# TWO-PHASE FLOW IN PARTIALLY SATURATED TUFF

K. Thomas Feldman, Jr.
Manager, Engineering Analysis Department
Duane A. Labreche
Project Engineer
RE/SPEC Inc.
Albuquerque, New Mexico

## ABSTRACT

A study was conducted to determine the relative importance of conduction heat transfer and two-phase flow resulting from heating of partially saturated tuff. To achieve this, a conceptual model of the heat transfer processes occurring near the boiling interface was developed and analyzed. In this model, heat from the nuclear waste causes the tuff temperature to increase until the water in the porous tuff boils. Hot vapor flows away from the boiling interface - some into the cooler tuff and some back toward the heat source. The hot vapor will condense in the cooler regions of the tuff and condensate liquid will return to the boiling region by capillary surface-tension forces. The processes of boiling, condensing and capillary wicking in a porous media are similar to the processes occurring in a heat pipe. Analysis that has been developed by the author for heat pipes was used in this study. The results of the analysis indicate that the heat transfer rate by heat pipe vapor flow is less than 0.1 percent of the total heat transfer rate for permeabilities less than approximately $10^{-13}$ $m^2$. Thus, it is apparent that the heat pipe heat transfer effect in the tuff only becomes significant for permeabilities that are equal to or greater than $10^{-13}$ $m^2$ which was found to occur in the highly fractured tuff in the J-13 well. For higher permeabilities approaching $10^{-9}$ $m^2$, corresponding to sand, the heat pipe heat transfer effect dominates, approaching 100% of the total heat transfer rate. The model of the heat pipe effect developed here is more detailed than the lumped parameter models typically being used since the combined conduction and two-phase flow processes occurring in the partially saturated tuff are modeled explicitly. Yet this heat pipe model is simple and does not require much computer time. Thus, it provides an improved model which can be adapted into larger finite element codes used to analyze repository problems.

## INTRODUCTION

The site currently being considered for a high level nuclear waste repository is located 300 meters below the surface in volcanic tuff at Yucca Mountain, Nevada. The tuff is porous and partially saturated with ground water. The heat transfer process from the hot nuclear waste to the tuff is complex because both conduction and boiling heat transfer occur. In order to reduce this complexity, analyses of the heat transfer processes typically use a simplified boiling model. However, because heat transfer rates from boiling and two-phase flow can be many times greater than that from conduction, it is important to determine the conditions under which boiling heat transfer and two-phase flow are significant.

The objective of this study was to determine the relative importance of conduction heat transfer and two-phase flow resulting from heating of partially saturated tuff. To achieve this, a conceptual model of the heat transfer processes occurring near the boiling interface was developed and is depicted in Fig. 1. In this model, heat from the nuclear waste causes the tuff temperature to increase until the water in the porous tuff boils. Hot vapor flows away from the boiling interface - some into the cooler tuff and some back toward the heat source. The hot vapor will condense in the cooler regions of the tuff and condensate liquid will return to the boiling region by capillary surface-tension forces. The processes of boiling, condensing and capillary wicking in a porous media are similar to the processes occurring in a heat pipe. Analyses that were developed for heat pipes (1) were used in this study of two-phase flow phenomena in partially saturated tuff.

The properties used in this analysis were taken from the Yucca Mountain Project Reference Information Base and the Site Characterization Plan (2) and are summarized in Table I.

## DEVELOPMENT OF THE MODEL

A one-dimensional conceptual model of the two-phase heat transfer problem was developed with the boiling interface initially stationary. This model was assumed to encompasses a volume of porous tuff that extends from the dry region near the heat source out into the partially saturated tuff. Steady state heat transfer rates, and liquid and vapor flow rates were assumed to occur in this region. Part of the vapor was assumed to flow toward the heat source and be lost, and the remainder to flow back into the cooler tuff to condense and recirculate. Based on assumed temperature gradients between the hot waste and the tuff, conduction and boiling heat transfer rates were calculated along with liquid and vapor flow rates.

The model was formulated by considering a heat balance at the boiling interface. By assuming steady state heat transfer processes, there are no transient effects and heat storage. Heat is transferred by conduction from the heat

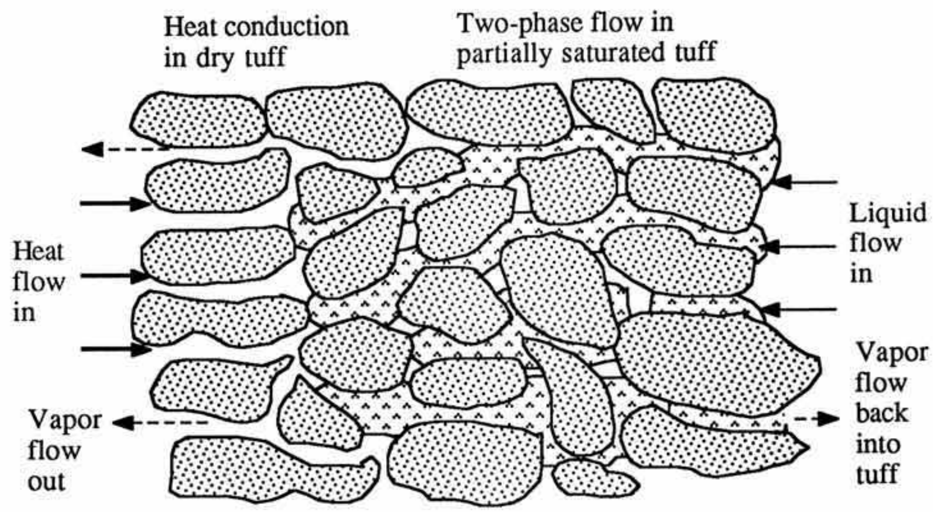


Fig. 1. Heat transfer and two-phase flow in porous media.

source through a layer of dry tuff to the boiling interface. The heat conduction rate into the tuff was determined from the assumed temperature gradient, based on earlier boiling isotherm work (5), and then proportioned according to the thermal resistances of the parallel conduction and boiling heat flow paths. The mass flow rate of vapor was determined from the heat flow rate of vapor by applying Darcy's Law (provides flow resistance) and the Clapeyron Equation (relates temperature and pressure differences).

## Steady State Heat Transfer Processes

Consider the conceptual model shown in Fig. 2 for this analysis. In Fig. 2(a) a schematic drawing of the physical system depicted in Fig. 1 is shown. In Fig. 2(b) the corresponding schematic mass flow diagram is shown, with vapor mass flow rate plotted as a function of length along the heat pipe. In Fig. 2(c) the corresponding schematic graph of temperature as a function of length is shown. In Fig. 2(d) the corresponding vapor pressure as a function of length is

**TABLE I**

**Thermophysical Properties of Tuff**

| Property | Range of Values | Rib Section |
|---|---|---|
| Porosity | 0.121 +/- 0.036 | 1.2.1.3 |
| Density | 2.550 +/- 0.032 kg/cu m | 1.2.1.3 |
| Thermal Conductivity (in situ saturation) | 1.910 +/- 0.083 W/mK | 1.2.2.4 |
| (dry) | 1.839 +/- 0.064 W/mK | 1.2.2.4 |
| Permeability (fractured TSw) | $8 \times 10^{-13}$ $m^2$ | DOE, 1988 (2) |
| (matrix TSw) | $1.93 \times 10^{-18}$ $m^2$ | DOE, 1988 (2) |
| Pore Radius | 0.0030 to 0.35 m | Klavetter, 1987 (3) |
| In Situ Saturation | 65 +/- 19 % | Montazer, 1984 (4) |

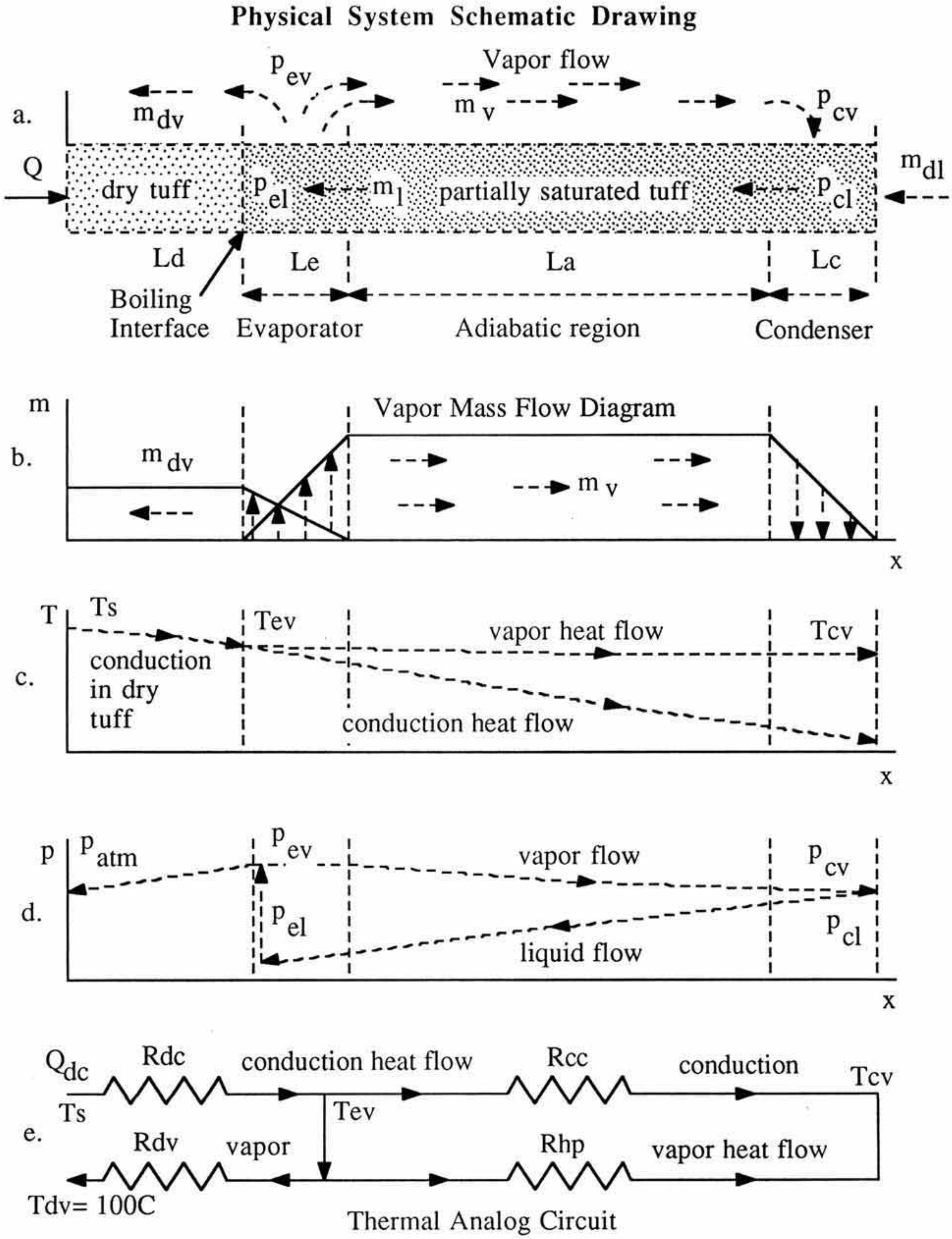


**Fig. 2. Heat flow schematics.**

shown, and in Fig. 2(e) the corresponding thermal resistance diagram for the heat transfer processes is shown.

Heat is transferred by conduction from the heat source through a length of dry tuff, $L_d$. This conduction heat transfer rate, $Q_{dc}$, is given by,

$$Q_{dc} = (T_s - T_{ev})/R_{dc} \qquad \text{(Eq. 1)}$$

where

$T_s$ = heat source temperature, and $T_{ev}$ = temperature at a distance $L_d$ into the dry tuff where the boiling interface occurs, $R_{dc}$ is the conduction thermal resistance (°K/Watt), given by,

$$R_{dc} = \frac{L_d}{k_{c\,dry} A_{tuff}} \qquad \text{(Eq. 2)}$$

where $k_{c\,dry}$ = thermal conductivity of the dry tuff, and $A_{tuff}$ = cross sectional area of tuff normal to the heat flow (a unit area is assumed, $A_{tuff} = 1\ m^2$).

The conduction heat transferred rate, $Q_{dc}$, is the heat supply for the boiling process. Thus, the total combined heat transfer rate by boiling and two-phase flow, $Q_t$, cannot exceed the conduction heat transfer rate, $Q_{dc}$.

At the boiling interface, the incoming conduction heat transfer rate $Q_{dc}$ is divided into two parts, part is transferred through the tuff by conduction, $Q_{cc}$, and part is transferred by boiling and two phase flow (heat pipe action), $Q_{hp}$. The total heat transfer rate through the heat pipe region is $Q_t$ and for this model $Q_t = Q_{dc}$.

$$Q_t = Q_{hp} + Q_{cc} \qquad \text{(Eq. 3)}$$

The region where the boiling occurs is called the evaporator. The temperature and vapor pressure in this region are given by $T_{ev}$ and $p_{ev}$ respectively. The temperature and pressure in the adjacent liquid phase are $T_{el}$ and $p_{el}$. The corresponding values in the condenser region are $T_{cv}$, $p_{cv}$, $T_{cl}$ and $p_{cl}$.

Eq. (3) may be written,

$$Q_t = (T_{ev} - T_{cv})/R_t + (T_{ev} - T_{dv})/R_{dvt} \qquad \text{(Eq. 4)}$$

In the heat pipe portion of the model, $R_t$ is the total thermal resistance to heat flow from the evaporator to the condenser.

$$R_t = \frac{1}{1/R_{hp} + 1/R_{cc}} \qquad \text{(Eq. 5)}$$

The heat pipe thermal resistance, $R_{hp}$, has three components,

$$R_{hp} = R_e + R_c + R_v, \qquad \text{(Eq. 6)}$$

where,

$R_e$ = the evaporator thermal resistance, (°K/Watt),
= $1/(h_e A_e)$, where $h_e$ is the evaporator heat transfer coefficient (estimated from experimental data) and $A_e$ is the cross-sectional area normal to the heat flow (taken to be $1\ m^2$),

$R_c$ = the condenser thermal resistance,
= $1/(h_c A_c)$, where $h_c$ is the condenser heat transfer coefficient (estimated from experimental data) and $A_c$ is the cross sectional area normal to the heat flow ($1\ m^2$),

$R_v$ = the thermal resistance for the vapor flow to the condenser (defined later),

$R_{cc}$ = the thermal resistance for the conduction heat transfer in the tuff along the length of the heat pipe,
= $L_{hp}/k_c A_{tuff}$,

$R_{dv}$ = the thermal resistance of the vapor flow through the dry tuff back to the heat source,

$T_{dv}$ = the vapor temperature corresponding to the vapor pressure at the heat source.

In the dry tuff portion of the model, the total thermal resistance of the vapor flow through the dry tuff toward the heat source is $R_{dvt}$ and is defined as,

$$R_{dvt} = R_{dv} + R_e$$

The second term, $R_e$, is the resistance for evaporation, as noted above. The general expression for thermal resistance to vapor heat flow is

$$R_v = \frac{\Delta T_v}{Q} \qquad \text{(Eq. 7)}$$

The temperature difference in the vapor flow, $\Delta T_v$, may be expressed in terms of the pressure difference, $\Delta p_v$, by using the Clapeyron equation.

$$\Delta T_v = \frac{T_v}{\rho_v h_{fg}} \Delta P_v \qquad \text{(Eq. 8)}$$

Here $T_v$ and $\rho_v$ are the mean temperature and density of the vapor, respectively, and $h_{fg}$ is the latent heat of vaporization. Introducing the Clapeyron Eq. (8) into Eq. (7) gives

$$R_v = \frac{T_v \Delta \rho_v}{Q \rho_v h_{fg}} \qquad \text{(Eq. 9)}$$

The pressure difference, $\Delta\rho_v$, can be related to the vapor mass flow by Darcy's law,

$$\Delta\rho_v = m_v R_{vf} \qquad \text{(Eq. 10)}$$

where $R_{vf}$, the vapor flow resistance in Darcy's Law, may be written (1),

$$R_{vf} = \frac{\mu_v L_{hp}}{\rho_v A_v K_v} \qquad \text{(Eq. 11)}$$

Here $L_{hp}$ is the length of the heat pipe section, $\mu_v$ and $\rho_v$ are the vapor viscosity and density respectively, $A_v$ is the cross sectional area to the vapor flow and $k_v$ is the permeability of the porous media. For the evaporation process, $Q = m_v h_{fg}$, so the vapor flow thermal resistance $R_v$ may now be written,

$$R_v = \frac{T_v R_{vf}}{\rho_v h_{fg}^{\ 2}} \qquad \text{(Eq. 12)}$$

Eqs. (11) and (12) give the thermal resistance to vapor flow from the evaporator to the condenser. Similar equations may be used with the length term $L_d$ to calculate the thermal resistance from the evaporator to the heat source. Using an assumed temperature gradient, based on earlier boiling isotherm work (5), the heat transfer rate can be calculated.

## SUMMARY OF RESULTS

The performance of the heat pipe system described by Eqs. (1) through (12) was determined for the range of permeabilities of interest. The heat transfer rate of the heat pipe vapor flow, $Q_{hp}$, was determined using the resistance in Eq. (6) and the total heat transfer rate in the heat pipe region, $Q_t$, (conduction plus heat pipe vapor flow) was calculated from Eq. (4). The results are shown in Fig. 3 as a ratio of $Q_{hp}/Q_t$ (in percent). The permeabilities of interest range from 1.9 x $10^{-18}$ $m^2$ for unfractured matrix TSw tuff, 3.2 x $10^{-16}$ for fractured TSw, 8.2 x $10^{-13}$ for fractured TSw from the J-13 well (2), to as high as $10^{-9}$ $m^2$ for sand (6).

The data shown in Fig. 3 indicate that the heat transfer rate by heat pipe vapor flow is less than 0.1 percent of the total heat transfer rate for permeabilities less than approximately $10^{-13}$ $m^2$. Thus, it is apparent that the heat pipe heat transfer effect in the tuff only becomes significant for permeabilities that are equal to or greater than $10^{-13}$ $m^2$. This permeability was found to occur in the highly fractured tuff in the J-13 well. For higher permeabilities approaching $10^{-9}$ $m^2$, corresponding to sand, the heat pipe heat transfer effect approaches 100% of the total heat transfer rate.

The mass flow rate of heat pipe liquid was also analyzed. No mass flow of vapor occurs when $T_s \leq 100$°C but as $T_s$ increases, the mass flow rate increases to a substantial value of 1376 kg/yr at $T_s$ = 200°C. A complete discussion of these calculations is given by Feldman (7).

The maximum velocity of the boiling interface velocity of heat pipe vapor was also analyzed. No mass flow of vapor occurs when $T_s \leq 100$°C, but as $T_s$ increases,the vapor velocity increases to a substantial value of 20 m/yr at $T_s$ = 200°C (7).

## CONCLUSIONS AND RECOMMENDATIONS

The results indicate that the two-phase flow and recirculation of condensate liquid through the porous partially saturated tuff, which is here called the heat pipe effect, is significant for the cases where the tuff is highly fractured or faulted. Here the heat pipe effect can be the dominate heat transfer process. For example, in a region where the permeability is that of sand ($10^{-9}$ $m^2$), the heat pipe effect is nearly nine times larger than the conduction heat transfer rate. The permeability of the tuff is by far the dominate variable in determining the magnitude of the heat pipe effect. In the unfractured matrix tuff where the permeability is very low ($10^{-18}$ $m^2$) the heat pipe effect is insignificant.

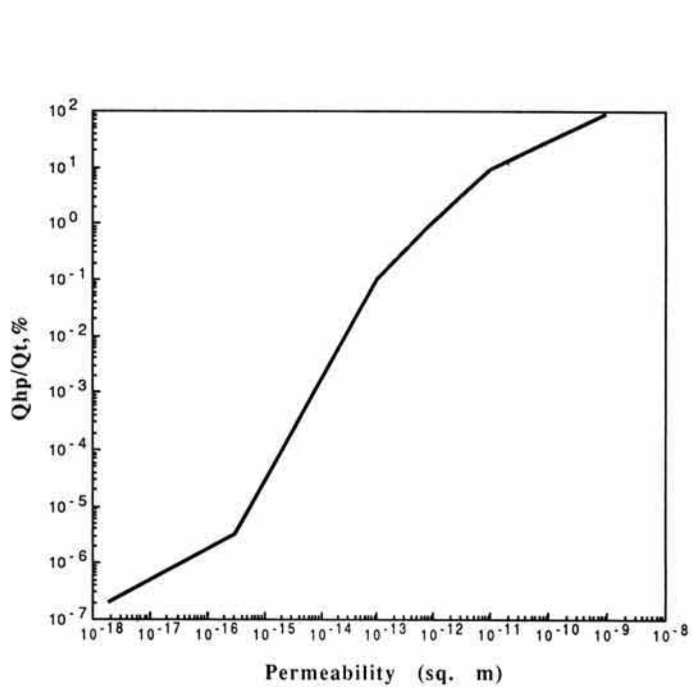


Fig. 3. Qhp/Qt vs. permeability.

The mass flow rate of vapor and liquid in the heat pipe two-phase flow process is directly proportional to the boiling heat transfer rate. Also, the velocity of the boiling interface is proportional to the boiling heat transfer rate. For regions of tuff with permeabilities on the order of $10^{-9}$ $m^2$ the boiling interface can have a velocity as high as 20 m/year.

The model of the heat pipe effect developed here is more detailed than the lumped parameter models currently being used since the combined conduction and two-phase flow processes occuring in the partially saturated tuff are modeled explicitly. This heat pipe model is simple, yet an improvement over the simplified boiling models, and does not require much computer time.

The logical next step for analysis is to review the size and density of fractures and faults known to occur in the repository tuff and determine the corresponding permeability of these regions. Then using this permeability data, the relative significance of the heat pipe effect in the repository can be determined.

Because of the potential dominance of the heat pipe effect in high permeability regions, this heat pipe model should be considered for incorporation into the finite element codes used to model the thermal response of the repository. Then a reexamination of the overall heat transfer process and transient temperature distribution in the repository should be considered using the new model.

## REFERENCES


1. K. T. FELDMAN, JR., Heat Pipe Technology, Technology Information Center, The University of New Mexico, Albuquerque, New Mexico (1976).
2. DOE, Site Characterization Plan, ORG/B-2, Department of Energy, Washington, D.C. (December 1988).
3. E.A. KLAVETTER and R.R. PETERS, An Evaluation of the Use of Mercury Porosimetry in Calculating Hydrologic Properties of the Tuff from Yucca Mountain, Nevada, Sandia National Laboratories, Albuquerque (March 1987).
4. P. MONTAZER and W.E. WILSON, Conceptual Hydrological Model of Flow in the Unsaturated Zone, Yucca Mountain, Nevada, United States Geological Survey Water Resources Investigation Report 84-4345, Lakewood, Colorado (1984).
5. E.S. HERTEL, JR. and S.V. PETNEY, Determination of the Location of the Boiling Isotherm for Ventilation System Studies, (PDM 78-018), TLM RSI(ALO)-0041, SLTR 89-4001 (January 25, 1989).
6. R. A. FREEZE and J. A. CHERRY, Groundwater, Prentice-Hall, Inc., Englewood Cliffs, New Jersey (1979).
7. K. T. FELDMAN, JR., Scoping Studies to Investigate Two-Phase Flow Phenomena in Partially Saturated Tuff, PDM 75-21, for Sandia National Laboratories (November 1989).