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TRANSIENT ANALYSIS OF HEAT PIPES
WITH APPLICATIONS TO SELECTED EXPERIMENTS
AND A CONCEPTUAL REACTOR DESIGN

DISSERTATION

Presented in Partial Fulfillment of the Requirements for
the Degree Doctor of Philosophy in the Graduate
School of The Ohio State University

By

Charles H. Bowers, B.Sc., M.Sc.

* * * * *

The Ohio State University
1973

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# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>ACKNOWLEDGEMENTS</td>
<td>ii</td>
</tr>
<tr>
<td>VITA</td>
<td>iii</td>
</tr>
<tr>
<td>LIST OF TABLES</td>
<td>vi</td>
</tr>
<tr>
<td>LIST OF FIGURES</td>
<td>vii</td>
</tr>
<tr>
<td><strong>Chapter</strong></td>
<td></td>
</tr>
<tr>
<td>1.0 INTRODUCTION</td>
<td>1</td>
</tr>
<tr>
<td>1.1 Description and Statement of Objectives</td>
<td></td>
</tr>
<tr>
<td>1.2 Description of a Typical Heat Pipe and Sonic Limit Explanation</td>
<td></td>
</tr>
<tr>
<td>1.3 Contribution to the Field of Heat Pipe Engineering</td>
<td></td>
</tr>
<tr>
<td>1.4 Organization and Content of the Dissertation</td>
<td></td>
</tr>
<tr>
<td>2.0 REVIEW OF PREVIOUS EXPERIMENTAL AND ANALYTICAL WORK</td>
<td>19</td>
</tr>
<tr>
<td>2.1 Mechanism of Heat Transfer at a Liquid Vapor Interface</td>
<td></td>
</tr>
<tr>
<td>2.2 Experimental Transient and Steady State Studies</td>
<td></td>
</tr>
<tr>
<td>2.3 Analytical Transient and Steady State Analyses</td>
<td></td>
</tr>
<tr>
<td>2.4 Summary with Main Conclusions</td>
<td></td>
</tr>
<tr>
<td>3.0 ANALYTICAL MODEL FOR THE STUDY OF THE TRANSIENT BEHAVIOR OF HEAT PIPES</td>
<td>42</td>
</tr>
<tr>
<td>3.1 General Approach and Methodology</td>
<td></td>
</tr>
<tr>
<td>3.2 General Definitions and Assumptions</td>
<td></td>
</tr>
<tr>
<td>3.3 Adoption of the Fluid Dynamics Equations for Use in the Heat Pipe Problem</td>
<td></td>
</tr>
<tr>
<td>3.4 Finite Difference Form and Solution Flow Diagram for Solution of the Equations</td>
<td></td>
</tr>
<tr>
<td>3.5 Boundary and Initial Conditions for Steady State</td>
<td></td>
</tr>
<tr>
<td>3.6 Burnout and Wick Thermal Resistance Analysis</td>
<td></td>
</tr>
</tbody>
</table>
3.7 Stability Analysis of the Finite Difference Equations
3.8 Summary

4.0 NUMERICAL RESULTS AND COMPARISON WITH SELECTED EXPERIMENTAL AND ANALYTICAL RESULTS ............104

4.1 Analytical Two Dimensional Steady State Investigations
4.2 Incompressible Comparison
4.3 Comparison with Experimental Results

5.0 REACTOR TRANSIENT STUDIES. ...............143

6.0 DISCUSSION OF RESULTS WITH CONCLUSIONS AND RECOMMENDATIONS. .........167

6.1 Discussion of Results
6.2 Limitations of the Technique and Recommendations for Future Study
6.3 Closing Statement

Appendices (Contain all experimental data)

A Flow Diagram for the Implicit Continuous Eulerian (ICE) Method as Adapted to the Heat Pipe Problem .................174

B Listing of the ICE code as written for this Dissertation .................186

C Experimental Work on a Stainless Steel Enclosed Mercury Working Fluid Heat Pipe .................227

D Experimental Work on a Stainless Steel Enclosed Sodium Working Fluid Heat Pipe .................239

E Input Instruction .................245

References .................255
# LIST OF TABLES

<table>
<thead>
<tr>
<th>Table</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>3-1</td>
<td>Variable Description</td>
<td>57</td>
</tr>
<tr>
<td>5-1</td>
<td>Dimensions of Evaporator with Annular Wick</td>
<td>152</td>
</tr>
<tr>
<td>5-2</td>
<td>Dimensions of Condenser</td>
<td>152</td>
</tr>
<tr>
<td>5-3</td>
<td>Values of the Boundary Variables During the Startup Transient as a Function of Time</td>
<td>162</td>
</tr>
</tbody>
</table>
### LIST OF FIGURES

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1-1</td>
<td>Cylindrical heat pipe structure</td>
<td>3</td>
</tr>
<tr>
<td>1-2</td>
<td>2MW(th) nuclear heat source (heatpipe-cooled)</td>
<td>5</td>
</tr>
<tr>
<td>1-3</td>
<td>Heatpipe-cooled reactor/boiler</td>
<td>6</td>
</tr>
<tr>
<td>1-4</td>
<td>A cutaway 3-D view of a typical cylindrical heatpipe</td>
<td>7</td>
</tr>
<tr>
<td>1-5</td>
<td>Axial static pressure profiles in a converging-diverging nozzle</td>
<td>10</td>
</tr>
<tr>
<td>1-6</td>
<td>Axial static temperature profiles in a heat pipe</td>
<td>11</td>
</tr>
<tr>
<td>2-1</td>
<td>Thermocouple positions used in the reference (14) experimental study</td>
<td>26</td>
</tr>
<tr>
<td>2-2</td>
<td>A comparison of evaporator wall temperatures (fluid: H_2O; base temperature: 180°F)</td>
<td>27</td>
</tr>
<tr>
<td>2-3</td>
<td>Possible pressure conditions in sodium heat pipe</td>
<td>30</td>
</tr>
<tr>
<td>2-4</td>
<td>Stream tube flow pattern</td>
<td>34</td>
</tr>
<tr>
<td>2-5</td>
<td>Stream tube coordinate map</td>
<td>35</td>
</tr>
<tr>
<td>3-1</td>
<td>Layout of variables and indices in the mesh</td>
<td>55</td>
</tr>
<tr>
<td>3-2</td>
<td>Nodal network</td>
<td>72</td>
</tr>
<tr>
<td>3-3</td>
<td>Typical evaporator wick region within a heat pipe</td>
<td>82</td>
</tr>
<tr>
<td>3-4</td>
<td>Comparison of Schwartz's (20) results and the correlated results of this analysis</td>
<td>86</td>
</tr>
<tr>
<td>Figure</td>
<td>Description</td>
<td>Page</td>
</tr>
<tr>
<td>--------</td>
<td>-----------------------------------------------------------------------------</td>
<td>------</td>
</tr>
<tr>
<td>4-1</td>
<td>Velocity Profiles, Beginning of Evaporator vs. Time (Low Mach Numbers)</td>
<td>106</td>
</tr>
<tr>
<td>4-2</td>
<td>Velocity Profiles for Near Choking Flow</td>
<td>107</td>
</tr>
<tr>
<td>4-3</td>
<td>The zero flow test (Velocity Vector Field)</td>
<td>109</td>
</tr>
<tr>
<td>4-4</td>
<td>The zero flow test continued (Velocity Vector Field)</td>
<td>110</td>
</tr>
<tr>
<td>4-5</td>
<td>The zero flow test continued (Velocity Vector Field)</td>
<td>111</td>
</tr>
<tr>
<td>4-6</td>
<td>The Symmetrical Heat Pipe</td>
<td>112</td>
</tr>
<tr>
<td>4-7</td>
<td>Initial velocity field guess for the incompressible problem</td>
<td>114</td>
</tr>
<tr>
<td>4-8</td>
<td>Velocity field for Re -.3</td>
<td>115</td>
</tr>
<tr>
<td>4-9</td>
<td>Velocity vector plot for an evaporator Re of -.56</td>
<td>116</td>
</tr>
<tr>
<td>4-10</td>
<td>Velocity vector field for a Re of -1.1</td>
<td>117</td>
</tr>
<tr>
<td>4-11</td>
<td>Velocity vector field for a Re of -2.338</td>
<td>118</td>
</tr>
<tr>
<td>4-12</td>
<td>Velocity vector field for a Re of -5.96</td>
<td>119</td>
</tr>
<tr>
<td>4-13</td>
<td>Total pressure loss for symmetrical heat pipes</td>
<td>121</td>
</tr>
<tr>
<td>4-14</td>
<td>Sodium Enthalpy Saturated Vapor vs. Temperature as found in El Wakil (36)</td>
<td>125</td>
</tr>
<tr>
<td>4-15</td>
<td>Velocity vector field and streakline plot of sodium pipe vapor space 2X10^-5 sec. after boundary conditions of start up are applied</td>
<td>128</td>
</tr>
<tr>
<td>4-16</td>
<td>Velocity vector field and streakline plot of sodium pipe vapor space 4X10^-5 sec. after beginning of start up.</td>
<td>129</td>
</tr>
<tr>
<td>4-17</td>
<td>Velocity vector field and streakline plot of sodium pipe vapor space 6X10^-5 sec. after start up.</td>
<td>130</td>
</tr>
<tr>
<td>Figure</td>
<td>Description</td>
<td></td>
</tr>
<tr>
<td>--------</td>
<td>-------------</td>
<td></td>
</tr>
<tr>
<td>4-18</td>
<td>Velocity vector field and streakline plot of sodium pipe vapor space 8x10^{-6} sec. after start up.........................131</td>
<td></td>
</tr>
<tr>
<td>4-19</td>
<td>Velocity vector field of sodium pipe vapor space 1x10^{-7} sec. after start up.....................132</td>
<td></td>
</tr>
<tr>
<td>4-20</td>
<td>Streakline plot of sodium pipe vapor space 1x10^{-7} sec. after start up...............................133</td>
<td></td>
</tr>
<tr>
<td>4-21</td>
<td>The applicable test results of Hg steel pipe...135</td>
<td></td>
</tr>
<tr>
<td>4-22</td>
<td>Experimental Mercury Heat Pipe Schematic........136</td>
<td></td>
</tr>
<tr>
<td>4-23</td>
<td>Initial formation of pressure front during start up of mercury heat pipe.........................139</td>
<td></td>
</tr>
<tr>
<td>4-24</td>
<td>Mercury heat pipe start up.................................140</td>
<td></td>
</tr>
<tr>
<td>4-25</td>
<td>Mercury heat pipe start up.................................141</td>
<td></td>
</tr>
<tr>
<td>5-1</td>
<td>Space Power Plant Design by Niederauer (45)....144</td>
<td></td>
</tr>
<tr>
<td>5-2</td>
<td>Incorporation of Tungsten-Lithium Heat Pipes into Reactor Design by Niederauer (45)........145</td>
<td></td>
</tr>
<tr>
<td>5-3</td>
<td>Nodal Model of Reactor Heat Pipe Used in the Analysis by the Computer Code Airos (48).......146</td>
<td></td>
</tr>
<tr>
<td>5-4</td>
<td>Neutron Density, Heat Flux, and Temperature Histories for a Start Up Transient.................150</td>
<td></td>
</tr>
<tr>
<td>5-5</td>
<td>Cutaway view of modelled heat pipe...............151</td>
<td></td>
</tr>
<tr>
<td>5-6</td>
<td>Heat pipe model used for reactor transient studies..........................153</td>
<td></td>
</tr>
<tr>
<td>5-7</td>
<td>Conditions at 4.4x10^{-4} sec. after application of &quot;at power&quot; boundary conditions to heat pipe....156</td>
<td></td>
</tr>
<tr>
<td>5-8</td>
<td>Conditions at 4.6x10^{-4} sec. after application of &quot;at power&quot; boundary conditions to heat pipe....157</td>
<td></td>
</tr>
<tr>
<td>5-9</td>
<td>Conditions at 5x10^{-4} sec. after application of &quot;at power&quot; boundary conditions to heat pipe....158</td>
<td></td>
</tr>
<tr>
<td>Figure</td>
<td>Description</td>
<td>Page</td>
</tr>
<tr>
<td>--------</td>
<td>-----------------------------------------------------------------------------</td>
<td>------</td>
</tr>
<tr>
<td>5-10</td>
<td>Conditions at $5.4 \times 10^{-4}$ sec. after application of &quot;at power&quot; boundary conditions to heat pipe</td>
<td>159</td>
</tr>
<tr>
<td>5-11</td>
<td>Final steady state streakline plot for the application of normal &quot;at power&quot; boundary conditions</td>
<td>160</td>
</tr>
<tr>
<td>5-12</td>
<td>The development of the velocity vectors within the vapor volume after application of start up boundary conditions for 24.187 sec.</td>
<td>164</td>
</tr>
<tr>
<td>5-13</td>
<td>Plate A represents conditions when the pipe burnout conditions are being closely approached. Plate B indicates the very high velocities present at time of burnout</td>
<td>165</td>
</tr>
</tbody>
</table>
1.0 INTRODUCTION

1.1 Description and Statement of Objectives

The field of energy production is quickly becoming one of mankind's critical concerns. As a result of this concern, nuclear energy has emerged as one of the primary energy sources of the future. Certainly a critical part of the energy production process is the transport of the energy from its point of production to a point where it can be safely utilized. This dissertation attempts to investigate, analytically, the use of the heat pipe as the path to transport heat. Since most of the available data for comparison purposes is related to liquid metal systems to be used in space nuclear reactor concepts, the study would initially appear to be restricted in applicability. It must be emphasized, however, that the heat pipe is not limited to the space application; it can be used in any process where there is a need to transport heat energy.

The objectives of this investigation are three-fold: (1) to develop a general transient vapor volume heat pipe numerical model, (2) to compare the calculations of the numerical model with the results of analytical and experimental investigations, and (3) to study the response of the model to situations which could occur within a proposed reactor core. The computerized model was created for the IBM 370 computer presently at the Ohio State University.
1.2 Description of a Typical Heat Pipe and Sonic Limit

Explanation

The heat pipe is a simple structure which can be thought of as a heat transformer or a temperature uniformer in that it can be used to transfer large quantities of heat and/or maintain an almost uniform temperature over its entire surface area. The high thermal conductance associated with this structure is attributable to two-phase fluid flow with capillary circulation.

Shown in Fig. 1-1 is a cylindrical heat pipe in a cutaway view by Cotter (1). The device is basically a hollow cylinder containing a capillary structure filled with a liquid and surrounding a vapor region. The end which the heat energy enters is designated the evaporator or the heater while the heat is taken out at the condenser. Notice that it is a self contained, simple engineering structure which exhibits a thermal conductance greatly in excess of that which could be obtained by the use of a homogeneous piece of any known metal. This property is achieved within the containing envelope by the evaporation of a liquid; transport of the vapor to the other end of the cylinder; condensation of the vapor and return of the condensate to the evaporator through a wick motivated by capillary forces such as appear when the corner of a towel
FIGURE 1-1 Cylindrical heat pipe structure (1)
is dipped into water. Considered in this dissertation is the description and analysis of a device such as this in the transfer of heat from the core of a nuclear power plant. An example of such a nuclear plant would be a space power plant of Fig. 1-2. This is basically a small, fast spectrum nuclear reactor which takes advantage of the very high thermal conductance peculiar to heat pipes. The heat pipe takes the place of an in core coolant channel to transport heat directly and efficiently to externally located thermionic diodes or to externally located boilers containing the working fluid of a Rankine cycle system.

Fig. 1-4 is an illustration of a typical heat pipe. The heat piping action becomes effective, i.e., greater than conduction, at internal vapor pressures as low as a hundredth of an atmosphere, and improves with increasing pressure. It is ordinarily possible in terrestrial applications to choose a working fluid which operates at a vapor pressure about equal to the ambient pressure at the desired operating temperature thereby minimizing the thickness of the pipe wall. In the case of a heat pipe to be used in a reactor space power supply this choice might be a liquid metal such as lithium or sodium. The question of the long-term compatibility of this type of system has not yet been completely investigated. In Fig. 1-4 the wick is
LRL 2 MW(t) NUCLEAR HEAT SOURCE
(HEATPIPE-COOLED)

FIGURE 1-2 2 MW(th) nuclear heat source
(heatpipe-cooled)  (2)
HEATPIPE-COOLED REACTOR/BOILER

FUELED CORE
REFLECTOR
CONTROL TUBES
BOILER HEAT PIPES
CENTRAL VAPOR DUCT

LIQUID METAL VAPOR TO TURBINE
LIQUID METAL FROM MAIN RADIATOR

FIGURE 1-3 Heatpipe-cooled reactor/boiler
FIGURE 1-4 A cutaway 3-D view of a typical cylindrical heat pipe
made up of longitudinal channels covered by a mesh. The heat enters the evaporator end of the heat pipe, which is in the core, and is carried to the condenser end where there is a heat sink (boiler). The vaporized working fluid condenses in this region and the heat is drawn off. The working fluid then continues back to the heat pipe evaporator end as a liquid within the wick and the process continues.

The use of heat pipes to remove heat from reactors represents a great change from conventional heat removal schemes. The use of heat pipes allows the replacement of the coolant loops, piping, and pumps of a conventional system by many independent coolant loops which are self-starting and self-regulating, using only a fraction of the coolant required in the conventional systems. The heat pipe in such an application will be investigated as one objective of this dissertation.

In a useful space power plant, the power level must be able to be easily and safely controlled and must be able to be varied between the maximum and zero; therefore, the transient behavior of the heat pipes become very important. In a reactor such as shown in Fig. 1-3, every power level change causes the heat input rate to vary which in turn causes the vapor stream within the pipe to accelerate or decelerate. This change in velocity is caused by the change in the addition rate of mass to the evaporator,
the change in the removal rate, and/or condensation rate of mass from the condenser. The resulting flow patterns within the vapor volume are similar to those of the flow in a converging-diverging nozzle and this analogy is explored below.

Velocity variations in a converging-diverging nozzle result from a constant mass flow through a variable area, whereas in a heat pipe, velocity variations result from a variable mass flow through a constant area. As shown in Fig.1-5, the pressure decreases in the converging section of the nozzle with a resulting increase in the velocity. In the diverging section, the velocity can either decrease with pressure recovery or increase further and become supersonic. The amount of pressure recovery for a properly shaped nozzle depends on the magnitude of the back pressure, $P_{a-d}$. The pressure decreases in the convergent section with an increase in velocity up to the throat. In the divergent section, a pressure recovery occurs with a decrease in the velocity. If the back pressure is lowered to $P_b$, the velocity becomes sonic at the throat and the maximum mass flow rate is attained. This is considered the critical, or choked, flow condition. Further reduction of the back pressure will not increase the flow rate. When the pressure is reduced to $P_c$, the velocity in the divergent section becomes supersonic and pressure recovery is often in the form of a shock front. There is one value of back pressure, $P_d$,
FIGURE 1-5. Axial Static Pressure profiles in a converging - diverging nozzle (3)
FIGURE 1-6. Axial Static Temperature profiles in a heat pipe. (3)
for a given area ratio, which results in a continuous acceleration of the gas over the length of the divergent section. Decreasing the back pressure below this value has no effect upon conditions in the nozzle section.

The vapor flow in a cylindrical heat pipe is quite similar to the flow characteristics encountered in the converging-diverging nozzle. Very high velocities, choked flow, and pressure recovery are evident in heat pipe operation with the particular situation dependent on the heat input and rejection rates. Kemme (22) has demonstrated these characteristics experimentally with a sodium heat pipe. The summarized results are shown in Fig. 1-6 as a plot of static temperature vs. heat pipe length. Note that Fig. 1-6 is related to a heat pipe and Fig. 1-5 is related to the converging-diverging nozzle. Heat pipe wall temperatures are plotted rather than pressure because, in heat pipe operation, temperature is the main parameter of concern and, because a two-phase system exists, the temperature and pressure profiles are similar. A constant heat input was supplied to the evaporator section, and heat rejection was controlled by varying the surface heat transfer coefficient of the outside of the pipe at the condenser. Curve A shows a subsonic flow condition with a slight temperature recovery in the condenser. The temperature dropped along the evaporator section as the vapor stream was accelerated due to mass
addition by evaporation. When the condenser temperature was lowered (Curve B) by increasing the heat rejection rate, the evaporator temperatures were also lowered, the velocity at the exit became sonic, and critical or choked flow conditions existed. Increasing the heat rejection rate further only lowered the condenser temperature because the heat transfer rate to that section could not be increased; this was due to the choked flow condition. The change in condenser temperature had no effect upon the evaporator temperature because the vapor was moving at the speed of sound at the evaporator exit and changes in condenser conditions could not be transmitted upstream to the evaporator section. This demonstrates the sonic limit; there is a maximum axial heat transfer rate due to choked flow and a fixed axial temperature drop along the evaporator associated with any given evaporator entrance temperature. This discussion was set forth by Deverall, et al.\(^2\).

The sonic condition at the adiabatic section is dictated by the ratio of the upstream pressure to that at the adiabatic section. This critical pressure ratio for a perfect gas undergoing heat pipe choking flow can be shown to be 1 plus the ratio of the specific heats. In the condenser section the static pressure associated with the temperature, which can be measured, gives an indication of the type of flow. If the static pressure at the beginning of the con-
denser section becomes lower than that at the adiabatic section, supersonic flow may be expected.

When the sonic limit is reached, it is possible depending on the wick design, for the liquid from the wick to be entrained in the vapor flow field. If this occurs, the liquid circulation to the evaporator is stopped and the evaporator will begin to rise quickly in temperature and ultimately melt if steps are not taken to retard the heat input. Because the entrainment limit is a function of the wick design, it can be investigated on a case basis, experimentally. The sonic limit mechanism can be analyzed in that it is a function of the vapor chamber conditions. The sonic limit should be completely understood and predicted for any reactor design using heat pipes so that proper wick construction can be utilized which will eliminate transient start up failures.

Two other mechanisms which also lead to a decrease in the working fluid circulation rate and thus drying of the wick in the evaporator are: (1) the capillary pumping limit and (2) the boiling limit. The capillary pumping limit occurs when the capillary forces in the wick in the evaporator are not sufficient to maintain circulation of the liquid. The boiling limit occurs at very high temperatures. Normally, when the heat flux becomes large enough to cause conditions for one or a combination of the above
four mechanisms to take place, then burnout occurs.

The primary objective of the numerical model, then, will be to analyze the flow field within the vapor volume both during normal steady state operation, in which the fluid is essentially incompressible, and during the start up or at power transient when sonic and compressible conditions can be evidenced within the vapor volume.

The overall purpose of this dissertation is to provide the basis for a code which the designer can use to investigate the response of a proposed heat pipe prior to construction. Further, all types of reactor transient situations can be investigated.

1.3 Contributions to the Field of Heat Pipe Engineering

There have been many steady state studies concerning the heat pipe, both analytical and experimental; however, until relatively recently very little interest has been expressed in the transient characteristics of these devices. Within the past few years analytical methods for transient analysis were developed by Cotter (4), DeMichele (6), and Sockol (5). Cotter (4) and Sockol (5) used a simple one-dimensional transient analysis while DeMichele (6) utilized a Von Mises transformation to provide a two dimensional model that could be used to investigate steady state choking phenomena that can ap-
pear under extreme start up conditions. Bankston and Smith, (7) used a stream function analysis which is a steady state analysis in two dimensions in which symmetrical heat pipes (heat pipes with one half of the length the evaporator and one half of the length the condenser) were investigated throughout the radial Reynold's number range of \(-0.1 < Re < 1000\). The radial Reynold's Number as used in this study is defined as follows:

\[
Re = \frac{\rho U_{wall} R}{\mu}
\]

(1-1)

where

- \(\rho\) is the vapor density at the evaporator entrance
- \(U_{wall}\) is the radial velocity at the evaporator entrance
- \(R\) is the vapor space radius
- \(\mu\) is the vapor viscosity

The Re is negative for evaporation and positive for condensation.

This dissertation improves upon the previous analytical treatments by providing a description of the transients evidenced by the heat pipe throughout the sequence.

1.4 Organization and Content of the Dissertation

Chapter I is a general introduction giving an explanation of the heat pipe and clearly stating the three objectives of the dissertation. The chapter ends with a discussion of the importance of this study to the heat pipe
Chapter 2 is a review of the analytical and experimental work done to date in the areas of: condensation and evaporation at the liquid vapor interface, and experimental and analytical steady state and transient studies in the specific area of vapor flow within a cylindrical heat pipe. The main conclusions are then summarized at the end of Chapter 2.

Chapter 3 is an adaption of the ICE (Implicit Continuous Eulerian) method of problem solution showing:

1. Assumptions
2. Derivation of finite difference fluid dynamics equations from the non-linear fluid dynamics equations in 2 dimensional cylindrical coordinates.
3. Boundary and initial conditions for the steady state problem.
4. Boundary and initial conditions for the transient problem.
5. Burnout and wick thermal resistance analysis.
6. Stability analysis of the finite difference equations.

Chapter 4 shows the results of comparison of the results of the ICE code calculations with known solutions.
Chapter 5 models an actual proposed reactor heat pipe design and analyzes the results of fast transient applications to the outer surface.

Chapter 6 discusses the results of the reactor heat pipe analysis and indicates how other useful studies can be made by modification of the code.

The references are included in one grouping together at the end of Chapter 6.

Finally, the Appendices include:

A. Flow diagram of the Implicit Continuous Eulerian Method proposed by Harlow (26) as adapted to the Heat Pipe Problem.

B. Listing of the ICE code as written for this dissertation.

C. Experimental work done by Deverall, et al, (3) on a stainless steel, enclosed, mercury-working fluid heat pipe.


E. Input Instruction.
2.0 REVIEW OF PREVIOUS EXPERIMENTAL AND ANALYTICAL WORK

There have been many studies concerning the startup and burnout characteristics of the heat pipe; however, there has never been, to the knowledge of the author, an attempt to make a finite-difference transient model. This is the primary objective of this dissertation. This section will review the analyses, both experimental and analytical, which have been completed prior to 1973 on the semi-transient heat pipe analyses and other associated studies upon which the previous analyses have been based.

2.1 Mechanism of Heat Transfer at a Liquid Vapor Interface

The mechanism of heat transfer at the liquid vapor interface has been the subject of extensive study (9-12). The Ferrell-Alleavitch (12) theory indicates that for values of heat flux below the critical, i.e., that value of the heat flux at which drying of the wick occurred and a large increase in the surface temperature was observed, the heat transfer causing the vaporization of the liquid from the wick covered surfaces was caused by conduction across a thin, liquid saturated film in contact with the
heated surface, which is maintained by the capillary forces existing at a liquid vapor interface within the wick. Ferrell and Johnson \((\text{2})\), an experimental investigation of this mechanism with a wick composed of monel and glass beads, supports the mechanism proposed by Ferrell and Alleavitch \((\text{12})\). At the critical heat flux the capillary forces are no longer sufficient to maintain a liquid saturated film at the heated surface. As the heat flux is increased, the liquid and beads in contact with the surface become superheated, i.e., acquire surface temperatures above the saturation temperature of the system. Eventually, the layer of liquid and beads at the surface become superheated to the extent that a liquid-vapor interface forms. The most probable location for the formation of this interface is at the minimum pore diameter formed by the layer of beads in contact with the surface. Heat flowing from the surface is conducted through the bead-liquid layer to the vaporization interface located at the minimum pore diameter in the first bead layer which suggests an effective wick conductivity made up of contributions from the water and wick structural material. As the liquid is drawn off into the vapor stream the surface tension of the remaining liquid draws more liquid to the interface. The flow of liquid to the interface is laminar and flows through the small porous passages in the wick. The over-
all result of the flow restriction caused by the porous material is to provide a relatively stagnant liquid layer on the heating surface. In another investigation by Chang (13), it has been shown that the capillary forces are unaffected by rapid vaporization occurring at the vapor liquid interface and can be predicted by taking into account the effect of temperature change on the surface tension and density. It was also concluded that once established, the interface location remains fixed until the critical heat flux is approached. The pressure change as given by Kunz (17) and Katzoff (18) across the interface for a fluid that completely wets a surface is described by:

\[ P_v - P_L = \frac{2 \gamma}{r_c} \]  

(2-1)

where

- \( P_v \) the vapor pressure within the pipe center
- \( P_L \) the liquid pressure inside the wick
- \( \gamma \) the surface tension which is a function of temperature
- \( r_c \) the liquid-vapor interface radius of curvature in the evaporator and can be approximated by \( \frac{d + \delta}{2} \) where
- \( d \) is the diameter of the wick element
- \( \delta \) is the pore diameter or distance between wick elements at the liquid vapor interface when \( P_v - P_L \) is a maximum.
The determination of $r_c$ is a critical calculation if burnout is to be predicted. The above approximation of $r_c$ is accurate only for screen wicks which are not compressed and is recommended by Tien and Sun (19). If the wick is highly compressed experimental studies like that of Kunz, et al (17) can be used to find $r_c$.

The final parameter in the wick calculations is the effective wick conductivity which is a combination of the conductivity of the metal of the wick and the conductivity of the liquid within the wick as indicated by Ferrell and Alleavitch (12).

Experimental studies by Schwartz (20) and Seban and Abhat (21) indicate that thermal conductivities can be measured for wicks. The effective value of wick conductivity is a function of temperature and, therefore, calculation of heat transfer across a heat pipe wick in which the working fluid is water must include allowances for this consideration.

The high thermal conductivity of liquid metal working fluids enables the evaporator to be operated without vaporization occurring within the wick structure and also with negligible thermal impedances across the evaporator wick.
In summary then, it is concluded that in order to properly describe the heat transfer process at the liquid vapor interface in a computer heat pipe study, certain variables must be accurately described:

1. The minimum pore radius of the wick surface.
2. The surface tension of the working fluid as a function of temperature.
3. The temperature of the liquid-vapor interface.
4. The effective wick conductivity for heat pipes with water as the working fluid.

The liquid vapor interface temperature can be assumed equal to the outer surface temperature if the working fluid is a liquid metal. If the working fluid is water, the thermal impedance of the wick must be considered and experimental investigations provide information for these values.

Looking at a pipe through startup relative to the liquid vapor interface (before any heat is applied to the evaporator and provided the pipe is horizontal, i.e., no gravity effects), the radius of curvature at the liquid-vapor interface meniscus in each mesh pore is essentially infinite and the pressure difference zero. At the application of heat (startup) the surface tension increases and the meniscus radius decreases. This maintains the increasing
pressure difference across the liquid vapor interface. This continues as long as heat is applied or until the meniscus radius reaches the maximum value described by:

\[ r = r_0 \sec \theta \quad (2-2) \]

where

\[ r_0 \]

is the minimum radius of curvature of the meniscus at the wetting angle of 0 degrees or for the complete wetted condition.

\[ \theta \]

is the wetting angle characteristic of the liquid being used. This is the angle that the liquid surface makes with an adjoining wall to which it is clinging.

Any increase in the heat flux after the meniscus radius has reached the minimum value will cause the liquid to detach from the structure and recede from the surface layer. This receding liquid layer allows the entry of vapor into the wick structure which cuts off the liquid flow within the wick. At this point boiling within the wick causes evaporator temperatures to rise rapidly and the pipe burns out. This is the capillary pumping limit.

Corman (10) concluded that in thick wicks, i.e., in wicks larger than 40 mils, the vaporization process can occur in the wick structure thereby producing a larger
liquid flow pressure drop. Corman did not verify this with actual observations.

To further support and modify somewhat the hypothesis of Corman (10) regarding the occurrence of the vaporization process in the wick structure, Fox (14) definitely established that the burnout process is begun by the initiation of film boiling within the wick structure. This observation was based on actual thermocouple measurements of the vapor volume and the outside surfaces of two cylindrical water heat pipes. This work is significant because of the very detailed nature of the instrumentation and is described in the next section.

2.2 Experimental Transient and Steady State Studies

There have been numerous experimental studies of the heat pipe. Most recent and by far the most detailed investigation thus far was performed by Fox (14). For this study a cylindrical heat pipe was instrumented with thermocouples both in the vapor space and on the outside surface. The working fluid was water. Both transient and steady state information were obtained. Shown in Fig. 2-1 is a diagram of one of the pipes used in this investigation and the thermocouple positions, located within the vapor volume. Two different wicks were studied; a 5 layer 100 mesh screen wick and a four layer wick which consisted of 3 layers of 100 mesh screen and one layer of 16 mesh screen, the 16 mesh screen being adjacent to the wall of the pipe. The
FIGURE 2-1. Thermocouple positions used in the reference (14) experimental study.
FIGURE 2-2. A comparison of evaporator wall temperatures (fluid: H₂O; base temperature: 160°F) (14)
latter wick is known as a dual wick. (15)

With a working fluid of water the temperatures were measured at power levels below calculated burnout and above calculated burnout. At power levels below the calculated burnout point, as shown in Fig. 2-2, it was found that the temperature of the outer surface of the pipe in the evaporator section minus the vapor volume temperature, (ΔT across the wick), at the same point for the 5 layer, 100 mesh screen were slightly lower than predicted from calculations which assumed that only saturated liquid exists inside the wick. However, the dual wick ΔT was far in excess of that predicted by the same calculations. The dual wick with the 16 mesh screen layer at the wall was concluded to permit film boiling below predicted burnout power levels confirming the statements made by Corman (10). The measurements obtained at these lower power levels for both the 5 layer, 100 mesh wick and the dual wick showed that the internal temperature distribution differed by no more than 2°F at any point within the vapor volumes at steady state. At power levels above the predicted burnout the large ΔT's appeared in the 5 layer, 100 mesh screen and adjacent thermocouples within the evaporator vapor volume indicated temperature differences of as much as 80°F at various times; the overall temperature distribution at these higher power levels was found to be ex-
tremely erratic with no apparent consistency.

The erratic temperature distribution occurring at the higher than burnout power levels was thought to be caused by the superheated vapor, formed next to the pipe wall, blowing through the 100 mesh portion of the wick causing a higher thermal resistance and thus a larger $\Delta T$ with erratic behavior.

The main thrust of this experimental study as applied here lies in the fact that for water working fluids the temperature drop across the wick can be successfully described as long as the heat flux is less than that value which could be expected to cause burnout, i.e., vaporization within the wick structure.

In another investigation in which a single thermocouple could be moved along the axis of a sodium heat pipe, McSweeny (16) reported that burnout could be detected by a large temperature decrease in the vapor section of the pipe, which is a possible manifestation of using a liquid metal as the working fluid.

In an experimental investigation performed by Kemme (22), a 137 cm. long stainless steel heat pipe was constructed with the following parameters:

- Outer diameter pipe 1.38 cm.
- Inner diameter pipe wall 1.2 cm.
- Wick inner diameter 1.14 cm.

Working fluid sodium
FIGURE 2-3. Possible pressure conditions in sodium heat pipe.
Evaporator length 14 cm.
Condenser length 110 cm.

The resulting vapor pressures are illustrated for the case of a heat input of 6.4 kW and a temperature distribution as shown at the bottom of Fig. 2-3. Note that pressure recovery occurs within the condenser. Also shown at the bottom of Fig. 2-3 is the pipe schematic with associated temperatures. The pressure curve was determined by conversion of the surface temperatures, assuming these temperatures represented saturated conditions.

Similar to the above sodium working fluid investigation, Dzakowic, et al, (8) researched the limiting sonic velocity. This investigation is analyzed for the code verification and therefore, this problem is described in detail in Appendix D.

Two main conclusions were confirmed by Dzakowic, et al (8) and include:

1. A sonic velocity limit exists at the adiabatic section during normal start up conditions and

2. A supersonic vapor velocity may occur in the condenser section under certain circumstances.

The second conclusion of supersonic vapor velocity occurs in the situation where condensation takes place at such a rate to cause the vapor stream to undergo expansion and acceleration from the point where the Mach number is 1 in the adiabatic section. The principles governing this situation are discussed in Chapter I.
2.3 **Analytical Transient and Steady State Analyses**

This section outlines the work found to date in the literature concerning analytical treatments of the heat pipe sonic and wicking limits and vapor flow distributions.

DeMichele (6) deals with the vapor compressible flow problem and the startup characteristics found in a heat pipe including mass injection and suction as occurs during condensation and evaporation. This was the first two dimensional analytical treatment of this problem in reference to the heat pipe; however, the analysis is based on steady flow equations. In general, there are three assumptions upon which DeMichele (6) based his study.

1. The flow is streamlined. This is based on the similarity of the heat pipe flow problem to the general flow problem with "blowing" and "suction". Both "blowing" and "suction" are two well known methods to stabilize high speed flow and thus DeMichele (6) does not consider the possibility of turbulence or separation occurring in the vapor space for the analysis.

2. The radial pressure gradient is small. When the radial terms of the general flow equations are evaluated, they are found to be at least three orders of magnitude smaller than the axial terms.
3. It is assumed that the flow originating in the evaporator remains streamlined and it can be approximated in $r$--$z$ coordinates as a bundle of concentric stream tubes, as shown in Fig. 2-4. Each tube could be thought of as a converging-diverging nozzle through which lies the path of the vapor originating in the evaporator at the base of that tube. A bundle of these tubes fit inside one another in a concentric framework and are numerically placed within a cylindrical shell. This is the picture of the basis for this development by DeMichele(6). The nozzle equations are then used to describe the conditions in each tube with the added constraint that all the tubes must fit within the cylindrical boundary of the pipe. The pressure, temperature, and velocity of the entering vapor must be known initially at the base of each tube.

The momentum, energy, and continuity equations were transformed from the $r$--$z$ coordinate system to a non-orthogonal coordinate system as shown in Fig. 2-5.
FIGURE 2-4. Stream tube flow pattern
FIGURE 2-5. Stream tube coordinate map
In a typical high flow problem solved by this method, as the Mach number continues to increase, the stream begins to cool, the velocity increases and the stream tubes contract. When a Mach number of unity is reached at the center of the pipe, the center of the stream tube begins to expand causing an increase in the rate of acceleration of the other stream tubes. As more and more of the stream tubes reach a Mach number of unity and begin to expand, the pressure begins a radical drop. Eventually, there comes a point in the analysis where no solutions to the transformed equations exists and this is the point of mass choking. The central stream tubes have by this time reached a supersonic velocity of approximately Mach 1.2 and the velocity pattern during this time has changed from a cosine function to one which is virtually slug flow. The high Mach number pattern is very much like the patterns of turbulent flow in that the velocity profile of the flow is very flat over the central region, but near the wall the velocity exhibits a rapid change.

DeMichele's solution is very good until choking is reached and then no solution to the problem exists. Further, each section of the pipe is investigated separately with the results of the evaporator and adiabatic zone analyses being used as initial conditions for the adiabatic zone and condenser analyses, respectively.

Gas dynamic choking of the flow in the vapor pas-
sage of a heat pipe can place a limitation on the maximum rate of heat transfer. In analytical studies of the sonic limit, Deverall (3) and Levy (23), treating the vapor as a pure monatomic perfect gas and assuming one-dimensional vapor flow in the evaporator region, derived equations for the maximum heat transfer rate. Dzakowic (8) pointed out that a vapor dissociation-recombination reaction for sodium could influence the performance of a sodium heat pipe; such reactions had been ignored previously. Levy (23) studied both this effect and that of the wall shear stress in the evaporator and adiabatic sections. DeMichele (6) concluded that dimer concentration has negligible effect on the problem. Busse (25) and DeMichele (6) found that for inviscid or high flow injection into a steady flow the velocity distribution approaches a cosine curve:

\[ u = u_0 \cos \left( \frac{r}{R} \right)^2 \]  

(2-3)

where

- \( u \) is the axial velocity at the point of interest on the radius
- \( u_0 \) is the maximum axial velocity occurring on the centerline
- \( r \) is the value of each point calculated in the vapor section. \( r \) is defined for \( 0 \leq r \leq R \) and \( R \) is the vapor space radius of the heat pipe.
DeMichele (6) indicated the sonic limit can occur at various points from evaporator exit to condenser entrance. Sockol (5) analytically described the movement of the choking point and verified the analysis by observation of a Lithium pipe startup. Sockol (5) observed that in a lithium heat pipe the starting temperatures can be visual observation of the temperature profile during startup. When sufficiently high heat input to the evaporator end of a pipe, the temperature rises to some intermediate level which remains constant as the "hot zone" increases in length. This uniform hot zone is the length of the pipe between the end of the evaporator end of the pipe and the steep temperature front which progresses down the pipe. When the temperature front reaches the other end of the pipe, the temperature of the pipe then increases to the steady state value. The lithium pipe typically operates in the steady state between the temperatures of 1000 to 1400°C.

Because it is logical to interpret that the sonic limit is coincident with this steep temperature front, Sockol assumed that the vapor velocity is sonic at this point throughout the analysis. When the hot zone fills the length of the pipe, the temperature increases to its steady state value. The movement of the temperature front was experimentally observed by Sockol (5) to be 1.1 cm/sec for a typical startup. Because this velocity is insigni-
ficant when compared to the sonic velocity of the vapor in the pipe it was assumed for Sockol's one dimensional analysis that the steady flow equations could be utilized. The basis of Sockol's analysis were the one dimensional conduction and radiation equations patterned after Cotter (4) for the temperature history description. The wall temperature of the hot zone, i.e., that distance to the point where the temperature abruptly drops, was assumed constant and the rest of the pipe remains at the initial temperature. The vapor flow was assumed to be laminar and a corresponding laminar friction factor was used in the condenser only. Wall friction was assumed negligible in the evaporator. The description of events within the condenser lacked information pertaining to the condensation immediately downstream of the temperature front.

At no time during this analysis was the Reynolds Number high enough to indicate the presence of turbulence. The analysis gave good agreement with measured temperature histories on a pipe surface; however, no definite conclusions could be drawn about flow distributions within the pipe.

2.4 Summary with Main Conclusions

In this chapter, the main aspects of the transient heat pipe description as pertaining to the wicking and sonic limits were discussed. The mechanism of heat trans-
fer within the evaporator was the first subsection. It was shown that a heat pipe may burnout due to the inability of the capillary action to maintain a large enough pressure differential across the liquid-vapor interface. This pressure differential varies directly as the surface tension and inversely with the meniscus radius—the maximum pressure differential being maintained by a minimum meniscus radius, which is identical to half the pore diameter of the wick. When the summation of the pressure drops in the vapor space and wick exceeds the liquid-vapor interface pressure differential, the vapor can enter the wick and the heat pipe begins the burnout process. This limit is the capillary pumping or wicking limit.

In the next subsection the experimental work was reviewed and various conclusions were identified, namely,

1. The effective conductivity of the wick liquid matrix in the evaporator can be measured with good accuracy for water working fluid heat pipes below the capillary pumping limit. The thermal impedance for liquid metal heat pipes is negligible.

2. The sonic limit occurs either in the adiabatic zone or may progress along the pipe during the initial startup depending on the type of startup.

3. The vapor velocity may become supersonic in
the condenser section.

In the third subsection, analytical work on transient heat pipe analysis was cited and explained. It was concluded that:

1. Analytically, there has been no two dimensional description of the sonic limit within the vapor for the entire vapor space in the steady state.

2. Analytically, there has been no description of the vapor flow for transient conditions, including the sonic limit.

3. Although the movement of the location of choking has been predicted, no analytical treatment has yet described the complex two dimensional vapor flow during such movement.
3.0 ANALYTICAL MODEL FOR THE STUDY OF THE TRANSIENT BEHAVIOR OF HEAT PIPES

3.1 General Approach and Methodology

Because the vapor flow velocities vary from almost zero to above the speed of sound, a compressible flow solution was needed that would allow calculations for all velocity ranges. To meet this requirement, a computer code was assembled which used the Implicit Continuous-Fluid Eulerian (ICE) method originally developed by Harlow and Amsden (26). The ICE method is based on an implicit finite-difference approximation to the full non-linear equations of fluid dynamics. This method has been shown to provide a numerically stable and efficient means of calculating transient, viscous fluid flows for Mach numbers ranging from zero to infinity (27). The method was adapted to calculate heat pipe flows.

In general, there are eight basic steps in the fluid dynamics calculation as proposed by Harlow and Amsden (26):

1. Put the fluid dynamics equations into finite differenced conservative form.

2. Divide the flow region into a Eulerian mesh of rectangular cells to which the conservative finite differenced equations can be applied.
3. For the mass conservation equation, the convective flux terms are expressed as functions of the advanced-time densities and velocities.

4. Through substitution for the pressure of a relationship containing the advanced time density the momentum conservation equations are expressed as a function of the unknown advanced velocities and densities.

5. After combination of the mass and momentum equations, in such a way as to eliminate advanced time velocities, a Poisson equation for the new densities results.

6. This is then converted to an equation for the pressure which produces the Poisson equation that is actually solved.

7. The new cycle densities and velocities can then be found based upon the new pressures.

8. Energy changes are then calculated based upon the newly calculated velocities and densities. During each cycle, the calculated values must be monitored for instabilities and the time increment adjusted accordingly.
3.2 General Definitions and Assumptions

The objective of this study is to develop an analytical description of the events which take place inside of a heat pipe during the periods of transient operation or startup. This analysis is also to be compared with data taken from experimental sources. The sources utilized for this comparison are described in Appendices C and D.

1. General Definitions

a. The geometry used in this analysis is exclusively cylindrical; however, the equations can be easily altered for other geometries if desired.

b. All working fluids considered are considered as Newtonian fluids, i.e., the viscous stress can be described as proportional to a linear velocity variation perpendicular to the direction of flow.

c. The Prandtl Number, (Pr), is taken to be a constant.

d. The axial heat transfer by conduction can be neglected, as concluded by Sun and Tien (37).

e. The pressure drop across the interface in the condenser is negligible. This assumes the liquid condenses in such a way as to cause the meniscus radius to be infinite in Eq. (2-1).

2. General Assumptions

a. The heat pipe exists in an environment as
dictated by the situation and both the temperature and the energy input and output are completely known.

b. Negligible heat conduction is considered to take place between the vapor and its surroundings.

c. The vapor is a perfect gas.

d. The vapor is considered as a 100% saturated vapor.

3.3 Adaptation of the Fluid Dynamics Equations for Use in the Heat Pipe Problem

The non-linear equations of fluid dynamics as written below in cylindrical co-ordinates including special stability terms as will be described in section 3.7 are:

Continuity:

\[
\frac{Dp}{Dt} - \rho (\nabla \cdot \mathbf{v}) + \frac{r}{r} \frac{\partial}{\partial r} (r \frac{\partial \rho}{\partial r}) + \Gamma \frac{\partial^2 \rho}{\partial z^2} \quad (3-1)
\]

where the \( \Gamma \) terms indicate corrections for negative mass diffusion, a purely artificial method to insure against instability of the equations (this will be explained in detail later). Expanding the substantial derivative of the density on the LHS and the \( \mathbf{v} \) vector on the RHS:

\[
\frac{D\rho}{Dt} = \frac{\partial \rho}{\partial t} + \mathbf{u} \cdot \frac{\partial \rho}{\partial \mathbf{x}} + \nu \frac{\partial^2 \rho}{\partial z^2} \quad (3-2)
\]
\[-\rho (\vec{\nabla} \cdot \vec{v}) = -\rho \left( \frac{1}{r} \frac{du}{dr} + \frac{dv}{dz} \right) \]

where

\[
\frac{v}{d}{\frac{dE}{d\gamma}} = \frac{dE}{d\gamma} - \rho \frac{dv}{d\gamma}
\]

\[
U \frac{dE}{dr} = \frac{dE}{dr} - \rho \frac{du}{dr}
\]

Combining and eliminating terms:

\[
\frac{dE}{d\gamma} = -\left[ \frac{1}{r} \frac{d}{dr} (\rho u) + \frac{d}{d\gamma} (\rho v) \right] + \frac{1}{r} \frac{d}{dr} (r f_{x})
\]

Momentum Equation:

r component: \( \rho \left( \frac{du}{d\gamma} + U \frac{du}{dr} + v \frac{dv}{d\gamma} \right) = -\frac{dE}{dr} \)

\[
-\frac{1}{r} \frac{d}{dr} (r Tr) - \frac{dTz}{d\gamma} + \rho g r
\]

z component: \( \rho \left( \frac{dz}{d\gamma} + U \frac{dz}{dr} + v \frac{dv}{d\gamma} \right) = -\frac{dE}{d\gamma} \)

\[
-\frac{1}{r} \frac{d}{dr} (r Trz) - \frac{dTz}{d\gamma} + \rho g z
\]

Where Trz is conventionally defined as the viscous flux of z-momentum in the r-direction and \( \rho g z \) is the body force.

Energy Equation:

\[
\rho \frac{De}{D\gamma} = -\left( \vec{\nabla} \cdot \vec{q} \right) - (\vec{\nabla} \cdot p \vec{v}) - (\vec{\nabla} \cdot (\vec{T} \cdot \vec{v}))
\]

where \( \vec{T} \) designates the stress tensor

\( e \) denotes the total energy/unit mass
To use the equations in a computer code, they must be changed to a finite difference representation based upon an Eulerian Mesh network and solved simultaneously.

In order to use equation (3-6) in either an implicit or explicit computer calculation scheme, it must be broken down. The RHS of equation (3-6) is weighted between a purely explicit form and an advanced time form utilizing a constant $\Theta$ such that:

$$0.0 \leq \Theta \leq 1.0$$

$$\frac{d\rho}{dx} = -\Theta \left[\text{advanced time form} \right] - (1-\Theta) \left[\text{same cycle form eqn. (3-6)}\right]$$

(3-10)

The purpose of this form is to provide the advanced time (cycle n+1) density, $\rho$, and velocities, $u$ and $v$, for $\Theta = 1$ or the same cycle density and velocities for explicit calculations ($\Theta = 0$). By setting the value of $\Theta$ to some value between 0 and 1 a centering calculation can be performed.
The \( n + 1 \) superscript denotes advanced time values and \( n \) denotes same cycle or time step values.

In order to most easily utilize these equations on a computer the conservative form is by far the most efficient formulation for calculation as indicated by Roache(39). Rewriting equation (3-7) to begin to establish a conservative form (explained in section 3.7) for 2 space dimensions (radial and axial), and Newtonian fluids:

\[
\frac{d\rho u}{dt} - \frac{\rho u}{dt} + \rho u \frac{du}{dr} + \rho v \frac{dv}{dy} = -\frac{d\rho}{dr} - \left[ \frac{1}{r} \frac{d}{dr} (r T_{rr}) + \frac{d}{dy} T_{ry} \right] + \rho g r
\]

(3-11)

where

\[
T_{rr} = -2\mu \frac{du}{dr} + \frac{2}{3} \mu \left( \frac{r}{r} \frac{du}{dr} + \frac{dv}{dy} \right)
\]

(3-12)

\[
T_{ry} = T_{gr} = -\mu \left( \frac{dv}{dy} + \frac{du}{dy} \right)
\]

(3-13)
\[
T_{yz} = -2u \frac{dv}{dy} + \frac{3}{2} \mu \left( \frac{1}{r} \frac{du}{dr} + \frac{dv}{dy} \right)
\]

(3-14)

Substituting the continuity equation and equations (3-12, and 13) into equation (3-11):

\[
\frac{du}{dt} + \mu \frac{d^2 u}{dr^2} + \frac{u}{r} \frac{du}{dy} + \rho u \frac{du}{dr} + \rho u \frac{dv}{dy} - \rho g r
\]

\[= - \frac{dp}{dr} - \left[ \frac{dTr}{dr} + \frac{Tr}{r} + \frac{dT_{rz}}{dy} \right]
\]

(3-15)

and:

\[
\frac{dp}{dt} + \frac{1}{r} \frac{dp}{dr} + \frac{dp}{dy} - \rho g r
\]

\[= - \frac{dp}{dr} - \left[ \frac{2 \mu}{dr} \left( -2 \mu \frac{du}{dr} - \lambda \left( \frac{1}{r} \frac{du}{dr} + \frac{dv}{dy} \right) \right) \right] \]

\[- \frac{2}{r} \mu \frac{du}{dr} - \frac{2}{r} \lambda \left( \frac{1}{r} \frac{du}{dr} + \frac{dv}{dy} \right) - \frac{\mu}{dy} \left[ \frac{dv}{dr} + \frac{du}{dy} \right]
\]

where, in accordance with Stokes Hypothesis:

\[\lambda = -\frac{2}{3} \mu\]

Proceeding in the same fashion with equation (3-8), rearranging, and with some manipulation, the
momentum equations become:

\[
\begin{align*}
\mathbf{r}: \frac{d}{dr} \left( \frac{D}{r} \right) + \frac{1}{r} \frac{d}{dr} \left( \frac{D}{r} \right) + \frac{d}{dy} \frac{d}{dy} \\
= - \frac{d}{dy} (p+q) + \mu \frac{d}{dy} \left( \frac{d}{dy} - \frac{d}{dr} \right) \\
\mathbf{y}: \frac{d}{dz} \left( \frac{D}{r} \right) + \frac{1}{r} \frac{d}{dr} \left( \frac{D}{r} \right) + \frac{d}{dy} \frac{d}{dy} =
\end{align*}
\]

\[(3-16)\]

\[
\begin{align*}
- \frac{d}{dy} (p+q) - \mu \frac{d}{dr} \left( r \frac{d}{dy} - \frac{d}{dr} \right)
\end{align*}
\]

\[(3-17)\]

where

\[
q = - (\lambda + 2 \mu) \left[ \frac{1}{r} \frac{d}{dr} \frac{d}{dy} \right]
\]

\[(3-18)\]

Rewriting equation (3-9) in a form which explains each term results in:

\[
\frac{d}{dt} \left[ \rho \left( i + \frac{1}{2} (u^2 + v^2) \right) \right] = - \left( \nabla \cdot \mathbf{q} \right) \mathbf{q} + \left( \nabla \cdot \mathbf{Q} \right)
\]

Rate of gain Rate of energy input Rate of energy
of energy per per unit volume by input per unit
unit volume convection volume by conduction
\( \mathbf{q} \) is the heat flux
vector
\[-(\nabla \cdot p \mathbf{v})\] Rate of work done on fluid per unit volume by pressure forces

\[-(\nabla \cdot \mathbf{T} \cdot \mathbf{v})\] Rate of work done on fluid per unit volume by viscous forces where \(\mathbf{T}\) designates the stress tensor

\(\mathbf{i}\) is internal energy per unit mass

\(u\) and \(v\) are the magnitude of the local fluid velocities.

Changes must be made to Equation (3-19) so that consistent quantities are used throughout the code. The quantity \(e\) is defined here as the total or stagnation specific energy:

\[e = 1 + \frac{1}{2} (u^2 + v^2)\] (3-20)

where \(\mathbf{i}\) is the specific internal energy

It is assumed that the heat conduction coefficient, \(k\), and the coefficient of viscosity are related to the constant specific heat, \(C_v\), in such a way that:

\[k \frac{\partial T}{\partial x} = \mu B \frac{\partial (1)}{\partial x}\] (3-21)

where \(B = \frac{k}{\rho r}\), dimensionless

\(k = \frac{C_p}{C_v}\); the ratio of the specific heat at constant pressure to the specific heat at constant volume
In this way it is possible to account for variable viscosity and conductivity. They both vary in such a way that the Prandtl number remains approximately constant.

Substituting these quantities into the energy equation (3-19):

\[
\frac{\partial \rho e_r}{\partial t} = - \left( \frac{1}{r} \frac{\partial}{\partial r} (\rho e_r u) + \frac{\partial}{\partial \theta} (\rho e_r v) \right) \\
- \left[ \frac{1}{r} \frac{\partial}{\partial r} \left( r (\mu B \frac{\partial}{\partial r}) \right) + \frac{\partial}{\partial \theta} \left( -\mu B \frac{\partial}{\partial \theta} \right) \right] \\
- \left[ \frac{1}{r} \frac{\partial}{\partial r} (r p u) + \frac{\partial}{\partial \theta} (\rho v) \right] \\
- \left[ \frac{1}{r} \frac{\partial}{\partial r} \left( r T r U + r T g v \right) \right] \\
+ \frac{\partial}{\partial \theta} \left( T g r U + T g g v \right)
\]

Looking at the last term of equation (3-22) and substituting equations (3-12 to 14):

Last term = - \left[ \frac{1}{r} \frac{\partial}{\partial r} \left( r u \left\{ 2 \frac{dv}{dr} - \frac{2}{3} \left( \frac{1}{r} \frac{\partial}{\partial r} (ru + \frac{dv}{dr}) \right) \right\} \right) \\
+ r v \left\{ -\mu u \frac{dv}{dr} - \mu u \frac{dv}{dr} \right\} \right] + \\
\frac{\partial}{\partial \theta} \left[ \left( -\mu u \frac{dv}{dr} - \mu u \frac{dv}{dr} \right) \\
- u u \left\{ 2 \frac{dv}{dr} - \frac{2}{3} \left( \frac{1}{r} \frac{\partial}{\partial r} (ru + \frac{dv}{dr}) \right) \right\} \right]
or, after further expansion:

\[-\left[ \frac{1}{r} \frac{d}{dr} \left( -2ru \frac{du}{dr} + \frac{2}{3} \mu \frac{d}{dr} (ru) 
+ \frac{2}{3} \mu r \frac{dv}{dr} - ruu \frac{du}{dr} - ruu \frac{dv}{d\theta} \right) \right]
+ \frac{d}{d\theta} \left[ (-ru \frac{dv}{dr} - ru \frac{du}{d\theta}) \right]
- ruu (2\frac{dv}{d\theta} + \frac{2}{3} \mu \nu (\frac{1}{r} \frac{d}{dr} (ru) + \frac{d\nu}{d\theta})) \right]
- \left[ \frac{1}{r} \frac{d}{dr} \left( \frac{2}{3} \frac{d}{dr} (ru) + \frac{2}{3} \mu \frac{dv}{d\theta} 
- 2 \frac{du}{dr} - ru \frac{dv}{dr} - ru \frac{du}{d\theta} \right) \right]
+ \frac{d}{d\theta} \left[ (-ru \frac{dv}{dr} - ru \frac{du}{d\theta}) \right]
- 2ru \frac{dv}{d\theta} + \frac{2}{3} \mu \left( \frac{v}{r} \frac{d}{dr} ru + \nu \frac{d\nu}{d\theta} \right) \right]

(3-23)
and substituting equation (3-22) into equation (3-23) and rearranging terms the energy equation becomes:

\[
\frac{\partial \rho e}{\partial t} + \frac{1}{r} \frac{\partial \rho u e}{\partial r} + \frac{\partial \rho v e}{\partial z} = \frac{1}{r} \frac{\partial}{\partial r} \left\{ r \left[ \beta u \frac{\partial i}{\partial r} \right] - p u - \left( \frac{\lambda}{\lambda + 2\mu} \right) \varphi U + \frac{\mu}{2} \frac{\partial}{\partial r} \left( 2 u^2 + v^2 \right) \\
+ \mu v \frac{\partial u}{\partial z} \right\}^2 + \frac{\partial}{\partial z} \left\{ \beta u \frac{\partial i}{\partial z} - p v - \left( \frac{\lambda}{\lambda + 2\mu} \right) \varphi V \right\}^2 \\
+ \frac{\mu}{2} \frac{\partial}{\partial z} \left[ u^2 + 2 v^2 \right] + \mu v \frac{\partial v}{\partial r}^2 \right\}
\]

(3-24)

where as before

\[
\varphi = \left( \lambda + 2\mu \right) \left( \frac{1}{r} \frac{\partial u r}{\partial r} + \frac{\partial v}{\partial z} \right)
\]

\[
e = \frac{1}{2} \left( u^2 + v^2 \right)
\]

3.4 Finite-Difference Form and Solution Flow Diagram for Solution of the Equations
FIGURE 3-1. Layout of variables and indices in the mesh.
Fig. 3-1 depicts one mesh point of a two-dimensional array which makes up the Eulerian grid used in the ICE method. The index \( i \) counts all cell centers from left to right while \( j \) counts them from bottom to top. For the purposes of the calculation, the fluid boundary velocities, \( U_{i+\frac{1}{2},j} \) and \( V_{i,j+\frac{1}{2}} \), become \( U(I,J) \) and \( V(I,J) \) respectively. Likewise, \( U_{i-\frac{1}{2},j} \) and \( V_{i,j-\frac{1}{2}} \), become \( U(I-1,J) \) and \( V(I,J-1) \) respectively.

The variables of the nodal network shown in Fig. 3-1 are summarized in Table 1. The variables are shown as they are subscripted in the code, the corresponding analytical symbol, and the definition.
<table>
<thead>
<tr>
<th>Code Variable</th>
<th>Analytical Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>RHO(I,J)</td>
<td>( \rho )</td>
<td>Th(\bar{\rho}) density of the vapor in the cell</td>
</tr>
<tr>
<td>P(I,J)</td>
<td>( p )</td>
<td>The pressure of the cell</td>
</tr>
<tr>
<td>E(I,J)</td>
<td>( e )</td>
<td>The total energy within the cell/unit mass</td>
</tr>
<tr>
<td>INTRNL(I,J)</td>
<td>( i )</td>
<td>The internal energy/unit mass</td>
</tr>
<tr>
<td>DT</td>
<td>( \Delta t )</td>
<td>The time step interval</td>
</tr>
<tr>
<td>R(I)</td>
<td>( r_1 )</td>
<td>The radial distance to node I</td>
</tr>
<tr>
<td>GZ</td>
<td>( \varepsilon_z )</td>
<td>The gravitational constant in the axial coordinate</td>
</tr>
<tr>
<td>GR</td>
<td>( \varepsilon_r )</td>
<td>The gravitational constant in the radial coordinate</td>
</tr>
<tr>
<td>GM</td>
<td>( 1 - K )</td>
<td>1- the specific heat ratio</td>
</tr>
<tr>
<td>VISCOS</td>
<td>( \mu )</td>
<td>Viscosity</td>
</tr>
<tr>
<td>LAMBEA</td>
<td>( \lambda )</td>
<td>(-\frac{2}{3} \mu)</td>
</tr>
</tbody>
</table>
Putting equation (3-10) into finite difference form and including a term $\gamma \tau u$ for stability purposes:

\[
\frac{\rho^{n+1} - \rho^n}{\Delta t} = \frac{\Theta}{R(i) \Delta x} \left[ \rho^{n+1} (i+\frac{1}{2}, j) \cdot u^{n+1} (i-1, j) \right. \\
\left. \times \left( R(i) - \frac{\Delta x}{2} \right) - \rho^{n+1} (i+\frac{1}{2}, j) \cdot u^{n+1} (i+1, j) \right] \\
\left. \times \left( \frac{1 - \Theta}{R(i) \Delta x} \right) \right] \\
+ \frac{\Theta}{\Delta z} \left[ \rho^{n+1} (i, j+\frac{1}{2}) \cdot v^{n+1} (i, j+1) - \rho^{n+1} (i, j+\frac{1}{2}) \cdot v^{n+1} (i, j-1) \right] \\
\left. \times \left( \frac{1 - \Theta}{\Delta z} \right) \right]
\]

\[
+ \frac{\gamma}{R(i) \Delta x \Delta z} \left[ (R(i) + \frac{\Delta x}{2}) \cdot (\rho^{n+1} (i+1, j) - \rho^n (i, j)) - (R(i) - \frac{\Delta x}{2}) \cdot (\rho^n (i-1, j) - \rho^n (i, j)) \right] \\
\left. \times \left( \frac{1}{\Delta z \Delta z} \right) \right] \\
+ \frac{\gamma}{\Delta z \Delta z} \left[ \rho^n (i, j+\frac{1}{2}) + \rho^n (i, j-\frac{1}{2}) - 2 \rho^n (i, j) \right]
\]

\[ (3-25) \]

The terms without subscripts imply values at the point $(i,j)$ while half-subscripted terms denote averages of the variable between nodes as illustrated in Figure 3-1. $\gamma$ is a constant term; a correction for destabilizing
negative mass diffusion. \( \tilde{\nabla} \) is the stress tensor not to be confused with \( \nabla \).

Because of the implicit feature of the code, it is convenient to separate those parts of the momentum equations that will contain no advanced time factors and therefore the time index, superscript \( n \), will not be included. All terms of equation 3-16 except \( \frac{\partial u}{\partial z} \) and \( \frac{\partial p}{\partial r} \) are included in the variable RICE (I, J):

\[
RICE (I, J) = \frac{1}{(R(I) + DR) \times DR} \left[ U \times (RHO \times R(I) \times U(I-1, J)) - RHO(I+1, J) \times R(I+1) \times U(I+1, J) \right] + \frac{1}{DZ} \left[ RHO(I + \frac{1}{2}, J - \frac{1}{2}) \times U(I, J + \frac{1}{2}) \times V(I + \frac{1}{2}, J - 1) - RHO(I + \frac{1}{2}, J + \frac{1}{2}) \times U(I, J + \frac{1}{2}) \times V(I + \frac{1}{2}, J) \right] ^{1/DR} [Q - Q(I+1, J)] + \text{VISCOS} \left[ \frac{U(I, J+1) - U}{DZ} - \frac{V(I+1, J) - V(I, J-1)}{DR} \right] - \frac{U - U(I, J-1)}{DZ} + \frac{V(I+1, J-1) - V(I, J-1)}{DR}
\]

where VISCOS = VISCOSITY

\[
GR = \text{gravity component in the r direction (Eqn.3-Q)}
\]

\( Q \) is defined in equation (3-24); in finite difference form:

\[
Q = -(LAMBDA + 2 \times VISCOS) \left( \frac{1}{R(I) \times DR} \left[ (R(I) + DR/2) \times U - (R(I) - DR/2) \times U(I-1, J) \right] \right)
\]
Likewise all terms of Equation (3-17) except $\frac{\partial p_y}{\partial z}$ and $\frac{\partial p}{\partial z}$ are included in the variable $S(I,J)$:

$$
S(I,J) = \frac{1}{R(I)\times DR} \left[ RHO(I-1/2, J+1/2)\times U(I-1, J+1/2)\times V(I-1/2, J)\times R(I) \\
- RHO(I+1/2, J+1/2)\times U(I+1/2, J)\times V(I+1/2, J)\times (R(I)+\frac{DR}{2}) \\
+\frac{1}{DZ} \left[ V(RHO\times V(I, J-1) - RHO(I, J+1)\times V(I, J+1)) \\
+\frac{1}{DZ} \left[ Q(I, J) - Q(I, J-1) \right] + GZ \times RHO(I, J+1/2) \\
\right]
$$

where the $\bar{}$ subscript terms represent averages of the variables immediately adjacent to the point. For example:

$$
RHO(I-1/2, J-1/2) = \frac{1}{4} \left[ RHO + RHO(I-1, J) + RHO(I, J-1) + RHO(I-1, J-1) \right]
$$

and the terms with no subscripts represent the value of the variable at the point $(I, J)$. 

\[ (3-27) \]
It is assumed here that any fluid considered can be described by the perfect gas equation of state in the form:

\[ p = (K-1) \rho \]  

(3-28)

The variable \( C(I,J) \) is defined as the partial derivative of the pressure with respect to the density:

\[ C(I,J) = \frac{\partial p}{\partial \rho} = GM \times INTRNL(I,J) \]  

(3-29)

where \( GM \) and \( INTRNL \) are the computer variables for \( (K-1) \) and \( \rho \), respectively.

To bring the two variables together, a new variable \( PBAR \) is defined which depends on the advanced time density and which represents the corrected value of pressure due to changes in the density and, indirectly, in the internal or heat energy.

\[ PBAR(I,J) = P(I,J) + C(I,J) \times (\rho(I,J))^{n+1} - \rho(I,J)^n \]  

(3-30)

The principal dependence of the equation of state upon the density is illustrated in equation (3-30). This equation is the source from which the advanced time density is calculated.

Equations (3-26 and 30) are combined into the radial component of momentum, equation (3-16), to yield
\[ r: \frac{\rho \left( I + \frac{1}{2}, J \right)^{n+1} u^{n+1} - \rho \left( I + \frac{1}{2}, J \right)^n u^n}{\Delta T} = \frac{\Phi}{\Delta R} (P_{\text{B}} - P_{\text{B}}(I+1, J)) \]

\[ + \frac{1-\Phi}{\Delta R} (\rho^n - \rho^n(I+1, J)) + \text{RISE}(I, J) \]

(3-31)

where \( \Phi \) is variable between 0 and 1 and \( P_{\text{B}} \) can be thought of as the advanced time value of the pressure.

The \( z \) component of momentum is found in the same manner except the \( J \) or axial subscript is now the key variable:

\[ z: \frac{\rho(I, J + \frac{1}{2})^{n+1} v^{n+1} - \rho(I, J + \frac{1}{2})^n v^n}{\Delta T} = \frac{\Phi}{\Delta Z} (P_{\text{B}} - P_{\text{B}}(I, J+1)) \]

\[ + \frac{1-\Phi}{\Delta Z} (\rho^n - \rho^n(I, J+1)) + \text{S}(I, J) \]

(3-32)

The finite difference continuity equation (3-25) has included the expansion of the mass flux term into two terms by use of the variable, \( \Theta \), resulting in the ability to use either the time centered finite difference form, \( \Theta = 0.5 \), or the advanced time form, fully implicit, \( \Theta = 1 \), or a combination of both.
At this point, the combination of continuity, equation (3-25), and momentum, equations (3-31, 32), leads to a Poisson equation in the variable PBAR.

\[
P_{\text{BAR}} \times \left[ \frac{1}{C_n} + 2 \Theta \phi \frac{\partial^2}{\partial t^2} \right] \frac{1}{DR^2} + \frac{1}{Dz^2} \right]
\]

\[
G + \Theta \phi \frac{\partial^2}{\partial t^2} \left[ \frac{(R(I) - DR)}{2} \right] PBAR(I-1, J) + \frac{(R(I) + DR/2)}{2} PBAR(I+1, J)
\]

\[
\frac{R(I) * DR^2}{R(I) * DR^2}
\]

\[
+ PBAR(I, J-1) + PBAR(I, J+1)
\]

\[
\frac{DZ^2}{DZ^2}
\]

(3-33)

In this equation advanced time density has been eliminated by use of the rearranged version of equation (3-30) and \( G \) is a variable composed of all the following same cycle values:

\[
G = G(I, J) = \frac{P(I, J)}{\Theta \phi (I, J)} + \Theta \phi \frac{\partial^2}{\partial t^2}
\]

\[
\left[ \frac{(1 - \phi)}{DR} \right] \left( \frac{(R(I) - DR)}{2} \right) \left\{ P(I-1, J) - P \right\}
\]

\[
\left( \frac{(R(I) + DR/2)}{DR} \right) \left\{ P - P(I+1, J) \right\}
\]

\[
+ (R(I) - DR/2) \times RICE(I-1, J) - (R(I) + DR/2) \times RICE
\]
\[
\frac{\Theta \cdot DT^2}{DZ} \left[ \frac{P(I,J-1)+P(I,J+1)-2 \cdot P \cdot (1-\Phi)}{DZ} \right] \\
+ (S(I,J-1)-S)] \\
+ \frac{DT}{R(I) \cdot DR} \left[ (R(I)-\frac{DR}{2}) \cdot \rho(I-\frac{1}{2},J) \cdot U(I-1,J) \\
- (R(I)+\frac{DR}{2}) \cdot \rho(I+\frac{1}{2},J) \cdot U \right] \\
+ \frac{DT}{DZ} \left[ \rho(I,J-\frac{1}{2}) \cdot V(I,J-1) - \rho(I,J+\frac{1}{2}) \cdot V \right] \\
+ \frac{\tau \cdot DT}{R(I) \cdot DR^2} \left[ (R(I)+\frac{DR}{2}) \cdot (\rho(I+1,J)-\rho) \\
- (R(I)-\frac{DR}{2}) \cdot (\rho - \rho(I-1,J)) \right] \\
+ \frac{\tau \cdot DT}{DZ^2} \left[ \rho(I,J+1) - \rho(I,J-1)-2 \cdot \rho \right]
\]

(3-34)

As before, the \((I,J)\) subscripts have been left out of the equations for brevity where no confusion can result.
Summarizing the steps to derive the Poisson equation for PBAR:

1. Multiply equation (3-25) by \( dt \)
2. Substitute \( (\rho u)^{n+1} \) into equation (3-25) from equation (3-31)
3. Substitute \( (\rho v)^{n+1} \) into equation (3-25) from equation (3-32)
4. Substitute equation (3-30) advanced time density into equation (3-25)
5. Form \( G(I,J) \) out of all terms which do not include PBAR
6. The resulting final equation in PBAR, equation (3-33) is then obtained.

Note that \( G(I,J) \) is given in equation (3-34) and is composed entirely of data available at the beginning of the cycle or as initialized.

Calculate PBAR from equation (3-33). The calculation is accomplished by an iterative procedure, a slightly modified form of which minimizes the truncational error. A method of solution, suggested by Hirt, (29) is used to solve for PBAR at each time step so that minimum error is introduced from cycle to cycle.
If equation (3-33) is first divided by the density and then rearranged so that $\text{PEAR}/\text{RHO}$ stands alone on the left hand side, equation (3-35) is obtained:

$$\frac{\text{PBAR}}{\text{RHO}} = \left[ \Theta \phi \frac{\Delta t^2}{R(I)} \left( \frac{\Delta r}{2} \right) \times \Delta r^2 \left( (R(I) - \frac{\Delta r}{2}) \times \text{SIGL} ight) + (R(I) + \frac{\Delta r}{2}) \times \text{SIGR} \right] + \Theta \phi \frac{\Delta t^2}{\Delta z^2} (\text{SIGB} + \text{SIGT}) \left[ \frac{1}{C} + 2 \Theta \phi \frac{\Delta t^2}{\Delta r^2 + \frac{1}{\Delta z^2}} \right]$$

(3-35)

where

$$\text{SIGL} = \text{PEAR}(I-1, J)/\text{RHO}(I-1, J)$$

$$\text{SIGR} = \text{PEAR}(I+1, J)/\text{RHO}(I+1, J)$$

$$\text{SIGB} = \text{PEAR}(I, J-1)/\text{RHO}(I, J-1)$$

$$\text{SIGT} = \text{PEAR}(I, J+1)/\text{RHO}(I, J+1)$$

To provide an efficient solution of $\text{PEAR}$ by iteration in problems with large spatial variations of density, it has been found that $\text{PEAR}$ can most efficiently be found by a transformation such that

$$\text{SIGIJ} = \frac{\text{PBAR}(I,J)}{\text{RHO}(I,J)}$$

This transformation allows a speedier convergence in that the variable $\text{SIGIJ}$ will remain of the same magnitude throughout the mesh even if large variations of pressure and density are present.

If equation (3-35) is iterated to a fine enough
convergence, little or no error will be introduced; however, this requires mammoth amounts of computer time on very large computers if the problems are to yield scientifically valuable results. Equation (3-35) is further rearranged:

\[ x = \text{Equation}(3-35) \times (1+\text{ALP}) \left( \frac{\text{SIGIJ}}{\text{ALP}} \right) \]  

(3-36)

where

\[ \text{SIGIJ} = \frac{\text{PBAR}(I,J)}{\text{RHO}(I,J)} \]

\[ x = \text{A new variable discussed below} \]

\[ \text{ALP} = \text{Overrelaxation parameter} \quad 0 \leq \text{ALP} \leq 1 \]

This equation now represents the difference between the RHS of equation (3-35) multiplied by \((1+\text{ALP})\) and the LHS multiplied by \(\text{ALP}\). Thus, it can be seen that during any cycle there is an influence from the previous cycle tending to provide a more rapid conversion to a new correct value. This is the key to the PBAR calculation.

With the overrelaxation parameter, \(\text{ALP}\), the convergence is generally speeded with the PBAR \((I,J)\) from step \(n\) being carried forward on the right hand side of the iteration equation.

After sufficient iteration a value of PBAR results for each node in the mesh. Equation (3-30) is then solved for the advanced time density:
\[
\begin{align*}
RHO^{n+1}(i,j) &= \left[ PBAR(i,j) - P(i,j) \right] / C(i,j) \\
+ RHO^n(i,j) \\
\end{align*}
\]

\[ \tag{3-37} \]

Where \( C(i,j) \) is \( \frac{\partial P}{\partial T} \) and is the square of the isothermal sound speed with a constant specific heat.

Rearranging equations (3-31 and 32), the new time advanced velocities can be calculated.

With all quantities but the energy for each node in the mesh for the cycle now known, the total and internal energies are set into finite-difference form and solved:

\[
RHO^{n+1} E^{n+1} = RHO^n E^n + DT \left\{ \frac{1}{R(i) \times \text{DR}} \left[ RHO(i-\frac{1}{2},j) \times U(i+\frac{1}{2},j) \right] \\
E(i-\frac{1}{2},j) \times (R(i) - \text{DR}/2) - RHO(i+\frac{1}{2},j) \times U \times E(i+\frac{1}{2},j) \times (R(i) + \text{DR}/2) \\
+ \frac{1}{\text{DR}} \left[ RHO(i,j-\frac{1}{2}) \times V(i,j-1) \times E(i,j-\frac{1}{2}) \right] \\
+ RHO(i,j+\frac{1}{2}) \times V \times E(i,j+\frac{1}{2}) \right] \\
+ RHO \times V(i,j-\frac{1}{2}) \times GR \\
+ \frac{1}{R(i) \times \text{DR}} \left[ (R(i) + \text{DR}/2) \left[ \frac{B \mu}{\text{DR}} (\text{INTRNL}(i+1,j) - \text{INTRNL}) \right] \right] \right. 
\]


\[- \text{PBAR} (I+\frac{1}{2}, J) \times U - \frac{\lambda}{\lambda+2\mu} U \times Q (I+\frac{1}{2}, J) \]

\[+ \frac{\mu}{2 \text{DR}} \left( 2 (U(I+\frac{1}{2}, J))^2 + V(I+1, J-\frac{1}{2})^2 - 2 U(I-\frac{1}{2}, J)^2 \right) - V(I, J-\frac{1}{2})^2 \] \[+ \frac{\mu}{DZ} \left( U(I, J+\frac{1}{2}) - U(I, J-\frac{1}{2}) \right) \]

\[- (R(I) - DR) \left[ \frac{B\times\mu}{\text{DR}} \right] \left( \text{INTRNL} - \text{INTRNL} (I-1, J) \right) \]

\[- \text{PBAR} (I-\frac{1}{2}, J) \times U(I-1, J) - \frac{\lambda}{\lambda+2\mu} U (I-1, J) \times Q (I-\frac{1}{2}, J) \]

\[+ \frac{\mu}{2 \text{DR}} \left( 2 U(I-\frac{1}{2}, J)^2 + V(I, J-\frac{1}{2})^2 - 2 U(I-\frac{1}{2}, J)^2 \right) - V(I-1, J-\frac{1}{2})^2 \] \[+ \frac{\mu}{DZ} \left( U(I-1, J+\frac{1}{2}) - U(I-1, J-\frac{1}{2}) \right) \]

\[- \text{INTRNL} \right] + \frac{1}{DZ} \left[ \frac{B\mu}{DZ} \right] \left( \text{INTRNL} (I, J+1) - \text{INTRNL} \right) \]

\[- \text{PBAR} (I, J+1) \times V - \frac{\lambda}{\lambda+2\mu} V \times Q (I, J+\frac{1}{2}) \]

\[+ \frac{\mu}{DZ} \left( U(I-\frac{1}{2}, J+1)^2 + 2 V(I, J+\frac{1}{2})^2 \right) - 2 U(I-\frac{1}{2}, J)^2 - 2 V(I, J-\frac{1}{2})^2 - U(I-\frac{1}{2}, J-1)^2 \]
\[-2 * V(I, J - \frac{3}{2})^2 + \frac{\mu}{D} \frac{U(I - \frac{1}{2}, J - \frac{1}{2})}{(V(I + \frac{\lambda}{2}, J - 1) - V(I - \frac{\lambda}{2}, J - 1))}\]

(3-38)

where

\(\mu\) is the computer variable VISCOUS
\(\lambda\) is the computer variable LAMBDAB

Equation (3-38) describes the specific total energy within the node. The specific internal or heat energy, INTRNL, within all nodes can now be obtained:

\[
\text{INTRNL}(I, J) = \mathcal{E}(I, J) - \frac{1}{8} \left[(U + U(I - 1, J))^2 + (V + V(I, J - 1))^2\right]
\]

(3-39)

As before the terms have been given subscripts in accordance with the computer subscripting as shown in Figure 3-1.
The cycle now returns to the beginning for another calculation. The entire procedure is shown in a flow diagram in Appendix A.

If the calculation is incompressible then the sound speed becomes essentially infinite and the density becomes constant throughout the mesh. For this situation the energy equation and equation of state do not enter the calculation and the calculation should proceed in such a way as to iterate on PBAR utilizing only the continuity and momentum equations, i.e., three equations in three unknowns, pressure, radial and axial velocities. The main change involved in the code is to set the pressure variable, $P$, equal to PBAR instead of calculating the pressure for each cycle by the equation of state. The iteration in PBAR essentially allows any changes in the pressure to be instantaneously transmitted throughout the mesh.

During each cycle the code tests all values of density within the mesh. If the variation of density is less than 1% throughout the mesh the incompressible flag is set and the calculation proceeds as discussed above. The user may designate the calculation incompressible in the input.
FIGURE 3-2. Nodal network
3.5 Boundary and Initial Conditions for the Steady State and Transient Analysis

The actual mesh which is established consists of more nodes than originally called for by the user. The number of nodes created by the user, IBAR × JBAR, is surrounded by a set of nodes called boundary nodes. This is illustrated in Figure 3-2 where IBAR, radial, and JBAR, axial, nodes are shown surrounded by IBAR+2 or IP2 radial and JBAR+2 or JP2 axial boundary nodes:

Fig. 3-2 translates into a mesh which is IP2 (IBAR +2) radial nodes by JP2 (JBAR +2) axial nodes.

The purpose of the boundary nodes is to provide values of the variables at the boundary to both simulate the inlet and outlet conditions and provide boundary conditions to realistically simulate the flow situations at the other nonflow boundaries.

1. Nonflow Boundaries- The rigid boundaries are considered as no-slip or free slip boundaries to the flowing vapor. Therefore, for all rigid boundaries the perpendicular velocities = 0 while the tangential velocities across the boundary are equal in magnitude and either
possess the same or opposite signs, depending on whether the surface is free or no slip. If the boundary is free slip as on the centerline of the pipe the axial velocity in the adjoining boundary cell is set equal to the axial velocity in the node immediately inside the mesh. If across a no-slip boundary, the boundary velocity is the negative of the inside velocity. The radial velocities at the centerline are zero. To insure that no other effects enter into the calculations all other variables are set equal across the rigid or centerline boundaries.

2. Inlet and Outlet or Flow Boundaries - The inlet and outlet boundaries require special attention. It is across these boundaries that the driving functions are maintained:

   A. Inlet Boundary

   To define the inlet boundary conditions the energy entering is a known input quantity and is applied to the boundary nodes comprising the user designated inlet. This value may either be a function of time or constant depending on the type of solution desired.

   In looking at an operating heat pipe, the heat is input over the outer surface of the evaporator. The energy
then penetrates to the liquid vapor interface at the boundary of the vapor volume where evaporation takes place allowing an increase in internal energy of the saturated vapor due to the latent heat of vaporization. At this point, the code begins the description of the problem assuming that the fluid entering the vapor volume is a perfect gas and a 100% saturated vapor. The energy of the fluid entering the vapor space is composed of the internal energy within the vapor and the kinetic energy imposed by the pressure gradients and heat flux existing near the interface.

The inlet enthalpy and the temperature are the input variables which are designated by the user. The enthalpy input value less the "pv" term and the corresponding temperature input defines the inlet conditions. The saturation pressure subroutine yields the correct input pressure at the inlet.

With the previous information, either the thermophysical subroutine or the equation of state can be used to find the density:

\[ \rho (I, J) = \frac{\text{INPUT PRESSURE}}{\text{INTRNL} (I, J) \times GM} \]

or

\[ \rho = \frac{p}{(K-1) \times \text{Internal energy}} \]

where \[ K = \frac{cp}{cv} \]

\( GM = K - 1 \)  \hspace{1cm} (3-40)
INTRNL (I,J) = the energy possessed by the vapor assuming negligible interactions between the molecules of gas (enthalpy minus pressure times specific volume).

The inlet velocity is assumed to be a function of the heat flux, density, and latent heat of vaporization, all known quantities for any time step. The pressure is calculated as mentioned above. The heat flux, temperature, and enthalpy can be an assumed function of time, and the latent heat of vaporization is known from correlations within the thermophysical property subroutine (31) as a function of temperature.

From pressure and internal energy calculations, the density can be found as indicated previously. The inlet vapor velocity can then be calculated (assuming no burnout conditions exist):

\[ U_w = \frac{q''}{hfg \rho} \]  \hspace{1cm} (3-41)

where \( U_w \) is the inlet velocity

\( q'' \) is the heat flux at the evaporator liquid vapor interface

\( \rho \) is the density as a function of input temperature
hfg is the latent heat (energy) of vaporization as a function of temperature.

By the combination of the internal and kinetic energies, the total energy possessed by the incoming fluid is known.

If a steady state problem is attempted, then the input inlet enthalpy and temperature will not change. If a transient problem is attempted, these values will be a function of time, as defined by the user-created time variation function which must be compiled within the code.

E. Outlet Boundary

The outlet boundary is handled in a somewhat different fashion than the inlet. If the user designates the calculation as incompressible or it is found to be incompressible during any cycle by a test on the density at each node, then the outlet boundary is treated differently than for a compressible calculation.

1. Outlet Conditions for the Incompressible Calculation

This situation occurs for a steady state calculation; hence, the velocity can be calculated by the relationship:

\[ U_{\text{OUT}} = U_{\text{IN}} \times \frac{(\text{Area of inlet})}{(\text{Area of outlet})} \] (3-42)

The outlet enthalpy and condenser temperatures are designated by the user, and should be saturation values for the fluid being used. The outlet density is calculated
from the equation of state using the user designated outlet temperature (pressure from the property subroutine) and internal energy. All other quantities are assumed equal across the boundaries.

This causes an immediate steady state mass flow condition to be imposed; the calculation then proceeds to supply the unique variation of pressure and energy throughout the mesh. Convergence for this problem is complete when the internal and total energy for the mesh does not change for two complete cycles at which time the code simply calculates the movement of the particles until the finish time as specified by the user in the input data.

2. Outlet Conditions for the Compressible Calculation

The compressible flow calculation can be either a steady state or transient calculation and it is assumed that the mass flow at the condenser entrance is the same as the mass flow at the outlet boundary. The flow is found by integration of the flow field across the vapor flow passage at the condenser inlet. The pipe outlet velocity is then calculated as follows:

\[ U \text{ (OUT)} = \frac{\text{Condenser inlet mass flow}}{\rho \text{ (OUT)} \times \text{Area (OUT)}} \]

(3-42)

The outlet enthalpy and temperatures are input by the user as for the inlet. The outlet density is calculated by the perfect gas relationship.
The problem is at a steady state if either the mass flows at the input, output, and condenser entrance attain the same values or, as in the incompressible calculation, the total energy value within the mesh, (the addition of the energy values at each node) remain constant for 2 cycles.

3.6 **Burnout and Wick Thermal Resistance Analysis**

Two necessary analyses to complete the overall approach to solution of the transient heat pipe problem are included which consider the following items:

1. For liquid metal working fluid pipes, burnout can be due also to the interface pressure rise in the evaporator being less than the sum of the vapor and liquid pressure drops and—

2. For water heat pipes, burnout normally occurs when the temperature drop across the evaporator wick becomes large and vaporization occurs within the wick. Therefore, prior to burnout the temperature drop across the wick must be known so that the vapor-liquid interface temperatures can be accurately described.

1. **Burnout Analysis**

For liquid metal pipes, the latent heat is relatively small thus necessitating a large mass flow rate thereby causing an increase in the vapor and wick liquid pressure drops. When the addition of these drops become larger than the pressure rise from the liquid to vapor in the evaporator,
the pipe is in the initial stages of burnout and is assumed to be burned out for this analysis. The first calculation results in the maximum pressure rise across the interface, as explained in Chapter 2. Repeating equation (2-1):

\[(P_V - P_L) \text{ Interface} = \frac{2 \gamma(T)}{rc}\]

(2-1)

where

\[\gamma(T)\] is the surface tension as a function of temperature, T.

\[rc\] is the minimum meniscus radius or the characteristic pore radius. Values of \[rc\] for various type wicks have been compiled by Phillips (32) and Kunz, et al (17).

\[P_L\] and \[P_V\] are the liquid and vapor static pressures on the appropriate side of the interface.

The surface tension is obtained from the Thermo-Physical Property Subroutine by Verljen (31).

The pressure drop of liquid in the wick is a function of the resistance of the wick to flow (permeability), the liquid velocity and viscosity. The liquid velocity is assumed to have a flat distribution across the wick.

In applying the analysis formulated by Hwang-Bo (30), it is assumed that for each time step in the wick:

\[\frac{dU_{wk}}{dz} \sim \frac{q_{wk}}{Awk}\]
where

$U_{wk}$ is the average velocity of the liquid in the wick
$q_{wk}$ is the volumetric flow rate of liquid in the wick
$A_{wk}$ is the total surface area of the wick and is proportional to the volume.

The forces in the wick during any time step consist of pressure forces opposing viscous forces:

Application of Darcy's Law assuming $U_{wk}$ to be an average wick velocity:

$$\frac{dP_l}{dz} = -\frac{\mu_l}{K_p} U_{wk}$$  \hspace{1cm} (3-44)

where

- $\mu_l$ is the liquid viscosity
- $\frac{dP_l}{dz}$ is the wick liquid pressure drop
- $K_p$ is the permeability and can be obtained from experimental studies conducted by Shen (33) and Langston (17, 34) for the wick used.

The vapor pressure drop is calculated from the code for each time step. With the three pressure drops calculated, the comparison for burnout can be performed.

2. Wick Thermal Resistance Analysis

For water heat pipes the thermal impedance is such that the $\Delta T$ across the liquid in the evaporator is significant. Therefore, the $\Delta T$ should be estimated so that the vapor inlet may be accurately described.

Figure 3-3 is a model of the wick evaporator region.
FIGURE 3-3. Typical evaporator wick region within a heat pipe
PLATE A. Cross-sectional view of 100-mesh sintered nickel screen
PLATE B. Top view of 100-mesh sintered nickel screen
As shown in the figure, the liquid flows along the inside wall increasing in temperature by convection while at the same time the wick is also conducting energy to the interface at some rate higher than the convective rate.

To describe this situation analytically the inside wall temperature and heat flux is used as the initial known conditions for the analysis. The bulk liquid temperature is guessed and this temperature is used to calculate the effective wick conductivity. The effective wick conductivity values were obtained experimentally by Schwartz (20) for a double layer 100 mesh stainless steel screen wick. A correlation was found which described this variation and was then put into subroutine form for use in this analysis.

The value of $h_c$, the surface heat transfer coefficient, is assumed to be the effective wick conductivity divided by the wick thickness. The final value of the wick surface heat transfer coefficient, $h_c$, is given by:

$$h_c = \frac{k_{\text{wick}}}{L}$$

(3-45)
where

\[ k \] is the experimentally determined effective wick conductivity based on the experimental results of Schwartz (20).

\[ L \] is the wick thickness.

The vapor-liquid interface is assumed to exist at the bulk temperature.

In the above way, the significant temperature drop across the wick of the evaporator of a water heat pipe is found. It must be emphasized that this is strictly applicable to one set of experimental results; however, it is used to provide an estimate of the general situation.

With a value calculated for \( hc \), the known wall temperature, and bulk temperature, a new value of heat input is calculated and compared to the known value. If the new calculated value differs, the bulk temperature is adjusted and the process is repeated. When the correct value of the heat input is calculated, the correct bulk temperature will be known.
To check the validity of this logic, the subroutine was used as a stand-alone program and compared with experimental results obtained by Schwartz (20) where the temperature drop across a typical wick was measured for a successive number of heat pipe heat loads. The type of wick used by Schwartz (described previously) was modelled in the subroutine and the results showed good agreement with the experimental results before inception of burnout. Figure 3-4 shows this comparison. The predicted values, shown by the line connected by circles in Figure 3-4, is high by 5% for the low heat loads and low by 10% for the higher heat loads when compared with Schwartz's results for the saturated wick in the evaporator section. At approximately 150 BTU/hr the actual pipe began to experience burnout with vapor entering the wick as evidenced by the rapidly increasing $\Delta T$. Therefore, the model is limited to the regime where burnout is not a consideration, i.e., $\Delta T$ values below 9 degrees. If, while using the code, the change in temperature across a wick of a heat pipe using water is calculated to be greater than 9 degrees a caution message is printed indicating that burnout would be possible. The user should recognize that if this message is received, the code results are questionable.
FIGURE 3-4. Comparison of Schwartz's (20) results and the correlated results of this analysis. The correlated results of this analysis have been superimposed (dots with circles connected by lines) on Schwartz's results for the saturated wick in the evaporator section.
3.7 Stability Analysis of the Finite Difference Equations

With a finite number of grid points representing the infinite medium of the molecular structure of a moving fluid, the problem representation can at best yield only an approximation; however, if the conservation laws are obeyed in the mesh the results can be very useful. First, the finite difference equations governing the problem must be written in conservative form, i.e., they must reflect the fact that the quantity in question can be in fact conserved throughout the mesh. Secondly, the stability must be insured during a problem; the stability criteria must be constantly monitored throughout any calculation. In this section, an explanation of these two points will be completed as summarized from the presentation by Harlow (24).

A. Conservative Form of Equations

Consider the simplified momentum equation written in 2 forms:

\[
\frac{d\rho u}{dx} + \frac{d}{dx}(\rho u^2 + p) = 0 \tag{3-46}
\]

and

\[
\frac{du}{dx} + u \frac{du}{dx} + \frac{1}{\rho} \frac{dp}{dx} = 0 \tag{3-47}
\]

where friction has been neglected.

If equation (3-46) is integrated between 2 fixed positions \(x_1\) and \(x_2\):
\[ \frac{d}{dt} \int_{x_1}^{x_2} \rho u dx + (\rho u^2 + p)_{x_2} - (\rho u^2 + p)_{x_1} = 0 \]  

(3-48)

which shows that the time rate of change of all the momentum within the interval is given by the difference between the momentum fluxes at the two ends. Momentum is obviously conserved and the in-cell momentum change with time is related to differences in the momentum fluxes at the cell edge.

The finite difference form of equation (3-47) is:

\[
\frac{u_j^{n+1} - u_j^n}{\delta t} + \frac{u_j^n}{\delta x} (u_j^n - u_{j-1}^n) + \frac{1}{\rho_j \delta x} (p_j - p_{j-1}) = 0
\]

(3-49)

which relates changes of velocity in the cell to terms which are not all cell edge quantities. While in some cases the latter form offers valid results, it is much more difficult to work with and cannot be summed over all cells, thus providing no check on total mesh momentum. In most cases the results obtained by the latter form will introduce erroneous results into the calculation. For these reasons, the equations are written in conservative form.

B. Stability Analysis

In the actual flow of fluid, at low speeds
the flow is laminar and as the speed is increased fluid-
dynamic instability will begin to occur, one example be-
ing the Von Karman vortex street evident in flow around
a cylinder. This fluid instability can be described by
the equations.

In the calculation of solutions to such prob-
lems as above and as completed in this dissertation, the
researcher must be concerned with numerical instability,
which is simply the result of a violation of natural
pheno\-mena by the equations. Consider the equation below
which describes the simplified mass conservation in a
constant speed fluid flow field:

$$\frac{d\rho}{dt} + u_0 \frac{d\rho}{dx} = 0 \quad (3-50)$$

where $u_0$ is the velocity parallel to the $x$ axis
and $\rho$ is the density. Recasting equation (3-50) into
finite difference form:

$$\frac{\rho_j^{n+1} - \rho_j^n}{\delta x} + \frac{u_0}{2\delta x} (\rho_j^{n+1} - \rho_j^{n-1}) = 0 \quad (3-51)$$

with a trial solution of:

$$\rho_j^n = \rho_0 + A^n e^{ik_j \delta x} \quad (3-52)$$

When equation (3-52) is substituted into equation (3-51)
the solution for $A^n$ is:
\[ A^{n+1} = A^n (1 - \frac{iU_0 St}{Sx} \sin kSx) \]  
(3-53)

Where \( A^{n+1} \) represents the solution at the next succeeding time step higher than \( n \). Being complex, the magnitude of the coefficient on \( A \) is found:

\[
A^n \text{ coefficient} = 1 + \left\{ \left( \frac{U_0 St}{Sx} \right) \sin kSx \right\}^2
\]  
(3-54)

Which always exceeds unity and thus the value of the coefficient for \( A^n \) will increase without bound for infinite time steps. For this reason the equation must be modified prior to use on a computer.

A well known modification is the donor cell technique used by Harlow (26) with equation (3-51) modified as follows:

\[
\frac{\rho_j^{n+1} - \rho_j^n}{Sx} + \frac{U_0}{Sx} (\rho_j^n - \rho_{j-1}^n) = 0; U_o > 0
\]  
(3-55a)

or

\[
\frac{\rho_j^{n+1} - \rho_j^n}{Sx} + \frac{U_0}{Sx} (\rho_{j+1}^n - \rho_j^n) = 0; U_o < 0
\]  
(3-55b)

If the trial solution, equation (3-52), is substituted into equations (3-55a), the result is:
\[(\rho_0 + A^{n+1}e^{i(k+1)jSx} - \rho_0 - A^n e^{ikjSx})/st \]
\[+ \frac{U_0}{Sx} (\rho_0 + A^n e^{ikjSx} - \rho_0 - A^n e^{ik(j-1)Sx}) = 0 \]

(3-56)

Dropping the \(\rho_0\) terms and dividing by \(e^{ikjSx}\) and rearranging:

\[A^{n+1} = A^n \left[1 - \frac{U_0 St}{Sx} (1 - e^{-ikSx}) \right] \]

(3-57)

The coefficient is complex and the magnitude is:

Coefficient of \(A^n = 1 - 2\lambda + 2\lambda^2 + 2(2-\lambda)\lambda \cos kSx \]

where

\[\lambda = \frac{U_0 St}{Sx} \]

(3-58)

For \(\lambda < 1\) the maximum value of \(l\) is obtained for the coefficient when \(\cos kSx = 1\), which provides a stable representation. If \(\lambda > 1\), the factor is greater then unity with \(\cos kSx = -1\). In conclusion, a conditionally stable equation is obtained with the constraint that:

\[\frac{U_0 St}{Sx} < 1 \]

(3-59)
which is the Courant condition. The previous stability analysis is known as either the Fourier or Von Neumann method and is particularly well suited to linear equations.

In application to the fluid dynamics equations, non linear equation representation, variable coefficients, viscosity and convection must be examined and the deleterious effects neutralized to get useful results.

In general, a few limitations can be deduced after some experience is gained in manipulation of the finite difference fluid dynamics equations:

1. The change in any cell value during a cycle must be small compared to the value of the quantity within the cell at the start of the cycle.

2. The fluid must move only a fraction of the cell width during any cycle. If this is not obeyed resolution will deteriorate.

3. The development with time of the flow field at any point in the mesh must be smooth.

Due to the limitations of the Fourier analysis technique already discussed, the stability of the equations considered for this problem must be investigated by a different method. Hirt(29) devised a method in which each term of the finite difference equation is expanded in a Taylor series about a point which yields the original equation plus some additional higher order terms which
represent the truncational error, or simply, the error incurred in the calculation because the computer did not consider these terms. For stability, the truncational error must be small. The Hirt method (29) yields results that both predict instabilities found by the Fourier method and additional instabilities not detectable by the Fourier technique.

Having now explained the analysis of stability, it remains to analyze, by the expansion of the momentum and continuity equations in Taylor series, the original equations which contain the $i \pm \frac{1}{2}$, and $j \pm \frac{1}{2}$ subscripts. The original subscripts must be used as they denote the value of these variables at the cell edges; the whole number subscripts, as explained previously, allow computerized representation.

The stability needs to be found only for the continuity and momentum equations because these equations contain the variables upon which direct iterations occur. Rewriting the continuity expression,

\[
\frac{\rho_{i,j}^{n+1} - \rho_{i,j}^n}{\delta t} = \frac{\Theta}{\delta x} \left( \rho u_{i-\frac{1}{2},j}^{n+1} r_{i-\frac{1}{2}} - \rho u_{i+\frac{1}{2},j}^{n+1} r_{i+\frac{1}{2}} \right) + \frac{\Theta}{\delta y} \left( \rho v_{i,j-\frac{1}{2}}^{n+1} r_{j-\frac{1}{2}} - \rho v_{i,j+\frac{1}{2}}^{n+1} r_{j+\frac{1}{2}} \right) + \frac{(1-\Theta)}{\delta z} \left( \rho u_{i,\frac{1}{2},j}^n r_{i-\frac{1}{2}} - \rho u_{i+\frac{1}{2},j}^n r_{i+\frac{1}{2}} \right)
\]
If each term is considered as a continuous function of \( x, t, \) and \( z \) and expanded in a Taylor series about the point \( x, t, \) and \( z, \) the truncation errors should be revealed. The axial dimension of the equation will be analyzed and the two dimensional stability conditions will be deduced from these results. Restating equation (3-60) in one dimensional (axial) form:

\[
\frac{\rho_i^{n+1} - \rho_i^n}{\delta x} = \frac{\theta}{\delta \theta} (\rho u_{i-\frac{1}{2}}^{n+1} - \rho u_{i+\frac{1}{2}}^{n+1}) + (1-\theta) (\rho u_{i-\frac{1}{2}}^n - \rho u_{i+\frac{1}{2}}^n) + \frac{\tau}{\delta z^2} (\rho_{i+1}^n + \rho_{i-1}^n - 2 \rho_i^n) \tag{3-61}
\]

Combining all terms of equation (3-61) and numbering the terms:

**Term 1**

\[
\frac{\rho_j^{n+1} - \rho_j^n}{\delta x} = \frac{1}{\delta z} (\rho u_{j-\frac{1}{2}}^n - \rho u_{j+\frac{1}{2}}^n)
\]

**Term 2**

\[
\frac{\rho_{i-\frac{1}{2}}^{n+1} - \rho_{i+\frac{1}{2}}^{n+1}}{\delta x} + (1-\theta) (\rho u_{i-\frac{1}{2}}^n - \rho u_{i+\frac{1}{2}}^n) + \frac{\tau}{\delta z^2} (\rho_{i+1}^n + \rho_{i-1}^n - 2 \rho_i^n)
\]
Term 3
\[\frac{\delta}{\delta x} \left[ \rho u_{j-\frac{1}{2}}^{n+1} - \rho u_{j+\frac{1}{2}}^{n+1} - \rho u_{j-\frac{1}{2}}^n + \rho u_{j+\frac{1}{2}}^n \right] \]

Term 4
\[\frac{T}{\delta^2} \left[ \rho_{j+1}^n + \rho_{j-1}^n - 2 \rho_j^n \right] \quad (3-62)\]

Expanding the Term 1 of equation (3-62):
\[\rho_{i}^{n+1} = \rho_{i}^n + \delta t \rho + \frac{\delta t}{2} \rho \cdot \cdot \cdot \]

where
\[\rho = \frac{\partial \rho}{\partial t} \quad \text{and} \quad \rho = \frac{\partial^2 \rho}{\partial t^2} \quad (3-63)\]

Substituting into the left hand side of equation (3-62):
\[\text{Term 1} = \rho \cdot \cdot \cdot + \frac{\delta t}{2} \rho \cdot \cdot \cdot \quad (3-64)\]

Expanding Term 2 of Equation (3-62)
\[\text{Term 2} = \frac{1}{\delta^2} \left[ \rho u_{j-\frac{1}{2}}^n - \rho u_{j+\frac{1}{2}}^n \right] \quad (3-65)\]

Expanding terms within brackets of equation (3-65) and dropping subscripts and superscripts on the right hand side:
\[\rho u_{j-\frac{1}{2}}^n = \rho u - \frac{\delta t}{2} \frac{\partial \rho u}{\partial x} + \frac{\delta^2}{4} \frac{\partial^2 \rho u}{\partial x^2} - \frac{\delta^3}{8} \frac{\partial^3 \rho u}{\partial x^3} \quad (3-66)\]
\[ \rho u_{j+\frac{1}{2}} = \rho u_j + \frac{\delta x}{2} \frac{\partial^2 \rho u}{\partial y^2} + \]
\[ \frac{\delta x^2}{4(2)} \frac{\partial^2 \rho u}{\partial z^2} + \frac{\delta x^3}{8(6)} \frac{\partial^3 \rho u}{\partial z^3} \]
\[ (3-67) \]

Combining equations (3-66 and 3-67):
\[ \rho u_j^n - \rho u_{j+\frac{1}{2}} = -\delta x \frac{\partial \rho u}{\partial z} - \frac{\delta x^3}{8(3)} \frac{\partial^3 \rho u}{\partial z^3} \]
\[ (3-68) \]

Dividing by \( \delta x \):
Term 2:
\[ \rho u_i^n - \rho u_{i+\frac{1}{2}} = -\frac{\partial \rho u}{\partial z} - \frac{\delta x^3}{8(3)} \frac{\partial^3 \rho u}{\partial z^3} \]
\[ (3-69) \]

Likewise Term 3:
Term 3:
\[ \frac{\Theta}{\delta z} \left[ \rho u_{j-\frac{1}{2}} - \rho u_{j+\frac{1}{2}} \right] \]

Expanding in separate parts:
\[ \rho u_{j-\frac{1}{2}} - \rho u_{j-\frac{1}{2}} = \delta t \frac{\partial \rho u_j^n}{\partial t} + \frac{\delta t^2}{2} \frac{\partial^2 \rho u_j^n}{\partial x^2} \]
\[ (3-71) \]
\[ \rho u_{j+\frac{1}{2}} - \rho u_{j+\frac{1}{2}} = \delta t \frac{\partial \rho u_{j+\frac{1}{2}}}{\partial t} \]
\[ (3-72) \]

Substituting equations (3-71 and 3-72) into equations (3-70)
Term 3:
\[ \Theta \frac{\delta t}{\delta z} \left[ \frac{\partial \rho u_j^n - \partial \rho u_{j+1}^n}{\partial t} \right] \]
\[ (3-73) \]
Expanding the terms of equations (3-73):

\[
\frac{d p u_{i}}{d t} = \frac{d p u}{d t} - \frac{\delta x}{2} \frac{d p u}{d x} + \ldots \quad (3-74)
\]

\[
\frac{d p u_{j+i}}{d t} = \frac{d p u}{d t} + \frac{\delta x}{2} \frac{d p u}{d x} + \ldots \quad (3-75)
\]

Substituting equations (3-74 and 3-75) into equation (3-73):

Term 3 = -\Theta \delta t \left[ \frac{d p u}{d x} \right] \quad (3-76)

Expanding term 4:

Term 4 = \frac{\delta z}{2} \left[ \rho_{j+i}^{n} + \rho_{j-i}^{n} - 2 \rho_{j}^{n} \right] \quad (3-77)

Expanding each term:

\[
\rho_{j+i}^{n} = \rho \frac{d \rho}{d z} \delta z + \frac{\delta z}{2} \frac{d^{2} \rho}{d z^{2}} + \rho \quad (3-78)
\]

\[
\rho_{j-i} = \rho - \frac{d \rho}{d z} \delta z + \frac{\delta z}{2} \frac{d^{2} \rho}{d z^{2}} \quad (3-79)
\]

Substituting equation (3-78 and 3-79) into equation (3-77)

Term 4 = \frac{\delta z}{2} \left[ \delta z \frac{d^{2} \rho}{d z^{2}} \right] = \rho \left[ \frac{d^{2} \rho}{d z^{2}} \right] \quad (3-80)
Combine equations (3-64, 69, 76, and 80):

\[
\frac{dP}{dt} + \frac{d^2P}{d^2t} = -\frac{dP}{d\gamma} - \frac{d^2P}{d\gamma^2} - \frac{d^3P}{d\gamma^3}
\]

\[
-\Theta \delta t \left[ \frac{dP}{d\gamma} \right] + \tau \frac{d^2P}{d\gamma^2}
\]

(3-81)

\[
\frac{dP}{dt} + \frac{dP}{d\gamma} = -\Theta \delta t \frac{dP}{d\gamma}
\]

-\frac{\delta t}{2} \frac{d^2P}{d^2t} - \frac{\delta t}{2} \frac{d^3P}{d^3t} + \tau \frac{d^2P}{d\gamma^2}

(3-82)

Rearranging the momentum equation:

\[
\frac{dP}{dt} = -\frac{dP}{d\gamma} - \frac{dP}{d\gamma^2}
\]

(3-83)

Substituting \(c^2 \delta \rho\) for \(d\rho\):

\[
\frac{dP}{dt} = -c^2 \frac{d^2P}{d\gamma^2} - \frac{dP}{d\gamma^2}
\]

(3-84)

Taking the derivative with respect to \(z\):

\[
\frac{d}{dz} \left( \frac{dP}{dt} \right) = -c^2 \frac{d^3P}{d^3\gamma} - \frac{dP}{d\gamma} \left[ \frac{d^2P}{d\gamma^2} \right]
\]

(3-85)

Where

\[
\frac{dP}{d\gamma} = u^2 \frac{dP}{d\gamma} + \rho \frac{d^2P}{d\gamma^2}
\]
Combining all terms of equation (3-85):

\[
\frac{\partial}{\partial t} \left[ \frac{\partial p}{\partial x} \right] = \frac{\partial^2 p}{\partial x^2} + 2 \frac{\partial\rho}{\partial x} \frac{\partial u}{\partial x} + \rho \frac{\partial}{\partial x} \left[ 2 \frac{\partial u}{\partial x} \right]
\]

(3-86)

Rearranging the continuity equation:

\[
\frac{\partial}{\partial t} \left( \frac{\partial p}{\partial x} \right) = - \frac{\partial p}{\partial x} \frac{\partial \rho}{\partial x} + \frac{\partial}{\partial x} \left[ \gamma \frac{\partial^2 p}{\partial x^2} \right]
\]

(3-88)

But

\[
\frac{\partial}{\partial x} \left( \gamma \frac{\partial^2 p}{\partial x^2} \right) = \gamma \frac{\partial^2 p}{\partial x^2} \frac{\partial \rho}{\partial x}
\]

= \gamma \frac{\partial^2 p}{\partial x^2} \left( \frac{\partial^2 p}{\partial x^2} - \frac{\partial p}{\partial x} - \frac{\partial u}{\partial x} \right)

(3-89)

Dropping all terms of higher order and substituting equation (3-87):

\[
\frac{\partial^2 p}{\partial x^2} = \left[ c^2 + u^2 \right] \frac{\partial^2 p}{\partial x^2} - \rho \frac{\partial^2 p}{\partial x^2} \frac{\partial u}{\partial x} + \frac{\partial^2 p}{\partial x^2} \frac{\partial u}{\partial x}
\]

(3-90)

Substituting equations (3-90) and (3-93) into equation (3-82):

\[
\frac{\partial^2 p}{\partial x^2} + \frac{\partial \partial u}{\partial x} = \Theta \delta t \left[ c^2 + u^2 \right] \frac{\partial^2 p}{\partial x^2} - \left[ c^2 + u^2 \right] \frac{\partial^2 p}{\partial x^2} \frac{\partial u}{\partial x}
\]

(3-91)
Expanding the third derivative:

\[
\frac{d^3 p}{d \xi^3} = \frac{d^2 p}{d \xi^2} \frac{dp}{d \xi} + \frac{dp}{d \xi} \frac{d^2 p}{d \xi^2} + \frac{dp}{d \xi} \frac{d^2 p}{d \xi^2} + \frac{dp}{d \xi} \frac{d^2 p}{d \xi^2} + \frac{dp}{d \xi} \frac{d^2 p}{d \xi^2}
\]

(3-93)

Keeping only the mass diffusion terms:

\[
\frac{d^3 p}{d \xi^3} = 3 \frac{d^2 p}{d \xi^2} \frac{dp}{d \xi} + \text{higher order}
\]

(3-94)

Combining terms by substituting equation (3-94) into equations (3-91) the truncational errors are revealed:
Truncation errors for the momentum equation are found in the identical manner and the resulting final equation is:

$$\frac{dp}{dx} + \frac{dpu}{dy} = [(2\theta-1)(c^2+u^2) \frac{St}{2} \left( -\frac{b^2}{8} - \frac{\tau St}{2} \right) \frac{du}{dy} + \tau] \frac{d^2p}{dy^2}$$

(3-95)

The objective of the stability analysis has now been partially accomplished with the uncovering of the truncational error terms. The final step is completed prior to each problem calculation by by picking $\Theta$, $\Phi$, and $\gamma$ such that the coefficients of the right hand sides of equations (3-95, 96) are zero. The stability conditions are summarized below:

$$\frac{\delta x^2}{8} \frac{du}{dy} \max < \gamma < \frac{\delta x^2}{4 \delta x^2}$$

$$\frac{\lambda}{\rho} > \frac{3}{2} U^2 \max St$$

$$\frac{\delta x^2}{2 \rho} + \frac{U \max \delta x^2}{\rho} \max \frac{dp}{dy}$$

(3-97)

(3-98)

where $\gamma$ has the dimensions of \( \text{distance}^2 \text{time}^{-1} \) or \( \text{in}^2 \text{sec}^{-1} \).
In addition, the particles must be limited to a movement which is less than one node per time step for the largest velocity. This is known as the Courant condition:

$$\frac{U_{\text{max}} S t}{\delta y} < 1$$  \hspace{1cm} (3-99)

Satisfying these conditions has thus far been adequate for these problems. The precise specification of stability criteria for the full two dimensional equations is very difficult; hence, the one dimensional results are applied.

In summary, the left hand side of equation (3-95) is the continuity equation as first discussed. After expansion of the finite difference representation into Taylor Series expansions, there are mass diffusion terms which appear. If the coefficients of these terms are negative, an error would be introduced which can be likened to the destruction of entropy, this manifests itself in numerical instability; however, if the terms are positive, the equation would be stable with a positive diffusion correction.

Equation 3-60 is the continuity equation used for this analysis. The additional coefficient is utilized to insure a positive mass diffusion term (where a positive mass diffusion ($\gamma > 0$) may sometimes be necessary for numerical stability).
3.8 Summary

In this chapter the conversion of the Navier-Stokes equations to the finite difference form for solution to the transient heat pipe two dimensional vapor flow problem was completed. The method used is adapted from the ICE method as proposed by Harlow (26); modified in form to calculate the flow around corners and within the vapor region of the pipe and to simulate conditions which exist immediately inside the vapor volume in the evaporator of a cylindrical heat pipe.

Through inclusion of the thermo-physical property subroutine by Varljen (31), any one of 12 working fluids can be chosen, 11 of which are liquid metals.

The applicable boundary conditions for steady state and transient solutions are discussed. It is possible to investigate all modes of the transient heat pipe problem up to the initiation of burnout with the proposed solution method.

The analyses applied to the burnout and thermal impedance calculations within the evaporator are described.

The final item covered was a stability analysis of the continuity and momentum finite difference equations from which the numerical stability criteria were established.
4.0 NUMERICAL RESULTS AND COMPARISON WITH SELECTED EXPERIMENTAL AND ANALYTICAL RESULTS

4.1 Analytical Two Dimensional Steady State Investigations

There have been many papers on the steady flow of a fluid within a hollow tube with suction and blowing at the porous wall boundaries. Yuan and Finkelstein (40) found an exact solution for the Navier Stokes equations for small and large injection and suction flows at the wall. The solution was in the form of a third order non-linear differential equation with a perturbation method being used to solve the equation for both small and large flows through the wall. The velocity at the beginning of the flow injection boundary was assumed to be fully developed Poiseuille flow. Wageman, and Guevara (41) then experimentally investigated the results of Yuan and Finkelstein with favorable results. Other works similar to those above (42, 43, 44) also found solutions to the problem of the vapor flows within heat pipes of semi-infinite length with only uniform evaporation or only uniform condensation at the boundaries. Bankston and Smith (7) and DeMichele (6) obtained solutions for two dimensional steady state vapor flow within a heat pipe. Bankston and Smith solved the Navier-Stokes equations in finite-difference form including
both suction at the outlet to simulate condensation and blow­ing at the inlet to simulate evaporation. Because the heat pipe has both condensation and evaporation, the results of Bankston and Smith were chosen as the main benchmark from which to establish the validity of the ICS method as applied to the steady state portion of the solution.

To establish an initial general comparison with the previous works, typical velocity profiles are shown in Figures 4-1 and 4-2. The velocity profiles are for both low and high Mach Numbers of vapor flow. The results of DeMichele (6), Yuan and Finkelstein (40), and others (41, 42) indicate that, at low Reynolds number, the velocity profiles in the condenser and evaporator tend to the Poiseuille form, \( u = u_0 \left( 1 - \frac{r}{\rho} \right)^2 \) as shown in Figure 4-1. As the Reynolds Number (Radial) tends to high values in the evaporator (large negative values) the velocity profiles tend to the form \( u = u_0 \cos \left( \frac{\pi}{2} \left( \frac{r}{\rho} \right)^2 \right) \). This was verified in the ICS code, as shown in Figure 4-2. Curves 3A and 5 coincide and represent the axial velocity distribution and the Poiseuille velocity distribution at the very beginning of the evaporator. These curves coincide to within 1%. The injection effects are fully indicated at the evaporator exit where the close proximity of curves 3B and 4 indicate the predicted cosine distribution has developed. For the condenser solution for large Re there is less agree­ment in the literature. The solution obtained from this
FIGURE 4-1

Velocity Profiles

Beginning of Evaporator

vs

Time (Low Mach Numbers)

1. Initial velocity distribution
   (constant)

2. Steady state velocity distribution
   (obtained by low flow ICE calculation)

3. The Poiseuille profile

Note that curves 2 and 3 exactly coincide
FIGURE 4-2
Velocity Profiles for
Near Choking Flow

1. Adiabatic Section
2. Condenser
3A. Evaporator no injection effects (coincides with 5)
3B. Modified flow at evaporator exit
4. $U \cdot \cos \left( \frac{\pi U}{H} \right)$ Knight and McInteer (exit values) (The $\Delta$ fall on the evaporator) (42)
5. $\circ \circ \circ \circ$ Poiseuille Parabolic Distribution
analysis indicates a solution between the cosine distribution and the Poiseuille distribution at the condenser entrance. The condenser inlet distribution is shown in Figure 4-2, curve 2.

To test for obvious results, the code was started with initially erratic velocity profiles but with constant pressure and zero flows at the boundaries. Shown in Figures 4-3 to 4-5 are selected cycles showing the smoothing and slowing down of the velocity vector field. The field never slowed completely to zero and began to show other instabilities as the flow became less than .01 in/sec; however, it does indicate that correct tendencies exist. As can be seen on the cycle 415 plot the only visible velocity vectors of any length indicate a flow along streamlines from left to right. The lower solid line is the vapor space centerline and is considered to be a free slip wall, i.e. no velocity gradient exists between the boundary node directly below the centerline and the visible node just above the centerline. As can be seen by the labeling, the problem used was the symmetrical heat pipe of Bankston and Smith (7) with a length to radius ratio of 20.

In view of the previous objective of comparison, a problem similar to one constructed by Bankston and Smith was investigated and the results of the two calculations are compared.
INITIAL RANDOM VELOCITY FIELD

BANKSTON AND SMITH INCORPORATE PROBLEM

FIGURE 4-3. The zero flow test—(Velocity Vector Field)
TIME = 0.006     CYCLE = 271

BANKSTON AND SMITH INCOMPRESSIBLE PROBLEM

FIGURE 4-4. The zero flow test continued (Velocity Vector Field)
FIGURE 4-5. The zero flow test continued (Velocity Vector Field)
FIGURE 4-6. The Symmetrical Heat Pipe
4.2 **Incompressible Comparison**

The problem chosen to model is that of a symmetrical heat pipe which is a pipe in which the evaporator and condenser lengths are identical and both are immediately adjacent to each other. The length to radius ratio is 20. Various problems were calculated with $-1 \gg Re \gg -10$, where $Re$ is the radial Reynolds Number in the evaporator and is negative due to the radial wall velocity being in the direction of decreasing radius.

Figure 4-6 illustrates the physical features of the pipe for the sample problem comparison. Shown in Figure 4-6 is a full cut away view of a symmetrical heat pipe with two sample streamlines of laminar fluid travel illustrated.

The results are summarized in Figures 4-7 to 4-12 where the velocity profiles are shown. In general, the flow assumed a streamline laminar field with the Poiseuille distribution being evident for low $Re$. As the $Re$ became increasingly negative, the velocity profiles resembled the cosine distribution as expected. In Figure 4-12 the velocity vectors are long enough to clearly indicate the formation of the cosine distribution. This calculation verifies the quantitative behavior of the numerical method for incompressible flows and shows good agreement with the results obtained by Bankston and Smith (7).
TIME = 0.000000    CYCLE = 0

BANKSTON AND SMITH INCOMPRESSIBLE PROBLEM

FIGURE 4-7. Initial velocity field guess for the incompressible problem. Each line represents a velocity vector where the evaporator and the condenser are to the left and right respectively. The plot is distorted for viewing, the length to radius ratio is actually 20.
FIGURE 4-8. Velocity field for Re = -0.3

For a Re of -0.3 the velocity vectors are largest in the adiabatic zone with the inlet and outlet effects on the flow field being small.
FIGURE 4-9. Velocity vector plot for an evaporator Re of -.56

The Poiseuille velocity distribution is dominant for this low value.
FIGURE 4-10. Velocity vector field for a Re of -1.1. As the Re becomes more negative the inlet effects become important.
TIME = 0.025010  CYCLE = 1470

BANKSTON AND SMITH INCOMPRESSIBLE PROBLEM

FIGURE 4-11. Velocity vector field for a Re of -2.338. The axial velocity at the end of the condenser has now begun to flow in the reverse direction (left) at the wall signifying the appearance of an adverse pressure gradient.
BANKSTON AND SMITH INCOMPRESSIBLE PROBLEM

FIGURE 4-12. Velocity vector field for a Re of -5.96. Now significant inlet and outlet effects are evident and the velocity vectors originating in the adiabatic zone clearly exhibit the cosine distribution.
Figure 4-13 is a comparison of the results of this calculation with the Bankston and Smith calculations and the accepted pressure loss for Poiseuille flow as represented by the solid line. As can be seen from the results, these calculations indicate good agreement when compared to the pressure loss that would be obtained by Poiseuille flow in the complete length of the pipe and as obtained by Bankston and Smith (7). The results indicate higher pressure loss values for higher Re (more negative). The higher values are indicative of a higher force within the pipe vapor space needed to overcome the increased velocity gradient at the wall.

The results documented in Figures 4-7 to 4-12 were calculated sequentially during the same computer run; therefore, the converged results of one run could be used as the initial guess for the next succeeding higher flow velocity calculation. Therefore, the time steps associated with each figure do not apply with each figure representing a steady state flow field at the stated Re.
v_w = radial velocity at evaporator wall
R = radius
L = evaporator length

FIGURE 4-13. Total pressure loss for symmetrical heat pipes compared with pressure loss which would occur if the flow was fully developed Poiseuille flow for the complete length of the heat pipe.
4.3 Comparison with Experimental Results

Having established the validity of the calculation (via Banlston and Smith (7)) for typical steady state incompressible problems, the solution is now extended to the compressible case. Appendices C and D are two experimental investigations for which sample problems are constructed. The two cases included in Appendices C and D are modelled in this section and the calculation results are compared with the actual experimental results.

a. Sodium Choking Experiment Comparison  The first transient problem investigated is described in Appendix D. Sodium is used as the working fluid and the case where 1440 watts were input to the evaporator with the following input parameters being used:

Card 1

IBAR = 6  
JBAR = 140  
$\Delta r = .06$  
$\Delta \gamma = 0.12$  
$\Delta t = 1.0-7$ sec.  
IPHM = 999  
PC = 0.0 (cylindrical)  
ALP = 0.1

Card 2  Label card
Card 3

BCB, BCR, BCT = -1 (No slip surface)

BCL = 1 (Free slip centerline)

GM = γ - 1

B = γ/pr = 1.667/ .8 = 2.09

λ = 4.25X10^-7 \text{lbm} \text{in-sec}^{-1}

μ = 6.37X10^-7 \text{lbm} \text{in-sec}^{-1}

εs = 2.0X10^-4 (convergence criteria)

γ = 2.0

Card 4

Various times shown in the plot

Θ = φ = 1. (fully implicit)

Card 5

L1 = 0

L2 = 58

L3 = 77

L4 = 140

NX B = NYB = 1

UL = UR = 20.

Card 6

NX = NY = 1

Two general initial blocks of vapor are described

Energy = 2157.6 \text{Btu/} \text{lbm}

PRESS = 2.0-3 \text{lbf/in}^2

WTMOL = 23.
Card 8

\[ Q_{IN} = 4920 \text{\textdegree} F \text{ or } 1440 \text{ watts} \]

\[ E_{INTRL} = 2270 \text{ to } 2264 \text{\textdegree} F \text{ linear variation at inlet} \]

\[ J_{FLUID} = 2 \text{ (sodium)} \]

\[ X_{ITINT} = 2257 \text{ to } 2235 \text{\textdegree} F \text{ linear variation at exit} \]

Card 9

\[ T_{EB} = 1410^\circ R \]

\[ T_{EE} = 1310^\circ R \]

\[ K\text{ONTER} = 1 \text{ (compressible)} \]

\[ T_{CB} = 1230^\circ R \]

\[ T_{CE} = 1000^\circ R \]

Card 10

\[ RW = .4425 \text{ in.} \]

\[ EP = 0.625 \]

\[ ID = 1 \text{ (not used)} \]

\[ W_{KMR} = 0.0012 \text{ in.} \text{ (taken from Kunz (17))} \]

\[ P_{ERMEA} = 1.0 \times 10^{-2} \text{ in.}^2 \text{ (taken from Kunz (17))} \]

Figure 4-14 is a plot of sodium enthalpy vs. temperature for the 100% saturated vapor state. These values are taken from El-Wakil (36) and used to describe the boundary conditions for the variable EINTRL and XITINT of card 8.

Results:

For a heat load of 1440 watts, the plots of Figures 4-15 to 4-20 indicate the code velocity vectors and streakline plots when the previous boundary conditions are
FIGURE 4-14
Sodium Enthalpy Saturated Vapor
vs Temperature as found in El Wakil (36)
imposed on the model. As can be seen, a region of low density progresses down the pipe with time at about the speed of sound. The low density area also is the area where velocities within the pipe become sonic and is marked with an indicator of $M = 1$ in the plots. The density increases and the velocity abruptly decreases across the pressure front as shown in the figures.

In viewing these figures, one can see the velocity fields in the evaporator for cycle 200 adjusting for the pressures and the erroneous initial first guess. As the calculation proceeds, the evaporator velocity vector field is seen to establish itself. The speed of sound in sodium is actually 2225 ft./sec. at $1500^\circ$ R for a saturated vapor with $\gamma = \text{constant}$ as indicated by Meisl and Shapiro (51). In looking at the plots, the pressure pulse progresses across the adiabatic zone, 1.8 in., (Plate B of Figure 4-15 to Plate B of Figure 4-17) in $6.0 \times 10^5$ sec. This indicates a sonic velocity, as calculated from the code of:

$$\frac{1.8}{6 \times 1.2} \times 10^5 = 2500 \frac{ft}{sec}$$

where 1.8 in. represents 15 nodes of the adiabatic zone, this indicates that the pressure pulse is proceeding with a velocity which is indicative of the sound speed.

Also to be observed are the distribution of the particles within the vapor space. The Poiseuille distribution is evident at the beginning of the evaporator and
then the cosine distribution at the evaporator exit as discussed in the previous sections.

Of particular interest are the high velocities found very close to each side of the adiabatic zone.
1. Plot (Plate A) of sodium pipe vapor space 2 x 10

2. Figure 4-15: Velocity vector field (Plate A) and streamline

STEEL-NR HP OZAKOMIC 1440 WATTs

TIME = 0.00002 CYCLE = 200

STEEL-NR HP OZAKOMIC 1440 WATTs

TIME = 0.00002 CYCLE = 200
FIGURE 4-16. Velocity vector field (Plate A) and streakline plot (Plate B) of sodium pipe vapor space $4 \times 10^{-5}$ sec. after beginning of start up.
FIGURE 4-17. Velocity vector field (Plate A) and streakline plot (Plate B) of sodium pipe vapor space $6 \times 10^{-5}$ sec. after start up. One can clearly recognize the area of high density of the shock front and the fluid profiles illustrate the Poiseuille and cosine velocity distributions as previously discussed.
FIGURE 4-18. Velocity vector field (Plate A) and streakline plot (Plate B) of sodium pipe vapor space \(8 \times 10^{-5}\) sec. after start up.
FIGURE 4-19. Velocity vector field of sodium pipe vapor space

$1 \times 10^{-4}$ sec. after start up
FIGURE 4-20. Streakline plot of sodium pipe vapor space

$1 \times 10^{-4}$ sec. after start up.
b. **Mercury Heat Pipe Experiment**

The problem tested was Test 1 of the experimental work completed on a steel-mercury heat pipe by Deverall (2). The specific internal energy of the mercury vapor at the inlet was set such that the interface inlet temperature was varied in a linear fashion identical with that of Figure 4-21. As previously discussed, the linear variation of boundary internal energies must be put in a functional form and compiled within the code. The actual statements of inlet and outlet internal energy variation used for this problem can be found in the Transient Section of the code edit in Appendix B.

The experimental heat pipe schematic is shown in Figure 4-22. This vapor space of the pipe was modelled with the ICE code with the comparison values given below. Note that the computer model was altered to shorten the pipe by neglecting the pipe lengths which protrude at either end of the experimental pipe.
FIGURE 4-21. The applicable test results of Hg steel pipe
FIGURE 4-22. Experimental Mercury Heat Pipe Schematic
Actual Experimental Model

Pipe length = 19 $\frac{5}{3}$ inches
Radius = 0.215 inches
Evaporator length = 7.25 inches
Condenser length = 7.5 inches
Working fluid = Mercury
\[ \gamma = 1.667 \]
\[ \rho = 1.245 \]
\[ \mu = 2.61 \times 10^{-6} \text{ lbm/sec-in} \]
\[ \lambda = -1.745 \times 10^{-6} \text{ lbm/sec-in} \]

Computer Model Vapor Space

DR = 0.043 \quad IBAR = 5
DZ = 0.095 \quad JBAR = 155
Length = 14.73 inches
Evaporator length = 6.84 inches
Condenser length = 7.03 inches
Working fluid = Mercury with same fluid properties

The evaporator will be modeled as 72 nodes and the condenser as 74 nodes. This is neglecting the pipe ends and shortening both the condenser and evaporator and allowing the 0.85 inch adiabatic zone.
Figure 4-21 is a temperature distribution in the heat pipe in which the condenser was carefully controlled so that heat was removed fast enough such that the sonic limit was evident at any particular heat input shown in Figure 4-21. This was the same experimental results described in Appendix C and illustrated in Figure C-3.

The evaporator and condenser boundary energies were described in the same axial variation as the temperature distribution for the 90 watt line of Figure 4-21.

The initial pressure description of the vapor space is chosen so that the values lie within the evaporator and condenser boundary values. This will insure a smooth start up numerically.

The wick parameters taken from Kunz, et al, (17, 34) for the 100 mesh screen of 3 layers are:

Minimum pore radius $5.2 \times 10^{-4}$ in.
Porosity 67.9%
Permeability $2.36 \times 10^{-7}$ in.$^2$

The input cards were set up in a similar manner to the input cards for the compressible problem of 4.3.
FIGURE 4-23. Initial formation of pressure front during startup of mercury heat pipe.
FIGURE 4-24. Mercury heat pipe startup
FIGURE 4-25. Mercury heat pipe startup
Figures 4-23 to 4-25 illustrate the troubles one has with this type of code. Figure 4-23 shows the initial formation of the pressure front. Figure 4-24 illustrates the movement into the adiabatic zone; however, notice in the condenser that the extreme boundary condition of a very low mercury saturated vapor pressure has caused very high condenser exit velocities. With these very high velocity gradients in the condenser, the fluid has been swept out and instabilities are beginning to be evident in the velocity field. Figure 4-25 occurred just prior to the problem terminating due to instabilities (high gradients in the condenser). Notice that the pressure front has progressed further down the pipe; however, the associated velocity field within the condenser has become more erratic. In Figure 4-25 the axial velocity at the centerline in the adiabatic zone has reached 5274 in/sec. For these conditions, Deverall (3) indicates that the sonic limit is 6470 in/sec. The flow, therefore, has not reached choking; however, if the stability problem is solved the results would indicate good agreement.
5.0 REACTOR TRANSIENT STUDIES

A split core reactor utilizing an out-of-core thermionic conversion system has been proposed by Niederauer (45) as a power source to be applied to future large space systems. Figure 5-1 illustrates the reactor and heat pipes protruding from each side of a core that is split in the center. The split core design allows control by variation of the gap size shown in Figure 5-1.

The heat pipes within this reactor are made of Tungsten with Lithium as the working fluid. A typical pipe is incorporated in the core as shown in Figure 5-2; the evaporator is cylindrical surrounded by fuel. The evaporator extends into a condenser of rectangular shape. The positioning of these pipes are clearly shown in Figure 5-1.

A complete discussion of the design referred to above can be found in References (45,46, and 47).

To investigate this design, a non-scale representation of an average fuel piece and heat pipe was constructed by Niederauer (46). This model was utilized in the reactor dynamics code AIROS (48) which was modified to handle non linear reactivity coefficients, heat transfer from radiators, heat pipe startup, and stepping of the startup front down the nodal model of the pipe. The heat pipe model was two dimensional, as shown in Figure 5-3, with 7 nodes completely describing the heat pipe.
FIGURE 5-1. Space Power Plant Design by Niederauer (45)
FIGURE 5-2. Incorporation of Tungsten-Lithium Heat Pipes into Reactor Design by Niederauer (45)
FIGURE 5-3. Nodal Model of Reactor Heat Pipe Used in the Analysis by the Computer Code Airos (48)
Attached to the reactor heat pipe is a radiator heat pipe being completely described by two nodes. The seven nodes describing the reactor heat pipe and the two nodes describing the radiator pipe do not include any of the vapor space. The heat flow is described by an unsteady energy balance equation taken on the evaporator or hot zone as it is called by Niederauer (46):

\[ nC \frac{dT_s}{dt} = (U_A)_{W_L} (T_W - T_L) - \omega_h \]  \hspace{1cm} (5-1)

where

\[ \omega = \text{mass flow rate} \]
\[ = A \theta \left[ \frac{k}{2(\kappa + 1)RT_l} \right] \frac{1}{2} \exp \left[ -h/RT_L \right] \]  \hspace{1cm} (5-2)

\[ m_l = \text{mass of the liquid in the hot zone} \]
\[ C = \text{specific heat of the coolant in the hot zone} \]
\[ U = \text{the overall heat transfer coefficient} \]
\[ A = \text{the vapor flow cross sectional area} \]
\[ B = \text{vapor constant} \]
\[ k = \frac{C_p}{C_v} = \frac{5}{3} \]
\[ R = \text{Gas constant} \]
\[ h = \text{heat of vaporization} \]

W and L indicate heat pipe wall and liquid coolant.

This is simply a statement that the change in energy stored within the working fluid must be equal to the energy gained in the evaporator minus the energy carried from the evaporator by the latent heat multiplied by the mass flow rate. The mass flow rate, \( \omega \), is obtained from
the one dimensional compressible steady flow approximations of Shapiro (49) with the assumptions that wall friction in the evaporator is negligible, the vapor flow speed is less than the movement of the sonic front, and the vapor flow is laminar. Integrating the flow equations between the start of the evaporator and that point where the Mach Number is 1.0 yields a mass flow equation in terms of $\dot{W}$, the mass flow rate at $M$, or Mach number=1.0. This resulting equation is evaluated at $M=0$ and a flow rate for the evaporator is found to be:

$$\dot{W} = p_o A \left[ \frac{8}{RT_l [2(8+1)]} \right]^{1/2}$$

where

- $p_o$ is the pressure in the evaporator at the end
- $A$ is the cross sectional area of the vapor space
- $T_l$ is the liquid temperature

When the equation, which approximates the properties of lithium vapor in the region of interest, is substituted, equation (5-2) follows.

The objective of this chapter then, is to look at the vapor flow within this pipe as described by the ICE method using the transient temperature description of the wick evaporator and condenser coolant nodes. The Airos (48) model used by Niederauer is shown in Figure 5-3. For the ICE analysis, Niederauer's results for nodes 11, 12, and 14 were used to define the at power boundary conditions and
the results for nodes 11 and 12 were used to define the start up boundary conditions at the evaporator and condenser of the ICE model. The transient evaporator inlet then is defined by Figure 5-4 which shows the reactivity, neutron density, heat flux, and temperature transients. The coolant temperature and heat flux are used for the evaporator boundary conditions. The condenser transients are not shown; however, the actual values of both evaporator and condenser were obtained from Neiderauer (50) in the form of computer printed tables. The U235 start up is investigated.

The coolant temperature vs time is assumed to be the parameter which dictates the conditions at the inlet. There is a lag in the temperature variation when compared to the heat flux vs time and for this reason the coolant temperature is used for the analysis.

The problem was set up as a final goal for this dissertation to show how this code might be used to aid in design problems. Figure 5-5 illustrates the actual cutaway reactor heat pipe and its surroundings imbedded within the reactor. For the purposes of the problem the heat pipe will be assumed completely cylindrical with the condenser area being equal to the actual rectangular condenser area thus providing the identical size heat sink.
FIGURE 5-4. Neutron Density, Heat Flux, and Temperature Histories for a Start Up Transient as found by an Airos (48) Analysis based upon Niederauer's Model. The Reactivity was Input for the Analysis.
FIGURE 5-5. Cutaway view of modelled heat pipe
The dimensions of the evaporator with the annular wick described by Niederauer (47) are shown in Table 5-1.

Table 5-1

Dimensions of Evaporator with Annular Wick

<table>
<thead>
<tr>
<th>Region</th>
<th>Region Thickness</th>
<th>Outer Radius</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wall</td>
<td>.020</td>
<td>.197</td>
</tr>
<tr>
<td>Liquid</td>
<td>.022</td>
<td>.176</td>
</tr>
<tr>
<td>Wick</td>
<td>.017</td>
<td>.155</td>
</tr>
<tr>
<td>Vapor</td>
<td></td>
<td>.135</td>
</tr>
</tbody>
</table>

The wick region is divided into 70 percent wick material (tungsten) and 30 percent coolant by volume. The rectangular condenser had the following dimensions as described by Niederauer (47) in Table 5-2.

Table 5-2

Dimensions of Condenser

<table>
<thead>
<tr>
<th>Description</th>
<th>Inch</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cutside width</td>
<td>.687</td>
</tr>
<tr>
<td>Cutside depth</td>
<td>.394</td>
</tr>
<tr>
<td>Solid wall thickness</td>
<td>.0315</td>
</tr>
<tr>
<td>Channel depth</td>
<td>.00394</td>
</tr>
<tr>
<td>Overall wall thickness</td>
<td>.0354</td>
</tr>
</tbody>
</table>
The channel region is equally divided between tungsten and Lithium.

From these dimensions it is seen that the evaporator has a vapor space radius of 0.135 in. The evaporator length is taken to be 4.8 in. and the condenser length is taken to be 3.0 in. The overall length of the pipe is taken to be 8.8 in. and the condenser is assumed cylindrical with an equal surface area to the actual. The evaporator is assumed to be surrounded by fuel over a length of 4.8 in. with 1.0 in. being surrounded by the axial reflector and adjoining void (which is taken to be the adiabatic zone). The pipe is shown in Figure 5-6.

FIGURE 5-6. Heat Pipe model used for reactor transient studies
The parameters for the pipe model are as follows:

\[ \text{Dr} = 0.027 \]
\[ \text{IBAR} = \frac{135}{0.027} \approx 5 \]
\[ \text{DZ} = 0.054 \]
\[ \text{JBAR} = \frac{8.8}{0.054} \approx 158 \text{ (node limit)} \]

or a total of \((7+2) \times (110+2) = 1008\) nodes will describe the model. The code is very unreliable for node length/width ratios greater than 2.0 and, due to computer limitations, the number of nodes in the axial coordinate must not exceed 158.

The other parameters for this study are as follows:

Lithium vapor viscosity: \(2 \times 10^{-6} \text{g/sec cm} \)

\[ \mu = 2 \times 10^{-4} \times 2.54/454 \frac{\text{lbm}}{\text{in} \cdot \text{sec}} = 1.12 \times 10^{-6} \frac{\text{lbm}}{\text{in} \cdot \text{sec}} \]

\[ \gamma = -\frac{3}{2} \mu = -0.747 \times 10^{-6} \frac{\text{lbm}}{\text{in} \cdot \text{sec}} \]

\[ \theta = \frac{c_p \mu}{k} = 1.663 \text{ Meisl, et al (51)} \]

The parameter \(B\) of the energy equation is given by:

\[ B = \frac{\theta}{Pr} \approx 0.792 \]

An example of the calculation of \(B\) can be seen by using the thermal conductivity, specific heat, and viscosity relationships of Varljen (31) found in the code edit Appendix B to calculate \(Pr\) at 1750° K.
Calculation of the stability variable, $\gamma$, yields:

$$\gamma = \frac{8x^2}{8} \frac{du}{dx} = \frac{(0.54)^2 10^4}{8} = 3.64$$

Evaporator extends for 89 nodes
Condenser extends for 56 nodes

Two basic situations will be considered for this analysis: (1) the pipe exists in a normal at power operation scheme, and (2) the startup transient is applied to the pipe.

For the application of the normal operational mode boundary conditions no violent or sonic conditions would be expected for the startup experienced by the code. Figures 5-7-
FIGURE 5-7. Conditions at 4.4 × 10^{-4} sec. after application of "at power" boundary conditions to heat pipe modelled after the design described by Niedersuer (45)
FIGURE 5-8. Conditions at $4.6 \times 10^{-4}$ sec. after application of "at power" boundary conditions to heat pipe modelled after the design described by Niederauer (45)
FIGURE 5-9. Conditions at \(5 \times 10^{-4}\) sec. after application of "at power" boundary conditions to heat pipe modelled after the design described by Niederauer (45).
FIGURE 5-10. Conditions at $5.4 \times 10^{-4}$ sec. after application of "at power" boundary conditions to heat pipe modelled after the design described by Niederauer (45)
FIGURE 5-11. Final steady state streakline plot for the application of normal "at power" boundary conditions. The streamlines are fully established.
5-11 document the vapor flow for these conditions. A normal operational heat input is applied to the evaporator and extracted through the condenser of Figure 5-6. The evaporator and condenser are represented by nodes 11 and 14 respectively, of Niederauer's model of Figure 5-3. As seen in Figures 5-7 to 5-10, the flow never passes through the choking limit and the code start up is relatively smooth. The streamlines can be clearly seen forming a continuous line from the evaporator to the condenser. The fluid previously in the condenser at the beginning of the problem is being pushed aside and cut off the pipe by the advancing flow. This, then, can be contrasted with the abrupt start-ups of the previous chapter. The density remains relatively constant throughout the vapor space and the flow remains stable.

For the start-up transient the model of Figure 5-3 is assumed to consist of the evaporator and the new condenser nodes numbers 12 and 14. In the ICE model, Figure 5-6, during start-up the condenser becomes essentially all areas at the initial temperature which, in addition to the designated condenser, also includes the 1.0 inch of reflector which acts as a condenser until it is up to the operational temperature. At this time, however, the pipe will be operating normally. The objective is to view the severity of the start-up during the initial period.

Shown in Table 5-3 is a temperature heat flux history during the most rapid rise during the start-up transient.
Table 5-3 is a sample of the conditions at the evaporator (node 11, Figure 5-3) and the condenser (nodes 12 and 14, Figure 5-3) at specific times after the startup is initiated. The pipe was assumed to include both nodes 12 and 14 which would be the most conservative assumption. Niederauer assumed the absence of node 14 for startup analyses.

During the startup transient, the evaporator temperature node 11, (Figure 5-3) varies between 300 K (ambient) and 1495.5°K., the severest portion of the transient. The ICE code is used to model the conditions existing at 24.187 sec. and is then allowed to run for a representative amount of time. The entire problem cannot be run due to the small time step for cycle advancement, as dictated by stability requirements.

Table 5-3
Values of the Boundary Variables During the Startup

<table>
<thead>
<tr>
<th>Transient as a Function of Time</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Time (sec)</strong></td>
</tr>
<tr>
<td>Node 11</td>
</tr>
<tr>
<td>24.187</td>
</tr>
<tr>
<td>25.187</td>
</tr>
</tbody>
</table>
The wick was taken to have a minimum pore radius of 0.003 inches as indicated by Bacigalupi (52) and the permeability was estimated to be $2.36 \times 10^7$ in$^2$.

The results of the calculation are shown in Figures 5-12,13 and illustrate the velocity vectors existing in the pipe as the boundary conditions of Table 5-3 are imposed. The time span represented by Figures 5-12,13 is 100 microseconds. The calculation reveals supersonic velocities in the condenser with choking at the throat. The high velocities shown are not the direct cause of failure. The calculation reveals that for the input wick parameters, the capillary pressure change is overcome by the sum of the pressure changes in the vapor and wick; a direct result of the severe boundary conditions. The flux value of Table 4 will not cause burnout; therefore, this calculation shows that previous results obtained by Niederauer based on simplifying approximations are not consistent. It must be understood that this portion of Niederauer's work was not the primary objective of his analysis and would not be expected to produce exacting results; however, the analysis presented here does point out a weakness which could be eliminated if a code such as this were employed. At $1410^\circ$K or $2078^\circ$F the sonic limit occurred at a flux of approximately 13,000 watts/cm. This is in agreement with other published values for lithium working fluid heat pipes. In actual practice, the flux controls the temperature at the interface and the temps.
FIGURE 5-12 The development of the velocity vectors within the vapor volume after application of startup boundary conditions for 24.187 sec.
FIGURE 5-13. Plate A represents conditions when the pipe burnout conditions are being closely approached. Plate B indicates the very high velocities present at the time of burnout.
present would adjust to cause the proper flux removal value and the pipe would not burn out. For the reasons above, it is concluded that the temperatures predicted during start up at the interface by the Airco code as modified by Niede-rauer (46) are not realistic for the heat flux being predicted.
6.1 Discussion of Results

In the essentially incompressible problem modelled after the pipe of Bankston and Smith (7), it was shown that the code obtains both velocity and pressure profiles which show good agreement with analytical approaches and the numerical stream function approach of Bankston and Smith (7). The other problem results indicate that for the higher Reynolds numbers the compressibility effects shown from experimental evidence to exist are predicted by the calculation.

In Chapter 5.0, the code was used to point out inconsistencies in the analysis obtained for the conceptual design of a space power supply. Herein lies an important reason for development and use of such a design tool; if the designer can obtain information which would be vital to any start up or transient in the conceptual design of any space power reactor, potentially costly problems can be avoided prior to construction. Further, all the design analyses can be cross-checked for consistency and any discrepancies can be more closely investigated to obtain optimum performance.
The code could possibly be used as a design tool to calculate flow fields within heat pipes designed to be used in any environment where severe thermal pulses are expected.

The high velocities attained during sonic limiting flow do not in themselves cause burnout. As more heat is added, the temperature rises and the sonic limit is increased; however, the possibility of failure due to entrainment and the wicking limit are still possibilities for any severe start up situation.

The flow of a compressible fluid around obstacles can also be studied with some modification to the present code.
6.2 **Limitations of the Technique and Recommendations for Future Study**

a. **Single Phase Assumption** This approach is exclusively a single phase analysis which neglects the phenomenon of condensation of the vapor as it passes through the region where the sonic limit is known to exist. This process, known as a condensation shock, can be divided into two general steps.

1. The formation of the liquid nuclei
2. The growth of these nuclei into larger drops

In an analytical treatment of a sodium working fluid heat pipe by Levy (53), this effect was investigated using both the classical theory and the newer Lothe-Pound theory for determination of the droplet nucleation rates. The results indicated that with both theories of droplet nucleation, the amount of vapor condensed was negligible upstream of the sonic point and had a minor influence on the sonic heat transfer rate. If this code is considered as a design tool, however, this effect must be incorporated for a conservative analytical treatment because this effect tends to reduce the heat transferred at any given temperature.

b. **Vapor Dissociation and Recombination** Implied in this analysis is the fact that the vapor consists of a pure monatomic gas. In reality some liquid metal vapors can undergo reactions where molecules composed of two atoms
are created and destroyed (such as with sodium, \(2\text{Na} \rightleftharpoons \text{Na}\)) with which comes a certain amount of heat production or absorption. The effects of this reaction have been shown analytically by Levy \((53)\) to be negligible; however, as with the vapor condensation process considered previously, the error introduced is in a nonconservative direction and should be considered if a design problem is attempted with this code.

c. **Boiling Limit for Liquid Metal Working Fluids**

The limit of boiling in the evaporator wick of a liquid metal pipe should be investigated and added to the analysis based upon future and more detailed investigations. The boiling limit for liquid metal heat pipes is a difficult problem primarily due to the fact that the heat flux for boiling in heat pipes can be larger than that for pool boiling depending on the capillary wick structure. Procedures such as are derived by Silverstein \((54)\) to calculate this limiting heat flux provide a starting point for such an analysis.

The boiling limit for fluids other than liquid metals can be predicted as described by Chun\((55)\).

Experimental investigations using neutron radiographic techniques were utilized by Moss\((56)\) to study the boiling limit in water heat pipes and an approximate theoretical description emerged.
It is restated here that presently the results of this code cannot be expected to yield valid results for water heat pipes above that heat flux for which vaporization within the wick structure would be expected to occur as discussed in Chapter 2.0. This conclusion is based on the experimental results of Fox, Carothers, and Thomson (14).

d. Transient Boiling Effects

In order to understand the transient flows which occur within the vapor phase, the liquid flow within the evaporator wick must be satisfactorily described. The liquid within the wick can be either stationary as in a start up from zero heat load or flowing steadily as would be the case in a heat pipe operating at a steady state which experiences a sudden increase in heat input. For these reasons the description of a transient boiling phenomenon must be considered. McLaughlin (57) indicates that the factors which influence the "transient boiling" curve (and thus the cooling efficiency of the heat pipe) include: the increase with time and the magnitude of the transient heat pulse, the heat flux existing prior to any heat pulse, the subcooling of the working fluid, and the thermal capacitance of the heated surface.

Recent work in the area of transient boiling phenomena has included investigations of transient pool boiling with various power transients, and quench cooling of hot bodies as indicated by McLaughlin (57). It has been
found that increased transient heat fluxes can be applied without reaching burnout when compared to a steady state heat flux application. As stated by McLaughlin (57) further experimental work is required to verify these statements for the geometries, boiling surfaces, and transient response of the heat pipe cooling system under general transient loads.

If implemented into the code, the boiling surface heat transfer rate could be an iteration variable in the solution scheme. The wall-saturation temperature difference \( (T_w - T_{sat}) \) could be input as a tabular function of boiling surface heat flux at a fixed saturation temperature. A correction could then be made within the program to account for the effect of saturation temperature. An experimentally obtained \( q_B vs. \Delta T_B \) curve could be used for the transient heat flux calculation.

e. Entrainment Limit

The entrainment limit can be approximated by the following equation as described by Kemme (58):

\[
\frac{\rho_v V^2 l}{2 \pi \gamma} = \frac{\text{Vapor inertial forces}}{\text{Surface tension forces}}
\]

\( (6-1) \)

where

- \( l \) is some representative characteristic length
- \( \gamma \) is the surface tension
\( V \) is the vapor velocity

\( \rho_v \) is the vapor density

When the vapor inertial forces exceed the liquid surface tension forces, i.e., equation (6-1) becomes larger than unity the liquid of the wick becomes entrained within the vapor. This generally does not occur until after the wicking limit has already caused burnout, however, consideration should be given to incorporation of such an analysis into the analysis.

6.3 **Closing Statement**

It is believed that this study might be the beginning of liquid metal heat pipe transient design tools for use in the creation of nuclear powered space exploration vehicles. It is the firm belief of this author that even though at the present time man has turned away from the challenge of space, his driving thirst for knowledge will eventually force him back into the unknown; and for the successful accomplishment of this quest, he will need a long lived and reliable nuclear power source using heat pipes.
APPENDIX A

FLOW DIAGRAM FOR THE IMPLICIT CONTINUOUS EULERIAN METHOD
AS ADAPTED TO THE HEAT PIPE PROBLEM
Subroutine Prog

Read IBAR, JBAR, DR, DZ, DT, PC, ALP

Read name
Read BCB, BCR, BCT, BCL, GM, B, LAMBDA, VISCOS, EPS, GR, GZ, TAU
Read T, TWPLT, WPRT, TW, TWFIN, LPR, THETA, PHI

PRINT ALL INFORMATION JUST READ

Calculate Radii for Use in Cylindrical Coordinates

\[
\begin{align*}
IPI &= IBAR + 1 \\
JPI &= JBAR + 1
\end{align*}
\]

Obtain information pertaining to inflow, outflow
Read TYPE L1 L2 L3 L4 NXB NYB UL UR

Set Boundary Velocities and Multipliers
Do 38 J = 1, JP2
   BCLT (J) = BCL
   ROS (J) = 1
   RCRT (J) = BCR
   U(IPI,J) = U(IP2,J) (INLET)
   U(IP2,J) = U(IPI,J) (OUTLET)

Generate Particles

Read IX NY XC YC XD YD UO VD, ENERGY, PRESS

\[
\begin{align*}
NX &= 0 \\
NO \quad & \rightarrow 80 \\
YES \quad & \rightarrow 40
\end{align*}
\]

INITIALIZE IN MESH QUANTITIES FOR THE FLUID clock DESCRIBED

Initial guess P, RHO, INTRNL, and E

Read TEMP, EINTRL, JFLUID, XITINT

INITIAL TEMPERATURE AND INTERNAL E.
AT THE INLET AND OUTLET, FLUID NO.
EXIT ENERGY

FRRT 250
Go to 645
Set All Velocities on Obstacle Boundaries if Obstacles Exist

Go to 250
If $\text{Cycle} = 0$
from 645

If $(\text{Cycle} > 0)$
from 700

250

Label cells
Boundary
Full
Surface
Empty
Obstacle
Depending on how
fluid has moved

Find Latent Heat
by Entering Subroutine TRANP

Adjust Inlet and outlet
transient internal
energy functions and evaporator
and condenser temperatures which
can be varied arbitrarily by the
user when the program is recompiled.

Use temperatures to
obtain inlet and outlet pressures from subroutine TRANP

Initialize or update RHO, PBAR, and the perpendicular
velocities across boundaries
using new pressures and boundary energies

Set INTRNL at inlet and outlet

Reflag cells
dependent on
fluid movement

NO $\text{Cycle} = 0$

YES

330

Initialize all parallel wall velocities

If $(\text{Cycle} = 0)$ Go to 500

NO $650$

Has problem converged

yes

400

113

400
This is the control section

Check if time plots, prints, and tape dumps are desired

Increment cycle and time

Plot
1. Velocity Vectors
2. Particle Positions

Go to 500
If cycle = 0
From 330

500 Initialize for this cycle

Has problem converged

501 Calculate Q
Calculate RICE
Calculate S

YES Cycle = 0

400 NO
Calculate G
Put last cycle RHO values into SIGMA array

YES Cycle ≠ 1

400 NO

Continued to next page

Bypass iteration scheme until energy calculation is performed
calculate $x^n$

Set all PBAR'S on non flow Boundary equal to PBAR'S just inside boundary

Is this the first iteration

NO

Is $\frac{\text{ABS}(x^n - x^{n+1})}{\text{ABS}(x^n + x^{n+1})} \leq \varepsilon$

YES GO TO 600

calculate $x^{n+1}$

calculate PBAR values

Set non flow PBAR and RHO values equal across boundaries

The error flag can be set at any iteration number desired

ITER is the iteration number
Is Incompressible flag set

No

Call TRANP to calculate Inlet SAT pressure as a function of temperature

Pressure at Inlet = TRANP pressure

Print Out Temperatures at Inlet and Outlet

Go to next page
If cycle = 0
From Initialization section which begins with statement 250

Calculate parallel velocities on the boundary

Calculate flow boundary velocities
Inlet-function of heat load
Outlet-incompressible dictated by continuity
compressible calculated from condenser inlet mass flow

calculate Radial Re

Set Boundary Velocities on Obstacles (This feature has not been documented or tested)
This will depend on the particular computer involved.
Calculate Energies (Total and Internal)
Set non flow boundary energies to values inside mesh (DO NOT SET INLET AND OUTLET VALUES)

Find total energy in mesh
\[ E_{\text{TOTAL}} = \sum_{i=2}^{IP1} \sum_{j=2}^{JP1} E(i,j) \]

Calculate \( C-(\text{Sound speed})^2 \) and Pressure within the mesh and across non flow boundaries

Go to next page
Find total internal energy in mesh

$$\text{ERINT}_{J=2}^{I=2} \leq \text{Internal energy (ij)}$$

Calculate inlet, outlet and condenser mass flow
Are they all equal?

No

400

Yes

Problem has converged
Set convergence flag

Velocity field
Stable

Yes

Cancel convergence flag

No

Increase calculational, print, and plot interval

400
APPENDIX B

IBM 370

LISTING OF THE ICE CODE AS

ADAPTED TO THIS PROBLEM
ICE IS THE IMPLICIT-CONTINUOUS FULERIAN METHOD FOR CALCULATION TRANSIENT COMPRRESSIBLE FLUID FLOW PROBLEMS WITHIN CYLINDRICAL GEOMETRIES. THE ICE METHOD WAS WRITTEN BY ANTHONY A. AMSDEN AND FRANCIS H. HARLOW OF LOS ALAMOS SCIENTIFIC LAB. THIS PROGRAM WAS WRITTEN BY C. H. BOWERS IN JUNE 1973 BASED ON THE ICE METHOD TO INVESTIGATE FLOW TRANSIENTS IN THE VAPOR VOLUME OF HEAT PIPES

COMMON A1(1200), A2(1200), A3(1200), A4(1200), A5(1200), A6(1200),
A7(1200), A8(1200), A9(1200), A10(1200), A11(1200), A12(1200),
1FLUX,QIN,KONER,WKMR,
2A3(1200), A14(1200), A,GM,ALP,B,BCB,BCL,BCLT(160), BCR,VI,FLAMB,DD,
2BCT,5ND,LAMBDA, COF1, COF2, Cols, CYCLE, DR, DH, DRODZ, DTHOLD,
3DROU, DT, DTODR, DTODZ, DZ, DZODR, EMP, EPS, FUL, GR, GZ,
4II(1200), IALL, IPT, IP , I P H M, JBAR, JPI, LENCOM, LPB,
5LPR, NAME(D), NIN, NP, VISCOS, NUMTO, O8, PC, R(40), RDR, RDR2,
6RDRDZ, RDZ, RDZ2, RIP(160), ROS(160), SUR, TPLT, TPRT, TT, TWFNP, TWPRT,
7JFLUID, TEMP, UNIVG, DEP, BCR(160), LL1, LL2, LL3,
7PRESS, TB, TEE, TCB, ICE ,
7TAU, THTFA, PHI, TEMPI, TEMPI, THTML, EINTRL, XITINT,
8TWPT, TWT, TYPE, UL, UR, W, XDIS, XP(14000), YDIS, YFIR,
9IEND

COMMON /RS/ RSTART
EQUIVALENCE (A1,F), (A2,UL), (A3, V), (A4, P), (A5, RICE), (A6, PBAR),
1(A7, RHO), (A8, S), (A9, G), (A10, C), (A11, SIGMA),
2(A12, INTNRL), (A13, E), (A14, KF), (A14, G)

REAL *NU
INTEGER CYCLE, TYPE
LOGICAL RSTART

--- INITIALIZE.
DO 6 J=1,1200
A2(J)=C.
A3(J)=C.
A4(J)=C.
A5(J)=C.
A6(J)=C.
A7(J)=C.
A8(J)=C.
A9(J)=C.
A10(J)=C.
A11(J)=C.
A12(J)=C.
A13(J)=C.
I1(J)=C.
A14(J)=C.
6 A1(J)=0.C
LPB=14000
### Release 1.1

```
C ----  IPAR = NO. OF INSIDE CELLS IN X OR R DIRECTION.
C ----  JBAR = NO. OF INSIDE CELLS IN Y OR Z DIRECTION.
C ----  FOR RESTART -
C ----  IBAR = 0
C ----  FOR TERMINATION -
C ----  IRAR = 0 OR EOF
5 READ (5,1000) IRAR,JBAR,DR,DZ,DT,IPHM,PC,ALP
      DH=DR/2.
      DD=386.4
10 RSTART = .FALS.
      IF(DR) 11,7,7
11 ISENT=1
7 CONTINUE
      IF(IBAR) 55,15,50
C ----  RESTART FROM DUMP TAPE = TAPE7.
15 WRITE ( 6,2000)
      WRITE (12,2000)
      RSTART = .TRUE.
      IF(ISENT.EQ.1) READ 8,FACTOR
8 FORMAT(F10.6)
      REWIND 7
      NDUMP = JBAR
      LENNEW = LENCOM
20 READ (.7) (A1(I),I=1,LENNEW)
25 WRITE ( 6,2002) NUMTD
      WRITE (12,2002) NUMTD
      GO TO 60
30 IF(LENNEW.EQ.LENCOM) GO TO 35
      WRITE ( 6,2004) LENCOM,LENNEW
      WRITE (12,2004) LENCOM,LENNEW
      GO TO 60
35 IF(NDUMP-NUMTD) 40,45,20
40 WRITE ( 6,2006) NDUMP
      WRITE (12,2006) NDUMP
      GO TO 60
```
45 WRITE (6,2008) NUMTD,NAME,T,CYCLE
46 WRITE (12,2008) NUMTD,NAME,T,CYCLE
47 IF (ISENT .EQ. 1) WFNF=WFNF*FACTOR
48 NUMTD = NUMTD+1
49 REWIND 7
50 IP2 = IBAR+2
51 JP2 = JBAR+2
53 ISENT=0
54 CALL PLOTE2
55 CONTINUE
56 REWIND 8
57 STOP
58 C ----- FORMAT.
59 1000 FORMAT (215,'E0.3,15,'E0.3)
60 2000 FORMAT (1H1)
61 2002 FORMAT ('FOF ON RESTART TAPE. LAST TAPE DUMP WAS NO. ',15)
62 2004 FORMAT ('C. RESTART COMMON LENGTH IS ',15,' AND CURRENT COMMON LENG
63 = 1TH IS ',15,'. THEY SHOULD BE THE SAME. ')
64
RELEASE 1.1
MAIN
DATE = 73111
C1/42/10

2006 FORMAT ('FO TAPE DUMP NO. ',15,' NOT FOUND')
2008 FORMAT ('FO RESTARTING FROM DUMP NUMBER ',15,'3X,6AR,' T=','F12.5,
11 CYCLE=',15)
END
COMMON A(12CC), A2(12CC), A3(12CC), A4(12CC), A5(12CC), A6(12CC), A7(12CC), A8(12CC), A9(12CC), A10(12CC), A11(12CC), A12(12CC),
1FLUX, QIN, KINFE, WKMR, RW, EP, PRFME,
2A13(12CC), A14(12CC), A15(12CC), A16(12CC), A17(12CC), A18(12CC), A19(12CC), A20(12CC), A21(12CC), A22(12CC), A23(12CC), A24(12CC),
3RCT, RNP, LAMPA, COF1, COF2, CCLS, CYCLE, DR, DH, DRODZ, DTHOLD,
4DROU, DT, DTCMR, DTMDZ, DZ, DZZDD, EMP, EPS, FUL, GR, GZ,
5L1(12CC), IALL, IBAR, IP1, IPHM, JBAR, JP1, LENCOM, LPB,
6LPR, NAME(6), IN, NP, VISCONS, NUMVR, OB, PC, R(40), RPR, RRD2,
7RDRDZ, RD?, RC?, RIL(160), ROS(160), SUR, TPLT, TPRT, TT, TWFIN, TWPLT,
8JFLUID, TEMP, UNIVG, DEF, BCR(160), LLL1, LLL2, LLL3,
9FREV, TST, TEE, TCE, TCH,
10TAU, THETA, PHI, TEMPT, TEMPI, WTMOL, EINTL, XINT, TINT,
11TWPT, TWT, TYPE, UL, UR, W, XDIS, XP(14000), YMIS, YFIR,
12IFND
COMMON /PR$/ PSTART
COMMON /C3ST/ L1, L2, L3, L4, L5, L6, L7
DIMENSION F(IP2, JP2), U(IP2, JP2), V(IP2, JP2), P(IP2, JP2),
1O(IP2, JP2), RICE(IP2, JP2), PBAR(IP2, JP2), RHO(IP2, JP2), S(IP2, JP2),
2G(IP2, JP2), C(IP2, JP2), SIGMA(IP2, JP2),
3INTRL(IP2, JP2), E(IP2, JP2),
4KIP(IP2, JP2), MSCP(2)
REAL INTRNL, LAMBDA
INTEGER CYCLE, TYPE
LOGICAL PSTART
1 FORMAT(1M1)
2 FORMAT(6A8)
3 FORMAT(' FLOW', 'I5)
4 FORMAT('E6.3, I2, E6.3')
5 FORMAT(' IBAR=*, I3, JBAR=*, I5, DR=*, F8.3, DZ=*, F8.3, D',
6 FORMAT(' IF12.5, IPHM=*, I2, PC=*, F8.3, ALP=*, F8.3)
7 FORMAT(' RCE-R=T-L=*, F5.1, 1X), GM=*, F8.3, 4X, B=*, F8.3, 6X,')
8 FORMAT('1DA=*, F10.3, VISCOS=*, E10.3, 4X, EPS=*, F8.5, 5X, GR=*, F8.3,')
9 FORMAT(' GZ=*, 2F8.3, 5X, TAU=*, E10.3)
7 FORMAT(' T=*, E12.5, THPLT=*, E12.5, THWPRT=*, F12.5, TWIT=*, E12.5,')
1 FORMAT(' TFIN=*, F12.5, LPR=*, I2, TTHETA=*, F8.3, PHI=*, F8.3)
8 FORMAT('15, E6.3, E5.2')
9 FORMAT(' NX=1, I2, ' NY=1, I3, ' XC=*, F8.3, 5X, ' YC=*, F8.3, XD=*
2 FORMAT(' F8.3, 1X, ' PRESSUR=*, F8.3, 1X, ' MOLECULAR WEIGHT FLUID=*, F6.2)
10 FORMAT(' O PARTICLE STORAGE EXHAUSTED')
11 FORMAT(' O PARTICLE STORAGE EXHAUSTED')
12 FORMAT(10I5, 2E6.3)
13 FORMAT(13)
14 FORMAT(3X,2(2X,I3),8(2X,E11.4))
17 FORMAT(* CYLINDRICAL CORRDS ALLOWS NC INFLOW*)
18 FORMAT(* NO OBST OR 1/O BOUNDARIES*)
20 FORMAT(* CYLINDRICAL CORRDS ALLOWS NC INFLOW*)
21 FORMAT(* NO OBST OR 1/O BOUNDARIES*)
22 FORMAT(2E10.4, I5, 5E10.4)
23 FORMAT(* ENERGY DESCRIPTION -- PIPE HEAT LOAD=*, F10.4, * PTU/HR*)
1 INLET INTERNAL ENERG Y=*, F10.4, * PTU/LB, FLUID NUMBER *=*, I2, * WILL
1 BE USED*, * THE EXIT INTERNAL ENERGY=*, F12.5, * BTU/LBM*)
24 FORMAT(* TEMP. AT EVAPORATOR BEGINNING=*, E10.3, * TEMP. AT THE EVAP
1 ORATOR EXIT=*, E10.3, * COMPRESSIBLE CALCULATION=*, I2, * YES=1, NO=0,
1 / * TEMP. AT THE CONDENSER INLET=*, F12.4, * TEMP. AT THE CONDENSER
2 FND=*, F12.4, * ALL TEMPS. DEG. F.*!)
25 FORMAT(* THE RADIUS TO THE WALL=*, E10.4, * THE FRACTION OF WICK VD
1 LUMF CONTAINING LIQUID = *, E10.4)
31ILITY=*, E10.4, * IN2*)
NPLCT = 0
PRINT 1
C ------- IS THIS A RESTART RUN.
RV=1E0*DR
NY=1
DROU=0.
UNIVG=0.
TEB=0.
TEE=0.
TCB=0.
TCE=0.
TEMP=212.
TEMP1=212.
CYCLE=0.
HVIP=0.
TEMPK=212.
TEMPI=212.
L1=0
L2=LL2-2
L3=LL3-2
L4=JP2-2
KONVPC=0
KON=C
IF (START) GO TO 404
READ 2, NAME
READ 3, SCH, HCR, RCT, BCL, GM, B, LAMBD, VISCS, EPS, GR, GZ, TAU
VI=VISCS
FLAME=LAMBD
IF (PC.EQ.0.) BCL=1.0
READ 4, T, TWPL, TWPT, TWT, TWF, LPR, THETA, PHI
IF (LPR.EQ.0.) GOTO 28
IF (LPR.EQ.3.) GOTO 27
WRITE (12,1)
WRITE (12,2) NAME
WRITE (12,5) ISAR, JSAR, DR, DZ, DT, IPHM, PC, ALP
WRITE (12,6) RE, RC, RCT, BCL, GM, B, LAMBD, VISCS, EPS, GR, GZ, TAU
WRITE (13) T, TWPL, TWPT, TWT, TWF, LPR, THETA, PHI
IF (LPR.EQ.0.) GOTO 28
WRITE (12,7) T, TWPL, TWPT, TWT, TWF, LPR, THETA, PHI
IF (LPR.EQ.3.) GOTO 27
PPC= DATE = 73111 01/42/10
R(1)= RIP(1)- 0.5*Y
DO 29 I=2, IP?
RIP(I)= RIP(I-1) + Y
RI(I)= R(I-1) + Y
CONTINUE
IP1=16AR+1
JP1= 9AR+1
IPHM=1PHM+1
IP2=2*IPHM
RDR=1./DR
RNP2=RNP*RDR
RDZ=1./DZ
RDZ2=RDZ*RDZ
DPRDZ=DR*RNDZ
DZGDZ=DZ*RDZ
PDRDZ=PDR*RDZ
VISCCITY IS IN LB/IN-SEC
CCF1 = (2.*VISCS*LAMBD)*RDR
CCF2 = (2.*VISCS*LAMBD)*RDZ
W=(1.+ALP)/(2.*(RDR2+RDZ2))
DTNP=DT*RDP
DTGDZ=DT*RDZ
TPLT = TWPLT
DTHPLN = DT
TPRT = TWPR T
T1 = TWT
TWPLT = TPLL + T
TWPR T = TPRT + T
TWT = T1 + T
CYCLE = C
NUMTD = 1
C  ---- INITIALIZE CELL FLAGS
DO 37 J = 2, JP1
   DO 37 I = 2, JP1
      F(I, J) = 4.
      IF(I.EQ.2) F(I-1, J) = 1.
      IF(J.EQ.2) F(I, J-1) = 1.
      IF(I.EQ.JP1) F(I+1, J) = 1.
      IF(J.EQ.JP1) F(I, J+1) = 1.
37 CONTINUE
C  ---- DETERMINE PROBLEM TYPE.
   READ 12, TYPF, L1, L2, L3, L4, L5, L6, L7, NXR, NYR, UL, UR
   LLI = L2 + 1
   LL3M1 = L3 + 1
   LL3 = L3 + 2
   XP = IBP*DR
   YT = JEAR*DZ
   X = .33*SQRT(ARS*GR)*XP
   Y = .33*SQRT(ARS*GZ)*YT
   UUU = UL
   RDU = AMAX1(X, Y, UUU, UR)
   DO 38 J = 1, JP2
      BCLT(J) = 3CL
      ROS(J) = 1.0
      38 BCRT(J) = GCR
   K = 1
   NP = 0
C  ---- TYPE NE = 0 FOR AN OBSTACLE PROBLEM
   IF(TYPF, NE, C) GOTO 100
   IF(LPR.GE.2) PRINT 18
   IF(LPR.EQ.1.CR, LPR.EQ.2) WRITE(12, 18)
C  ---- GENERATE PARTICLES
   40 READ R, NX, NY, XC, YC, XD, YD, UN, V0, ENERGY, PRESS, WTMOL
   IF(NX.GE.1) GOTO 50
   IF(LPR.EQ.1.CR, LPR.EQ.2) WRITE(12, 9) NX, NY, XC, YC, XD, YD, UN, V0
   ENERGY = PRESS + WTMOL
   IF(LPR.GE.2) PRINT 9, NX, NY, XC, YC, XD, YD, UN, V0
   ENERGY = PRESS + WTMOL
C  ---- GAS CONSTANT IS CALCULATED BASED ON FLUID USED FT-LBF/(LBM-F).
UNIVG = 1544.3/W1MOL
XC=XC+RDS
YC=YC+RDZ
XD=XD+RD
YD=YD+ROZ
XTE=1./NX
YTE=1./NY
Y=YTE*.5
DRCU=AMAX1(DFGU,AR5(UC),AR5(VO))
50 X=XTEN*.5
55 IF(YD.LT.1.e-9)GOTO 72
IF((X.GE.XC) .AND. (X.LE.XD) .AND. (Y.GE.YC) .AND. (Y.LE.YD)) GO TO 60
GO TO 70
60 I=X+2.
J=Y+2.
IF(F(I,J).LT.OE)GOTO 76
XP(K)=X
XP(K+1)=Y
K=K+2
IF(K.GT.LPR)GOTO 75
NP=NP+1
P(I,J)=FUL
IF AND F IN BTU/LB
C
INITIALIZE P,RHO,INTRNL,E,AND PRAR IN THE MESH
INTRNL(I,J)=ENERGY
RHO(I,J)=PRESS/(GM*INTRNL(I,J)*778.)
SIGMA(I,J)=RHO(I,J)
RHO(IP,I,J)=RHO(IP-1,J)
P(I,J)=PRESS
C(I,J)=0.0
F(I,J)=INTRNL(I,J)+UL**2+V0**2/2.
2 /DEF
PBAR(I,J)=P(I,J)+RHO(I,J)*1.e+05
1.*.G001
IF(F(I+1,J).EQ.BND .AND. J.GE.LL2) U(I,J)=U(I,J)*ROS(J)+UO*(1.-ROS(J))
IF(F(I+1,J).EQ.BND .AND. J.LT.LL2) U(I,J)=UL
IF(F(I+1,J).NE.BND .AND. F(I+1,J).NE.OB) U(I,J)=U0
IF(F(I-1,J).EQ.BND .AND. F(I-1,J).NE.OB) U(I-1,J)=UO
IF(F(I,J-1).EQ.BND .AND. F(I,J-1).NE.OB) V(I,J-1)=V0
IF(F(I,J+1).EQ.BND .AND. F(I,J+1).NE.OB) V(I,J)=V0
70 X=X+XTE
IF(X.LT.IPAB)GOTO 55
Y=Y+YTE
IF(Y.LT.JBAR)GOTO55
GOTO40
72 \( Y_1 = (Y - Y_C)^2 \)
\( X_1 = (X - X_C)^2 \)
IF \((Y_1 + X_1) \leq (XD^2)\) GOTO 60
GOTO 70

75 PRINT 11
WRITE(12,11)
GOTO 9999

80 IF(DROU.NE.0.) DROU=1.2*DR/DR0U
READ 22,Q1N,FINTRL,JFLUID,XITINT
PRINT 21,Q1N,FINTRL,JFLUID,XITINT
READ 22,TEB,TEE,KONER,TCR,TCF
PRINT 23,TFB,TEE,KONER,TCB,TCF

TEMP=(TEB+TEE)/2.
READ 22, RW,EP,INUM,WKMR,PERMFA.
PRINT 24, RW,EP,INUM,WKMR,PERMFA.

73 CONTINUE

JFLUID MATERIAL
1 LITHIUM 7 NAK-78
2 SODIUM 8 STRONTIUM
3 POTASSIUM 9 SILVER
4 RUBIDIUM 10 FLUORIDE
5 CESIUM 11 LITHIUM FLUORIDE
6 MERCURY 12 WATER

CONTINUE

DO 82 J=2,JPI
INTRNL(1,J)=INTRNL(2,J)
INTRNL(IP2,J)=XITINT
1 -P(IP1,J)/(PHO(IP1,J)*778.*12.)
IF(J,LT,LL2) INTRNL(IP2,J)=FINTRL
1 -P(IP1,J)/(PHO(IP1,J)*778.*12.)
F(IP2,J)=INTRNL(IP2,J)+0.5*U(IP2,J)**2/DEF
F(1,J)=F(2,J)
82 CONTINUE

DO 83 I=2,JPI
INTRNL(I,JP2)=INTRNL(I,JP1)
INTRNL(I,1)=INTRNL(1,2)
F(I,1)=F(1,2)
F(I,JP2)=F(I,JP1)
83 CONTINUE

IF(LPR.GE.2) PRINT 10, NP
IF(LPR.EQ.1 OR LPR.EQ.2) WRITE (12,10) NP
ASSIGN 25O TCKRET
GOTO 645
C ----- SETUP FOR OBSTACLE WITH INFLOW/OUTFLOW

100 CONTINUE
IF(LPR.GE.2) PRINT 19, TYPE, L1, L2, L3, L4, L5, L6, L7, NX5, NYB, UL, UR
IF(LPR.EQ.1 .OR. LPR.EQ.2)
1 WRITE (12,19) TYPE, L1, L2, L3, L4, L5, L6, L7, NX5, NYB, UL, UR
IL=L5+2
I0=L6+1
JTE=L7+1
DO 104 J=2, JTE
DO 104 I=1, IP
104 F(I,J)=0.0
Y=L1*N
XDIE=0.5/AMAX0(NX U, 1)
103 YFIN= L4*N
YDIS=1.0/AMAX0(NYB, 1)
YFIR = (1.5*YDIS)+L1
MIN=NYB*(L2-L1)
COLS=0.0
DC 110 J=1, IP2
V(I,J)=V(I,J)*PCL
IF(J.GE.(L1+2).AND.J.LE.(L2+1)) GOTO 107
106 IF(J.GE.(L3+2).AND.J.LE.(L4+1)) GOTO 109
GOTO 110
107 U(IP2,J)=UL
U(IP1,J)= UL*(R(IP2)+DH)/(R(IP1)+DH)
BCT(J)=1.
GOTO 106
108 U(IP1,J)=UR*
U(IP2,J)= UR*(R(IP1)+DH)/(R(IP2)+DH)
BCT(J)=1.
IF(UJ).NE.0.) GOTO 110
RNS(J)=0.
BCT(J)=1.
110 CONTINUE
GOTO 40
120 PRINT 17
WRITE (12,17)
GOTO 9999

C ----- REFLAGGING

250 DO 255 J=1, JP2
DO 255 I=1, IP2
255 IF(I,J)=0
IF(ITER.EQ.1) EPS=EPS/1.1
C UPDATE DENSITY, P, PBAR, AND PERPENDICULAR BOUNDARY VELOCITIES
TEMPKI=(TEMPI-32.)*5./9.+273.
TEMPKT=(TEMPT-32.)*5./9.+273.
CALL HVAP(TEMPKI, HVIP, JFLUID)
HVIP = HVIP * 0.4299
HILAT = 1600.
IF (ABS(HVIP) * HILAT) HVIP = 0.45.
IF (ABS(HVIP) < ELOW) HVIP = 0.45.
CALL HVAP(TEMPKT, HVFP, JFLUID)
HVFP = HVFP * 0.4299
IF (ABS(HVFP) * HILAT) HVFP = 0.45.
IF (CYCLE.AF. AND. NY.EQ.0) PRINT 251, JFLUID, TEMP1, HVIP, TEMPT, HVFP
251 FORMAT (2, FLUID NO., *, 12, *, INLET TEMPERATURE = *, E12.4, *, DEC F --
1 INLET LATENT HEAT = *, F12.4, *, PTU/LBM, */ XT LATENT TEMPERATURE = *, E12.4, *
2 * DEC F -- EXIT LATENT HEAT = *, E12.4, *, PTU/LBM)
252 CONTINUE
DO 111 J = 1, JP2
1 IF (J.GE.LL3. AND. UR.EQ.0) INTRNL(IP2, J) = INTRNL(IP1, J)
1 IF (J.GE.LL3. AND. UR.EQ.0) P(IP2, J) = P(IP1, J)
C XXXXX TRANSIENT SECTION - MODIFIED BY USER XXXXX
C XXXX THE BELOW FUNCTIONS CAN BE XXXX
C XXXX INPUT TO DESCRIBE ANY PROBLEM XXXXX
INTRNL(IP2, J) = INTRNL(IP1, J)
IF (RHO(IP1, J).EQ.0) GO TO 259
C XXXX INLET (DEVERALL)
IF (JFLUID.EQ.0. AND. J.LT.LL2. AND. RHO(IP2, J).GT.0.)
1 INTRNL(IP2, J) = 0.1 INTPL-P(IP2, J)
1 /(RHO(IP2, J)*770.*12.)/(LL2-1-J)/(LL2-1)*1.3
1 IF (RHO(IP2, J).EQ.0. AND. J.LT.LL2. AND. RHO(IP2, J).GT.0.)
1 INTRNL(IP2, J) = 0.1 INTPL-P(IP2, J)/(RHO(IP2, J)*770.*12.)
C DZAKOWIC 1440 INLET
IF (JFLUID.EQ.2. AND. J.LT.LL2. AND. RHO(IP2, J).GT.0.)
1 INTRNL(IP2, J) = 0.1 (LL2-1-J)/(LL2-1)*5.
1 IF (RHO(IP1, J).EQ.0) INTRNL(IP2, J) = 0.1 (LL2-1-J)/(LL2-1)*5.
C INLET (KESSLER)
C IF (JFLUID.EQ.12. AND. J.LT.LL2) INTRNL(IP2, J) = PUT IN
C INLET (EANKSTON) STAYS THE SAME
C OUTLET (DEVERALL)
C IF (JFLUID.EQ.6. AND. J.GE.LL3) INTRNL(IP2, J) = 0.1 (JP2-J)/
1 (JP2-LL3)
C OUTLET CONDITIONS HAVE BEEN CALCULATED IN THE ENERGY SECTION FO
C UP=0 -- IF UR .NE. 0 THE OUTLET ENERGY IS SPECIFIED HERF
C XXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXX
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RHC(1, J) = PHC(2, J)  
PBAR(1, J) = PBAR(2, J)  

V(1, J) = V(2, J)  

DO 112 I = 1, IP2  
P(I, JP2) = P(I, JP1)  
P(I, 1) = P(I, 2)  
V(I, 1) = 0.  
V(I, JP1) = 0.  
V(I, JP2) = 0.  

RHC(I, 1) = RHC(I, 2)  
RHC(I, JP2) = RHC(I, JP1)  
PBAR(I, 1) = PBAR(I, 2)  
PBAR(I, JP2) = PBAR(I, JP1)  

CONTINUE

NPT = 0  
K = 1  
I = XP(K) + 2.  
J = XP(K + 1) + 2.  
KF(I, J) = K  
K = K + 2  

IF (NPT LT NPT) GO TO 260  

1 PRINT 261, NPT, T, CYCLE  

261 FORMAT (3, THE NUMBER OF PARTICLES IN THE MESH = 1, 15, AT TIME = 1, 1E12, 4, CYCLE = 1, 15)  

DO 265 J = 2, JP1  
DO 265 I = 2, IP1  
IF (F(I, J) NE SUR) GO TO 265  
IF (KF(I, J) NE 0) GO TO 265  
F(I, J) = EMP  
IF (F(I + 1, J) EQ EMP) U(I, J) = 0.  
IF (F(I - 1, J) EQ EMP) U(I - 1, J) = 0.  
IF (F(I, J + 1) EQ EMP) V(I, J) = 0.  
IF (F(I, J - 1) EQ EMP) V(I, J - 1) = 0.  

CONTINUE

265 CONTINUE

DO 267 J = 2, JP1  
DO 267 I = 2, IP1  
IF (KF(I, J) EQ 0) GO TO 267  
IF (F(I, J) EQ FUL) GO TO 267  
F(I + 1, J) EQ EMP OR F(I - 1, J) EQ EMP OR F(I, J + 1) EQ EMP OR F(I, J - 1) EQ EMP  
F(I, J) = SUR  
GO TO 270

GO TO 270

CONTINUE
PHASE 1, STEP 2 - JUST OUTSIDE TANGENTIAL VELOCITIES.

IF (CYCLE .EQ. 0) GO TO 220

GOTO 65C

--- PHASE 1, STEP 3 - ACROSS THE WALL TANGENTIAL VELOCITIES.

IF (CYCLE .EQ. 0) GO TO 640

DO 350 I=2, IHR

U(I,1)=U(I,2)*FCE

U(I,JP2)=U(I,JP1)*BCT

IFLOW=1

DO 360 J=2, JBAK

V(I,J)=V(2,J)*CLT(J)

V(JP2,J)=V(JP1,J)*BCRT(J)

IF (CYCLE .EQ. 0) GO TO 500

GOTO 65C

CONTINUE

--- CONTROL SECTION.

IF (CYCLE .EQ. 0) GO TO 415

IF (TWPLT .LE. T+.001*DT) GO TO 405

IF (TWPR .LE. T+.001*DT) GO TO 407

IF (TWPRT .LE. T+.001*DT) GO TO 409

IF (TWFIN .LE. T+.001*DT) GO TO 413

CYCLE=CYCLE+1

IF (ITER .LE. 2 .AND. CYCLE .GT. 5 .AND. JFLUID.EQ.12) DT=DT*1.05

KEEP DT PFLOW LIMIT

IF (TAU .LT. 0.)

1

CRITER=DR**2*0.25/ABS(TAU)

IF (DT .GE. CRITER) DT=CRITER-0.1*C CRITER

T=T+DT

GOTO 500

TWPLT=TWPLT+TPLT

ASSIGN 401 TO KRET

GOTO 420

TWPR=TWPR+TPR

ASSIGN 402 TO KRET

GOTO 420

TW=TW+TT

ASSIGN 403 TO KRET

GOTO 456

T=T2

ASSIGN 412 TO KRET

GOTO 450

RFTUPN
415 ASSIGN 416 TO KRET
GOTO 420
416 ASSIGN 404 TO KRET
GOTO 430

C ---- PARTICLE AND VELOCITY VECTOR PLOTS.

420 CONTINUE
CALL FRAME(NPLOT,TYPE,JBAR,IBAR,DZ,DR,NAME,T,CYCLE)
NPP = 2*NPP
CALL PLOT(XP,NPP)
CALL FRAME(NPLOT,TYPE,JBAR,IBAR,DZ,DR,NAME,T,CYCLE)

PROG         DATE = 73111             01/42/10

CALL VPLOT(F,U,V,IP2,JP2,DPOR,EMP)
GO TO KRET,(250,416,404,413,403,402,401,640,400)

C ---- EDITS ON PAPER OR MICROFILM OR BOTH.

430 IF(LPR.EQ.0) GO TO KRET,(250,416,404,413,403,402,401,640,400)
INDIC=0
MDDF=1
DO 431 NCNT=1,2
   LINMAX = 0
   LINEP = 0
   LINEF = 0
   IF(LPR.GE.2)
      1 CALL HEAD(PAPER,NAME,T,CYCLE,LINEP,INDIC)
   IF(LPR.LE.2)
      1 CALL HEAD(MODE,NAME,T,CYCLE,LINEF,INDIC)
   1 CALL HEAD(PFILM,NAME,T,CYCLE,LINEF,INDIC)
   DO 440 JJ=1,JP2
       J = JP2 + 1 - JJ
       IF(LINEP.EQ.LINMAX)
          1 CALL HEAD(MODE,NAME,T,CYCLE,LINEP,INDIC)
       IF(LINFF.EQ.LINMAX)
          1 CALL HEAD(MODE,NAME,T,CYCLE,LINEFF,INDIC)
       IF(LPR.GT.2) GO TO 435
   IF(NCNT.EQ.1)
      1 WRITF(12,14) I,J,F(I,J),U(I,J),V(I,J),P(I,J),G(I,J),
      1 RHO(I,J),PBAR(I,J),C(I,J)
   IF(NCNT.EQ.2)
      1 WRITF(12,14) I,J,F(I,J),RISE(I,J),S(I,J),E(I,J),SIGMA(I,J),
      1 INTRNL(I,J),Q(I,J),XP(I)
   LINEF = LINEF + 1
435 IF(LPR.LT.2) GO TO 440
   IMJNUM = 1 + IBAR + J
   IF(NCNT.EQ.1)
      1 WRITF(6,14) I,J,F(I,J),U(I,J),V(I,J),P(I,J),G(I,J),
      1 RHO(I,J),PBAR(I,J),C(I,J)
   IF(RHOAV.EQ.0.) RHOAV = 1.
IF(AVEU.EQ.0) AVEU=1.0E+25
IF(CYCLE.EQ.2) GO TO 436
UDEMA=(P(I,2)-P(I,J))*((R(IP1)+DH)/(DZ*FLOAT(L4-L1)))**2
1/(RHOAV *AVEU**2) *DD

436 CONTINUE
WRITE(6,14) I,J,F(I,J),RICE(I,J),S(I,J),F(I,J),SIGMA(I,J),
1NTRNL(I,J),D(I,J),UDEMA
LINEP = LINFP+1

440 CONTINUE

431 CONTINUE
GO TO KRET,(250,416,404,413,403,402,401,640,400)

C ---- TAPF DUMP.
450 WRITE(12,15) NUMTD,T,CYCLE
NUMTD = NUMTD+1
GO TO KRET,(250,416,404,413,403,402,401,640,400)

C ---- CALCULATION OF RICE, S, AND CONVECTIVE FLUX, AND Q
500 CONTINUE
IF(KONFTER.NF.0) GO TO 502
DO 501 J=2,JP1
DO 501 I=2,IP1

502 CONTINUE
IF(KON.FEQ.1) GO TO 760
C ---- Q HAS BEEN VERIFIED
503 CONTINUE
C ---- Q IS IN LBM/IN**2-SEC**2
DO 521 J=2,JP1
DO 521 I=2,IP1
Q(I,J) = -C0F1/R(I) *((R(I)+DH)*U(I,J)-(R(I)-DH)*U(I-1,J)
1-C0F2*(V(I,J)-V(I,J-1))

509 IF(RHO(I,J).EQ.0.) GO TO 521
521 CONTINUE
DO 522 J=2,LL1
DO 522 I=2,IP1
Q(IP2,J) = Q(IP1,J)

522 CONTINUE
DO 523 J=LL3,JP1
DO 523 I=2,IP1
Q(IP2,J) = Q(IP1,J)

523 CONTINUE
IFLOW=2
C ---- RICE AND S ARE IN LBM/IN**2-SEC**2
DO 520 J=2,JP1
DO 520 I=2,IP1
IF(F(I,J).GE.EMP) GOTO 520
IF(F(I+1,J).GE.EMP) GOTO 510
C ---- PICE HAS BEEN VERIFIED
515 RHOAVP= (((RHO(I,J)+RHO(I,J+1))/2.+(RHO(I+1,J)+RHO(I+1,J+1))/2.)/2
C ---- CALCULATE PBAR

530 CONTINUE

IFLOW=4

DO 540 J=2,JP1
DO 540 I=2,IP1
IF(F(I,J).NE.FUL)GOTO 525
GOTO 540
535 INTRL(I,J)=0.

540 CONTINUE

ITER=0

IND=1
DO 602 J=1,JP2
DO 602 I=1,IP2
602 SIGMA(I,J)=PHO(I,J)
GOTO 550
550 CONTINUE

IF(PBCLFM GOES UNSTABLE TRY TO RECYCLE STARTING AGAIN

IF(ITER.EQ.100 ) IDUM=IDUM+1
IF(ITER.EQ.100 .AND.IDUM.LE.5) DT=DT/1.2

604 CONTINUE

IF(IND.EQ.100 ) IND=0

IF(IND.EQ.0)GOTO600

IND=C
ITER=ITER+1

DO 570 J=2,JP1
DO 570 I=2,IP1
IF(F(I,J).NE.FUL)GOTO 570
IF(PH0(I+1,J).EQ.0.) GO TO 561
SIGR =PBAR(I+1,J)/RHO(I+1,J)
561 IF(RHO(I,J-1).EQ.0.) GO TO 562
SIGB = PBAR(I,J-1)/RHO(I,J-1)
562 IF(PH0(I-1,J).EQ.0.) GO TO 563
SIGL = PBAR(I-1,J)/RHO(I-1,J)
563 IF(RHO(I,J+1).EQ.0.) GO TO 566
SIGT = PBAR(I,J+1)/RHO(I,J+1)
566 IF(RHO(I,J).EQ.0.) GO TO 567
SIGIJ = PBAR(I,J)/PH0(I,J)
567 CONTINUE

IF(TYPE.EQ.0)G0 T0 565
IF(F(I+1,J).EQ.0) SIGR=PBAR(I,J)/RHO(I,J)
IF(F(I,J-1).EQ.0) SIGB=PBAR(I,J)/RHO(I,J)

PROG DATE = 73111 01/42/10

IF(F(I-1,J).EQ.0) SIGL=PBAR(I,J)/RHO(I,J)
IF(F(I,J+1).EQ.0) SIGT=PBAR(I,J)/RHO(I,J)

C X HAS BEEN VERIFIED

IF(C(I,J).EQ.0 .OR.RHO(I,J).EQ.0.) GO TO 570

565 X=(G1(I,J)/RHO(I,J)+ THEMA*PHI*DT*DT* (((R(I)-DH)*
1 SIGL + (R(I)+DH)*SIGR) / (R(I)+DR*D)
2 +(SIGB + SIGT) / (DZ*DZ))
2 *(1.+ALP)
3 \[ \frac{1}{2} \cdot C_{ij} + 2 \cdot \Theta \cdot \Phi \cdot D \cdot T \cdot (R_{D2} + R_{D2}) - \Delta \cdot \Sigma_{ij} \]

\[ Y = \text{ABS}(X) - \text{ABS}(\Sigma_{ij}) \]

\[ Z = \text{ABS}(X) + \text{ABS}(\Sigma_{ij}) \]

\[ \Sigma_{ij} = X \]

\[ \text{PRAR} (i, j) = \Sigma_{ij} \cdot \text{RHO} (i, j) \]

\[ \text{RHO} (i, j) = \frac{R_{I} (i, j)}{C_{I} (i, j) \cdot 386.4 + \Sigma_{I} (i, j)} \]

615 CONTINUE

620 CONTINUE

C ---- CALCULATE TIME ADVANCED VELOCITIES

600 IF (LPR.EQ.1 .OR. LPR.EQ.2) WRITE (12,16) T, CYCLE, ITER

IF (LPR.EQ.2) PRINT 16, T, CYCLE, ITER

IF (LPR.EQ.1 .OR. LPR.EQ.2) WRITE (12,16) T, CYCLE, ITER

IF (CYCLE.EQ.2) PRINT 16, T, CYCLE, ITER

IF (LPR.EQ.2) TFMP = TFMP

IF (LPR.EQ.1 .OR. LPR.EQ.2) TERNP = TERNP

IF (CYCLE.EQ.2) PRINT 16, T, CYCLE, ITER

IF (LPR.EQ.2) TFMP = TFMP

IF (LPR.EQ.1 .OR. LPR.EQ.2) TERNP = TERNP

IF (LPR.EQ.2) PRINT 16, T, CYCLE, ITER

IF (JFLUID.NE.6 .AND. JFLUID.NE.12) TFMP = 5.9/(TEMP - 32) + 273.

IF (JFLUID.NE.6 .AND. JFLUID.NE.12) TFMP = 5.9/(TEMP - 32) + 273.

IF (NKT.EQ.0) CALL PSAT(TEMP, X, JFLUID)

IF (NKT.EQ.0) CALL PSAT(TEMP, X, JFLUID)

IF (JFLUID.NE.6 .AND. JFLUID.NE.12) TFMP = 5.9/(TEMP - 32) + 273.

IF (JFLUID.NE.6 .AND. JFLUID.NE.12) TFMP = 5.9/(TEMP - 32) + 273.

IF (X.EQ.0) GO TO 612

IF (X.EQ.0) GO TO 612

IF (NKT.EQ.0) GO TO 606

GO TO 55B

GO TO 55B
IF(ABS(DIFR).GT.0.9) AND ABS(DIFR).LT.1.02) PRINT 607, P(IP2, LL2-2), X, TEMPI
1, E12.4, * TEMPERATURE = E12.4, * DF6PEFS
GO TO 608
607 FORMAT ( 'P INTERFACE USING INTNL = ', ITL2.4, ' P INTERFACE TRANP = ', E12.4, ' P INTERFACE TEMP = ', E12.4, ' DF6PEFS = ')
GO TO 608
608 IF(X.EQ.0.0) GO TO 609
DIFR = P(IP2, LL3+2)/X
IF(AES(DIFR).GT.0.9) AND ABS(DIFR).LT.1.02) PRINT 609
1, P(IP2, LL3+2), X, TEMPI
609 FORMAT ( 'EXIT INTERFACE PRESSURE =', E12.4, ' TRAPN PRESSURE =', E12.4, ' TEMPERATURE OF GP =', E12.4)
607 CONTINUE
607 CONTINUE
KNT = KNT + 1
IF(KNT.GT.50) PRINT 617
IF(KNT.GT.50) GO TO 632
617 ' WARNING TEMPERATURE ITERATION HAS HIT THE LIMIT'!
IF(AES(DIFR).GT.0.9) AND ABS(DIFR).LT.1.02) GO TO 632
631 IF(AES(DIFR).EQ.1.0) FNCREN = 5.
1 - 2, E725
IF(AES(DIFR).LT.1.02) FNCREN = 1
1 + 0.5725/(KNT+1)
IF(KNT.NE.0) TEMPI = TEMPI + FNCREN
IF(KNT.NE.0) TEMPI = TEMPI + FNCREN
GO TO 606
632 IF(AES(DIFR).LT.1.02) FNCPEN = 5.
1 - 3, E725 - 0.14725*KTR
IF(AES(DIFR).EQ.1.0) FNCREN = 1
1 - 1, E725 * KTR
IF(KNT.NE.0) TEMPI = TEMPI - FNCPEN
IF(KNT.NE.0) TEMPI = TEMPI - FNCPEN
GO TO 606
637 CONTINUE
637 CONTINUE
IF(KNT.NE.0) GO TO 636
KNT = 1
GO TO 606
636 CONTINUE
636 CONTINUE
IF(JPAR.LE.0.4) GO TO 649
IF(CYCLE.GT.2) CALL SCRFNM(P, IP2, JP2)
649 CONTINUE
IF(CYCLE.GT.2 AND JFLUID.EQ.12) TEMPI = TEMPI
DC 620 J = 2, JP1
DC 620 I = 2, JP1
SP = (SIGMA(1,J)+SIGMA(1+1,J))/2.
SP = (SIGMA(1,J)+SIGMA(1+1,J))/2.
RHCP = 2./(PHCI(1,J)+RHCP(1+1,J))
RHCP = 2./(RHCP(1,J)+RHCP(1+1,J))
```
IF(F(I,J).GE.EMP)GOTO 620
IF(F(I+1,J).GE.EMP)GOTO 610
C U AND V HAVE BEEN VERIFIED
IF(F(I+1,J).EQ.FQ.BND.AND.J.GE.(L2+2).AND.J.LT.(L3+1)) GOTO 610
L = 1.0 - PHI
1 + (1.*DTRG*(F(I,J) - P(I+1,J)) + RHOCIP.DT*RIQPCE(I,J) + U(I,J)
2*SIP*RHOCIP
C FIX UP FOR IMPOSSIBLE U VELOCITY VARIATIONS AS A RESULT OF INSTABILITY

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<th>DATE = 73111</th>
<th>01/42/10</th>
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<td>610 IF(J.EQ.IP1.OR.F(I,J+1).EQ.EMP) GOTO 620</td>
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<td>V(I,J) = DT<em>DO</em>RHOCIP*(PHI<em>DTRG</em>(PBAR(I,J) - PBAR(I,J+1))</td>
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|       | 1 + (1.0 - PHI)*DTRG*(I,P(I,J) - P(I,J+1)) + RHOCIP*DT* S(I,J) + V(I,J)
| 620 CONTINUE
| IFFLOW = 7 |
| ASSIGN 640 TO KRF |
| GO TO 330 |
| 640 IF(INP.EQ.1) GOTO 626 |
| DO 621 J = 1, IP2 |
| U(IP1,J) = U(IP1,J)*RHO(J) + U(IP1-1,J)*(1.0 - RHO(J))*(R(IP1-1) + |
|        + DH)/R(IP1)+DH)
| IF(KONTER.EQ.1.AND.J.GE.LL3) GOTO 621 |
| U(IP1,J) = U(IP2,J)*(R(IP2)+DH)/(R(IP1)+DH)
| 621 CONTINUE
| 626 CONTINUE |
| NAV = 0 |
| AVEU = 0 |
| RHOTOT = 0 |
| AVMASS = 0 |
| CORL = (L2-L1)*DZ |
| DO 622 J = 2, IP1 |
| C U(IP2,J) SHOULD BE CALCULATED HERE BY Q/(HFG*RHO) |
| U(IP2,J) = U(IP1,J)*(R(IP1)+DH)/(R(IP2)+DH)
| IF(J.LT.LL2) NAV = NAV + 1 |
| IF(RHO(IP1,J).EQ.0.0.OR.HVIP.EQ.0.0. OR.HVEP.EQ.0.0) GO TO 622 |
| IF(RHO(IP2,J).EQ.0.0) GO TO 622 |
| IF(J.LT.LL2) U(IP2,J) = QIN/(3600.*HVIP*RHO(IP1,J)*6.28*RHOCORL) |
| 1*RHO(IP1,J)/RHO(IP2,J) |
| IF(J.GE.LL3) U(IP2,J) = QIN/(3600.*HVEP*RHO(IP1,J)*6.28*RHOCORL) |
| 1*(L4-L3)*DZ |
| 1*RHO(IP1,J)/RHO(IP2,J) |
| IF(KONTER.EQ.0.0.AND.J.LT.LL2) U(IP2,J) = -CNDFLW/(RHO(IP2,J) |
| 1*6.28* (R(IP1)+DH)*CORL)
| IF(KONTER.EQ.0.0.AND.J.GE.LL2) U(IP2,J) = CNDFLW/(RHO(IP2,J) |
| 1*6.28* (R(IP1)+DH)*(L4-L3)*DZ) |
```
IF(KONT > GT.1 OR KONT < EQ.0) GO TO 641
IF (KONER < LT.L1 AND J < LT.LJ) GO TO 641
CALCULATE UL(J,J) OUT BY ASSUMING IT = TOTAL CONDENSER IN FLOW
(U(J,J) = U(J,J) + ROTA
t = (R(J+1,J)*RTA) OVER DENSITY OUT)
TOTAL CONDENSER IN FLOW IS CALCULATED ABOVE 705
IF(RHO(J,J) < EQ.0.) GO TO 641
U(J+1,J) = ABS(CNF/(RHO(J,J)+RTA)*(L4-L3)*DZ))
U(J,J) = U(J,J) + RHO(J,J)/U(J,J)
TOTAL CONDENSER IN FLOW IS CALCULATED ABOVE 705
IF(P < EQ.P1 OR J < EQ.J) GO TO 641
U(J,J) = AGS(CNF/RHO(J,J)) * 6.28 * (P(J,J) + DH) * (L4-L3) * DJ
U(J,J) = U(J,J) + RHO(J,J)*U(J,J)
GO TO 622
AVEU = AVEU/NAV
RHO = RHO/NAV
PROG DATE = 73111 01/42/10
PV = AVEU*FLAT(J*1-1)*DR*RHO_AV/VISCOS
IF (CYCLE < EQ.0) GO TO 370
ASSIGN 70C TO KRET
IF (ITYPE < EQ.0) GO TO KRET,(250,416,404,413,403,402,401,640,400)
DO 647 J = 2, J P1
DO 647 I = 2, I P1
IF (F(I,J) < EQ.0) GO TO 647
IF (F(I,J+1) < EQ.0 AND F(I,J+1) < EQ.0) X(I,J) = X(I,J+1) * RCB
IF (F(I,J+1) < EQ.0 AND F(I,J+1) < EQ.0) V(I,J) = V(I,J+1) * RCB
IF (F(I+1,J) < EQ.0 AND F(I+1,J) < EQ.0) V(I+1,J) = V(I+1,J) * RCB
647 CONTINUE
IF (FLOW = 16) GO TO KRET,(250,416,404,413,403,402,401,640,400)
655 CONTINUE
IF (CYCLE < EQ.1) GO TO 661
FLOW = 13
670 CONTINUE
STOTH = ETOTAL
IF (CYCLE < EQ.1) GO TO 661
671 DO 680 J = 2, J P1
DO 680 I = 2, I P1
680 CONTINUE
VV = (V(I,J)+V(I+1,J-1))/2.
VVPA = ((V(I+1,J)+V(I+1,J-1))/2)**2.*
UU = (U(I,J)+U(I-1,J))/2.
V1PH = (V(I,J)+V(I+1,J)+V(I,J-1)+V(I+1,J-1))/4.
IF (I.EQ.0) GO TO 678
VECTOR = (2.*((U(I,J)+U(I-1,J)))/2)**2. + ((V(I,J)+V(I,J-1))/2)**2.*
1 2.*((U(I-1,J)+U(I-1,J-1))/2)**2. -((V(I-1,J)+V(I-1,J-1))/2)**2.*
2 **2.*
GO TO 679
2 **2.*
GO TO 679
679 CONTINUE
VECT = ((V(I,J)+V(I+1,J-1)+V(I-1,J)+V(I,J-1))/4.
VECT1 = ((U(I,J)+U(I+1,J+1))/2)**2. +2.*((V(I,J)+V(I,J+1))/2)**2.*
1**2. -UU**2.*
U1JPH = (U(I,J)+U(I-1,J)+U(I-1,J+1))/4.
U1JMH = (U(I,J)+U(I-1,J)+U(I-1,J-1)+U(I-1,J-1))/4.
FAVP = (E(I+1,J)+E(I+1,J))/2.
1*DEF
IF(J.EQ.2) GO TO 676
SPECV=((V(I,J)+V(I,J-1))/2)**2.*
GO TO 677
676 SPECV=((V(I,J)+V(I,J-1))/2)**2.*
677 CONTINUE
RHOAVM = (RHO(I,J)+RHO(I-1,J))/2.
RHOAVP = (RHO(I,J)+RHO(I+1,J))/2.
UAVP = (U(I,J)+U(I+1,J))/2.
PROG = DATE = 73111 01/42/10
UAVM = (U(I,J)+U(I-1,J))/2.
VAVP = (V(I,J)+V(I+1,J))/2.
VAVM = (V(I,J)+V(I,J-1))/2.
EAVM = (E(I,J)+E(I-1,J))/2.
1 *DEF
F(I,J+1)=E(I,J+1)*DEF
E(I,J-1)=E(I,J-1)*DEF
F(I,J)=E(I,J)*DEF
IF(RHO(I,J).EQ.0 或 R(I,0).EQ.0.) GO TO 660
C_EF IS VERIFIED
VISCOS=VI
LAMBDA=FLAMB
EF =RDZ* (( B*VISCOS*RDZ*(INTRNL(I,J+1)-INTRNL(I,J)))
1 *DEF =PPAR(I,J)+PPAR(I+1,J+1)/2.*V(I,J)*DD-LAMBDA/ {LAMBDA+2.}
1*VISCOS)
2 *V(I,J)*Q(I,J)+Q(I+1,J+1)/2. + VISCOS*RDZ/2.*{VECT1}
3 +VISCOS*RDZ*U1JPH*{(V(I,J)+V(I+1,J))/2. -(V(I,J)+V(I-1,J))/2.)}
4 -(B*VISCOS*RDZ*DEF*{INTRNL(I,J)-INTRNL(I,J-1)}-{PPAR(I,J)}
4 + PPAR(I+1,J-1)/2.*DD
5 *V(I,J+1)-LAMBDA/ {LAMBDA+2.} *VISCOS) *V(I,J-1)*Q(I,J)+Q(I,J-1)
(61.2+VISCO5+RDR*(U(1,J-1)+U(I-1,J-1))/2.*-(V(I,J-1)+V(I-1,J-1))/2.)

C E(I,J) IS VERIFIED
662 E(I,J)=1./RHO(I,J)*(SIGMA(I,J)*E(I,J)+DT* (DR/R(I))*(RDAVM*U(I-1,J)+QEAVM*(R(I)-DH)-RHOAVP*(U(I,J)+EAVP*(R(I)+DH))+RDZ*
2*(RHO(I,J)+RHO(I-1,J))/2.*V(I,J-1)*(E(I,J)+E(I-1,J))/2.*-(RHO(I,J)+
3**RHC(I,J+1))/2.*V(I,J)*E(I,J-1)+E(I,J+1))/2.)*RHO(I,J)*V1SC0S*RDZ
4/R(I)*(R(I)-DH)*(RDRB*VISCO5*(INTNL(I+1,J)-INTNL(I,J)))
5*DEF - DN*
6*(PBAR(I,J)+PBAR(I+1,J))/2.*U(I,J)-LAMBDA/(LAMBDA+2.*VISCO5)*U(I,J)
6+O(I,J)+Q(I+1,J))/2.*VISCO5/(2.*DR)*(2.*(U(I+1,J)+U(I,J))/2.*
7+2.*VVP - 2.*UU**2.- VV**2.)*VISCO5*RDZ
8*V1P5*O(U(I,J)+U(I,J+1))/2.-O(U(I,J)+U(I,J-1))/2.)
9 - (R(I)-DH)* (B*VISCO5*RD)*(INTNL(I,J)-INTNL(I-1,J))
5*DEF
1 - (PRAR(I,J)+PRAR(I+1,J))/2.*U(I,J)-LAMBDA/(LAMBDA+2.*VISCO5)*U(I,J)
6+O(I,J)+Q(I+1,J))/2.*VISCO5/(2.*DR)*V1ECT*(U(I+1,J)+U(I+1,J))/2.-
7(U(I,J)+U(I,J-1))
1/2.) ) } + EE)

E(I,J) = E(I,J)/DEF
E(I,J+1)=E(I,J+1)/DEF
E(I,J-1)=E(I,J-1)/DEF

IF(J.EQ.2) E(I,J)= E(I,J)
1 IF(J.EQ.2) E(I,J)=E(I,J)
2 IF(I.EQ.2) E(I,J)=E(I,J)
3 IF(I.EQ.IP1.AND.J.GE.EL2.AND.J.LT.EL3) E(IP2,J)=E(IP1,J)

DO NOT CALCULATE PIN,OUT OR INTRNL IN,OUT

IF(J.EQ.IPI .AND .J.GE.EL2 .AND .J.LT.EL3) INTRNL(I,J)=G*M*778.*336.4
IF(KONTER.GE.TX) GO TO 663
IF(INTRNL(I,J))=GM*778.*336.4

C IF(1.EQ.1) P(I,J)=P(I,J)
1 IF(J.EQ.2) INTRNL(I,J)=INTNL(I,J)
2 IF(I.EQ.2) INTRNL(I,J)=INTNL(I,J)
1 IF(J.EQ.IPJ1) INTRNL(I,J1)=INTNL(I,J1)
1 IF(J.EQ.IPJ2) INTRNL(I,J2)=INTNL(I,J2)

C(I,J) = INTPNL(I,J)*G*M*778.*336.4

IF(KONTER.EQ.GO TO 663
IF(I.EQ.IPJ1.AND.J.GE.EL2.AND.J.LT.EL3) P(IP2,J)=P(IP1,J)
1 IF(J.EQ.IPJ2) P(I,J)=P(I,J)

660 CONTINUE
661 CONTINUE

IFLOW=14
EPINTH=PRINT
PRINT=10
NC 659 J=2, JP1

PROG DATE = 79111 01/42/10

IF(J.EQ.2) P(I,J)=P(I,J)
1 IF(J.EQ.2) P(I,JP1)=P(I,JP1)
1 IF(I.EQ.IP1.AND.J.GE.EL2.AND.J.LT.EL3) P(IP2,J)=P(IP1,J)
1 IF(J.EQ.IPJ1) P(I,J1)=P(I,J1)
DC 659 I=2,IP1
ERINT=ERINT+INTRNL(I,J)
659 CONTINUE
IF(NY.EQ.0) PRINT 713,ETOTAL,ERINT
713 FORMAT(' TOTAL ENERGY=',E15.8,' BTU/LB AND TOTAL INTERNAL ENERGY='
1,E15.8,' BTU/LB')
IFLOW=15
UKIN=C.
DO 712 J=2,LL1
UKIN=UKIN+U(IP2,J)*RHO(IP2,J)*6.28*(R(IP1)+DH)*DZ
IF(J.EQ.(LL1-2))UKCONT=U(IP2,J)*RHO(IP2,J)
712 CONTINUE
CNDFLW=0.
DO 709 I=2,IP1
709 CNDFLW = CNDFLW + V(I,LL3)*RHO(I,LL3)*(4.14159*((R(I)+DH)**2-(P(I)
1-DH)**2)
UKOUT=0.
DO 711 J = LL3,JP1
UKOUT= UKOUT+U(IP2,J)*RHO(IP2,J)*6.28* DZ*(R(IP1)+DH)
711 CONTINUE
705 CONTINUE
HTLOAD=-AVMASS*HVIP*6.28*RV*C0RL*3600.
HEAFLB=0.
DO 707 I=2,IP1
HEAFLB=HEAFLB+3.14159*RHO(I,L3)*HVIP*V(I,L3)
1*(R(I)+DH)**2-(P(I)-DH)**2
707 CONTINUE
HEAFLB=HEAFLB*0.2929/(R(IP1)+DH)**2*.0496
IF(NY.EQ.0)
1PPINT 706,HEAFLB, HTLOAD
706 FORMAT(' THE AXIAL HEAT FLUX WATTS/CM2=',F12.4,' THE EVAPORATOR
1 HEAT LOAD BTU/HR=',E12.4,' THE VAPOR SPACE HEAT LOAD BTU/HR=',
1E12.4)
IF(ABS(ERINT-EINTH).LE.0.9 AND .ABS(ETOTH-ETOTAL).LE.0.9 AND .
1(UKOUT-UKIN).LE.0AND.CYCLE.GE.50) KON=1
ENCK=ABS(UKIN/CNDFLW)
IF(ENCK.GT.9.9 AND ENCK.LT.1.1 AND KON.EQ.1) PRINT 714,UKOUT,
1UKIN,CNDFLW
IF(ENCK.GT.9.9 AND ENCK.LT.1.1) KON=1
IF(NY.EQ.0) PRINT 714, UKOUT, UKIN, CNDFLW
714 FORMAT(' THE TOTAL EXIT MASS FLOW LBM/SEC=',E12.4,' TOTAL INLE
1 MASS FLOW LBM/S=',E12.4,' COND. INLET=',E10.3)
IF(KON.EQ.0) GO TO KRET,(250,416,404,413,403,402,401,640,400)
C CHECK RHO FOR S.S. RHO MUST = CONSTANT THROUGH MESH
RMAX=0.
RMIN=1.E+25
DO 719 J=2,JP1
719
RMIN = \text{MIN}(RMIN, RHO(IP1-2, J))

RMAX = \text{MAX}(RMAX, RHO(IP1-2, J))

\text{CHECK} = \text{APS}(RMIN/RMAX)

\text{IF(CHECK} \geq 0.99) \text{KON\textlbrace T\textrbrace = 0}

\text{KCONVRG = KCONVRG + 1}

\text{PRINT 717, CYCLE, T, KCONVRG, TAU, RE}

\text{IF(KON\textlbrace T\textrbrace = 0, AND \text{KCONVRG} > 300) KON = 0}

\text{717 FORMAT(/' FOR CYCLE\textlbrace I\textrbrace, T, AND \text{THETA, CHECK} \text{ID}', TIMES \text{TAU}, \text{RE}, /')}

\text{IF(KON\textlbrace T\textrbrace = 0) DT = DT * 10.}

\text{IF(KON\textlbrace T\textrbrace = 1) Trem\textlbrace T\textrbrace = Trem\textlbrace T\textrbrace * 10.}

\text{IF(KON\textlbrace T\textrbrace = 1) TFIN = TFIN * 10.}

\text{IF(KON\textlbrace T\textrbrace = 1) TPR = TPR * 10.}

\text{GO TO KREF(250, 416, 404, 413, 403, 402, 401, 640, 400)}

\text{C --- MOV\textlbrace F, PACK AND INPUT PARTICLES.}

\text{700 NPT = 0}

\text{NPN = 0}

\text{IF(LC = 11}

\text{K = 1}

\text{KN = 1}

\text{710 IF(NPT < NP) GO TO 735}

\text{ID = XP(K) \text{+ 2.}}

\text{HPX = ID - 1, - XP(K)}

\text{HMX = 1, - HPX}

\text{JR = XP(K) + 1, + 1.5}

\text{HPY = JR - 0.5, - XP(K + 1)}

\text{HM = 1, - HPY}

\text{UK = HPX \text{+ HMY \text{+ U(ID - 1, JR + 1) \text{+ HMX \text{+ HMY})}}}

\text{*U(ID, JR + 1) \text{+ HFX \text{+ HPY \text{+ U(ID - 1, JP) \text{+ HMX \text{+ HPY \text{+ U(ID, JR})}}}}}

\text{IR = XP(K) \text{+ 1.5}}

\text{HPX = IR - 0.5, - XP(K)}

\text{HMX = 1, - HPX}

\text{JD = XP(K + 1) \text{+ 2.}}

\text{HPY = JD - 1, - XP(K + 1)}

\text{HM = 1, - HPY}

\text{VX = HPX \text{+ HPY \text{+ V(IR, JD) \text{+ HMX \text{+ HMY \text{+ V(IR, JD + 1) \text{+ HPX \text{+ HPY}}}}}}}

\text{*V(IR, JD + 1) \text{+ HMX \text{+ HPY \text{+ V(IR, JD - 1) \text{+ HFX \text{+ HPY \text{+ V(IR, JD - 1) \text{+ HFX \text{+ HPY}}}}}}}

\text{XP(K) = XP(K) \text{+ UK \text{+ DTDR} \text{+ DTODZ}}}

\text{XP(K + 1) = XP(K + 1) \text{+ VK \text{+ DTODZ}}}

\text{L = XP(K) \text{+ 2.}}

\text{J = XP(K + 1) \text{+ 2.}}

\text{IF(TYPE \text{= 0) GO TO 715}

\text{IF(XP(K) \text{GE IPAR) GO TO 720}

\text{715 KN = KN \text{+ 2}}

\text{NPN = NPN \text{+ 1}}

\text{720 IF(F(I, J) \text{EQ EMP) GO TO 760}
213

730  K=K+2
    NPT=NPT+1
    GOTO 710

735  NP=NPN
    IF(TYPE.EQ.0) GOTO 250

740  X=-AVEU*RDRI*(T+DT)-XDIS*(2.*COLS+1.)
    IF(X.LT.0.*AND.KON.EQ.1) GO TO 113
    IF(X.LT.0.) GOTO 250

750  XP(KN)=X
    XP(KN+1)=Y
    KN=KN+2
    I=X+2
    J=Y+2
    NP=NP+1
    Y=Y+YDIS
    IF(F(I,J).NE.EMP) GOTO 755
    F(I,J)=SUR
    U(I,J)=UL

755  IF(KN.GT.LPB) GOTO 75
    IF(NP.LT.NPN) GOTO 750
    GOTO 740

760  F(I,J)=SUR
    IF(F(I+1,J).EQ.EMP) U(I,J)=U(ID,JD)
    IF(F(I-1,J).EQ.EMP) U(I-1,J)=U(ID-1,JD)
    IF(F(I,J+1).EQ.EMP) V(I,J)=V(ID,JD)
    IF(F(I,J-1).EQ.EMP) V(I,J-1)=V(ID,JD-1)
    GOTO 730

4999  CONTINUE
    RETURN
    END
SUBROUTINE HEAD (MODE, LABEL, T, N, LIN, INDIC)
DIMENSION LABEL(6)
NTAPE = 12
IF(INDIC.EQ.2000) WRITE(NTAPE,2000) LABEL, T, N
IF(INDIC.EQ.1000) WRITE(NTAPE,1000) LABEL, T, N
LIN = 3
RETURN
1000 FORMAT (1H1,2X,6A6,2X,2HT=,E12.5,2X,6HCYCLF=,I5,1H1,4X,1HJ,
16X,6HF(I,J),8X,6HU(I,J),6X,6HV(I,J),5X,6HP(I,J),5X,10H G(I,J)
2,4X,8HRHO(I,J),7X,6HPBAR(I,J),3X,6HC(I,J)).
2000 FORMAT (1H1,2X,6A6,2X,2HT=,F12.5,2X,6HCYCLE=,I5,1H1,4X,1HJ,
16X,6HF(I,J),5X,9HRICE(I,J),5X,6HS(I,J),9X,6HE(I,J),5X,
110HRHO-OLD,7X,6HINTRNL,7X,6HQP(I,J),5X,12HUN DIM PRESS)
END

SUBROUTINE FRAME (NPLT,KIND,NX,NY,DX,DY,LABEL,TIMF,NCYCLF)
COMMON /OBST/ L1,L2,L3,L4,L5,L6,L7
COMMON /PLT/ XO,YO,SCALE,DXS,DYS
DATA XSIZE /6.0/, YSIZE /8.0/,
1 PXSIZE /8.5/, PYSIZE /11.0/
DIMENSION LABELL(6)

IF(NPLT.GT.0) GO TO 10
C --- INITIALIZATION.
XMAX = NX*DX
YMAX = NY*DY
C --- DETERMINE OP NTATI ON.
MAXPPB = 0
MAXPLT = C
IF(XMAX.LT.YMAX) MAXPRB = 1
IF(XSIZE.LT.YSIZE) MAXPLT = 1
IF(MAXPRB.EQ.MAXPLT) GO TO 5

TEMP = XSIZE
XSIZE = YSIZE
YSIZE = TEMP
TEMP = PXSIZE
PXSIZE = PYSIZE
PYIZES = TEMP

5 XSCALE = XSIZE/XMAX
YSCALE = YSIZE/YMAX
SCALE = AMIN1(XSCALE,YSCALE)
XINC = XMAX*SCALE
YINC = YMAX*SCALE*6.
YO = 3.-PXSIZE
YO = 20.
NPLPAG = 28./PYSIZE
NPLSET = 2*NPLPAG
SIGN = -1.
DXS = DX*SCALE
DYS = DY*SCALE

IF(KIND.EQ.0) GO TO 10

C ----- OUTLINE FOR OBSTACLE PROBLEM.
X1 = L5*DXS
X2 = L6*DXS
Y1 = L1*DXS
Y2 = L2*DXS
Y3 = L3*DXS
Y4 = L4*DXS
Y5 = L7*DYS

C ----- DRAW OUTLINE AND LABEL IT.
10 IF(MOD(NPLT,NPLSET).NE.0) GO TO 15
XO = XC+2.*PXSIZE
YO = 28.
15 NPLT = NPLT+1
IF(SIGN.EQ.-1) YC = YO-PYSIZE
XO = XO+SIGN*PXSIZE
SIGN = -1.*SIGN
YLAB = YO-.4
XLAB = XO+.5*PXSIZF-.5
CALL SYMBOL (XLAB, YLAB, .14, 7HTIME = , 0.0, 7)

CALL SYMBOL (XLAB, YLAB, .14, .14, 7HTIME = , 0.0, 7)
FRAME DATE = 73111 01/42
XLAB = XLAB+-.96
CALL NUMBER (XLAB, YLAB, .14, TIME, 0.0, 6)
XLAB = XLAB+1.75
CALL SYMBOL (XLAB, YLAB, .14, 8HCYCLE = , 0.0, 8)
XLAB = XLAB+1.12
CYCLE = NCYCLE
CALL NUMBER (XLAB, YLAB, .14, CYCLE, 0.0, -1)
XLAB = XO
YLAB = YO-.8
DO 20 I = 1,6
CALL SYMBOL (XLAB, YLAB, .14, LABEL(I), 0.0, 8)
20 XLAB = XLAB+1.12
IF(KIND.NE.0) GO TO 25

C ----- SIMPLE OUTLINE.
CALL PLOT (XO, YO, 3)
CALL PLOT (XO+XINC, YO, 2)
CALL PLOT (XO+XINC, YO+YINC, 2)
CALL PLOT (XO+XINC, YO+YINC, 2)
CALL PLOT (XO, YO, 2)
RETURN

C ----- OBSTACLE OUTLINE.
25 CALL PLOT(XO,YO,3)
CALL PLOT(XC,YO+YINC,2)
CALL PLOT(XO+Y1,YO+YINC,2)
CALL PLOT(XO+Y2,YO+YINC,3)
CALL PLOT(YO+Y3,YO+YINC,2)
CALL PLOT(XO+XINC,YO+YINC,2)
CALL PLOT(XC+Y1,YO+YINC,2)
CALL PLOT(XC+XINC,YO+YINC,2)
CALL PLOT(XO,YO,2)
RETURN
END

SUBROUTINE PLOT (X,MANY)

PARICLE PLOTTING.

COMMON /PLT/ XG,YG,SCALE,DXS,DYS
DIMENSION X(MANY)

IF(MANY.LT.2) RETURN
DO 5 N = 1,MANY,2
XP=XO+X(N+1)*DXS
YP=YD+X(N)*DYS
X1=XP
X2=XP
Y2=YP
Y1=YP
CALL PLOT (XP,YP,3)
CALL PLOT (XP,YP,2)
5 CONTINUE
RETURN
END

SUBROUTINE VPLT (F,U,V,IP2,JP1,DV,EMP)

VELOCITY VECTOR PLOTTING.

COMMON /PLT/ X0,Y0,SCALE,DXS,DYS
DIMENSION F(IP2,JP2), U(IP2,JP2), V(IP2,JP2)

IP1 = IP2-1
JP1 = JP2-1
DO 5 J = 2,JP1
DO 5 I = 2,IP1
IF(F(I,J),GE,EMP) GO TO 5
X1=X0+(J-1.5)*DXS
Y1=YD+(I-1.5)*DYS
IF(DV.LE.1.E-05) ELFAC=500.
IF(DV.GT.1.E-05) ELFAC=1.
5 CONTINUE
\[ X_2 = X_1 + 0.5 \times (V(I-1,J)+V(I,J)) \times DV * SCALE / 10. \]

\[ Y_2 = V_1 + 0.5 \times (U(I-1,J)+U(I,J)) \times DV * SCALE / 10. \]

1*ELFAC
CALL PLOT (X1,Y1,3)
CALL PLOT (X2,Y2,2)
5 CONTINUE
RETURN
END

SUBROUTINE TCNL (T,X,JF)

C SAT. LIQUID THERMAL CONDUCTIVITY (W/C)

GO TC (1,2,3,4,5,6,7,8,9,10,11,12), JF
1 T=1.8*T-459.67
   X = 0.49998+2.7992E-4*T+2.2565E-8*T*T-2.4606E-11*T*T*T
   GO TO 13
2 T=1.8*T-459.67
   X = 1.4298-6.1292E-4*T+1.7127E-7*T*T-3.0633E-11*T*T*T
   GO TO 13
3 T=1.8*T-459.67
   X = 0.9689-4.7904E-4*T+1.3778E-7*T*T-2.4884E-11*T*T*T
   GO TO 13
4 T=1.8*T-459.67
   X = 1.4298-6.1292E-4*T+1.7127E-7*T*T-3.0633E-11*T*T*T
   GO TO 13
5 T=1.8*T-459.67
   X = 0.9689-4.7904E-4*T+1.3778E-7*T*T-2.4884E-11*T*T*T
   GO TO 13
6 T=1.8*T-459.67
   X = 0.49609-4.7904E-4*T+1.3778E-7*T*T-2.4884E-11*T*T*T
   GO TO 13
7 X=1.84235F-1+2.05547E2/T-1.062331E5/T**2+1.60138E7/T**3-4.43661E9/T**4
   GO TO 13
8 X=3.91471E—1—1.084549E-4*T+1.58502E2/T
   GO TO 13
9 X=-1.118324+4.745658E-3*T-2.795012E-6*T*T+7.594127E-10*T**3-1.99E-14*T**4
   GO TO 13
10 CALL OUT
   GO TO 13
11 CALL OUT
   GO TO 13
C H2O CORRELATION GOOD 273 T 373 DEG.K IN BTU/H-F-F
12 X=(0.0743 + 9.00E-04*T)
   X = X*.7087
   GO TO 13
C CONVRT WATTS/IN-F TO WATTS/CM-C
13 X = X*.7087
   IF(JF.LE.5) T=(T+459.67)/1.8
   RETURN
END

TCNL DATE = 73111 01/4
SUBROUTINE CSPL (T, X, JF)

SAT LIQUID SPECIFIC HEAT (W-SEC/G-K)

GO TO (1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12) JF

LATENT HEAT OF FUSION SIMULATION - ASSUME THAT SPECIFIC HEAT INC

BY AN AMOUNT EQUAL TO THE LATENT HEAT OVER A TEMP. RANGE OF 44C

TO INSURE THE TEMPERATURE PASSES THROUGH THE REGION

IF(T.LT.298.) GO TO 15
IF(T.LT.348.) GO TO 16
IF(T.LT.440.) GO TO 17
IF(T.LT.470.) GO TO 18
IF(T.LT.800.) GO TO 19

X = 4.294012 - 1.210551E-4*T

GO TO 13

15 X = 0.814 * 0.2389

GO TO 13

16 X = 0.872 * 0.2389

GO TO 13

17 X = 0.95 * 0.2389

GO TO 13

18 X = 103.2/30.*0.2389

GO TO 13

19 X = 0.2389

ABOVE TAKF FROM RARE METALS HANDBOOK, REFINHOLD CO., LONDON, 1961

GO TO 13

2 X = 1.63249 - 8.359602E-4*T + 4.637195E-7*T*T

GO TO 13

3 X = 9.512849E-1 - 4.860081E-4*T + 3.122763E-7*T*T

GO TO 13

4 T = 1.8*T - 459.67
X = 4.187 * (0.09915 - 3.106E-5*T + 1.299E-6*T*T)

GO TO 13

5 T = 1.8*T - 459.67
X = 4.187 * (0.08543 - 9.605E-5*T + 5.985E-8*T*T)

GO TO 13

6 X = 1.519435F - 1.5970309E-5*T + 5.301029E-8*T*T

GO TO 12

7 X = 1.072183 - 4.990436E-4*T + 3.125257E-7*T*T

GO TO 13


GO TO 13

9 CALL OUT

GO TO 13

10 CALL OUT

GO TO 13

11 CALL OUT

GO TO 13

12 X = 4.197

CP IN WATT-SEC/GM DEG K

13 RETURN

END
SUBROUTINE RHDL (T, X, JF)
SAT LIQUID DENSITY (G/CM3)
GO TO (1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12), JF
1 X = 5.46398E-1 - 9.381799E-5*T + 9.318741E-9*T*T
GO TO 13
2 X = 1.013338 - 2.350445E-4*T - 9.861048E-10*T*T
GO TO 13
3 X = 9.083578E-1 - 2.244534E-4*T - 1.274617E-8*T*T
GO TO 13
4 X = 1.575802 - 3.074245E-4*T + 3.837297E-8*T*T
GO TO 13
WADD TR 61-96
GO TO 13
5 X = 1.985607 - 4.549765E-4*T - 8.955005E-8*T*T
GO TO 13
6 X = 1.438176E-1 - 2.861766E-3*T + 3.763475E-7*T*T
GO TO 13
7 X = 9.3800519589E-1 - 2.3937338810E-4*T + 3.5881034579E-9*T*T
GO TO 13
8 X = 2.8259 - 3.69951IE-4*T
GO TO 13
9 X = 1.040067E-1 - 9.070670E-4*T
GO TO 13
10 X = 2.277 - 3.26E-4*T
GO TO 13
11 X = 2.359 - 4.95E-4*T
GO TO 13
12 X = -254.259 + 3.6568*T - 0.015728*T*T + 3.00121E-05*T**3
1 - 2.1657E-08*T**4
X = X*454./(1728.*2.54**3)
WATER DENSITY GOOD LBM/FT3 GOOD 288.5 T 388.5 DEG K
13 RETURN

SUBROUTINE TFNS (T, X, JF)
SAT LIQUID SURFACE TENSION (DYN/CM)
GO TO (1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12), JF
1 X = 4.544968F2 - 1.356226E-1*T + 1.615487E-6*T*T
GO TO 13
C TAYLOR, PROG. NUC. ENG. V-2
2 X = 7.75895262 - 1.356323*T + 9.517612E-4*T*T - 2.375630E-7*T**3
GO TO 13
220

3 $T = 1.8 \times T - 459.67$
   $X = 1.459 \times E + 3 - 2.8149 \times E - 4 \times T$
   GO TO 13

4 $X = 1.347196 \times E + 2 - 5.606006 \times E - 2 \times T - 1.513351 \times E - 5 \times T \times T$
   GO TO 13

5 $T = 1.8 \times T - 459.67$
   $X = 76.4 - 0.3 \times (T - 83)$
   GO TO 13

6 $X = 4.876255 \times E + 2 + 1.132794 \times E - 3 \times T - 2.458797 \times E - 4 \times T \times T$
   GO TO 13

7 $X = 1.201687 \times E + 2 - 3.890232 \times E - 2 \times T$
   GO TO 13

8 CALL OUT
   GO TO 13

9 $X = 1.0850000 \times E + 3 - 1.400000 \times E - 1 \times T$
   GO TO 13

10 $X = 330 - 0.1 \times T$
   GO TO 13

11 $X = 373.2 - 0.109 \times T$
   GO TO 13

C SURF TENSION DYNE/CM GOOD
273 T 373 DEG K

12 $X = -1.47566 \times E + 0.6 + 4.5737 \times T - 621.3 \times T \times T + 4.86528 \times T \times T \times T$
   $-0.024239 \times T \times T \times T + 7.9771 \times E - 0.5 \times T \times T \times T - 1.73564 \times E - 0.0 \times T \times T \times T$
   $+2.4093 \times T \times T \times T - 1.937 \times E - 13 \times T \times T \times T + 6.8769 \times E - 17 \times T \times T \times T$
   $-1.73564 \times T \times T \times T \times T - 6.8769 \times E - 17 \times T \times T \times T$
   $+2.4093 \times T \times T \times T \times T - 1.937 \times E - 13 \times T \times T \times T \times T$
   $+6.8769 \times E - 17 \times T \times T \times T \times T$
   RETURN

END

MAIN DATE = 73111
SUBROUTINE PSAT (T, X, JF)
VAPOR PRESSURE (BARS)

1 $X = 1.01356 \times 10^{-5} \times (4.88 \times E - 1787.9 \times T)$
   GO TO 13

2 $X = \exp (9.983178 - 1.091806 \times E / T - 6.862319 \times E / T \times T \times T)$
   GO TO 13

3 $X = \exp (9.191863 - 9.030992 \times E / T - 4.33038 \times E / T \times T \times T)$
   GO TO 13

4 $T = T + 1.8$
   $X = 0.06895 \times 10^{-5} \times (5.20071 - 6994.6 \times T)$
   GO TO 13

5 $X = \exp (8.636035 - 7.715273 \times E / T - 3.846408 \times E / T \times T \times T)$
   GO TO 13

C PSAT HG VAPOR TAKEN FROM REF 49 FL WAKIL IN LBF/IN2 420 T 1530
CHANGE TEMP TO RANKINE

6 $T = T + 460$
BLT = ALUG (T)
   $X = 0.19337 \times \exp (-13711 / T - 0.8 \times BLT + 24.356)$
CHANGE TEMP BACK TO F
T = T-460.
GO TO 13
7 CALL OUT
GO TO 13
8 CALL OUT
GO TO 13
9 X = -2.026076 - 7.430453E-3 * T + 1.525201E-5 * T * T - 8.751015E-9 * T * T * T + 1 + 1.622201E-12 * T * T
BOHODANSKY AND SCHINS, J. PHYS. CHEM. VOL 71 NO 2
GO TO 13
10 X = .001333 * EXP(26.2075 - 26.9552E3/T)
GO TO 13
11 X = .001333 * EXP(20.256 - 26.270E3/T)
GO TO 13
MAKE INTO DEGREES FAHRENHEIT FOR THIS CALCULATION FROM KAPLN-M-S3G-RES
12 CONTINUE
CONVERT TO BARS 1 BAR = 14.5037738 LB/IN^2 GOOD 32F T 660F
X = 4.02E6 * EXP(-8.338E3/(T+459.6))
1 CM^3/GM = 1/62.42796 FT^3/LB
1 JOULE/GM = 0.429922614 BTU/LB
13 RETURN
END

MAIN
DATE = 73111

SUBROUTINE HVAP (T, X, JF)
SUBROUTINE FAT OF VAPORIZATION (J/GM)
GO TO (1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12), JF
1 X = 2.639E+4 - 5.325 * T + 6.25E-4 * T * T
WEATHERFORD, WADD TR 61-96
GO TO 13
2 X = 4.178649E3 + 2.829841E-1 * T - 4.765964E-4 * T * T
GO TO 13
3 X = 2.269079E3 - 1.318445E-1 * T - 2.003039E-4 * T * T
GO TO 13
4 X = 1.013000F3 - 2.080000E-1 * T
WEATHERFORD, WADD TR 61-96
GO TO 13
5 X = 6.0503G2E-2 - 6.543721E-2 * T - 5.902942E-5 * T * T
GO TO 13
6 X = -8.37285E-11 * T * T * T + 1.719541E-07 * T * T * T - 1.2451E-04 * T * T
1 + 5.08665E-03 * T + 312.089
SUBROUTINE FAT HG VS T KELVIN
GO TO 13
7 CALL OUT
GO TO 13
B CALL OUT
GO TO 13
9 X=2.584048E3-9.285714E-2*T
GO TO 13
10 X=4.6700E3+5.3995E-1*T
GO TO 13
11 X=1.0306E4-1.0039*T
GO TO 13
LATENT HEAT H2O GOOD 273 T 377 DEG. K ERGS/GM
12 X=-0.62173E+13 +1.0276E+11*T -6.3592E+08*T*T +1.5673E+06*T**3
1 +6.2942E+02*T**4 -11.6*T**5 +.022906*T**6 15003F-04*T**7
X=X/(10.**7) :
13 RETURNEND '  - -  -  -  -
SCREEN DATE = 73111
SUBROUTINE SCREEN(IP, IP2, JP2)
COMMON A1(1200), A2(1200), A3(1200), A4(1200), A5(1200), A6(1200),
A7(1200), A8(1200), A9(1200), A10(1200), A11(1200), A12(1200),
1FLUX, QIN, KONIER, WKMRA, RH, EP, PERMEA,
2A13(1200), A14(1200), A15(1200), A16(1200), A17(1200), A18(1200), A19(1200),
3RGT, BND, LAMBD, GOF, GOF2, COLS, CYCLE, DR, DH, DRODZ, DTHOLD,
4DROU, DT, DTODR, DTODZ, DZ, DZODR, EMP, EPS, FUL, GR, G7,
511(1200), IALL, JBAR, JP1, IPHM, JBAR, JP1, LENCOM, LPB,
6LP, NAME(6), IN, N, VISCS, NUMTO, PB, PB, R(40), RDR, RDR2,
7RDZ, RDZ, RDZ2, RIP(160), ROS(160), SUR, T, TPLT, TPRT, IT, THFIN, TPRT,
8JFLUID, TEMP, UNIVG, DEF, BCRF(160), LL1, LL2, LL3,
9PRESS, TEB, TEE, TCB, TCE, TAU, THETA, PHI, TEMPT, TEMPI, WMOL, FINTRL, XINTINT,
10TWPRT, TWT, TYPE, UL, UR, W, XDIS, XP(14000), YDIS, YFIR,
11IEND
COMMON /OBST/ L1, L2, L3, L4, L5, L6, L7
DIMENSION P(IP2*JP2)
INTEGER CYCLE
111=1
IF(JFLUID .NE. 12) GO TO 44
UNIT=8 THIS SUBROUTINE IN BTU-DEG F- IN- SEC
****** THE FOLLOWING IS AN EXPLANATORY LIST OF NOTATIONS USED **
QIN IS THE HEAT INPUT/HR IN BTU/HR
CORI IS THE EVAPORATOR LENGTH
RW IS THE RADIUS TO THE OUTSIDE OF THE WICK OR THE WALL RADIUS
TEMP. IS THE TEMPERATURE BEING CONSIDERED FOR THAT PROBLEM
QRADAL IS THE RADIAL HEAT FLUX AT THE EVAPORATOR
HVOP IS THE LATENT HEAT OF EVAPORATION BTU/LBM
VISCL IS THE LIQUID VISCOSITY G/CM-SEC AND VISCG IS GAS VISCOSIT
RHAL IS THE DENSITY OF WATER AT A SPECIFIC TEMP
IF(JFLUID.NE.12) TEMP=TEMPI
IF(JFLUID.NE.12) GO TO 44
IF(ICYCLE.LE.3) TRW=TEMP
IF(ICYCLE.LE.3) TEMP=TRW-8.

44 CONTINUE
40 CONTINUE
JF=JFLUID
IF(JF.NE.12) TRW=TEMP
MJ=0
PI=3.14159
CORL=(L2-L1)*DZ
IF(OIN.NE.0.) GO TO 43
41 QIN=FLUX*2.*PI*RW*CORL
43 CONTINUE
HPL = JBAR*DZ
HPFL = HPL/CORL
VOL= PI*EP *(RW**2-RV**2)*HPL
EP = THE FRACTION OF WICK VOL WHICH IS LIQUID
QPIPE=QIN/3600.
D=2*RW
DRADAL=QPIPE/(2.*PI*RW*CORL)
RV=DP*IBAR
HYDD=2*(RW-RV)
TRW=TRW+5.

50 CONTINUE
IF(ICYCLE.LE.3) TEMP=TRW-8.

SCREEN DATE = 73111

TRW = 5.79*(TRW -32.)*373.
TEMP= 5.79*(TEMP-32.)*373.
PRINT 1000,TEMP,TRW
1000 FORMAT(5,TEMP=1,E10.4,DEG K AND TRW =.E10.4)
CALL HVAP(TEMP,HVOP,JF)
CONVERT J/G TO BTU/LBM
HVOP=HVOP* 9.4783E-04*454.
IF(JF.EQ.12) HVOP=HVOP+100.

CALL RHOL(TEMP,RHAL,JF)
CONVERT G/CM3 TO LBM/IN3
RHAL=RHAL*2.54**3/454.

CALL VISL(TEMP,VISCL,JF)
CALL VISL(TRW, VISCL1,JF)
CONVERT G/S-CM TO LBM/SEC-IN
VISCL1= VISCL1*2.54/454.
VISCL = VISCL *2.54/454.

CALL CP(SL(TEMP,CP,JF)
CONVERT W-SEC/G-K TO BTU/LBM-F
CP= CP*454.*100./(180.)*9.47E-4
CALL TCNL(TEMP,EKH20,JF)
CONVERT W/CM-K TO BTU/SEC-IN-F
EHK20=EKH20/(3600.*12.4)

CALL WICK (TEMP,Fkwck,JF)
CONVERT BTU/H-F-F TO BTU/SEC-IN-F
EKWCK=EKWCK/(3600.*12.)
TEMP=(TEMP-273.)*9./5.+32.
TRW=(TRW-273.)*9./5.+32.
PRINT 2000,RHAL,HVOP,VISCL1,CP,EKH20,EKWCK
2000 FORMAT(* RHAL,HVOP,VISCL1,CP,EKH20,EKWCK=','7(E9.2,2X))
AA1=2*PI*R*C
V=QPIPE/(RHAL*AA1*HVOP)
AA2=PI*(RHW**2-RV**2)*EP
U=AA1*AA2
IF(JF,NE,12) GO TO 45
RE=RHAL*HV0D/VISCL
PR=CP*VISCL/EKH20
SIEDER-TATE E Q N. EL WAKIL P. 250
HC=EH20/HYDD+0.023*RF**C.E*PR**.4*(VISCL1/VISCL)**0.14
1 +EKWCK/(RHW-RV)
PRINT 4000,RF,PR
4000 FORMAT(* RE AND PR= * ,2(E12.4,3X))
IF(TRW.EQ.0.0.) TRW=TEMP+5. __________
CiNEW=HC*2*PI*RW*C0RL*(TRW-TEMP)
IF (ABS(QPIPE/CNEW).LT.1.03.AND.ABS(JQPIPE/QNEW).GT.0.97)GO TO 10
IF (QPIPE=QNEW20,10,30
10 CONTINUE
DI = HYDD*RHAL*U/EKH20*CP*D/CORL
WRITE (6,941)I,JP,CORL,HPL,QPIPE,QRADAL,TEMP
94 FORMAT(* fluid * ,3X. CORL (IN),5X. HPL (UN),7X. QPIE (BTU/S) ,4X. TEMP (F) ,11X. 16.9X, 2 A3,5E15.7//) _ _  _______ _________________________________________
FIND TRO-TRV
EIMP=1/(2*PI*RW*HC) . . . ........................
DELTAT = EIMP/CORL*QPIPE
WRITE (6,600)DELTAT,QPIPE,TEMP
600 FORMAT(* THE TEMP. DIFF. FROM RW TO RV IS * ,F12.6,' THE DEGREES FOR OP
1/PE='F12.6,' AND A TEMP. OF * ,F12.6)
IF(DELTAT.GT.9.) PRINT 601
601 FORMAT(* -- CAUTION--- THE DELTA TEMP ACROSS THE WICK INDICATES
A POSSIBILITY OF BURNOUT EXISTS*)
60 RETURN
45 CONTINUE
TEMP= 5./9.*((TEMP-32.)+273.
CALL TENS(TEMP,X,JF)
TEMP=(TEMP-273.)*9./5.+32.
CONVRT DYN/CM TO LBF-IN D*1.0197E-03 =GMS
X=X*1.0197E-03*2.54/454.
CELPIN = 4X/WKMR
DLPWK = VISCL*U/(PERMFA*CD)*HPL
DLPVP = P(IP1,ZI) - P(IP1,JP2)
IF(CELPIN+DLPWK+DLPVP) GO TO 46
PRINT 3000, CELPIN, DLPWK, DLPVP
5000 FORMAT ('; THE INTERFACE DELTA P=1,F12.4; THE WICK DELTA P=1,F12.4; AND THE VAPOR SPACE DELTA P'=1,F12.4)
RETURN
46 PRINT 3000, CELPIN, DLPWK, DLPVP
PRINT 3500
3500 FORMAT ('PERMUTATION HAS OCCURRED -- THE PROBLEM IS TERMINATED')
RETURN
20 CONTINUE
D1 = HYD*PHAL*U/EKH2C*CP*P/CORL
WRITE (6,94) II1, JP, CORL, HPL, OPIPE, QNFW, TEMP
WRITE (6,95) HC, QIN, QNFW, TEMP, D1
WRITE (6,96) II1
94 FORMAT (5X, 'ITERAT', '17, 'OPIPE-QNEW NEG')
IF (ABS(QPIPE/QNFW).GT. .92 ) GO TO 31
      TEMP=TEMP+1
      GO TO 32
31 TEMP=TEMP+.01
32 II1=II1+1
IF (II1. LT. 25 ) GO TO 60
   GO TO 50
20 WRITE (6,94) II1, JP, CORL, HPL, OPIPE, QNFW, TEMP
D1 = HYD*PHAL*U/EKH2C*CP*P/CORL
WRITE (6,95) HC, QPIPE, QNFW, TEMP, D1
WRITE (6,97) II1
57 FORMAT (5X, 'ITERAT', '17, 'OPIPE-QNEW POS')
II1=II1+1
IF (II1. LT. 25 ) GO TO 60
      IF (ABS(QPIPE/QNFW).LE.1.08 ) GO TO 33
      TEMP=TEMP-1
      GO TO 34
33 TEMP=TEMP-.01
34 GO TO 50
END

WICK DATE = 73111

SUBROUTINE WICK(T, X, JF)
CONDUCTIVITY OF WICK SCHWARTZ 200 MESH S.S. WICK BTU/HR-F-E
GOOD 300 T 372 DEG K
12 T = -54.96 +.45922*T -1.26577E-03*T*T +1.15975E-06*T**3
RETURN
END
VISL

DATE = 73111

SUBROUTINE VISL(T,X,JF)

SAT LIQUID VISCOSITY (G/SEC-CM)

GO TO (1,2,3,4,5,6,7,8,9,10,11,12), JF

1 X = 2.924347E-3 - 2.648556/E3/T + 2.995261E3/T**2 - 5.302574E5/T**3

GO TO 13

2 X = -1.04205E-3 + 4.11188/E - 1.74178E3/T**3 + 4.954036E5/T**4

GO TO 13

3 X = -4.390586E-4 + 2.028652/E - 5.410948E2/T**2 + 1.646804E5/T**3

GROSSE, SCIENCE VOL 147

GO TO 13

4 X = 1.559395E-3 - 1.12291/E + 1.00179E3/T**2 - 8.367401E4/T**3

WEATHERFORD, WADD TR 61-96

GO TO 13

5 T = 1.8T - 459.67

X = 239.5/E - 5.9911*ALOG(T) + 4.32449

X = 1.3*EXP(X)

GO TO 13


GO TO 13

7 X = 2.187261E-4 + 8.019051E-1/7 + 2.142332E2/T**2 + 2.596423E4/T**3

GO TO 13

8 X = 1.05417E-3 - 5.576295/E + 2.100457E4/T**2

GO TO 13


GROSSE, J. INORG. NUCL. CHEM VOL 23

GO TO 13

10 X = EXP(-18.695 + 2.6466E+4/T + 1.471E+6/T**2)

FROM J. OF CHEM PHYS VOL 48 NO 1 JAN 68

GO TO 13

11 X = 5.86e-4*EXP(3936./T)

FROM MOSEL WORK (1965) AND THE REACTOR HANDBOOK

GO TO 13

W2O IN GM/SEC-CM GOOD 283 T 373 DFG K

12 X = -2.32111E+ -7.332E-03*T*T + 1.4151E-06*T**3 + 6.93806E-05

1*T**5 = -3.635E-05*T**5 + 1.161E-12*T**6 - 2.7393E-16*T**7

2 - 1.8945E-18*T**8 + 2.7248E-21*T**9

RETURN

END
APPENDIX C

EXPERIMENTAL INVESTIGATIONS
ON A MERCURY WORKING
FLUID- STEEL HEATPIPE
Recent work done at a national laboratory utilized a cylindrical heat pipe with a screen wick and mercury as the working fluid. (3) This heat pipe was constructed in order to determine the heat-transfer capability of a mercury system and to compare its operational limitations with theoretical limitations. With good wetting, mercury becomes a suitable working fluid yielding high heat-transfer rates in the 200°C to 400°C temperature range. In addition, the mercury heat pipe characteristics would probably be representative of any of the various liquid metal working fluid heat pipes which are being considered for use in nuclear space power plants. Their main use comes in transferring heat from a nuclear core directly and efficiently to thermionic diodes or to externally located boilers containing the working fluid of a Rankine cycle system.

This heat pipe, shown in Figure C-1, was made of Type 347 stainless-steel tube with a length of 19-5/8 inches, an outer diameter of 0.6 inch, and an inner diameter of 0.495 inch. Three layers of 100 mesh, Type 304 stainless-steel screen pressed against the inner wall of the pipe were used as the wick. The screen was forced against the inner wall by a steel ball which was pushed through the pipe before the pipe was sealed. The pipe was sealed at each end by caps which were welded into place.
In one of these end caps, a \( \frac{1}{2} \) inch outer diameter tube was installed which could be connected to a bellows valve for loading and sealing the pipe.

1. **Advantages of the Mercury Heat Pipe**

   It is known that heat pipes can be constructed to operate anywhere within any given part of the temperature range from \(-50^\circ\text{C}\) to \(2000^\circ\text{C}\) provided a proper working fluid is chosen for the particular range as pointed out by Deverall (35) and Arcella (38). It becomes difficult to find a suitable fluid for the \(200^\circ\text{C}\) to \(400^\circ\text{C}\) range in that temperatures in this range are too high for water. Water vapor pressure at these temperatures is high and presents containment problems; while the use of alkali metals is not recommended because their vapor pressures are too low to sustain operation.
FIGURE C-1. Stainless steel-mercury heat pipe
Figure C-2. Screen wick under increasing magnification
For this temperature range mercury is the best working fluid; however, it is very corrosive and, in a pure state, exhibits poor wetting characteristics. Its wetting characteristics were modified as will be discussed.

2. Wick Description

Shown in the plates of Figure C-2 are progressively enlarged views of a cross section of the wick. The wick is made of three layers of Type 304 stainless steel 100 mesh screen pressed together on the inside wall of the heat pipe. Plate I shows a cutaway view of the screen structure; plate II shows an enlarged close-up view within the same area; and plate III is an enlarged view within the area of plate II immediately next to the wall of the pipe. These plates will be discussed in connection with the deposit which is seen in the photos.

3. Working Fluid

The heat pipe was injected with 177 gms. of clean mercury and subsequently connected to a vacuum system and evacuated overnight to remove the noncondensible gases. The assembly was then sealed by closing the bellows valve. The next step was to insure that the mercury filled the volume between the heat pipe wall and the inner screen radius. This consisted of wetting-in at 500°C for two days by wrapping a tape heater around the full length of the pipe. It was noted, after the
wetting-in process, that there was still some noncondensible gas in the pipe. Adequate wetting was verified by radiographs taken immediately before the beginning of the testing.

4. **Heat Source and Sink**

Heat was supplied by a helical Nichrome wire which was enclosed in insulator beads and wound around the evaporator over a length of 7 inches. The Nichrome wire was surrounded by a two inch diameter tube, filled with insulation, and sealed at the ends with Alundum cement.

Heat was rejected to a water-cooled gas-gap calorimeter fitted with a "delta T" meter across the inlet and outlet connections to provide heat balance measurements. The gas gap volume contained a small capillary tube through which helium could be fed to replace the air, thus increasing the thermal conductivity across the gap. When the gap was completely full of helium, the thermal conductivity was increased by a factor of four over that present when air was contained in the gap. See Figure C-1 for illustration of the heat source and sink.

5. **Instrumentation**

Shown in Figure C-1 are the 10 thermocouple locations. Three thermocouples were attached to the
heater winding to monitor heater temperature, and seven thermocouples were spot welded along the length of the pipe. To record the thermocouple information, a sixteen point recorder was used.

6. Test Procedure

The testing was accomplished under conditions which demonstrate the factors affecting startup. For the test, each data set, i.e., each line designated by a power transferred in watts, was obtained after the heat pipe was allowed to reach a temperature equilibrium. The procedure was as follows: the heater was turned on and the pipe gradually heated to various temperature levels until the maximum or steady state temperature was reached. Readings were then taken before proceeding to the next level. The temperature differences of the "ΔT" meter across the calorimeter inlet and outlet connections were measured in millivolts by a potentiometer while the flow rate of the water through the calorimeter was determined by timing with a stopwatch the amount of time needed to fill a known volume. The heat transfer rates shown in the Figure C-3 for each power level, were taken from the calorimeter measurements.
Figure C-3. Experimental test results for the mercury working fluid-steel heat pipe
Burnout Test  This test was made with helium in the calorimeter gap and no noncondensable gas in the pipe. The pipe evaporator was gradually heated to various temperature levels and allowed to establish a steady state. This steady state, however, represented a startup situation in that there was a high temperature gradient along the pipe characteristic of sonic vapor velocities.

The results of this test for various values of heat transferred are shown in Figure C-3. Helium in the calorimeter gap resulted in high heat removal rates which resulted in the low vapor densities which characterize the sonic limit. With the heat input increased to 250 watts, the liquid in the wick was swept into, or entrained in, the vapor cutting of the liquid flow to the evaporator. With no liquid to carry the heat away the evaporator overheated.

From this experiment, data is obtained which imposed known heat inputs and outputs during startups which can be compared with the analytical approach developed for this dissertation. The primary purpose of the preceding experimental work was to demonstrate the effect of sonic vapor velocities and related startup problems which influence heat pipe design.

7. Postmortem Investigation of a Life Test Heat Pipe

A similar heat pipe was constructed to investi-
gate the effects of operation at 330°C for 10,000 hours. At the end of the 10,000 hours the operation of the pipe was still essentially isothermal over the entire length. After completion of the 10,000 hours the pipe was sectioned at various points and the wick structure and tube wall examined.

The results of this examination showed that mass transfer had occurred. As shown in Figure (C-2) Plate II, practically all the screen wires were covered with a coating which appeared quite uniform throughout the structure. Subsequent analysis revealed this to be an iron-nickel-chromium alloy similar to the original Type 304 stainless-steel except that the manganese and chromium concentrations were lower. The coating on the wall of the tube was predominantly iron with less chromium and nickel than the original 347 stainless steel. It was observed that heavier corrosion occurred at the crossover points of the screen wires. These points incurred large pressures or stress concentrations when the ball was forced down the length of the pipe. This concentration can be seen in Figure C-2 plate III.

Radiographs were taken immediately after the test which revealed an accumulation of material had formed in a small section of the evaporator. This accumulation, shown in Figure C-2 plate I, possessed a crystalline
structure, was quite porous, and did not penetrate into the other two layers of the screen. The make up of this material had a distribution of elements completely different from that of the original stainless steels used for the pipe and wick consisting mainly of nickel with high percentages of manganese. The constituents of the deposit, having some solubility in mercury, were apparently carried to the evaporator and deposited there when the mercury vaporized.

The previous discussion of the life test is offered here to point out that effects of corrosion must be taken into account for any heat pipe analysis as discussed in Chapter 6.
APPENDIX D

EXPERIMENTAL INVESTIGATIONS

ON A SODIUM WORKING

FLUID-STEEL HEAT PIPE
FIGURE D-1. Heat pipe used to investigate sonic limit using sodium as a working fluid
Shown in Figure D-1 is a schematic diagram of the sodium working fluid-steel heat pipe used in the research by Dzakowic et al, (8), to investigate the vapor velocity limit using sodium as the working fluid.

The following specifications were used for construction of this pipe:

- Stainless steel- sodium working fluid
- Effective length- 16.5"
- Outer diameter 1.05"
- Inner diameter .892"
- Wick 5 layers of compressed 60 mesh screen with a total thickness of .0825"
- Evaporator length 7"
- Condenser length 7.5"
- Adiabatic length 2"

As stated above, the wick consisted of five layers of 60 mesh screen(0.0825 inch thick). The permeability and porosity of this wick were not cited in the paper by Dzakowic et al(8); therefore, these values were taken from Kunz, et al(17) and were as follows in Table D-1:
<table>
<thead>
<tr>
<th>Sample No.</th>
<th>Mesh Size</th>
<th>Wire diameter/in</th>
<th>Equivalent capillary pore diameter microns</th>
<th>Porosity %</th>
<th>Friction Factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>M7 Ref. (17)</td>
<td>50</td>
<td>.009</td>
<td></td>
<td>609</td>
<td>62.5</td>
</tr>
<tr>
<td>Dzakowic Ref. (8)</td>
<td>60</td>
<td>.008</td>
<td></td>
<td>Not Reported</td>
<td>Not Reported</td>
</tr>
</tbody>
</table>
FIGURE D-2. Axial Temperature Profiles for Two Representative Heat Loads
The heat input was a 5KW heater 7" in length and the condenser was surrounded by a calorimeter which closely controlled the heat removal rate as in Appendix C.

Axial temperature profile measurements were obtained from 10 chromel-alumel thermocouples which were distributed along the pipe outer length.

Figure D-2 represents the temperature profiles obtained for two representative heat inputs of 1440 watt and 2020 watts, respectively. At the lower heat transport rate, the temperature profile shows a continual decrease from the peak evaporator temperature to the cold end of the condenser. The corresponding pressure (equilibrium saturation vapor pressure) is also decreasing. Because the ratio of the pressure upstream to that in the adiabatic section is larger than the value expected for the critical pressure ratio for a converging-diverging nozzle, the sonic velocity is found at the adiabatic section exit. The continuous expansion of the vapor in the condenser, as deduced from the condenser temperature profile, suggests that supersonic velocities might occur in this area. The temperature profile corresponding to the higher heat transport rate shows a constant downstream temperature from the adiabatic section, which indicates that pressure recovery is evident with decelerating vapor flow.
APPENDIX E

ADAPTED ICE INPUT INSTRUCTION
The input consists of 11 cards which describe the following general categories:

1. Pipe dimensions
2. Boundary conditions for the non-flow boundaries
3. Vapor properties
4. Start and finish times plus print times
5. Specification of flow boundaries, input and output velocities.
6. Specification of initial fluid properties and velocities for each section of the mesh.
7. Energy input specifications at evaporator and condenser.
8. Specification of wick properties and dimensions
9. Termination information

Card 1

Cols. Format: (2I5, F8.3, I5, 2F8.3)
1-5 IBAR = $I$, the number of inside cells in the $r$ direction.
6-10 JBAR = $J$, the number of inside cells in the $z$ direction.
11-18 DR = $\Delta r$, the cell size in the $r$ direction.
19-26 DZ = $\Delta z$, the cell size in the $z$ direction.
27-34 DT = $\Delta t$, the time step (sec.)
35-39  IPHM = The cycle after which the value of φ and θ are changed to PC
40-47  PC = The value of φ and θ after cycle IPHM
48-55  ALP = α, the overrelaxation parameter in the PBAR iteration scheme

Card 2

Cols. Format:(6A8)

Columns 2 through 48 of this card are used for problem identification on prints and plots. Column 1 cannot be used because it is treated as a carriage control. If desired, the card may be entirely blank, but it must always be included.

Card 3

Cols. Format:(4F3.1, 8F8.3)

1-3  BCB  Boundary condition for rigid wall (or wall section) on bottom, right, top, left mesh boundaries, respectively.
4-6  BCR
7-9  BCT
10-12 BCL
13-20 GM = 1 - γ where γ is the ratio of the specific heats cp/cv
21-28 B = the constant γ/pr applied to the conduction terms
29-36 λ = -2/3 μ
<table>
<thead>
<tr>
<th>Col.</th>
<th>Code</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>37-44</td>
<td>$\mu$</td>
<td>the viscosity ($\text{LBM (in.-sec.)}$)</td>
</tr>
<tr>
<td>45-52</td>
<td>EPS</td>
<td>$\epsilon$, the convergence criterion for PBAR calculation. Usually $2 \times 10^{-4}$.</td>
</tr>
<tr>
<td>53-60</td>
<td>GR</td>
<td>Gravity in the r direction (+ or -). ($\text{in./sec.}^2$)</td>
</tr>
<tr>
<td>61-68</td>
<td>GZ</td>
<td>Gravity in the z direction (+ or -). ($\text{in./sec.}^2$)</td>
</tr>
<tr>
<td>69-74</td>
<td>$\tau$</td>
<td>stability constant</td>
</tr>
</tbody>
</table>

**Card 4**

Format: \((5F8.3,12,2F8.3)\)

1-8 \(t_0\), the starting time, usually zero.

9-16 TWPLT \(\text{Intervals of problem time between micro-film plots. (sec.)}\)

17-24 TWPRT \(\text{Intervals of problem time between cell prints. (sec.)}\)

25-32 TWT \(\text{Intervals of problem time between tape dumps. (sec.)}\)

33-40 TWFIN \(\text{Problem finish time. (sec after start)}\)

41-42 LPR \(\text{Printing control for cell prints and plots on printer set 3. Other options can be incorporated}\)

43-50 THETA \(\Theta\) (Time centering of calculation)

51-58 PHI \(\phi\) (Time centering of calculation)
Card 5
Format: (515, 2F8.3)

<table>
<thead>
<tr>
<th>Cols.</th>
<th>TYPE</th>
<th>0 = no inflow, outflow, 1 = inlet and outlet</th>
</tr>
</thead>
<tbody>
<tr>
<td>1-5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>6-10</td>
<td>L1</td>
<td>Dimensions shown in Fig.E-1. These are given in integer numbers of cells to emphasize that all points must coincide with cell boundaries. Number of particles per cell to be created at the inflow boundary, in the r and z directions.</td>
</tr>
<tr>
<td>11-15</td>
<td>L2</td>
<td></td>
</tr>
<tr>
<td>16-20</td>
<td>L3</td>
<td></td>
</tr>
<tr>
<td>21-25</td>
<td>L4</td>
<td></td>
</tr>
<tr>
<td>41-45</td>
<td>NXB</td>
<td></td>
</tr>
<tr>
<td>46-50</td>
<td>NYB</td>
<td></td>
</tr>
<tr>
<td>51-58</td>
<td>UL</td>
<td>Prescribed inflow velocity. (in./sec.)</td>
</tr>
<tr>
<td>59-65</td>
<td>UR</td>
<td>Prescribed outflow velocity. Outflow becomes continuative if UR = 0.0. (in./sec.)</td>
</tr>
</tbody>
</table>
FIGURE E-1. Specification of Inlet and Outlet
Card 6

Format: *(215, 8F8.3, F5.2)*

<table>
<thead>
<tr>
<th>Columns</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1-5</td>
<td>(NX) = Number of particles/cell to be created for this part in the (r) and (z) direction</td>
</tr>
<tr>
<td>6-10</td>
<td>(NY) = Part dimensions (in.) See Fig. E-2</td>
</tr>
<tr>
<td>11-18</td>
<td>(XO) =</td>
</tr>
<tr>
<td>19-26</td>
<td>(YG) =</td>
</tr>
<tr>
<td>27-34</td>
<td>(XD) =</td>
</tr>
<tr>
<td>35-42</td>
<td>(XD) =</td>
</tr>
<tr>
<td>43-50</td>
<td>(U_o) = Initial velocity components of the part</td>
</tr>
<tr>
<td>51-58</td>
<td>(V_o) = These velocities will be applied to those cells the part initially occupies. (in./sec.)</td>
</tr>
<tr>
<td>59-66</td>
<td>\text{ENERGY} = The initial internal energy for that part of the mesh BTU/IBM</td>
</tr>
<tr>
<td>67-74</td>
<td>\text{PRESS} = Initial pressure in this part of mesh LBF/in^2</td>
</tr>
<tr>
<td>75-80</td>
<td>\text{WTMOL} = Molecular weight of fluid</td>
</tr>
</tbody>
</table>

This card is repeated as many times as needed to describe the mesh initial conditions.
FIGURE E-2. Specification of fluid sections
Card 7 consists of a card with a zero in columns 1-5. This card signifies the end of the mesh description 6 cards. NY should be set to 1 for minimum printout information. PRESS should be repeated from card 6.

Card 8

Cols.

<table>
<thead>
<tr>
<th>Col</th>
<th>Name</th>
<th>Format: (2F10.4,15, 2F10.4)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1-10</td>
<td>QIN</td>
<td>Heat load at the evaporator (BTU/HR)</td>
</tr>
<tr>
<td>11-20</td>
<td>EINTRL</td>
<td>Inlet enthalpy (BTU/Lbm)</td>
</tr>
<tr>
<td>21-25</td>
<td>JFLUID</td>
<td>Number of fluid being used in the problem as shown in table below:</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>JFLUID</th>
<th>MATERIAL</th>
<th>JFLUID</th>
<th>MATERIAL</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>LITHIUM</td>
<td>7</td>
<td>NAK-78</td>
</tr>
<tr>
<td>2</td>
<td>SODIUM</td>
<td>8</td>
<td>STRONTIUM</td>
</tr>
<tr>
<td>3</td>
<td>POTASSIUM</td>
<td>9</td>
<td>SILVER</td>
</tr>
<tr>
<td>4</td>
<td>RUBIDIUM</td>
<td>10</td>
<td>BERYLLIUM FLORIDE</td>
</tr>
<tr>
<td>5</td>
<td>CESIUM</td>
<td>11</td>
<td>LITHIUM FLUORIDE</td>
</tr>
<tr>
<td>6</td>
<td>MERCURY</td>
<td>12</td>
<td>WATER</td>
</tr>
</tbody>
</table>

Card 9

Cols.

<table>
<thead>
<tr>
<th>Col</th>
<th>Name</th>
<th>Format: (2F10.4, I5, 2F10.4)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1-10</td>
<td>TEB</td>
<td>Temperature (°F) of the evaporator entrance at the liquid vapor interface</td>
</tr>
<tr>
<td>11-20</td>
<td>TEE</td>
<td>Temperature (°F) of the evaporator exit at the liquid vapor interface</td>
</tr>
<tr>
<td>21-25</td>
<td>KONTER</td>
<td>1 or 0 If = 1 the calculation is compressible and if 0 the calculation is incompressible. If, during the cal-</td>
</tr>
</tbody>
</table>
In calculation, it is determined that the density is constant throughout the mesh, then the value of KONTER will be set to 0.

<table>
<thead>
<tr>
<th>26-35</th>
<th>TCB</th>
<th>=</th>
<th>Temperature (°F) of the condenser entrance at the liquid vapor interface</th>
</tr>
</thead>
<tbody>
<tr>
<td>36-45</td>
<td>TCE</td>
<td>=</td>
<td>Temperature (°F) of the condenser exit at the liquid vapor interface</td>
</tr>
</tbody>
</table>

**Card 10**

<table>
<thead>
<tr>
<th>Cols.</th>
<th>Format:</th>
<th>(2F10.4, 15, 2F10.4)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1-10</td>
<td>RW</td>
<td>The inner wall radius (this is larger than the vapor radius) (in.)</td>
</tr>
<tr>
<td>11-20</td>
<td>EP</td>
<td>Porosity of wick or the fraction of the wick which is liquid.</td>
</tr>
<tr>
<td>21-25</td>
<td>ID</td>
<td>Not used</td>
</tr>
<tr>
<td>26-35</td>
<td>WKMR</td>
<td>Wick Minimum pore radius (in.)</td>
</tr>
<tr>
<td>36-45</td>
<td>PERMEA</td>
<td>Permeability (in.²)</td>
</tr>
</tbody>
</table>

**Card 1 or 11**

<table>
<thead>
<tr>
<th>Cols.</th>
<th>Format (2I5, 3F8.3, I5, 2F8.3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1-5</td>
<td>IBAR</td>
</tr>
</tbody>
</table>

Start of a new problem if $> 0$; Signifies end of problem if $< 0$. 
REFERENCES


42. B.W. Knight and B. B. McInteer, "Laminar Incompressible Flow in Channels with Porous Walls," LADC-5039, 1965, Los Alamos Scientific Laboratory, Los Alamos, N.M.


