THE EFFECTS OF A NON-CONDENSABLE GAS
ON PRESSURIZER INSURGE TRANSIENTS

by

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B.S. in Nuclear Engineering
Purdue University (1980)

Submitted to the Department of Nuclear Engineering
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at the

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December 1983

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Chairman, Nuclear Engineering Department Committee

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ABSTRACT

An analysis of the effects of the presence of a non-condensable gas
on insurge transients in a simulated pressurizer of a light pressurized
water reactor is presented. Experimental data and supporting analytical
model calculations are used to develop recommendations for modeling
the effects of non-condensables on wall heat transfer, which is the
significant physical phenomenon affected by the presence of the gas.
These data were taken at low pressure where the effect of wall
condensation heat transfer on the system pressure response is greatly
exaggerated from that of a full scale PWR pressurizer. Under these
conditions, insurge transients with a non-condensible gas present (in
relatively small quantities) result in peak pressures significantly
higher than similar transients without any gas. The presence of a gas
of molecular weight similar to water is observed to produce a more rapid
pressure rise than for gases of molecular weight much different from
water. This effect can be satisfactorily calculated by appropriate
modeling of the vapor-gas distribution within the pressure vessel
and modifying the wall heat transfer coefficient to account for the
presence of a non-condensable gas.

Thesis Supervisor: Dr. Peter Griffith
Thesis Reader: Dr. Neil Todreas
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<tr>
<td>A</td>
<td>vessel fluid cross sectional area</td>
</tr>
<tr>
<td>Bi</td>
<td>Biot number</td>
</tr>
<tr>
<td>g</td>
<td>gravitational acceleration</td>
</tr>
<tr>
<td>J</td>
<td>thermal - mechanical energy conversion factor</td>
</tr>
<tr>
<td>h</td>
<td>average wall heat transfer coefficient</td>
</tr>
<tr>
<td>H</td>
<td>total height of pressurizer</td>
</tr>
<tr>
<td>i</td>
<td>enthalpy</td>
</tr>
<tr>
<td>k</td>
<td>thermal conductivity</td>
</tr>
<tr>
<td>L</td>
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<tr>
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<tr>
<td>P</td>
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<td>heat transfer rate</td>
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<tr>
<td>( w )</td>
<td>mass flow rate</td>
</tr>
<tr>
<td>( x )</td>
<td>distance (through vessel wall)</td>
</tr>
<tr>
<td>( X )</td>
<td>non-condensable gas mass fraction</td>
</tr>
<tr>
<td>( z )</td>
<td>elevation above the bottom of the pressurizer</td>
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Greek symbols:

- \( \alpha \) thermal diffusivity
- \( \gamma \) gravitational potential
- \( \lambda \) eigenvalue
- \( \mu \) viscosity
- \( \mu_c \) chemical potential
- \( \rho \) mass density
- \( \theta \) non-dimensionalized time
- \( \Gamma \) condensate film mass flow rate per unit width
- \( \nabla \) del operator

Subscripts:

- \( f \) liquid (saturated)
- \( fg \) difference between saturated liquid and vapor values
- \( g \) vapor (saturated)
- \( i \) component-i
- \( nc \) non-condensable gas
- \( o \) initial value
- \( r \) reduced
- \( w \) wall
- \( \infty \) bulk
1. INTRODUCTION

The pressurizer response to pressurized water reactor (PWR) transients, not involving a rupture in the primary coolant system (PCS) plays an important role in determining the pressure history of the PCS. Since the pressure history can determine the timing of safety system responses, it is important to develop an accurate model of the pressurizer for computer simulations of postulated transients. Such a model must adequately represent the significant physical phenomena.

Existing mathematical models of pressurizer systems have been principally derived from the general forms of the governing conservation equations, retaining as much detail as deemed necessary for the transients under consideration. Lack of a complete and consistent data base has hindered development of a truly "universal" model of pressurizer transients. The data used for model verification has typically been that of full scale power systems which have undergone operational transients of one form or another. Although this type of data has the distinct advantage of being truly "typical" in the sense of scaling and operating conditions, it suffers from a lack of adequate data measurement to completely characterize the physical phenomena which were taking place. A problem, therefore, arises in evaluating a pressurizer model due to the possibility of over-looking an important effect which has yet to appear in the scattered data available.

One specific topic which has attracted no attention in the analytical or experimental investigations of pressurizer transients is the effect of non-condensable gases on insurge transients. Several types of non-condensable gas exist in the PCS as a part of standard water chemistry procedures. Although it is recognized that the
relative concentrations of such gases are small it is not known how much gas in the vapor space of the pressurizer would significantly affect the transient response of the pressurizer.

It has been repeatedly documented that the presence of a very small amount of non-condensable gas in steam dramatically reduces heat transfer to a cold condensing wall. In transients where wall condensation in the pressurizer is a significant physical phenomena, it could, therefore, be expected that the presence of even a small amount of gas in the pressurizer could significantly affect the response of the system.

The purpose of this investigation is to experimentally determine the magnitude of non-condensable gas concentration required to significantly alter the pressure-rise response of a pressurizer system during an insurge transient. These experiments will be performed under conditions which exaggerate the influence of wall condensation on the pressure history of a pressurized vessel (low pressure and high surface area to volume ratio) relative to those of a full scale PWR pressurizer. Gases of molecular weights similar to and different from that of water will be analysed and a recommended method of calculating the pressure response in the presence of a non-condensable gas will be presented.
2. HISTORICAL PERSPECTIVE

Early pressurizer transient models were developed usually assuming either isentropic compression of the vapor or continuous thermal equilibrium between the vapor and liquid phases. Under certain conditions, (e.g. slow surge flow rate or low pressure) both of these assumptions have been shown to yield erroneous results [1,2,3]. Deviations from existing data were attributed to the neglect of vapor condensation on the vessel walls. Several more mechanistic models were then developed based upon considering the pressurizer as a lumped parameter thermodynamic system [3,4,5,6]. This approach allows the liquid and vapor regions to be modeled separately, each including the effects of non-equilibrium. In each of these analyses, heat loss from the vapor, due to condensation heat transfer to the vessel walls, was modeled. The analysis by Drucker and Gorman [5], and a recent one presented by Baggoura and Martin [7], model the wall heat transfer by assuming the inner wall surface to always be at the saturation temperature corresponding to the system pressure, and the outer wall adiabatic. The wall energy content (and, thus, the change in vapor energy content) is predicted by integrating the wall temperature distribution.

The TOPS pressurizer model, developed at Westinghouse [6], included the effects of wall condensation by applying Nusselt laminar film theory to estimate a wall heat transfer coefficient; the vessel wall temperature was, therefore, calculated to be different from the system saturation temperature. Saedi [3] recently investigated the relative magnitudes of the physical phenomena associated with surge transients, and initiated a data base (at low pressure) for a model, developed by Kim [8], at MIT. Saedi also applied Nusselt theory in calculating the
effect of including wall condensation in a simple thermodynamic pressurizer model with good results when compared with his data.

None of the above models include the effects of a non-condensable gas, primarily due to a lack of data with which comparisons could be made. Included in this analysis is a series of experiments (at low pressure) which insurge relatively cold water into a pressurized vessel with a known initial non-condensable gas mass fraction. A modification to a general pressure vessel model is then recommended which consists of a modified Nusselt heat transfer coefficient to account for the reduced partial pressure of vapor at the condensate interface in the presence of a non-condensable gas.
3.0 THEORY

The practical application of this study hinges on the likelihood of existence of non-condensable gases in the pressurizer of a PWR. The fact that the pressurizer is both the highest elevation in the system and at saturation conditions (with the balance of the primary coolant subcooled) ensures that non-condensables, if present, will collect in the pressurizer. The source of introduction of non-condensables into the pressurizer during steady state operation is a continuous spray which passes primary coolant through the pressurizer at approximately 1-10 gpm (depending on the specific plant considered). The source of the continuous spray can be either a coolant loop cold leg, or the makeup supply tank.

As the spray droplets fall through the vapor space of the pressurizer they increase in temperature, forcing dissolved gas to be stripped from the water. This process has been analyzed experimentally and analytically and summarized by Cohen [10]. A rough estimate of the volume of non-condensable gas stripped from the continuous spray, for a given the spray flow rate and inlet gas concentration, can be made. Assuming 100% of a dissolved gas in the spray is driven out of solution in the pressurizer, a volume fraction of approximately 3% can develop over a year's continuous operation (for a spray coolant flow rate of 1.5 gpm and gas concentration of 20 Scc/kg). Actual gas stripping efficiencies have been cited as being 82% [10].

Two specific gases of practical concern are hydrogen and nitrogen. Hydrogen gas is dissolved in the makeup supply tank of PWRs which continuously supplies coolant to the PCS during normal operation. The purpose of adding hydrogen is to maintain a high enough concentration
in the primary coolant to eliminate, by recombination, free oxygen formed from hydrolysis in the core. Oxygen is undesirable for material considerations, namely corrosion. The potential for nitrogen gas in the PCS exists by considering transient or accident conditions. If an event were to take place involving the actuation of an accumulator (an emergency core coolant injection tank), pressurized with nitrogen, a large amount of nitrogen would be added to the system when the accumulator emptied. In this case, the gas injection location is in the cold leg, however, over time, some of this gas will be transported to the pressurizer.

The actual amount of non-condensable gases in the pressurizer may be small; however, the experimental results of this study indicate that only small amounts of certain types of gases are required to significantly alter the response of the pressurizer to insurge transients. The experimental results are limited to transients during which condensation heat transfer is an important phenomenon. The effects of the presence of a non-condensable gas on condensation heat transfer are summarized below.

The physical process for vertical, film-wise condensation heat transfer involves the motion of vapor at a temperature \( T_\infty \) toward a surface at \( T_w < T_\infty \). The latent heat of evaporation is transferred to the wall as the vapor condenses and the liquid condensate flows down the wall in a laminar film (for the range of operating parameters considered here). Even for small driving potentials \( (T_\infty - T_w) \), the rate of heat transfer can be significant because of the very large heat transfer coefficients associated with this process (typically of the order of \( 10^3 \text{ Btu/hr-ft}^2\text{-R} \)). Ignoring the slight subcooling of the liquid film due
to the temperature gradient across it, the rate of heat transfer and condensate mass flow rate are related by a simple energy balance:

\[ Q = \bar{h} \ A \ (T_\infty - T_w) = \int P \]

where \( P \) is the perimeter of a cylindrical vessel, and \( \Gamma \) the film mass flow rate per unit width. The average heat transfer coefficient may be evaluated using the result originally derived by Nusselt [11],

\[ \bar{h} = 0.943 \left[ \frac{\rho_f g (\rho - \rho_f) i}{L \mu_f (T_\infty - T_w)} \frac{k}{fg} \right]^{3/4} \]

A summary of the boundary layer analysis culminating in Nusselt's result, with corrections for vapor superheat, film subcooling, and interfacial shear is presented in many standard textbooks [12].

Early experimental work in steam condensation by Othmer [13], demonstrated a significant reduction in heat transfer due to the presence of a very small amount of air. Othmer developed an empirical equation to correlate his results which was useful for applications, but provided no insight as to the phenomenon at work. A theoretical approach by Sparrow and Lin [14], describes the phenomenon as a simultaneous heat and mass transfer process. Their theory is based on the conservation laws of the laminar film and vapor-gas mixture (each considered as a boundary layer) and satisfactorily predicts the results of Othmer.

The effect of a non-condensable gas may be summarized as follows. Consider a "cold" surface brought into contact with a homogeneous mixture of vapor and non-condensable gas. When vapor condenses on the surface, forming the liquid film, the gas is left behind at the inter-
face. This increases the local concentration of the gas at the interface (therefore, decreasing the vapor concentration) and a concentration gradient for mass diffusion is initiated. The vapor-gas mixture diffuses toward the interface and the gas diffuses away from the interface. For a given bulk non-condensable gas concentration, the gas concentration at the interface increases until an equilibrium is attained where the gas diffusion rate away from the interface equals the rate at which gas is brought toward the interface by the diffusion of the vapor-gas mixture. Partial pressure may be substituted for concentration in the above process without loss of generality, and Figure 3.1 depicts the profiles of the vapor and gas partial pressures at the steady state condition. Assuming the local fluid temperatures are those corresponding to the saturation temperatures at the local partial pressure of the vapor, the effect of the presence of the non-condensable gas is readily apparent. The thermal driving force ($T_{\infty} - T_w$) is significantly reduced due to the decreased vapor pressure at the interface. In the absence of a non-condensable gas the interface and bulk temperatures are approximately the same. As verified in Othmer's experiments, heat transfer decreases with increasing bulk non-condensable gas concentration (decreasing interfacial vapor pressure).

The results of either Othmer or Sparrow could be used to evaluate, quantitatively, the decrease in condensation heat transfer due to the presence of a non-condensable gas. The data of Othmer [13], however, is presented as a function of non-condensable bulk volume fraction, whereas the analytical results of Sparrow are presented in terms of bulk mass fraction. For the reasons discussed later, the latter is preferred. For bulk fluid conditions consistent with those of the present experimental system (approximately 700°R) the ratio of the
Figure 3.1 - Vapor and non-condensable gas distribution near the vertical condensate film.

Figure 3.2 - Degradation of the wall heat transfer coefficient in the presence of a non-condensable gas.
actual heat transfer to the Nusselt film theory heat transfer (in the absence of non-condensables) is plotted as a function of the bulk non-condensible gas mass fraction in Figure 3.2. These values correspond to the results presented by Sparrow [15] for a driving potential of approximately 2°R. The decrease in heat transfer is significant (more than 50% with less than 1% by mass air) and is reasonably approximated by an exponential curve over the mass fraction range shown.

The application of these data to an analytical model of the pressurizer is discussed later when the overall thermodynamic model is presented. It is evident, however, that when wall heat transfer is an important physical phenomenon, significant changes in the pressure response of a pressurizer are anticipated by introducing a non-condensible gas.
4. EXPERIMENTAL PROGRAM

The experimental apparatus, data acquisition system, and basic transient procedures used in this analysis were identical to those of Saedi [3], and Kim [8]. Since the objective of this analysis centered around the effect of a non-condensable gas, the apparatus was slightly modified to allow for the introduction of the gas into the system and an accurate method of measuring the injected gas mass.

4.1 DESCRIPTION OF EXPERIMENTAL APPARATUS

A schematic diagram of the experimental apparatus is shown in Figure 4.1. The system consists of two stainless steel tanks, an surge line with flow control, service lines, a gas injection system and a data acquisition system. One of the steel tanks (primary tank) models the pressurizer volume and the other (storage tank) serves as a reservoir for "cold" injection water.

The primary tank is instrumented with thermocouples along the full length of the center of the fluid volume, a visual pressure gage, and a pressure transducer. Along with a pressure transducer in the surge line (measuring flow rate), the pressure and temperature signals from the primary tank are scanned and recorded at a frequency of approximately 3.3 Hz by a mini-computer. The primary tank is an 8 in. inner diameter pipe, 45 in. in height. It is heavily insulated to keep heat loss at a minimum, and equipped with an external gage glass to accurately measure the initial liquid level. The primary tank is also equipped with immersion heaters capable of approximately 9 kW total power.

The storage tank is filled with approximately 70°F water to a
FIGURE 4.1 - Schematic Diagram of Apparatus
level also measured with an external gage glass. This tank is pressurized with nitrogen and maintained at an approximately constant pressure during the transient. Since the pressure in the primary tank rises during an insurge, and flow control was provided by a control valve fixed in position during an experiment, the flow rate into the primary tank was not maintained precisely constant. The insurge rate was approximately 0.5 in. per second (level rise rate of the primary tank) for all of the transients simulated. The actual insurge flow rate was measured and provided as a boundary condition for subsequent calculations.

The non-condensable gas injection system consists of a small, thin-shelled storage tank (pressurized by, then isolated from a larger gas cylinder), and an injection line in the bottom of the primary tank. The top of the injection line in the primary tank is above the initial water level in the tank. The pressure (and, therefore, the mass) of the gas is measured in the gas storage tank.

A detailed listing of the experimental equipment and its characteristics is given in Appendix A. For details of the primary tank construction, the reader is refered to the thesis of Saedi [3].

4.2 EXPERIMENTAL PROCEDURES

Several precautionary procedures were followed prior to initiating each of the transient experiments. The primary and storage tanks were filled to the required levels with tap water. Steam was bubbled through the primary tank to assist the immersion heaters in bringing the tank up to saturation conditions. When the tank reached saturation, the steam line was isolated and the water allowed to boil (at 1 atm pressure)
with the vent line open for approximately 15 minutes. This process was intended to rid the system of dissolved gases in the water. The vent valve was then isolated and the primary tank brought up to the desired operating pressure of 77 psia. This pressure was chosen (in conjunction with the 70°F surge water) to scale the cold to hot water density ratio of the full scale system.

For the experiments with an initial non-condensable gas fraction in the vapor, the gas was injected into the vapor space after the tank had been purged of dissolved gases, and before the system had reached the desired operating pressure. The gas was bled into the vapor space from the small storage tank by slowly opening the connecting isolation valve. The storage tank had been previously pressurized with the gas and allowed to achieve thermal equilibrium with the environment. The desired amount of gas was injected by measuring the drop in the storage tank pressure. The final pressure in the storage tank was recorded after the tank had again achieved thermal equilibrium with the environment. The gas was bled into the system as slowly as possible to reduce local density gradients as much as possible. In the case of helium as a gas this procedure was thought to be insufficient and a special fan was built in the vapor space to forcibly mix the gas and steam, prior to initiating the surge. This special procedure is discussed in Section 5.3 in more detail. In all cases, the system was brought up to the desired operating pressure with the gas already introduced to the system.

When the tank reached the operating pressure, the immersion heaters were turned off and the system was allowed to equilibrate. The time constant for the steel vessel wall is approximately 60 s. Therefore,
several minutes were allowed to achieve a steady state condition. When
the system had stabilized, a check of environmental heat loss was made
by measuring the heater power required to maintain a constant system
pressure. This value was consistently found to be 1.3 kW. The system
was then brought to approximately 80 psia and the heaters turned off.
The transient would be initiated when the tank had returned to 77 psia.
This final step allowed enough time for the immersion heaters to
dissipate their remaining heat capacitance prior to initiating the
transient.

An insurge was initiated by opening the quick-opening valve at
the base of the pressurizer. The data acquisition system was initiated
prior to beginning the insurge to allow some steady state data to be
taken. This provided a measure of consistency in the initial conditions
among the experiments in a given series. An outline of the initial
conditions simulated is given in the following section.

4.3 EXPERIMENT INITIAL CONDITIONS

The position of the flow control valve and storage tank back-
pressure remained constant for all of the experiments performed. Two
types of insurge transients were simulated, distinguished only by the
initial water level in the pressurizer (primary tank). The first type
of insurge was initiated from an initial level of +17 in. (above the
bottom of the tank). This corresponds to a z/L of approximately 0.38.
The insurge was terminated when the water level reached +34 in.
(z/L = 0.76). This type of insurge is referred to in the following
discussions as a "partially full tank" experiment.

The second type of insurge was initiated from an initial level
near the bottom of the tank (these experiments are subsequently referred
to as "empty tank" insurges). These data were of interest based on the
qualitative results of Saedi, which showed dramatic reductions in
system pressure due to the cold surge liquid breaking through the
vapor/liquid interface, greatly increasing condensation heat transfer.
Based upon his experimental data, this phenomenon was significant only
when the initial liquid level in the tank was of the order of the inlet
sparger height [3]. This mechanism of heat transfer is discussed in
detail in the theses of Kim [8] and Kang [9]. Since the effect of a
non-condensable gas on condensation applies to heat transfer at a vapor-
liquid interface as well as at a wall, experiments were run for initial
levels of +1 in. and +2 in. to determine the effects of a gas on the
initial pressure reduction reported by Saedi. These "empty tank"
insurges were terminated when the pressurizer level reached +17 in.

A matrix of the experimental data taken and presented in this work
is given in Table 4.1. Three different gases were used, individually,
to observe the effect of molecular weight (relative to that of water).
For each gas, experiments with similar initial mass fractions in the
vapor space were performed. Mass fraction is used in the discussions
of experimental results to be consistent with other published works on
the effects of non-condensables on condensation heat transfer [13],
[14], [15], although mole fraction and volume fraction (stratified at
the initial pressure) are given in Table 4.1, as perhaps preferred by
others.
<table>
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<tr>
<th>EXPERIMENT IDENTIFICATION</th>
<th>INITIAL LEVEL (in.)</th>
<th>FINAL LEVEL (in.)</th>
<th>TYPE OF GAS USED</th>
<th>INITIAL MASS FRACTION (%)</th>
<th>INITIAL MOLE FRACTION (%)</th>
<th>INITIAL VOLUME FRACTION (%)</th>
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5. EXPERIMENTAL RESULTS

The most useful means of gleaning information from the data produced from this study is through analysis by comparison. That is, an individual pressure history curve offers little insight until it is compared (overlaid) with another set of data from an experiment which differs only by a single known initial condition. For this reason, data from the experiments listed in Table 4.1 are presented below primarily in a comparative format. For completeness, and the possible desire of the reader to use some data for further analysis, a tabulated form of data from some experiments is given in Appendix B.

5.1 PARTIALLY FULL TANK EXPERIMENTS

The overall trends of data for all of the experiments were very similar, differing only in the slope (and ultimate peak value) of the system pressure history. As a means of presenting the typical response of the system during an individual experiment, the data for the "base case" insurge transient are described below. The pressure histories of insurge transients with non-condensable gas present in the vapor space are then presented in comparison to this base case. It should be noted that the plotted pressure response of the base case described below depicts all data points recorded and is, therefore, more noisy than the subsequent comparative plots of pressure histories in which a smooth curve has been drawn through the data.

5.1.1 BASE CASE INSURGE RESPONSE

The recorded pressure history for the "base case" insurge transient (no non-condensable gas present - INS.BCR) is shown in Figure 5.1. An
almost identical response was measured in successive transients with the same boundary conditions to verify repeatability. The duration of the insurge was approximately 32s. On the time scale used in this figure, the pressure rise appears linear. The slope actually increases slightly after approximately 20s (this is shown more clearly on subsequent figures). The pressure reaches a peak value of approximately 94 psia then drops first sharply then gradually after the insurge is terminated. The measured insurge mass flow rate for this experiment is shown in Figure 5.2. The flow rate data for this experiment are typical (both in magnitude and duration) of flow rates for all subsequent experiments.

The measured axial temperature profile for the base case experiment is shown in Figure 5.3 at steady state and for three times during the transient. The corresponding elevation of the liquid level in the pressurizer is indicated at the right margin of the figure. The overall thermal response can be described as a parfait of three layers: a shrinking vapor volume, and a "hot" liquid layer which is lifted by a basically stratified "cold" insurge liquid layer. Very little mixing is indicated. The vapor temperature is shown to increase slightly as the system pressure increases and follows the corresponding saturation values very closely. The bulk liquid temperature on the other hand remains fairly constant at its initial value except at low elevations where the insurge water is highly subcooled. These results are consistent with the findings of Saedi and allow for significant simplification in modeling (Section 6.).

5.1.2 TRANSIENTS WITH NON-CONDENSABLES

As outlined in Table 4.1, three types of gas were used individually, each in varying mass fractions. The objective of using the
Figure 5.2 - Mass flow rate of water into the primary tank.
(Base Case Insurge Transient)
Figure 5.3 - Axial fluid temperatures during the Base Case Insurge.
gases chosen, helium (molecular weight, \( M = 4 \)), nitrogen (\( M = 28 \)), and argon (\( M = 40 \)), was to examine the effect of gases of molecular weight close to and much different from that of water (\( M = 18 \)).

The pressure histories for the experiments with helium, nitrogen, or argon gas present are shown in Figures 5.4, 5.5, and 5.6, respectively. The initial bulk non-condensible mass fraction for a given transient is labeled on each curve. The base case insurge, described above, is also plotted on each figure for comparison. Also indicated on each plot (as a reference maximum slope) is a calculated pressure history for adiabatic compression of the vapor (assuming constant properties over the range of pressures considered). The most obvious difference in the results, among the cases with different gases, is the large variation in peak pressure. For the same initial non-condensible gas mass fraction and insurge flow rate, the cases with nitrogen exhibit much higher peak pressures than those with either helium, or argon, especially those with helium.

The variation in pressure histories would appear even more dramatic if transients with equivalent initial gas volume fractions were compared. For example, referring to Table 4.1, experiments INS.NC9 (helium initially 53% by volume) and N2I.N11 (nitrogen initially 14% by volume), correspond to the largest mass fraction experiments in Figures 5.4 and 5.5, respectively. The peak pressure with nitrogen initially at 14% by volume is much larger than that for helium even at 53% by volume. Obviously, volume fraction is a very misleading standard of the effects of a non-condensible gas on the pressure history.

As was shown quantitatively in the Section 3.0, the difference
Figure 5.4 - Measured pressure response of insurge transients with helium present.
Figure 5.5 - Measured pressure response of insurge transients with nitrogen present.
Figure 5.6 - Measured pressure response of insurge transients with argon present.
in the condensation heat transfer coefficient with and without a non-condensable gas is so large, even for very small amounts of gas, that one may think of the latter as being about zero. It is, therefore, apparent that now the gas distributes itself within the vapor space becomes a key factor in predicting the degree of degradation of wall heat transfer. If the gas is completely stratified (at the top or bottom of the vapor space), a finite amount of the vessel wall is affected by the presence of the gas and the remaining portion, exposed to vapor only, is still open for significant heat transfer. It, on the other hand, the gas is evenly distributed throughout the vapor space, the entire length of the vessel wall (within the vapor space) is affected and subject to a significant reduction in heat transfer from the vapor may be anticipated.

In fact, it may be anticipated that the completely mixed condition would lead to a greater reduction in the magnitude of wall heat transfer than the stratified condition, for the following reasons. Two parameters directly affected by the presence of a non-condensable gas, are the wall heat transfer coefficient and the length of vessel wall available for heat transfer. The magnitude of wall heat transfer is directly proportional to both of these variables:

\[ Q \propto hL \]

Under the fully mixed assumption, the full length of the vessel wall (L) is available for heat transfer, but subject to a reduced condensation heat transfer coefficient \( (h_r) \). In the stratified condition, the wall length available for condensation heat transfer is reduced \( (L_r) \), but a very large heat transfer coefficient \( (h) \) is applicable over that length. As shown in Figure 3.2, the condensation heat transfer
coefficient decreases dramatically for very small amounts of gas, certainly more than the exposed wall area would be reduced if the gas were to stratify. In other words, the ratio

\[
\frac{Q \text{ (mixed)}}{Q \text{ (stratified)}} = \frac{h_r L}{h L_r} < 1 ;
\]

the completely mixed assumption is anticipated to yield lower wall heat transfer rates than the stratified assumption and lead to higher peak pressures for a given insurge.

The above discussion provides a plausible explanation for the results of the partially full tank experiments. If helium stratifies within the vapor space, but nitrogen (and to a lesser extent argon) remains well mixed with the vapor, the degradation of condensation heat transfer would be less significant with helium than either nitrogen or argon. An insurge transient with nitrogen would resemble an adiabatic compression more closely than with helium or argon, resulting in a higher peak pressure. An experimental method of determining the relative degree of stratification was, therefore, pursued to confirm this hypothesis as discussed below.

5.2 EMPTY TANK EXPERIMENTS

As mentioned in Section 4.3, insurge transients with very low initial levels are of interest because of the large initial pressure reductions due to rapid condensation on the cold insurge wake. Kang [9] experimentally determined the interface heat transfer coefficient as a function of insurge flow rate and initial level. For initial levels on the order of the height of the inlet sparger, the heat transfer coeff-
coefficients were found to be very large (greater than $10^3 \text{ Btu/hr-ft}^2\text{°R}$).

The large condensation rates (and associated decreases in pressure), typical of "empty tank" insurges also provide a useful environment for determining the relative stratification of helium and nitrogen in the vapor space. Since helium is lighter than vapor, if the vapor and helium stratify, the helium will occupy the uppermost volume of the pressurizer with the steam below it. The empty tank insurge response would then not be strongly affected by the presence of the gas. On the other hand, if a gas and vapor remain well mixed, interface condensation heat transfer will be degraded (through mechanisms similar to those of vertical condensation) and the initial pressure reduction suppressed.

As in the partially full tank experiments, a repeatable base case response was experimentally determined for two initial levels: 1 in. and 2 in. Several insurge experiments were then performed with different initial gas mass fractions of helium and nitrogen. The measured pressure histories over the insurge period (approximately 30 s in all cases) are shown in Figures 5.7 and 5.8 for initial levels of 1 in. and 2 in., respectively. The initial bulk mass fractions of gas are labeled for each case.

Two results are immediately evident. The initial decrease in pressure, due to interface condensation heat transfer, is significantly greater with helium than nitrogen of the same initial mass fraction. It is also evident that not much nitrogen (-5% by mass) is required to entirely suppress the effect of the insurge jet. Based on this experimental data alone, it would appear that indeed helium stratifies at the top of the pressurizer, and nitrogen remains well mixed.
Figure 5.7 - Measured pressure response of empty tank insurge transients with helium or nitrogen present.
Figure 5.8 - Measured pressure response of empty tank insurge transients with helium or nitrogen present.
A valid question at this point, however, is: Is the stratification of helium an actual (realistic) phenomenon, or merely an artifice of the method of gas injection? The reasons for this concern, and the experimental work done to provide the answer are discussed in the following section.

5.3 HELIUM STRATIFICATION

If we accept the experimental evidence described above as "proof" of helium stratification, this results must be justified in light of the fact that, thermodynamically, this configuration should not occur. To demonstrate this fact, the thermodynamic conditions for equilibrium of a system, composed of a mixture of two ideal gases in a gravity field [16], are derived below.

The gravity potential ($\gamma$) of a system in a gravity field of magnitude $g$ is defined such that:

$$g = -\nabla \gamma$$

For a one-dimensional gravity field, the difference in the gravity potential ($\gamma - \gamma_0$) corresponding to an elevation difference $(z - z_0)$ is then:

$$(\gamma - \gamma_0) = g (z - z_0); \quad \text{for } z > z_0.$$ 

The chemical potential of component $i$ in a mixture of ideal gases, at partial pressure $p'_i$, is (on a mass basis):

$$\mu_{ci} = \frac{1}{M_i} \left[ R \, T \, \ln(p'_i) + \tilde{f}(T) \right]$$

where $R$, $M_i$, and $\tilde{f}(T)$ are the universal gas constant, the gas component molecular weight, and a function of temperature (for the particular gas), respectively.
For equilibrium, the system temperature and total potential must be constant and the same at every elevation within the system:

\[ \mu(z) + \gamma(z) = \mu(z_0) + \gamma(z_0). \]

Therefore, if the equilibrium partial pressure (or mole fraction) of a component is known at one elevation \((z_0)\), the mole fraction at any other elevation \((z)\) is given by:

\[
y(z) = \frac{p_i'(z)}{p_{\text{total}}} = \frac{p_i'(z_0) \exp \left[ \frac{-(\gamma - \gamma_0) M}{R T} \right]}{p_{\text{total}}}.
\]

For the experimental conditions in this study, the calculated difference in equilibrium mole fraction of helium gas, between the top and bottom of the vapor space (elevation difference of about 28 in), is negligible;

\[
y(He, z = +17 \text{ in}) = 0.999 \quad y(He, z = +45 \text{ in})
\]

The experimental results are, therefore, inconsistent with the thermodynamic conditions for equilibrium of an isolated system.

A plausible reason for the inconsistency lies in the method of injecting the helium into the system. As described in Section 4.2, the gas is injected into the vapor space through a tube, connected to a storage tank. Although the gas is bled into the system slowly, it is still concentrated at the point of injection. A significant buoyant force could result which may have driven the helium to the top of the pressurizer. This undesirable explanation was found not to be the case by the following experiment.
A small fan was built and mounted inside the vapor space through one of the viewing ports in the primary tank. The helium experiments were repeated, but this time, after the helium was injected, the fan was run for five minutes to forceably mix the vapor and helium. After the system was allowed to achieve equilibrium at constant pressure (heaters operated at 1.3 kW to offset heat loss), the transient was initiated. The measured transient response, however, showed no change from the original experimental data. Recorded peak pressures were completely consistent with those of the original experiments. Experimental methods were, therefore, ruled out as being the cause of the different pressure response of the transients with helium compared to those with either nitrogen or argon.

The source of the discrepancy between the experimental results and conditions for thermodynamic equilibrium, derived above, is that the equilibrium conditions apply for an isolated system. The experimental apparatus in this study and, to a lesser extent, a full scale pressurizer are not isolated systems due to the finite steady state environmental heat loss. Wall condensation draws a non-condensable gas to the wall where it accumulates in high enough concentrations to consider buoyancy forces between the vapor and gas. In the case of helium (or hydrogen), the gas molecular weight is significantly lower than that of vapor and the gas "floats" to the top of the pressurizer. Nitrogen and vapor, being of more comparable molecular weights, do not move as much axially. Argon, being much heavier than vapor "sinks" to the lower regions of the "vapor space" and also exhibits some stratified behavior (the peak pressures for experiments with argon were lower than those with equivalent mass fractions of nitrogen).

These arguments are entirely qualitative and require some analyt-
ical support. The results of modeling the pressurizer experiments are described in the next section and will reinforce the experimental implications of helium statification. The difference in wall heat transfer assuming fully mixed conditions or stratified conditions will be shown to be very large and provides good analytical support for the observed system responses.
6. EXPERIMENTAL DATA ANALYSIS

Several interesting phenomena were indicated by the experimental results discussed in the previous section. Among the most important were:

(i) Stratification of subcooled insurge liquid below the steady state (hot) liquid volume,

(ii) Significantly higher peak pressures for insurge transients with a non-condensable gas present, than for an equivalent transient without gas present,

(iii) Strong dependence of the peak pressure on the type of gas considered (for a specified initial non-condensable gas mass fraction).

The first of these phenomena is strongly supported not only by the data of this study, but also that of others [3],[8]. If one reviews the analysis by Kang [9], it is evident that this result is not limited to the geometry of this experimental apparatus but can be described in terms of the fluid mechanics of the insurge flow through the inlet sparger. The second is a measured fact of this study alone. The third is an unanticipated result which requires some analytical support to justify it physically.

An analytical model of the M.I.T. test vessel was, therefore, developed to lend some credence to the argument that the distribution of a non-condensable gas in the vessel vapor space influences the pressure response of the system. This model, described in the next section, was used to calculate the transient pressure responses for the bounding configurations of the vapor/gas mixture (1. assuming a uniform mixture of the vapor and gas, and 2. assuming the vapor and gas are
completely stratified above the liquid interface). A comparison of the actual (measured) pressure response for each experiment to the bounding calculated responses, also described below, provides the required analytical support to the proposed cause of the observed variation in peak pressure with different non-condensable gases. Conclusions as to the "typicality" of the above results are also presented in light of the scaling difference between the experimental apparatus and the modeled full scale system.

6.1 THERMODYNAMIC MODEL OF THE PRESSURIZER

In the development of the thermodynamic model of a PWR pressurizer, and, in particular, the experimental apparatus of this study, the following guidelines were imposed:

The model must be

(i) Simple in content and application,

(ii) Easily adaptable for sensitivity studies in modeling the effect of non-condensables, and

(iii) Consistent with the modeling approach of Kim [8].

The latter requirement allowed the results of this analysis to be incorporated with Kim's model. The derivation of the general thermodynamic relation for calculating the pressure response of the system is, therefore, very similar to that of Kim and is only summarized here. A significantly different approach to modeling wall heat transfer was taken here, however, and is described in detail.

A deformable control volume, defined as the region of space occupied by water vapor, is taken to be the thermodynamic system of interest. The control volume representation of the experimental system
is shown in Figure 6.1(a). The physical size of this volume depends on the model used, as illustrated in the adjacent Figure 6.1(b). For the completely stratified model, the system volume is that above the liquid interface less the volume of non-condensable gas, as determined by the ideal gas law. Under the fully mixed assumption, however, the entire volume above the liquid interface is part of the system and the state variables are determined as a function of the partial pressure of vapor. Interacting with the system are the vessel walls, and a neighboring deformable control volume, defined as the balance of the total pressurizer volume.

The insurge flow is treated only as a boundary condition for the increase of volume of the neighboring, liquid control volume. This simplification can be made for initial liquid levels well above the inlet sparger height since the vapor space remains essentially insulated from the cold insurge liquid by the initial pressurizer liquid volume. The enthalpy of the liquid in the neighboring control volume at the interface is, therefore, taken as that of saturated liquid at the system total pressure.

The conservation equations and state relations for the thermodynamic system are:

Mass Balance: \[ \frac{dm_1}{dt} = w_{21} \]

Energy Balance: \[ \frac{dU_1}{dt} = w_{21} i_2 + Q_{\text{wall}} - p \frac{dV_1}{dt} \]

State Relations: \[ U = m i - p' V \]
\[ i = i(p') \]
\[ v = v(p', i) = \frac{V}{m} \]
Figure 6.1 - Control volume modeling configurations of the experimental apparatus.
Note that \( p \) indicates total pressure and \( p' \) the vapor partial pressure.

Additional Condition: Total volume of pressurizer is constant,

\[
V = V + V + V_{\text{tot}} + 1 + 2 \quad \text{NC}
\]

where the subscripts 1, 2, NC refer to the vapor, liquid, and gas regions, respectively. Note that for the fully mixed model

\[
V = 0 \quad \text{NC}
\]

because \( V \) becomes the mixture volume.

It is assumed that the system follows the saturation line during an insurge, therefore, the Clausius-Clapeyron relation is employed to relate pressure and temperature. The non-condensable gas is always assumed to behave as an ideal gas.

The final result for the system pressure in terms of the insurge volumetric flow rate \( \dot{V}_{\text{insurge}} \), wall heat transfer \( Q_{\text{wall}} \), and state properties is:

\[
\frac{\partial p}{\partial t} = \frac{J}{V} \left[ \frac{v}{g} \frac{Q}{\text{wall}} + i \frac{\dot{V}}{\text{fg} \text{insurge}} \right] - \left[ \frac{1}{J m} \frac{\partial v_g}{1 \partial i} \right] - \left[ v + J \frac{\text{fg} \partial v_g}{g \partial p} \right]
\]

Solving the pressure equation requires a simultaneous solution of the wall heat transfer problem (to supply the term \( Q_{\text{wall}} \)), as outlined below.
6.1.1 WALL HEAT TRANSFER

Two configurations of the vapor and non-condensable gas mixture were modeled to calculate the bounding pressure responses of the experimental system. A schematic diagram of the "completely stratified" and "fully mixed" model configurations is shown in Figure 6.1(b), next to the overall control volume representation. The significant differences in the analytic models of these bounding configurations is the length of the vessel wall subject to condensation heat transfer and the relative magnitude of the wall heat transfer coefficient (with the Nusselt theory coefficient taken as the maximum possible). In both cases, the thermal capacity of the non-condensable gas was ignored.

The two bounding wall heat transfer models are characterized as follows:

(i) Completely Stratified - The length of vessel wall corresponding to the non-condensable gas volume at the current pressure, as determined by the ideal gas law, is modeled as an adiabatic surface (labeled $\Delta Z$ in Figure 6.1(b)). The remaining wall length, adjacent to vapor only ($H-\text{LVL}-\Delta Z$ in Figure 6.1(b)), available for heat transfer as governed by the Nusselt theory heat transfer coefficient.

(ii) Fully Mixed - The entire length of the vessel wall above the current liquid level ($H-\text{LVL}$ in Figure 6.1(b)) is available for heat transfer, but subject to a degraded heat transfer coefficient. The exponential curve fit of Sparrow's analysis, shown in Figure 3.2, is used to determined the heat transfer coefficient as a function of bulk non-condensable gas mass fraction.
In both models, an adiabatic boundary condition is applied to the outside wall of the vessel and a 1.3 kW constant heat loss is "hardwired" as a negative $Q$ term in the pressure equation. The wall latter condition introduced a slight problem in calculating the insures with a large mass fraction of nitrogen as pointed out later.

A solution to the transient conduction problem in the vessel wall must be solved to calculate the energy loss of the system due to wall heat transfer. This requires the application of Duhammel's theorem to calculate the wall temperature profile due to the time varying boundary condition. The problem geometry, governing differential equation and boundary conditions are:

**Governing Equation:**

\[
\frac{\partial^2 T}{\partial x^2} = \frac{\partial T}{\partial \theta} \quad ; \quad \text{where, } T = \frac{T - T_\infty}{T_0 - T_\infty}, \quad \theta = \frac{\alpha t}{L}, \quad \bar{x} = \frac{x}{L}
\]

**Boundary conditions:**

1. \( \frac{\partial T}{\partial x} (\bar{x}=0) = 0 \)
2. \( \frac{\partial T}{\partial x} (\bar{x}=1) = -Bi \times \bar{T} \)

where, \( Bi = \frac{hL}{k} \)

**Initial condition:**

\( T(\bar{x},0) = 1 \)

It is important to note that the temperature of the inner wall of the vessel, \( T_w \), is not assumed to be at the saturation temperature of the system, as modeled by other investigators. Here, the inner wall temperature is required to allow calculation of the heat transfer rate.
from the following relation:
\[
\frac{Q}{\text{wall}} = h \left( T_{\text{wall}} - T_{\text{bulk}} \right)
\]

Solving the conduction equation with the standard method of separation of variables (assuming a solution of the form \( T = X(x)\Theta(\theta) \)), the resulting eigencondition is,
\[
\lambda_n \tan \lambda_n = Bi
\]
and the solution for the nondimensionalized temperature profile for a single step change in the fluid temperature is,
\[
T(\tilde{x},\theta) = 2 \sum_{n=1}^{5} \left[ \frac{Bi \cos(\lambda_n)}{\lambda_n^2 + (Bi \cos(\lambda_n))^2} \right] \cos(\lambda_n \tilde{x}) \exp\left(-\lambda_n^2 \theta \right)
\]

The first five terms of this series are used as a sufficient solution. At each time step, the set of solutions for previous step changes are summed (each over their respective period of application) to determine the "current" temperature distribution (Duhammel's theorem).

6.1.2 STATE VARIABLES

Fluid properties and state variables of water are calculated from the curve fit equations used in the RETRAN-02 code [17]. These equations are based on the 1964 ASME steam tables and are easily adapted for evaluation on the computer. The specific quantities evaluated are:
\[ i(p), i(p), T(p,i), v(p,i), v(p,i), \mu(T,v) \]
\[ f(f), g(g), sat(f), f(f), g(g) \]

6.1.3 NON-CONDENSABLE GAS PROPERTIES

Several properties of the non-condensable gas are required to determine the solution of the pressure equation. For the stratified model, the volume occupied by the gas is required to determine the
vapor volume (and length of vessel wall available for heat transfer). For the fully mixed model, the bulk non-condensible mass fraction and gas partial pressure are required to determine the wall heat transfer coefficient and vapor partial pressure, respectively. Each of these quantities is easily calculated using the ideal gas law. The initial non-condensible gas mass (or mass fraction) and gas molecular weight are known quantities.

\[
V = \frac{m \cdot R \cdot T}{g \cdot n_c \cdot p \cdot M}
\]

Gas Mass Fraction (as a function of p):

\[
X = \frac{1}{M(g) + \frac{p (H - LVL) A M(v)}{m \cdot R \cdot T}}
\]

Parameters

\[
m = \text{mass of gas} \quad g
\]
\[
R = \text{universal gas constant}
\]
\[
T = \text{system temperature}
\]
\[
p = \text{system pressure}
\]
\[
M(g) = \text{gas molecular weight}
\]
\[
M(v) = \text{vapor molecular weight}
\]
\[
A = \text{vessel fluid X-sectional area}
\]

6.1.4 SOLUTION PROCEDURE

A descriptive flow chart of the calculation procedure for solving the pressure equation is as follows:

(i) Initialize all variables to stated initial conditions.

(ii) Determine the liquid level at the new time based on the boundary condition insurge volumetric flow rate.

(iii) Guess the new time pressure.

(iv) Evaluate all parameters based on assumed new pressure; state properties, vessel wall temperature, wall heat loss.
(v) Calculate new time pressure from pressure equation.
(vi) Compare with guessed pressure, iterate for convergent solution and advance in time.

An iterative type solution is required since the pressure equation and wall heat transfer problem are coupled.

6.2 COMPARISON OF BOUNDING CALCULATIONS WITH EXPERIMENTAL RESULTS

The pressure response for each of the experiments (initiated with a partially full tank) was calculated using both of the bounding models (completely stratified and fully mixed). The calculated results are compared to the corresponding measured pressure history of the base case insurge transient and each of the partially full tank experiments with a non-condensable listed in Table 4.1, in Figures 6.2 through 6.10.

As anticipated, the fully mixed model always predicts a higher peak pressure for all gases (the same degradation curve for the heat transfer coefficient is used for all gases). The two experiments with helium agree well with the pressure response calculated with the completely stratified model. The experiments with nitrogen are, in general, in better agreement with the fully mixed model, and the pressure histories with argon are somewhere in between. As proposed in Section 5.1.2, the lower peak pressures for the experiments with helium as the non-condensable gas can be explained in terms of the configuration the vapor/gas mixture assumes at steady state.

As shown in Figure 6.6 and 6.7, the experimental pressure histories are not always bounded by the calculated results and have peak pressures above those calculated by the fully mixed model. This is a result of the contradictory modeling technique of assuming an adiabatic outer vessel wall and then "hardwiring" a constant value for heat loss. If
Figure 6.2 - Measured and calculated pressure history of the Base Case insurge transient.
Figure 6.3 - Measured and calculated pressure history of the insurge with 3% helium by mass.
Figure 8.4 - Measured and calculated pressure history of the insurge with 20% helium by mass.
Figure 6.5 - Measured and calculated pressure history of the insurge with 3% nitrogen by mass.
Figure 6.6 - Measured and calculated pressure history of the insurge with 10% nitrogen by mass.
Figure 6.7 - Measured and calculated pressure history of the insurge with 20% nitrogen by mass.
Figure 6.8 - Measured and calculated pressure history of the insurge with 3% argon by mass.
Figure 6.9 - Measured and calculated pressure history of the insurge with 10% argon by mass.
Figure 6.10 - Measured and calculated pressure history of the insurge with 20% argon by mass.
the heat loss is set to zero and the calculation rerun, a steeper pressure rise is calculated which does bound the measured result. In reality, some intermediate value for heat loss is probably correct.

These particular results simply illustrate the magnitude of the degradation of the wall heat transfer coefficient for relatively large amounts of nitrogen. It is important to note that this problem did not occur for the cases with either helium or argon. The former having a molecular weight much smaller than water, and the latter somewhat larger. In both cases, the gas becomes less mixed with (or more stratified from) the water vapor. One could, therefore, extrapolate these results by predicting that a very heavy (high molecular weight) gas, such as fission product gases, would stratify between the liquid volume and the vapor.

6.3 THE EFFECTS OF SCALING

Application of the experimental and analytical results of this study to a full scale PWR system requires consideration of the effects of non-typical operating conditions and geometry. Namely, the experimental apparatus was operated at approximately 80 psia, and has a surface area to volume ratio of 0.5 (in$^{-1}$). A full scale PWR pressurizer operates at 2210 psia and has a surface area to volume ratio of approximately 0.05 (in$^{-1}$). The key characteristics of the system which are affected by these scaling differences are the density ratio of water and environmental heat loss.

The density ratio of water ($\frac{\rho_f}{\rho_g}$) is approximately 300 at 80 psia and only 6 at 2210 psia. Conversely, the reduction in specific volume associated with a given mass condensation rate at 80 psia is
much larger than at 2210 psia. Experiments at low pressures, therefore, tend to overemphasize the effects of condensation heat transfer on the system pressure response. This effect is important to keep in mind when transferring the effects, measured at low pressure, to predictions of the response at high pressure. A model which can predict the effects of change in condensation heat transfer at low pressure should, however, do well at high pressure where the overall effect of condensation heat transfer is less important.

The larger surface area to volume ratio in the experimental apparatus also has the effect of increasing the effects of wall condensation on the system pressure response. For a given volume of vapor, the wall surface area available for condensation is proportionally larger than in the full scale system. Environmental heat loss is also proportionally larger. A non-typical result which may occur due to the relatively high heat loss is the stratification of gases lighter than vapor (e.g. hydrogen or helium). This effect was noted by Cohen [10], from previous experimental work in gas-vapor separation, and is consistent with the stratification of helium, observed in the partially full tank experiments of this study, at steady state operation.
7. CONCLUSIONS

An experimental study of the response of a small scale pressure vessel to insurge transients in the presence of a non-condensable gas has been performed. A significant change in the pressure history of the system was measured for relatively low amounts of non-condensable gas. The cause of the change in the pressure response of the system was investigated by modeling thermodynamic characteristics of the bounding configurations of the vapor and non-condensable gas mixture. The following conclusions are drawn based upon the experimental and analytical results:

(i) For insurge transients, during which condensation heat transfer is a significant physical phenomenon, the pressure response of the M.I.T. test vessel is significantly affected by the presence of a non-condensable gas.

(ii) Significant reductions in wall heat transfer can occur when very small amounts of a non-condensable gas are present.

(iii) The magnitude of the increase in peak pressure with a non-condensable gas is dependent upon the type of gas considered. Gases of molecular weight close to that of water (18) cause the greatest increase in peak pressure, in contrast to gases of molecular weight much different from water.

(iv) The difference in the effect on the system pressure response for gases of different molecular weight (but of equal bulk mass fraction) is due to the configuration the specific vapor-gas mixture assumes in steady state operation. The pressure response of the M.I.T. pressure vessel can be predicted using a simple thermodynamic (first law) model with
a heat transfer model which accounts for the presence of a non-condensable gas as follows:

1. For Helium (molecular weight = 4):
   Assume complete stratification of the vapor and gas. Wall heat transfer is modeled only over a length of the vessel wall proportional to the stratified volume of vapor. A Nusselt theory heat transfer coefficient is satisfactory to estimate the thermal resistance of the condensate film.

2. For Nitrogen (molecular weight = 28):
   The gas and vapor are fully mixed. Wall heat transfer is modeled over the entire length of the vessel wall, but the Nusselt theory heat transfer coefficient is degraded as a function of bulk non-condensable gas mass fraction. The degradation factor may be estimated by the analysis of Sparrow.

3. For other gases, the stratified model is appropriate if the molecular weight of the gas is significantly different from that of water. If the gas has molecular weight similar to water, the fully mixed assumption is more appropriate. In either case, the fully mixed model yields a higher peak pressure.
8. REFERENCES


APPENDIX A - SUMMARY OF EXPERIMENTAL APPARATUS CONSTRUCTION

Primary Tank

Dimensions: I.D. = 8.0 in., Internal Height = 45.0 in.
Wall thickness = 21/64 in.

Material: ASTM A-312 Stainless Steel

Thermocouples: Iron - Constantan (Omega Type T)
Locations - distance above bottom of tank (in.):
4, 7, 10, 13, 16, 19, 22, 25, 28, 31, 33, 36, 39, 42

Elevation of Non-condensable gas injection tube outlet:
19 in. above the bottom of the tank.

Secondary Tank

Dimensions: I.D. = 8.0 in., Internal Height = 57.0 in.

Material: ASTM A-312 Stainless Steel

Non-condensable Gas Storage Tank

Internal Volume = 0.18 cuft.

Insurge Flow Measurement - Orifice

Inlet opening diameter = 0.218 in.

Geometry: Inlet side abrupt, Outlet side tapered 40

Plate thickness = 1/8 in.
APPENDIX B - LISTING OF SELECTED EXPERIMENTAL DATA

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