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SODEMME: A NATURAL CIRCULATION THERMAL-HYDRAULICS CODE FOR HTGR TRANSIENT ANALYSIS

LESTER G. EPEL

Date Published - August 1977

HTGR SAFETY EVALUATION DIVISION
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UPTON, NEW YORK 11973

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TABLE OF CONTENTS

	<u>Page</u>
ABSTRACT	iv
LIST OF FIGURES	v
1. INTRODUCTION	1
1.1 General Description	1
1.2 Program Execution	2
2. MAJOR FEATURES	3
3. METHOD OF SOLUTION	6
3.1 Conservation of Linear Momentum (Steady State)	6
3.2 Conservation of Mass	9
3.3 Conservation of Energy	9
3.4 Conservation of Linear Momentum (Transient)	10
3.5 Pumping, Friction, and Heat Transfer	11
3.5.1 Pressure Increase Due to a Machine	11
3.5.2 Friction Factor	12
3.5.3 Heat Transfer Rate	13
3.5.3.1 Steady State Heat Transfer	13
3.5.3.2 Time Dependent Heat Transfer	15
4. DESCRIPTION OF INPUT	18
5. SAMPLE PROBLEM	28
6. NOMENCLATURE - SYMBOLS USED IN EQUATIONS	30
ACKNOWLEDGEMENTS	32
FIGURES 1-6	33
REFERENCES	50
APPENDIX A - Subroutine HTRNSFR	52
FIGURES A1-A2	62
APPENDIX B - Geometric Relations in Plena	63

ABSTRACT

The SODEMME code provides a Solution to the One Dimensional Energy, Mass and Momentum Equations for a system of "volumes" and "segments" that comprise a circulating system containing a gaseous coolant. Input consists of a description of the system's geometry, the user's choice of some control variables, specification of the initial and time-dependent boundary conditions and setting various options for items such as pumping head, heat transfer and frictional effects. The program automatically finds the steady state solution corresponding to the initial conditions and then proceeds to compute the transient solution corresponding to the input boundary conditions.

The present code is designed to operate with helium as the coolant but can very easily be altered to function with any gas by changing a few program statements. It is somewhat unique in that it is particularly suitable for solving natural circulation problems. The program has been written in modular form so that applications to diverse systems can be accommodated in the future by rewriting the subroutine that computes the heat transfer in particular components of the system being studied. The version described in this report is applicable to the 3000 MWth HTGR design.

LIST OF FIGURES

<u>LIST OF FIGURES</u>		<u>Page</u>
Figure 1	Segment i.	33
Figure 2	User Options for Terms in the Balance Equations.	33
Figure 3	Calculational Model for Sample Problem.	34
Figure 4	Input Listing for Sample Problem.	35
Figure 5	Thermal Barrier Cover Plate Temperature Response to Graphite Surface Temperature Increase.	37
Figure 6	Output Listing for Sample Problem.	39
Figure A-1	HTGR Unit Cell.	62
Figure A-2	Equivalent One-Dimensional Cell.	62

1. INTRODUCTION

1.1 General Description

In support of the HTGR safety analysis effort, the transient natural circulation code SODEMME has been developed to predict thermal-hydraulic conditions inside the reactor primary cooling system during events such as a loss of forced circulation (LOFC) incident. The primary coolant flow rate and the reactor vessel cavity surface temperatures, which this program computes, are important in determining the amount of fission product release, first to the cavity itself and then to the containment building from the primary circuit, in the aftermath of such an occurrence.

Existing codes that deal with HTGR fluid flow and heat transfer have either been designed for specific components (e.g., SUPERHEAT¹, CORCON²) or to simulate the entire power plant (e.g., TAP³, RECA⁴, CHAP⁵) with no emphasis on the details of primary circuit surface temperature evolution. Another class of thermal-hydraulic code is available that is neither specific for a given component nor intended to simulate entire systems. These codes are general purpose programs that solve general multichannel and/or network problems (e.g., POKE⁶, FLAC⁷) but none of these have the characteristics that make them suitable to be converted into a transient, natural circulation code for HTGR application.

SODEMME was developed to provide a means of computing flow rates and surface temperatures along a series of segments that are joined together to make a complete loop (i.e. the end of the last segment joins with the beginning of the first segment). Each of the segments may have their own individual effects on the flow and temperature of the circulating helium

by virtue of their frictional and heat transfer characteristics.

The code was planned so that the small pressure (and density) changes that are inherent in natural circulation processes could be accommodated; nevertheless, it is possible to include the effects of a mechanical circulator, such as a blower, in the computation. The major limitation of the program is that multichannel flow, in which the thermal-hydraulic characteristics of the parallel paths are dissimilar, cannot be accommodated. However, if the parallel channels are identical, so that they can be replaced by an "equivalent" single segment, the problem then fits within the scope of the program.

1.2 Program Execution

SODEMME was written in Fortran IV for the Brookhaven National Laboratory CDC 7600 system, operating under Scope 2. With the present Dimension Statements it requires 32K (octal) of small core memory to execute. The running of a problem requires that the core be preset to zero and utilizes a library file called CUBIC, which extracts the three roots of a cubic algebraic equation. The running time naturally depends on the problem definition and the user options chosen during input preparation. The sample problem, described in Section 5 of this report, required 415 seconds of execution time. With the accounting system currently in effect at the BNL Central Scientific Computing Center, and using a "normal" priority to run the problem, the approximate cost of solving the sample problem was 2¢ per reactor second.

The program is available on magnetic tape or as a source or object deck from the Brookhaven National Laboratory HTGR Safety Code Library.

2. MAJOR FEATURES

SODEMME can be used to compute flow rates, temperatures, pressures, heat transfer, etc. in any closed circuit comprised of a number of connected segments which have specified geometry and frictional and heat transfer characteristics. All of the programming except the last subroutine, HTRNSFR, is rather general and written without regard to any specific system. HTRNSFR was written to simulate heat transfer in various components of the 3000 MWth HTGR primary circuit (core, plena, cross ducts, steam generator) and so is "hard-wired" for this specific design. Since the program is written in modular form, other gas-cooled systems could be simulated by replacing HTRNSFR with another subroutine written to calculate heat transfer in components of that system. Alternatively, if the amount of heat transferred in components of such a system can be stated explicitly as a function of time, subroutine HTRNSFR is not required at all. Furthermore, a system cooled by a gas other than helium could be considered by simply replacing a few of the gas constants in the code with those of the other gas.

The program usually starts a given problem by finding a steady state solution corresponding to specified initial conditions. Besides the geometric data supplied to the code during input, information on pumping, friction factor, and heat transfer for each segment must be specified. During steady state each of these quantities may be specified in either of two ways:

- . The pressure change due to a "pump" can be a constant value or it may be specified in tabular form as a function of flow rate.

- . The friction factor can be a constant value or the code can compute it automatically from Moody's⁸ correlation.
- . The heat transferred to the helium can be a constant value or can be computed by one of the pre-programmed options available in HTRNSFR.

These three quantities are important because they influence the momentum equation used by the steady state subroutine, STST, to obtain pressure drops around the circuit for a specified range of flow rates. The code ensures that for each flow rate considered, the temperature of the helium exiting the last segment equals the temperature of the helium entering the first segment within 1% (this criterion can be changed by altering EPS2 in the Fortran). After a range of flow rates and their corresponding pressure drops have been computed by STST, subroutine TERPOL selects the one flow rate that yields a net pressure drop around the complete circuit of zero. The steady state solution so derived is then utilized as the initial condition for the transient portion of the calculation.

The transient calculation, performed by subroutine TRNSNT, involves an accounting of mass and energy flows into and out of volumes. An explicit method of solution is used to advance the time for each volume and find updated values of density and specific energy at the end of the time step. These updated values facilitate the computation of thermodynamic pressures for each volume via the perfect gas laws. The new pressures, in turn, are used in the momentum equation to find new flow rates.

Because of the inherent differences in the algorithms used to obtain the steady state solution compared to those used for the transient portion,

slight discrepancies in many of the variables occur when the transient part of the problem "takes over" from the steady state solution. This artifact of the computation can be overcome by making the transient portion of the problem at first simulate the initial, time-independent conditions until all of the variables approach steady state values again. The actual transient can then be computed starting from this "new" steady state solution.

As in the case of the steady state computation, input information on pumping, friction factor, and heat transfer for each segment is essential during the transient portion of the calculation. Each of these quantities may be specified in either of three ways. The two input options used during the steady state portion of the problem are available during the transient calculation, but in addition, the pressure change due to a mechanical device, the friction factor, and the heat transfer to the helium may be specified in tabular form as a function of time. These quantities are specified for a number of discrete time steps in such a table and subroutine POLATE performs a linear interpolation to obtain values for intermediate times, as needed. The transient portion of the calculation continues by alternately updating pressures and flow rates. The code does mass and energy balances on all of the volumes, utilizing updated information on pumping, friction factor, and heat transfer and then computes thermodynamic pressures in those volumes. The code then uses these pressures to update flow rates in all of the segments. This process is continued until the specified completion time of the problem is reached.

3. METHOD OF SOLUTION

SODEMME obtains a time-dependent thermal-hydraulic description of a closed circuit by integrating a set of differential equations that describe mass, momentum, and energy conservation. Auxiliary algebraic relationships dealing with pumping, frictional effects, and heat transfer, as well as with the equation of state for helium, are used in conjunction with the conservation equations. As in any computer program, the physical problem is broken up into discrete spacial and temporal increments. The fluid circuit is divided into an integral number of "segments" and "volumes" while time is divided into time steps, during which all of the parameters of the system are assumed to be constant. The formal representation of the conservation laws and their transformation to simpler forms, which are the bases for the main algorithms in SODEMME, are presented in this section.

3.1 Conservation of Linear Momentum (Steady State)

The general momentum theorem⁹ for a control volume states that the sum of the forces acting on the control volume is equal to the net rate of momentum efflux from the control volume plus the rate of accumulation of momentum within the control volume. Formally, this can be written as*

$$\Sigma \vec{F} = \iint \vec{v} \rho (\vec{v} \cdot \vec{n}) dA + \frac{\partial}{\partial t} \iiint \rho \vec{v} dV \quad (1)$$

where $\vec{n}$ is the outward directed (from the control surface) unit normal. This vector equation can be simplified to a scalar equation in the x direction if one is willing to consider only the one-dimensional aspects:

*See listing of abbreviations used in the equations in Section 6.

$$\Sigma F_x = \iint v_x \rho (\vec{v} \cdot \vec{n}) dA + \frac{\partial}{\partial t} \iiint \rho v_x dV \quad (2)$$

During steady state the time derivative vanishes and Equation (2) becomes

$$\Sigma F_x = \iint v_x \rho (\vec{v} \cdot \vec{n}) dA \quad (3)$$

The left-hand side of Equation (3) has several possible components: pressure forces, forces due to a mechanical device in the control volume, frictional forces, and gravitational forces. If the direction of fluid flow along the control volume is the reference direction, Equation (3) can be written (see Figure 1)

$$\begin{aligned} (P_i - P_{i+1}) A + \Delta p_p A - f \frac{L}{D} \frac{w^2}{2g_c \rho A} - K \frac{w^2}{2g_c \rho_i A} \\ - \lambda A \rho \sin \theta = \frac{w^2}{g_c A} \left(\frac{1}{\rho_{i+1}} - \frac{1}{\rho_i} \right) \end{aligned} \quad (4)$$

Notice that the control volume is referred to as Segment i in Figure 1 and that the segment is associated with two volumes, i and i+1. Volume i (cross hatched in Figure 1) is comprised of one half of segment i and one half of the preceding segment. Volume i+1 consists of the other half of segment i plus one half of the succeeding segment. Certain properties such as pressure, temperature, density, internal energy, enthalpy, etc. are considered to be associated with volumes in the program and indeed, are assumed concentrated at the volume nodal points. Other parameters such as flow area, equivalent diameter, length, relative roughness, angle of inclination, orifice constant, friction factor, pump head, heat input, flow rate, etc. are associated with segments and are

constant over the length of the segment at any given time. The entire flow circuit is made up of overlapping segments and volumes of this nature. The reason for this will become apparent later when the mass and energy conservation equations are applied. While the momentum equation is applied to segments, as typified by Equation (4), the mass and energy equations are more readily applied to volumes.

Notice also that the frictional term involves the average density in the segment. The fourth term on the left hand side of Equation (4), which represents forces associated with entrance losses to the segment, depends on the fluid density at the inlet of the segment, however. This is somewhat arbitrary in that a force representing outlet losses from the segment could also be included. Instead, this latter term is included as the inlet loss for the succeeding segment.

Equation (4) is combined with the gas laws so that it can be rewritten in terms of pressures and temperatures rather than in terms of pressures and densities. The temperatures are dependent only on the heat transferred to the segment and are thus readily calculable. By making use of such auxiliary relations, and considering P_i to be known, Equation (4) can be cast into a cubic equation in P_{i+1} . During execution of the steady state portion of SODEMME, this cubic equation is solved segment by segment, starting with the input value of P_i for the first segment, until P_{i+1} for the last segment is found. The program repeats this procedure for a range of flow rates and builds up a catalog of flow versus circuit-pressure-drop values. For a closed loop P_{i+1} for the last segment must equal P_i for the first segment. Subroutine TERPOL finds the flow rate that satisfies this condition by linear interpolation

among the cataloged values. This flow rate and the corresponding pressures, temperatures, densities, etc. constitute the initial conditions for the transient portion of the computation.

3.2 Conservation of Mass

With respect to the control volume of Section 3.1, the law of conservation of mass states that the rate of accumulation of mass within the control volume equals the net rate of mass influx into that control volume. This rather obvious fact can be formalized as

$$\iint \rho (\vec{v} \cdot \vec{n}) dA + \frac{\partial}{\partial t} \iiint \rho dV = 0 \quad (5)$$

The integral form of Equation (5) applied to volume i in the one-dimensional approximation is

$$\frac{\partial m_i}{\partial t} = w_{i-1} - w_i \quad (6)$$

and is readily calculable since the flow rates in segments $i-1$ and i are known. For a given time step, then, an updated mass (and therefore density) can be derived for each volume.

3.3 Conservation of Energy

One statement of the familiar first law of thermodynamics is¹⁰ that the difference between heat added to a control volume and the work done by the fluid in a control volume equals the net rate of energy efflux from the volume plus the rate of energy accumulation within the volume. The formal representation of this law is

$$\frac{\delta Q}{dt} - \frac{\delta W}{dt} = \iint e \rho (\vec{v} \cdot \vec{n}) dA + \frac{\partial}{\partial t} \iiint e \rho dV \quad (7)$$

where e is the specific total energy and Q and W are the heat into and work out of the control volume, respectively. The work term consists of

pressure work, shaft work, and shear work. If (as is true for the control volumes used in SODEMME) all parts of the control surface not at fixed walls are normal to the velocity of any fluid crossing the surface, the shear work is zero. Further, in the absence of electricity, magnetism, and capillarity, the specific total energy is comprised of kinetic, potential, and internal energies. The first two generally can be neglected in comparison with the third in this type of application and this approximation serves to further simplify Equation (7). The integrated, one-dimensional form of the energy equation used for volume i becomes

$$\frac{\partial(m_i u_i)}{\partial t} = w_{i-1} h_{i-1} - w_i h_i + \frac{\delta Q_i}{dt} - \frac{\delta S_i}{dt} \quad (8)$$

where h is the specific enthalpy and S is the shaft work. In natural circulation cases the shaft work is zero and consequently the last term can be neglected*. Equation (8) provides a means of updating the internal energy in the volumes that make up the loop during successive time steps, given the flow rates, enthalpies, and heat transferred. The internal energy so found, together with the updated density from Equation (6), makes it possible to compute an updated thermodynamic pressure for each volume. This pressure is then used to compute a new flow rate as described in the following section.

3.4 Conservation of Linear Momentum (Transient)

The one-dimensional momentum equation, represented by Equation (2), can be written in integral form for the time dependent case using the same

*Even when there is shaft work supplied to the gas by devices such as fans or blowers, that amount of energy usually is negligible compared to the other energy terms.

principles as in Section 3.1

$$\begin{aligned}
 (P_i - P_{i+1}) A + \Delta p_p A - f \frac{\ell}{D} \frac{w^2}{2g_c \rho A} - K \frac{w^2}{2g_c \rho_i A} \\
 - \ell A \bar{\rho} \sin \theta = \frac{w^2}{g_c A} \left(\frac{1}{\rho_{i+1}} - \frac{1}{\rho_i} \right) + \frac{\ell}{g_c} \frac{\partial w}{\partial t}
 \end{aligned} \tag{9}$$

The time dependent term, which was not present in the steady state form (see Equation [4]), can be evaluated because all of the other terms in Equation (9) are now known. In this manner, new flow rates are established for each of the segments and then the cycle of updating densities and internal energies in the volumes is repeated. These lead to new volume pressures, which lead to new segment flow rates and so on.

3.5 Pumping, Friction, and Heat Transfer

The balance equations are used to continually update the quantities of interest in the circulating system as described above. However, the momentum equation requires values of pressure rise due to a mechanical device and of friction factors; the energy equation requires values of heat transferred to the volumes. These values are supplied as described below and a summary of the user's options for supplying these terms to the code is given in Figure 2.

3.5.1 Pressure Increase Due to a Machine

During the steady state portion of the calculation, when the initial conditions are being found, there are two ways in which a pressure increase due to a mechanical device, such as a blower or circulator, can be specified. The user can specify some constant value during input preparation or alternatively may specify a curve of pressure increase

versus flow rate. This curve is specified by inputting up to 10 pairs of data points. The code will interpolate to obtain pressure increases for flow rates between the tabular values by using subroutine POLATE*.

During the transient portion of the calculation both of these input alternatives are available together with a third. The user may, if he desires, specify a pressure increase versus time curve in the same way as the pressure increase versus flow rate curve is specified.

3.5.2 Friction Factor

The friction factor used in the momentum equation is defined by the Darcy-Weisbach¹² equation

$$\Delta p = f \frac{L}{D} \frac{\rho v^2}{2g_c} \quad (10)$$

During the steady state portion of the computation there are two ways of specifying this parameter. The user has the choice of specifying some constant value during input preparation or, if he wishes, allowing the code to automatically compute f . This is done by subroutine FRICT**. Subroutine STST, using current values of pressure, temperature, and flow rate for any segment, calculates the corresponding Reynold's number for that segment. This information, together with the relative roughness for the segment, is used by FRICT to find the friction factor by reference to Moody's tables.

During the transient part of the computation the two input

*This subroutine was adapted from the LWR thermal-hydraulic code RELAP-3B.

See Reference 11.

**This subroutine was adapted from the fluid flow analysis code FLAC.

See reference 7.

specification choices are still available and in addition there is a third choice. Again, the user may specify a friction factor versus time curve by inputting up to 10 pairs of points that describe the curve.

3.5.3 Heat Transfer Rate

3.5.3.1 Steady State Heat Transfer

The heat transfer rate required by the energy conservation equation, is provided to the code during the steady state computation in one of two ways. The first method is by specifying some constant value of heat rate. The alternative is to allow the code to compute the heat transferred to the various segments via the built-in heat transfer algorithms contained in subroutine HTRNSFR. During the steady state analysis there are four options the user has available. The models for these options (and for the six options available during the transient analysis) are detailed in Appendix A and so will be only outlined here. These algorithms were written specifically with the 3000 MWth HTGR in mind and so can legitimately be used only in such a system. If a different system is under study the user must either specify the heat transfer rate explicitly or re-program this subroutine to reflect the characteristics of the other design.

Besides computing the segment heat transfer rates, the algorithms in HTRNSFR also return calculated segment surface temperatures. Of course if HTRNSFR is not called upon during program execution, the surface temperatures are not computed by the code and asterisks will appear in the output where surface temperatures are usually printed.

A brief description of the four steady state options is given here.

Option 1 - This option applies to the coolant channels in the core. The model is based on flow in a constant area, constant surface temperature channel with a constant heat transfer coefficient. The graphite channel surface temperature is specified during input and the fluid temperature tends to approach this temperature asymptotically.

Option 2 - This option is based on flow in ducts that are lined with an HTGR-type thermal barrier. The temperature dependence of the kaowool thermal conductivity is taken into account and it is assumed that the cool side of the barrier is maintained at 100°F.

Option 3 - This option computes the heat transfer to the helium in the plenum when the heat transfer rate from the graphite reflector to the helium is specified during input. It provides the heat transfer contribution from the thermal barrier which is modeled as a circular disc whose cool side is maintained at 100°F. Only convective heat transfer to the helium is considered from the thermal barrier and reflector surfaces. The lesser of these two surface temperatures is returned as the surface temperature for the segment. The method of defining segment parameters such as flow area, equivalent diameter, and length for this type of segment, which is needed during the input phase, is discussed in Appendix B.

Option 4 - This option is equivalent to Option 3 with only one difference. Instead of specifying the heat transfer contribution of the reflector to the helium explicitly, this option requires the reflector surface temperature to be specified during input instead. The rest of the modeling is as in Option 3.

3.5.3.2 Time Dependent Heat Transfer

During the transient portion of the computation the heat transfer rate is supplied to subroutine TRNSNT in one of three ways for each segment. The first is, as in the case of steady state, by specifying some constant value during input. The code considers this constant to be applicable to the entire course of the time-dependent calculation if this option is chosen. The second alternative is to provide a curve of heat transfer rate versus time by specifying up to 10 pairs of points, just as in the time-dependent specification of pressure increase and/or friction factor. The third alternative is to allow the program to compute the transient heat transfer rate. Six options are provided for the user if he chooses this alternative as described below.

Option 1 - This option is the time-dependent analog of Option 1 in Section 3.5.3.1. Instead of specifying the graphite channel surface temperature as some constant value during input as was done there, however, the channel surface temperature must be specified as a function of time. Up to 10 pairs of data points may be prescribed to define this function.

Option 2 - This option is the time-dependent analog of Option 2 in Section 3.5.3.1. First, the thermal barrier cover plate initial temperature is found from the steady state conditions. Then, assuming that all of the thermal resistance is in the insulation and all of the thermal capacitance is in the cover plate, the cover plate temperature and heat transfer to the helium is computed as a function of time by a forward difference technique. In this model the thermal conductivity of the insulation is assumed to be a constant. This treatment

of the thermal barrier is used in Options 3 and 4, below also.

Option 3 - This option is the time-dependent analog of Option 3 in Section 3.5.3.1. Instead of specifying the heat transfer rate from the reflector as a constant, however, this heat transfer rate (which is subsequently combined with that from the thermal barrier) is specified as a function of time using up to 10 pairs of data points. The graphite reflector surface temperature is computed and compared to the thermal barrier cover plate temperature. The smaller of these is returned to TRNSNT as the segment surface temperature.

Option 4 - This option is the time-dependent analog of Option 4 in Section 3.5.3.1. Instead of specifying the reflector surface temperature as a constant, however, this parameter must be stipulated as a function of time, using up to 10 pairs of data points. At any time the smaller of the two surface temperatures (thermal barrier or graphite reflector) is returned as the significant surface temperature.

Option 5 - This option provides a primitive model for the transient temperature behavior in the steam generators/reheaters. The entire assemblage of tubes, sheets, and shells in the economizers, evaporators, superheaters, and reheaters has been homogenized in this model. It is assumed that no steam/water is present in the assemblies during this portion of the transient. The metal is considered to be isothermal and changes temperature with time due solely to convective heat transfer with the helium. The thermal capacitance of the heat exchanger metal is thus taken into account in a rudimentary fashion.

Option 6 - This option simulates transient heat transfer in the HTGR core. A one-dimensional cell was designed that approximates the two-dimensional triangular cell typically used in HTGR heat transfer calculations. First the initial temperature profile is computed through the fuel and graphite in this cell. Then, from a power versus time curve supplied during input, the transient temperature distribution through the fuel and graphite is computed. The heat transfer to the helium is found at each time step by the convective boundary condition at the graphite-helium interface.

During the execution of these 6 time-dependent heat transfer options the user has the capability of using a time step larger than the time step used in the other parts of SODEMME. In this way he can avoid having the code do the laborious heat transfer calculations every time the mass, momentum, and energy equations are updated. The only restriction on the size of this time step has to do with numerical instabilities that could occur. This question is addressed in Appendix A where criteria on maximum time step sizes are developed. It is shown there that, in general, the stability criteria are not unduly restrictive for practical situations encountered in the HTGR and that time steps on the order of 100 times the "basic" time step are reasonable for the heat transfer calculations.

It is noted that when the user takes advantage of this opportunity to decrease computing time, the program updates the flow- or time-dependent pressure increase (Section 3.5.1), the flow- or time-dependent friction factor (Section 3.5.2), and the time-dependent

heat transfer rate (Section 3.5.3.2) according to the same schedule. That is, if the heat transfer rate is updated every 50 time steps, for example, an updated friction factor will also be computed every 50 time steps, if automatic calculation of the friction factor is another of the chosen options.

4. DESCRIPTION OF INPUT

This section describes the input requirements for SODEMME. The input is divided into the data needed for solution of the steady state portion and that needed for the transient part. It is possible to run only one or the other of these portions of the code, if desired.

There are, in general, 11 types of input cards required for the steady state part and 13 types of cards for the transient part. Some of these are not always necessary, depending on the problem definition. Most card types involve several individual cards, depending on the number of segments that require input in that category. When more than one card is required, they are inserted consecutively, starting with the one for the first segment and continuing with any others that may be required, until the one for the last segment has been included. The only card types that invariably need a single card are card types 1, 2, 11, 12, and 24. All of the others could, depending on the specific problem, require more than a single card.

The details of the input requirements follow. The first 11 card types are related to the steady state portion of the problem; the next 13 card types are concerned with the transient part.

1. Title Card

8A10

TITLE Any 80 column (or less) alphanumeric description
to be printed as the heading on the output.

2. Control Parameter Card

3F10.5, 4I10

WFIRST The smallest flow rate to be used in the steady
state computation (pounds/sec).

WLAST The largest flow rate to be used in the steady
state computation (pounds/sec).

DELTAW The difference between successive flow rates used
in the series of computations starting with WFIRST
(pounds/sec).

NMAX The number of segments in the representation of the
circuit. Up to a maximum of 30.

NOSTST If the user does not wish to compute the steady
state solution, this parameter is set to 1.

NOTRAN If the user does not wish to compute the transient
solution, this parameter is set to 1.

IPUNCH If relevant data at the end of the transient calcu-
lation is to be punched for possible restart of the
problem, this parameter is set to 1.

3. Segment Parameter Card
(one for each segment)

I10, A10, 6F10.5

J Segment number (1 through 30).

SNAME Segment name (up to 10 characters).

A Cross sectional flow area of the segment (inch²).

D Equivalent diameter of the segment (inches).

L Length of the segment (feet).

K Entrance loss coefficient of the segment
(dimensionless).

RELRF The segment relative roughness (dimensionless).

THETA The angle the segment makes, in the direction of
flow, with the rightward pointing horizontal
vector (degrees).

If NOSTST=1, then card types 4 through 11 are not required and the next card is card type 12.

4. Segment Indicators Card 7I10, F10.5
(one for each segment)

INDPC If the pressure increase due to a mechanical device
for this segment is to be a constant, this indicator
is set to 1.

INDPVW If the pressure increase due to a mechanical device
for this segment is to be tabulated against flow
rate, this indicator is set to 1.

INDFC If the friction factor for this segment is to be a
constant, this indicator is set to 1.

INDFVW If the friction factor for this segment is to be
computed automatically by the code, this indicator is
set to 1.

INDQC If the heat transfer for this segment is to be a
constant, this indicator is set to 1.

INDQVW If the heat transfer for this segment is to be computed
automatically by the code, this indicator is set to 1.

NOPTSS If INDQVW=1, this parameter must be set to the
desired steady state heat transfer option
(1 through 4).

HTCSS If INDQVW=1, the steady state heat transfer coefficient
is required (Btu/hr ft² F).

If INDPG=1 for any segment, a card of the following type for each such
segment is required.

5. Constant Pressure Increase Card F10.5

PPUMP Pressure increase (pounds/in²).

If INDFC=1 for any segment, a card of the following type for each such
segment is required.

6. Constant Friction Factor Card F10.5

F Friction factor (dimensionless).

If INDQC=1 for any segment, a card of the following type for each such
segment is required.

7. Constant Heat Rate Card F10.5

Q Heat transfer rate to the helium (Btu/sec).

If INDPVW=1 for any segment, cards of the following type for each such
segment are required*.

8. Pressure Increase vs. Flow Card I10, 10X, 6F10.5/(8 F10.5)

NE The number of entries in the table (2 times the

*As many as 3 cards may be needed to specify the 10 pairs of data points
that could be used to define a complete table for each segment. If
exactly 3 pairs of data points are used for any segment, the single card
used should be followed by one blank card.

number of data points).

PVW(1) Pressure increase (pounds/in²).

PVW(2) Flow rate (pounds/sec).

PVW(3) Pressure increase (pounds/in²).

PVW(4) Flow rate (pounds/sec).

etc.

If INDQVW=1 and NOPTSS=3 for any segment, a card of the following type for each such segment is required.

9. Reflector Heat Rate Card F10.5

QG The rate of heat transfer from the graphite reflector to the helium in the plenum (Btu/sec).

If INDQVW=1 and NOPTSS=1 or 4 for any segment, a card of the following type for each such segment is required.

10. Graphite Surface Temperature Card F10.5

TSG The graphite surface temperature (F).

11. Inlet Pressure/Temperature Card 2F10.5

P(1) The inlet pressure to the first segment (pounds/in²).

T(1) An estimate of the inlet temperature to the first segment (F).

This concludes the input for the steady state portion of the program.

The remaining cards constitute the input for the transient part.

12. Control Parameter Card 3F10.5, 2I10

TFIRST The time at which the transient portion is considered to begin. This will be zero except for restart problems (sec).

TLAST	The time at which it is desired to end the problem (sec).
DELTAT	The time step used in advancing the history of the problem (sec).
ICALC	The number of time steps that shall pass before pumping head, friction factors, and/or heat transfer effects are updated.
IPRINT	The number of time steps that shall pass before the variables are printed as output.

13. Segment Indicators Card
(one for each segment)

10I5, F10.5

INDPC	If the pressure increase due to a mechanical device for this segment is to be a constant, this indicator is set to 1.
INDPVW	If the pressure increase due to a mechanical device for this segment is to be tabulated against flow rate, this indicator is set to 1.
INDPVT	If the pressure increase due to a mechanical device for this segment is to be tabulated against time, this indicator is set to 1.
INDFC	If the friction factor for this segment is to be a constant, this indicator is set to 1.
INDFVW	If the friction factor for this segment is to be computed automatically by the code, this indicator is set to 1.

INDFVT If the friction factor for this segment is to be tabulated against time, this indicator is set to 1.

INDQC If the heat transfer for this segment is to be a constant, this indicator is set to 1.

INDQVW If the heat transfer for this segment is to be computed automatically by the code, this indicator is set to 1.

INDQVT If the heat transfer for this segment is to be tabulated against time, this indicator is set to 1.

NOPTTR If INDQVW=1, this parameter must be set to the desired transient heat transfer option (1 through 6).

HTCTR If INDQVW=1, the transient heat transfer coefficient is required (Btu/hr ft² F).

If INDPC=1 for any segment, a card of the following type for each such segment is required.

14. Constant Pressure Increase Card F10.5

PPUMP Pressure increase (pounds/in²).

If INDFC=1 for any segment, a card of the following type for each such segment is required.

15. Constant Friction Factor Card F10.5

F Friction factor (dimensionless).

If INDQC=1 for any segment, a card of the following type for each such segment is required.

16. Constant Heat Rate Card F10.5

Q Heat transfer rate to the helium (Btu/sec).

If INDPVW=1 for any segment, cards of the following type for each such segment are required*.

17. Pressure Increase vs. Flow Card I10, 10X, 6F10.5/(8F10.5)

NE The number of entries in the table (2 times the number of data points).

PVW(1) Pressure increase (pounds/in²).

PVW(2) Flow rate (pounds/sec).

PVW(3) Pressure increase (pounds/in²).

PVW(4) Flow rate (pounds/sec).

etc.

If INDPVT=1 for any segment, cards of the following type for each such segment are required*.

18. Pressure Increase vs. Time Card I10, 10X, 6F10.5/(8F10.5)

NE The number of entries in the table (2 times the number of data points).

PVT(1) Pressure increase (pounds/in²).

PVT(2) Time (sec).

PVT(3) Pressure increase (pounds/in²).

PVT(4) Time (sec).

etc.

If INDFVT=1 for any segment, cards of the following type for each such segment are required*.

19. Friction Factor vs. Time Card I10, 10X, 6F10.5/(8F10.5)

NE The number of entries in the table (2 times the

*See footnote on bottom of page 21.

number of data points).

FVT(1) Friction factor (dimensionless).

FVT(2) Time (sec).

FVT(3) Friction factor (dimensionless).

FVT(4) Time (sec).

etc.

If INDQVT=1 for any segment, cards of the following type for each such segment are required*.

20. Heat Rate vs. Time Card I10, 10X, 6F10.5/(8F10.5)

NE The number of entries in the table (2 times the number of data points).

QVT(1) Heat transfer rate to the helium (Btu/sec).

QVT(2) Time (sec).

QVT(3) Heat transfer rate to the helium (Btu/sec).

QVT(4) Time (sec).

etc.

If INDQVW=1 and NOPTTR=3 for any segment, cards of the following type for each such segment are required*.

21. Reflector Heat Rate vs. Time Card I10, 10X, 6F10.5/(8F10.5)

NE The number of entries in the table (2 times the number of data points).

QGV(1) Heat transfer rate to the helium (Btu/sec).

QGV(2) Time (sec).

QGV(3) Heat transfer rate to the helium (Btu/sec).

*See footnote on bottom of page 21.

QGV(4) Time (sec).

etc.

If INDQVW=1 and NOPTTR=1 or 4 for any segment, cards of the following type for each such segment are required*.

22. Graphite Surface Temperature vs. Time Card I10, 10X, 6F10.5/(8F10.5)

NE The number of entries in the table (2 times the number of data points).

TSGVT(1) Graphite surface temperature (F).

TSGVT(2) Time (sec).

TSGVT(3) Graphite surface temperature (F).

TSGVT(4) Time (sec).

etc.

If INDQVW=1 and NOPTTR=5 for any segment, a card of the following type for each such segment is required.

23. Initial SG Metal Temperature Card F10.5

TSTEEL Initial temperature of the steel in the steam generators/reheaters (F). If this is made equal to zero (or left blank) the helium temperature at time = 0 will be assigned to the metal.

If INDQVW=1 and NOPTTR=6 for any segment, a card of the following type for each such segment is required.

24. Fractional Power vs. Time Card I10, 10X, 6F10.5/(8F10.5)

NE The number of entries in the table (2 times the number of data points).

*See footnote on bottom of page 21.

FRPOW(1)	Fraction of 3000 MWth.
FRPOW(2)	Time (sec).
FRPOW(3)	Fraction of 3000 MWth.
FRPOW(4)	Time (sec).
etc.	

If the case being run is a re-start of a previous problem (i.e. NOSTST=1) then the cards that were punched by the previous run, that contain relevant data for continuing the problem, are included here (after Card Type 24).

5. SAMPLE PROBLEM

The computational model for an illustrative problem is depicted in Figure 3. This model consists of 9 segments representing, in order of flow direction during natural circulation, the bottom plenum, the core, the upper plenum, the upper cross ducts, the outer flow annulii in the steam generators, the main steam generator bundles, the inner flow annulii in the steam generators, the reheater bundles, and the lower cross ducts of an HTGR.

The numerical values used in the segment parameter cards are not intended to accurately represent the 3000 MWth HTGR. It proved very difficult to obtain design values for such components as the main steam generator bundle, for example, and so various types of information from diverse sources were blended together. For example, reference 1, pages 4-6, states that the design Reynold's number for the helium side in the main steam generator is on the order of 40000. Also, reference 13 gives

the design helium velocity in the Fort St. Vrain main coil as 80 fps. Assuming that this velocity is representative of the HTGR main steam bundle, these two pieces of information make it possible to estimate the equivalent diameter of this segment in the model. Although some of the numerical values used may only approximate the true values, this sample problem nevertheless serves to illustrate the application of the computer program.

In the sample problem, the initial conditions are that the lower reflector, core, and upper reflector graphite are at 1000 F, 1500 F, and 1700 F, respectively, and that the main steam bundle removes 5000 Btu/second from the helium. Heat transfer Option 2 was selected for the upper and lower cross ducts, Option 1 was used for the core, and Option 4 for the top and bottom plena. Segments 5, 7, and 8 were assumed to be adiabatic. There was no pumping action in any segment and the friction factors were specified explicitly*.

After the steady state solution was derived, it was desired to let the code find a "new" steady state (see page 4) via the algorithm in subroutine TRNSNT. Thus the initial conditions were maintained for the first 1000 seconds into the transient. For the next 900 seconds (i.e. quarter hour) the lower reflector, core, and upper reflector graphite temperatures were assumed to ramp linearly to 1200 F, 1700 F, and 1900 F, respectively.

*Actually, the sample problem was first run utilizing the option that calculates friction factors automatically. Then, in order to save computing time, the values of friction factors so computed by the program were used for the sample problem.

All of the other boundary conditions were assumed to remain unchanged.

The listing of the input for this problem is shown in Figure 4.

The resulting thermal barrier cover plate temperatures are shown in Figure 5 for this 15 minute transient. It is clear that the response of these surfaces is rather sluggish compared to the 200°F rise that the graphite surfaces underwent during the same time span. Because of this lag in following the increase in graphite temperature, the heat rejecting parts of the system, although increasing the amount of heat they extract from the helium, cannot keep pace with the increase in the amount of heat supplied to the helium. Reference to the next-to-last column in the output (Figure 6) for the transient part of the calculation indicates that whereas the net amount of heat into the gas was essentially zero at the beginning of the ramp, this parameter had risen to about 700 Btu/second at the end of the 15 minute time interval. The corresponding rise in fluid temperature and pressure throughout the system is evident from the second and third columns in the output.

6. NOMENCLATURE - SYMBOLS USED IN EQUATIONS

Symbols

a	constant used in thermal conductivity equation
A	area for heat transfer or fluid flow
b	constant used in thermal conductivity equation
C _p	specific heat of gas
C _s	specific heat of solid
D	equivalent diameter

f	friction factor
F	force
g	volumetric heat generation rate
g_c	gravitational constant
h	heat transfer coefficient
k	thermal conductivity
K	entrance loss coefficient
l	length of segment
m	mass of fluid
M	mass of steel
Δp	pressure difference
p	wetted perimeter of segment
P	pressure
q	rate of heat transfer
s	distance between nodal points
Δt	time step
T	temperature
Δx	thickness of kaowool insulation
u	internal energy
v	velocity
V	volume
w	rate of mass flow

Subscripts

c	coolant water
f	fluid (helium)

h	for heat transfer
i	beginning of segment (also used as an index)
l	left side
o	end of segment
p	due to pump
r	right side
s	at the surface

Greek

δ	thickness of thermal barrier cover plate
ρ	density
θ	angle of inclination in direction of flow

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Figure 1. Segment 1.

(Steady State Portion)	
Pressure Increase	Constant
	Tabulated as a Function of Flow Rate
Friction Factor	Constant
	Computed in Subroutine FRICT
Heat Transfer	Constant
	Computed in Subroutine HTRNSFR
(Transient Portion)	
Pressure Increase	Constant
	Tabulated as a Function of Flow Rate
	Tabulated as a Function of Time
Friction Factor	Constant
	Computed in Subroutine FRICT
	Tabulated as a Function of Time
Heat Transfer	Constant
	Computed in Subroutine HTRNSFR
	Tabulated as a Function of Time

Figure 2. User options for terms in the balance equations.

Figure 3. Calculational model for sample problem.

Figure 4. Input listing for sample problem.

SAMPLE PROBLEM FOR USER MANUAL							
7.0	10.0	1.0	9	0	0	0	
1BOT.PLENUM	48280.	150.	17.0	15.8	.00001	-180.	
2CORE	18581.	.826	28.6	3.34	.002	90.	
3TOP PLENUM	48280.	150.	17.0	238.7	.00001	0.	
4TOP DUCTS	24295.	71.8	10.8	.5	.00002	0.	
5ANNULUS	9600.	20.	21.	1.	.00008	-90.	
6MAIN STEAM	28800.	.9	21.	2.	.00015	90.	
7CENTERTUBE	19200.	40.	33.6	2.	.00004	-90.	
8REHEATERS	26400.	1.5	5.	2.	.00015	90.	
9BOT. DUCTS	24295.	71.8	10.8	30.	.00002	-180.	
1	0	1	0	1	4	2.0	
1	0	1	0	1	1	17.0	
1	0	1	0	1	4	2.0	
1	0	1	0	1	2	4.0	
1	0	1	0	1	0		
1	0	1	0	1	0		
1	0	1	0	1	0		
1	0	1	0	1	0		
1	0	1	0	1	2	4.0	
0.0							
0.0							
0.0							
0.0							
0.0							
0.0							
0.0							
0.0							
.029							
.384							
.030							
.031							
.034							
.546							
.032							
.273							
.029							
0.							
-5000.							
0.							
0.							
1000.							
1500.							
1700.							
720.	900.						
0.0	1900.	.001	100	100000			
1	0	0	0	1	0	4	2.0
1	0	0	0	0	1	0	17.0
1	0	0	0	1	0	4	2.0
1	0	0	0	0	1	0	4.0
1	0	0	1	0	0	0	
1	0	0	1	0	0	0	
1	0	0	1	0	0	0	
1	0	0	1	0	0	0	
1	0	0	1	0	0	0	
1	0	0	1	0	0	2	4.0
0.0							
0.0							
0.0							
0.0							
0.0							
0.0							
0.0							
0.0							
0.0							
0.0							
0.0							
.029							
.384							
.030							
.031							
.034							
.546							
.032							
.273							
.029							
0.							
-5000.							
0.							
0.							
6	1000.	0.0	1000.	1000.	1200.	1900.	
6	1500.	0.0	1500.	1000.	1700.	1900.	
6	1700.	0.0	1700.	1000.	1900.	1900.

Figure 5. Thermal barrier cover plate temperature response to graphite surface temperature increase.

Figure 6. Output listing for sample problem.

SAMPLE PROBLEM FOR USER MANUAL

IN THIS PROBLEM THERE ARE 9 VOLUMES/SEGMENTS

THE FOLLOWING TABLE SUMMARIZES THE GEOMETRIC DESCRIPTION OF THE CIRCULATING SYSTEM

SEG. NO.	SEG. NAME	AREA (SQ IN)	EQ. DIA. (IN)	LENGTH (FT)	LOSS COEF.	REL. ROUGH.	INCLIN. (DEG)
1	ROT. PLENUM	48280.00000	150.00000	17.00000	15.80000	.00001	-180.00000
2	CORF	18581.00000	.82600	28.60000	3.34000	.00200	90.00000
3	TOP PLENUM	48280.00000	150.00000	17.00000	238.70000	.00001	0.00000
4	TOP DICTS	24295.00000	71.80000	10.80000	.50000	.00002	0.00000
5	ANNULUS	9600.00000	20.00000	21.00000	1.00000	.00008	-90.00000
6	MAIN STEAM	28800.00000	.90000	21.00000	2.00000	.00015	90.00000
7	CENTERTUBE	19200.00000	.800000	33.60000	2.00000	.00004	-90.00000
8	REFEATERS	26400.00000	1.50000	5.00000	2.00000	.00015	90.00000
9	ROT. DICTS	24295.00000	71.80000	10.80000	30.00000	.00002	-180.00000

THE OPTIONS USED IN THE STEADY STATE PORTION OF THIS PROBLEM ARE GIVEN IN THE TABLE BELOW

NO.	CONST. HEAD	HEAD VS FLOW	CONST. FF	FF VS FLOW	CONST. HEAT	HEAT VS CALC	HT TRANS OPT	HTC (BTU/HR SQ-FT FT)
1	1	0	1	0	0	1	4	2.0000
2	1	0	1	0	0	1	1	17.0000
3	1	0	1	0	0	1	4	2.0000
4	1	0	1	0	0	1	2	4.0000
5	1	0	1	0	1	0	-0	-0.0000
6	1	0	1	0	1	0	-0	-0.0000
7	1	0	1	0	1	0	-0	-0.0000
8	1	0	1	0	1	0	-0	-0.0000
9	1	0	1	0	0	1	2	4.0000

FOR THE STEADY STATE PORTION OF THIS PROBLEM THE CONSTANT PUMP HEAD, FRICTION FACTOR, AND HEAT TRANSFER OPTIONS ARE SHOWN BELOW

SEGMENT NO.	PUMP HEAD (LBS/SQ-FT)	FRICTION FACTOR	HEAT INPUT (BTU/SEC)
1	0.0000	.0290	*****
2	0.0000	.3840	*****
3	0.0000	.0300	*****
4	0.0000	.0310	*****
5	0.0000	.0340	0.0000
6	0.0000	.5460	-5000.0000
7	0.0000	.0320	0.0000
8	0.0000	.2730	0.0000
9	0.0000	.0290	*****

THE RESULTS FOR A FLOW RATE OF 7.00000 POUNDS PER SECOND ARE AS FOLLOWS

Figure 6 (continued).

NO.	SFG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1	ROT. PLENUM	859.6636	720.0000	0.0000	.0290	2.0000	-29.6343	660.8355
2	COPE	856.2768	720.0000	0.0000	.3840	17.0000	5632.5781	1500.0000
3	TOP PLENUM	1500.0000	719.9661	0.0000	.0300	2.0000	-105.6660	1091.4386
4	TOP DUCTS	1487.9239	719.9660	0.0000	.0310	4.0000	-333.7673	1241.3249
5	ANNULUS	1449.7791	719.9660	0.0000	.0340	0.0000	0.0000	*****
6	MAIN STEAM	1449.7791	719.9863	0.0000	.5460	0.0000	-5000.0000	*****
7	CENTERTURE	878.3505	719.9615	0.0000	.0320	0.0000	0.0000	*****
8	REHFATERS	878.3505	720.0080	0.0000	.2730	0.0000	0.0000	*****
9	ROT. DUCTS	878.3505	720.0011	0.0000	.0290	4.0000	-163.5105	757.5432

THE TOTAL LOOP INVENTORY IS 5106.76 POUNDS
THE CIRCUIT CYCLE TIME IS 729.54 SECONDS

THE RESULTS FOR A FLOW RATE OF 8.00000 POUNDS PER SECOND ARE AS FOLLOWS

NO.	SFG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1	ROT. PLENUM	937.6808	720.0000	0.0000	.0290	2.0000	-81.1769	715.1362
2	COPE	929.5631	720.0000	0.0000	.3840	17.0000	5704.3693	1500.0000
3	TOP PLENUM	1500.0000	719.9670	0.0000	.0300	2.0000	-105.6660	1091.4386
4	TOP DUCTS	1489.4334	719.9669	0.0000	.0310	4.0000	-334.2296	1242.4928
5	ANNULUS	1456.0104	719.9669	0.0000	.0340	0.0000	0.0000	*****
6	MAIN STEAM	1456.0104	719.9872	0.0000	.5460	0.0000	-5000.0000	*****
7	CENTERTURE	956.0104	719.9632	0.0000	.0320	0.0000	0.0000	*****
8	REHFATERS	956.0104	720.0071	0.0000	.2730	0.0000	0.0000	*****
9	ROT. DUCTS	956.0104	720.0005	0.0000	.0290	4.0000	-183.2969	820.5843

THE TOTAL LOOP INVENTORY IS 4933.08 POUNDS
THE CIRCUIT CYCLE TIME IS 616.63 SECONDS

THE RESULTS FOR A FLOW RATE OF 9.00000 POUNDS PER SECOND ARE AS FOLLOWS

NO.	SFG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1	ROT. PLENUM	998.7250	720.0000	0.0000	.0290	2.0000	-121.6975	757.2458
2	COPE	987.9074	720.0000	0.0000	.3840	17.0000	5761.0415	1500.0000
3	TOP PLENUM	1500.0000	719.9677	0.0000	.0300	2.0000	-105.6660	1091.4386
4	TOP DUCTS	1490.6075	719.9675	0.0000	.0310	4.0000	-334.5893	1243.4012
5	ANNULUS	1460.8662	719.9675	0.0000	.0340	0.0000	0.0000	*****
6	MAIN STEAM	1460.8662	719.9878	0.0000	.5460	0.0000	-5000.0000	*****
7	CENTERTURE	1016.4218	719.9643	0.0000	.0320	0.0000	0.0000	*****
8	REHFATERS	1016.4218	720.0064	0.0000	.2730	0.0000	0.0000	*****
9	ROT. DUCTS	1016.4218	720.0001	0.0000	.0290	4.0000	-199.0887	869.3280

Figure 6 (continued).

THE TOTAL LOOP INVENTORY IS 4808.36 POUNDS
THE CIRCUIT CYCLE TIME IS 534.26 SECONDS

THE RESULTS FOR A FLOW RATE OF 10.00000 POUNDS PER SECOND ARE AS FOLLOWS

NO.	SEG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1	ROT. PLENUM	1047.7988	720.0000	0.0000	.0290	2.0000	-154.3913	790.8632
2	CORF	1035.4475	720.0000	0.0000	.3840	17.0000	5806.9062	1500.0000
3	TOP PLENUM	1500.0000	719.9681	0.0000	.0300	2.0000	-105.6660	1091.4386
4	TOP DUCTS	1491.5467	719.9679	0.0000	.0310	4.0000	-334.8771	1244.1278
5	ANNULUS	1464.7566	719.9679	0.0000	.0340	0.0000	0.0000	*****
6	MAIN STEAM	1464.7566	719.9881	0.0000	.5460	0.0000	-5000.0000	*****
7	CENTERTURE	1064.7566	719.9650	0.0000	.0320	0.0000	0.0000	*****
8	REFRATERS	1064.7566	720.0058	0.0000	.2730	0.0000	0.0000	*****
9	ROT. DUCTS	1064.7566	719.9997	0.0000	.0290	4.0000	-211.9718	908.1443

THE TOTAL LOOP INVENTORY IS 4714.39 POUNDS
THE CIRCUIT CYCLE TIME IS 471.44 SECONDS

THE LINEAR INTERPOLATION TO OBTAIN ZERO PRESSURE DROP AROUND THE ENTIRE CIRCUIT RESULTS IN A FLOW RATE OF 9.08960 POUNDS PER SECOND

THE RESULTS FOR A FLOW RATE OF 9.08960 POUNDS PER SECOND ARE AS FOLLOWS

NO.	SEG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1	ROT. PLENUM	1003.5531	720.0000	0.0000	.0290	2.0000	-124.9094	760.5624
2	CORF	992.5595	720.0000	0.0000	.3840	17.0000	5765.5419	1500.0000
3	TOP PLENUM	1500.0000	719.9677	0.0000	.0300	2.0000	-105.6660	1091.4386
4	TOP DUCTS	1490.7001	719.9676	0.0000	.0310	4.0000	-334.6176	1243.4728
5	ANNULUS	1461.2495	719.9676	0.0000	.0340	0.0000	0.0000	*****
6	MAIN STEAM	1461.2495	719.9878	0.0000	.5460	0.0000	-5000.0000	*****
7	CENTERTURE	1021.1863	719.9643	0.0000	.0320	0.0000	0.0000	*****
8	REFRATERS	1021.1863	720.0063	0.0000	.2730	0.0000	0.0000	*****
9	ROT. DUCTS	1021.1863	720.0001	0.0000	.0290	4.0000	-200.3489	873.1615

THE TOTAL LOOP INVENTORY IS 4798.87 POUNDS
THE CIRCUIT CYCLE TIME IS 527.95 SECONDS

Figure 6 (continued).

THE TIME STEP USED IN THIS PROBLEM IS .00100 SECONDS
 A SUMMARY OF VOLUME AND SEGMENT CONDITIONS WILL APPEAR EVERY 100000 TIME STEPS
 THE TIME-DEPENDANT OR FLOW-DEPENDANT BOUNDARY CONDITIONS ARE UPDATED EVERY 100 TIME STEPS

THE OPTIONS USED IN THE TRANSIENT PORTION OF THIS PROBLEM ARE GIVEN IN THE TABLE BELOW

NO.	HEAD=C	HEAD=FLOW	HEAD=TIME	FF=C	FF=FLOW	FF=TIME	HEAT=C	HEAT=CALL	HEAT=TIME	H. T. OPT	HTC (BTU/HR SQ-FT F)
1	1	0	0	1	0	0	0	1	0	4	2.0000
2	1	0	0	1	0	0	0	1	0	1	17.0000
3	1	0	0	1	0	0	0	1	0	4	2.0000
4	1	0	0	1	0	0	0	1	0	2	4.0000
5	1	0	0	1	0	0	1	0	0	-0	-0.0000
6	1	0	0	1	0	0	1	0	0	-0	-0.0000
7	1	0	0	1	0	0	0	0	0	-0	-0.0000
8	1	0	0	1	0	0	1	0	0	-0	-0.0000
9	1	0	0	1	0	0	0	1	0	2	4.0000

FOR THE TRANSIENT PORTION OF THIS PROBLEM THE CONSTANT PUMP HEAD, FRICTION FACTOR, AND HEAT TRANSFER OPTIONS ARE SHOWN BELOW

SEGMENT NO.	PUMP HEAD (LBS/SQ-FT)	FRICTION FACTOR	HEAT INPUT (BTU/SEC)
1	0.0000	.0290	*****
2	0.0000	.3840	*****
3	0.0000	.0300	*****
4	0.0000	.0310	*****
5	0.0000	.0340	0.0000
6	0.0000	.5460	-5000.0000
7	0.0000	.0320	0.0000
8	0.0000	.2730	0.0000
9	0.0000	.0290	*****

FOR SEGMENT NUMBER 1 THERE ARE 3 DATA POINTS IN THE GRAPHITE SURFACE TEMPERATURE VERSUS TIME TABLE...

1000.00000 0.00000 1000.00000 1000.00000 1200.00000 1900.00000

FOR SEGMENT NUMBER 2 THERE ARE 3 DATA POINTS IN THE GRAPHITE SURFACE TEMPERATURE VERSUS TIME TABLE...

1500.00000 0.00000 1500.00000 1000.00000 1700.00000 1900.00000

FOR SEGMENT NUMBER 3 THERE ARE 3 DATA POINTS IN THE GRAPHITE SURFACE TEMPERATURE VERSUS TIME TABLE...

1700.00000 0.00000 1700.00000 1000.00000 1900.00000 1900.00000

Figure 6 (continued).

THE REMAINDER OF THIS OUTPUT SUMMARIZES CONDITIONS IN THE SEGMENTS/VOLUMES DURING THE TRANSIENT PORTION OF THE CALCULATION

TIME INTO THE TRANSIENT = 0.000 SECONDS

NO. SEG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LRS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1 ROT. PLENUM	1003.55	720.0000	9.089604	0.0000	.0290	2.00	-124.91	760.56
2 CORE	992.56	720.0000	9.089604	0.0000	.3840	17.00	5765.54	1500.00
3 TOP PLENUM	1500.00	719.9677	9.089604	0.0000	.0300	2.00	-105.57	1091.44
4 TOP DUCTS	1490.70	719.9676	9.089604	0.0000	.0310	4.00	-334.62	1243.47
5 ANNULUS	1461.25	719.9676	9.089604	0.0000	.0340	0.00	0.00	*****
6 MAIN STEAM	1461.25	719.9676	9.089604	0.0000	.5460	0.00	-5000.00	*****
7 CENTERTUBE	1021.19	719.9643	9.089604	0.0000	.0320	0.00	0.00	*****
8 REHEATERS	1021.19	720.0063	9.089604	0.0000	.2730	0.00	0.00	*****
9 ROT. DUCTS	1021.19	720.0001	9.089604	0.0000	.0290	4.00	-200.35	873.16

TIME INTO THE TRANSIENT = 100.000 SECONDS

NO. SEG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LRS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1 ROT. PLENUM	1030.27	733.5372	10.207084	0.0000	.0290	2.00	-154.31	755.90
2 CORE	1009.31	733.5372	10.235446	0.0000	.3840	17.00	6278.37	1500.00
3 TOP PLENUM	1505.49	733.5051	10.182102	0.0000	.0300	2.00	-114.08	1085.81
4 TOP DUCTS	1501.00	733.5049	10.138939	0.0000	.0310	4.00	-354.96	1238.75
5 ANNULUS	1475.24	733.5049	10.118269	0.0000	.0340	0.00	0.00	*****
6 MAIN STEAM	1477.92	733.5253	10.089499	0.0000	.5460	0.00	-5000.00	*****
7 CENTERTUBE	1085.55	733.5023	10.033678	0.0000	.0320	0.00	0.00	*****
8 REHEATERS	1076.07	733.5434	10.077315	0.0000	.2730	0.00	0.00	*****
9 ROT. DUCTS	1066.25	733.5372	10.119583	0.0000	.0290	4.00	-268.57	869.79

TIME INTO THE TRANSIENT = 200.000 SECONDS

NO. SEG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LRS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1 ROT. PLENUM	1053.85	737.2874	10.438131	0.0000	.0290	2.00	-179.78	752.82
2 CORE	1026.94	737.2878	10.506978	0.0000	.3840	17.00	6212.62	1500.00
3 TOP PLENUM	1503.05	737.2557	10.473159	0.0000	.0300	2.00	-114.29	1080.53
4 TOP DUCTS	1497.40	737.2555	10.442642	0.0000	.0310	4.00	-355.78	1234.54
5 ANNULUS	1471.44	737.2555	10.429546	0.0000	.0340	0.00	0.00	*****
6 MAIN STEAM	1474.10	737.2761	10.404625	0.0000	.5460	0.00	-5000.00	*****
7 CENTERTUBE	1087.23	737.2529	10.400400	0.0000	.0320	0.00	0.00	*****
8 REHEATERS	1085.98	737.2941	10.396045	0.0000	.2730	0.00	0.00	*****
9 ROT. DUCTS	1085.08	737.2879	10.395649	0.0000	.0290	4.00	-291.02	870.05

TIME INTO THE TRANSIENT = 300.000 SECONDS

Figure 6 (continued).

NO.	SEG.	NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LRS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1	ROT.	PLENUM	1064.73	739.9081	10.654731	0.0000	.0290	2.00	-191.86	750.76
2	CORE		1042.23	739.9081	10.6498769	0.0000	.3840	17.00	6122.02	1500.00
3	TOP	PLENUM	1501.88	739.8759	10.678266	0.0000	.0300	2.00	-115.62	1075.56
4	TOP	DUCTS	1495.18	739.8758	10.659317	0.0000	.0310	4.00	-358.16	1230.56
5	ANNULUS		1469.19	739.8757	10.650829	0.0000	.0340	0.00	0.00	*****
6	MAIN	STEAM	1470.99	739.8964	10.634682	0.0000	.5460	0.00	-5000.00	*****
7	CENTER	TUBE	1092.93	739.8732	10.635959	0.0000	.0320	0.00	0.00	*****
8	REHEATERS		1091.47	739.9143	10.638541	0.0000	.2730	0.00	0.00	*****
9	ROT.	DUCTS	1090.69	739.9082	10.640081	0.0000	.0290	4.00	-297.20	871.11

TIME INTO THE TRANSIENT = 400.000 SECONDS

NO.	SEG.	NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LRS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1	ROT.	PLENUM	1070.27	741.3240	10.773888	0.0000	.0290	2.00	-198.25	749.23
2	CORE		1051.05	741.3240	10.799013	0.0000	.3840	17.00	6060.34	1500.00
3	TOP	PLENUM	1501.04	741.2918	10.787911	0.0000	.0300	2.00	-117.13	1070.90
4	TOP	DUCTS	1493.53	741.2917	10.777110	0.0000	.0310	4.00	-360.92	1226.87
5	ANNULUS		1467.30	741.2917	10.772089	0.0000	.0340	0.00	0.00	*****
6	MAIN	STEAM	1468.48	741.3124	10.762242	0.0000	.5460	0.00	-5000.00	*****
7	CENTER	TUBE	1095.95	741.2891	10.761982	0.0000	.0320	0.00	0.00	*****
8	REHEATERS		1095.16	741.3303	10.763523	0.0000	.2730	0.00	0.00	*****
9	ROT.	DUCTS	1094.69	741.3241	10.764804	0.0000	.0290	4.00	-300.75	872.49

TIME INTO THE TRANSIENT = 500.000 SECONDS

NO.	SEG.	NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LRS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1	ROT.	PLENUM	1073.43	742.0760	10.853323	0.0000	.0290	2.00	-202.07	748.00
2	CORE		1056.01	742.0760	10.867687	0.0000	.3840	17.00	6031.51	1500.00
3	TOP	PLENUM	1500.56	742.0438	10.861695	0.0000	.0300	2.00	-118.84	1066.56
4	TOP	DUCTS	1492.48	742.0436	10.855630	0.0000	.0310	4.00	-364.06	1223.50
5	ANNULUS		1465.99	742.0436	10.852679	0.0000	.0340	0.00	0.00	*****
6	MAIN	STEAM	1466.72	742.0644	10.846820	0.0000	.5460	0.00	-5000.00	*****
7	CENTER	TUBE	1097.48	742.0411	10.846659	0.0000	.0320	0.00	0.00	*****
8	REHEATERS		1097.08	742.0822	10.847402	0.0000	.2730	0.00	0.00	*****
9	ROT.	DUCTS	1096.84	742.0761	10.847999	0.0000	.0290	4.00	-301.63	873.98

TIME INTO THE TRANSIENT = 600.000 SECONDS

NO.	SEG.	NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LRS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1	ROT.	PLENUM	1075.21	742.4814	10.903033	0.0000	.0290	2.00	-204.39	746.98
2	CORE		1058.79	742.4814	10.911199	0.0000	.3840	17.00	6017.65	1500.00
3	TOP	PLENUM	1500.30	742.4492	10.907980	0.0000	.0300	2.00	-120.62	1062.51
4	TOP	DUCTS	1491.84	742.4490	10.904553	0.0000	.0310	4.00	-367.34	1220.44
5	ANNULUS		1465.11	742.4490	10.902773	0.0000	.0340	0.00	0.00	*****

Figure 6 (continued).

6 MAIN STEAM	1465.57	742.4698	10.899256	0.0000	.5460	0.00	-5000.00	*****
7 CENTERTURE	1098.32	742.4465	10.899197	0.0000	.0320	0.00	0.00	*****
8 REHEATERS	1098.10	742.4876	10.899574	0.0000	.2730	0.00	0.00	*****
9 ROT. DUCTS	1097.98	742.4815	10.899888	0.0000	.0290	4.00	-301.18	875.45

TIME INTO THE TRANSIENT = 700.000 SECONDS

NO. SEG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LBS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1 ROT.PLENUM	1076.22	742.6981	10.934588	0.0000	.0290	2.00	-205.87	746.10
2 CORE	1060.36	742.6981	10.939309	0.0000	.3840	17.00	6011.74	1500.00
3 TOP PLENUM	1500.16	742.6659	10.937593	0.0000	.0300	2.00	-122.39	1058.76
4 TOP DUCTS	1491.44	742.6657	10.935613	0.0000	.0310	4.00	-370.51	1217.70
5 ANNULIIS	1464.48	742.6657	10.934489	0.0000	.0340	0.00	0.00	*****
6 MAIN STEAM	1464.78	742.6865	10.932294	0.0000	.5460	0.00	-5000.00	*****
7 CENTERTURE	1098.76	742.6632	10.932253	0.0000	.0320	0.00	0.00	*****
8 REHEATERS	1098.65	742.7043	10.932450	0.0000	.2730	0.00	0.00	*****
9 ROT. DUCTS	1098.58	742.6982	10.932617	0.0000	.0290	4.00	-300.15	876.82

TIME INTO THE TRANSIENT = 800.000 SECONDS

NO. SEG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LBS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1 ROT.PLENUM	1076.83	742.8129	10.955732	0.0000	.0290	2.00	-206.88	745.32
2 CORE	1061.25	742.8129	10.958560	0.0000	.3840	17.00	6010.06	1500.00
3 TOP PLENUM	1500.08	742.7807	10.957653	0.0000	.0300	2.00	-124.08	1055.27
4 TOP DUCTS	1491.18	742.7805	10.956464	0.0000	.0310	4.00	-373.46	1215.25
5 ANNULIIS	1464.01	742.7805	10.955710	0.0000	.0340	0.00	0.00	*****
6 MAIN STEAM	1464.22	742.8014	10.954260	0.0000	.5460	0.00	-5000.00	*****
7 CENTERTURE	1099.00	742.7780	10.954233	0.0000	.0320	0.00	0.00	*****
8 REHEATERS	1098.94	742.8102	10.954334	0.0000	.2730	0.00	0.00	*****
9 ROT. DUCTS	1098.90	742.8130	10.954421	0.0000	.0290	4.00	-298.89	878.07

TIME INTO THE TRANSIENT = 900.000 SECONDS

NO. SEG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LBS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1 ROT.PLENUM	1077.21	742.8730	10.970804	0.0000	.0290	2.00	-207.61	744.62
2 CORE	1061.78	742.8730	10.972589	0.0000	.3840	17.00	6010.52	1500.00
3 TOP PLENUM	1500.04	742.8408	10.972117	0.0000	.0300	2.00	-125.68	1052.03
4 TOP DUCTS	1490.98	742.8406	10.971362	0.0000	.0310	4.00	-378.15	1213.07
5 ANNULIIS	1463.64	742.8406	10.970823	0.0000	.0340	0.00	0.00	*****
6 MAIN STEAM	1463.79	742.8614	10.969882	0.0000	.5460	0.00	-5000.00	*****
7 CENTERTURE	1099.12	742.8381	10.969783	0.0000	.0320	0.00	0.00	*****
8 REHEATERS	1099.09	742.8702	10.969834	0.0000	.2730	0.00	0.00	*****
9 ROT. DUCTS	1099.07	742.8730	10.969879	0.0000	.0290	4.00	-297.59	879.20

TIME INTO THE TRANSIENT = 1000.000 SECONDS

Figure 6 (continued).

NO. SEG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LBS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1 BOT. PLENUM	1077.45	742.9035	10.982238	0.0000	.0290	2.00	-208.19	743.99
2 CORE	1062.10	742.9035	10.983427	0.0000	.3840	17.00	6012.01	1500.00
3 TOP PLENUM	1500.02	742.8713	10.983189	0.0000	.0300	2.00	-127.17	1049.03
4 TOP DUCTS	1490.83	742.8711	10.982677	0.0000	.0310	4.00	-378.58	1211.13
5 ANNULUS	1463.32	742.8711	10.982268	0.0000	.0340	0.00	0.00	*****
6 MAIN STEAM	1463.45	742.8919	10.981502	0.0000	.5460	0.00	-5000.00	*****
7 CENTERTUBE	1099.18	742.8686	10.981489	0.0000	.0320	0.00	0.00	*****
8 REHEATERS	1099.16	742.9097	10.981514	0.0000	.2730	0.00	0.00	*****
9 BOT. DUCTS	1099.16	742.9036	10.981538	0.0000	.0290	4.00	-296.35	880.20

TIME INTO THE TRANSIENT = 1100.000 SECONDS

NO. SEG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LBS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1 BOT. PLENUM	1079.82	745.1138	11.160715	0.0000	.0290	2.00	-199.61	743.45
2 CORE	1064.62	745.1138	11.138108	0.0000	.3840	17.00	6370.71	1522.20
3 TOP PLENUM	1512.91	745.0817	11.168101	0.0000	.0300	2.00	-130.26	1046.49
4 TOP DUCTS	1498.35	745.0815	11.179770	0.0000	.0310	4.00	-390.75	1209.64
5 ANNULUS	1468.74	745.0815	11.181399	0.0000	.0340	0.00	0.00	*****
6 MAIN STEAM	1467.04	745.1023	11.177540	0.0000	.5460	0.00	-5000.00	*****
7 CENTERTUBE	1104.55	745.0789	11.184401	0.0000	.0320	0.00	0.00	*****
8 REHEATERS	1102.69	745.1201	11.181427	0.0000	.2730	0.00	0.00	*****
9 BOT. DUCTS	1102.01	745.1139	11.178005	0.0000	.0290	4.00	-298.88	881.17

TIME INTO THE TRANSIENT = 1200.000 SECONDS

NO. SEG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LBS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1 BOT. PLENUM	1087.29	750.6315	11.178408	0.0000	.0290	2.00	-196.05	743.18
2 CORE	1071.05	750.6315	11.150603	0.0000	.3840	17.00	6598.12	1544.42
3 TOP PLENUM	1535.04	750.5994	11.169446	0.0000	.0300	2.00	-142.15	1045.03
4 TOP DUCTS	1517.81	750.5992	11.181978	0.0000	.0310	4.00	-417.04	1209.67
5 ANNULUS	1485.39	750.5992	11.184973	0.0000	.0340	0.00	0.00	*****
6 MAIN STEAM	1481.32	750.6200	11.187765	0.0000	.5460	0.00	-5000.00	*****
7 CENTERTUBE	1117.43	750.5966	11.193497	0.0000	.0320	0.00	0.00	*****
8 REHEATERS	1113.91	750.6378	11.193966	0.0000	.2730	0.00	0.00	*****
9 BOT. DUCTS	1112.26	750.6316	11.193220	0.0000	.0290	4.00	-310.76	882.65

TIME INTO THE TRANSIENT = 1300.000 SECONDS

NO. SEG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LBS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1 BOT. PLENUM	1098.96	757.5855	11.097105	0.0000	.0290	2.00	-196.50	743.39
2 CORE	1080.78	757.5855	11.075384	0.0000	.3840	17.00	6727.25	1566.64
3 TOP PLENUM	1557.88	757.5534	11.086678	0.0000	.0300	2.00	-154.18	1044.76
4 TOP DUCTS	1539.90	757.5532	11.094046	0.0000	.0310	4.00	-444.22	1211.68
5 ANNULUS	1505.28	757.5532	11.095325	0.0000	.0340	0.00	0.00	*****

Figure 6 (continued).

6 MAIN STEAM	1500.75	757.5740	11.097678	0.0000	.5460	0.00	-5000.00	*****
7 CENTERTUBE	1133.19	757.5506	11.102977	0.0000	.0320	0.00	0.00	*****
8 REHEATERS	1128.94	757.5918	11.104996	0.0000	.2730	0.00	0.00	*****
9 ROT. DUCTS	1126.84	757.5856	11.105610	0.0000	.0290	4.00	-327.12	885.14

TIME INTO THE TRANSIENT = 1400.000 SECONDS

NO. SEG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LBS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1 ROT. PLENUM	1113.05	765.1522	11.014845	0.0000	.0290	2.00	-199.11	744.21
2 CORE	1093.11	765.1522	11.001633	0.0000	.3840	17.00	6817.84	1588.87
3 TOP PLENUM	1580.47	765.1201	11.007368	0.0000	.0300	2.00	-165.39	1045.59
4 TOP DUCTS	1561.91	765.1199	11.010926	0.0000	.0310	4.00	-468.83	1215.50
5 ANNULUS	1525.35	765.1199	11.010797	0.0000	.0340	0.00	0.00	*****
6 MAIN STEAM	1520.94	765.1407	11.010951	0.0000	.5460	0.00	-5000.00	*****
7 CENTERTUBE	1150.17	765.1174	11.015829	0.0000	.0320	0.00	0.00	*****
8 REHEATERS	1145.58	765.1585	11.018320	0.0000	.2730	0.00	0.00	*****
9 ROT. DUCTS	1143.29	765.1523	11.019412	0.0000	.0290	4.00	-344.43	888.79

TIME INTO THE TRANSIENT = 1500.000 SECONDS

NO. SEG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LBS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1 ROT. PLENUM	1128.47	773.0431	10.949422	0.0000	.0290	2.00	-202.73	745.68
2 CORE	1107.21	773.0430	10.941231	0.0000	.3840	17.00	6891.62	1611.09
3 TOP PLENUM	1602.89	773.0109	10.944381	0.0000	.0300	2.00	-175.91	1047.44
4 TOP DUCTS	1583.67	773.0107	10.945703	0.0000	.0310	4.00	-490.96	1220.91
5 ANNULUS	1545.34	773.0107	10.944794	0.0000	.0340	0.00	0.00	*****
6 MAIN STEAM	1541.03	773.0315	10.943440	0.0000	.5460	0.00	-5000.00	*****
7 CENTERTUBE	1167.78	773.0082	10.948072	0.0000	.0320	0.00	0.00	*****
8 REHEATERS	1163.01	773.0493	10.950686	0.0000	.2730	0.00	0.00	*****
9 ROT. DUCTS	1160.61	773.0431	10.951929	0.0000	.0290	4.00	-361.38	893.59

TIME INTO THE TRANSIENT = 1600.000 SECONDS

NO. SEG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LBS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (BTU/SEC)	SURF. TEMP. (DEG F)
1 ROT. PLENUM	1144.65	781.1349	10.895457	0.0000	.0290	2.00	-206.79	747.79
2 CORE	1122.40	781.1349	10.890305	0.0000	.3840	17.00	6955.25	1633.31
3 TOP PLENUM	1625.21	781.1028	10.891857	0.0000	.0300	2.00	-185.87	1050.21
4 TOP DUCTS	1605.30	781.1026	10.891851	0.0000	.0310	4.00	-511.05	1227.69
5 ANNULUS	1565.32	781.1026	10.890529	0.0000	.0340	0.00	0.00	*****
6 MAIN STEAM	1561.07	781.1234	10.888351	0.0000	.5460	0.00	-5000.00	*****
7 CENTERTUBE	1185.80	781.1001	10.892817	0.0000	.0320	0.00	0.00	*****
8 REHEATERS	1180.91	781.1412	10.895472	0.0000	.2730	0.00	0.00	*****
9 ROT. DUCTS	1178.43	781.1350	10.896778	0.0000	.0290	4.00	-377.57	899.45

TIME INTO THE TRANSIENT = 1700.000 SECONDS

Figure 6 (continued).

NO. SEG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LRS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (RTU/SEC)	SURF. TEMP. (DEG F)
1 ROT. PLENUM	1161.32	789.3609	10.847701	0.0000	.0290	2.00	-211.04	750.53
2 CORE	1138.29	789.3608	10.844435	0.0000	.3840	17.00	7011.43	1655.53
3 TOP PLENUM	1647.50	789.3287	10.844927	0.0000	.0300	2.00	-195.35	1053.84
4 TOP DUCTS	1626.87	789.3285	10.844085	0.0000	.0310	4.00	-529.43	1235.69
5 ANNULUS	1585.37	789.3285	10.842543	0.0000	.0340	0.00	0.00	*****
6 MAIN STEAM	1581.12	789.3494	10.839930	0.0000	.5460	0.00	-5000.00	*****
7 CENTERTURE	1204.09	789.3261	10.844256	0.0000	.0320	0.00	0.00	*****
8 REHEATERS	1199.11	789.3671	10.846913	0.0000	.2730	0.00	0.00	*****
9 ROT. DUCTS	1196.58	789.3609	10.848244	0.0000	.0290	4.00	-392.87	906.30

TIME INTO THE TRANSIENT = 1800.000 SECONDS

NO. SEG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LRS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (RTU/SEC)	SURF. TEMP. (DEG F)
1 ROT. PLENUM	1178.34	797.6819	10.803336	0.0000	.0290	2.00	-215.34	753.87
2 CORE	1154.64	797.6819	10.801316	0.0000	.3840	17.00	7063.22	1677.76
3 TOP PLENUM	1669.75	797.6496	10.801053	0.0000	.0300	2.00	-204.40	1058.27
4 TOP DUCTS	1648.42	797.6496	10.799650	0.0000	.0310	4.00	-546.34	1244.74
5 ANNULUS	1605.49	797.6496	10.797987	0.0000	.0340	0.00	0.00	*****
6 MAIN STEAM	1601.22	797.6704	10.795144	0.0000	.5460	0.00	-5000.00	*****
7 CENTERTURE	1222.57	797.6471	10.799349	0.0000	.0320	0.00	0.00	*****
8 REHEATERS	1217.52	797.6881	10.801948	0.0000	.2730	0.00	0.00	*****
9 ROT. DUCTS	1214.96	797.6820	10.803322	0.0000	.0290	4.00	-407.27	914.04

TIME INTO THE TRANSIENT = 1900.000 SECONDS

NO. SEG. NAME	INLET TEMP. (DEG F)	INLET PRES. (PSI)	FLOW RATE (LRS/SEC)	PUMP HEAD (PSI)	FRIC. FAC.	H.T. COEF. (B/HR SQFT)	HEAT TRANS. (RTU/SEC)	SURF. TEMP. (DEG F)
1 ROT. PLENUM	1195.63	806.0746	10.760854	0.0000	.0290	2.00	-219.61	757.79
2 CORE	1171.32	806.0746	10.759689	0.0000	.3840	17.00	7110.51	1699.98
3 TOP PLENUM	1691.99	806.0424	10.758895	0.0000	.0300	2.00	-213.06	1063.43
4 TOP DUCTS	1669.96	806.0423	10.757116	0.0000	.0310	4.00	-561.97	1254.73
5 ANNULUS	1625.70	806.0423	10.755398	0.0000	.0340	0.00	0.00	*****
6 MAIN STEAM	1621.40	806.0631	10.752440	0.0000	.5460	0.00	-5000.00	*****
7 CENTERTURE	1241.14	806.0398	10.756535	0.0000	.0320	0.00	0.00	*****
8 REHEATERS	1236.10	806.0808	10.759125	0.0000	.2730	0.00	0.00	*****
9 ROT. DUCTS	1233.51	806.0746	10.760446	0.0000	.0290	4.00	-420.81	922.58

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APPENDIX. A

Subroutine HTRNSFR

This appendix describes the details of the heat transfer models incorporated in subroutine HTRNSFR. Subroutine HTRNSFR is the only portion of SODEMME that is not programmed to allow for an arbitrary problem definition. Instead it has been coded to simulate the geometry and materials of the 3000 MWth HTGR design. In particular, the options the user has in utilizing this subroutine deal with heat transfer in either the core, the cross ducts, the plena, or the steam generators. Thus this part of SODEMME is "hard wired" for 3000 MWth HTGR application; to simulate problems in other systems the user must avoid utilizing this subroutine (which can be done for problems in which the heat transfer in each segment is given explicitly) or alternatively, provide a substitute subprogram that solves the heat transfer problems germane to those systems.

Subroutine HTRNSFR has two major tasks that it must perform for any segment:

- . compute the heat transferred to the helium
- . compute the surface temperature in that segment.

Further, the subroutine must perform these calculations during steady state operation when SODEMME is establishing the initial conditions, and then later, during the transient portion, it must supply these two variables for transient heat transfer conditions. A total of ten options, four for steady state and six for transient conditions, have been coded into HTRNSFR to accomplish this. These ten options are in addition to the two simpler ones

at the disposal of the user; viz. an arbitrary constant amount of heat transferred to the helium or alternatively, an arbitrary temporal function specified during input. The remainder of Appendix A is devoted to describing the four steady state models first, and then the six transient models.

Steady State Options

Option 1

The first option available in HTRNSFR is the only one that is not tailored to any particular kind of HTGR component. It calculates heat transfer in a constant flow area channel having a constant surface temperature. If it is assumed that the heat transfer coefficient and gas properties are independent of position along such a channel, the outlet gas temperature can be found in terms of the other parameters of the flow system

$$T_o = T_s + (T_i - T_s) \exp - (hpl/w C_p) \quad (A-1)$$

For this option the channel surface temperature must be specified during input; it is returned by the subroutine without alteration. The heat transferred into the gas is simply calculated from

$$q = w C_p (T_o - T_i) \quad (A-2)$$

Option 2

This option is used to calculate heat transfer and surface temperatures in ducts lined with the thermal barrier used in the HTGR. It is appropriate to use this option when it is desired to simulate the cross ducts that connect the plena to the steam generator cavities.

It is assumed that the thermal barrier consists of a 1/4 inch thick

Hastelloy cover plate over a 2 3/4 inch thick kaowool blanket. The side of the kaowool blanket away from the cover plate is cooled by water circulated through the liner cooling system. It is assumed that this side is always maintained at 100°F. The thermal conductivity of the kaowool is taken to be linear with temperature over the range likely to be of interest in practical applications

$$k = a + b T \quad (A-3)$$

In the units used internally by the code a good fit to kaowool thermal conductivity data is achieved when

$$a = 1.654 \times 10^{-5}$$

$$b = 3.404 \times 10^{-8}$$

By matching the heat flux across the interface between the helium and the cover plate to the heat flux through the insulation, the cover plate temperature (i.e. the surface temperature) can be determined. Continuity of heat flux results in a quadratic equation for T_s

$$\frac{b}{2\Delta x} T_s^2 + \left(\frac{a}{\Delta x} + h\right) T_s - \frac{a}{\Delta x} T_c - \frac{b}{2\Delta x} T_c^2 - h T_i = 0 \quad (A-4)$$

Once the surface temperature is obtained the heat transferred to the helium is found simply from the convection equation

$$q = h p \ell (T_s - T_i) \quad (A-5)$$

Option 3

This option addresses heat transfer in the plena of the HTGR. The plenum is modeled as two circular surfaces, one a graphite surface (the reflector), and the other a thermal barrier having exactly the same characteristics as the thermal barrier lining the ducts. The heat transfer and

surface temperature calculation for the latter is therefore the same as in Option 2.

In this option, however, the amount of heat transferred from the reflector to the helium must be specified during input preparation. This amount of heat is assumed independent of the heat transfer situation prevailing in the thermal barrier. The surface temperature of the reflector is then found from the convection equation. Finally, the surface temperature of the graphite is compared to the surface temperature of the cover plate on the thermal barrier. The lesser of these two temperatures is printed in the output as the relevant surface temperature*.

Option 4

Option 4 is very similar to Option 3 in that heat transfer in the plenum is of interest. The only difference between this model and the one in Option 3 is that in this model the surface temperature of the reflector, instead of the amount of heat transferred from the reflector, must be specified during input preparation. The heat transferred from the thermal barrier is again computed as in Options 2 and 3 and the heat transferred from the reflector is computed from the convection equation. Here again the reported surface temperature is the smaller of the two surface temperatures (i.e. the thermal barrier cover plate or the graphite reflector).

*The lower surface temperature is relevant in this version of the code because interest is focused on the possibility of fission product condensation on PCRV internals.

Transient Options

These options compute transient heat transfer in various components of an HTGR. The next four options are the transient analogs of the four options outlined above. The fifth one is specific to the steam generators in the HTGR and the sixth option is used for the core when the reactor power is specified as a function of time. In Options 2-6 the options first compute the steady state solutions consistent with the steady state boundary conditions and then use these as the initial conditions for the transient portions. Since an explicit method of solution is employed some attention must be paid to the time step used so as not to violate stability criteria. The equations used in these options will be outlined in some detail so that the limitations on the allowable time step will be apparent. It will be seen that the upper limit on the time step for transient heat transfer calculations is many times larger than the time step that would normally be used for the other parts of the SODEMME program and so this consideration does not impose an undue restriction on the user.

Option 1

This option calculates heat transfer to a fluid flowing in a constant flow area channel whose surface temperature is constant along the entire channel length but is an arbitrary function of time. If the same assumptions concerning the other parameters are made as in Option 1 in the steady state model, it is easy to show that Equation (A-1) is valid for the time-dependent case if T_s is replaced by its time-dependent equivalent, $T_s(t)$. The amount of heat transferred to the helium during any time step is likewise still correctly given by Equation (A-2), where T_o is replaced by its equivalent, $T_o(t)$.

Option 2

This option computes the transient behavior of the thermal barrier lining the duct walls in an HTGR. The same assumptions regarding size and thermal properties of the barrier that are used in Option 2 during steady state are used again here and the surface of the kaowool opposite the cover plate is assumed to be kept always at 100°F, also as in the steady state case.

First the initial thermal barrier cover plate temperature is calculated, using the same relationship as during steady state [i.e. Equation (A-4)]. After obtaining this temperature, the assumption is made that all of the thermal barrier capacitance is in the cover plate and all of the thermal resistance is in the kaowool. It is further assumed, for the transient part of the analysis, that the thermal conductivity of the kaowool is temperature-independent. Using an explicit forward differencing technique it is easy to set up the heat balance for the cover plate. The cover plate temperature is updated by repeatedly applying the resulting equation:

$$T'_s = (h T_f + \frac{k}{\Delta x} T_c) \frac{\Delta t}{\delta \rho C_s} + [1 - (h + \frac{k}{\Delta x}) \frac{\Delta t}{\delta \rho C_s}] T_s \quad (A-6)$$

This relation also yields the stability criterion for the maximum allowable time step. Since the term in brackets in Equation (A-6) must be positive for all values of the parameters the time step must obey inequality (A-7)

$$\Delta t \leq \frac{\delta \rho C_s}{(h + \frac{k}{\Delta x})} \quad (A-7)$$

The heat transfer rate is obtained at any time by applying the convective law to the updated cover plate surface temperature.

Option 3

This option is the time-dependent analog of Option 3 among the steady state algorithms. The thermal barrier cover plate temperature is found as a function of time exactly as in Option 2 above; i.e. by first obtaining the steady state solution and then stepping along in time using a finite difference technique. The contribution of the reflector to the heat input is specified as a function of time explicitly and so HTRNSFR simply adds this component to the heat transferred by the thermal barrier.

The surface temperature of the thermal barrier is determined as just explained. The concomitant reflector surface temperature is obtained from the convection equation. As in the steady state situation, the lesser of these two surface temperatures is printed in the output under that designation.

Option 4

This option is the time-dependent analog of Option 4 among the steady state algorithms. The treatment of the thermal insulation is once again as in Option 2 above but it is the surface temperature of the reflector, rather than its heat flow, that is specified in the input as a function of time. Utilization of the convection equation supplies the latter quantity. The surface temperature that is printed is the lesser of the two surface temperatures, as before.

Option 5

This option provides a simple model for the main steam generators during transients. There is no steady state analog; the heat transferred during the steady state portion of the computation must be a constant, specified as input. If the heat transfer during the transient portion is

not specified as some constant or as an explicit function of time, this option will compute heat transfer in the steam generators based on the following premises:

- . The steam generators and reheaters are homogenized into 6 parallel, 72 foot long assemblies containing a total of 3.15×10^6 pounds of steel and having a total of 201600 square feet of heat transfer surface.

- . The steam generators and reheaters have no water or steam in the modules. Any heat transfer that takes place is between steel and helium only.

- . There is no temperature variation radially through the tube and/or shell walls. All resistance to heat flow takes place in the helium boundary layer.

The programming is such that any initial steel temperature may be specified during input to serve as the initial condition for each segment using this option. If an initial temperature is not specified, the code proceeds by assigning the surrounding helium gas temperature to the steel. Using a forward differencing technique similar to the one used for the thermal barrier, the heat balance for the heat exchanger metal yields

$$T'_s = \frac{h A T_f}{M C_s} \Delta t + (1 - \frac{h A}{M C_s} \Delta t) T_s \quad (A-8)$$

The maximum allowable time step is found, as before, by observing that the term in parentheses in Equation (A-8) must always be positive. Therefore

$$\Delta t \leq \frac{M C_s}{h A} \quad (A-9)$$

Option 6

For some applications of the code the core power as a function of time may be the variable specified instead of the core graphite surface temperature. Option 6 has been included in HTRNSFR for this eventuality. This option utilizes an "equivalent" unit cell which has the heat transfer properties of the original unit cell. It first computes the steady state temperature distribution through the fuel and graphite and then, from this initial temperature profile, the code proceeds to obtain the transient temperature distribution, again by using a forward difference method.

Figure A-1 shows a representative unit cell from the core of an HTGR. Although the thermal problem is two dimensional in principle, studies have shown that in practice it can be closely approximated by a one-dimensional model. This comes about because the graphite in the right angle region of the unit cell is not influential in conducting heat from the fuel to the coolant and can be almost completely neglected. This option therefore utilizes the pseudo-one-dimensional equivalent unit cell shown in Figure A-2. In this equivalent cell the fuel rod has been kept intact and the lengths of the fuel-graphite interface, the coolant-graphite interface, and the graphite web are preserved. The only important parameter that is changed is the volume of graphite. Since this is important only as a thermal inertia term during transients, the thermal diffusivity of the graphite was modified to compensate for the volume change resulting from this mapping into an equivalent cell.

In this conduction model the fuel is divided into five concentric cylindrical regions of equal width and the graphite is divided into four

equal width slabs. The steady state, finite difference solution for each of the resulting ten nodal points is easily obtained by application of the Poisson Equation. The transient heat conduction equation for an internal point, using forward differences, is in general

$$k_l A_l \frac{T_l - T}{S_l} + k_r A_r \frac{T_r - T}{S_r} + gV = \rho C_s V \frac{T' - T}{\Delta t} \quad (A-10)$$

Solving this equation for T' results in a linear equation in T_l , T_r and T in which the coefficient of T is

$$1 - \frac{k_l A_l}{\rho C_s S_l V} \Delta t - \frac{k_r A_r}{\rho C_s S_r V} \Delta t \quad (A-11)$$

For reasons of numerical stability this coefficient must always be positive and that consideration yields the limitation on the size of the time step

$$\Delta t \leq \frac{\rho C_s V}{\frac{k_l A_l}{S_l} + \frac{k_r A_r}{S_r}} \quad (A-12)$$

Similarly, for a point on the graphite-coolant interface, a similar heat balance results in another stability criterion

$$\Delta t \leq \frac{\rho C_s V}{\frac{k_l A_l}{S_l} + h A_r} \quad (A-13)$$

For the numerical values appropriate to the HTGR core materials and geometry the more restrictive of these two conditions occurs at the graphite-coolant interface (Equation A-13) but even there the critical time step is about 175 milliseconds. This is two orders of magnitude higher than the typical time steps used in the other subroutines in SODEMME and so is not in any sense debilitating in terms of executing the code.

Figure A-1. HTGR unit cell.

Figure A-2. Equivalent one-dimensional cell.

APPENDIX B

Geometric Relations in Plena

A system containing a circulating gas is represented by SODEMME as a series of connected segments, each of which is cylindrical (although not necessarily circular) and has an individual flow area, equivalent hydraulic diameter, and length. These three parameters, among other descriptive geometric items, are required as input to the program. For most components of an HTGR, such as the coolant channels, cross ducts, and steam generators, the three parameters are easily evaluated from their basic definitions. For a component such as the plenum, however, some care must be exercised in defining these parameters so that their definition is consistent with the intent of the algorithms in the code.

The flow area, equivalent diameter and length are used by the program to derive three other useful quantities for each segment: the wetted perimeter, the volume, and the heat transfer area. For each segment the code computes these derived quantities in terms of the input parameters by using

$$p = 4A/D$$

$$V = A\ell \tag{B-1}$$

$$A_h = 4A\ell/D$$

Equations (B-1) are obvious for items such as coolant channels, cross ducts, and other components through which the flow is in a generally axial direction. The question becomes: what input values of A , D , and ℓ are appropriate for the plena, so that physically meaningful values of p , V , and A_h will ensue?

If it is assumed that the fluid flows in a generally radial direction in the plenum, which is modeled as a circular cylinder of radius R and height d , suitable values of p , V , and A_h in terms of R and d can be formulated

$$\begin{aligned} p &= 2\pi R \\ V &= \pi R^2 d \\ A_h &= 2\pi R^2 \end{aligned} \tag{B-2}$$

Combining Equations (B-1) with (B-2) makes it possible to solve for A , D , and ℓ in terms of R and d . The result is

$$\begin{aligned} A &= \pi R d \\ D &= 2d \\ \ell &= R \end{aligned} \tag{B-3}$$

Equations (B-3) should therefore be used when preparing input values of A , D , and ℓ for segments representing plena. In this way the coding will yield numerical values of p , V , and A_h which are meaningful within the context of the physical problem being simulated. Furthermore, these relations are consistent with intuition and hence are easily remembered. For radial flow from the center of a cylinder to the outside radius, the simplest definitions for flow area, equivalent diameter, and length along which the fluid must travel, coincide exactly with the relationships shown in Equations (B-3).