SIMULATION OF PIPING LENGTHS
IN SODIUM SYSTEMS

A. L. Gunby

December 1970

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Printed in the United States of America
Available from
National Technical Information Service
National Bureau of Standards,
U.S. Department of Commerce
Springfield, Virginia 22151
Price: Printed Copy \$3.00; Microfiche \$0.65

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ABSTRACT

Several models for sodium piping were examined, using example pipe sizes and example flow and temperature transients. A reference model (alternating central difference and three-point backward differences at each axial mesh point) was selected to describe the effect of pipe wall stored heat on coolant thermal transients. Model simplifications were investigated. This investigation resulted in a model using a transport delay in series with a single-node for stored heat effects, with sodium and metal volumes altered by appropriate fractions.

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SIMULATION OF PIPING LENGTHS IN SODIUM SYSTEMS

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INTRODUCTION

The FFTF heat transport system will use large-diameter sodium piping with outer diameters ranging from 8 inches to 28 inches. Stored heat in the steel piping walls constitutes about 20 to 30% of the total stored heat of sodium and steel. This characteristic will affect thermal transients as they move along a pipe and should be accounted for in FFTF simulation work. Up to this time, the effect has been neglected in order to compute conservative (i.e., more severe) thermal transients for conceptual design. Accordingly, various approaches to modeling pipe stored heat effects were studied in an effort to find a reasonable approach to incorporating the effect in the several FFTF system simulations now in use.

Previous work by Westinghouse on FFTF simulation used a nodal heat balance technique with separate metal and sodium regions and seven axial segments (nodes). This effort was an extension of earlier Los Alamos work.⁽¹⁾ The approach of the study reported here was to compare a variety of modeling techniques, each using 10 nodes and separate metal and sodium regions, in order to determine a suitable reference model which could be used as a base of comparison for model simplifications. Simplification is necessary due to the limits of computing time and computer size when the large system simulations are used.

A particular pipe size and flow rate were selected, comparable to values encountered in current FFTF designs. Five modeling techniques were studied by using two thermal transients, one with constant flow and the other with varying flow. A reference model was selected based on its close approximation to the expected actual transient. Simplified models were then studied in the same manner and compared to the reference case. Selection of a simplified model for the system simulations was then made based on the needs of the simulation. Finally, the selected simplified

model was examined for its response to other selected transients and for a range of piping sizes.

SUMMARY AND CONCLUSIONS

In simulating sodium systems, stored heat in the steel piping may have a significant effect on thermal transients, particularly for large pipes. A thermal transient entering a length of pipe will be altered by the time it reaches the end of the pipe, due to transfer of heat between the pipe and sodium. Accordingly, the effect of piping stored heat was studied using a range of pipe sizes and assumed inlet transients in an effort to find a reasonably simple modeling approach for large system simulation. Results will be applied to the several FFTF system simulations now in use.

Several modeling techniques were studied to determine the best approach for a base, or reference model. An arbitrary pipe length was divided into nodal segments and temperatures for each segment using different finite differencing schemes. Overall, the best model for the pipe length was shown to be a mixed difference approach, in which the temperature gradient at each point along the length is computed with central differences alternating with three-point backward differences.

Simplified models were then explored to reduce complexity of a total system simulation. The combination of metal and sodium stored heat, and the reduction of the number of nodes were studied with the following result: a simplified model consisting of a pure transport delay in series with a single node combining metal and sodium, and with sodium and metal volumes reduced by an appropriate factor. Comparison of this simple model to the reference model showed good results for a much less complex simulation.

The conclusions of this study may be summarized as follows:

- Simulation of a pipe with flowing sodium may be accomplished with reasonable accuracy using a single transport delay and a single heat storage node in series.
- Volume factors should be adjusted to match the pipe size, based on preliminary calculations before inserting the pipe model into a total system model.

REFERENCE MODEL SELECTION

MODEL DESCRIPTION

In order to develop a pipe heat model sufficiently simple to be usable in the FFTF system simulations, a detailed model is needed as a reference for comparison. The model should be sufficiently detailed to give a good representation of the expected effects.

For the purpose of comparison, a hypothetical pipe section was selected and divided into ten axial segments, or nodes. Two nodes were used radially, one for metal and one for sodium. Material properties and heat transfer characteristics of each node (using the tube-side correlation of Dwyer⁽²⁾) were variable with respect to temperature and flow. Several different approaches to the modeling were examined, with heat balances performed on either a nodal length or at a mesh point.

Nodal Heat Balance

Figure 1 shows the model for a nodal heat balance approach with the sodium outlet temperature of node j designated as T_{cj} . Performing a heat balance on node j , we have Equations (1) and (2).

FIGURE 1. Pipe Heat Model, Nodal Heat Balance

$$\frac{dT_{mj}}{dt} = - \frac{(UA')_j}{(\rho V C)_{mj}} \left(T_{mj} - \frac{T_{cj} + T_{cj-1}}{2} \right) \quad (1)$$

$$\begin{aligned} \frac{dT_{cj}}{dt} = & - \frac{W}{(\rho V C)_{cj} (L/10)} (T_{cj} - T_{cj-1}) \\ & + \frac{(UA')_j}{(\rho V C)_{cj}} \left(T_{mj} - \frac{T_{cj} + T_{cj-1}}{2} \right) \end{aligned} \quad (2)$$

for $j = 1 \dots 10$.

Steel properties of node j (density, specific heat, conductivity) are determined from T_{mj} . Sodium properties of a coolant node with nodal outlet temperature T_{cj} are determined by an average nodal temperature T_{cjj-1} , defined in Equation (3). Sodium properties of interest are density, specific heat, viscosity, and sodium film coefficient.

$$T_{cjj-1} = \left(\frac{T_{cj} + T_{cj-1}}{2} \right) \quad (3)$$

Reynolds number is computed using the viscosity at each node, and is then used along with nodal sodium temperature as an input to the heat transfer correlation to get h_{jj-1} , the nodal film coefficient. For the heat transfer resistance of the pipe wall, the metal temperature is assumed to be located midway through the pipe wall, that is,

$$D_m = \left(\frac{D_o + D_i}{2} \right) \quad (4)$$

Thus, the overall heat transfer coefficient (based on inside area of the pipe) is expressed as

$$U_j = \left\{ \frac{1}{h_{jj-1}} + \frac{D_i \ln (D_m/D_i)}{k_{mj}} \right\}^{-1} \quad (5)$$

In the above equations the following are constants: A' , V' , L , D_o , D_i and D_m . The film coefficient calculation is described in Reference 2.

Equations (1), (2) and (3) define the nodal heat transfer model for ten axial segments over pipe length, L . A second class of models will be studied as well, by performing the heat balance at mesh points instead of over a node volume. Four such approaches are described below.

General Differencing Technique

Differencing techniques are needed when the mesh point approach of Figure 2 is used, since the temperature derivative must be approximated.

FIGURE 2. Pipe Heat Model, Mesh Point Approach

The basic heat balance equations for the mesh point approach are expressed as follows:

$$(\rho V' C)_{mj} \frac{dT_{mj}}{dt} = - (UA')_j (T_{mj} - T_{cj}) \quad (6)$$

$$(\rho V' C)_{cj} \frac{dT_{cj}}{dt} = - WC_{cj} \left(\frac{\partial T_{cj}}{\partial x} \right) + (UA')_j (T_{mj} - T_{cj}) \quad (7)$$

Note that areas and volumes are used on the basis of per unit length (prime superscript). The fundamental question then becomes the approximation of the partial derivative $(\partial T_{cj} / \partial x)$. Equations (6) and (7) reduce to:

$$\frac{dT_{mj}}{dt} = - \frac{(UA')_j}{(\rho V' C)_{mj}} (T_{mj} - T_{cj}) \quad (8)$$

$$\frac{dT_{cj}}{dt} = - \frac{W}{(\rho V C)_{cj}} \left(\frac{\Delta T_{cj}}{n \Delta x} \right) + \frac{(UA)_{cj}}{(\rho V C)_{cj}} (T_{mj} - T_{cj}) \quad (9)$$

Two-Point Backward Difference

For this approach, the temperature derivative is approximated by the temperature difference across a single axial segment divided by the segment length, Δx , that is,

$$\Delta T_{cj} = T_{cj} - T_{cj-1} \quad \text{for } j = 1 \dots 10., \text{ and for } n = 1. \quad (10)$$

Note that $\Delta x = L/10$, and that Equation (9) is identical to Equation (2) except for the coolant temperature in the heat transfer term. Thus, the two-point backward difference model (backward with respect to flow direction) is like a series of perfect mixing chambers set end-to-end, with coolant properties and heat transfer defined by each chamber's outlet temperature.

Central Difference

A central difference approximation is expressed as follows:

$$\Delta T_{cj} = T_{cj+1} - T_{cj-1} \quad (11)$$

$$\text{for } j = 1 \dots 9,$$

with $n = 2$ so that $n \Delta x = L/5$. The outlet node requires a backward difference, and the three-point backward approach was selected as follows:

$$\Delta T_{cj} = 3T_{cj} - 4T_{cj-1} + T_{cj-2} \quad (12)$$

$$\text{for } j = 10,$$

also with $n = 2$.

Three-Point Backward Difference

As shown previously, three temperatures are used to form the difference:

$$\Delta T_{cj} = 3T_{cj} - 4T_{cj-1} + T_{cj-2} \quad (13)$$

for $j = 2 \dots 10$,

with $n = 2$. Since the first mesh point does not have two preceding temperatures to use, a central difference was selected to represent the derivative at that point.

$$\Delta T_{cj} = T_{cj+1} - T_{cj-1} \quad (14)$$

for $j = 1$,

also with $n = 2$. Note that for all models, coolant inlet temperature is always the $j-1$ node with respect to $j = 1$, that is,

$$T_{j-1} = T_i, \text{ for } j = 1 \quad (15)$$

Mixed Difference

Previous work on IHX simulation⁽³⁾ showed the usefulness of alternating the central and three-point backward difference techniques. Accordingly, the approach was applied to the problem of modeling piping stored heat.

$$\Delta T_{cj} = T_{cj+1} - T_{cj-1} \quad (16)$$

for $j = 1, 3, 5, 7, 9$

$$\Delta T_{cj} = 3T_{cj} - 4T_{cj-1} + T_{cj-2} \quad (17)$$

for $j = 2, 4, 6, 8, 10$

with $n = 2$.

MODEL RESPONSES

To check the responses of the foregoing models, two example transients were selected:

- Inlet temperature ramp with constant flow
- Combined inlet temperature and flow ramp.

Each model was subjected to these transients, with the results as shown in Figures 3 through 6. As shown in the previous section, all five models were of the same complexity in terms of nodal detail and variability with respect to flow and temperature.

Figures 3 and 4 show the responses to the inlet temperature ramp. The selected ramp has no relationship to expected transients on any system, but is steep enough to provide good definition in response. A pure transport delay for the selected pipe size would merely displace the $-20\text{ }^{\circ}\text{F/sec}$ ramp by 16.5 seconds later in time. No change to the input wave form would occur. Accounting for the stored heat in pipe walls, however, produced the effects of reduced transient rate and a considerable "tail" on the outlet temperature transient. Comparison of the two figures shows the following:

- The three-point backward and central difference models each respond with uncharacteristic overshoots.
- The nodal heat balance and two-point backward models each respond with a shallower slope and a shorter transport time than does the mixed difference model.

Figures 5 and 6 show the same models responding to the combined temperature and flow ramp. With reduced flow, the pipe has a greater opportunity to release its heat to the coolant and mitigate the transient. Comparison of the figures reveals the following:

- The three-point backward model responds with an uncharacteristic initial temperature increase to an inlet temperature decrease.

FIGURE 3. Sodium Response to Inlet Temperature Ramp, Detailed Models (#1)

FIGURE 4. Sodium Response to Inlet Temperature Ramp, Detailed Models (#2)

- NOTES: 1. 26" O.D., 25" I.D., 150 FT LONG.
 2. DWYER HEAT TRANSFER CORRELATION.
 3. 10-NODE MODELS.
 4. A= NODAL HEAT BALANCE MODEL.
 B= MIXED DIFFERENCE MODEL.
 C= 2-POINT BACKWARD DIFFERENCE MODEL.

FIGURE 5. Sodium Response to Combined Inlet and Flow Ramps, Detailed Models (#1)

FIGURE 6. Sodium Response to Combined Inlet and Flow Ramps, Detailed Models (#2)

- The nodal heat balance and two-point backward models show a significantly shallower slope and a shorter transport time than does the mixed difference model.

Figures 7 and 8 show the responses to the test transients for the nodal heat balance and mixed difference models, as well as the expected results for a pure transport delay. In addition to ten nodes, the nodal heat balance model was examined for 20 nodal segments over the same pipe length. Comparison of the curves gives the following:

- The ten-node mixed difference model responds with a steeper transient and a more appropriate transport delay than either model of the nodal heat balance approach
- Further increase in number of nodes should make the models equivalent.

REFERENCE MODEL SELECTION

The expected outlet temperature transient from a pipe in thermal interaction with the coolant will begin as the pure transport delay transient does, but will gradually pull away to produce a "tail" on the curve. This result is due to the build-up of the metal-coolant ΔT as the transient proceeds in time. Thus, the selection of a reference model depends on the best approximation to the foregoing qualitative description of the expected transients.

Accordingly, the ten-node mixed difference model was selected as the reference model for the following reasons:

- The central difference and three-point backward difference models are not acceptable due to their anomalous responses to the test transients.
- The nodal heat balance model with a much larger number of nodes appears to be equivalent to the mixed difference model. Thus, the less complex model is preferred.

FIGURE 7. Sodium Response to Inlet Temperature Ramp, Best Candidate Models

- NOTES 1 26" O D, 25" I D, 150 FT LONG
 2 DWYER HEAT TRANSFER CORRELATION
 3 A= MIXED DIFFERENCE MODEL, 10 NODES
 B= NODAL HEAT BALANCE MODEL, 10 NODES
 C= NODAL HEAT BALANCE MODEL, 20 NODES

FIGURE 8. Sodium Response to Combined Inlet and Flow Ramps, Best Candidate Models

SIMPLIFIED MODELS

A simplified approach to pipe stored heat effects is needed because of the large number of pipe lengths included in the several FFTF system simulations. For a total system simulation with two complete heat transport systems, each having a suction and discharge leg on either side of a pump, the number of piping segments is twelve. If a pure transport delay were used, then twelve integrations would be required; if the ten-node reference model were used, 240 integrations would be needed. Clearly, some accuracy must be sacrificed for simpler models.

ALTERNATIVE MODELS

The spectrum of model simplifications may be covered by the following general categories:

- Reduction in number of axial segments
- Lumping of metal stored heat with that of sodium
- Combination of transport delay and simplified heat transfer expression.

After preliminary evaluation, several such models were selected for continued study as discussed in the following paragraphs.

Ten-Node Lumped Metal Model

The reference mixed difference model was selected for checking the effect of pipe metal "lumping". Stored heat of the steel pipe was included in the value for coolant stored heat, thus eliminating the metal temperature equations. Coolant temperature equations then reduce to:

$$\frac{dT_{c_j}}{dt} = - \frac{W}{(\rho V^c)_j} \left(\frac{\Delta T_{c_j}}{2\Delta x} \right) \quad \dots \quad (18)$$

for $j = 1 \dots 10$,

with $(\rho V^c)_j$ including steel and sodium. The ΔT_{c_j} values are computed as before in Equations (16) and (17). For each pipe modeled this way, ten integrations are needed.

Two-Node Heat Balance Model

Two axial segments were used for this model, employing equations (1) through (5) only for $j = 1, 2$. Equation (2) becomes

$$\begin{aligned} \frac{dT_{c_j}}{dt} &= \frac{W}{(\rho V C)_{c_j} (L/2)} (T_{c_j} - T_{c_{j-1}}) + \frac{(UA')_j}{(\rho V C)_{c_j}} \cdot \cdot \cdot \\ &\quad \left(T_{mj} - \frac{T_{c_j} + T_{c_{j-1}}}{2} \right) \cdot \cdot \cdot \end{aligned} \quad (19)$$

for $j = 1, 2$.

Material properties and heat transfer coefficients are calculated at each axial position as before. Four integrations are needed for each pipe length with this approach.

Two-Node Difference Model

Reducing the ten-node models for central, three-point backward and mixed difference techniques results in a single two-node model. The first mesh point in the flow direction uses central differencing while the other uses the three-point backward approach. Thus Equations (8), (9), (11), and (12) apply, for $j = 1$ and 10 only, and for $n\Delta x = L$. As before, properties are computed at each mesh point. Four integrations are required for each simulated pipe length.

Transport Delay and Simple Heat Transfer

Reducing the number of integrations even further requires a fresh approach. Accordingly, a flow-dependent transport delay is followed by a simple expression denoting the effect of stored heat on the transient. The stored heat effect was simulated two ways: (1) a metal and a coolant node, and (2) a single coolant node with metal stored heat lumped together with coolant. Figure 9 shows the arrangement for both approaches.

FIGURE 9. Simplified Pipe Heat Model Using Transport Delay

The transport delay portion is common to both Model A and B, and is expressed as follows:

$$T_d = T_i (t - \tau) \quad \dots \quad (20)$$

with τ determined by the volume integration,

$$V = \int_{\tau-T}^t Q(t) dt \quad \dots \quad (21)$$

For the initial traverse of the pipe,

$$\tau = t, \text{ that is, for } V \geq \int_0^t Q(t) dt \quad \dots \quad (22)$$

$$Q(t) = \frac{W}{\rho} \quad (23)$$

The pipe volume, V , is computed from pipe inside dimensions.

Model A assumes the coolant behaves as it would in a perfect mixing chamber with an adjoining metal node.

$$\frac{dT_m}{dt} = - \frac{UA}{(\rho VC)_m} (T_m - T_o) \quad \dots \quad (24)$$

$$\frac{dT_o}{dt} = \frac{W}{(\rho V)_c} (T_d - T_o) + \frac{UA}{(\rho VC)_c} (T_m - T_o) \quad \dots \quad (25)$$

Model B assumes the coolant behaves as it would in a perfect mixing chamber with the metal stored heat added to that of the coolant.

$$\frac{dT_o}{dt} = \frac{W}{(\rho \bar{V})_c} (T_d - T_o) \quad \dots \quad (26)$$

where $\bar{V}$ is an equivalent sodium volume, combining metal and sodium stored heat.

The expressions for the volumes of Model B (both the transport delay volume, V , and the mixing volume, $\bar{V}$) were adjusted to an appropriate fraction as discussed in the next section. The expressions for these effective fractions are as follows:

$$V = K_1 (L) \frac{\pi}{4} (D_i)^2 \quad \dots \quad (27)$$

$$\bar{V} = K_2 \left[(L) \frac{\pi}{4} (D_i)^2 + (L) \frac{\pi}{4} (D_o^2 - D_i^2) \left(\frac{\rho_m C_m}{\rho_c C_c} \right) \right] \quad \dots \quad (28)$$

where L , D_o , and D_i are pipe dimensions and the K 's are the fractions. The $(\rho C)_m / (\rho C)_c$ ratio is determined for the initial temperature. As before, material properties and heat transfer as noted in Equations (23) through (26) are allowed to vary with temperature and flow. Note that Model A uses three integrations per pipe length while Model B uses two per pipe length.

MODEL RESPONSES

Each simplified model was tested with the previously described input transients. Pipe dimensions, flow rates and initial temperatures remain unchanged for the purposes of comparison.

Lumped Metal Models

Figure 10 shows the lumped metal model response to the temperature ramp with constant flow. Since metal mass is added to that of sodium and the model eliminates separate metal nodes, the net effect is to increase the effective size of the sodium nodes and lengthen the transport delay with respect to the reference case. However, the maximum rate of temperature change for each case of Figure 10 is the same as the rate of change for the respective model with separate metal nodes (see Figure 3). The omission of separate metal nodes only changes the apparent transport delay and the character of the "tail" of the curve.

An overshoot characteristic is noted for the mixed difference response. Apparently the presence of separate metal nodes and heat transfer to the coolant acts as a stabilizing influence on the reference model. Although the ten-node lumped metal models still use a large number of integrations, the results point to omission of metal nodes as a possible path for model simplification.

Two-Node Models

The second approach to model simplification is the reduction of the number of nodes. Figures 11 and 12 show the responses of two-node models compared to the reference model for the same test transients. As expected, rates of change are much less than for the detailed model, outlet temperatures begin responding too early, and the mixed difference model produces an anticipatory error. More nodes would improve the accuracy but would also defeat the objective of simplicity. The third simplified model is built upon the results of the previous two approaches.

FIGURE 9. Simplified Pipe Heat Model Using Transport Delay

FIGURE 11. Comparison of Two-Node Models to Reference Model, Inlet Temperature Ramp

FIGURE 12. Comparison of Two-Node Models to Reference Model, Combined Inlet and Flow Ramps

Transport Delay and Simple Heat Transfer Models

Both models described in Equations (20) through (26) were examined using the usual test transients while varying the K-factors (Equations (27) and (28) for effective volume. Figures 13 and 14 show the responses of Models A and B for the usual two transient test cases with K's = 1.0 (full metal and sodium volumes). It is clear that the added metal node does not give sufficient accuracy to offset the cost of more integrations; hence, Model B was pursued as having the most promise. The results of Figures 13 and 14 were compared to the reference model results, showing the potential of a good curve match by adjusting the volume K-factors.

Figures 15 and 16 show the same two transients for a spectrum of K's in the effort to align with the reference model results. Alignment is surprisingly good except for one basic difficulty: choosing the desired K's for a good match with constant flow produces transient results which differ markedly for the flow-ramp case. One must choose his K's based on what he desires from the simulation, that is, whether his transient studies will consider constant or variable flow and in what manner the flow will vary. It is clear that a perfect match cannot be achieved even for the constant flow case, as the "tail" of the curve is lost when we eliminate the separate metal node. However, as in all simulations, a judicious choice can be made.

SIMPLIFIED MODEL SELECTION

Visual examination of these results shows that a reasonable choice for a simplified model of pipe stored heat effects is the transport delay with a single lag (Model B) and $K_1 = 0.85$, $K_2 = 0.3$. Numerical analysis by statistical deviation of simple model results from those of the reference model is not appropriate, since one must tailor the simulation to his analytical needs. For FFTF simulation, transient rates and transport delays are important and lead to the following:

- Equivalent transport delay (that is, delay until outlet temperature begins to change) for constant flow case ($\pm 10\%$ of reference case value).

FIGURE 13. Response of Simplified Models to Inlet Temperature Ramp

FIGURE 14. Response of Simplified Models to Combined Inlet and Flow Ramps

NOTES: 1. CONSTANT FLOW = 1650 LB/SEC
 2. 26" O.D., 25" I.D., 150 FT LONG.
 3. MODEL B WITH VOLUME FRACTIONS AS MARKED.

FIGURE 15. Response of Simplified Model to Inlet Temperature Ramp, Varying Effective Pipe Volume

FIGURE 16. Response of Simplified Model to Combined Ramp, Varying Effective Pipe Volume

- Equivalent maximum outlet temperature transient rate for constant flow case ($\pm 10\%$ of reference case value).
- Equivalent transport delay for the variable flow case ($\pm 10\%$ of reference case value).
- Equivalent maximum outlet temperature transient rate for the variable flow case ($\pm 30\%$, chosen because rates are extremely small and of low significance).

For these criteria, the K-factors of 0.85 and 0.3 are satisfactory. The simplified model is now tested with different pipe size and more extreme transients.

SIMPLIFIED MODEL CHECK OUT

The first check to be made concerned the effect of variable coolant density. Sodium density for both the transport delay and the stored heat term were assumed to be constant through a transient and equal to the initial value. The usual test cases were repeated with the following results:

- Differences in transient rates between the constant- ρ and variable- ρ models are negligible.
- Maximum deviation between the two models at any point in time is 4 °F for the inlet temperature ramp and 5°F for the combined flow and inlet temperature ramp.

All subsequent runs were performed with the constant density model (Model B; constant- ρ).

Effect of Pipe Size

Figures 17 through 20 show the effect of changing pipe size on the previously examined outlet temperature transients. Both transient cases were imposed on a relatively small, short pipe and then on a large, long one, using the same volume factors of $K_1 = 0.85$ and $K_2 = 0.3$.

FIGURE 17. Response to Temperature Ramp for Small Pipe

FIGURE 18. Response to Combined Ramp for Small Pipe

FIGURE 19. Response to Temperature Ramp for Large Pipe

FIGURE 20. Response to Combined Ramp for Large Pipe

The best agreement between simplified and detailed reference models occurs for the constant flow case with both pipe sizes (Figures 17 and 19). Although the "tail" of the curve disappears for the simple model's results, rates of change and delay times are in good agreement. For control study simulation which generally does not require large flow changes, the simple model should be adequate regardless of pipe size.

Combined temperature and flow ramps, however, do show some deviations as noted previously. For the small pipe case (Figure 18), the general character of the curve is maintained well, although the stored heat "tail" is diminished for the simple model. The inflection characteristic results from the early part of the transient reaching the outlet before the inlet transient is complete.

The large pipe case, however, (Figure 20) shows outlet transients differing by as much as 50 °F. Clearly, the simple simulation for large amounts of piping stored heat and long transport delays must be reviewed for the particular pipe and the expected simulated transients. From Figures 19 and 20, it appears that K_1 is reasonable at 0.85, while K_2 could be increased, perhaps to 0.5. In summary, the results of Figures 17 through 20 show that the simple model is applicable to a wide range of pipe sizes, with only minor adjustments needed in the volume K-factors.

Effect of Other Transients

The final check of simplified simulation is shown in Figures 21 and 22 for the responses to (1) a mild flow and temperature transient such as may be expected in the FFTF primary cold leg following scram, and (2) an extreme flow and temperature transient as a severe test of the model. Although deviation occurs between the reference and simplified models (up to 20 °F), the results show clearly the improvement in using the simplified model compared to the pure transport delay. With the simple model, the character of the detailed model responses is preserved and a great saving is achieved in model complexity. For simulations using limited computing equipment such as are currently used for FFTF analysis, the simplified model is an excellent choice.

FIGURE 21. Response to Mild Flow and Temperature Decay

FIGURE 22. Response to Extreme Flow and Temperature Ramps

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NOMENCLATURESymbolsUnits

A'	Heat Transfer Area Per Unit Length	ft
C	Heat Capacity	Btu/lb °F
D	Diameter	ft
h	Film Coefficient	Btu/ft ² °F sec
k	Conductivity	Btu/ft °F sec
L	Pipe Length	ft
Q	Volumetric Flow	ft ³ /sec
K	Volumetric Fraction	--
T	Temperature	°F
U	Overall Heat Transfer Coefficient	Btu/ft ² °F sec
V'	Volume Per Unit Length	ft ²
V	Pipe Volume	ft ³
w	Coolant Flow Rate	lb/sec
x	Axial Distance Along Pipe	ft
ρ	Density	lb/ft ³
τ	Transport Delay Time	sec

Subscripts

j, j-1, j+1	Mesh Point Position (General Symbol)
1...j...10	Mesh Point Property
12...jj-1...89	Nodal Property (Between Mesh Points j and j-1)
c	Coolant
d	Delayed
i	Inlet and Inside
m	Metal
o	Outlet and Outside

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