ACRSR-1969

October 23, 2001

Dr. William D. Travers
Executive Director for Operations
U.S. Nuclear Regulatory Commission
Washington, D.C. 20555-0001

Dear Dr. Travers:

SUBJECT: EPRI REPORT ON RESOLUTION OF NRC GENERIC LETTER 96-06 WATERHAMMER ISSUES

During the 486th meeting of the Advisory Committee on Reactor Safeguards, October 4-6, 2001, we completed our review of the Electric Power Research Institute (EPRI) proposed approach to resolution of issues associated with waterhammer events occurring in low-pressure containment cooling systems in pressurized water reactors (PWRs), pursuant to the requirements specified in NRC Generic Letter (GL) 96-06. We also discussed this matter with representatives of the NRC staff and EPRI during our 485th meeting on September 5-7, 2001. In addition, our subcommittee on Thermal-Hydraulic Phenomena reviewed this matter during meetings held on November 17, 1999, January 16-17, 2001, and August 22-23, 2001. During our review, we had the benefit of the documents referenced.

RECOMMENDATION

The application of the proposed EPRI methodology should not be approved until there is a better demonstration that it provides results that are bounding for realistic plant configurations and scenarios.

DISCUSSION

Fan cooler units (FCUs) are installed in PWRs to cool the containment during normal operation and, for some designs, to cope with the design-basis loss-of-coolant accident (LOCA) or main steamline break (MSLB) event simultaneous with a loss of offsite power (LOOP). In this design-basis event, the cooling water will stop circulating through the FCU and the fans driving containment air and steam across the FCU heat transfer surfaces will coast down. These conditions will last at least 30 seconds until the emergency diesel generators can start and the load sequencing will restart the service water pumps. This 30-second window is enough time for the water in the FCUs to boil and drain to create steam-and-air-filled void regions. When the FCU pumps return to power, a “slug” of water entering the voids creates conditions for a possible waterhammer event that could break the system piping, thus potentially causing loss of the cooling function, containment flooding, and creation of a containment bypass path.

The NRC issued GL-96-06, in part, to address the above concerns. The GL referenced NUREG/CR-5220, which provides a conservative approach for evaluating waterhammer events.

EPRI and a number of utilities elected to pursue a less conservative approach than that specified in this report.

The EPRI methodology includes an analytical model of the closure of an air and steam pocket, which could cushion the impact of the incoming water slug, thus spreading out the pressure spike and reducing its maximum amplitude. The EPRI model hypothesizes that the pocket contains dissolved air that is released during boiling, and steam that is left uncondensed before the waterhammer event occurs.

The determination of the two major parameters (air release fraction and steam condensation on the water/void interfaces) that affect the reduction in severity of the waterhammer, was done by EPRI in scaled experiments that were intended to represent conditions in real FCUs.

To determine the air release fraction, EPRI conducted two series of experiments with two different configurations of the test apparatus. In the first configuration, water in a simulated FCU heat exchanger tube was allowed to drain into a header as boiling occurred in the tube due to heat addition from steam external to the tube. The difference between the oxygen content of the drained water and the initial concentration before the experiment started was said to be an indication of the air released. EPRI ran a number of such tests at two different boiloff rates and recommended the lower mean value of the results as the air release fraction under these conditions.

To simulate conditions that would exist in some parts of the FCU, the second configuration had the header release path filled with air-saturated water to a height of two-feet above the heat exchanger tube drain point, thus inhibiting drainage. In these experiments, the steam and released air bubbled up through the two-feet of water before being released. The dissolved oxygen content of the two-foot column before and after the 30-second test period was considered to be an indication of the air-release fraction. EPRI recommended using the lower mean value from the two sets of fractions of air released.

EPRI also experimentally determined the condensation rates of steam during column closure events in a two-inch diameter pipe. The real FCU waterhammer events are believed to generally take place in piping that is 10-inches to 16-inches in diameter. The results, therefore, must be scaled up to represent plant FCUs. EPRI developed a scaling model to do so.

Our discussion focused on the prototypicality of these experiments, the adequacy of the scaling model, and the appropriateness of the condensation and air-release models (see the attached discussion by ACRS Member Graham Wallis). We find that EPRI's conceptual model is oversimplified, and we are uncertain how it can be applied to plant-specific scenarios and configurations.

The combination of a LOCA and an independent LOOP is a very low probability event (some estimates have placed the frequency at a value as low as 10^{-9}/yr). A risk-informed view could conclude that such an event is not risk significant. This LOCA/LOOP event, however, has been defined as a design-basis accident, and coping with it must be considered as a compliance issue at this time. We would, however, support efforts to use risk-informed approaches to

determine whether the FCU waterhammer requirements associated with the LOOP/LOCA event should be modified.

Sincerely,

/RA/

George E. Apostolakis
Chairman

References:
1. U. S. Nuclear Regulatory Commission, Generic Letter 96-06, "Assurance of Equipment Operability and Containment Integrity During Design-Basis Accident Conditions," September 30, 1996
2. EPRI Letter dated December 15, 2000, transmitting EPRI Report, TR-113594, Volumes 1 & 2, "Resolution of Generic Letter 96-06 Waterhammer Issues," December 2000 (Proprietary).
3. EPRI Letter dated July 10, 2001, to J. Tatum, U. S. Nuclear Regulatory Commission, Subject: Resolution of Generic Letter 96-06 Waterhammer Issues, EPRI Report TR-113594, Volumes 1&2, Revised Sections.
4. U. S. Nuclear Regulatory Commission, NUREG/CR-5220, Vol. 1, "Diagnosis of Condensation-Induced Waterhammer," October 1988.

Attachment: Comments on EPRI Reports and Presentations on GL-96-06 (Fan Cooler Waterhammer Issues) by ACRS Member Graham Wallis, October 2, 2001.

COMMENTS ON EPRI REPORTS AND PRESENTATIONS ON GL 96-06
(FAN COOLER WATERHAMMER ISSUES)
Graham Wallis, October 2, 2001

In its earlier meetings with EPRI (November 17, 1999, January 16-17, 2001, and August 22-23, 2001), the Thermal-Hydraulic Phenomena Subcommittee raised three major issues: the justification for the air release model, the validity of the condensation model, and the relationship of these models to the actual events that have to be analyzed in a nuclear plant. EPRI responded to the first two of these at the Thermal-Hydraulic Phenomena Subcommittee meeting on August 22-23, 2001, and at the September 6-7, 2001 ACRS meeting.

I still have concerns over the third issue and how it influences the proper conclusions regarding the first two. EPRI has had little to say about plant scenarios. I am unconvinced that the work reported so far actually solves the real problem.

Layout of a Plant

The EPRI scenario is based on idealized sketches of a plant (Figures 2-1 and 2-2 of the Technical Basis Report). This is further reduced to a very simplified diagram in Figure 9-5 of the Report, which is also Figure 5-2 of the User's Manual. Analysis is based on the motion of a single slug compressing a single bubble in a single large pipe closed at the downstream end. All of the air released in the boiling process is assumed to be in this bubble.

The actual plant conditions are far from this conceptual model. Plants typically have three or four fan coolers. In the plants that I am familiar with, these are connected in parallel to ring headers, running around the periphery near the inner wall of containment, that supply water to the coolers and remove it from them. These ring mains also supply service water to other parts of the plant, therefore they are much bigger than is needed to carry the flows to the coolers themselves.

The coolers may each have similar designs, but the piping that connects them to the ring headers is likely to differ between coolers because of the need to avoid other structures in the vicinity. This connecting piping contains various bends and pipes of several orientations, which may include a vertical U-bend. The coolers are not necessarily installed in a symmetrical pattern with respect to the main supply lines or exhaust lines to the ring headers, which are represented by the vertical branches inside containment in EPRI's sketches.

The fan cooler module sketched in Figure 2-3 of the Technical Basis Report has 48 tubes per pass, each consisting of three 112-inch lengths of 5/8-inch tubing. The total fluid volume in this, assuming that 5/8 is the internal diameter of the tubes, is $3 \times 48 \times 112/1728 \times \pi/4 \times (5/8)^2 = 2.86$ cubic feet. According to the sketch, there are six modules stacked above each other. This would lead to an overall height of about 20 feet, which my colleagues who are familiar with plants tell me is too big. Accepting EPRI's figure, the total volume of the cooler is $6 \times 2.86 = 17.2$ cubic feet.

The ring mains have diameters as large as 16 inches. If the nominal value of the diameter is taken as close enough to the actual physical value, the volume of one of these is $\pi/4 \times (16/12)^2$ = 1.4 cubic feet per foot length. Since these mains are perhaps 200 feet long, the total volume of water that is initially in them is large compared with the volume of water in the fan coolers. This water is initially at around the containment temperature of perhaps 100° F and it is not significantly heated by the LOCA atmosphere during the 30 seconds of fan cooler voiding, because the main pipes are large and insulated. Some plants use chilled water, which is initially at or about 50^0 F

LOOP Alone

EPRI does not explain the details of how voids form in a LOOP scenario, but some events can be inferred. Looking at the LOOP discussion on pages 5-2 to 5-5 of their report, we see that plants with open-loop cooling water systems and either top- or bottom-draining FCUs experienced waterhammers during a LOOP. Apparently, this has not happened in plants that have closed-loop cooling water systems.

In a LOOP without a LOCA, the water is not heated and voids form at close to zero pressure. The voids contain a mixture of air and steam with such low density that they might be considered to be empty for some purposes.

We are told that waterhammer occurred in the main piping in both systems. Now, if the void were only in the top of the cooler with a bottom drain, turning on the pump would cause closure in the cooler unless the void actually extended to the main piping. So, I infer that the draining of the main piping, when there is almost a complete vacuum in the cooler, is so rapid that the voids extend into the main pipes before the pumps come on. Condensation does not occur because the temperature of 100° F is the same as the saturation temperature corresponding to the pressure (about 1 psia).

The key point is that in the time before the pumps come on, with an open-loop cooling water system, there is enough draining to the outside world to create significant voids in the main pipes as well as in the cooler, even when there is almost a complete vacuum sucking the water back. These voids are not going to look like EPRI's with vertical ends. They are more likely to be composed of long, stratified regions, with perhaps a bubble-like "nose" at the ends spreading along the pipe, as well as some regions of dispersed smaller bubbles.

I don't know what the assumptions are behind the typical plant's analysis of a LOOP without LOCA. If the void formation in the cooler is homogeneous, then any fluid leaving the cooler will carry air and steam with it, as I will hypothesize for the LOCA. If there is significant phase separation, then the cooler alone must void to some extent before voids reach the mains, depending on the details of the piping.

LOOP-LOCA

In a LOOP-LOCA event, the pumps circulating water to the fan coolers run down and stop. As the containment heats up, heat transfer to the fan coolers is very effective as the fans are still coasting down. Eventually, the water in them reaches saturation temperature corresponding to the local pressure and is ready to boil. Probably, the elevation differences from top to bottom of the cooler are small enough, especially if the height estimated by my colleagues who are

familiar with these devices is less than half that shown by EPRI, that the conditions in the cooler may be taken as approximately uniform.

The water cannot boil unless room is made for the bubbles that are formed. Otherwise the pressure would rise, but no significant vaporization would occur. Voids open up when water flows out of the open ends of the piping system in an open-loop arrangement or when the level rises in the head tank in a closed-loop system. If there is a check valve at the pump in an open-loop system, water must flow out of the discharge, which will not be closed as shown in Figure 9-5 of the EPRI Report and in Figure 5-2 of the User's Manual.

The engineer faced with analyzing what happens next has to perform a transient dynamic analysis of the motion of the water, given a driving pressure (saturation?) in the cooler, the resistance and inertia of the fluid in all the flow paths, and the exhaust or head pressure. This may also involve some heat transfer calculations for the cooler, but probably this is not the limiting process, at least at the start.

Growth of the voids in the coolers leads to ejection of a steam/air/water mixture into the ring headers which are initially filled with water around the containment temperature of 100° F (or less if the water is chilled) and was not warmed up by the LOCA. The first fluid ejected is mostly water; later it is mostly steam. When this mixture emerges into the cold water, the steam tends to condense. This does not occur in a stagnant system. The water in these lines is itself set in motion to varying degrees in response to the flow out of the coolers. The flow out of one cooler displaces water past the other downstream coolers. The order of magnitude of the volumetric cocurrent flow of cold water is the same as that of the mixture leaving the cooler; therefore, new cold water is continually brought into contact with the fluid leaving the coolers.

Even if the flow in the main should be stagnant at one of the entry points, a large bubble of steam will spread by gravity along the top of the pipe in both directions at a speed given approximately by the standard formula for a large "Slug Flow" bubble, 0.5 times the square root of gD, or about 3.3 ft/s for a 16-inch pipe. As the bubble spreads out, this speed reduces roughly in proportion to the square root of the height of the vapor in the stratified region behind the bubble nose. This continually exposes new cold water at the ends of the bubble, and also stirs up the hot and cold water mixture. As long as the pressure is maintained above the saturation pressure corresponding to some effective value of the water temperature in the main, most of the steam entering the ring main(s) will condense leaving bubbles of air spread out along the top of the pipe. Estimates of the rate and extent of this condensation will be given later.

Eventually, the coolers may dry out, stopping the production of steam. If boiling is sufficiently vigorous, as it is described to be by EPRI, then the flow in the cooler may be approximated as homogeneous, with the air, steam, and water moving together at the same velocity. For this case, if the heat transfer and pressure conditions are approximately uniform throughout the cooler, it is possible to calculate the amount of steam that will have been driven into the ring headers by the time the cooler goes dry (see the Appendix to this discussion). The same result is predicted if the fluids in the cooler are well–mixed. For a pressure of one atmosphere in the cooler (closed system), the volume of steam driven out is 6.4 times the volume of the cooler, V_c, and the volume of water driven out is roughly the same as the volume of the cooler. For a pressure of 8 psia in the cooler (open-loop system, corresponding closely to a pressure of one-half atmosphere), the amount of vapor driven out is about 7 times the volume of the cooler.

This may seem like a lot of steam, but it has little mass. The mass is $6.4V_c/v_g$ in the first case. If it is condensed by water initially at 100° F the capacity of the cold water to condense steam may be evaluated from an energy balance in which the water is heated to the saturation temperature in the limiting case. The volume of water that it takes to condense all the steam is

$$V_w = 6.4V_c\, v_f/v_g\, h_{fg}/ (212 - 100)\, C_p =$$
$$6.4\, (.01672/26.8) \times 970.4/(112 \times 1) = 0.0346V_c \text{ cubic feet.}$$

Therefore, it takes a volume of water equal to about 3.5% of the volume of the cooler to condense all the steam that is ejected from the cooler if these events occur at around atmospheric pressure. Using the cooler volume of 17.2 cubic feet calculated earlier, this water volume is close to 0.6 cubic feet or the amount of cold water in about a five-inch length of the ring main. The amount of cold water available in the ring mains is two orders of magnitude larger than this.

For the case of the open-loop cooling systems, EPRI states on page 6-5 that a typical pressure during voiding is 15 inches of mercury. For convenience in using steam tables, I'll assume that this corresponds to 8 psia.

Using the result in the Appendix, the amount of steam ejected from a cooler that is boiled dry homogeneously at 8 psia is 17.2 cu.ft. x 7 / 47.35 cu.ft./lb. = 2.54lb.

The amount of heat transfer it takes to condense this steam is 2.54lb x 988.5 BTU/lb = 2514 BTU. This corresponds to heating 30 pounds of water from 100° F to the saturation temperature of 182.8° F. This is about the amount of water contained in a 4.25-inch length of 16-inch pipe.

The volume of fluid ejected in boiling the cooler dry homogeneously at 8 psia is made up of seven times the cooler volume of steam and one cooler volume of water. Thus, the total volume ejected is 17.2 x 8 = 137.6 ft^3. The velocity of this if it filled a 16-inch pipe and occurred uniformly over 20 seconds is around 5 ft/s. In the smaller exhaust pipe from the cooler, this velocity will be bigger and will stir up the water in the main considerably, thereby enhancing mixing and condensation.

If the heat transfer coefficient is 72,000 BTU/hr.ft^2 F and it acts over an area of twice the main pipe cross-sectional area, then the time taken to condense this steam with a temperature difference of 82.8° F is

$$2514 \text{ BTU/lb} \times 3600 \text{ s/hr} / 72000 \text{ BTU/hr.ft}^2\text{F} / 82.8°\text{F} / 2 / 1.4 \text{ ft}^2 = 0.54 \text{ s.}$$

If the actual effective temperature difference is only 10° F, then the time is 4.5 s.

These results suggest that there is plenty of time for essentially complete condensation to occur in the mains during the voiding of the coolers, which takes around 20 seconds.

One may argue how efficient the mixing is in the ring main, but it is very likely that with so little cold water required, the steam will indeed all be condensed if the pressure is indeed 8 psia. The hot water, equal to about the cooler volume, that also enters the ring main is dispersed and mixed and is unlikely to inhibit condensation. Moreover, it is mostly ejected at the start of the process, while almost pure steam emerges over the latter stages of the discharge and it will be condensed very rapidly.

Should the cooler boil dry, the steam that is left in the cooler at the end of the process has a concentration of air equal to the initial concentration of air in the water. This is because the flow was homogeneous (the same result is obtained if it is considered to be well mixed). The mass of water corresponding to this volume is V_c/v_g whereas the initial mass of water was V_c/v_f. The ratio of these masses is 1600 at atmospheric pressure. Therefore, an estimate that one half of the air initially in the cooler is in the remaining steam bubble would be off by a factor of about 800. The air that was emitted from the water is almost all in the ring main. This illustrates that one cannot simply measure the amount of air removed from a water sample by boiling in a simple laboratory test, one has to work out where it goes in an actual plant. The discussion on pages 5-5 and 5-6 about "the void" does not take into consideration that there are different voids produced at different times. They have different histories and contain different amounts of air.

When the pumps are turned on, water flows into the fan coolers and will rapidly condense the vapor that is there. This is particularly true in the open-system with a check valve at the pump. Since in this case, no fluid comes from the cooler to the upstream ring main during the vaporization phase, the water that enters has not been preheated at all by steam or hot water. It meets steam with almost no air in it. If there is still boiling occurring in a fan cooler that has not fully drained by the time the pumps come on, then a balance must be computed between condensation in one part and boiling in another. There will also be somewhat more air in the steam in this case. I suspect that the condensation will overwhelm any vapor formation because the water flow rate is high and dispersed over many tubes.

In a **closed cooling water system**, or an open system with insufficient downstream hydrostatic suction to support a vacuum, some water from the downstream ring main may be sucked back into some or all coolers in response to rapid condensation in the cooler and the resultant depressurization. A dynamic system analysis has to be performed to decide the direction of flow of this water. Because the air bubbles and warm water have earlier been convected away, this water may be representative of the original cold water in the main.

If water enters a cooler from both ends, with the predominant flow at the entry because the pump head is far greater than the small difference between the vacuum in the cooler and the gravitational head downstream, the final column closure will occur in the cooler. Probably, the last vestiges of steam will be in the downstream header. They will be condensed by multiple jets emerging from the tubes, and will not produce a coordinated waterhammer, though there may be some bangs as individual small bubbles condense.

It is possible that conditions may favor the emergence of some steam into the ring main during the filling of the cooler. If the pressure is maintained at around the 8 psia value used by EPRI, this steam will be condensed on arrival in the cold water environment and will not produce a large bubble in the ring main. There may be some waterhammers in the piping leading from the cooler to the main, perhaps in the U-bends or other places favorable to the trapping of a steam bubble. This is a plant-specific question that is not answered by the EPRI work. Since there have probably been waterhammers in the cooler system during startup and LOOP testing, this event has probably already been shown not to damage the plant.

Even in the very unlikely event that steam survives to make a significant bubble in the ring main, there will be four of these bubbles, not one, and their interaction may tend to reduce the intensity of any waterhammer.

In plants with an **open loop cooling water system,** the rate of draining to the outside world plays a key role in the LOCA scenario. As the coolers boil, the pressure in the voiding region is kept up above the almost complete vacuum in the LOOP case. Therefore, the draining to the outside world is more rapid than in a LOOP alone. With this much water being lost, there must be a corresponding volume of voids formed and it will be larger than in the LOOP case.

In the LOOP event, the pressure is around 1 psia and voids immediately enter the mains if the voiding in the coolers is homogeneous. There, they will not condense and may coalesce to form large bubbles. On the other hand, though voids form more rapidly in the LOOP-LOCA event because of boiling, the voids that enter the mains are removed by condensation as long as the pressure is maintained sufficiently above the saturation pressure corresponding to the temperature in the mains. No waterhammer will occur in the mains unless there is a mechanism for voids to persist there.

Voids will form in the coolers but, as argued above, they will not extend to the main pipes if the steam formed (at 8 psia?) is being consumed almost at once by condensation on the cold water in the mains. What has to happen for voids to persist in the mains is for the pressure to drop to the point where condensation is no longer completely effective. In the extreme case, where approach to equilibrium is rapid, the energy balance is dominated by the heat capacity of the cold water in the mains. The pressure will drop to around the saturation temperature at 100° F, or 1psia, as in the LOOP. There will be a kind of “ejector” sucking steam out of the cooler. The flow rates and pressures at various points have to be calculated from an integrated analysis of the boiling, condensation, flow, and mixing phenomena. EPRI ran their tests at about one-half an atmosphere on the basis of the assertion that this was the pressure in the coolers during a LOCA-LOOP. This assertion must be the result of some analysis that is not described in their report and which needs to be explained.

A possible scenario for waterhammer in the main piping of a plant with an open-loop cooling water system is as follows. The coolers boil and voids form. Steam is ejected into the mains and it condenses, leaving air bubbles and warm water in parts of the mains. Because the mains are draining faster than in a LOOP, the voids in the fan cooler grow faster, making more steam. The pressure in the fan coolers has to be calculated by considering interaction between the rate of boiling, the rate of condensation, and the rate of draining of the main pipes to the outside world. This essential role of the rate and amount of draining to the outside world in an open loop system is missing from EPRI’s report. Indeed, Figure 9-5 in the report, that shows the downstream end of the main pipe to be closed, is misleading as it precludes what is actually the governing process in void growth.

After a while, and especially if the fan coolers become dry, the draining to the outside world may drive the pressure low enough that condensation is no longer so effective in the mains. The pressure may also drop to the point where water in the mains that has been heated by condensation and by mixing with hot water ejected from the fan coolers may flash to form steam. At that time, the steam in the coolers will contain very little air, but the steam formed in the mains will probably mix with the air that was left there by earlier condensation. The point is that the histories of local pressures, temperatures, and air and steam volume fractions have to be calculated from some plant-specific system mode. There may be voids in several locations, each containing different amounts of air. Because there are several coolers and the flow and heat transfer histories differ in the mains downstream of each, it is not clear that subsequent events can be modeled conservatively by assuming a single bubble collapse in a single

location. There may be a succession of column closures, with perhaps some of them involving steam with little air in it.

In contrast to the above, it seems unlikely that either LOOP or LOCA can lead to waterhammer in a closed-loop system because the pressure is maintained above atmospheric by the head tank (assuming it is above the coolers). I note that EPRI only mentions LOOP waterhammers in open-loop systems (pages 5-2 to 5-5). Steam voids are unlikely to get into the mains without condensing there during a LOOP-LOCA if there is a head tank because the key mechanism of a vacuum pulled by the outside world is absent and draining of the mains does not occur. There is plenty of cold water to condense the steam at the prevailing pressure and direct contact condensation should ensure that it occurs rapidly enough that little steam can survive in the main pipes.

There are still questions about the way in which the cooler voids. Does some sort of phase separation occur? How does it differ in the LOOP and LOOP-LOCA cases? Can it be assumed that the cooler is a single uniform node, or should it be represented by several nodes stacked vertically and horizontally? Boiling tends to make things homogeneous; does “void formation” at 1 psia do the same? I expect that void formation occurs throughout the fluid, as in a depressurized champagne bottle.

EPRI has not provided an analysis of the phenomena that occur in an actual plant. The very simplified problem that was addressed seems to have been formulated by a group of experts making estimates in setting up a Phenomena Identification and Ranking Table (PIRT). Perhaps they did have access to plant-specific analyses, but they haven’t shown them to us nor explained the assumptions behind them. We and the staff deserve to see some plant analyses and may need to understand their bases in order to be able to tell if and how the proposed generic solution is relevant or leads to conservative predictions.

A code such as RELAP may have trouble modeling these events because it does not have a good representation of direct contact condensation, the mixing and stratification processes in the main pipes, nor the phase separation that may occur in the coolers.

It is reasonable to argue that waterhammer in a LOOP event would be expected to be more severe than that in a LOOP/LOCA because there is essentially no cushioning in the former case. However, this argument depends on a suitable demonstration that the other key parameters, such as closure velocities and void locations, are sufficiently similar for the two events in a particular plant.

APPENDIX

Let M be the mass of fluid in the cooler at a steam quality x. The volume of the cooler is related to its contents by

$$V_c = M\,(xv_g + (1-x)\,v_f) \qquad (1)$$

Let a mass –dM, containing a mass dM_g of steam leave the cooler. Then

$$-x\,dM = dM_g \qquad (2)$$

Eliminating x from (1) and (2) we get

$$dM_g = -dM/v_{fg}\,(V_c/M - v_f) \qquad (3)$$

Integrating from M_1 at the start to M_2 at the end of the process, the mass of steam ejected is

$$M_g = v_f/v_{fg}\,(M_2 - M_1) - V_c/v_{fg}\,\ln(M_2/M_1) \qquad (4)$$

If the cooler is initially full of water and ends up full of steam,

$$V_c = M_1 v_f = M_2 v_g \qquad (5)$$

Tidying up between (4) and (5),

$$M_g = M_1\,(v_f/v_{fg})\,(\ln\,(v_g/v_f) - 1) \qquad (6)$$

Or, in terms of volumes,

$$V_g = V_c\,(v_g/v_{fg})\,((\ln(v_g/v_f) - 1) \qquad (7)$$

Since v_g is very close to v_{fg} at low pressures, the first factor is about unity.

At one atmosphere, v_g= 26.8 cu.ft/lb and v_f= 0.01672 cu.ft./lb. Then (7) reduces to

$$V_g = V_c\,(7.4 - 1) = 6.4\,V_c.$$

Equation (1) is valid if the conditions of each fluid element are the same and they are each homogeneous (no mixing). It is equally valid if the contents of the cooler are well-mixed. The physics and phase distributions do not change from one case to the other, the conditions being uniform in each case.

The same result can be deduced by analyzing the special case of voiding at constant pressure of a long pipe heated with a uniform heat flux. It is not necessary that the heat flux be constant with time. In this case, the actual void history of each fluid particle is also predicted, perhaps lending more credibility to the analysis until it is realized that the actual geometry does not matter.

The nomenclature used in the above equations is as follows:

C_p	specific heat of water
D	pipe diameter
g	acceleration due to gravity
h_{fg}	latent heat
M	mass
v_f	specific volume of water
v_g	specific volume of steam
v_{fg}	change of specific volume in evaporation
V_c	volume of cooler
V_w	volume of water
x	steam quality