initializes arrays for a sub-interval
POOLFL	Pressure coefficients for a pool
PRESDR	Pressure drop in a flow element
PRIMAR	Main driver for the module

<b>Routine</b>	<b>Description</b>
PRIMR1	Simple PRIMAR-1 option, with user-specified driving head vs. time, or flow vs. time
PRMADJ	Adjusts inlet and outlet plenum variables to account for the differences between estimated and computed channel flows
PRMEND	Puts final sub-interval results into permanent arrays for storage
PRMPRT	Prints PRIMAR-4 results
PRSRZF	Pressure coefficients for a pressurized (expansion tank)
PRSTEP	Finds the next PRIMAR time-step size or decides to cut back the current step size
PUMPFL	Pump head and flow coefficients
PUMPFN	Pump impeller speed and head calculation at the end of a sub-interval
RUPSKF	Pressure coefficients for a pipe rupture sink, guard vessel with cover gas
RUPSRF	Pipe rupture source pressure coefficients
SELECT	Chooses average and final inlet/outlet temperature into/from a compressible volume, according to segment flow directions
SSADHX	Steady-state air dump heat exchanger
SSARDX	Steady-state air dump DRACS heat exchanger
SSBYPS	Bypass channel steady-state coolant and wall temperatures, given steady-state flow rate and steady-state power
SSCKVL	Steady-state check valve pressure coefficients
SSCPNL	Steady state null transient driver
SSDRAC	Steady-state DRACS temperatures
SSIHX	Steady-state IHX temperatures
SSIHXC	Steady-state IHX temperatures, simple model
SSLQSG	Steady-state initialization of the elements in a liquid segment
SSPMLP	Steady-state initialization of primary loop flows, pressures, and temperatures
SSPRM4	Steady-state initialization of PRIMAR-4 variables -- driver for the steady-state routines
SSPRPL	Initialization for file 15 binary output
SSPRSR	Steady-state pressure and pump head for one segment of an intermediate loop
SSPUMP	Steady-state initialization of pump parameters
SSP4PR	Steady-state PRIMAR-4 printout
SSP4TH	Initialization of node volumes, VOLNDT(ITGP), coolant and wall temperatures, and liquid and wall temperature arrays for pipe-type

<b>Routine</b>	<b>Description</b>
	temperature groups
SSSCLP	Steady-state initialization of the intermediate sodium loops, and also calls the steam generator initialization routines
SSSTDR	Steady-state steam generator driver
SSSTGN	Steady-state steam generator initialization of simple table look-up option
SSSTG1	Steady-state steam generator initialization of detailed model
SSVALV	Steady-state valve pressures
STEPFN	Finishes liquid flows and temperatures for a sub-interval
STEPGS	Gas flow between compressible volumes and storage tanks for a sub-interval
STEPLQ	Driver for liquid flow and pressure calculations
STEPTM	Driver for liquid temperature calculations
STGNFL	Steam generator flow coefficients, sodium side
STRATV	Stratified volume calculations
STRGVH	Calculates gravity heads in a stratified volume
SUBSIZ	Sets the PRIMAR time sub-interval size
TSDRCS	DRACS temperatures
TSIHX	IHX temperatures, shell and tube sides, and gravity head
TSIHXC	IHX temperatures and gravity head, simple model
TSPRPL	Writes binary arrays to file 15 for later use
TSSTGN	Steam generator temperatures and gravity head, simple model
TSSTG1	Steam generator temperatures and gravity head
VALVAJ	Steady-state valve pres. drop adjustment. If needed, between compressible volumes connected by several liquid segments
VALVFL	Valve pressure coefficients

### 5.12.3 Subroutine Calls

Table 5.12-2 lists the PRIMAR-4 module subroutines that call other subroutines, and also the subroutines that are called. PRIMAR is the driver for the transient-state subroutines, and SSPMR4 is the driver for the steady-state subroutines.

The table that gives the subroutines called by each PRIMAR-4 module subroutine is inverted to provide each subroutine called with a list of those calling it. This is shown in Table 5.12-3.

Table 5.12-2. PRIMAR-4 Subroutine Calls

Calling Subroutines	Called Subroutines				
BYPSTM	INVRT3	LINES			
COREFL	LINES				
CUTBAK	CMCOPY				
CVTEMP	LINES				
GRVHED	LINES				
IHXSHF	PIPEFL				
IHXTBF	PIPEFL				
INCPVF	SELECT				
INITST	CMCOPY				
INPLNF	SELECT				
LIQCV	SELECT				
LIQFIN	LINES				
LQGSFN	LINES				
OUTNGF	SELECT				
OUTPLF	SELECT				
PIPEFL	LINES				
PIPTMP	LINES				
PMPBLF	SELECT				
PMSTRT	CMCOPY				
POOLFL	SELECT				
PRESDR	LINES				
PRIMAR	CRDTMP	CUTBAK	DATMOV	INITST	LINES
	PMSTRT	PRIMR1	PRMADJ	PRMEND	PRMPRT
	PRSTEP	STEPFN	STEPGS	STEPLQ	STEPTM
	SUBSIZ	TSPRPL			
PRIMR1	INTIRP	LINES			
PRMADJ	LINES				
PRMEND	CMCOPY				
PRMPRT	LINES				
PRSRZF	SELECT				
PUMPFL	AVRGIT	INTIRP	PIPEFL	TBLKUP	
PUMPFN	TBLKUP				

Calling Subroutines	Called Subroutines					
P4EDT	LINES					
SSBYPS	LINES					
SSIHX	INVRT3	LINES				
SSIHXC	LINES	SSIHX				
SSLQSG	LINES	PRESDR	SSADHX	SSCKVL	SSSTGN	
	SSVALV					
SSPMLP	LINES	PRESDR	SSARDX	SSBYPS	SSCKVL	
	SSDRAC	SSIHXC	SSPUMP	SSVALV		
SSPRIM	DATMOV	LINES				
SSPRM4	DATMOV	LINES	P4EDT	SSPMLP	SSPRPL	
	SSP4CV	SSP4PR	SSP4TH	SSSCLP	VALVAJ	
SSPRPL	LINES					
SSPRSR	LINES	SSPUMP				
SSPUMP	PRESDR					
SSP4CV	LINES					
SSP4PR	LINES					
SSP4TH	LINES					
SSSCLP	LINES	PRESDR	SSLQSG	SSPRSR	SSSTDR	
SSSTGN	LINES					
STEPFN	LIQFIN	LQGSFN	PUMPFN			
STEPLQ	CKVFL	COREFL	DRACSF	IHXSHF	IHXTBF	
	INCPVF	INPLNF	LINES	LIQCV	OUTNGF	
	OUTPLF	PIPEFL	PMPBLF	POOLFL	PRSRZF	
	PUMPPFL	RUPSKF	RUPSRF	STGNFL	STRGVH	
	VALVFL					
STEPTM	BYPSTM	CVTEMP	GRVHED	LINES	PIPTMP	
	STRATV	TSDRCS	TSIHXC	TSSTGN		
STGNFL	PIPEFL					
TSIHX	LINES					
TSIHXC	INTIRP	LINES	TSIHX			
TSSTGN	INTIRP	LINES	TSSTG1			
VALVFL	AVGVAL	PIPEFL				

Table 5.12-3. PRIMAR-4 Inverse Subroutine Calls

<b>Called Subroutines</b>	<b>Calling Subroutines</b>				
AVGVAL	VALVFL				
AVRGIT	PUMPFL				
BYPSTM	STEPTM				
CKVFL	STEPLQ				
CMCOPY	CUTBAK	INITST	PMSTRT	PRMEND	
COREFL	STEPLQ				
CRDTMP	PRIMAR				
CUTBAK	PRIMAR				
CVTEMP	STEPTM				
DATMOV	PRIMAR	SSPRIM	SSPRM4		
DRACSF	STEPLQ				
DRACTF	STEPLQ				
GRVHED	STEPTM				
IHXSHF	STEPLQ				
IHXTBF	STEPLQ				
INCPCV	STEPLQ				
INITST	PRIMAR				
INPLNF	STEPLQ				
INTIRP	PRIMAR1	PUMPFL	PUMPFN	TSIHXC	TSSTGN
INVRT3	BYPSTM	SSIHX			
LINES	BYPSTM	COREFL	CVTEMP	GRVHED	LIQFIN
	LQGSFN	PIPEFL	PIPTMP	PRESDR	PRIMAR
	PRIMR1	PRMADJ	PRMPRT	P4EDT	SSBYP5
	SSIHX	SSIHC	SSLQSG	SSPMLP	SSPRIM
	SSPRM4	SSPRPL	SSPRSR	SSP4CV	SSP4PR
	SSP4TH	SSSCLP	SSSTGN	STEPLQ	STEPTM
	TSIHXC	TSIHXC	TSSTGN		
LIQCV	STEPLQ				
LIQFIN	STEPLQ				
LQGSFN	STEPFN				
OUTNGF	STEPLQ				
OUTPLF	STEPLQ				

<b>Called Subroutines</b>	<b>Calling Subroutines</b>				
PIPEFL	IHXSHF	IHXTFB	PUMPFL	STEPLQ	STGNFL
	VALVFL				
PIPTMP	STEPTM				
PMPBLF	STEPLQ				
PMSTRT	PRIMAR				
POOLFL	STEPLQ				
PRESDR	SSLQSG	SSPMLP	SSPUMP	SSSCLP	
PRIMR1	PRIMAR				
PRMADJ	PRIMAR				
PRMEND	PRIMAR				
PRMPRT	PRIMAR				
PRSRZF	STEPLQ				
PRSTEP	PRIMAR				
PUMPFL	STEPLQ				
PUMPFL	STEPFN				
P4EDT	SSPRM4				
RUPSKF	STEPLQ				
RUPSRF	STEPLQ				
SELECT	INCPVF	INPLNF	LIQCV	OUTNGF	OUTPLF
	PMPBLF	POOLFL	PRSRZF		
SSADHX	SSLQSG				
SSARDX	SSPMLP				
SSBYPS	SSPMLP				
SSCKVL	SSLQSG	SSPMLP			
SSDRAC	SSPMLP				
SSIHX	SSIHXC				
SSIHXC	SSPMLP				
SSLQSG	SSSCLP				
SSPMLP	SSPRM4				
SSPRPL	SSPRM4				
SSPRSR	SSSCLP				
SSPUMP	SSPMLP	SSPRSR			
SSP4CV	SSPRM4				

<b>Called Subroutines</b>	<b>Calling Subroutines</b>	
SSP4PR	SSPRM4	
SSP4TH	SSPRM4	
SSSCLP	SSPRM4	
SSSTDR	SSSCLP	
SSSTGN	SSLQSG	
SSVALV	SSLQSG	SSPMLP
STEPFN	PRIMAR	
STEPGS	PRIMAR	
STEPLQ	PRIMAR	
STEPTM	PRIMAR	
STGNFL	STEPLQ	
STRATV	STEPTM	
STRGVH	STEPLQ	
SUBSIZ	PRIMAR	
TBLKUP	PUMPFL	PUMPFN
TSDRCS	STEPTM	
TSIHX	TSIHXC	
TSIHXC	STEPTM	
TSPRPL	PRIMAR	
TSSTGN	STEPTM	
TSSTG1	TSSTGN	
VALVAJ	SSPRM4	
VALVFL	STEPLQ	

## 5.13 Input/Output Description

### 5.13.1 Input Description

The input description can be divided into the input needed for the PRIMAR-1 option and the input needed for the PRIMAR-4 option.

#### 5.13.1.1 PRIMAR-1 Option

For the PRIMAR-1 option two variables need to be supplied for Block 1 and all the variables for Block 14. The two variables for Block 1 are NPRES, location 19, and IPRION, location 27.

### 5.13.1.2 PRIMAR-4 Option

For the PRIMAR-4 option, one variable needs to be supplied for Block 1, eight variables for Block 14, and all of the variables for Blocks 3 and 18 that are appropriate for the model being considered. The one variable for Block 3 is IPRION, location 27. The eight variables for Block 14 are

Variable Name	Location in Block 14
PX	1
ZPLENL	87
ZPLENU	88
DZBCGL	90
DZBCGU	91
XMMSI	93
XMMSO	94
TIMMIX	95

Block 3 contains the integer input for PRIMAR-4 and Block 18 contains the floating-point input, as described in the input listing in Chapter 2. Only input information pertinent to the model being calculated needs to be entered. For example, if no intermediate loop is present, no information for the intermediate loop in either Block 3 or in block 18 need be entered. Any variable not entered is assumed to be zero, since all of the input blocks are zeroed out before any input information is read in. Also, any extraneous information, which may be left over from an old input deck, is ignored, provided all entries are made correctly. That is, if the number of compressible volumes in an intermediate loop is entered as zero, then any compressible volume information supplied for that intermediate loop will be ignored.

Suggested values for some of the PRIMAR input variables are as follows:

Note 1: The default values for the pump head coefficients in option 2 for the centrifugal pump are:

0.63381, 0.46016, -2.40040, 3.17937, -1.77304, .46236, -.04625,  
 431.96699, -574.61438, 301.00029, -75.46586, 8.67550, -0.26062, -0.01596,  
 6171.9821, -4958.9692, 1406.3329, -126.17344, -13.21712, 3.24505, -.16925

Note 2: The default values for the pump torque coefficients in option 2 for the centrifugal pump are:

-.68437, 2.77599, -5.39880, 6.85412, -4.07579, 1.08133, -0.10476,  
 -1154.9471, 1858.4915, -1237.6683, 436.01653, -85.57377, 8.86277,

-0.37830, -379.81080, 726.14914, -496.25029, 167.64136, -30.36692,  
2.83119, -0.10682

Note 3: Recommended values for the torque loss coefficients (APMPHD(K,IPMP) for K=1,...,11) in option 2 for the centrifugal pump are:

0.01, 0.0, -73.13, 0.00268, 0., 0.07, 0.00383, 0.01071, 0.01406, 0.01,  
0.268

An example of input Block 3 and input Block 18, the integer and floating-point input blocks for PRIMAR-4, is shown below for the example used in Section 5.13.2.

### 5.13.2 Example of a Primary Loop

As an aid to the user in preparing PRIMAR-4 input data, the primary loop in Fig. 5.13-1 is described.

This primary loop consists of five compressible volumes, CV1,...,CV5, five liquid segments, S1,...,S5, fourteen elements, E1,...,E14, distributed among the liquid segments, nine temperature groups, T1,...,T9, of liquid flow elements, and three gas segments. The compressible volumes are:

CV1 = inlet plenum

CV2 = outlet plenum with cover gas

CV3 = pump bowl and cover gas

CV4 = compressible gas volume, no liquid

CV5 = compressible gas volume, no liquid

The elements are:

E1 = core subassemblies

E2, E4, E5, E7, E8, E10, E12, E13, E14 = pipes

E3, E6 = bypass channels

E9 = pump impeller

E11 = shell side of intermediate head exchanger

The elements in liquid segment S5, for example, are E9, E10, E11, E12, E13, and E14.

The assumed directions of flow for the liquid, as well as for the gas, segments are indicated in the diagram. It is this assumed direction of flow that is used in choosing the inlet and outlet compressible volumes for a liquid or for a gas segment. For example,

CV3 is the inlet compressible volume and CV1 is the outlet compressible volume for liquid segment S5, while CV3 and CV4 are the inlet and outlet compressible volumes for gas segment G2. It is also with respect to the assumed directions of the segments that positive and negative flow rates are interpreted. Finally, it should be observed that positive inlet flow to a liquid or gas segment is flow out of the compressible volume at the segment inlet.

The multiplicity of 3 that appears in Fig. 5.13-1 where G2 joins CV4, where E8 joins CV2, and where E14 joins CV1 is present because in this example there are actually three identical primary loops attached to the respective compressible volumes, and instead of performing three identical calculations, the code computes one primary loop and multiplies the appropriate flows by 3. It should be noticed, however, that the multiplicity for liquid segment S4 is 3 at the inlet and 1 at the outlet, for liquid segment S5 it is 1 at the inlet and 3 at the outlet, and for gas segment G2 it is 1 at the inlet and 3 at the outlet. The multiplicity for all other liquid and gas segments must be set to 1 at both ends.

The liquid temperature calculations, which are performed after the liquid flow calculations have been completed, are done by temperature groups. This allows for the grouping of adjacent elements whose temperatures are computed in a similar way, such as elements E12, E13, and E14, all of which are pipes in temperature group T10, and also elements E9, and E10, a pump and connecting pipe, in temperature group T8. Where such grouping is inappropriate, a single element comprises a temperature group, such as the bypass channels as T2 and T5 and the intermediate heat exchanger as T11.

Although not shown in the figure, if an intermediate loop were to be added, it would be structured in a fashion similar to the primary loop, and it would contain an element representing the tube side of the intermediate heat exchanger and would be located next to E11 in the diagram.

### 5.13.3 Output Description

The PRIMAR output information is largely self-explanatory. Specimens of three output printouts for the example in Section 5.13.2 are displayed. The first is an input edit in which all of the input variables in Blocks 3 and 18 are echoed back. The second is a print of the steady-state values calculated by the PRIMAR steady-state routines. Finally, the third is the print of the values calculated by the transient routines at a later time step.

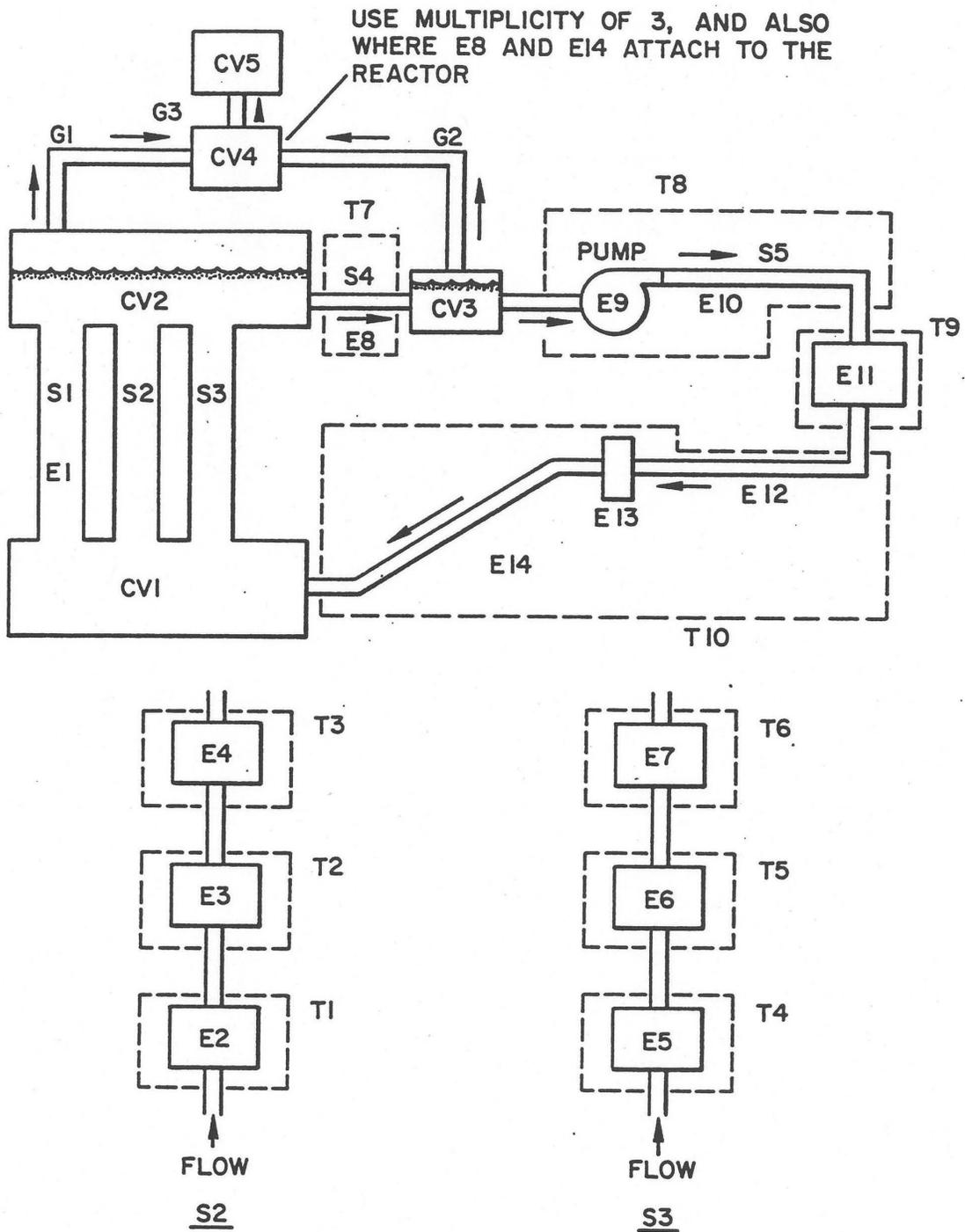


Figure 5.13-1. Example of a Primary Loop



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ELEMENT	AREAL Hm*2	DI/ELEH H	ROUGIIL H	BENDNH	G2PRDR	HALLHC J/H/K	HALLH J/H/K
1	1.000000+00	1.000000+00	5.000000-05	0.0	0.0	1.0955000+05	1.8353000+05
2	1.000000-01	1.000000-01	5.000000-05	0.0	0.0	1.0955000+05	1.8353000+05
3	1.000000-02	1.000000-01	5.000000-05	0.0	0.0	1.0955000+05	1.8353000+05
4	1.000000-02	1.000000-01	5.000000-05	0.0	0.0	1.0955000+05	1.8353000+05
5	1.000000-02	1.000000-01	5.000000-05	0.0	0.0	1.0955000+05	1.8353000+05
6	1.000000-02	1.000000-01	5.000000-05	0.0	0.0	1.0955000+05	1.8353000+05
7	1.000000-02	1.000000-01	5.000000-05	0.0	0.0	1.0955000+05	1.8353000+05
8	6.2072600-01	8.890000-01	5.000000-05	0.0	0.0	1.6549000+05	2.7929000+05
9	2.680000-01	5.842000-01	5.000000-05	0.0	0.0	1.0955000+05	1.8353000+05
10	2.680000-01	5.842000-01	5.000000-05	0.0	0.0	1.0955000+05	1.8353000+05
11	1.9360000+00	3.770000-02	5.000000-05	0.0	0.0	1.0955000+05	1.8353000+05
12	2.680000-01	5.842000-01	5.000000-05	6.8000000+00	0.0	1.0955000+05	1.8353000+05
13	2.680000-01	5.842000-01	5.000000-05	1.0000000+00	0.0	1.0955000+05	1.8353000+05
14	2.680000-01	5.842000-01	5.000000-05	6.8000000+00	0.0	1.0955000+05	1.8353000+05

COMPRESSIBLE VOLUMES IN PRIMARY LOOPS

VOLUME	TYPE	DESCRIPTION	VOLLE6 Hm*3
1	1	INLET PLENUM	8.6400000+01
2	7	OUTLET PLENUM WITH COVER GAS	1.3000000+02
3	9	PUMP BOHL AND COVER GAS	1.6000000+01
4	11	COMPRESSIBLE GAS VOLUME, NO LIQUID	1.0000000+00
5	11	COMPRESSIBLE GAS VOLUME, NO LIQUID	1.0000000+02

VOLUME	PRESGO PA	ALPIAP 1/PA	ALPIAT 1/K	ZCVL H	AREATN Hm*2	TREFCV K	TALGAS S
1	0.0	1.1000000-09	0.0	-1.6000000+00	0.0	6.6090000+02	0.0
2	1.0070000+05	0.0	0.0	3.0000000+00	2.7000000+01	0.0	1.0000000+00
3	1.0070000+05	0.0	0.0	4.8360000+00	3.0700000+00	0.0	1.0000000+00
4	1.0070000+05	0.0	0.0	1.4360000+01	1.0000000+00	8.0620000+02	1.0000000+00
5	1.0070000+05	0.0	0.0	1.4360000+01	1.0000000+01	8.0620000+02	1.0000000+00

LIQUID SEGMENTS IN PRIMARY LOOPS

LIQUID SEGMENT	FIRST ELEMENT	NUMBER OF ELEMENTS	JCVL ONE THO	FLOSSL KG/S	ZINL H	CVLHLT ONE	THO
1	1	1	1	4.7850000+03	-2.9960000+00	1.0000000+00	1.0000000+00
2	2	3	1	1.2900000+02	-2.9960000+00	1.0000000+00	1.0000000+00
3	5	3	1	3.0300000+02	-2.9960000+00	1.0000000+00	1.0000000+00
4	8	1	2	1.7390000+03	4.7560000+00	3.0000000+00	1.0000000+00
5	9	6	3	1.7390000+03	4.8360000+00	1.0000000+00	3.0000000+00

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GAS SEGMENTS IN PRIMARY LOOPS

SEGMENT	JCVG QIE THO	XLENG H	AREASG HM*2	ONE	CVGHIT THO	DHSEGG M	ROUGHG H
1	2 4	1.500000*01	3.491500*03	1.000000*00	1.000000*00	6.667500*02	0.0
2	3 4	1.500000*01	3.491500*03	1.000000*00	3.000000*00	6.667500*02	0.0
3	4 5	5.000000*00	6.658400*03	1.000000*00	1.000000*00	9.207500*02	0.0

PUMPS

IELPMP	IELRPH	PHPSR KG-HM*2	HEADR FA	PHPSPR RAD/S	PHPEFR KG/S	PHPEFR TORQUE	TRKLSG J-S/KG	AMHTTK SEC	EPSCAV
9	1	1.200000*03	1.126750*06	1.166660*02	1.932000*03	9.000000*01	1.000000*01	0.0	0.0
1	0								
1	1	1.229000*00	11	5.640000*01	1	1.000000*00	11	0.0	11
2	1	-1.800000*01	12	3.300000*02	2	0.0	12	1.000000*02	12
3	1	1.920000*01	13	5.530000*01	3	0.0	13	1.000000*03	13
4	1	-2.800000*01	14	-1.520000*01	4	3.000000*00	14	2.500000*01	14
5	1	3.900000*02	15	-1.900000*02	5	4.000000*00	15	3.200000*01	15
6	1	3.090000*00	16	8.000000*03	6	5.000000*00	16	4.000000*02	16
7	1	-2.200000*01	17	8.000000*03	7	6.000000*00	17	4.010000*02	17
8	1	1.144000*01	18	0.0	8	8.000000*00	18	1.000000*04	18
9	1	8.050000*02	19	0.0	9	1.000000*01	19	1.000000*04	19
10	1	6.020000*02	20	3.240000*02	10	1.200000*01	20	1.900000*01	20

TEMPERATURE GROUPS

TEMP. GROUP	NUMBER NODES	FIRST ELEMENT	LAST ELEMENT
1	4	2	2
2	7	3	3
3	4	4	4
4	4	5	5
5	7	6	6
6	4	7	7
7	20	8	8
8	16	9	10
9	41	11	11
10	30	12	14

BYPASS CHANNELS

CHANNEL	NTLBSY	IDKTYP	IELBYP
1	7	1	3
2	7	1	6





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\*\*\*\*\* STEADY-STATE PRIMAR-4 PRINTOUT \*\*\*\*\*

COMPRESSIBLE VOLUMES

C.V.	LIQUID PRESSURE	GAS PRESSURE	LIQUID TEMP.	GAS TEMP.	LIQUID MASS	GAS MASS	GAS VOLUME	TOTAL VOLUME	LIQ.-GAS INTERFACE	REF. PRESSURE	REF. TEMP. OR HEIGHT	TYPE
1	1028019.0	0.0	599.1500	660.9000	75390.0	0.0	0.0	86.40000	-1.60000	1028019.0	599.15000	1
2	155300.0	100700.0	794.7701	794.7701	57907.6	36.53147	60.00000	130.00000	9.74830	0.0	0.0	7
3	136791.9	100700.0	794.7701	794.7701	8603.4	3.40960	5.60000	16.00000	9.29679	0.0	0.0	9
4	0.0	100700.0	808.2000	808.2000	0.0	0.59874	1.00000	1.00000	14.36000	0.0	808.20000	11
5	0.0	100700.0	808.2000	808.2000	0.0	59.87404	100.00000	100.00000	14.36000	0.0	808.20000	11

COMPRESSIBLE VOLUMES (CONTINUED)

ICV	ALPHAP	ALPHAT	ZCVL	AREAIN	BTAPNA	BTAPNA	BTATNA	RHONAR	HVALL	AVALL	CHVALL
1	0.110000-08	0.0	-0.160000+01	0.0	0.213000-09	-0.280000-09	-0.280000-03	0.844000+03	0.500000+05	0.198000+02	0.759000+07
2	0.0	0.0	0.300000+01	0.270000+02	0.213000-09	-0.280000-03	0.844000+03	0.844000+03	0.100000+04	0.393000+02	0.130000+09
3	0.0	0.0	0.483600+01	0.307000+01	0.213000-09	-0.280000-03	0.844000+03	0.844000+03	0.500000+05	0.640000+01	0.245000+07
4	0.0	0.0	0.143600+02	0.100000+01	0.0	-0.280000-03	0.844000+03	0.844000+03	0.500000+05	0.100000+01	0.383000+06
5	0.0	0.0	0.143600+02	0.100000+02	0.0	-0.280000-03	0.844000+03	0.844000+03	0.500000+05	0.218000+02	0.835000+07

COMPRESSIBLE VOLUMES (CONTINUED)

ICV	VOLG50	ZLQREF
1	0.0	0.0
2	6.0000000+01	7.1557030+00
3	5.6000000+03	5.9091660+00
4	1.0000000+00	0.0
5	1.0000000+02	0.0

LIQUID SEGMENTS

ISGL	FLOSSL	FLOSL2	ZINL	CVLHLT 1	CVLHLT 2	JCVL 1	JCVL 2	TSLI 1	TSLI 2
1	4.7850000+03	4.7848500+03	-2.9960000+00	1.0000000+00	1.0000000+00	1	2	0.0	0.0
2	1.2900000+02	1.2899600+02	-2.9960000+00	1.0000000+00	1.0000000+00	1	2	5.9915000+02	6.0796930+02
3	3.0300000+02	3.0299050+02	-2.9960000+00	1.0000000+00	1.0000000+00	1	2	5.9915000+02	6.0290300+02
4	1.7390000+03	1.7389450+03	4.7560000+00	1.0000000+00	1.0000000+00	2	3	7.9477010+02	7.9477010+02
5	1.7390000+03	1.7389450+03	4.8360000+00	1.0000000+00	3.0000000+00	3	1	7.9477010+02	5.9915000+02

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A TABLE FOR LIQUID ELEMENTS

ISGL	IELL,ZOUTEL	GRAVID	TELE 1	TELE 2	XLENEL	AREAEL	DHELEN	ROUGH	BENDNH	G2PRDR	HALLMC	HALLH	DPRSEL
1	1	3.000	0.0	0.0	5.996	1.0000	1.0000	0.0000	0.0	0.0	109550.0	183530.0	0.0
2	2	0.145	599.150	599.150	3.141	0.0100	0.1000	0.0000	0.0	6.7	109550.0	183530.0	741010.2
3	3	1.100	599.150	607.969	0.955	0.0100	0.1000	0.0000	0.0	0.0	109550.0	183530.0	15912.8
4	4	2.700	607.969	607.969	1.600	0.0100	0.1000	0.0000	0.0	0.0	109550.0	183530.0	76732.0
5	5	0.145	599.150	599.150	3.141	0.0100	0.1000	0.0000	1.000	-0.5	109550.0	183530.0	324713.6
6	6	1.100	599.150	602.903	0.955	0.0100	0.1000	0.0000	0.0	0.0	109550.0	183530.0	87457.1
7	7	2.700	602.903	602.903	1.600	0.0100	0.1000	0.0000	1.000	0.0	109550.0	183530.0	421460.9
8	8	4.836	794.770	794.770	33.528	0.6207	0.8890	0.0000	1.000	0.0	165490.0	279290.0	3653.1
9	9	4.836	794.770	794.770	1.000	0.2680	0.5842	0.0000	0.0	0.0	109550.0	183530.0	0.0
10	10	9.326	794.770	794.770	24.384	0.2680	0.5842	0.0000	10.900	0.0	109550.0	183530.0	114065.6
11	11	1.794	599.150	599.150	8.820	1.9360	0.0377	0.0000	0.0	0.0	109550.0	183530.0	2409.2
12	12	-0.900	599.150	599.150	22.360	0.2680	0.5842	0.0000	6.800	0.0	109550.0	183530.0	71282.9
13	13	-0.900	599.150	599.150	1.000	0.2680	0.5842	0.0000	1.000	0.0	109550.0	183530.0	9327.7
14	14	-2.990	599.150	599.150	22.360	0.2680	0.5842	0.0000	6.800	0.0	109550.0	183530.0	71282.9

RESTART NUMBER 1 HAS BEEN WRITTEN ON LOGICAL UNIT 17 ON MAIN TIME STEP 0

2/07/84 03.22.44

OUTLET PLENUM WITH COVER GAS

SAS4A 0.0

\*\*\*\*\* PRIHAR-4 RESULTS \*\*\*\*\*

MAIN TIME STEP 50 PRIHAR TIME STEP 50 TIME 4.7496310+01 SECONDS

LIQUID-GAS INTERFACE

COMP. VOL.	TEMPERATURE K	MASS KG	SUM3	SUM4	MASS CHANGE RATE	ZINTER M	HALL TEMP. K
1	5.9915000+02	7.5267240+04	3.1123040+05	-5.1950820+02	-5.5011580-02	-1.6000000+00	5.9915000+02
2	7.1082580+02	5.8506350+04	1.6984350+05	-2.8161760+02	1.1953050+01	9.7283520+00	7.9406300+02
3	7.9473780+02	8.3312170+03	8.0443680+04	-1.0521750+02	-3.9660130+00	9.1895820+00	7.9475060+02
4	8.0820000+02	0.0	0.0	0.0	0.0	1.4360000+01	8.0820000+02
5	8.0820000+02	0.0	0.0	0.0	0.0	1.4360000+01	8.0820000+02

COMPRESSIBLE VOLUMES, GAS

COMP. VOL.	TEMPERATURE K	MASS KG	XISUM	THISUM
1	6.6090000+02	0.0	3.6533640+01	7.1213510+02
2	7.1213110+02	3.6533640+01	3.4099090+00	7.9431060+02
3	7.9428940+02	3.4093090+00	5.9580970-01	8.0369370+02
4	8.0918480+02	5.9563360-01	5.9874080+01	8.0820100+02
5	8.0820220+02	5.9874050+01	0.0	0.0

LIQUID SEGMENTS

LIQ. SEG.	FLOW RATE KG/S	TEMP. IN K	TEMP. OUT K
1	0.0	5.9915000+02	7.1082580+02
2	1.6180860+01	5.9915000+02	6.0137770+02
3	3.8104240+01	5.9915000+02	6.0012740+02
4	1.6691650+02	7.1082580+02	7.9460860+02
5	1.9052410+02	7.9473780+02	5.9915000+02

GAS SEGMENTS

GAS SEG.	FLOW RATE KG/S
1	-6.9646820-05
2	-3.4566850-05
3	-2.9712290-05

PUMP	HEAD	FLOW	PUMPS SPEED	PUMP TORQUE	MOTOR TORQUE	NORM. SPEED	NORM. FLOW	NORM. HEAD
1	14590.78	190.9241	12.9891	0.0119484	0.0	0.111431	0.0988220	0.0129494

PIPE-TYPE	TEMPERATURES, K	TEMPERATURE GROUP	FIRST ELEMENT	LAST ELEMENT	COOLANT	HALL	COOLANT	HALL
J	599.150	599.150	2	599.150	3	599.150	4	599.150
J	599.150	599.150	1	599.150	2	599.150	3	599.150



SAS4A 0.0

OUTLET PLEINH WITH COVER GAS

2/07/84 03.22.44

IHX TEMPERATURES, K		TEMPERATURE GROUP 9		PRIMARY SIDE ELEMENT NUMBER 11		INTERMEDIATE SIDE ELEMENT NUMBER 0	
J	SHELL COOLANT	TUBE COOLANT	J	SHELL COOLANT	TUBE COOLANT	J	SHELL COOLANT
1	0.0	599.150	2	0.0	599.150		
3	794.769	599.149					
	599.149						

PIPE-TYPE TEMPERATURES, K		TEMPERATURE GROUP 10		FIRST ELEMENT 12		LAST ELEMENT 14	
J	COOLANT HALL	J	COOLANT HALL	J	COOLANT HALL	J	COOLANT HALL
1	599.149	2	599.149	3	599.149	4	599.150
6	599.150	7	599.150	8	599.150	9	599.150
11	599.150	12	599.150	13	599.150	14	599.150
16	599.150	17	599.150	18	599.150	19	599.150
21	599.150	22	599.150	23	599.150	24	599.150
26	599.150	27	599.150	28	599.150	29	599.150
				23	599.150	24	599.150
				28	599.150	29	599.150

LIQUID ELEMENT PRESSURE DROPS, PA		TEMPERATURE GROUP 10		FIRST ELEMENT 12		LAST ELEMENT 14	
ELEM.	GRAVITY HEAD	GRAVITY HEAD	GRAVITY HEAD	GRAVITY HEAD	GRAVITY HEAD	GRAVITY HEAD	GRAVITY HEAD
1	0.0	26791.6	8143.2	1295.9	13639.2	5472.7	26791.6
6	1457.3	13643.9	653.6	-15148.8	0.0	1492.3	36326.8
11	40.4	-62567.0	0.0	946.1	-17822.7		

## 5.14 Data Management

Data management for PRIMAR-4 in Version 3.0 of the SAS4A/SASSYS-1 codes is much simpler than it was for earlier versions of the codes. In earlier versions, large blocks of data were moved around between working memory, where it was used, and a storage area, where it was kept when not currently needed. This earlier data management scheme made it more feasible to use disk storage for the storage area in order to reduce memory requirements. The earlier scheme also made it easier to use a computer with an explicit two level memory architecture including a small fast memory and a larger, slower memory. Explicit two level memories are not currently used, and it has never been feasible to run the SAS series of codes using disk storage for the storage area, because disk speeds are too slow to keep up with central processor speeds for these codes. Also, computer memory is now plentiful and cheap. Therefore, in Version 3.0, all PRIMAR-4 variables other than a few local variables are stored in the COMMON blocks where they are used and never moved to a storage area. Most temporary PRIMAR-4 variables that do not need to be saved from step to step are put in temporary storage areas in blank common and in labeled common WORKSP. These temporary storage areas are shared with other modules in the codes. Input variables are stored in their own separate labeled common blocks (INPMR4, PRIMIN, and PMR4IN). Computed variables that need to be saved for later use are stored in labeled common CMPMR4.

## REFERENCES

### NOTICE

Several references in this document refer to unpublished information. For a list of available open-literature citations, please contact the authors.



## APPENDIX 5.1: IHX MATRIX COEFFICIENTS

The coefficients in Eq. 5.4-32 for the  $j$ -th vertical section of the shell in terms of the quantities defined in Section 5.4.2.2 are as follows:

$$a_1(j) = (\rho c)_{SH} d_{SH} + \theta_{2S} \Delta t H_S(j) + \theta_{2S} \Delta t (hA)_{snk} / P_S \quad (\text{A5.1-1})$$

$$a_2(j) = -\frac{1}{2} \theta_{2S} \Delta t H_S(j) \quad (\text{A5.1-2})$$

$$a_3(j) = -\frac{1}{2} \theta_{2S} \Delta t H_S(j) \quad (\text{A5.1-3})$$

$$a_4(j) = -\Delta t H_S(j) T_{SH3}(j) + \Delta t H_S(j) \bar{T}_{CS3}(j) + \Delta t (hA)_{snk} [T_{snk} - T_{SH3}(j)] / P_S \quad (\text{A5.1-4})$$

$$\bar{T}_{CP3}(j) = \frac{1}{2} [T_{CS3}(j) + T_{CS3}(j+1)] \quad (\text{A5.1-5})$$

where

$\theta_{2S}$  = the degree of implicitness for the shell-side coolant channel

$\Delta t$  = the time interval

The coefficients in Eq. 5.4-33 for the  $j$ -th vertical section of the shell-side coolant for normal flow (downward) are:

$$e_1(j) = \frac{1}{2} A_c \bar{\rho}_{CS}(j) \bar{c}_{CS}(j) + \Delta t \frac{\bar{c}_{CS}(j)}{\Delta z(j)} \theta_{2S} |w_{S4}| + \Delta t P_S H_S(j) \frac{1}{2} \theta_{2S} + \Delta t S P_{ST} H_{ST}(j) \frac{1}{2} \theta_{2S} \quad (\text{A5.1-6})$$

$$e_2(j) = 0 \quad (\text{A5.1-7})$$

$$e_3(j) = -\Delta t P_S H_S(j) \theta_{2S} \quad (\text{A5.1-8})$$

$$e_4(j)=0 \quad (\text{A5.1-9})$$

$$e_5(j)=-\Delta t S P_{ST} H_{ST}(j)\theta_{2S} \quad (\text{A5.1-10})$$

$$e_6(j)=0 \quad (\text{A5.1-11})$$

$$e_7(j)=\frac{1}{2} A_{CS} \bar{\rho}_{CS}(j)\bar{c}_{CS}(j)-\Delta t \frac{\bar{c}_{CS}(j)}{\Delta z(j)} \theta_{2S} |w_{S4}| \quad (\text{A5.1-12})$$

$$+\Delta t P_S H_S(j)\frac{1}{2}\theta_{2S} +\Delta t S P_{ST} H_{ST}(j)\frac{1}{2}\theta_{2S}$$

$$e_8(j)=-\Delta t \frac{\bar{c}_{CS}(j)}{\Delta z(j)} \left\{ \theta_{1S} |w_{S3}| + \theta_{2S} |w_{S4}| \left[ T_{CS3}(j) - T_{CS3}(j+1) \right] \right\} \quad (\text{A5.1-13})$$

$$+\Delta t P_S H_S(j) \left\{ T_{SH3}(j) - \frac{1}{2} [T_{CS3}(j) + T_{CS3}(j+1)] \right\}$$

$$+\Delta t S P_S H_{ST}(j) \left\{ T_{TU3}(j) - \frac{1}{2} [T_{CS3}(j) + T_{CS3}(j+1)] \right\}$$

The same coefficients for reversed flow (upward) in the shell-side coolant channel are:

$$e(j)=\frac{1}{2} A_{CS} \bar{\rho}_{CS}(j-1)\bar{c}_{CS}(j-1)+\Delta t \frac{\bar{c}_{CS}(j-1)}{\Delta z(j-1)} \theta_{2S} |w_{S4}| \quad (\text{A5.1-14})$$

$$+\Delta t P_S H_S(j-1)\frac{1}{2}\theta_{2S} +\Delta t S P_{ST} H_{ST}(j-1)\frac{1}{2}\theta_{2S}$$

$$e_2(j)=-\Delta t P_S H_S(j-1)\theta_{2S} \quad (\text{A5.1-15})$$

$$e_3(j)=0 \quad (\text{A5.1-16})$$

$$e_4(j)=-\Delta t S P_{ST} H_{ST}(j-1)\theta_{2S} \quad (\text{A5.1-17})$$

$$e_5(j)=0 \quad (\text{A5.1-18})$$

$$\begin{aligned}
 e_6(j) = & \frac{1}{2} A_{CS} \bar{\rho}_{CS}(j-1) \bar{c}_{CS}(j-1) - \Delta t \frac{\bar{c}_{CS}(j-1)}{\Delta z(j-1)} \theta_{2S} |w_{S4}| \\
 & + \Delta t P_S H_S(j-1) \frac{1}{2} \theta_{2S} + \Delta t S P_{ST} H_{ST}(j-1) \frac{1}{2} \theta_{2S}
 \end{aligned} \tag{A5.1-19}$$

$$e_7(j) = 0 \tag{A5.1-20}$$

$$\begin{aligned}
 e_8(j) = & -\Delta t \frac{\bar{c}_{CS}(j-1)}{\Delta z(j-1)} \left\{ (\theta_{1S} |w_{S3}| + \theta_{2S} |w_{S4}|) [T_{CS3}(j) - T_{CS3}(j-1)] \right\} \\
 & + \Delta t P_S H_S(j-1) \left\{ T_{SS3}(j-1) - \frac{1}{2} [T_{CS3}(j-1) + T_{CS3}(j)] \right\} \\
 & + \Delta t S P_{ST}(j-1) \left\{ T_{TU3}(j-1) - \frac{1}{2} [T_{CS3}(j-1) + T_{CS3}(j)] \right\}
 \end{aligned} \tag{A5.1-21}$$

The terms  $e_9(j)$  and  $e_{10}(j)$  have been added to Eq. 5.4-33 because they appear during the solution of the simultaneous equations. These arrays are set to zero before the solution is begun.

In addition, the boundary conditions for normal shell-side coolant channel flow are

$$e_1(j \max) = 1; e_{2,3,4,5,6,7}(j \max) = 0; e_8(j \max) = \Delta T_{CS}(j \max) \tag{A5.1-22}$$

For reversed primary channel flow, they are

$$e_1(1) = 1; e_{2,3,4,5,6,7}(1) = 0; e_8(1) = \Delta T_{CS}(1) \tag{A5.1-23}$$

and for both cases, they are

$$\begin{aligned}
 e_2(1) = 0; e_4(1) = 0; e_6(1) = 0; \\
 e_3(j \max) = 0; e_5(j \max) = 0; e_7(j \max) = 0
 \end{aligned} \tag{A5.1-24}$$

The coefficients in Eq. 5.4-34 for the  $j$ -th vertical section of the tube are:

$$\begin{aligned}
 c_1(j) = & (\rho c)_{TU} \frac{1}{2} (P_{ST} + P_{TT}) d_{TU} + \Delta t \theta_{2S} P_{ST} H_{ST}(j) \\
 & + \Delta t \theta_{ST} P_{TT} H_{TT}(j)
 \end{aligned} \tag{A5.1-25}$$

$$c_2(j) = -\frac{1}{2} \Delta t \theta_{2S} P_{ST} H_{ST}(j) \quad (\text{A5.1-26})$$

$$c_3(j) = -\frac{1}{2} \Delta t \theta_{2S} P_{ST} H_{ST}(j) \quad (\text{A5.1-27})$$

$$c_4(j) = -\frac{1}{2} \Delta t \theta_{2T} P_{TT} H_{TT}(j) \quad (\text{A5.1-28})$$

$$c_5(j) = -\frac{1}{2} \Delta t \theta_{2T} P_{TT} H_{TT}(j) \quad (\text{A5.1-29})$$

$$\begin{aligned} c_6(j) = & -\Delta t [P_{ST} H_{ST}(j) + P_{TT} H_{TT}(j)] \\ & + \Delta t P_{ST} H_{ST}(j) \frac{1}{2} [T_{CS3}(j) + T_{CS3}(j+1)] \\ & + \Delta t P_{TT} H_{TT}(j) \frac{1}{2} [T_{CT3}(j) + T_{CT3}(j+1)] \end{aligned} \quad (\text{A5.1-30})$$

The coefficients in Eq. 5.4-35 for the j-th vertical section of the tube-side coolant for normal flow (upward) are:

$$\begin{aligned} f_1(j) = & \frac{1}{2} A_{CT} \bar{\rho}_{CT}(j-1) \bar{c}_{CT}(j-1) + \Delta t \frac{\bar{c}_{CT}(j-1)}{\Delta z(j-1)S} \left| w_{T4} \right| \theta_{2T} \\ & + \Delta t P_{TT} H_{TT}(j-1) \frac{1}{2} \theta_{2T} \end{aligned} \quad (\text{A5.1-31})$$

$$f_2(j) = -\Delta t P_{TT} H_{TT}(j-1) \theta_{2T} \quad (\text{A5.1-32})$$

$$f_3(j) = 0 \quad (\text{A5.1-33})$$

$$\begin{aligned} f_4(j) = & \frac{1}{2} A_{CT} \bar{\rho}_{CT}(j-1) - \Delta t \frac{\bar{c}_{CT}(j-1)}{\Delta z(j-1)S} \left| w_{T4} \right| \theta_{2T} \\ & + \Delta t P_{TT} H_{TT}(j-1) \frac{1}{2} \theta_{2T} \end{aligned} \quad (\text{A5.1-34})$$

$$f_5(j)=0 \quad (\text{A5.1-35})$$

$$f_6(j)=-\Delta t \frac{\bar{c}_{CT}(j-1)}{\Delta z(j-1)S} \left\{ \left( |w_{T3}| \theta_{1T} + |w_{T4}| \theta_{2T} \right) [T_{CT3}(j)-T_{CT3}(j-1)] \right\} \\ + \Delta t P_{TT} H_{TT}(j-1) \left\{ T_{TU3}(j-1) - \frac{1}{2} [T_{CT3}(j-1) + T_{CT3}(j)] \right\} \quad (\text{A5.1-36})$$

The same coefficients for reversed flow (downward) in the intermediate coolant channel are:

$$f_1(j)=\frac{1}{2} A_{CT} \bar{\rho}_{CT}(j) \bar{c}_{CT}(j) + \Delta t \frac{\bar{c}_I(j)}{\Delta z(j)S} |w_{T4}| \theta_{2T} \\ + \Delta t P_{TT} H_{TT}(j) \theta_{2T} \quad (\text{A5.1-37})$$

$$f_2(j)=0 \quad (\text{A5.1-38})$$

$$f_3(j)=-\Delta t P_{TT} H_{TT}(j) \theta_{2T} \quad (\text{A5.1-39})$$

$$f_4(j)=0 \quad (\text{A5.1-40})$$

$$f_5(j)=\frac{1}{2} A_{CT} \bar{\rho}_{CT}(j) \bar{c}_{CT}(j) - \Delta t \frac{\bar{c}_{CT}(j)}{\Delta z(j)S} |w_{T4}| \theta_{2T} \\ + \Delta t P_{TT} H_{TT}(j) \frac{1}{2} \theta_{2T} \quad (\text{A5.1-41})$$

$$f_6(j)=-\Delta t \frac{\bar{c}_{CT}(j)}{\Delta z(j)S} \left\{ \left( |w_{T3}| \theta_{1T} + |w_{T4}| \theta_{2T} \right) [T_{CT3}(j)-T_{CT3}(j+1)] \right\} \\ + \Delta t P_{TT} H_{TT}(j) \left\{ T_{TU3}(j) - \frac{1}{2} [T_{CT3}(j) + T_{CT3}(j+1)] \right\} \quad (\text{A5.1-42})$$

The terms for  $f_7(j)$  and  $f_8(j)$  have been added to Eq. 5.4-35 because they appear during the solution of the simultaneous equations. These arrays are also set to zero before the solution is begun.

Also, the boundary conditions for normal tube-side coolant channel flow are

$$f_1(1)=1; f_{2,3,4,5}(1)=0; f_6(1)=\Delta T_{CT}(1) \quad (\text{A5.1-43})$$

For reversed tube-side channel flow, they are

$$f_1(j \text{ max})=1; f_{2,3,4,5}(j \text{ max})=0; f_6(j \text{ max})=\Delta T_{CT}(j \text{ max}) \quad (\text{A5.1-44})$$

and for both cases, they are

$$\begin{aligned} f_2(1)=0; f_4(1)=0; \\ f_1(j \text{ max})=0; f_5(j \text{ max})=0 \end{aligned} \quad (\text{A5.1-45})$$

## APPENDIX 5.2: IHX MATRIX SOLUTION ALGORITHM

The solution of the matrix represented by Eqs. 5.4-32 through 5.4-35 and whose coefficients are given in Appendix 5.1 is accomplished by Gaussian elimination, making use of the zeros present in the matrix. It is presented as an algorithm as it is coded in subroutine TSIHX. The arrows in the following mean replacement of what is on the left by the expression on the right.

JMAX= the number of nodes in the primary and intermediate coolant

JMAX -1= the number of nodes in the shell and tube

1. Set  $j = 1$ .

2. Multiply Eq. 5.4-32 by  $1/a_1(j)$

$$a_2(j) \rightarrow \frac{a_2(j)}{a_1(j)}; a_3(j) \rightarrow \frac{a_3(j)}{a_1(j)}; a_4(j) \rightarrow \frac{a_4(j)}{a_1(j)}; a_5 = \frac{a_5(j)}{a_1(j)}; a_1(j) \rightarrow 1;$$

3.  $e_1(j) \rightarrow e_1(j) - e_3(j)a_2(j)$

$$e_7(j) \rightarrow e_7(j) - e_3(j)a_3(j);$$

$$e_8(j) \rightarrow e_8(j) - e_3(j)a_4(j);$$

$$e_3(j) \rightarrow 0;$$

4. Multiply non-zero coefficients in Eq. 5.4-33 by  $1/e_1(j)$ .

$$e_5(j) \rightarrow \frac{e_5(j)}{e_1(j)}; e_7(j) \rightarrow \frac{e_7(j)}{e_1(j)};$$

$$e_8(j) \rightarrow \frac{e_8(j)}{e_1(j)}; e_{10}(j) \rightarrow \frac{e_{10}(j)}{e_1(j)}$$

$$e_1(j) \rightarrow 1$$

5.  $c_1(j) \rightarrow c_1(j) - c_2(j)e_5(j)$

$$c_3(j) \rightarrow c_3(j) - c_2(j)e_7(j)$$

$$c_4(j) \rightarrow c_4(j) - c_2(j)e_{10}(j)$$

$$c_6(j) \rightarrow c_6(j) - c_2(j)e_8(j)$$

$$c_2(j) \rightarrow 0$$

$$f_1(j) \rightarrow f_1(j) - f_7(j)e_{10}(j)$$

$$f_3(j) \rightarrow f_3(j) - f_7(j)e_5(j)$$

$$f_6(j) \rightarrow f_6(j) - f_7(j)e_8(j)$$

$$f_8(j) \dashrightarrow f_8(j) - f_7(j)e_7(j)$$

$$f_7(j) \dashrightarrow 0$$

$$e_1(j+1) \dashrightarrow e_1(j+1) - e_6(j+1)e_7(j)$$

$$e_4(j+1) \dashrightarrow e_4(j+1) - e_6(j+1)e_5(j)$$

$$e_8(j+1) \dashrightarrow e_8(j+1) - e_6(j+1)e_8(j)$$

$$e_9(j+1) \dashrightarrow e_9(j+1) - e_6(j+1)e_{10}(j)$$

$$e_6(j+1) \dashrightarrow 0$$

6. Multiply non-zero coefficients in Eq. 5.4-34 by  $1/c_1(j)$ .

$$c_3(j) \dashrightarrow \frac{c_3(j)}{c_1(j)}; c_4(j) \dashrightarrow \frac{c_4(j)}{c_1(j)}$$

$$c_5(j) \dashrightarrow \frac{c_5(j)}{c_1(j)}; c_6(j) \dashrightarrow \frac{c_6(j)}{c_1(j)}$$

$$c_1(j) \dashrightarrow 1$$

7.  $f_1(j) \dashrightarrow f_1(j) - f_3(j)c_4(j)$

$$f_8(j) \dashrightarrow f_8(j) - f_3(j)c_3(j)$$

$$f_5(j) \dashrightarrow f_5(j) - f_3(j)c_5(j)$$

$$f_6(j) \dashrightarrow f_6(j) - f_3(j)c_6(j)$$

$$f_3(j) \dashrightarrow 0$$

$$e_1(j+1) \dashrightarrow e_1(j+1) - e_4(j+1)c_3(j)$$

$$e_9(j+1) \dashrightarrow e_9(j+1) - e_4(j+1)c_4(j)$$

$$e_{10}(j+1) \dashrightarrow e_{10}(j+1) - e_4(j+1)c_5(j)$$

$$e_8(j+1) \dashrightarrow e_8(j+1) - e_4(j+1)c_6(j)$$

$$e_4(j+1) \dashrightarrow 0$$

$$f_1(j+1) \dashrightarrow f_1(j+1) - f_2(j+1)c_5(j)$$

$$f_4(j+1) \dashrightarrow f_4(j+1) - f_2(j+1)c_4(j)$$

$$f_7(j+1) \dashrightarrow f_7(j+1) - f_2(j+1)c_3(j)$$

$$f_6(j+1) \dashrightarrow f_6(j+1) - f_2(j+1)c_6(j)$$

$$f_2(j+1) \rightarrow 0$$

8. Multiply non-zero coefficients in Eq. 5.4-35 by  $1/f_1(j)$ .

$$f_5(j) \rightarrow \frac{f_5(j)}{f_1(j)}; f_6(j) \rightarrow \frac{f_6(j)}{f_1(j)};$$

$$f_8(j) \rightarrow \frac{f_8(j)}{f_1(j)}; f_1(j) \rightarrow 1$$

9.  $e_1(j+1) \rightarrow e_1(j+1) - e_9(j+1)f_5(j)$

$$e_{10}(j+1) \rightarrow e_{10}(j+1) - e_9(j+1)f_5(j)$$

$$e_8(j+1) \rightarrow e_8(j+1) - e_9(j+1)f_6(j)$$

$$e_9(j+1) \rightarrow 0$$

$$f_1(j+1) \rightarrow f_1(j+1) - f_4(j+1)f_5(j)$$

$$f_7(j+1) \rightarrow f_7(j+1) - f_4(j+1)f_8(j)$$

$$f_6(j+1) \rightarrow f_6(j+1) - f_4(j+1)f_6(j)$$

$$f_4(j+1) \rightarrow 0$$

10. Set  $j \rightarrow j+1$

11. If  $j < JMAX$ , go to step 2

12. Multiply the non-zero coefficients in Eq. 5.4-35 by  $1/f_1(j)$

$$f_6(j) \rightarrow \frac{f_6(j)}{f_1(j)}; f_7(j) \rightarrow \frac{f_7(j)}{f_1(j)}; f_1(j) \rightarrow 1$$

13.  $e_1(j) \rightarrow e_1(j) - e_{10}(j)f_7(j)$

$$e_8(j) \rightarrow e_8(j) - e_{10}(j)f_6(j)$$

$$e_{10}(j) \rightarrow 0$$

14.  $e_8(j) \rightarrow \frac{e_8(j)}{e_1(j)}; e_1(j) \rightarrow 1$

15.  $\Delta T_{CP}(j) = e_8(j)$

$$\Delta T_{CT}(j) = f_6(j) - f_7(j)\Delta T_{CS}(j)$$

16. Set  $j \rightarrow j-1$

17. If  $j < 1$ , go to step 23

18.  $\Delta T_{CT}(j) = f_6(j) - f_5(j)\Delta T_{CT}(j+1) - f_8(j)\Delta T_{CS}(j+1)$

19.  $\Delta T_{TU}(j) = C_6(j) - C_3(j)\Delta T_{CS}(j+1) - C_4(j)\Delta T_{CS}(j+1) - C_5(j)\Delta T_{CT}(j+1)$

20.  $\Delta T_{CS}(j) = e_8(j) - e_5(j)\Delta T_{TU}(j) - e_7(j)\Delta T_{CS}(j+1) - e_{10}(j)\Delta T_{CT}(j)$

21.  $\Delta T_{SH}(j) = a_4(j) - a_2(j)\Delta T_{CS}(j) - a_3(j)\Delta T_{CS}(j+1)$

22. Go to step 16

23. End

### APPENDIX 5.3: COVER GAS FLOW AND PRESSURE ALGORITHM

The main steps in the cover gas treatment in Section 5.7 may be summarized in the following algorithm:

1. Initialize. Set  $I_n = 0$ .
2. Start a loop on  $I_o$  for  $I_o = I_{CV1}, \dots, I_{CV2}$ . Set up  $INEW(I_o)$ ,  $IOLD(I_n)$ , where  $I_o$  is the old compressible volume number and  $I_n$  is the new compressible volume number for the compressed arrays.
3. Is there gas in this compressible volume? Is  $ITYPCV(I_o) = 2, 3, 4, 5$ , or 8? If so, go to step 5.
4. No gas in this compressible volume.  $INEW(I_o) = 0$ ,  $p_4(I_o) = p_3(I_o)$ ,  $T_4(I_o) = T_3(I_o)$ ,  $m_4(I_o) = 0$ . Go to the end of the loop, step 6.
5. Gas in this compressible volume.

$$I_n = I_n + 1$$

$$INEW(I_o) = I_n$$

$$IOLD(I_n) = I_o$$

$$p'_3(I_n) = p_3(I_o), m'_3(I_n) = m_3(I_o),$$

$$V'_3(I_n) = V_3(I_o), V'_4(I_n) = V_4(I_o),$$

$$\tau'(I_n) = \tau(I_o)$$

6. End of loop on  $I_o$ . Loop back to step 3 for the next value of  $I_o$ .

7.  $I_{nmax} = I_n$

8. Start a loop on  $I_n$  for  $I_n = 1, \dots, I_{nmax}$

Compute the adiabatic expansion and heat flow, as well as a temporary array  $c(i)$  (see below).

9. Re-calculate  $p'_3(I_o)$  giving.

$$p'_3(I_n) = p'_3(I_n) \left\{ 1 - \gamma \left[ \frac{V'_4(I_n) - V'_3(I_n)}{V'_4(I_n)} \right] \right\}$$

Calculate  $T'_3(I_n)$ .

$$\text{Set } c(I_n) = \frac{p'_3(I_n) \gamma \Delta t_s}{m'_3(I_n) T'_3(I_n)}$$

Also set  $c(I,J) = 0$ , with  $c(I_n, I_n) = 0$ ,

and  $d(I_n) = 0$ .

10. End of loop on  $I_n$ . Loop back to step 9 for the next value of  $I_n$ .

11. Is there more than one gas compressible volume? Is  $I_{nmax} > 1$ ? If so, go to step 13.

12. Go to step 25.

13. Start loop on  $I$  for  $I = 1, \dots, I_{nmax}-1$ .

Initialize  $e_{ij}, F_{1ij}, F_{2ij}$ .

14. Start loop on  $J$  for  $J = I_n + 1, \dots, I_{nmax}$

15. Set  $c(I,J) = 0.0, c(J,I) = 0.0, F_1(I,J) = 0.0,$

$F_1(J,I) = 0.0, F_2(I,J) = 0.0, F_2(J,I) = 0.0$

16. End of loop on  $J$ . Loop back to step 11 for additional values of  $J$ .

17. End of loop on  $I$ . Loop back to step 14 for additional values of  $I$ .

18. Are there any gas segments? Is  $ISG1 > 0$ ?

If not, go to step 25.

19. Start loop on  $I_s$  for  $I_s = ISG1, \dots, ISG2$ .

Compute gas flow between compressible volumes.

20. Find  $I_{ni}$  and  $I_{no}$ , the inlet and outlet compressible volumes for gas segment  $I_s$ .

$I_{ni} = INEW(JNODG(1, I_s))$

$I_{no} = INEW(JNODG(2, I_s))$

21. Iterate routine to obtain  $F'_o, F'_1, F'_2, T'_{ij}$  (See remarks after Eq. 5.5-12).

22.

$$\begin{aligned}
 F_1(I_{ni}, I_{no}) &= F_1(I_{ni}, I_{no}) + F_1' \\
 F_1(I_{no}, I_{ni}) &= F_1(I_{no}, I_{ni}) + F_1' \\
 F_2(I_{ni}, I_{no}) &= F_2(I_{ni}, I_{no}) + F_2' \\
 F_2(I_{no}, I_{ni}) &= F_2(I_{no}, I_{ni}) + F_2' \\
 c(I_{no}, I_{no}) &= c(I_{no}, I_{no}) - e(I_{no})T_{ij} F_1' \\
 c(I_{no}, I_{ni}) &= c(I_{no}, I_{ni}) - e(I_{no})T_{ij} F_1' \\
 c(I_{ni}, I_{ni}) &= c(I_{ni}, I_{ni}) - e(I_{ni})T_{ij} F_2' \\
 c(I_{ni}, I_{no}) &= c(I_{ni}, I_{no}) - e(I_{ni})T_{ij} F_2' \\
 d(I_{no}) &= d(I_{no}) + e(I_{no})T_{ij} F_0' \\
 d(I_{ni}) &= d(I_{ni}) - e(I_{ni})T_{ij} F_0' \\
 F_{os}(I_{ni}) &= F_{os}(I_{no}) + F_0' \\
 F_{os}(I_{ni}) &= F_{os}(I_{no}) - F_0' \\
 F_{os}(I_s) &= F_0' \\
 F_{s1}(I_s) &= F_1' \\
 F_{s2}(I_s) &= F_2'
 \end{aligned}$$

23. End of loop on  $I_s$ . Loop back to step 20 for the next value of  $I_s$ .

24. Solve the matrix equation for  $\Delta p(i)$  (see Eq. 5.5-20).

25. Start loop on  $I_n$  for  $I_n = 1, \dots, I_{nmax}$

26. 
$$\begin{aligned}
 p_4'(I) &= p_3'(I) + \Delta p_i \\
 m_4'(I) &= m_3'(I) + \Delta t_x F_{os}(I)
 \end{aligned}$$

27. End of loop on  $I_n$ . Loop back to step 26 for additional values of  $I_n$ .

28. If  $I_{nmax} = 1$ , go to step 34.

29. Start loop on  $I$  for  $I = 1, \dots, I_{nmax} - 1$ .

30. Start loop on  $J$  for  $J = I + 1, I_{nmax}$ .

31. 
$$\begin{aligned}
 m_4'(I) &= m_4'(I) + \Delta t_s [F_1(J, I)\Delta p(I) + F_2(J, I)\Delta p(J)] \\
 m_4'(J) &= m_4'(J) + \Delta t_s [F_1(I, J)\Delta p(J) + F_2(I, J)\Delta p(I)]
 \end{aligned}$$

32. End of loop on  $J$ . Loop back to step 31 for additional values of  $J$ .

33. End of loop on  $I$ . Loop back to step 30 for additional values of  $I$ .

34. For  $I = 1, \dots, I_{nmax}$  set

$$T_4'(I) = \frac{p_4'(I) V_4'(I)}{n_4'(I) R}$$

35. Start loop on  $I_n$  for  $I_n = 1, \dots, I_{nmax}$

36.  $I_o = IOLD(I_n)$

$$p_3(I_o) = p'_3(I_n)$$

$$m_3(I_o) = m'_3(I_n)$$

$$T_3(I_o) = T'_3(I_n)$$

37. End of loop on  $I_n$ . Loop back to step 36 for additional values of  $I_o$ .

38. If  $ISG1 = 0$ , go to step 40.

39. For  $I_s = ISG1, \dots, ISG2$

$$I_1 = INEW(JNODG(1, I_s))$$

$$I_2 = INEW(JNODG(2, I_s))$$

$$F_{g4}(I_s) = F_{so}(I_s) + F_{s1}(I_s)\Delta p(I_2) = F_{s2}(I_s)\Delta p(I_1)$$

40. Optional debugging print-out.

41. Return

## APPENDIX 5.4: AIR BLAST HEAT EXCHANGER STACK MOMENTUM EQUATION

This appendix derives an expression for the air mass flowrate through the natural convection stack. The stack contains an opening at its base through which air is drawn in, the air passes over the finned tubes of the air blast heat exchanger and then rises to be exhausted at the top of the stack.

The one-dimensional steady-state momentum equation for flow in a channel of uniform cross section is

$$\frac{dP}{dz} = \frac{-d}{dz}(\rho v^2) - \rho g \sin \theta - \tau \frac{P_w}{A}, \quad (\text{A5.4-1})$$

where

$p$ =pressure

$\rho$ =density

$v$ =velocity

$\tau$ =wall shear stress

$P_w$ =wetted perimeter

$A$ =flow area

$\theta$ =channel inclination relative to horizontal

Integrating Eq. (A5.4-1) gives the pressure change along the channel

$$\Delta P = -\left(\frac{w}{A}\right)^2 \left(\frac{1}{\rho_o} - \frac{1}{\rho_i}\right) - \rho_m g \ell \sin \theta - \frac{K}{2\rho_m} \left(\frac{w}{a}\right)^2 \quad (\text{A5.4-2})$$

where

$w$ =channel mass flowrate

$\ell$  =channel length

$\rho_o$ =outlet density

$\rho_i$ =inlet density

$\rho_m$ =mean density

$K$ =flow loss coefficient

Using Eq. (A5.4-2), the pressure change from stack inlet to above the heat exchanger is

$$\Delta P = - \left( \frac{w}{A_R} \right)^2 \left[ \frac{K_{SI}}{2\rho_c} \left( \frac{A_R}{A_{SI}} \right)^2 + \frac{K_{HX}}{A_{HX}} \left( \frac{A_R}{A_{HX}} \right)^2 \right] \quad (\text{A5.4-3})$$

where

$A_{SI}$ =stack inlet cross-sectional area

$A_R$ =riser cross-sectional area

$A_{HX}$ =flow area at heat exchanger

$K_{SI}$ =stack inlet loss coefficient

$K_{HX}$ =heat exchanger loss coefficient

$\rho_c$ =inlet air density

The gravity and acceleration terms have been neglected.

Similarly, the pressure change from the start of the riser to the stack outlet is

$$\Delta P = - \left( \frac{w}{A_R} \right)^2 \frac{(K_{SO} + K_R)}{2\rho_h} - \rho_h g \ell \quad (\text{A5.4-4})$$

where

$K_{SO}$ =stack outlet loss coefficient

$K_R$ =riser loss coefficient

$\rho_h$ =riser air density

$\ell$  =riser length

The pressure change from the stack outlet through the outside air back to the stack inlet is approximately

$$\Delta P = \rho_c g \ell \quad (\text{A5.4-5})$$

The above three pressure changes, Eqs. A5.4-3 through A5.4-5, must sum to zero since they are taken around a closed circuit. Solving for the air flowrate yields

$$w^2 = \frac{(\rho_c - \rho_h) g \ell A_R^2}{\frac{K_{SI}}{2\rho_c} \left( \frac{A_R}{A_{SI}} \right)^2 + \frac{K_{HX}}{2\rho_c} \left( \frac{A_R}{A_{HX}} \right)^2 + \frac{(K_{SO} + K_R)}{2\rho_h}} \quad (\text{A5.4-6})$$



## APPENDIX 5.5: INTERPOLATION FROM USER-SUPPLIED TABLES

A number of user-supplied tables are used in the SAS4A/SASSYS-1 codes. These tables are of the form  $Y(i)$  as a function of  $X(i)$ . The default option is to use linear interpolation between points in the table to obtain  $Y$  as a function of  $X$ . If  $X$  is between  $X(J)$  and  $X(J+1)$ , then linear interpolation gives

$$Y(X) = Y(J) + Z[Y(J+1) - Y(J)] / [X(J+1) - X(J)] \quad (\text{A5.5-1})$$

where

$$Z = X - X(J) \quad (\text{A5.5-2})$$

Linear interpolation has the advantages that it is simple to do and it never exhibits the wild behavior and spurious oscillations that higher order interpolation schemes can give. On the other hand, linear interpolation has some disadvantages. One disadvantage is that if only a few widely spaced points are used for a table, linear interpolation may not be very accurate between points. Another disadvantage is that linear interpolation gives slope discontinuities at the table points. These slope discontinuities can cause corresponding slope discontinuities in plots of powers, temperatures, and flow rates calculated by the codes. Sometimes these slope discontinuities correspond to real physical events, such as scrambling the control rods or tripping the pumps; but usually they are numerical artifacts of the interpolation scheme. In order to address this problem, alternative interpolation schemes are available for some of the tables used by the code. Table 5.5-1 lists the interpolation options.

For the third order fit,

$$Y(X) = C_{1J} + C_{2J}Z + C_{3J}Z^2 + C_{4J}Z^3 \quad (\text{A5.5-3})$$

where the coefficients  $C$  are picked such that:

- a)  $Y$  is continuous at  $X(J)$  and  $X(J+1)$ .
- b) The slope of  $Y$  is continuous at internal points.
- c) The value of  $F$  is minimized, where

$$F = \int_{X_{\min}}^{X_{\max}} \left( \frac{d^2 y}{dx^2} \right)^2 dx \quad (\text{A5.5-4})$$

Table A5.5-1. Table Interpolation Options for a Table of  $Y(i)$  vs.  $X(i)$ , as Determined by the User-Specified Parameter IFIT

$IFIT(K) = 0$	Use linear fits
1	3rd order fits
2	3rd order fits with slope discontinuities at points where $X(i+1) - X(i) < .001$
3	Linear fit to $\log(Y)$
4	3rd order fit to $\log(Y)$
5	3rd order fit to $\log(Y)$ with slope discontinuities
$K = 1$	Power vs time or user specified reactivity vs time
2	PRIMAR-1 pump head vs time or channel flow vs time
3	PRIMAR-1 inlet temperature vs time
4	PRIMAR-4 pump head, motor torque, or pump speed vs time

Note: For all other tables, linear interpolation is used.

In this equation  $X_{min}$  and  $X_{max}$  are the first and last points in the table if  $IFIT = 1$  or  $4$ . If  $IFIT = 2$  or  $5$ , then the table is broken up into ranges, with the boundaries between ranges occurring at the points where the slope discontinuities occur. In this case, the fitting is done separately for each range; and  $X_{min}$  and  $X_{max}$  are the points at the ends of the range. There are some special cases or exceptions:

1. For all fitting options, if  $X$  is less than the first entry in the table, then  $Y$  is set equal to the first value of  $Y$  in the table. Also, if  $X$  is greater than the last entry in the table, then the last value of  $Y$  is used.
2. For  $IFIT = 2$  or  $5$ , in an interval where a slope discontinuity occurs ( $X(i+1) - X(i) < .001$ ) linear interpolation is used from  $X(i)$  to  $X(i+1)$ .
3. If there are only two points in a range, then linear interpolation is used.
4. For  $IFIT = 1, 2, 4$  or  $5$ , if there are only 3 points in a range, then a parabola is fitted to the 3 points to determine the values of  $C$  in the range. Third order fits are only used if there are at least 4 points in a range.

Care should be taken when using third order fits, since they can result in spurious wiggles. Fig. A5.5-1 shows the results of a third order fit to the power level in the SHRT-17 test in EBR-II. In this case, the third order fit to the power produces wiggles, whereas the third order fit to the logarithm of the power produces a smooth curve.

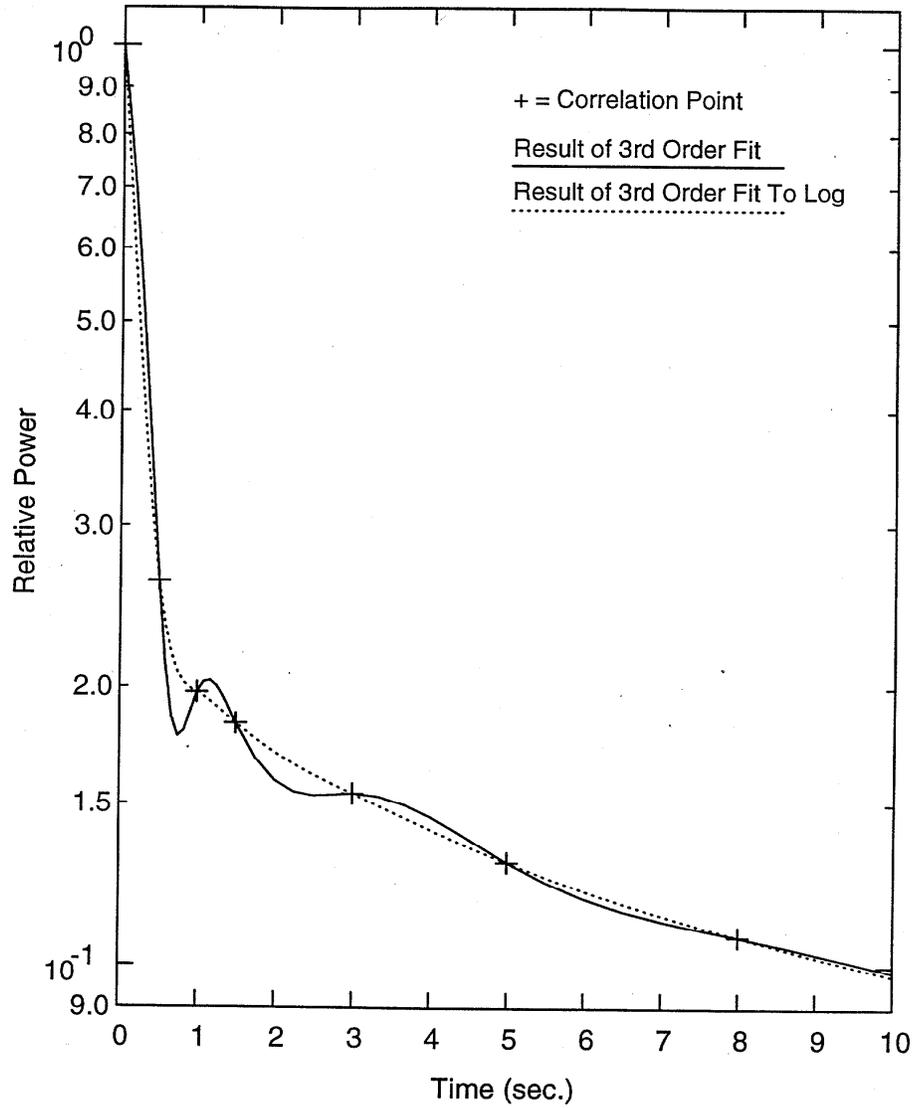


Figure A5.5-1. Fit to SHRT-17 Normalized Total Power, Early Times



## APPENDIX 5.6: OPTIONAL EULERIAN SOLUTION FOR PIPE TEMPERATURES

As mentioned in Section 5.4.1, a Eulerian calculation can be used to speed up the pipe temperature calculation if the coolant moves more than two nodes in a time step. This Eulerian speed-up has only been implemented for flow in the nominal direction; if flow reversal has occurred in a pipe, then the Eulerian calculation is not used.

For the Eulerian calculation, Eq. 5.4-1 for the coolant is replaced by

$$\rho_c c_c A_c \frac{\partial T_c}{\partial t} + w c_c \frac{\partial T_c}{\partial z} = P_{er} h_{wc} (T_w - T_c) \quad (\text{A5.6-1})$$

Eq. 5.4-2 is still used for the wall. Finite differencing of Eq. A5.6-1 gives

$$\begin{aligned} & \rho_c c_c A_c \left[ \frac{T_{c6j} + T_{c6j+1} - T_{c5j} - T_{c5j+1}}{2\delta t} \right] \\ & + \frac{w c_c}{\Delta t} \left[ \theta_1 (T_{c5j+1} - T_{c5j}) + \theta_2 (T_{c6j+1} - T_{c6j}) \right] = \\ & P_{er} h_{wc} \left\{ \theta_1 \left[ T_{w5j} - \frac{T_{c5j} + T_{c5j+1}}{2} \right] \right. \\ & \left. + \theta_2 \left[ T_{w6j} - \frac{T_{c6j} + T_{c6j+1}}{2} \right] \right\} \end{aligned} \quad (\text{A5.6-2})$$

Similarly, finite differencing of Eq. 5.4-2 gives

$$\begin{aligned} \frac{M_w C_w}{\delta t} (T_{w6j} - T_{w5j}) = P_{er} h_{wc} & \left[ -\theta_1 \left( \frac{T_{c5j} + T_{c5j+1}}{2} - T_{w5j} \right) \right. \\ & \left. + \theta_2 \left( \frac{T_{c6j} + T_{c6j+1}}{2} - T_{w6j} \right) \right] \\ & + (hA)_{snk} (T_{snk} - \theta_1 T_{w5j} - \theta_2 T_{w6j}) \end{aligned} \quad (\text{A5.6-3})$$

Eq. A5.6-3 can be rewritten as

$$T_{w6j} = B_{w0j} + B_{w1j} (T_{c6j} + T_{c6j+1}) \quad (\text{A5.6-4})$$

where

$$B_{w1j} = \frac{P_{er} h_{wc} \theta_2 \delta t}{2 d_w} \quad (\text{A5.6-5})$$

$$d_w = M_w C_w + \theta_2 \delta t [P_{er} h_{wc} + (hA)_{snk}] \quad (\text{A5.6-6})$$

and

$$B_{w0j} = - \left\{ M_w C_w T_{w5j} + P_{er} h_{wc} \theta_1 \delta t \left[ \frac{T_{c5j} + T_{c5j+1}}{2} - T_{w5j} \right] + (hA)_{snk} \delta t (T_{snk} - \theta_1 T_{w5j}) \right\} / d_w \quad (\text{A5.6-7})$$

Similarly, Eq. A5.6-2 can be rewritten as

$$T_{c6j+1} = B_{c0j} + B_{c1j} T_{c6j} + B_{c2j} T_{w6j} \quad (\text{A5.6-8})$$

where

$$B_{c0j} = \left\{ \Delta z \rho_c c_c A_c + (T_{c5j} + T_{c5j+1}) + 2\theta_1 \delta t w c_c (T_{c5j} - T_{c5j+1}) + 2\theta_1 \delta t P_{er} h_{wc} \left[ T_{w5j} - \frac{T_{c5j} + T_{c5j+1}}{2} \right] \right\} / d_c \quad (\text{A5.6-9})$$

$$d_c = \rho_c c_c A_c \Delta z + 2\theta_2 dt \left( w c_c \frac{P_{er} h_{wc}}{2} \Delta z \right) \quad (\text{A5.6-10})$$

$$B_{c1j} = - \left\{ \rho_c c_c A_c \Delta z + 2\theta_2 \delta t \left( \frac{P_{er} h_{wc}}{2} \Delta z - w c_c \right) \right\} / d_c \quad (\text{A5.6-11})$$

and

$$B_{c2j} = \frac{2\theta_2 \delta t \Delta z P_{er} h_{wc}}{d_c} \quad (\text{A5.6-12})$$

Eqs. A5.6-4 and A5.6-8 can be combined to give

$$T_{c6j+1} = B_{cc0j} + B_{cc1j} T_{c6j} \quad (\text{A5.6-13})$$

where

$$B_{cc0j} = \frac{B_{c0j} + B_{c2j} B_{w0j}}{1 - B_{c2j} B_{w1j}} \quad (\text{A5.6-14})$$

and

$$B_{cc1j} = \frac{B_{c1j} + B_{c2j} B_{w1j}}{1 - B_{c2j} B_{w1j}} \quad (\text{A5.6-15})$$

Note that  $B_{cc0j}$  and  $B_{cc1j}$  can be calculated before the temperatures at the end of the sub-interval are known.

The pipe inlet temperature at the end of the step is used to set the first coolant temperature,  $T_{c61}$ . The code marches along the pipe, using Eq. A5.6-13 to calculate  $T_{c6j+1}$  after  $T_{c6j}$  has been calculated. After the coolant temperatures have been calculated, Eq. A5.6-4 is used to calculate the wall temperatures.

