BNWL-1319
UC-80

2-
4-70

DYNAMIC DIGITAL COMPUTER SIMULATION
OF THE FFTF
DUMP HEAT EXCHANGER
R. N. MADSEN
APRIL, 1970

AEC RESEARCH &
DEVELOPMENT REPORT

ROUTE TO	FILE NO.	DATE	FILED DATE

LEGAL NOTICE

This report was prepared as an account of Government sponsored work. Neither the United States, nor the Commission, nor any person acting on behalf of the Commission:

A. Makes any warranty or representation, expressed or implied, with respect to the accuracy, completeness, or usefulness of the information contained in this report, or that the use of any information, apparatus, method, or process disclosed in this report may not infringe privately owned rights; or

B. Assumes any liabilities with respect to the use of, or for damages resulting from the use of any information, apparatus, method, or process disclosed in this report.

As used in the above, "person acting on behalf of the Commission" includes any employee or contractor of the Commission, or employee of such contractor, to the extent that such employee or contractor of the Commission, or employee of such contractor prepares, disseminates, or provides access to, any information pursuant to his employment or contract with the Commission, or his employment with such contractor.

PACIFIC NORTHWEST LABORATORY

RICHLAND, WASHINGTON

operated by

BATTELLE MEMORIAL INSTITUTE

for the

UNITED STATES ATOMIC ENERGY COMMISSION UNDER CONTRACT AT(45-1)-1830

3 3679 00061 8977

BNWL-1319
UC-80 Reactor
Technology

DYNAMIC DIGITAL COMPUTER SIMULATION
OF THE FFTF
DUMP HEAT EXCHANGER

By

R. N. Madsen

Reactor and Plant Technology Department
FFTF Division

April 1970

BATTELLE MEMORIAL INSTITUTE
PACIFIC NORTHWEST LABORATORIES
RICHLAND, WASHINGTON 99352

Printed in the United States of America
Available from
Clearinghouse for Federal Scientific and Technical Information
National Bureau of Standards, U.S. Department of Commerce
Springfield, Virginia 22151

Price: Printed Copy \$3.00; Microfiche \$0.65

DYNAMIC DIGITAL COMPUTER SIMULATION
OF THE FFTF
DUMP HEAT EXCHANGER

R. N. Madsen

ABSTRACT

The complex problem of analytically modeling the FFTF extended-surface sodium-to-air dump heat exchanger (DHX) for dynamic simulation is investigated in detail. The development of an extremely detailed reference model followed by the determination of an idealized yet adequate simple model is described. In connection with this study, a detailed investigation of transient fin efficiency for fins of constant thickness is reported.

CONTENTS

ABSTRACT	iii
LIST OF FIGURES	vii
1. Introduction	1.1
2. Summary	2.1
3. Reference Simulation Development	3.1
3.1 Model Description	3.1
3.1.1 Nodalized Representation of the DHX	3.2
3.1.2 Calculation of Admittances and Capacitances	3.3
3.1.3 Heat Transfer Coefficients	3.9
3.1.4 Stability	3.12
3.2 Verification of the Reference Simulation	3.13
3.2.1 Internal Consistency Verification	3.13
3.2.2 External Verifications	3.15
3.2.3 Check of Transient Calculations	3.18
4. Idealized Model Development	4.1
4.1 Model Description	4.1
4.1.1 Numerical Methods for Calculating Spatial Temperature Derivatives	4.4
4.1.2 Calculation of Constants and Heat Transfer Coefficients	4.5
4.2 Parametric Examination of the Three-Radial-Node Model	4.7
4.3 Transient Calculations for Extended Surfaces	4.9
4.3.1 Transient Fin Efficiency	4.13
4.3.2 Model Description for 6-Radial Nodes	4.15
4.3.3 Refinements To The DHX Simulations	4.20

CONTENTS (Continued)

5. Discussion.	5.1
NOMENCLATURE.	5.4
APPENDIX A - Thermophysical Properties.	A.1
APPENDIX B - Dimensional Data for the Dump Heat Exchanger	B.1
APPENDIX C - Exact Calculation of Steady State Fin Temperature, Heat Transfer and Fin Efficiency	C.1
APPENDIX D - Verification of TAP Model Transient Calculations by the Methods of Myers, <u>et al</u>	D.1
APPENDIX E - The Time Dependent Solution for the Temperature and Heat Transfer in an Extended Surface of Constant Thickness	E.1
REFERENCES	

FIGURES

3.1	Actual DHX Tube Arrangement	3.2
3.2	Diagram of DHX Tube Sections.	3.3
3.3	Nodal Arrangement -- Typical for Each Section	3.5
3.4	Radial Temperature Distribution in a Fin.	3.16
4.1	6-Axial Node, 3-Radial Node Model	4.3
4.2	Comparison of Predicted Fluid Exit Temperatures for 3-Radial Node Model vs. TAP Reference	4.10
4.3	Ω vs Time Based on Tabulated Values Compared With Selected Results From TAP Calculations For a Typical Loss-of-Power Transient	4.12
4.4	4-Axial Node, 6-Radial Node Model	4.16
4.5	Node Dimensions for 6-Radial Node Model	4.19
4.6	TAP Reference Model Compared With 4- and 8-Axial-Node, 6-Radial-Node Models for a Typical Loss-of-Power Transient	4.21
4.7	Algebraic Air Node Calculations for Various Lag Time Constants vs Explicit Integration.	4.23
5.1	Test Case - Ramp of Air Flow to 10% in 1 sec.	5.2
5.2	Test Case - Ramp of Sodium Flow to 10% in 10 sec.	5.3
A.1	Thermophysical Properties of Liquid Sodium.	A.2
A.2	Thermophysical Properties of Dry Air.	A.3
D.1	Unstepped-Fluid Response (Large $\bar{c}_w^*$) for $P = 0.5528$ and $M = 32.792$	D.4
D.2	Stepped-Fluid Response (Large $\bar{c}_w^*$) for $P = 0.5528$ and $M = 32.792$	D.4
D.3	Comparison of TAP Reference Model Response With MM-3 Report Results for a Step Change in Sodium Temperature from 765°F to 1000°F.	D.5
E.1	Differential Fin Element.	E.1

DYNAMIC DIGITAL COMPUTER SIMULATION
OF THE FFTF DUMP HEAT EXCHANGER

R. N. Madsen

1. INTRODUCTION

Digital computer simulation of the transient performance of the heat transport system, including the dump heat exchanger (DHX), is an essential part of the FFTF analysis program. These simulations are being utilized for safety, component design and process control evaluations. This report presents the results of an investigation into the complex problem of determining an adequate digital computer model to represent the extended-surface, sodium to air dump heat exchanger.

Studies^(1,2) performed during conceptual design determined the superiority of a cross-flow sodium-to-air heat dump for the FFTF. Cross-flow heat exchangers are, however, difficult to analyze. Hopkinson⁽³⁾ has shown that cross-flow heat exchangers may be adequately represented by concentric-tube counter-current simulations. Myers, et al.,⁽⁴⁾ have reported some solutions to the dynamics of cross-flow heat exchangers. While these are excellent starting points, no results, or even performance data, were found by which the determination of adequate DHX simulation might be made. Analog-hybrid computer simulations during conceptual design⁽⁵⁾ had included a simple DHX simulation. The purpose of this study was thus two-fold: first, the development of a means for evaluating the accuracy of transient DHX simulations; secondly, the determination of a simple yet adequate DHX simulation for use in systems analysis.

To provide the means for evaluating DHX simulations, since it was not desirable or expedient to develop an original computer code, available methods and computer codes were investigated for applicability. The result was the selection of the Thermal Analyzer Program TAP,⁽⁶⁾ which was used to develop a very detailed reference simulation of a cross-flow exchanger.

Simplified models suitable for inclusion in the simulation of large systems and consisting of numerical approximations of a relatively few differential equations, were then developed for solution with the dynamic analysis code, DYNASYAR.⁽⁷⁾ These simple models were compared with the detailed reference simulation, and the development of an idealized but adequate DHX model was realized.

The following sections document the development of the reference TAP model, the various simplified models investigated, refinements developed to improve these simulations, and typical results.

2. SUMMARY

The capability for accurately determining the transient temperature response of the dump heat exchanger (DHX) is essential for an adequate simulation of the heat transport system. A study has been made in which a detailed, complex simulation of the DHX was developed to serve as a reference model and then a simple, yet adequate, idealized model was developed for inclusion in the heat transport system digital simulation.

The reference simulation was developed by using the computer code, TAP, which uses about 300 nodes, or equivalent integrations, to simulate a typical DHX tube of n passes. On the basis of several tests, coupled with the use of as much detail as possible, this reference TAP simulation was considered sufficient in the absence of any performance data for determining the validity of more simple DHX simulations. These simplified DHX simulations, suitable for inclusion in the simulation of the entire heat transport system, were developed for solution with the dynamic analysis code, DYNASYAR. For these models, the effects of various node meshes and such approximations as average-temperature-based calculations were studied in detail. Because the preliminary transient results of the simple DHX models differed significantly from the TAP reference, a detailed investigation of transient and steady state fin efficiency was made and is presented in detail. From this investigation, the concept of fin efficiency was found to be not applicable to transient heat transfer calculations in extended surfaces. Instead of using fin efficiency, it was necessary to simulate the extended surface explicitly.

The idealized, yet adequate model of the DHX resulting from this study is a concentric-tube, counter-current representation where the total effective length of a DHX tube of n passes is represented by only 4 nodes. The radial heat transfer is represented by 6 nodes: 1 sodium, 1 base-tube, 3 fin and 1 air node. Heat transfer coefficients and thermo-physical properties are calculated on the basis of average temperatures.

3. REFERENCE SIMULATION DEVELOPMENT

The Thermal Analyzer Program (TAP)⁽⁶⁾ generally solves the partial differential equation:

$$\nabla^2 T(x,y,z) + \frac{Q}{K}(x,y,z) = \frac{1}{\alpha} \frac{\partial T}{\partial x}(x,y,z) \quad (3-1)$$

This code, therefore, will handle a variety of problems which can be described in terms of potentials, sources, sinks and storage. Though designed specifically for heat transfer analysis, the TAP code uses the nomenclature of the electrical analogy in which the heat flow and geometry are represented by an equivalent network of admittances, Y , capacitances, C , and source terms, Q . The admittances are analogous to the UA terms, and the capacitances are analogous to the ρVc terms of heat transfer.

The finite difference approximation used by TAP requires that the problem to be analyzed be broken into mesh points, each representing a small volume or node. The code has the capability of using up to 1000 admittances and 500 capacitances in any conceivable network arrangement. Both steady state and transient analyses may be performed. Additional aspects of the TAP code will be described as we discuss the DHX model in detail.

3.1 MODEL DESCRIPTION

The DHX is a multi-pass sodium-to-air finned-tube heat exchanger consisting of four modules for each of the three heat transport circuits. Within each module, only the cross-flow heat exchanger will be of interest here. The physical dimensions of the finned-tube have been tentatively established and are presented in Appendix B. The DHX

consists of heat exchanger modules of 10 tube passes with one row of tubes per pass. (The DHX was altered to eight tube passes part way through this study. This change will be discussed in Section 4.3.3.)

To achieve the greatest possible detail without prohibitive mode size, we assumed that each tube of 10 passes would perform exactly like every other tube. Thus, only one tube needed to be simulated in detail. This also permitted the DHX size to be a variable as the number of tubes in each DHX could be specified. Since one row per pass results in a staggered sodium flow path, we also assumed that the air flow path was staggered, effectively resulting in straightening out the flow path without altering the physical dimensions

FIGURE 3.1. *Actual DHX Tube Arrangement*

3.1.1 Nodalized Representation of the DHX

This single tube, consisting of 10 passes was divided into 30 sections, each section consisting of five feet of finned tubing with 300 fins. A diagram of the tube showing the sectional divisions is given in Figure 3.2. Within each section a centrally located fin was

FIGURE 3.2. *Diagram of DHX Tube Sections*

considered representative of the average thermal processes taking place in that section. This fin was assumed to be symmetrical about a radial plane through its center. This half fin was then represented by eight circumferential metal nodes and two associated fluid nodes. These nodes and sections were connected to form the representation of the actual thermal flow paths. Figure 3.3 presents this arrangement together with the typical network of admittances. The air coolant flow stream was divided into three equal flow paths as is also shown in Figure 3.2. Perfect mixing was assumed at the exit in order to obtain the average exit air temperature.

For simplicity of calculation we initially arranged the fin nodes so that each node had the same surface area in contact with the coolant air stream. During the evaluation of the model we determined that this was not the best node division. (See Section 3.2.1) The nodal division given in Figure 3.3 is that which we finally selected.

Each of the fins, which represent the average thermal behavior of each section, was then scaled to represent the entire section. The TAP code treats solid node temperatures as being the temperature at the physical center of each node, and fluid temperatures as the arithmetic means of the inlet and outlet temperatures to that node.

3.1.2 Calculation of Admittances and Capacitances

Each of the nodes presented in Figure 3.3, and likewise each of the sections, were connected by the proper thermal admittance (reciprocal

FIGURE 3.3. Nodal Arrangement - Typical for Each Section

resistance). These connections are indicated in the figure using the symbol of a resistor.

Since the greatest possible accuracy was desired, the thermophysical properties of each node were computed on the basis of the local temperature at that node, as were all the heat transfer coefficients. Thus, the only averaging used was the assumption that a five foot section of the DHX could be represented by a fin located at the midpoint of that section.

The following is a summary of the manner in which each of the admittances was calculated:

- Connected metal nodes.

These admittances were calculated as:

$$Y_i = \frac{\bar{A}k}{L} \quad (3-2)$$

where $\bar{A}$ = logarithmic mean area (axial conduction along the tube did

not require the use of the log mean area),

k = thermal conductivity,

L = distance between nodes.

Since the fin is not integral with the tube but is to be bonded in some yet undetermined manner, and since the fin material is not identical with the tube material, the definition of the admittance for these nodes was extended. For admittances connecting T_{120+j} and T_{180+j} :

$$Y_i = \left[\frac{L_t}{k_t \bar{A}_t} + R_c + \frac{L_f}{k_f \bar{A}_f} \right]^{-1} \quad (3-3)$$

where R_c = contact resistance, initially assumed to be negligible,
 j = section number.

- Metal nodes connected to sodium coolant.

Metal nodes T_{60+j} and T_{90+j} were connected to the sodium node T_{2j} by admittances

$$Y_i = \left[\frac{L_t}{k_t \bar{A}_t} + \frac{1}{h_i A_i} \right]^{-1} \quad (3-4)$$

where h_i = sodium heat transfer coefficient,

A_i = inside surface area.

- Metal nodes connected to air coolant.

Metal nodes T_{150+j} , T_{180+j} , T_{210+j} , T_{240+j} , and T_{270+j} were connected to the appropriate adjacent air node by admittances

$$Y_i = \left[\frac{L_f}{k_f \bar{A}} + \frac{1}{h_{of} A_o} \right]^{-1} \quad (3-5)$$

where h_{of} = air side heat transfer coefficient,

A_{of} = outside surface area.

- Adjacent air nodes (not in the direction of flow) were connected by admittances

$$Y_i = \left[\frac{k_a A}{L} \right] \quad (3-6)$$

These were currently used only as conduction terms but were included to incorporate the possible simulation of turbulent mixing.

● Flowing Fluids.

To account for flowing fluids, the TAP code assumes that the fluid node temperature is the arithmetic mean of the inlet and outlet temperature. For a given fluid node (see sketch), T is represented exactly by the partial differential equation

$$(\rho V c) \frac{\partial T}{\partial t} + W c \frac{\partial T}{\partial x} = Y_i \nabla^2 T + Q. \quad (3-7)$$

Assuming the temperature distribution is one-dimensional and expressing the spatial derivative in difference form, Equation (3-7) becomes

$$(\rho V c) \frac{dT}{dt} = \sum_{j=1}^n q_j + QV - Wc (T_{m+1} - T_{m-1}) \quad (3-8)$$

where,

Q = heat generation rate (zero in this case)

q = heat flow from or to adjacent nodes.

Other symbols have already been defined.

The TAP code assumes that the right-hand side of Equation (3-8) may be described by a series of terms of the form $Y_i(T_i - T)$, where Y_i is the admittance between the two nodes, T_i , the temperature of the adjacent node, and T , the temperature under consideration. The $\sum q$ terms are already of this form, and QV , though this analysis has none, may be easily made of this form. For each fluid node the inlet temperature, T_{m-1} is known. Thus, if an intermediate temperature is defined midway between each node and the next following node,

$$T_m = 1/2(T_{m-1} + T_{m+1}); \quad (3-9)$$

Then, rearranging,

$$T_{m+1} = 2T_m - T_{m-1}. \quad (3-10)$$

Consequently,

$$-Wc(T_{m+1} - T_{m-1}) = Wc(2T_m - T_{m-1} - T_{m-1}) = 2Wc(T_m - T_{m-1}), \quad (3-11)$$

which is of the required form if

$$Y_i = 2Wc. \quad (3-12)$$

Thus, the fluid nodes are represented in the same manner as other nodes with $Y_i = 2Wc$ and the average temperature computation is done internally in the code.

3.1.3 Heat Transfer Coefficients

The heat transfer coefficients used in Equations (3-4 and 3-5) are functions of both temperature and flow.

Sodium side coefficient

For flow inside tubes the Nusselt number relationship is

$$N_{Nu} = \frac{h_i D_i}{k}. \quad (3-13)$$

The Lyon-Martinelli expression for Nusselt number with the Peclet number corrector as developed by Dwyer⁽⁸⁾ was used with a conservative constant of 5. This empirical expression for the Nusselt number is

$$N_{Nu} = 5.0 + 0.025 (\bar{\psi} N_{Pe})^{0.8} \quad (3-14)$$

where N_{Pe} is the Peclet number. The $\bar{\psi}$ factor as developed by Dwyer⁽⁸⁾ represents the average, effective ratio of the eddy diffusivity of heat

transfer to that for momentum transfer and is defined as

$$\bar{\psi} = 1 - \frac{1.82}{N_{Pr} \left(\frac{\epsilon_m}{\nu} \right)_{\max}^{1.4}} \quad (3-15)$$

where N_{Pr} is the local Prandtl number and $\left(\frac{\epsilon_m}{\nu} \right)_{\max}$ is obtained from Dwyer's work and is a function only of Reynolds number.

The TAP code has the capability for using tabulated data and also includes a section in which arithmetic operations, including manipulation of tabulated data and some logic functions, may be set up for calculating admittances and any other value in the code. By using these provisions and the design data given in Appendix B, the Reynolds Prandtl and Peclet numbers were determined as functions of flow rate and temperature for each section, and Equations (3-13, 3-14, and 3-15) were used to determine the sodium side heat transfer coefficient, h_i , at each location. This coefficient was then used in Equation (3-4) to determine the value of the admittance. No fouling is included for the sodium side of the DHX.

Air side coefficient

Data obtained by Jameson⁽⁹⁾ have been arranged by Kern⁽¹⁰⁾ in the form of a Sieder-Tate factor, j . This coefficient is defined as

$$j = \frac{h_{oc} D_e}{k_a N_{Pr}^{1/3}} \quad (3-16)$$

where h_{oc} = clean air film heat transfer coefficient,

D_e = equivalent diameter,

k_a = thermal conductivity of air,

N_{Pr} = Prandtl Number.

Kern also presents the j -factor as a sole function of Reynolds number

$$j = 13.44 \left(\frac{N_{Re}}{10^3} \right)^{0.72322} \quad (3-17)$$

Thus, the air side film coefficient may be expressed by

$$h_{oc} = \frac{j k N_{Pr}^{1/3}}{D_e} = 13.44 \left(\frac{N_{Re}}{10^3} \right)^{0.72322} k N_{Pr}^{1/3} \left(\frac{1}{D_e} \right) \quad (3-18)$$

The only calculation in the above expression which is not straightforward is the calculation of the equivalent diameter, which is also used in the calculation of Reynolds number. The equivalent diameter is

$$D_e = \frac{2(A_{fin} + A_{tube})}{\pi \chi} \quad (3-19)$$

where A_{fin} is the fin heat transfer area, and A_{tube} is the tube-to-air heat transfer area. This sum is calculated by

$$A_{fin} + A_{tube} = \left[2\pi (r_f^2 - r_t^2) N_f + 2y_b r_f N_f + r_t (1 - 2y_b N_f) \right] \quad (3-20)$$

where N_f is the number of fins per unit length

r_f is the radius of the fin end, and

r_t is the tube outer radius.

The projected perimeter, χ , is calculated by

$$\chi = 2N_f (D_f - D_o) + 2 \quad (3-21)$$

A fouling resistance is also included for the air side:

$$h_{of} = \frac{h_{oc}/R_{of}}{h_{oc} + 1/R_{of}} \quad (3-22)$$

where h_{of} = fouled film coefficient,

R_{of} = fouling resistance.

The necessary operations were coded so that Equation (3-18) was used to determine the heat transfer coefficient as a function of flow rate and temperature for each node for which it is required.

Capacitance - or Heat Storage

The terms $(\rho V c)_i$ in the general heat transfer equations (for example, see Equation (3-7)) are designated capacitances in the TAP code and likewise are based upon local temperature. These heat storage terms are included for all nodes except the air nodes. We neglected the effect of the heat storage in the air because of its effect on the determination of the time increment used by the code for transient calculations.

For transient calculations, the critical time increment is determined by:

$$\Delta t_c = \Delta t_{\min} = \frac{C_i}{\sum Y_{o,i}} \quad (3-20)$$

where C_i is the capacitance of node i , and

$Y_{o,i}$ is the sum of all admittances connecting node i with any other node.

Since C_i for the air nodes is small ($C_s/C_a \doteq 400$ and $C_m/C_a \doteq 1700$) the $\Delta t_{\min}$ was on the order of milliseconds. This would be extremely time consuming. By neglecting these very small capacitances, a much larger critical time increment is calculated and very little accuracy is lost.

3.1.4 Stability

The critical time increment described above is the largest stable

time increment. To prevent incipient instability it is best to use some fraction of this critical time increment. Dray⁽⁶⁾ recommends that $0.8 \Delta t_c$ be used. A check of this model for the range $0.1 \Delta t_c < \Delta t < 0.9 \Delta t_c$ indicated less than 1 °F variation in calculated temperatures and no stability problems using $0.9 \Delta t_c$. Thus, we will use this value for all transient calculations.

3.2 VERIFICATION OF THE REFERENCE SIMULATION

The planned use of this TAP model as a reference simulation made the accuracy of the simulation very essential. Consequently, we used many methods to verify the calculations and results of the TAP simulation. These checks fit into broad divisions, internal and external verification.

3.2.1 Internal Consistency Verification

Heat balances were made in which we compared not only the steady state $Wc(T_{in} - T_{out})$ expressions for each of the fluids but also the heat transfer across each of the fluid-metal boundaries -- the $Y\Delta T$ terms over all the applicable areas. Results of this comparison demonstrated a typical maximum deviation from the arithmetic mean of about 0.4% which is of the same order of magnitude as the uncertainty in the physical properties data.

Insofar as was possible, we verified the calculations of this model with the exact analytical solutions for the equations of heat transfer and temperature distribution. Though the temperature distribution in the DHX is much too complex for explicit analytic solution, it is possible to solve the differential equation for the radial temperature distribution

and heat transfer in a fin with given boundary temperatures. We must necessarily assume that the physical properties are constant. To accomplish this, another version of the model was developed where the calculation of all heat transfer coefficients and thermophysical properties were changed to functions of average temperatures, either average sodium, metal or air. There was almost no change (<0.2%) in the predicted heat transfer duty. Though the more detailed simulation continued as the reference simulation, the model based on average temperature calculations was used for comparison with these exact solutions. The equation for the steady temperature in a fin is a modified Bessel equation of the form

$$y_r \frac{d^2\theta}{dr^2} + \frac{y_r}{r} \frac{d\theta}{dr} + \frac{dy_r}{dr} \frac{d\theta}{dr} - \frac{h}{k} \theta = 0. \quad (3-21)$$

The solution to this equation is given in detail in Appendix C together with the calculation for the heat transfer from the fin, or any portion thereof, to the air. Initial checks indicated that a finer node structure was necessary near the base of the fin in lieu of the equal area arrangement which we had first assumed. Consequently, adjustments in the location of the fin node boundaries were made. Likewise, our comparison of heat transfer from given fin nodes with the exact heat transfer indicated better agreement could be obtained by adjusting the location of the node temperature slightly in from the center of the node. This location was determined for a given node such that

$$2\pi \int_{r_i}^{r_o} r \theta dr = (Y\Delta T)_n; \quad (3-22)$$

that is, so that the heat transfer predicted by the TAP model for node n , $(Y\Delta T)_n$, is as nearly equal to the exact heat transfer, $2\pi \int_{r_i}^{r_o} r \theta dr$, as possible. Since the heat transfer will always be in the same direction, we deemed this to be a valid adjustment to the model and one which did not sacrifice any generality. We have included the final node boundaries and temperature locations in Figure 3.3. A specific solution for a location near the hot end of the DHX is compared with the TAP calculation in Figure 3.4. The comparison indicated in Equation (3-22) is presented in Table 3.1 for this same fin. These typical results demonstrate the excellent agreement we obtained.

Table 3.1 Nodal Heat Transfer Comparison

r_i , in.	r_o , in.	$2\pi h \int_{r_i}^{r_o} r \theta dr$, Btu/hr	$Y T$, Btu/hr
0.95	1.05	11.931	11.871
1.05	1.20	16.391	16.353
1.20	1.40	19.906	19.834
1.40	1.725	31.441	31.574
Total		79.669	79.632

3.2.2 External Verifications

The term *external check* is meant to indicate an independent verification of the model performance. For our case this primarily consisted of using the FFTF steady-state Heat Transport System performance and cost optimizing code, ADOPT,⁽¹¹⁾ to enable an independent audit of steady state performance and heat transfer calculations. The ADOPT code calculates

FIGURE 3.4. Radial Temperature Distribution in a Fin

an average overall U-factor based on film coefficients, thermal conductivity to determine the resistance between adjacent nodes, and then the temperature at each node. These temperatures enable an overall duty to be calculated both by summing the heat transferred across a surface, and by the enthalpy change of the fluid. The TAP reference model does not use fin efficiency but an equivalent value may be determined by using the heat transfer rates. (Appendix C)

A comparison of heat transfer coefficients and fin efficiency was computed in the ADOPT code and our reference TAP model is presented in Table 3.2. These values are based upon the fluid flows and inlet temperatures currently proposed for initial operation and are for identical DHX unit. The ADOPT calculations are for 400 MW_t reactor operation with three loops or 133.33 MW_t per loop. The TAP calculations predict a duty of 131.484 MW_t,

Table 3.2 Comparison of ADOPT and TAP Heat Transfer Coefficients

	<u>ADOPT Value</u>	<u>TAP Value</u>
h_{ic}	3962.	3949.
h_{of}	13.289	13.253
Ω (fin efficiency)	0.584	0.584

less than 1.5% away from the ADOPT value. This result is well within the uncertainties associated with the calculation methods and parameters used.

3.2.3 Check of Transient Calculations

There were few methods by which we could verify the transient calculations of the TAP code. Calculations were performed using various step changes and ramps of both flow rates and inlet temperatures as forcing functions. No anomalous behavior was indicated. Valid new steady state end points were indicated and the effects of fluid dwell time were observed to be accurate. However, the most significant bases for confidence in the transient predictions of this simulation were two-fold. First was the minimization of simplifying approximations. Wherever possible we chose to model features in detail rather than to make simplifying assumptions. The other significant check was the agreement with the results of Myers, et al.⁽⁴⁾ Though their results employed many simplifying assumptions and presented parametric results in a very general way, it was possible to verify that the TAP model was computing transient results in general agreement with their predictions. A detailed description of this check of the TAP reference is presented in Appendix D.

With these verifications we have a high level of confidence in the TAP model and we have attained the first of our two-fold objective -- the determination of a means for evaluating DHX simulation.

4.0 IDEALIZED MODEL DEVELOPMENT

The determination of an idealized yet adequate simulation of the DHX which is suitable for inclusion in heat transport system studies was the second part of our two-fold objective. As indicated previously, Hopkinson⁽³⁾ has conducted investigations of cross-flow heat exchanger dynamics in which frequency responses and transfer functions were studied and compared with actual physical performance data. One of the most applicable conclusions of his work was the determination that a concentric-tube, counter-current approximation to the actual cross-flow exchanger is a valid and satisfactory approximation. This was essentially the method used by Flowers and Gerhardstein⁽⁵⁾ in their early analog-hybrid computer investigation of the dynamics of the FFTF Heat Transport System. In consequence of this previous work, the search for an adequate model began with a counter-current, concentric-tube, three-radial-node model similar to that used by Flowers and Gerhardstein.

4.1 MODEL DESCRIPTION

The concentric-tube true counter-current simplification which was used may be thought of as straightening out the return bends of the heat exchanger and then constraining the outer fluid -- air in our case -- to flow in a true counter-current manner such that the flow area and heat transfer contact area is identical with the actual cross-flow arrangement. The physical dimensions were assumed the same as for the reference TAP simulation and are presented in Appendix B.

Our first assumption was that the concept of fin efficiency could be used to represent the heat transfer processes occurring in the fins. Consequently, three radial, circumferential nodes were assumed, one each for the sodium, metal and air. Ensuing work established the need for alteration of the radial node structure. This change will be discussed later in this section in its actual place in the development of this model. Because of the use of fin efficiency, the metal node temperature calculated was arranged to be the temperature at the outside of the tube and at the base of the fin. This is the temperature on which the fin efficiency calculation is based.

The number of axial, or longitudinal nodes parallel to the direction of flow was a parameter which was investigated. Initially the number of nodes was somewhat arbitrarily set at six.

No plenum effects at the inlet or outlet were considered as these may be added to any simulation and were not of specific interest here. Also, the effect of axial conduction was neglected.

The nodal representation for this six-node model is shown in Figure 4.1. Sodium temperatures are indicated by T_{sj} , lumped metal temperatures by T_{fj} , and the air temperatures by T_{tj} . The weight flow rates for sodium and air are W_s and W_t , respectively.

FIGURE 4.1. 6-Axial Node, 3-Radial Node Model

The positive x-direction is indicated with L_x being the total effective length of the DHX, that is, the number of passes times the effective length per pass. The length per node is ΔL_x . The parameters T_{s1} , T_{t7} , W_s and W_t are the forcing functions and are used as tabulated functions of time.

Application of the one-dimensional heat equation to each of the mesh points indicated in Figure 4.1 results in the following set of equations describing the heat transfer:

Sodium:

$$\left[\rho V \hat{C}\right]_{sj} \frac{dT_{sj}}{dt} = - \left[W \hat{C}\right]_{sj} \frac{\partial T_{sj}}{\partial x} - \left[UA\right]_{sj} \left[T_{sj} - T_{fj}\right] \quad (4-1)$$

for $j = 2, 3, \dots, 7$.

Metal:

$$\left[\rho V \hat{C}\right]_{fj} \frac{dT_{fj}}{dt} = \left[UA\right]_{sj} \left[T_{sj} - T_{fj}\right] - \left[UA\right]_{tj} \left[T_{fj} - T_{tj}\right] \quad (4-2)$$

for $j = 1, 2, \dots, 7$.

Air:

$$\left[\rho V \hat{C}\right]_{tj} \frac{dT_{tj}}{dt} = \left[UA\right]_{tj} \left[T_{fj} - T_{tj}\right] + \left[W \hat{C}\right]_{tj} \frac{\partial T_{tj}}{\partial x} \quad (4-3)$$

for $j = 1, 2, \dots, 6$.

The primes indicate constants which are in terms of unit length or one tube.

4.1.1 Numerical Methods for Calculating Spatial Temperature Derivatives

For solution with the DYNASYAR code we must express the axial temperature gradients, $\frac{\partial T}{\partial x}$ terms, in the above equations by a numerical approximation. A study of Intermediate Heat Exchanger modeling techniques⁽¹²⁾ demonstrated that the most stable technique for that simulation was an alternating central and three-point backward difference combination which has been referred to as a "mixed" difference method. This method necessitates the use of an even number of nodes. By using this approximation, Equations (4-1, 4-2, and 4-3) become, respectively,

$$\frac{dT_{sj}}{dt} = - \left[\frac{W'}{\rho V'} \right]_{sj} \frac{\Delta T_{sj}}{2\Delta L_x} - \left[\frac{UA'}{\rho V \dot{C}} \right]_{sj} [T_{sj} - T_{fj}] \quad (4-4)$$

$$\text{for } j = 2, 3, \dots, 7$$

$$\frac{dT_{fj}}{dt} = \left[\frac{UA'}{\rho V \dot{C}} \right]_{sj} [T_{fj} - T_{tj}] + \left[\frac{UA'}{\rho V \dot{C}} \right]_{tj} [T_{fj} - T_{tj}] \quad (4-5)$$

$$\text{for } j = 1, 2, \dots, 7$$

$$\frac{dT_{tj}}{dt} = \left[\frac{UA'}{\rho V \dot{C}} \right]_{tj} [T_{fj} - T_{tj}] + \left[\frac{W'}{\rho V'} \right]_{tj} \frac{\Delta T_{tj}}{2\Delta L_x} \quad (4-6)$$

$$\text{for } j = 1, 2, \dots, 6$$

In these equations the ΔT_j expressions are

$$\Delta T_{sj} = T_{s(j+1)} - T_{s(j-1)} \quad \text{for } j = 2, 4, 6 \quad (4-7)$$

$$\Delta T_{sj} = 3T_{sj} - 4T_{s(j-1)} + T_{s(j-2)} \quad \text{for } j = 3, 5, 7 \quad (4-8)$$

$$\Delta T_{tj} = T_{t(j+1)} - T_{t(j-1)} \quad \text{for } j = 2, 4, 6 \quad (4-9)$$

$$\Delta T_{tj} = -3T_{tj} + 4T_{t(j+1)} - T_{t(j+2)} \quad \text{for } j = 1, 3, 5 \quad (4-10)$$

In the above $\Delta L_x/n$ where n = number of nodes and (as previously mentioned) L_x is the total effective length. A'_s and V'_s are the sodium-side heat transfer area and sodium volume per unit length, respectively; A'_t and V'_t are the same quantities for the air-side, and V'_f the total metal volume per unit length, both fins and tube.

Once the coefficients are explicitly determined, the above Equations (4-4 through 4-6), are directly solvable using the dynamic analysis code, DYNASYAR. However, we caution that these equations are not the final result, but were expanded in the final DHX simulation. We have presented this much detail because it is still generally applicable and will assist in making clear the reasons why this simulation was not completely adequate.

4.1.2 Calculation of Constants and Heat Transfer Coefficients

The constants which are used in the foregoing equations are calculated in the following manner.

- A'_{sj} Heat transfer area per unit length, inside tubes.

$$A'_{sj} = \pi N_t D_i \quad (4-11)$$

where N_t = Number of tubes per pass.

- V'_{sj} Volume per unit length, sodium side.

$$V'_{sj} = \frac{\pi}{4} D_i^2 N_t \quad (4-12)$$

- V'_{fj} Tube and fin metal volume per unit length.

$$V'_{fj} = N_t \left[V'_{\text{tube}} + V'_{\text{fin}} \right] \quad (4-13)$$

$$= N_t \left[\frac{\pi}{4} (D_o^2 - D_i^2) + N_f 2y_b \frac{\pi}{4} (D_f^2 - D_o^2) \right] \quad (4-14)$$

where D_o = outside tube diameter, ft.,

D_i = inside tube diameter, ft.,

D_f = fin diameter, ft., and

N_f = number of fins per foot of length.

- A'_{tj} Air-side heat transfer area per unit length.

$$A'_{tj} = N_t \left[A'_{\text{fin}} + A'_{\text{tube}} \right]$$

$$= N_t \left[N_f \frac{\pi}{2} (D_f^2 - D_o^2) + \pi y_b N_f D_f + \pi D_o (1 - N_f y_b) \right] \quad (4-15)$$

- V'_{tj} Volume per unit length, air side.

$$V'_{tj} = N_t \left[V'_{\text{hex}} - V'_{\text{tube}} - V'_{\text{fin}} \right]$$

where V'_{hex} is the volume of the assumed concentric tube which defines the actual air flow volume associated with a given tube.

$$V'_{tj} = N_t \left[0.866 D_p^2 - \frac{\pi}{4} D_o^2 - N_f y_b \frac{\pi}{4} (D_f^2 - D_o^2) \right] \quad (4-16)$$

where D_p is the pitch diameter, ft.

The overall heat transfer coefficient, U_{sj} , includes the sodium-side film coefficient, h_{sj} , and the resistance due to the tube out to the location of the metal node temperature at the base of the fin. This coefficient is given by

$$U_{sj} = \left[\frac{1}{h_{ij}} + \frac{D_i \ln(D_o/D_i)}{2k_t} \right]^{-1} \quad (4-17)$$

The calculation of h_{ij} is performed exactly as for the TAP reference model as given in Section 3.1.3, Equation (3-13).

The metal-to-air heat transfer coefficient U_{tj} , for the three-radial node model, includes the use of the concept of fin efficiency, Ω (Appendix C). The overall heat transfer coefficient is

$$U_{tj} = \left[\Omega A_{fin} + A_{tube} \right]_j \frac{h_{ofj}}{A_{tj}} \quad (4-18)$$

As was the case with the inside film coefficient, this outside film coefficient, h_{ofj} , is computed exactly as for the cross-flow exchanger as given in Section 3.1.3, Equation (3-22). The values of fin efficiency, Ω , which were initially used were developed by Gardner⁽¹³⁾ and are also found in Kern.⁽¹⁰⁾ As given in these references, Ω is a function of fin type, geometry and N where $N = \sqrt{\frac{h}{ky_b}}$.

4.2 PARAMETRIC EXAMINATION OF THE THREE-RADIAL-NODE MODEL

Following a detailed verification of coefficient calculations and the comparison of steady state performance predictions with the reference TAP model, a program was conducted to investigate the effect of various model parameters on the steady-state and transient results. The parameters of particular interest were those which determine the complexity of the model, namely the number of axial nodes and the detail associated with the calculation of coefficients and thermophysical properties. We investigated 3-radial-node models with 12, the initial 6, and 4 axial nodes. The 12

nodes were considered a sufficiently large number to determine whether there was an improvement over the initial 6. With "mixed differences" requiring an even number of nodes, and recognizing 2 nodes were insufficient to determine a realistic axial temperature gradient, 4 axial-nodes was the minimum investigated. Coefficient and thermophysical properties were calculated on the basis of individual node temperatures and on the basis of average temperatures; also we examined a combination of average-temperature-based heat transfer coefficients and node-temperature dependent properties.

This investigation demonstrated that the least complicated model, the 4-axial, 3-radial-node model with average-temperature-based coefficients and properties, predicted results for various steady-state and transient conditions which were nearly identical -- within the accuracy of the methods used -- to the models with the greater number of axial nodes and more complicated property and coefficient calculations. Steady-state predictions of the temperature distribution with distance along the sodium flow path were also nearly identical for cases of balanced and moderately unbalanced ($\sim 3:1$) flow rates. For cases of extremely unbalanced ($\sim 50:1$) flow rates, there existed an oscillation which was smaller for the 12 node model and which became more pronounced with a decreasing number of nodes. However, the most essential predictions, fluid exit temperatures of sodium and air, were almost completely unaffected by the number of nodes assumed. Transient response to sudden ramps of temperature, air flow, or sodium flow, and some combinations of these parameters were nearly identical. However, as we have previously alluded, the most significant discovery was

that none of these models predicted results which agreed with the reference simulation for the transient behavior of the DHX, particularly in the first 10 sec of a transient. It is not practical to present a complete documentation for all the cases studied; one typical result is presented in Figure 4.2. In this figure the sodium and air DHX exit temperatures are compared for a simulated loss-of-electrical-power incident. The exit air temperature in the 3-node simulation is observed to have a much more rapid rate of rise and levels out at the new steady state value much sooner than for the reference simulation. This type of behavior was typical of all the 3-radial node models studied. Consequently, an explanation and a more correct method of DHX simulation was sought.

4.3 TRANSIENT CALCULATIONS FOR EXTENDED SURFACES

The complex problem of representing transient heat transfer in the fin was thoroughly investigated as a result of the differences which existed between the reference simulation and the simpler models. As has previously been alluded in section 4.1, this investigation led to the determination that the values of fin efficiency which were developed by Gardner⁽¹³⁾ and which enabled us to use only 3 radial nodes were applicable only to steady state calculations of heat transfer.

To determine the true nature of fin efficiency as a function of time, we used the definition of fin efficiency, $\Omega = q/q_0^*$ to determine the fin efficiency for each section of the TAP model. Although Ω is not used in

*See Appendix C for the derivation and discussion.

FIGURE 4.2. Comparison of Predicted Fluid Exit Temperatures for 3-Radial Node Model vs. TAP Reference

the TAP calculations, this enabled us to compare the transient behavior of Ω with the tabulated data used in the 3-radial-node models. Some typical results of these calculations are shown in Figure 4.3** for a loss-of-electrical-power incident occurring at $t = 3$ sec. The inadequacy of the tabulated data is immediately apparent. Not only is Ω a much different function of time than is obtained from the tabulated data, but fin efficiency during a transient is also very location dependent as may be observed from the figure.

It is important to emphasize that the values developed by Gardner are perfectly valid at steady state. The agreement prior to the start of the transient is evident in Figure 4.3. Likewise, agreement also exists for the steady state end point of this transient. The fin efficiency calculations for all sections approach the indicated steady state value of 0.867.

The reasons for these differences may be understood if we examine the basic equation for time-dependent heat transfer in a fin. The equation for the temperature at steady state is given as Equation (3-21). However, the time-dependent equation requires another term. The complete equation, thus:

$$y_r \frac{d^2\theta}{dr^2} + \frac{y_r}{r} \frac{d\theta}{dr} \frac{dy_r}{dr} \frac{d\theta}{dr} - \frac{h}{k} \theta = \frac{\rho c y_r}{k} \frac{d\theta}{dt} \quad (4-19)$$

The derivation of this equation for fins of constant thickness is given in Appendix E together with a detailed development of the calculation of Ω .

For time dependent calculations Ω becomes not just the straightforward function of geometry and $N = \frac{h}{ky_b}$ but is:

** Refer to Figure 3.2 for section identification. The results in Figure 4.3 are for an 8 pass DHX.

FIGURE 4.3. Ω vs Time Based on Tabulated Values Compared With Selected Results From TAP Calculations For a Typical Loss of Power Transient

$$\Omega = \frac{4\pi y_b}{hA_f(T_b - T_a)} \left\{ r_1 k \left(\frac{d\theta}{dr} \right)_{r=r_1} - \int_{r_1}^{r_2} \rho c r \frac{d\theta}{dt} dr \right\} \quad (4-20)$$

where θ = temperature difference, $T - T_a$, at any point r in the fin

y_r = one-half the fin thickness

h = fin-to-air heat transfer coefficient

A_f = surface area of the fin for heat transfer

r_1 = radius at base of fin

r_2 = radius at end of fin + y_b

T_b = temperature at base of fin

T_a = air temperature

ρ, c, k = thermophysical properties of the fin.

At steady state the integral term vanishes as $\frac{d\theta}{dt} = 0$ and the slope $\left(\frac{d\theta}{dr} \right)$ at rate $r=r_1$ is a constant for every point in the DHX. Therefore, at steady state Equation (4-20) reduces to the expression for fin efficiency tabulated by Gardner⁽¹³⁾. (Refer also to the discussion of steady state Ω in Appendix C.)

4.3.1 Transient Fin Efficiency

Having demonstrated the inadequacy of the 3-radial node model using tabulated fin efficiency, and having determined the cause, the application of Equation (4-20) to the DHX simulation was the logical next step. Difficulties were encountered in directly applying these results to the DHX simulation.

One is tempted to immediately think that because the variable time appears explicitly only in one of the two terms, Equation (4-20) consists

of a steady-state portion minus a transient correction which diminishes with increasing time. Such an assumption is not valid because the other term, $\left(\frac{d\theta}{dr}\right)_{r=r_1}$, contains time implicitly due to the temperature difference at the base of the fin being a function of time. Thus, this derivative is also changing.

Examination of Figure 4.3 will disclose a previously mentioned and important factor: Ω is not only a function of time, but also of position in the DHX. In spite of this variation of Ω with position, there was the possibility that an "average" Ω could be found which would preserve the simplicity yet still produce sufficiently valid results. Assuming that such an average might be obtainable, we examined methods of calculating Ω as given in Equation (4-20).

To permit calculation of $\left(\frac{d\theta}{dr}\right)_{r=r_1}$, we determined three additional temperature points radially outward halfway through the DHX. Numerical approximations to $\left(\frac{d\theta}{dr}\right)_{r=r_1}$ were then made. The second portion of the expression poses a different problem: $\frac{d\theta}{dt}$ is not constant with position, just as $\frac{d\theta}{dr}$ is not constant with time. It is really impossible to truly separate these variables.

If there were a typical location where $\frac{d\theta}{dt}$ was representative of the fin, then $\frac{d\theta}{dt}$ could be removed from the integral and the second term within the brackets would be:

$$\frac{\rho c V}{4\pi h} \frac{d\theta}{dt} \quad (4-21)$$

(Feedback is a consideration here as $\frac{d\theta}{dt}$ is being used in an expression to compute $\frac{d\theta}{dt}$.)

These attempts to determine a transient Ω were not successful. This was primarily due to stability considerations in the calculation of derivatives immediately following a disturbance to the system and also due to the inaccuracies associated with the many assumptions which have been described.

4.3.2 Model Description for 6-Radial Nodes

The alternative to using fin efficiency and just the 3 radial nodes is to model the fins explicitly by means of additional radial nodes. To determine the temperature distribution curve for the fins, at least 3 radial nodes were necessary. So that the number of nodes would be kept at a minimum, we initially assumed this minimum number. The number of axial nodes remained at the previously determined 4. Subsequent investigations have verified that these 6-radial nodes are sufficient for adequate representation of the DHX. This model is essentially the previously described counter-current, concentric-tube model with the fins described by 3 nodes, and one node each for the sodium, the base tube and the air. A depiction of the nodal arrangement of this model is given in Figure 4.4. The nomenclature is identical with that given Section 4.1, with the extension of 3 additional subscripts. All three fin nodes and the tube nodes are connected with the air nodes.

Applying the one dimensional heat equation to each of these mesh points, and using the previously described "mixed" difference technique for calculating the axial temperature gradients, we obtain the following equations for each mesh plane, j :

FIGURE 4.4. 4-Axial, 6-Radial Node Model

Sodium:

$$\frac{dT_{sj}}{dt} = - \left[\frac{W}{\rho V \hat{C}} \right]_s \frac{\Delta T_{sj}}{2\Delta L_x} - \left[\frac{UA}{\rho V \hat{C}} \right]_s [T_{sj} - T_{fj}] \quad (4-22)$$

for $j = 2, \dots, 5$.Base Tube:

$$\begin{aligned} \frac{dT_{fj}}{dt} = & \left[\frac{UA}{\rho V \hat{C}} \right]_s [T_{sj} - T_{fj}] - \left[\frac{UA}{\rho V \hat{C}} \right]_{ft} [T_{fj} - T_{tj}] \\ & - \left[\frac{UA}{\rho V \hat{C}} \right]_{fb} [T_{bj} - T_{tj}] \end{aligned}$$

for $j = 1, \dots, 5$. (4-23)Fin:

$$\begin{aligned} \frac{dT_{bj}}{dt} = & \left[\frac{UA}{\rho V \hat{C}} \right]_{fb} [T_{fj} - T_{bj}] - \left[\frac{UA}{\rho V \hat{C}} \right]_{bc} [T_{bj} - T_{cj}] \\ & - \left[\frac{UA}{\rho V \hat{C}} \right]_{bt} [T_{bj} - T_{tj}] \end{aligned} \quad (4-25)$$

for $j = 1, \dots, 5$.

$$\begin{aligned} \frac{dT_{cj}}{dt} = & \left[\frac{UA}{\rho V \hat{C}} \right]_{bc} [T_{bj} - T_{cj}] - \left[\frac{UA}{\rho V \hat{C}} \right]_{cd} [T_{cj} - T_{dj}] \\ & - \left[\frac{UA}{\rho V \hat{C}} \right]_{ct} [T_{cj} - T_{tj}] \end{aligned} \quad (4-25)$$

for $j = 1, \dots, 5$

$$\frac{dT_{dj}}{dt} = \left[\frac{UA}{\rho V \hat{C}} \right]_{cd} [T_{cj} - T_{dj}] - \left[\frac{UA}{\rho V \hat{C}} \right]_{dt} [T_{dj} - T_{tj}] \quad (4-26)$$

for $j = 1, \dots, 5$.

Air:

$$\begin{aligned}
 \frac{dT_{tj}}{dt} = & \left[\frac{UA}{\rho V \hat{C}} \right]_{ft} [T_{fj} - T_{tj}] + \left[\frac{UA}{\rho V \hat{C}} \right]_{bt} [T_{bj} - T_{tj}] \\
 & + \left[\frac{UA}{\rho V \hat{C}} \right]_{ct} [T_{cj} - T_{tj}] + \left[\frac{UA}{\rho V \hat{C}} \right]_{dt} [T_{dj} - T_{tj}] \\
 & + \left[\frac{W}{\rho V} \right]_{t} \frac{\Delta T_{tj}}{2\Delta L_x}
 \end{aligned} \tag{4-27}$$

for $j = 1, \dots, 4$.

In these equations the ΔT_j expressions are given by:

$$\Delta T_{sj} = T_{s(j+1)} - T_{s(j-1)} \quad \text{for } j = 2, 4 \tag{4-28}$$

$$\Delta T_{sj} = 3T_{sj} - 4T_{s(j-1)} + T_{s(j-2)} \quad \text{for } j = 3, 5 \tag{4-29}$$

$$\Delta T_{tj} = T_{t(j+1)} - T_{t(j-1)} \quad \text{for } j = 2, 4 \tag{4-30}$$

$$\Delta T_{tj} = -3T_{tj} + 4T_{t(j+1)} - T_{t(j+2)} \quad \text{for } j = 3, 5 \tag{4-31}$$

Since we have determined the adequacy of average-temperature-based properties and coefficients, no distinction of nodes is made for the constants in the above equations. The constants and the heat transfer coefficients are calculated as described in Section 4.1.2 with the exception that the surface area between the fin and the air nodes and the fin metal volume is now divided according to the nodal dimensions shown in Figure 4.5.

FIGURE 4.5. Node Dimensions for 6-Radial Node Model

The node temperature locations for the outer two nodes are determined in the same manner as is discussed in Section 3.2.1 for the TAP simulation. Table 4.1 compares the exact heat transfer for a typical steady state case with the heat transfer calculated on the basis of the above node dimensions and temperature locations. (The values in Table 4.1 are for $x=0$, where x is the distance from the sodium inlet; the values in Table 3.1 are for $x=7.5$.)

Table 4.1 Nodal Heat Transfer Comparison

r_i , in.	r_o , in.	$2\pi h \int_{r_i}^{r_o} r_o dr$, Btu/hr	$hA_j \Delta T_j$, Btu/hr
0.95	1.05	12.244	12.220
1.05	1.35	32.193	32.066
1.35	1.725	36.980	37.459
Total		81.417	81.745

The effect of increasing the number of axial nodes could not be inferred completely from the previously reported study of the 3-radial

node models, though this gave good reason to expect the effect would not be great. Consequently, results were obtained for an 8-axial node model.

Predictions of fluid exit temperatures using both the 4- and the 8-axial node, 6-radial node models are compared in Figure 4.6 with the predictions of the TAP reference simulation. These results should be contrasted with those for the 3-radial node model as given in Figure 4.2 for the identical transient incident. The agreement with the TAP reference is excellent and well within the confidence level of the data and calculation procedures used.

4.3.3 Refinements to the DHX Simulations

In the latter stages of this investigation the basic design of the DHX was altered slightly and new initial temperature and flow conditions were determined. The only change which had to be made to the DHX simulations was in the number of tube passes. This necessitated removing the last six sections of the TAP model (Figure 3.2) along with a corresponding revision of some of the calculations programmed. The change of the simple DYNASYAR simulations required only a very minor change of the effective length constant. The 8-pass arrangement is the reference standard at the present time.

A second refinement made a significant change in the manner in which the air temperatures are calculated in the DYNASYAR models. A somewhat standard assumption in many heat exchanger analyses involving liquid-to-air heat transfer is that the heat storage capacity ($\rho V c$ product) of the air is negligible compared with the corresponding terms for the liquid and the

FIGURE 4.6. TAP Reference Model Compared with 4- and 8-Axial Node, 6-Radial Node Models for a Typical Loss-of-Power Transient

metal. This makes the left hand side of Equation (4-27) identically zero, and the resulting equation may be solved algebraically. This procedure is used in the TAP reference simulation but since the DYNASYAR code cannot handle algebraic feedback or initial conditions of algebraic functions, the differential equation was initially used.

During the testing of the 6-radial node, 4 axial node model, an in-ordinate calculation time for some transients was observed. In an effort to improve speed of the simulation a "pseudo-algebraic" method for the calculation of the air node temperatures as successfully employed. This method consists of using the algebraic equation to compute the air node temperatures and then using a simple lag differential equation with small time constant to impose an initial condition and to prevent algebraic feedback. This feature is essentially the differential equation

$$\tau \frac{dT_2}{dt} + T_2 = T_1 \quad (4-32)$$

where T_1 is the solution of the algebraic equation and T_2 is the solution of the differential equation. For a sufficiently small τ , T_2 should be close enough in value to T_1 to prevent loss of accuracy and yet not be as limiting a time constant as that imposed by the very small air heat capacity.

The results of this method are presented in Figure 4.7 in which the explicit integration method is compared with the pseudo-algebraic method for $\tau = 0.1, 0.5$ and 1.0 . A value of $\tau = 0.1$ yields results that are identical with the explicit integration with ± 0.5 °F at nearly every point yet is computationally twice as fast.

FIGURE 4.7. Algebraic Air Node Calculations for Various Lag Time Constants vs. Explicit Integration

5. DISCUSSION

The concentric-tube, counter-current DHX representation developed herein, in which the typical thermal behavior of one tube of n-passes is represented by 4-axial and 6-radial nodes and in which all heat transfer coefficients and thermophysical properties are based on average properties, is an adequate, idealized model of the DHX. The predictions of this simple model compare excellently with those of the reference TAP simulation and are well within the accuracy of the data and correlations used.

In addition to the simulated loss-of-electrical-power transient which has been used to illustrate many of the conclusions obtained, a number of other transient investigations were conducted. Among these was a program of test cases which was designed to reveal the performance of the DHX model over an extended range of hypothetical conditions. Two of these test cases are presented in Figures 5.1 and 5.2. In these figures the calculated fluid exit temperatures of the TAP reference simulation are compared with those of the 6-radial node model -- the one finally adopted for simulation work. An arbitrary linear ramp of air flow rate from 100% to 10% in 1 second is shown in Figure 5.1 and a similar ramp of sodium flow rate, only occurring over a period of 10 sec, is presented in Figure 5.2. All other inlet conditions are constant for the duration of the transient. These results further substantiate the adequacy of this simple representation of the dump heat exchanger.

FIGURE 5.1. Test Case - Ramp of Air Flow to 10% in 1 Sec.

FIGURE 5.2. Test Case - Ramp of Sodium Flow to 10% in 10 Sec.

NOMENCLATURE

<u>Symbol</u>	<u>Description</u>	<u>Units</u>
A	Area	ft ²
$\bar{A}$	Logarithmic Mean Area	ft ²
C	Capacitance	Btu/°F
c	Specific Heat	Btu/lbm°F
D	Diameter	ft
h	Heat Transfer Film Coefficient	Btu/hr-ft ² -°F
I	Modified Bessel Function	
i	$\sqrt{-1}$	
J	Bessel Function	
j	Sieder-Tate Heat Transfer Factor	
K	Modified Bessel Function	
k	Thermal Conductivity	Btu/hr-ft-°F
L	Length	ft
N	Bessel Equation Parameter; Numerical Constant	
n	Bessel Equation Order	
N_{Pe}	Peclet Number	
N_{Pr}	Prandtl Number	
N_{Re}	Reynolds Number	
Q	Heat Generation Rate	Btu/hr-ft ³
q	Heat Flux	Btu/hr-ft ²
q'	Total Heat Transfer Rate	Btu/hr
R	Thermal Resistance	hr-ft. ² -°F/Btu

<u>Symbol</u>	<u>Description</u>	<u>Units</u>
r	Radius	ft
T	Temperature	°F
t	Time	sec
U	Heat Transfer Coefficient	Btu/hr-ft ² °F
V	Volume	ft ³
v	Velocity	ft/sec
W	Weight Flow Rate	lb/hr
Y	Admittance	Btu/hr°F
y_b	One-Half Fin Thickness	ft
<u>Greek Letters</u>		
θ	Temperature Difference	°F
Λ	Derivative Function	
ρ	Density	lbm/ft ³
χ	Projected Perimeter	ft/ft
ψ	Average value of the ratio of diffusivity for heat transfer to diffusivity for momentum transfer.	
Ω	Fin Efficiency	
<u>Subscripts</u>		
a	air	
b	base of fin, fin node	
c	clean, fin node	
d	fin node	

Subscripts (continued)

e	equivalent
f	fin, fouled, node including base fin temperature
i	inside
i, j, m, n	indices
m	metal
o	outside
s	sodium
t	tertiary or air, tube

APPENDIX ATHERMOPHYSICAL PROPERTIES

The thermal properties of both sodium and air which are used in simulations performed are given in Figures A-1 and A-2. Though the data are presented as continuous functions here, they are included in the simulations as tabular data for which linear interpolation is used. There are sufficient points in the tables so that the linear interpolation introduces little error.

The sources of data for these curves are as follows:

Figure A-1. Sodium Properties

All data: *Liquid Metals Handbook, Sodium-NaK Supplement*,
AEC, June 30, 1967.

Figure A-2. Air Properties

Specific Heat: Perry, Chilton, and Kirkpatrick. *Perry's Chemical Engineers' Handbook*, 4th ed., p. 3-206. McGraw-Hill Book Company, Inc., New York.

Viscosity: Same as Specific Heat.

Thermal Conductivity: Kreith, *Principles of Heat Transfer*, p. 535. International Textbook Company, Scranton, Pa., 1958. (Sources given on p. 539.)

Density: Calculated from the ideal gas law using $R = 53.3 \text{ ft}^2/\text{R}$ and $P_{\text{atm.}} = 14.4 \text{ psia}$.

Prandtl Number: Calculated from other values given in this table.

FIGURE A.1. Thermophysical Properties of Liquid Sodium

FIGURE A.2. Thermophysical Properties of Dry Air

APPENDIX BDIMENSIONAL DATA FOR THE DUMP HEAT EXCHANGER

The following data were used to define the physical characteristics of the Dump Heat Exchanger which was simulated:

Outside Tube Diameter, in.	1.9
Tube Wall Thickness, in.	0.109
Height of Fins, in.	0.75
Outside Diameter of Fins, in.	3.4
Tube Pitch, Triangular, in.	3.775
Fin Thickness, in.	0.050
Fin Pitch, Fins Per Foot	60.
Total Tube Length, ft.	18.
Effective Tube Length, ft.	15.
Fouling Factor, Sodium Side, hr. ft. ² /Btu	0
Fouling Factor, Air Side, hr. ft. ² /Btu	0.003
Number of Passes	10 and 8
Number of Tubes Per Pass	variable

APPENDIX CEXACT CALCULATION OF STEADY STATE FIN TEMPERATURE HEAT TRANSFER
AND FIN EFFICIENCY

The explicit equation for the steady state temperature at any point in a fin is given by Equation (3-21) which we repeat here for convenience:

$$y_r \frac{d^2\theta}{dr^2} + \frac{y_r}{r} \frac{d\theta}{dr} + \frac{dy_r}{dr} \frac{d\theta}{dr} - \frac{h}{k} \theta = 0 \quad (C-1)$$

where θ = temperature difference, $T - T_a$, at any point r in the fin,

y_r = one-half the fin thickness at location r ,

h = fin-to-air heat transfer coefficient,

k = fin thermal conductivity.

For the case where fins are of constant thickness, $y_r = y_b = \text{constant}$ and it follows that $\frac{dy_r}{dr} = 0$. With this term eliminated Equation (C-1) is a modified Bessel equation for which the standard form is

$$\frac{d^2\theta}{dr^2} + \frac{1}{r} \frac{d\theta}{dr} + \left(N^2 i^2 - \frac{n^2}{r^2} \right) \theta = 0, \quad (C-2)$$

where $i = \sqrt{-1}$ and

n = order of the equation.

For our particular case Equation (C-1) is of the form given in Equation (C-2) if we set $N^2 = \frac{h}{ky_b}$ and the order $n = 0$. The solution of this equation is of the form

$$\theta = c_1 J_0(iNr) + c_2 Y_0(iNr)$$

or

$$\theta = c_1 J_0(Nr) + c_2 K_0(Nr). \quad (C-3)$$

For $\theta_o = T_b - T_a$

where T_b = temperature at base of fin, and

T_a = air temperature,

and $\theta = T - T_a$,

r_1 = radius at base of fin where $T = T_b$, and

r_2 = outside fin radius $+y_b$,

then

$$\theta = \theta_o \frac{I_0(Nr)K_1(Nr_2) + I_1(Nr_2)K_0(Nr)}{I_0(Nr_1)K_1(Nr_2) + I_1(Nr_2)K_0(Nr_1)}. \quad (C-4)$$

Equation (C-4) is used to determine the temperature at any point radially out the fin. We have made the tacit assumption that average conductivity and heat transfer coefficients will be used and that there is no temperature gradient across the width of the fin.

The heat transfer from one side of a fin is given by

$$q = 2\pi h \int_{r_i}^{r_j} r \theta dr. \quad (C-5)$$

The above integral may be evaluated by observing that Equation (C-2) may be alternately written as

$$\frac{d}{dr} \left[r \frac{d\theta}{dr} \right] = \frac{h}{ky_b} r \theta. \quad (C-6)$$

Solving for $r\theta$ and differentiating gives

$$r\theta dr = \frac{ky_b}{h} \frac{d}{dr} \left[r \frac{d\theta}{dr} \right] dr. \quad (C-7)$$

On substitution in Equation (C-5) the integral may be immediately evaluated resulting in

$$q' = \frac{2\pi h}{N^2} \left(r \frac{d}{dr} \right)_{r_i}^{r_j} \quad (C-8)$$

The derivative in Equation (C-8) may be evaluated by differentiating Equation (C-4). Then, if we define a function $\Lambda = f(r)$, such that

$$\Lambda(r) = \frac{1}{N\theta_o} \left(\frac{d\theta}{dr} \right) = \frac{I_1(Nr)K_1(Nr_2) - I_1(Nr_2)K_1(Nr)}{I_0(Nr_1)K_1(Nr_2) + I_1(Nr_2)K_0(Nr_1)}, \quad (C-9)$$

then the heat transfer may be computed by

$$q' = 2\pi h \frac{\theta_o}{N} \left(r \Lambda(r) \right)_{r_i}^{r_j} \quad (C-10)$$

For $r_i = r_1$ and $r_j = r_2$, the above equation reduces to

$$q' = -2\pi h \frac{\theta_o}{N} r_1 \Lambda(r_1) \quad (C-11)$$

which gives the total heat transfer from one side of the fin. Equation (C-10) may be used to determine the heat transfer between any two radii.

Fin efficiency is a method of avoiding the above complex solution for heat transfer at steady state. Since $\Lambda(r_1)$ is constant for steady state, the heat transfer becomes a function only of h , θ_o , N and the geometry of the fin. Let us define a quantity q'_o such that

$$q'_o = 2\pi h (r_2^2 - r_1^2) \theta_o. \quad (C-12)$$

This quantity represents the heat transfer which could occur from an isothermal fin of temperature $T=T_b$. The ratio q'/q'_o is defined as fin

efficiency Ω , and is solely a function of h , N and the geometry of steady state. Gardner⁽¹³⁾ presents tabulations of Ω for various geometries and fin types.

APPENDIX DVERIFICATION OF TAP MODEL TRANSIENT CALCULATIONS BY THE METHODS
OF MYERS, ET AL.

In an effort to extend the understanding of cross-flow heat exchangers, Myers, Mitchell and Norman⁽⁴⁾ at the University of Wisconsin performed an investigation of the transient response of such exchangers with one fluid mixed. (Hereafter for simplicity, this work will be referred to by its report number, MM-3.) In their book on compact heat exchangers, Kays and London⁽¹⁴⁾ devote one chapter to transient performance, but dwell primarily on counter-flow and duct arrangements. Using methods somewhat similar to those of Kays and London, but much better adapted to transient studies, Myers, et al.⁽⁴⁾ developed general cross-flow differential equations and appropriate non-dimensional parameters. These equations were modified to the "one-fluid-mixed" case and then, having made some generalizing assumptions, several limiting cases for which exact transient solutions may be determined are presented.

Since we were desirous of comparing the TAP reference to one of these "limiting" solutions, it would be well to present the assumptions made in the MM-3 report. These were:

- The fluid flows are uniform across each flow passage (one-dimensional).
- The heat transfer is one-dimensional (longitudinal conduction in fluids and in the wall is assumed to be zero).
- The conduction resistance through the wall is negligible.

- The following quantities are constant (independent of temperature, time, and position): fluid capacitance, wall capacitance, thermal resistance, and flow rate.

Only the first assumption was made in the development of the TAP reference.

The most pertinent solution presented in the MM-3 report is for cases where the wall capacitance is large relative to the fluid capacitance. For this case, in the nomenclature of the MM-3 report, the solution of the general equations for the case of a step change in the temperature of the hot fluid is given by:

$$t_h^* = (\theta^* - 1) \left\{ \phi_0[B(\theta^* - 1), A] + e^{-A} e^{-B(\theta^* - 1)} I_0(2\sqrt{B(\theta^* - 1)A}) \right\} \quad (D-1)$$

$$t_c^* = \frac{P}{1 - e^{-P}} \int_{x^*=0}^1 e^{-Px^*} \phi_0[B(\theta^* - 1), Ax^*] dx \quad \text{for } \theta^* \geq 1 \quad (D-2)$$

Computer programs are presented in the MM-3 report for numerically integrating Equation (D-2) and for computing ϕ_0 . By using these basic methods, a solution of these equations was determined for parameters derived from the DHX data given in Appendix B.

It would be well at this point to draw attention to the fact that the MM-3 report makes no attempt to deal with extended surface exchangers. The use of extended surfaces (circumferential fins in this case) adds a significant degree of complexity to the heat transfer processes which occur. Any comparison of results should make allowance for this significant difference in the exchangers investigated. Because of these extended surfaces involved, we investigated what we felt were two limiting cases.

The first considers that the entire metal capacitance has an effect on both the air and sodium temperature calculations. For this case the constants have the values: $A = 32.239$, $B = 9.0637$, $P = 0.5528$; hence, $M = A+P = 32.792$. The other case considers that the metal capacitance in the fin primarily affects the air temperature calculations while the tube capacitance in the base tube primarily affects the sodium. The only constant changed is the value of B , which is $B = 12.623$ for the t_c^* calculation and $B = 32.128$ for the t_h^* calculation. The actual behavior is probably somewhere between these two limiting cases. The results for these two cases are given in the accompanying Figures D-1 and D-2 (similar to Figure 5 and Figure 7 of the MM-3 report). The temperature predictions based on these results for a step change in the sodium inlet temperature from 765 to 1000 °F are given in Figure D-3.

When one considers the assumptions made in the MM-3 report and the much more complex heat transfer processes occurring in the actual extended surface DHX, the results presented in Figure D-3 are really quite excellent. The results of the air exit temperature calculations support the conclusions arrived at in Section 4.3 of the report. Likewise, the general shape and location in time of the sodium temperature results lend affirmative support to the validity of the TAP reference model.

FIGURE D.1. Unstepped-Fluid Response (Large $\bar{\alpha}_w^*$)
for $P = 0.5528$ and $M = 32.792$

FIGURE D.2. Stepped-Fluid Response (Large $\bar{\alpha}_w^*$)
for $P = 0.5528$ and $M = 32.792$

FIGURE D.3. Comparison of TAP Reference Model Responses With MM-3 Report Results for a Step Change in Sodium Temperature from 765 to 1000 °F

APPENDIX ETHE TIME DEPENDENT SOLUTION FOR THE TEMPERATURE AND HEAT TRANSFER
IN AN EXTENDED SURFACE OF CONSTANT THICKNESS

The accompanying Figure E-1 gives a sketch of a differential element of the fin with the associated sign convention. The one-dimensional Fourier conduction equation in the radial direction is given by:

FIGURE E.1 *Differential Fin Element*

$$q = -kA \frac{d\theta}{dr} . \quad (E-1)$$

The associated heat transfer area is

$$A = (2\pi r)2y_b = 4\pi r y_b . \quad (E-2)$$

Substituting in the first equation and differentiating with respect to r ,

$$dq = \frac{d}{dr} -4\pi k y_b r \frac{d\theta}{dr} dr . \quad (E-3)$$

With respect to the differential element shown above,

$$dq = \text{heat transfer to air} + \text{heat stored}$$

$$dq = -2h \theta dA_s - \rho c dV \frac{d\theta}{dt} \quad (E-4)$$

In this expression,

$$dA_s = \text{surface area} = 2\pi r dr \quad (E-5)$$

and

$$dV = 2\pi r dr (2y_b) = 4\pi y_b r dr . \quad (E-6)$$

The method used in determining the above differentials neglect second order contributions. This is the usual assumption. Replacing the differentials of Equation (E-4) with their definitions in Equations (E-5) and (E-6), we obtain:

$$dq = 4\pi h \theta r \, dr - 4\pi \rho c y_b r \frac{d\theta}{dr} \, dr. \quad (E-7)$$

Equating the right hand sides of Equations (E-7 and E-3) gives the following result after some simplification:

$$\frac{d}{dr} \left[k y_b r \frac{d\theta}{dr} \right] = h \theta r + \rho c y_b r \frac{d\theta}{dt}, \quad (E-8)$$

or

$$\frac{d^2\theta}{dr^2} + \frac{1}{r} \frac{d\theta}{dr} - \frac{h}{k y_b} \theta = \left(\frac{\rho c}{k} \right) \frac{d\theta}{dt}. \quad (E-9)$$

For steady state, the right-hand side vanishes as $\frac{d\theta}{dt} = 0$ and Equation (E-9) is identical with Equation (3-21) in the test for fins of constant thickness for which $\frac{dy_b}{dr} = 0$. This is the equation for the temperature at any time in the fin.

The solution of Equation (E-9) is not straightforward by any means, but, fortunately, it is not necessary to accomplish our purpose. Switching temporarily to heat transfer calculations, the total heat transferred from the fin to the air is given by:

$$q' = 2 \int_{r_1}^{r_2} \theta h \, dA = 4\pi \int_{r_1}^{r_2} h \theta r \, dr \quad (E-10)$$

where r_1 = radius at base of fin,

r_2 = radius at fin end $+y_b$.

However, from Equation (E-8) we obtain

$$\frac{d}{dr} \left[r \frac{d\theta}{dr} \right] = \frac{h}{ky_b} r\theta + \frac{\rho cy_b}{k} r \frac{d\theta}{dt} \quad (E-11)$$

which may be solved for r as follows:

$$r\theta = \frac{ky_b}{h} \frac{d}{dr} \left[r \frac{d\theta}{dr} \right] - \frac{\rho cy_b}{h} r \frac{d\theta}{dt} . \quad (E-12)$$

These manipulations assume that average-temperature-based coefficients and properties are used so that they are not functions of location. They may be functions of temperature. Substituting this expression in Equation (E-10) yields

$$q' = 4\pi h \int_{r_1}^{r_2} \left\{ \frac{ky_b}{h} \frac{d}{dr} \left[r \frac{d\theta}{dr} \right] - \frac{\rho cy_b}{h} r \frac{d\theta}{dt} \right\} dr. \quad (E-13)$$

The first term may be integrated immediately. The result is

$$q' = 4\pi h \left[\frac{ky_b}{h} \left(r \frac{d\theta}{dr} \right)_{r_1}^{r_2} - \frac{\rho cy_b}{h} \int_{r_1}^{r_2} r \frac{d\theta}{dt} dr \right] . \quad (E-14)$$

One further simplification; by using for r_2 the radius at the edge of the fin plus y_b , as indicated, then $r \left(\frac{d\theta}{dr} \right)_{r_2} = 0$ and the above equation is still more simplified. This is the expression for the total heat transfer rate from a fin of constant thickness to be air.

Employing the previously given definition of fin efficiency as

$$\Omega = q'/q'_o \quad (E-15)$$

where,

$$q'_o = 2\pi h (r_2^2 - r_1^2) \theta_o = h A_f (T_b - T_a), \quad (E-16)$$

then the fin efficiency as a function of time is given by

$$\Omega = \frac{4\pi y_b}{hA_f(T_b - T_a)} \left\{ r_1 k \left(\frac{d\theta}{dr} \right)_{r=r} - \int_{r_1}^{r_2} \rho c r \frac{d\theta}{dt} dr \right\}. \quad (E-17)$$

This is the equation which is found in the text.

REFERENCES

1. H. G. Johnson. Water Heat Sink for the FFTF Main Heat Removal System, BNWL-CC-691. Battelle-Northwest, Richland, Washington, June 10, 1966.
2. R. A. Harvey, H. G. Johnson, M. A. McLaughlin, and E. F. Witeck. Technical Basis for FFTF Main Heat Transport System Concept and Concept Configuration Selections, BNWL-554. Battelle-Northwest Richland, Washington, September 1967.
3. A. Hopkinson. Some Numerical Results on Theoretical Models for the Dynamics of a Cross-Flow Heat Exchanger, AEEW-R-304. Atomic Energy Establishment, Winfrith, England, August 1963.
4. G. E. Myers, J. W. Mitchell, and R. F. Norman. The Transient Response of Cross-Flow Heat Exchangers With One Fluid Mixed, Report No. MM-3. Department of Mechanical Engineering, University of Wisconsin, October 1, 1965.
5. C. D. Flowers and L. H. Gerhardstein. Analog-Hybrid Dynamic Simulation of the FFTF Reactor and Heat Transport System, BNWL-707. Battelle-Northwest, Richland, Washington, April 1968.
6. B. J. Dray. Thermal Analyzer Program (TAP) User's Manual, NAA-SR-TDR-11812, North American Aviation, Downey, California, January 11, 1966.
7. C. L. Goss. Engineers Manual on Fortran IV Version of Dynamic Systems Analyzer (DYNASYAR), DUN-M-17. Douglas United Nuclear, Inc., Richland, Washington, March 1968.
8. D. E. Dwyer. "Eddy Transport in Liquid-Metal Heat Transfer," A.I.Ch.E Journal, 9, p. 261, March 1963.
9. S. L. Jameson. "Tube Spacing in Finned-Tube Banks," Transactions ASME, 67, p. 633, 1945.
10. D. Kern. Process Heat Transfer, 1st ed., p. 555. McGraw-Hill Book Company, Inc., New York, 1950.
11. C. L. Goss and H. G. Johnson. ADOPT, Program for Steady State Process Design and Cost Optimization of FFTF Heat Removal Systems Using Heat Dumps to Air, BNWL-308. Battelle-Northwest, Richland, Washington, September 1966.
12. A. L. Gunby. Intermediate Heat Exchanger Modeling for FFTF Simulation, BNWL-SA-2818. Battelle-Northwest, Richland, Washington, September 15, 1969.

REFERENCES (Continued)

13. K. A. Gardner. "Efficiency of Extended Surface Heat Exchangers," Transactions ASME, 67, pp. 621-633, 1945.
14. W. M. Kays and A. L. London. Compact Heat Exchangers, McGraw-Hill Book Company, Inc., New York, 1964.

DISTRIBUTIONOFFSITENo. of
Copies

1	<u>AEC Chicago Patent Group</u> GH Lee, Chief
28	<u>AEC Division of Reactor Development and Technology</u> Director, RDT Asst Dir for Nuclear Safety Analysis & Evaluation Br, RDT:NS Environmental & Sanitary Engr Br, RDT:NS Research & Development Br, RDT:NS Asst Dir for Plant Engrg, RDT Facilities Br, RDT:PE Components Br, RDT:PE Instrumentation & Control Br, RDT:PE Liquid Metal Systems Br, RDT:PE Asst Dir for Program Analysis, RDT Asst Dir for Project Mgmt, RDT Liquid Metals Projects Br, RDT:PM FFTF Project Manager, RDT:RE Asst Dir for Reactor Engrg, RDT Control Mechanisms Br, RDT:RE Core Design Br, RDT:RE (2) Fuel Engineering Br, RDT:RE Fuel Handling Br, RDT:RE Reactor Vessels Br, RDT:RE Asst Dir for Reactor Tech, RDT Coolant Chemistry Br, RDT:RT Fuel Recycle Br, RDT:RT Fuels & Materials Br, RDT:RT Reactor Physics Br, RDT:RE Special Technology Br, RDT:RT Asst Dir for Engrg Standards, RDT EBR-II Project Manager, RDT:PM
1	<u>AEC Idaho Operations Office</u> <u>Nuclear Technology Division</u> CW Bills, Director
1	<u>AEC San Francisco Operations Office</u> <u>Director, Reactor Division</u>

No. of
Copies

4	<u>AEC Site Representatives</u> Argonne National Laboratory Atomics International General Electric Company Westinghouse Electric Corporation
215	<u>AEC Division of Technical Information Extension</u>
3	<u>Argonne National Laboratory</u> RA Jaross LMFBR Program Office NJ Swanson
1	<u>Atomic Power Development Association</u> Document Librarian
5	<u>Atomics International</u> FFTF Program Office
1	<u>Liquid Metal Engineering Center</u> RW Dickinson
2	<u>Liquid Metal Information Center</u> AE Mitter
2	<u>Babcock & Wilcox Company</u> Atomic Energy Division SH Esleeck GB Garton
1	<u>BNW Representative</u> RM Fleischman (ZPPR)
1	<u>Combustion Engineering</u> 1000 MWe Follow-On Study WP Staker, Project Manager
1	<u>Combustion Engineering</u> Mrs. Nell Holder, Librarian
5	<u>General Electric Company</u> Advanced Products Operation Karl Cohen (3)
	<u>Nuclear Systems Programs</u> DH Ahmann (2)

No. of
Copies

2	<u>Gulf General Atomic Inc.</u> <u>General Atomic Division</u> D Coburn
1	<u>Idaho Nuclear Corporation</u> JA Buckham
1	<u>Oak Ridge National Laboratory</u> WO Harms
1	<u>Stanford University</u> <u>Nuclear Division</u> Division of Mechanical Engrg R Sher
1	<u>United Nuclear Corporation</u> <u>Research and Engineering Center</u> RF DeAngelis
5	<u>Bechtel Corporation</u> JJ Teachnor
10	<u>Westinghouse Electric Corporation</u> <u>Atomic Power Division</u> Advanced Reactor Division SL Additon (4) FM Heck TB McCall HS McCreary PC Miller CF Wolfe

ONSITE HANFORDNo. of
Copies

1	<u>Bechtel Corporation</u> MO Rothwell (Richland)	
1	<u>Westinghouse Electric Corporation</u> JD Herb (Richland)	
3	<u>RDT Asst Dir for Pacific Northwest Programs</u> TA Nemzek	
2	<u>AEC Richland Operations Office</u> JM Shivley	
1	<u>AEC Chicago Patent Group</u> RK Sharp (Richland)	
3	<u>Battelle Memorial Institute</u>	
52	<u>Battelle Northwest</u>	
	JM Batch	Legal-703 Bldg.
	AL Bement	Legal-ROB, 221-A
	RD Benham	BNW-Technical Information (5)
	NE Carter	BNW-Technical Publications (3)
	WL Chase	FFTF File (703) (10)
	JC Cochran	FFTF TPO (703)
	DL Condotta	
	JF Erben	
	EA Evans	
	CD Flowers	
	AL Gunby	
	LD Gustafson	
	RA Harvey	
	PL Hofmann	
	HG Johnson	
	HE Little	
	RN Madsen (5)	
	WB McDonald	
	JS McMahon	
	RE Peterson	
	JM Seehuus	
	DD Stepnewski	
	GH Strong	
	JH Westsik	
	LA Whinery	
	TW Withers	
	B Wolfe	