14.3 REACTOR COOLANT SYSTEM PIPE RUPTURE (LOSS OF COOLANT ACCIDENT)

14.3.4 Containment Integrity Analysis

14.3.4.5 Containment Integrity Analysis - Background Information

14.3.4.5.1 LOCA Mass and Energy Release Data

14.3.4.5.1.1 Model Description

Mass and energy release rate transients generated for the TMD pressure calculation are supported by an extensive investigation of short term blowdown phenomena. The SATAN-V code was used to predict early blowdown transients. The study concerned a verification of the conservatism of the SATAN-V calculated transients. This verification was accomplished through two approaches: a review of the validity of the SATAN-V break model, and a parametric study of significant physical assumptions.

The SATAN-V code uses a control volume approach to model the behavior of the Reactor Coolant System resulting from a large break in a main coolant pipe. Release rate transients are determined by the SATAN-V break model which includes a critical flow calculation and an implicit representation of pressure wave propagation.

The SATAN-V critical flow calculation uses appropriately defined critical flow correlations applied for fluid conditions at the break element. For the early portion of blowdown, subcooled, saturated, and two-phase critical flow regimes are encountered. SATAN-V uses the Moody (Reference 25) correlation for saturated and two-phase fluid conditions and a slight modification of the Zaloudek (Reference 27) correlation for the subcooled blowdown regime.

Since most short term blowdown transients are characterized by a peak mass and energy release rate that occurs during a subcooled condition, the Zaloudek application is particularly significant. The Zaloudek correlation is modified to merge to Moody predicted mass velocities at saturation in the break element. This correlation appears in the critical flow routine of SATAN-V in the form:

$$G_{crit} = CK1\sqrt{(5.553*10^5)(P - C_1 P_{sat})}$$

where:

G_{crit}	=	critical flow in lb. mass/sec-ft^2
P	=	reservoir pressure (psia)
P_{sat}	=	reservoir saturation pressure (psia)
C_1	=	constant where $.5 < C_l < 1$

$CK1 = \sqrt{\frac{.1037}{1 - C_1}}$ = constant adjusted such that when $P = P_{sat}$, G_{crit} from Zaloudek matches the SATAN-V Moody critical flow calculated at zero quality. For the present analysis, C_l equals 0.9 and CK_l equals 1.018. The modification also more conservatively accounts for the phenomena of increasing mass velocity with increasing degrees of subcooling. The slope of the subcooled G vs. P curve is steeper for the modified correlation. The low quality portion of the SATAN-V critical flow model is presented in Figure 14.3.4-143. The Moody saturation line corresponds to the condition upstream in the break element where quality equals zero and pressure equals saturation pressure. Thus when pressure equals saturation pressure in the break element the Zaloudek and Moody critical flow values are equal. When pressure exceeds saturation pressure in the break element, the modified Zaloudek is used for the critical flow calculation. The steep slope of the Zaloudek G vs. P line indicates the over-accounting for the subcooling effect.

14.3.4.5.1.2 Comparison to other Critical Flow Models

The Henry-Fauske critical flow correlation was considered for comparison (References 16, 28, and 29). This correlation models flow nonequilibrium via an approach which includes an empirical parameter. This parameter describes the deviation from equilibrium mass transfer and depends on flow geometry.

The value is selected for a particular configuration based on the range of throat equilibrium qualities. The value for constant area ducts is used in the present analysis. This choice is based on the worst possible double-ended break geometry described below.

For cold leg and hot leg breaks, the majority of the flow, about 65%, comes from the vessel side of the break. For this side, the geometry may be described as an entrance nozzle and a straight pipe of approximately 12 feet in length with a diameter of 29 inches. This length of pipe represents the distance from the reactor vessel to the periphery of the biological shield. No

double-ended break can occur within the biological shield because of the restricted movement within the pipe annulus. Hence the constant area value is appropriate.

Like the SATAN-V model, the Henry-Fauske correlation yields a G_{crit} in terms of upstream conditions and like the SATAN-V model it also exhibits a steeper slope of the G vs. P line for subcooled conditions. As can be seen in Figure 14.3.4-143, the Henry-Fauske saturated liquid line is below the Moody saturated line (SATAN-V model) for pressures greater than about 1000 psia.

For short term blowdown calculations, the significant pressure region is from 1000 psia to 1800 psia, with increased emphasis on subcooled conditions for the 1000 psia end. Subcooled mass velocity versus pressure is given for the two fluid temperatures corresponding to P_{sat} = 1000 and P_{sat} = 1800. It is clear from the figure that the slope of the Zaloudek G vs. P line is steeper in both cases. This increased sensitivity coupled with the higher value for Moody at saturation causes the SATAN-V model to predict higher mass velocities. Hence the SATAN-V model is a more conservative treatment of critical flow than the Henry-Fauske model.

In the original FLASH model,(Reference 30) the Moody correlation was extended to subcooled conditions. This treatment is employed in many blowdown codes and thus it is appropriate to compare the SATAN-V model to these values. This is illustrated in Figure 14.3.4-144. Again, the Zaloudek treatment yields higher mass velocities and the SATAN-V model is more conservative.

14.3.4.5.1.3 Comparison to Experimental Data

The margin included in the modified Zaloudek prediction of subcooled critical flow rates is demonstrated by a review of experimental subcooled critical flow data. Figures 14.3.4-145 and -146 present a plot of measured vs. predicted critical flow values for Zaloudek's own data.(References 27 and 31) The figures indicate that when the modified correlation is applied to Zaloudek's data, the predicted critical flow values are significantly higher than measured flow rates.

The margin associated with the SATAN-V critical flow calculation may also be demonstrated by a review of the low quality data presented by Henry in ANL-7740 (Reference 29). Exit plane quality, in terms of the Moody model, is determined as a function of upstream conditions by assuming an isentropic expansion to exit plane (i.e., critical) pressure. The lowest exit plane qualities where the Moody model is applied in the SATAN-V code occur for expansion from saturated liquid conditions; a plot of these are shown in Figure 14.3.4-147. For exit plane

qualities above the line, the Moody model is used in the SATAN-V code. Below the line, the Modified Zaloudek model is used.

Henry's comparison between data and model shows that for the range of exit plane quality greater than 0.02, the Moody model overpredicts the data, hence is conservative.

For the region below 0.02, it is appropriate to compare Henry's results with the Modified Zaloudek model, as used in the SATAN-V code. This is done in Figure 14.3.4-148 for all of Henry's data points. As can be seen, the Zaloudek model overpredicts the flow. A discharge coefficient of 0.6 would be more reasonable than the 1.0 value used in SATAN-V.

14.3.4.5.1.4 Application to Transient Conditions

The Zaloudek correlation was developed for stagnation (reservoir) pressure and quasi-steady-state critical flow conditions. It is extended to application in the SATAN-V break element and transient flow conditions. This extension is justified because of the following considerations.

The pressure in the break element differs from the value in a nearby large volume because of three effects:

1. Pressure drop due to friction
2. Pressure drop due to spatial acceleration (momentum flux)
3. Pressure drop due to the transient

The friction term in the reactor application is quantifiable; this term is less important than the other two. The sensitivity of the break flow rate to fluid friction was evaluated via a parametric study. For the purposes of this study, an analysis was made wherein the frictional resistance between the vessel and the break was reduced from the design values by a factor of one hundred. Over the period from 0.0 to 60 milliseconds (which includes the peak break flow), the integrated mass flow differed by less than 18 lbm from the design friction case; the total release over this period was about 5000 lbm.

Spatial acceleration is the major source of pressure drop upstream of the break between the reservoir and the pipe, causing steep pressure gradients in the approach region to critical flow. This term is not calculated explicitly in the SATAN-V code. Spatial acceleration is accounted for by the use of critical flow correlations (Zaloudek or Moody) which contain this effect. No credit is taken for pressure drop due to spatial acceleration for elements other than the break element. Hence the pressure calculated by SATAN-V may be interpreted as a stagnation pressure which is the appropriate pressure for the Zaloudek and Moody models.

Prior to the occurrence of the peak release rate, the break element and upstream reservoir pressures differ as a result of the transient described by pressure wave propagation. The applicability of the SATAN-V break model to this situation is verified by the code's ability to match recorded semi-scale transients. SATAN simulations of LOFT transients support the SATAN-V transient calculation. Figure 14.3.4-149 presents a comparison of LOFT pressure transients recorded near the break to the SATAN-V model of the LOFT break element transient. The graphs demonstrate the ability of the SATAN-V code to track pressure waves in the broken pipe.

Moreover, the critical flow correlation is implemented in the present analysis by combining the correlation with the appropriate momentum equation. This provides a model for predicting break flow acceleration vis-a-vis a quasi-steady simulation. This is found to have little effect on containment pressure but is a more physical representation.

Thus the SATAN-V break model is supported by subcooled critical flow data, by comparison to other correlations, and by ability to simulate short term transients.

14.3.4.5.1.5 Parametric Studies

With confirmation of the conservatism of the SATAN-V break model, a series of parametric studies were undertaken to identify the blowdown transient corresponding to the most severe TMD results. A series of basic sensitivities were first studied to set the scope of the more detailed investigations. The assumptions of break size, break type and break location were considered. The results of this analysis were evaluated using the TMD code.

14.3.4.5.1.6 Break Size, Type and Location

A break of an area corresponding to twice the coolant pipe area was the most severe for mass and energy release. For this size break both double-ended guillotine and double-ended split type breaks were considered. These break types differ in that the split allows full communication between approach regions at each side of the break while the guillotine models a complete severance of two ends of a broken coolant pipe.

SATAN-V transients were generated for both type double-ended breaks with the guillotine break resulting in higher mass and energy release rates. The split type break is less severe because flow is reduced from the loop side of the break. This is because communication makes the break element pressure higher than would occur for the loop end in a guillotine rupture. The higher break element pressure yields a smaller pressure gradient for driving loop side flow. The vessel end is relatively unaffected by break type because a choked condition remains at the nozzle. In particular, the split type break results in a 10,000 lbm/sec reduction in peak mass flow rate.

The influence of break location on TMD peak pressure was considered by generating blowdown transients for possible worst break locations. The results indicated that a double-ended break in the pump suction leg was clearly less severe for short term blowdown release rates and that no such clear decision could be made between hot and cold leg breaks.

More detailed parametric studies were continued for the cold leg and the hot leg double-ended guillotine breaks. The two locations produce intrinsically different TMD pressure responses and therefore must be dealt with in separate parametric surveys.

14.3.4.5.1.7 Hot Leg Nodal Configuration

A study of the SATAN-V nodal configuration has been applied to the hot leg double-ended guillotine break. It was found that for this break the nodal configuration of the broken hot leg and the upper plenum are significant to short term transients. Spatial convergence was achieved for the upper plenum after the addition of four nodes to the standard SATAN-V two node upper plenum model. These nodes are hemispherical shells arranged concentrically from the broken hot leg nozzle and approximate the propagation of the pressure wave in the upper plenum. They are significant in that they specify the inertial response of the upper plenum. Spatial convergence was demonstrated because doubling the number of nodes yielded less than a one percent change in break flow at all times.

Sensitivity to nodal configuration in the broken hot leg pipe was also investigated. Models with from 4 to 16 nodes were used to generate transients. Increasing the number of nodes was found to give a better simulation of pressure wave propagation in the pipe.

14.3.4.5.1.8 Cold Leg Studies

The cold leg break transient was also reviewed in terms of significant parameters.

The Reactor Coolant System behavior is different for cold leg breaks and the peak containment pressure occurs later for cold leg breaks.

14.3.4.5.1.9 Nodal Configuration

For the cold leg break the nodal configuration of the broken cold leg and the downcomer is significant to the transient. Spatial convergence was achieved with the addition of three additional nodes to the standard SATAN-V model. These are annular rings arranged concentrically from the broken cold leg nozzle and model propagation of the pressure wave in the downcomer.

As in the hot leg sensitivity, from 4 to 16 pipe node models were tried for the cold leg transient. Again, more nodes gave a better simulation of pressure wave propagation in the broken pipe.

14.3.4.5.1.10 Pump Modeling

For the time period of interest, the variation in pump inlet density is small and the variation in pump speed is small. This model was found to have no effect.

14.3.4.5.1.11 Summary

From the hot leg and cold leg studies, the design basis mass and energy release rates have been finalized. The mass and energy release rate transients for all the design cases are given in Figures 14.3.4-133 to -142. All cases are generated from the SATAN-V break model consisting of Moody-Modified Zaloudek critical flow correlations applied at the break element. Since no mechanistic constraints have been established for full guillotine pipe rupture, an instantaneous pipe severance and disconnection is assumed for all transients. Assumptions specific to the presented transients are discussed in section 14.3.4.3.1.1.

Figures 14.3.4-137, -138 -139, and -140 present mass and energy release rate transients for hot leg and cold leg split type breaks of a single ended pipe area. For breaks of this size, the split type break is used as a design basis and this choice is justified by a generic study of the effect of break type on short term release rates. A discussion of this study and of break type influence was given as a response to question 6.71 to the Catawba PSAR (USNRC Docket No's. 50-413 and 50-414). It is sufficient for this discussion to note that for single ended breaks, a split type break results in higher release rates.

Differences in blowdown mass and energy release rates between hot leg and cold leg single ended split breaks result from the influence of the hot water in the upper plenum and hot legs. For a cold leg single ended split, the flashing fluid in the upper plenum and hot legs sustains flow to the break from both the vessel and from the loop through the broken loop pump. This flashing, then, acts to maintain a subcooled blowdown for the cold leg break.

For the hot leg single ended split no such pressurization effect occurs at the break. Flashing fluid in the hot leg and upper plenum, rather, results in an extensive two-phase blowdown condition. The broken leg pump continues to remain effective during the hot leg split transient and thus draws flow away from the break.

The hot leg and cold leg double ended release rate transients presented in the figures discussed above are the result of a guillotine type break. This basis is again justified as a result of the generic break type study referenced above. The study indicated that for breaks of twice the coolant pipe area, a guillotine type break resulted in the highest release rates.

An explanation of the differences in the release rate transients presented for hot leg and cold leg double-ended breaks is complicated by the fact that these are guillotine type breaks. Since the guillotine break models a complete separation of the broken pipe, conditions at each end of the break must be considered individually. The total release rate is then the sum of contributions from each end.

Flashing of the fluid in the hot legs again accounts for the higher mass flow rates observed for the cold leg double-ended break in comparison to the hot leg double-ended transient. However, two other influences are significant for breaks of this type and area.

For the cold leg guillotine, the increased break area requires higher flows if a subcooled blowdown condition is to be maintained at the break. A subcooled blowdown occurs at the vessel end of the broken pipe but because of the broken loop pump resistance to increased flow, a two-phase blowdown occurs at the loop end of the break.

Since for both hot leg and cold leg breaks the loop side of the break experiences a two-phase blowdown, the loop layout geometry determines the difference in their release rates. Higher release rates are observed for the loop side of hot leg break because it is fed from the reservoir of water in the inlet plenum of the steam generator. No such supply of water exists at the loop side of the cold leg break. In fact, flow to the cold leg loop side is restricted by the resistance of the broken loop pump.

The differences in release rates for the double-ended break are thus the result of two effects. A higher vessel side mass flow rate for the cold leg break results from a subcooled blowdown maintained by the pressurizing effect of flashing hot leg fluid.

A lower loop side mass flow is observed for the cold leg break because of the differences accountable to loop layout geometry. However, since the subcooled blowdown effect dominates the total release rate, the cold leg double-ended guillotine still results in highest total mass discharge rates.

14.3.4.5.2 Experimental Verification

14.3.4.5.2.1 Early Tests

The performance of the TMD Code was verified against the 1/24 scale air tests and the 1968 Waltz Mill tests. For the 1/24 scale model the TMD Code was used to calculate flow rates to compare against experimental results. The effect of increased nodalization was also evaluated. The Waltz Mill test comparisons involved a reexamination of test data. In conducting the reanalyses, representation of the 1968 Waltz Mill test was reviewed with regard to parameters

such as loss coefficients and blowdown time history. The details of this information are given in Reference 14.

14.3.4.5.2.2 1973 Waltz Mill Tests

14.3.4.5.2.2.1 Test Purpose

The Waltz Mill Ice Condenser Blowdown Test Facility was reactivated in 1973 (Reference 2) to verify the ice condenser performance with the following redesigned plant hardware scaled to the test configuration:

1. Perforated metal ice baskets and new design couplings.
2. Lattice frames sized to provide the correct loss coefficient relative to plant design.
3. Lower support beamed structure and turning vanes sized to provide the correct turning loss relative to the plant design.
4. No ice baskets in the lower ice condenser plenum opposite the inlet doors.

The primary objective of these tests was to determine the transient heat transfer and fluid flow performance of the ice condenser design and to confirm that conclusions derived from previous Waltz Mill tests had not been significantly changed by the redesign of plant hardware. Consequently, the design of the test hardware was configured to provide heat transfer and fluid flow characteristics which were equivalent to those in the plant design. It should be noted that test hardware was not representative of structural characteristics for the plant design since structural response to blowdown was not one of the test objectives. In addition, responses of lower, intermediate, and upper deck doors to blowdown were not included in the test objectives.

14.3.4.5.2.2.2 Test Facility

The Waltz Mill Ice Condenser Blowdown Facility consists of a boiler, receiver vessel, and instrumentation room, and also ice storage and ice machine rooms which are used in conjunction with the ice technology facility. Figure 14.3.4-150 shows the general arrangement of the facility. The boiler and receiver vessels are connected by a 12" schedule 160 pipe in which is located a rupture disc assembly.

The boiler is 3 feet in diameter and 20 feet long, mounted on a structural frame. It can be heated electrically to pressurize a maximum of 117 cubic feet of water to an allowable maximum of 1586 psig pressure at 600°F. Strip heaters mounted on the outside of the boiler shell provide the heat. The flow rate from the boiler is controlled by an orifice located in the piping between the boiler and receiver vessel.

The 12" piping between the boiler and receiver vessel is heated by strip heaters attached to the outside surface of the pipe. Figure 14.3.4-151 shows the piping is arranged into three sections as far as flow and heater capability are concerned. This permits operating the piping and sections of the piping at various subcooled temperatures relative to the boiler.

Figure 14.3.4-152 shows the internal arrangement of the receiver vessel. The ice chest section contains eight ice baskets, 12" diameter by 36 feet high, arranged in a 2 x 4 array. Lattice frames are located at six foot levels of the ice baskets. The baskets set on a lower support structure with flow blockage areas proportional to the plant. Turning vanes are located below the ice baskets and direct the flow entering the lower inlet up through the ice baskets. The vessel is divided into lower and upper compartments. The flow enters the lower compartment from the 12" pipe diffusers, is directed into the ice chest, past the ice baskets and then vents into the upper compartment. The ice chest is wood. All metal surfaces are insulated to limit the heat transfer to these surfaces.

Figure 14.3.4-153 shows the location and typical arrangement for the temperature and pressure measurements that will be made inside the receiver vessel ice chest. The outputs from the transducers are connected to a data acquisition system with scanning rates of 2000 samples per second or 200 samples per second.

14.3.4.5.2.2.3 Test Procedure

The ice baskets are filled in the penthouse at the top of the receiver vessel by a blower system before being lowered into the ice chest. Prior to installing ice baskets, the receiver vessel and building is cooled down by an air recirculation and refrigeration system. A lattice frame is installed after each six foot array of ice baskets and a hold down bar attached through the ice chest walls to prevent basket uplift. After all baskets are installed, the receiver vessel top manhole is closed and the boiler then brought to test conditions.

The boiler is evacuated and filled with demineralized water and the heatup started by energizing the strip heaters. As the water heats in the boiler and expands, it is vented through a letdown heat exchanger. The initial fill of the boiler is measured as well as the water relieved so that the total amount of water in the boiler and piping is always known. Water is circulated between the boiler and downstream piping during heatup by the recirculation system to the various sections of the 12" piping (Figure 14.3.4-151). Subcooled conditions can, thus, be obtained for the water preceding the saturated water in the boiler itself. By using the heaters, recirculating systems, and letdown system, test energy conditions are obtained.

The flow from the boiler and subcooled piping to the receiver vessel is controlled by an orifice plate located in front of the rupture disc assembly. By varying the size of orifice, the blowdown rate can be changed in accordance with the test plans. It is calculated that the maximum orifice required is 5.5".

After the boiler has reached test pressure and temperature, the blowdown is initiated by the rupture disc assembly. This is a double disc assembly with the pressure between the discs normally at about half the boiler operating pressure. The rupture disc burst pressure rating is 60 - 75% of the boiler operating pressure. The pressure between the discs is provided by a high pressure gas cylinder of nitrogen. At blowdown, the gas pressure is quickly released from the cavity between the discs by venting it into the downstream side of the rupture disc, causing the discs to rupture and the water upstream of the discs to be released into the receiver vessel.

At the time the pressure is started to vent from between the rupture discs, the data acquisition systems is actuated so that data is recorded throughout the blowdown transient. Data recording continues for ten seconds at high speed and then is reduced to a 1/10 speed for five minutes.

A preliminary set of test conditions is presented in Table 14.3.4-37.

14.3.4.5.2.2.4 Results

Confirmation of the predicted ice condenser pressure performance was determined by comparing test results with TMD code predictions for the appropriate test conditions and configuration. Initially, the TMD code predictions were based on assumptions that provided best agreement with previous Waltz Mill test results (e.g., 30% entrainment).

The TMD Code has, as a result of the 1973 test series, been modified to match ice bed heat transfer performance.

14.3.4.5.3 Short Term Containment Response

14.3.4.5.3.1 Results Based on 1973 Waltz Mill Tests

A number of analyses have been performed to determine the various pressure transients resulting from hot and cold leg reactor coolant pipe breaks in any one of the six lower compartment elements. The analyses were performed using the following assumptions and correlations:

1. Flow was limited by the unaugmented critical flow correlation.
2. The TMD variable volume door model, which accounts for changes in the volumes of TMD elements as the door opens, was implemented.
3. The heat transfer calculation used was based on performance during the 1973-74 Waltz Mill test series. A higher value of the ELJAC parameter has been used and an upper bound on calculated heat transfer coefficients has been imposed (see Reference 3).

14.3.4.5.3.2 Subcritical Flow Model Studies

For high Mach number subsonic flow, the TMD momentum equation incorporates a compressibility multiplier to account for compressibility effects resulting from area changes, and uses an average density along constant area flow paths.

With these modifications, both inertial and density effects are modeled by the TMD computer code.

A description of the compressibility multiplier, its derivation and application, is presented in this section. A brief description of the method by which the polytropic exponent (a necessary parameter in the compressibility multiplier approach) is calculated is also provided.

These effects have been examined for the D. C. Cook plant short term transient analysis by comparing previous analyses where these methods were not used to analyses using these methods.

For the plant the worst case RCS pipe break is a DEHL rupture in the lower compartment element 6. Results are presented also, for comparison purposes, for a DECL rupture in element 6.

The results of the short term pressure analysis are summarized in Table 14.3.4-38. The values given in parentheses are those pressures calculated on the same basis but without using a compressibility multiplier. As can be seen from the table, the effects of the modifications to the TMD code are minimal.

Consideration was given to determining the effect of a varying polytropic exponent of the flow mixture across the throat section of a flow path. This was done by lowering the steam-water polytropic exponent calculated by the code by 5, 10 and 20%. The lowered polytropic exponent variance computer runs were made for a DEHL break in lower compartment element #6. The results are presented in Table 14.3.4-39 and it is apparent that the polytropic exponent variance has virtually no effect on the results.

14.3.4.5.3.3 Derivation of the Compressibility Multiplier

The system under study is shown in Figure 14.3.4-154. The flow assumptions are:

1. Steady flow
2. Zero gravity effects
3. Isentropic conditions
4. Fluid is an ideal gas
5. Channel wall is non-conducting (no heat transfer)

The resulting compressibility multiplier is:

$$y = \left[r^{2/\gamma} \left(\frac{\gamma}{\gamma - 1} \right) \left(\frac{(1-r)\left(\frac{\gamma-1}{\gamma}\right)}{1-r} \right) \right]^{1/2} \left[\frac{1-B^4}{1-B^4 r^{2/\gamma}} \right]^{1/2}$$

The choked mass flow rate is:

$$\dot{m} = ay \left[\frac{2g\rho_1 (P_1 - P_2)}{1 - B^4} \right]^{1/2}$$

where

$$B = (a/A)^{1/2}$$

We next apply the compressibility multiplier to the friction term of the TMD momentum equation written as:

$$\Delta P = \left(\frac{K + f_1 / D}{2\rho g} \right) \left(\frac{\dot{m}^2}{a^2} \right) \qquad (3)$$

Incorporating the compressibility multiplier into the TMD momentum equation, eqn. (3) takes on the form:

$$\Delta P = \frac{(K + f_1 / d)\dot{m}^2}{2\rho g y^2 a^2} \tag{4}$$

Coupling eqn. (4) with the inertia term presently used in TMD, the momentum equation for general flow systems (non-steady state) appears as:

$$\Delta P = \frac{L}{A}\frac{dm}{dt} + \frac{(K + f_1 D)\dot{m}^2}{2\rho g y^2 a^2} \tag{5}$$

It should be noted that the TMD (Reference 14) computer code also employs a critical flow correlation as a check on sonic flow conditions. This critical flow correlation has not been modified as a result of this present work.

The compressibility multiplier as it is used in eqn. (4) (and in TMD) is calculated by the code; the only information needed as input is the B factor.

The polytropic exponent is also calculated within the code, dependent upon the flow mixture conditions.

14.3.4.5.3.4 Choked Flow Characteristics

The data in Figure 14.3.4-155 illustrate the behavior of mass flow rate as a function of upstream and downstream pressures, including the effects of flow choking. The upper plot shows mass flow rate as a function of upstream pressure for various assumed values of downstream pressure. For zero back pressure ($P_d = 0$), the entire curve represents choked flow conditions with the flow rate approximately proportional to upstream pressure, P_u. For higher back pressure, the flow rates are lower until the upstream pressure is high enough to provide choked flow. After the increase in upstream pressure is sufficient to provide flow choking, further increases in upstream pressure cause increases in mass flow rate along the curve for $P_d = 0$. The key point in this illustration is that flow rate continues to increase with increasing upstream pressure, even after flow choking conditions have been reached. Thus choking does not represent a threshold beyond which dramatically sharper increases in compartment pressures would be expected because of limitations on flow relief to adjacent compartments.

The phenomenon of flow choking is more frequently explained by assuming a fixed upstream pressure and examining the dependence of flow rate with respect to decreasing downstream pressure. This approach is illustrated for an assumed upstream pressure of 30 psia as shown in

the upper plot with the results plotted vs. downstream pressure in the lower plot. For fixed upstream conditions, flow choking represents an upper limit flow rate beyond which further decreases in back pressure will not produce any increase in mass flow rate.

The augmented choked flow relationship used in TMD is based on experimental data obtained for choked two-phase flow through long tubes, short tubes, and nozzles. The short tube data was cited by Henry and Fauske in Reference (16). Henry and Fauske conclude that an identical discharge coefficient may be applied to two-phase critical flow through sharp-edged orifices and short tubes to represent the actual critical flow rate through each geometry. On this basis, since the augmented choked flow correlation is based on short-tube data, it is applicable to sharp-edged orifices as well. Figure 14.3.4-156, from Reference (17), presents experimental data for two phase critical flow through several different geometries. The dashed line on the graph represents the augmented homogeneous equilibrium critical flow relationship used in TMD. Below a quality of 0.2 the augmentation correlation is not applicable. 0.62 is the highest quality at which critical flow is calculated by TMD to occur in a major flow path following a DEHL break in the Cook containment. It is apparent that the augmented critical flow calculated by TMD is conservative within the quality range of interest.

Carofano and McManus (18) have published data for the two-phase flow of air-water and steam-water mixtures. Actually, water vapor was present in the gas phase of the so-called air-water test, making it in effect an air-steam-water test. The data presented in Reference (18) demonstrates that the ratio of experimental air-(steam)-water critical flow values to homogeneous equilibrium model predictions is equal to or greater than the ratio of steam-water experimental critical flow values to homogeneous equilibrium model predictions. Therefore augmentation factors derived by comparing steam-water data to the homogeneous equilibrium model may be used in air-steam-water calculations.

14.3.4.5.3.5 Early Sensitivity Studies

The TMD computer code was used to establish peak pressures and peak pressure differentials for double-ended hot and cold leg breaks, double-ended steam line breaks in the steam generator and fan room enclosures, a 6 inch spray line break for the pressurizer enclosure, and a single-ended pipe break in the reactor cavity of the D. C. Cook Plant. These cases were analyzed with and without augmentation of the calculated homogeneous equilibrium critical mass flow rates, to study the sensitivity of compartment pressures to augmentation. The double-ended hot leg break was assumed to occur in node 6, and the double-ended cold leg break was assumed to occur in node 1 of the TMD model network given in Figure 14.3.4-20. These were the worst locations for a hot leg and a cold leg break, respectively.

The pressure response to a hot leg break is only slightly affected by augmentation; the cold leg break pressure response exhibits significant sensitivity to augmentation. Since the hot leg break parameters are limiting for the D. C. Cook Plant, omitting augmentation increases the design basis peak operating deck ΔP less than 5%.

No change in the compartment peak pressures or pressure differentials occurred when unaugmented critical flow was used in analyzing the D. C. Cook Plant fan room and steam generator enclosure. Removing augmentation increased the peak pressure and the peak differential pressure in the pressurizer enclosure by 27% and 25%, respectively, in the upper reactor cavity by 17.5% and 19.5%, respectively, and in the lower reactor cavity by 13% and 8%, respectively.

The reason that there is no change in the peak pressures in the steam generator enclosure is that both the peak pressure and peak differential pressure are due to inertia. The fan room pressures remain constant because of high resistances in the flow paths from the fan room to the lower compartment which prevent choking.

In the reactor cavities and pressurizer enclosure, peak pressures occur in the transient coincidental with choking, and therefore a significant change in calculated pressures will occur when the critical flow model is changed (see Table 14.3.4-40).

A number of analyses have been performed using 100 percent moisture entrainment to determine the various pressure transients resulting from hot and cold leg reactor coolant pipe breaks in any one of the six lower compartment elements. The maximum peak pressure and differential pressure for all cases have been determined for each compartment element. Figure 14.3.4-157 is representative of the upper and lower compartment pressure transients that result from a hypothetical double ended rupture of a reactor coolant pipe for the worst possible location in the lower compartment of the containment, a hot leg break (DEHL) in element 6.

In addition, a series of TMD runs investigated the sensitivity of peak pressures to variations in individual input parameters for the design basis blowdown rate and 100% entrainment. This analysis used a DEHL break in element 6, and investigated effects from blowdown sensitivity to addition of the compressibility factor in the momentum equation. Table 14.3.4-36 gives these results.

The sensitivity study results demonstrate that variations in the plant geometric parameters and in ice bed loss coefficients, both of which are known with a high degree of accuracy, have little effect on the peak pressure calculated by TMD for DEHL break in element 6. However, variations in blowdown and entrainment, which are not known with great accuracy, greatly affect

the pressure calculated. The highly conservative values used in the design basis analysis ensure a conservative prediction of the peak break compartment pressure.

14.3.4.5.4 Ice Condenser Performance Criteria

The performance of the ice condenser containment is demonstrated by results and analysis of ice condenser tests performed on a full-scale section test at the Westinghouse Waltz Mill Site. These tests confirmed the ability of the ice condenser to perform satisfactorily over a wide range of conditions, exceeding the range of conditions that might be experienced in an accident inside the containment.

The ice condenser containment performance has been evaluated by testing the following important parameters. A partial list of parameters tested include blowdown rate, blowdown energy, deck leakage, compression ratio, drain performance, ice condenser hydraulic diameter, dead-ended volumes and long term performance. Analytic models have been developed to correlate and supplement these test results in the evaluation of the containment design. The results indicate that the analytical models are conservative and that the performance of the ice condenser containment is predictable relative to these variables.

The layout of the reactor containment compartments and ice condenser provides for effective and efficient use of the ice condenser to suppress pressure buildup.

The lower (Reactor Coolant System) compartment is bounded by the divider barrier such that essentially all of the energy released in this compartment is directed through doors at the bottom of the ice condenser.

Seals are provided on the boundary of compartments and hatches in the operating deck to prevent steam from bypassing the ice condenser.

Layout, size, and flow communication among compartments is arranged to minimize the containment volume compression ratio.

The ice bed geometry provides sufficient ice heat transfer area and flow passages so that the magnitude of the pressure transient resulting from an accident does not exceed the containment design pressure for all reactor coolant pipe breaks sizes up to and including the hypothetical double-ended severance of the reactor coolant piping.

The initial containment peak pressure and peak asymmetrical containment pressure loads are determined by analysis. The analytical results are experimentally verified by comparison with the ice condenser tests. This analysis supplements the experimental proof of performance tests and provides pressure transients for application of the plant design.

The final peak pressure occurring at or near the end of blowdown is determined by a containment volume air compression calculation. A method of analysis of the final peak pressure was developed based on the results of full-scale section tests.

Steam bypass of the ice condenser during the postulated RCS blowdown is to be avoided. The divider deck and any other leakage paths between the lower and upper compartments are reasonably sealed to limit bypass steam flow. For the containment, the analysis considered bypass area as composed of two parts: a conservatively assumed leakage area around the various hatches in the deck, and a known leakage area through the deck drainage holes for spray located at the bottom of the refueling cavity.

Flow distribution to the ice condenser for any RCS pipe rupture that opens the ice condenser inlet doors, up to and including the double-ended RCS pipe rupture, is limited such that the maximum energy input into any section of the ice condenser does not exceed its design capability. The door port flow resistance and size provides this flow distribution for breaks that fully open the ice condenser inlet doors. For breaks that partially open the inlet doors, the lower inlet doors proportion flow into the ice bed limiting maldistribution.

For large pipe breaks, the containment final peak pressure is mainly determined by the displacement of air from the lower compartment into the upper compartment. Only a small amount of steam bypasses the ice condenser by passing through the operating deck and into the upper compartment. This steam bypass then adds a small amount to the final peak pressure.

For small pipe breaks, which generate less than the pressure drop required to fully open the spring-hinged, ice condenser inlet doors and result in the door performance being in the flow proportioning range, a larger than normal fraction of the break flow will pass through the deck by way of the divider deck bypass area and into the upper compartment. Unlike a postulated large break, which sweeps all of the air out of the lower compartment, the smaller breaks do not. Below some break size, only a portion of the lower compartment air will be displaced by the break flow. Also, for breaks less than approximately l0,000 gpm, the ice condenser inlet doors will open; however, only a fraction of the air may be displaced from the lower compartment by the incoming steam. For these small energy release rates, operation of the containment spray system eventually is required to limit the containment pressure rise.

Another case has been examined where it is postulated that a small break loss-of-coolant accident precedes a larger break accident which occurs before all of the coolant energy is released by the small break, (i.e., a double accident). During the small break blowdown, some quantity of steam and air will bypass the ice condenser and enter the upper compartment via

leakage in the divider deck. The important design requirement for the case of a double accident is that the amount of steam leakage into the upper compartment must be limited during the first part (small break) of the accident so that only a small increase in final peak pressure results for the second part (double-ended break) of the postulated accident. The steam which reaches the upper compartment will then add to the peak pressure for the second part of the accident. Therefore, the containment spray system is used to limit the partial pressure of steam in the upper compartment due to deck bypass. The key elements which determine the double accident performance are the ice condenser lower doors, which open at a low differential pressure to admit steam to the ice condenser and limit the bypass flow of steam and thus the partial pressure of steam in the upper compartment, and the sprays which condense this bypass flow of steam and limit the partial pressure of steam in the upper compartment to a low value, less than 2 psia. The containment spray set point actuation pressure has been set at 3 psig to limit steam partial pressure to less than 2 psia in the upper compartment for the double accident use.

After a LOCA, the ice condenser has sufficient remaining heat absorption capacity such that, together with the containment spray system, subsequent assumed heat loads are absorbed without exceeding the containment design pressure. The subsequent heat loads considered include reactor core and coolant system stored heat, residual heat, substantial margin for an undefined additional energy release, and consideration of steam generators as active heat sources.

The primary purpose of the Containment Spray System is to spray cool water into the containment atmosphere in the event of a loss-of-coolant accident, thereby ensuring that containment pressure cannot exceed the containment design pressure. Protection is afforded for all pipe break sizes up to and including the hypothetical instantaneous circumferential rupture of a reactor coolant pipe. Adequate containment heat removal capability for the Ice Condenser Containment is provided by two separate full capacity containment spray systems. The Containment Spray System is designed based on the conservative assumption that the core residual heat is continuously released to the containment as steam, eventually melting all ice in the ice condenser. The heat removal capability of each spray system is sized to keep the containment pressure below design after all the ice has melted and residual heat generated steam continues to enter the containment. The spray system is designed to keep the pressure below the design pressure with adequate margin.

14.3.4.5.4.1 Inlet Door Performance

14.3.4.5.4.1.1 Introduction

The ice condenser inlet doors form the barrier to air flow through the inlet ports of the ice condenser for normal plant operation. They also provide the continuation of thermal insulation around the lower section of the crane wall to minimize heat input that would promote sublimation and mass transfer of ice in the ice condenser compartment. In the event of a loss-of-coolant incident that would cause a pressure increase in the lower compartment, the doors open, venting air and steam relatively evenly into all sections of the ice condenser.

The inlet doors are essentially pairs of insulated composite panels vertically hinged to a rectangular shaped angle section frame that has a center post. This assembly is fastened to the crane wall support columns which frame the ports through the crane wall from the containment lower compartment to the ice condenser compartment.

The door panels are made of composite steel sheets and urethane foam construction, comprising a total thickness of 7 inches to provide proper insulating characteristics. Each door is mounted to the frame with ball bearing hinges. The door panels are normally held shut against a bulb type gasket seal by the differential pressure produced by the higher density cold air of the ice condenser, sealing against loss of the ice condenser air to the lower compartment.

The door panels are provided with tension spring mechanisms that produce a small closing torque on the door panels as they open. The magnitude of the closing torque is equivalent to providing a one pound per square foot pressure drop through the inlet ports with the door panels open to a position that develops full port flow area.

The zero load position of each spring mechanism is set so that with zero differential pressure across the door panels the gasket seal holds the door slightly open. This provides assurance that all doors will be open slightly and relatively uniformly, prior to development of sufficient lower compartment pressure to cause flow into the ice condenser, therefore eliminating significant inlet maldistribution for very small incidents. For larger incidents the doors open fully and flow distribution is controlled by the inlet ports.

14.3.4.5.4.1.2 Design Criteria

Normal Operation

a. Doors shall be instrumented to allow remote monitoring of their closed position.

b. Doors shall be capable of being inspected to determine that they are functioning properly.

c. The inlet doors shall limit the leakage of air out of the ice condenser to the minimum practical limit.

d. The inlet doors shall restrict the heat input into the ice condenser to the minimum practical limit.

e. Normal maintenance and inspection must be performed in a manner that does not hinder the ice condenser performance or availability.

Accident Conditions

a. All doors shall open to allow venting of energy to the ice condenser for any leak rate which results in a divider deck differential pressure in excess of the ice condenser cold head.

   The force required to open the doors of the ice condenser is sufficiently low such that the energy from any leakage of steam through the divider barrier can be readily absorbed by the containment spray system without exceeding containment design pressure.

b. Doors and door ports shall limit maldistribution to 150% maximum, peak to average mass input for the accident transient which provides adequate margin in the design ice bed loadings. This is used for any reactor coolant system energy release of sufficient magnitude to cause the doors to open. The inlet doors of the ice condenser are designed to open and distribute steam to the ice condenser in accordance with design basis above, for any postulated loss-of-coolant accident.

c. The doors are designed to eliminate the possibility of doors remaining closed, even for small break conditions. In particular, two degrees of freedom of rotation are incorporated in the hinges and the sealing gasket is designed to pull out for a postulated condition of sticking. The gasket material is itself selected to prevent sticking.

d. The basic performance requirement for lower inlet doors for design basis accident conditions is to open rapidly and fully, to ensure proper venting of released energy into the ice condenser. The opening rate of the inlet doors is important to ensure minimizing the pressure buildup in the lower compartment due to the rapid release of energy to that compartment.

e. Ice condenser doors shall be protected from direct steam jet following a postulated steam line break.

14.3.4.5.4.1.3 Performance Capability

Normal Operation

The normal operation mode for the ice condenser inlet doors is to serve as an insulated barrier to natural convection heat and air flow through the ice bed, providing a sufficient insulating value to limit heat input into the ice bed. In addition the design and performance of the doors must be consistent with ensuring continuous availability of the ice condenser function.

The importance of normal operation criteria is the establishment of design parameters that provide for long term ice bed life, and for constant ice condenser availability for plant protection. In this context two inlet door design parameters affect these factors. They are heat conductivity and leak tightness.

Heat input is the parameter of prime concern, as it is the major factor influencing ice bed sublimation. Leakage out of the ice bed has been reduced to an insignificant amount. The heat input is a calculated value using a two dimensional heat conductance computer program that has been verified by tests.

The inlet door leakage is predicted by tests to be significantly less than the 50 CFM total used for the ice condenser design. This predicted leakage value has negligible effect on reinforcing the convective flow developed in the ice bed, therefore not affecting sublimation rates significantly. The effect of the make-up air entering the ice condenser due to this leakage is also negligible on refrigeration load or ice condenser air handling unit coil defrost frequency.

Seismic analyses associated with response data for the Cook Nuclear Plant shows that the ice condenser inlet doors will not be opened by the maximum seismic forces. This is due to the very low frequency of the rotational or opening mode, at which the response is negligible, and the ice condenser cold head pressure holds the doors closed.

Figure 14.3.4-158 shows the door opening characteristics as a function of door differential pressure based on a linear spring constant. Notably, there is no special significance to be

attached to a linear spring constant, and detail design of the door and spring system indicated that non-linear spring characteristics changed the release rate at which maximum maldistribution would occur, but did not change the maximum maldistribution value. The performance characteristics to be expected from the inlet doors would be typical of those shown in Figure 14.3.4-158.

The effect of maximum variation of door proportioning characteristics indicates significantly less maldistribution than the 150% limit.

Importantly, and as discussed in other reports, the ratio of maximum to average flow of steam into the ice condenser for pipe break sizes large enough to fully open the doors is limited by the door ports themselves to a reasonably low value, about 116 percent of the average.

The equilibrium position of the inlet door panels with zero differential pressure is slightly open (about 3/8 inch), which provides a small flow area at each door for uniform inlet flow into each segment of the ice bed. The doors are designed to eliminate the possibility of doors remaining closed, even for small break conditions. In particular two degrees of freedom of rotation are incorporated in the hinges, and the sealing gasket is designed to pull out for a postulated condition of sticking. The gasket material is itself selected to prevent sticking.

Consideration is, however, given in the analysis of ice condenser performance to a hypothetical case of stuck doors at which the most severe of the above postulated malfunctions is overcome by the force on the door. Even in this hypothetical case the door panels would rupture, providing a sufficient flow path into the ice condenser to permit the ice condenser to function to limit containment pressure below design limits.

It is recognized that the springs are an important part of the lower ice condenser doors. These spring assemblies are designed such that the failure of any spring will not significantly change the operating characteristics of the ice condenser doors. This objective has been achieved in a practical manner by the use of four separate tension springs per door, which provides redundancy and assures adequate opening characteristics.

Accident Conditions

The basic lower inlet door performance requirement for design basis accident conditions is to open rapidly and fully, to insure proper venting of released energy into the ice condenser. The opening rate of the inlet doors is important to insure minimizing the pressure buildup in the lower compartment due to the rapid release of energy to that compartment. The rate of pressure rise and the magnitude of the peak pressure in any lower compartment region is related to the

confinement of that compartment, and in particular the active volume and flow restrictions out of that compartment. The time period to reach peak lower compartment pressure due to the design basis accident is a fraction of a second. It is dependent upon flow restrictions and proximity to the break location. The opening rate of the inlet doors is wholly dependent upon the inertia of the door and the magnitude of the forcing function, which is the pressure buildup in the lower compartment due to the energy release. The ice condenser inlet door inertia is slightly less than the doors tested in the ice condenser full scale section tests. These tests demonstrate that door inertia has essentially no effect on the initial peak pressure.

The maximum inlet door structural loading is due to the design basis accident for the doors adjacent to the lower compartment in which the release occurs. Structural analysis for maximum loaded conditions shows that all door members remain well below allowable stress levels. Further verification of the structural adequacy of the door is provided by the proof load testing carried out on the full prototype doors.

The necessary performance of the ice condenser is further ensured by the door design incorporating a low pressure fail open characteristic. Even if it is postulated that the doors were held rigidly along the bottom edge, they would fail open at a differential pressure sufficiently low to allow venting from the lower compartment well within the limits of pressure capability of the structures.

14.3.4.5.4.2 Top and Intermediate Deck Door Performance

The doors enclosing the top of the ice condenser and forming the roof of the upper plenum are of a non-rigid design lighter than the intermediate deck doors. These top doors are supported by the ice condenser bridge crane support structure. The crane support structure consists of radial beams spanning the ice condenser annulus at the top of the crane wall.

The intermediate deck doors enclose the ice compartment and forming the floor of the upper plenum. These doors are supported by the lattice frame support columns.

The intermediate deck door panels are a 2 1/2 inch foam plastic core with bonded sheet metal facings. These doors are hinged horizontally and are normally closed. The top deck doors are flexible foam bonded to high strength foil steel. These doors are hinged (clamped) on the crane wall side of the top deck and are normally closed. On an increase in pressure in the ice condenser compartment, these doors will open as required, allowing air to flow into the upper containment volume.

14.3.4.5.4.2.1 Design Criteria

Normal Operations

1. The top deck will be provided with a total vent area of approximately 20 ft^2.
2. Doors will limit heat input within their immediate vicinity to the minimum practical limit.
3. Doors will be capable of being inspected during plant shutdown to determine that they are functioning properly.

Accident Conditions

1. All doors will open fully for a low differential pressure loading.
2. All doors will be light-weight to have a minimum effect on the initial peak pressure.
3. Doors will be of simple mechanical design to minimize the possibility of malfunction.
4. Doors will not be required to remain either open or closed following an accident.

14.3.4.5.4.2.2 Performance Capability

On an increase in pressure in the ice condenser compartment, these doors will open as required to allow air to flow into the upper compartment. The primary design criterion for these doors is their insulating capability to limit heat flow. The flow area provided by the open doors is that area available in the compartment, considering area reduction by support structures. Both the inertia of the door with the desired insulation capability and the available flow area have been modeled in the ice condenser door tests.

The mean heat input to the plenum through the top deck is about 4.5 Btu/hr-ft^2. This heat input is removed from the plenum ambient by the ice condenser air handling units and does not affect ice bed sublimation. The effect of this heat load on refrigeration heat load and plenum ambient conditions has been investigated and provides for operation well within the ice condenser design operating parameters.

Incorporation of the vents imposes no operational problem to the ice condenser. The lower doors will effectively seal the ice condenser compartment and limit any flow of air through the compartment to a negligible value. Therefore tight seals are not necessary at the intermediate

deck doors. A balanced flapper is provided to minimize migration of moisture into the ice condenser through the vents.

The open flow area through each deck for air flow due to an incident is slightly larger than the ice condenser test ratio equivalent, providing slightly less resistance to air flow. The slightly larger flow area does not produce any significant change in ice condenser performance, other than to assure flow resistances slightly less restrictive than the reference design.

The top and intermediate deck and doors have been analyzed for all loading combinations. This structural analysis shows that all members remain below allowable stress levels.

14.3.4.5.4.3 Vent Design and Performance

14.3.4.5.4.3.1 Introduction

The upper and intermediate doors are not required to remain open following the reactor coolant system blowdown and also are not required to open for small breaks. For these situations, a vent was provided through both the top and intermediate deck to allow air to flow into or from the ice condenser compartment as required. This vent air first passes into the fan cooler plenum at the top of the ice condenser compartment where it mixes with the cooling air. The temperature and humidity of the vent air that passes from this plenum into the ice condenser compartment are therefore about the same as the average temperature and humidity of the air in the condenser compartment.

Accordingly, sublimation or frosting in the ice bed due to this vent flow of air will be limited to a negligible value. Specific performance requirements are given below both for large breaks and for small breaks.

14.3.4.5.4.3.2 Large Break Performance Requirements

Following the reactor coolant system blowdown, the vents were designed to allow air to return from the upper compartment into the ice condenser without imposing an excessive pressure drop across the upper and intermediate doors. The maximum pressure decay rate and therefore the maximum reverse flow rate of air results from the case where reactor residual heat is not released to the containment following the reactor coolant system blowdown. The pressure decay for this case was measured in a full-scale section test. In this test, the pressure decayed from 6.2 psig to 4.5 psig during the one minute period immediately following the reactor coolant system blowdown. From this pressure decay rate, the plant equivalent flow rate of air was calculated to be 98 lbm/sec. This flow rate developed a pressure drop across each of the upper doors of 0.28 psi, which was well within the structural capability of the upper and intermediate doors and

support structures. Further, in this calculation it was conservatively assumed that no air flowed through the deck or through the containment air recirculation fan duct.

14.3.4.5.4.3.3 Small Break Performance Requirements

For small breaks which generate less than the required opening pressure of the upper and intermediate doors, the vent was designed to limit the flow of steam through the deck to an acceptable level during the period of air flow through the condenser and into the upper compartment. For breaks less than approximately 5000 gpm, the full-scale section tests have shown that only a fraction of air is displaced from the lower compartment. The 20 sq. ft. vent area in both the top and intermediate deck provides a low resistance air flow path through the ice condenser to the containment upper compartment for these small break conditions.

14.3.4.5.4.4 Drain Design and Performance

14.3.4.5.4.4.1 Introduction

Drains are provided at the bottom of the ice condenser compartment to allow the melt-condensate water to flow out of the compartment during a loss-of-coolant accident. These drains are provided with check valves that are counter-weighted to seal the ice condenser during normal plant operation and to prevent steam flow through the drains into the ice condenser during a loss-of-coolant accident. These check valves will remain closed against the cold air head (1 psf) of the ice condenser and open before the water level rises to the point where it can interfere with the operation of the lower inlet doors, as described in Chapter 5.3.

For a small pipe break, the water inventory in the ice condenser will be produced at a rate proportional to the rate of energy addition from the accident. The water collecting on the floor of the condenser compartment will then flow out through the drains and through the doors, which are open during the blowdown.

For a large pipe break, a short time (on the order of seconds) will be required for the water to fall from the ice condenser to the floor of the compartment. Therefore, it is possible that some water will accumulate at the bottom of the condenser compartment at the completion of the blowdown. Such water accumulation could exert a back pressure on the inlet doors, requiring an additional pressure rise in the lower compartment to open the doors and admit steam to the ice condenser. However, results of full-scale section tests indicated that, even for the design blowdown accident, a major fraction of the water drained from the ice condenser, and no increase in containment pressure was indicated even for the severe case with no drains.

14.3.4.5.4.4.2 Large Break Performance Requirements

A number of tests were performed with the reference flow proportional-type door installed at the inlet to the ice condenser, the reference-type hinged door installed at the top of the condenser. Tests were conducted with and without the reference water drain area, equivalent to 15 ft^2 for the plant,[1] at the bottom of the condenser compartment.

Tests were conducted with various assumed blowdown conditions. These tests were performed with the maximum reference blowdown rate, with an initial low blowdown rate followed by the reference rate, with a low blowdown rate alone, and with the maximum reference blowdown rate followed by the simulated core residual heat rate.

The results of all of these tests showed satisfactory condenser performance with the reference type doors, vent, and drain for a wide range of blow-down rates. Also, these tests demonstrate the insensitivity of the final peak pressure to the water drain area. In particular, the results of these full-scale section tests indicated that, even for the reference blowdown rate, and with no drain area provided, the drain water did not exert a significant back pressure on the ice condenser lower doors. This showed that a major fraction of the water had drained from the ice condenser compartment by the end of the initial blowdown. The effect of this test result is that containment final peak pressure is not affected by drain performance.

Although drains are not necessary for the large break performance, approximately 13 ft^2 of drain area are required for small breaks.

14.3.4.5.4.4.3 Small Break Performance Requirements

For small breaks, water will flow through the drains at the same rate that it is produced in the ice condenser. Therefore, the water on the floor of the compartment will reach a steady height which is dependent only on the energy input rate.

To determine that the 12.63 ft^2 drain area met these requirements, the water height was calculated for various small break sizes up to a 30,000 gpm break. Above 30,000 gpm the ice condenser doors would be open to provide additional drainage. The maximum height of water required was calculated to be 2.2 ft above the drain check valve. Since this height resulted in a water level which was more than 1 ft below the bottom elevation of the inlet doors, it was

[1] As noted in Chapter 5.3.5.1.6., the D.C. Cook ice condenser floor drains have a flow area of approximately 18 ft^2. Since the reference tests were performed at Waltz Mill both with and without the reference plant drain area of 15 ft^2, the D.C. Cook ice condenser drain configuration is bounded by the reference tests.

concluded that water will not accumulate in the ice condenser for this condition and that a 12.63 ft^2 drain will give satisfactory performance.

14.3.4.5.4.4.4 Normal Operational Performance

During normal plant operation, the sole function of the valve is to remain in a closed position, minimizing air leakage across the seat. To avoid unnecessary contamination of the valve seat, a 1 ½ -inch drain line is connected to the 12 inch line immediately ahead of the valve. Any spillage or defrost water will drain off without causing the valve to be opened.

Special consideration has been given in the design to prevent freezing of the check valves and to minimize check valve leakage.

To minimize the potential for valve freezing, a low conductivity (transite) section of pipe is inserted vertically below the seal slab, while the horizontal run of pipe (steel) is embedded in a warm concrete wall before it reaches the valve. The valve itself is in the upper region of the lower compartment, where ambient temperature is generally above the freezing temperature.

The valve is held in a closed position by virtue of its design as an almost vertical flapper with a hinge at the top. The slight (10°) angle from the vertical holds the flap in place by gravity.

To reduce valve leakage to an acceptable value, a sealant was applied to the seating surface after installation of the valves. Tests show that this will reduce leakage to practically zero. Maximum allowable leakage rate would be approached as a limit only if all the sealant were to disappear completely from all the valves, which is unlikely. Sealant is replaced as necessary.

14.3.4.6 Changes from Base Containment Analyses: Note Concerning Tables and Figures

If an evaluation or partial re-analysis is needed for some change from one of the base containment analyses of record, and this results in changes to information appearing in the UFSAR, a text description of the new work is provided here in Section 14.3.4. However, unless specifically indicated otherwise, the associated tables and figures for Section 14.3.4 are taken from the base analysis of record, and not any subsequent specific evaluation or partial re-analysis.

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