**ANALOG-HYBRID DYNAMIC SIMULATION
OF THE FFTF REACTOR
AND HEAT TRANSPORT SYSTEM**

C. D. FLOWERS
AND
L. H. GERHARDSTEIN

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3000 STEVENS DRIVE P. O. BOX 999, RICHLAND, WASHINGTON 99352

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Reactor Technology

ANALOG-HYBRID DYNAMIC SIMULATION
OF THE FFTF REACTOR
AND HEAT TRANSPORT SYSTEM

By

C. D. Flowers

Reactor and Systems Analysis Section
FFTF Project Department

And

L. H. Gerhardstein

Computer and Control Section
Applied Physics and Electronics Department

April 1968

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RICHLAND, WASHINGTON

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C. D. Flowers and L. H. Gerhardstein

ABSTRACT

A mathematical model has been developed for investigating the dynamic response and control characteristics of the Fast Flux Test Facility (FFTF) Reactor and heat removal systems. The model was developed for solution on the analog-hybrid computer facility at Pacific Northwest Laboratory (PNL). The mathematical modeling techniques used to simulate each system component are discussed in detail, and sample transient results are presented. Prospective uses of the simulation are also discussed.

One complete heat removal circuit (out of four) is simulated, and the total heat is accounted for by a multiplying factor. The reactor simulation includes point kinetics with temperature feedbacks, single-node representation of fuel, coolant, leakage, and reflector regions, and inlet and outlet mixing plenums. Multinode heat exchanger models are employed to represent the intermediate and air-dump heat exchangers. The system model also includes simulation of coolant transport time lags, sodium pumps, motors, electromagnetic clutches, air fans, and sodium and air hydraulics.

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0. INTRODUCTION

The Fast Flux Test Facility will be the main facility for testing fuels and structural materials in the USAEC's Liquid Metal Fast Breeder Reactor (LMFBR) Program. The dynamic behavior of the reactor and its associated heat removal systems is critical in the total design of the plant. The system must be controllable and it must be designed to survive the expected system transients. To analyze the system dynamic characteristics, a dynamic simulation model of the plant has been developed for solution on the analog-hybrid computer at Pacific Northwest Laboratory.⁽¹⁾ This report provides a detailed description of the mathematical models used to simulate each system component. The model is basically an expansion of the simplified model described in Reference 2 and accounts for:

- Reactor Kinetics and Feedbacks
- Heat Transfer
- Sodium and Air Hydraulics
- Coolant Transport Time Lags
- Rotating Equipment.

Typical transient results from the simulation are also given. Actual design data are not included in the report since:

1) the design is only conceptual at this time, and 2) the purpose of the report is to describe the mathematical modeling methods.

The model will be used to provide transient information in the following areas:

- Component Design
- Safety System Design
- Process Control and Data Handling Design
- Operator Training.

Temperature and flow transient information will be provided for the design of system components. Such information is required, for example, in heat exchanger and reactor vessel procurement specifications. Safety studies will include evaluation of system response to the credible range of disturbances, and determination of instrument trip limits needed to limit system component transients. Process control studies will aid in demonstrating and choosing the best control scheme for the system. The model may also be utilized to analyze and demonstrate data handling systems prior to installation in the facility. Finally, the simulation will be used for a comprehensive operator training program.

1. SUMMARY

A mathematical model of the FFTF Reactor and heat removal system has been developed to study the plant dynamic characteristics. The model is programmed on the analog-hybrid facility at PNL. The purpose of this report is to provide a detailed description of the mathematical modeling techniques used to simulate each system component. Sample transient results are also included.

Due to the limitation of computing equipment, only one complete heat removal circuit (out of four) has been simulated. The total heat is accounted for by applying the appropriate multiplying factor. One heat removal circuit consists of a primary sodium loop, a secondary sodium loop, and a tertiary air heat dump system. The total simulation includes models of: 1) reactor, 2) multinode intermediate heat exchanger (IHx), 3) multinode dump heat exchanger (DHx), 4) primary, secondary and tertiary coolant hydraulics, 5) coolant transport time lags, 6) sodium pumps, motors and clutches, and 7) air blowers. The complete system model has been programmed

and debugged, and is available for transient studies. The addition of a second heat removal circuit to the simulation is planned for investigating multiloop interaction and control.

2. PLANT DESCRIPTION

The FFTF heat removal system consists of four independent, parallel cooling circuits that have a common flow path through the reactor vessel. Each circuit consists of three major systems as shown in Figure 2.1.1.

- A radioactive primary sodium system.
- A secondary sodium system.
- A tertiary air-cooled heat dump system.

The plant is designed to allow full-load reactor operation using any three of the four cooling circuits.

2.1 Reactor Description

Figure 2.1.2 shows the internals of the FFTF split core arrangement. Figure 2.1.3 shows a cross-sectional view of the core. The primary sodium coolant enters the inlet nozzles and flows through downcomers to a high-pressure inlet plenum. There is a three-way flow split from the inlet plenum:

- Main sodium flow through the reactor tubes (containing fuel subassemblies, open loops, and control elements) to remove heat generated by the fuel, axial reflectors, and control elements
- Leakage flow around the outside of the subassemblies (including the flow up through center of core around the closed loops)
- Radial reflector coolant flow that removes heat generated in the radial reflector regions.

The three flows dump into a common mixing plenum at the subassembly outlet. From the outlet mixing plenum, the sodium flows to the four parallel heat removal circuits.

FIGURE 2.1.1. FFTF Heat Removal System Arrangement--One Circuit, Typical of Four (Inlet and outlet coolant temperatures are labeled for the major components)

FFTF SPLIT CORE ARRANGEMENT PIPED INLET

Neg 0674071-1

FIGURE 2.1.2. Reactor Vessel Internals

Neg 0673276-2

(Simulation only accounts for driver fuel, open loops, closed loops, and reflectors.)

FIGURE 2.1.3. Reactor Core Cross Section (Conceptual Design)

5

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The primary sodium coolant receives heat from the driver fuel elements, open loops, and axial and radial reflector material. The closed loops in the center of the core have separate heat removal systems and, for the present design, are assumed to add negligible heat to the primary coolant. The core design is extremely flexible at this stage of conceptual design, thus only ranges can be given for the various core parameters. The core is expected to generate 300 to 400 MW with a core delta-T of 300 to 400 °F. Core volume can range from 800 to 1100 liters with a height of ~3 ft. The fuel material for the initial core is a mixed plutonium-uranium oxide and is assembled on a triangular pitch in pin bundles. The core zone is surrounded by axial and radial reflector regions as shown in Figures 2.1.2 and 2.1.3.

2.2 HEAT REMOVAL CIRCUIT DESCRIPTION

Each of the four heat removal circuits is identical and each operates independently. The primary loop of each circuit removes heat from the reactor core and vessel internals. The heat is transported to the secondary sodium system via an intermediate heat exchanger (IHX). The IHX in each circuit is a vertical, single-pass, shell and tube, counterflow heat exchanger, with primary sodium flow on the shell side. A single stage, free surface, centrifugal pump drives the primary coolant. Variable pump speed is provided and is presently assumed to be included by eddy current drive couplings. This method of pump speed control is not final in the design and is subject to change. Auxiliary pony motors are provided for pump operation at ~10 to 15% of rated pump speed. An elevation difference between the reactor core and the IHX provides natural circulation flow in the primary loop to remove decay heat following an emergency shutdown.

The secondary sodium loop receives heat from the tube side of the IHX. This heat is removed by the air heat dump system (DHX) in each circuit. The secondary sodium pumps are similar to the primary pumps, and are located in the cold leg of each circuit. Variable pump speed control on the secondary pumps is also provided by eddy current clutch couplings. Sodium expansion tanks are included in the secondary cold legs of each circuit to allow for sodium thermal expansion and to provide the required pump suction head. A natural circulation head is also provided in the secondary loops by a height difference between the IHX and DHX.

Forced draft air heat dumps remove heat from the secondary sodium loops. The heat dumps for each of the four heat removal circuits are divided into four modules, each with its own heat exchanger coil and blower. The heat exchanger sections consist of multipass finned tubes, containing secondary sodium, that are cooled by cross-flowing air. The DHX is designed so that each module can be operated independently of the other parallel modules. Air flow is provided by electric motor driven blowers of either axial flow or centrifugal type. The blowers are equipped with variable-position inlet vanes for flow control. Blower discharge dampers are provided to allow manual control during low load conditions. Heated air is exhausted from the DHX modules through insulated stacks that are equipped with low-leakage dampers for blockage of air flow. Each of the four heat removal circuits is designed so that three circuits will handle the entire maximum heat load.

3. MATHEMATICAL MODELS OF THE FFTF SUBSYSTEMS

The mathematical bases for the models used in the FFTF dynamics simulation are presented in this section. Where detailed analysis is necessary, refer to the appendices. The general assumptions made in developing the models are presented in each subsection discussion.

3.1 REACTOR MODEL

The reactor model simulates the heat generated and transmitted within the reactor vessel. The model is nodalized to simulate the regions where heat is stored or generated and where primary sodium coolant flows. Figure 3.1.1, p. 14, represents the nodalized reactor model.

3.1.1. Heat Generation

The total power (PL_T) within the reactor vessel originates in the core zone and consists of neutron (both prompt and delayed) power (PL_N) and fission product decay power (PL_F):

$$PL_T = PL_N + PL_F \quad .$$

The total reactor power results from heat generation in the following regions within the reactor vessel:

- Driver Fuel and Open Loops
- Closed Loops
- Radial Reflectors
- Axial Reflectors.

The driver fuel elements and the open loops are combined into an equivalent number of average driver fuel elements (N_{ER}). Each open loop is assumed to produce a given fraction of the heat generated in an average driver fuel element (e.g., 12 open loops are assumed to be equivalent to six driver elements in the present simulation).

The closed loops are in the core zone and their total heat generation consists of neutron and fission products. The closed loops, however, have their own heat removal systems and are assumed to be thermally insulated from the primary sodium coolant. Therefore, the closed-loop heat generation is subtracted from the total heat generation (PL_T) when determining driver fuel heat generation. The closed-loop power (PL_{CR}) is unknown at this time and is assumed to be a fraction (γ_{CR}) of the total power:

$$PL_{CR} = \gamma_{CR} PL_T \quad . \quad (3.1-1)$$

The fission product heat is confined within the active core zone. Therefore, the reflector regions are heated only by the bombardment of neutrons that escape from the active core zone. The radial reflector heat generation (PL_{RR}) is assumed to be a fraction (γ_{RR}) of the neutron power level:

$$PL_{RR} = \gamma_{RR} PL_N \quad . \quad (3.1-2)$$

The axial reflector heat is included with the driver fuel heat generation since the axial reflectors are not modeled as separate nodes.

The heat generated in the equivalent driver fuel elements (open loops plus drivers) and axial reflectors is the remainder of the total heat. Heat generated in the driver fuel and open loops (PL_{FR}) is the total power minus the closed-loop power and the radial reflector power:

$$\begin{aligned} PL_{FR} &= PL_T - PL_{CR} - PL_{RR} \\ &= (1 - \gamma_{CR}) PL_F + (1 - \gamma_{CR} - \gamma_{RR}) PL_N \quad . \quad (3.1-3) \end{aligned}$$

3.1.2. Reactor Kinetics - Neutron Power

The neutron power is calculated from the standard space independent reactor kinetics equations⁽³⁾ with the assumption that

$$\delta k = \frac{k_{\text{eff}} - 1}{k_{\text{eff}}} \approx k_{\text{eff}} - 1$$

Therefore,

$$\frac{d(PL_N)}{dt} = \frac{\delta k - \beta}{\ell^*} (PL_N) + \sum_{i=1}^3 \lambda_i C_i \quad (3.1-4)$$

where

$$\frac{dC_i}{dt} = \frac{\beta_i}{\ell^*} (PL_N) - \lambda_i C_i \quad (3.1-5)$$

Algebraic manipulation of these equations yields an alternate form that is better suited for analog simulation of reactors with small neutron lifetimes, ($\ell^* = 3.5 \times 10^{-7}$ sec for the FTR). The alternate equations that have been simulated are

$$PL_N + \frac{\ell^*}{\beta} \frac{d(PL_N)}{dt} = \frac{\delta k}{\beta} (PL_N) + \sum_{i=1}^3 \frac{\ell^* \lambda_i C_i}{\beta} \quad , \quad (3.1-6)$$

$$\frac{d \left(\frac{\ell^* \lambda_i C_i}{\beta_i} \right)}{dt} = \lambda_i (PL_N) - \lambda_i \left(\frac{\ell^* \lambda_i C_i}{\beta_i} \right) \quad , \quad (3.1-7)$$

$$\frac{\ell^* \lambda_i C_i}{\beta} = \frac{\beta_i}{\beta} \left(\frac{\ell^* \lambda_i C_i}{\beta_i} \right) \quad (3.1-8)$$

Three reduced delayed neutron groups are used. The delayed group constants are calculated with the computer code prepared by Albrecht.⁽⁴⁾ The code uses six equivalent groups with either the first or sixth group isolated and the remaining five groups reduced to two groups. For short-time response

to fast transients, the short-lifetime group is isolated as one of the three groups, and for long-time, slow transients, the long-lifetime group is isolated.

3.1.3. Reactivity Feedbacks

The following terms are included in the reactivity feedback model for the reactor:

1. Sodium temperature feedback, δk_C
2. Fuel Doppler feedback, δk_D
3. Structural feedbacks,
 - a. Tube deflection due to radial core expansion, δk_E
 - b. Tube deflection due to fuel shroud tube bowing, δk_B
4. Miscellaneous control inputs
 - a. Scram reactivity, δk_S
 - b. Control reactivity, δk_{CR} , for power level control
 - c. Reactor state reactivity—for steady-state δk balance, δk_z .

The sodium temperature feedback coefficient is small and negative for the FTR core, and the reactivity feedback is a function of the average shroud tube coolant temperature in the core zone:

$$\delta k_C = \alpha_C T_{ACTR} \quad . \quad (3.1-9)$$

The Doppler feedback coefficient is also negative and is a function of the average fuel temperature. The Doppler feedback coefficient is expressed as follows:

$$\alpha_D = T \frac{dk}{dT} \quad . \quad (3.1-10)$$

Integrating this expression between a reference fuel temperature (T_{AFRZ}), in degrees Fahrenheit, and the average calculated fuel temperature (T_{AFR}), in degrees Fahrenheit, we obtain the Doppler reactivity feedback term

$$\delta k_D = \alpha_D \left\{ \ln[(T_{AFR} + 460)/(T_{AFRZ} + 460)] \right\} \quad . \quad (3.1-11)$$

Any reference fuel temperature (T_{AFRZ}) can be used in this term as long as the reactivity balance at steady-state power yields a zero total δk . The value for T_{AFRZ} normally used is the steady-state initial condition value of average fuel temperature, T_{AFR} .

The structural feedback due to radial core expansion provides a negative feedback from the temperature of the shroud tubes and spacers. The feedback is assumed to be a function of the average shroud coolant temperature. This appears to be a reasonable assumption since the shroud tube material temperature will follow the tube coolant temperature quite closely. The equation for radial core expansion feedback is

$$\delta k_E = \alpha_E (T_{ACTR} - T_{ACTRZ}) \quad . \quad (3.1-12)$$

As with the Doppler feedback equation, any reference average coolant temperature (T_{ACTRZ}) can be used as long as the total reactivity balance is correct.

The shroud tube bowing is assumed to contribute a positive reactivity feedback with increasing core ΔT . This feedback is caused by the inward (towards core centerline) bowing of the shroud tubes caused by a transverse temperature gradient across the tube diameter. This temperature gradient is proportional to the core radial flux profile and can be approximated mathematically as a function of the axial core coolant ΔT ($T_{CTR} - T_{CIP}$). The reactivity feedback due to bowing is

$$\delta k_B = \alpha_B (T_{CTR} - T_{CIP}) \quad . \quad (3.1-13)$$

The miscellaneous control reactivities are either forced onto the system by controllers or by the systems engineer to study a transient or to achieve a steady-state reactivity balance. These terms, therefore, have no fixed mathematical representation at this time.

The total reactivity (δk) utilized in Equation (3.1-4) is the sum of the individual reactivity feedbacks:

$$\delta k = \delta k_C + \delta k_D + \delta k_E + \delta k_B + \delta k_{CR} + \delta k_S + \delta k_Z \quad . \quad (3.1-14)$$

3.1.4. Fission Product Power

A portion of the total heat generated in the core zone is due to decay of fission products. At full power steady-state operating conditions the fission products contribute only about 5% of the total core power. However, in less than 1 min after a reactor scram, the fission product decay heat becomes the major portion of the total heat generated. Decay heat generation curves⁽⁵⁾ have been used to obtain a model representing the fission product decay heat (following shutdown) as a fraction of full power, steady-state neutron power level.

This relationship is shown in Figure 3.1.2 and applies to a plutonium fueled fast reactor (100 day exposure) and includes the power generation due to beta and gamma decay of all fission product isotopes. The data used to obtain this curve are preliminary and are subject to future revision.

To simulate the fission product decay, three groups are assumed to approximate the decay curve, represented by the asymptotes shown in Figure 3.1.2. The equations for the three fission product decay groups are

$$\frac{d(PL_{F1})}{dt} = \lambda_{F1} (\beta_{F1} PL_N - PL_{F1}) \quad , \quad (3.1-15)$$

$$\frac{d(PL_{F2})}{dt} = \lambda_{F2} (\beta_{F2} PL_N - PL_{F2}) \quad , \quad (3.1-16)$$

$$\frac{d(PL_{F3})}{dt} = \lambda_{F3} (\beta_{F3} PL_N - PL_{F3}) \quad . \quad (3.1-17)$$

FIGURE 3.1.1. Nodalized Reactor Model

FIGURE 3.1.2. Fission Product Decay Heat Generation Following Reactor Scram

The total fission product heat is then the sum of three groups:

$$PL_F = PL_{F1} + PL_{F2} + PL_{F3} \quad . \quad (3.1-18)$$

3.1.5. Reactor Thermal Model Description

The block diagram for the reactor thermal model is shown in Figure 3.1.1. Primary coolant enters the inlet plenum from the reactor inlet nozzle and is distributed to the subassembly shroud tubes, leakage region, and radial reflectors. A node for the stainless steel lower support structure is included adjacent to the inlet plenum to account for heat transfer between the inlet plenum, where perfect mixing is assumed, and the lower support structure. Gamma heating in the lower support structure is assumed to be negligible. The liquid sodium coolant entering the tubes flows upward through the core region, removes heat from the fuel and axial reflectors, and exits into the outlet plenum. Liquid sodium flowing in the radial reflectors removes the heat generated in the radial reflector material and exits into the outlet plenum. Heat transfer is included between the adjacent regions in the reactor model (small arrows in Figure 3.1.1).

The radial reflector material heat capacity is accounted for by converting the reflector material volume to an equivalent sodium volume and lumping it with the radial reflector coolant. The radial reflector heat generation is included in the radial reflector coolant equation. The lower axial reflector material volume and the sodium volume in the shroud tubes up to the bottom of the core zone are lumped with the inlet plenum volume; likewise, the upper axial reflector material, fuel fission gas plenum volume, and sodium volume (inside of shroud tube) from the core zone outlet to the shroud tube outlet are lumped with the outlet plenum volume. Perfect mixing is assumed in the outlet plenum and heat

transfer to a metal node is included. The metal node includes the volume of the tubes that extend up through the outlet plenum and the volume of all other steel in the outlet plenum region.

3.1.6. Reactor Thermal Model Equations

Linear coolant temperature distributions are assumed through each of the three parallel flow paths, and the rate of change of the nodal outlet temperature is assumed to be equal to the rate of change of the average nodal temperature. The flow rates in the tubes, leakage, and reflector are assumed to split from the total flow in the inlet plenum as a constant percentage of total reactor flow (W_R).

$$\text{Tube Flow:} \quad W_{TR} = a_{TR} W_R \quad , \quad (3.1-19)$$

$$\text{Leakage Flow:} \quad W_{LR} = a_{LR} W_R \quad , \quad (3.1-20)$$

$$\text{Radial Reflector Flow:} \quad W_{RR} = a_{RR} W_R \quad ; \quad (3.1-21)$$

$$a_{TR} + a_{LR} + a_{RR} = 1 \quad . \quad (3.1-22)$$

Thermal properties (density and specific heat) are assumed to be constant, and axial heat conduction in the coolant is neglected.

The mixing equation for the reactor inlet plenum temperature (T_{CIP}) is (perfect mixing assumed):

$$\frac{dT_{CIP}}{dt} = \frac{W_R C_{CIP}}{(\rho VC)_{IP}} (T_{CIR} - T_{CIP}) - \frac{(UA)_{LSR}}{(\rho VC)_{IP}} (T_{CIP} - T_{LSR}) \quad (3.1-23)$$

where T_{CIR} = reactor inlet temperature, °F

$(\rho VC)_{IP}$ = density x volume x specific heat—(inlet plenum sodium-assumed constant)

$(UA)_{LSR}$ = overall heat transfer conductance, metal-sodium.

The equation for the lower support structure temperature (T_{LSR}) is:

$$\frac{dT_{LSR}}{dt} = \frac{(UA)_{LSR}}{(\rho VC)_{LSR}} (T_{CIP} - T_{LSR}) \quad (3.1-24)$$

where $(\rho VC)_{LSR}$ = density x volume x specific heat—(lower support metal-assumed constant).

The heat transfer conductance between the metal and sodium, $(UA)_{LSR}$, is assumed to be constant with temperature and includes the metal thermal conductivity and surface area. The heat transfer resistance of the sodium film at the metal surface is neglected. This is a reasonable assumption since liquid sodium has excellent heat transfer properties.

The equations for the average fuel temperature and the subassembly shroud tube coolant temperature are written for an average equivalent driver fuel subassembly. Therefore, the following volumes and areas are per equivalent driver element, per unit length: V'_{FR} , A'_R , V'_{CTR} , and A'_{TR} . The leakage coolant temperature and radial reflector coolant temperature equations are written for the total leakage and reflector volumes. Therefore, V_{CLR} , A_{RR} , and V_{RR} are total volumes and areas. The average equivalent driver fuel temperature (T_{AFR}) is obtained from:

$$\frac{dT_{AFR}}{dt} = \frac{b_1 PL_{FR}}{N_{ER}(\rho V'C)_{FR} L_{FR}} - \frac{(UA')_R}{(\rho V'C)_{FR}} (T_{AFR} - T_{ACTR}) \quad (3.1-25)$$

where $(UA')_R$ = overall heat transfer conductance/unit length—(fuel to coolant), discussed below

N_{ER} = number of equivalent driver fuel elements

$T_{ACTR} = \frac{T_{CTR} + T_{CIP}}{2}$ = average coolant temperature

L_{FR} = fuel length, ft

$(\rho V'C)_{FR}$ = density x (fuel volume/unit length) x specific heat—(fuel region-assumed constant)

b_1 = conversion constant, Btu/sec-MW.

The heat transfer conductance (per unit length) between the fuel and the shroud tube coolant, $(UA')_R$, includes a fuel resistance, a clad resistance and a clad surface resistance. The fuel to coolant heat transfer coefficient is simulated as a function of fuel temperature as explained in Appendix A.

The shroud tube coolant outlet temperature equation includes heat removed by the coolant, heat transfer from the fuel, and heat transfer to the leakage. The clad, fuel pin spacers, and subassembly shroud tube wall are all lumped with the tube coolant. This is accomplished by calculating equivalent density and specific heat terms on a volume weighted basis as follows:

$$(\rho C)^*_{\text{equivalent}} = (\rho C)_{\text{coolant}} (VF)_{\text{coolant}} + (\rho C)_{\text{metal}} (VF)_{\text{metal}}$$

where $(VF)_{\text{coolant}}$ = coolant volume fraction

$(VF)_{\text{metal}}$ = metal volume fraction (includes clad, spacers, and tube wall).

Note: $(\rho C)^*$ denotes lumped coolant and metal properties.

The tube coolant outlet temperature (T_{CTR}) is calculated by the following equation

$$\begin{aligned} \frac{dT_{CTR}}{dt} = & - \frac{W_{TR} C_{CTR}}{N_{ER} L_{FR} (\rho V' C)^*_{CTR}} (T_{CTR} - T_{CIP}) \\ & + \frac{(UA')_R}{(\rho V' C)^*_{CTR}} (T_{AFR} - T_{ACTR}) \\ & - a_{TLR} \frac{(UA')_{TR}}{(\rho V' C)^*_{CTR}} \left(\frac{T_{CTR} - T_{CLR}}{2} \right) \end{aligned} \quad (3.1-26)$$

where $(UA')_{TR}$ = overall heat transfer conductance/unit length—(tube coolant to leakage coolant), discussed below

T_{CLR} = leakage region coolant outlet temperature.

The last term accounts for heat transferred from the shroud tube coolant, through the tube wall, to the leakage coolant. The temperature difference for heat transfer results from taking the difference between the average tube and leakage coolant temperatures. Since the adjacent coolant regions have the same inlet temperature, the driving force between the tube coolant and leakage coolant becomes one-half the difference between the node outlet temperatures. A length ratio (a_{TLR}) accounts for the portion of the tube (bottom of core to tube outlet) where heat transfer from the shroud tube coolant to the leakage coolant actually occurs:

$$a_{TLR} = \frac{\text{active heat transfer length}}{\text{active core length}} = \frac{L_{FR} + L_{UR}}{L_{FR}} \quad (3.1-27)$$

The shroud tube coolant to leakage coolant heat transfer conductance, $(UA')_{TR}$, is assumed to be a constant with temperature. The term consists of inside and outside film coefficients, and the tube wall thermal conductivity.

The bypass leakage outlet coolant temperature (T_{CLR}) equation includes heat removed by the leakage coolant and heat transfer from the shroud tube coolant. Heat transfer between leakage coolant and radial reflector coolant was found to be negligible and, therefore, has not been included.

$$\begin{aligned} \frac{dT_{CLR}}{dt} = & - \frac{W_{LR} C_{CLR}}{(\rho VC)_{CLR}} (T_{CLR} - T_{CIP}) \\ & + a_{TLR} \frac{N_{ER} L_{FR} (UA')_{TR}}{(\rho VC)_{CLR}} \left(\frac{T_{CTR} - T_{CLR}}{2} \right). \end{aligned} \quad (3.1-28)$$

The heat transfer conductance between the tube coolant and leakage coolant is multiplied by the number of equivalent driver elements and the core length since the total leakage volume (V_{CLR}) is used in this region.

The radial reflector coolant outlet temperature equation includes heat removal by the coolant, and heat generation in the reflector material. The reflector material is lumped with the coolant by calculating equivalent $(\rho C)^*$ terms as before.

$$\frac{dT_{CRR}}{dt} = - \frac{W_{RR} C_{CRR}}{(\rho VC)^*_{RR}} (T_{CIP} - T_{CRR}) + \frac{b_1 PL_{RR}}{(\rho VC)^*_{RR}} \quad (3.1-29)$$

The three flows (tube, leakage, and reflector) are combined in the outlet mixing plenum. Constant and equivalent specific heats are assumed for each flow and also for the mixed flow. This is reasonable since the outlet temperatures from each region will not be greatly different. The outlet plenum temperature equation is

$$\begin{aligned} \frac{dT_{COR}}{dt} = & \frac{W_R}{(\rho V)_{OR}} (a_{TR} T_{CTR} + a_{LR} T_{CLR} + a_{RR} T_{CRR} - T_{COR}) \\ & - \frac{(UA)_{USR}}{(\rho VC)_{OR}} (T_{COR} - T_{USR}) \quad (3.1-30) \end{aligned}$$

The equation for the outlet plenum metal is

$$\frac{dT_{USR}}{dt} = \frac{(UA)_{USR}}{(\rho VC)_{USR}} (T_{COR} - T_{USR}) \quad (3.1-31)$$

The outlet plenum coolant to metal heat transfer conductance, $(UA)_{USR}$, is assumed constant and includes a metal thermal conductivity resistance similar to the lower support plate heat transfer coefficient. Resistance due to the surface film conductance is neglected in the heat transfer conductance.

3.2 INTERMEDIATE HEAT EXCHANGER MODEL

The intermediate heat exchanger (IHx) to be used in each FFTF coolant loop will be of the tube-in-shell, counter flow type. A multinode model is used to describe heat transfer from primary to secondary as shown in Figure 3.2.1. After

FIGURE 3.2.1. Basic Configuration and Coordinate System for the IHX Model

studying the transient response of several number of nodes, a four-node approach was selected as a reasonable compromise between accuracy and available analog computing equipment.

Heat transfer is described by the usual partial differential equations, in which temperature is a function of both time and axial position. Axial heat conduction is neglected in both tube wall and coolant, and heat storage in the tube metal is divided equally between the stored heat in primary and secondary sodium. Density and specific heat are assumed constant with temperature over the range of interest in each particular problem. The basic equations for the coolant temperature at each node(i) then appear as follows:

Primary Side

$$(\rho V' C)_{Pi}^* \frac{\partial T_{CPi}}{\partial t} = - W_P C_{CPi} \frac{T_{CPi}}{\partial x} - (UA')(T_{CPi} - T_{CSi}), \quad (3.2-1)$$

Secondary Side

$$(\rho V' C)_{Si}^* \frac{\partial T_{CSi}}{\partial t} = W_S C_{CSi} \frac{\partial T_{CSi}}{\partial x} + (UA')(T_{CPi} - T_{CSi}), \quad (3.2-2)$$

where the $(\rho V' C)^*$ coefficients include tube metal heat storage. The terms V' and A' represent volume and area per unit length.

A finite difference approximation is used to represent the axial temperature gradient $(\partial T / \partial x)$ for each node. Several such approximations were studied by comparing temperature transients for realistic response of the coolant outlet temperatures. The best approximation of the axial temperature distribution was found to be a combination of a "central difference" approximation at each inlet node and "three-point backward" differences at the remaining three nodes. The term "backward" is used in this context to indicate a direction opposite to that of coolant flow. Note that five temperatures are used to describe the heat transfer of four nodes (Figure 3.2.1). The finite difference approximations are as follows:

Central Difference

$$\frac{\partial T_i}{\partial x} = \frac{T_{i+1} - T_{i-1}}{2\Delta L_X} \quad (3.2-3)$$

Three-Point Backward

$$\frac{\partial T_i}{\partial x} = \frac{3T_i - 4T_{i-1} + T_{i-2}}{2\Delta L_X} \quad \text{.....primary} \quad (3.2-4)$$

$$= \frac{-3T_i + 4T_{i+1} - T_{i+2}}{2\Delta L_X} \quad \text{.....secondary} \quad (3.2-5)$$

where $\Delta L_X = L_X/4$ is the length between node points.

The above equations are combined to give the following complete set of expressions for IHX heat transfer. Subscripts are included for clarity:

C = coolant, P = primary, S = secondary, and X = IHX.

$$(\rho V' C)_{PX}^* \frac{dT_{CPXi}}{dt} = \frac{2W_P C_{CPX}}{L_X} (\Delta T)_{CPXi} - (UA')_X (T_{CPXi} - T_{CSXi}) \quad (3.2-6)$$

for $i = 2, 3, 4, 5$, and

$$(\rho V' C)_{SX}^* \frac{dT_{CSXi}}{dt} = -\frac{2W_S C_{CSX}}{L_X} (\Delta T)_{CSXi} + (UA')_X (T_{CPXi} - T_{CSXi}) \quad (3.2-7)$$

for $i = 1, 2, 3, 4$, where

$$(\Delta T)_{CPX2} = -T_{CPX1} + T_{CPX3} \quad (3.2-8)$$

$$(\Delta T)_{CPX3} = 3T_{CPX3} - 4T_{CPX2} + T_{CPX1} \quad (3.2-9)$$

$$(\Delta T)_{CPX4} = 3T_{CPX4} - 4T_{CPX3} + T_{CPX2} \quad (3.2-10)$$

$$(\Delta T)_{CPX5} = 3T_{CPX5} - 4T_{CPX4} + T_{CPX3} \quad (3.2-11)$$

$$(\Delta T)_{CSX1} = -3T_{CSX1} + 4T_{CSX2} - T_{CSX3} \quad (3.2-12)$$

$$(\Delta T)_{CSX2} = -3T_{CSX2} + 4T_{CSX3} - T_{CSX4} \quad (3.2-13)$$

$$(\Delta T)_{CSX3} = -3T_{CSX3} + 4T_{CSX4} - T_{CSX5} \quad (3.2-14)$$

$$(\Delta T)_{CSX4} = -T_{CSX3} + T_{CSX5} \quad (3.2-15)$$

Areas and volumes are easily computed from design data as shown in Appendix B. The heat transfer coefficient is a function of coolant flows and average temperatures, and is computed using the methods of Dwyer⁽⁶⁾ for fluid flow within and around tube bundles. A complete description of the computation of $(UA')_X$ and its simulation is found in Appendix B.

3.3 TRANSPORT DELAY MODEL

Liquid sodium coolant is transported by the primary and secondary coolant loop piping. The time delay in the temperature leaving a section of piping with respect to the inlet temperature is simulated by a variable flow, no-mixing time delay program on the digital portion of the hybrid computer.⁽⁷⁾ The theory is similar to that of the digital transport delays developed by Walker.⁽⁸⁾

Time delays are calculated between the reactor and IHX in the primary loop, and between the IHX and DHX in the secondary loop. The cold leg time delays include the pipe sections on both the upstream and downstream sides of the pumps in primary and secondary loops. The following assumptions were made in simulating the piping time delays.

1. No mixing occurs within the pipe section.
 2. There is no axial heat conduction.
 3. There are no heat gains or losses to the piping.
 4. Coolant density within a piping leg is assumed constant.
- With these assumptions, the pipe section outlet temperatures at time (t) are defined as:

Primary Loop

$$\text{IHX Inlet Temperature} = T_{\text{CPX1}}(t) = T_{\text{COR}}(t - \tau_{\text{CPH}}) \quad (3.3-1)$$

Reactor Vessel Inlet

$$\text{Temperature} = T_{\text{CIR}}(t) = T_{\text{CPX5}}(t - \tau_{\text{CPC}}) \quad (3.3-2)$$

Secondary Loop

$$\text{DHX Inlet Temperature} = T_{\text{CSY1}}(t) = T_{\text{CSX1}}(t - \tau_{\text{CSH}}) \quad (3.3-3)$$

$$\text{IHX Inlet Temperature} = T_{\text{CSX5}}(t) = T_{\text{CSY4}}(t - \tau_{\text{CSC}}) \quad (3.3-4)$$

where T_{COR} = reactor vessel outlet temperature

T_{CPX5} = IHX primary outlet temperature

T_{CSX1} = IHX secondary outlet temperature
 T_{CSY4} = DHX secondary outlet temperature
 τ = time delay for piping section of concern.

For a constant flow rate, the transport delay time (τ) is

$$\tau = \frac{\rho V}{W} \quad (3.3-5)$$

where ρ = coolant density
 V = pipe section volume
 W = mass flow rate.

If the flow rate is varying, the delay time is obtained from the solution of the integral equation

$$\int_{t-\tau}^t W(t) dt = \rho V \quad (3.3-6)$$

The digital time delay program in Reference 7 solves Equation (3.3-6) for each of the four transport delays in the simulation.

3.4 HYDRAULICS MODELS FOR PRIMARY AND SECONDARY LOOPS

The primary and secondary coolant loops consist of stainless steel piping, fittings, and sodium pumps. The hydraulics model for the primary and secondary loops was derived from Newton's second law of motion applied to a viscous fluid and nonsteady flow (see Appendix C). The basic energy equation for the hydraulics is

$$\rho \frac{dW}{dt} = \rho H_p - \int \rho dZ - \Delta P_f \quad (3.4-1)$$

where $\rho \frac{dW}{dt}$ = inertance pressure drop
 W = loop flow rate
 ρH_p = pump pressure
 $\int \rho dZ$ = natural circulation head
 ΔP_f = frictional pressure loss.

The model development is based on the following assumptions:

1. The fluid viscous properties are governed by a smooth pipe relationship.
2. Coolant density and viscosity are assumed to be constant in all sections of the loop being considered. (The maximum error in this assumption for the entire temperature range of interest is $\sim 3\%$.)

A functional diagram of the hydraulics model is shown in Figure 3.4.1.

FIGURE 3.4.1. Primary/Secondary Coolant Loop Hydraulics Model Functional Diagram

The inertance coefficient, ℓ (sec^2/ft^2), is given by the following summation over the loop:

$$\ell = \frac{1}{g_c} \sum \frac{(\Delta L)_i}{A_i} \quad (3.4-2)$$

where g_c is the gravitational acceleration constant, $(\Delta L)_i$, is the length of section i , and A_i is the section area. Calculation of ℓ is explained in Appendix C.

The natural circulation head is approximated by

$$\int \rho dZ = (\Delta Z) \left(\frac{\partial \rho}{\partial T} \right) (T_{AH} - T_{AC}) \quad (3.4-3)$$

where T_{AH} = average hot-leg temperature of loop (linear average)

T_{AC} = average cold-leg temperature of loop (linear average)

ΔZ = elevation difference of active elements

$\frac{\partial \rho}{\partial T}$ = coolant density temperature coefficient (this term is negative - see Appendix C).

The friction pressure loss term of Equation (3.4-1) is given by

$$\Delta P_f = C_f W^{1.8} \quad (3.4-4)$$

(The pressure loss coefficient C_f includes all pressure losses in a loop and is explained in Appendix C.) The driving force for the primary and secondary hydraulics is the head developed by the sodium coolant pumps (H_p) which is obtained from the pump simulation.

3.5 SODIUM PUMP MODEL

The primary and secondary coolant is transported by head produced in the centrifugal coolant pumps. The driving power for each pump is developed in a squirrel cage induction motor and is transmitted to the pump by an electromagnetic

(eddy-current) clutch assembly. The pump speed and coolant flow rate are controlled by varying the clutch excitation current.

Pump performance curves are required to simulate the primary and secondary pumps. Since the design is not fixed, actual pump performance curves are not available. Therefore, typical centrifugal pump performance curves have been assumed for the present simulation. The performance curves assumed for the primary pumps are shown in Figures 3.5.1 and 3.5.2. Similar curves have been assumed for the secondary pumps. The curves presently being used will be replaced by the actual pump performance curves when they become available. Figure 3.5.1 gives the primary pump head as a function of flow and speed and Figure 3.5.2 gives the pump efficiency as a function of normalized flow and speed. The efficiency curve applies to both primary and secondary pumps.

The following empirical equation was found to adequately represent the pump head curves:

$$H_p = H_{DCP} \left(\frac{N_p}{N_{DP}} \right)^2 + a_1 W^{1.8} \quad (3.5-1)$$

where H_p = pump head
 H_{DP} = pump design head
 H_{DCP} = pump design shutoff head (at zero flow, full speed)
 N_p = pump speed
 N_{DP} = pump design speed
 W = loop flow rate
 W_D = loop design flow rate
 a_1 = curve fit constant

The curve fit constant (a_1) is calculated from the shutoff head and the pump design conditions as follows:

at the design operating point, $N_p = N_{DP}$, $H_p = H_{DP}$, and $W = W_D$.

FIGURE 3.5.1. Head Curve for Primary Centrifugal Pump Simulation

FIGURE 3.5.2. *Typical Pump Efficiency Curves*

Therefore,

$$a_1 = \frac{H_{DP} - H_{DCP}}{(W_D)^{1.8}} \quad (3.5-2)$$

The reduced-speed head curves calculated by Equation (3.5-1) obey the affinity laws quite well as shown by the circled data points in Figure 3.5.1. These points were calculated from the assumed full speed design curve using the affinity laws.

A more complex relation is necessary to simulate the pump efficiency curves shown in Figure 3.5.2. The following empirical form was found adequate to represent these curves:

$$E_p = \left[a_2 \left(\frac{W}{W_D} \right) \left(\frac{N_{DP}}{N_p} \right) - a_3 \left(\frac{W}{W_D} \right)^2 \left(\frac{N_{DP}}{N_p} \right)^2 \right] \left(1 - e^{-a_4 N_p / N_{DP}} \right) \quad (3.5-3)$$

where E_p = pump efficiency.

Other terms are previously defined.

The pump torque is calculated from the head, efficiency, speed and flow as

$$T_p = \frac{H_p W}{b_2 N_p E_p} \quad (3.5-4)$$

where $b_2 = (2\pi) \frac{\text{radians}}{\text{revolution}} \cdot \frac{1 \text{ min}}{60 \text{ sec}} = \frac{\pi}{30}$.

The flow (W) is calculated in the hydraulics simulation (Section 3.4).

The pump shaft speed is obtained from the dynamic relationship for rotating bodies defined by Newton's first law. This can be expressed as a torque balance on the pump as follows:

$$\sum T = \frac{I_p b_2}{g_c} \frac{d N_p}{dt} \quad (3.5-5)$$

where g_c = gravity constant

I_p = moment of inertia of the pump shaft, pump impeller and clutch electromagnet assembly.

The summation of torques is equal to the torque provided by the clutch (T_c) minus the pump load torque (T_p). Therefore, the equation becomes

$$\frac{d N_p}{dt} = \frac{g_c}{I_p b_2} (T_c - T_p) \quad (3.5-6)$$

The clutch torque is calculated in the next section.

3.6 PUMP DRIVE ASSEMBLY MODEL

The pump drive assembly consists of the induction motors and the eddy-current clutch coupling. Figure 3.4.1 shows the motor, clutch and pump arrangement for the primary and secondary loops.

The dynamics of the induction motor can be described by Newton's first law of motion as applied to rotating bodies which will reduce to the following torque balance:

$$\frac{dN_M}{dt} = \frac{g_c}{I_M b_2} (T_M - T_{ML}) \quad (3.6-1)$$

where N_M = motor speed in rpm

I_M = motor moment of inertia

T_M = motor torque (provided by electrical input)

T_{ML} = motor load torque.

The motor torque is obtained from a typical motor torque curve as shown in Figure 3.6.1. Similar curves have been assumed for primary and secondary pump motors. These curves will be replaced by the actual motor torque curves when they are available. A typical load torque curve for one steady-state operating motor speed is also shown in Figure 3.6.1. The torque difference between the two curves for a given speed is the acceleration torque which is described by Equation (3.6-1).

The motor torque curve in Figure 3.6.1 can be expressed as a function of the maximum torque, motor slip, and slip at maximum torque by the following equation:

$$T_M = \frac{2 T_M^*}{\frac{S_M}{S_M^*} + \frac{S_M^*}{S_M}} \quad (3.6-2)$$

FIGURE 3.6.1. Motor Torque Curve with a Typical Load Torque

where S_M = motor slip, rpm
 S_M^* = motor slip at maximum torque
 T_M^* = maximum torque.

The motor slip is calculated as

$$S_M = \frac{N_S - N_M}{N_S} \quad (3.6-3)$$

where N_S is the synchronous speed of the motor.

The motor load torque (T_{ML}) is the only remaining undefined term in Equation (3.6-1). By performing a free-body torque balance on the motor, clutch, and pump impeller, we find that the clutch output torque is equal to the motor load torque:

$$T_C = T_{ML} \quad (3.6-4)$$

The clutch torque (T_C) is determined by the difference in motor shaft speed and pump shaft speed, and by the amount of electromagnetic coupling between the two shafts. The shaft speed difference is called the clutch slip speed (N_{SC}) and is represented as

$$N_{SC} = N_M - N_P \quad . \quad (3.6-5)$$

The degree of coupling in the clutch is determined by the amount of excitation current that is supplied to the electromagnetic windings. By varying this current, the pump speed becomes a controllable value.

As with the motor and pump, typical electromagnetic clutch torque curves have been assumed until actual data is available. The curves, shown in Figure 3.6.2, provide clutch torque as a function of slip speed and percent excitation current. The breakpoint slip (N_{SC}^*) is defined as the slip speed above which the clutch torque is constant for a constant excitation current.

For simulation, the clutch torque curves in Figure 3.6.2 have been broken down into two curves (functions) such that

$$T_C = T_C^* \cdot F_{1C} \cdot F_{2C} \quad (3.6-6)$$

where T_C^* = maximum clutch torque

F_{1C} = percent excitation function

F_{2C} = percent breakpoint slip function.

The curves that apply to Equation (3.6-6) are shown in Figure 3.6.3. The curves shown in Figures 3.6.2 and 3.6.3 apply to the primary loop clutches. Similar curves with a different maximum clutch torque have been assumed for the secondary clutches.

FIGURE 3.6.2. Typical Electromagnetic Clutch Torque Curve

(a) Excitation Current Dependency

(b) Slip Speed Dependency

FIGURE 3.6.3. Clutch Torque Functions; $T_C = T_{MAX} F_{1C} F_{2C}$,
 $Slip = N_M - N_P$, Slip (Breakpoint) as in
 Figure 3.6.2 = 400 rpm = N_{SC}^*

3.7 DUMP HEAT EXCHANGER MODEL

Heat is removed from the secondary sodium loops by air-cooled dump heat exchangers (DHX). Each DHX unit (one per loop) is divided into four separate modules as explained in Section 2.2. All modules are assumed to operate identically, therefore, only one module is simulated. The total heat is accounted for by applying the appropriate multiplying factor. Each DHX module contains multipass finned tubes arranged on a triangular pitch as shown in Figure 3.7.1.

* Reference Concept

FIGURE 3.7.1. DHX Module Arrangement (Four Modules Per DHX)

Air enters at the bottom, flows across the tubes and exits through stacks at the top. The sodium enters the tube sections at the top on the air outlet end and exits at the bottom.

Cross-flow heat exchangers are difficult to simulate due to their unsymmetrical and multidimensional temperature distributions. Preliminary simulation studies were made of several cross-flow models and a simple counter-current model (with and without log-mean temperature difference, LMTD, correctors for cross flow). These studies indicated that a counter-current representation (without LMTD correctors) provides adequate overall agreement with more complex crossflow models.⁽⁹⁾ Therefore, due to the limit on available computing equipment, a three-node counter-current model was chosen for the DHX simulation. To obtain a counter-current model from the crossflow arrangement shown in Figure 3.7.1, the tube sections are assumed to be straightened to their total length (90 ft for reference concept — six 15 ft passes per tube section). The straightened tubes are then assumed to be arranged on a triangular pitch so that the air volume per foot of tube is the same as in the actual crossflow arrangement. Finally, the air is assumed to flow parallel to the tubes in a counter-flow manner. The total effective heat transfer area is unchanged (corners connecting the tube passes are neglected in determining the effective heat transfer area). The counter-current node arrangement is shown in Figure 3.7.2.

With the counter-current assumption, the DHX modelling technique is nearly identical to that used for the intermediate heat exchanger, except there are three axial nodes instead of four. Fluid temperature and velocity profiles are assumed to be uniform over any cross-section normal to the direction of flow and axial heat conduction is neglected. Since the metal temperature approaches the sodium temperature, the metal heat storage is included with the sodium. Air is assumed an

FIGURE 3.7.2. DHX Model Node Arrangement

ideal gas and the effective air temperature for heat transfer is the average of the inlet and outlet temperatures at any cross-section. Heat losses from the air duct are also neglected. Density and specific heat for both air and sodium are assumed constant with temperature for any given problem. Although air density varies with temperature, the air heat storage term $(\rho V' C)_{Ti}$ is small and the air temperature transient response is not affected appreciably.

The basic partial differential equations for the coolant temperatures at each nodal point (i) are as follows:

Secondary Side

$$(\rho V' C)_{Si}^* \frac{\partial T_{CSi}}{\partial t} = - W_S C_{CSi} \frac{\partial T_{CSi}}{\partial y} - (UA')_Y (T_{CSi} - T_{CTi}) \quad (3.7-1)$$

Tertiary Side

$$(\rho V' C)_{Ti} \frac{\partial T_{CTi}}{\partial t} = W_T C_{CTi} \frac{\partial T_{CTi}}{\partial y} + (UA')_Y (T_{CSi} - T_{CTi}) \quad (3.7-2)$$

where the $(\rho V' C)_{Si}^*$ coefficients for the secondary sodium side include the tube metal heat storage. As before, the V' and A' terms represent volumes and areas per unit length. The DHX finite difference scheme for approximating the $\partial T / \partial y$ terms is similar to the scheme used for the IHX simulation except there are only three axial nodes. The nodal arrangement and finite difference approximations scheme are shown in Figure 3.7.2. The notations for "central" and "three-point backward" differences are the same as those given in Equation (3.2-3), (3.2-4), and (3.2-5) for the IHX model. However, the length between nodal points is expressed as $\Delta L_Y = L_Y / 3$ since there are only three DHX nodes. Substituting the difference approximations into the general equations, we obtain the following complete set of expressions for one DHX module. Subscripts are included for clarity:

C = coolant, S = secondary, T = tertiary, and Y = DHX.

Secondary Side:

$$(\rho V' C)_{SY}^* \frac{dT_{CSYi}}{dt} = \frac{3W_S C_{CSY}}{2N_{MY} L_Y} (\Delta T)_{CSYi} - (UA')_Y (T_{CSYi} - T_{CTYi}) \quad (3.7-3)$$

for $i = 2, 3, 4$;

Tertiary Side:

$$(\rho V' C)_{TY} \frac{dT_{CTYi}}{dt} = \frac{3W_{FT} C_{CTY}}{2L_Y} (\Delta T)_{CTYi} + (UA')_Y (T_{CSYi} - T_{CTYi}) \quad (3.7-4)$$

for $i = 1, 2, 3$, where N_{MY} is the number of modules per DHX, W_S is the secondary flow per loop, and W_{FT} is the air flow per module.

The temperature differences for node (i) are:

$$(\Delta T)_{CSY2} = -T_{CSY1} + T_{CSY3} \quad (3.7-5)$$

$$(\Delta T)_{CSY3} = 3T_{CSY3} - 4T_{CSY2} + T_{CSY1} \quad (3.7-6)$$

$$(\Delta T)_{CSY4} = 3T_{CSY4} - 4T_{CSY3} + T_{CSY2} \quad (3.7-7)$$

$$(\Delta T)_{CTY1} = -3T_{CTY1} + 4T_{CTY2} - T_{CTY3} \quad (3.7-8)$$

$$(\Delta T)_{CTY2} = -3T_{CTY2} + 4T_{CTY3} - T_{CTY4} \quad (3.7-9)$$

$$(\Delta T)_{CTY3} = -T_{CTY2} + T_{CTY4} \quad (3.7-10)$$

The equations are written for one module of the total DHX unit. Therefore, the areas and volumes correspond to one module and are calculated from the design data as explained in Appendix D. The overall heat transfer conductance $(UA')_Y$ is based on the tube inside surface area, and includes an inside film coefficient, a tube wall resistance and an outside film coefficient. The outside air film coefficient is calculated for air flowing across finned tube banks. The description of the overall heat transfer conductance $(UA')_Y$ can also be found in Appendix D.

3.8 TERTIARY COOLANT HYDRAULICS MODEL

The tertiary coolant system provides atmospheric air to the shell side of the DHX to remove heat from the secondary sodium. Each DHX module has its own tertiary coolant system that consists of a heat exchanger section, an inlet filter, a fan, a modulating damper or valve, and a stack exit. A schematic of the assumed arrangement is shown in Figure 3.8.1. The reference concept employs variable inlet vanes on the fan for capacity control. However, due to the lack of data for fans with this type of flow control, we have temporarily assumed damper control by a butterfly valve located downstream

FIGURE 3.8.1. Schematic of the Tertiary Coolant Hydraulic System for One DHX Module

of the fan. This makes air flow a function of the valve angle of opening for a given fan speed. The fan speed is assumed to be a constant in the present model, and a motor is not simulated. Only full speed and zero speed are presently considered. The fan, however, is simulated so that a motor can be easily added at a later time. More accurate fan data will be incorporated in the model when it is available.

Newton's second law of motion, applied to a compressible fluid, is used to simulate the tertiary hydraulics. Air is assumed to behave as an ideal gas. Also, energy due to coolant inertia is assumed to be negligible. This is a reasonable assumption since the coolant mass for air is small. The energy balance for the open system therefore reduces to the following pressure balance:

$$H_{FT} = \Delta P_{IT} + \Delta P_{FFT} + \Delta P_{VT} + \Delta P_{YT} + \Delta P_{ST} - \Delta P_{DT} + \Delta P_{EST} \quad (3.8-1)$$

where H_{FT} = fan head, in. of water

ΔP_{IT} = inlet loss, in. of water

ΔP_{FFT} = inlet filter loss, in. of water

ΔP_{VT} = butterfly valve loss, in. of water

ΔP_{YT} = DHX shell side loss, in. of water

ΔP_{ST} = stack friction loss, in. of water

ΔP_{DT} = natural draft, in. of water

ΔP_{EST} = stack exit loss, in. of water.

Fan performance curves are required to simulate the fan and thus obtain the fan head (H_{FT}). As stated earlier, actual fan curves are presently not available for fans with variable inlet vane control. Therefore, typical curves for fans without this feature are being used in the fan simulation. These curves are shown in Figure 3.8.2. The curves are normalized to the system design conditions. Curves for lower speeds are obtained from the full-speed design curve by applying the fan laws. Manufacturer's curves are used that provide data only down to $0.8 W_{FT}/W_{DFT}$. For $W_{FT}/W_{DFT} < 0.8$ the curves are assumed to take the shape of typical fan performance curves and are shown by dashed lines in Figure 3.8.2.

Since a variable-speed motor may be simulated in the future, the fan head is simulated as a function of both flow and speed. The following empirical equation was found to adequately represent the fan head curves in Figure 3.8.2.

FIGURE 3.8.2. Normalized Fan Characteristics (Typical)

$$\frac{H_{FT}}{H_{DFT}} = a_1 \left(\frac{N_{FT}}{N_{DFT}} \right)^2 - a_2 \left(\frac{W_{FT}}{W_{DFT}} - a_3 \frac{N_{FT}}{N_{DFT}} - a_4 \right)^2 \quad (3.8-2)$$

where N_{FT}/N_{DFT} = normalized fan speed (constant input in present analog model)

W_{FT}/W_{DFT} = normalized air flow/DHX module

H_{FT}/H_{DFT} = normalized fan head

a_1, a_2, a_3, a_4 = curve fit constants.

Equation (3.8-2) provides an adequate representation for all cases except the following. With the fan off and with zero flow (valve closed), Equation (3.8-2) will still yield a fan head. This is not possible if an airtight system is assumed. To solve this dilemma, we let the constant $a_4 = 0$ when $N_{FT}/N_{DFT} = 0$. With this condition included, Equation (3.8-2) provides an adequate representation of the assumed fan performance curves.

Since the coolant inertia terms are neglected in the energy balance Equation (3.8-1), we no longer have a differential equation for calculating the tertiary flow (W_{FT}) as we do in calculating the primary and secondary sodium flows (Appendix C). Therefore, the air flow is calculated from the expression that describes the butterfly valve characteristic. Input to this expression is the valve pressure drop (ΔP_{VT}) which is calculated by rearranging Equation (3.8-1) as follows:

$$\Delta P_{VT} = H_{FT} + \Delta P_{DT} - \Delta P_{IT} - \Delta P_{FFT} - \Delta P_{YT} - \Delta P_{ST} - \Delta P_{EST} \quad (3.8-3)$$

The tertiary flow per module (W_{FT}) is then calculated by the following equation.⁽¹⁰⁾

$$W_{FT} = 124 C_v D^2 \sqrt{\frac{P_{YT} \Delta P_{VT}}{(T_A + 460)}} \quad (3.8-4)$$

where C_v = flow coefficient (function of valve angle of opening)

D = valve diameter, ft

P_{YT} = line pressure = atmospheric, psi

T_A = air inlet temperature, °F.

The flow coefficient, C_v , is obtained from Figure 3.8.3, which provides the flow coefficient as a function of the valve angle of opening, θ . The assumption that P_{YT} = atmospheric is reasonable since the fans operate at very low head (less than 10 in. water).

FIGURE 3.8.3. *Butterfly Valve Coefficients (Typical)*

The only terms left to define are the pressure losses through the system. Duct losses are neglected.

Inlet and Exit Losses, ΔP_{IT} , ΔP_{EST} :

Starting with Bernoulli's equation and assuming constant density, we obtain the following expression for pressure loss.

$$\frac{P_1 - P_2}{\rho} = \frac{\Delta P}{\rho} = \frac{V_2^2 - V_1^2}{2g_c} + F \quad (3.8-5)$$

where F = friction pressure loss between points 1 and 2
 (point 2 is downstream of point 1)
 (potential energy is zero for inlet and exit).

At the duct inlet, $V_1 = 0$ (relative to V_2) and the inlet friction loss is known to be⁽¹¹⁾

$$F_{IT} = 0.5 \frac{V_2^2}{2g_c} \quad (3.8-6)$$

Substituting in Equation (3.8-5), the inlet pressure loss is

$$\Delta P_{IT} = \frac{\rho V_2^2}{2g_c} + \frac{0.5 \rho V_2^2}{2g_c} = \frac{1.5 \rho V_2^2}{2g_c} \quad (3.8-7)$$

Substituting

$$V_2 = \frac{W_{FT}}{\rho A_{IT}} \quad (3.8-8)$$

into Equation (3.8-7), we get the inlet pressure loss, in inches of water:

$$\Delta P_{IT} = \frac{1.5 b_3 W_{FT}^2}{2g_c \rho A_{IT}^2} \quad (3.8-9)$$

where A_{IT} = inlet area
 b_3 = conversion constant, in. water/lb-ft².

By the ideal gas law, density is expressed as

$$\rho = \frac{P_A}{R(T_A + 460) b_4} \quad (3.8-10)$$

where P_A = inlet atmospheric pressure, psi
 T_A = air inlet temperature, °F
 R = ideal gas constant for air, ft-lb_f/lb_m-°R
 b_4 = conversion constant, ft²/in.².

Substituting Equation (3.8-10) in (3.8-9), we get

$$\Delta P_{IT} = \frac{1.5(b_5)(b_3) R(T_A + 460)}{2g_c P_A A_{IT}^2} W_{FT}^2 \quad (3.8-11)$$

or

$$\Delta P_{IT} = \frac{C_{GT} (T_A + 460)}{A_{IT}^2} W_{FT}^2 \quad (3.8-12)$$

where $C_{GT} = \frac{1.5 R b_5 b_3}{2g_c P_A} \quad (3.8-13)$

Equation (3.8-11) is used to represent the inlet pressure loss. To obtain the exit loss at the top of the stack, we let $V_2 = 0$ (relative to V_1) in Equation (3.8-5). This yields

$$\Delta P_{EST} = - \frac{V_1^2}{2g_c} + \rho F_{EST} \quad (3.8-14)$$

The exit friction loss is known to be equal to one velocity head.⁽¹¹⁾ Therefore, substituting into Equation (3.8-14), we get

$$\Delta P_{EST} = - \frac{\rho V_1^2}{2g_c} + \frac{\rho V_1^2}{2g_c} = 0 \quad (3.8-15)$$

Equation (3.8-15) states that the change in kinetic energy at the exit is equal to the exit friction loss and the net pressure change is zero.

Inlet Filter Loss, ΔP_{FFT} :

The pressure loss through the inlet filter is assumed to be a function of the filter design loss at rated capacity and the flow squared as follows:

$$\Delta P_{FFT} = \Delta P_{DFT} \left(\frac{W_{FT}}{W_{DFT}} \right)^2 \quad (3.8-16)$$

where ΔP_{DFT} = filter loss at design flow, inches of water
(assumed to be a constant)

W_{DFT} = design flow.

Valve Loss, ΔP_{YT} :

The valve pressure drop is described by Equation (3.8-4).

DHX Shell Side Loss, ΔP_{YT} :

The DHX shell side pressure loss, ΔP_{YT} , is calculated using the Gunter and Shaw⁽¹²⁾ correlation for pressure drop in cross-flow through finned tube bundles.

The following expression applies:

$$\Delta P_{YT} = \frac{f_{YT} H_{YT} (D'_{ev}/P)^{0.4} \left(\frac{W_{YT}}{A_{CT}} \right)^2}{2g_c \bar{\rho}_{YT} D'_{ev} \phi} \quad (3.8-17)$$

where f_{YT} = friction factor (explained below)

H_{YT} = length of the flow path through the tube array, ft

D'_{ev} = equivalent diameter (explained below)

P = tube pitch, ft (see DHX section)

A_{CT} = free-flow area for a single bank of tubes located at right angles to the air flow (i.e., crossflow arrangement), ft

$\phi = (\mu_B/\mu_{wall})^{0.14}$ = air viscosity correction factor ($\phi = 1.0$ is assumed)

μ_B = air viscosity at bulk average air temperature

μ_{wall} = air viscosity at wall temperature

$\bar{\rho}_{YT}$ = average air density based on average DHX air temperature, lb/ft³.

The friction factor (f_{YT}) is assumed to be a constant in the simulation and is obtained from Curve b in Figure D-2, Appendix D. The equivalent diameter (D'_{ev}) is calculated as follows:

$$D'_{ev} = \frac{4 \times \text{net free volume}}{\text{frictional surface}}$$

$$D'_{ev} = \frac{4[0.867 P^2 - (\pi D_o^2/4) - \pi (D_{Fins}^2 - D_o^2) y_b N_{Fins}/4]}{A'_1 + A'_2} \quad (3.8-18)$$

where A'_1 is fin surface area per unit length, A'_2 is the bare tube surface per unit length, D_o is the tube outside diameter, D_{Fins} is fin diameter, y_b is fin thickness, and N_{Fins} is the number of fins per unit length. The Reynolds number used to obtain f_{YT} in Figure D-2 (Appendix D) is also based on this same equivalent diameter (D'_{ev}).

Stack Friction Loss, ΔP_{ST} :

The stack friction loss in inches of water is calculated by the following expression:⁽¹³⁾

$$\Delta P_{ST} = C_{ST} W_{FT}^2 \quad (3.8-19)$$

$$\text{where } C_{ST} = \frac{f_{ST} H_{ST}}{2g_c (5.19) D_{ST} \rho_{ST} (\pi D_{ST}^2/4)^2} \quad (3.8-20)$$

f_{ST} = Darcy friction coefficient for steel stack (from Moody diagram with $\epsilon = 0.00015$ for steel stack—assumed constant in simulation)

H_{ST} = stack height, ft

D_{ST} = stack diameter, ft

ρ_{ST} = air density at DHX air outlet temperature, lb/ft³.

Natural Draft, ΔP_{DT} :

Natural draft occurs in the stack because of a difference in gas density at the stack inlet and outlet. This draft (in inches of water) is calculated as a function of the DHX outlet temperature (T_{CTY1}) and the atmospheric air temperature as follows:⁽¹⁴⁾

$$\Delta P_{DT} = C_{DT} \left(\frac{1}{T_A + 460} - \frac{1}{T_{CTY1} + 460} \right) \quad (3.8-21)$$

where $C_{DT} = 0.52 P_A H_{ST}$

P_A = atmospheric pressure, psi

H_{ST} = stack height, ft

T_A = atmospheric air temperature, °F

T_{CTY1} = DHX air outlet (stack inlet) temperature, °F.

This completes the definition of the terms. For simulation purposes, all of the equations have been combined to calculate the tertiary air flow (W_{FT}) as a function of valve angle of opening and DHX air outlet temperature. Curves for two cases (fan on and fan off) have been calculated by hybrid function generators and are shown in Figures 3.8.4 and 3.8.5. With the fan on, the flow curves are essentially the same for all DHX air outlet temperatures. This is because the natural draft is a small percent of the total flow when the fan is on. With the fan off, however, the air outlet temperature greatly affects the air flow, since the natural draft is the main driving force.

4. SAMPLE TRANSIENT RESULTS

Simulation results for the following two typical transients are presented here to demonstrate the system model performance: 1) manual reactor scram with full primary and secondary flows, and air fan off with valve open; 2) manual reactor scram with simultaneous primary, secondary, and

FIGURE 3.8.4. DHX Air Flow Per Module Versus Valve Angle of Opening for Fan at Full Design Speed

FIGURE 3.8.5. DHX Air Flow Per Module Versus Valve Angle of Opening for Fan Off

tertiary flow decays. The system variables of interest for the first transient are presented in Figures 4.1 and 4.2, while the results of the second transient are given in Figures 4.3 and 4.4. The initial steady-state conditions, given below, were identical for both transients.

Steady State Conditions:

Total Reactor Power	$PL_T = 400 \text{ MW}$
Reactor Vessel Inlet Temperature,	$T_{CIR} = 622 \text{ }^\circ\text{F}$
Reactor Vessel Outlet Temperature,	$T_{COR} = 907 \text{ }^\circ\text{F}$
Primary Loop Flow Rate (four loops),	$W_P = 1102 \text{ lb/sec-loop}$
IHX Secondary Inlet Temperature,	$T_{CSX5} = 526 \text{ }^\circ\text{F}$
IHX Secondary Outlet Temperature,	$T_{CSX1} = 804 \text{ }^\circ\text{F}$
Secondary Loop Flow Rate (four loops),	$W_S = 1126 \text{ lb/sec-loop}$
DHX Air Inlet Temperature,	$T_A = 50 \text{ }^\circ\text{F}$
DHX Air Outlet Temperature,	$T_{CTY1} = 410 \text{ }^\circ\text{F}$
Tertiary Air Flow Per Module (16 modules),	$W_{FT} = 252 \text{ lb/sec-module.}$

The reactivity insertion graph appears in all figures for each transient to establish a common time zero, when the transient begins. The reactor scram is simulated by a negative 15\$ reactivity step for both sample transients.

The sample results indicate that the complete model performs in a reasonable manner. The model steady-state values provide a heat balance that is accurate to ~1%. Time delays function properly and all loop feedbacks and interactions react as expected. Flow decays and reactor responses are also reasonable. Additional transient studies indicate that all portions of the simulation respond in a reasonable manner, and that the model is ready for application.

FIGURE 4.1. Hybrid Simulation Model Results for Manual Scram with Full Primary, Secondary, and Tertiary Flows (Results from Output Channels 1-8). (Remaining results for this transient are in Figure 4.2.)

FIGURE 4.1. Hybrid Simulation Model Results for Manual Scram with Full Primary, Secondary, and Tertiary Flows (Results from Output Channels 1-8). (Remaining results for this transient are in Figure 4.2.)

FIGURE 4.2. Hybrid Simulation Model Results for Manual Scram with Full Primary, Secondary, and Tertiary Flows (Results from Output Channels 9-16). (Results for channels 1-8 are in Figure 4.1.)

Time After Start of Transient, sec

FIGURE 4.2. Hybrid Simulation Model Results for Manual Scram with Full Primary, Secondary, and Tertiary Flows (Results from Output Channels 9-16). (Results for channels 1-8 are in Figure 4.1.)

FIGURE 4.3. Hybrid Simulation Model Results for Manual Scram with Immediate Primary, Secondary, and Tertiary Flow Decays (Results from Output Channels 1-8). (Remaining results for this transient are in Figure 4.4.)

FIGURE 4.3. Hybrid Simulation Model Results for Manual Scram with Immediate Primary, Secondary, and Tertiary Flow Decays (Results from Output Channels 1-8). (Remaining results for this transient are in Figure 4.4.)

FIGURE 4.4. Hybrid Simulation Model Results for Manual Scram with Immediate Primary, Secondary, and Tertiary Flow Decays (Results from Output Channels 9-16). (Results for channels 1-8 are in Figure 4.3.)

FIGURE 4.4. Hybrid Simulation Model Results for Manual Scram with Immediate Primary, Secondary, and Tertiary Flow Decays (Results from Output Channels 9-16). (Results for channels 1-8 are in Figure 4.3.)

TABLE 4.1. *Sample Transient Design Data*

Numerical values in this table are of a conceptual nature and are subject to future revision.

Reactor Core

Number of Equivalent Driver Fuel Subassemblies (N_{ER})	82
Prompt Neutron Power (PL_N)	380 MW
Fission Product Power (PL_F)	20 MW
Fraction of Total Power Generated in Closed Loops (γ_{CR})	0
Fraction of Neutron Power Generated in Radial Reflector (γ_{RR})	0.0526
Driver Fuel Subassembly Data	
Fuel Composition (UO_2 , PuO_2)	20 to 25% PuO_2
Clad Material	TP 304 SS
Active Fuel Height	34 in.
Fuel Geometry	Hexagonal Rod Cluster (Triangular Pitch)
Pin Diameter	0.25 in.
Number of Pins Per Subassembly	217
Fuel Pin Pitch to Diameter Ratio	1.12
Subassembly Duct Dimensions	Hexagonal 4.5 in. Across Flats, 0.150 in. Thick
Reactor Vessel Flow Split	
Driver Fuel Subassemblies	0.90
Leakage	0.05
Radial Reflector	0.05

Intermediate Heat Exchanger (4 Units - One Per Loop)

Design Duty Per Unit	133 MW _t
Total Surface	3810 ft ²
Log-Mean Temperature Difference	125 °F
Tube OD	1.0 in.

TABLE 4.1. (contd)Intermediate Heat Exchanger (4 Units - One Per Loop) (contd)

Tube Wall Thickness	0.065 in.
Tube Pitch (Triangular)	1.375 in.
Number of Tubes Per Unit	858
Effective Tube Length	16 ft

Dump Heat Exchanger (4 Units with 4 Modules Per Unit)

Design Duty Per Unit	133 MW _t
Outside Finned Surface Per Unit	301,300 ft ²
Log-Mean Temperature Difference	322 °F
Tube OD	1.0 in.
Tube Wall Thickness	0.065 in.
Tube Pitch (Triangular)	2.75 in.
Fin OD	2.25 in.
Fin Thickness	0.022 in.
Fin Spacing	8 Fins/in.
Number of Tubes Per Unit	748
Number of Tube Passes	6
Number of Tube Rows Per Pass	2
Effective Tube Length	15 ft

Primary Loop Piping (Primary Pump in Cold Leg)

Reactor to IHX	
Length	93 ft
Pipe ID	23.25 in.
IHX to Primary Pump	
Length	43 ft
Pipe ID	23.25 in.
Primary Pump to Reactor	
Length	123 ft
Pipe ID	17.25 in.

Secondary Loop Piping (Secondary Pump in Cold Leg)

IHX to DHX	
Length	422 ft
Pipe ID	15.25 in.
DHX to Secondary Pump	
Length	172 ft
Pipe ID	15.25 in.

TABLE 4.1. (contd)Secondary Loop Piping (Secondary Pump in Cold Leg) (contd)

Secondary Pump to IHX	
Length	247 ft
Pipe ID	15.25 in.

Primary Sodium Pumps

Capacity Rating	12,000 gal/min
Total Dynamic Head	440 ft
Pumping BHP	1480
Pump Assembly Inertia	1318 lb-ft ²
Drive Motor Assembly Inertia	4173 lb-ft ²
Maximum Operating Speed	870 rpm

Secondary Sodium Pumps

Capacity Rating	12,000 gal/min
Total Dynamic Head	250 ft
Pumping BHP	870
Pump Assembly Inertia	718 lb-ft ²
Drive Motor Assembly Inertia	2754 lb-ft ²
Maximum Operating Speed	1170 rpm

Air Blower (One Per DHX Module)

Capacity Rating	279,000 ft ³ /min
Equivalent Pressure	7.6 in. Water
Blower BHP	372
Efficiency	79%
Speed	1170 rpm

NOMENCLATURE

		<u>Units</u>
A	Surface or flow area	ft^2
A'	Area per unit length	ft^2/ft
a	Constant	
b	Constant	
C	Specific heat	$\text{Btu}/\text{lb}-^\circ\text{F}$
C_f	Pressure loss coefficient	$\frac{\text{lb}(\text{sec})^{1.8}}{\text{ft}^2(\text{lb})^{1.8}}$
C_i	Power due to concentration of ith decay group	MW
C_v	Valve flow coefficient	
D	Diameter	ft
D_e	Equivalent diameter	ft
F	Analog function generator	
f	Darcy friction factor	
g_c	Acceleration of gravity	$\text{lb}_m\text{-ft}/\text{lb}_f\text{-sec}^2$
G	Mass velocity	$\text{lb}/\text{sec-ft}^2$
H	Height	ft
H_p	Pump head	psi
H_{FT}	Fan head	in. of water
h	Film coefficient for Heat Transfer	$\text{Btu}/\text{hr-ft}^2\text{-}^\circ\text{F}$
I	Mass Moment of Inertia	lb-ft-sec^2
j	Sieder and Tate heat transfer factor	
k	Thermal conductivity	$\text{Btu}/\text{sec-ft-}^\circ\text{F}$

		<u>Units</u>
δk	Reactivity feedback	
L	Length	ft
N_p	Pump speed	rpm
N_M	Motor speed	rpm
N_{FT}	Fan Speed	rpm
N_i	Number of converging flow paths	
P	Pressure	psi
PL	Power level	MW
p	Tube pitch	ft
P_e	Peclet number	
P_r	Prandtl number	
P_w	Wetted perimeter	ft
R	Heat transfer resistance/unit length or ideal gas constant	$\text{sec-ft-}^\circ\text{F/Btu}$ $\text{ft-lb}_f/\text{lb}_m\text{-}^\circ\text{R}$
r	Radius	ft
Re	Reynolds number	
S_M	Motor slip	
T	Temperature or torque	$^\circ\text{F}$ or ft-lb
t	Time	sec
U	Overall heat transfer coefficient	$\text{Btu/sec-ft}^2\text{-}^\circ\text{F}$
u,v	Velocity	ft/sec
V	Volume	ft^3
V'	Volume/unit length	ft^3/ft
W	Weight flow rate	lb/sec

		<u>Units</u>
x	Distance or length	ft
y_b	Fin thickness	ft
Z	Elevation	ft
<u>Greek Letters</u>		
α	Reactivity feedback coefficients	δk or $\delta k/^\circ F$
β_i	Delayed neutron fraction for ith Group	
β	Effective delayed neutron fraction = $\sum \beta_i$	
γ	Heat generation fraction	
θ	Valve angle of opening	Angular degrees
λ_i	Decay constant for ith Group of delayed neutrons	sec^{-1}
μ	Viscosity	lb/ft-sec
λ	Coolant inertance coefficient	sec^2/ft^2
λ^*	Prompt neutron lifetime	sec
ρ	Density	lb/ft^3
τ	Time delay for coolant flowing through a pipe section	sec
ϕ	$(\mu/\mu_W)^{0.14}$	
$\bar{\psi}$	Average value of ratio of diffusivity for heat transfer to diffusivity for momentum transfer	
Ω	Fin efficiency factor	

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APPENDIX A

REACTOR FUEL TO COOLANT HEAT TRANSFER COEFFICIENT

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REACTOR FUEL TO COOLANT HEAT TRANSFER COEFFICIENT

The overall heat transfer conductance $(UA')_R$ between the fuel and the coolant consists of the following heat transfer resistances: 1) fuel resistance, 2) fuel-clad gap resistance, 3) clad resistance, and 4) sodium film resistance. The following equation applies:

$$(UA')_R = \frac{n_p}{R_{\text{fuel}} + R_{\text{gap}} + R_{\text{clad}} + R_{\text{film}}} \quad (\text{A-1})$$

where n_p = number of fuel pins per subassembly.

The number of pins per subassembly is included because the resistances are all based on the heat transfer area per fuel pin. With the factor n_p included, the total $(UA')_R$ is then defined for one subassembly, as desired. Figure A-1 shows the fuel pin cross-section used to obtain the resistance terms.

r_1 = Fuel Radius
 r_2 = Clad Inside Radius
 r_3 = Clad Outside Radius

FIGURE A-1. Fuel Pin Cross-Section

The individual resistances are defined as follows.

Fuel Resistance, R_{fuel} :

Since only one radial node is simulated in the fuel region, we have assumed that the average fuel temperature is located at the volume average radius, $0.707r_1$, where r_1 = fuel radius. The fuel resistance then is

$$R_{\text{fuel}} = \frac{1}{8\pi k_f} \quad (\text{A-2})$$

where k_f = fuel thermal conductivity, Btu/sec-ft-°F. A length term does not appear since the overall coefficient is calculated on a per unit length basis.

Gap Resistance, R_{gap} :

A constant gap coefficient, h_g (Btu/sec-ft²-°F), is assumed. The gap resistance is defined as

$$R_{\text{gap}} = \frac{1}{2\pi r_2 h_g} \quad (\text{A-3})$$

$R_{\text{gap}} = 42.0$ ft-sec-°F/Btu for present reference concept.

Clad Resistance, R_{clad} :

Cylindrical heat transfer resistance is assumed to calculate the clad resistance.

$$R_{\text{clad}} = \frac{\ln(r_3/r_2)}{2\pi k_{\text{clad}}} \quad (\text{A-4})$$

where k_{clad} = clad thermal conductivity, Btu/sec-ft-°F.

Outside Film Resistance, R_{film} :

The clad surface film resistance is calculated by the following expression

$$R_{\text{film}} = \frac{1}{2\pi r_3 h_o} \quad (\text{A-5})$$

where h_o = sodium film coefficient, Btu/sec-ft²-°F.

The fuel resistance provides the bulk of the total heat transfer resistance in Equation (A-1) ($\sim 70\%$). The gap resistance provides $\sim 25\%$, assuming a gap coefficient of $1500 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$. Therefore, the fuel clad and film resistances contribute only about 5% to the total. The clad, fuel, and film resistances for the reference concept are shown in Figures A-2, A-3, and A-4. The clad conductivity is plotted as a function of coolant temperature since the clad and coolant are nearly the same temperature. Fuel thermal conductivity data for 93% TD (Theoretical Dense) $\text{PuO}_2\text{-UO}_2$ fuel were used to obtain Figure A-3. Reference 15 was used to obtain the film resistances in Figure A-4. The curves in Figure A-4 apply only to turbulent flow. The sodium film coefficients, however, will remain quite high even with laminar flow, and the overall conductance will not be affected appreciably whether the flow is laminar or turbulent.

Since we have assumed a constant gap coefficient, it is obvious from Figures A-2, A-3, and A-4 that the only parameter that will cause significant change in the overall heat transfer conductance, $(UA')_R$, is the variation of the average fuel temperature. Therefore, constant clad and film resistances have been assumed. This makes the overall heat transfer conductance, $(UA')_R$, a function of the average fuel temperature only, and Figure A-5 results. This curve is simulated to represent the fuel to coolant heat transfer.

FIGURE A-2. *Clad Heat Flow Resistance as a Function of Coolant Temperature*

FIGURE A-3. *Fuel Heat Flow Resistance as a Function of Average Fuel Temperature*

FIGURE A-4. Coolant Film Heat Flow Resistance as a Function of Tube Flow Rate and Coolant Temperature

FIGURE A-5. Overall Fuel-Coolant Heat Transfer Conductance as a Function of Average Fuel Temperature

APPENDIX B
IHX CONSTANTS AND HEAT TRANSFER COEFFICIENTS

APPENDIX B

IHX CONSTANTS AND HEAT TRANSFER COEFFICIENTS

VOLUMES AND AREAS

The geometry of the IHX is assumed to be the triangular pitch arrangement of Figure B-1.

FIGURE B-1. IHX Tube Geometry

From the above configuration, the following parameters for the analog-hybrid model may be calculated:

Shell-Side Flow Volume Per Unit Length (primary flow):

$$V'_{CPX} = \frac{N_{TX}}{4} (2\sqrt{3} p^2 - \pi D_o^2) \quad , \quad (B-1)$$

Tube-Side Flow Volume Per Unit Length (secondary flow):

$$V'_{CSX} = \frac{N_{TX}}{4} (\pi D_i^2) \quad , \quad (B-2)$$

Tube Metal Volume Per Unit Length:

$$V'_{MX} = N_{TX} \left(\frac{\pi}{4} \right) (D_o^2 - D_i^2) \quad , \quad (B-3)$$

where N_{TX} = number of tubes. The proper coefficients for the

simulation may be computed from the above three expressions by lumping half the metal volume with each sodium volume.

Stored Heat Per Unit Length, Primary Side:

$$(\rho V' C)_{PX}^* = (\rho C)_{Na} V'_{CPX} + \left(\frac{1}{2}\right) (\rho C)_M V'_{MX} \quad , \quad (B-4)$$

Stored Heat Per Unit Length, Secondary Side:

$$(\rho V' C)_{SX}^* = (\rho C)_{Na} V'_{CSX} + \left(\frac{1}{2}\right) (\rho C)_M V'_{MX} \quad , \quad (B-5)$$

where $(\rho C)_{Na}$ and $(\rho C)_M$ are the products of density and specific heat for sodium and tube metal, respectively. Values of ρ and C are selected based on average temperatures at steady-state conditions.

Heat Transfer Area Per Unit Length:

$$A'_X = N_{TX} \pi D_o \quad . \quad (B-6)$$

Heat transfer area is arbitrarily based on the tube outside diameter. Appropriate corrections are included in the heat transfer coefficient for the inside film coefficient and the tube metal heat resistance.

HEAT TRANSFER COEFFICIENT

The IHX heat transfer coefficient is computed from the following basic expression:

$$U_X = \left[\frac{1}{h_o} + \frac{1}{h_i (D_i/D_o)} + \frac{D_o \ln(D_o/D_i)}{2k_M} \right]^{-1} \quad (B-7)$$

where h_o and h_i are the outside and inside film coefficients, respectively, and k_M is the thermal conductivity of the tube metal. The parameters h_o and h_i are functions of temperature and flow, while k_M is a function of tube temperature only. Each of the three terms in the above expression is simulated as a function of the appropriate temperature and flow. The

last term, corresponding to the heat transfer through the tube is shown in Figure B-2 as a function of the wall temperature (assumed to be an average of the average sodium temperature on each side). The other two terms require more extensive derivation.

FIGURE B-2. Heat Transfer Coefficient Through IHX Tube Wall

FILM COEFFICIENTS

Film coefficients for sodium flow inside and outside the IHX tubes are obtained as functions of flow and temperature. The basic work of correlating experimental information to mathematical expressions was performed by O. E. Dwyer and M. W. Maresca.⁽⁶⁾ The Nusselt number relations are as follows:

$$h_i = \frac{k_i}{D_i} (Nu_i), \text{ and } h_o = \frac{k_o}{D_e} (Nu_o) \quad , \quad (B-8)$$

where k_i and k_o are sodium thermal conductivities at average temperatures inside and outside the tube, respectively; Nu_i and Nu_o are inside and outside Nusselt numbers, respectively. The equivalent outside diameter (D_e) is based on the following:

$$D_e = 4 \frac{(\text{flow area})}{(\text{wetted perimeter})} = D_o \left[\frac{2\sqrt{3}}{\pi} \left(\frac{p}{D_o} \right)^2 - 1 \right] \quad (\text{shell side}) \quad . \quad (B-9)$$

For turbulent flow and pitch/diameter ratios between 1.3 and 3.0, the Nusselt numbers may be expressed as

$$\text{Tube Side: } Nu_i = A + 0.025 (\bar{\psi}_i Pe_i)^{0.8} \quad , \text{ and} \quad (B-10)$$

$$\begin{aligned} \text{Shell Side: } Nu_o = 6.66 + 3.126 (p/D_o) + 1.184 (p/D_o)^2 \\ + 0.0155 (\bar{\psi}_o Pe_o)^{0.86} \quad , \end{aligned} \quad (B-11)$$

where Pe_i and Pe_o are the inside and outside Peclet numbers, respectively. The constant (A) accounts for data scatter and is given the value $A = 5.0$. The $\bar{\psi}$ terms represent the average, effective ratio of the eddy diffusivity of heat transfer to that for momentum transfer and are defined as follows:

$$\bar{\psi} = 1 - \frac{1.82}{Pr(\epsilon_M/\nu)_{\max}^{1.4}} \quad (B-12)$$

where Pr is the Prandtl number, and $\bar{\psi}$ may only be positive. The factor (ϵ_M/ν) is obtained from Dwyer's work,⁽¹⁶⁾ as presented in Figure B-3. The familiar Peclet, Prandtl, and Reynolds numbers are as follows:

$$\left. \begin{aligned} Pe &= \frac{4C(W)}{k(P_w)} , \\ Pr &= \frac{\mu C}{k} , \\ Re &= \frac{vD\rho}{\mu} , \end{aligned} \right\} \quad (B-13)$$

where C is specific heat, μ is viscosity, ρ is density, W is flow rate, P_w is wetted perimeter, v is fluid velocity, and D is the diameter (D_i for the tube side fluid and D_e for the shell side fluid).

Design data for the IHX and appropriate sodium temperatures and flows are used to determine the foregoing parameters with the end result of inside and outside film coefficients as functions of flow and temperature. These results for the current IHX design are shown in Figures B-4 and B-5. On either side of the IHX, sodium temperature is assumed to be the average temperature.

The overall heat transfer coefficient (U_X) is then computed from the information of Figures B-2, B-4, and B-5. At present, these families of curves are simulated on the analog computer with curve-fitting techniques; future modifications to the program will transfer such computations to the digital portion of the hybrid computer. It must be emphasized that this analysis is valid only for fully developed turbulent flow. Heat transfer will change markedly below $Re \sim 3000$, corresponding to flow rates of about 2% of FFTF design flow. For current plans, the simulation is adequate; however, should laminar flow regions be of interest in future work, the model may easily be altered to account for it.

FIGURE B-3. Value of $\left(\frac{\epsilon_M}{\nu}\right)_{\max}$ for Fully Established Turbulent Flow of Liquid Metals Through Circular Tubes, and Rod Bundles with Equilateral Spacing

FIGURE B-4. Shell Side IHX Inverse Film Coefficient

FIGURE B-5. Tube Side IHX Inverse Film Coefficient

APPENDIX C

PRIMARY AND SECONDARY COOLANT LOOPS HYDRAULICS THEORY

APPENDIX CPRIMARY AND SECONDARY COOLANT LOOPS HYDRAULICS THEORY

The hydraulics model used in the FFTF Dynamics Hybrid Simulation was derived from Newton's second law of motion applied to a viscous noncompressible fluid and nonsteady flow. The equation is

$$-\frac{g_c}{\rho} \frac{\partial P}{\partial S} - g \frac{\partial Z}{\partial S} - \frac{g_c \tau}{\rho} \frac{2}{r} = \frac{\partial}{\partial S} \frac{u^2}{2} + \frac{\partial u}{\partial t} \quad , \quad (C-1)$$

where S = axial distance along flow path

r = radius of flow path

u = fluid velocity

τ = viscous force.

The following derivation assumes that all loop pressure losses can be expressed as equivalent straight pipe pressure losses. Therefore, the following relations apply

$$\tau = \frac{h_L \rho r}{2L} \quad (C-2)$$

$$h_L = f(L/D) u^2 / 2g_c \quad (C-3)$$

$$u = \frac{W}{\rho A} \quad (C-4)$$

where h_L = head loss

f = Darcy's friction factor

L/D = pipe length/diameter ratio.

If we assume that the weight flow rate is the same at any point in a pipe section and that the coolant density is constant in the pipe section, substitution of Equation (C-2), (C-3), and (C-4) into Equation (C-1) yields the following equation:

$$-\frac{\partial P}{\partial S} - \rho \frac{\partial Z}{\partial S} - f \frac{W^2}{2g_c D A^2 \rho} = \frac{W^2}{2g_c \rho} \frac{\partial}{\partial S} \left(\frac{1}{A^2} \right) + \frac{1}{g_c A} \frac{W}{t} \quad (C-5)$$

Integration over a section with constant cross-sectional area then yields

$$-(\Delta P)_i - \rho_i (\Delta Z)_i - \frac{f_i W_i^2 (\Delta L)_i}{2g_c D_i A_i^2 \rho_i} = \frac{(\Delta L)_i}{g_c A_i} \frac{\partial W_i}{\partial t} \quad (C-6)$$

where i indicates parameters and variables for the i th section.

Equation (C-6) assumes friction and inertial losses are for uniform pipe sections. For a closed loop, $\sum (\Delta P)_i = 0$, and $\sum (\text{Kinetic Energy}) = 0$. Therefore, summation of Equation (C-6) around a closed loop yields

$$\rho H_p - \int \rho dZ - \frac{W^2}{2g_c} \sum \frac{f_i (\Delta L)_i}{D_i A_i^2 \rho_i} = \frac{1}{g_c} \left[\sum \frac{(\Delta L)_i}{A_i} \right] \frac{dW}{dt} \quad (C-7)$$

where W = flow per loop

H_p = pump head encountered by flowing fluid

$\int \rho dZ = \sum \rho_i (\Delta Z)_i$ = natural circulation pressure head.

Let the inertance coefficient, ℓ , be defined by the following relation

$$\ell = \frac{1}{g_c} \sum \frac{(\Delta L)_i}{A_i} \quad (\text{explained in detail later}) \quad (C-8)$$

and the friction pressure drop $(\Delta P)_f$ be defined as

$$(\Delta P)_f = \frac{W^2}{2g_c} \sum \frac{f_i (\Delta L)_i}{D_i A_i^2 \rho_i} \quad (C-9)$$

The following assumption was made: f_i is governed by a smooth pipe relationship ($r_o/\epsilon \approx 10^4$ for stainless steel) and turbulent flow. By inspection of the Moody Diagram, the Darcy friction factor (f_i) for turbulent flow, can be approximated by the following empirical equation

$$f_i = K_f \text{Re}_i^{-\alpha_f} \quad (\text{C-10})$$

where Re_i = Reynolds number of the section

$$K_f = 0.184$$

$$\alpha_f = 0.2.$$

The Reynold's number can be written as

$$\text{Re}_i = \frac{4W}{P_{wi}\mu_i} \quad (\text{C-11})$$

where P_{wi} is the wetted perimeter of the flow cross section, and μ_i is the coolant viscosity for the section. Substitution of Equations (C-10) and (C-11) in Equation (C-9) yields

$$(\Delta P)_f = \frac{K_f(W)^{2-\alpha_f}}{g_c(2)^{3+2\alpha_f}} \sum \frac{(\Delta L)_i (P_{wi})^{1+\alpha_f}}{A_i^3} \frac{(\mu_i)^{\alpha_f}}{\rho_i} \quad (\text{C-12})$$

Assuming constant density and viscosity for all sections in the loop, and substituting the known constants, Equation (C-12) becomes (units are in lb, ft, and sec):

$$(\Delta P)_f = (2.709 \times 10^{-4}) \frac{(\mu)^{0.2}}{\rho} \left[\sum \frac{(P_{wi})^{1.2} (\Delta L)_i}{A_i^3} \right] W^{1.8} \quad (\text{C-13})$$

Equation (C-13) can be expressed more simply as

$$(\Delta P)_f = C_f W^{1.8} \quad (\text{C-14})$$

$$\text{where } C_f = (2.709 \times 10^{-4}) \frac{(\mu)^{0.2}}{\rho} \sum \frac{(P_{wi})^{1.2} (\Delta L)_i}{A_i^3} \quad (\text{C-15})$$

Rather than summing up equivalent straight pipe pressure losses as calculated in Equation (C-13), a simpler approach may be taken. If the primary and secondary pressure losses and flows at steady state are known from design

conditions, Equation (C-14) can be used to calculate the pressure loss coefficient (C_f) for both primary and secondary loops as follows.

Primary Loop:

$$C_{fp} = \frac{(\Delta P)_{fp}}{W_p^{1.8}} \quad , \quad (C-16)$$

Secondary Loop:

$$C_{fs} = \frac{(\Delta P)_{fs}}{W_s^{1.8}} \quad , \quad (C-17)$$

where $(\Delta P)_{fp}$ = total primary loop pressure loss, lb/ft²
 $(\Delta P)_{fs}$ = total secondary loop pressure loss, lb/ft²
 W_p = primary flow per loop, lb/sec
 W_s = secondary flow per loop, lb/sec.

The inertance coefficients for the primary and secondary loops are calculated from Equation (C-8). The inertance terms for pipe sections can be calculated directly. In the heat exchangers (IHx and DHX) the inertance is calculated as the inlet-to-outlet length divided by the total flow area. The reactor inertance is more difficult to calculate, however, since nonuniform parallel flow paths are involved. The total primary flow splits three ways in the reactor vessel:

1) driver fuel and open loop tubes, 2) bypass leakage, and 3) radial reflector. An equivalent reactor vessel inertance (ℓ_{RV}) can be calculated as follows:

$$\frac{1}{\ell_{RV}} = \frac{1}{\ell_1} + \frac{1}{\ell_2} + \frac{1}{\ell_3} \quad (C-18)$$

where ℓ_1 = inertance of driver fuel and open loop tubes
 ℓ_2 = inertance of leakage region
 ℓ_3 = inertance of radial reflector region.

Each individual inertance is calculated as follows.

Tube Inertance:

$$l_1 = \frac{1}{g_c} \frac{\Delta L_T}{N_{ER} \bar{A}_T} \quad (C-19)$$

where ΔL_T = total tube length through reactor

N_{ER} = number of equivalent driver fuel elements

$\bar{A}_T$ = average flow area per fuel and open loop tube for total tube length (calculated on a length weighted basis to account for different tube cross-sections along the tube length);

$$\bar{A}_T = \frac{\sum A_j \Delta L_j}{\Delta L_T} \quad (C-20)$$

where A_j = flow area of tube section j

ΔL_j = length of tube section j with flow area A_j .

Leakage Inertance:

$$l_2 = \frac{1}{g_c} \frac{\Delta L_L}{\bar{A}_L} \quad (C-21)$$

where ΔL_L = bypass leakage length through reactor

$\bar{A}_L$ = average leakage flow area through reactor.

Radial Reflector Inertance:

$$l_3 = \frac{1}{g_c} \frac{\Delta L_{RR}}{\bar{A}_{RR}} \quad (C-22)$$

where ΔL_{RR} = radial reflector length through reactor

$\bar{A}_{RR}$ = average radial reflector flow area through reactor.

Substituting Equations (C-19), (C-21), and (C-22) into Equation (C-18), we get

$$\frac{1}{\ell_{RV}} = g_c \left(\frac{N_{ER} \bar{A}_T}{\Delta L_T} + \frac{\bar{A}_L}{\Delta L_L} + \frac{\bar{A}_{RR}}{\Delta L_{RR}} \right) \quad (C-23)$$

If we assume that the lengths of the three paths are identical (reasonable for FTR), Equation (C-23) reduces to

$$\frac{1}{\ell_{RV}} = g_c \left(\frac{N_{ER} \bar{A}_T + \bar{A}_L + \bar{A}_{RR}}{\Delta L_R} \right) \quad (C-24)$$

where $\Delta L_R = \Delta L_T = \Delta L_L = \Delta L_{RR}$.

Inverting Equation (C-24), we get

$$\ell_{RV} = \frac{1}{g_c} \frac{\Delta L_R}{N_{ER} \bar{A}_T + \bar{A}_L + \bar{A}_{RR}} \quad (C-25)$$

Since the flow per loop is calculated in Equation (C-7), the inertance term for the reactor vessel must be multiplied by the number of primary loops, (N_{PL}). Therefore, the inertance term for the reactor vessel in Equation (C-8) is

$$\ell_{RV} = \frac{1}{g_c} \frac{N_{PL} \Delta L_R}{N_{ER} \bar{A}_T + \bar{A}_L + \bar{A}_{RR}} \quad (C-26)$$

The natural circulation term is approximated by

$$\int \rho dZ = (\Delta Z) \frac{\partial \rho}{\partial T} (T_{AH} - T_{AC}) \quad (C-27)$$

where T_{AH} and T_{AC} are the average hot and cold leg temperatures, respectively, and $(\partial \rho / \partial T) \simeq -0.0082 \text{ lb/ft}^3\text{-}^\circ\text{F}$ (slope of sodium density versus temperature curve). The term ΔZ is the net elevation difference between the core and IHX in the primary loop, and between the IHX and DHX centerlines in the secondary loop.

Substituting Equations (C-8), (C-14), and (C-27) into Equation (C-7), we obtain the expression that represents primary and secondary sodium hydraulics:

$$\rho \frac{dW}{dt} = \rho H_p + 0.0082 (\Delta Z)(T_{AH} - T_{AC}) - C_f W^{1.8} \quad (C-28)$$

where all terms are defined previously.

APPENDIX D

DHX CONSTANTS AND HEAT TRANSFER COEFFICIENTS

APPENDIX DDHX CONSTANTS AND HEAT TRANSFER COEFFICIENTS

The DHX finned tubes are arranged on an equilateral triangular pitch as shown in Figure D-1.

FIGURE D-1. DHX Tube and Fin Geometry

DHX CONSTANTS

All volumes and areas are calculated per foot of tube length and are denoted by V' and A' . From Figure D-1 the following DHX parameters may be calculated.

Shell-Side Flow Volume Per Unit Length (Air Flow):

$$V'_{CTY} = N_{TMY} [V'_{Hex} - V'_{Tube} - V'_{Fins}]$$

$$V'_{CTY} = N_{TMY} \left[0.866 p^2 - \frac{\pi}{4} D_o^2 - N_{Fins} \cdot y_b \cdot \frac{\pi}{4} (D_{Fins}^2 - D_o^2) \right], \quad (D-1)$$

Tube-Side Flow Volume Per Unit Length (Secondary Sodium):

$$V'_{CSY} = N_{TMY} \cdot \frac{\pi D_i^2}{4} \quad , \quad (D-2)$$

Tube Metal Volume Per Unit Length:

$$\begin{aligned} V'_{MY} &= N_{TMY} [V'_{Tube} + V'_{Fins}] \\ V'_{MY} &= N_{TMY} \left[\frac{\pi}{4} (D_o^2 - D_i^2) + N_{Fins} \cdot y_b \right. \\ &\quad \left. \cdot \frac{\pi}{4} (D_{Fins}^2 - D_o^2) \right] \end{aligned} \quad (D-3)$$

where N_{TMY} = number of tubes per DHX module

N_{Fins} = number of fins per unit length

Other nomenclature in Figure D-1.

To obtain the stored heat terms used in the simulation, the tube metal heat storage is lumped with the secondary sodium heat storage.

Secondary Stored Heat Per Unit Length, Tube Side (Sodium):

$$(\rho V'C)^*_{SY} = (\rho C)_{Na} V'_{CSY} + (\rho C)_M V'_{MY} \quad , \quad (D-4)$$

Tertiary Stored Heat Per Unit Length, Shell Side (Air):

$$(\rho V'C)_{TY} = (\rho C)_{Air} \cdot V'_{CTY} \quad (D-5)$$

where $(\rho C)_{Na}$, $(\rho C)_M$, and $(\rho C)_{Air}$ are the products of density and specific heat for sodium, tube metal, and air, respectively. These properties are assumed constant and are based on the average steady-state temperatures, with the tube metal assumed to be at the same temperature as the tube side secondary sodium.

Heat Transfer Area Per Unit Length:

The heat transfer area is based on the tube inside surface area since it is easily calculated. The outside

film coefficient and tube wall resistance are both corrected to account for this assumption.

$$A'_Y = N_{TMY} \pi D_i \quad (D-6)$$

HEAT TRANSFER COEFFICIENT

The overall DHX heat transfer coefficient is calculated by the following formula:

$$U_Y = \left[\frac{1}{h_i} + \frac{D_i \ln (D_o/D_i)}{2 k_M} + \frac{1}{h_o} \right]^{-1} \quad (D-7)$$

where h_i and h_o are the tube inside and outside film coefficients, respectively, k_M is the base tube metal thermal conductivity, and D_i and D_o are inside and outside tube diameters, respectively.

The inside film coefficient (h_i) is calculated in the same manner as the IHX inside film coefficient (see Appendix B). The tube wall coefficient is for simple cylindrical heat transfer and is based on the tube inside surface. The metal thermal conductivity is a function of tube temperature which is assumed to be identical to the tube side sodium temperature. The shell-side air film coefficient requires further explanation.

Shell-Side DHX Air Film Coefficient

The shell-side film coefficient is calculated for air flowing across finned tube banks and is based on the tube inside surface. Transverse fins of uniform thickness are assumed. Data generated by Jameson⁽¹⁷⁾ has been arranged by Kern⁽¹⁸⁾ in the form of a Sieder-Tate heat transfer factor, as shown in Figure D-2 (curve b in Figure D-2 is used for the air pressure loss calculation in the tertiary hydraulics model, Section 3.8). The "j" factor in Figure D-2 is plotted against the air flow Reynolds number and is represented by the following expression:

film coefficient and tube wall resistance are both corrected to account for this assumption.

$$A'_Y = N_{TMY} \pi D_i \quad . \quad (D-6)$$

HEAT TRANSFER COEFFICIENT

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$$U_Y = \left[\frac{1}{h_i} + \frac{D_i \ln (D_o/D_i)}{2 k_M} + \frac{1}{h_o} \right]^{-1} \quad (D-7)$$

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G_s = Air mass velocity determined from free-flow area
for single bank of tubes at right angles to air flow
for cross-flow heat exchanger

De_v = (4) (Net Free Volume/Frictional Surface)

De = (2) (Fin Area + Bare Tube Area)
(π) (Projected Perimeter)
(f) (G_s^2) (L)

$$\Delta P = \frac{(5.22 \times 10^{10}) (De'_v) (S) (\phi_s)}{\left(\frac{De'_v}{S_T}\right)^{0.4} \left(\frac{S_L}{S_T}\right)^{0.6}}$$

FIGURE D-2. Transverse-Fin Heat Transfer and Pressure Drop

$$j = \frac{h_{oc} D_e}{k_{air} (Pr)^{1/3}} \quad (D-8)$$

where h_{oc} is the clean air-film-coefficient (fin efficiency not included), k_{air} is air conductivity, Pr is air Prandtl number, and D_e is outside equivalent diameter. All air physical properties are a function of the DHX (inlet-outlet)

average air temperature. The air flow Reynolds number (abscissa in Figure D-2) is calculated as:

$$R_e = \frac{D_e G_s}{\mu} \quad , \quad (D-9)$$

where G_s is the mass velocity (units, lb/sec-ft²) determined from the free-flow area for a single bank of tubes located at right angles to the air flow (i.e., as in the cross-flow arrangement). The equivalent diameter (D_e) in the "j" factor equation and the Reynolds number equation is calculated as follows:

$$D_e = \frac{2(A'_1 + A'_2)}{\pi \chi_1} \quad (D-10)$$

where A'_1 = fin surface per unit length

$$= \frac{2\pi}{4} (D_{Fin}^2 - D_o^2) N_{Fins}$$

A'_2 = bare tube surface per unit length

$$= \pi D_o - \pi D_o N_{Fins} y_b$$

χ_1 = projected perimeter per unit length

$$= 2N_{Fins} (D_{Fins} - D_o) + 2(1 - N_{Fins} y_b)$$

(all terms are given in Figure D-1). The projected perimeter is the sum of all external distances in the plan view of a transverse finned tube. The surface on the outer edge of the fin is neglected since the fins are usually very thin (on the order of 0.03 in. or less).

From the Reynolds number, the "j" factor is obtained from Figure D-2. A clean air-film-coefficient (h_{oc}) is then calculated from the "j" factor by rearranging Equation (D-8). This clean air-film-coefficient is then degraded to account for fouling:

$$h_{of} = \frac{h_{oc}/R_{foul}}{h_{oc} + 1/R_{foul}} \quad . \quad (D-11)$$

The actual outside film coefficient (h_o) based on the tube inside surface area is then calculated from the fin efficiency curves given in Figure D-3 for annular fins of constant thickness. The fin efficiency (Ω) from Figure D-3 corrects the fin surface area and, therefore, is multiplied times A'_1 in the following equation for h_o :

$$h_o = (\Omega A'_1 + A'_2) \frac{h_{of}}{\pi D_i} \quad (D-12)$$

To simulate the overall DHX heat transfer coefficient, the inside and outside film coefficients are plotted as functions of flow and average coolant temperature in the same manner as in the IHX model. The metal conductivity term is

FIGURE D-3. Efficiency of Annular Fins of Constant Thickness

also treated the same as it was in the IHX model. The curves are then simulated by curve-fitting techniques on the analog computer. The overall coefficient (U_Y) is obtained by summing the terms and inverting as per Equation (D-7). Both air and sodium film coefficients are calculated for turbulent flow only in the present model. Future modifications can easily be made, if necessary, to account for laminar flow.

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	C. D. Flowers (10)	G. G. Thieme
	L. H. Gerhardstein (5)	J. C. Tobin
	A. L. Gunby	K. G. Toyoda

Battelle-Northwest (contd)

M. A. Vogel
R. C. Walker
J. H. Westsik
L. A. Whinery
N. G. Wittenbrock
M. R. Wood
F. W. Woodfield
G. A. Worth
FFTF File (5)
Technical Information Files (5)
Technical Publications (2)

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